Proceedings of the Conference on Coal Feeding Systems

Held at the California Institute of Technology
Pasadena, California
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FOREWORD

This volume attempts to preserve in written form the presentations made and the discussions held at the Coal Feeding Systems Conference, held on June 21-23, 1977, at the California Institute of Technology, Pasadena, California. Organized by the Jet Propulsion Laboratory under ERDA Contract Award #763-10202-0-341, under direction of Mr. R. R. Fleischbein, P.E., of the Major Facilities Project Management Division, the conference attracted approximately 260 representatives from government, industry, and universities, 30 of whom made formal presentations.

Different methods have been employed in arriving at the written accounts included in these proceedings, depending upon the materials submitted by the authors. Some complete papers were submitted, and other presentations have been reconstructed or abstracted from a tape recording of the sessions. Editorial judgment has been used in an effort to include as much information of general interest as possible, and to publish the proceedings quickly. Since time constraints did not allow the authors to check the final copy and editors to go over the material very carefully, the coordinator assumes responsibility for any errors contained herein.

The conference fulfilled its major objective by providing a forum to discuss the challenges of coal feeding, to acquaint feeder developers with current operations, and to inform potential users about ongoing operations. The successful realization of this conference is the result of a cooperative spirit, the sharing of expert opinions, and timely response by all the participants. Mr. James Powell, Mr. R. R. Fleischbein, P.E., of ERDA, Mr. Arthur Murphy, Mr. R. Phen, and Ms. Diane Osman of JPL are gratefully acknowledged for their essential roles in the planning, organizing, and running of this conference. My thanks to Dr. Bruce Murray of JPL, Mr. Harvey Weisenfeld of ERDA, and Mr. Harvey Proctor of the Southern California Gas Company for giving us some of their valuable time. Ms. Dorris Wallenbrok's efforts in preparing the proceedings are very much appreciated.

Ram Manvi, Coordinator
Energy Conversion Systems
  Section
Jet Propulsion Laboratory

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WELCOME ADDRESS

B. C. Murray

Director, Jet Propulsion Laboratory
Pasadena, California

WELCOME ADDRESS

Dr. Bruce Murray
Jet Propulsion Laboratory

Here on the Caltech campus, in a part of the world which does not
use much coal directly, it is significant that there should be a gather-
ing representing most of the key industries, governmental and private
groups, and national labs concerned with one of the real bottlenecks in
coal utilization--coal feeding. The reason this is happening is three-
fold. The United States government has recognized the impending dislo-
cation of our economy and our way of life by the energy shortage and has,
I believe, made the correct decision to use all the power at its dis-
posal in an attempt to mitigate the effects of this coming transition.
Federal funds are being allocated, which is very good, though it presents
a dilemma, because this is the first time the United States government
has subsidized technological development in the area of high technology
where it is not its own customer.

JPL and many of the national labs which are represented here grew
up in an environment in which the government was its own customer: in
weaponry for World War II, for space (as, for example, JPL), or for
nuclear reactors or other major technological endeavors.

Now, we all, I think. recognize that we are in a different world.
But it is not yet clear how we go from here. The details are not yet
apparent as to how one uses federal subsidies. One would hope for en-
lightened tax policies and regulation policies to, in fact, induce
change in the private sector at the required rate.

We have not yet learned how to acquire a harmonious balance be-
tween the forces of the market place and the necessity for long-term
planning on the scale we are discussing. So in a way we are all experi-
menters--experimenting not just in the particular aspect of coal feed-
ing and coal processing, but in the more general aspect of how we evolve

a society to cope with a totally new sort of problem. Although I suspect there may be considerable anguish in the process, I do believe it is an area in which we can succeed, and we must.

The fact that we are on the Caltech campus indicates that the Institute itself is anxious about the future, and researchers who normally would be doing rather more esoteric things are worrying about why coal burns exactly as it does, or what the chemistry of coal is, and how it really interacts in various processing schemes.

JPL is a sponsor and coordinator of this because as a large technological institution, JPL also feels that anxiety. We sense the need to have some practical product of our activities as well as the dramatic and rather esoteric exploration of Mars, Jupiter, or Saturn.

My personal formula for JPL (incidentally, I have been there for only a little over a year, so we will see how good the formula is) is: the practical and pizzazz. Coal is certainly one of the most practical things we can work on, and indeed it has a very high priority. We are particularly pleased to be able to participate in a catalytic role, helping to bring people together, and we hope this kind of spread of interest continues to involve such diverse things as space laboratories that are in transition and industrial groups that have had the traditional problem of handling coal for years, as well as the national labs and various institutes associated with this.

. . . .

There will not be a coffee break this morning. My thought was that we had better use that time by opening this morning's session to questions and answers from the floor after the three speakers have given their overviews. So I would encourage you to think more in the terms of this being a participation meeting. It is a large meeting, and I think we need to structure it a little to make this happen, but we will be looking forward to hearing the overviews and looking for alternative or

counterviews which may be expressed from the floor, or at least questions
that clarify and further elaborate. The purpose of this conference is
to expose a variety of viewpoints, and we hope, new ideas or thoughts
that cause other people to have new ideas, not to reach a consensus.

On a national scale, the driving force in this area is the Energy
Research and Development Agency, and the key person in the area of
today's interest is Mr. Harvey Weisenfeld, Deputy Director of Major Fa-
cilities Project at ERDA.

I made a quick call this morning to find out a bit more about
Harvey's background. I don't know if ᴵ should tell you that he really
comes from the nuclear world, but I believe we recognize that the nu-
clear people have been struggling with large project problems for many
years, and their experience, again between the government and private
interface, is particularly relevant as the federal government's efforts
go into largescale and new areas. Harvey was educated ᴵt Drexel Uni-
versity. Preceding that, he was employed by Bechtel Corporation. He
is a member of the American Institute of Plant Engineers, and he is
listed in Who's Who in American Government.

It is with pleasure, then, that I introduce Harvey Weisenfeld.

WELCOME ADDRESS

H. Weisenfeld
Deputy Director, Major Facilities
Project Management Division

U. S. Energy Research and
Development Administration

Washington, D. C.

HARVEY WEISENFELD
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
WASHINGTON, D.C.

Thank you very much, Dr. Murray. I don't take any shame in being in the nuclear industry. I firmly believe that one of the key solutions to the Nation's crisis in energy lies with nuclear and fossil fuels; so I feel I am fortunate in being in both fields. It is a pleasure to be here this morning, and I thank you for the opportunity of addressing this important conference.

The return to coal as an important basic energy source is essential if the Nation is to survive the current energy crisis we are in today. To this end, the development of reliable and economic coal feed systems will go a long way to assure the successful use of coal, especially within the ERDA-sponsored programs, to bring coal conversion plants to a reality. As you may know, ERDA is endeavoring to commercialize large coal conversion plants by sharing the cost and the high technical risk with industry. The first step of this ERDA program to commercialization will be with a coal demon-stration plant program.

To put the importance of an effective coal feeding system for demonstration and commercial plants in its proper perspective, I'd like to relate to you some data that was released recently in Washington. Under the National Energy Plan, coal production will increase from the current 600 million tons per year to over a billion by 1985. Assuming about 2% of this coal will be

initially consumed by ERDA-sponsored programs, you can see that for each percent of savings in a demonstration plant, a savings of 200,000 tons of coal or an equivalent of one million barrels a year of oil would be realized. As we estimate, coal feed systems represent about 8 - 15 percent of the total coal demonstration plant capital cost. When we realize that each demo plant processes approximately more than 5,000 tons per day and costs about 400 million dollars each, it is easy to realize the significant savings and energy efficiency that could be realized. This potential savings, coupled with a further savings in a commercial plant program, which would process about 30,000 tons per day, at a cost of 1.2 billion each, will help tremendously if the Nation and ERDA are to meet the overall objectives of the National Energy Plan and reduce our dependence on foreign oil.

Our Division, Major Facility Project Management, has the responsibility for the Fossil Energy Demonstration Plant Program within ERDA. Our current program has either in design or the active planning stage eight projects representing a total Government investment of about 1.5 billion dollars. We expect industry will equally cost share and invest approximately this same amount. Since coal feeding systems will play an important role in the success of our program, we have undertaken a coal feeder development program with the following objectives: it should be capable of operating at high pressures; reliable, with minimum downtime; easy to maintain; and have low capital cost and low operating cost. Of course, this is all motherhood, ans as a result, I don't plan to speak too long because motherhood statements have a tendency to lose their impact.

At the present time, there appear to be only two systems that have the potential of operating at these elevated pressures above 500 psi. One is the slurry pump, and the other is the lockhopper system. Slurry systems do not appear desirable in coal gasification processes because the heat of vaporization loss costs the process 3 - 5 percent of its efficiency. Although lockhopper systems are available today, our experience indicates them to be costly both in initial cost and in operating cost. Therefore, dry feed systems appear to us to offer a solution that will meet the high pressure of 1,500 psi objectives.

We believe that an effective coal feeding program for our demo program, and eventually the commercialization program, can make significant break-throughs in the acceptance of these plants by industry for, after all, they will have the responsibility to invest the money to build these plants in the long run. We feel that this program could have a significant savings in the capital cost and the operating cost as well as the efficiency. We cannot stress too much its importance to our program.

This will conclude what I have to say, and I want to thank you very much for the opportunity to be here. It is a very exciting program that you are involved with, and I wish you the greatest success. Thank you very much.

AN OVERVIEW OF ERDA ACTIVITIES

J. Powell

U. S. Energy Research and
Development Administration
Washington, D. C.

OVERVIEW

J. POWELL

ERDA

It is a real pleasure to be here and be involved in this, the first but certainly no⁺ the last, conference directed to coal feeding systems. ERDA itself is hitting its third birthday, but it is doubtful that it will ever survive under that name and reach that milestone. A new Department of Energy appears viable. The House and Senate bills have passed, and it's now in conference and the ERDA functions will be and are planned to be assimilated within this new department. I am sure this information you know, but I think it is appropriate to mention here. From where we sit, there appears to be little change in the conduct of the coal feeder program once this marriage occurs. We anticipate there may be increased emphasis for the program, and we certainly hope so.

The problem with coal feeding against pressure heads is as old as the first rudimentary 1800 coal gasification systems. Even today, we have only a few approaches to accommodating the dry coal feeding problem, and here we are a hundred years later with almost the same type of problem. ERDA has recognized this problem, and we have had the present feeder program underway for about three years. At the present time, three contractors and two of the Energy Research Centers are involved, and of course there is some peripheral involvement by the others. I won't dwell on the results that we have achieved to date in this program but will let the active participants present our case for your evaluation, use, and comment.

Here are just a few truisms that help guide the program: All coal conversion processes require feeding systems. Each process has some peculiarities which require different feeding conditions. Dry solids transfer against pressure heads is definitely a challenge. State-of-the-art operational coal feeder systems are limited. A universal feeder design is not yet in sight, and it might never be.

Reliable feeders and feed systems will enhance the probabilities of achieving a synthetic fuel industry in this country within this century.

Major Facilities and Program Management, a division of Fossil Energy, which in turn is an administration of ERDA, have the responsibility for this coal feeder development. The feeder program has been proceeding from concept to bench, bench to pilot, and that is just about the point where we are now, then pilot scale on to demonstration and/or commercial use. Usage of any advanced feeders in the early demonstration plants early in the 1980's would be most gratifying and productive to all of us that are involved in this program, and also to the plant process economics, as Harvey mentioned earlier.

By your presence at this meeting, I believe all of you today know that you and we collectively have a stake in making these systems a reality. The program needs your involvement, and this conference provides all of us a good, solid starting point. Thanks again for your attendance, and I would like to meet many of you while I am here. Thank you.

AN OVERVIEW OF EPRI ACTIVITIES

H. Gilman

Electric Power Research Institute
Palo Alto, California

Ladies and Gentlemen

The Electric Power Research Institute is a non-profit research organization supported by the utility industry. The Fossil Fuels and Advanced Systems Division of EPRI is involved in the development of new coal processes. We are organized into program areas concerned with Coal Liquefaction, Coal Gasification, and Fluidized Bed Combustion. Associated with these processes are component development programs. Papers concerning EPRI projects in the development of coal feeders and slurry feed pumps will be given during this Conference.

The slides show a list of some of the accomplishments in these program areas that we anticipate during the years to come.

We are participating with ERDA and others in several coal liquefaction pilot plant projects such as the H-Coal Process, Exxon EDS Process, and the SRC Process at Wilsonville, Alabama. Future commercial liquefaction plants will require high volume slurry feed pumps and at that time it will be advantageous to use centrifugal pumps instead of the reciprocating pumps presently being used at the liquefaction pilot plants. EPRI has a project with the Rocketdyne Division of Rockwell International to develope a suitable high pressure, high volume centrifugal slurry feed pump. A description of

this project will be presented by Dr. Don Davis of Rocketdyne.

Feeders for gasification plants and pressurized fluid-
ized bed boilers are another equipment development area in
which we are focusing attention. We have a dry feeder project
with GATX in which we are exploring the concept of adapting
the commercial Fuller-Kenyon pump to a coal and limestone
injection feeder primarily for a pressurized fluidized boiler.
A presentation of this project will be presented during the
Conference by Dr. Fields.

EPRI also has a contract with General Electric for the
development of a high pressure coal and tar paste feeder for
a gasifier. The extensive development work on this project
will be presented by Dr. Furman of GE.

There is a great deal of emphasis today on process
development; but as operators know, if you can't make the
equipment work, or if men can't run the plants, the process
isn't worthwhile. It is important that equipment development
keep pace with the process work going on so that the necessary
machines will be available when the process is ready. In
recognition of this goal, EPRI enthusiastically endorses this
Feeder Conference.

I appreciate this opportunity to give you a brief over-
view of some of the program areas in progress at EPRI and to
introduce you to three of the projects concerned with develop-
ment of specialized coal feeders.

LIQUEFACTION

RESEARCH   AREAS

PILOT PLANTS & DEMONSTRATIONS

SRC - WILSONVILLE
SRC - DEMONSTATION PROGRAM
H-COAL DEMONSTRATION PROGRAMS

SUPPORTING DEVELOPMENT EFFORTS

COMPONENT DEVELOPMENT
MINERAL MATTER REMOVAL FROM LIQUIDS
CATALYST DEVELOPMENT
SRC BENCH SCALE INVESTIGATIONS
SRC ANALYTICAL SUPPORT
ENGINEERING EVALUATIONS

LONG RANGE PROJECTS

LIQUID PHASE METHANOL SYNTHESIS
$ZnCl_2$ CATALYSIS FOR BOILER FUEL
LIQUIFIED COAL UPGRADING
NEW PROCESSES-EXXON

GASIFICATION

RESEARCH   AREAS

SHORT RANGE TECHNOLOGY

LURGI TEST FACILITY- COMBINED CYCLE
ADVANCED FIXED BED:
GEGAS
BGC SLAGGING GASIFIER
HOT GAS CLEAN-UP -
SYSTEM DEVELOPMENT
GENERAL PLANNING:
PROCESS ANALYSIS
ENGINEERING EVALUATIONS
COED CHAR GASIFICATION &
COMBUSTION

ADVANCED GASIFIERS

COMBUSTION ENGINEERING GASIFIER
BABCOCK & WILCOX GASIFIER
TEXACO
FOSTER/WHEELER

EQUIPMENT DEVELOPMENT

o ANTICIPATE ULTIMATE COMMERCIAL NEEDS

o ENCOURAGE INDUSTRY PARTICIPATION

o PROVIDE FRONT-END RISK CAPITAL FOR NEW AS

o EVALUATE EQUIPMENT OPERATING EXPERIENCE AND
   NEED FOR DESIGN IMPROVEMENTS
   - o PUMP & FEEDERS
   - o SOLIDS DISTRIBUTORS
   - o ASH REMOVAL EQUIPMENT
   - o HOT GAS CLEAN-UP
   - o COMMINUTION
   - o PNEUMATIC TRANSPORT
   - o VALVES
   - o MATERIALS HANDLING
   - o FILTERS

FEEDER DEVELOPMENT AREAS

Dry Solid Feeder

    o Low Btu Gasification
        o Moving Bed
        o Entrained
        o Fluidized Bed

    o Fluidized Bed Combustion
        o Atmospheric
        o Pressurized
        o Fast Fluid Bed

Paste Feeder - (formed with tars)

    o Low Btu Gasification
        o Moving Bed - Lurgi
    o Hydrogen manufacture for liquefication
        o Ash Laden Column BTMS

FOSSIL FUEL AND ADVANCED SYSTEMS

DIRECTOR
R. Balzhiser

ASS.JTANT DIRECTOR FOR EXTERNAL AFFAIRS
G. Hill

ASSISTANT TO DIRECTOR
R. Zeren

ASSISTANT TO DIRECTOR FINANCE AND ADMINISTRATION
G. Mader

MATERIALS SUPPORT
R. Jaffee

FOSSIL-FUEL POWER PLANTS DEPARTMENT	ADVANCED FOSSIL POWER SYSTEMS DEPARTMENT	ENERGY MANAGEMENT AND UTILIZATION TECHNOLOGY DEPARTMENT	NEW ENERGY RESOURCES DEPARTMENT
Director, G. Hill	Director, D. Spencer	Director, F. Kalhammer	Director, P. Bos (acting)
Air Quality Control K. Yeager	Power Plant Requirements and Analysis R. Loth	Energy Storage T. Schneider	Fusion W. Gough
Water Quality Control and Heat Rejection J. Maulbetsch (acting)	Clean Gaseous Fuels N. Holt	Fuel Cells and Chemical Energy Conversion Technology A. Fickett	Geothermal V. Roberts
Fluidized Combustion and Coal Cleaning G. Ehrlich	Clean Liquid and Solid Fuels R. Wolk	Energy Utilization and Conservation Technology O. Looney (acting)	Solar J. Cummings (acting)
Fossil Plant Performance and Reliability D. Anson (acting)	Power Generation Gas Turbines MHD V. Cooper		

Figure 1 The ranges of busbar power cost for coal fuel technologies are most easily compared in grou; s four options for direct coal firing,two liquefaction processes, and four gasification processes In each group, the present -or most nearly developed—option is at the left Thus, among the direct-firing options each results in successively more costly power than power from low-sulfur coal without controls Among the gasification processes, however, the more advanced—and, at the moment, more uncertain   versions produce markedly cheaper power

AN OVERVIEW OF AGA ACTIVITIES

D. Crawford

C. F. Braun & Co.
Alhambra, California

AN OVERVIEW OF A.G.A. ACTIVITIES

I have been asked to appear before you this morning to present an overview of the activities of the American Gas Association in the field of coal conversion, with particular reference to those activities which pertain directly to coal feeding systems, since that is the main subject of this conference. Perhaps I should begin by telling you a bit about the American Gas Association and its current involvement in coal conversion technology.

The American Gas Association is an association of companies involved in the US gas industry which includes gas transmission and gas distribution companies. The current membership includes over 300 companies, representing about 90 percent of all gas sales.

On August 3, 1971, an agreement was entered into between the Office of Coal Research (OCR) of the US Department of the Interior and the A.G.A. to provide for the organization, joint operation and mutual funding of an accelerated program of coal gasification research, covering the first phase of the program through the pilot plant stage. Under this agreement, the A.G.A. would contribute up to $10 million per year, commencing in the 1972 fiscal year, to match, on a 1/3 to 2/3 ratio, Federal funding appropriated through Congress. On this basis, total joint funding of $30 million annually would be provided. The agreement was to cover a period of eight years from the date of signing.

In January 1975, the OCR was removed from the Department of the
Interior and incorporated within a new agency, the Energy Research
and Development Administration, which had been established to
unite all Federal energy research and development programs
within one agency. Included in these research and development
programs was the OCR-A.G.A. joint program, now referred to as the
ERDA-A.G.A. Coal Gasification Pilot Plant Research Program.

The Joint Program currently encompasses five different major coal
gasification processes, four basically different methanation
schemes, one major project for hydrogen generation, and related
technical evaluation, materials development and commercial scale
planning work. In addition, close technical liason is being
maintained with the Synthane project, although this project is
being funded solely by ERDA.

Since neither ERDA nor the A.G.A. has a technical staff sufficiently
large to monitor all the various aspects of this major cooperative
effort with the degree of thoroughness considered essential, the
agreement and the subsequent guidelines provided for a single
Technical Evaluation Contractor to provide the required in-depth
technical evaluations and reviews of all contract research projects
and proposals. In March 1972, C F Braun & Co was awarded a

contract as the Technical Evaluation Contractor. In this role,
Braun acts as a technical extension of both the ERDA and the A.G.A.
staffs, with the major responsibility of closely monitoring the
technical aspects of every project, providing complete and
thorough interchange of technical information between and among
all of the contractors, and recommending prompt termination of
any project which does not meet the program objectives. In
addition, Braun recommends processes, systems and equipment from
the various pilot plants as the basis for the best overall
design of commercial demonstration coal gasification plants.

As an employee of Braun, I have the responsibility for the design
and evaluation of coal handling, storage, size reduction, drying
and feed preparation equipment and systems as they relate to
coal gasification processes. And it is with this brief background
that we can begin to discuss the activities of the A.G.A. in the
areas of interest to us.

As I mentioned previously, the ERDA-A.G.A. joint program is
sponsoring work on five different major coal gasification
processes, as well as keeping in touch with a sixth process.
These processes are all designed to produce pipeline quality gas
from coal. For the purposes of our discussion here, we can, I

believe, define the term "pipeline quality gas" as gas which will be interchangeable with natural gas. Bulletin 36 of the A.G.A. gives the complete acceptability criteria for this product.

Production of liquid and gas fuels from coal has been around for a long time. A man named Faur built a gas producer in Germany as early as 1832, and Mr F Bergius was studying the liquefaction of coal between 1910 and 1917. After the close of World War I, the I G Farben syndicate began working on coal liquefaction, and had a commercial plant in operation in 1927. Fisher and Tropsch were studying catalytic reduction of carbon monoxide to various hydrocarbon liquids between 1923 and 1933. In the field of coal gasification, the first commercial Lurgi plant was built in 1936. The Winkler process was developed in Europe over 50 years ago, the Wellman-Galusha process has been commercial for over 30 years, and the Koppers-Totzek process was developed commercially in 1949. It can be seen from this that coal gasification and liquefaction technology is not a recent development. We usually refer to the Lurgi, Winkler, Wellman and Koppers-Totzek processes as "first generation" processes. These four processes have all been proven as operating processes. However, none of them have operated on a commercial basis in the United States for the production of high Btu gas from US coals.

In the joint program, ERDA-A.G.A. has funded the development of
processes which we will refer to here as "second generation"
processes.   These processes include, in no particular order,

    IGT HYGAS, Steam-Oxygen
    IGT HYGAS, Steam-Iron
    Consol CO_2 Acceptor
    BCR BI-GAS

The Battelle Agglomerating Ash project had been included in this
group, but was transferred to ERDA's Coal Conversion and
Utilization program, effective August 20, 1974.

As I mentioned earlier the joint program has maintained close
technical liason with the Synthane project, and has completed
tests on four US coals in a modified Lurgi gasifier at Westfield,
Scotland.   For the purpose of this report, then, we will consider
five processes as "second generation" - HYGAS Steam-Oxygen, HYGAS
Steam-Iron, CO_2 Acceptor, BI-GAS, and Synthane.

These processes were developed by various groups and process
developers, and commercial projections have been studied.   Pilot
plants have been constructed and operated for various lengths of
time, and with varying degrees of success.   I've mentioned the
Lurgi gasifier tests at Westfield, which successfully
demonstrated that American coals could be gasified in a modified
Lugi gasifier, and that the medium Btu gas normally produced

could be methanated to high Btu pipeline quality gas and
interchanged with natural gas. In addition, IGT has had a 70-ton
per day pilot plant operating on the HYGAS Steam-Oxygen process
since 1974, and Consol has been operating a 40-ton per day pilot
plant on the CO_2 Acceptor Process since mid-1974. The ERDA
Synthane pilot plant started up for its first trial run in March.
Construction of the IGT HYGAS Steam-Iron pilot plant has been
completed, but it is still in the process of starting up, and
has not yet had a trial run.

These processes have also been studied by Braun as part of our
work under the technical evaluation contract with ERDA-A.G.A.
Our studies have included, among others, the following.

   Commercial concept designs for each process in order to
     estimate the capital and gas costs for each process, using
     different coals.

   Safety assurance studies.

   Process trade-off studies.

   Areas and specific items of equipment which may require
     mechanical development.

Many of our studies have already been published, including studies
on

Packed Bed Reactor Shapes.

Preliminary Coal Grinding Tests for the Design of Coal Grinding
Systems in Coal Gasification Plants.

Rotary Feeders.

Factored Estimates for Western Coal Commercial Concepts.

For those of you who are interested, Report FE-2240-12 is a
published index of Braun reports.

Let us take a more detailed look at these second generation
processes in which we are interested. In second generation
processes the coal which is to be gasified is fed into some type
of fluidized bed or entrained bed within a reactor vessel.
Generally second generation processes are designed to operate at
elevated temperatures and pressures, with the result that both
process equipment and operations are required to conform to new
parameters.

To refer specifically to areas in which these new parameters
have affected both operations and equipment, let us consider the
coal feed requirements for these processes. Without going into
specific details here, it can be generally said that second
generation coal gasification processes require coal reduced to
a finer particle size, with specific upper and lower size

distribution limits. The first generation processes, with the exception of Koppers-Totzek, generally operate with a graded lump coal, or a coarser particle size. Another area which affects operations is the requirement for continuous injection of the feed coal into vessels operating under high pressures and temperatures. In several cases, these second generation processes operate at temperature above 1500° F and at pressures of 1000 psig or above. Once a reactor vessel is brought up to operating conditions, the process should continue at those conditions.

These requirements, then, of continuous feeding of small, carefully sized coal particles at substantial tonnages, into a high temperature and pressure environment, are the reason for our attendance at these sessions.

Let us consider the actual coal feed requirements for the various second generation processes, as shown in Table 1. I have previously referred to a series of factored estimates of these processes recently completed by Braun for ERDA-A.G.A. This work was done for various processes, using different coals, in order to establish relative orders of magnitude of the cost of pipeline quality gas produced by these processes. In order to establish a common datum or base, the factored estimates were made

GASIFIER FEED REQUIREMENTS
TABLE I

TYPES OF COAL	HYGAS STEAM-OXYGEN		CO2 ACCEPTOR		BI-GAS	
	Tyler Mesh	Wt %	Tyler Mesh	Wt %	Tyler Mesh	Wt%
WESTERN SUBBITUMINOUS	+8m -8m +100m -100m	0 85 min 15 max	-8m -100m	98 10 max	+100m -100m	1 99
EASTERN BITUMINOUS	+8m -8m +100m -100m	0 85 min 15 max	N.A.		+100m -100m	1 99
LIGNITE	N.A.		-8m -100m	98 10 max	N.A.	

TYPES OF COAL	SYNTHANE		LURGI		HYGAS STEAM-IRON	
	Tyler Mesh	Wt %	Tyler Mesh	Wt %	Tyler Mesh	Wt %
WESTERN SUBBITUMINOUS	-6mx0	100	-1 1/4"x9m	100	+8 -8m +100m -100m	0 85 min 15 max
EASTERN BITUMINOUS	+20m -20 +200m -200 +325 -325 m	1 69 30min min	-1 1/4"x0 -28m	99 10 max	+8m -8m +100m -100m	0 85 min 15 max
LIGNITE	N.A.		-2"x1/4"	100	N.A.	

on the conceptual design of a commercial scale plant. The

hypothetical plant was sized to produce 250 billion Btu per day

of pipeline quality gas. This is, of course, equivalent to 250

million SCF per day of gas with a value of 1000 Btu per SCF.

The quantity of coal required to produce this gas varies with the

type of coal used and with the process. It would also vary with

other factors, such as the coal moisture content and the actual

size consist, but in general we are talking in terms of between

500 and 1000 tons of coal perhour for our definition of a

commercial scale plant. This means that the commercial scale

plant will require between 12,000 and 24,000 tons of coal per

day for continuous operation and that is a substantial quality.

For purposes of comparison, a 550 to 600 megawatt coal-fired power

plant will consume about 250 tons of coal per hour, or 6000 tons

per day.

Each one of these processes has a particular requirement for the

coal feed, not only in terms of tonnage, but also in terms of the

coal particle size consist and moisture content. The figures

given in Table 1 for the feed size consist for each of the

processes were used in Braun's development of the factored estimates.

These figures were provided by the various process developers, and

represent the best knowledge available at the time the figures

were released. As you will notice, these various requirements each have two controls - control of absolute top size, as expressed by "0% plus X-mesh", and control of the percentage of fines. Even though the different processes may define "fines" as a different size, all the specifications represent an effort to limit the percentage of undersize particles to quantities well below those which would normally be produced in a coal size reduction operation.

Control of the top size of coal particles must be accomplished because particles larger than the optimum size will require a longer residence time in a gasifier environmen to become completely reacted. Information currently available from at least two pilot plants, Synthane and HYGAS Steam-Oxygen, indicate that both processes are reducing their top size requirements. Synthane is in the process of reducing their feed size from 20 mesh to 40 or 50 mesh, with the -20 +40 fraction recycled to the grinding mill. HYGAS reports that "pretreatment operation was dramatically improved by removing the +14 mesh coal from the feed to the pretreater." This particular test was with a bituminous (Illinois #6) coal, and subsequent tests have used this finer feed on both bituminous and subbituminous coals.

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Control of the percentage of fines must be accomplished because fine particles will not remain in the gasifier environment, but will be elutriated overhead into the process areas. If these particles have not been completely reacted, because of their short residence time in the gasifier, the process will have a reduced efficiency. In addition, quantities of fines carried downstream will overload the subsequent solids-gas separation systems, and cause severe operating problems. Pilot plants have reported that fines carryover has been a problem, at times causing downstream scrubbing sections to be nearly inoperable.

As we can see, the coal feed requirements for these second generation coal gasification processes are quite restrictive. Large quantities of pulverized coal, with a closely controlled size consist which may or may not be rigorously defined at this time, will be required for commercial scale plants.

Once the required tonnages of coal have been produced in the required size consist, it then becomes apparent that this coal feed must not be changed by the mechanism by which it is injected into the gasifier vessels. The various pilot plants, to which we have already referred, have taken different approaches toward resolving the problems of feeding coal into high temperature and

pressure gasifier vessels. In the HYGAS pilot plant coal is
slurried with a light oil and pumped into a vessel at pressure.
The slurry oil is vaporized, to be recovered and returned to the
feed unit. One of the papers to be presented this afternoon will
deal with this subject in more detail. The BI-GAS pilot plant also
uses a slurry feed system, although here the coal is slurried with
water, pumped into a flash dryer at pressure, and then gravity fed
into the gasifier.

The synthane pilot plant uses a lockhopper feed system and we will
hear more about this particular system in a paper which will also
be given this afternoon. The CO_2 Acceptor pilot plant also uses
lockhoppers, but at a lower pressure.

If the properly-sized particles are further broken down, excess
fines will be created. We have already seen that fines created
by the feed preparation method are going to cause process
problems, and any excess fines created in the feed injection
process can only add to that problem. By the same token, any
agglomeration of particles which will cause oversized particles
to be injected will create a different set of problems, but they
will still be problems. In addition to these problems, we also
have the specific problem of exactly how to move a substantial

quantity of pulverized coal across a temperature and pressure barrier continuously, without affecting the process on the downstream side of the barrier. In brief, then, the problems of coal feed injection into gasifier vessels may be stated as follows.

1 Continuous feeding of a large tonnage of pulverized coal from ambient temperature into a high temperature environment.

2 Continuous feeding of a large tonnage of pulverized coal from atmospheric pressure into a high pressure environment.

3 Continuous feeding of a large tonnage of pulverized coal without changing the established particle size range or size consist during feeding unless these changes can be accurately controlled.

These problems are real problems, and are of real concern to those of us working in the field of coal gasification. We feel that a successful solution or solutions to these problems will materially affect our efforts to develop and construct operating coal conversion plants. These problems have been recognized for some time, although I cannot say that efforts have been underway to solve them since they were first recognized. In a few instances, the original process work included "black box" feeders at the critical points, with the hope that someone else would

determine the configuration and content of these black boxes by
the time they were needed. This approach has seldom worked in
other areas, and is not likely to work on coal gasification.

One of the early projects assigned by the ERDA-A.G.A. Joint
Program to Braun was to identify the various areas of commercial
scale coal gasification programs which might require mechanical
development. In a series of reports, written in mid-1973 and
recently updated, Braun included a report recommending mechanical
development of dry coal feed injection systems. In particular, it
was recommended that parallel mechanical development programs be
undertaken for such items as lock-hopper systems, piston feeders,
extruders and screw pumps. I quote from that report.

"The development work should be aimed initially at technical
feasibility only. Further development of commercial scale
designs should be made only after (1) it has been demonstrated
that (a) particle size and fluidity of the feed thus injected
is not unacceptably altered by the injection equipment and (b)
that equipment life and continuity of operation will be
reasonable, and (2) an economic study has been made to cull
those designs that appear to be unduly expensive."

Several ERDA contracts are currently active in these areas, and we are looking forward with a great deal of interest to the reports of these investigations.

SESSION I

PROCESS OPERATIONS EXPERIENCE

EXPERIENCE IN FEEDING COAL INTO A
LIQUEFACTION PROCESS
DEVELOPMENT UNIT

S. Akhtar

S. Friedman

N. J. Mazzocco

P. M. Yavorsky

U. S. Energy Research and Development
Administration, Pittsburgh Energy
Research Center
Pittsburgh, Pennsylvania

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ABSTRACT

Thiry years of experience in coal liquefaction at PERC provides con-
siderable background technology on feeding coal into pressurized units.
Performance of the preferred feed system as now used serves as a basis
to compare new developments. Flowable coal-oil slurry is fed by positive
displacement plunger pumps into high-pressure liquefaction units. Records
of over 600 hours of continuous unit operations without repairs to the
feed system attest to its reliability. SYNTHOIL process development oper-
ations with slurries of various concentrations, coal, and recycle vehicle
oils will be reviewed. Methods for minimizing settling, plugging and
erosion will be discussed.

"Experience in Feeding Coal into a Liquefaction Process Development Unit"

Sayeed Akhtar, Sam Friedman, Nestor J. Mazzocco and Paul M. Yavorsky
Energy Research and Development Administration
Pittsburgh Energy Research Center
4800 Forbes Avenue
Pittsburgh, PA 15213

INTRODUCTION

A system for preparing coal slurry and feeding it into a high-pressure liquefaction plant has been in use at the Pittsburgh Energy Research Center (PERC) for over thirty years. The system was developed for the liquefaction units operated at the Center during the 1940's to provide supporting research and development for the Bureau of Mines coal lique-faction pilot plant (60 TPD) at Louisiana, Missouri. Although operation of the Missouri pilot plant was discontinued in 1953, research and develop-ment on coal liquefaction has continued almost uninterrupted at PERC and the coal feeding system is still essentially the same with minor materials modifications. The experimental units currently in use for research on production of low-sulfur, low-ash utility fuel oils by SYNTHOIL and other processes are equipped with this coal feeding system. It is the purpose of this presentation to describe the system which has been used for long con-tinuous runs and to review the experience of operating it.

DESCRIPTION OF THE SYSTEM

The system for preparing coal slurry and pumping it into a high-pressure liquefaction plant is shown in figure 1. It is based on positive displace-ment pumping of coal slurried in a recycle vehicle oil. It consists of three parts: A slurry mix tank where the coal slurry is prepared, a slurry feed tank that maintains the suspension of slurry during feeding, and a slurry feed pump which pressurizes the slurry into the pressurized vessels of the plant. Of paramount importance is the circuit used to keep slurry moving closely by the intake valves of the high-pressure pump to prevent any solids settling in transfer lines to the pump. The slurry mix tank, shown to the left in the figure, is mounted on a scale and is equipped with an agitator and a circulating pump. The tank is connected to an inert gas supply so that the slurry preparation may be conducted under an inert gas blanket. All connections to the tank have flexible hose couplings to enable accurate transmission of weight changes to the scale.

The circulating pump is a commercial* progressing cavity Moyno pump[1]. The convoluted rotor and stator are made of tool steel and the rotor has a

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*Reference to commercial items does not constitute endorsement by the U.S. Government but is for identification purposes only.

chrome finish. Neither part is plastic lined; there is metal-metal contact. The capacity of the pump is 3gpm which gives a velocity of 1.2 ft/sec to the slurry in the circulation loop of 1-inch ID pipe. This is adequate pipeline velocity to assure suspension of the minus 100-mesh coal in the oil.

The slurry feed tank, shown to the right of the slurry mix tank, is also mounted on a scale and is equipped with an agitator and a circulating Moyno pump. Like the mix tank, the feed tank also is connected to an inert gas supply and has flexible hose endings for all connections.

The slurry feed pump, shown to the right in figure 1, is a modified Milton Roy plunger-type metering pump[2]. The modification of the head, along with other details of the pump assembly, is shown in figure 2. The pump is a duplex unit with a common drive. The plunger is attached to the crosshead by a floating connection which permits lateral movement and assures alignment of the plunger with the axis of the stuffing box. A metal gland and a lantern ring provide close guidance of the plunger to prevent excessive wear of the packing. The plunger is made of low-carbon steel and is plated with chromium, 0.002-inch thick. Both the stuffing box and the plunger have surface finishes of 60 µ-inch or better. The plunger withstands slurry wear very well as seen by the good condition of the one photographed in figure 3 after 6000 hours of use.

The packing is chevron or "V" type, self sealing, and fabricated from teflon. Lubrication of the packing is essential to reduce friction. Grease lubricant is injected with a grease gun at least once every day.

The suction and discharge valve assemblies are our own modifications of the pump. The ball valves were designed at PERC. The ball and seat, assembled as a unit, are screwed into the pumphead and seated against a gasket. The arrangement permits easy replacement of the valve, when necessary. The ball is made of a hardened chromium steel and the seat of stainless steel 410, hardened to about 350 Brinell. The pump is provided with pressure relief valves, one directly in the pumphead on the discharge side and one in the piping on the suction side. The relief valve on the suction side is necessary to protect the low-pressure piping from over pressurization if the ball valves on both the suction and discharging sides become faulty.

The pump is provided with a gas vent in the cylinder body to bleed out air when priming the pump. The same vent is also used to bleed out organic vapors or steam if the pump is vapor locked by volatiles from the vehicle oil or moisture from coal.

The capacity of the pump has been varied from 2 lb/hr to 70 lb/hr by using one or both sides of the pump, interchanging plungers of 1/4-inch to 5/8-inch diameter, and adjusting the length of the stroke from 1/2-inch to 4-inches. As an additional means of varying the capacity, the pump is equipped with a variable speed drive but it has been used at only one speed, corresponding to a frequency of 28 strokes/minute.

The piping for the feed system and the safety features in it are shown in figure 1. The high-pressure line from the feed pump to the plant is provided with a reverse flow check valve to prevent the plant pressure from blowing back if the ball valve on the outlet of the pump fails to

seat properly. The high-pressure line is also provided with a pressure senstive cut-off switch to interrupt supply of power to the feed pump if the line, or any point downstream of it, developes a constriction. A manually operated oil pump, shown on the extreme right in figure 1, is then used to hydraulically break open the constriction by applying pressures up to 10,000 psig. To minimize the risk of constriction developing in the slurry line due to settling of coal, horizontal piping is avoided and the length of the line is kept as short as practicable. To facilitate preparation and feeding of very viscous slurries, the system can be steam heated. The mix and feed tanks are steam-jacketed and all pipings involved in slurry transfer are steam-traced. The head assembly of the feed pump is provided with 1/4-inch holes for inlet and outlet of steam.

OPERATION OF THE FEED SYSTEM

The vehicle oil is transferred to the mix tank and the agitator and circulating pump started. Pulverized coal is added, manually in small quantities through the open top while the tank is blanketed with inert gas (nitrogen) by maintaining an outward flow of ne gas. After the addition of coal is complete, the tank is covered and pressurized with 5 psig of the inert gas. Agitation and circulation of the slurry is continued for 2 hours to ensure thorough mixing, after which the slurry is transferred to the feed tank by diverting the circulation stream to the tank. Agitation and circulation of the slurry is continued non-stop in the feed tank also under 5 psig of the inert gas. The slurry is supplied to the feed pump from the circulation loop as shown in figure 1. The slurry is supplied at about 6 psig or higher to ensure an adequate flow of the slurry to the suction side of the feed pump. A gauge in the circulation loop measures the pressure of the circulating slurry. A pressure of less than 6 psig on the gauge indicates that the rotor and stator in the circulating pump are eroded and should be replaced. The erosion rates of the rotor and stator depend on the type and amount of mineral matter in coal. With a West Virginia coal containing 3 percent ash, the rotor and stator had a service life of 4,000 hours while with a Kentucky coal containing 16 percent ash, the service life was reduced to 1600 - 1800 hours.

This system has been used with a wide variety of coals and vehicle oils: coals ranging in rank from low-volatile bituminous to lignite and vehicle oils from anthracene oils and coke oven tars to process-derived recycle oils. The coal is dried and pulverized to 70 percent thru 200-mesh, U. S. standard seive, and 99 percent thru 100-mesh. Immediately before use, the pulverized coal is screened through a 50-mesh sieve to reject particles retained by this sieve and to break up agglomerated clusters of pulverized coal.

Drying of coal is of no fundamental significance to liquefaction per se. However, wet coals are difficult to grind, and pulverized coal containing more than 10 percent moisture generates foam when added to recycle oil at 200 ° F or more. The nuisance is most pronounced with subbituminous coals and lignites, which usually retain about 20 percent moisture after conventional drying and grinding. If foaming is excessive, a good portion of the slurry in the mix tank can spill over to the floor. Excessive foaming is controlled by adding the coal slowly and in small quantities to the recycle oil. It is important to add the coal in small quantities for another reason also. If added too rapidly, the coal particles agglomerate into internally dry lumps

which do not break up easily. These lumps usually float on the oil but occasionally are sucked into the circulating loop and plug up the valve at the bottom of the tank or freeze the rotor of the circulation pump. A suitable rate for adding coal to oil without foaming or lumping must be determined empirically since the wetting characteristics of coals and oils differ. Frequently, some pulverized coal agglomerates stick to the walls of the tank. As a routine practice, therefore, when the addition of coal is complete, the walls are scraped to break up the coal agglomerates and the scrapings added to the slurry.

The slurry must be homogenized thoroughly in the mix tank before its transfer to the feed tank. An improperly mixed slurry can easily cause malfunction of the ball valves on the feed pump. Two hours of mixing time is liberal allowance to ensure thorough mixing, although, with many coal-oil combinations, a far shorter duration may be sufficient.

The slurry preparation and feeding are conducted in an inert atmosphere. This precaution is desirable since the viscosity of coal-derived oils, and therefore of slurries of coals in such oils, increases by exposure to air. An investigation[3] of the aging characteristics of product oils from the SYNTHOIL process has shown that the rate of increase of viscosity under different gases varies in the order nitrogen < air < oxygen, and at different temperatures in the order 86° F < 113° F < 141° F. Furthermore, the higher the initial viscosity of the oil, the larger the rate of increase of its viscosity. Since slurry preparation and feeding are frequently conducted at about 200° F and the initial viscosity of the slurry is high, the rate of increase of the viscosity will be substantial if the slurry is prepared and fed exposed to air. It may also be noted that the agitation and circulation of the slurry in the mix and feed tanks are extremely vigorous, and if slurry preparation and feeding were conducted in air, the air-slurry contact would be far more intimate than under the experimental conditions of the aging study. Consequently, the rate of increase of the viscosity of the slurry would be larger than the results of the aging study might suggest.

CONCENTRATION OF COAL IN FEED SLURRY

Maximization of coal concentration in the slurry is a desirable objective since the higher the concentration of coal, the higher the plant throughput for a given slurry feed rate. However, the slurry must be pumpable, a quality determined by its viscosity and stability. If the viscosity is too high, the slurry cannot be circulated or introduced into the feed pump, while if the viscosity is too low, coal will segregate from the slurry and deposit in the inlet and outlet valves of the feed pump and/or downstream of the pump in the high-pressure line to the plant, before hydrogen meets the slurry and coal starts dissolving. Any of these occurrences will disrupt the feeding system.

The viscosity and stability of a coal slurry depend on the nature of the vehicle oil and the concentration of coal in the slurry. Although all coal liquefaction processes are based on process-derived oils, the latter are not all identical. The vehicle oils in some processes are distillate fractions of the liquefaction product, while in others, the distillation bottoms or the whole product oil itself. The vehicle oils in various coal liquefaction processes, their viscosities and the concentration of

coal in pumpable slurries are given in table 1. The vehicle oils consisting of distillate materials are low-viscosity liquids that give pumpable slurries with up to 33 percent coal. At higher coal concentrations, the slurries are too unstable although viscosity is no problem. The vehicle oils consisting of heavy oils, or whole product oil, are viscous liquids containing asphaltenes and some benzene insolubles. These liquids give pumpable slurries with up to 45 to 50 percent coal. At higher concentrations of coal the slurries are excessively viscous. Indeed, if the viscosity of the vehicle oil is more than 100 SSF (at 130° F) and slurries containing more than 35 percent coal are to be prepared, the slurry preparation and feeding system must be heated. The practical upper limit of temperature for our feeding system is about 230° F, at ve which volatilization of the low-boiling hydrocarbons in the vehicle oil causes vapor-locking of the feed pump.

OPERATING EXPERIENCE

The experience of operating the coal slurry preparation and feeding system with the 1/2-TPD SYNTHOIL unit[4,5,6] over the past three and a half years is summarized in table 2. Every run made with this unit is listed. The pump was always newly overhauled prior to each run. (See footnote to table 2 for details of overhauling). During this history of operation, slurries of five different coals in indigenous recycle oil, a solution of SRC in SRC solvent, and the SYNTHOIL product oil without any coal were processed through the SYNTHOIL unit in the 26 separate runs of table 2. Seven of these runs were of 500 to 700 hours duration, ten of 200 to 425 hours, and nine of 8 to 169 hours. None of these runs was terminated because of feed pump failure. One side of the duplex pump was always ready to run whenever the other side needed overhaul.

The pumping time with each head during the runs is tabulated in column 6 of table 2. When one side of the duplex pump falters as observed by any decrease of feed rate, the other side is put into service and the defective side is overhauled by changing the pumphead ball valves and packing assembly (see figure 2) and cleaning the inlet manifold with light oil. The overhauled side of the pump is then available on standby. The ball valves and the pressure relief valve from the defective pumphead are later cleaned and examined. If not scarred, they are ready for reuse; if scarred they are replaced with new balls and valve seats. Most of the time, only valve cleaning is needed.

In conclusion, note from table 2 that runs of 500 hours and 616 hours duration have been accomplished with a single side of the pump assembly, requiring no overhaul or switchover to the other side of the pump in that time. Also the switchover and overhaul technique, without interruption of a run can be practiced indefinitely for very long continuous runs.

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TABLE 1.- Vehicle oils and concentrations of coal
in feed slurries in various liquefaction
processes

Vehicle oil*	Viscosity	Approx. max. concentration of coal in pumpable slurry	Processes
Distillate of boiling range 400°-700° F	10-15 centi-stokes at 140° F	33	SRC, H-coal
Hydrogenated distillate of boiling range 400°-700° F	Unknown	33	Pott-Broche, CSF, Exxon
Heavy oils (distillation bottoms), deashed	10-50 SSF at 180° F	50	Bergius, U. S. Bureau of Mines coal liquefaction R & D
Whole product oil, deashed	5-400 SSF at 180° F	45[7]	SYNTHOIL

*All vehicle oils are process-derived materials.

TABLE 2. -Coal slurry pumping history for all runs of the 1/2-TPD SYNTHOIL Unit

Coal	Concentration of coal in feed paste, wt pct	Slurry feed rate, lb/hr	Plant pressure, psig	Total Pumping time, both heads, hr	Pumping time between overhauls*
Hvab, Homestead Mine, Kentucky	35	25	4,000	500	0-249 hrs right side, 249-259 hrs left side, 259-456 hrs right side, 456-500 hrs left side.
do do	35	25	2,000	500	0-130 hrs right side, 130-500 hrs left side.
Mvb, Clearfield County, PA	35	25	4,000	36	0-36 hrs right side, no overhaul.
Hvab, Homestead Mine, Kentucky	35	25	4,000	400	0-53 hrs right side, 53-114 hrs left side, 114-500 hrs right side.
Hvbb, Herron Mine, Illinois	35	30	4,000	30	0-30 hrs left side, no overhaul.
Hvab, Homestead Mine, Kentucky	35	50 (both sides of pump)	4,000	100	0-100 hrs both sides on, no overhaul.
do do	35	30	4,000	200	0-151 hrs left side, 151-200 hrs right side.
Hvbb, Spencer County, Indiana	35	25	4,000	20	0-20 hrs right side, no overhaul.
do do	20	25	4,000	8	0-8 hrs right side, no overhaul.
Hvab, Ireland Mine, W. VA.	35	25	4,000	500	0-1 hr left side, 1-500 hrs right side.
do do	35	25	2,000	300	0-300 hrs right side, no overhaul.
do do	35	25	4,000	500	0-500 hrs right side, no overhaul.
do do	35	25	2,000	169	0-281 hrs right side
do do	35	18	2,000	112	no overhaul.
do do	35	25	4,000	375	0-375 hrs right side, no overhaul.

TABLE 2.-Coal slurry pumping history for all runs of the 1/2-TPD SYNTHOIL Unit-
Continued

No coal. Rehydrogenated product oil from a previous run		35	4,000	34	0-34 hrs right side, no overhaul.
Hvab, Ireland Mine, W. VA.	35	25	4,000	200	0-200 hrs right side, no ov..rhaul.
50 wt pct SRC in SRC-Solvent		25	4,000	96	0-128 hrs right side,
Pure SRC-Solvent		25	4,000	32	no overhaul.
Hvab, Ireland Mine, W. VA.	35	25	4,000	712	0-320 hrs left side, 320-712 hrs right side.
Hvab, Sinclair Mine, Kentucky	35	25	4,000	279	0-279 hrs right side, no overhaul.
do do	35	25	4,000	169	0-39 hrs left side, 39-169 hrs right side.
do do	35	25	4,0C0	425	0-78 hrs right side, 78-94 hrs left side, 94-425 hrs right side.
do do	35	25	4,00C	365	0-365 hrs right side, no overhaul.
do do	35	25	4,000	621	0-616 hrs right side, 616-621 hrs left side.
do do	35	25	4,000	98	0-98 hrs left side, no overhaul.
do do	35	25	4,000	226	0-226 hrs left side, no overhaul.
do do	35	25	4,000	564	0-12 hrs right side, 12-158 hrs left side, 158-302 hrs right side, 302-564 hrs left side.

*The overhaul consists of changing the pump head that includes the valves and changing the packing; also cleaning the feed inlet manifold with light oil. The ball valves and the pressure relief valve of the removed pumphead are thoroughly cleaned for reuse. If valves are scarred, balls and seats are replaced.
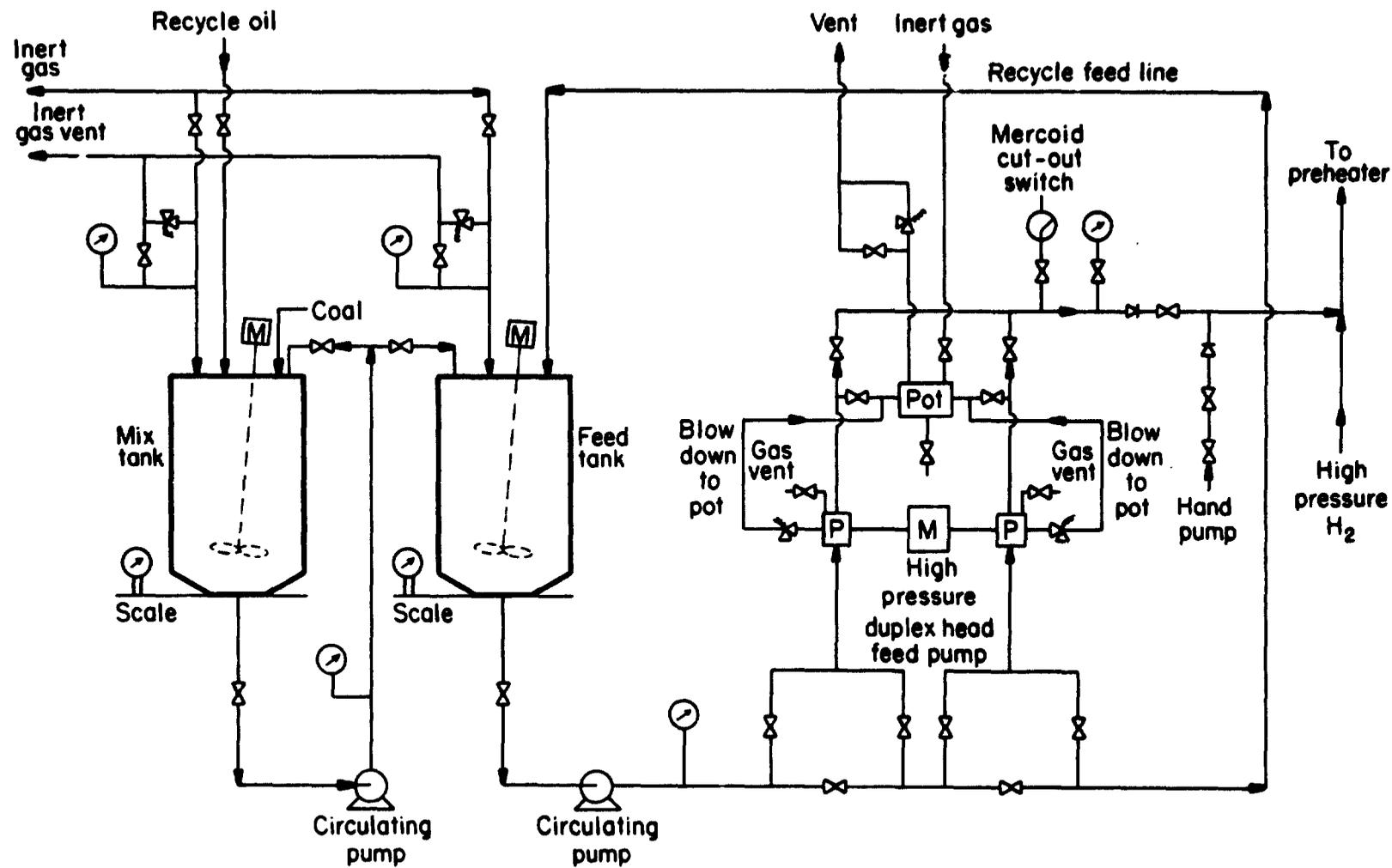
Figure 1- Coal slurry preparation and feeding system.

Safety valve

Crosshead

Floating rod

Discharge

Gland

Vent plug

Grease port

Steam
jacketed

Ball type
check valves

Suction

"Vee" packing

Lantern ring
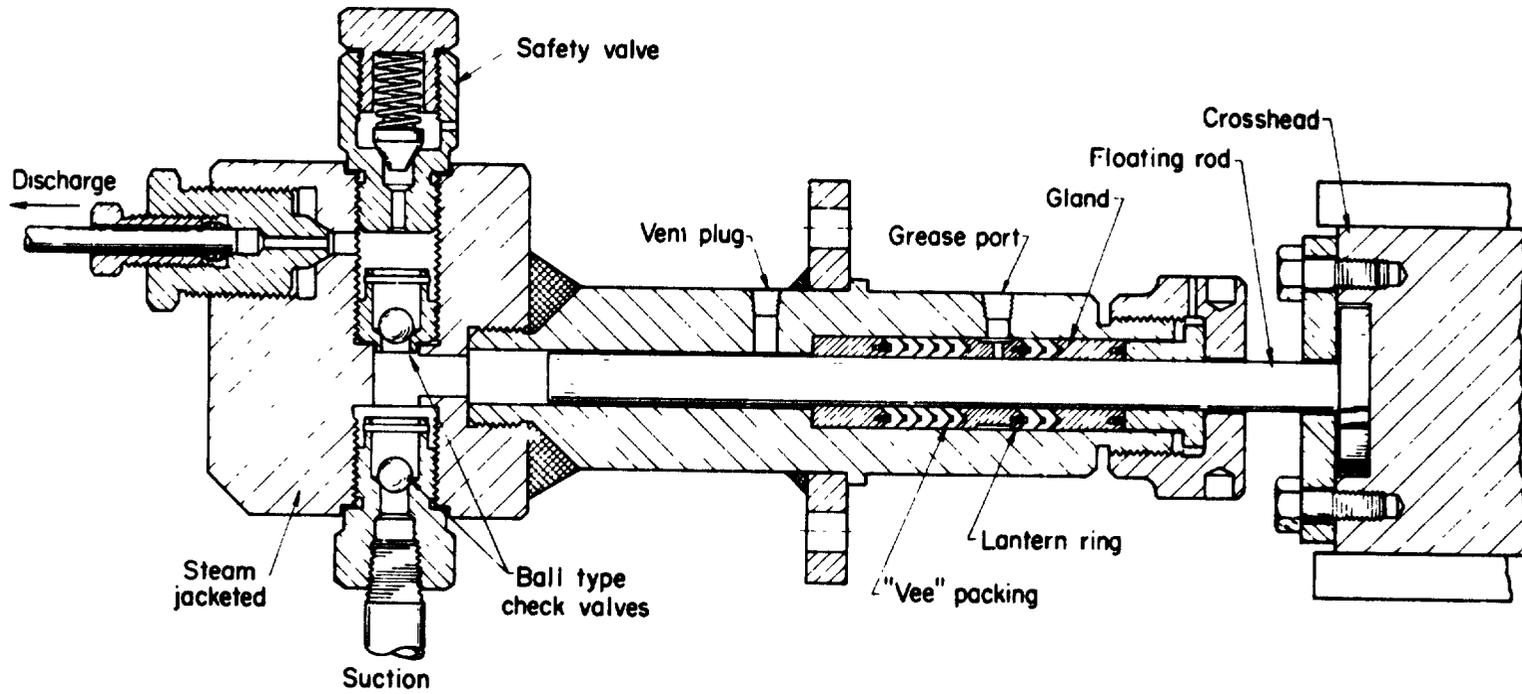
Figure 2-High pressure feed pump assembly
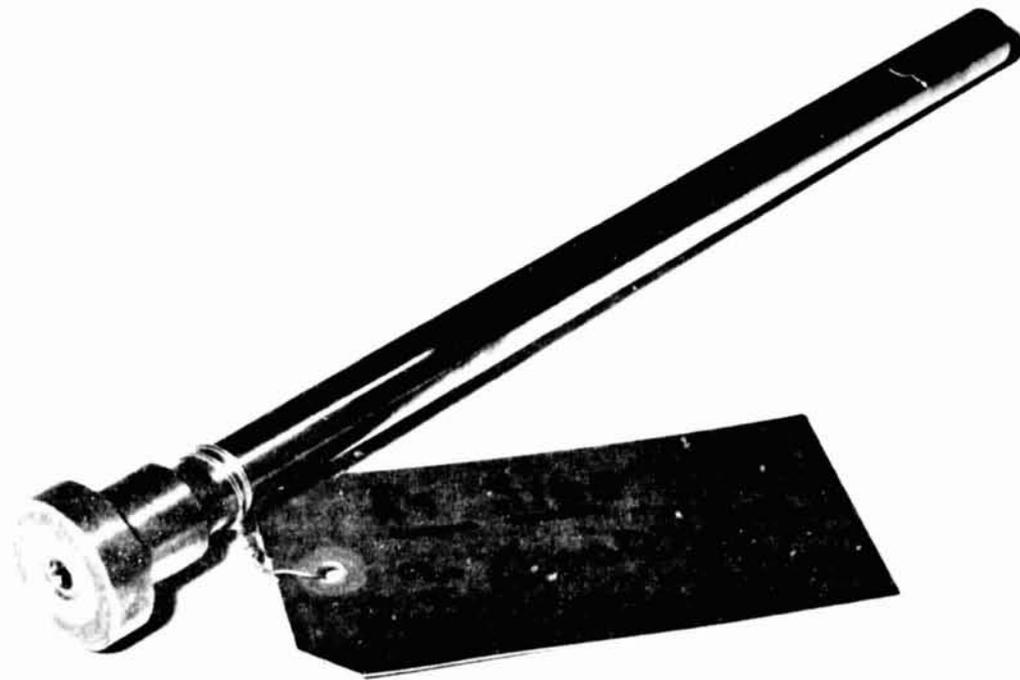
Figure 3-Photograph of the pump plunger after 6,000 hours of use.

THE PETROCARB PNEUMATIC FEEDING
SYSTEM--A PROVEN METHOD FOR
FEEDING PARTICULATE SOLIDS AT
CONTROLLED RATES

H. Reintjes

Petrocarb, Inc.
New York, New York

THE P E T R O C A R B PNEUMATIC FEEDING SYSTEM — A PROVEN METHOD FOR FEEDING PARTICULATE SOLIDS AT CONTROLLED RATES

by Harold Reintjes, President
    PETROCARB, INC.
    New York, New York

ABSTRACT

The Petrocarb Injection System is a pneumatic feeding system having proven capability for feeding solids at controlled rates into processes which operate up to 60 atmospheres pressure. A single system can be provided which can feed a single or multiple feed points with substantially equal distribution between the various feed lines. The system is completely automatic and is normally supplied with a solids rate control system.

A summary description of the system elements is presented together with an outline of the principal features of the system.

The paper refers to early development which started approximately twenty years ago and outlines various commercial applications.

The history of successful applications and experience leads to the conclusion that the Petrocarb Injection System is capable of feeding dry solids into most of the processes being developed for utilizing coal.

1. INTRODUCTION

Petrocarb has supplied Injectors for feeding particulate solids at controlled rates since 1959. Most applications of this early technology were in the iron making industry where hundreds of units are in use today on a world-wide basis.

Some of the basic requirements for the feeders used in these applications are that they:

a) be capable of supplying constant and reproducible flow rates for short intervals of a few minutes duration
b) have the ability to feed against back pressure such as developed by a head of several feed of molten iron
c) be available in a range of sizes and be capable of feeding solids at rates from a few pounds per minute to several tons per hour
d) can be easily and quickly stopped and started
e) use high solids to gas ratios in order to conserve expensive gases such as argon and to minimize metal splash from submerged lances.

Technological developments of the early 1960's prompted the introduction of more sophisticated custom designed systems capable of feeding against pressures of over 60 atmospheres, such as may be encountered in coal gasifiers and combustors, at rates up to about 150 tons per hour in a single, shop fabricated feed system. Early applications required only a single feed line but as technology was advanced, units having multiple lines, feeding continuously, were developed and commercialized. A brief description of a basic Petrocarb Coal Injection System, together with some applications, follows.

2. DESCRIPTION OF PETROCARB INJECTION SYSTEM

Please refer to Figure No. 1, which is a simplified flow diagram of a typical system for feeding against intermediate pressures. Modifications of this system are sometimes used depending on the particular application. The Petrocarb Injection System provides continuous and automatic injection of prepared solids into a process reactor at any designated rate, distributed uniformly among the feed lines. The Primary Injector is the heart of the system. This unit is a pressure vessel of proprietary design with multiple feed outlets, one for each injection nozzle of the reactor. Each of the multiple feed outlets from the Primary Injector has an individual injection air line, equipped with both gas and solids flow indicating instruments. After initial adjustments are completed, substantially equal quantities of solids will be delivered to each nozzle.

The total injection rate is a direct function of the differential pressure between the continuously pressurized Primary Injector and the point of delivery of the solids. The characteristic is stable, smooth, and repeatable, thereby providing a means of controlling solids feed rate without introducing variable restrictions or other equipment in the solids stream, with their attendant problems. The Primary Injector is mounted on load cells which provide necessary signals for a weight rate control system. In automatic mode, the desired feed rate is automatically maintained as set by the operator at the control panel. The operator can override the weight rate control action by placing the system in manual mode and setting the differential pressure controller set-point manually. In some installations the more sophisticated provision for automatic resetting of the rate control instrument is unnecessary.

Above the Primary Injector is the Storage Injector, which automatically replenishes the solids fed from the Primary Injector without interrupting or disturbing the injection process. This functions as a lock hopper. Lock hoppers, traditionally, have been somewhat unreliable for the same reasons that one cannot assume that a vessel or storage bin, provided with an inlet and outlet, assures that solids will flow from the vessel when the outlet valve is opened.

Our Storage Injector overcomes such problems, and is an integral part of the system. When the Storage Injector is emptied of material, it is automatically depressurized and refilled either by a Feed Injector located at grade level below a prepared materials storage bin, or from an elevated storage bin. When a Feed Injector is used, material from the storage bin is fed and conveyed to a receiver-filter above the Storage Injector where the solids and conveying gas are separated. Solids flow is by gravity into the Storage Injector and the filtered transport gas is discharged to the atmosphere.

The size of the solids batch, mechanical equipment, valves, and time increment for each function are all designed and coordinated to provide feed to the process, up to the maximum design rate, while maintaining a reserve capability in the Primary Injector. All operations are automatic, being supervised and controlled by a specially designed logic system. Solids flow in any feed line can be stopped or started by the manual operation of a remote switch for that line. Also the flow can be stopped automatically in an individual line or any desired multiple thereof by programmed process requirements or system safeguards.

A section from a strip chart recorder showing the net weight of coal being fed from a Primary Injector mounted on load cells, with the system functioning in the equivalent of the manual mode, is shown in Figure 2. Note the smooth line with only one irregularity which was caused by automatic stopping of coal feed for a brief period because of a process requirement. The feed rate can be commanded to follow a downstream process signal such as temperature, bed level, or production rate.

3. APPLICATIONS

Let us now look at some of the applications of the Petrocarb Injection System where the process application demands continuous feed under automatic control.

Petrocarb pioneered the first commercial sized blast furnace coal injection system at the Hanna Furnace Corporation No. 2 Furnace in 1961. This was a merchant iron plant and economics unfortunately dictated the closing down of this furnace. The coal injection operation was described in a paper [1] by Strassburger et al in 1962.

In 1963 the National Coal Board in England purchased a Petrocarb Coal Injection System which was installed at the Stanton & Staveley Ltd. No. 5 Blast Furnace. This system was operated successfully for about three years and was the subject of a prize winning paper by E. M. Summers at the AIME Iron Making Conference in 1964. [2] The installation and operation were technically very successful, but unfavorable economics resulting from the relative costs of coal and fuel oil at the time, plus the discovery of oil in the North Sea, prompted the discontinued operation of the coal injection system and abandonment of a plan to install coal injection systems on blast furnaces throughout the U.K. The injection unit was dismantled and moved to Leatherhead where it has been used in test operations by National Research Development Corporation, London, England, for feeding coal in a pressurized fluid bed combustor experimental program. A portion of this work was sponsored by the U.S. Office of Coal Research and the British Coal Utilization Research Association Ltd. (BCURA) and is covered in an OCR Report. [3]

Since 1967 two zinc slag fuming furnaces have been in substantially continuous operation at Port Pirie, South Australia, using the Petrocarb Injection System. The furnaces require dependable feeding of pulverized coal which serves a dual role of providing heat, and acting as a reducing agent in the process. Each of these two furnaces has thirty (30) submerged tuyeres which are fed with coal from a single injection unit (one for each furnace). The most difficult problem

in the design of this system was to cope with the feed rate of three to
five pounds per minute in each of the thirty lines. A larger system would
have been less difficult to design. This operation was the subject of a
paper by I. D. Brett presented in 1968. [4]

Petrocarb's Injection System is used in a modern copper-nickel plant located
in Botswana which uses the Outokumpu Flash Smelting Process. Pulverized coal
is being fed to multiple burners to produce heat for the process. It is
essential that the coal injection equipment used to feed coal into the burners
provide smooth, steady flow and uniform distribution among the burners. In
this facility, the coal feed rate is automatically controlled by a single
rate-setting on the control panel. The rate control instrumentation auto-
matically adjusts the instrument primarily responsible for the delivery rate
of coal. The net weight of coal in the Primary Injector is continuously recorded
on a strip-chart so that the operator has visual evidence of the performance of
the feeding system.

A copper and nickel smelter complex being constructed in Russia, which also
uses the Outokumpu Flash Smelting Process, will utilize Petrocarb Injection
Systems for feeding pulverized coal to the process. This application i
interesting in that the coal is to be used as a reducing agent for converting
sulfur dioxide in the high temperature exit gas from each of two smelters (one
copper and the other nickel) to elemental sulfur which will be condensed and
recovered in waste heat boilers.

The application of Petrocarb Injection Units for high feed rates has been fully
demonstrated in a multiple unit installation in Venezuela where very abrasive
iron ore is fed at the rate of about 77 tons per hour in each unit against a
pressure of three atmospheres. As a result of this installation, hardware has
been developed which has good abrasion resistance. Reference is made to the
units in Venezuela as the reason for the selection of the Petrocarb System by
The Lummus Company for use on the Synthane Coal Gasification Project in a paper
by R. T. Whitehead in 1974. [5]

The use of coal as a replacement for oil and gas in the production of electric
power, and the production of pipe-line gas from coal, is a subject with which
we are all concerned. Many different processes are being investigated. While
the basic approach to solving the problems associated with each process is quite
varied, there is one common denominator — each must have a dependable coal
feeding system.

Petrocarb is involved with major pilot projects as the supplier of solids
feeders dealing with

        a) Pressurized fluid bed combustor development,
        b) MHD reactor development, and
        c) Coal gasification

An example of the latter is the Synthane Process for high pressure coal
gasification being developed by the Pittsburgh Engineering Research Center
(ERDA). This application is the subject of a paper by Mr. Robert Lewis et al. [6]

With reference to pressurized fluid bed combustor projects, Petrocarb is
supplying coal and dolomite feed systems to Curtiss-Wright on its current
contract with ERDA, and has been awarded a contract to supply the coal
dolomite feeding system for the Grimethorpe Project in England which is
being co-sponsored by ERDA, the National Coal Board of England, and West
Germany.[7]

4. STATUS OF FEEDER DEVELOPMENT

Petrocarb's current position is that it can provide dependable feed systems
using existing knowledge and equipment components without large scale development
work. We believe our feeder designs will be able to keep up with any realistic
requirements of the coal utilization program without large scale government
funding. These systems can readily be scaled up to large throughputs and in-
corporate components which for the most part are currently being manufactured
or utilize proven designs. Certainly there is a requirement for specialized
know-how in the design of a dependable feeding system. Such far ors as solids
to gas ratios, feed rates, reactor pressure, particle size and particle size
distribution, flow characteristics of the solids, and moisture content, all
enter into the design of a dependable feed system. Furthermore there are no
known published data which could be utilized to treat all of such variables
and permit one to design a successful installation. Fortunately, Petrocarb's
experience (as partially related above) plus a considerable amount of development
work carried out in its own test facility permits it to confidently claim the
position as stated above.

PETROCARB CONTINUOUS INJECTION SYSTEM
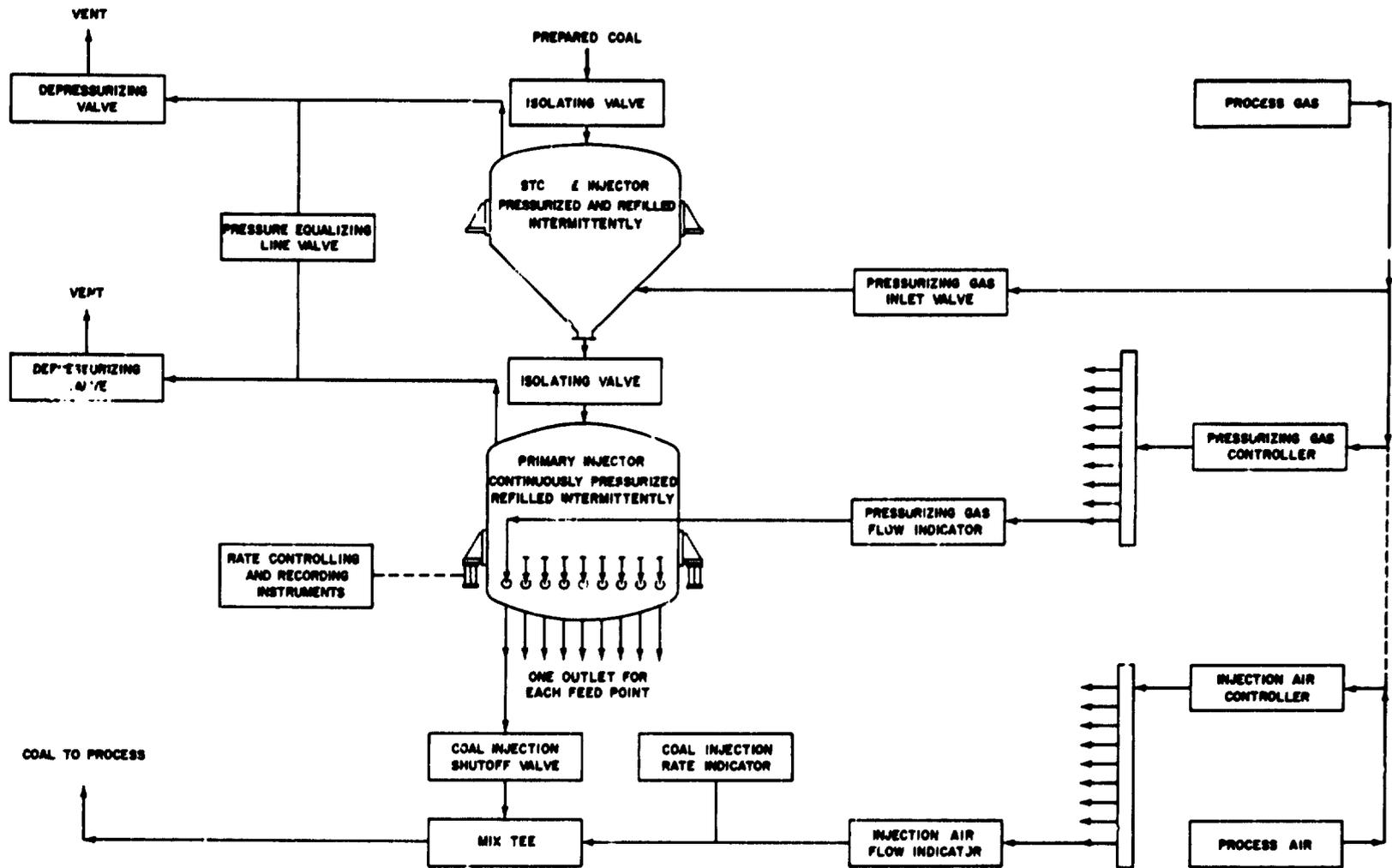
Figure No. 1 - Simplified Flow Diagram of a Typical Petrocarb System for feeding against intermediate pressures.

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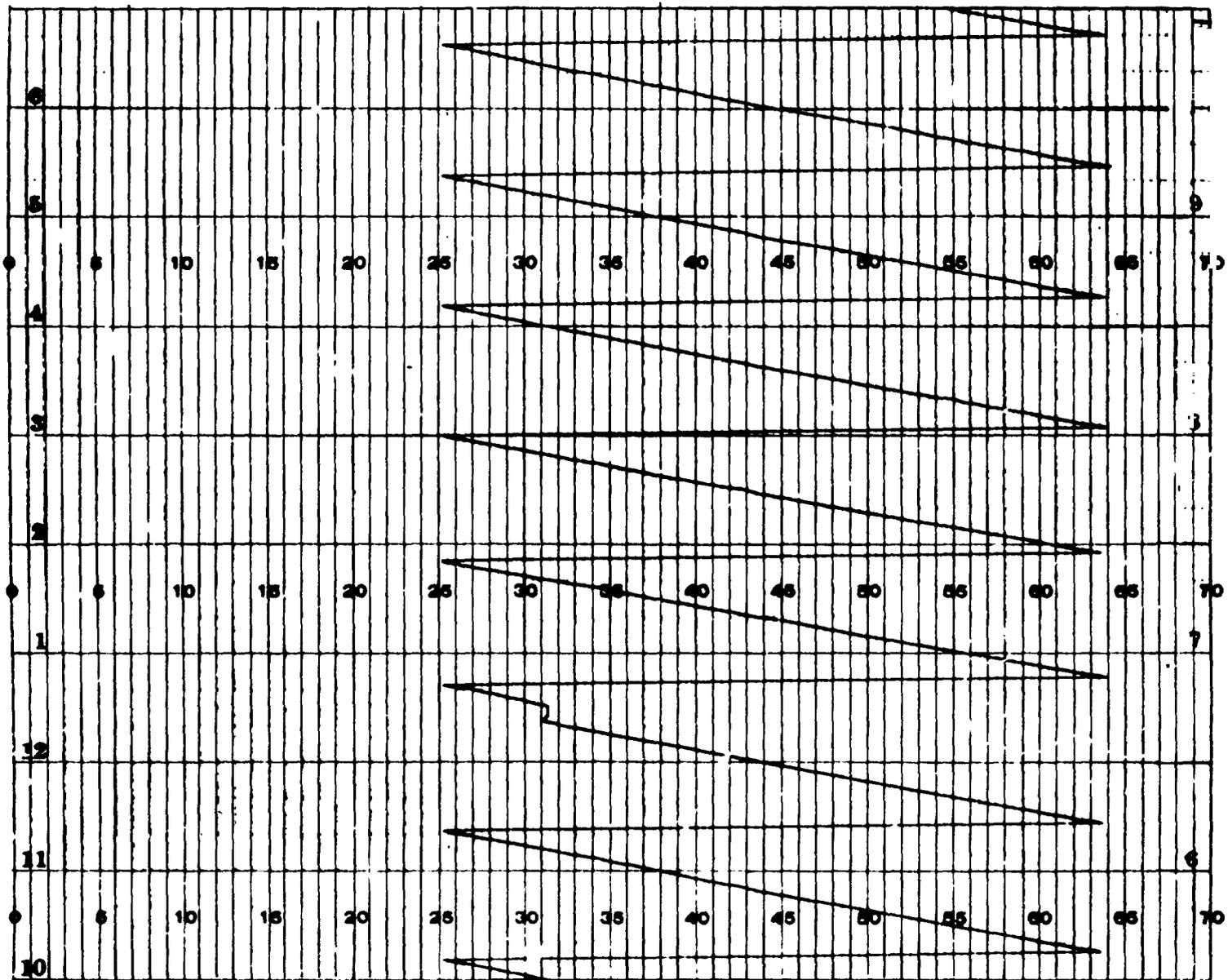
Figure No. 2 - Section from a strip chart recorder showing the net
weight of coal being fed from a Primary Injector mounted
on load cells.

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COAL PRESSURIZATION AND FEEDING--
USE OF A LOCK HOPPER SYSTEM

R. Lewis

R. R. Santore

D. Dubis

U. S. Energy Research and Development
Administration, Synthane Pilot Plant
Pittsburgh, Pennsylvania

COAL PRESSURIZATION AND FEEDING **N78-13244**

USE OF A LOCK HOPPER SYSTEM

INTRODUCTION

The SYNTHANE process is a high pressure coal gasification system developed by the Pittsburgh Energy Research Center of the U. S. Energy Research and Development Administration (E.R.D.A.), formerly a part of the U. S. Bureau of Mines. It was designed to convert bituminous coal, subbituminous coal and lignite into a satisfactory substitute for natural gas with a heating value of 950 BTU's per cubic foot. A 72 ton per day SYNTHANE Pilot Plant has been constructed in South Park Township near Pittsburgh, Pennsylvania.

A necessary preliminary step in high pressure coal gasification processes is to take ground coal at atmospheric pressure and feed it to the gasification system at operating pressure. For reasons that are not pertinent to this report a decision was made to use lock hoppers at SYNTHANE. Accordingly, a proprietary system was purchased from Petrocarb, Incorporated. This system was designed to feed coal at pressures up to 1000 psig at rates of 1.67 to 5.0 tons per hour.

This report will discuss some of the specific problems experienced with the operation of the Petrocarb system at the SYNTHANE Pilot Plant. It will also review the modifications made to improve its performance.

SUMMARY

The SYNTHANE gasifier has been operated periodically from July, 1976 to the present time (January 31, 1977). During this period more than 750 tons of coal have been fed to it through the Petrocarb system. Including functional tests, this represents approximately 1000 cycles on the single train and 500 cycles on the dual train portions of the system.

After an initial testing and shakedown period the performance of the Petrocarb unit is considered satisfactory. With certain limitations it can be depended upon to feed coal to the gasifier within the design feed range. For a system of this degree of mechanical complexity, maintenance is reasonable.

Major problems encountered during the initial testing and operation have been eliminated. Brief interruptions of coal feed will probably continue to occur occasionally due to the difficulty of completely eliminating minor breakdowns.

DESCRIPTION OF SYSTEM

The system consists of a weigh hopper, two lock hoppers (storage injectors), a feed hopper (primary injector), interconnecting piping, valves, instruments and controls, all furnished by Petrocarb, Incorporated (see Appendix A-1, Drawing E438-A-007 Rev. 6). It is designed for automatic operation and operates on a demand basis triggered by a low level probe in the primary injector. The system may also be operated in a manual mode.

Coal enters the system at atmospheric pressure from the pulverized coal storage bin (FE-103). Batches of up to 2500 pounds drop by gravity to the weigh hopper (FE-106). The weigh hopper is mounted on load cells and weighs each batch and then allows it to fall through open valves into an empty lock hopper (FE-104A or B). The lock hopper is then isolated by closing the inlet valves. Pressurizing gas (SYNTHANE uses CO_2) is introduced into the lock hopper to raise its pressure to equal that of the pressurized feed hopper (FE-105). Valves below the full lock hopper then open to allow the batch to fall by gravity into the pressurized feed hopper. The empty lock hopper is again isolated by closing the outlet valves. The gas is cross-vented to the second lock hopper which is at atmospheric pressure and has now been charged with coal. When the pressures of the two lock hoppers have essentially equalized the cross-pressurization valves are closed and the remaining gas in the empty lock hopper is vented. Additional gas is then pumped into the full lock hopper to raise it to system pressure and the cycle is repeated using alternate lock hoppers. The cross-pressurization procedure reduces the total amount of gas needed to pressurize a lock hopper but some gas is unavoidably wasted.

When coal feed to the gasifier is required, the primary injector discharge valve (XCV-26) is opened and coal flows continuously from the primary injector into the coal conveying line to the gasifier. Carbon dioxide is used as the conveying medium. The coal feed rate for any given transport line velocity is controlled by varying the pressure differential between the primary injector and the gasifier. Carbon dioxide required to maintain the system pressure enters the primary injector through nozzles located just above the discharge valve. These gas inlet nozzles are designed to keep the coal partially fluidized and continuously flowing. Coal flow may be stopped at any time by closing the primary injector discharge valve.

EXPERIENCES

Prestartup

The system was installed at SYNTHANE during 1974 but testing was not started until July, 1975. Poor storage practices during construction and the amount of time the system remained idle before use caused many problems during the initial pressure testing.

Valves

Valves have been more troublesome than any other single component. Correction of valve malfunctions in the Petrocarb system has required more time and effort than any other problem.

With one or two exceptions all valves in the Petrocarb system are ball valves. Due to the experimental nature of the plant various materials were specified for the balls used in the Petrocarb valves. The most prevalent material used is 440C stainless steel but some balls were coated with tungsten carbide (LW-IN40) or ceramic (LC-4). The larger 6 inch and 8 inch valves were all hard face coated and have not been a problem. The smaller 2 inch through 4 inch gas handling valves have balls of all three materials. For valves of this size we have obtained no better service with the hard face coated balls and have therefore standardized on the cheaper 440C stainless steel for the ball material. All valve seats are stellite on 316 stainless steel and show little or no wear (see Appendix A-2, Cutaway Valve Sketch).

During the first pressure test numerous valves leaked and several would not rotate. Inspection showed that these valves were obstructed with construction debris or the balls and seats were frozen in place due to corrosion of the steel bodies caused by moisture. All these valves were removed and made serviceable by cleaning, relapping of balls and seats or replacing badly corroded balls and seats.

Methods of reducing the corrosion of the carbon steel valve bodies have been investigated. In conjunction with the valve manufacturer three methods have been suggested and are being considered: (1) valve bodies can be produced of a material less susceptible to corrosion, such as stainless steel, (2) bodies can be electroless nickel coated or (3) an internal body sleeve can be installed in a critical area.

It has been determined that high gas velocities and rapid valve actuation in venting and cross-pressurization service caused many valve malfunctions. Phenomena such as spring retainer wear and deformation, axial ball and stem movement, retraction and cocking of seats, all of which contribute to valve leakage, can be attributed to the above factors.

Actuator speeds were appreciably reduced. The valves were initially opening or closing in less than two seconds. This time factor has now been increased to 10 to 15 seconds for all the valves in question. Vent orifice diameters have been reduced and orifices have been relocated down stream of the vent valves. After these two changes were made most problems with these valves have been eliminated.

Two separate failures of shafts twisting off the balls have occurred. Initial investigation indicated a crack started at the root of a sharp corner and may have been deepened by corrosive attack. Subsequent additional torsional loads applied by the actuator in rotating the ball ultimately caused failure.

At the time the Petrocarb system was purchased it was felt that valve fail-ures due to erosion might be frequent, causing delays in plant operation. An investigation was conducted into state-of-the-art valves and new and novel valve designs[1]. However, after seven months of operation and a system throughput of over 750 tons of coal there have been no valve failures due to erosion. The performance of the 6 inch and 8 inch coal handling valves has been particularly satisfactory. This can probably be attributed to: (1) the fact that these valves are purged with CO_2 before opening and closing and (2) the valves open and close against no differential pressure.

Other system modifications have been made in an effort to improve operation. Piping configurations have been changed to eliminate areas where moisture can condense and be trapped. Isolation valves were installed in the vent lines to separate the two lock hoppers. If one lock hopper system fails, it can now be isolated and repaired while the other lock hopper is still operating.

Electro-Mechanical Components

The Petrocarb system requires numerous pressure switches, relays and micro-switches to function, particularly in an automatic sequencing mode. We have been unable to operate this system consistently in an automatic sequencing mode because of frequent malfunctions of the mechanical or electrical com-ponents. On occasion, only one train of lock hoppers has been used. The fact that the system is capable of continued service in spite of component malfunctions is considered an advantage.

Initially, individual batch weights of coal in the weigh hopper were in-accurate and inconsistent because the load cells were very sensitive and were affected by many physical factors. Adjustments to the load cells and re-vision to supports of piping and vessels corrected the problem. Accurate and repetitive weights are now regularly obtained.

The weight totalizer[2] has not functioned properly. This may be due to electrical interferences because of the unit's location. Arc suppressors have been installed in an effort to eliminate this problem[3]. In addition, batch weights are now being totalized by the data acquisition computer.

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[1] A considerable amount of information was obtained and several valves were purchased and are on hand. A separate report on this subject will be issued in the near future.

[2] Totalizes and indicates the total weight of individual weigh hopper batches.

[3] At the time of this writing these revisions have not been tested.

Operations

This system was designed to feed dry ground coal to the gasifier. Operation has been interrupted frequently by two recurring problems, both of which cause plugging of the discharge venturi below the primary injector.

(a) **Foreign Material.** Although the vessels were originally cleaned during start-up operations, construction and maintenance debris has continued to come through, lodging in the venturi and interrupting coal flow. A screen was installed in the weigh hopper in an effort to correct this problem.

(b) **Wet Coal.** This problem has been eliminated by disconnecting the Petrocarb vent from a common vent line to the thermal oxidizer. This vent was originally shared by Petrocarb and the fluidizing steam discharge from the char cooler. It was found that steam from the char cooler entered the lock hoppers and the dry coal storage bin through the vent valves and condensed in these vessels, wetting the coal. A separate line from the Petrocarb system to the thermal oxidizer is being installed.

This system was designed to feed coal at a maximum rate of five tons per hour with a turn down ratio of 3 to 1. To date, we have been unable to achieve turn down rates below 2-1/2 to 1, or approximately two tons per hour. Rates in the 2 to 5 tons per hour range have generally been found to be reproducible and reliable. Rates below two tons per hour have been erratic. The current range is adequate for Montana subbituminous coal. However, when operating with caking coals requiring pretreatment, lower feed rates such as 1-1/2 tons per hour may be required. Discussions were undertaken with Petrocarb, Inc. to determine what changes might be necessary to achieve lower rates. Petrocarb's position is that a feed rate of 1.67 tons per hour should be attainable with the present system. At feed rates below two tons per hour wide fluctuations in the fluidizing gas to the primary injector have been noted. Petrocarb feels this may be limiting the turn down ratio. They have suggested improving the control of this fluidizing gas and recommended fine tuning of the various control instruments involved.

FEATURES OF THE SYSTEM

The Petrocarb system feeds dry ground coal by entraining it in a pressurized transport gas stream. In the gasification unit the coal can be separated from the transport gas without any additional energy input.

When the system is operating in an automatic mode it requires little operator attention. When operating in a manual mode, some additional operator attention is required. The fact that it can be run in a manual mode is considered an advantage.

Parallel lock hopper trains conserve on pressurizing gas and also provide some redundancy allowing for continued operation when one train is inoperable.

The Petrocarb unit contains many electrical and mechanical components
necessitating a continual maintenance effort. When in the stand-by mode,
normally between runs, daily cycling of valves and pressure testing of
the system before start-up is required. If kept idle for longer periods
of time additional rechecking of the instrumentation is also necessary
to assure reliable performance.

For the Petrocarb unit installed at SYNTHANF we have been unable to find
a means of instantaneously measuring the coal flow rate. Therefore, feed
rates are obtained by averaging batch weights over a period of time.

Coal flow rates are not absolutely constant and are estimated to fluctuate
approximately $\pm 5\%$.

As previously discussed, coal feed rates can be controlled within a given
range. The feed rate range is primarily a function of the pressure differ-
ential between the primary injector and the gasification unit. The rate is
controlled within the design limits by varying this pressure differential.
Petrocarb has stated that the best way to lower the operating range and still
retain high solids to gas ratios, with a reasonable line velocity, is to
change the line size in preference to introducing point restrictions or in-
creasing gas flow. It is therefore inconvenient to change the range because
a new diameter feed line would have to be installed.

This type of feeding system consumes considerable energy, mostly in the com-
pression of gas, a significant portion of which is lost through venting,
leakage and passage of gas out with the coal.

Capability of scale-up is inherent in this type of system. Larger size com-
ponents are commercially available.

90° operation – ¼ turn with manual lever. Gear
operator, Pneumatic, Electric, or Hydraulic
Actuators.

Ball Stem.
One piece ball and stem
provides maximum strength
& ease of operation.

Full Port.
Maximum flow at minimum
pressure drop.

Ball. Chrome, Nickel, or
Stellite surface lapped
against the suitable seating
material.

Body and ball – in various
materials suitable for up to
10,800 psi and 1200°F.

Seats – metal or elastomer
encased in the rigid spool.
Exceptional sealing under the
most adverse conditions.

All parts interchangeable
for like valves of the same
size and pressure class.

Trunnion Bearings.
Permit low operating
torque and exact
ball alignment.

Spring retainer-springs
provide pre-load to seal
tightly from vacuum to
10.800 psi. Compensate for
seat wear.

End connections – available
in a variety of Union Nut
connections, integral weld
and flanged ends.

FIG. 1 CUTAWAY OF TYPICAL VALVE

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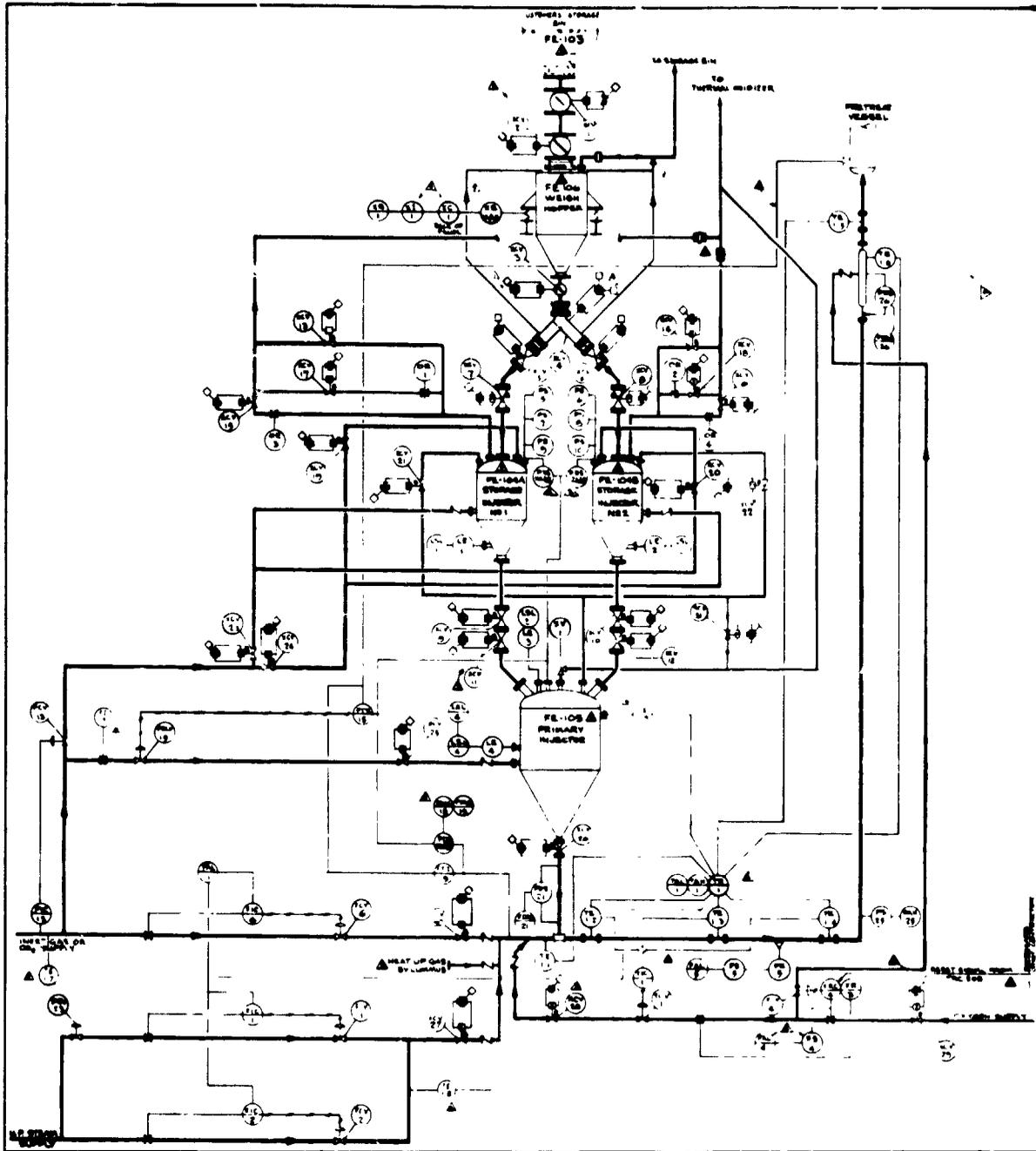
FIG. 2   SCHEMATIC FLOW DIAGRAM
FOR HIGH PRESSURE
COAL FEED SYSTEM

COAL GASIFICATION--NEW CHALLENGE FOR THE BEAUMONT ROTARY FEEDER

J. Stelian

Beaumont Birch Co.
Pittsburgh, Pennsylvania

ABSTRACT

Energy, its production and consumption, is a prime concern to every individual, all corporations, and the United States Government.

90% of the known fossil fuel reserves of the U.S. exist in the form of coal - much of it having a sulfur content too high for direct burning in conventional power plants without treatment of the coal or the combustion products. The utilization of this coal in an efficient and environmentally acceptable manner is important to help meet our national energy requirements.

The future use of coal may depend on our ability to successfully convert coal to clean gaseous and liquid fuels.

One important part in the coal gasification process is played by the specially designed rotary feeders which - in almost any coal gassification method - provide the regulator and airlock in the feeding of the coal or coal char at high pressure and in some applications at high temperatures. These units must be of an airlock sealed type to withstand the differential pressure.

The coal or coal char is discharged from a reactor or pressurized lock hopper through a rotary feeder into a pressurized transport line. This flow must be controllable - a very important factor in such a process. The rotary feeder, placed between these two systems, plays a very essential dual role.

All these factors constitute a serious challenge to the various materials used in the manufacturing process. A discussion of these issues should be a timely undertaking to professionals in the coal gasification field.

Energy, its production and consumption is a prime concern of every individual, all corporations and the United States Government. Shortages of energy have already influenced the behavior and growth pattern of the nation, and will continue to do so until long term solutions to the problem are achieved.

The continuous increase in the world population and the industrialization of many third world countries combined with the daily increase of energy consumption per capita in the entire world, requires that all countries make long range plans concerning their future energy needs. Research and development, conservation and finding new and more efficient forms of energy are the principle targets for all the industrialized nations in the next 10 to 15 years.

The U.S. has to be in the forefront of this battle.

As President Carter clearly stated in his energy message:

"The heart of our energy policy problem is that our demand for fuel keeps rising too quickly while our production goes down, and our primary means of solving this problem is to reduce waste and inefficiency.

In the same message President Carter paid great attention to our large reserves of coal and the necessity of converting many present user of oil and natural gas to coal.

His message was clear:

"Although coal now provides only 18% of our energy needs, it makes up 90% of our energy reserves" - and calls on all of us

to increase the use of coal by 400 million tons - or 65%
in industry and utilities by 1985.

As mentioned above, by current estimates - nearly 90%
of the known fossil fuel reserves of the United States exist
in the form of coal - much of it having a sulfur content too
high for direct burning in conventional power plants without
treatment of the coal or the combustion products. The utilization
of this coal in an efficient and enviromentally acceptable manner
is important to help meet our national energy requirements.

The conversion of coal to clean gaseous and liquid fuels will
produce the big break-through in solving the energy shortage.

As you well know there are many different methods of
conversion and the product of conversion can be oil or gas
defined as low, medium or high BTU type.

Today's enviromental regulations say either remove SO2
from combustion gases or remove the sulfur before burning.

One answer: Gasify the coal - then desulfurize the gas.

Presently, different processes to convert coal into gas are
beginning to emerge and the research to find more sophisticated
one is very extensive.

To find the most economical method to gasify coal is
presently one of the most important projects of many major
engineering and research companies throughout the United States.
The present crisis situation and the outlook for the future show how

important the finding of the right solution is, and to find it in record time before all our present resources of energy are gone.

Coal and coal char present some different problems to the engineer in the material handling field.

The equipment must be designed to handle large volumes of coal and to be resistant to its corrosive nature, especially when moisture is present.

Strip mined coal also has impurities such as rock; sand which must also be taken into account.

We are talking about a sizeable capital investment for coal handling equipment even before the gasification stage.

In the future, we feel that the development plants will prove to be much more economical than the development of small coal handling units for each individual user.

Presently, each utility company, hospital, school, etc. has it's own coal handling equipment The future will prove that it will be more economical for these users to obtain the coal in a gasified form than trying either to burn coal or gasify it themselves.

Based on the above, I feel that we are ready to make these large investments to handle vast amounts of coal in some strategically located coal gasification plants.

One important part of the coal gasification process is played by the specially designed rotary feeders. The rotary feeders provide the regulator in the feeding of the coal or char at high pressure

and in some applications at high temperatures.

Our Company - Beaumont Birch Co. - considers itself fortunate to have had the opportunity to furnish the rotary feeders for a number of pilot plant and laboratory research units - in the past 5 to 10 years.

We intend to continue furnishing the specialized rotary feeders for any future pilot plants and also transplanting our experience in the rotary feeders field to build larger units for demonstration plants and hopefully in the near future commercial plants.

The Beaumont Birch - special designed rotary feeders were and are used in the following locations and processes:

CO_2 Acceptor Process

A number of 4" size special rotary feeders designed to operate in a high pressure system are used at the demonstration plant in Rapid City, S.D. - developed by Consolidation Coal Co. and built by Stearns-Roger Co.

The rotary feeders handle coal and dolomite, and some units were designed to handle high temperature char.

Cogas Process & Project Coed

Beaumont Birch Co. supplied the necessary rotary feeders for both processes.

The Cogas Process being based on the Coed experience, we developed the first rotary feeder used in the Coed Project.

Westinghouse - OCR Project at Waltz Mills, Pa.

Eleven high pressure rotary feeders especially designed to handle coal, char and dolomite at specified rates, temperatures and pressures are used at the Waltz Mills location.

At Hydrocarbon Research Co. in Trenton, NJ a team of researchers were using 2 - 4" size Beaumont Birch Co. rotary feeders in a pilot plant operation designed on the basis of the Bureau of Mines "Synthane" process for coal gasification under pressure. The two rotary feeders were working alternately. They were used under a pressurized lock hopper at 450 PSI, and were feeding the coal into a super heated st am transport line which discharged the coal into a reactor. The super heated steam transport line was also under 450 PSI pressure, with maximum differential pressure between the two systems of 25 PSI.

These processes and users are the better known but the list does not end here. A number of companies are using our rotary feeders in small coal gasification plants such as a Texas Cement Co.

Or they are doing different studies related to the coal gasification field as:

Argonne National Laboratory in Argonne, Ill., Exxon Co. USA in their research laboratory in Baton Rouge, La. -- C. E. Lummus Co. At Bloomfield, NJ etc.

Presently we are working on some special - 2" size units to be used by Rockwell International - Atomics International Div in their coal gasification pilot plant to operate at Santa Susana Laboratory.

These feeders are designed to operate in a 300 PSI pressure

system - between a pressurized lock hopper and gasifier maintaining

a minimum amount of leakage across the rotary feeders.

As a sister operation, our feeders are used successfully by

FMC Corp. - Industrial Chemical Div in their Kemmerer, Wy. plant

in their process to obtain coke.

We consider the process similar, because the feeders have to

handle coke dust, or coal. Actually the first unit used by

FMC Corp. at Kemmerer, was transferred from FMC's Coed Plant

in Princeton, NJ. FMC stresses that it's process at Kemmerer,

gives a high quality coke from any single type of coal from

anthracite to lignite.

All the jobs have in common the following factors:

1) All handle: coal, limestone, dolomite, char - products almost

equally abrasive and in some cases corrosive.

2) They require a very accurate regulation of product fed in order

to obtain an optimal reaction.

We are dealing with a product with many built in variables as:

bulk density, variable size (from $\frac{1}{2}$" down to mesh sizes).

The feeders are able to react to changes in bulk density and

size of product by changing the speed at which they are working.

All rotary feeders are sized based on volume. The displacement

figure of every size of rotary feeder is very accurate and in most

cases when handling mesh size products the efficiency of the units can

be considered 100%.

Even when handling larger size products, by making a few test runs, the efficiency factor of the unit can be established to an exact number.

3) They require a "perfect seal" or almost perfect seal. Based on our experience in the rotary feeders field, we can estimate rather accurately the amount of leakage which can be expected across the rotary feeder (airlock).

Every unit is tested in our manufacturing facilities for leakage under a differential pressure of 10 PSI or higher if requested.

The amount of leakage varies from less than 1 CFM for a 2" size unit to 3 CFM for a 12" size, plus the volume of the empty returning pockets.

4) - They are custom designed to various pressure systems as required by the process.

5) - They are custom designed to the various temperatures as required in process.

6) - They are sized to the capacity requirements of the process - using in every application the optimal size for the process.

We combined all the design factors for every job, and it resulted in a wide variety of rotary feeders designed - variety reflected in size, material of construction, type of drive, machining details, choice of auxiliary materials as: packing, adjustable

blades, etc.

The most common size used until now is the 4", because the capacity requirements were rather small in the pilot plant operations.

In some applications even the 4" size was to large and we had to use only a 2" size unit.

An exception to the above is the 10" size unit furnished to a Texas Cement Co. where they are gasifying coal for their internal use.

As we hopefully step into the next stage of coal gasification - demonstration and commercial units, the size of rotary feeders (airlocks) should grow also.

Based on the experience of some pilot plants, there is a limit to the size of the rotary feeders to be used.

Being a rotating type of equipment with many parts required to be maintained in good condition for constant good performance, the manageable size should be of concern in the future feeder design.

We feel that the 12" size rotary feeder is large enough (capacity 150 cu. ft./hr. for each RPM - 3,000 cu. ft./hr. @ 20 RPM - max. recommended) - but still manageable for this type of application.

The practical experience chooses the use of multiple 12" size units in place of one or two very large units.

Also, the multiple unit approach can alleviate the complete

;hut-down situation   In a multiple unit system the chances

of a complete breakdown is rather remote.

The pilot plant units were constructed of:

A - Cast Steel - Type WCA, WCB, WCC with 0.40 - 0.50 carbon, with

body hardened to 4⁵0 brinell for improving the wear resistance at:

1) CO_2 Acceptor Process - To Handle spent dolomite at

300° F design temperature, 33⁵ PSIG pressure system.

2)   Project Coed - To handle pulverized coal at 215° F

design temperature 100 PSIG pressure system.

4) Low BTU Gas - Pilot Plant Project at Waltz Mills, Pa.

(Westinghouse Co.) to handle coal, char, dolomite at temperatures

ambient to 200° F, 300 PSIG system.

5)   Atomics International - Santa Susana - To handle

coal at ambient temperature 300 PSIG system.

6)   C. E. Lummus Co. - Bloomfield, NJ - To handle coal

at temperatures rang⸴ng from 75 to 100° F, operating pressure

180 PSIG.   The unit was designed for a max. pressure system of

300 PSIG.

7)   FMC Kemmerer, Wy. - To handle coke at 100 and 350°

F and 100 PSIG pressure system.

B - Series 300 Stainless Steel at:

1)   Coed Process - To handle char at 600° F.

2) CO_2 Acceptor Process - To handle fresh dolomite and

fresh lignite at 600 and 700° F - 320 PSIG pressure system.

3) California Research Co. - To handle char at 1600° F.

4) Argonne National Laboratory, Argonne, Ill. - To handle crushed coal, limestone, different types of coal (Pittsburgh, Illinois and San Juan), lignite (South Dakota type)

At ambient temperature in a 135 PSIG system with ero or very small differential pressure across the feeder.

5) Exxon Res. & Dev. Co. - Research Laboratory, Baton Rouge, La. - To handle char at 1200° F.

At Linden, NJ - To handle coal at 300° F.

6) Low BTU Gas - Pilot plant project - To handle fly ash, dolomite, spent dolomite, ash, gas fines, and char at 600° F, 300 PSIG pressure system.

7) FMC Kemmerer, Wy. - Coke Processing Plant - To handle coke at 400° F.

On the above jobs - for temperatures 600° F and above a type 309 or 310 stainless steel was used which combined successfully the wear and heat resistance.

Below 600° F, type 304 stainless steel was used.

Without ever materializing as a job, we were actively involved in the design of very special rotary feeders to be used in the bi-gas process - feeders for handling both coal and char.

The extreme conditions - 1200 PSIG system and 600° F for coal and 1200 PSIG system and 1600° F for char presented some

very difficult problems concerning the choosing of the right material of construction, example HH, HK type stainless steel.

Even for the pilot plant operation the size of the rotary feeders was rather large.

The body and the end plates had to be designed with extremely thick walls; large and very thick flanges. The large size of these castings can cause machining and handling problems. The approximate weight of the pilot plant unit designed to handle char at 1600° F, 1200 PSIG system is 11,000 lbs. For a good performance at these conditions we also had to design a very sophisticated type seal.

In order to design a commercial size rotary valve for the bi-gas and synthane processes - we recommend the design, manufacturing and testing of a pilot plant size rotary valve. Only after building and testing such a special rotary valve, will we be able to learn about the problems and try to improve the critical wear points of the unit.

When we speak of rotary feeders constructed of, we mean that the major cast parts: body, rotor and end plates are cast from the respective material.

1) All the materials entering in the construction of the rotary feeder as shaft, bearing, adjustable blades and packing seals should be chosen to be the optimal type for the design conditions.

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In all the fields - metals, plastics, packing etc. day
by day new developments occur in producing a better, more
wearable type of product.

There are some wear points concerning the type of materials
used or specific design but the new impulse given to R & D in the
coal gasification should eliminate all these points and generate
an optimal unit for the application.

In many applications the rotary feeders provide the
regulator in the feeding of the coal or coal char at high
pressure and in some applications high temperatures.

2) The rotary feeder is placed between a pressurized
lock hopper and a pressurized transport line. The rotary feeder
not only provides a seal between these two systems, it also
regulates the flow of coal or coal char from one system to another.

In most cases the rotary feeder is driven by a variable
speed drive which changes the capacity of the rotary feeder
by changing the output speed of the variable speed drive.

The development of SCR drives - using DC motors opens the
road for even a wider regulation of speed, capacity and volume.

In many pilot plant applications the pressure in the lock
hopper and the transport line may vary as required by the specific
process - from 150, 250, 375 to 450 PSI, having a maximum differential
pressure between the two systems of 25 PSI.

Also the temperature especially for coal char handling,

may be very high 600, 700, 900, 1200 or in some cases even 1600° F.

As mentioned previously for pilot plant operations, in most cases the rotary feeders are made of type SAE 1045 cast steel having the body hardened to 450 Brinell for a better wear resistance.

The connecting flanges, inlet and outlet are designed for the pressure rating of the specific system.

To achieve the airlock properties of the rotor, the major part of the rotary feeder, is equipped with two sets of adjustable tips and the feeder is also provided with two different types of seals - periphery and shaft seals.

When the operating temperature is higher than 600 - 700° F, we recommend the use of different type of stainless steel - type 309 or 310

If the pressure and temperature requirements combined require an even higher quality type of material, we recommend the use of stainless steels with high allowable stress at elevated temperatures.

Example - Cast HK (25 CR:20 Ni) This alloy has become highly popular for steam reforming and ethylene pyrolysis plants - for good reason - it not only provides high tensile strength and excellent resistance to hot gas corrosion in the crucial 1500 - 2000° F range, but offers outstanding rupture and creep properties. At 1600° F, for example, HK's 10,000 hour rupture strength is approximately 4000 PSI as compared with about 2500 PSI for it's

closest wrought equipment.

Similar to HK, all heat resistant alloys - HH HT & HX
would be used extensively in making the rotary feeders used at
high temperatures and high pressures

For long term service, materials should be ductile, easily
cast and welded if necessary, small grained, able to retain an
erosion resistant smooth surface, resistant to high temperature
and high pressure;

H_2S (Hydrogen Sulfide) H_2 Attack: and immune to stress
corrosion cracking. A good candidate for use could be as mentioned
chrome-moly alloys with and without protective coatings.

Coal, especially that found in the Eastern part
of the United States contains substantial sulfur. Knowing that
the need for coal is much bigger in the East than in the other
parts of the Country, we have to be prepared to deal with the
excessive sulfur. During processing, much of this is converted
to hydrogen sulfide which becomes increasingly corrosive to carbon
steel at temperatures greater than 450 - 550° F. We can draw upon
the experience of petroleum refineries, which have collected
extensive data on hydrogen sulfide corrosion at temperatures
from ambient to about 1000° F and at operating pressures to 3,500
PSI.

The combination of H_2S and H_2 is particularly troublesome,
since the 5 CR - ½ MO and 9 CR - 1 MO steels, commonly used to

resist sulfur corrosion at moderately high temperature show little improvement over carbon steel as their resistance to H_2S/H_2 atmospheres. A chromium content of at least 12% is required. The series 300 of austenitic stainless steels (min. 18 CR - 8 Ni) have excellent resistance and are normally specified for the H_2S/H_2 enviroment.

At temperatures between 800 and 1550° F the austenitic stainless steels tend to sensitize and precipitate carbides along grain boundaries. Based on that experience at high temperature it is necessary to add stabilizing elements such as columbium, tantalum or titanium to tie up the carbon.

Another route to go to improve the life of the major components could be hard chrome plating, coating with wear - abrasion - corrosion resistant type agent, etc.

Although the way to obtain high BTU gas is by using high pressures, high temperatues, from a maintenance and cost point of view it is recommended to use the rotary airlock feeders at lower temperatures and less harsh conditions.

Tungsten carbide could be the answer for a critically wearing surface, however, even tungsten carbide can be destroyed within 500 hours of operation. With good maintenance and high quality workmanship a 2000 - 3000 hours service can be reached.

Because of the critical nature of rotary feeders - airlock - valves performance in coal gasification plants, a program to develop

improved units - especially concerning material of construction

should be the major item in the next R & D program.

As we know, there are many methods in use to gasify coal with

some methods the operating conditions are not excessive.

In Coed pyrolysis, coal is heated in fluidized bed reactors.
In successive stages over a temperature range of 600 to 1100° F at a

pressure of 10 PSIG. Char gasification takes plate at temperatures

up to 1800° F. No serious material problems were experienced at

the Coed Pilot Plant during the course of it's operation in 1970-74

at Princeton, N.J.

The greatest challenges to the Materials Engineer exist

in the construction of the critical parts of a coal gasification

plant as: high temperature reactor, hot gas quench systems etc.

In the coal handling equipment, the major problem appears to be

erosion caused by highly abrasive coal particles. This is

compounded by the fact that the coal oil slurry must be fed

continuously or semi-continuously into the high pressure,

high temperature reactor assembly in direct linkage with the

rotary feeder.

Wear of the rotary - airlocks - valve system must be

minimized to prevent loss of pressure and temperature. Even

in the limited number of hours operation in the pilot plants

more conventional wear resistant stainless and alloy steel

did not prove to be completely adequate.

The finding of the ideal material of construction should be the major challenge especially when the pilot plant will be transformed in demonstration and commercial size plants.

In coal char handling there is also a corrosion factor present. Alloys such as incoloy alloy 625, hastelloy alloy "C" which displayed high resistance to hot corrosion and stress at high temperature could find a considerable use in the manufacturing of major components in contact with coal char.

The good performance of the rotary feeders consists of three major features:

1. Choosing the right material of construction.

2. Using the right combination of materials for the adjustable tips to achieve a continuous tight seal (almost rubbing type contact) between the body and the rotor with the edges equipped with the blades.

3. Using the right periphery seal. Seal provided between the body bore; rotor shoulder at both ends, consisting of packing rings or manufactured rings and following rings. In the case of using packing for periphery seals the follower ring is adjustable without taking the feeder apart.

The shaft seal-standard on most rotary feeder designs has only a secondary role in the Beaumont feeder design. The shaft seal has to perform well only if the periphery seal fails.

The ideal seal - minimum leakage it can be obtained only when a pair of adjustable tips are used - a base blade - flexible type manufactured of nylon, adiprene, teflon, ryton, vespel, etc. and a supporting blade facer tip made of hardened steel, tool steel, stainless steel etc. If the flexible type of blade can not be used due to temperature limitations, a rubbing contact never can be achieved due to the metal to metal contact.

We are very optimistic that the temperature limitations in using plastic based products is raised higher as days go by, and by the time that a feeder for a commercial coal gasification plant will be built we will have the necessary type of material for blades to use up to 1200 - 1500° F. Already Du Pont's Vespel product raised the temperature limitation to 800° F, although the cost of vespel presently is too excessive for use as adjustable blades.

Also concerning the periphery seals, we hope that with our R & D program, we will be able to develop a longer lasting type seal- minimizing the cost of maintenance and replacement.

It is very important that the rotary feeder be used in the conditions to which it was designed for. If the design temperature is much higher than the operating temperature, we will face an excessive amount of leakage. Machining clearances are calculated to the design temperature figures. The sealing material used as packing may become flexible

only at certain temperature. If the feeder will never reach
that temperature the packing may remain rigid not being able to
provide an adequate seal.

The Beaumont type airlock rotary feeders have some specific
design characteristics:

The inlet of the body will be fitted with a V-shear
plate.

This device, an integral part of the body reduces shock
or the horsepower requirements. It also reduces by approximately
90% the amount of material sheared in passing through the feeder.
A specially designed outlet (part of the body) permits the continuous
discharge of material from the feeder.

The feeder rotor is welded to an oversize steel,
stainless steel or special alloy shaft, and both the rotor and
shaft are machined while turning on centers. This construction and
method of machining assure the rotor of always being concentric
on the rotor shaft. It also prevents the rotor from ever working
loose on the rotor shaft. With this design, the seals (periphery)
and adjustable rotor blade tips are the only parts of the rotor
coming in contact with the feeder body.

As mentioned previously, the periphery seals are adjusted
through plugged holes in each feeder end plate with a torque
wrench supplied with the feeder. The torque wrench is to obtain
the maximum seal with the minimum drag on the motor.

With a good preventive maintenance program related to the adjustment and replacement of seals and blades a maximum service and good performance could be obtained from the feeder.

Each end plate is equipped with pipe connections used to inert gas or air between the end plates and the end of the rotor to balance the pressure within the feeder, or to scavenge out the space with air steam, etc.

By using these purge connections, we are able to keep the abrasive - corrosive fine powdered material out of the seal area. The life of the seal is drastically shortened if the product handled would leak into the seal area acting as a wearing agent. The parts which are in contact - body, rotor, seal have an acceptable lifetime and will wear out only if an outside wearing agent as fine coal would act as a friction agent.

The cover plate over the service door on the returning side of the feeder is also fitted with two pipe plugs - these plugs can be removed and pipelines connected for relieving the pressure or purging out the returning empty rotor pockets.

We have followed very closely the performance of the rotary feeders used in different existing pilot plant operations, and below we wish to summarize some of the conclusions learned:

The wear and the maintenance requirements of every rotary feeder is in direct relation to the working conditions as:

The wear was progressive with the size of the unit.

Temperature of the material handled - the unit should be used at the designed temperature assuring this way the ideal clearance between the different rotating parts. Also knowing the correct operating temperature will enable us to use the right type of material for cast parts, adjustable tips, and packing.

In one pilot plant project material as "Ryton" used for adjustable tips proved to provide the best wear resistance. On one job after 6 weeks use the tips did not show any wear.

It is very important that before starting to use any rotary feeder a very concentrated maintenance study should be made in order to know as well as possible both the ways of adjusting the units and also the replacing of different parts.

We strongly recommend that the person who would be in charge of the maintenance of these units should visit our shop and actually witness the assembly process testing and adjustment.

The periphery seals can be maintained for a very long time in good working conditions by purging the space between the end plates and the rotor with gas air - at a 1 - 2 PSI higher pressure than the inlet (equal on both end plates).

The shaft seals taking a secondary role to the periphery seals do not represent any maintenance problem.

It is an established practice (in order to cut maintenance cost) that each time the tips are replaced the packings are replaced also.

Consulting a few users of our units in different pilot plant projects the following running time without maintenance was found.

CO$_2$ Acceptor Process

100 hours continuous running time was obtained with the complete plant operating.

Cogas - Coed

The unit used for coal handling required adjustment about every 2 - 3 weeks.

The coal char handling unit did not require any adjustment for a period of 2 - 3 months.

Coke Handling

The units are running almost 4 - 5 months without any adjustment.

The conclusion which we could draw from these experimental units can be summarized as: If our recommendations concerning operation and maintenance are closely followed and after a short learning period in every new job the rotary feeders can be operated continuously with good performance for approximately 90 days.

As the units grow in size the wear of the major cast components as the body can be reduced by sleeving it with a much higher quality material as chrome plated sleeve inconel, etc. The cost of a sleeve represents only a fraction of the new body cost.

We are sure that many specialists involved in the coal

gasification jobs, consider the rotary feeder a troublesome

type of equipment. It has been found in some cases that the

blade tips w...r rapidly with the result that coal or coal char

flushes through without satisfactory flow control. In addition

in some cases the seals failed due to abrasion from coal particles.

Due to the fact that the rotary valves are a critical

component of a lock hopper system being a controlled rate feed

device our job is to improve both the grade of materials used

and also to minimize or completely eliminate the weak points.

We feel confident that with Beaumont Birch Co's long

history in manufacturing rotary feeders and other types of

equipment for material handling systems, especially coal, we

will succeed.

Our rotary feeders played a major role in obtaining good

performance in different coal gasification projects, and we hope

to continue to do so when we face the next step - commerical size

plant.

In the commercial size plants we would like to make the

following comments and recommendations concerning size and type:

CO_2 Acceptor Process

In handling either lignite, western sub-bituminous coal or

dolomite, our 12" size S.T.T. Mark II periphery sealed type feeder

would be able to give a good performance. This statement based

on the facts learned from using our 4" size similar rotary feeders
at the pilot plant working based on this process - at Rapid City,
S.D.

We feel that the 12" size is large enough - but still
manageable for this type of application.

We would recommend certain improvements to improve wear
resistance and reduce leakage, in order to have a better flow
control.

Bi-Gas Process & Synthane Process

In order to be able to recommend the use of a rotary feeder
in this application - a test unit is required to be manufactured
and experience gained for a scale up unit.

The test unit should be tried with both char and coal.

We c   t conclude our notes about the rotary feeders
without men...oning the important part the complex drive systems
play in both existing pilot respectively laboratory units and
for the future use in demonstration and commer﹒﹒al units.

The drives have to be able to translate the variation
in weight to variation in speeds, with a rather high percentage
of accuracy.

We are dealing with a product with to many variables
and a 100% designed system has to have ways to deal with it.
The variation in size and bulk density require that the rotary
feeder adjust itself instantly. A fast change in speed of the

feeder would deliver the right amount of coal or coal char into the reactor.

There are no limits concerning the sophisticated control systems which can be used in the coal gasification field.

In some aspects coal gasification represents a challenge to solve all the problems which we may be facing today.

But with a joint effort and especially economic backing the existing problems can be solved. Today's technology - concerning especially materials, presently is at such a level that there are no unsolvable problems.

The usual cliche - we could put a man on the moon why can't we in this particular case - successfully and economically gasify or liquefy coal - the answer is we can. We hope that the next few years will be the years when coal gasification plants will be a common site be that a power plant, industrial user or any other facility which would either replace the scarce oil or natural gas.

We hope that the discussions at this conference already may give some solutions to the problems which today may delay the construction of the commercial size coal gasification plants.

We are ready for the challenge and so I think is the entire industry.

DEVELOPMENT OF COAL-FEEDING SYSTEMS AT THE MORGANTOWN ENERGY RESEARCH CENTER

J. M. Hobday

U. S. Energy Research and Development
Administration, Morgantown Energy Research
Center
Morgantown, West Virginia

DEVELOPMENT OF COAL-FEEDING SYSTEMS
AT THE
MORGANTOWN ENERGY RESEARCH CENTER

John Mark Hobday*

ABSTRACT

The Morgantown Energy Research Center (MERC) has developed a variety
of systems since 1945 for feeding crushed and pulverized coal into coal
conversion reactor vessels. This paper describes their past and current
work on these systems including pneumatic methods for feeding pulverized
coal, slurry feeders, and coal pumps, methods for steam pickup, and a method
for drying a water-coal slurry in a steam-fluidized bed subsequent to feeding
the coal into a reactor vessel.

*Mechanical Engineer, Morgantown Energy Research Center, Morgantown,
 West Virginia

1. INTRODUCTION

The renewed emphasis on coal as a primary energy source and the investigation of a variety of conversion processes on a larger scale have resulted in increased research and development artivi.y in coal feeding techniques and equipment. Processes for delivering/feeɩ'ɩng the coal into the gasifier have usually been developed incidental tc the process with varying degrees of success. The purpose of this paper is therefore to review and summarize MERC's experience and research on coal feeding systems in the last 30 years.

The development of coal feeding systems at MERC is divided into two parts for this presentation:

• A historical discussion covering the period from 1945-1965

• A discussion of progress from 1966 to date

Since our time is limited, my presentation does not contain detailed data. However, for those who are interested in the system's problems and current progress, a comprehensive bibliography is included which lists reports and documents generated by MERC on coal-feeding systems.

2. HISTORY (1945-1965)

The Bureau of Mines at Morgantown has done considerable research on a variety of coal-feeding systems including research on atmospheric and pressurized coal gasifiers as well as developing a coal-fired gas turbine. Most of these processes required the feeding of pulverized coal either to the gasifier (atmospheric or pressurized) or turbine combustor.

During this early stage of coal-feeding development, M. gantown successfully fed coal at varying rates from 2 grams (laboratory scale) to 2 tons (pilot scale) per hour from vacuum to 600 psig. Morgantown ;as also awarded four patents, three on feeders and one on unique contr s for a feeder.

In the following paragraphs, I will discuss those feeding systems which Morgantown has either used or conducted research on during the 20 years from 1945-1965.

2.1 FLUIDIZED FEEDER

Purpose

The objective of using the fluidized feeder in coal gasification was to develop a system that will continuously feed pulverized coal at a uniform rate to a reactor with operating pressures up to 600 psig. Developmental work extended from the 1950's to the early 1960's.[4,12,17,21,26]

Features

The initial development at Morgantown started with a laboratory-scale feeder which injected coal at atmospheric pressure. As the investigations progressed, larger feeders were built until satisfactory performance was demonstrated at pilot scale. When the emphasis shifted to pressure gasification, the fluidized feeder was adapted for use at pressure; however, feeding the pulverized coal at pressure created definite problems.

A schematic diagram of the fluidized-feeder system is depicted in Figure 1. Coal is fluidized in the pressurized feeder by inert gas (carbon dioxide plus nitrogen) and recycled by the recycle compressor which maintains a bed density of about 20 lbs/ft^3. A dense, fluidized stream flows continuously through a calibrated coil to the pressure gasifier. Excess fluidized gas leaves the top of the feeder through an internal centrifugal dust collector; passes through a knockout chamber for further de-dusting; is recompressed by the compressor; and is cooled before re-entry into the feeder. A small amount of inert gas is added continuously to replace the conveying gas leaving with the coal.

The Morgantown pilot-scale fluidized coal feeder, used with the pressure gasifier, was equipped for zone fluidization which limits the circulation of fluidizing gas and prevents the carryover of particles into the recycle system, particularly the recycle compressor. Zone fluidization functions (Figure 2) as follows. Fluidizing gas enters as two separate streams, auxiliary fluidizing gas and main fluidizing gas. Auxiliary fluidizing gas provides limited fluidization of the main coalbed and ensures migration of finely divided coal particles from the main bed into the central column. Main fluidizing gas creates a fully fluidized zone within the central column. Fluidized coal is withdrawn from the central zone through a feeder discharge
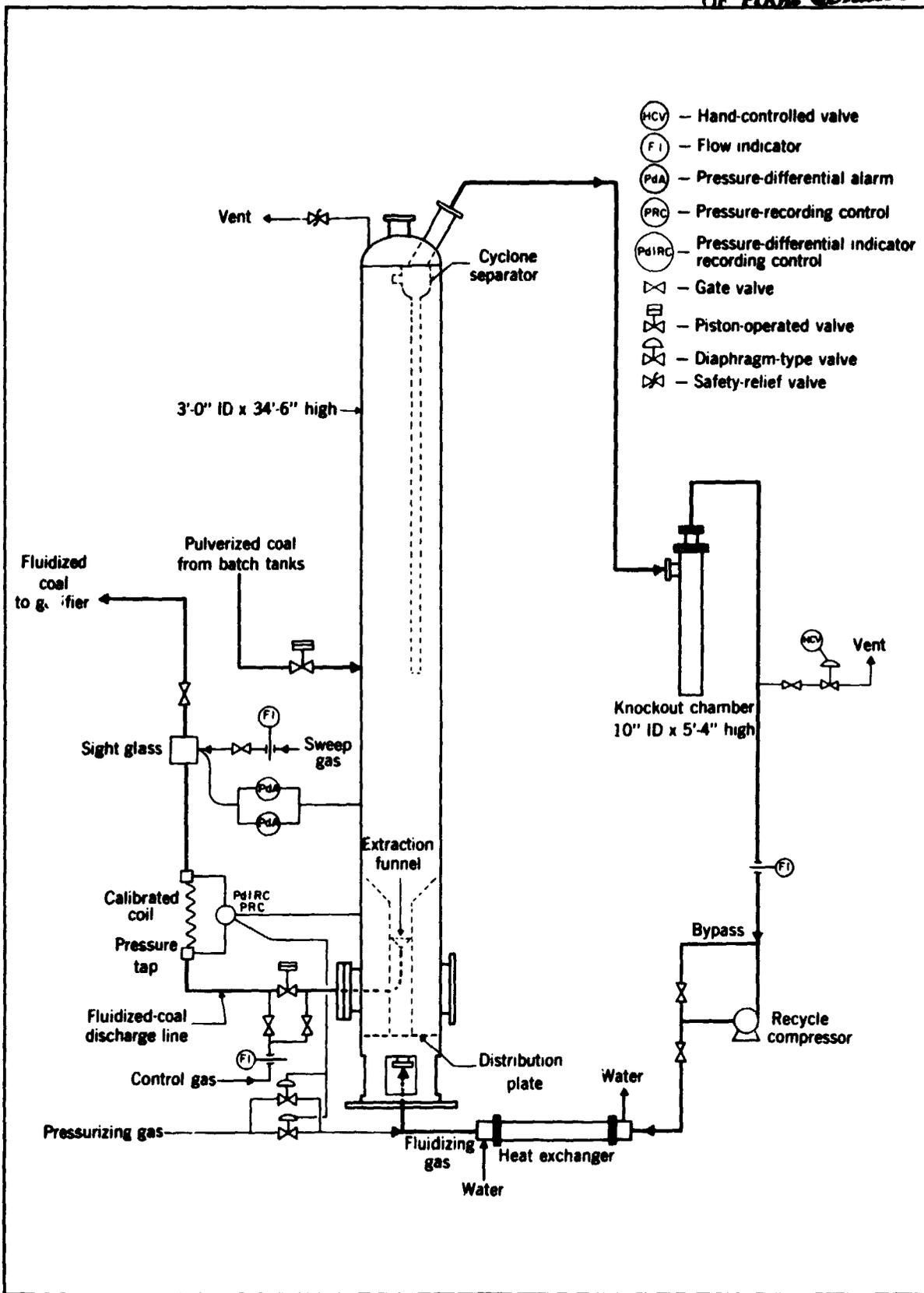
Figure 1. Pilot-Scale Fluidized Feeder Utilizing Zone Fluidization
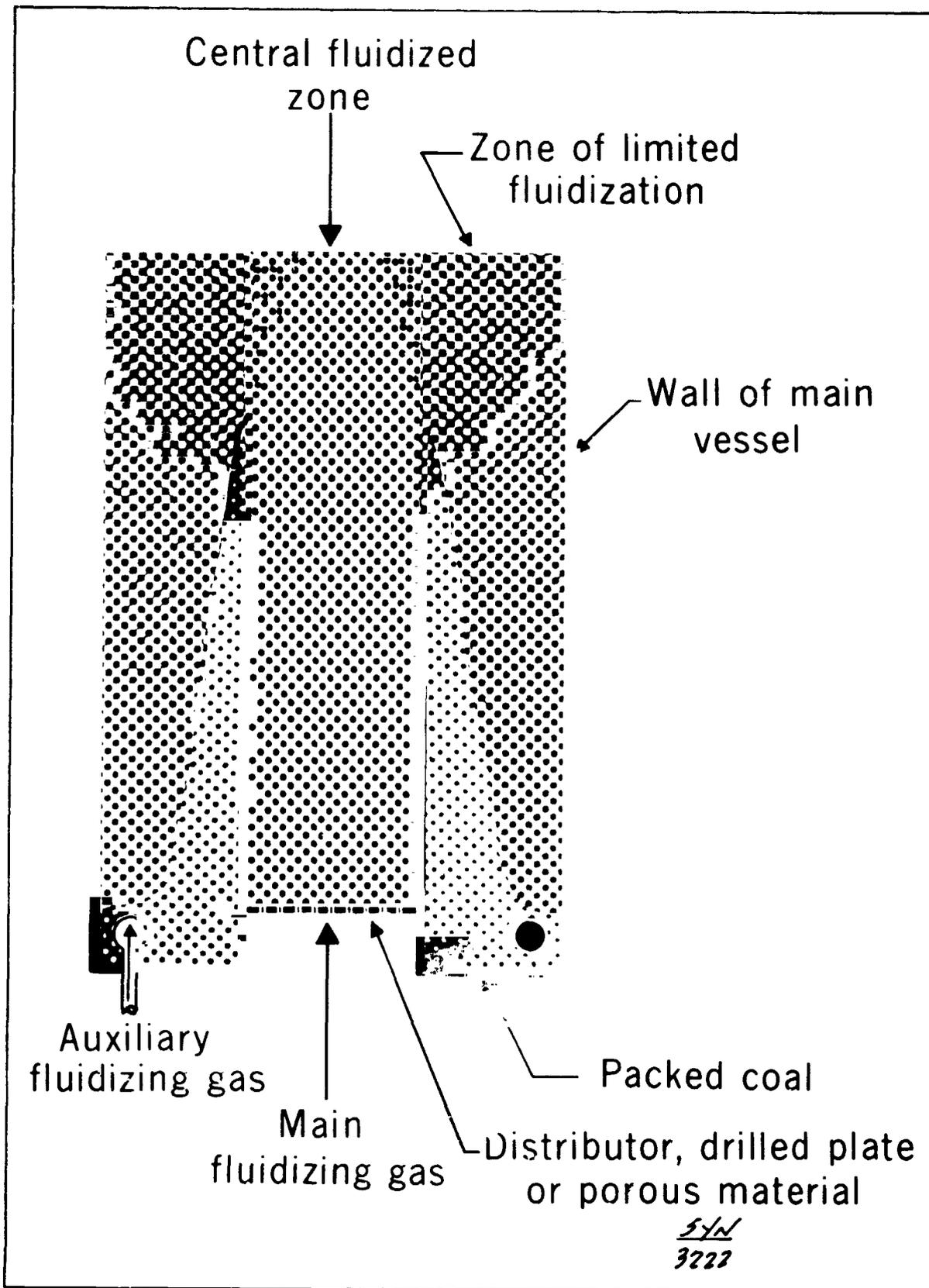
Figure 2. Zone Fluidization Technique

line. The gas distribution plate near the bottom of the continuous feeder supports a 6-foot high central fluidizing column, with a 13-inch diameter, topped by a 70^0 funnel. The 60^0 extraction funnel shields the coal delivery tube, preventing erratic flow.

Results

The fluidized coal-feeding equipment satisfactorily demonstrated an accurate method of feeding coal to a pressure vessel at 450 psig with a capacity of 1 to 2 tons/hour (tph). Two coal metering devices, calibrated coil and weigh cell system, worked satisfactorily and fluidizing gas requirements were low. Equipment was also developed for storing and transferring pulverized coal. Scale tanks were used for transferring coal between buildings. They were pressurized at the top with the coal and gas leaving at the bottom. The valves at the exit controlled the flow. Transfer rates of 1 to 8 tph were achieved.

Figure 3 shows the fluidized feeder on the weigh cells. The feeder is the tall cylindrical vessel. The recycle-gas line runs from the top of the feeder to the bottom. A surge tank for the inert gas supply is in the foreground, and the calibration receiver with conical bottom is on the scales at the right.

2.2 PULSE FEEDER

Purpose

During the late 1950's, Morgantown gained experience in pulse feeders through an improvement to a laboratory size apparatus. This work was performed while studying dust collection.[34,35]

Features

The pulse feeder is a laboratory type feeder for entraining powdered solids in a pulsating gas stream, wherein the powder is fluidized in a vertical tube and is withdrawn intermittently at the bottom.

The system operates as follows. The pulsating air is admitted to the feeder at a controlled rate through the orifice just above the dust-outlet nozzle. At each impulse peak, a small quantity of the carrier air flows up through the dust-outlet nozzle and the dust bed and escapes through a bleed-air tube above the dust bed. At the low-pressure point in each impulse, the
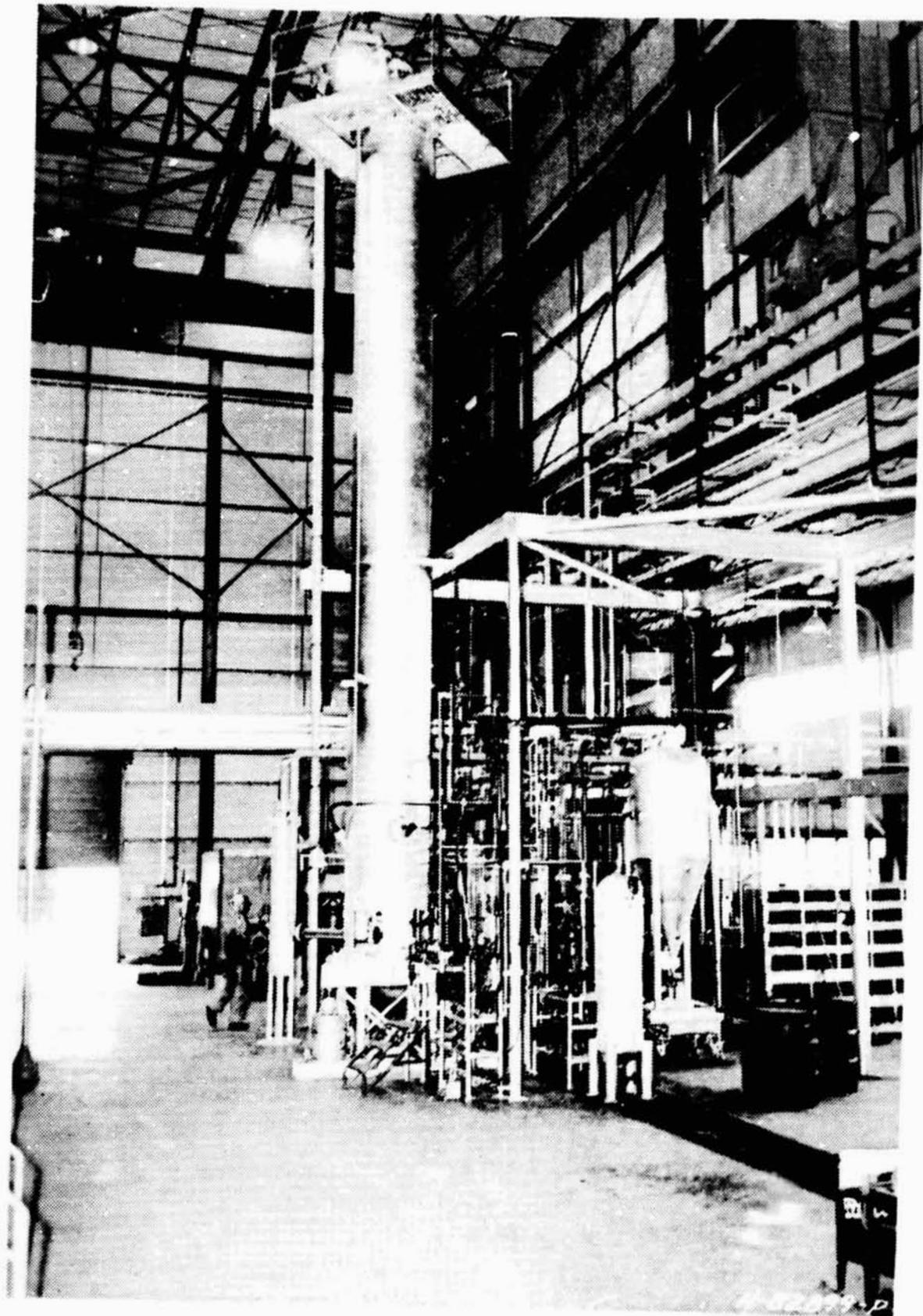
Figure 3. Fluidized Coal Feeder and Related Equipment

air flow through the nozzle reverses, and a small increment of dust is entrained by the pulsating carrier-air stream. This operation results in a continuous flow of pulsating air through the bleed-air line and an intermittent discharge of dust from the outlet nozzle. Feed rate will remain constant if the average operating pressure and the pressure differential across the dust-outlet nozzle and dust bed are maintained constant. See Figure 4 for an illustration of the system.

Results

Findings showed that the system had a range of feed rates from 1 to 200 gm/hr. However, the system may not be feasible for scale-up.
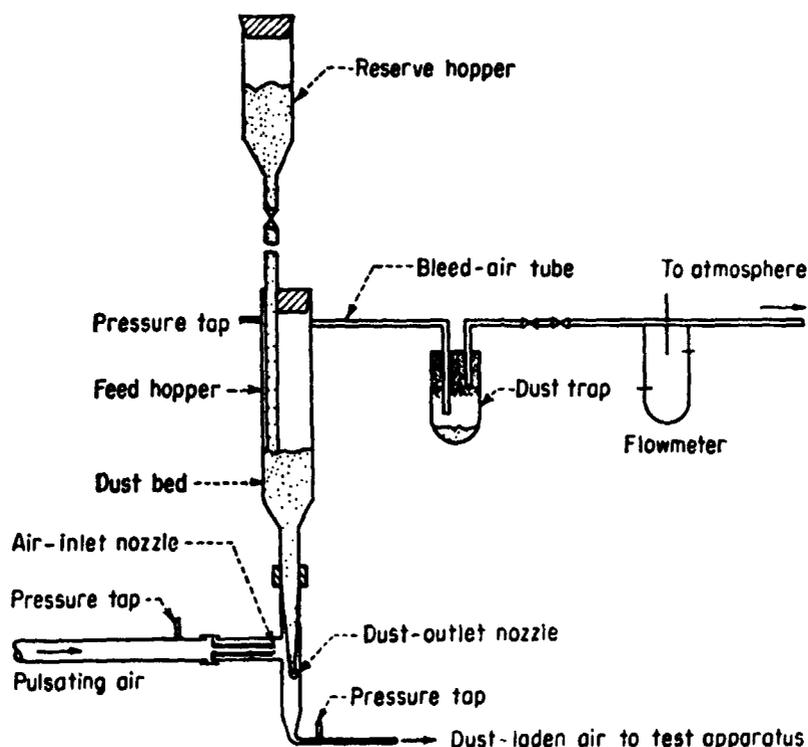
Figure 4. Pulse Feeder

2.3 BAILEY FEEDER (STEAM PICKUP)

Purpose

It became necessary to develop a reliable coal-feeding system in conjunction with the Bureau's programs on coal gasification. The Bailey feeder (also referred to as the steam pickup coal feeder) was modified and incorporated into the pilot plant, which gasified pulverized high-volatile A bituminous coal with oxygen and superheated steam.

The primary purpose of operating the pilot plant, equipped with steam-pickup feeding, was to determine the operability of the gasifier fed by this method as compared to the results obtained when the gasifier was fed with the fluidized coal feeder. Development extended from the late 1950's to the early 1960's.

Features

A Bailey feeder (star-wheel feeder) was used to feed a mixture of coal and steam to the pressure gasifiers.[36,38,50] Figure 5 illustrates this process. Lock hoppers were used for pressurizing the coal. Between the hoppers was a 4-inch motor-driven ball valve. The system operated with 70% through 200-mesh coal. The star-wheel feeder was mounted at the bottom of the main hopper (Figure 6), and regulated the coal flow to the water-cooled tee where the superheated steam and coal mixed.

Results

The system's capacity was from 1 to 2 tons of coal per hour. It had an advantage over the fluidized feeder in that it required no recycle gas. However, its disadvantage was that close control of the differential pressure across the feeder was required to maintain uniform rates.

2.4 ORIFICE FEEDER

In support of a Bureau of Mines' program on coal gasification process development during the late 1960's, an orifice-type feeder was developed. A patent was subsequently issued.[32]

Features

The orifice feeder was a laboratory-type feeder applicable to continuous uniform coal feeding. As shown in Figure 7, the feeder can discharge into a pneumatic pickup line or a screw conveyor into a fluidized bed and directly into a pressurized reactor.[3,22,32]

The device consists of a chamber with multiple, vertically spaced orifices aligned and decreasing progressively in size from top to bottom. Material is introduced through the entry at the top of the chamber. When using this feeder to introduce coal into a pressurized reactor, it has been found advantageous to maintain a slight positive gas pressure, relative to the reactor pressure. Lock hoppers may also be conveniently used with this system.
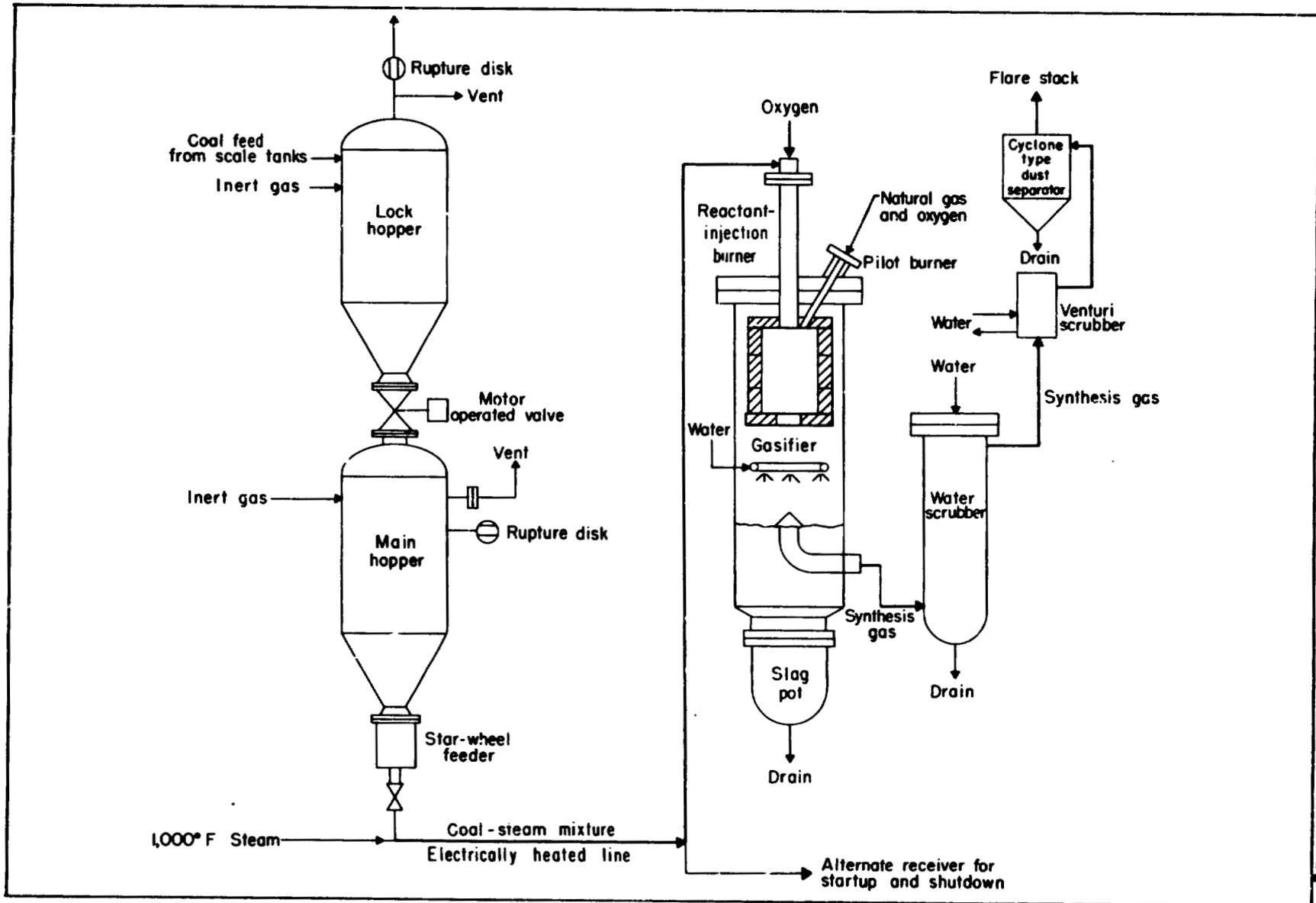
Figure 5. Pressure Gasification Pilot Plant Equipped with Steam Pickup Coal Feeder

Pulverized
coal

Fluffer wheel
(rotating)

Top feeder apron
(stationary)

Feeder wheel
(rotating)

Spacer for
feeder wheel
(stationary)

Bottom feeder apron
(stationary)

To stream of
superheated steam
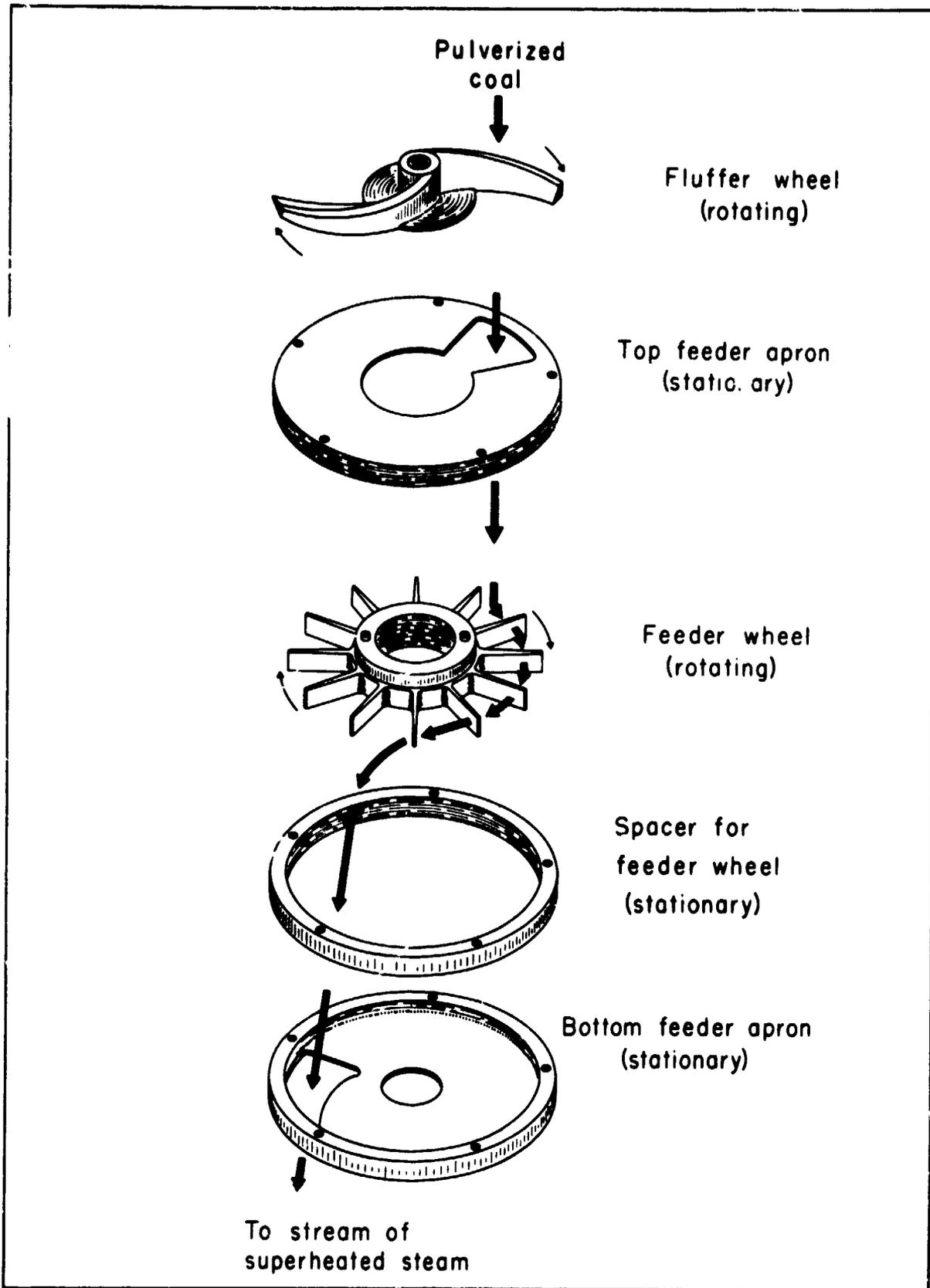
Figure 6. Exploded View of Star-Wheel Feeder

INVENTOR

HARRY G. LUCAS

BY
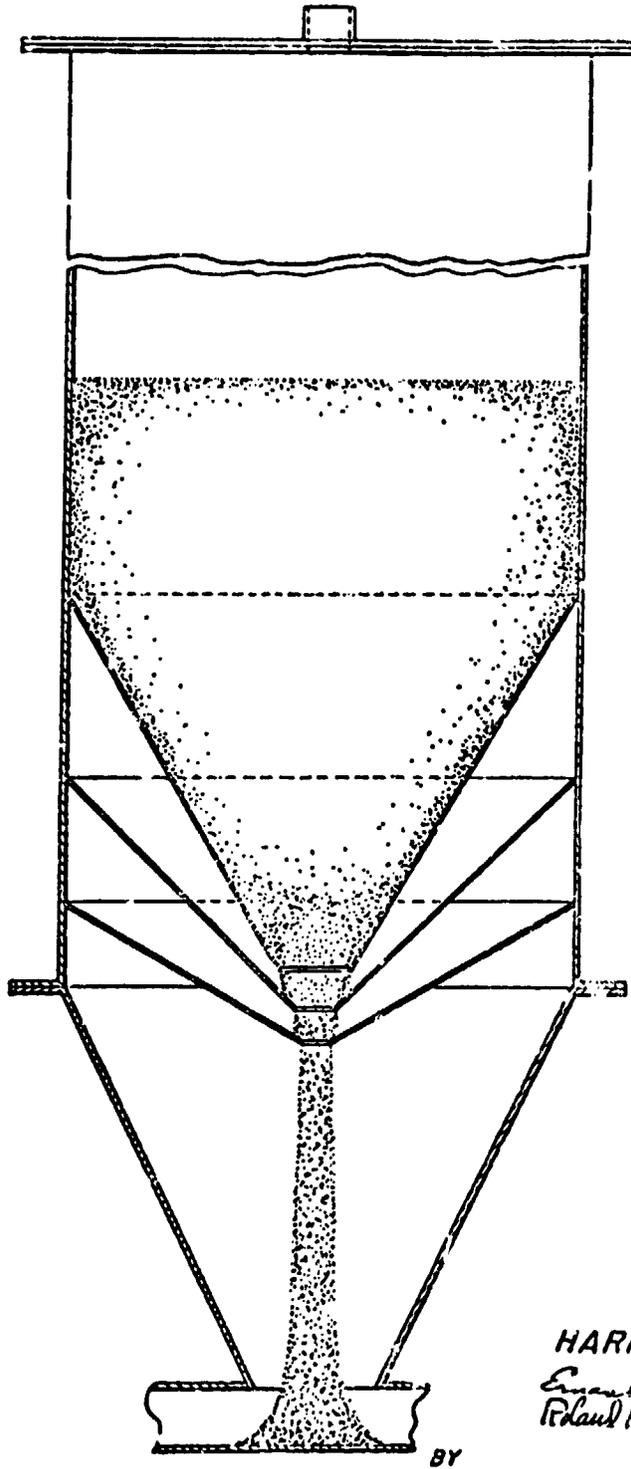
ATTORNEYS

Figure 7. Orifice Feeder

Results

Coal char experience demonstrated a constant feed rate of 4 gm/hr to 27.5 kg/hr into a fluidized bed. The system solved the problem of uncontrolled flow rate due to nonuniformity of particle size. The inventor also claimed that the system has a scale-up capability.

2.5 LDC COAL-FEEDING SYSTEM

Purpose

The Bureau of Mines conducted a comprehensive research and development program on a direct coal-fired gas turbine plant. The LDC coal-feeding system is one of the pneumatic systems tested for pulverizing and feeding coal to the combustors. The feeding system was previously developed and operated by Locomotive Development Committee, Bituminous Coal Research, Inc., for use in locomotives utilizing coal. For the Bureau's purpose of evaluating the effects of coal firing on turbine life, this system was installed in pre-paration for the 1500-hour coal-fired turbine run during the fall of 1962.

Features

The LDC feeding system supplies 3500 lb/hr coal at 86 psig to the turbine's combustors. Coal crushed to 3/16-inch x 0 and containing about 3% moisture and 6% ash is fed into the top of the coal pump and fills the pockets of the rotor. The coal is rotated to tne bottom of the pump, where air sweeps the coal from the pockets at 90 psig and carries it into the pulverizer. The pulverizer operates at approximately 86 psig and 3600 RPM and reduces the coal to 90% through 200 mesh. The pulverizer requires 100 hp when coal is ground under pressure as compared to 35 hp when coal is ground at atmospheric pressure, primarily because of increased windage losses. The coal/air mixture leaving the pulverizer flows through the 2½-inch pipe to the combustors. This system contains relatively compact equipment. Figure 8 shows the coal-feeding system;[40,43] Figure 9 illustrates the coal pump.

Results

The overall system was seriously limited by the pulverizer's operating life; erosion of the parts resulted in an unbalanced rotor. The pump also had problems with seals and running clearance.
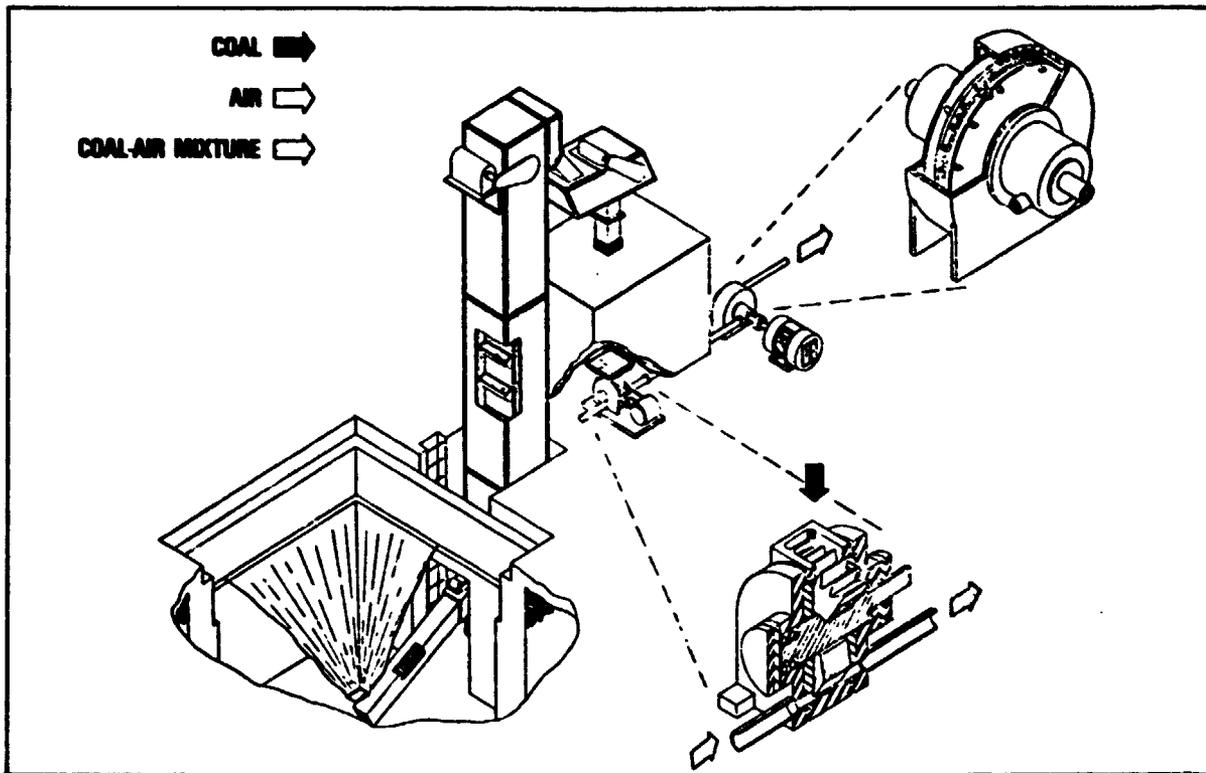
Figure 8. Coal Feeding System Using LDC Components

Later reports indicate that the Koppers Company improved the seals so that a higher degree of success was obtained while operating at a pressure of approximately 50 psig.

2.6 FULLER-KINYON PUMP

Purpose

The Fuller-Kinyon pump was installed to determine if it would work at 90 psig pressure differential and feed directly to the process, thereby eliminating the need for lock hoppers.

Features

Basically, the pump (Figures 10 and 11) consists of a high-speed screw with a gradually reducing pitch section at the delivery end. The coal being conveyed is advanced by the screw from the hopper section into a short barrel section where it is compressed to form a seal against blowback. The coal is then discharged into a stream of inert gas which carries the coal to the process.

COAL AND AIR

AIR
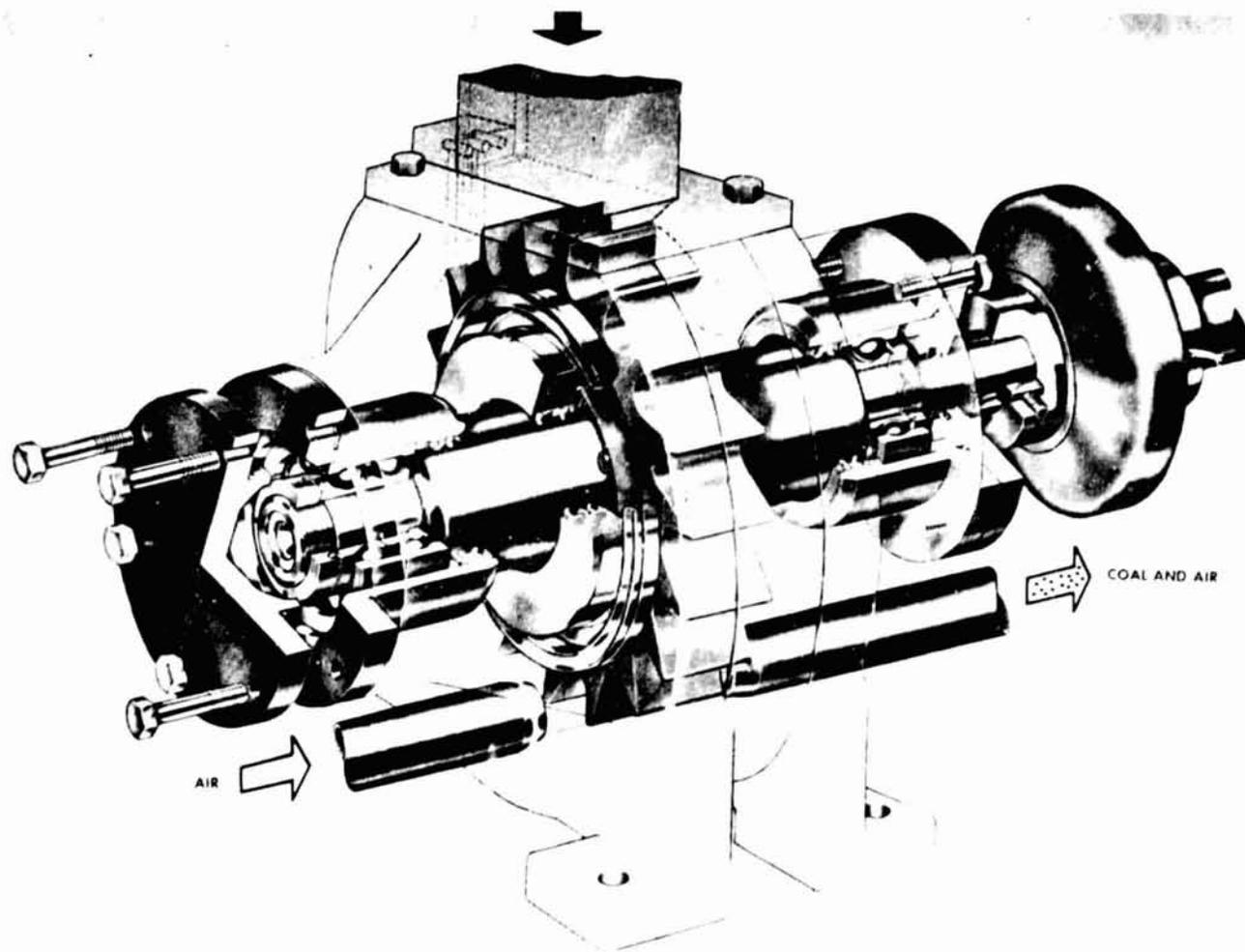
FIGURE 9. L.D.C. COAL PUMPS
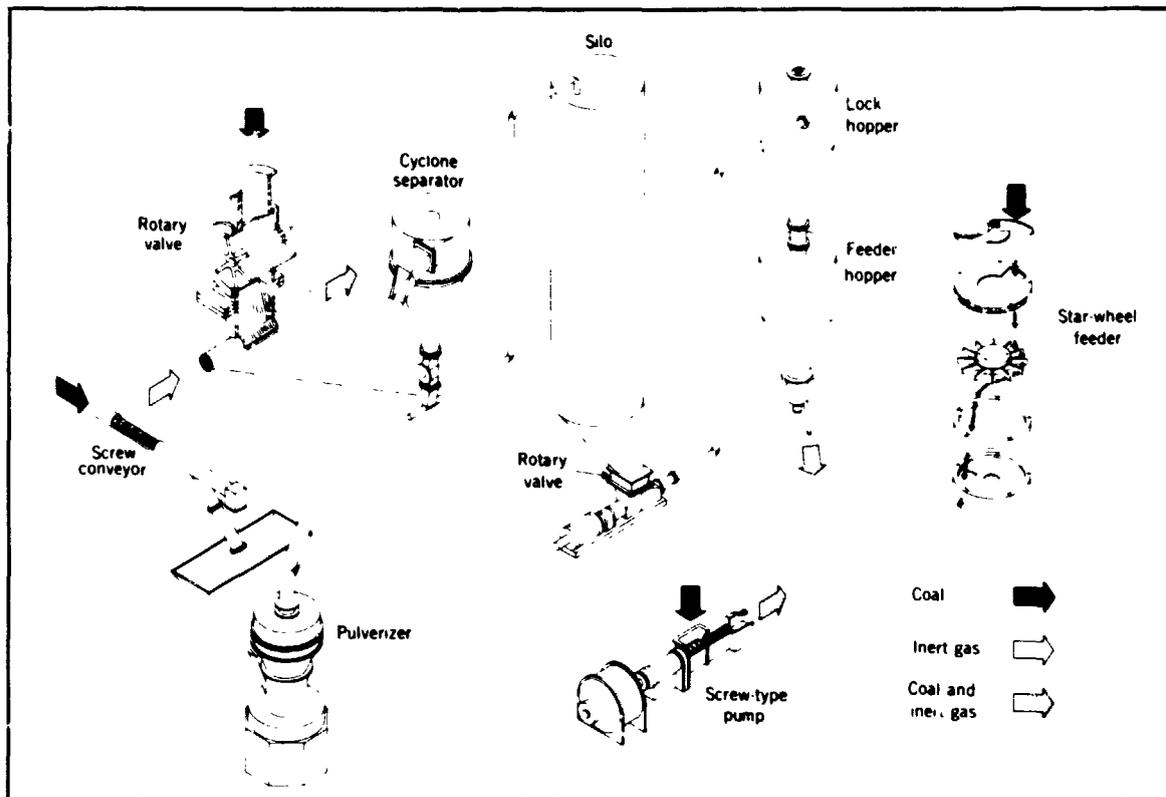
Figure 10. Star-Wheel Coal Feeding System

Results

The coal-feeding system with the Fuller-Kinyon pump satisfactorily fed approximately 3500 pounds of coal per hour to the turbine combustor with a pressure differential of 15 psig. The Fuller Company is currently working on improving the pressure rating.
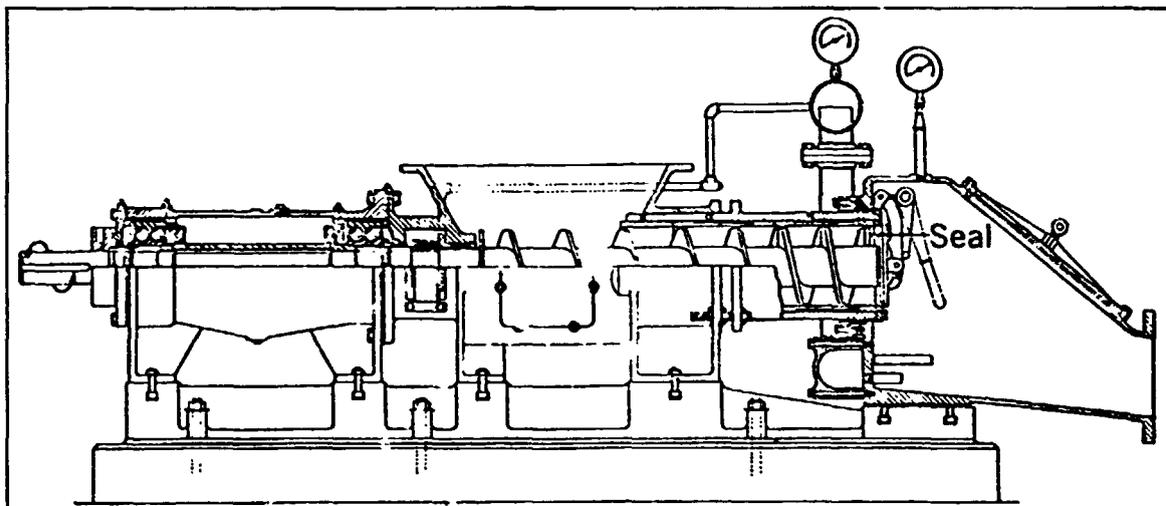
Figure II. Fuller-Kinyon Pump Schematic

2.7 STAR-WHEEL COAL FEEDER WITH INERT GAS AS THE CARRIER GAS

Purpose

To ove come the pui erizer problems, the star-wheel coal feeder system was developed, installed, and tested for the same 1500-hour test run on a gas turbine in 1962. One objective of this program was to determine the system's reliability.

Features

The system (Figure 10) contained several standard commercially available feeders.[35,37,38,40,44,45] The first was a standard screw conveying 3/16-inch x 0 coal; the second was a rotary valve operating at 7 to 15 psig differential. Another rotary valve at the bottom of a 10-ton capacity storage silo was designed and built a* MERC and operated with balanced pressures. The Fuller-Kinyon (screw-type) pump operated with a 15 psig differential pressure. The lock hopper star-wheel feeder combination was used with the star-wheel feeder operating as described in the steam-pickup or Bailey feeder.

Results

The system was automated, completely reliable, and satisfactorily transferred pulverized coal to the coal-fired gas turbine for about 2000 hours. The demonstrated rate was 3500 pounds of coal per hour. The disadvantages of the system were the cost of inert gas and the complexity of the system.

2.8 PETROCARB FEEDER

Purpose

During the early 1960's, the Bureau of Mines gained experience with the Petrocarb feeder while investigating the feasibility of pneumatic transport of pulverized coal in small lines. The program's objective was to develop basic information on pressure drops in a pneumatic system.

Features

The Petrocarb system installed at MERC consisted of a weigh hopper and a receiver hopper. The weigh hopper and receiver hopper were connected by 250 feet of 2-inch or 3-inch pipe, and operated as a batch system. Inert gas (88% N_2, 12% CO_2) was used as the conveying medium.

Results

The system had the capacity to feed coal at a rate of 60 tph at pressures up to 100 psig. Since this is a sophisticated rate control system, it depends largely on instrumentation. The system performed satisfactorily for a complete series of tests which lasted about four months.

3. RECENT PROGRESS (1966-1977)

The feeders discussed in this section are recent or on-going projects at the Morgantown Energy Research Center.

3.1 LOCK HOPPER SYSTEM

Purpose

From mid-1960 to early 1970, the lock hopper coal-feeding system was used in the MERC gas producer. The feeding system was to deliver coal (2 inches x 0) at pressures up to 300 psig.

Features

In the lock hopper operation, coal is dropped from a feed hopper into a lock hopper at atmospheric pressure. The lock hopper is then pressurized to slightly above the coal gasifier operating pressure. The coal is discharged from the lock hopper, the hopper is isolated, and the pressurizing gas is vented in preparation for the next feeding cycle. In the first setup on the producer, the coal was choke fed to the gasifier by lock hoppers. In the second (present) setup (Figure 12), the coal is transferred through a rotary valve and an inclined screw into the gasifier. The screw is MERC-designed and fabricated and serves as a scraper rather than a feeder.

Results

Based on our previous experience, the lock hopper system is the most reliable and trouble-free system for feeding dry solids. However, the first problem occurred when this system was used with our gas producer and required larger valves (10 inch) to facilitate feeding of large size coal. The system operates, but valve maintenance has been a constant problem. The system rate is 1 ton per hour for pressures up to 300 psig.
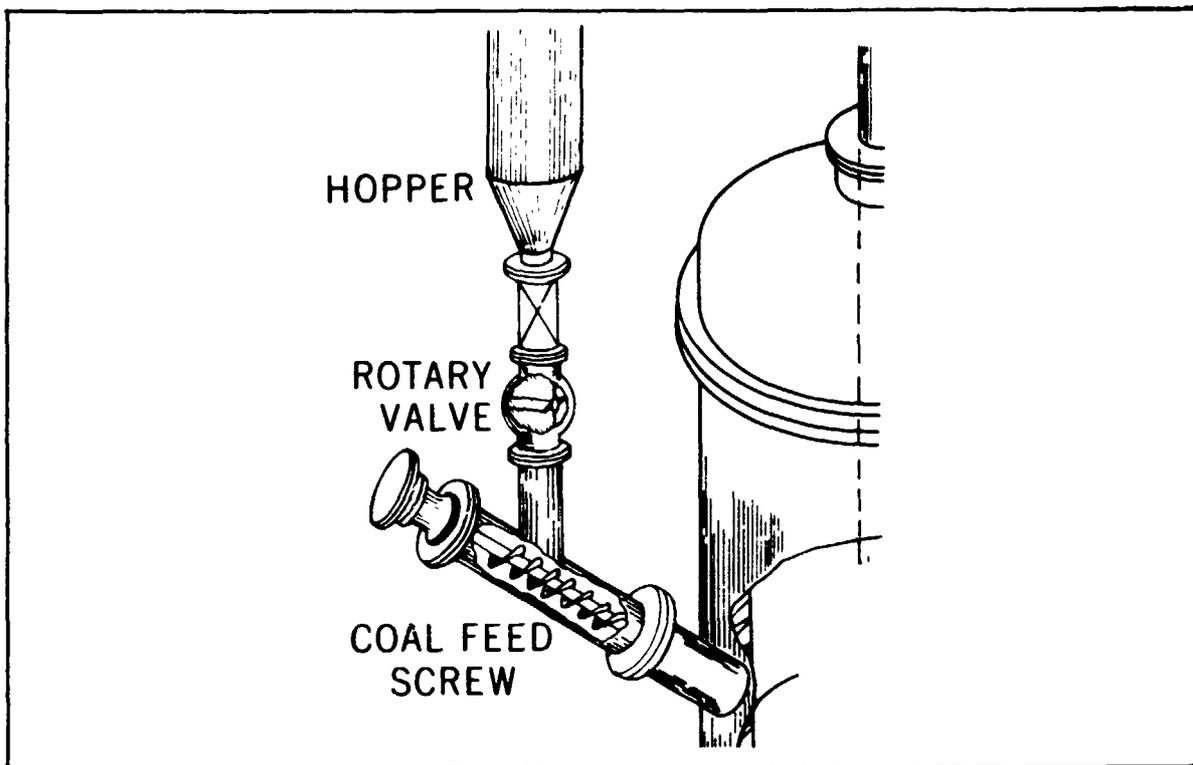
Figure 12. Lock Hopper System with MERC Gas Producer

3.2 VALVE DEVELOPMENT FOR LOCK HOPPER SYSTEM

Purpose

Over the past two years, a valve R&D effort has been underway at MERC. The objective is to develop valves which will be operable, easily maintainable, and have a long life when used in lock hopper systems in coal conversion plants. This development program was initiated to solve the continuing problems of using larger valves and higher pressures in lock hopper systems.[10]

The project was divided into two programs:

● Prototype Valve Test Program

● State-of-the-Art (SOA) Valve Test Program

The objective of prototype valve testing is to design an efficient lock hopper valve capable of handling solids with long life and minimum maintenance cost. The valve designs should also have scale-up capability for use in pilot plant testing and subsequently for commercial plant use.

The prototype valves being furnished ERDA by contract will be available for evaluation in May 1978, with units for pilot plant operation to be available a year later. To bridge this gap, commercially available designs or slight design modifications are required to provide valves for the current pilot plants and early demonstration plants. The SOA Program has been initiated to help identify appropriate valves.

The objective of SOA valve testing is to generate a valve life cycle and engineering data base that will provide a comparative baseline from which to measure improvements achieved by the prototype valves.

Features

For the purpose of this testing, a generic classfication was developed to identify types of valves for specific lock hopper applications. Figure 13 depicts typical lock hopper valve applications and indicates the four different types of service:

- Type I - Feed system valves generally located at the inlet to a loading or injection lock hopper

- Type II - Feed system valves typically located between the loading lock hopper and the gasifier or reactor

- Type III - Discharge system valves usually located at the gasifier or reactor outlet. These discharge into the char or ash lock hopper

- Type IV - Slurry discharge system valves located at the ash lock hopper outlet.

MERC has the capability to test valves with solids up to 600 psig and ambient temperature. By fall of this year, that capability will be increased to 1200 psig and $600^{\circ}F$. Two additional systems are currently under design and scheduled for completion next year, one for testing Type IV slurry discharge valves and the other for testing with solids at temperatures up to $2000^{\circ}F$. MERC also has a metrology laboratory to support this program and works closely with Oak Ridge National Laboratory, Argonne National Laboratory, and the Bureau of Mines for materials selection and failure analysis.

Results

SOA valves have been selected and ordered from nine supplying companies. Selection was made on the basis of a technical evaluation of those design

features that showed promise of reducing the primary valve failure mode of seat scoring. Eight valves have been through a screening test and the metrology laboratory. Tests with solids are scheduled for June 1977.
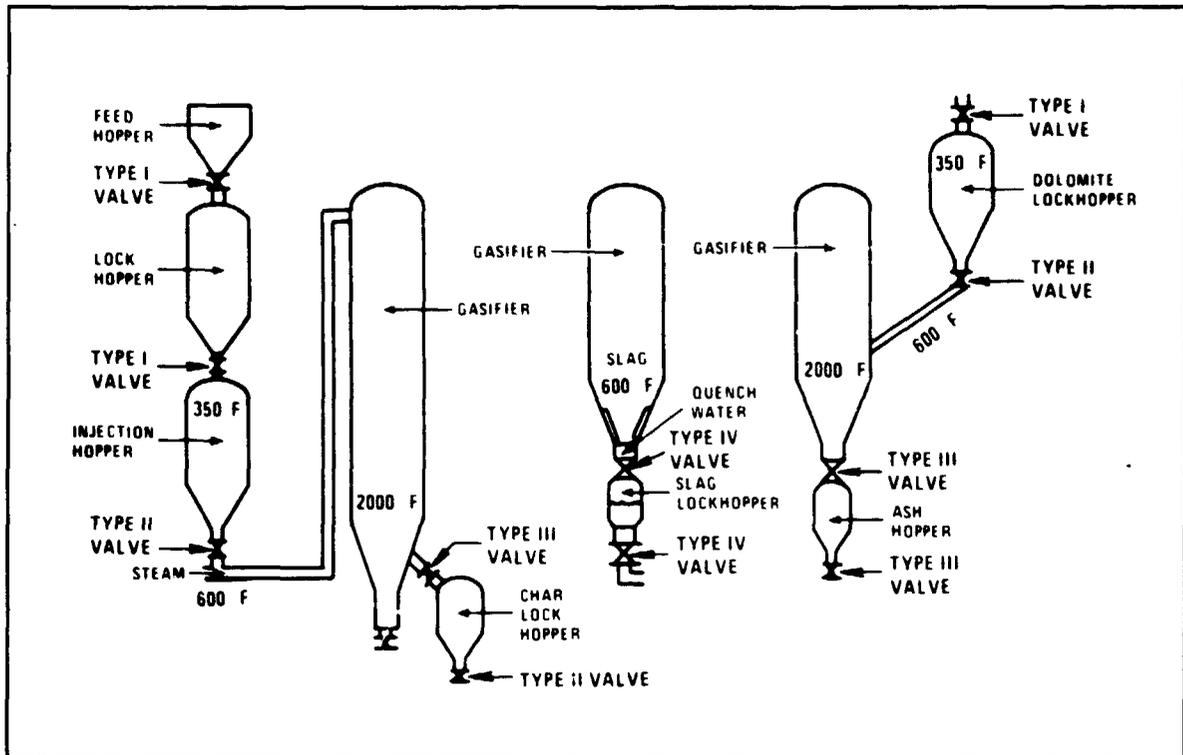
Figure 13. Typical Lock Hopper Valve Applications

3.3 SLURRY FEEDER

Purpose

The development of a steam-dried slurry-feed system is a program which started in FY 1977 and is expected to be completed in three years. The objective is to develop a coal-feed system for high-pressure processes using crushed coal. The method is to pump a coal-water slurry to achieve operating pressure and then dry the slurry in a vertical lift dryer using superheated steam. Steam was selected since it is anticipated that it will be used in other parts of the conversion process. This development program provides technology support for plants such as Bi-Gas.

Earlier work on coal-in-water suspensions fed to a pressurized gasifier started in the early 1960's. Although research and development work has been intermittent, the MERC programs, involving coal-slurry feeding, continued step-by-step to try to find solutions for problems encountered in earlier work.

Previous work with slurry feeding at pressures to 450 psig showed that major problems arose when indirect heat exchange was used to vaporize the water in the slurry. Temperatures had to be limited to about 600° to 650°F to avoid problems from plasticity of the coal. Also, even at these relatively low temperatures, problems occurred when part of the ash dissolved in the water and was deposited downstream on the inside wall of the steaming section.

Features

This current slurry-feeding project uses superheated steam to dry a pressurized coal-water slurry in an entrained bed, thereby eliminating problems of indirect heating. MERC is developing the system (Figure 14) in a pilot plant nominally operating at rates of 300 to 1000 lb/hr of coal-water (50wt% of coal) slurry. The slurry will be pumped into a vessel at 250 psig where it will be contacted by superheated steam. Water in the slurry will be vaporized, creating a stream of dry coal contained in superheated steam at pressure. The two components will be split in cyclone separators and steam will be vented. Eventually, steam will be recycled through a compressor and superheater.
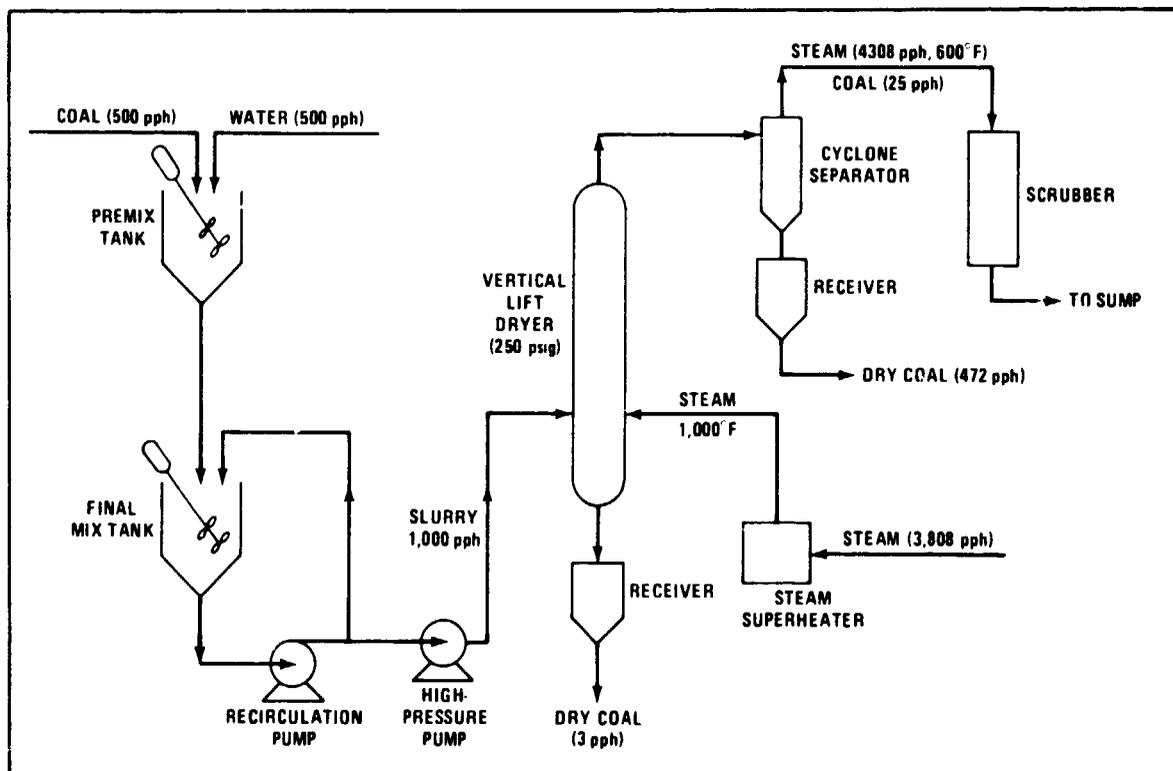
Figure 14. Steam-Dried Coal Slurry Feed System

The series of MERC development work on coal slurries are discussed in detail in several references listed in the bibliography.[8,9,10,11,25,51,55]

Results

The design is complete, and construction of the system has started. Operation is scheduled to begin in early October 1977.

3.4 ROTARY PISTON FEEDER

Purpose

The research program on new design concepts for coal pumps was initiated locally to improve dry coal-feeding systems. Detailed designs of feeding mechanisms will be developed, and sealing wear as well as lubrication problems will be investigated. A low pressure (100 psig) prototype will be built and tested.

Features

The pump will consist of a rotary feeder with a piston (Figure 15) that seals the feeder void after the coal is dumped into the pressurized vessel, thereby preventing gas loss. Various means of activating the piston to give the desired feeding action into the pressurized chamber will be investigated. The present design uses a fixed cam with a roller follower to activate the piston and piston spring. The main advantages of the system are its compact size, simplicity in operation, minimum use of gas, and variable capacity depending on the number of cylinders used in the unit. In addition, this pump may be used on all sizes of coal from run-of-mine to pulverized.

Results

This program is in process.[10] Shop drawings for the prototype are complete, and construction on the prototype will begin this year. Cost of this program to date is less than $50,000.

3.5 EXTRUSION FEEDER

Purpose

In 1970, MERC's program on formed coke investigated production of a uniformly sized blast-furnace fuel using a hot extruder. While working on this program, it was recognized that the same technique may be applied to high-pressure coal feeding.

FEEDING COAL INTO PRESSURE VESSELS

CONCEPT: Rotary Piston Feeder with Controlled Cylinder Volume

DATE 3/9/73
D C SHERREN
D C STRIMBECK

NOTES

1 CONCEPT SHOWN AS SINGLE PISTON FEEDER MULTIPLE PISTON FEEDER WILL IMPROVE FEED RATES (SEE CHART)

2 BORE & STROKE MAY BE DETERMINED AS TO FEED DESIRED COAL SIZES SHOWN HERE AS BORE = 5" & STROKE = 2" WITH WHEEL DIA = 18"

3 FIXED CAM MAY BE DESIGNED FOR SIMPLE HARMONIC MOTION OF ROLLER FOLLOWER

4 FEED RATE MAY BE VARIED WITH BORE & STROKE (FIXED) AND SPEED (MAX APPROX 30 RPM SEE CHART)

5 MECHANICAL SEALS TO BE DEVELOPED FOR MAXIMUM PRESSURE DIFFERENTIAL ACROSS WHEEL

PISTON & RINGS
OPEN CYLINDER

SECTION B B
SHOWING PISTON/WHEEL DRIVE AND FIXED CAM ARRANGEMENT

PISTON/WHEEL DRIVE

FIXED CAM

ROLLER FOLLOWER

APPROXIMATE* RATES (LB/HR) FOR A ROTARY PISTON FEEDER WITH MULTIPLE CYLINDERS

SPEED (RPM)	CYL BORE (IN)	CYL STROKE (IN)	COAL RATE IN POUNDS/HOUR			
			1 CYL	2 CYL	4 CYL	8 CYL
10	2	2	60	120	240	480
	5	2	400	800	1600	3200
	5	5	1000	2000	4000	8000
20	2	2	120	240	480	960
	5	2	800	1600	3200	6400
	5	5	2000	4000	8000	16000
30	2	2	180	360	720	1440
	5	2	1200	2400	4800	9600
	5	5	3000	6000	12000	24000

*ASSUMING BULK DENSITY OF COAL = 30 LB/FT^3 AND CYLINDER VOLUME IS FILLED (FOR PARTIAL FILLING REDUCE RATES BY PERCENT OF VOID VOLUME)

COAL FEED PORT (LOW PRESSURE)

OPEN CYLINDER

PISTON & RINGS
PISTON SPRING
FIXED CAM
ROLLER FOLLOWER
ROTATING PISTON/WHEEL
PISTON ROD WITH FOLLOWER AND SEALS

COAL DISCHARGE PORT (HIGH PRESSURE)

SECTION A A (1)
INTAKE POSITION
ROTATION 0° (360°)

NOTE AS WHEEL ROTATES FROM THE CLOSED POSITION (270°). THE PISTON IS WITHDRAWN FROM TDC TO BDC BY ACTION OF THE FIXED CAM CYLINDER FILLS IN THIS POSITION

OPEN COAL FILLED CYLINDER

SECTION A A (2)
DISCHARGE POSITION
ROTATION 90°

NOTE WHEEL ROTATES WITH PISTON AT BDC TO 90° (FEEDING) ON THE FIXED CAM PISTON SPRING CAUSES FOLLOWER TO FORCE MOVEMENT AGAINST HIGH PRESSURE DIFFERENTIAL

SECTION A A (3)
DISCHARGE POSITION
ROTATION 180°

NOTE WITH PISTON IN DISCHARGE POSITION GAS CARRY OVER IN THE CYLINDER IS LIMITED TO THE LEAKAGE AROUND SEALS IF WHEEL IS PRESSURIZED INTERNALLY WITH INERT GAS OR SOME OTHER BUFFER FLUID LOSSES FROM DISCHARGE ENVIRONMENT WILL BE NON EXISTENT

COAL DISCHARGE INTO PRESSURIZED PROCESS

SECTION A A (4)
CLOSED POSITION
ROTATION 270°

NOTE CYLINDER VOLUME IS CLOSED AND SEALED NO GAS CARRY OVER TO INLET SIDE OF WHEEL PISTON BEGINS TO WITHDRAW TO BDC FROM THIS POSITION BY ACTION OF SPRING LOADED FOLLOWER

Figure 15. Rotary Piston Feeder

Features

A commercially available laboratory-size extruder (Figures 16 and 17) was improved for use in the Formed Coke Program. It consists of a powerful drive system and a wide selection of speed ranges, a modified barrel and auger, and automatic temperature controls. The barrel is a 2½-incn, schedule-40, stainless steel pipe electrically heated by resistance coils wrapped around the pipe and insulated to prevent appreciabl_ heat loss. The barrel is divided into two sections, conveying and forming. Each section requires about 1500 watts for heating the barrel to proper operating temperature. The auger is heated by a cartridge-type heater located within the center of the shaft.

ELECTRIC-RESISTANCE-COILS

AUGER-TYPE EXTRUDER
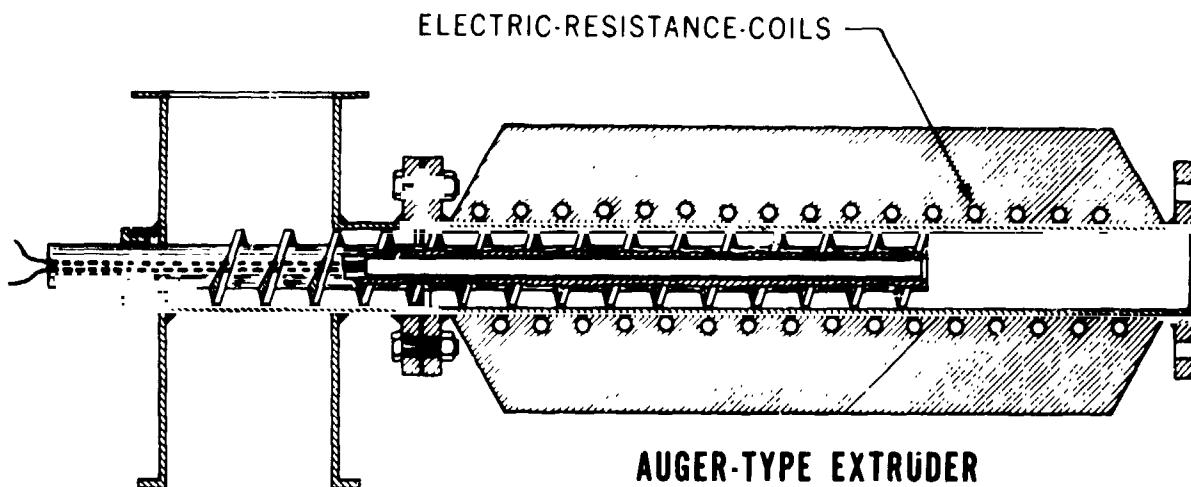
Figure 17. Extruder Schematic

Results

The problems encountered were coke clogging of the auger or barrel, premature coking, excess volatile matter buildup which caused "blowouts," and incomplete heat transfer throughout the coal mixture. To solve these problems, coals were blended, preheating was tried, and variable screw depths were used in conjunction with tapered barrels. These attempts were only partially successful. The program emphasis shifted and the program was stopped. As a result, the initial plan of developing the extruder as a feeder was not pursued.
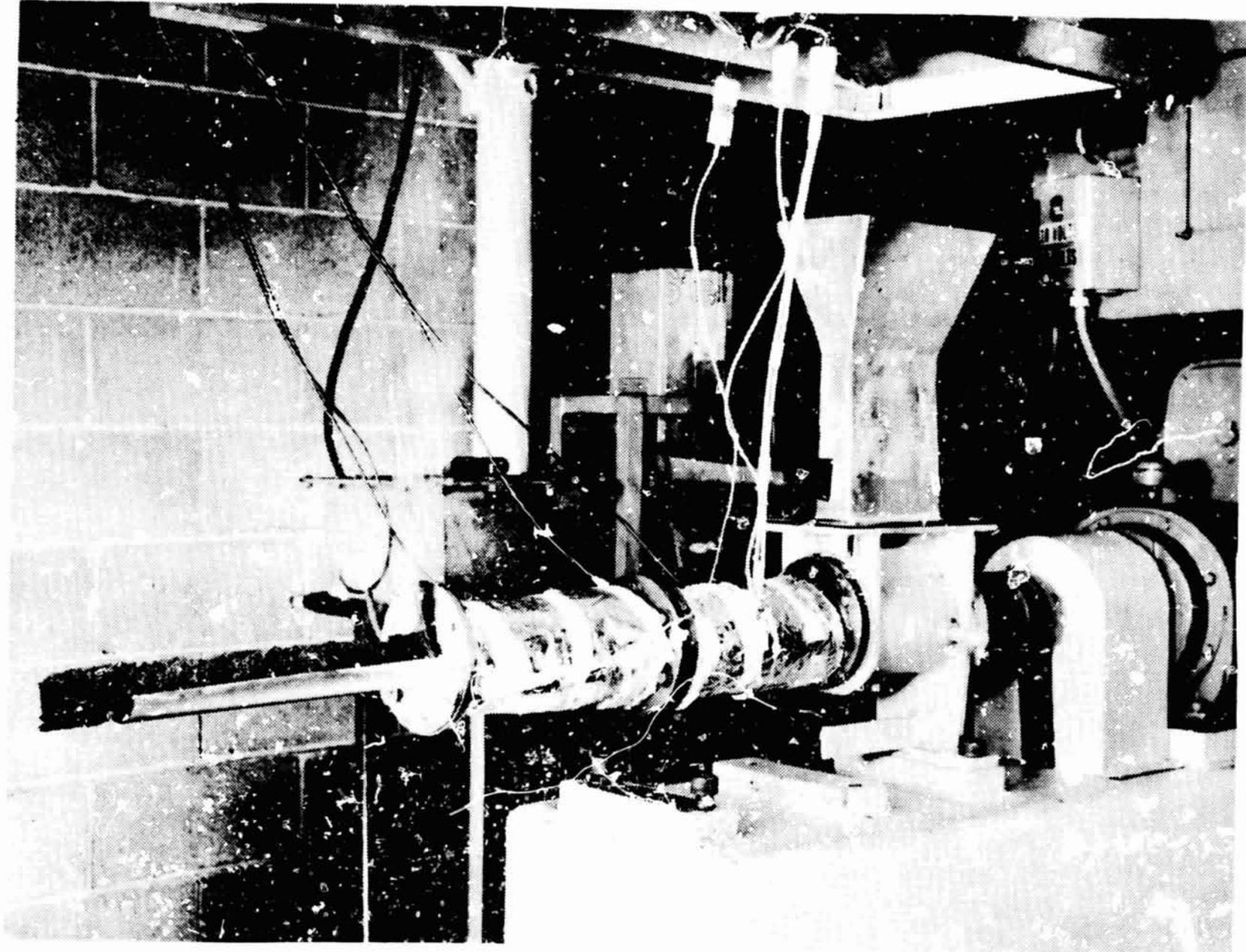
Figure 16   Auger Extruder

3.6 FUTURE TESTING

Starting in 1978, MERC will begin full-scale testing of two coal extrusion systems. An 1 to 5 tph high-pressure extruder being developed by Ingersoll Rand will be installed in the 42-inch pressurized fixed-bed gas producer. Tests will be conducted in the 100 to 300 psig range. A smaller 100 pph atmospheric-pressure unit will be installed in the 18-inch atmospheric fixed-bed gasifier. The objective of both of these tests is to provide in-field ope· .ing data and experience with both extrusion systems. Design, procureme..t, and fabrication of the modifications are underway. Delivery of the extruders is planned for January 1978.

4. CONCLUSION

Among the coal-feeding systems discussed, the lock hopper system is the most widely used. Such a system is very successful, especially when using smaller valves (less than 4 inch) and at lower pressures (up to 450 psig). For the future, in addition to improvements in lock hoppers, other feeding systems need to be developed which are smaller and which have low capital and operating costs, require minimum pressurizing gas, and include mechanical design requiring minimum maintenance.

The current activities on coal-feeding development at the Morgantown Energy Research Center are presented in Figure 18 (exclusive of positive-displacement piston-type coal pump). This chart identifies various alternatives to feeding coal at pressure. However, each of these systems needs further development before it can be adapted to a particular coal conversion process.

LOCK HOPPER SYSTEM

POSITIVE DISPLACEMENT
PISTON - TYPE COAL PUMP

ALTERNATIVE COAL FEED SYSTEMS

EXTRUDER FEEDER

GROUND COAL TO
PRESSURE - FEEDING
SYSTEM

CHARGE HOPPER

FEED HOPPER

COAL

COAL PREPARATION

LOCK HOPPER FEEDER

COAL

GASIFICATION

ATMOSPHERIC

LIQUEFACTION

PSIG

PYROLYSIS

COAL CONVERSION

SLURRY HEATING
AND
FLASHING SYSTEM

ROTARY PISTON FEEDER

SLURRY FEEDER (COAL + WATER)

Figure 18.   Alternative Coal Feed Systems

77-55

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COMPARATIVE DESCRIPTION OF COAL
FEEDING SYSTEMS FOR FIXED BED
PRESSURE GASIFICATION

E. Funk

Kamyr Inc.
Glens Falls, New York

R. Reimert

Lurgi Mineralöltechnik GmBh
Frankfurt, West Germany

ABSTRACT

The trend in coal gasification can be characterized as increasing
pressure, increasing throughput per gasification unit and, as far as possible,
the use of run of mine coal.

With regard to these criteria coal feeding systems are discussed which
are capable of feeding 20-100 T/H and the range of pressure is up to
100 bar. Most emphasis is placed on dry feeding systems, of which com-
mercial proven and those being developed are dealt with.

The systems outlined are subdivided into continuous and intermittent
and the influence of each system on lock gas losses and reactor design is
shown.

Finally a cost estimate based on a Lurgi gasifier as example is pre-
sented which indicates the areas of preferred application and permits con-
clusions to be drawn regarding the economics of the various systems.

The presentation consists of two parts which will be delivered by
Dr. Rainer Reimert, Lurgi Mineraloeltechnik GMBH, Frankfurt/Main, and
Erwin Funk, Kamyr Inc. Glens Falls.

COMPARATIVE DESCRIPTION

OF COAL FEEDING SYSTEMS

FOR FIXED BED PRESSURE GASIFICATION

by

Erwin Funk, Kamyr Inc., Glens Falls, N.Y., USA.

Rainer Reimert, Lurgi Mineralöltecnnik GmbH,
Frankfurt (Main),
Federal Republic of Germany.

The subject of this paper is briefly "Coal Feeding Systems for Fixed Bed Pressure Gasification". The Lurgi lock hopper system which has been successfully applied for this duty for many decades is discussed first. Thereafter a pump is presented which can handle solids in dry condition and which has been developed to a pilot scale. The discussion over the dry feeding systems concludes with a comparison of the costs of investments.

The slurry systems are dealt with next and the necessary requirements for their use in connection with fixed bed gasification are outlined. At last Erwin Funk will present the Kamyr feeder system which has already proved its merits for pipeline feeding and which we hope can soon be tried out for fixed bed gasification.

1. Lurgi Lock Hopper System

Figure 1 shows a lock with closures which operates discontinuously. Coal travels from a bunker through the open top closure into the atmospheric lock, the bottom closure being closed. A vertically movable filling pipe prevents overfilling of the lock. When the lock is filled, the top closure is also closed and the lock pressurized with gas to

reactor pressure. External gas or cooled product gas is used for pressurizing. In the latter case a lock gas cooler is required in plants without complete condensation to avoid a pressure drop within the lock due to cooling and condensation of vapours, which besides could lead to undesired agglutination.

The contents of the lock is discharged through the opened bottom closure into an intermediate hopper which in the case of fixed bed gasification is arranged direct in the reactor above the coal distributor. Unloading is then in accordance with the coal consumption at the pertinent gasifier output and the level is indicated either by a level gauge or by temperature measurement. After complete unloading the bottom closure is closed and the lock vented via a valve.

The vent gas is collected in a gas holder and used, for instance, for firing. It can also be recompressed and fed to the product gas system. The gas holder or compressor are common for a plant consisting of several gasifiers.

Figure 2 presents the pressure-time chart of one lock cycle which was measured on a lock of roughly 5 m^3 volume in the Westfield Plant. 100% on the abscissa correspond to about 5 minutes. This time obviously depends on the coal flowing properties. 4-7 lock cycles per hour are operated in commercial plants. The diagram shows the 4 phases of one cycle: charge, pressurize, feed, vent. The hold point during venting at about 275 psig denotes the pressure test for checking the leak-proofness of the bottom closure.

The closures have hard-faced metallic seating surfaces.
For safety reasons, the closures are so installed that the working
pressure keeps the closure closed which produces an additional
sealing effect. For example, the working pressure of the new
Mark IV gasifier produces a pressing force of about 50 tons.

Figure 3 illustrates a lock connected to the gasifier.
The coal intermediate hopper mentioned already earlier and
serving as storage compartment is arranged between bottom
closure and coal distributor. It is also evident that by making
a few modifications the same system is used as ash lock.

Figure 4 shows a view of the coal locks in the Dorsten
Plant. While in this old plant the lock operations were
controlled manually by the "Captain's Wheel" in the foreground,
modern coal locks operate fully automatically.

The volume of the old Dorsten coal lock was about 5 m^3 and
the design pressure 27 bar. The coal lock for the improved
Mark IV Gasifier at SASOL now has a volume of about 15 m^3
which permits a coal feeding rate of about 70 t/h at a plant
pressure of 30 bar. The maximum working pressure applied sc
far, with the same design principle, was 80 bar in a pilot
plant for lignite hydrogasification in the fluid bed at Union
Kraftstoff Wesseling wh   has operated trouble-free already for
some months.

The losses of compressed gas during lock venting increase with increasing reaction pressure. These losses can be reduced by about 30-40% by using parallel connected double locks with reciprocal pressurizing and venting. The incorporation of additional parallel operated locks would reduce the losses only by a few more percent. The reduction of the vent gas losses is realized at the expense of an increased investmen. for vessels, valves and instruments.

Another but very important advantage of the double lock is its high coal throughput. In view of the parallel connection it is possible to operate the gasifier at 70% of its previous capacity in the event one of the locks fails.

The principle of the double lock with reciprocal pressurizing and venting will be applied in a pilot plant for two-stage fixed bed gasification at maximum 100 bar which is scheduled to go on stream at Dorsten in 1979. This arrangement is shown on the diagram in Figure 5.

The cycle times for the lock system operating at 100 bar are illustrated in Figure 6 for various schemes. The reciprocal pressurizing and venting increase the cycle time of the double lock again by about 30% versus a double lock operated independently. A series connected double lock with reciprocal pressurizing and venting is shown as the last case where, however, only a little increase in throughput can be realized compared to a single lock. The time for feeding the gasifier is assumed to be 100 s in each case.

2. Piston-Type Pump System

The LURGI lock hopper system has the advantages

- that it can handle all sizes and grades of coal in dry
  condition, and practically without size degradation,
- that it can overcome high pressures in one stage, and
- that the system has proved successful during many years of
  commercial operation,

but shortcomings are that the lock operates discontinuously and
that vent gas occurs.

The variations in the coal supply caused by discontinuous
feeding have no negative effects on the fixed bed reactor in
view of its large coal volume. With a view to reducing the
increased output of vent gas particularly at high reactor
pressures another principle was invented about 30 years ago.
This system is being realized at present.

Figure 7 presents the patent drawing of a system in which
the vent gases are reduced to a minimum. It operates similar
to the Lurgi lock except that the top closure has to be regarded
as displacer. Cylinder 'c' is filled from coal bunker 'a'
via dosing facility 'b' with piston 'd' in its upper end position
and bottom closure 'e' closed. This closure can be arranged
similar to that of the Lurgi lock. The cylinder is filled only
so far that piston 'd' can still be moved downwards beyond the
feed port without resistance. Hereby the coal is not pressed.
The piston seals the cylinder compartment holding the coal,
which is then pressurized with product gas through line 'h'.
After pressure equalization bottom closure 'e' can be opened
and the coal drops into reactor 'f'. Piston 'd' follows up

to the lower cylinder end thereby removing coal remnants from the cylinder. After closing the bottom closure only a very small volume, compared to the volume fed, remains which contains gas at reactor pressure. After withdrawal of the piston this gas is available at sub-atmospheric pressure and causes only negligible emission.

A pump operating according to the principle outlined is currently under fabrication for a feed rate of some 20 t/h. The system still has to be tried out for feeding coal into pressure vessels to clarify in particular wear and power consumption. The bottom closure is designed as rotating cylinder with horizontal borehole. The piston front face is adapted to the configuration of the cylinder to keep the empty volume of the lock low when the piston is in its lower end position.

To avoid that the piston always has to be moved over the feed slot, an improvement of the system provides for feeding the coal direct through the piston into the cylinder.

Apart from the drastic reduction of the vent gas rate a further advantage of the system is the quasi-continuous conveying which enables a reduction of the hopper volume in the gasifier resulting in a lower gasifier height.

3. Comparison of Costs

A comparative cost estimate shows that a piston type pump also has economic advantages at least when low throughputs are involved. The investment costs of some Lurgi lock systems and of the piston type pump are plotted in Figure 8 versus the coal throughput. The Sasol lock with a throughput of 50 tons/hr was taken as reference point = 100%. The curves have been established by extrapolation based on the designed throughput. Those costs, which are influenced by the throughput, have been calculated using the power 0.7.

Remarkable are the high costs estimated for the double lock system for the Lurgi pressure gasifier Type Ruhr 100. This is due to the higher investment for material for the lock vessels and for piping and instrumentation required for reciprocal pressurizing and venting. Also the higher plant pressure adds up to the investment costs. Furthermore, these costs have to be viewed under the aspect of partial vent gas recovery for which otherwise an additional compressor station would have to be provided.

The curve for the piston type pump is at the lower end. It should be considered that commercial experience with this system is not yet available so that the actual costs are likely to be higher compared to the original estimate. The resistance of the material selected to the attack by erosion will play a major rôle.

The extrapolation of the costs considers that the capacity of these pumps will presumably be limited so that se eral pumps would have to be arranged in parallel. The cost advantage of the piston type pump is therefore mainly in the range of low coal throughputs.

4. Brief Summary of Dry Feeding Systems

Before discussing the slurry systems a brief summary of the features of the dry feeding systems should be given.

- The Lurgi lock has proved its merits on a commercial scale for pressures up to 80 bar and throughputs of 70 tons/hr. It can handle all coal grades and sizes.

- The quantity of vent gas which increases with increasing pressure can be reduced by the provision of double locks which results in higher investment costs but also in savings in energy. The vent gas can be recovered completely by recompression.

- The provision of a piston type pump for handling dry solids allows quasi-continuous opera..on and reduces the quantity of vent gas drastically. Such pump can have price advantages when low throughputs are involved.

5. Slurry Systems

Apart from the unavoidable vent gas losses, the dry feeding systems c te additional losses due to leakages and emissions during filling rt the locks. These gases which contain mainly

CO and H_2S as harmful components can be largely disposed of by exhausting and subsequent incineration. The permissible CO and H_2S emission limits could nevertheless be reached when severe environmental pollution control requirements exist.

For this reason and for the continuous feeding in the event of very high throughputs per unit slurry systems can be a valuable alternative, provided of course that the gasification process remains largely unaffected. Counter current flow of coal and gasification agent are typical for fixed bed gasification. Therefore, liquids introduced with the coal do not take part in the reaction unless very high boiling hydrocarbons are involved. They are evaporated in the reactor top section which reduces the possibility of waste heat recovery from the raw gas. A rather complete separation of the transport fluid from the coal is therefore desirable. The separated liquid should be recirculated as it contains coal fines and dissolved gases. The moist coal entering the gasifier top section must have adequate flowing properties.

Fixed bed gasification of bituminous coal of the preferred sizes of 3 - 50 mm yields a residual liquid content in the coal of less than 15% after separation so that water can be used as transport fluid. The residual water content of other coal grades would have to be determined on a case-to-case basis.

Erwin Funk will now present the Kamyr slurry feeding
system and its many advantages versus conventional slurry systems.

Finally I would thank my colleagues not named here who helped
in preparing this paper.
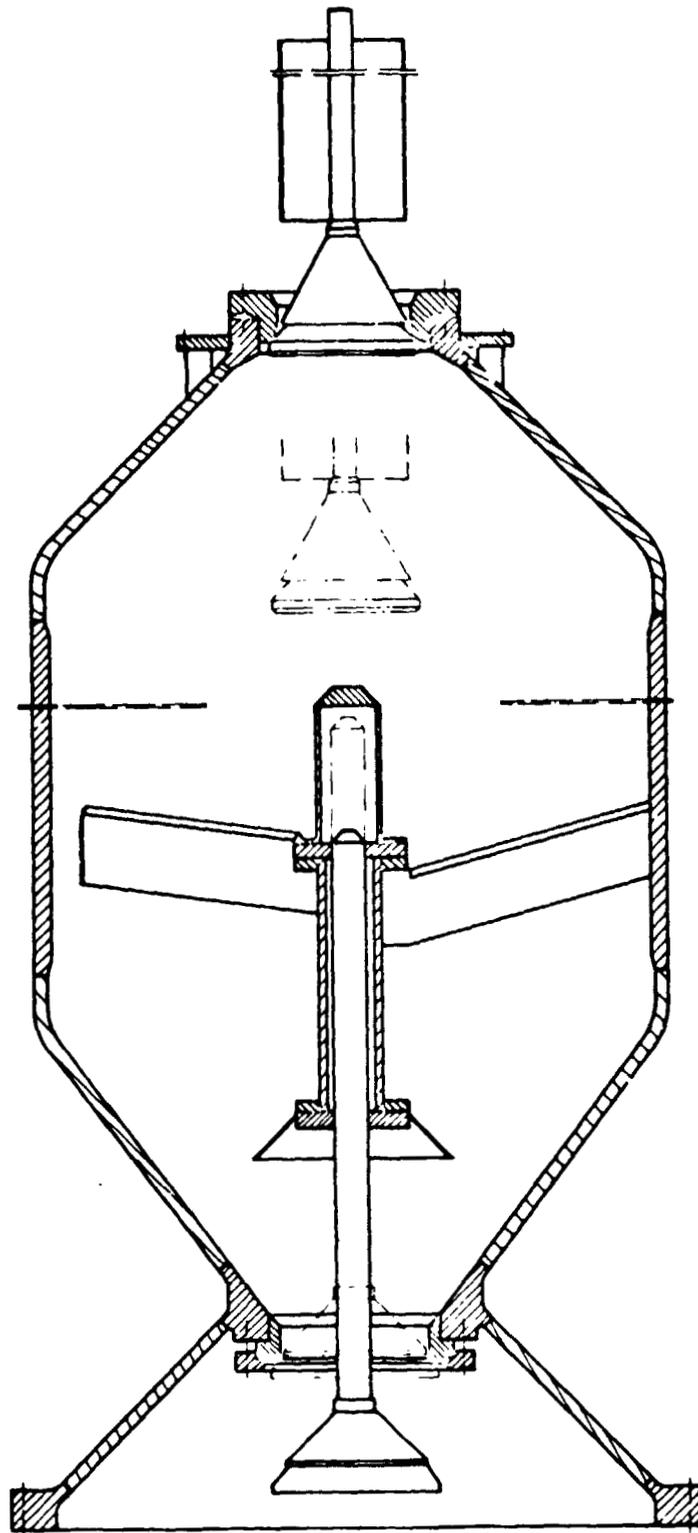
FIG. 1 LURGI COAL LOCK

FIG. 2   COAL LOCK CYCLUS

FEED COAL

COAL LOCK

RECYCLE TAR

DRIVE

STEAM

SCRUBBING COOLER

DISTRIBUTOR

GAS

GRATE

GRATE DRIVE

WATER JACKET

STEAM + OXYGEN

ASH LOCK
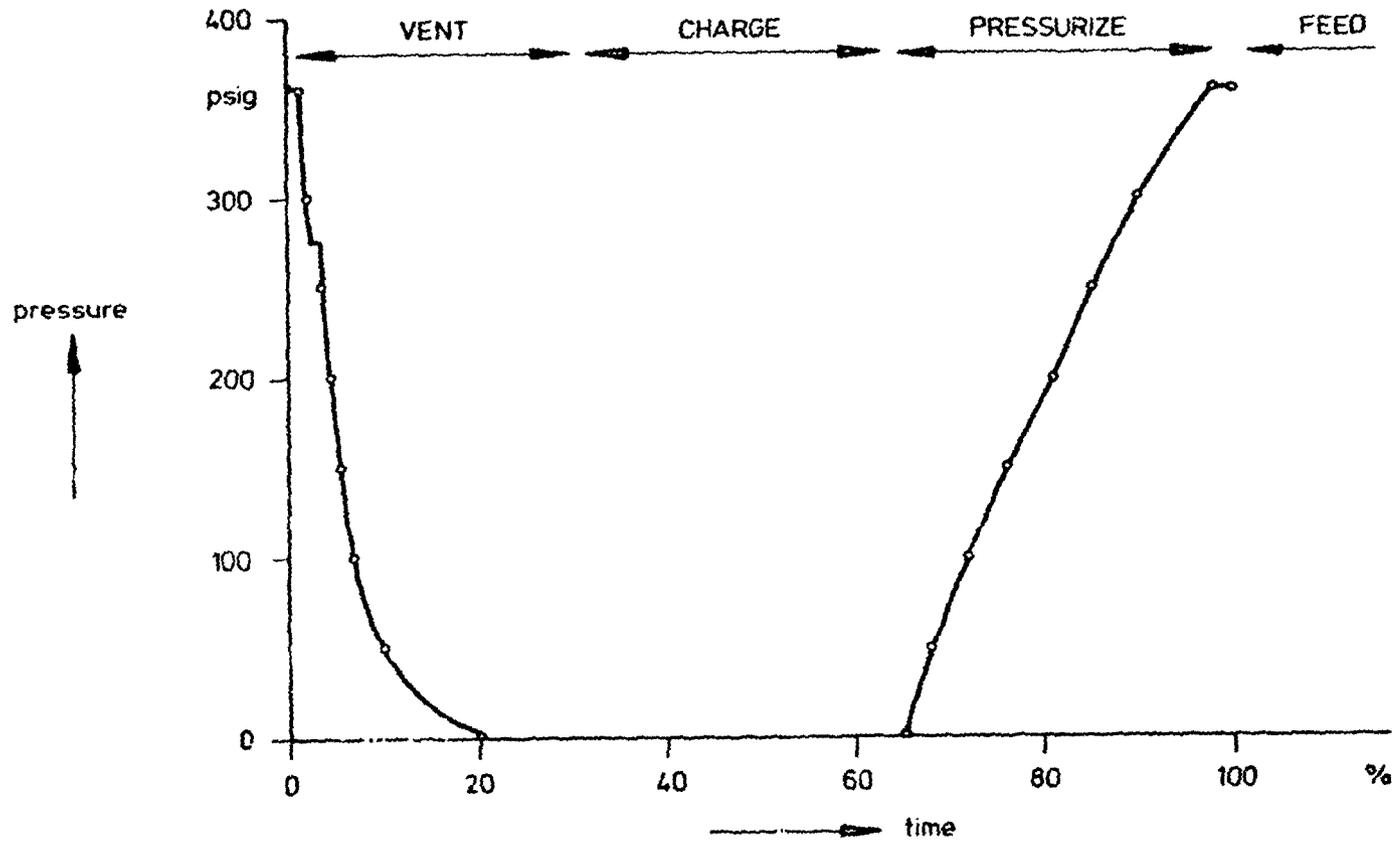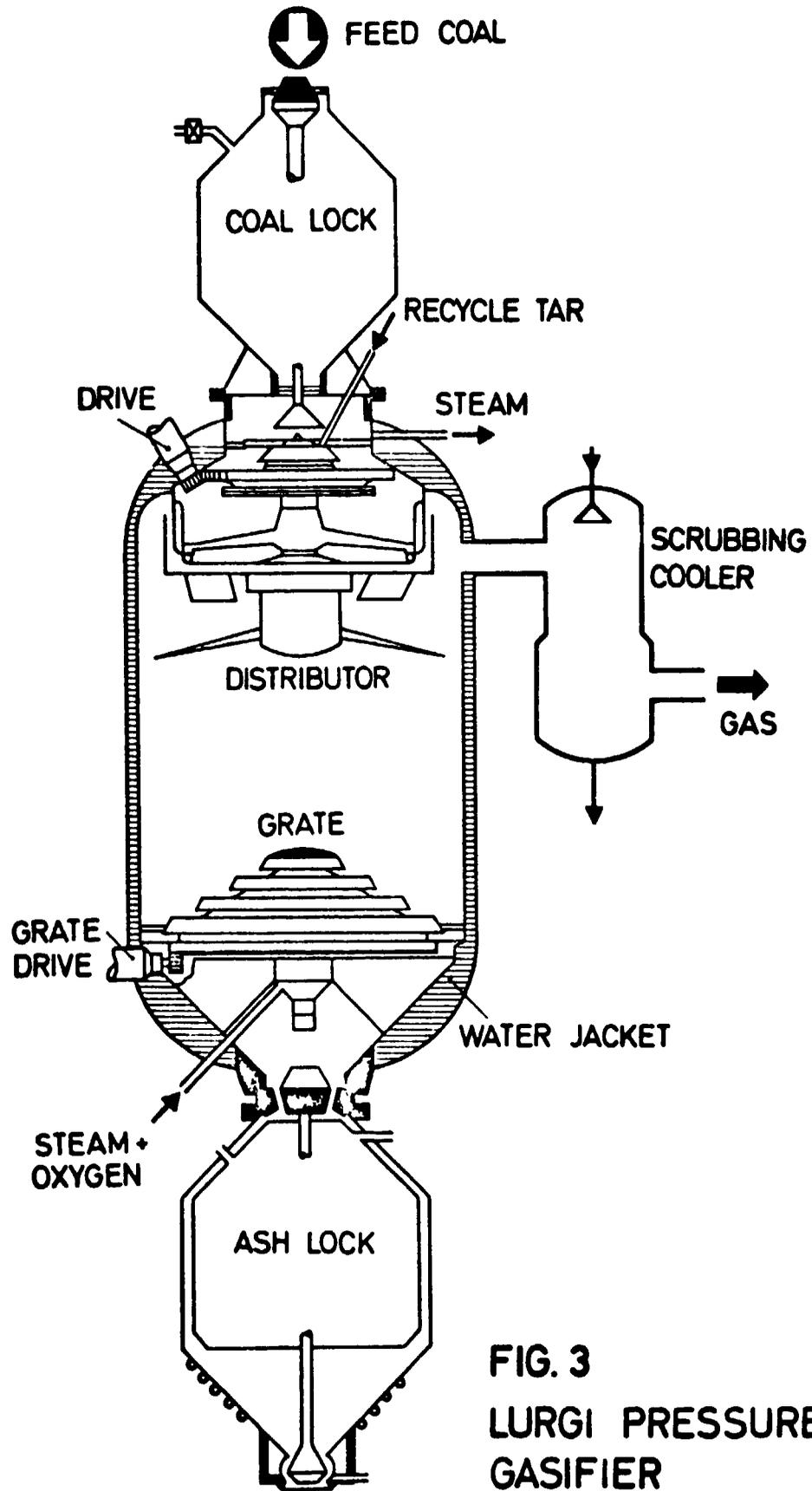
FIG. 3

LURGI PRESSURE GASIFIER
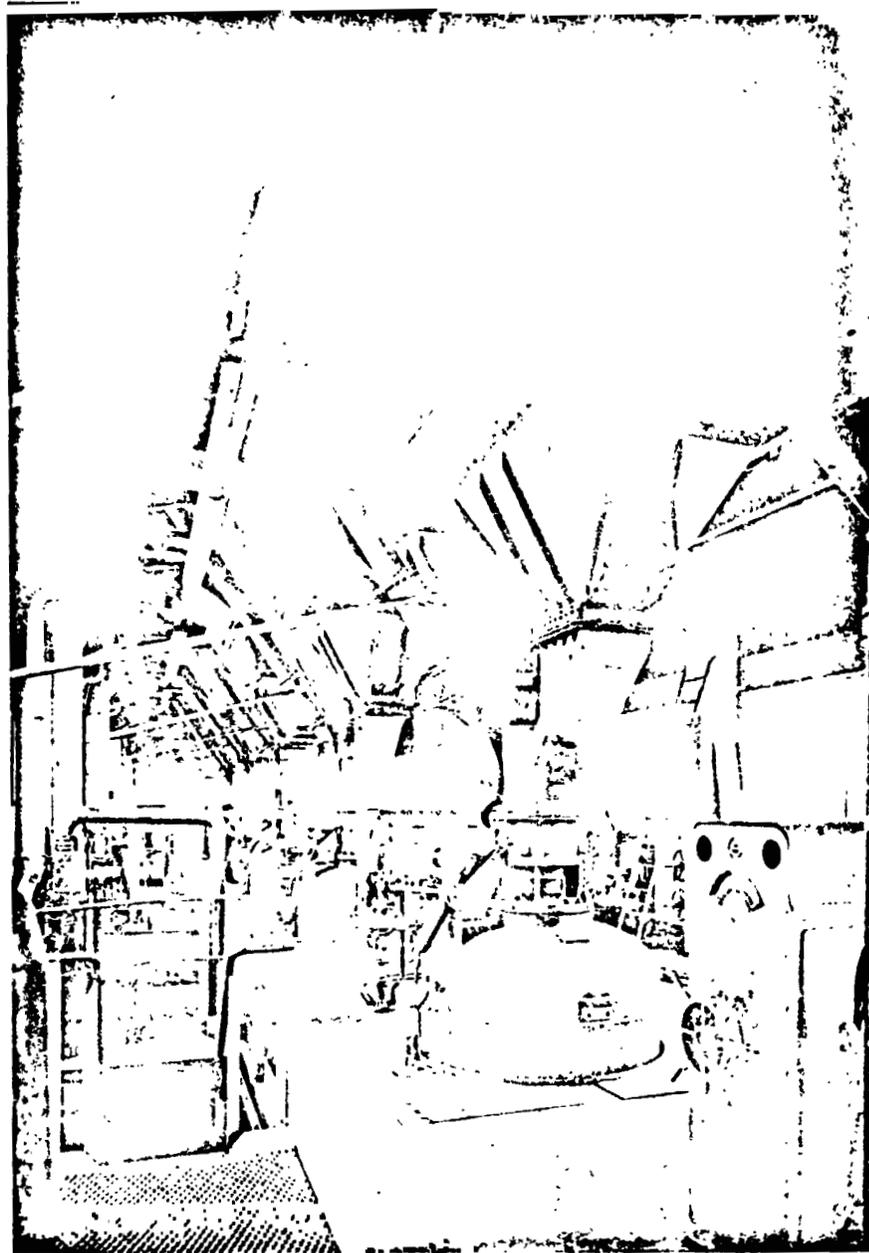
FIGURE 4 COAL LOCKS AT DORSTEN

1 LOCK GAS COOLER
2 GASIFIER
3 VENT GAS TANK
4 COAL LOCK
5 COAL BIN
6 SUCK OFF FAN

RAW GAS

BURNER

RECOMPRESSION
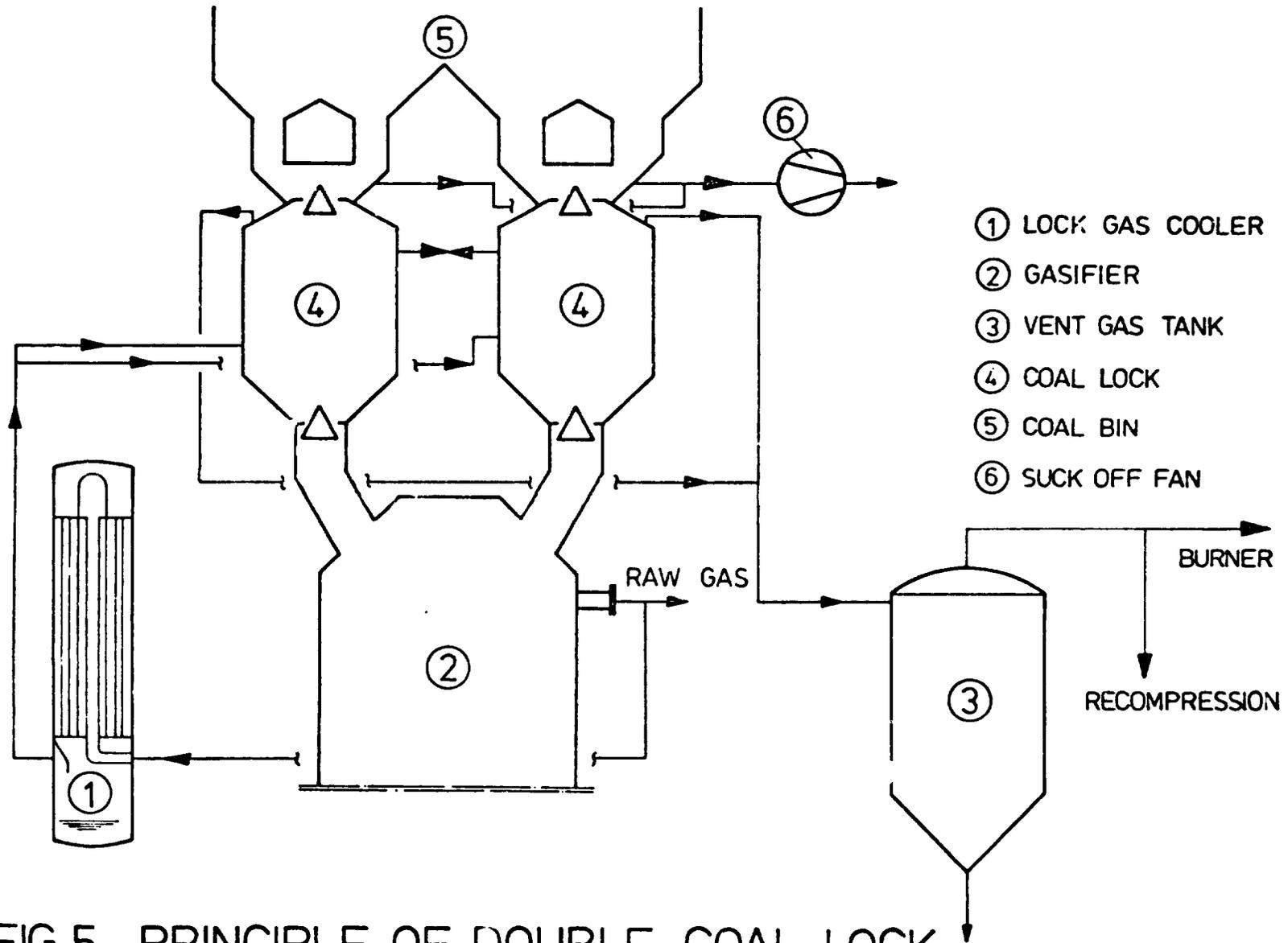
FIG.5 PRINCIPLE OF DOUBLE COAL LOCK

COAL LOCK ARRANGEMENT			PERIOD OF LOCK CYCLUS	
			SECONDS	%
SINGLE LOCK			594	100
DOUBLE LOCK	PARALLEL	WITHOUT *	347	58
		WITH *	450	76
	IN SERIES	WITH *	550	93

* RECIPROC/ LY   PRESSURIZING   AND   VENTING

FIG. 6 COMPARISON OF CYCLUS PERIODS
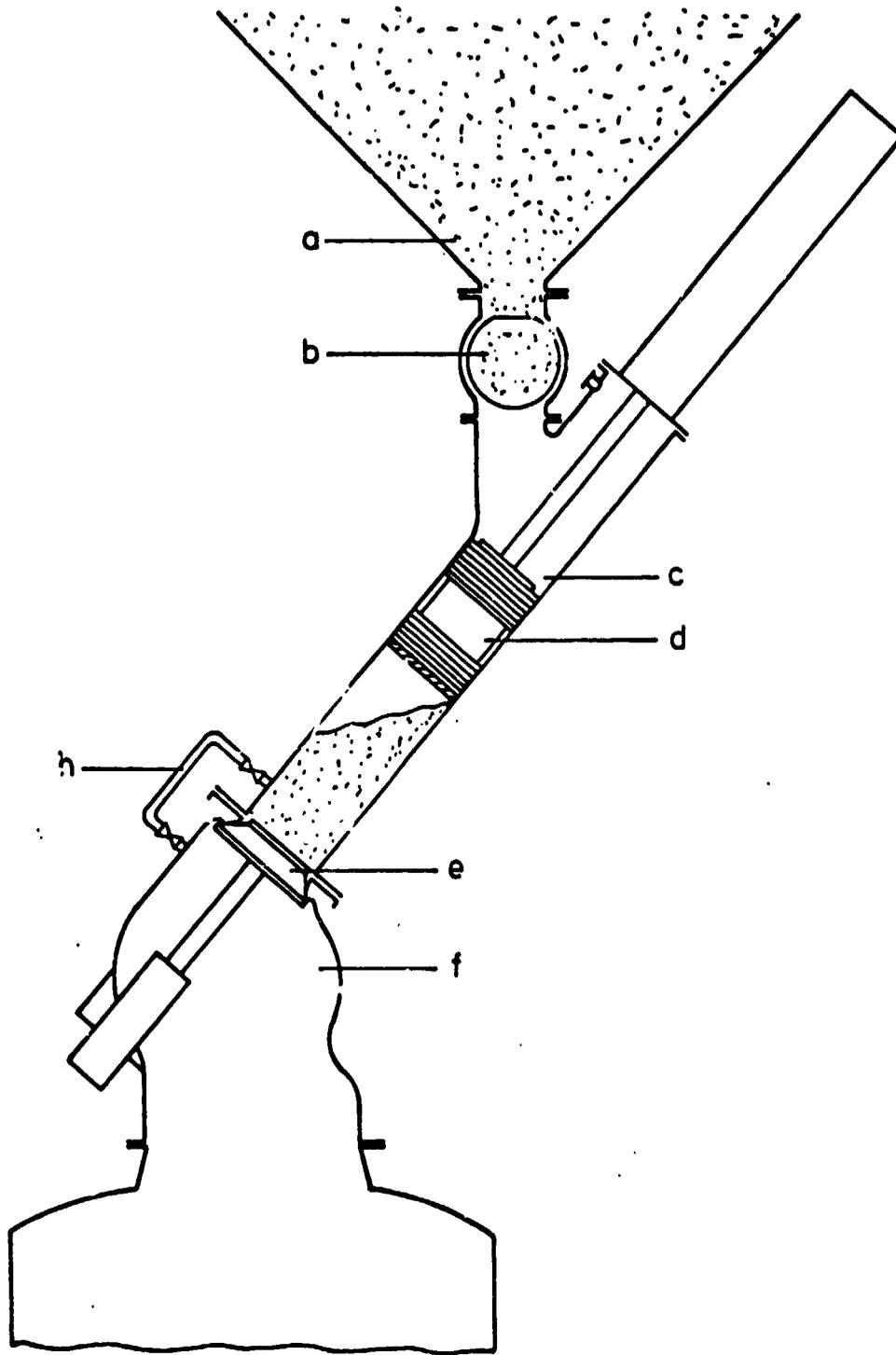
FIG. 7

SCHLEPPER SYSTEM FOR REDUCED VENT GAS
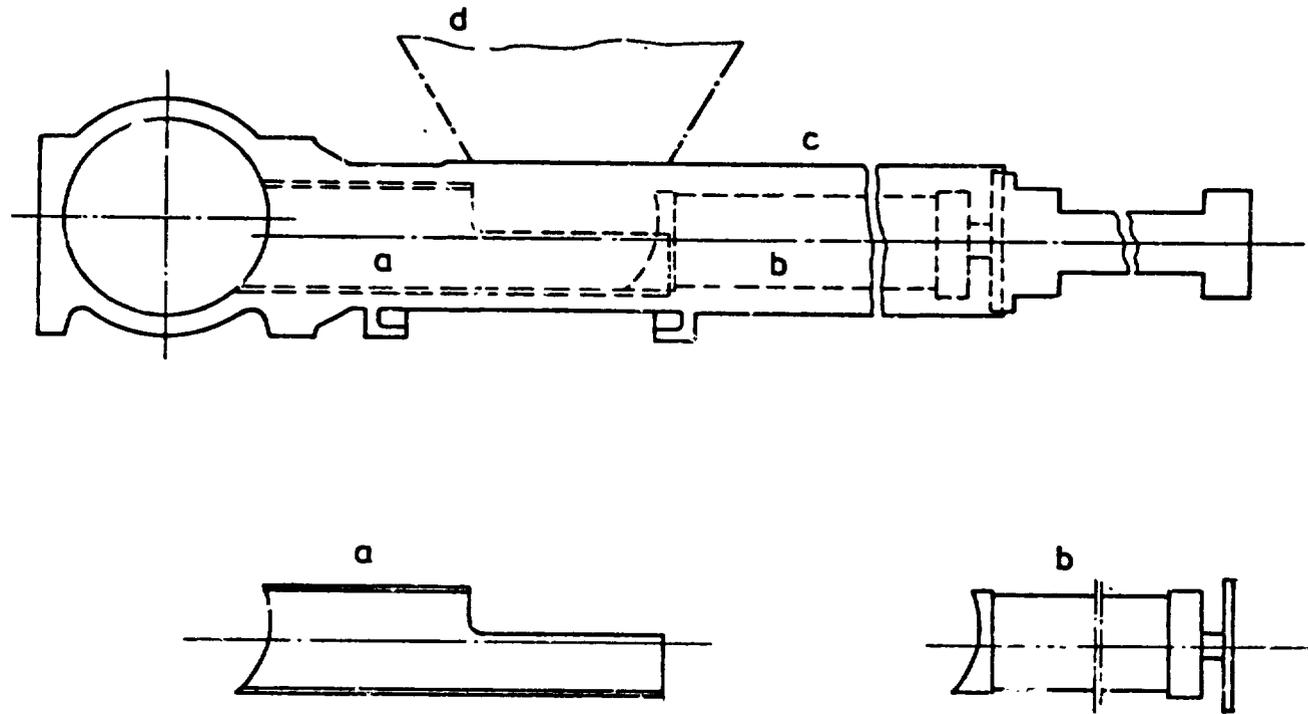
a LINER
b PISTON
c PUMP HOUSING
d FEEDING HOPPER

FIG.8 PISTON TYPE PUMP FOR DRY SOLIDS

77-55
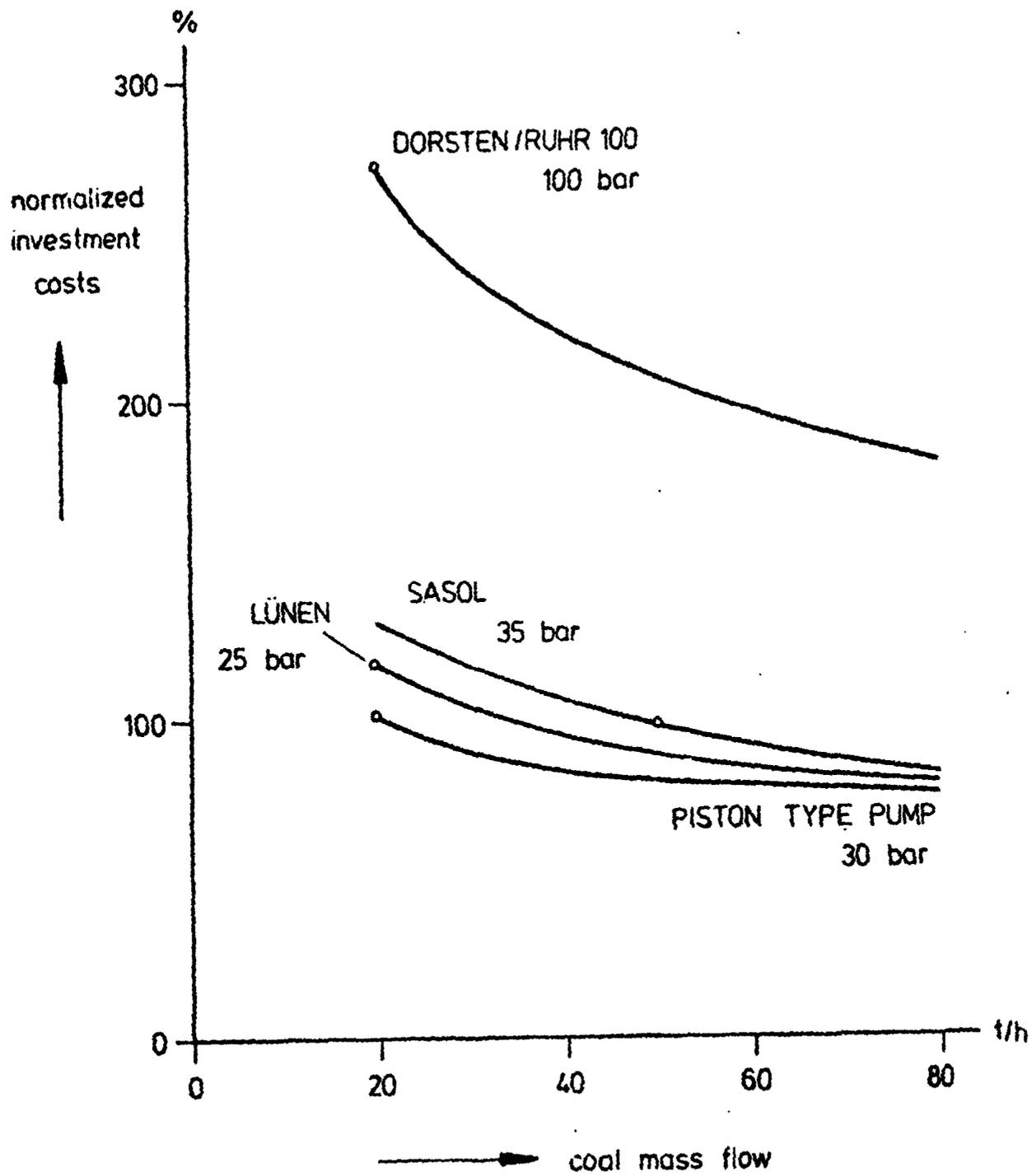
FIG. 9 COMPARISON OF INVESTMENT COSTS
OF DIFFERENT FEEDER SYSTEMS

Kamyr Steam-Phase Digester
with Two-Stage Diffusion Washer
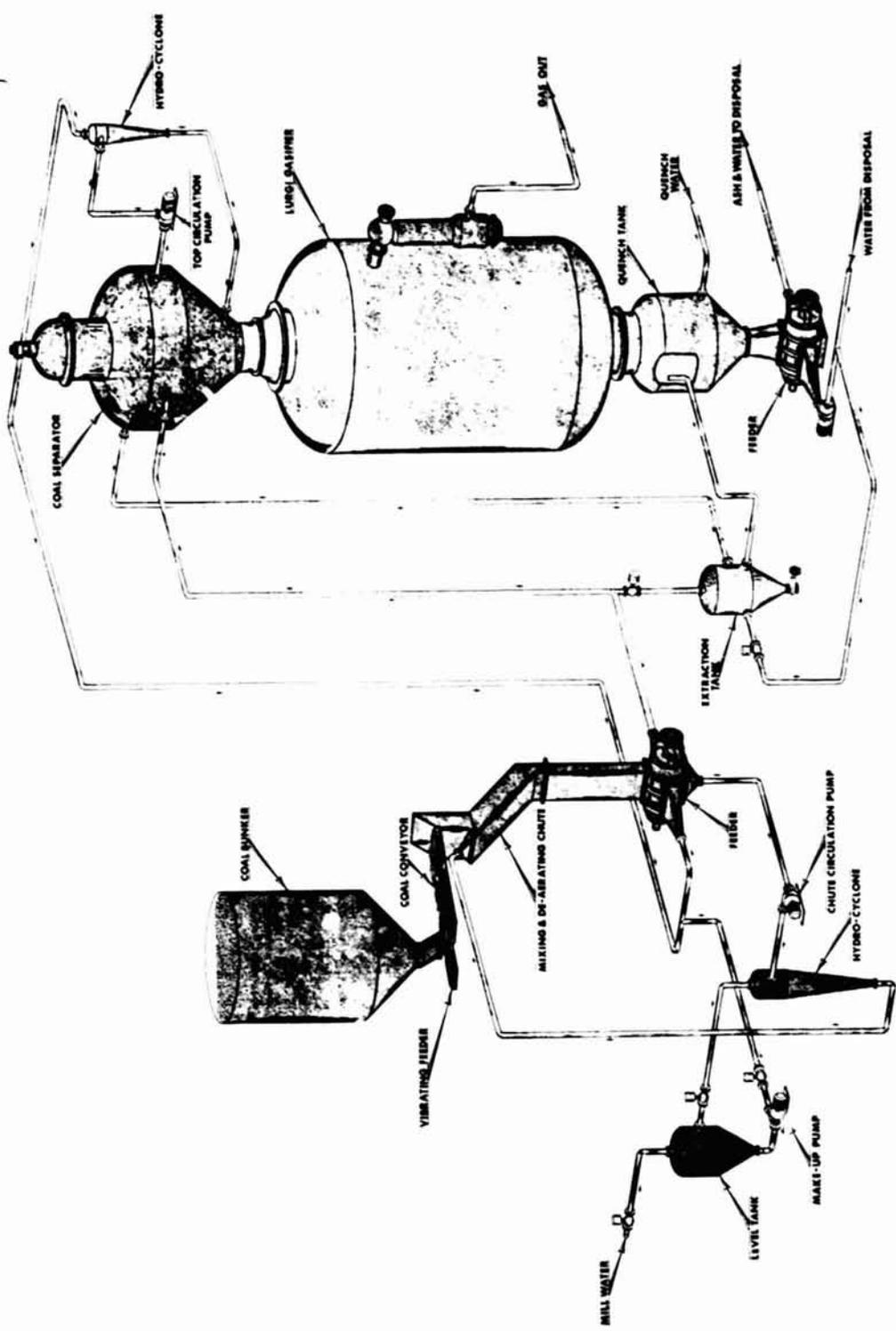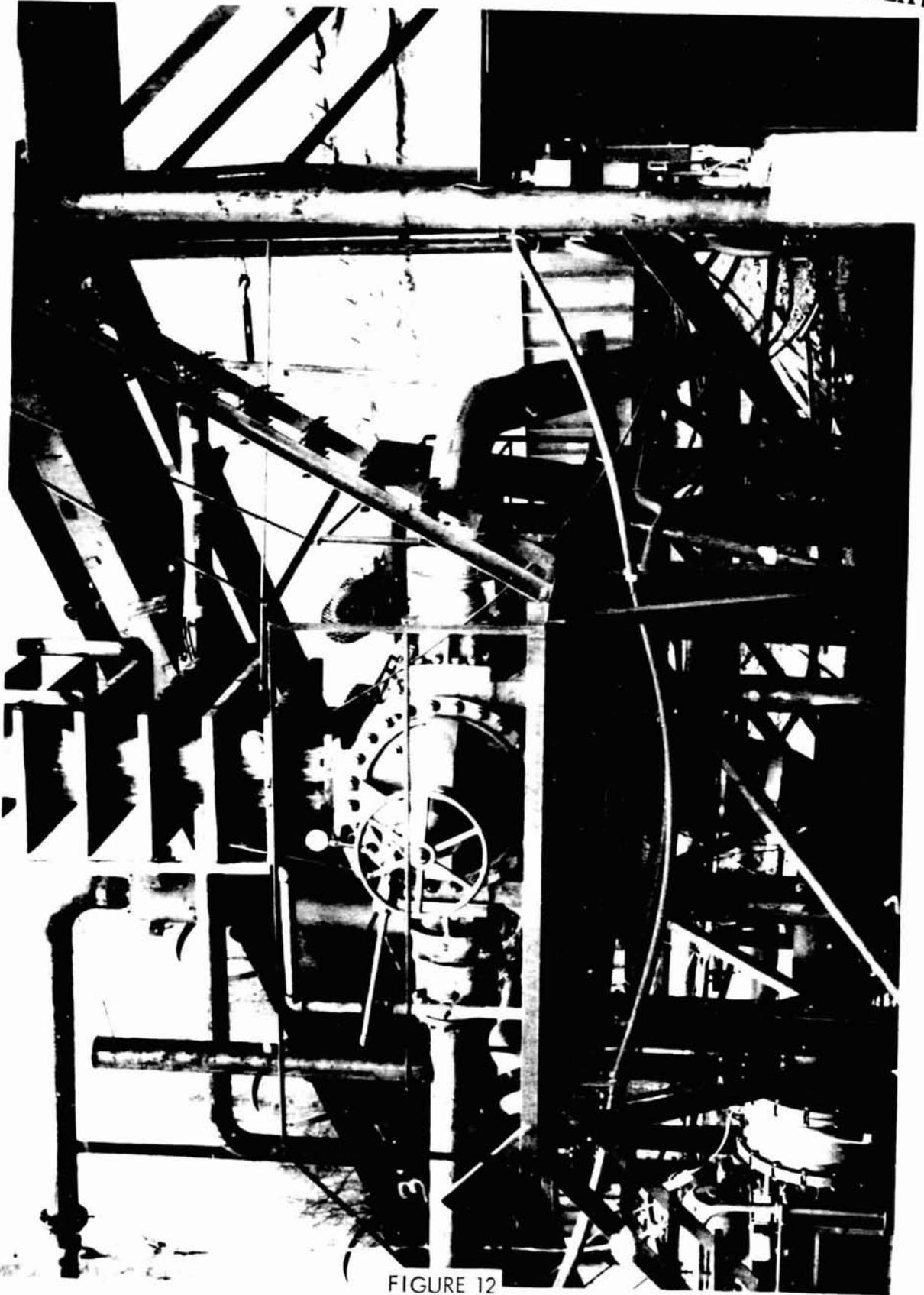
FIGURE 10
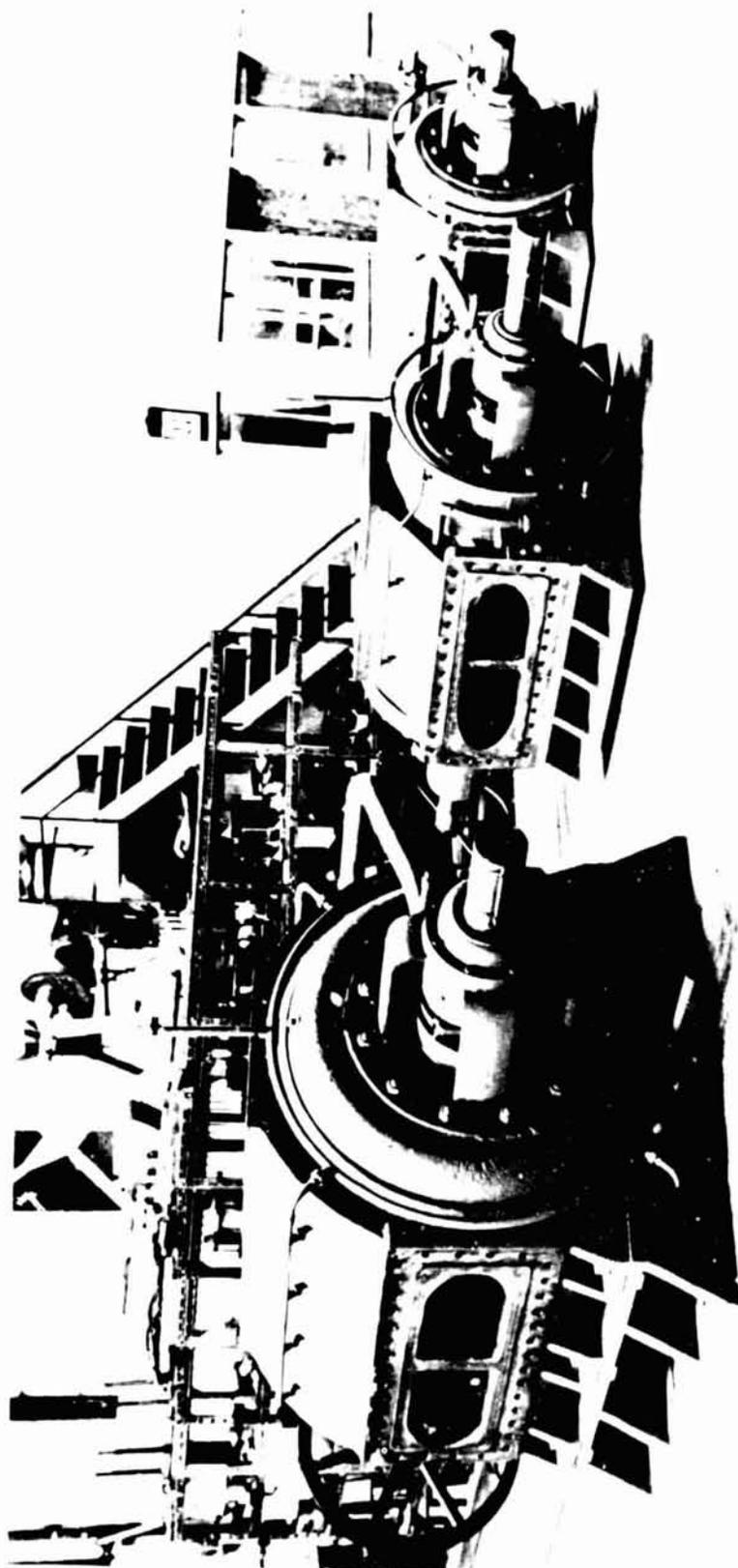
FIGURE 11

FIGURE 12

FIGURE 13

CONTINUOUS LUMP COAL

SLURRY FEEDING

Slurry feeding of coal to gasifiers is most often thought of in terms of fine particles with high solids to liquid concentrations to minimize thermal penalties inherent with vaporization and condensation recovery of the liquid. A method is available to feed lump coal to gasifiers where the majority of the transport liquid is easily drained.

Lurgi and Kamyr have been working on adapting a proven method of feeding wood chips to a method for feeding lump coal. The wood chip feeding system years ago caused a revolutionary conversion of pulp making from a batch method to a continuous method. Worldwide, more than 250 major pulp mills are utilizing this continuous feeding method to feed over 40 Million tons of wood chips annually to pressure digesters.

The wood chip feeding system is illustrated in Figure 10 which is a reprint from Kamyr's Digester Sales Brochure. The object is to feed wood chips without passing the chips thru pumping devices. Additional objectives are to feed the chips without gas back flow and without pressure letdown of the liquid transporting circuit. The basic concept involves establishing a closed loop high pressure liquid stream. Chips are injected into this stream by the high pressure feeder and transported through pipe (1) to the top of the digester where the chips are mechanically separated from the transport liquid (NaOH solution). The liquid recirculates thru pipe (2) to continue the feeding.

The feeding system for lump coal is very similar and is shown in Figure 11 for a Lurgi moving bed lump coal gasifier. The separator is a redesigned version of those available for chip separation.

Kamyr and Lurgi began a cooperative evaluation of this feeding system in June 1974. The first step was a technical evaluation to determine feasibility. Next the key element, the high pressure feeder, had to be proven capable of transmitting coal. In November of 1974, Kamyr began design work for a pilot facility to test the operational capabilities of the high pressure feeder with a two-fold purpose of proval for gasification and for deep mine coal lifting or short distance pipelining. A photo of this pilot installation is displayed in Figure 12. The pilot installation was located at an Appalachian coal preparation plant where metallurgical coal is cleaned. Raw 4" x 0 coal was transmitted in a 10 inch pipe from a storage silo to the top of the preparation plant using a small Kamyr high pressure feeder. The coal was transported 450 feet horizontally and then 115 feet vertically. Raw coal rates were varied between 100 and 215 tons per hour and at velocities between 4 and 14 feet per second. Data was collected on pressure drop and size degradation. The raw coal contained approximately 35% refuse which was in the form of clay and hard shale stone. Some stones with long dimensions as great as 7 inches were transported.

The success of the feeder testing program meant that one more test phase was required; a separator design was to be established. The separator needed to be designed in such a fashion to prevent liquid discharge into the gasifier under failure conditions. An engineering and laboratory evaluation began in the Fall of 1975 and continued into the Spring of 1976. Two test models were fabricated and operated on coal. The testing included evaluation of surface moisture carryover to the gasifier. The surface moisture ranged from 13% with 1/4" x 0 particles to 3.2% with very coarse particles. The separator involves mechanically lifting the settled coal particles from the transport liquid. Water has been used in all tests; however, the transport liquid could be an oil or waste liquor.

The feeding system is totally automatic. Once liquid flows are established, the feed rate is governed by a gasifier demand signal to the bunker vibrating feeder shown in Figure 11. The high pressure feeder does not control the rate; it merely performs the transfer from low to high pressure. The high pressure feeder is not a star wheel. It is a pressure balanced rotary type of transfer valve. It consists of one moving part, the tapered rotor, with a plurality of diametrically penetrating holes that allow a continuous downward vertical flow through the feeder while at the same time allowing an independent high pressure horizontal flow thru the feeder. Coal is introduced into the downward flowing stream, stopped in the rotor by

screens in the lower port, and subsequently transferred by the turning rotor into the high pressure stream. The pressure balance of the high pressure feeder is a very important feature. This pressure balance can be visualized in Figure 11. The hydraulic pressures at the upper connection and lower connection are nearly equal except for a couple of feet of static head between the two connections. The hydraulic pressures at each horizontal side connection are equal. The opposing pressures all around the feeder are equal; therefore, the rotor is free turning within the housing requiring very small turning power. The absence of high hydraulic side loads on the rotor allow operation at high pressures. The pilot facility feeder operated with a 10 horsepower motor. A high pressure gasifier would require this same power for the feeder.

Three sizes of feeders are shown in the photograph of Figure 13. The far right feeder of the photo is the size used at the pilot facility. The largest high pressure feeder which is not shown has the capability of transferring coal at the rate of 500 tons/hr. Operating pressure designs for pulp mills are 350 psig. Designs have been finalized for 450 psig and designs for 1000 and 1500 psig are in progress.

The lump coal feeding system offers the following advantages:

1. The system is automatic and continuous.

2. No gas compressions are involved.

3. A liquid seal is created between the gasifier and the feeding system eliminating gas leakage from the gasifier.

4. Very large coal rates are readily available, thus allowing for bigger gasifiers.

5. No coal storage areas are required in the gas plant. The feeders may be located at a separate coal bunkering area.

6. Since the high pressure feeder is pressure balanced, large pressures can be achieved which is not possible with conventional rotary feeders where side loads are inherent.

The system now remains to be commercially demonstrated on

a coal gasifier.

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SLURRY PUMPING TECHNIQUES FOR FEEDING HIGH-PRESSURE COAL GASIFICATION REACTORS

W. G. Bair

P. B. Tarman

Institute of Gas Technology
Chicago, Illinois

N78-13248

ABSTRACT

Operating experience in pumping coal and coal char slurries at pressures up to 1500 psig at the Institute of Gas Technology Pilot Plants will be discussed. Coal feed at rates up to 3.7 tons/hr using aromatic light oil and water mediums have been pumped successfully at solids concentrations as high as 50 weight percent. Slurry preheating and vaporization at temperatures up to $600^{\circ}F$ have been achieved with a high pressure char-water slurry feed stream. The design specifications for the mixing tanks, pumps, piping and slurry heaters will be discussed and system operating experience and maintenance will be reviewed. Pressure drops and minimum flow velocity data on water-lignite slurries also will be discussed.

1. Introduction

The Institute of Gas Technology has developed the HYGAS® and Steam-Iron Processes to convert coal to a high-Btu substitute natural gas and for producing hydrogen. These processes require reactor pressures in the range of 1000 to 1500 psig, which is typical of other coal gasification processes for producing substitute natural gas. An integral part of all these high-pressure processes is the mechanism for pressurizing and feeding the coal into the reaction zone. The conventional way to feed solids into a pressurized reactor is to use lockhopper feeding systems. However, lockhoppers require large volumes of pressurizing gas and tend to leak through their pressure sealing valves. These valves must operate under dusty and abrasive atmospheres, while sealing at high press. Because lockhoppers had disadvantages in feeding solids to our reactor systems, we decided to investigate using a slurry feed system to pump the coal slurry to high pressure for direct introduction into the reactors.

2. Design Considerations for the HYGAS® Slurry Feeding Systems

The HYGAS Process uses a fluidized-bed contacting system. The preferred solids feed is in the particle size range of 10 to 100 mesh USS., which easily lends itself to feeding solids into the reactor using slurry pumping. Four major areas of design must be considered once the particle size range has been determined for a slurry pumping system.

1. The combined heat balance of the feed and reactor systems must be considered. Dry, pulverized solids are needed for fluidized-bed contacting, therefore, the slurry medium must be vaporized and removed from the solids. The heat duty this requires can be handled in many ways, but it is an important design consideration because the method chosen affects the overall process.

2. Erosion characteristics of the particular slurry must be considered. The systems for piping and pumping, and the components of the overall feed system, must be designed with erosion-resistance in mind.

3. The velocity requirements for the particular slurry must be determined to maintain the proper flow without solids settling and causing plugging in the pipes. However, too high a velocity will cause increased erosion rates and higher horse-power requirements for pumping.

4. Instrumentation for measurement and control of the slurry pumping system must be considered.

The elements of the slurry pumping system which are incorporated in the HYGAS Pilot Plant are shown in Figure 1. Initial heat balance considerations for the HYGAS slurry feed system indicated there was insufficient sensible heat in the reactor itself to evaporate a water slurry and prepare a dry coal feed. Therefore, an aromatic light oil, which is a byproduct of the HYGAS reaction, was selected as the slurry medium. The heat of vaporization for the aromatic oil is approximately 25% that of water and sufficient heat is available in the HYGAS reactor to completely evaporate the oil, if a small amount of preheat is added to the slurry before it is pumped into the reactor.

Char is added (by weighing) to a 3000 gallon slurry mix tank with a top-entering mixer. The slurry is completely pumpable at concentrations of up to about 50 wt % coal or coal char in oil. The thoroughly mixed slurry is withdrawn at the bottom of the mix tank. A low-pressure centrifugal pump circulates the slurry through the suction manifold of a high-pressure positive-displacement plunger pump which pumps it up to reactor pressure. The centrifugal slurry-feed pump operates at a discharge pressure of 25 to 35 psig and has a capacity 2 to 3 times the maximum capacity needed for reactor feed. ⌐ excess slurry is returned to the slurry mix tank; it aids in the agitation and maintaining a homogeneous slurry. The high-pressure plunger pumps are single-stage, reciprocating, and raise the slurry from the inlet pressure of approximately 25 psig to the reactor pressure of 1000 to 1500 psig. The slurry is pumped through a high-pressure steam heat exchanger to preheat the oil-coal mixture. Then slurry is pumped through a simple spray head inside the reactor and into a fluidized-bed drying zo There, the slurry vehicle is vaporized, leaving the dry coal available for feed into the lower stages of the reactor. The vaporized slurry oil exits with the raw gas stream out of the top of the reactor. It is recovered in downstream condensing and separation equipment and recycled back to the slurry mix tank and remixed with more fresh coal feed.

Initial experiments with the slurry feed system indicated that velocities of 5 to 10 feet/sec were sufficient to keep the slurry thoroughly mixed while in the piping system, thus preventing settling of solids and eventual plugging of the pipes. Therefore, after the feed requirements and total volumetric
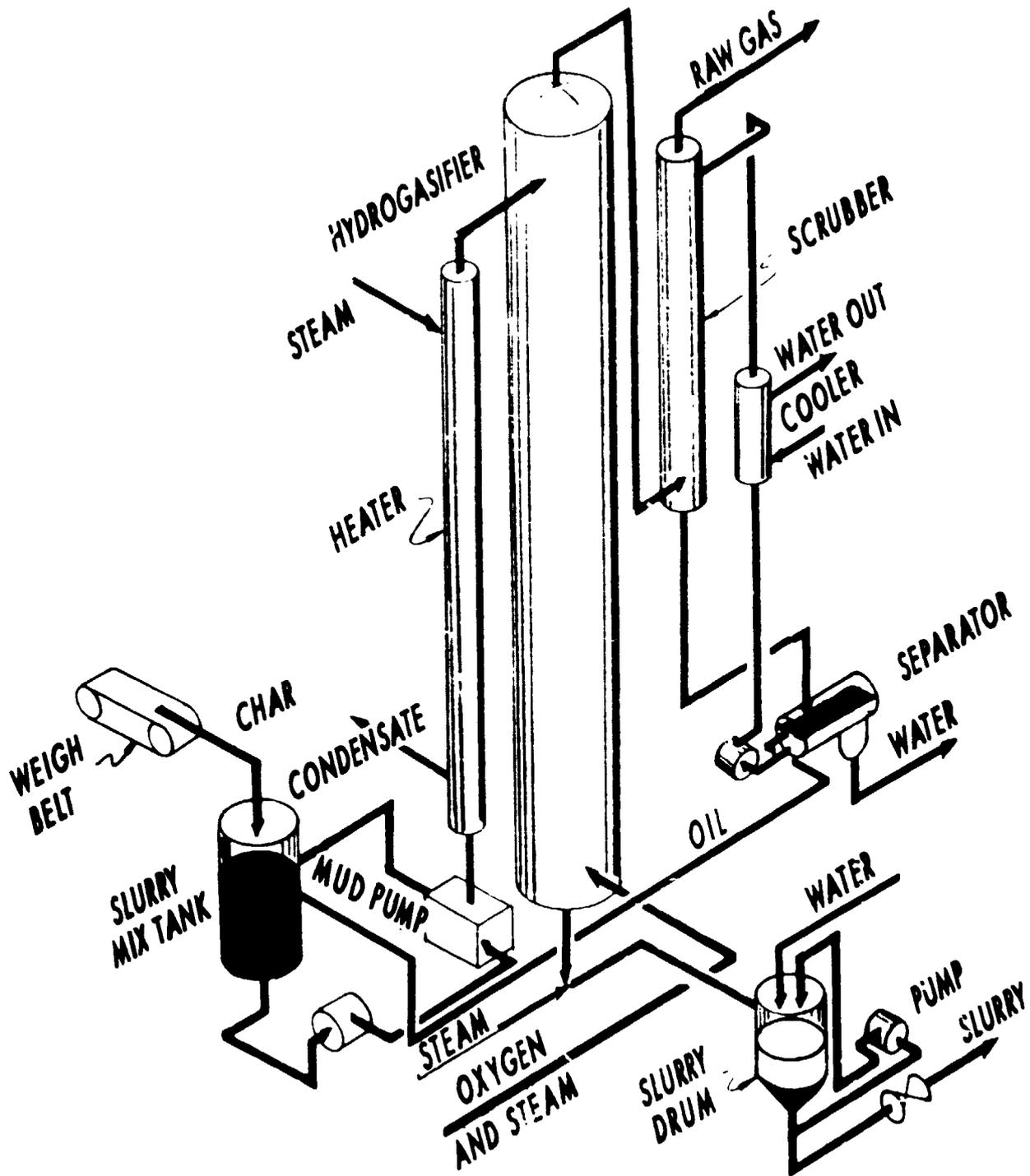
Figure 1.   SCHEMATIC DIAGRAM OF SLURRY FEED SYSTEM
OF THE HYGAS® REACTOR

flow rates were set by reactor design considerations, the piping was sized to obtain a 5 ft/sec flow rate with an average reactor feed. The piping system layout uses 90 degree directional changes. All control valves and shutoff valves were installed in short, vertical lengths of pipe to prevent accumulation of solids in valve seats if an emergency caused shutdown of the slurry system before it could be properly drained. The 90 degree directional changes were made up of tees and/or crosses. Long-radius elbows with long sweep distances were avoided because of their poor erosion resistance.

Initial equipment specifications of the HYGAS pumping equipment called for a chromia-oxide coating on the impeller and casings of the centrifugal pumps, to combat the erosive conditions we knew would exist. The reciprocating pump was to have polyurethane elastomer check valves; these had been very successful in coal and water slurry pumping systems.

Instrumentation was designed to measure the flow in the low-pressure circulation system using venturi flow meters and nuclear density measurement. The combination of flow and slurry density could then be calculated to give a check on the mass feed rate measured by the weigh belt. We also successfully applied the venturi meters to the high-pressure slurry feed system downstream of the positive-displacement plunger pump.

3. HYGAS® Operating Experience

The slurry feed system used in the HYGAS Pilot Plant has evolved into a highly reliable and maintenance-free operation. However, several initial problems were overcome to obtain the final design configurations routinely used today.

Initially, the low-pressure slurry circulation pumps operated at too high a speed (3500 rpm) and the chromia-oxide erosion-resistant coating was ineffective for the particle size range and the type of slurries being handled. Erosion rate is proportional to a power of the speed: Some investigators have suggested that this could be as high as the fifth or sixth power. We found that reducing the pump speed from 3500 to 1750 or slightly lower caused a significant decrease in erosion rate in the centrifugal pumps. Weld-applied stainless steel or stellite coatings also successfully combated erosion on the impellers and the casing of the centrifugal pumps.

Considerable development effort has been expended to improve check valve designs for the reciprocating plunger pumps. The original polyurethane elastomer check valve material was attacked by the aromatic light oil used in our slurry system and did not hold up at all in operation. The next design used on area-type, metal-to-metal check valve where hardened steel replaced the elastomer. This gave very poor wear resistance and quickly deteriorated. Next, we tried a hemispherical check valve. This gave much better service life; the concept of a line seal rather than an area seal should be used in slurry-pumping systems. The line contact is much less susceptible to hold-up of particles under the sealing surface, which leads to leakage and fast erosion of the check valve parts. Recently, experiments with full ball type check valves in the reciprocating pumps gave very good results. The full ball check valve allows random selection of a different sealing surface for each pump stroke. This prevents the excessive wear on a particular part of the check valve which occurs with the hemisperical-type design.

Carbon steel piping has given adequate erosion resistance and we have had essentially no problems with slurry pumping systems in carbon steel pipe at the 5 to 10 ft/sec velocity.

The instrumentation has worked satisfactorily. Venturi meters have been proven to be erosion-resistant and quite adaptable to measuring slurry flow rates. In adapting any type of a differential pressure meter to this service, it is important that the pressure tap lines to the transmitter be purged with a clear liquid, in our case oil, to keep them free of particles that would plug the taps and interfere with the differential pressure readings.

The centrifugal pumps that we now use are relatively low-speed (1400 to 1500 rpm) and are typically sand or gravel-type pumps constructed of Ni-hard material. They have removable wear plates so that the impellers and certain parts of the wetted end of the casing can be replaced as erosion occurs, rather than applying an overlay of stellite or other hard-faced material to the pump casing directly. All the pumps that we now use have double mechanical seals, which have been flushed with a clear seal flush liquid, rather than packed seals. The mechanical seal is definitely preferred for slurry pumping service.

4. Experimental Data

The oil-coal system used for slurry feeding to the HYGAS reactor is operating quite satisfactorily. However, consideration of other types of slurry

vehicle systems led us to investigate a water-coal slurry system. We designed
a series of tests to determine the lowest practical velocity for slurry pumping
and the maximum concentration of coal in water that could be handled. A test
loop was set up as shown in Figure 2. The slurry mix tanks and the positive
displacement pumps of the HYGAS Pilot Plant were briefly used and lignite
coal-water slurries were prepared in about 15% to 50% coal in water. The
test loop consisted of two straight runs of pipe and a series of return bends
made up with 90 degree weld ells and 5-foot straight pipe sections. The entire
test loop was installed in a single horizontal plane to avoid head differences.
We wanted to determine if a minimum velocity existed, at which the slurry
solids tend to settle out in the pipeline. We also wanted to determine the
pressure drop per unit length of pipe, for design of large-scale slurry systems,
to obtain an idea of the horsepower required for pumping. Pressure taps were
set up over two of the straight pipe sections and across the return bend section
to obtain an idea of the pressure drops there. Data from these tests are
presented in Figures 3 and 4. These figures illustrate the typical curves of
head loss as a function of velocity of the slurry liquid. The plots are on log-
log paper and are compared to the pressure drop exhibited by the system with
pure water. The data is fairly uniform and indicates that as the slurry
concentration is increased from 17.3 wt % to 53.1 wt % lignite coal in water,
the pressure drop increases with the slurry concentration and as a function of
the velocity in the pipe. We deliberately studied the low flow range because
we were interested in minimum velocity requirements. At the lowest flow
rate obtainable with the given equipment, slightly less than 2 ft/sec, no
plugging occurred because of settling of solids from the flowing stream. Our
general conclusions are that velocities as low as 2 ft/sec are sufficient for the
particle size range used in these tests and for up to about 50% coal in water.

The particle size range is shown in Figure 5 for three samples selected
from the slurry mix tank at different time periods. All of these data were
accumulated over a two-week period and the slurry was simply returned to
the mix tank after it had gone through the test loop. We sampled the slurry
at the start, the middle, and the end of the test series. There was very
little deterioration of the slurry particle size through impact and erosion; for
all practical purposes the materials are identical. The average particle size
range was between 35 and 45 mesh.
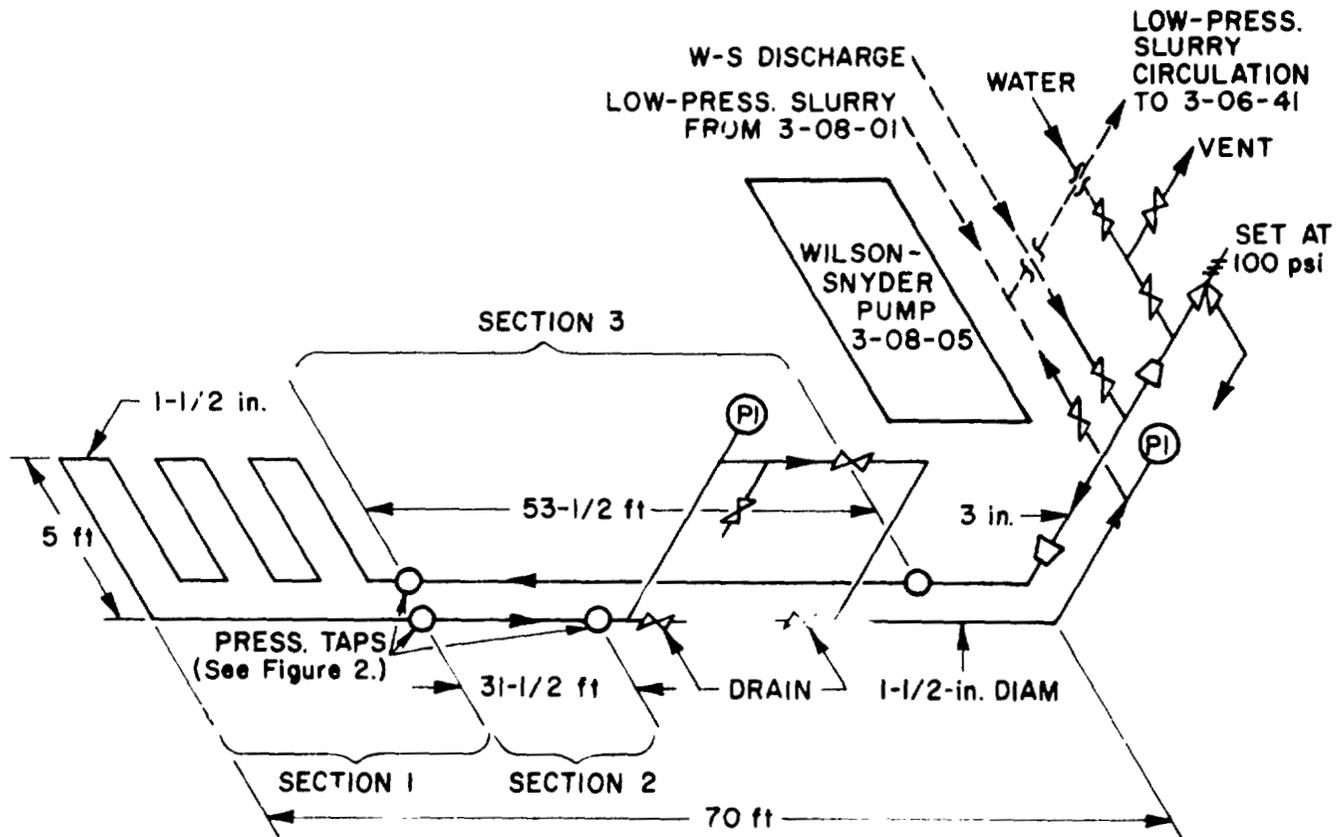
Figure 2. SLURRY TRANSPORT TEST LOOP
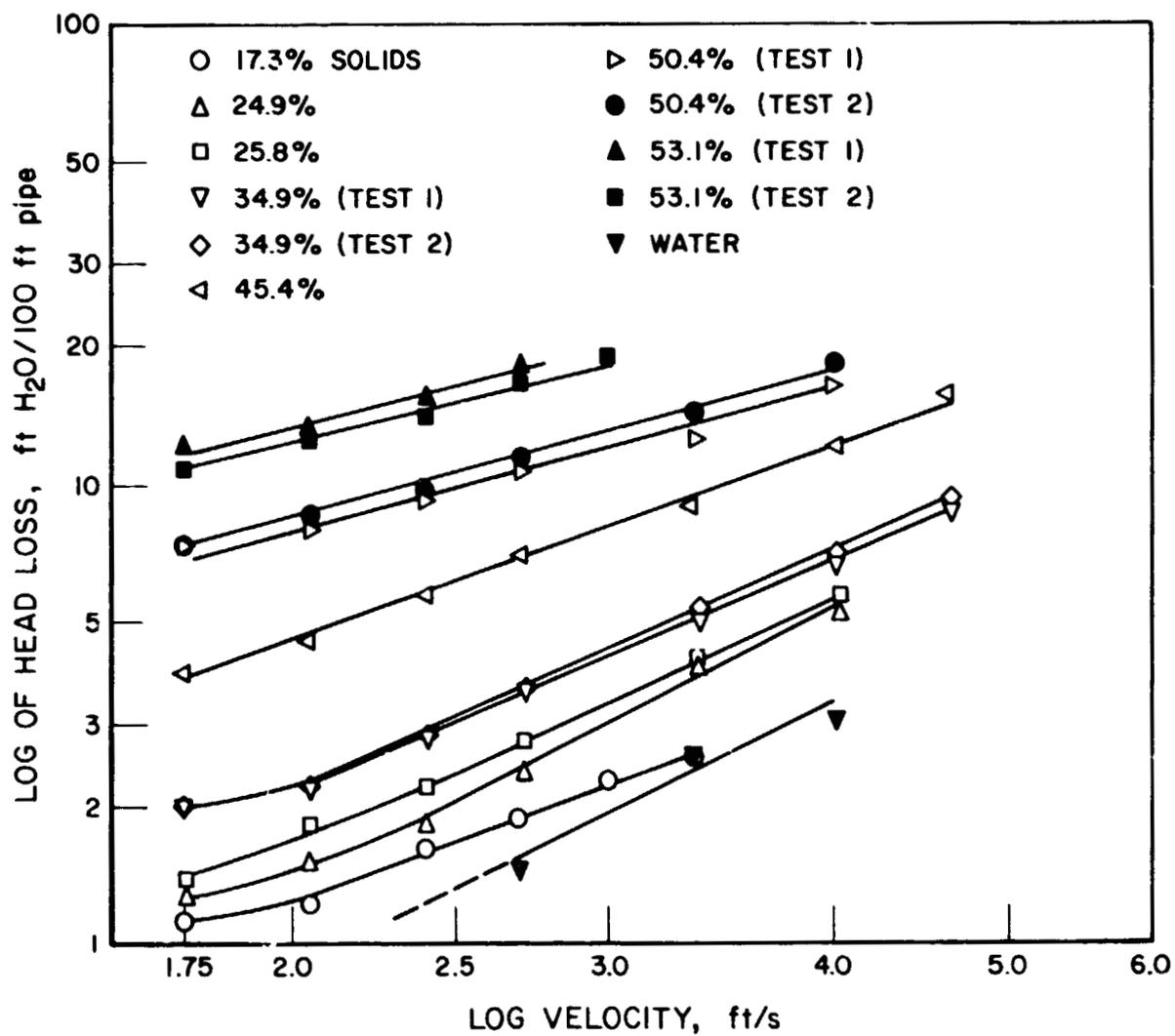
Figure 3. PRESSURE LOSS OF LIGNITE-WATER SLURRY IN 1-1/2 in. SCHEDULE 80 PIPE – DIFFERENTIAL PRESSURE INDICATOR 1
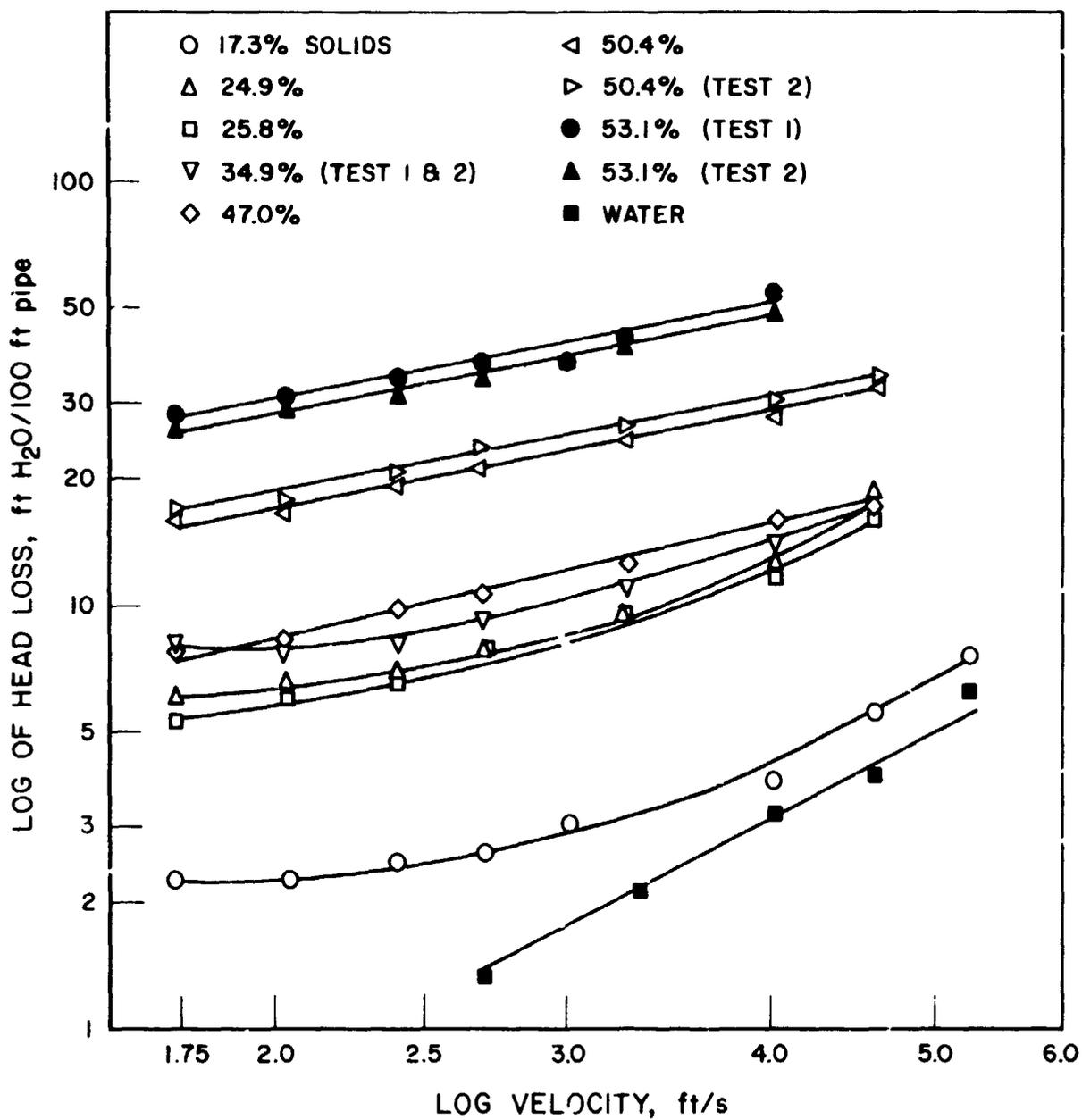
Figure 4. PRESSURE LOSS OF LIGNITE-WATER SLURRY IN 1-1/2 in. SCHEDULE 80 PIPE — DIFFERENTIAL PRESSURE INDICATOR 2

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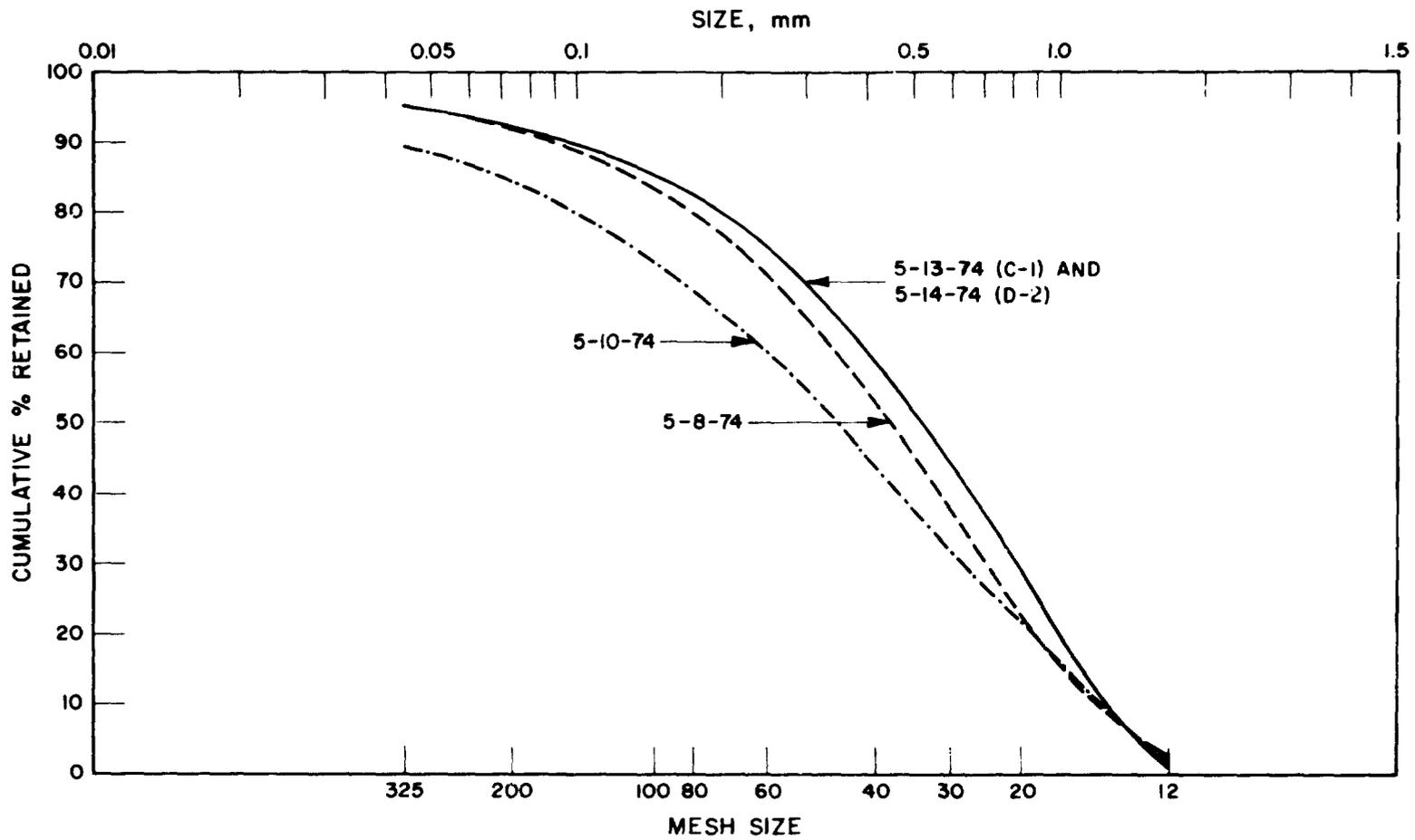
Figure 5. TYPICAL PARTICLE-SIZE DISTRIBUTION

We found pulsation dampeners highly desirable on the suction and discharge lines of the positive-displacement reciprocating pumps. The original piping system for the pilot plant did not contain these pulsation dampeners. Excessive vibration problems occurred during several tests, especially at extremely high slurry-feed rates. Pulsation dampeners were later installed on both the suction and discharge lines. The discharge dampener was fabricated from a 2-foot piece of 6-inch Schedule 160 pipe and installed in a vertical position, as close as possible to the actual pump discharge. A high-pressure nitrogen cushion is maintained in the top of the dampener by adding nitrogen as necessary. The low-pressure suction stabilizer is a 2-foot section of 8-inch schedule 40 carbon steel pipe installed in the pump suction line, as close as possible to the inlet. Again, a blanket of low-pressure nitrogen is maintained on the top of the pulsation dampener to provide a cushion for volumetric changes.

5. The Steam-Iron Slurry Feed System

Unlike the HYGAS pilot plant, the Steam-Iron pilot plant uses a slurry vaporizer in its high-pressure char feed system. Vaporizing the slurry water allows the direct feeding of dry char to a high-pressure producer reactor. This moves the evaporative heat load from the producer reactor; in commercial operation the high-level heat in the producer off gas could be used for other purposes. This would allow using low level plant heat to evaporate the slurry medium. In the Steam-Iron pilot plant, however, the slurry vaporizer is fired with natural gas because the objective is to prove system operability, not demonstrate heat economies.

Figure 6 shows the location of the slurry vaporizer in the char feed system. The char-water slurry is prepared in a mix-tank and the slurry is pumped to high pressure using equipment similar to that described for the HYGAS process. The char-water slurry is fed to the bottom of a helical coil contained inside an 11-ft diameter, 30-ft high, refractory-lined fired heater. Here, the water is vaporized and the resulting steam transports the char 100 ft, to the top of the producer reactor. The heater is designed to supply about 50°F superheat in the exit steam to inhibit condensation in the exit piping. In addition, the exit piping is steam-traced for winter operation.

The temperature of the slurry heater is controlled by the exit steam temperature, which is used to adjust the firing of 4 burners located in the base of the slurry heater. The burners are automatically shut down by a safety switch located in the stack of the slurry heater.

In operation, the approximately 200 psi pressure drop across the slurry heater means the slurry pump system operates at about 1200 psig when the producer operates at 1000 psig. Discharge pressures and flows of the high-pressure slurry pumps are continuously measured. Excessively high discharge pressures (indicating the onset of plugging within the coil) or high flow (indicating tube rupture), will automatically shut down the system. In shutdown, the slurry feed system is isolated from the reactor system, the high pressure pumps are shut down, and the fuel to the slurry heater is cut off. After a 10-second delay, the entire contents of the slurry coil and exit piping are rapidly discharged into a high-pressure holding pot located upstream of the slurry heater. This procedure effectively back-blows the coil and tends to dislodge any plug which may have formed. It also prevents possible sintering of a plug which would otherwise overheat because of the large quantity of heat stored within the coil and the refractory walls of the furnace.

The helical coil consists of three sections with increasing internal pipe diameters. The bottom coil preheats the slurry to its vaporization temperature. The middle coil supplies the evaporative duty and the final coil superheats the steam above its boiling point. The preheat portion of the coil is standard 1-1/2 inch Schedule XX pipe with a 1.100-inch internal diameter. This pipe size was chosen to ensure a velocity of 2 to 5 ft/s; this had been previously determined as the minimum velocity necessary to prevent settling of the char over the range of operating conditions expected. The top portion of the coil is 2-1/2-inch Schedule 160 pipe with an internal diameter of 2.125 inches. This diameter was chosen to limit the exit steam velocity to about 25 to 50 ft/s and thereby minimize erosion in the exit transport piping system. The middle portion of the coil is 2-1/2-inch Schedule XX pipe with an internal diameter of 1.7 1 inches. All piping within the heater is 1-1/4 chrome-1/2 moly steel (ASME SA 335-p 11). Stainless steel construction was unnecessary because operating temperatures will be below 650°F.
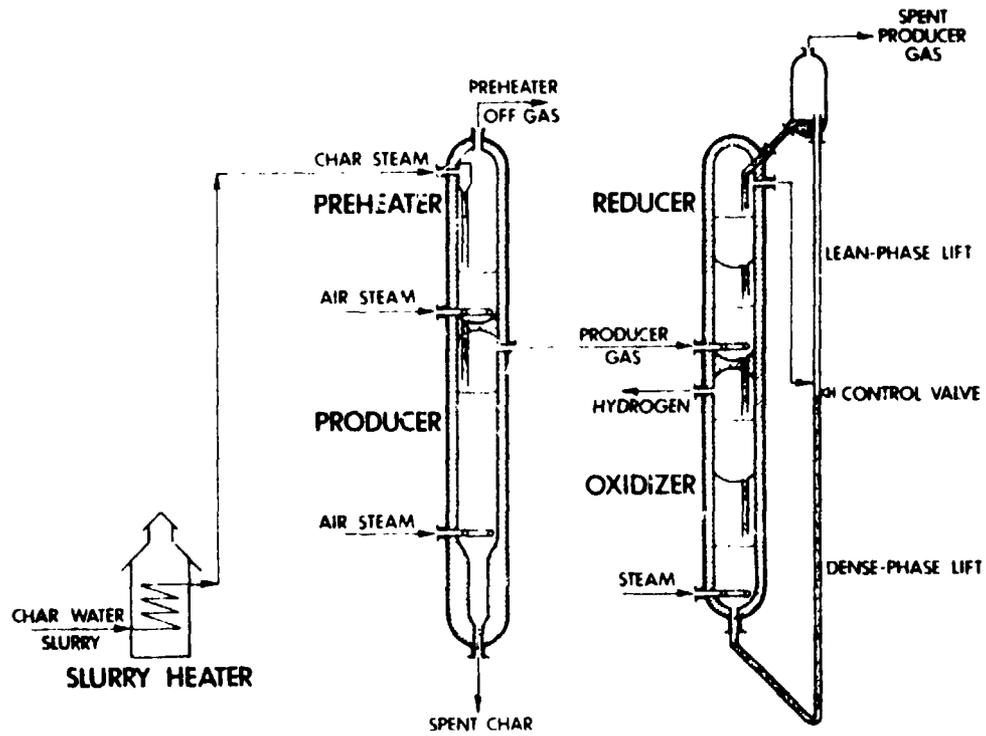
Figure6. LOCATION OF SLURRY VAPORIZER IN THE
STEAM-IRON SYSTEM

As in the HYGAS plant, all turns in the external piping are made with crosses instead of elbows. These allow rodding of the pipe in both directions and, more importantly, provide a pocket of solids which forms its own elbow for the high velocity char. This design greatly minimizes erosion which would be severe if regular piping elbows had been used.

The slurry heater is designed to feed from 1 to 206 tons/hr of char at concentrations of 20 to 43 wt % solids and at pressures of 500 to 1000 psig. Operating velocities range from 2 to 5 ft/s at the inlet to 25 to 50 ft/s at the exit. The inlet velocity is sufficient to prevent settling of the char and subsequent plugging for char particles ranging in size from 10 to 80 mesh. The exit velocities are sufficient to transport the char in lean-phase through the exit piping to the top of the producer reactor, but are low enough to minimize erosion of the transport piping. In designing the coiled heater, heat transfer coefficients similar to those for preheating and vaporizing water were used.

The coil was initially tested using coke and with direct discharge to the atmosphere. This testing at low pressure caused excessive velocities in the super heat coil section and portions of the uppermost coil were eroded. This coil was replaced and operating pressures were increased to within the design range using a temporary restriction orifice.

In subsequent operations, the system was reconnected to the reactor and the slurry vaporizer has worked very well. Complete vaporization has been achieved at char feed rates up to 1.75 tons/hr and concentrations up to 32 wt %. Accumulated operating time to date has been about 2000 hours with steady-state periods as long as 200 hours. Overall operation of the system has been very smooth and only momentary plugging of the coil has been experienced. In all instances, the plugs were easily cleared.

6. Conclusions

A vaporizing water-slurry feed system and a nonvaporizing oil-slurry feed system have been successfully applied to high-pressure coal gasification reactors. Initial operating problems have been overcome and valuable data on slurry-system design has been obtained. Slurry systems will have many applications for solids feeding in the emerging high-pressure coal processing technology.

7. Acknowledgements

KEYNOTE ADDRESS

COAL GASIFICATION: AN ESSENTIAL
ELEMENT IN ENERGY DEVELOPMENT

H. A. Proctor

Chairman of the Board, Southern
California Gas Co.
Los Angeles, California

Nearly everybody is talking about energy these days...
and I'm certainly no exception. My colleagues and I have been
devoting a lot of time and effort to stimulating public aware-
ness of the energy shortage. We've also been urging prompt and
responsible political action. Such great concern is justified...
because the future health and vitality of America are at stake.
A central factor in the energy equation is that coal, including
coal conversion, must assume an increasing share of the nation's
energy production.

My company, Pacific Lighting Corporation, has a keen interest
in coal gasification because of the benefits it can provide to
our 3.4 million natural gas customers in Southern and Central
California. Further, we believe that coal gasification is the
most realistic, environmentally sound way that California can
participate in President Carter's program to expand the use
of our nation's vast coal reserves. This is one of the reasons
why I was so pleased to accept the invitation to be your key-
note speaker tonight.

I'm going to relate to you our experiences...spread over
the past six years...in conceiving, planning and seeking
authorization for a proposed coal gasification plant in
Northwest New Mexico. I'll begin with an overview of the
project and its objective.

Our proposed plant site is situated on the Navajo Indian
Reservation, near Farmington, New Mexico. The coal gas plant
is designed to produce an average daily yield of 250 million
standard cubic feet of synthetic gas, or SNG. As such, it
could become the nation's first full-scale, high-Btu coal
gasification facility. It will cost about $1.3 billion, in
January 1, 1977 dollars. The gasification processing plant
and related facilities will be built and operated by Western
Gasification Company, or WESCO. WESCO is a joint venture of
subsidiaries of Pacific Lighting, Los Angeles, and Texas
Eastern Corporation, headquartered in Houston.

The primary objective of the WESCO project is to replace
declining gas deliveries to customers of Transwestern Pipeline
Company, a gas transmission subsidiary of Texas Eastern. Only
a short pipeline span from the plant site is needed to feed the
gas into the Transwestern pipeline system. Three-quarters of
the plant's output will be delivered for use by customers of
Southern California Gas Company, Pacific Lighting's gas
distribution utility subsidiary. The remaining 25 per cent
will go to Cities Service Gas Company, which serves Midwest
markets. In Southern California, the SNG will help relieve a
crippling shortage of natural gas forecast for the 1980's.
Natural gas currently provides nearly half of all non-
transportation energy used in Southern California. Yet gas
supplies available to this area have been declining sharply
for several years. The WESCO plant can make a major contribution
to alleviating this situation. For example, the SNG output of
the initial plant would be sufficient to satisfy the needs of

830,000 residential gas consumers. On an annual basis, this volume is equivalent to the energy output of seven Hoover Dams, or 1-1/4 times the electricity produced in 1976 by the Los Angeles Department of Water and Power, the nation's largest municipal electric utility. One additional gasification unit eventually could be built at the WESCO plant site. The two units would provide a total productive capacity of 500 million standard cubic feet of SNG per day. Production at this latter rate would equal, in a single year, more than one-fourth of the total energy generated by nuclear power in the United States during 1976.

For a variety of reasons, coal gasification is an attractive new alternative for helping to meet this country's future energy needs. A key factor is, of course, coal's position as the nation's most abundant energy resource. Although coal comprises more than 80 per cent of the remaining U.S. fossil fuel potential, it presently accounts for less than 20 per cent of U.S. fuel consumption. Also significant is the fact that coal gasification can be accomplished with currently available technology.

The chemical conversion of coal into synthetic gas offers several benefits to the energy consumer. High efficiency, for example. With present technology, the efficiency of the coal gasification process is about 70 per cent. The overall energy efficiency—from mine through ultimate residential user—is approximately 40 per cent. By way of comparison, that's 1-1/4 times the overall efficiency obtained by converting coal to electricity in a conventional electric power plant. And since it requires less coal and water to provide equivalent amounts of energy, gasification conserves these valuable resources.

Gasification also compares favorably to electric production on the basis of economics. Since the WESCO plant will replace declining gas supplies, it can utilize existing gas transmission, distribution and utilization facilities. An equivalent electric system...built to replace these decreasing gas deliveries... would require six times the capital investment of the WESCO plant.

The cost of energy from new nuclear or coal-fired electric generating plants...delivered to the point of use in Southern California...has been about $12 per million Btu's. That's according to an analysis published last year by the staff of the California Public Utilities Commission. Based on a similar time frame, the delivered cost of synthetic gas from the WESCO project would have been less than $3 per million Btu's. By January of 1977, the cost of SNG had risen to $3.68. Coal gasification can be expected to retain a significant cost advantage for the ultimate consumer...even assuming continued delays and inflation.

Coal gasification also promises environmental acceptability

...a result of low pollution levels. Comparing production of a given amount of energy, emissions of pollutants from coal gasification are significantly lower than from the combustion of coal in an electric generating plant. Gasification plants burn as little as 15 per cent of the total coal consumed. The remaining coal is reacted chemically in enclosed pressure vessels. Since the impurities are removed, there is essentially no air pollution.

Finally, coal gasification promises a major new source of energy for domestic consumption without dependence on foreign suppliers and without a negative impact on the U.S. balance of payments. In the case of the WESCO plant, nearly all the capital required for construction and operation will be spent in the United States.

Let's take a look now at the evolution of coal gasification and the process as it is practiced today.

There are three basic ways to treat coal to obtain off-gasses which are combustible: pyrolysis, hydrogenation, and gasification. Of the three, gasification is the best suited for subsequent upgrading to produce a high-Btu gas. What coal gasification does, in effect, is restructure the carbon-hydrogen ratio.

In the early 1960's, the Department of the Interior formed the Office of Coal Research. Since that time, four major gasification research and development projects have been undertaken in this country. They include (1) Hygas, Institute of Gas Technology, Chicago; (2) BiGas, Bituminous Coal Research, Home City, Pennsylvania; (3) CO_2 Acceptor, Consolidated Coal Company, Rapid City, South Dakota; and (4) Synthane, Bureau of Mines, Brucetown, Pennyslvania. The purpose of these projects...all in the pilot plant stage...is to develop technology that could lead to a more efficient and less expensive method of gasifying coal to produce high-Btu synthetic gas for use in the United States. Under the demonstration schedule for these projects, commercial plants will not be available until the 1990's.

Three commercial gasification processes are currently in use around the world: Winkler, Koppers-Totzek, and Lurgi. The Lurgi process, developed in Germany, is the only one that operates above atmospheric pressure.

Since the first Lurgi plant was constructed in 1936, a total of 16 plants have been built throughout the world. The largest operating plant is in Sasolburg, South Africa. Another plant--about twice its size--is under construction at the same location.

The Lurgi process consists of a series of chemical reactions and conversion steps in which carbon from coal is combined with hydrogen to form methane. Successive steps in the process

are arranged to gasify the coal, reject the ash, clean and cool the gas, increase the heating value by methanation, and then compress the product SNG into a pipeline.

The first section of the gasification process is the commercially proven Lurgi gas producer. The gas is produced by the reaction of coal and oxygen in the presence of excess steam at a pressure of 400 to 450 psig. The WESCO plant will have 24 gasifiers. The coal enters the gasifier through a coal lock hopper in a batch sequence. A rotating grate distributes the fresh coal uniformly over the coal bed. As the coal moves down the reactor, it is successively preheated, dried, devolatized, gasified and combusted. The resultant crude gas is then cooled and scrubbed to remove impurities.

At this point, the crude gas enters a shift conversion unit. In this step, carbon monoxide is catalytically converted to carbon dioxide and additional hydrogen is produced. The gas steam is now in the proper chemical balance for methanation.

Methanation is the step that catalytically converts the gas into essentially pure methane, or CH_4 . Extensive laboratory and pilot plant testing of methanation has been completed by Lurgi and other companies. Although methanation has not been used in a commercial-size plant, it has been tested and proven in pilot plants. In fact, methanated gas produced in a demonstration plant in Westfield, Scotland, was introduced into the Scottish gas grid system for use in homes in and around the city of Fife. Lurgi and others are now ready to guarantee a commercial-size methanation unit.

After methanation, the gas undergoes dehydration and final CO_2 removal. The product SNG consists of 97 per cent methane, with a heating value of 980 Btu's per standard cubic foot. The SNG is compressed to 1,000 psig and sent to market by pipeline. It is completely interchangeable with existing pipeline gas.

Other phases of the Lurgi process are designed to purify the SNG by removing by-products and to clean up plant emissions.

To give you a better idea of the size and complexity of the WESCO project, here are some representative facts and figures:

The plant is designed to convert approximately 9-1/2 million tons of coal each year into SNG. WESCO has contracted with Utah International Inc. to supply the necessary coal for the first plant. There is an option on coal for a possible second plant. At the same time, Utah International will assign its existing water rights to WESCO. About 8,000 acre-feet of water per year will be used in the gasification process, as well as for mine use and selective irrigation. Fluor Engineers and Constructors, Inc. has been retained by WESCO to design and construct the first plant. It will require some three years to build after all the necessary approvals

have been obtained and financing arranged. I'll have more to say about this aspect later. In addition to coal and water, the plant will use 5,800 tons of oxygen each day...to be extracted from the atmosphere. Hourly steam requirements from in-plant boilers will be slightly over two million pounds. Seventy megawatts of electrical power will be required to operate the plant, including a river pumping station.

The plant layout is typical of most process-type plants. Units are arranged on a flow-line basis from gas production through final product compression. The plant will measure approximately one mile long and one-third mile wide. The plant site...including plant, holding ponds and coal preparation area...will occupy approximately 600 acres. All of the major process units are in a multiple-train configuration to provide maximum flexibility and on-stream productivity. The plant layout allows for the addition of the second gasification plant alongside the first. Many of the facilities...such as the administration building, change house, cafeterias, warehouse and maintenance facilities...would be common to both plants. Although the plant site appears level, its size and slight grade will require that 1.5 million cubic yards of earth be moved during site preparation for the first plant.

At the peak of construction activity, 5,300 people will be employed. The total construction payroll will be $240 million in 1977 dollars. The Company has established a goal of Navajo employment averaging about 50 per cent during construction. Some classifications are expected to reach 95 per cent Navajo employment.

Plant operation will involve approximately 600 employees. WESCO has agreed to an affirmative action program at all job levels, including those in management. Training programs for Navajos will begin early, with the expectation that 58 per cent of the workers will be Navajos at the end of the first two years. After 10 years of operation, about 87 per cent Navajo employment is expected. Another 500 employees will be required for Utah International's coal mine. The plant and mine operating payroll will be approximately $22 million annually.

Local taxes collected during the first five years are projected to total $49 million. Coal revenues for the Navajo Nation should be some $5 million each year from WESCO, plus $1.9 million from Utah International.

In addition to synthetic gas, the plant also will produce by-products with an estimated annual value of $41 million. These by-products will include carbon dioxide, sulfur, crude phenols, naphtha, tar oils and ammonia.

The environmental considerations of the WESCO project are extensive and complex. Battelle Columbus Laboratories

was selected to prepare the Environmental Impact Report. That study covered every conceivable aspect of the natural and human environment. It included design and operation features of the plant, mine and pipelines, as well as such socio-economic factors as the attitudes of the area residents and the effects on their lifestyles. The attention being given to environmental considerations is evident in the fact that $189 million in 1977 dollars is included in the project estimate for this purpose. The best proven environmental engineering technology available will be required to meet or better state and federal regulatory standards for atmospheric emissions and waste water quality.

Battelle has also organized and implemented a detailed "base line" environmental study to document the existing pre-construction conditions for the plant site and neighboring area. This will be an ongoing study to monitor and compare environmental data from before, during and after construction of the plant.

One of the most important environmental programs is the land reclamation proposed by Utah International and presented to the New Mexico Surfacemining Commission. The mining plan states that the reclamation will be three spoil piles behind the active mining operations. In a typical year, 280 acres will be mined. The mined area will be refilled to resemble a gently undulating topography that was determined to best resist wind and water erosion.

Utah International is continually striving to improve the techniques of land reclamation. The firm has been experimenting with vegetation test plots to determine the optimum means of revegetating the reclaimed land. All factors concerned with establishing growth are being considered. Once the land has been reclaimed and vegetation established, it will be turned back to the Navajos for grazing livestock.

If you're impressed by the magnitude of the design, engineering and construction problems inherent in the WESCO project...you haven't heard anything yet. These complex considerations have been nearly overshadowed by the headaches involved in gaining regulatory authorization to build this pioneering coal gasification plant. WESCO's first plant requires more than 70 separate approvals from no less than 30 different agencies at the federal, state, and local levels.

In September of 1971, Pacific Lighting and Texas Eastern agreed to undertake a joint study to determine the economic, technical and environmental feasibility of a coal gasification plant. That study was successfully concluded in July, 1972. Next came various preparatory activities such as negotiations and signing of a coal contract with Utah International and the preparation of an environmental impact report by Battelle Columbus Laboratories. After everything was in order, an application for project authorization was filed with the Federal Power Commission in February, 1973. From that point began an all-too-familiar pattern: regulatory delay followed

by more delays, compounded by continuing cost inflation.

I'm going to briefly describe for you the major approvals required for the project and how they have progressed.

In September, 1973, the FPC decided to limit its jurisdiction over the project. It would begin where the synthetic gas from WESCO's plant was introduced into Transwestern Pipeline Company's existing interstate transmission system. Accordingly, an amended application was filed that November. Three months later, the FPC completed public hearings on the project. But it wasn't until April, 1975...some 27 months after filing of the initial application...that the FPC approved the transportation and sale of SNG. In its decision, the Commission spoke of coal gasification in glowing terms. Unfortunately, the FPC didn't provide conditions which would enable the project sponsors to obtain financing. Petitions for a rehearing from us as well as the California Public Utilities Commission produced an amended order in November, 1975 with conditions that would finally permit project financing. But financing was by no means assured, as I will explain shortly.

Meanwhile, the Department of Interior's Bureau of Reclamation began preparation of an environmental impact statement in late 1973 when it became apparent that Interior rather than FPC would be the lead agency. Public hearings on the draft weren't held until March, 1975. And the final environmental impact statement was filed by the Secretary of Interior with the Council on Environmental Quality in January of last year.

The New Mexico Surfacemining Commission...following public hearings....issued a permit for operation of the coal mine to Utah International in July, 1974. Utah made a basic commitment to reclaim and revegetate the land to at least equal to the existing grazing capacity.

Permit authority to construct the gasification plant was received from the New Mexico Environmental Improvement Agency in September, 1974. The agency had to be satisfied that atmospheric emissions from the WESCO plant would not exceed ambient air standards established by the Environmental Protection Agency. This permit had a one year life, and refilings have been necessary.

Negotiations for a business site lease agreement with the Navajo Nation were initiated with the Tribal Administration in March, 1973. After nearly three years, we now expect the agreement to be considered by the Navajo Tribal Council in its next session convening in July. Just this past week, we made a formal presentation on the lease agreement to the Council.

Aside from the lease agreement with the Navajos, financing is now the key to development of the coal gasification project. Consider that due in large part to inflationary pressures and regulatory delays, the estimated cost to the WESCO project has

increased from initial estimates of less than $500 million to about $1.3 billion in current dollars. Each day of further delay adds another $290,000 to construction costs.

Simply stated, the coal gasification plant cannot be built by WESCO...or by private industry in general...without federal financial assistance. Our financial advisors have made it clear that the credit risks of the project are too great for lenders to assume without an additional source of credit-- namely, a federal loan guarantee. They recite these reasons: (1) the technology is, in part, untried on a commercial scale; (2) the cost of building a commercial plant is huge; and (3) the completion and operation of such a plant are subject to future restrictive governmental actions...such as a ban on strip- mining of coal.

A federal loan guarantee program would facilitate the development of a coal gasification industry through private investment, thereby limiting direct, tax-supported government financial participation. Actual expenditure of governmental funds would only be required in the unlikely event that the plant were not completed or failed to operate satisfactorily. A federal loan guarantee also would permit lower project financing costs and, therefore, lower costs to consumers.

Based on receipt of a federal loan guarantee, plans for financing construction of the WESCO plant envision 75 per cent debt capital with 25 per cent equity capital provided by the sponsors. The federal loan guarantee would apply only to the debt portion of the financing.

For the third straight year, Congress is considering loan guarantee legislation to encourage synthetic fuel development. If enabling legislation is finally authorized by Congress this year, the WESCO coal gasification project...the nation's first on a full-size commercial scale...could be producing SNG by 1983.

SESSION II

ERDA COAL FEED PROGRAM

DEVELOPMENT OF DRY COAL FEEDERS

J. H. Bonin

D. E. Cantey

A. D. Daniel, Jr.

J. W. Meyer

Lockheed Missiles and Space Co., Inc.
Lockheed Palo Alto Research Laboratory
Sunnyvale, California

ABSTRACT

The objective of this ERDA sponsored program is to generate analytical and test data to permit confident design and fabrication of equipment to feed coal into pressurized environments. These feed systems must be compatible with coal conversion demonstration plant requirements, and should lead to their use in commercial applications. A three phase program is in progress: concepts selection, laboratory scale development, and pilot plant evaluation. Results through the laboratory scale phase are reviewed.

Based on feeder system performance and economic projections, four concepts were selected: two approaches using rotating components, a gas or steam driven ejector and a modified standpipe feeder concept. Concept selection was limited to dry coal feeders which did not produce gross changes in coal physical properties. Lockhopper systems were excluded in the selection of candidates for development.

Test facilities were installed and development testing of critical components was accomplished. Design procedures and performance prediction techniques were developed and verified.

OBJECTIVE AND SCOPE OF WORK

The reliable feeding of large quantities of dry pulverized coal into pressurized reactors poses a challenging problem. Presently, some installations are using lockhoppers. However, at the higher operating pressures and for large throughputs, which will require large valves, these systems are beyond the state-of-the-art, or at best inefficient. Based on the available evidence, the reliability of these systems will also impact plant operatic .. Slurry systems using either process derived oil or water are in use or being contemplated. The slurries must be dried before further processing which has not been demonstrated for large size applications. This drying step clearly is detrimental to the overall plant efficiency. At present, no system is commercially available to feed large quantities of dry pulverized coal into pressurized reactors at the large rates projected for future gasification plants. The objective of the program is to generate sufficient analytical and test data to enable the confident design and fabrication of coal feeders which are compatible with demonstration plant requirements and commerical applications. The program is being performed in the following three phases

Phase I. <u>Selection of Concepts</u>. This phase, of six months duration, was designed to review potential candidates and equipment, synthesize designs, assess fundamental problem areas and define laboratory evaluation techniques.

Phase II. <u>Laboratory Scale Feeder Development</u>. During this phase of the program, laboratory size feeders were built and tested in a continuous loop test facility. The data resulting from laboratory testing will permit confident design of pilot plant size equipment.

Phase III. <u>Pilot Plant Evaluation</u>. During this phase of the program, feeders compatible with existing pilot plants will be designed, built, installed, tested, and evaluated. The data resulting from this effort will be sufficient to permit confident design of commercial size feeders.

At the present time, the program is near the completion of Phase II. In the following three sections the program results are discussed.

PHASE I EFFORT

Establish Requirements

To facilitate comparison of feeders and to establish uniform operating conditions for the several concepts to be evaluated, operating requirements were defined early in the evaluation. These requirements are shown in Table 1. The system elements required to take dry pulverized coal from an atmospheric bin and to deliver it in a dry pulverized form to a high pressure storage bin is shown schematically in Fig. 1. The only large state-of-the-art feeder presently able to handle pulverized coal in dry form is the lockhopper. No work to develop an improved lockhopper system or improved components for a lockhopper is planned under the present program.

For pressurization, it is assumed that for all gasification plants, process gases can be made available such as CO_2 in high BTU plants. These could be bled after cleanup and be available at high pressure (80% of reactor pressure will be assumed). The gases are also assumed to be cooled to room temperature.

For scalup consideration and sizing of equipment, consideration was given to future commercial size equipment requirements. Single reactor vessels having throughput rates of 180 tons/hr are being considered. It is assumed that such installations would, at a minimum, require three feeder systems sized such that two feeder systems are capable of supplying the full throughput, if one of the feeder systems requires repairs.

Patent and Literature Survey and State of the Art Review

A limited survey was conducted to establish prior art of solids feeder systems. About 50 patents, dating back to 1932, were examined and the open literature was surveyed through the Lockheed DIALOG (computerized information retrieval system) Index files.

Field Trips. On-site visits were made to the Morgantown Energy Research Center, the Argonne National Laboratory, the Hy-Gas facility of the Institute of Gas Technology in Chicago, the Bi-Gas Pilot Plant at Homer City, Pa., and the Synthane Pilot Plant at Bruceton, Pa. The purpose of these visits was to get a first hand look at the feeding equipment being used and to have an opportunity to discuss operating problems with the operators of these devices.

Table 1
COAL FEEDER OPERATING REQUIREMENTS

Pressure:	150 to 1500 psi
Coal Size:	Fine up to 1/8 in. size
High Pressure Hopper:	Hopper should have capability to store 1-hr flow throughput. This permits orderly plant shutdown during emergencies.
Temperature:	350^OF maximum
Moisture:	Coal is dry and should stay dry
Bulk Density of Coal	35 lb of coal/ft^3 0.56 g/cc, void fraction: 0.60
	25 lb of coal/ft^3 0.40 g/cc, void fract. n (fluidized): 0.71
Gas Properties for Pressurization Gas:	Use thermodynamic properties of CO_2 or process gas for calculations
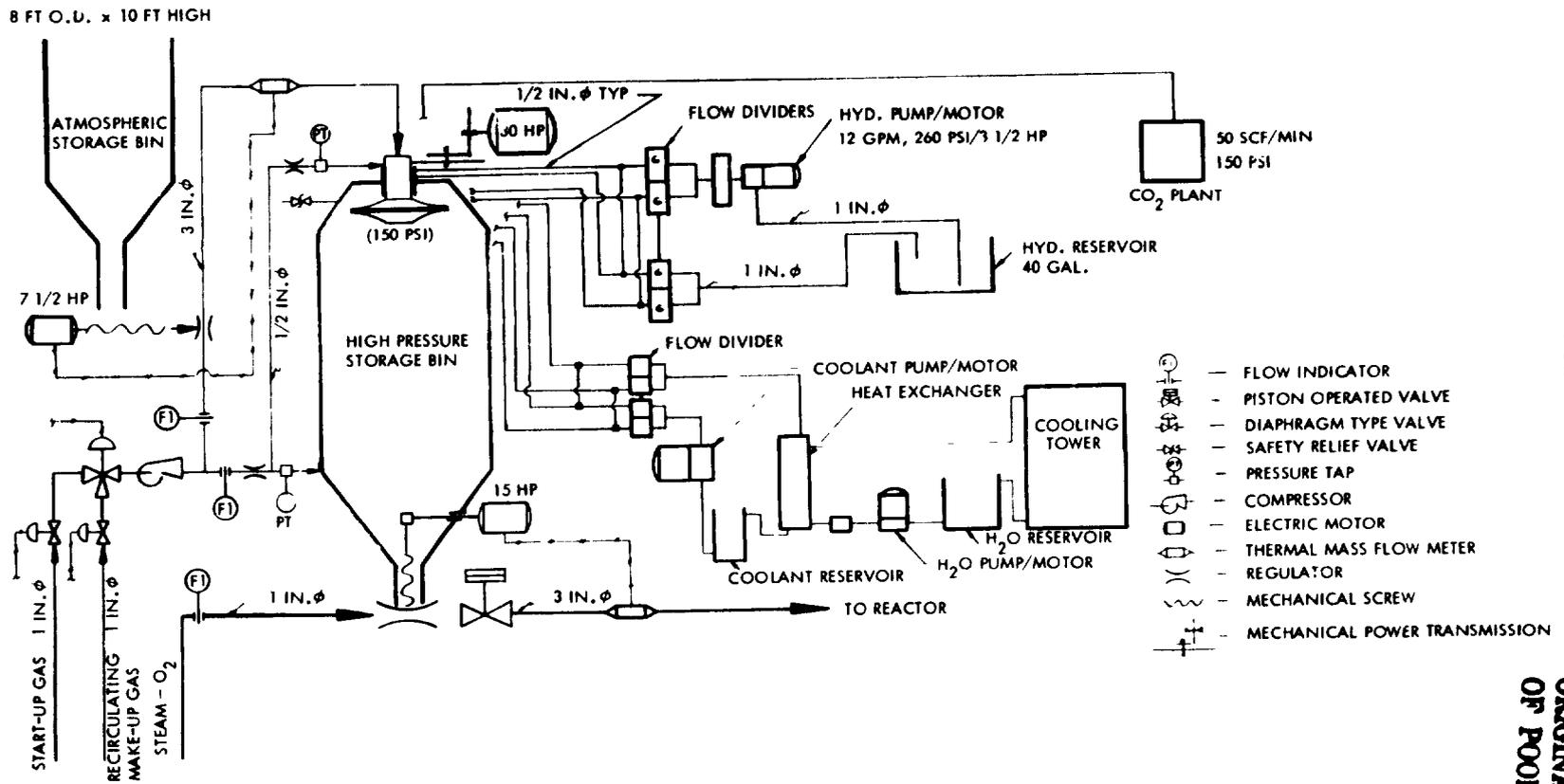
Fig. 1 Coal Feeder Schematic

At the Morgantown Energy Research Center, many diverse feeders have been used for small-scale experimental purposes. The large-scale fixed bed gasifier, the largest operational unit visited at the center, uses lockhoppers for pressurizing the coal to the reactor pressure level (300 psi).

The Argonne Fluidized Bed small-scale combustor used for laboratory type investigations also used a lockhopper type of coal pressurization scheme to achieve a pressure level of about 15 atm (225-psi level).

The IGT Hy-Gas process uses a process-derived light oil to slurry the coal and pump it at the 1000-psi level into the fluidized bed dryer section of the reactor. If dry coal could be fed into the reactor, the thermal efficiency of the cycle could be increased since the heat required to vaporize the oil could be used to heat other process streams.

The Bi-Gas Plant uses a water slurry system to pressurize the coal. Before transfer to the high-pressure storage bin, the slurry must be dried. Most of the heat required is supplied by an external heat source, and nearly 1000 BTU are required for each pound of coal to be dried. The plant efficiency could be raised significantly if dry pulverized coal could be fed directly into the high pressure bin.

The Synthane Plant is designed to use high-pressure lockhoppers. Design details cannot be made available, and few test data have been reported to date. Valve leakage problems can be anticipated at high operating pressures.

Concepts Considered

After reviewing the current practice, conceptual designs of dry pulverized coal feeders were developed. To focus attention on the more promising concepts, a pre-screening effort eliminated systems having obviously inferior potential compared with candidates selected for further consideration. The following fifteen concepts emerged from this process:

1. Fluid Dynamic Lock, based on the use of a bladeless centrifugal compressor
2. Kinetic Extruder, based on a rotating channel to impart centrifugal force to the coal particles
3. Ball Conveyor, using gravity forces to feed coal
4. Roller Pump, using an elastomeric roller for sealing
5. Gear Feeder, using the gear pump principle

6. Convolute Feeder, using a Root's blower type geometry
7. Centrifugal Compressor, using a conventional bladed impeller
8. Rotary Pump, using a Wankel engine type rotor
9. Piston Pump, using a reciprocating piston
10. Coal Pump, using a liquid-actuated displacement piston
11. Ejector, using a gas-driven jet pump
12. Lockhopper, using stationary pressure vessels
13. Screw Type Extruder, using plastic extruder technology
14. Positive Displacement Compressor, using gas compressor technology
15. Mechanical Conveyor, using solids handling technology

Coal Feeder System Synthesis and Economics

Based on an assessment of potential system performance, documented in Ref. 1, four concepts were selected for detail evaluation and incorporated into feeding systems for gasification plants. Feeders based on use of plastic extrusion technology were eliminated from consideration because development of this class of devices was already in progress by ERDA under separate contract. For similar reasons, lock-hopper feeders were also eliminated from consideration. However, work was performed on lockhopper systems sufficient to establish a basis for comparison of potential performance. The systems selected used the following concepts which will be described in detail in the discussion of the Phase II activity.

- Ejector
- Kinetic Extruder
- Ball Conveyor
- Fluid Dynamic Lock

Two types of gasification plants were used in the study, both with a nominal input rate of 50 tons/hr of dry pulverized coal to the reactor vessel. One plant shown schematically in Fig. 2 was designed for the production of low BTU gas at an assumed reactor pressure of 150 psi. The other plant, shown schematically in Fig. 3 was designed to operate at 1500 psi and was designed to produce high BTU gas.

As indicated in Figs. 2 and 3, the product gases represent an output rate of 1095×10^6 BTU/hr for the high BTU plant and 1253×10^6 BTU/hr for the low BTU plant. Each design uses two feeder trains of 25 tons/hr capacity.
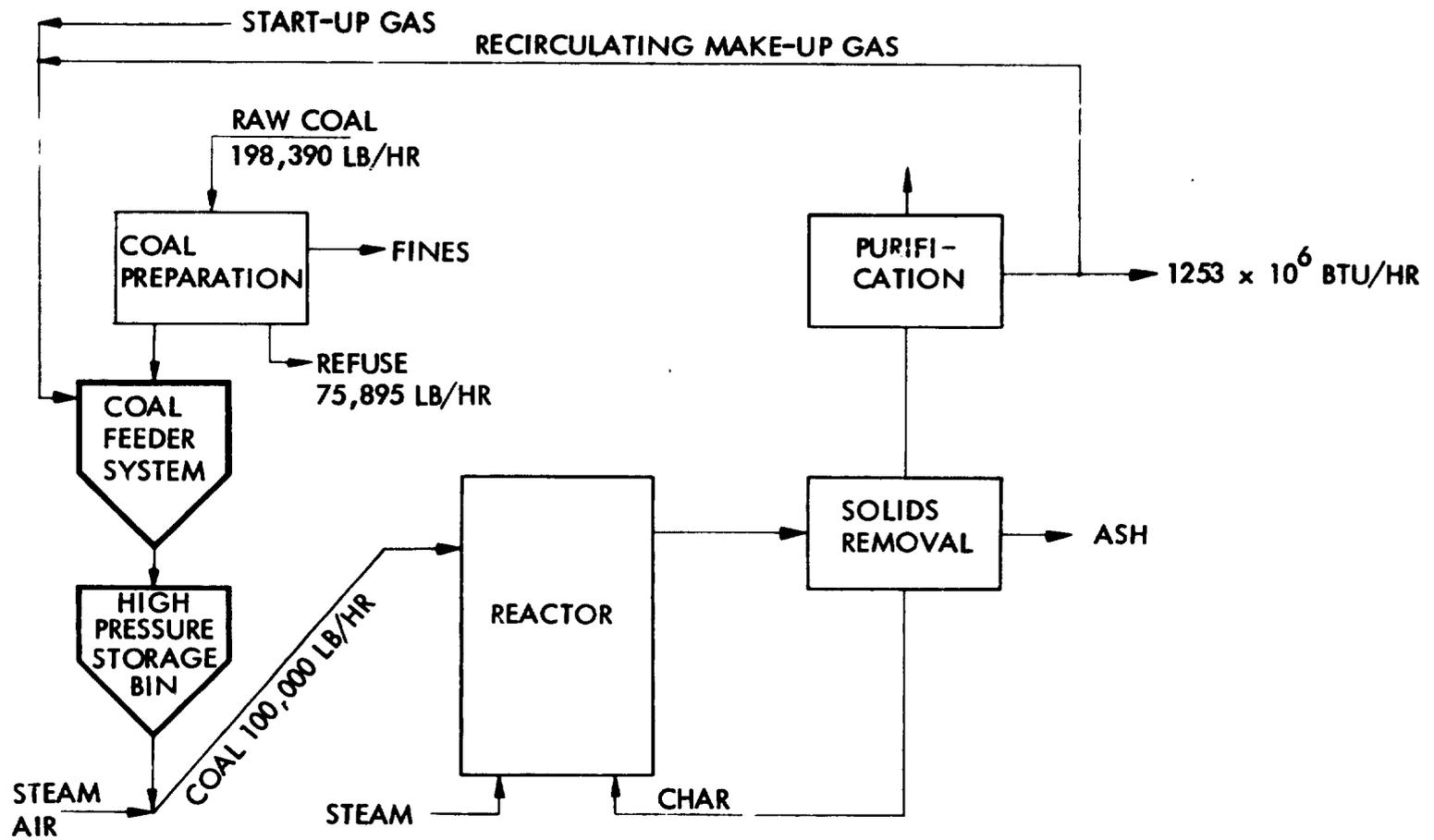
Fig. 2 Coal Gasification Process (Low BTU)

START-UP GAS

RECIRCULATING MAKE-UP GAS

RAW COAL
198,390 LB/HR

COAL
PREPARATION

FINES

SHIFT
CONVERSION

PURIFI-
CATION

METHAN-
ATION

REFUSE
75,895 LB/HR

STEAM

1095×10^6 BTU/HR

COAL
FEEDER
SYSTEM

HIGH
PRESSURE
STORAGE
BIN

COAL 100,000 LB/HR

REACTOR

SOLIDS
REMOVAL

ASH

OXYGEN
PLANT

AIR

STEAM
OXYGEN

STEAM

CHAR
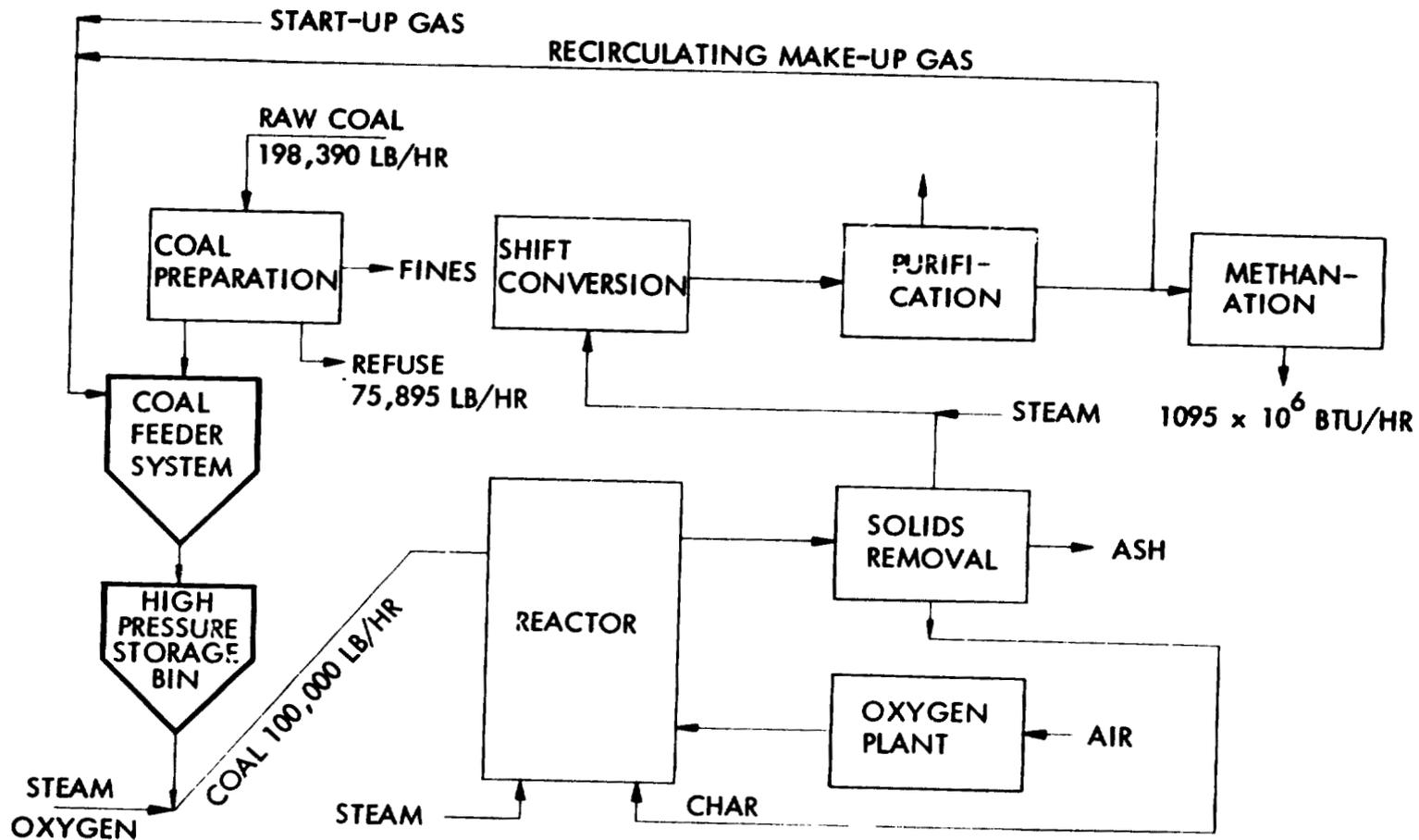
Fig. 3 Coal Gasification Process (High BTU)

For each system, flow diagrams were prepared and the cost of the major equipment was estimated, using the performance parameter derived during concept evaluation. Next, the erected cost of the feeder was determined by considering the need for ancillary equipment, foundations, structure, labor. etc. The direct operating cost was determined by calculating the energy requirements and operating labor cost. Electric energy was charged at $0.025 per k-Wh, and all-up labor cost was taken at $20 per labor hour. Maintenance and annual overhaul costs were determined, based on equipment complexity and estimated equipment costs. Based on these figures, the total annual cost of owning and operating the feeder system can be calculated. This cost was subsequently used to determine the contribution of the feeder system to the cost of the product. Details of this equipment sizing and the subsequent economic analysis have been presented in Ref. 1.

Feeder Systems Evaluation and Selection

The feeder system concepts were evaluated by considering such issues as technical feasibility, the requirement to develop new manufacturing technology, the technological risks involved, projected service life, maintenance and reliability, equipment costs, space requirements, and energy consumption. An evaluation and comparison matrix considering all these factors is difficult to develop when the equipment used involves wide differences in operating principles.

In the final analysis, the most important criterion for the selection of equipment is cost. To evaluate feed systems, we have therefore used the following method:

- It is assumed that the selected systems will perform as predicted.
- Development costs are not recovered by future commercial sales.
- All evaluation criteria are expressed in monetary terms.
- The feeder system used does not affect the cost of the balance of the plant.
- The figure of merit is the contribution of the feed system to the product cost (dollars/million BTU).

As indicated, the cost of the system is determined from preliminary designs of the major components. The energy consumption is based on performance calculations while the cost figures reflect costs associated with the following factors:

- Reliability
- Safety
- Maintainability
- Ease of operation
- Wear

The result of the cost analysis are shown in Table 2 for the low-pressure, low-BTU gasification plant, and in Table 3 for the high-pressure, high-BTU plant.

Table 2

COST DATA FOR FEED SYSTEMS: LOW BTU PRODUCT GAS (150 PSI)

	Concept	Equipment Cost (Erected) ($/ton/hr.)	Total Operating Cost of Fee ($/million)
1.	Ejector	$ 30,918	$ 0.126
2.	Kinetic Extruder	18,973	0.072
3.	Ball Conveyor	45,000	0.117
4.	Fluid Dynamic Lock	52,106	0.116

Table 3

COST DATA FOR FEED SYSTEMS: HIGH BTU PRODUCT GAS (1500 PSI)

	Concept	Equipment Cost (Erected) ($/ton/hr)	Total Operating Cost of Feeder ($/million BTU)
1.	Ejector	$ 112,094	$ 0.298
2.	Kinetic Extruder	36,173	0.154
3.	Fluid Dynamic Lock	55,303	0.293

To establish a reference point, an attempt was made to use data from Ref. 2 to estimate the cost of a high-pressure slurry and of a high-pressure lockhopper system, using the same groundrules which were applied to the systems contemplated here. The results indicated that the novel systems are economically viable and that if throughput can be increased beyond the conservative figures used here, a significant performance advantage might be achieved. The results of the effort represented in Tables 2 and 3 will have to be reassessed using the results obtained from Phase II of the program.

At the conclusion of Phase I, it was recommended that the four systems selected be carried forward into the laboratory testing phase. It had been shown that the systems were economically viable, but insufficient data existed to refine the designs or construct feeder systems with a high confidence of achieving efficient operation. Obtaining these data is the objective of the Phase II effort.

PHASE II EFFORT

Test Facility

A special test loop was designed and constructed for test and evaluation of the feeders. The equipment has been installed in the Energy Systems Test Facility at Lockheed's Sunnyvale plant. The 1000 ft^2 facility was originally designed and equipped for testing high-speed energy-storage type fly wheels.

The coal feeder test loop is installed as shown in Fig. 4 using the larger of the two spin pits. Fresh coal is loaded into the low-pressure tank and pneumatically transferred to the upper tank. From here it enters the feeder under evaluation and is discharged into the lower high-pressure tank. The coal is transferred pneumatically back into the upper tank which is also designed to withstand the high pressures.

The three vessels incorporate provision for zone fluidization to provide leveling of the coal surface and to assist in dense phase transfer from the bottom of the tanks under slight pressure differentials. To accommodate the great range of test conditions, flow to the fluidization manifold has been divided into three zones. Each can be separately controlled. The center section adjacent to the transfer line inlet fluidizes a 6-in.-diameter section of the bed. Gas is fed through twelve each 1/32-in. ports at a nominal flow rate of 1 cfm. This section is surrounded by a second manifold feeding an array of four circular tubes with a total of 48 ports, each of 3/64-in. diameter. The third manifold feeds two circular tubes having 40 ports, each of 1/16-in. diameter. The nominal flow rate for the number two manifold is 5 cfm, and the number three manifold is 8 cfm. Identical zone fluidization systems are used in all three tanks.

The pressurization and fluidization gas are supplied by a tube trailer, and the vent gases are cleaned by passing through a bag filter house before venting to the atmosphere. The high pressure vessels are designed for a maximum operating pressure of 1500 psi at a maximum temperature of 450°F. They have a capacity of about 40 ft^3.

DUST COLLECTOR

BALL CONVEYOR TEST STAND

UPPER HIGH PRESSURE TANK

LOW PRESSURE TANK

AIR TURBINE

FLUIDIZED COAL TRANSPORT PIPE

INSTRUMENTATION AND MAT'L SAMPLE PORTS

FEEDER ROTOR

LOWER HIGH PRESSURE TANK
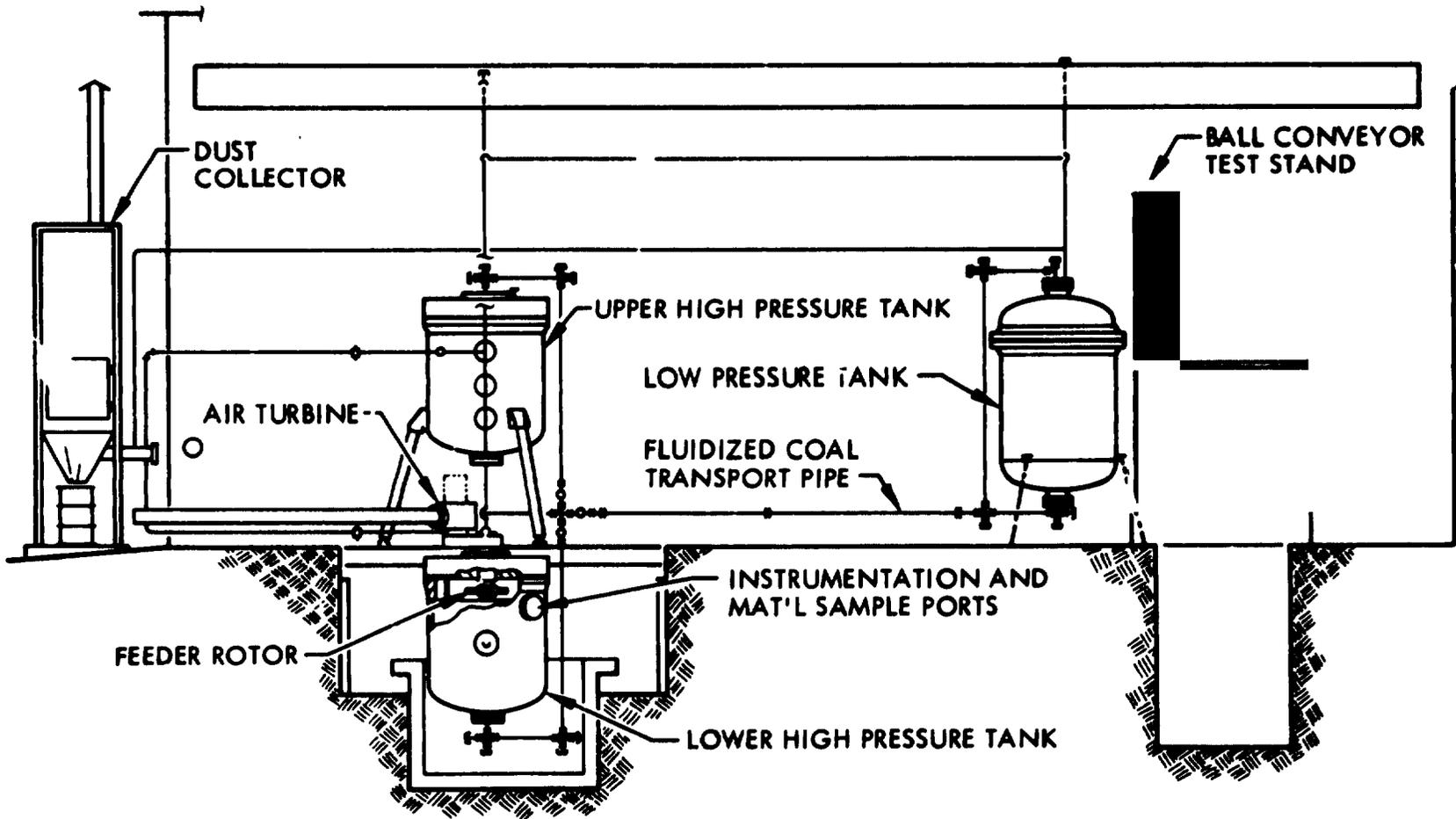
Fig. 4 Facility Layout - Coal Transfer Lines

Ejector

The use of compressed gas-driven ejectors or jet pumps offers the possibility of a coal feeder with no mechanical moving parts in the coal-handling section of the unit. Theoretical calculations, performed during the Phase I effort, indicated that pumping energy requirements for a feeder of this type may be competitive with those of other dry pulverized coal feeder candidates. In addition, staging concepts were investigated which indicated the possibility of multistage ejector units which could be driven by a central recycling gas compressor and a low-pressure gas cleanup unit. Based on these encouraging theoretical results, the ejector approach was selected as one of the four concepts identified for experimental evaluation under Phase II of the program.

Figure 5 presents a schematic of the coal ejector and a description of the operating principle. Driving gas is introduced into the ejector mixing section from an annular nozzle surrounding the coal inlet pipe. The annular driver jet (primary) nozzle configuration was selected to simplify the geometry of the secondary, coal flow inlet into the ejector mixing section. Gas from the high-pressure supply accelerates and drops in pressure as it flows into the converging section of the primary nozzle. At the annular nozzle throat, the flow velocity has increased to the local speed of sound (Mach 1) and the flow continues to accelerate and drop in pressure as it expands through the diverging section in supersonic flow. At the exit of the primary nozzle (mixing section entrance), the driver gas has a high velocity and Mach number greater than 1 and a static pressure somewhat smaller than the pressure in the coal flow at the entrance to the mixing section.

Coal flows from the coal supply reservoir with relatively low velocity and enters the mixing section at a pressure which is lower than the supply reservoir pressure by an amount equal to the flow pressure drop in the coal feedline. This pressure drop is a function of coal flow rate, feedline geometry, and design.

In the mixing section, the coal is accelerated by momentum transferred from the high-velocity driver gas. As the mixing of the two phases proceeds, the coal velocity increases and the driver gas velocity decreases with a corresponding rise in pressure until a uniform mixture of coal and gas at equal velocity is achieved at the outlet of the mixing section. The velocity of the mixture is subsonic, but is still appreciable.
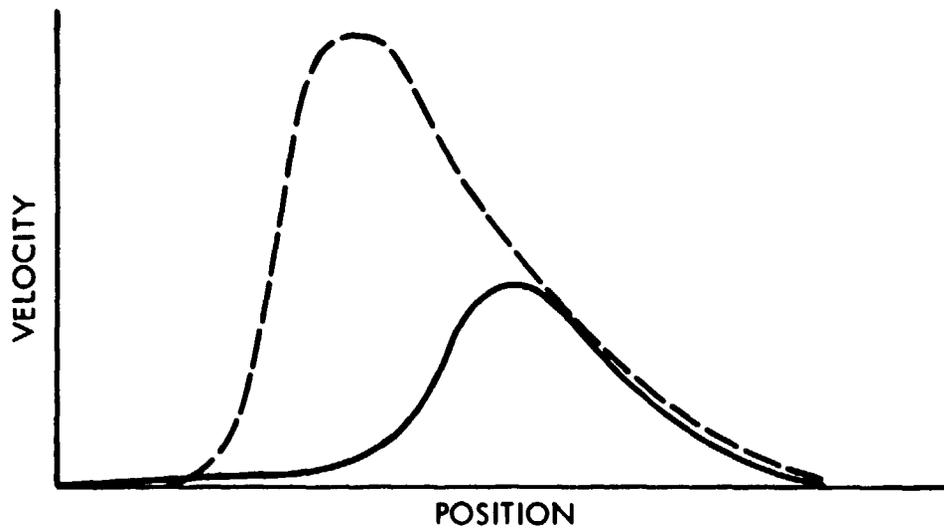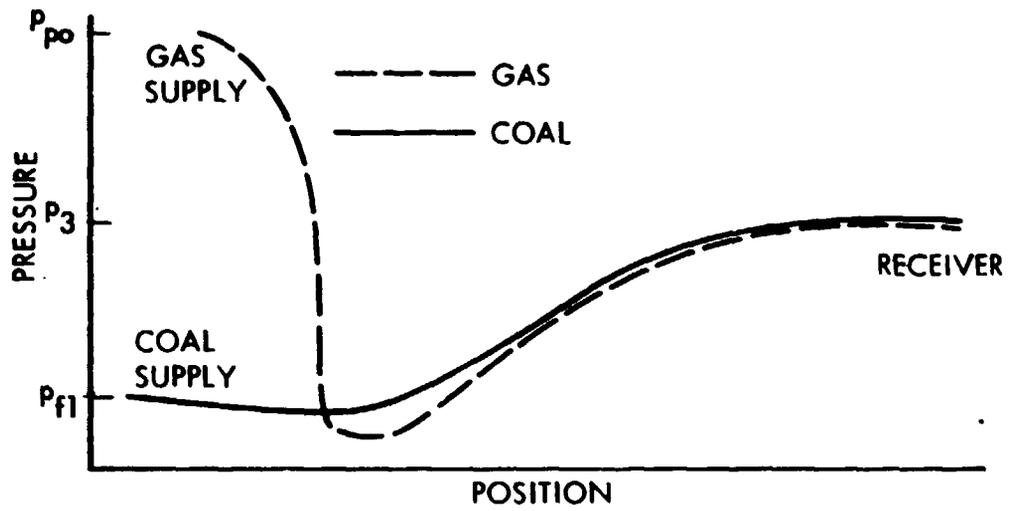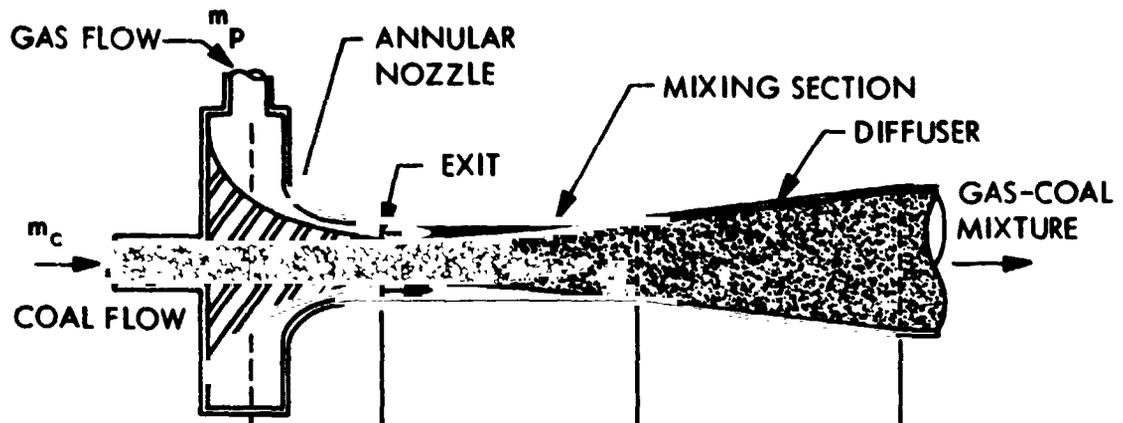
Fig. 5   Ejector Schematic

Flow of this subsonic mixture through a diverging section (diffuser) results in deceleration of the mixture with corresponding transfer of kinetic energy into a further increase in pressure of the flow.

The net result of this process is the transfer of coal from the low pressure reservoir to the high pressure receiver vessel. Mechanical work must be expended to maintain the gas supply at elevated pressure. The minimum work required is that associated with pumping the driver gas from the receiver pressure back to the ejector supply pressure.

The development work comprised an analytical and an experimental phase. The analytical effort resulted in a computer aided design procedure which is used to trade off design options and to evaluate the ejector performance. This mathematical treatment of the ejector makes use of the conventional control volume approach based on conservation of mass, momentum, and energy, and assumes that the gas properties are defined by the perfect gas relationships. Friction factors were derived from experimental data. The theoretical development is described in detail in Ref. 3.

Two basic ejectors were built for conducting the experiments. The first unit was a bench scale device capable of handling about 200 lb/hr of coal. This unit has been operated with room temperature nitrogen gas and also with saturated steam as the driving medium. The test flow diagram for these measurements is illustrated in Fig. 6 as arranged for testing with the steam driver. A larger 1000 lb/hr ejector unit was built and operated with room temperature nitrogen at the Test Facility to investigate size scaling effects.

These tests have verified the analytic design procedure for driver gases which exhibit no condensation effects and for a saturated steam driver in operating regimes where condensation effects are negligible. A typical comparison of test data and predicted performance for the bench scale device driver by saturated steam is shown in Figure 7. Symbols used in this figure are identified in Figure 6. Friction factors used for these performance predictions were obtained from experimental results with this ejector unit driven by room temperature nitrogen gas. The "design operating point" identified in Figure 7 is defined by the intersection of the lower branch of the theoretical mixing section outlet pressure curve with the secondary inlet pressure line as discussed in Ref. 3. The increased performance as compared with predictions to the right of the design operating point in Figure 7 is attributed to condensation effects in the steam driver which are not accounted for in the present theoretical model.
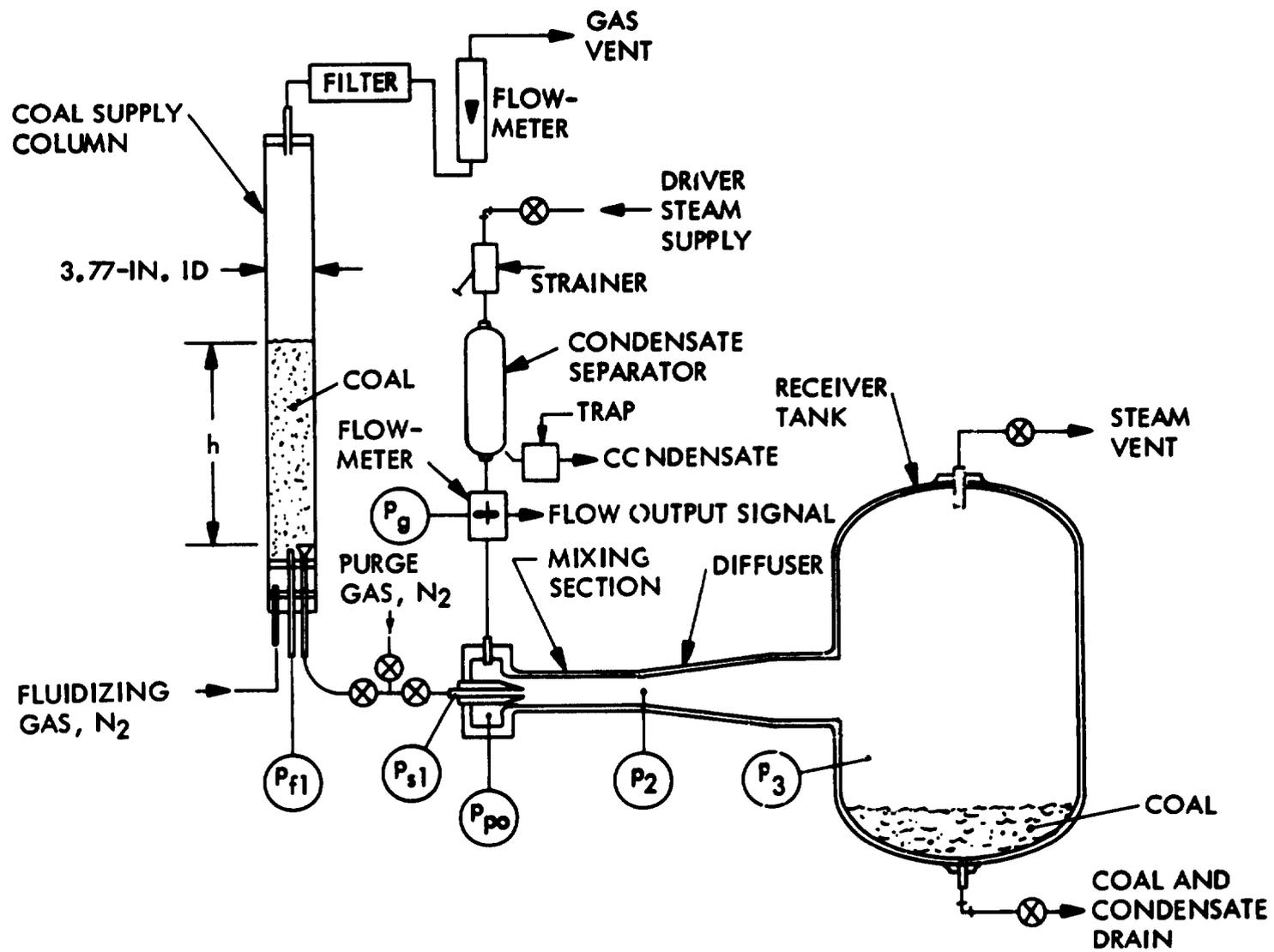
Fig. 6 Laboratory Test Schematic - Steam Driven Ejector
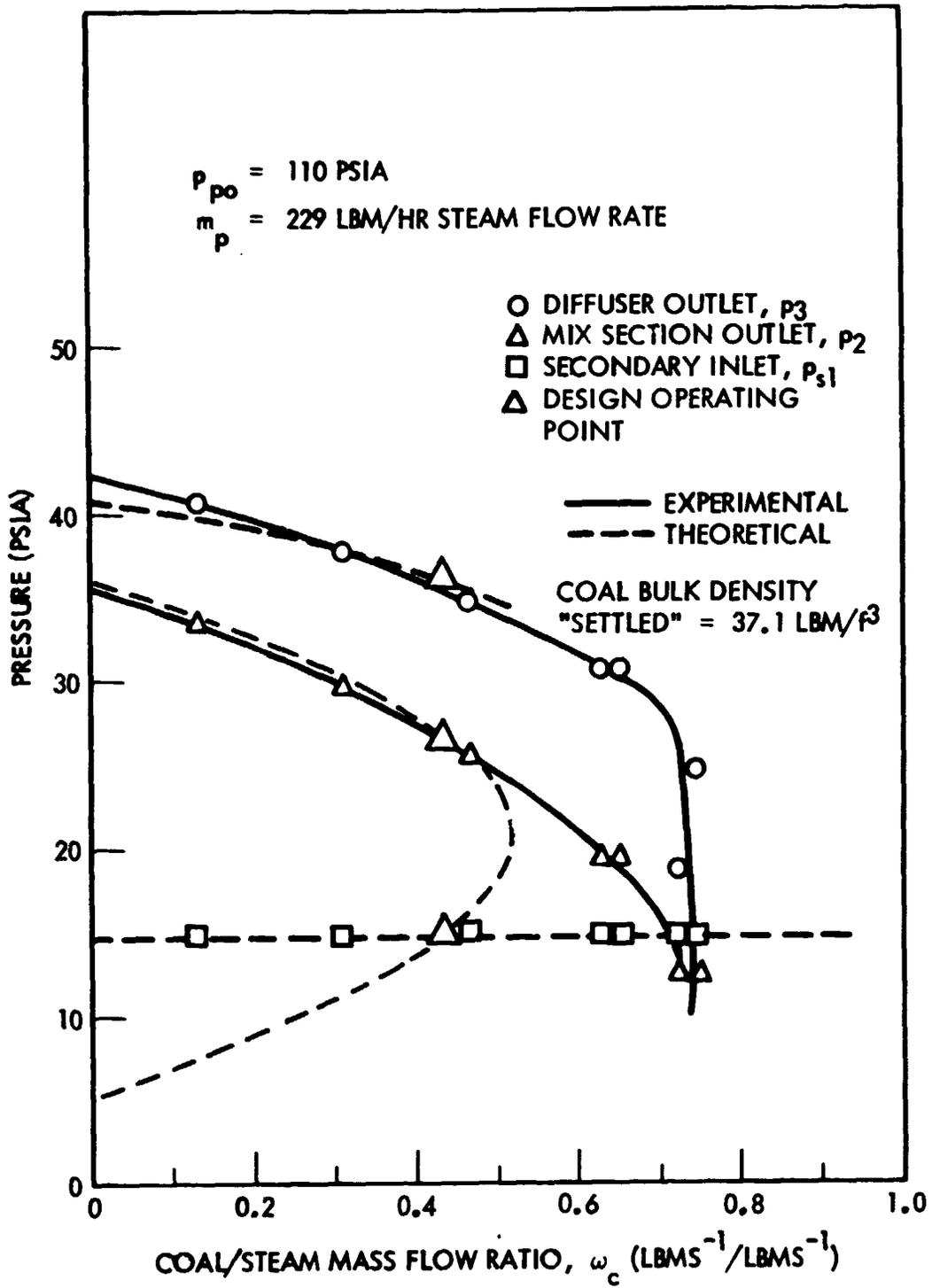
Fig. 7 Ejector Performance with 110-psia Steam Drive

Using a computer implementation of the theoretical model, performance can be predicted for a variety of ejector geometries, operating conditions, and scale sizes and optimum configurations can be selected. This procedure has been used to define ejector designs that achieve a high ratio of coal throughput and pressure increase for a given power expenditure. The result of such a study is shown in Figure 8 for independent ejector stages operated in series to achieve an overall system pressure ratio requirement, P_{co}, defined as the ratio of the coal bed pressure at the outlet of the multistage system to the coal pressure at the inlet of the system. Each stage is assumed to have the same coal pumping pressure ratio, P_c, which is treated parametrically in Figure 8. The driver gas examined here is a mixture of N_2 and CO_2 typical of inert gas generator products and is assumed to have a stagnation temperature, $T_{po,} = 135^O F$ at the ejector driver inlet. The minimum compression work is expressed in BTU equivalents of mechanical work and friction factors used in the predictions were scaled to correspond to the size of units required for coal throughputs of the order of 50 tons-per hour.

Similar calculations were carried out for different driver gas conditions and lines of minimum work are shown in Figure 9 for three different gases. For the elevated driver gas temperature (T_{po}) cases, it was assumed that the driver gas exhaust from each stage was cooled to a temperature, T_o, of $135^O F$ prior to recompression and reheating.

These curves clearly show that for a given pressure differential, the ejector requires relatively large power at low pressures and operates more efficiently at high pressures.

Direct comparison of steam and room temperature nitrogen gas drive data in the region where steam condensation is not significant shows a performance advantage for the steam. In the condensation region at high coal-to-steam ratios, performance can be achieved with steam which is not possible with nitrogen. Theoretical performance comparisons between steam and nitrogen in the region where steam condensation is not significant shows the steam advantage to be due to the higher steam temperature, i.e., nitrogen drive at the corresponding saturated steam temperature produces about the same performance as steam.

In summary, an analytical tool has been developed and verified by experiments which permit the evaluation of ejector feed systems for design trade-off studies. At
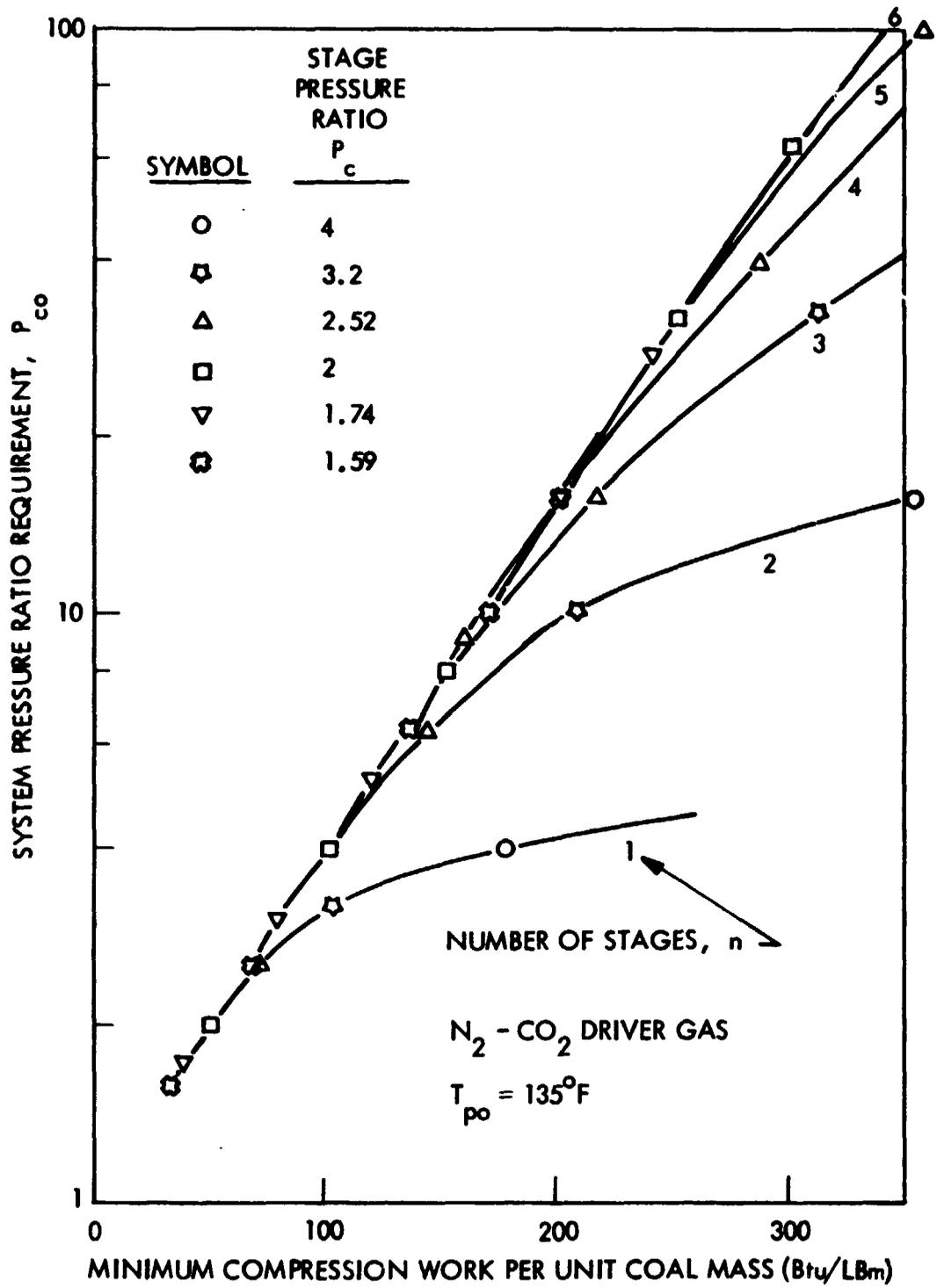
Fig. 8 Multistage Performance with Optimum Stages
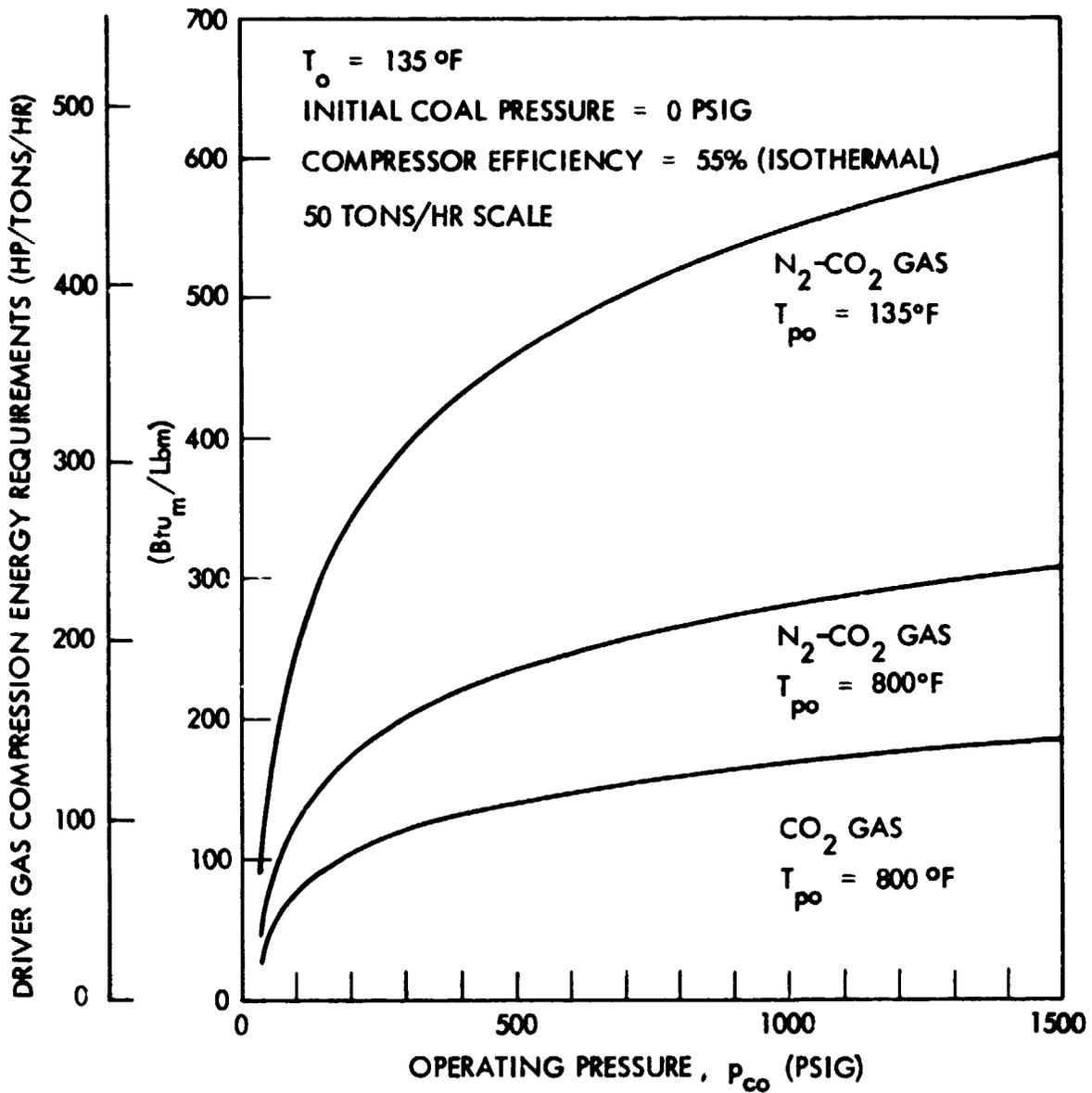
Fig. 9 Ejector System Compression Energy Requirements

this time, the ejector appears well suited as a booster or topping stage in high pressure systems. The use of steam as the driving fluid should also be explored if it proves compatible with the process under consideration. Present plans call for the evaluation of ejectors as a booster stage for Pilot Plant application.

Kinetic Extruder

The kinetic extruder shown in Fig. 10 uses centrifugal force to compact the solids particles and move them continuously through channels in a high speed rotor. The coal packed in the converging channels forms the gas seal. Excess gas at the channel entrances is removed through a vent line.

It should be noted that the forces acting on the particles are predominantly body forces caused by the centrifugal force field. Thus the particles are not pushed as by a cylinder or feed screw through the flow channel and bridging or similar phenomena do not interfere in the flow of particles through the channel. This concept offers a good chance of achieving high pressure levels (1500 psi) with a minimum number of stages.

To obtain stable operating conditions, the kinetic extruder must be designed to maintain a balance between the relatively low bulk density flow of coal through the feed tube, the packed bed coal flow through the sprue and the gas flow through the vent line. In addition, attention must be paid to the design of the transition region where the vertical downward flow in the feed pipe changes to the predominantly radial flow in the sprue. This region must be designed to handle the required coal flow rate to ensure that the flow rate controlling choke point is located at the sprue exit. If the choke point is located in the feed pipe or the transition region, the coal plug forming the gas seal in the sprue can not be maintained and blowback will result.

Computer based analytical tools have been developed to guide the design of the kinetic extruder. The design of the sprue shape is based on a mathematical model which treats, in one dimensional form, the percolation of gas into a moving, porous coal bed. For a given channel geometry, one obtains gas flow and pressure distribution as a function of the delivery pressure and the coal flow rate through the channel. A well designed channel has low gas leakage flow and a pressure gradient distribution which is nearly linear, but peaks toward the sprue exit.
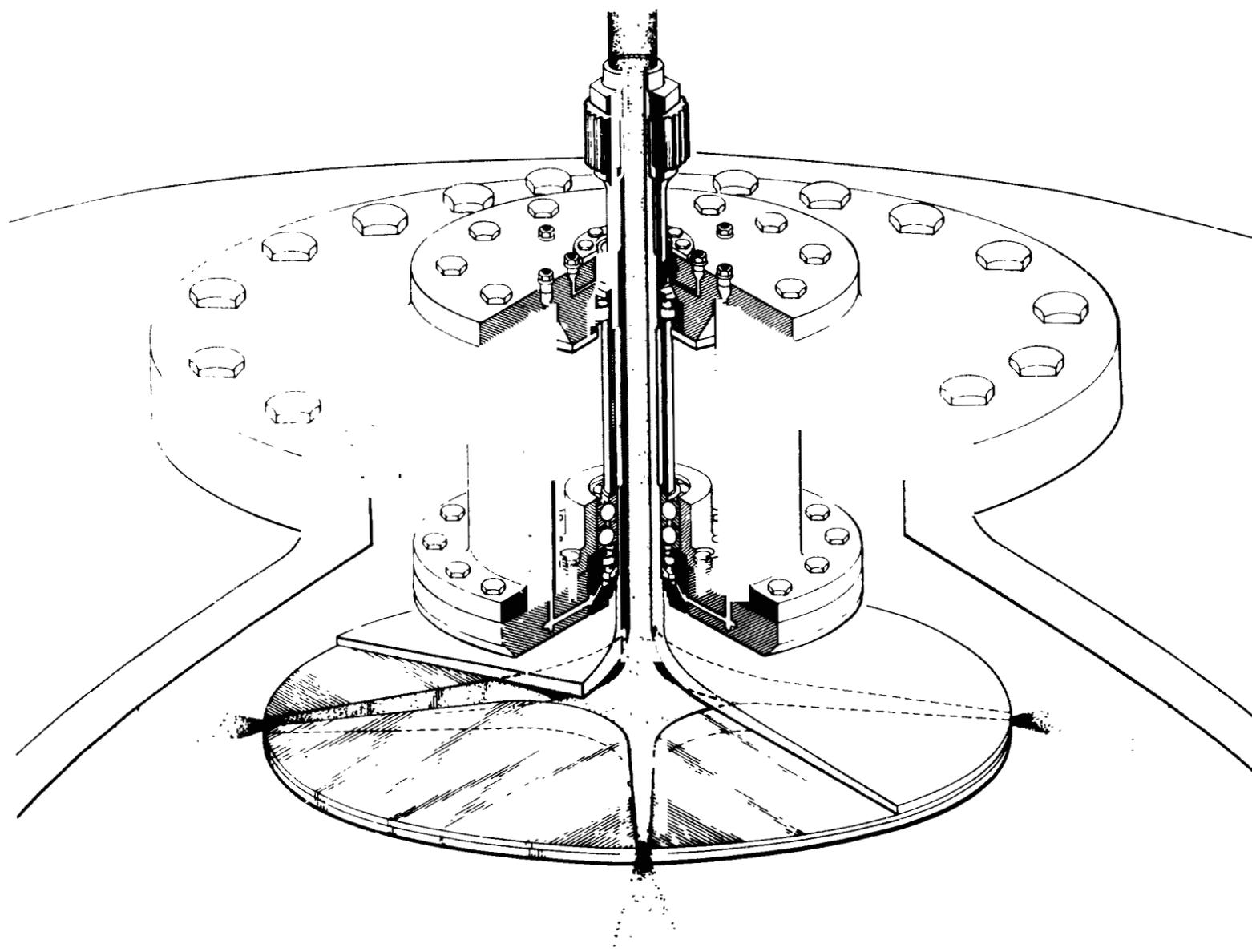
Fig. 10 Kinetic Extruder Concept

The coal flowrate predictions shown in Fig. 11 are derived from two sets of theoretical considerations which we term "pressure controlled" and "friction controlled". At sufficiently high delivery pressure, in the "pressure controlled" regime, the interparticle solids forces are negligible in comparison to the gas pressure forces and the coal flowrate is determined from the balance between the gas pressure gradient and the centrifugal body force at the sprue choke point.

In the "friction controlled" regime, the coal flowrate is calculated from a modified bin flow equation, which accounts for the large certrifugal forces. The flow rate is assumed independent of delivery pressure in this regime.

The "friction controlled" and "pressure controlled" solutions are matched at the point where they both yield the same coal flowrate. As shown in Fig. 11, the kinetic extruder coal delivery rate is predicted to be independent of back pressure up to a critical value and to then fall off rapidly as the "pressure controlled" mechanism takes over.

Two kinetic energy feeders were built for the experimental phase of the program. The test setup is shown schematically in Fig. 12. As indicated, the rotor is mounted inside the lower tank. The test rig is fully instrumented and key data are preserved on a strip chart recorder. The rotor is attached to a hollow drive shaft. The drive shaft is driven through a gear box. Rolling element bearings and face seals are used to seal the assembly, as shown in Fig. 13. For initial testing, an existing Barbour Stodwell air turbine was used to supply the input power. This has now been supplanted by a variable displacement hydraulic pump.

The first rotor head tested is shown in Fig. 14. Test results indicated that the transition zone was rather ineffective and tended to form the choke point. This wheel could not provide the required coal flow and mechanical difficulties were encountered. In particular, coal dust penetrated the space between the stationary feed tube and the rotating drive shaft. The resulting friction caused overheating as well as damage to the face seal near the tube flange.

The kinetic extruder was redesigned to overcome the observed mechanical difficulties. The new Model 2 is shown in Fig. 15. A bearing and face seal have been provided to prevent coal from entering the space between the stationary feed tube and the rotor shaft; the seal is buffered by purging nitrogen gas flow through a labyrinth passage. The transition zone has been enlarged and coal enters the rotor
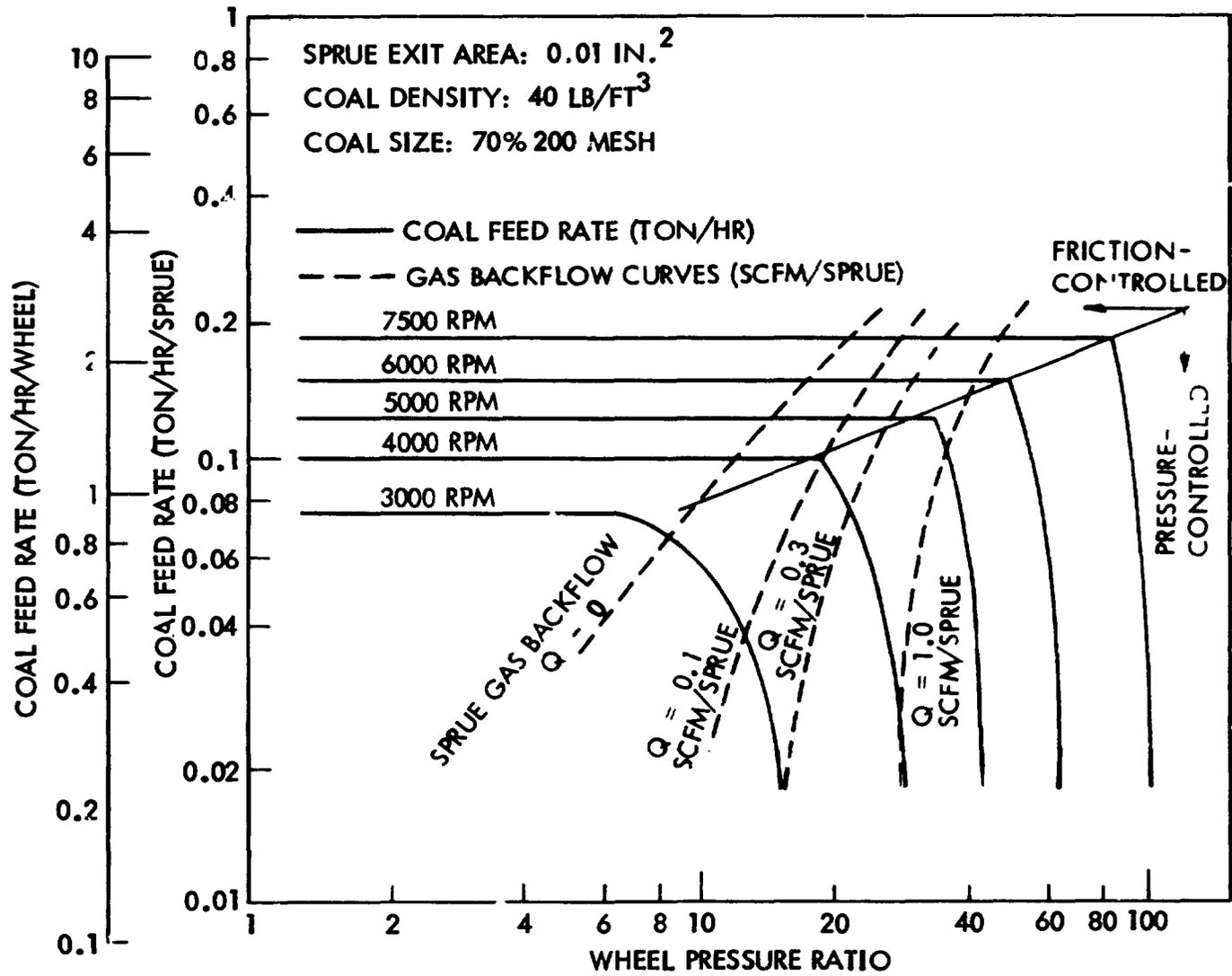
Fig. 11 Performance for Kinetic Extruder No. 2
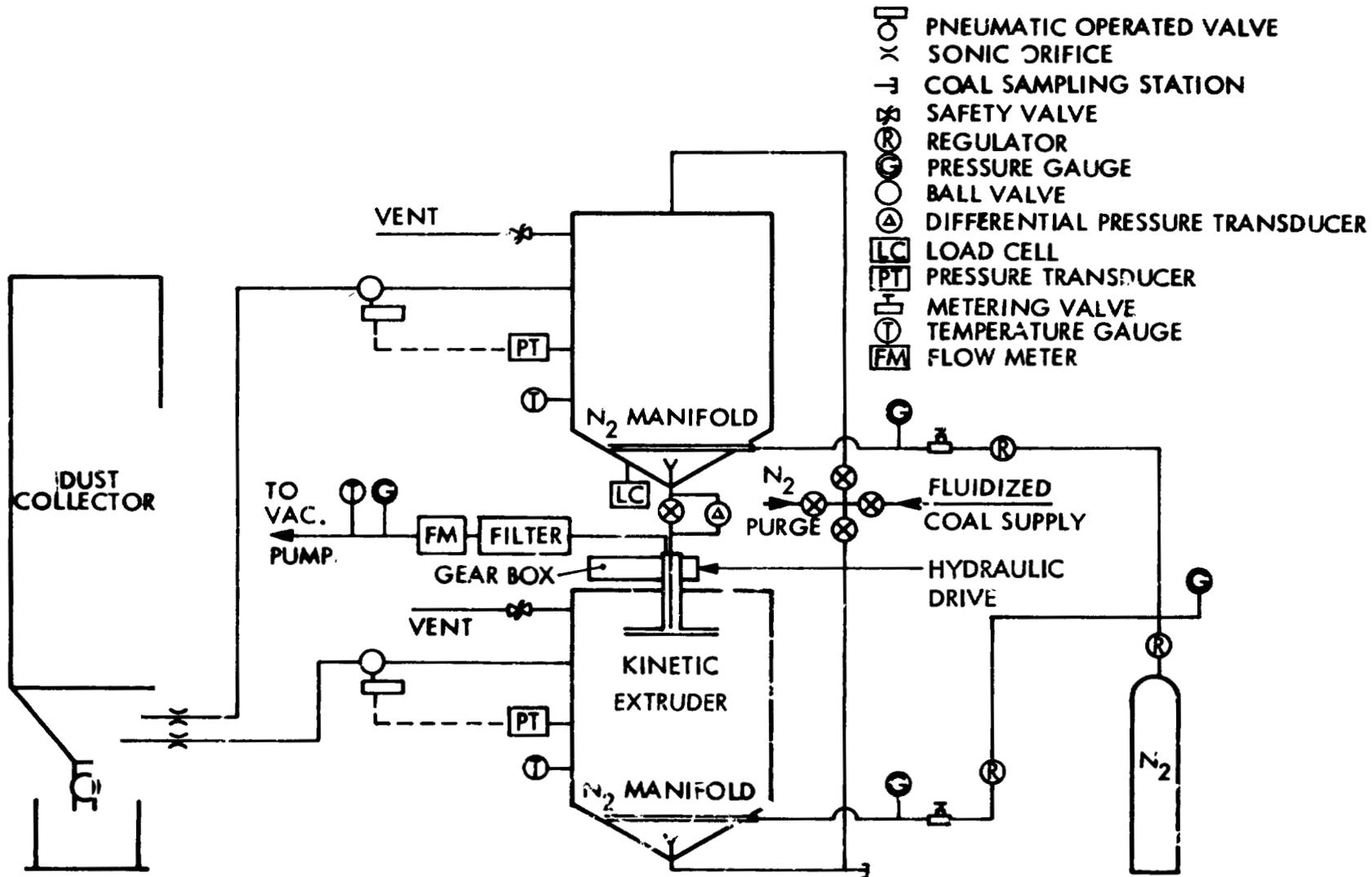
Fig. 12 Kinetic Extruder Test System Schematic

Legend:
- ⊟ PNEUMATIC OPERATED VALVE
- ⋈ SONIC ORIFICE
- ⊣ COAL SAMPLING STATION
- ⋈ SAFETY VALVE
- Ⓡ REGULATOR
- Ⓖ PRESSURE GAUGE
- ◯ BALL VALVE
- △ DIFFERENTIAL PRESSURE TRANSDUCER
- LC LOAD CELL
- PT PRESSURE TRANSDUCER
- ⊔ METERING VALVE
- Ⓣ TEMPERATURE GAUGE
- FM FLOW METER

TURBINE INTERFACE

INPUT SHAFT ℄

FEED TUBE FLANGE
INTERFACE

PRESSURE TANK
INTERFACE

FACE SEAL

℄ OUTPUT SHAFT
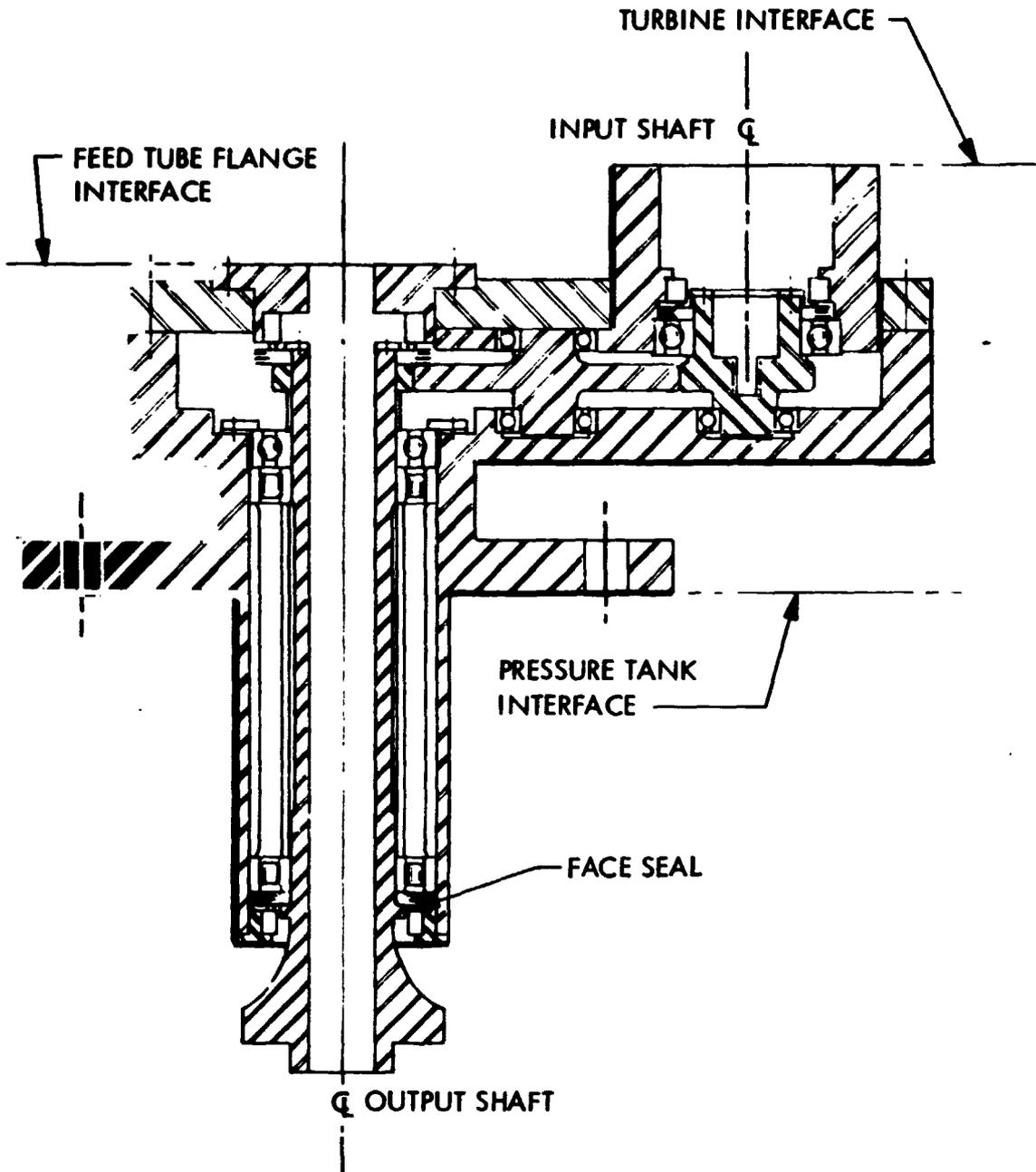
Fig. 13   Rotor Drive System
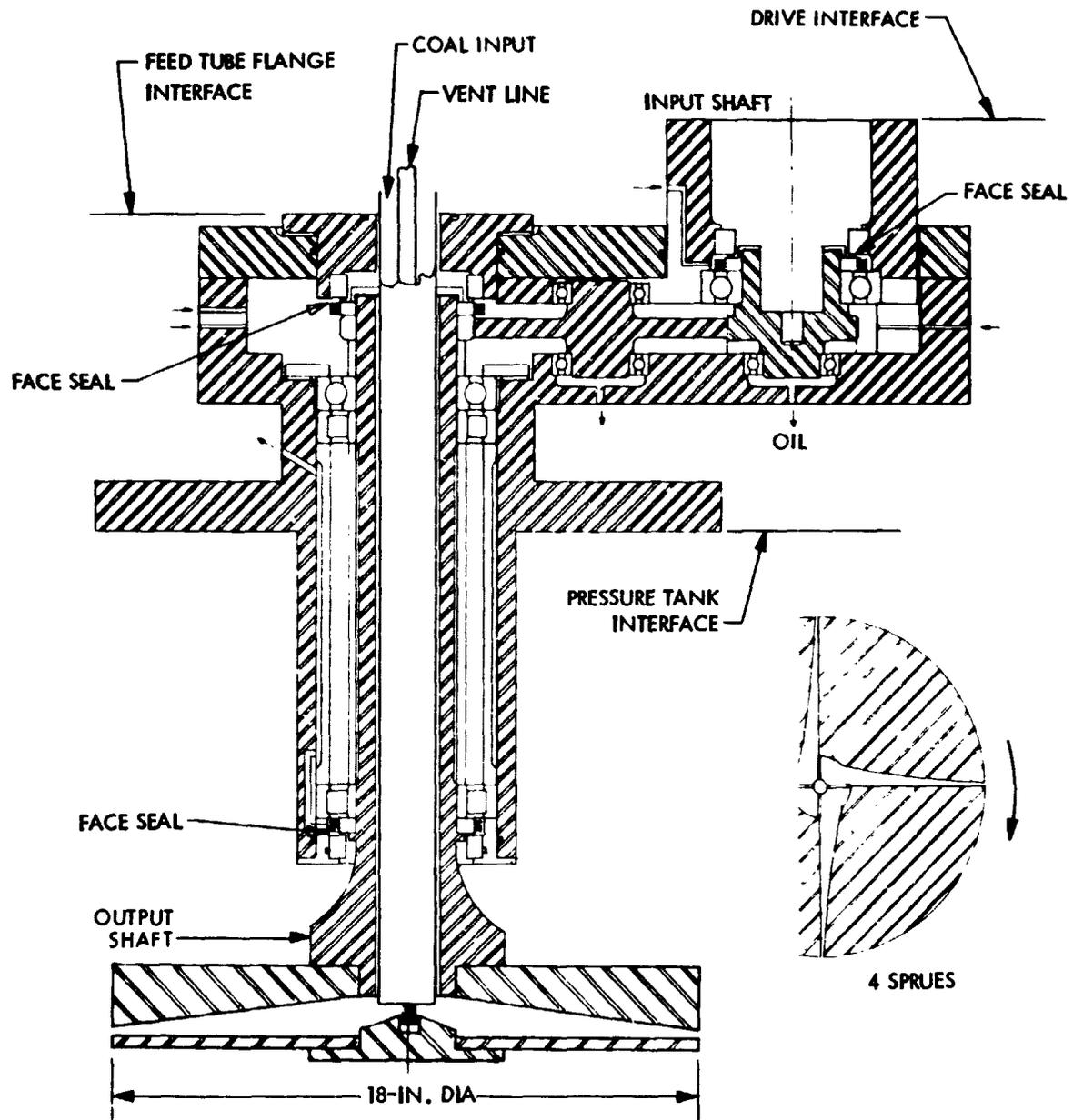
Fig. 14 Kinetic Extruder – Model No. 1 Configuration
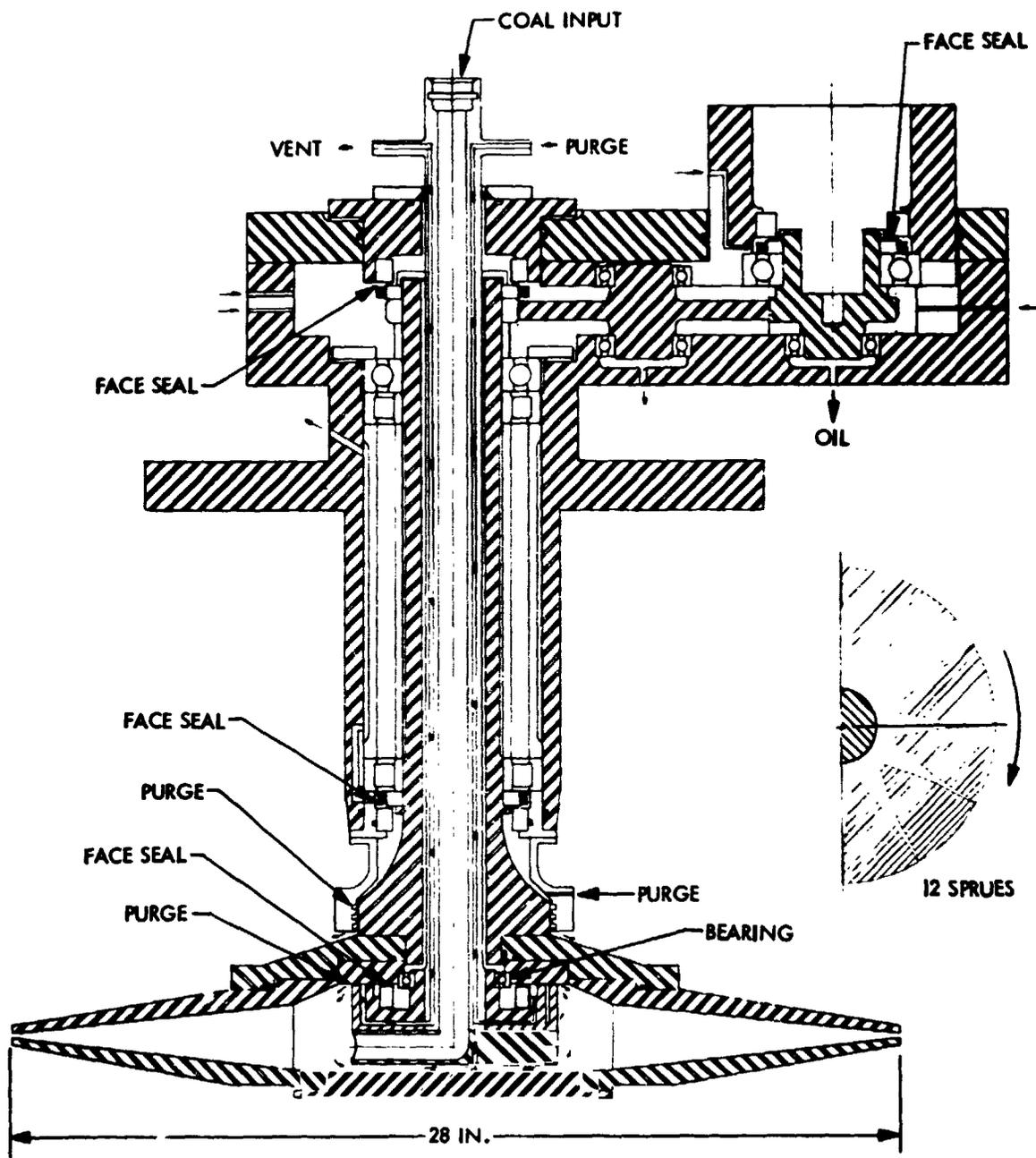
Fig. 15 Kinetic Extruder – Model No. 2 Configuration

well removed from the center line in a radial direction. The sprues are double
tapered to increase the operating pressure range and to keep gas infiltration low.
The rotor was designed to be compatible with the existing gear box/drive shaft
system. Therefore, the bearing and sealing systems are not optimally designed
based on present experience and will need to be reconfigured to increase the
reliability of the system. However, sufficient test data and experience have
been obtained to enable the design of Pilot Plant equipment. Results obtained for
the Kinetic Extruder Model No. 2 are illustrated in Figure 16. The predicted per-
formance is shown for two wheel speeds, the modified sprue configuration, and a
permeability of $6 \times 10^{-12} ft^2$. The data for a number of different wheel speeds
is generally in agreement with the predictions and follows the predicted trends with
speed. Nominally, this wheel would then pump 1 ton per hour into a pressure of 150
psia with 12 sprues and a wheel speed of 3500 RPM.

Based on design studies performed thus far, the kinetic extruder has good
potential for large throughputs of coal grinds up to 1/8 inch in particle size. Finer
coals can be fed at lower throughputs and higher pressures. Multistaging of the
kinetic extruder has been considered. Results indicate that best performance is
achieved in the lower stage. Further trade-offs are required before final recom-
mendations can be made on the potential for hybrid systems which might
incorporate the ejector as the final stage, for example.

Ball Conveyor

The ball conveyor is basically a standpipe filled with descending large metal
balls. Coal is sandwiched in the voids between the balls as they move down the pipe.
The weight of the column overcomes the static pressure, and the downward motion
of the column counterbalances the gas flow up the standpipe. On the return leg of
the standpipe, a liquid lock or gland seal is provided to prevent gas leakage. The
basic elements of a ball conveyor feeder system are shown in Fig. 17. Tests of
the pressure sealing portion of the system - the standpipe containing the ball-coal
column have been completed. Using steel balls, such a feeder can sustain a pressure
differential of 1.6 psi/ft of standpipe.

A computer model was developed based on the percolation of a gas through a
porous coal bed having coupled multiple cavities. The model permits introducing

Fig.16 Kinetic Extruder Model No. 2 Test Data. Inlet pressure 14.7 psia, fine coal, outermost aeration port open, and sprues modified with epoxy filler, sprue exit area 0.006 in^2.

VENT SYSTEM

BALL METERING
SYSTEM

OVERFLOW
RETURN

FLUIDIZED
COAL
INLET

BALL RETURN
ELEVATOR
& DRYER

8 IN. PIPE

PRESSURIZED
COAL STORAGE
VESSEL

FLUID STAND PIPE

BALL/FLUID
SEPARATOR

TO REACTOR

FLUID PUMP

Fig. 17   Ball Conveyor

pressurization gas at any location along the standpipe and also allows for the formation of channels within a loosely packed coal bed. Experimental setups were constructed to verify the predictive capability of the model and to obtain data on coal permeabilities, column mobility, and friction factors. Development tests were designed to answer the following questions:

(1) Can the gas leakage rate be kept at low values?

(2) Can friction forces be kept low?

(3) Will the coal wedge between the balls and the pipe and cause ball hangups?

Two different configurations were used for the ball conveyor simulator. The first model was built around an 8-in. lucite tube so that visual observations were possible. However, this setup was not equipped to make column descent tests against pressure since relatively lightweight balls (bowling balls) were used. Instead, descent tests at zero pressure differential and gas leakage tests with the ball column held in place were performed separately. After favorable results from the first test series, the test rig was reconfigured with heavy steel balls in order to make descent tests against realistic pressure gradients.

● Transparent Tube Configuration Tests

Dynamic and static experiments were performed with the transparent ball conveyor tube. In the dynamic tests, the balls were moved by a hydraulic piston and frictional resistance was determined as a function of coal packing density in the ball column cavities. The static tests consisted of pressure and gas flow rate measurements with stationary balls in order to determine the overall permeability of the column as a function of packing density. The test results were positive in that the ball/coal column still retained its mobility when packed tightly enough to be nearly impermeable to gas flow. The tests also indicated that in order to avoid channeling and the loss of an effective gas seal, the balls forming the column must be slightly separated. This assures that the coal in the cavity between the balls remains tightly packed. If the balls are touching the coal has a tendency to fluidize and the capability of the column to form a gas seal is rapidly lost.

● **Steel Tube Configuration Tests**

In the next series of tests, the experimental apparatus was modified to allow for motion of the ball column against gas pressure. This test rig is illustrated in Fig. 18. Heavy steel balls were used so that the pressure gradient and friction forces were the same as in an actual system. The lucite tube used for visual observation during the initial tests was replaced by a steel tube. Friction was measured by putting a load cell directly under the ball column. These modifications allowed close simulation of conditions in an actual recirculating system. A set of 5-in. steel ball-mill balls was used for the tests. These balls are hot forged, have rough surfaces, and are inexpensive. For example, a typical ball had a mean diameter of 5.096 in. with an rms deviation of 0.022 in. Tubes of 5.250 in. and 5.375 in. ID were used. These tests indicated that the column moves freely and the balls do not lock-up provided the radial clearance is larger than the coal particle size. Under these conditions frictional forces equal about 25 percent of the column weight and were insensitive to the pressure difference across the ball column. Pressure differences of 1.6 psi per foot of column can be maintained with steel balls. Figure 19 summarizes these data.

Design tools and experimental procedures have been developed which permit assessment of the ball conveyor as a potential feeder candidate. The operating regime is shown in Fig. 20. Feed stock particle size distribution and the permeability of the coal are important parameters. With low permeability coal, it is desirable to provide pressurization gas along the standpipe. Several concepts have been considered for coal loading into the column and also for the ball let-down system. These two subsystem functions require development before an all-up ball conveyor system can be designed and built.

Fluid Dynamic Lock

Concept Principle

The fluid dynamic lock (FDL) is shown schematically in Fig. 21. It basically is a centrifugal compressor in which a dense coal-laden gas stream is accelerated outward between two closely spaced rotating disks. Momentum is imparted to the fluid by the disk skin friction. This scheme eliminates severe blade wear problems encountered when conventional radial or axial compressors are used with particle-laden gases.
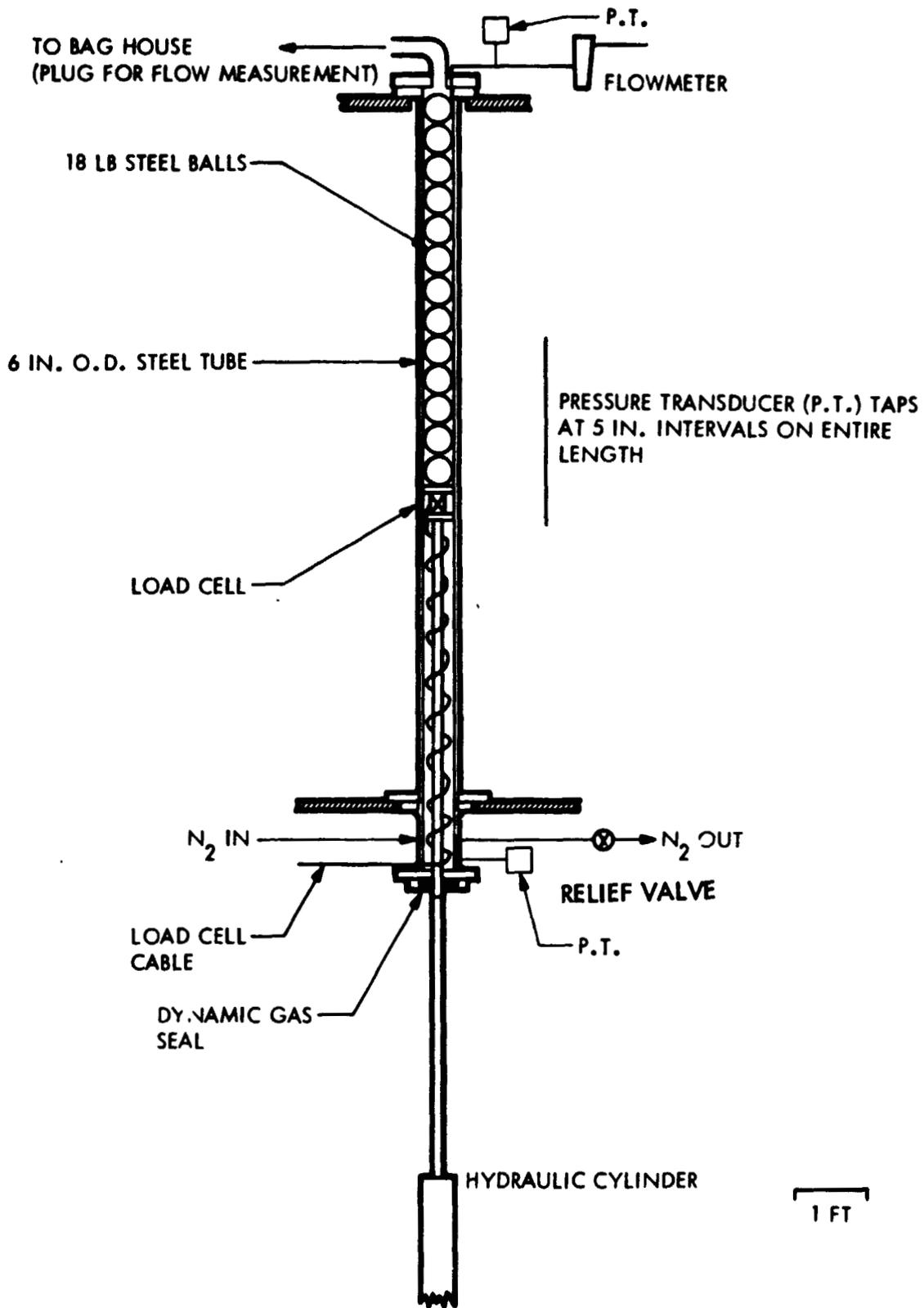
Fig. 18   Ball Conveyor Test Rig
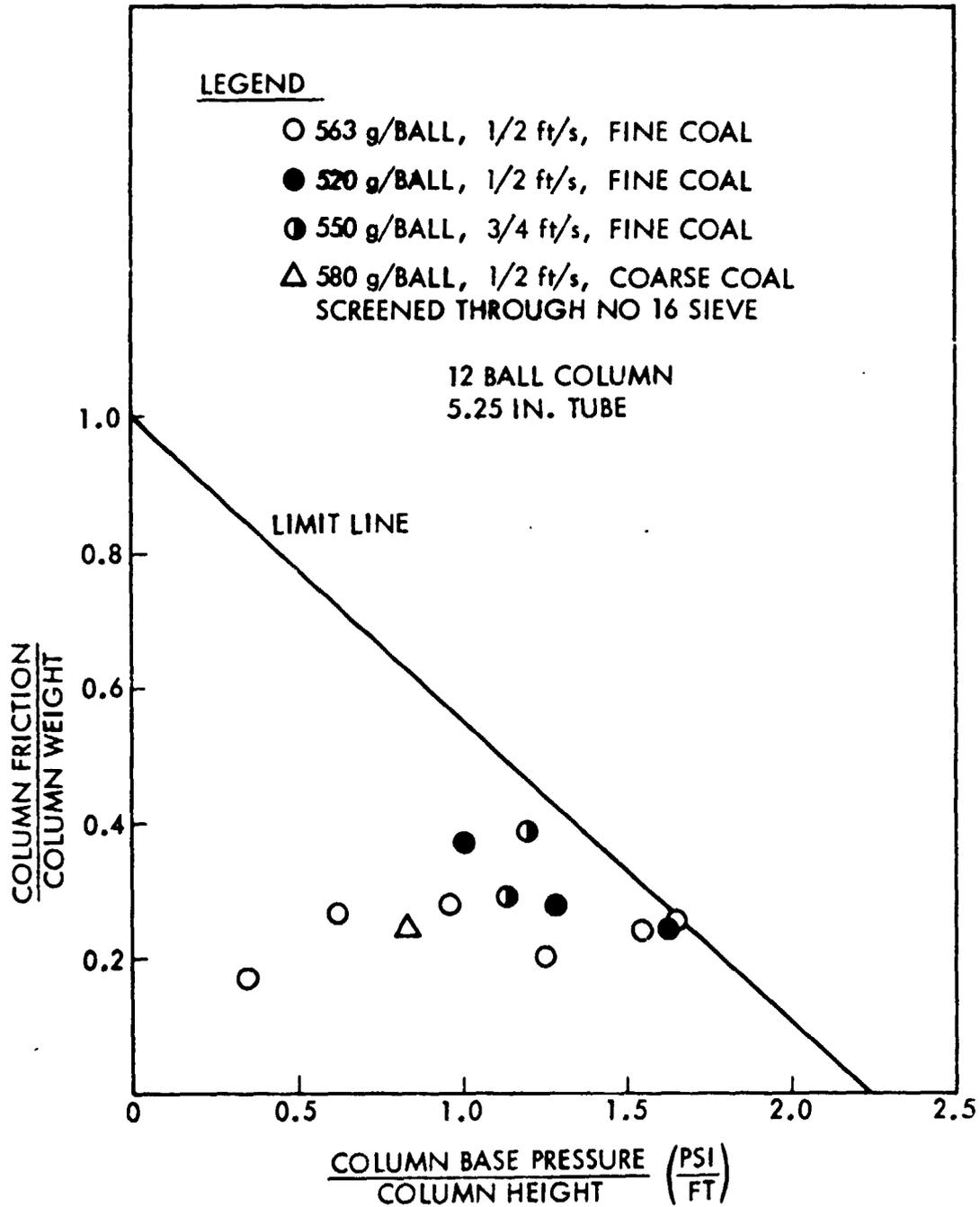
Fig. 19 Ball Conveyor Test Results
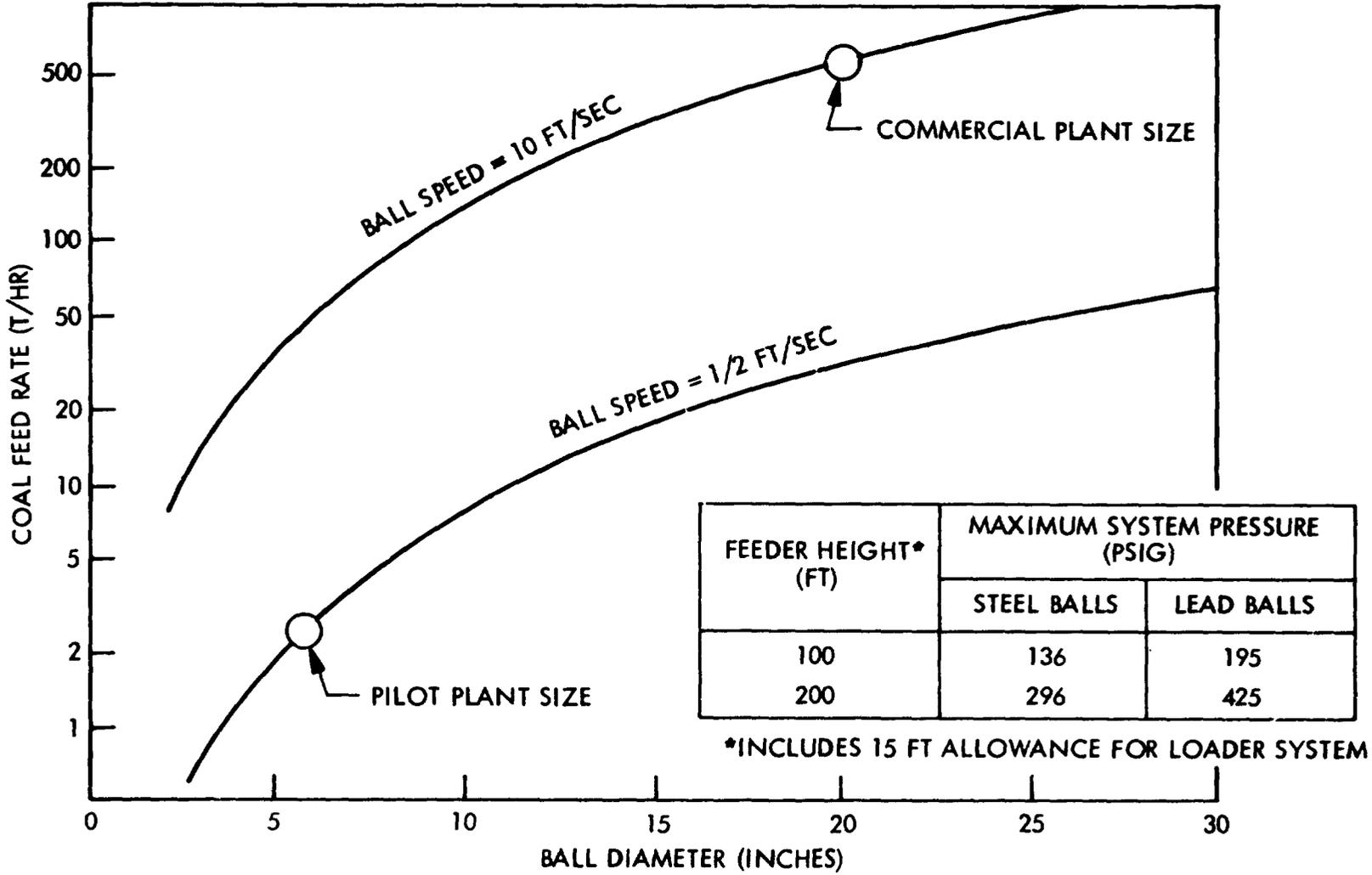
Fig. 20 Ball Conveyor Feed Rates

The figure plots COAL FEED RATE (T/HR) on the vertical axis (logarithmic, from below 1 up past 500) against BALL DIAMETER (INCHES) on the horizontal axis (0 to 30). Two curves are shown: the upper curve labeled "BALL SPEED = 10 FT/SEC" with a point marked "COMMERCIAL PLANT SIZE" near 20 inches / ~500 T/HR, and the lower curve labeled "BALL SPEED = 1/2 FT/SEC" with a point marked "PILOT PLANT SIZE" near 6 inches / ~2 T/HR.

FEEDER HEIGHT* (FT)	MAXIMUM SYSTEM PRESSURE (PSIG)	
	STEEL BALLS	LEAD BALLS
100	136	195
200	296	425

*INCLUDES 15 FT ALLOWANCE FOR LOADER SYSTEM

Fig. 21 Fluid Dynamic Lock — Rotor Assembly

Evaluation of the PDL concept has mainly relied on mathematical modeling. A test rotor has been designed, built, and tested in our feeder test facility to verify the predicted trend. To analyze the disk flow field, a very complete computer model has been generated by Professor Warren Rice of the Arizona State Univeristy under subcontract to Lockheed.

Consideration of multiple disk turbomachinery for various applications requires detailed knowledge of the flow between parallel corotating disks, which is the fundamental element of this bladeless type of turbomachinery. For single-phase laminar flow between corotating disks, numerical solutions of various models of the flow have been made and substantiated experimentally. The results have enabled calculation of predicted performance and the design of multiple disk turbines, pumps, and compressors using single-phase fluids. The calculations show that properly designed multiple disk turbomachines can have efficiency and performance comparable with that of conventional turbomachines. It has been shown that the efficiency of multiple disk turbomachines is higher for laminar than for turbulent flow.

Mathematical modeling of three-dimensional multiphase flows to practically any desired degree of sophistication has been presented in the literature. Modeling of the flow is relatively straightforward for laminar flow of a Newtonian fluid with a sparse population of solid particles, supplied uniformiy around the periphery of parallel corotating disks. The resulting system of equations constituting the modeling has been solved numerically on the computer by Professor Rice. However, there are severe limitations for use of the program in design investigations because long computer run times are needed to compute a single flow case with specified conditions at the flow inlet, and it is necessary to repeat calculations using variable mesh sizes to establish accurate results.

Because of these computational difficulties, a simpler model was developed which yields sufficiently accurate results but at far less expense than is possible using a three-dimensional problem solution program. Furthermore, it allows computation of two-phase turbulent flow between disks which is required to accurately model the flow. The analysis is one dimensional and treats the two-phase fluid in a bulk-parameter manner. This approach has been widely used for calculation of two-phase flows, but without the presence of centrifugal force field. The analysis is useful for both laminar and turbulent flow and for incompressible and compressible primary fluid with solid particles.

The computer program was used to size the test hardware. The performance predictions are shown in Figs. 22 and 23 as a function of the coal lo.....ng and the spacing between the disks.

A fluid dynamic lock was designed and 'uilt which is interchangeable with the kinetic extruder model No. 1. During the test runs, the same mechanical difficulties were encountered as with the kinetic extrud~ . The limited test data, however, indicated that for practical distances between the disks, only pressure ratios far less than the desired value of two were obtainable. Thus many stages are required to deliver coal at elevated pressure.

Design tools have been developed and verified by te:..s which permit the evaluation of the performanc. potential of the fluid dynamic lock in Pilot Plant use. Based on st..dies carried out to date, this device does not appear to be a strong candidate for coal feeding. The need for narrow disk spacing limits application to very fine coal grinds and the limited pressure rise per stage forces the use of many stages, increasing power consumption and equipment cost.

The device should be considered as a recompression unit for recirculating fluidizing gases in fluidized bed reactor. The available design procedures can be used to evaluate the fluid dynamic lock for this type of application.

SUMMARY

The present program has resulted in design procedures which permit confident evaluation of the four feeder systems considered by Lockheed for ..oal conversion plant application. Because of the variety of coal feed stocks, feed rates, and pressure levels being considered and the variety of proposed conversion processes, it is not possible to select one feeder system as superior. A trade-off must be conducted to select the proper candidate for a specific use. For a Pilot Plant of the Synthane type, for instance, the kinetic extruder, possibly in conjunction with a booster ejector final stage, is a leading candidate.
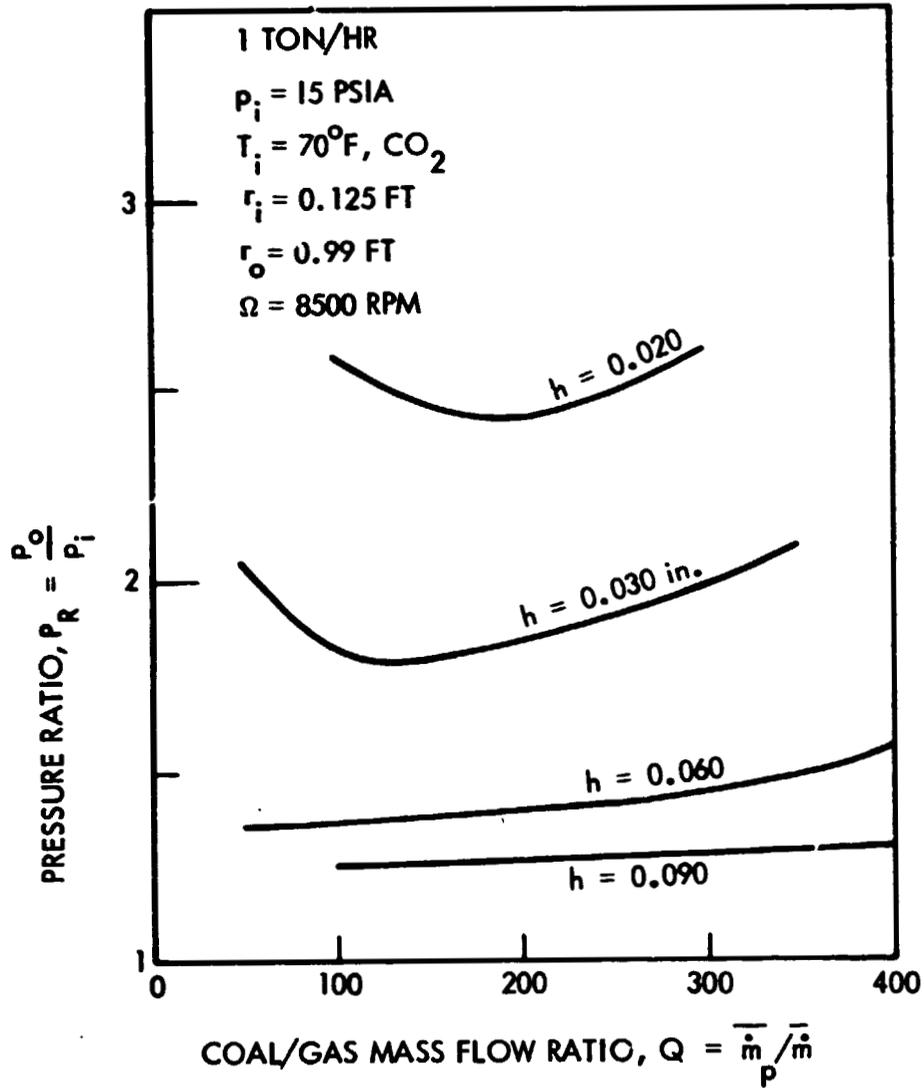
Fig. 22 FDL Performance Predictions. Pressure Ratio As a Function of
Coal/Gas Flow Rate Ratio and Disk Spacing.

1 TON/HR

$p_i = 15$ PSIA

$T_i = 70^\circ$F, CO_2

$r_i = 0.125$ FT

$r_o = 0.99$ FT

$\Omega = 8500$ RPM

$Q = 100$

$Q = 200$

$Q = \overline{\dot{m}}_p / \overline{\dot{m}} = 300$

COAL/GAS MASS
FLOW RATIO

EFFICIENCY PARAMETER

$hp/(ton/hr)/\ln P_R$
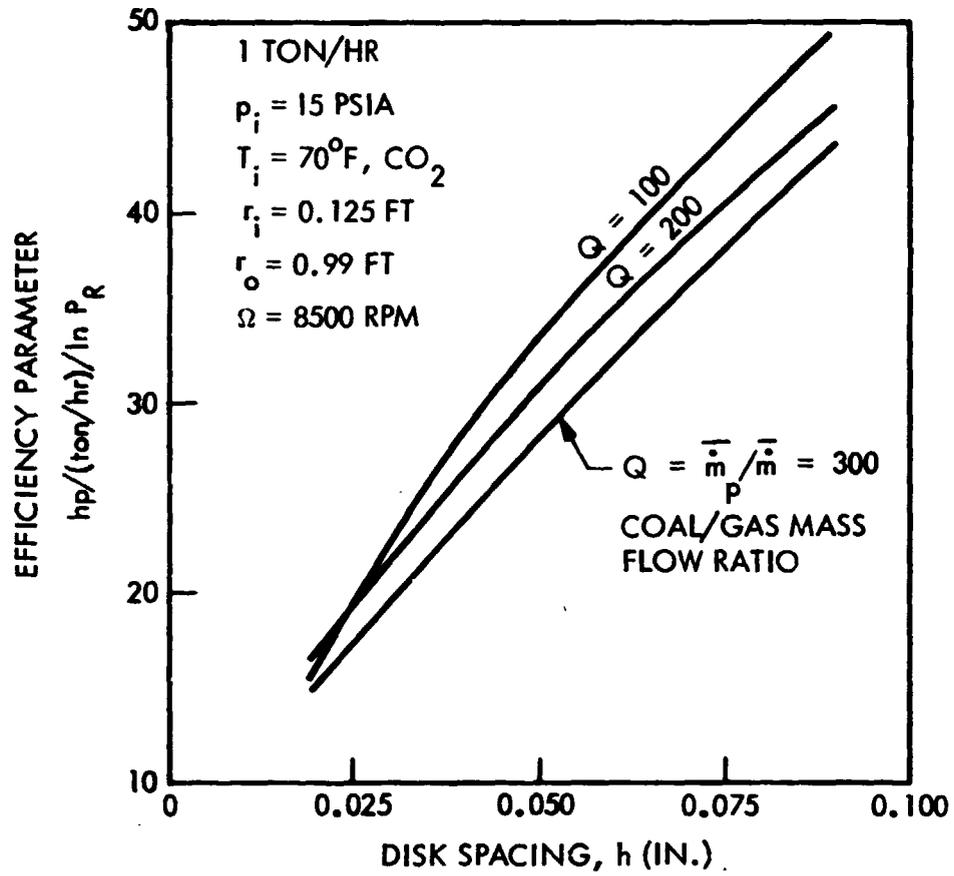
DISK SPACING, h (IN.)

Fig. 23 FDL Performance Predictions. Efficiency Parameter as a Function
of Coal/Gas Flow Rate Ratio and Disk Spacing

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2. "Feeding Coal to Pressurized Systems," E. J. Ferretti, Blaw-Knox Chemical Plants Div., Dravo Corp., <u>Chemical Engr.</u>, Dec. 1974

3. <u>Coal Feeder Development Program, Quarterly Report July - September 1976</u>, Report FE-1792-20, 30 September 1976

DRY COAL FEEDER DEVELOPMENT
PROGRAM AT INGERSOLL-RAND
RESEARCH, INC.

D. K. Mistry

T. N. Chen

Ingersoll-Rand Research, Inc.
Princeton, New Jersey

ABSTRACT

Ingersoll-Rand Research, Incorporated is developing a dry coal screw feeder under contract to the Energy Research and Development Administration for feeding coal into coal gasification reactors operating at pressures up to 1500 psig. The program consists of laboratory development of 1.5" and 5.5" diameter screw feeders followed by field testing of the large feeder at a pilot plant. A description of both feeders, their associated test systems and the test results to date on the small feeder under several different modes of operation are presented. In addition, three new piston feeder concepts and their technical and economical merits are discussed.

INTRODUCTION

With an increasing emphasis being put on the development of coal gasification systems, a number of problems have been identified which demand immediate attention. One of these is the process by which coal is delivered to the pressurized reactor; especially those operating at 300-1500 psig. Although the pilot plants are currently designed to use either a lock hopper or a slurry feed system, both have serious shortcomings and questionable long range adaptability to commercial gasification plants.

The lock hopper is particularly undesirable because it involves a cumbersome installation and requires frequent maintenance of valves. In addition to these, it tends to be expensive and inefficient because the gas must be compressed to the reactor pressure level before coal can be introduced into the reactor.

On the other hand, whereas the slurry feed concept is more economical and advantageous in performance over the lock hopper method, it tends to adversely affect the energy balance because of the thermal inefficiency due to the heat of vaporization of the carrier liquid. The slurry carrier also creates problems in pumping, condensation, subsequent separation and continual makeup.

Recognizing these difficulties in the current feeder technology for coal conversion processes, ERDA has set as one of its goals, the development of an efficient and reliable dry coal feeder. Such a system must be capable of delivering against high back pressures and be economically suitable for future commercial scale gasification plants.

To achieve this objective, the Fossil Energy Division of ERDA awarded a contract to work on a three phase program to Ingersoll-Rand Research, Inc. The now completed first phase established the feeder requirements for coal gasification plants, developed new concepts for delivering coal, and made recommendations about necessary equipment. The three new feeder concepts generated during this phase are briefly discussed in this paper with respect to general concept and operation.

The objective of Phase II which is currently being carried forward, involves the development of 1.5 and 5.5-inch diameter coal screw feeders in the laboratory. These prototypes are establishing performance criteria and information needed for subsequent commercial scale-up. The progress made in this part of the program is the main subject of this paper. The knowledge gained during the Phase II will improve success of the Phase III field testing of the 5.5-inch screw feeder at an existing pilot plant.

COAL SCREW FEEDER OPERATION

A general schematic of the coal screw feeder is shown in Figure 1. The feeder consists of a feed hopper, a screw, a barrel with heaters, a nozzle and a screw drive and injection system. The screw rotates in a tightly fitted barrel. The sized coal (-8 mesh) is gravity fed to the screw from the hopper. The coal particles are conveyed along the barrel and at the same time compacted to provide the necessary gas seal against back leakage. This pumping action is the result of a force balance between the coal and barrel frictional force, coal and screw frictional force and the pressure gradient in the coal along the screw.

The coal is subsequently discharged into the high pressure vessel by an extrusion or injection mode. In addition, the feeder can be operated with and without external heating of the coal through the barrel. To date, coal has been successfully pumped in the following modes: (1) Extrusion mode, with external heat and (2) Injection mode, without external heat.

During the extrusion mode, with external heat operation, coal is heated in the barrel to its semi-plasticized condition with electric heaters. As the coal particles move forward, they are compacted and agglomerated, forming a cylindrical plug in the nozzle, at the discharge end of the screw. These agglomerated coal particles provide a seal against elevated gas back pressure. The vapors generated from the heated coal are vented through an opening in the barrel near the intake area.
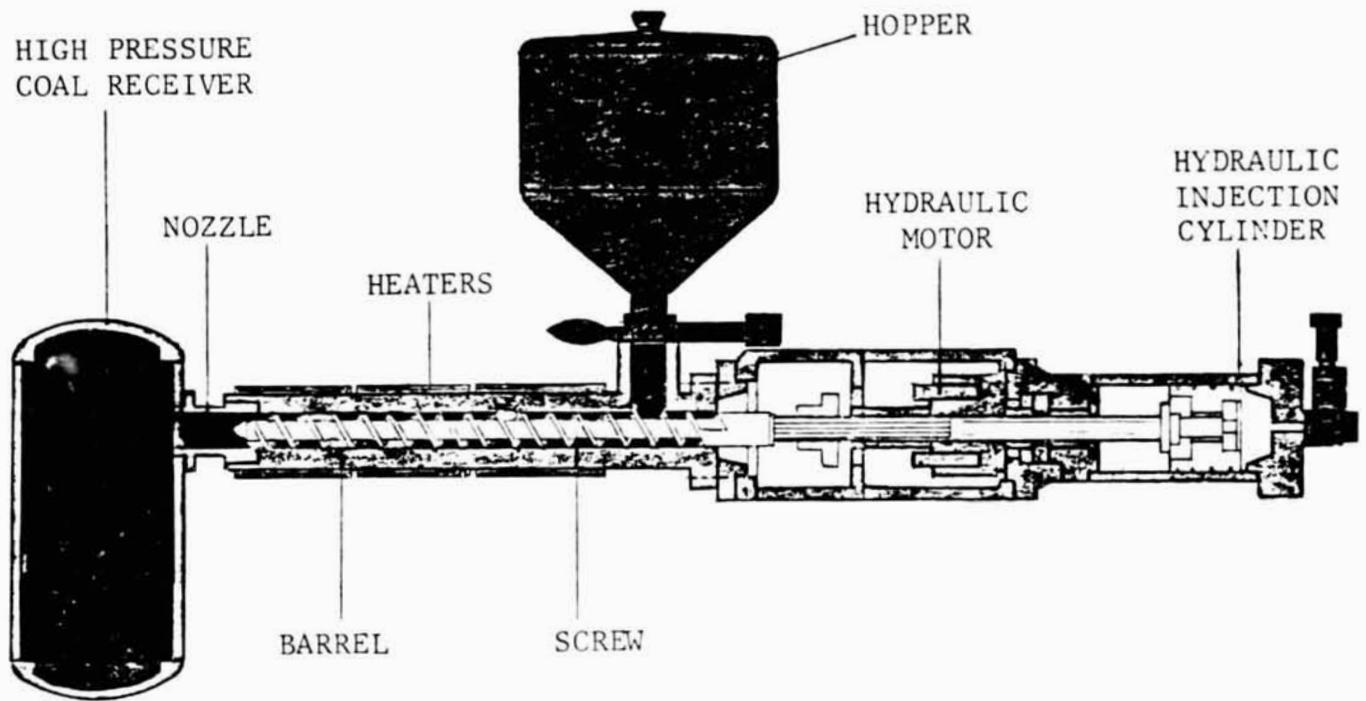
COAL SCREW FEEDER - GENERAL SCHEMATIC

Figure 1

In the case of injection mode without external heat, the coal particles are compacted as they move forward in the barrel forming a compacted plug in the nozzle at the discharge end of the screw. The axial thrust force on the screw causes the screw to travel backwards. The screw rotation is then stopped and the coal plug is injected by ramming the screw forward. The compaction of the particles due to both screw rotation and ramming action, provides the sealing against elevated gas back pressure.

MAJOR AREAS OF INVESTIGATION AND TECHNICAL APPRCACH

In order to successfully develop a coal screw feeder, several areas must be carefully investigated during the laboratory and pilot plant testing. The information acquired from this work will be used in developing a reliable scale-up method which will be the basis for designing large feeders for commercial plant service. The major areas requiring investigation relate to the characteristics of the coal as well as the design and operating characteristics of the machine. Of particular importance are the physical and chemical properties of the selected coal type; e.g. size, moisture content, friction coefficient, viscosity, packing coking, volatile matter etc. Important feeder design and operating parameters are screw geometry, screw and barrel wear, coal output, power consumption etc. In addition, the sealing capability of the feeder should be investigated up

to 1500 psig gas back pressure. In view of the practical
difficulties in developing a large commercial size coal
feeder, the present approach is to develop two sizes of
screw feeders of significantly different coal delivery
capacity. A small feeder, 1½" diameter screw, was selected
for the initial testing effort because of its manageable
size, flexibility, ease of rapid change of design and oper-
ating parameters and capability for expediting testing.
This feeder is being used to experimentally study the effect
of the design and operating variables on the performance.

The second and larger feeder, 5½" diameter screw, now
reaching development testing status will be particularly
used to study the size effect and help establish the design
scale-up method necessary for designing a feeder for a
commercial scale gasification plant. In addition, pre-
pilot plant testing will be carried out in the laboratory
prior to its installation at a pilot plant for field testing.

Theoretical analysis is being carried out in parallel
with development testing and is being used to guide experi-
mentation and to develop a scale-up method.

1½" DIAMETER SCREW FEEDER

DESCRIPTION OF EQUIPMENT

The 1½ inch diameter screw feeder used in the first series of tests is a standard Negri-Bossi, V-12 model injection molding machine and was purchased from Ingersoll-Rand's IMPCO Division. As shown in Figure 2, the machine has a main control box, coal storage hopper, a barrel containing the rotating screw, a coal receiver, a hydraulic drive system and is powered by a 20 hp electrical motor.

The surfaces of the barrel and the rotating screw are nitrided to minimize potential wear from the coal. The barrel is designed to withstand 20,000 psig pressure. In addition, the barrel is equipped with five electrical heater bands which have a total heat capacity of 6 Kw and are automatically controlled according to the temperature settings at three locations along the barrel. The operating time of each heater is recorded to give the heat input.

The screw is driven by a swash plate hydraulic motor mounted co-axially with the screw. The rotational speed can be ried from 0 to 225 rpm with a maximum torque of 400 ft-lb.

At the discharge end of the barrel is a straight cylindrical nozzle, and a specially designed, high pressure coal extrudate receiver mounted on the clamp. When the
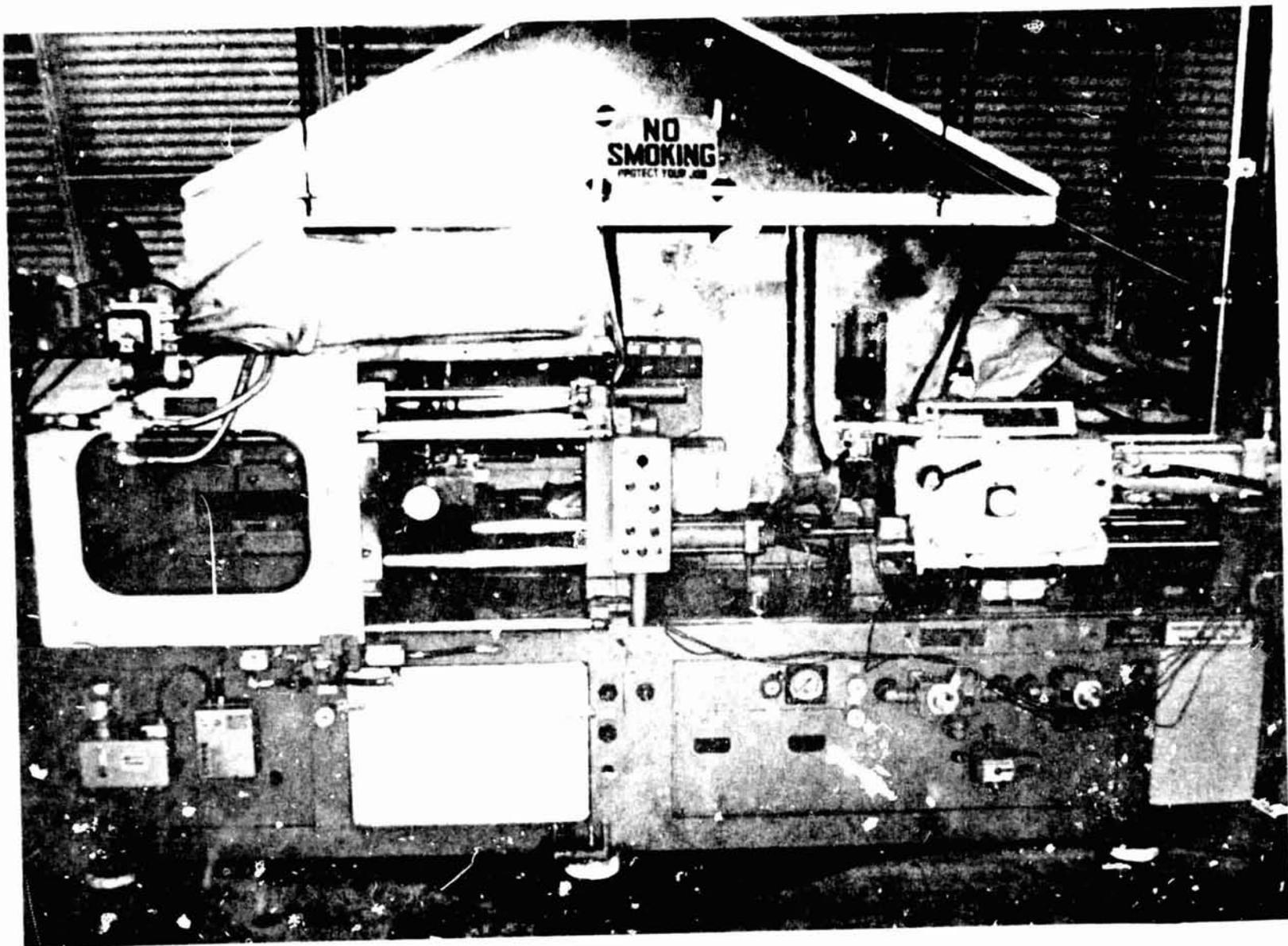
Figure 2: 1 1/2" DIAMETER SCREW FEEDER

clamp is in the open position, coal is extruded against atmospheric condition and can be observed continuously during operation. On the other hand, in order to extrude against a gas back pressure, the clamp is closed. The high pressure coal receiver engages with the nozzle and the receiver is then pressurized with nitrogen to the desired pressure level.

TESTING

The 1½" diameter screw feeder has been tested successfully in pumping a variety of coals against elevated gas back pressures.

The coals tested have included samples from the Pittsburgh seam, the Pittsburgh #8 seam and the Illinois #6 seam. In some cases, the coal was used as received, in others it was dried and screened. In all cases, the coal particles were sized to less than 8 mesh; while in some tests 30 mesh was set as a minimum and in others as received fines were included. Gas back pressure was varied up to 1500 psig and the rotational speed of the screw was also a variable.

The testing was done in two modes; extrusion, with external heat and injection, and without external heat. The testing in these modes has b. extensive and critical to the continuing development of the dry coal screw feeder.

EXTRUSION MODE WITH EXTERNAL HEAT

The extrusion mode using external heat, initially pre-
sented some operational difficulties because of the volatile
vapors generated from the heating of the coal. These vapors
tended to form pockets within the coal moving along the
barrel. Periodically the compressed vapors discharged with
a loud puff; shooting hot coal from the barrel's discharge
end. Additionally, some of the volatile vapors tended to
escape back to the feed hopper, condense on the incoming
coal, and cause bridging problems in the intake area.
Proper venting of the vapors through a hole in the barrel
near the intake area was found to be effective in minimizing
these difficulties.

Temperature distribution along the barrel is one of the
important operating parameters which affects the compaction
of the coal. When coal became highly compacted in the screw,
excessive frictional torque sometimes caused the screw to
stall. On the other hand, insufficient compaction may have
resulted in a loose coal plug that could not seal against the
back pressure. Nevertheless, a steady state operation was
achieved by controlling the temperature along the barrel.

Four screw configurations as shown in Figure 3 were
evaluated for performance. The standard thermoset screw,
#1 in Figure 3, which came with the 1½" diameter machine
was initially tested for performance. It was found that this

DISCHARGE   INTAKE

1. STANDARD THERMOSET SCREW

2. INCREASING VOLUME SCREW

3. THERMOSET WITH SMALLER CHANNEL DEPTH

4. THERMOSET WITH SMALLER PITCH LENGTH

* - NO. OF TURNS
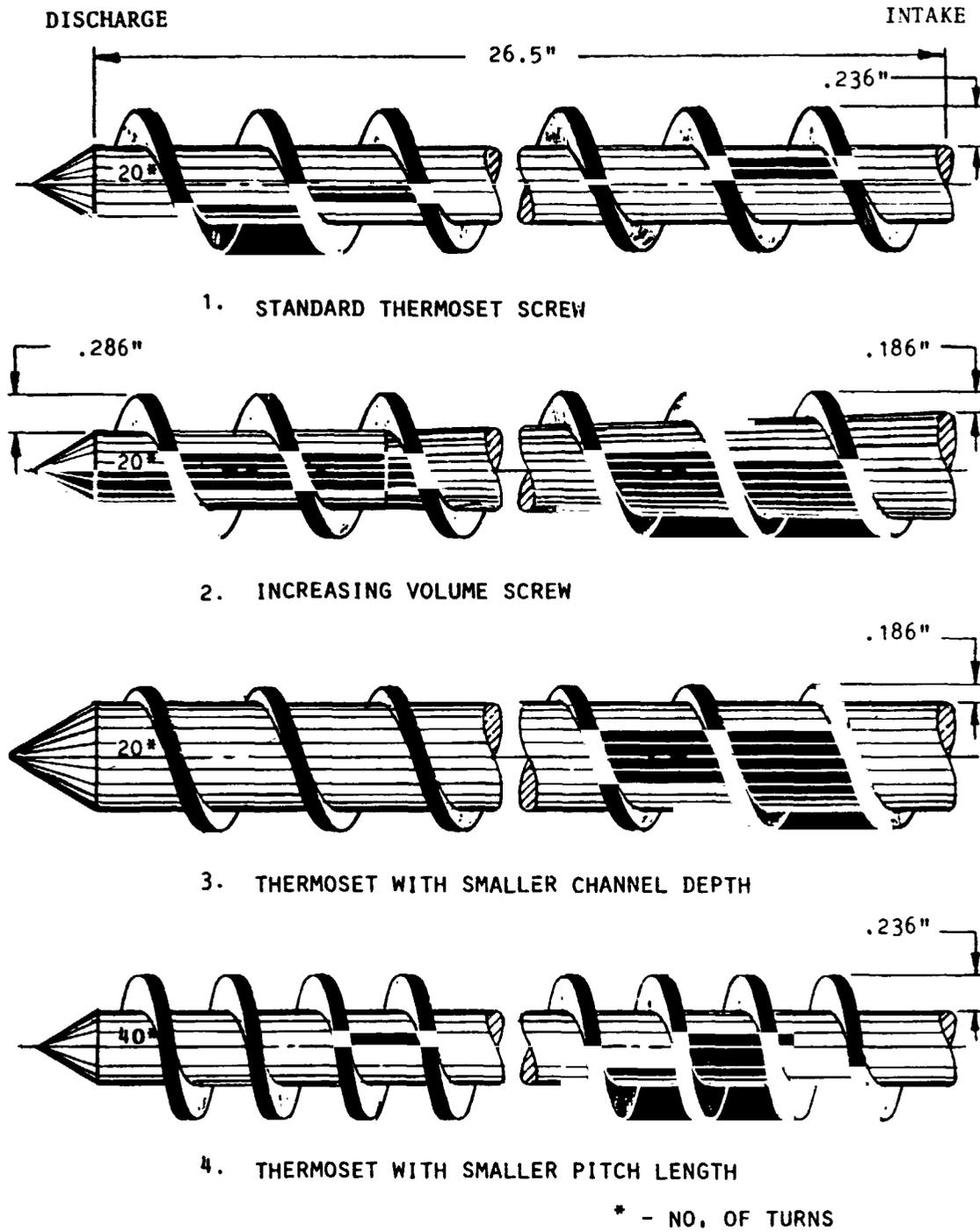
SCREW CONFIGURATIONS
1 1/2" DIAMETER SCREW FEEDER

Figure 3

screw was extremely sensitive to barrel temperature such that a high barrel temperature resulted in poor quality extrudate and low setting caused screw stalling. Only by shortening the screw and reducing the rotation speed to 10 rpm could a plug be extruded in a steady state operation. The energy required was found to be excessive and the coal feed rate lower than desirable. No other testing was subsequently done with this screw.

Performance of the thermoset screw with smaller channel depth, #3 in Figure 3, and that of the thermoset screw with smaller pitch length, #4 in Figure 3, was also not satisfactory. Although steady state operation was possible with the smaller channel depth screw, the output was somewhat lower and the specific power consumption higher than achieved with the increasing volume screw (#2 in Figure 3). With the smaller pitch screw (#4) problems developed because of the higher rotational speeds needed for operation. Steady state operation was never truly achieved.

The most acceptable screw configuration is the increasing volume screw, #2 in Figure 3. Not only was steady state operation possible, but the output and specific power consumption were most favorable. Tests were carried out at 20 and 30 rpm and pressures from 0 to 900 psig. The results of these latter tests are discussed below.

Coal Output

The steady state operation tests with increasing volume screw were carried out with Illinois #6 coal. As indicated in Figure 4, which relates coal output to gas back pressure at screw rotational speeds of 20 and 30 rpm, coal output is somewhat independent of gas back pressure and is approximately proportional to rotational speed.

Power Consumption

It is evident from Figure 5 that mechanical power increases with gas back pressure and the performance improves as the screw rotational speed is decreased. It should be noted that the power here includes neither the frictional power of the machine nor the electrical power used to heat the coal.

The electrical heat input to the coal was determined by recording the total time the heaters were turned on. The total energy input during the steady state operations with the Pittsburgh #8 seam coal is shown in Figure 6, where mechanical, electrical and total specific power consumption are plotted versus back pressure. The graph clearly indicates that only a minor portion of the total power is consumed mechanically in driving the screw; however, a large amount of heat energy is used to heat the coal and evaporate the r isture in the input coal. It should be pointed out that as far as the total process energy balance is concerned this energy input to heat the coal does not represent a

COAL OUTPUT VERSUS GAS BACK PRESSURE

Figure 4

SPECIFIC MECHANICAL POWER CONSUMPTION
VERSUS GAS BACK PRESSURE

Figure 5

1½" DIAMETER SCREW
EXTRUSION MODE
WITH EXTERNAL HEAT
PITTSBURG SEAM #8 COAL
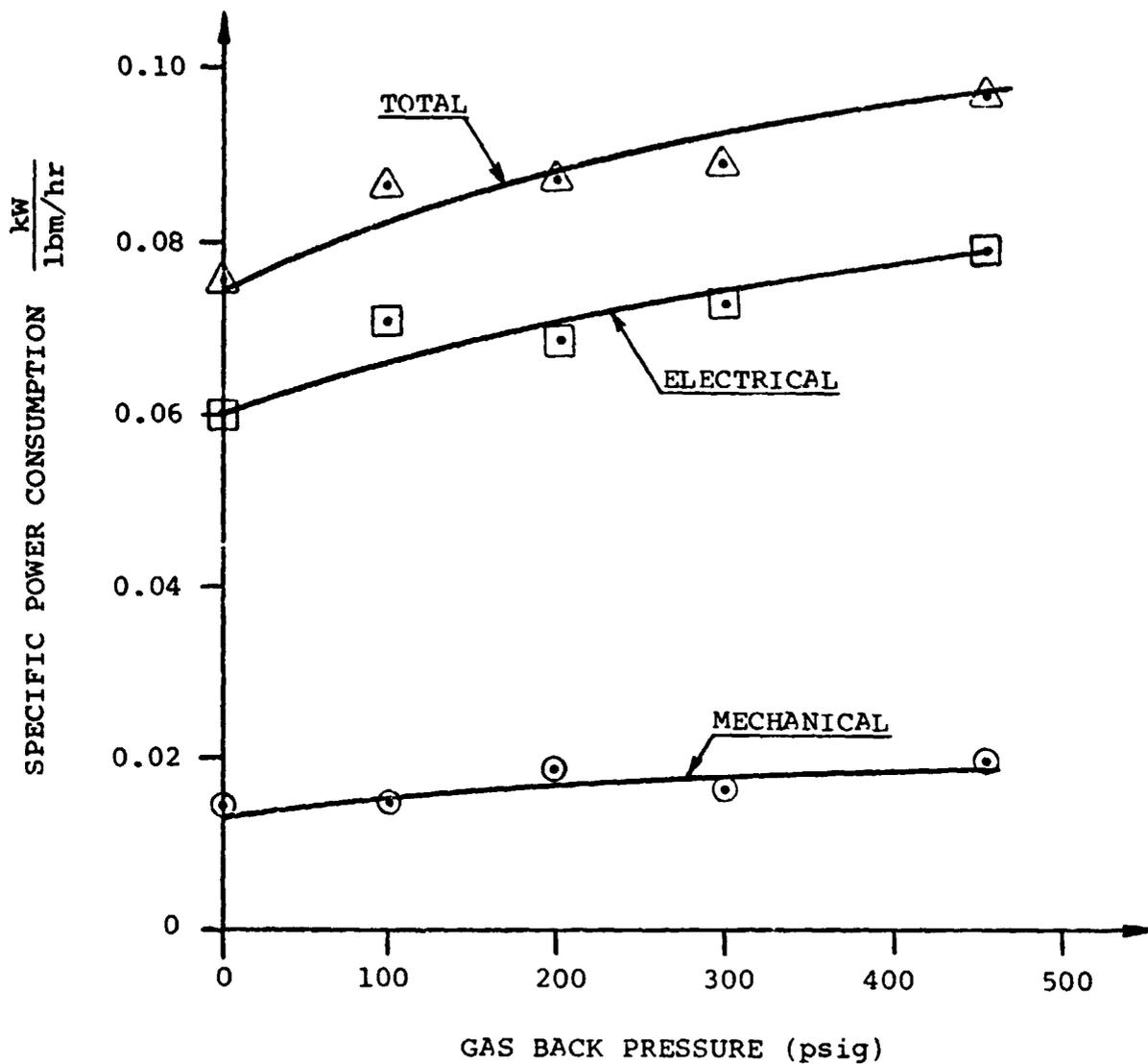
SPECIFIC POWER CONSUMPTION VERSUS GAS BACK PRESSURE

Figure 6

thermal inefficiency. On the contrary, this may be viewed as a preheat process combined with the feeding. In addition, since some volatiles are driven off from the coal, it offers possibility of converting agglomerating coal to non-agglomerating coal which is desirable for certain processes.

Moisture Content

The testing to date indicates that the moisture content of the incoming coal significantly affects the feeder performance with respect to power consumption and vapor generation. The higher moisture content requires more heat for evaporation and the vapors generated is undesirable from an operational standpoint. For efficient operation in the extrusion mode, the input coal should have a low moisture content.

INJECTION MODE WITHOUT EXTERNAL HEAT

As outlined earlier, the injection mode involves the discharge of a compacted coal plug through the ramming action of the screw. Study of this operation was initially done using a standard thermoset screw with a length to diameter (L/D) ratio of 20. Preliminary tests indicated the importance of the L/D ratio with respect to the torque, power and tendency of the screw to stall. Because steady state operation could not be achieved with the L/D ratio of 20, the feeder was redesigned with a new barrel and a constant channel depth screw having an L/D ratio of approximately 6. Although

with this modification feeder operation was improved, the continuous increase of barrel temperature due to heat addition from internal friction made the steady state operation impossible. It became apparent that the barrel temperature must be controlled and a water jacket was designed around the barrel. Steady state operation was achieved after this modification.

Coal Output

A cycle in the injection mode includes all the operations from filling the screw and forming the coal plug to injecting the plug into the receiver. Although the screw can be operated at higher rotational speeds to improve filling time of the cycle, the limiting factor for the cycle is the injection portion. It was found that if the number of cycles per unit time is increased beyond a certain level, the coal will not be sufficiently compacted to provide an effective seal against gas back pressure. An upper limit of coal feed rate, therefore, may exist. The coal output versus back pressure is shown in Figure 7 for 217 and 143 cycles/hour. This test, run on the Illinois #6 seam coal, clearly shows that output is independent of gas back pressure and is directly proportional to the cycle rate.

Power Consumption

The plot of the specific power consumption versus gas back pressure is shown in Figure 8. Whereas specific power

1½" DIAMETER SCREW
INJECTION MODE
WITHOUT EXTERNAL HEAT
ILLINOIS #6 COAL

COAL OUTPUT VERSUS GAS BACK PRESSURE

Figure 7
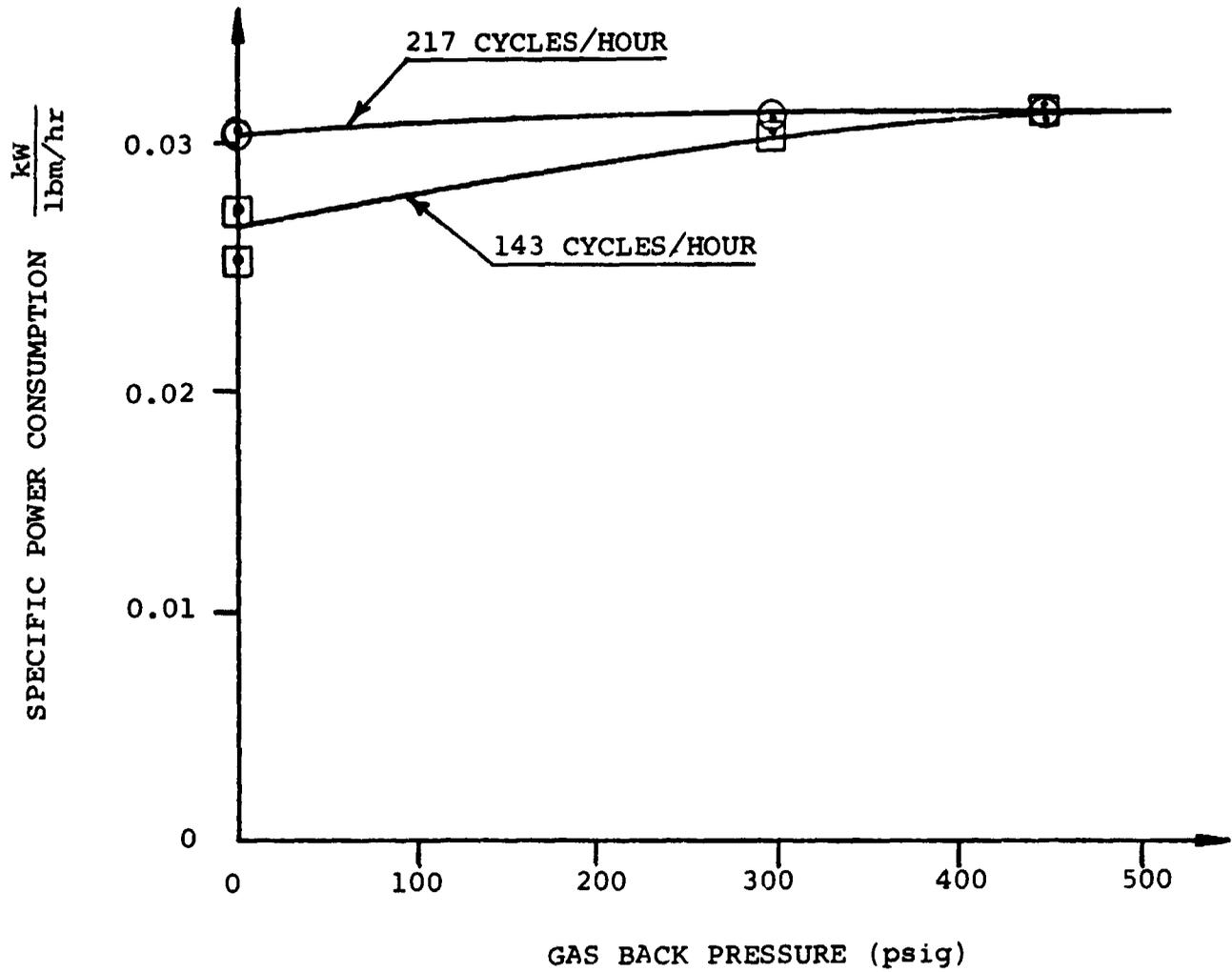
SPECIFIC POWER CONSUMPTION VERSUS
GAS BACK PRESSURE

Figure 8

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increases with higher gas pressure in the case of the 143 cycle/hour tests, there is no significant increase observable for the 217 cycle/hour rate. It is believed that the variation in the moisture content of the input coals used during this series of tests was the cause for the difference in the general behavior.

Effect of Moisture

Tests were also carried out to determine the effect of input coal (-8 mesh) moisture content on the feeder performance. The data is presented in Figure 9 which compares the results of 3.0% and 7.5% moisture in the input coal. It is clear that the higher moisture coal (7.5%) requires a specific power consumption of approximately 33% less than for a lower (3.0%) moisture coal. Other tests using fine coal (-30 mesh) indicate the same general trend, but for slightly different moisture contents: 5.8% and 10.7%.

EXTRUDATE CHARACTERISTICS

Figure 10 shows the typical extrudate from the 1½" diameter feeder operating in the extrusion mode with external heat and the injection mode without heat. The external heating with extrusion mode tends to form plasticized skin surrounding a compacted and slightly agglomerated core (see A in Figure 10).

1½" DIAMETER SCREW
INJECTION MODE
WITHOUT EXTERNAL HEAT
ILLINOIS #6 COAL
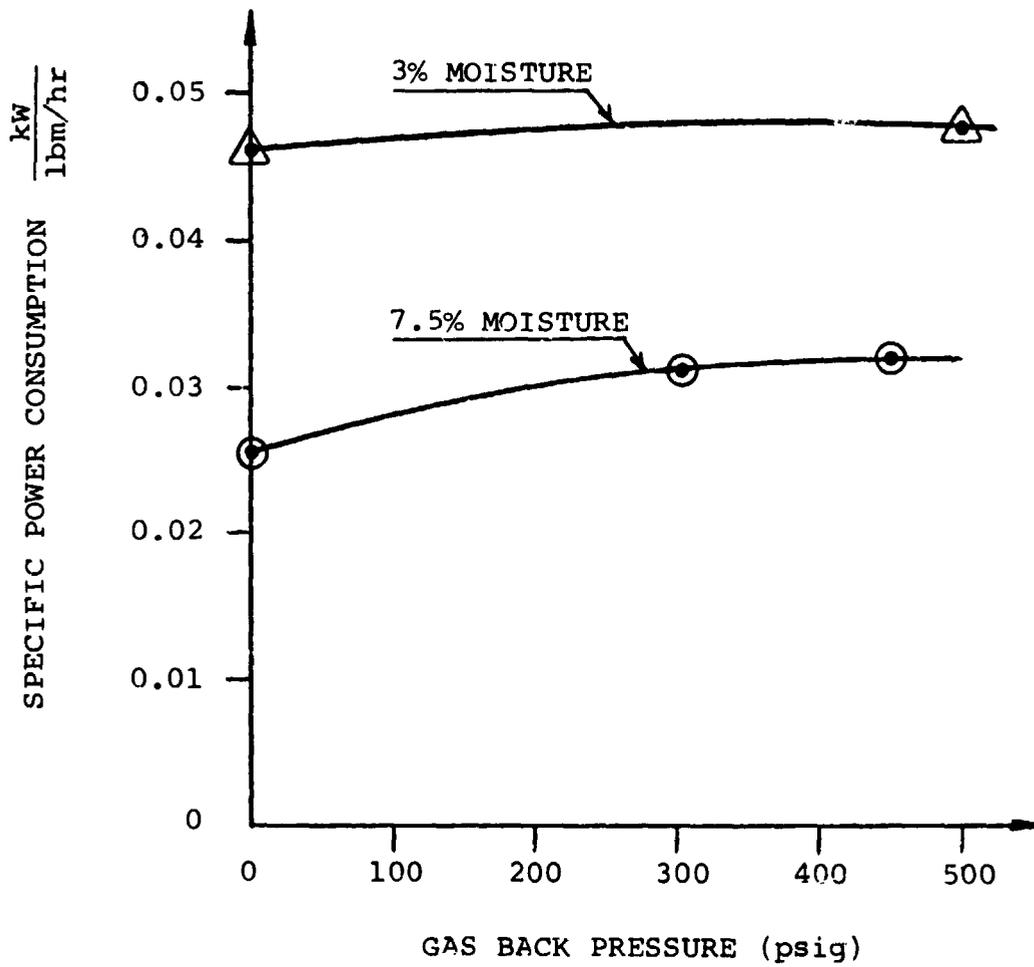
SPECIFIC POWER CONSUMPTION VERSUS
GAS BACK PRESSURE

Figure 9
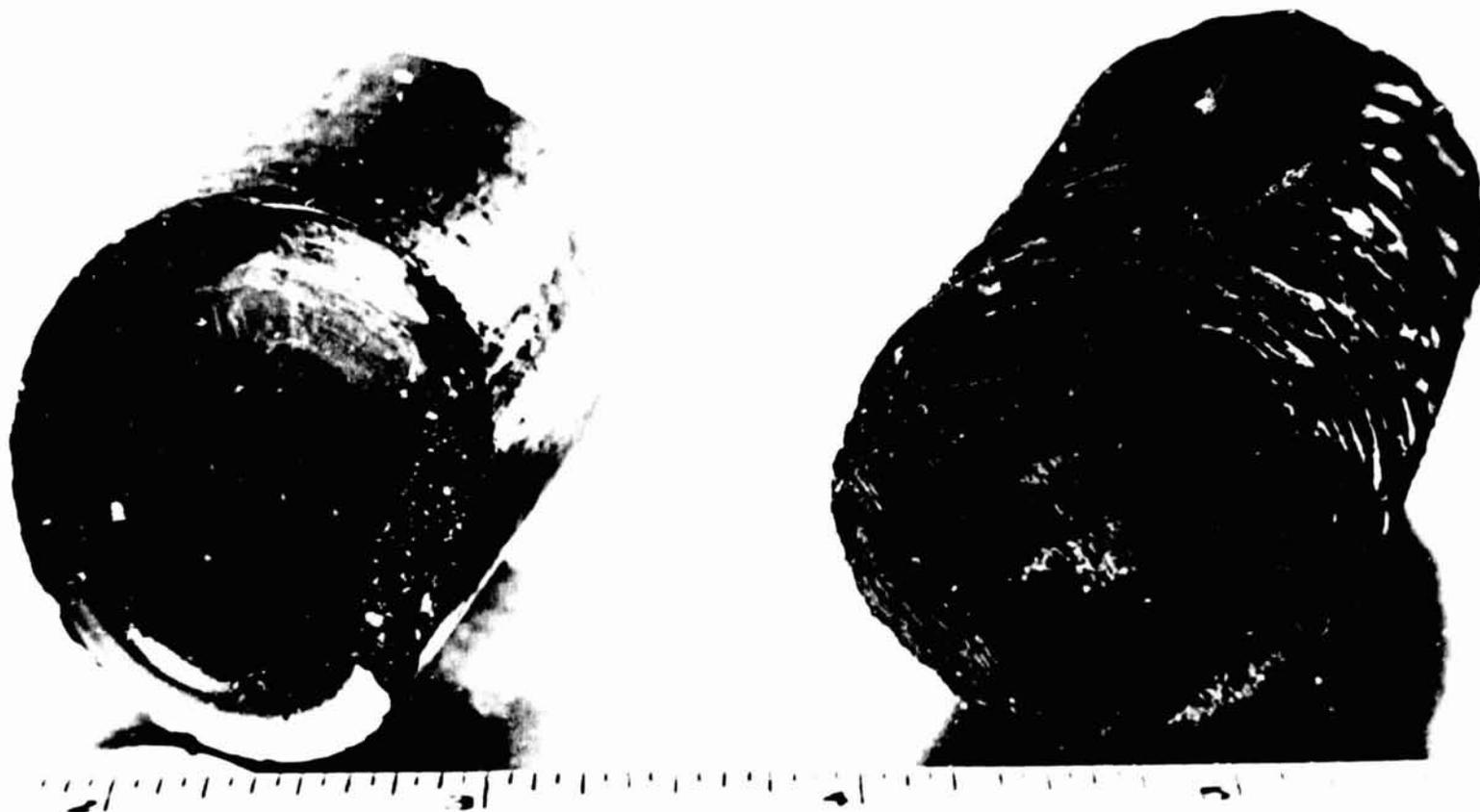
Figure 10: TYPICAL EXTRUDATE

In contrast, the extrudate plug (see B in Figure 10) from the injection mode, which did not involve external heating, tends to be glazed on the outside cylindrical surface and is of a highly compacted, granular structure in cross section.

Tests on the extrudates are being currently carried out at the Coal Research facility at Pennsylvania State University to determine the Proximate and Ultimate Analysis, the Free Swelling Index and the Hardgrove Grindability Index. The samples will also be subjected to a $1000^{\circ}F$, 300 psig pressure environment to determine their ability to retain their integrity.

CONCLUSIONS

Several conclusions have been drawn from the development testing to date with the $1\frac{1}{2}$" diameter screw feeder.

General

Various types and sizes of coal can be successfully pumped on a continuous basis to elevated gas back pressure with screw feeding. The sealing capability can be achieved either by taking advantage of coal plasticizing properties (extrusion mode with external heat) or by compacting coal particles (injection mode without external heat). The screw geometry, coal properties, input coal moisture content etc.

all have significant effect on the feeder performance. Coal output is essentially independent of back pressure and is nearly proportional to screw speed or cycle rate.

Extrusion Mode with External Heat

- Coal properties with respect to temperature, pressure and time are extremely important to proper operation of the screw feeder.

- Proper barrel venting is required to maintain consistent coal flow.

- Temperature distribution along the barrel is important.

- Input coal moisture content strongly affects performance and must be kept to a minimum.

- Increasing volume screw configuration with length to diameter ratio of 20 yields best performance.

Injection Mode without External Heat

- Increase in input coal moisture content significantly reduces power consumption.

- The screw length to diameter ratio should be approximately 6 for successful operation.

- Barrel temperature should be maintained below $200^{\circ}F$.

5½ INCH DIAMETER SCREW FEEDER

DESCRIPTION OF EQUIPMENT

Concurrently with the testing program of the 1½ inch diameter screw feeder, work has been progressing with the installation and preparations for tests with the 5½ inch diameter feeder. It is a specially modified, standard IMPCO 1500 injection molding machine fabricated by the IMPCO Division of the Ingersoll-Rand Company (Figure 11).

To meet the requirements necessary to feed coal at elevated gas back pressure, the machine has been extensively redesigned in the electrical and hydraulic systems. The screw flights are surface hardened with colmonoy and the surfaces are protected from corrosion with flash chrome plate. The Xaloy liner is shrunk into the barrel to protect the surface from wear. The barrel is heated in a similar manner as the 1½" diameter screw feeder with total heat input capability of 70 kw. The screw is internally heated with a specially designed 20 kw cartridge heater. The screw is driven by a vane-type hydraulic motor mounted coaxially with the screw. The screw speed can be varied from 0 - 60 rpm. The total rated feeder power is 200 hp.

The schematic of the feeder test system is shown in Figure 12. Two major system components are a high pressure coal receiver (vessel) and an extrudate break-up device. The system is designed for a coal feed rate of 1.0 ton/hour
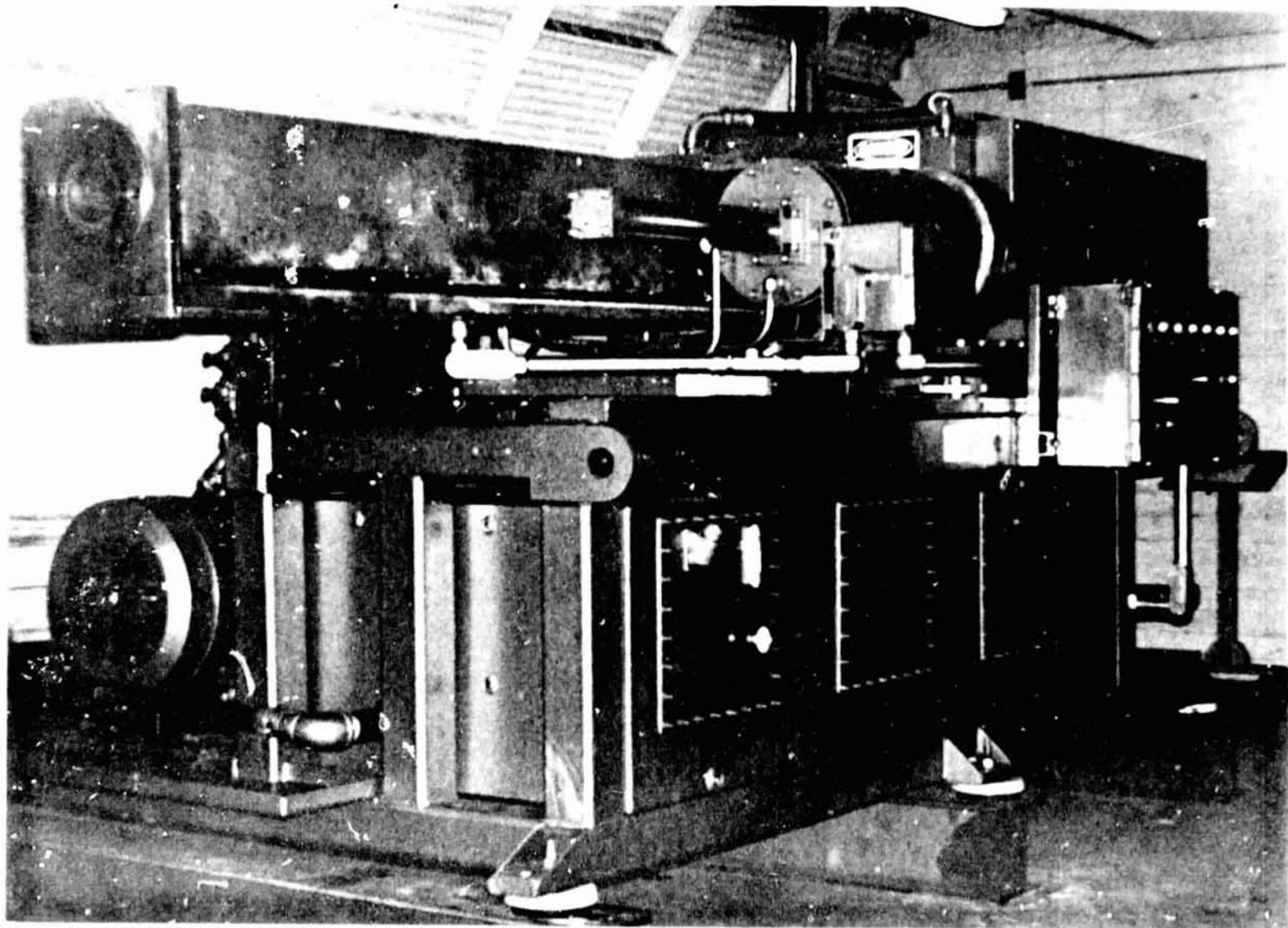
Figure 11: 5 1/2" DIAMETER SCREW FEEDER

COAL PLUG BREAKER

NITROGEN BLANKET ON WATER

NITROGEN FEED

WATER TANK

WATER PUMP

HIGH PRESSURE COAL RECEIVER

GAS VENT TO ATMOSPHERE

TO EXHAUST FAN

VESSEL SCREW CONVEYOR

ENCLOSURE-HOOD

BARREL HEATER

5 1/2" DIA SCREW

CARTRIDGE SCREW HEATER

BARREL

ATMOSPHERIC COAL EXTRUDATE CONTAINER

HOPPER

COAL FEED CONVEYOR

COAL FEED BIN

5 1/2" DIA SCREW COAL FEEDER
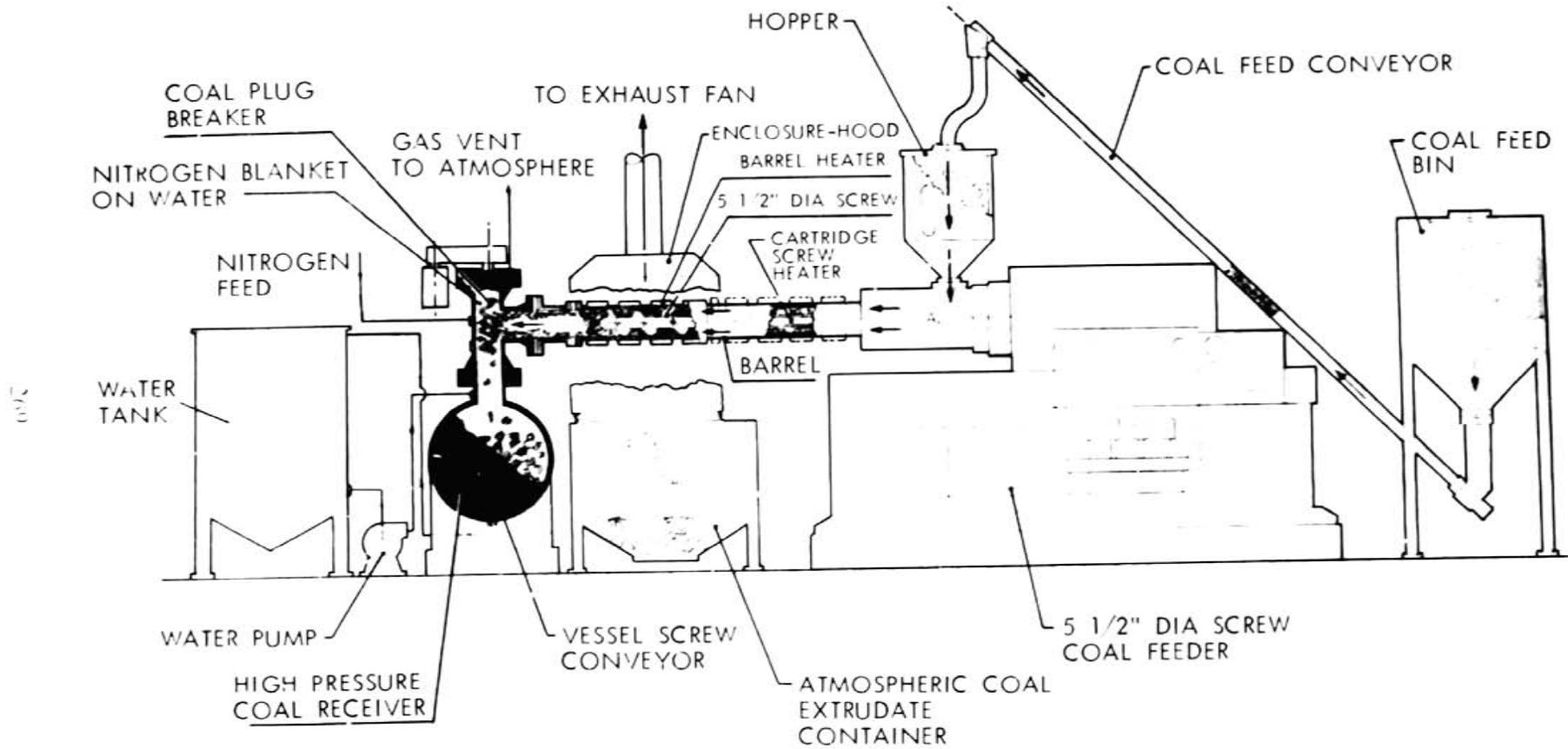
TEST SYSTEM SCHEMATIC
FOR 5 1/2" DIA SCREW COAL FEEDER

Figure 12

with the capability for operation up to 5 tons/hour and back
pressures of 1500 psig. The extruded coal from the feeder is
conveyed to the high pressure coal receiver through a trans-
ition piece and the coal breaker. The breaker has a continu-
ous, helical flight type cutter with sharp cutting edges and
is mounted on a vertical shaft which can be rotated at varying
speeds. The coal plug is fragmented by the breaker into small
pieces (1" - 2") so that it can be readily removed from the
vessel.

In operation, the high pressure coal receiver is filled
with water and pressurized to the desired level with nitro-
gen gas in the space between the feeder and the receiver.
The water is used to minimize the volume of high pressure
gas for safety and economy. Moreover, the water level is
maintained constant within the vessel as the coal
extrudate fills the vessel. When the vessel is filled to
the desired level with coal extrudate, the remaining water
is drained and the gas in the vessel is vented to atmosphere.
The coal is discharged from the vessel to the outside of
the laboratory with a screw conveyor located at the bottom
of the vessel.

The feeder and its test system are shown in Figure 13 as
installed in the laboratory.
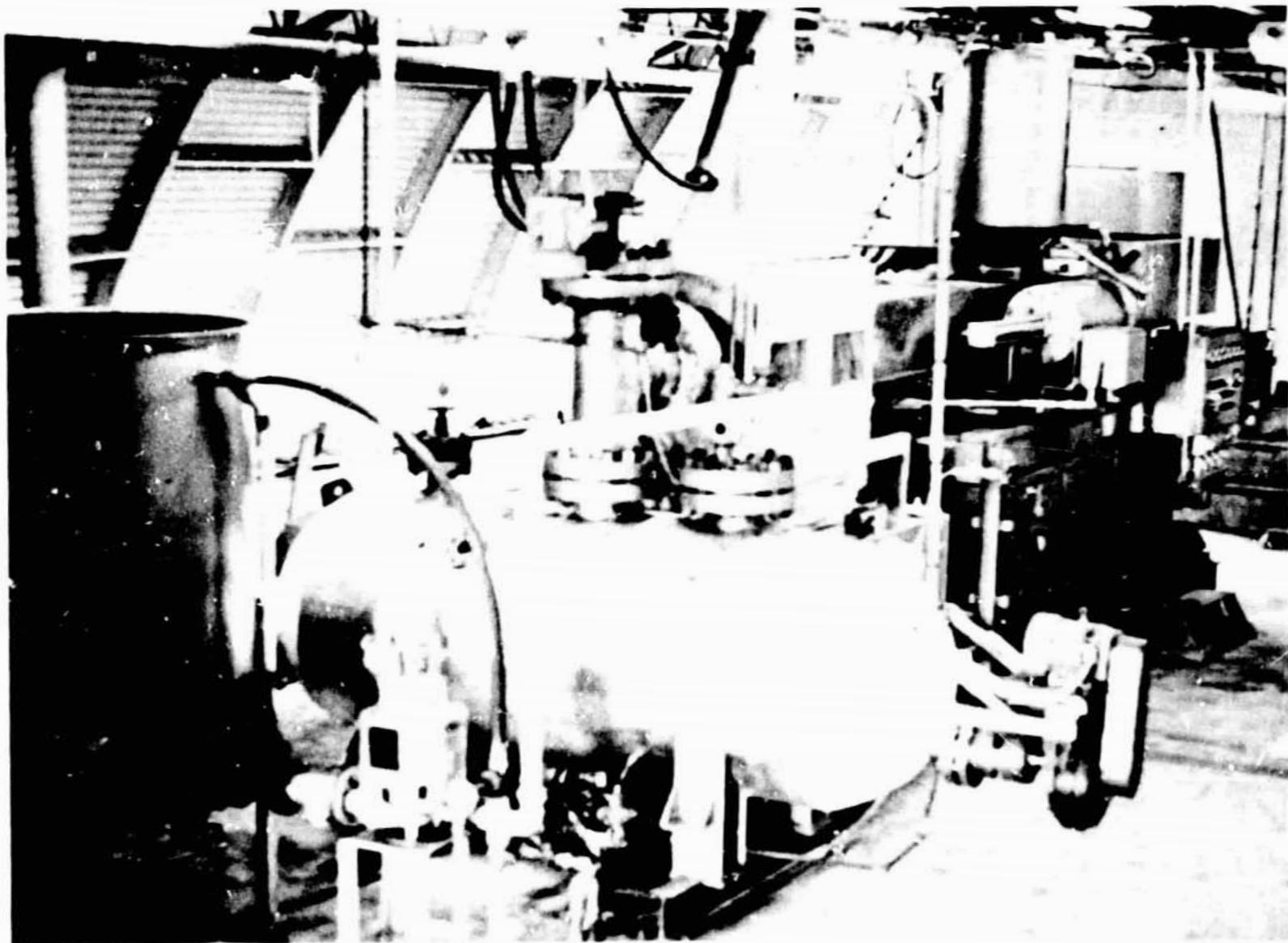
Figure 13: 5 1/2" DIAMETER SCREW
FEEDER AND TEST SYSTEM

PRESENT STATUS AND FUTURE PLAN

Testing with the 5½" diameter screw feeder, which began in March 1977, has been limited thus far largely to the "shaking down" of the feeder's subsystems. These tests have already provided valuable information for certain design alterations to the feeder and system which should greatly improve overall operation.

In addition, the feeder has been operated against atmospheric pressure in the extrusion mode (with and without external heat) and injection mode (without external heat). A coal plug has been successfully formed and extruded against atmospheric conditions (see Figure 14).

It is expected that a full development series of tests will be carried out with this feeder over the next several months. These tests will measure the coal output, power consumption and extrudate quality at elevated gas back pressure levels to simulate anticipated field conditions. The performance in different modes of operation will be compared. In addition, pre-pilot plant tests will be carried out in the laboratory. Following these tests, the feeder will be installed at a pilot plant for field testing. The experimental, stirred bed reactor at ERDA's Morgantown Energy Research Center appears at this time to be the most suitable installation for the pilot plant demonstration.

Figure 14:   EXTRUDATE ISSUING FROM 5 1/2" DIAMETER SCREW
FEEDER

PERFORMANCE IMPROVEMENT

A mathematical model was developed and a parameter
study was carried out to relate the feeder performance to
the design and operating parameters. The purpose of the
study was to obtain an in depth understanding of the
feeding process. Analytical results have indicated that,
for a given screw geometry, reducing the friction between
the coal and the screw is the most effective way
to increase the pumping rate and decrease the power
consumption. Figures 15 and 16 show that reducing the screw
friction coefficient from .32 to .2 almost doubled the coal
output rate and decreased the specific power by 50%.
Accordingly, methods for providing surface lubrication for
the screw are being investigated. The heating of the 5½"
diameter screw also tends to decrease the friction between
the screw and coal.

1½" DIAMETER SCREW
SCREW SPEED - 30 RPM
f_b - 0.32
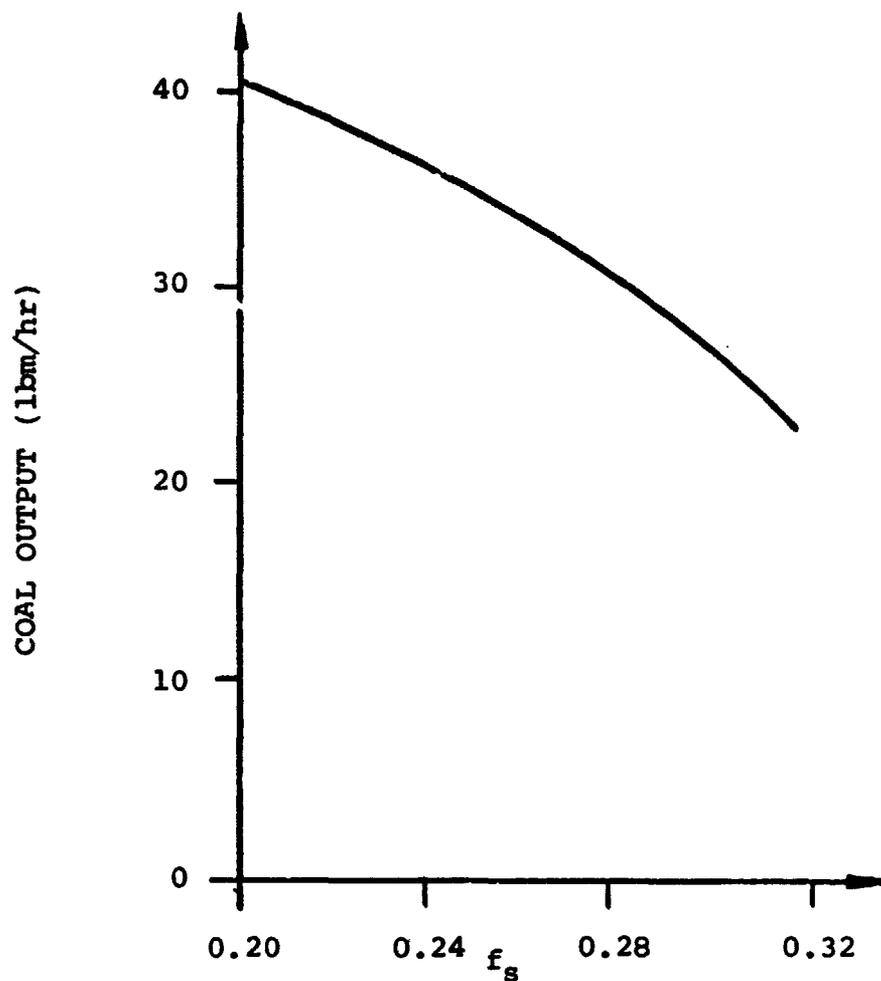
COAL OUTPUT VERSUS FRICTION COEFFICIENT BETWEEN
COAL AND SCREW (f_s)

Figure 15
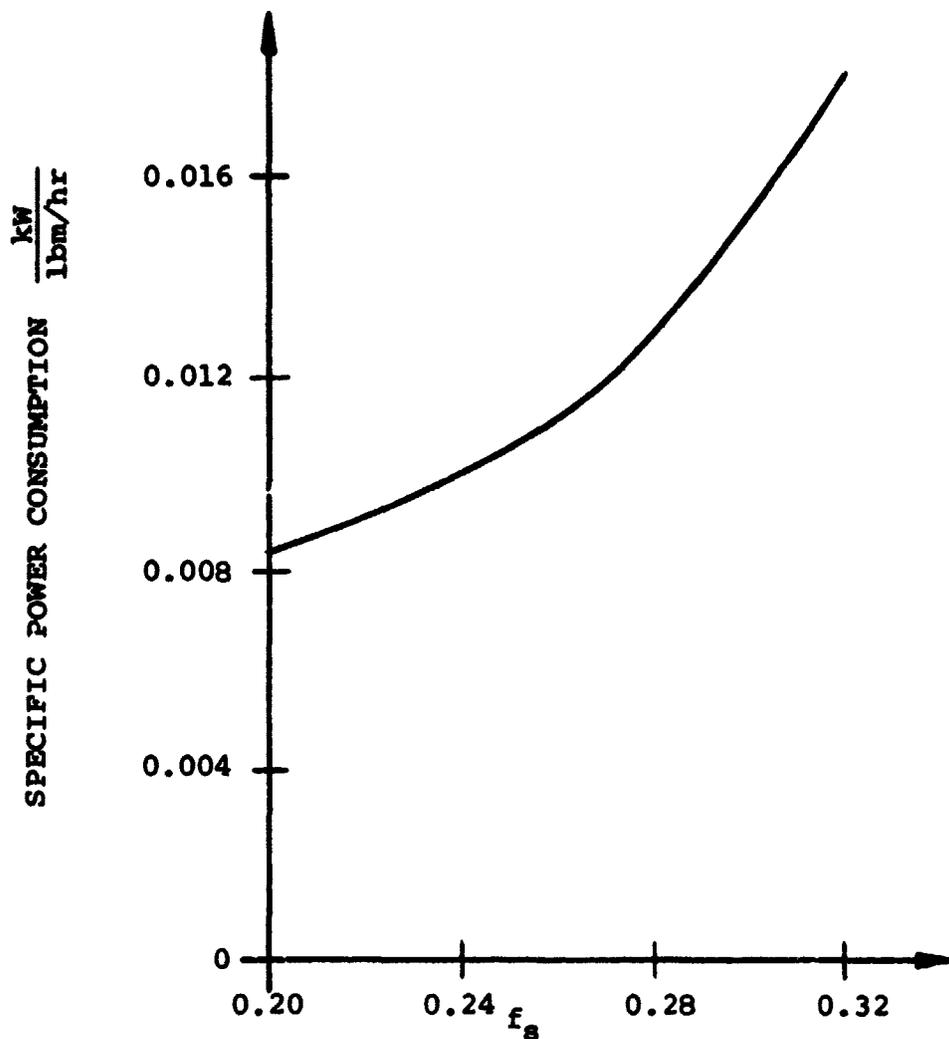
SPECIFIC POWER CONSUMPTION VERSUS FRICTION
COEFFICIENT BETWEEN COAL AND SCREW (f_s)

Figure 16

OTHER FEEDER CONCEPTS DEVELOPED

As a part of the Phase I study some effort was directed to the development of new feeder concepts. Three new concepts were generated and are briefly presented here for completeness.

Classified as piston feeders, each uses gravity to charge and unload the coal. However, they differ in their design and operation. A description of each concept is included below.

Piston Feeder - Single Acting

The single acting piston feeder utilizes two co-axial delivery pistons which are in turn actuated by drive pistons activated by a pneumatic or hydraulic power supply. The sequence of events in a working cycle is as follows: load, advance, delivery, volume reduction, retreat and open for reloading.

As can be seen in Figure 17, there are six steps in this single acting, piston feeder cycle. In the first, the coal from the feed hopper fills the cavity between pistons 1 and 2. This is followed by the simultaneous advance of both pistons which first closes the inlet port and then brings the cavity to the delivery position. In step 4, piston 1 is driven to engage piston 2 and complete the volume reduction. This displaces the pressurized gas back

FEED HOPPER — INLET PORT

STEP 1 LOAD

DISCHARGE PORT — PISTON NO. 1 — PISTON NO. 2

STEP 2&3 ADVANCE & DELIVERY

STEP 4 VOLUME REDUCTION

STEP 5 RETREAT

STEP 6 OPEN & LOAD
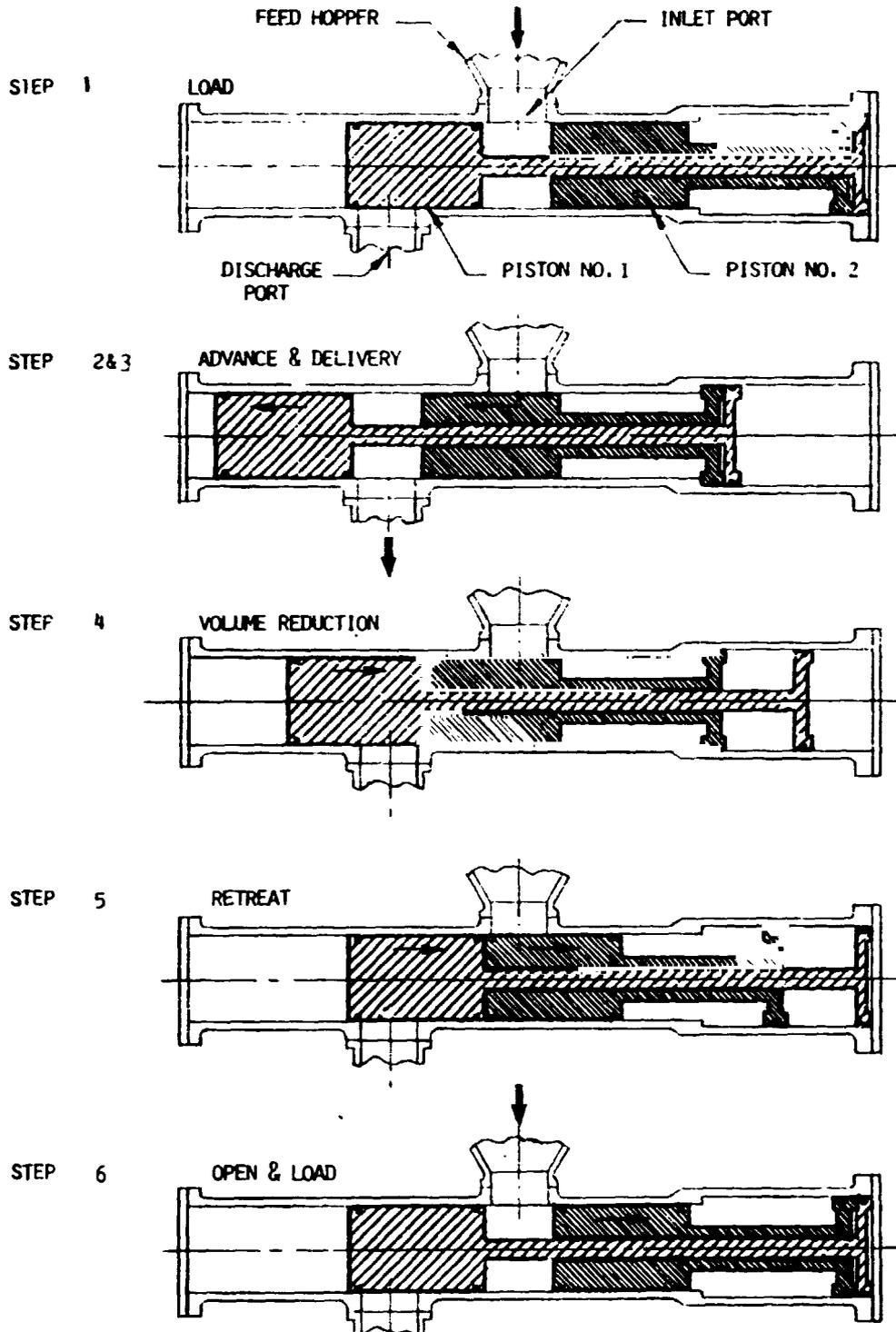
OPERATING SEQUENCE
PISTON FEEDER, SINGLE ACTING
Figure 17

into the system. In the fifth step both pistons translate back, restoring piston 1 to its original position. Finally, piston 2 is returned to its original position and the cycle starts again.

Piston Feeder - Double Acting

In the second piston feeder concept, the cylinder body contains two pistons at a fixed distance from each other joined by a common shaft. As shown in Figure 18, a floating piston lies inbetween, which, on its positioning in the cycle allows for alternate loading from the two feed hoppers. Again a power supply motivates and sequences the cycle. A summary of the operation is: open and load cavity A, advance and delivery, vclume reduction and load cavity B, advance in the opposite direction and delivery.

This cycle is broken into the following steps as shown in Figure 18. Beginning with the charging of cavity A, the second step shows the floating piston and the fixed piston - shaft assembly advancing to expcse cavity A in step 3. Following this action, the floating piston is driven to the left and this returns pressurized gas back into the system and allows cavity B to be filled. In the next two operations, the fixed and floating pistons are both advanced to the left and bring cavity B to the discharge part. The cycle is ready again when the floating piston returns to the right side, and cavity A opens.

FEED HOPPER (TYP)

PISTON-SHAFT ASSEMBLY

CAVITY A

STEP 1 OPEN & LOAD

FLOATING PISTON

DISCHARGE PORT

STEP 2&3 ADVANCE & DELIVERY

STEP 4 VOLUME REDUCTION & LOAD

CAVITY B

STEP 5&6 RETREAT & HOLD (FOR DELIVERY)
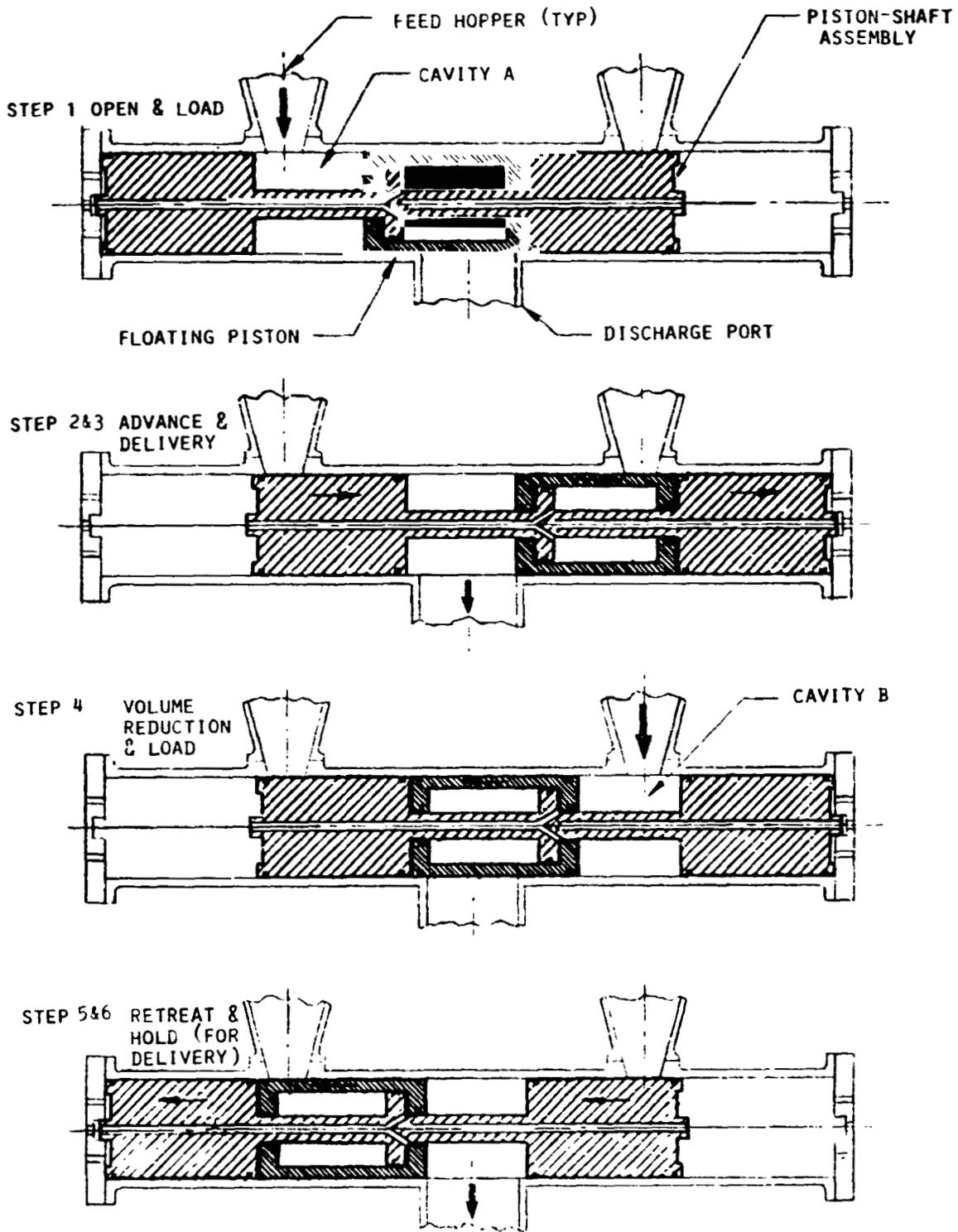
OPERATING SEQUENCE
PISTON FEEDER, DOUBLE ACTING

Figure 18

Rotary Valve Piston Feeder

The third piston feeder, shown in Figure 19, involves a single piston located within a rotating inner cylinder which is perforated on one side at its far end. This turns within the main housing which has an inlet hopper on top and a coal discharge port on the bottom. The process is sequenced and powered as in the other concepts. The delivery cycle involves the following sequence of events: open and load, rotate, delivery, volume reduction, rotate and retreat.

As Figure 19 indicates, step 1 involves the filling of the chamber with coal from the storage hopper. The cylinder then is rotated by a rotary actuator to a position where its opening corresponds to the discharge port. This action delivers the coal. The piston simultaneously moves forward to complete the volume reduction step. The hollow cylinder is then rotated back to the inlet port followed by piston withdrawal within the inner cylinder. The feeder is then ready to receive a new charge of coal.

Advantages of New Piston Feeders

An overall comparison was made of the piston feeder concepts against lockhoppers, slurry pumps and screw feeders to establish their relative merit. From this comparison, it became apparent that the piston feeder approach as a class offers significant benefits to developers of coal conversion systems.

STEP 1 — OPEN & LOAD
HOLLOW CYLINDER — FEED HOPPER — INLET PORT
PISTON — DISCHAPGE PORT

STEP 2&3 — ROTATE & DELIVER

STEP 4 — VOLUME REDUCTION

STEP 5 — ROTATE & RETREAT
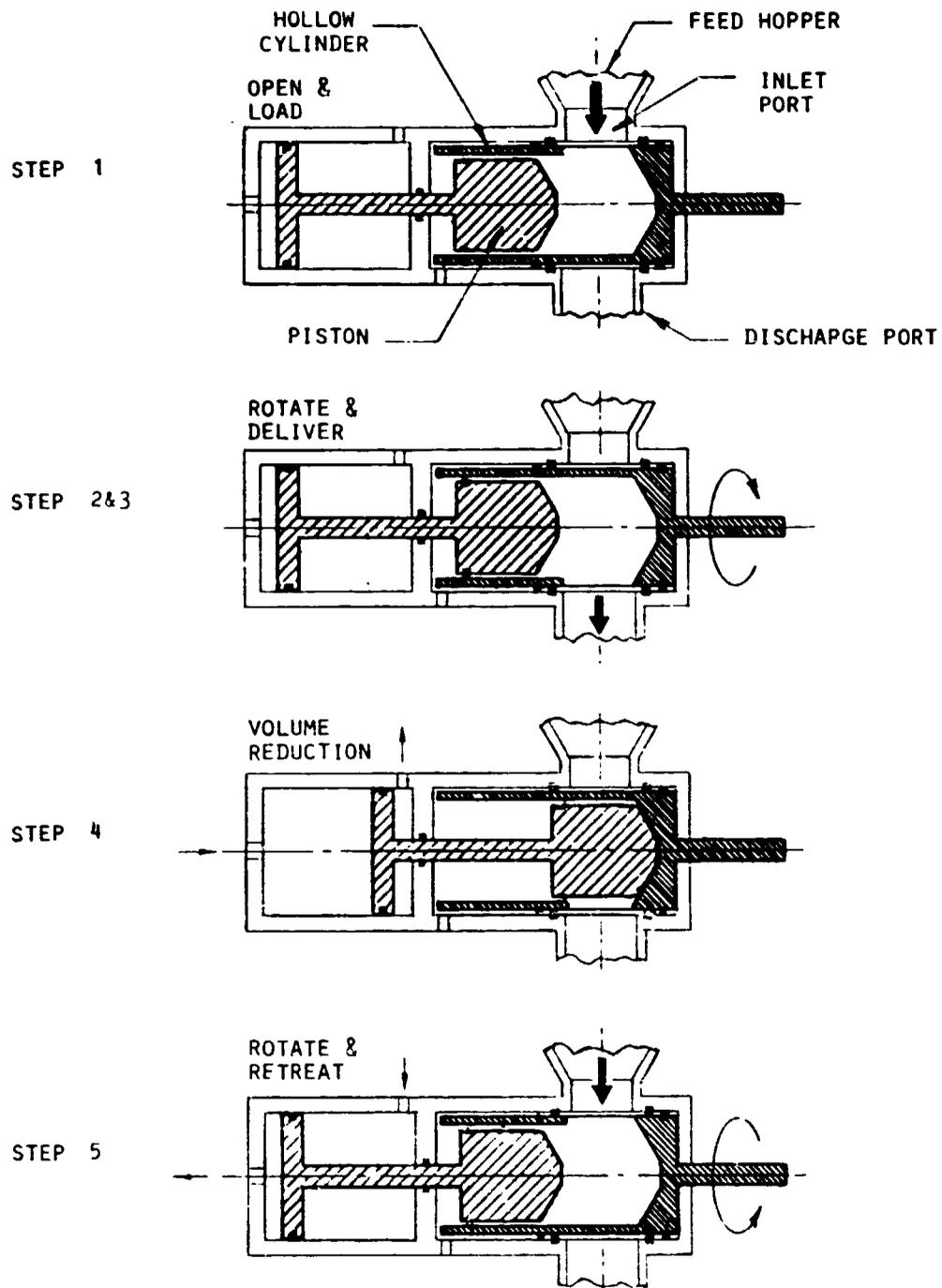
OPERATING SEQUENCE
ROTARY VALVE PISTON FEEDER
Figure 19

While not at the development stage of lockhoppers and slurry pump systems at this time, the piston feeder concepts should provide improvements in overall process operation because of the need for fewer auxiliary subsystems and simpler control systems. For example, no pressurization gas loss from the system will be incurred to degrade process efficiency (as occurs with lockhopper systems). Accordingly, there will be no requirement to collect, scrub, clean, and recompress such gas, thus eliminating much auxiliary equipment. Similarly, the need for substantial coal preparation and slurry mixing equipment necessary with any slurry system is obviated. The heat of vaporization penalty to the process is not present.

Next, the coal will not be physically or chemically changed (i.e., crushed, compacted, agglomerated, devolatilized, etc) during the delivery process. Furthermore, the concepts will accept all coal types (i.e., caking, non-caking, bituminous, non-bituminous, etc.) as well as a wide range of coal feed particle sizes.

A very unique benefit is the inherent safety in a condition of system malfunction. If a power failure or system component failure is encountered at any stage in the delivery cycle, there can be no "blow back" from the reactor of pressurization gas through the feeder. Finally, the piston-type feeders may be easily scaled up to future commercial scale feeders.

A preliminary economic evaluation has been made on the projected commercial use of piston feeders. Taking the concept of the piston feeder and scaling it up for installation at a conversion plant with a capacity of 15,000 tons/day and reactors operating at 1500 psig has indicated that considerably lower costs are expected as compared to existing equipment operating in similar situations.

ACKNOWLEDGEMENT

FOSTER-MILLER'S DEVELOPMENT OF DRY COAL FEED SYSTEMS

J. C. Harding

Foster-Miller Associates, Inc.
Waltham, Massachusetts

PREFACE

This report was prepared as an account of work sponsored by the
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ABSTRACT

A critical problem with all coal conversion processes which operate above atmospheric pressure is the delivery of coal to the process. The techniques now employed for feeding pressurized processes, lock hoppers and slurry feed systems, have serious shortcomings which render them questionable choices for commercial coal conversion applications.

Foster-Miller is developing alternative (second generation) dry coal feeder systems for pressurized conversion processes under contract to the Energy Research and Development Administration. Four feeder concepts were carried through a laboratory scale development program. These concepts included:

1. A centrifugal solids feeder
2. A fluidized piston feeder
3. A linear pocket feeder
4. A compacted coal plug feeder

The centrifugal feeder conveys pulverized coal from atmospheric pressure to a high pressure receiver by subjecting the coal to high centrifugal forces. The fluidized piston feeder is analogous to a reciprocating slurry pump, but with a fluidized feed stream rather than a slurried feed stream. The linear pocket feeder concept employs a series of sealed pockets to move the coal into the pressurized receiver. The design is suitable for a wide range of coal sizes. The coal plug feeder is a very simple design in which the coal feed is compacted into briquettes to limit high pressure gas losses.

Laboratory model testing of all concepts has been completed with encouraging results. Prototype pilot plant scale versions of concepts 1-3 have been built and are now under test. Promising feed systems will ultimately be installed on ERDA sponsored pilot scale conversion plants such as Synthane and Bi-Gas.

DEVELOPMENT OF DRY COAL FEED SYSTEMS

1.   Introduction

The objective of this program is to develop alternative dry coal feed systems for pressurized coal conversion processes. The specifications for this development are as indicated in Table 1. It should be noted that these specifications, as presently approved, apply to pilot plant applications. The eventual objective of the ERDA program is to develop coal feeders for demonstration plant applications. Feeder design considerations beyond the basic design specifications include power consumption, gas losses, gas added to the process, reliability, maintainability, wear, coal preparation required before feeding, coal degradation caused by the feeder, metering capability, development requirements and packaging flexibility.

Table 1. Specifications for Coal Feeder Development

Parameters	Value
Feed Rate	3 to 5 tons per hour
Feed Size	Fine - 3/4 inch
Feed Pressure	100-1500 psi
Temperature	As required

The program plan to accomplish the above objective is broken down into five phases which include:

Phase I   - Coal Feeder Conceptual Development
Phase II   - Pilot Scale Coal Feeder Development
Phase III - Pilot Plant Feeder System
             Qualification Testing
Phase IV   - Pilot Plant Coal Feed r Development
Phase V   - Feed System Installation and Test

The first phase has been completed having been conducted from June to December of 1975. Phase II, The Pilot Plant Scale Coal Feeder Development stage, involves translating the concepts from Phase I into actual hardware and verifying projected performance capabilities. This phase was initiated in September of 1975 and should be completed by October of 1977. Phase III, Pilot Plant Feeder System Qualification Testing, involves the construction and test of feed systems for eventual pilot plant installation. In Phase III these feed systems would be operated on a test facility at Foster-Miller Associates (FMA). Phase III is expected to be carried out during 1978 and 1979. Phases IV and V involve the refurbishing of the pilot plant feed system, (Phase IV), and the actual installation and testing of the feed system on a pilot plant, (Phase V). It is anticipated that the initiation of the Phase IV and V efforts would occur about 1980. The present FMA program is funded through Phase II.

2. Background

There are a large number of coal conversion processes which represent potential applications for feed systems. In Table 2, gasification processes are broken down generically in terms of bed type. This breakdown by gasification reactor bed type gives an application hierarchy in terms of feed size with entrained bed reactors requiring the finest feed size (typically 70 percent thru 200 mesh) and fixed bed/stirred bed reactors requiring the coarsest feed size. (Typically 1 3/4 x 1/4 inch for a Lurgi reactor) It should be noted that this generic breakdown leads to no hierarchy in terms of either pressure or state of development. In each class of gasifiers there are both developed and experimental processes, as well as processes which operate at both atmospheric and high pressure conditions. In this paper, feeder application will be related to gasification bed type.

Table 2. Generic Classification of Gasification Reactors

Type	Examples	Coal Size
Entrained Bed	Bi-Gas, Koppers - Totzek	Pulverized (70 percent - 200 mesh)
Fluidized Bed	Hygas, Synthane	Minus 8-20 mesh (to 1/8 inch for direct combustion)
Fixed Bed/ Stirred Bed	Lurgi, G.F. gas	Coarse (1-3/4 x 1/4 inch 1/16 inch)

3. Phase I Results

The objective of Phase I was to evaluate all possible ways of feeding coal to pressurized reactors and to recommend several techniques for development into hardware. As a result of the Phase I effort, FMA generated a series of concepts which cover the possible range of applications. Parenthetically, FMA also concluded, as a result of Phase I effort, that there is no such thing as a universal feeder. The result of the Phase I effort was a recommendation for continued development of four different feeder concepts. These concepts can broken down roughly into two classes; mechanically sealed and coal sealed. The four concepts are as follows:

1. Linear Pocket Feeder   Mechanically
2. Fluidized Piston Feeder   Sealed

3. Centrifugal Feeder
4. Coal Plug Feeder   Coal Sealed

Each of these concepts is described in detail below.

3.1 Linear Pocket Feeder

The first of the FMA concepts to be discussed is the linear pocket feeder illustrated schematically in figure 1. This feeder is essentially an enclosed tubular conveyor with very tight clearance. The device is pressure balanced across the receiving vessel so that the only driving forces required are those necessary to overcome friction. Referring to figure 1, the feeder can be broken down into five different sections; the feed section, the sealing section, the unloading section, the gas transfer section, and the return section.
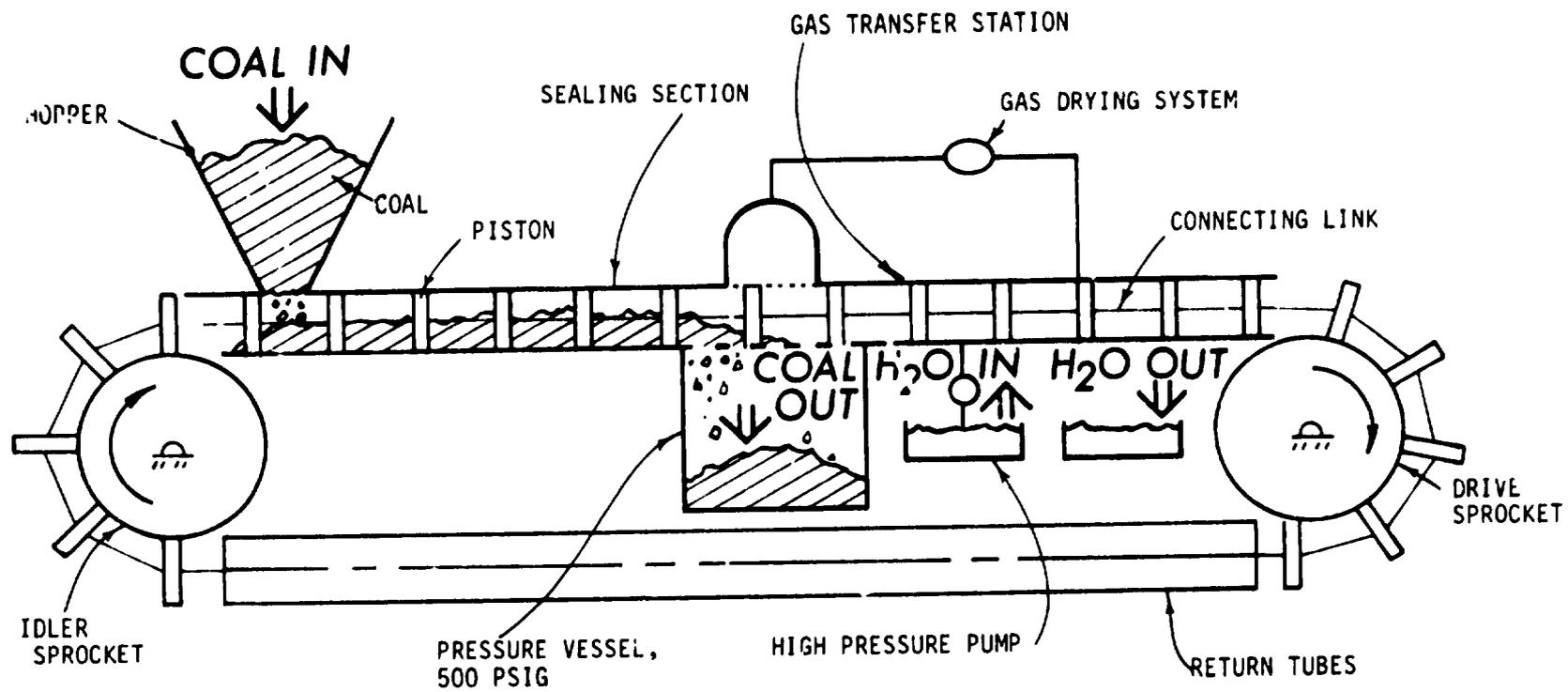
Figure 1. Schematic of Linear Pocket Feeder

In the feed hopper section coal is fed into the conveyor. This is followed by the sealing section where the conveyor flights pass through a close fit tube where they seal against the receiver vessel pressure. In the coal discharge section, the coal is dropped out of the conveyor and into a pressurized vessel. As the conveyor moves out of the discharge section, the space between the conveyor flights remains filled with high pressure gas. In the fourth section, the gas displacer section, high pressure water is pumped into the conveyor to displace the gas back into the receiver. As the conveyor comes out of the gas displacer section, the water is dumped into a sump and recycled. At this point the conveyor flights are dried off and pass thru a return section, which is merely a guide section to return the conveyor flights to the coal feed section.

In analyzing the linear pocket feeder functionally, it can be seen that this feeder or any other feeder has three basic functions to perform. The first of these functions is sealing against pressure, the second is transporting the coal, and the third is metering the coal. In the case of the linear pocket feeder, the sealing function is performed by means of a series of mechanical seals, the transportation function by the positive displacement action of the conveyor, and metering is controlled by the linear speed of the conveyor.

The linear pocket feeder is a straight forward feeder concept featuring redundant sealing and a simple efficient gas displacer technique. The feeder is suitable for all sizes and types of coals. The linear pocket feeder concept is suitable for near term applications, specifically, fixed and stirred bed gasification reactors and pressurized fluidized bed direct combustors.

3.2 Fluidized Piston Feeder

A cross sectional drawing of the fluidized piston feeder concept is presented in figure 2. The moving parts of the feeder
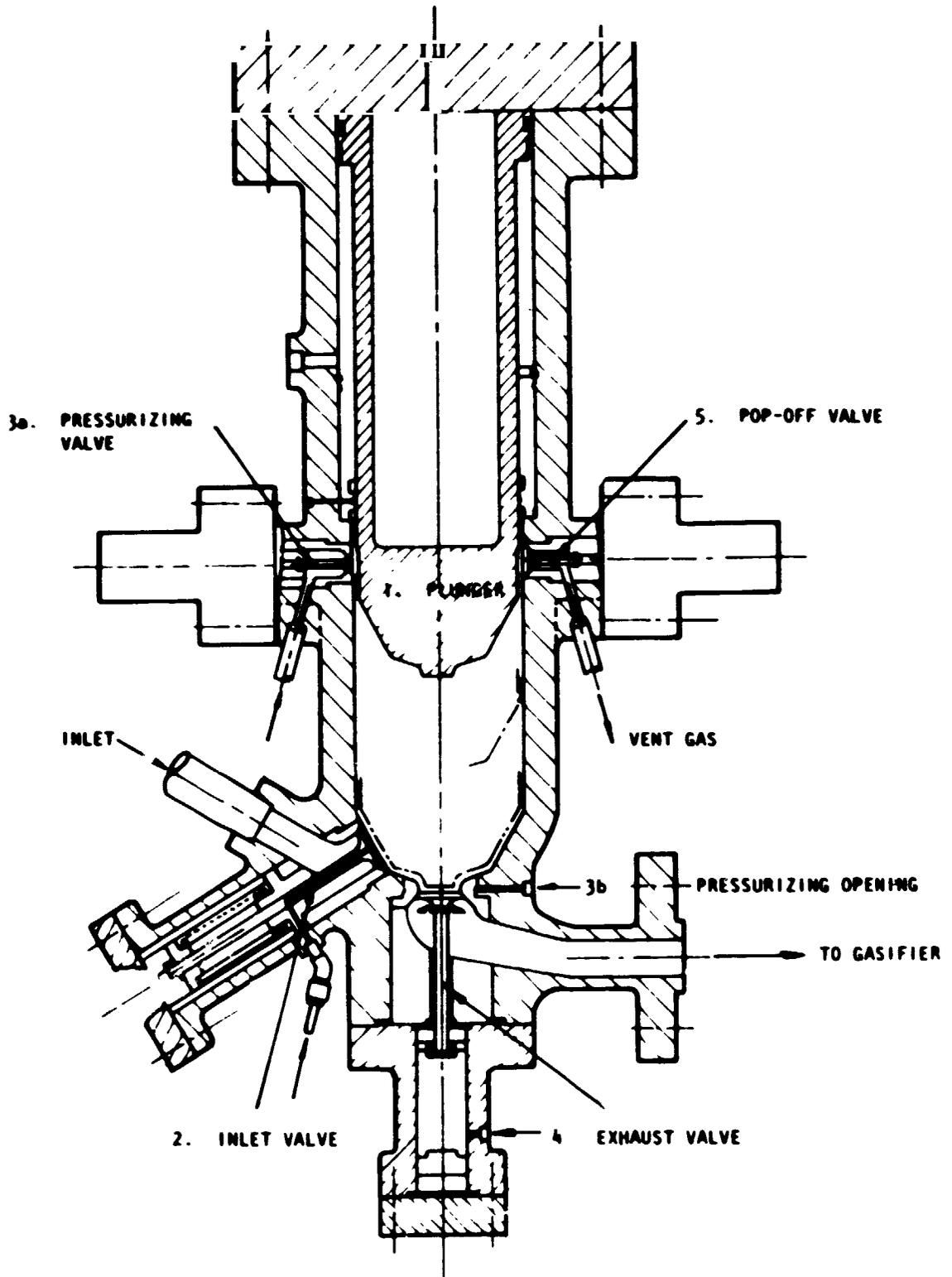
Figure 2. Fluidized Piston Feeder

consist of valves for coal inlet and discharge, a pressurizing valve,
a pressure relief (the "pop-off" valve), and a displacer piston.
The operating sequence of the feeder is depicted in figure 3. In
figure 3a a feed stroke has been completed. Figure 3b illustrates
a coal intake stroke during which the coal inlet valve is opened,
the displacer piston is withdrawn and a charge of fluidized coal is
drawn into the feed cylinder. In figure 3c the pressurizing valve
has opened, allowing the feed cylinder pressure to reach a value
slightly above the pressure of the receiving vessel. At this point,
the discharge valve is opened and the coal is flushed on out of the
cylinder. It should be noted that the coal is not pushed by the
piston but is flushed out by the gas. The piston serves as a gas
displacer only. After the coal is flushed out of the cylinder, the
displacer piston is stroked all the way down, the intake exhaust
valve closes, and the residual gas in the cylinder is vented through
the pop-off valve (figure 3d).

Analyzing the concept functionally, sealing is performed
mechanically by valves, coal transport is accomplished pneumatically,
and metering, as in any batch type device, is controlled by the
cycle rate of the feeder.

This concept has been characterized as a "small, high
cycle rate lock hopper". It is important in evaluating the con-
cept to appreciate how the fluidized piston feeder differs from the
conventional lock hopper. The device does indeed cycle much more
rapidly than a typical lock hopper. This high cycle rate is made
possible because the coal is handled in a fluidized state. The
high cycle rate of the piston feeder allows the device to be much
smaller than a lock hopper of comparable feed rate. The small size
of the device, in turn, makes it easier to adapt a displacer piston
to the design to limit gas losses. Furthermore, by using pneumatic
techniques to transport coal, more packaging flexibility is possible
than with the gravity transport mode of the lock hopper.

COAL

PURGE
GAS

INTAKE

PRESSURIZING
GAS

COAL/GAS

VENT GAS

DISCHARGE

VENT
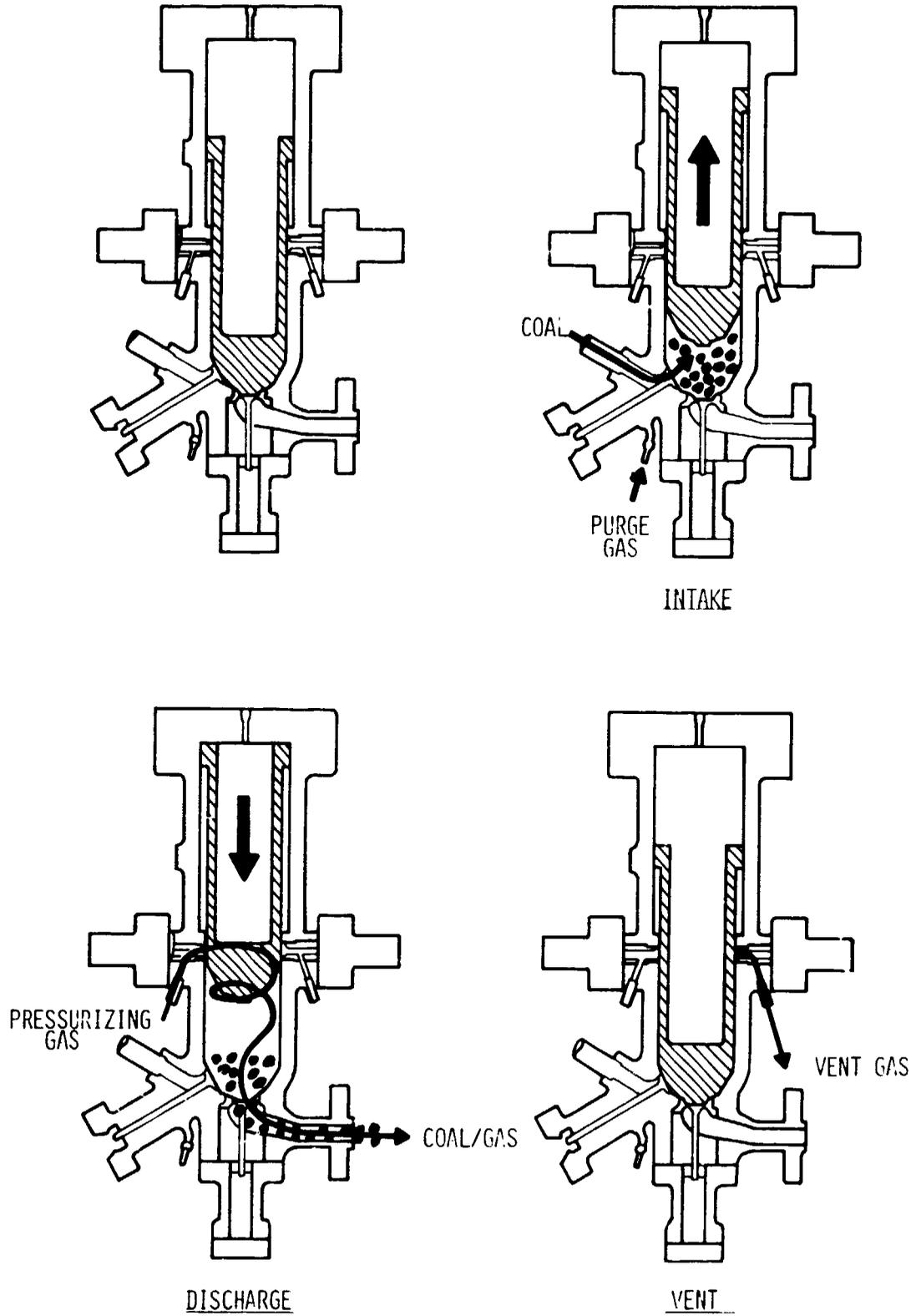
Figure 3. Fluidized Piston Feeder Operational Sequence

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Key development areas for the piston feeder concept are valv·s and seals. The concept should be capable of feeding minus 1/4 inch coals making it suitable for fluidized bed and entrained bed gasification applications.

3.3 Centrifugal Feeder

The centrifugal feeder is one of FMA's two "coal sealed" feeder co..cepts. The concept can be viewed as a centrifugal pump for solids or, alternatively, as a standpipe which has been reduced in length by placing it in a high "g" field created by centrifugal force. The centrifugal feeder concept is illustrated schematically in figure 4. The feeder consists of an atmospheric pressure supply hopper, a pipe which supplies coal to the feeder head (downc.)mer), and the feeder head. An open tube is provided to allow the gas which is entrained in the coal flowing to the feeder head to be vented to the atmosphere (figure 5). As coal is supplied to the feeder head through the stationary downcomer, it is picked up by an impeller and driven into the feeder passages (termed sprues). Figure 6 illustrates the manner by which the coal creates a dynamic seal as it moves through the feeder sprues. The moving coal is permeable and gas can leak through the coal (Vg). For the feeder to seal, the coal flow rate (Vc) must equal or exceed the gas flow rate. In the ideal "null" mode (Vc=Vg) there is no net gas loss from the receiver and no gas is conveyed into the receiver with the coal. In practical applications the feeder would probably be run with a slight net gas input to the receiver to ensure stable operation.

In functional terms, the centrifugal feeder concept operates in a continuous mode, seals with moving coal, transports by centrifugal force, and meters by controlling the rotational speed of the feeder head. While the centrifugal feeder is mechanically simple, the successful development of the device requires the solution of subtle and complicated twc-phase flow problems. The concept is basically suitable for fine coal feed in entrained bed gasification applications.

NORMAL LOW-PRESSURE FEED

PRIMING/EMERGENCY VALVE

PRIMING COMPRESSOR

DE-PRIMING VALVE

COAL SUPPLY
VESSEL

FEED ROTOR

FEED
VESSEL

FEEDER DRIVE MOTOR
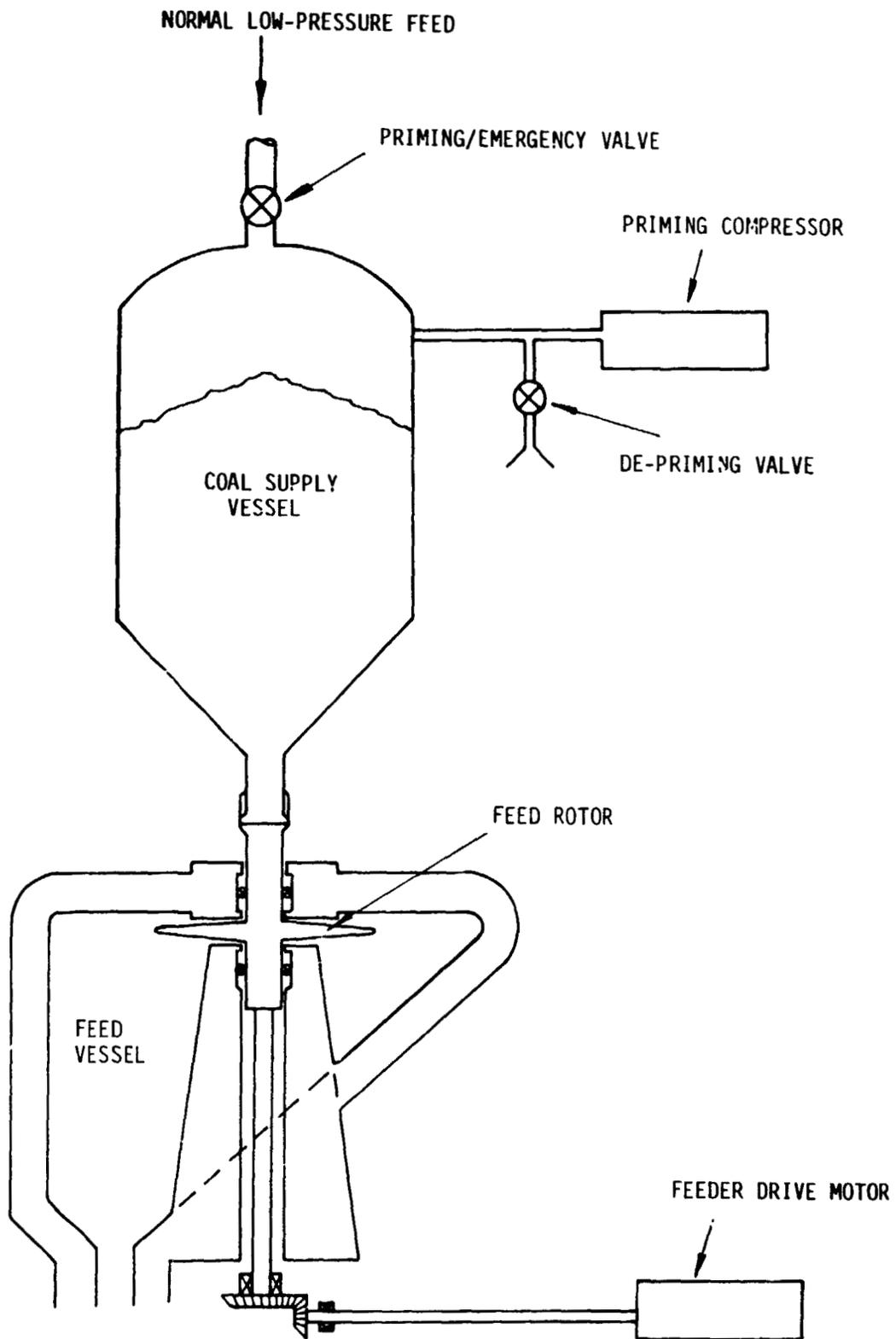
Figure 4. Schematic Diagram of a Centrifugal Feeder System

298
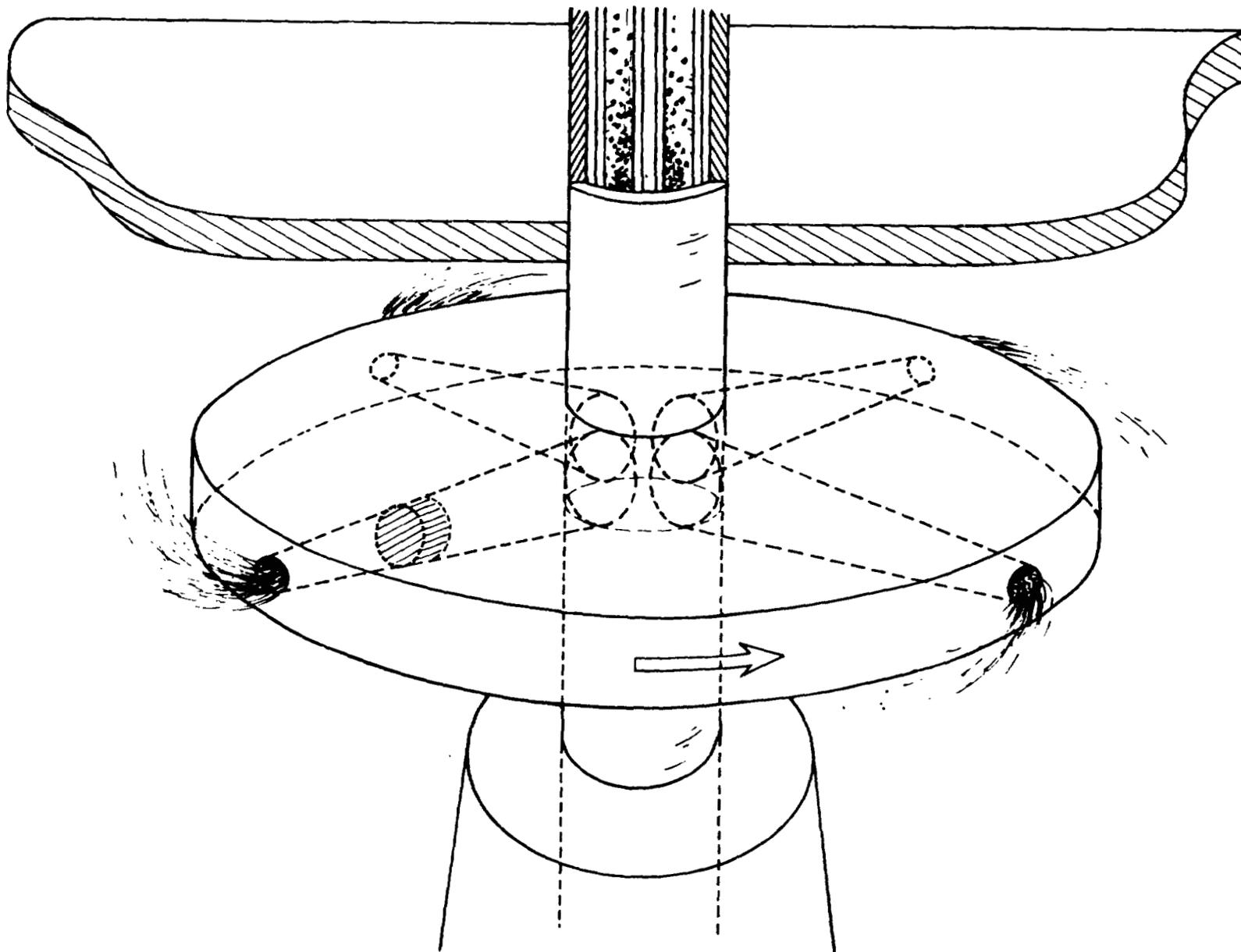
Figure 5. Schematic Representation of the Centrifugal Feeder Rotor
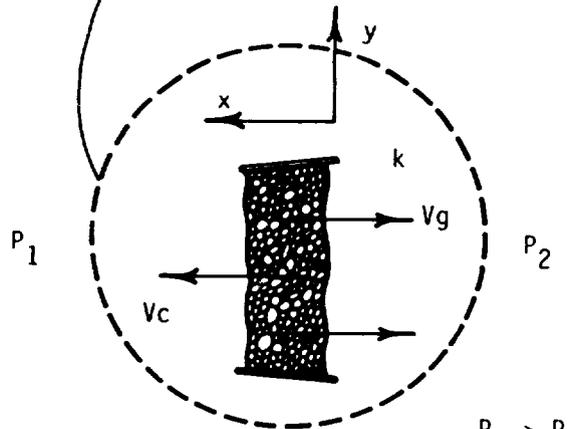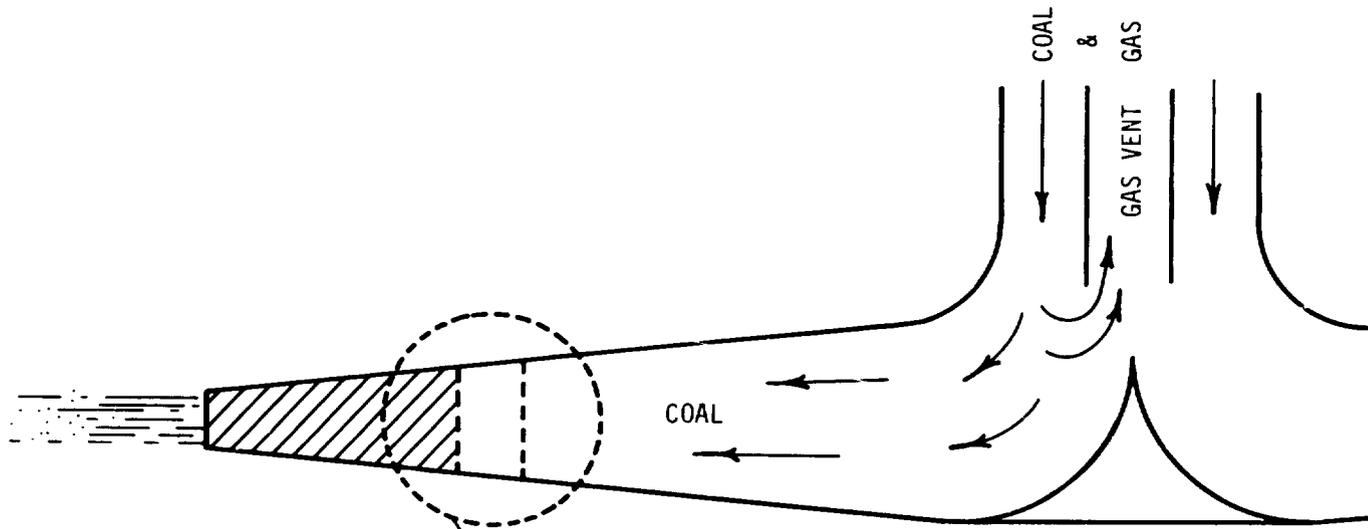
Figure 6. Operating Principle of Centrifugal Feeder

3.4 Coal Plug Feeder

The coal plug feeder concept also uses the coal itself
to create a pressure seal. In figure 7 a reciprocating screw
feeder configuration is illustrated. Other configurations are also
possible. The operation of the machine is depicted in figure 8.
In the top view a feed stroke has been completed. In the middle
view a new charge of coal is fed into the barrel of machine by the
rotation of the feed screw as the plunger mechanism retracts. In
the bottom view, the plunger reciprocates forward creating a new
briquette to add to the series of briquettes formed in previous
cycles. With successive cycles briquettes are formed and pushed
progressively into the receiver. The briquettes do not form an
absolute seal but rather a low permeability plug.

Functionally, the coal plug concept is a batch type
feeder, which seals with compacted coal. The coal is transported
with positive displacement mechanical action and metering is con-
trolled by cycle rate. The coal plug concept gives a very simple
feeder configuration. However, the concept has very high power
requirements and potentially severe wear rate problems The feeder
is suitable for feeding pulverized coal to entrained bed gasifiers.
This application requires that means be provided to break up the
briquettes in the high pressure receiver.

3.5 Phase I Summary

In table 3, a relative comparison of the four FMA feeder
concepts is presented. The Phase I recommended concepts are com-
pared in terms of feed pressure capability, feed size, application,
and feed cycle. Development work following the Phase I effort
indicates that the linear pocket feeder may be suitable for pres-
sures to 1,000 psi or higher.

HIGH PRESSURE
RECEIVER

COAL
SUPPLY

GAS
BLEED

SCREW

BALL SPLINE

HYDRAULIC
CYLINDER
RAM

CHECK VALVE

TAPER

SCREW DRIVE GEAR

FEED COAL

LOAD FEEDER
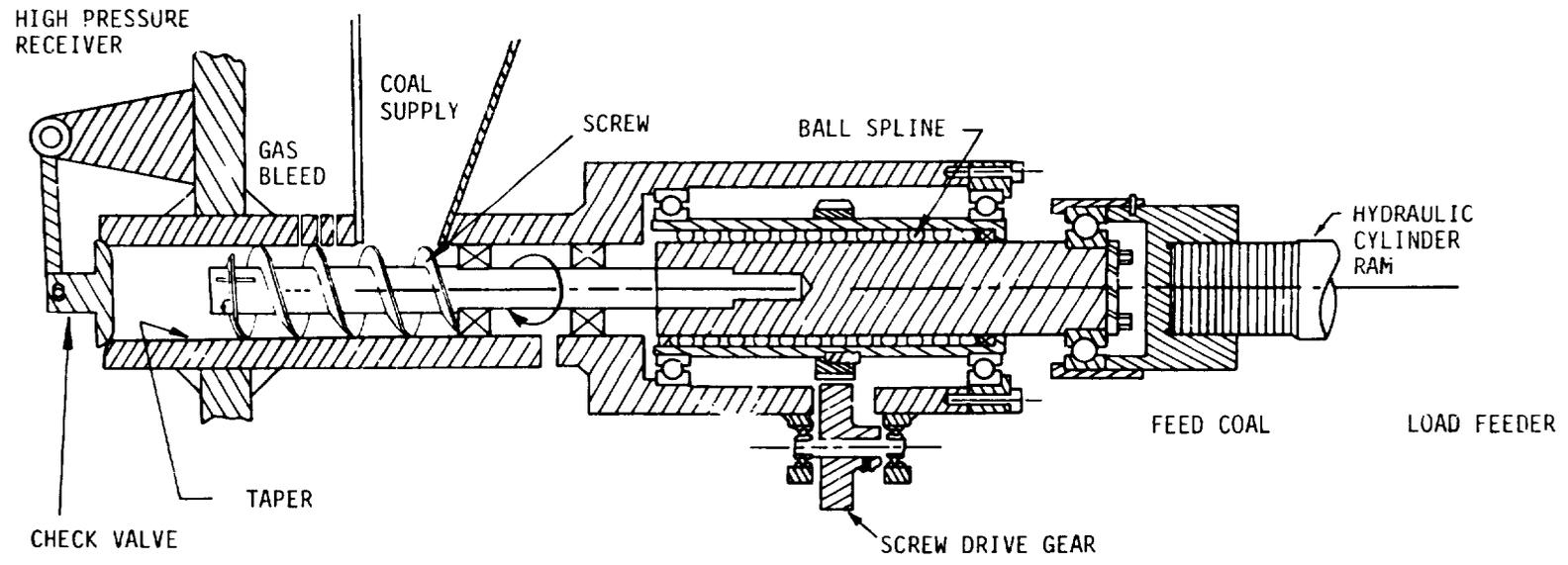
Figure 7.   Cross Section of a Reciprocating Screw Coal Plug Feeder

1. COMPLETION OF FEED STROKE

2. LOADING OF COAL CHARGE

3. FEED STROKE
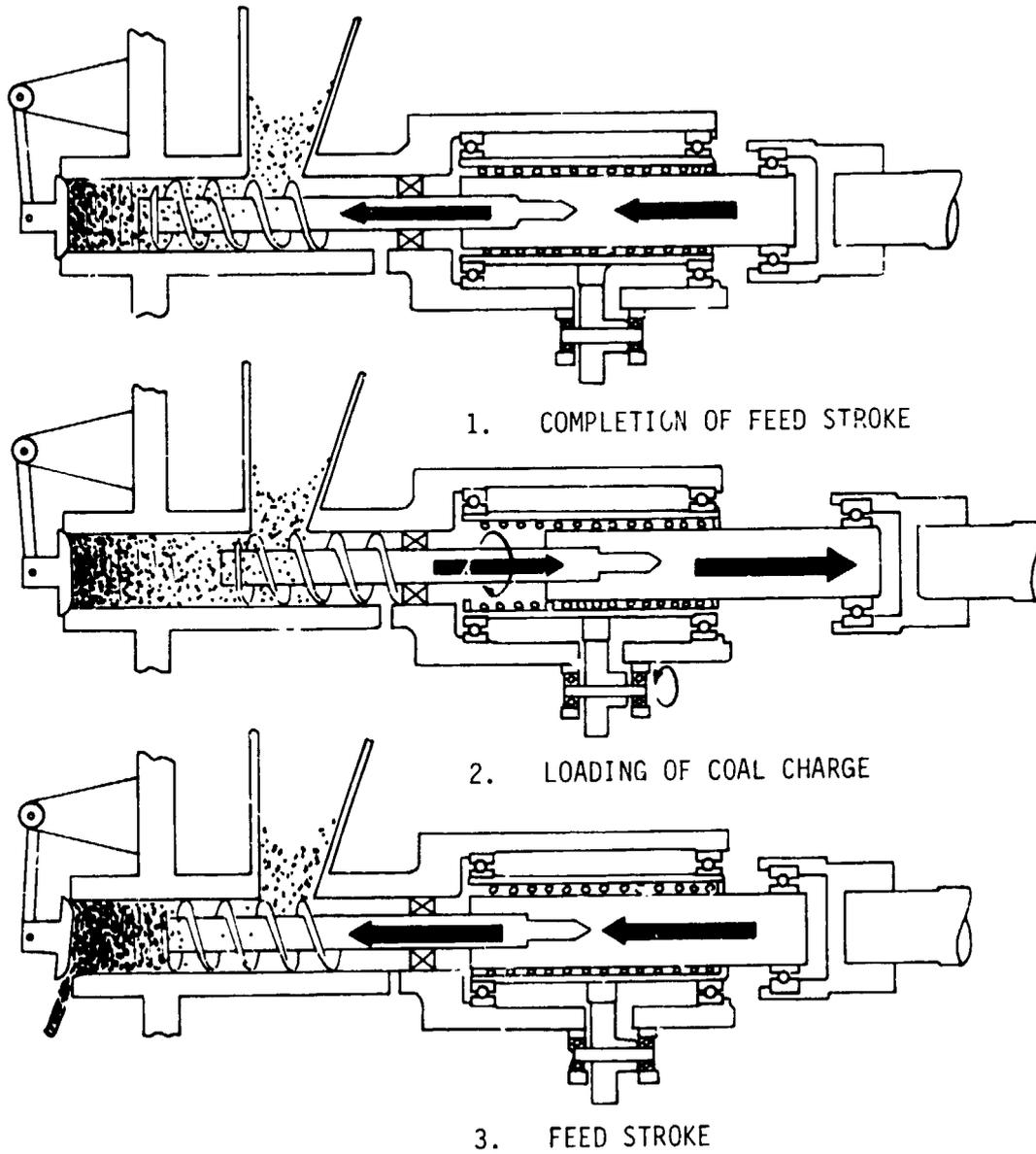
Figure 8. Operating Cycle of a Reciprocating
Screw Coal Plug Feeder

Table 3. Comparison of Foster-Miller Coal Feeder Concepts

Features	Centrifugal Feeder	Fluidized Piston Feeder	Coal Plug Feeder	Linear Pocket Feeder
Feed Pressure Differential (Maximum)	1000 psi	1,500 psi +	1,000 – 1,500 psi	500 psi
Feed Size	Fine*	Medium-Fine	Fine	Coarse-Fine
Application	Entrained Bed	Entrained Bed Fluidized Bed	Entrained Bed	All
Feed Cycle	Continuous	Intermittent	Intermittent	Continuous

* The feed pressure differential performance of the centrifugal feeder will decrease as feed size increases.

4.   Phase II Results

The Phase II program effort is essentially the hardware eval-
uation of the concepts developed in Phase I.   In conducting this
phase two scaling steps have been carried out.   Bench scale models
of all four concepts were designed, built and tested.   Following
this work, pilot plant scale prototypes of three of the four con-
cepts have been designed and built and are now under test.

4.1   Bench Scale Testing

The bench scale t sting will be discussed only briefly
in this paper.   Details of the bench scale test program are pre-
sented in reference 2.   The bench scale centrifugal feeder model
was used to perform coal flow studies and to determine if the
required outlet limited (full sprue) flow mode could be achieved.
This flow condition is necessary if the device is to be capable of
sealing against pressure.   For the piston feeder development a glass
model of the feeder was built and tested to study possible devel-
opment problems.   A full scale, two flight, sealing section model
for the linear pocket feeder was developed to test leakage rates
and drive forces.   For the plug feeder, a small bench scale appar-
atus was tested to determine leakage, plug stability, driving forces,
compaction forces required etc.   At the conclusion of the bench
scale testing, essentially all four concepts were deemed to be
technically feasible.   However, because of funding limitations, it
was necessary to eliminate one of the four feeder concepts.   The
plug feeder concept development was suspended.   Basically, the
considerations influencing this decision included the high power
requirements of the device, the lack of flexibility in end use, and
duplication of effort with respect to other feeder development pro-
grams being conducted.   It should be noted that in the bench scale
testing, the plug feeder apparatus was able to feed reliably against
pressures to 1200 psi with a stable plug and with very low leakage
rates.   The leakage rates were, in fact, substantially less than
lock hopper venting losses under comparable conditions.

4.2 Prototype Testing

Prototype feeders have been designed, built, and are now being tested for the centrifugal, piston, and linear pocket concepts. The prototype hardware is described and test results to date are discussed below.

4.2.1 Centrifugal Feeder Prototype

Testing of the centrifugal feeder prototype was initiated in mid November 1976. The test facility is illustrated in figure 9. The feeder rotor is located within the domed head of the large vessel on the left side of the support structure. The coal supply runs down a fluidized standpipe from a weigh hopper on the roof of the test facility. The weigh hopper is not visible in the photograph. The specifications for the prototype feeder are given in Table 4. The 200 psi limit for the prototype feeder was selected as being sufficient to prove the validity of the concept while keeping testing costs and component lead times to reasonable figures.

Table 4. Prototype Centrifugal Feeder Specifications

Parameter	Value
Feed Rate	Up to 1 ton per hour
Pressure Differential	Up to 200 psi
Speed	2000 RPM (nominal) 3000 RPM (maximum)
Rotor Diameter	24 inches
Downcomer Diameter	3 inches
Sprue Outlet Diameter	Up to 0.5 inches
Sprue Inlet Diameter	Up to 2.5 inches
Coal Size	70 percent through 200 mesh
Coal Moisture	1.5 percent (presently in use)
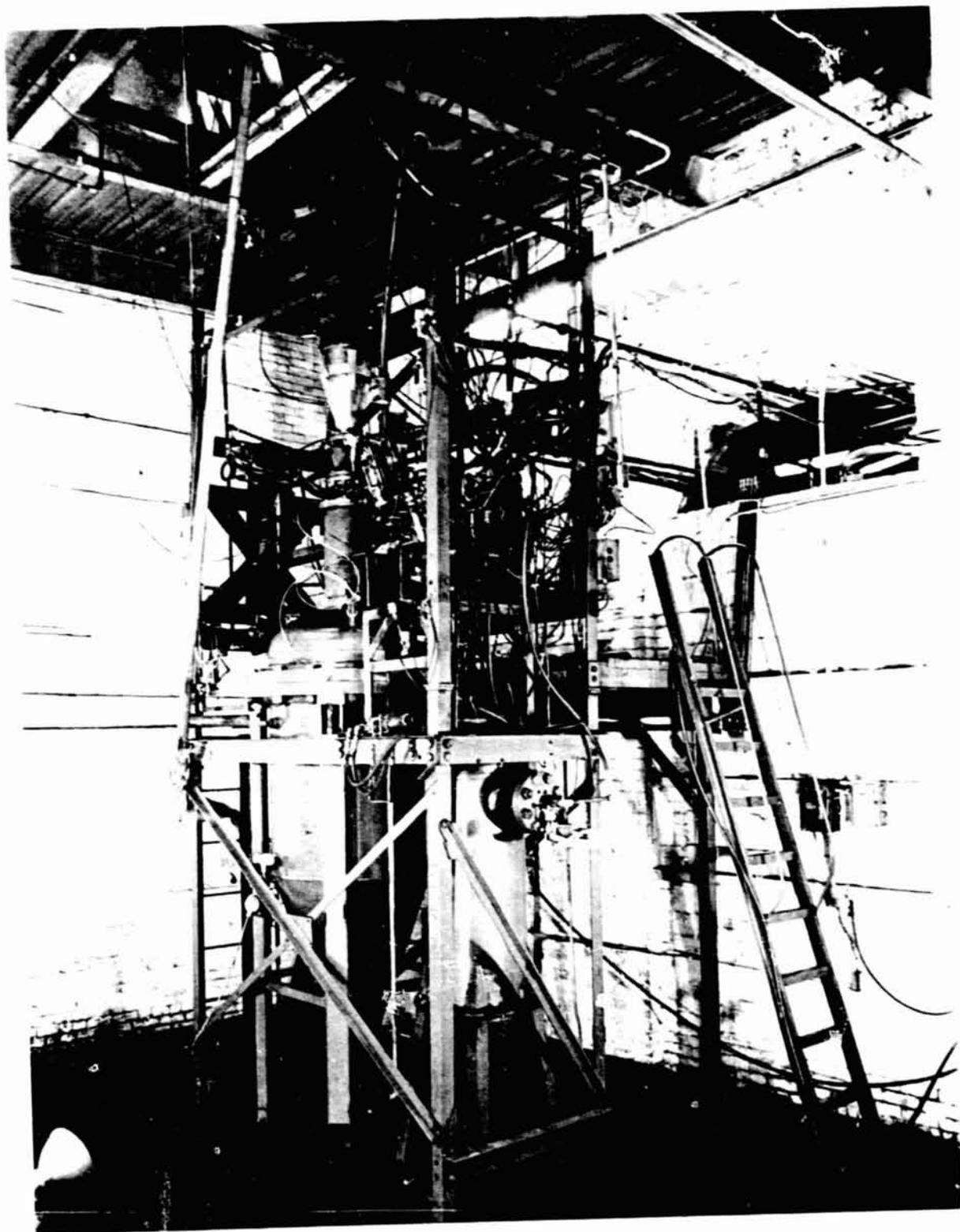
Figure 9.   Prototype Feeder Test Facility

Testing of the prototype centrifugal feeder has progressed through two stages. In conjunction with the testing, an analytical study of the centrifugal feeder concept is being carried out. This study has been computerized to allow rapid comparisons between test results and the analytical model of feeder performance.

The first series of feeder tests were plagued by problems in maintaining reliable coal feed to the inlet of the centrifugal rotor. These problems have been overcome by relocating and fluidizing the coal supply plumbing. The results of the series A feeder tests are summarized in Table 5. These results indicated that the prototype feeder operated at high feed rates with little or no pressure differential (Run 2, Table 5) and that feed rates dropped off very rapidly as the pressure differential increased (Run 6).

The analytical model of feeder performance was revised substantially and a rationale was developed which explained the results achieved in the series A tests. Based on this rationale, detail changes were made in the centrifugal rotor design and a second series of tests were initiated. The results to date in the B series test program are presented in Figure 10. The test results correlate very well with the data derived from the analytical feeder model.

Figure 10 presents test results and theoretical predictions for two sets of sprues, one with a 1/4 inch outlet diameter and a set with .150 inch openings. Referring to Figure 10, the 1/4 inch set has achieved pressure differentials to about 40 psi with a feed rate of about one ton per hour. The .150 inch set has achieved pressure differentials to 185 psi at feed rates of approximately one quarter ton per hour. Further testing will involve one more sprue set designed to achieve the 200 psi pressure differentail and one ton per hour feed rate projected for the prototype.

Table 5. Summary of Centrifugal Feeder Test Results
Test Series A

Run No.	Sprue Outlet Diameter (In.)	Rotor Speed (RPM)	Receiver Vessel Pressure (PSI)	Flowrate (LB/HR)	Comments
1	0.5	2000	20	-	Material flow test with "Speedy Dry"
2	0.188	2000	--	12,000	Temporary fluidization in rotor hub
3	0.5	2000	4	-	No sealing beyond 4 psi
4	0.375	2000	1.1	-	Break in coal delivery to hub. Coal seal breaks when pressure still building up
5	0.375	550	25	2,400	Coal fluidized in feed hopper, feed lines and rotor hub
6	0.375	1500	75	-	No discernable coal flow
7	0.375	2700	25	1,600	One sprue operational. Large foreign particle in the other

CENTRIFUGAL FEEDER PERFORMANCE
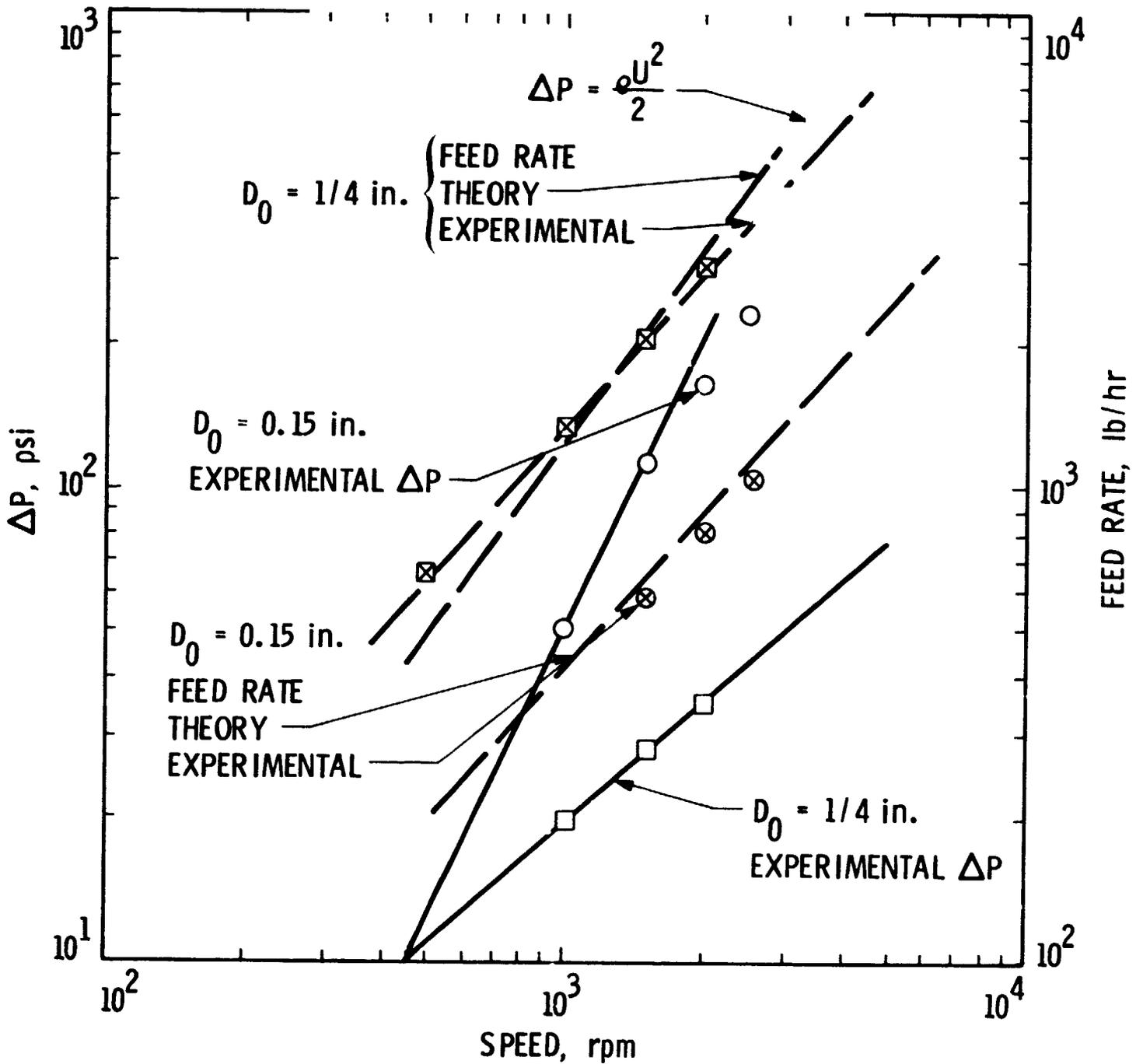
Figure 10. Prototype Centrifugal Feeder Test Results (Series B)

4.2.2 Fluidized Piston Feeder Prototype

The piston feeder prototype is essentially similar to the concept illustrated in Figure 2. The most significant change in the prototype is the use of a double inlet valve. The inlet valve set has a coarse valve which performs the function of stopping the coal flow from the supply hopper and a fine valve which performs the gas sealing function. The specifications for the prototype feeder are presented in Table 6.

Table 6. Prototype Fluidized Piston Feeder
Design Specifications

Parameter	Value
Feed Rate	1 ton per hour
Pressure Differential	1000 psi (1500 psi max.)
Feed Hopper Pressure	0-20 psi
Receiver Pressure	Up to 1500 psi
Coal Size	70 percent thru 200 mesh (1/4 x 0 max.)
Cylinder Diameter	4 inch
Stroke Length	Up to 8 inch

The performance which has been achieved to date with the prototype feeder is presented in Table 7 and Figure 11. The drop off in feed rate with increasing receiver pressure which is evident in Figure 11 is indicative of valve leakage problems. At lower pressure differential, the density of the material throughput has been above projections dropping off to below projections at higher differentials. A slight improvement in throughput occurs with increased supply hopper pressure. To date, the unit has fed at differentials up to 500 psi. Feed rate at this differential was approximately 200 pounds per hour. Development is continuing

FEED HOPPER PRESSURE (PSI)
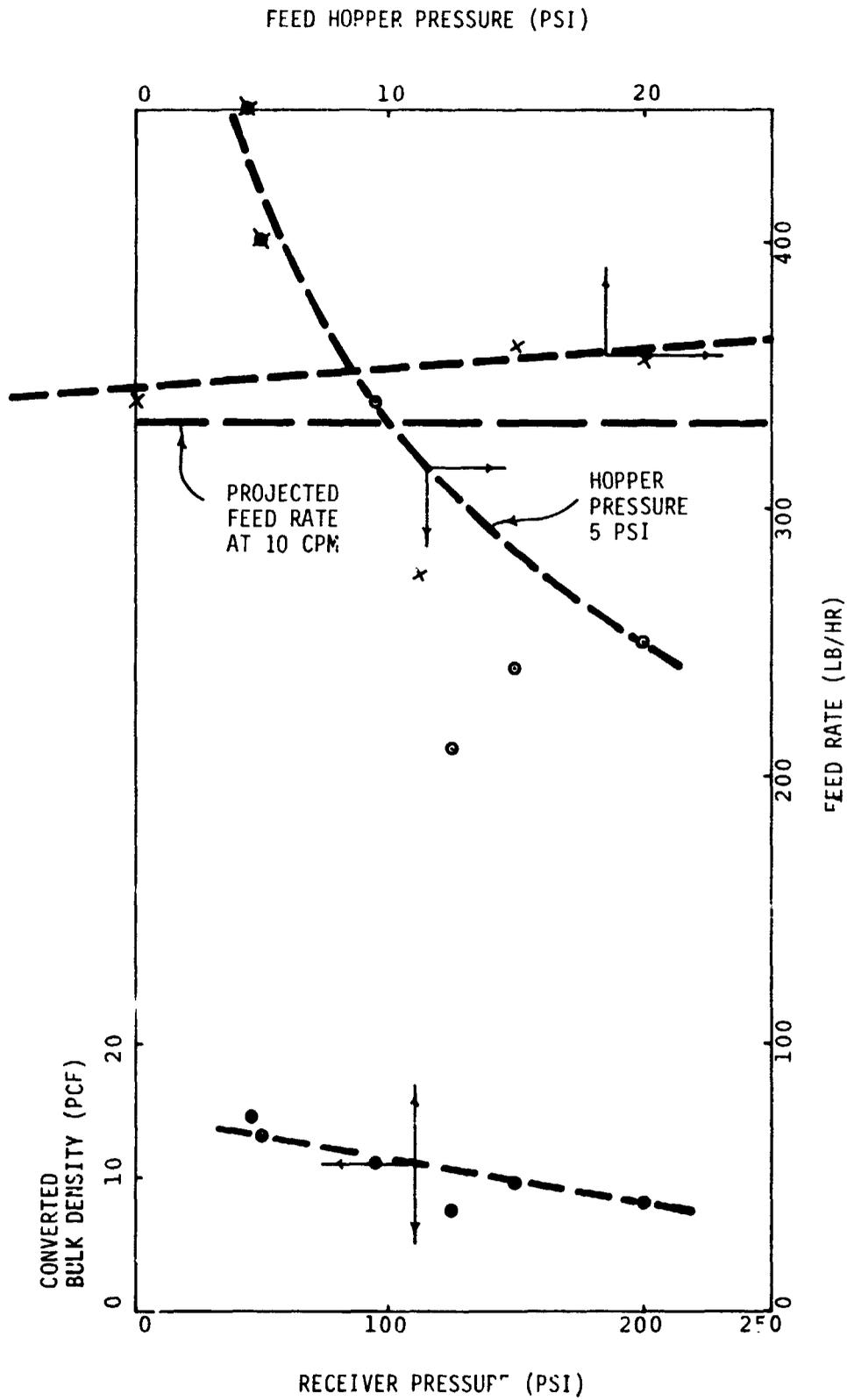
Figure 11.   Prototype FP Feeder Performance

Table 7. Fluidized Piston Feeder: Typical Test Results

Stroke Length: 8 inches

Run No.	Feed Hopper Press. (psi)	Line Fldzn. Press. (psi)	Valve Purge Press. (psi)	Cylinder Press. (psi)	Receiver Vessel Press. (psi)	Cycle Time (sec)	Run Time (min)	Feed Rate (lb/hr)	Estimated Feed Density (PCF)
1	5	10	40	80	45	6.7	12	450	14.5
2	10	10	40	85	45	10.0	12	265	12.72
3	15	-	60	80	50	6.7	12	360	11.61
4	20	-	60	80	50	6.7	13.5	356	11.47
5	10	10	50	140	95	6.8	8	323	10.5
6	5	-	60	150	125	7.04	12	210	7.09
7	7	10	60	180	145	6.8	7.5	304	9.75
8	5	10	60	250	200	6.8	6	250	8.16

Stroke Length: 6½ inches, intake valve modified.

Run No.	Feed Hopper Press. (psi)	Line Fldzn. Press. (psi)	Valve Purge Press. (psi)	Cylinder Press. (psi)	Receiver Vessel Press. (psi)	Cycle Time (sec)	Run Time (min)	Feed Rate (lb/hr)	Estimated Feed Density (PCF)
9	5	10	60	350	300	6.7	6	380	15.1
10	5	10	60	450	400	6.7	6	160	6.35
11	5	20	100	550	500	6.7	6	200	7.94

*Estimated feed density (converted density) refers to the mass of coal in the cylinder per unit cylinder volume. Please note that the maximum density possible is that of coal at minimum fluidization.

centered around improvement of valve performance. The piston feeder
has also been modeled analytically. The computer model allows
the effect of such variables as valve timing, cycle rate, and coal
input density to be evaluated.

4.2.3 Linear Pocket Feeder Prototype

The configuration of the linear pocket feeder
prototype is illustrated in Figure 12. The feeder is 16 feet in
length from sproket axle to sproket axle and about 18 feet in length
overall. The prototype is functionally equivalent to the basic
concept discussed earlier. The specifications for the prototype
feeder are presented in Table 8. The details of the pistons or
flights of the conveyor are illustrated in Figure 13. The alumina
wear ring takes the brunt of the mechanical wear and forces invol-
ved in breaking and feeding the coal. The metal hook type sealing
rings act as redundant seals.

Table 8.   Design Specifications
                       for the
           Linear Pocket Feeder Prototype

Parameter	Value
Flow Rate of Coal	0-5 tons per hour @ 2.5 feet per second 50 percent loading
Maximum Operating Pressure	500 psig
Diameter	3.5 inches
Coal Size	3/4 inch
Temperature	Ambient
Chain Speed	0-10 feet per second
Minimum Wear Life of Critical Components	8000 hours

Coal Input Hopper

Return Tubes

Drive Assembly

Drive Sprocket

Sealing Section

Idler
Sprocket

Water Outlet
Station

Gas Transfer
Station

Piston
Assembly

Coal
Unloading
Station
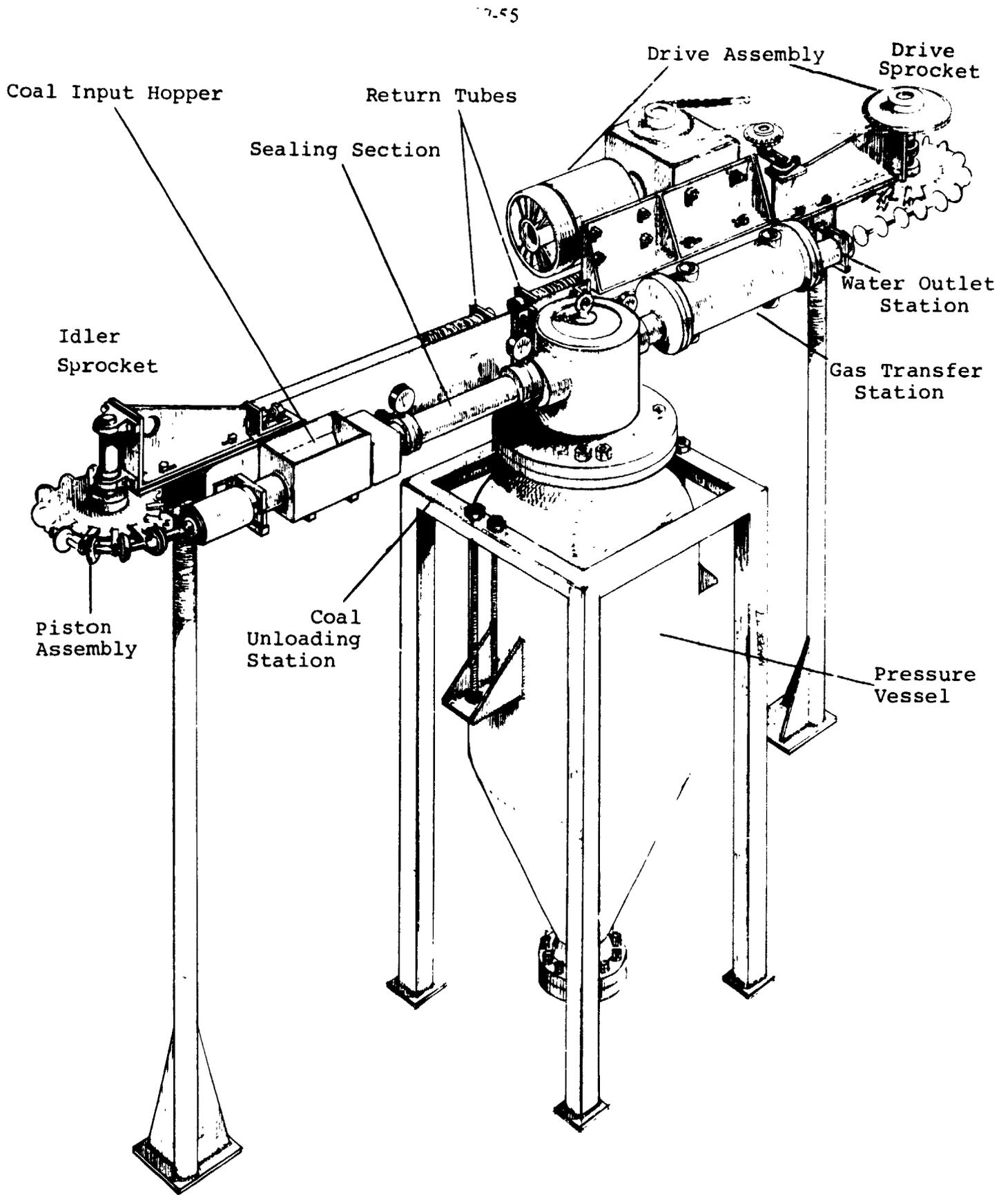
Pressure
Vessel

Figure 12.   Prototype Linear Pocket Feeder
(LFP) Presently Being Tested

315

DIRECTION OF MOTION
OF CHAIN

ALUMINA WEAR RING

3-1/2

CONNECTING LINK
6

PISTON
BODY

$\frac{13}{16}$ | 1 | $\frac{7}{16}$

HOOK JOINT
SEALING RINGS
(2)
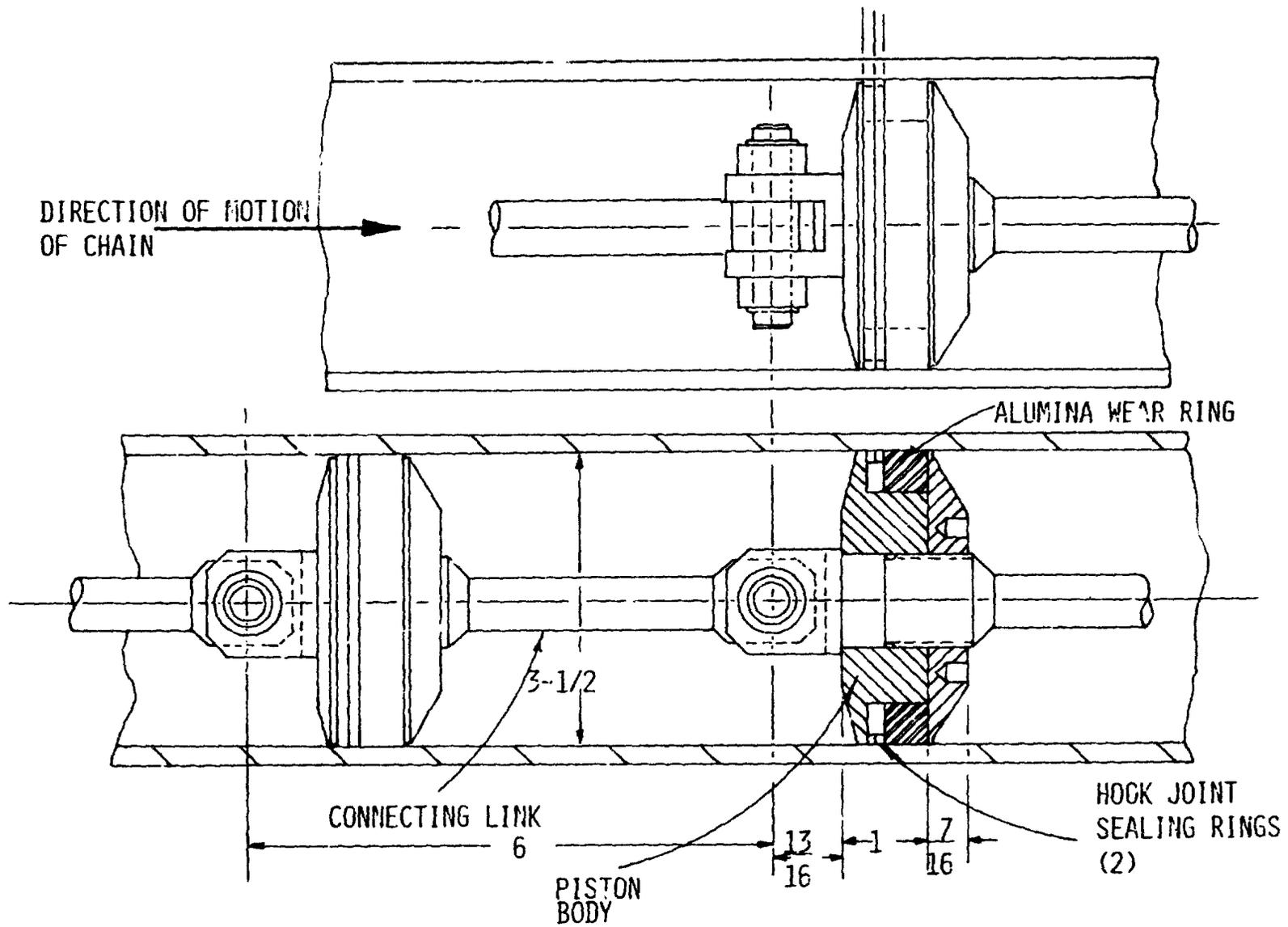
Figure 13. Linear Pocket Feeder Piston Details

At this time, the pressure vessel components of the linear pocket feeder have not been received, so testing to date has been limited to running coal without a pressure differential imposed on the device. In this testing, the feeder is operating with no significant problems. Line pulling force is about as predicted and coal feed has been fed against zero pressure differential. The feeder has been operated continuously for five hours, without coal, with no probelms being noted. Pressure testing of the feeder is scheduled for August 1977.

4.3 Phase II Summary

The remaining FMA feeder concepts are compared in Table 9. A consideration not noted in the table is the anticipated development time required for each of the feeder concepts. It is estimated that the linear pocket feeder can be developed for near term applications (approximately 1 to 3 years), the piston feeder for mid term applications (approximately 2 to 5 years), and the centrifugal feeder will probably require the most de..' w...t (3 to 5 years plus).

Table 9. Comparison of Foster-Miller Coal Feeder Concepts

Features	Linear Pocket Feeder	Fluidized Piston Feeder	Centrifugal Feeder
Feed Pressure Differential (Maximum)	500 psi +	1,500 psi +	1000 psi
Feed Size	Coarse - Fine	Medium-Fine	Fine
Application	All	Entrained Bed Fluidized Bed	Entrained Bed
Feed Cycle	Continuous	Intermittent	Continuous

5. Future Development

In Figures 14 to 16, the three feeder concepts are depicted in a form suitable for field applications. The linear pocket feeder illustrated is a 3½ inch inside diameter unit suitable for feed rates to five tons per hour. In Figure 15 the fluidized piston feeder is illustrated as it would be constructed for field applications. The unit illustrated would have a cylinder diameter of 2.5 feet and stroke of 3.125 feet and would feed at the rate of 5 tons per hour with a cycle time of 36 seconds. The centrifugal feeder design of Figure 16 eliminates the pressure vessel receiver used for the prototype. The feeder head is enclosed in a case which could be plumbed to a gas feed line to the reactor. Coal is conveyed out of the case continuously and carried into the process. The design illustrated employs a 3 foot diameter h d and would be suitable for flow rates of 3 to 5 tons per hour.
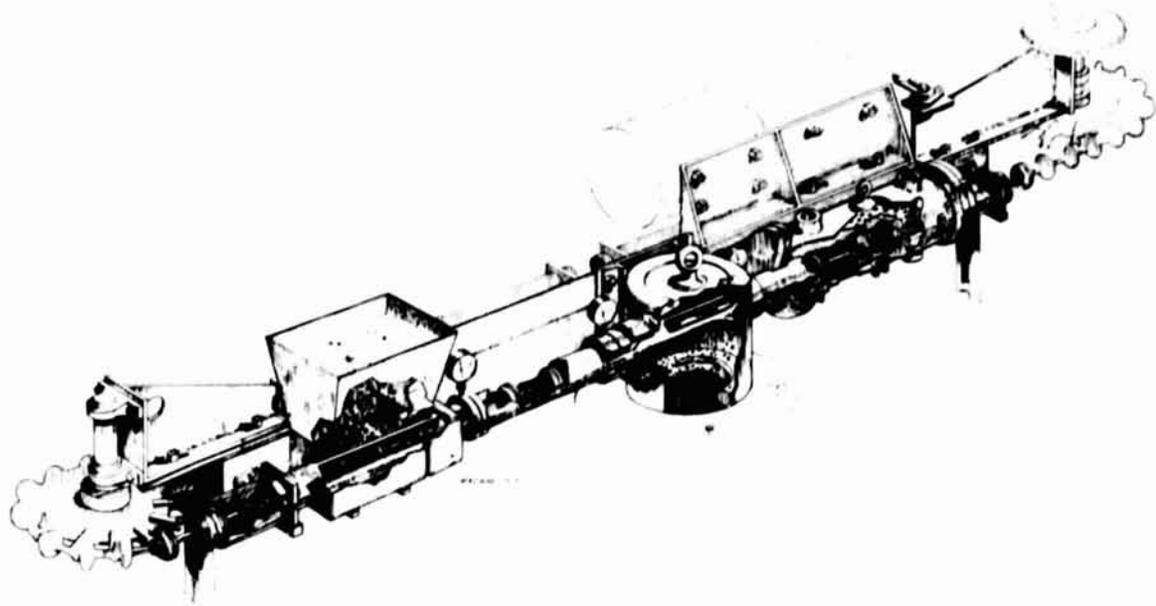
Figure 14.   Linear Pocket Feeder
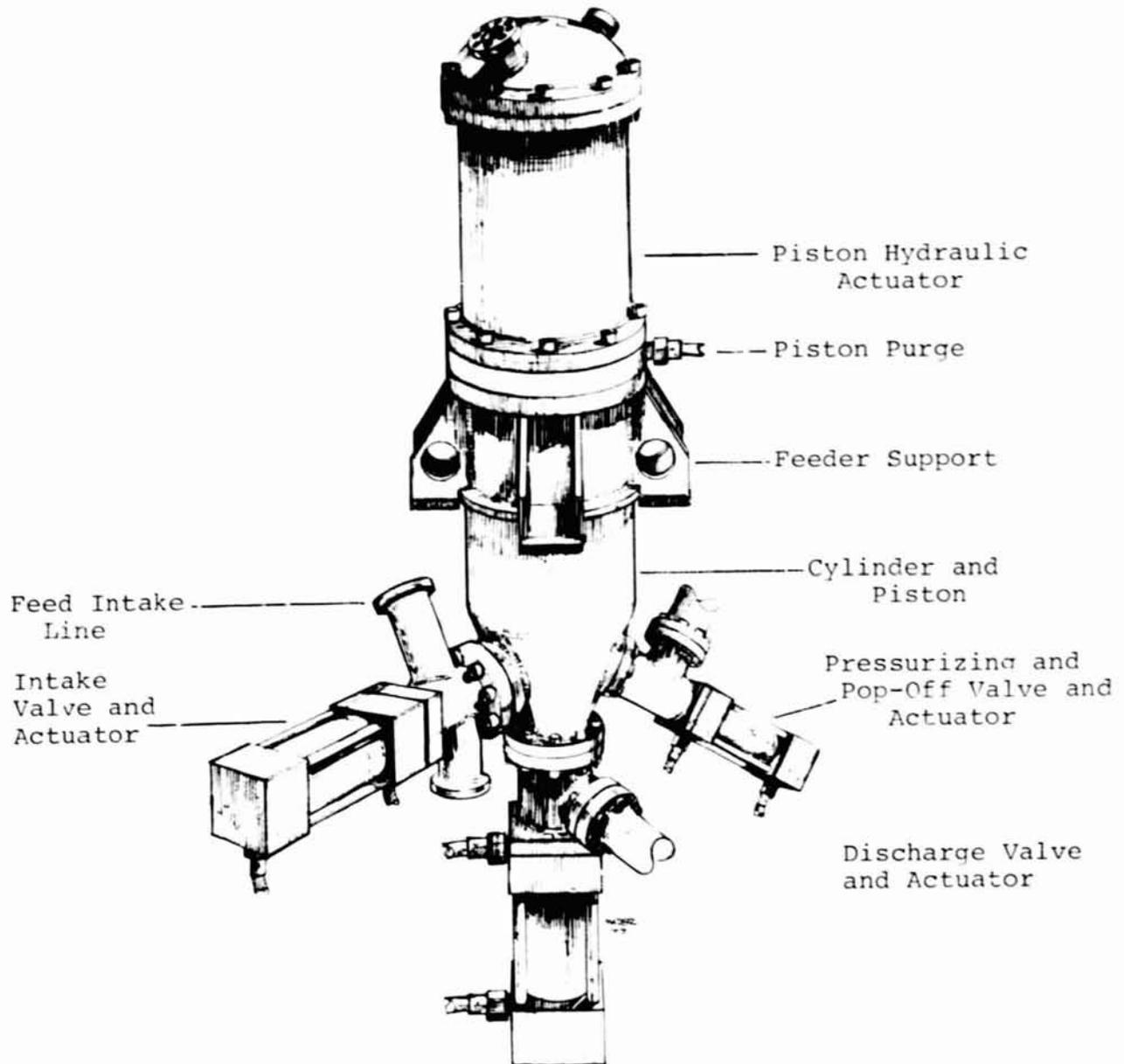
Figure 15. Fluidized Piston Feeder

Coal Feed

Flex Coupling

Vent

Gas Inlet

Downcomer

Head

Sprue

Gas Inlet

Coal/Gas Outlet

MADER '77
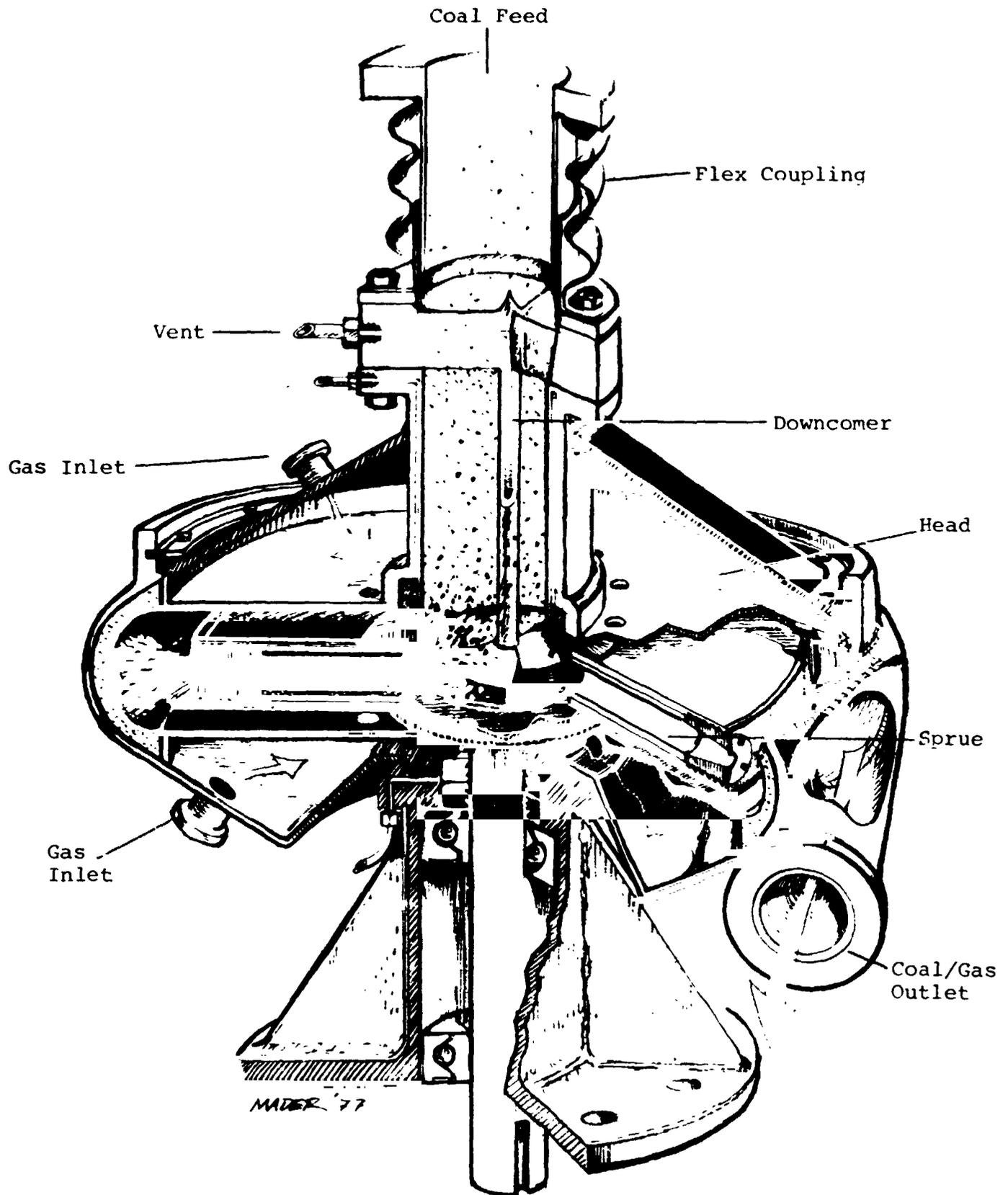
Figure 16. Centrifugal Feeder

REFERENCES

1. Coal Feeder Development. Phase I Report, Report FE-1793-10, March 1976

2. Coal Feeder Development. Phase II, Second Quarterly Technical Progress Report. For the Period 1 October 1975 to 30 June 1977, Report FE-1793-26, July 1977.

EVALUATION OF ERDA-SPONSORED COAL FEED SYSTEM DEVELOPMENT

R. L. Phen

W. K. Luckow

L. Mattson

D. Otth

P. Tsou

Jet Propulsion Laboratory
Pasadena, California

ABSTRACT

In March of 1977 the Jet Propulsion Laboratory began to provide staff

support to the ERDA coal feeder development program. An initial task in that

support effort was the evaluation of the coal feeders under development by

ERDA. The objective of the evaluation was to recommend to ERDA those coal

feed systems which should continue to receive development support as the pro-

gram progressed into the pilot-scale phase, and to recommend the development

actions to be undertaken for the selected feeders. The evaluation was based

upon criteria such as technical feasibility, performance (i.e. ability to

meet process requirements), projected life cycle costs and projected develop-

ment cost. An evaluation methodology was developed which incorporated the

evaluation criteria. Using this methodology, an initial set of feeders were

selected based on the feeders' cost savings potential compared with baseline

lockhopper systems. Additional feeders were considered for selection based

on: 1) increasing the probability of successful feeder development, 2) appli-

cation to specific processes and 3) technical merit. This paper presents the

results of the evaluation, lists the feeders recommended for continued devel-

opment and outlines a coal feeder development program.

INTRODUCTION

In response to the need for improved coal feeders, ERDA has sponsored a program of coal feed system development. Included in the program are feeder developments by three contractors: Foster-Miller Associates (FMA), Ingersoll-Rand Research, Inc. (IRR), and Lockheed Missiles and Space Company (LMSC). These contractors identified approximately a dozen feed system concepts which promised improved performance and reduced cost when compared with existing lockhopper and slurry pump coal feeders. Critical components and subsystems of these concepts are now being evaluated and tested by the contractors in preparation for a pilot-scale system demonstration effort which will begin about October 1977.

The objective of the JPL coal feed system evaluation is to recommend to ERDA those feed systems which should receive continued development support as the program proceeds into the pilot-scale phase and to identify those development actions which should be undertaken for each of the selected feeders.

The coal feed systems considered in the evaluation are listed in Table 1 which includes the development contractor; a brief description of the feeders, their characteristics and development status.

EVALUATION APPROACH

The criteria for the evaluation included:

- Technical feasibility
- Performance, i.e., ability to meet process requirements
- Projected life cycle costs
- Projected development risk and costs

Table 1. Coal Feed Systems

System	Developer	Schematic Drawing	Description	Pressure Limitations
• Positive Displacement	Foster-Miller		Cycled cavity piston fluidized coal feeder	1500 psi
• Centrifugal Feeder	Foster-Miller		Rotating centrifugal fluidized coal pump	1500 psi
• Linear Pocket Feeder	Foster-Miller		Tubular conveyor with coal conveyed to high pressure by a chain of interconnected pistons	1500 psi
• Screw Feeder	Ingersol-Rand		Type of auger which conveys coal axially down its length as the screw is rotated	1500 psi
• Single Acting Piston Feeder	Ingersol-Rand		Two coaxial delivery pistons operate in a common cylinder housing	1500 psi
• Rotary Valve Piston Feeder	Ingersol-Rand		Coal is transferred to high pressure by a piston sleeve rotation	1500 psi
• Kinetic Extruder Feeder	Lockheed		Rotating centrifugal fluidized coal pump	1000 psi (single stage) 1500 psi (two stages)
• Standpipe-Ball Conveyor Feeder	Lockheed		Standpipe filled with metal balls which conveys coal in the spaces between the balls	300 psi
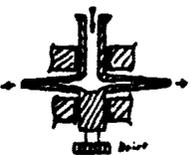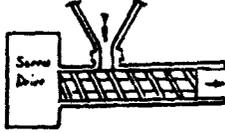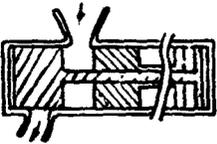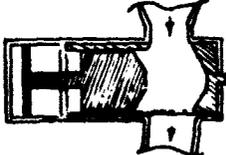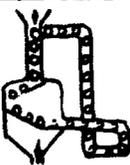
Table 1. Coal Feed Systems
(Continuation 1)

System	Coal Type, Size and Preparation Requirements	Development Status	Development Uncertainties
• Positive Displacement	• Any type • Size - fine/medium	Prototype in test	• Purging gas requirements may become large in large factors • Valve sequencing and sizing • Materials selection for seals and valve seats
• Centrifugal Feeder	• Any type • Size - fine	Prototype in test	• Pressure sealing dependent on coal properties • Sprue design uncertain • Feed throttling for control or throughput • Rotating seals
• Linear Pocket Feeder	• Any type • Size - medium/coarse	Prototype being assembled	• Incomplete filling generates back leakage and may limit pressure capability • Gas/liquid interface in water section • Wear and survival of rings and chain
• Screw Feeder	• Bituminous-agglomerating (for heated screw) • Size - up to 1" • Drying to 3-4% moisture	Prototype/pilot slave in test	• Possibly large power requirements • High pressure crusher to reduce extrudate to required size • Scale up of feeder with respect to heat input to coal • Screw/barrel wear • Operating parameters to provide throughput with minimum power
• Single Acting Piston Feeder	• Any type • Size - fine to coarse	Concept only	• Sealing and material wear • Purging coal from cavity • Coal jamming or piston/sleeve interface during loading and unloading
• Rotary Valve Piston Feeder	• Any type • Size - fine to coarse	Concept only	• Same as single acting piston feeder
• Kinetic Extruder Feeder	• Any type • Size - fine	Prototype in test	• Same as centrifugal feeder
• Standpipe-Ball Conveyor Feeder	• Any type • Size - fine to coarse	Bench tests	• Control of ball spacing and feeding mechanism • Purging gas out of U_2 feed line

Table 1. Coal Feed Systems
(Continuation 2)

System	Developer	Schematic Drawing	Description	Pressure Limitations
● Fluid Dynamic ●	Lockheed		Rotating bladeless turbine	2:1 pressure ratio per stage
● Gas-Solids Injector, Feeder	Lockheed		Gas-solids injector pump	2:1 pressure ratio per stage
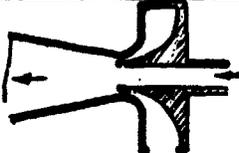
Table 1. Coal Feed Systems
(Continuation 3)

System	Coal Type, Size and Preparation Requirements	Development Status	Development Uncertainties
● Fluid Dynamic Lock Feeder	● Any type ● Size - find	Prototype tests	● Parasitic skin drag on disk requires high power ● Rotating face and bearing seals ● Coal flow through machine ● Wear on bearings, seals, disks
● Gas-Solids Injector, Feeder	● Any type ● Size - fine/medium	Prototype tests	● Wear in nozzle throat ● Compressor seals and bearings

Factors contributing to each feeders' relative capabilities in the above categories is shown in the methodology flow diagram given in Figure 1. The approach illustrated included the following steps:

(1) Analyze the technical feasibility of each feed system.

(2) Compare feeder performance capability vs feed system requirements.

(3) Determine feed system applicability to expected coal conversion processes.

(4) Evaluate expected feed system costs relative to baseline lock-hopper system.

(5) Select feed systems for future development which, from the cost analysis, show the best chance of achieving low cost and wide application to future processes, for specified R&D cost limitations.

(6) Consider recommending an expanded set of feeders as a means of increasing the probability of feed system commercialization.

(7) Examine specific applications as a reason for continuing development of a concept which was not otherwise selected.

(8) Review the feed systems selected on the basis of the cost analysis and modify this set based on the technical assessment.

DATA ACQUISITION

Data to accomplish the evaluation was obtained from the three feeder contractors and additional subcontractors as listed in Table 2. All data were analyzed by JPL for use in the evaluation.
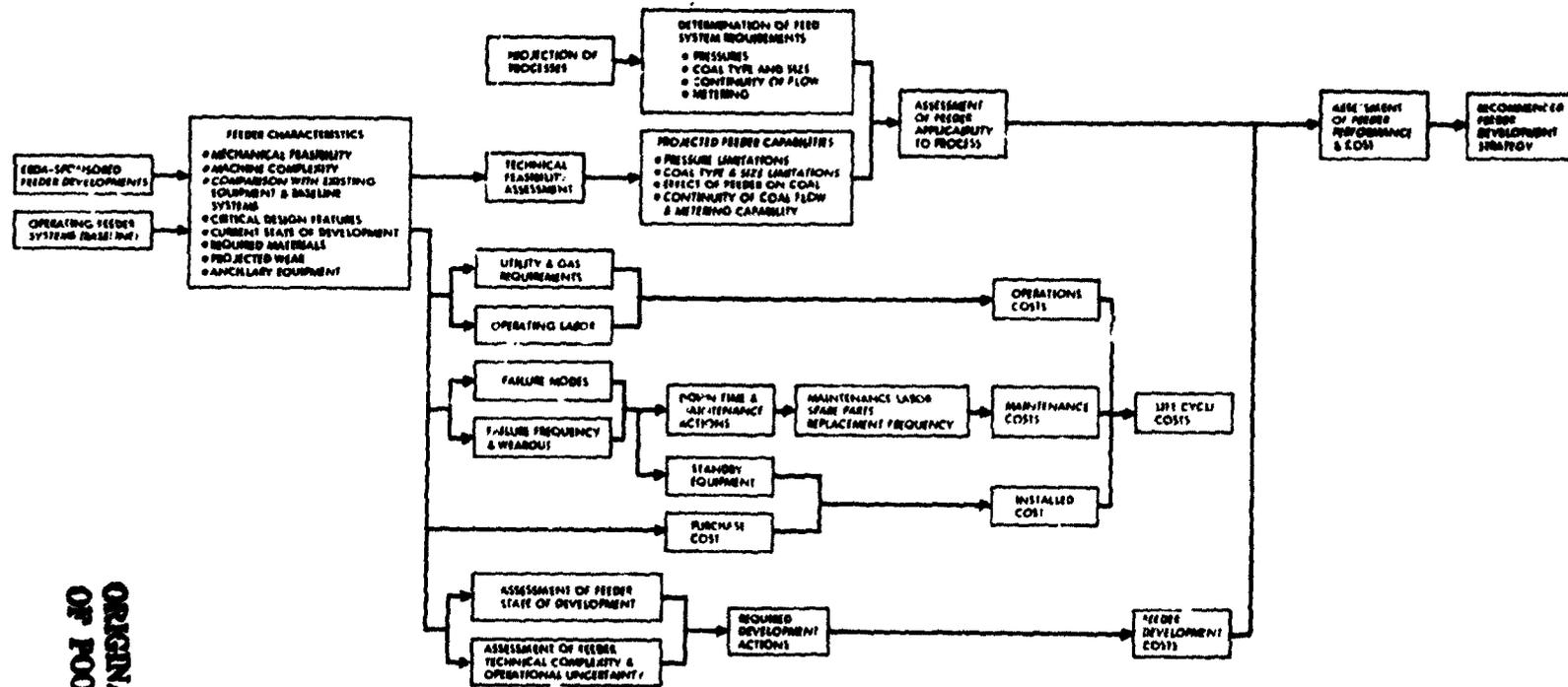
Figure 1. Coal Feed System Evaluation Methodology

FEED SYSTEM REQUIREMENTS

As a foundation for the coal feed systems development program, performance goals were established for the feed systems based upon the requirements of future coal conversion processes. The feed system requirements are the following:

- pressure – 150 to 1500 psi

- coal size – fines to coarse (2 inches)

  - the feeder should not affect coal size consist or

    properties, but should deliver coal as required to

    the process

- continuous flow should be provided

- coal metering capabilities are required

- lifetime – 20 years

The above requirements were developed by analysis of the conversion processes which were anticipated to achieve future commercialization. Further review of these processes enables classification of them into generic types based on their operating pressure and feed size consist.

The coal size and delivery pressure capabilities of the feed systems were matched against the generic requirements of the processes to establish the compatibility of the candidate feeders and the various conversion processes. Generic process conditions were determined by analysis of processes characteristics and are shown in Table 3. Application of the candidate feeders to the generic process conditions is shown in Table 4.

DEVELOPMENT UNCERTAINTY RANKING AND RELIABILITY

The development status and development problem areas have been used to estimate the commercialization potential for each feeder. The following

DATA	SOURCE [1]
TECHNICAL ASSESSMENT ● TECHNICAL DESCRIPTIONS ● STATE OF DEVELOPMENT ● DEVELOPMENT PROBLEM AREAS ● RELIABILITY	CONTRACTORS CONTRACTORS CONTRACTORS KAMAN SCIENCES
PROJECTED PERFORMANCE ● PRESSURE ● COAL TYPE & SIZE LIMITATIONS ● EFFECT OF FEEDER ON COAL	CONTRACTORS CONTRACTORS CONTRACTORS
PROCESS APPLICABILITY	JPL
PROCESS IMPACT	INTERNATIONAL SCIENCE AND TECHNOLOGY
PROJECTED COSTS ● INSTALLED ● OPERATING ● MAINTENANCE ● DEVELOPMENT	CONTRACTORS, ICARUS[2] CONTRACTORS, JPL CONTRACTORS, JPL[2] JPL
PROBABILITY OF SUCCESSFUL DEVELOPMENT	JPL

(1) IN ALL CASES DATA WAS ASSESSED BY JPL
(2) RELIABILITY DATA FROM KAMAN SCIENCES WAS USED TO DETERMINE STANDBY EQUIPMENT, SPARES, DOWNTIME, ETC.

Table 2. Data Sources

Table 3. Process Classification

Process	Size		Pressure					Remarks
	Lump	Pulver-ized	atm.	150	500	1000	1500	
HYGAS		X					X	
Lurgi	X				X			
Woodall-Duckham	X		X					
COGAS		X	X					
Texaco		X			X			
U-GAS		X			X			
AFBC		X	X					
SRC		X				X		Slurry feed
H-Coal		X						2250–2700 psi, slurry feed
Exxon Donner Solvent		X						2000 psi, slurry feed
BIGAS		X					X	Slurry feed
Synthane		X				X		
Mcdowell-Wellman	X				X			
Agglomeration Burner		X		X				
CO_2 Acceptor		X		X				
Synthoil		X						2–4000 psi, Slurry feed
AI Molten Salt		X		X				

Table 4. Feeder/Process Combinations

Feed System	Process							
	Lump			Pulverized				
	atm	150	500	atm	150	500	1000	1500
Positive Displacement Feeder	S	S	S	+	+	+	+	+
Centrifugal Feeder	S	S	S	+	+	+	+	+
Linear Pocket Feeder	+	+	+	+	+	+	P	P
Screw Feeder								
Heated	+	+	+	S	S	S	S	S
Unheated	+	+	+	+	+	+	+	+
Single Acting Piston Feeder	+	+	+	+	+	+	+	+
Rotary Valve Piston Feeder	+	+	+	+	+	+	+	+
Kinetic Extruder Feeder	S	S	S	+	+	+	+	+
Standpipe Ball Conveyor Feeder	S	S	S	+	+	P	P	P
Fluid Dynamic Lock Feeder	S	S	S	+	+	+	+	+
Gas-Solids Injector Feeder	S	S	S	+	+	+	+	+
Lockhopper	+	+	+	+	+	+	+	+
Slurry Pump	S	S	S	+	+	+	+	+

+ - Compatible feeder/process combinations

S - Incompatible feeder/process combinations due to feeder's inability to provide required coal size consist

P - Incompatible feeder/process combinations due to feeder's inability to feed to required pressure.

are the estimates of probability of successful commercialization assuming continued development.

Positive Displacement	0.80
Centrifugal	0.65
Linear Pocket	0.80
Screw	0.90
Single Acting Piston	0.75
Rotary Valve Piston	0.75
Kinetic Extruder	0.65
Standpipe-Ball Conveyor	0.60
Fluid Dynamic Lock	0.65
Gas-Solids Injector	0.85

Reliability analysis of the feeders were conducted by Kaman Sciences, Corp. The results of this analysis are summarized in Table 5. The table shows pertinent failure rate and availability data, and the number of redundant systems required per gasifier to achieve 95% availability. Note that the IRR screw feeder has the best reliability and that the FMA positive displacement pump has severe projected reliability problems due to its complexity and the large number of feeders required for a plant. It is important to note that the most significant contributors to most of the feed systems' unreliability were ancillary equipment. Therefore, feed system considerations should receive greater attention in the future development program.

Table 5. Summary of Feed-System Reliability Considerations[1]

Feed System	Failures in 10^6 Hours	Availability per Feeder or Train	Availability per Gasifier	Feeders or Trains per Gasifier	Required Backup Feeders or Trains per Gasifier for 95% Availability
Screw Feeder					
Unheated	1,290	0.96	0.85	4	1
Heated	1,490	0.96	0.85	4	1
Gas-Solids Injector	3,060	0.93	0.93	1	1
Ball Conveyor Feeder	3,710	0.92	0.85	2	1
Centrifugal Feeder	3,890	0.91	0.91	1	1
Kinetic Extruder Feeder	4,230	0.91	0.83	2	1
Single Acting Piston Feeder	4,800	0.90	0.66	4	2
Rotary Piston Feeder	4,850	0.89	0.63	4	2
Linear Pocket Feeder	7,530	0.85	0.61	3	2
Fluid Dynamic Lock Feeder	10,180	0 80	0.64	2	2
Positive Displacement Feeder	39,540	0.51	0.13	3	3 (2)
Lockhopper	5,120	0.89	0.79	2	2
Slurry Pump	2,430	0.95	0.90	2	1

(1) Data provided by Karran Sciences, Corp.

(2) Adding three additional feeder banks/gasifier would only increase the availability to 52%. Additional feeders required to achieve 95% availability were not determined.

Table 6. Technical Assessment Ranking

Development Uncertainty Ranking	Ability to Meet Requirements	Reliability Ranking	Overall Ranking
1. Screw	1. Single Acting and Rotary Piston	1. Screw	1. Screw
2. Gas-Solids Ejector	2. Screw	2. Gas-Solids Ejector	2. { Single Acting Piston / Rotary Piston
3. { Positive Displacement / Linear Pocket	3. { Positive Displacement Piston / Centrifugal/Kinetic Extruder / Linear Pocket	3. { Ball Conveyor / Centrifugal / Kinetic Extruder	3. { Centrifugal / Kinetic Extruder / Linear Pocket / Positive Displacement / Gas-Solids Ejector
4. { Single Acting Piston / Rotary Valve Piston	4. { Gas-Solids Ejector / Fluid Dynamic Lock	4. { Single Acting Piston / Rotary Valve Piston	4. Fluid Dynamic Lock
5. { Centrifugal / Kinetic Extruder / Fluid Dynamic Lock	5. Ball Conveyor	5. Linear Pocket	5. Ball Conveyor
		6. Fluid Dynamic Lock	
		7. Positive Displacement	

EVALUATION RESULTS

The feed systems were ranked preliminarily based upon the technical
factors:

- development uncertainty

- ability to meet requirements

- reliability

The ranking is shown in Table 6. The technical data, which was used to rank
the feeders was also incorporated in the cost analysis. The technical ranking
was used to check the results of the cost analysis to assure that feeders
selected on a cost basis included those with high technical ranking.

Cost analyses formed the foundation for the initial selection of feed
systems. Costs were provided by the three contractors and independently by
Icarus Corporation. The installed costs provided by the contractors and Icarus
were in good agreement, typically within 35% of each other for each feeder.
The evaluation reported here was based on the costs provided by the contrac-
tors. Sensitivity analyses have established that the same feeder selection
is obtained if the costs provided by Icarus are used.

Capital, operations and maintenance costs were used to calculate life
cycle costs for each feeder. These costs are shown in Tables 7-10 for vari-
ous reactor pressures.

Development costs were determined by assessment of the feeder develop-
ment status. The assessment is summarized in Table 11 and the costs are sum-
marized in Table 12.

Using the capital, installation, operating and maintenance costs given
in Tables 7-10, life cycle costs were calculated for each feed system. Cost
savings, ΔC, for individual feeders and for feeder sets compared with the

Table 7. Cost ($1,000 1977 Dollars) Summary for Commercial Plant
625 TPH Throughput 150 psi Pressure

Feeder	R&D	Installed	Operating		Maintenance		Number of Banks	TPH/ Bank	Stage/ Bank	No. Feeders
			Labor	Utility	Labor	Mat'l				
Foster Miller										
Positive Displacement		5,740	240	75	14.4	250	9	70	14*	126*
Linear Pocket		2,151	250	240	15	150	9	70	1	9
Centrifugal		1,971	160	138	15	36	3	210	1	3
Ingersol Rand										
Heated Screw		5,533	350	1,180	605	573	12	52	1	12
Cold Screw		4,646	350	1,180	605	481	12	52	1	12
Single Acting Piston		4,210	350	89	605	435	12	52	1	12
Rotary Valve Piston		2,835	350	89	605	293	12	52	1	12
Lockheed										
Kinetic Extruder		4,864	240	182	39	447	6	104	1	6
Standpipe Ball		11,520	350	383	230	922	6	104	1	6
Fluid Dynamic Lock		13,360	240	2,479	120	1,216	6	104	4	24
Injector		7,931	240	2,043	32	761	3	210	2	6
Lockhopper		4,080	350	271	147	98	6	104	1	6
Slurry Pump		4,675	350	7,642	168	112	6	104	1	6

*Number of cylinders per bank and feeder.

Table 8. Cost ($1,000 1977 Dollars) Summary for Commercial Plant
625 TPH Throughput 500 psi Pressure

Feeder	R&D	Installed	Operating		Maintenance		Number of Banks	TPH/ Bank	Stage/ Bank	No Feeders
			Labor	Utility	Labor	Mat'l				
Foster Miller										
Positive Displacement		6,820	240	240	17	340	9	70	14*	126*
Linear Pocket		2,150	250	833	15	150	9	70	1	9
Centrifugal		2,358	160	405	15	40	3	210	1	3
Ingersol Rand										
Heated Screw		6,577	350	1,593	605	680	12	52	1	12
Cold Screw		5,690	350	1,593	605	589	12	52	1	12
Single Acting Piston		5,613	350	295	605	589	12	52	1	12
Rotary Valve Piston		4,033	350	295	605	391	12	52	1	12
Lockheed										
Kinetic Extender		6,043	240	456	49	556	6	1C4	1	6
Sandpipe Ball										
Fluid Dynamic Lock		17,344	350	4,185	156	1,578	6	104	6	36
Injector		9,632	240	2,859	39	925	3	210	4	12
Lockheed		5,316	350	1,355	191	128	6	104	1	6
Slurry		4,791	350	7,976	172	114	5	104	1	6

*Number of cylinders per bank and feeder.

Table 9. Cost ($1,000 1977 Dollars) Summary for Commercial Plant
625 TPH Throughput 1000 psi Pressure

Feeder	R&D	Installed	Operating		Maintenace		Number of Banks	TPH/ Bank	Stage/ Bank	No. Feeders
			Labor	Utility	Labor	Mat'l				
Foster Miller										
Positive Displacement		8,140	240	851	17	343	9	70	14*	126*
Linear Pocket										
Centrifugal		3,355	160	788	15	178	3	210	1	3
Ingersol Rand										
Heated Screw		8,561	350	2,154	605	886	12	52	1	12
Cold Screw		8,053	350	2,154	605	833	12	52	1	12
Single Acting Piston		8,419	350	591	605	871	12	52	1	12
Rotary Valve Piston		5,670	350	591	605	587	12	52	1	12
Lockheed										
Kinetic Extruder		7,727	240	848	61,895	711	6	104	1	6
Standpipe Ball										
Fluid Dynamic Lock		23,039	350	6,622	207	2,097	6	104	7	42
Injector		12,062	240	4,025	48	1,158	3	210	6	18
Lockhopper		7,074	350	3,388	255	170	6	104	1	6
Slurry		4,912	350	8,603	177	11^	6	104	1	6

*Number of cylinders per bank and feeder.

Table 10. Cost ($1,000 1977 Dollars) Summary for Commercial Plant
625 TPH Throughput 1500 psi Pressure

Feeder	R&D	Installed	Operating		Maintenance		Number of Banks	TPH/ Bank	Stage/ Bank	No. Feeders
			Labor	Utility	Labor	Mat'l				
Foster Miller										
Positive Displacement		8,140	240	851	17	343	9	70	14*	126*
Linear Pocket										
Centrifugal		3,355	160	788	15	178	3	210	1	3
Ingersol Rand										
Heated Screw		11,077	350	2,950	605	1,146	12	52	1	12
Cold Screw		11,074	350	2,950	605	1,146	12	52	1	12
Single Acting Piston		13,217	350	886	605	1,452	12	52	1	12
Rotary Valve Piston		9,450	350	886	605	816	12	52	1	12
Lockheed										
Kinetic Extruder		9,409	350	1,239	75	866	6	104	2	12
Standpipe Ball										
Fluid Dynamic Lock		28,728	350	9,060	259	2,614	6	104	7	42
Injector		14,250	350	5,191	57	1,368	3	210	7	21
Lockhopper		8,771	350	6,030	316	211	6	104	1	6
Slurry		5,797	350	8,923	209	139	6	104	1	6

*Number of cylinders per bank and feeder.

Table 11. Coal Feeder Development Assessment

Feeder	Development Status	Scale-ability	Machine Complexity*	Development Risk
Ball Conveyor	Bench Tests	Poor	C	High
Kinetic Extruder	Proto in Test	Poor	S	High
Fluid Dynamic Lock	Proto in Test	Poor	S	High
Ejector	Proto in Test	Good	S	Low
Centrifugal	Proto in Test	Poor	S	High
Positive Displacement Piston	Proto in Test	Good	A	Low
Linear Pocket Feeder	Proto being assembled	Good	C	Low
Screw	Proto/Pilot Sizes in Test	Poor	S	Low
Single Acting Piston	Paper Concept	Good	A	Low
Rotary Piston	Paper Concept	Good	A	Low

*S - Simple
A - Average
C - Complex

baseline lockhopper systems were determined. The following three parameters were then used to select the most promising feeders.

$\Sigma\Delta C$ - The life cycle cost difference between the candidate feeder and the baseline (lockhopper) feeder summed over the process applications. A maximum value of this parameter represents the objective of the plant developer who seeks to minimize costs.

Table 12. Estimate Feed System Development Costs

Feed System	Relative Development Costs (Million $)		
	Phase III	Phase IV	Total[1]
Positive Displacement Feeder	1.3	3.3	8.0
Centrifugal Feeder	0.6	2.2	2.3
Linear Pocket Feeder	0.5	1.4	1.6
Screw Feeder	1.5	6.1 (heated) 5.8 (unheated)	6.4 6.1
Single Acting Piston Feeder	2.2	8.7	9.1
Rotary Valve Piston Feeder	1.6	6.2	6.5
Kinetic Extruder	1.4	5.5	5.8
Standpipe Ball Conveyor	4.0	15.7	16.5
Fluid Dynamic Lock Feeder[2]	4.2	16.8	17.6
Gas-Solids Injector[2]	1.4	5.6	5.8

[1] Constant dollars

[2] Staged systems

L - Cost leverage = $\Sigma\Delta C$/development costs. A maximum value of this parameter represents the goals of ERDA which seeks the maximum return for its development funding.*

R - Realizability. The probability of successful commercialization.

Figure 2 shows how these three parameters change with increased development funding, and with different select of feeder sets. All combinations

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*Note that the values of L show relative differences between systems. The actual value of L may be 10-50 times the number shown depending on how many plants derive economic benefit from use of the new feeder/gasifier systems.
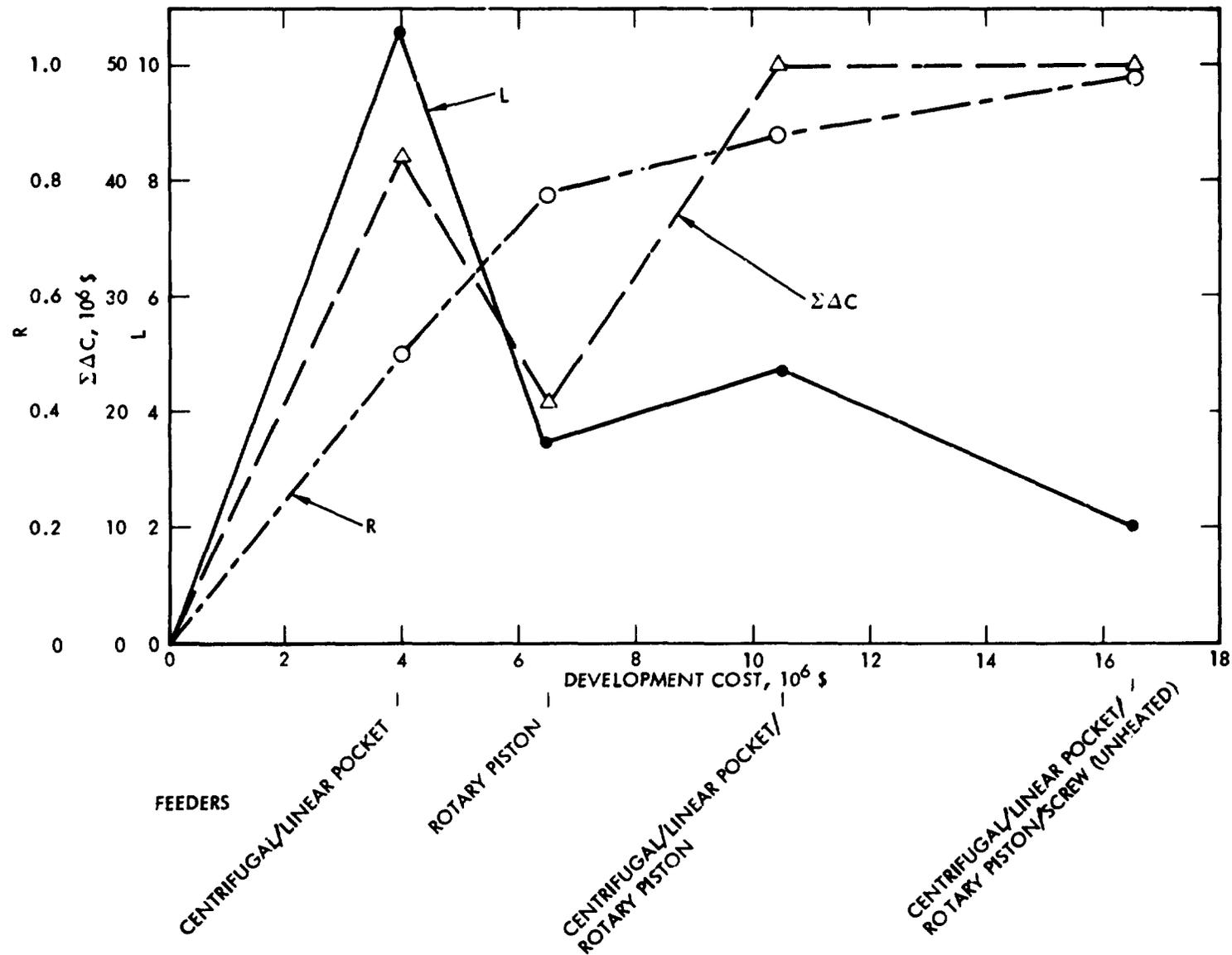
Figure 2.   Feed System Selection Parameters

of feeder sets which could meet all process conditions were examined. Figure 2 shows the most promising combinations. The sets shown provide the best choice, i.e., they optimize one or all of the three decision parameters for the range in development costs. The figure illustrates the following:

(1) The feeder set which maximizes L is the centrifugal (or kinetic extruder) and linear pocket feeder. This set also provides a high value for $\Sigma\Delta C$. However, there would be a high risk that these feeders would not realize commercialization (low R).

(2) The rotary valve piston feeder is predicted to have a higher probability of commercialization than the combination of the centrifugal and linear pocket feeder, but its predicted high life cycle and development costs result in lower $\Sigma\Delta C$ and L values. Actually, considering cost inaccuracies, the rotary piston and the set of centrifugal/linear pocket feeders probably have comparable values for $\Sigma\Delta C$ and L.

(3) Because of the low values for R which would result if only one feeder or feeder set was developed, it is recommended that parallel developments be undertaken to increase the probability of feed system commercialization. Parallel development of feed systems will reduce the parameter L as shown in the figure, because development costs are increasing faster than corresponding increases in cost savings, $\Sigma\Delta C$. By combining the centrifugal/linear pocket and rotary piston feeders, increased realizability is achieved; but it is not until a third parallel development, the unheated screw, is added that an acceptably high value for R is achieved.

(4) The positive displacement feeder, if added to the above set, would only slightly increase the commercialization realizability, but would increase the development cost by about 30%. The additional cost for little gain, coupled with the feeder's projected low reliability, leads to the recommendation that development of the positive displacement feeder be discontinued, or limited to testing of the present system and concentration on improving the system's reliability.

(5) None of the other feeder systems offer any additional cost or realizability advantages over the four selected in (3) above. Additionally, none of the other feeders was determined to have advantages for specific applications or redeeming technical features which would recommend its selection.

RECOMMENDED SYSTEMS

As a result of the above analysis the following feed systems are recommended for further development:

FMA centrifugal feeder or LMSC kinetic extruder

FMA linear pocket feeder

IRR rotary valve piston feeder

IRR unheated screw feeder

The recommended actions for each feeder and the bases for these recommendations are summarized in Table 13 and detailed in the Coal Feed System Development Plan, JPL Report No. 5030-94. For all selected feeders the development uncertainty is high. Continued evaluation of the selected concepts is required and is reflected in the Development Plan.

Table 12. Recommended Coal Feed System Development Actions

Feed System	Recommended Action	Basis for Recommendation
Positive Displacement Feeder	Discontinue major development effort. Limited testing of available equipment and design analysis to verify cost and reliability assessment.	• No cost advantage relative to selected systems • Serious reliability problem
Centrifugal/Kinetic Extruder Feeder	Continue component testing to verify concept functional capability, and pressure ratio potential.[1]	• Potential low cost system for high pressure processes using fine coal. • System simplicity
Linear Pocket Feeder	Conduct pilot-scale development. Assess sealing, leakage and pressure capability. Verify coal metering to the pockets and water lock design.[2]	• Potential low cost system for low pressure systems (to 500 psi) using fine to coarse coal
Screw Feeder	Conduct pilot scale development. Emphasize the unheated screw.[3]	• Provides parallel development alternative to other recommended developments to increase probability of commercial feed system development. • One of only two feeders capable of meeting all process requirements (piston feeders are only in conceptual stage of development).
Single Acting Piston Feeder	Discontinue development efforts in favor of rotary piston feeder development.	• Cost savings potential is not as great as rotary piston. Development problems may be easier, however.
Rotary Valve Piston Feeder	Conduct component development, emphasizing piston sealing and wear, solids loading and unloading to prevent jamming and system design to minimize power requirements.	• Potential cost savings compared to baseline. • Potential application to all process requirements.
Standpipe Ball Conveyor Feeder	Discontinue development.	• Very complex • High cost compared to baseline systems. • Applicable only to low pressures (below 150 psi)
Fluid Dynamic Lock Feeder	Discontinue Development	• Complex staging required to reach even 150 psi. • High cost compared to baseline systems.
Gas-Solids Injector Feeder	Discontinue development[4]	• Complex staging required to reach even 150 psi. • High cost compared to baseline systems.

[1] Because of development uncertainties parallel development efforts should be considered.

[2] Recommendation contingent on results of prototype testing results.

[3] This system has questionable cost advantages. Requires application analysis during Phase III to determine best applications.

[4] This system should be analyzed for application to low pressure systems.

The reliability assessment performance by Kaman Sciences pinpointed the ancillary equipments as the critical elements in regard to feed system reliability. Therefore, system aspects should receive greater attention in the continuing program.

The process impact study conducted in conjunction with International Science and Technology revealed the potential sensitivity of the processes to feeder characteristics. These results emphasize the need to view the feeder as but one equipment of an integrated coal conversion plant.

FUTURE FEED SYSTEM DEVELOPMENT

The coal feed system development program has the objective to provide the coal conversion process plant designer several feeder options which could result in technical advantages and cost savings over conventional lockhopper and slurry pump systems. Basic to the feeder development are the program elements of strategy which include:

(1) Maintaining open options - by continuing with parallel feeder development programs to increase the probability of successful development, and providing for the development of new concepts if they have advantages over other systems being developed.

(2) Involve decision makers - such as architect/engineering firms, utilities, and process developers in the pilot and demonstration phases, to assure that the feeders are tested against real process requirements and that the results will be rapidly disseminated throughout the industry.

(3) Component testing and resolution of common problems - by centralized testing to avoid duplication of effort.

(4) Utilization of process pilot plants for testing to demonstrate process compatibility. The schedule of Figure 3 reveals that some of the pilot plant processes will have completed process demonstration and be available for component tests at the time pilot scale feeders become available for test.

(5) Centralized demonstration test facility is suggested for duration testing of demonstration scale feeders. Plant designers will require such testing before they will commit to the incorporation of the feeders in demonstration or commercial plants.

(6) Feeder integration into process demonstration plants is required as a commercialization step, yet their development schedule is lagging the demonstration plant schedule as shown in Figure 3. Late introduction of feeders into the demonstration plants, or introduction into second generation plants, should be considered.

(7) Cost sharing by the contractors during the demonstration phase is recommended to stimulate contractor interest and introduce marketing considerations into the program.

The feed system evaluation and the strategic elements provided the basis for the plan which is summarized in the schedule of Figure 4. Shown in the figure are schedules for the specific feeder developments and related support tasks. Key features of the program illustrated by the schedule are the following:

- Component development of the centrifugal and kinetic extruder feeders is shown continuing in parallel until a better understanding of the concept is obtained. Then a single pilot plant-scale effort is recommended.

Figure 3.   Overall Feeder/Process Schedule

77-55

MILESTONES	FY 77	FY 78	FY 79	FY 80	FY 81	FY 82	FY 83

1. FEEDER DEVELOPMENTS
2.   KINETIC EXTRUDER — COMPONENT / PILOT
3.   CENTRIFUGAL
4.   LINEAR POCKET — COMPONENT / PILOT
5.   SCREW — PILOT
6.   ROTARY PISTON (OR SINGLE-
7.   ACTING PISTON) — COMPONENT / PILOT
8.   POSITIVE DISPLACEMENT — RELIABILITY / PILOT
9.   EJECTOR — PILOT
10.   OTHER FEED SYSTEMS
11.   SYSTEM DEVELOPMENT
12.
13. SUPPORT TASKS
14.   FEEDER REQUIREMENTS
15.   PROCESS INTERACTIONS
16.   COMMON COMPONENT TESTING
17.   SYSTEMS ANALYSIS AND
18.   EVALUATION
19.   AD HOC ACTIVITIES
20. FEEDER TEST FACILITY — DESIGN / CONSTRUCTION
21.
22.
23.

LEGEND:
△ BEGIN ACTIVITY
▽ COMPLETE ACTIVITY
▬▬▬ ACTIVITY TO BE DETERMINED

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Figure 4. Coal Feed Development Schedule

- Pilot-scale development of the linear pocket feeder is recommended if prototype testing is successful. The development effort should concentrate on seal effectiveness and life, metering of coal to the pockets, seal tube life and the effectiveness of the water lock or gas-water transfer subsystem.

- Continued pilot scale testing of the screw feeder is recommended with emphasis on the unheated design.

- Accelerated component testing of the rotary piston feeder is recommended, followed immediately by pilot-scale development (if component testing is successful). This will permit completion of the pilot phase of the development on a schedule consistent with the other feed system developments.

- It is the intent to reduce the number of feeders under development to a minimum set upon completion of the pilot scale phase of the program. Therefore, demonstration efforts for each of the feed systems are shown "to be determined" after the pilot phase.

- The positive displacement feeder is shown undergoing continued testing to obtain a better understanding of the capabilities, projected costs and reliability. A decision to proceed into pilot scale development will be made when the capabilities of the centrifugal/kinetic extruder feeders are determined by the component tests. If the tests of these two feeders are successful, it will be recommended that the development of the positive displacement feeder, which is a backup system, be discontinued.

- The ejector is shown subject to applications analysis prior to pilot scale development. Pilot plant development will only be recommended if special applications are found.

- Other feed systems development will be undertaken when promising feeder concepts are identified.

- Feeder systems development is recommended to be conducted in conjunction with the specific feeder development efforts.

- Support tasks will be performed, as required, to guide the development efforts.

- The need for a demonstration-scale feeder test facility will be analyzed. If the facility is needed, design and construction will follow.

SESSION III

ADVANCED CONCEPTS, EXPERIMENTS

CONTINOUS HIGH PRESSURE LUMP COAL
FEEDER DESIGN STUDY

S. F. Fields

GARD, Inc.

N   . Illinois

CONTINUOUS HIGH PRESSURE LUMP COAL FEEDER DESIGN STUDY

S. F. Fields
GARD, INC., Niles, Illinois

The purpose of this project was to try to develop a continuous lump coal dry feeder for a pressurized fluidized bed combustor. The approach was to adapt the commercially available Fuller-Kinyon pump to feed coal against a pressure differential of 100 psi or more. The pump was modified and tests performed at various pressure differentials, with differently pitched screws, various screw rotational speeds, and various seal lengths and configurations.

Successful operation of the modified Fuller-Kinyon pump was generally limited to pressure differentials of 60 psi or less. Although the results of this project are not conclusive, test data and observations were made that indicated that higher pressure differentials could be attained by further modifications of the test setup. In particular, it is recommended that further testing be performed after replacing the 40-horsepower pump motor presently in the test setup with a motor having a significantly higher power rating (thereby allowing pump operation with longer seals and at higher pressure differentials than those tested so far).

PURPOSE OF PROJECT

In the process of fluidized bed combustion of coal, if the fluidized bed combustor is operated suitably pressurized, overall power generating efficiency can be increased by expansion of the hot flue gases through a gas turbine in a combined cycle plant. However, in this case, there is a need for equipment to continuously and reliably feed large quantities of coal and dolomite into the pressurized combustor.

A screw-type feeder such as that manufactured by the Fuller Company for the low pressure differentials used in pneumatic conveying systems in the cement indu'ry (the Fuller-Kinyon pump) appears to offer potential for this application provided the pump can be modified and upgraded to meet system and process requirements. Thus the general purpose of this project has been to establish the feasibility of using a screw-type feeder to feed lump coal and limestone continuously into an appropriately pressurized vessel. Specifically. the possibility of adapting the existing design of the Fuller-Kinyon pump to this application has been investigated.

DESCRIPTION OF STANDARD FULLER-KINYON PUMP

The Fuller-Kinyon pump is used extensively in pneumatic conveying systems for injecting materials into pressurized pneumatic conveying lines. In normal applications, the materials conveyed are dry, pulverized, and free-flowing, and have a fineness of at least 100% passing 50 mesh, 75% passing 100 mesh, 60% passing 200 mesh, and 45% passing 325 mesh screens. The Fuller-Kinyon pump has operated with such materials at pressure differentials as high as 50 psi. It is currently available in 7 standard sizes with capacities ranging from 8 to 200 tons per hour (in terms of portland cement).

The Fuller-Kinyon pump is shown in Figure 1 (cut-away view) and Figure 2 (general assembly drawing - exploded view). The construction and operation of the pump for pneumatic conveying system applications are briefly summarized in the following (items followed by numbers in parentheses are correspondingly called out in Figure 1).

All parts are mounted on a cast iron base (1). The material to be conveyed enters the hopper (2) by gravity from the sources of supply, and is advanced through the barrel (3) by the impeller screw (4), the latter being directly driven through a flexible coupling connected to the driving motor. (The barrel is protected by renewable wear-resistant liners, and the screw flights are also protected with a special alloy to give a maximum of service.) As material advances through the barrel, it is compacted by the decreasing pitch of the impeller screw flights, and is further increased in density by the space or "seal" between the terminal flight of the impeller screw and face of the check valve disc (5). The exact density required is further controlled by an adjustment of the seal length by means of the screw jacks (6). (When there is no material in the hopper, the check valve prevents the flow of gas back into the barrel and screw.) The material then enters the check valve body or mixing chamber (7) wherein it is fluidized by compressed air introduced through a series of air jets (8) and from there enters the transport line.

The pump impeller screw is positioned in a hollow shaft (9) which is supported by ball bearings (10 and 11) in a single bearing housing (12). The supporting shaft is rotated through an eccentric lock collar (13). There are no packing glands. The material in the hopper is sealed from the bearings when the pump is in operation and under pressure by means of a Graphitar air-cooled, fan-type seal ring (15), in the chamber (16). The seal ring is kept clear of material from the hopper by means of compressed air supplied through air piping (17) from the header (18). Finally, the hopper section has a catch basin on the bottom and a clean-out door (19) on each side.
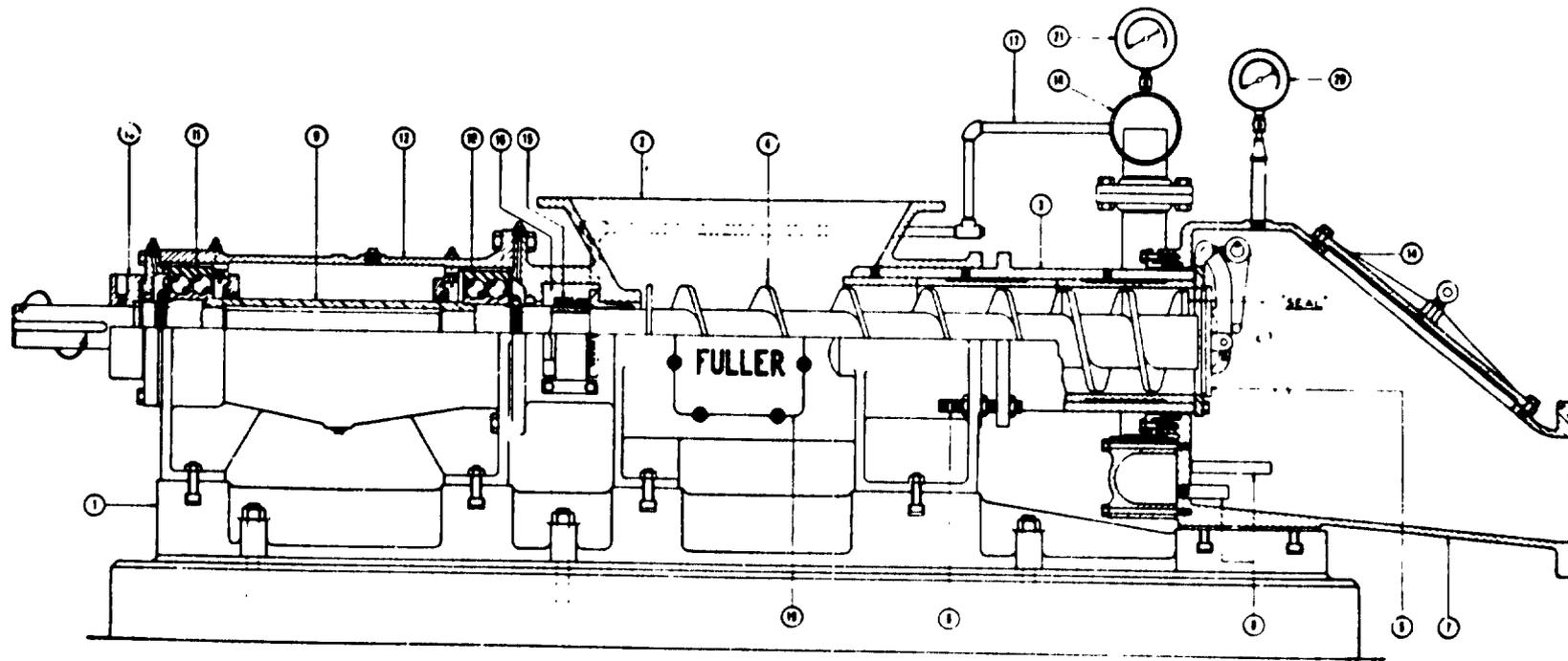
FULLER

Figure 1 Fuller-Kinyon Pump
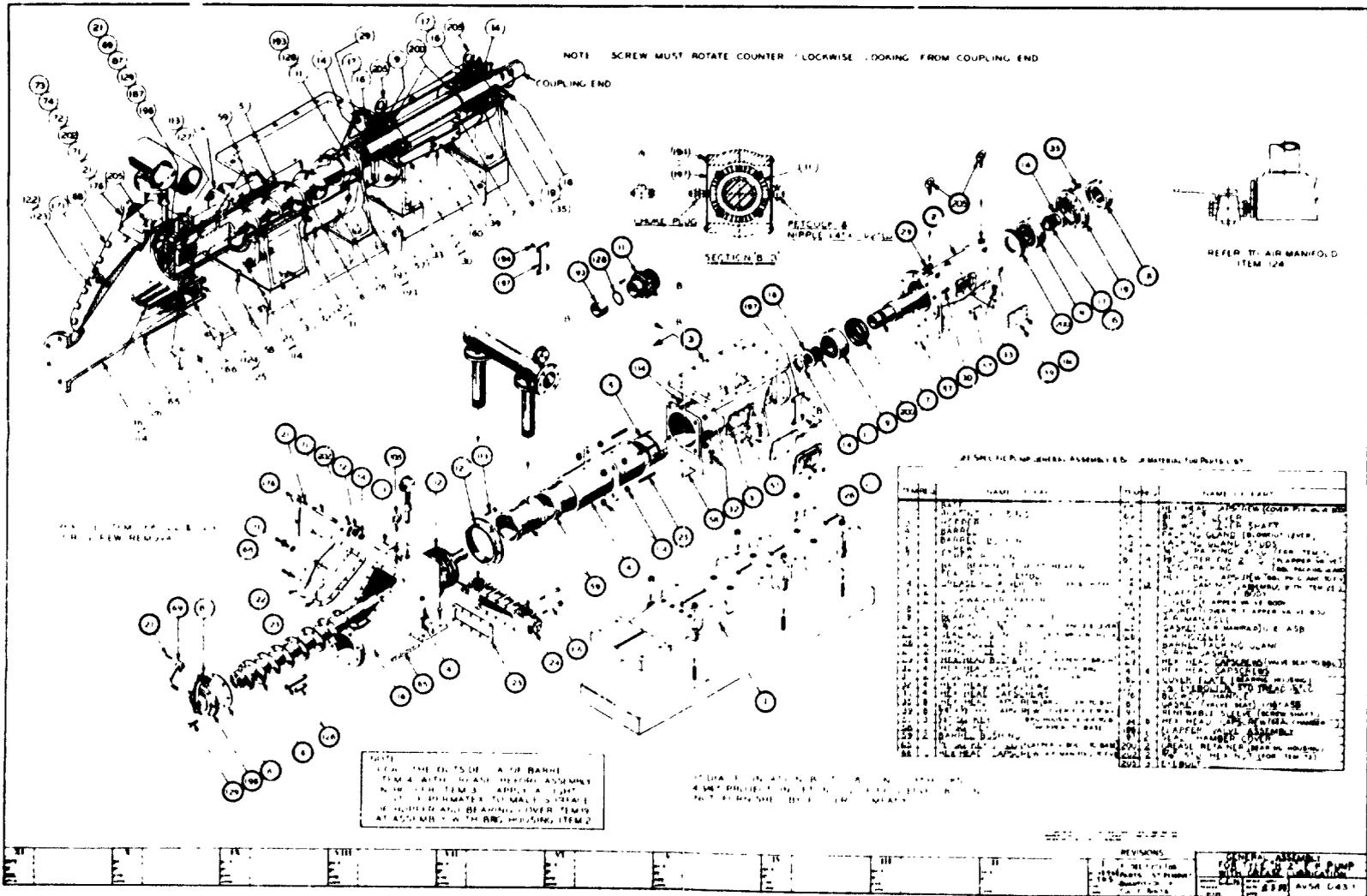
Figure 2 General Assembly for Type "H-2" Fuller-Kinyon Pump

DESCRIPTION OF THE DESIGN CONCEPT INVESTIGATED

The major modification to the Fuller-Kinyon pump investigated during this project involved: 1) removal of the check valve, the mixing chamber, and the air jets from the output end of the pump, and 2) extension of the barrel (in various configurations) beyond the terminating flight of the screw to provide additional length of extruded material for sealing purposes.

The object of this modification was to preserve the material transfer capability of the Fuller-Kinyon pump while at the same time obtaining, for materials much coarser than normally handled by the pump, a sufficient length of extruded material to provide, together with the material contained in the screw, an acceptable and reliable seal against pressure differentials much higher than those normally imposed on the pump. It appeared that with this modification other alterations in the existing design of the Fuller-Kinyon pump might not be required.

BACKGROUND WORK

Prior to testing of the design concept, considerable effort was expended: 1) analyzing the available background data on the Fuller-Kinyon pump to determine scaling relationships and the effects on pump performance of variation of such parameters as type of feed material, screw rotational speed, screw pitch, and screw compression ratio; 2) analyzing the mechanics associated with forced motion of a slug of granular material in a tube (as it relates to the concept of screw-type feeder operation involving downstream pressure sealing by a continuously extruded slug of feed material); and 3) making estimates of the various types of power consumption associated with the Fuller-Kinyon pump. (All of these analytical efforts are summarized in detail in Reference 1.)

Item 1 was aimed primarily at providing a basis for defining the tests to be conducted later in the project with a modified Fuller-Kinyon pump. This analysis led to the following conclusions. First, constant-pitch screws should be used in order to avoid possible material compression problems due to feed material non-uniformity and poor compaction characteristics. Second, values of the initial screw helix angle (the angle whose tangent is the ratio of the initial screw pitch to π times the screw outer diameter) should be chosen which

appeared likely, on the basis of available data, to maximize pump volumetric efficiency for the feed materials of interest. Third, tests should be run over a significant range of obtainable screw rotational speed in order to establish an output-versus-rpm characteristic curve for the feed materials of interest. (This data would of course be of particular interest when considering applications requiring a variable feed rate capability.)

Item 2 was directed toward aiding in the design of seal configurations which could provide the required seal without generating excessive loads on the pump. Based on this analysis, it was concluded that it would be most advantageous to conduct tests with a seal enclosure first consisting of a tube of constant cross-section (having a length somewhat less than the critical length for jamming) and then perhaps two somewhat different divergent, truncated cones.

Item 3 was intended to provide a basis for estimating power consumption with the Fuller-Kinyon pump for the application under investigation. Estimates were m e for the pump of the maximum rate of doing work: 1) against a pressure differential, 2) in imparting axial motion to the feed material, 3) in imparting rotational motion to the feed material, and 4) associated with feed material shear at the terminating flight of the screw. (No method was established to estimate the power consumed by friction occurring within the barrel of the pump.) These estimates indicated a power requirement f the 4-inch diameter Fuller-Kinyon pump (excluding the power consumed by friction occurring within the barrel of the pump) of up to about 30 horsepower f r operation with an effective seal against a gas pressure of 100 psig.

Certain required materials properties tests were also performed. Some of these were required to complement the analytical results. For example, the analysis under item 2 required a determination of the length of feed material moving in a smooth pipe at which the phenomenon of jamming occurs. (This defines the maximum obtainable seal length for feed material in a smooth pipe.) Other tests were required to determine, for example, leak rate as a function of the length of feed material in a column and the pressure differential across the column. (These and other materials properties tests are summarized in detail in Reference 1.)

Three basic conclusions were made based on the results of these tests. First, only very limited compaction of the materials of interest (corresponding to a 12% reduction in volume) can occur before the onset of actual mechanical compression. Second, reasonably long material slug lengths can be moved in tubes of constant circular cross-section. For example, "free" movement of slugs of material up to a length-to-diameter ratio of approximately 7 can be expected for a mixture of typical moist coal and limestone (4:1 by weight), and "free" movement of considerably longer slugs of such material can be realized in slightly divergent truncated cones. Third, relatively long material slug lengths appear to be required to obtain sufficient sealing against the pressure differentials of interest on this project. (It should be noted here that, for the materials tested, slugs of the dry materials jammed at shorter length-to-diameter ratios but required less length to achieve a given leak rate than did slugs of the moist materials.)

With respect to the third conclusion, the actual slug length (of extruded feed material) required with a modified Fuller-Kinyon pump is of course highly dependent on the amount of feed material contained within (and the degree of "filling" in) the pump barrel and screw, the compaction of the feed material as it advances along the screw and in the seal, and the feed material velocity through the pump and seal. Thus, the effectiveness of the seal provided by a Fuller-Kinyon pump operating with an extruded feed material seal can only be accurately determined by testing with the pump itself.

FULLER-KINYON PUMP TESTS

In order to develop the design concept, establish its performance characteristics, and evaluate it in terms of feasibility, approximately 150 tests were pe formed during this project with a modified 4-inch diameter Fuller-Kinyon pump. The 4 n diameter Fuller-Kinyon pump, which is the smallest such pump marketed, was chosen primarily to satisfy restrictions in the laboratory with regard to power availability, receiver pressure vessel size (as determined by pump output and desired test time), available coal store e space for the test program, and equipment availability. In pneumatic conveying system applications, this pump has a rated capacity of up to 8 tons per hour for Type 1 cement. It was felt that test results obtained with this pump could be scaled to the 100-ton-per-hour range with reasonable accuracy.

Tests were conducted primarily with three essentially non-compressing screws. One was a 4-1/4" constant-pitch screw (referred to as a 4-1/4" x 4-1/4" pitch screw). The second was a 3-1/4" x 3" pitch screw which offered a significantly reduced screw helix angle in comparison with that of the first screw. (The pitch of all flights on this screw was 3-1/4 inches with the exception of a 3-inch pitch for the last flight.) The third was a 1-1/2" constant-pitch screw (referred to as a 1-1/2" x 1-1/2" pitch screw). Otherwise, the three screws were similar, having an overall length of the flights of 27-3/8 inches, a flight thickness of 1/2 inch, and a shaft diameter of 2-1/8 inches.

The first two screws were slightly modified for the test program with the addition of a "terminal taper" at the output end of the screw to prevent the formation of a void region in the extruded feed material seal. For each screw, the terminal taper consisted of a cone which was welded to the output end of the screw shaft. Each cone was 4 inches in height and had a 2-1/8-inch diameter base.

Seal enclosure configurations tested included straight-pipe sections, two divergent truncated cones, and straight-pipe sections combined with these cones.

Tests were conducted at screw rotational speeds between 1160 rpm (the normal operating speed for the Fuller-Kinyon pump) and 570 rpm. Primarily for convenience, -1/4" coal was, in general, used as the feed material. However, some tests were also conducted with -1/4" coal mixed with -1/4", +1/32" limestone (20% by weight). Pressure differentials ranged from zero to 90 psi.

The Fuller-Kinyon pump laboratory test setup is best described by the schematic drawing shown in Figure 3 and the photographs of the test setup which follow (Figure 4). A close-up of one of the seal enclosures (test sections) installed in the test setup is shown in Figure 5.

General test procedure began with pneumatic conveying of the feed material to the storage bin suspended from a load cell above the modified 4-inch diameter Fuller-Kinyon pump. For each test, enough feed material was placed in the storage bin to maintain a sufficient feed material head above the pump for a continuous uniform supply to the screw throughout the test. At the beginning of a test, the 40-horsepower pump motor was started, and once the
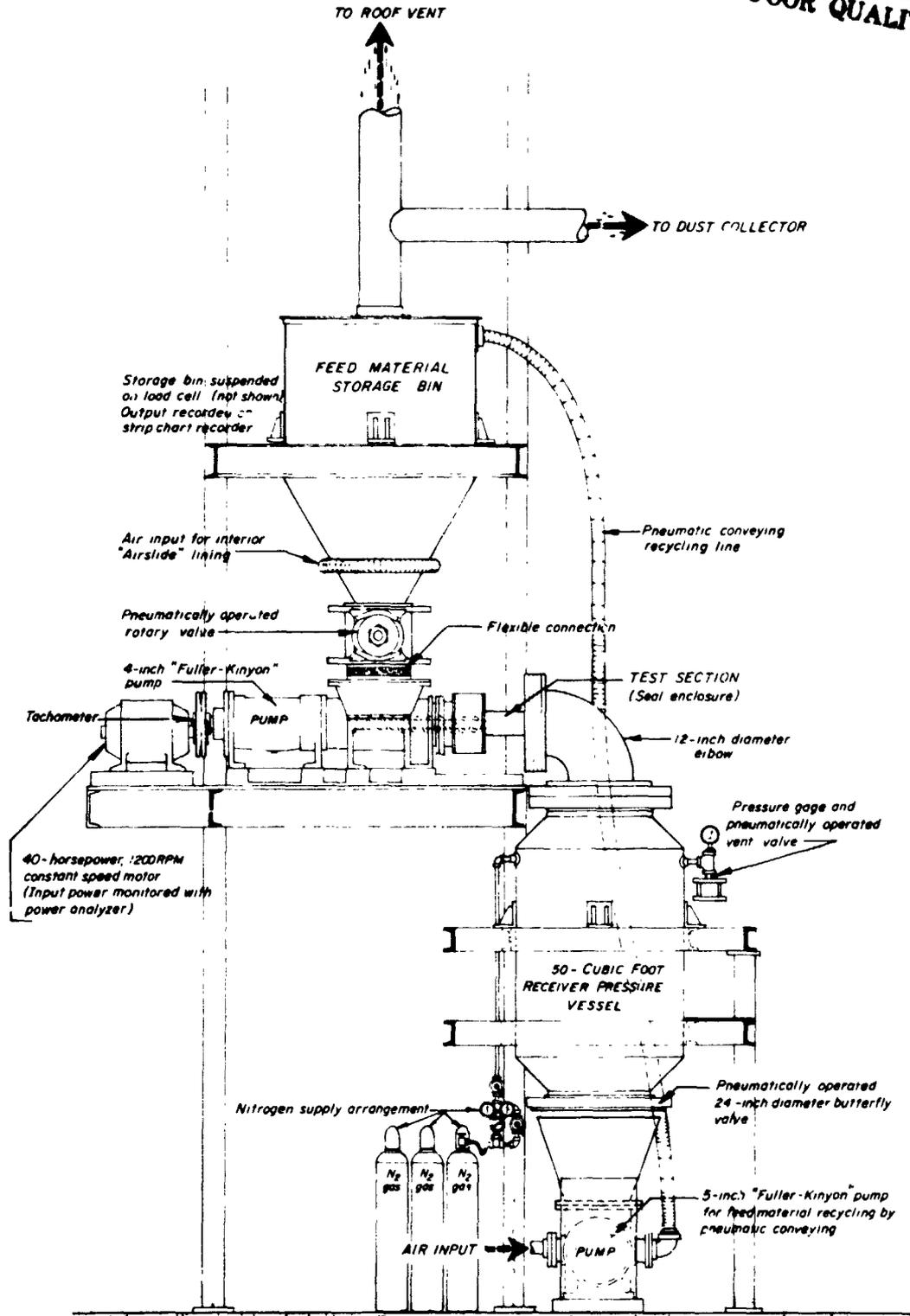
Figure 3   Fuller-Kinyon Pump Laboratory Test Setup Schematic
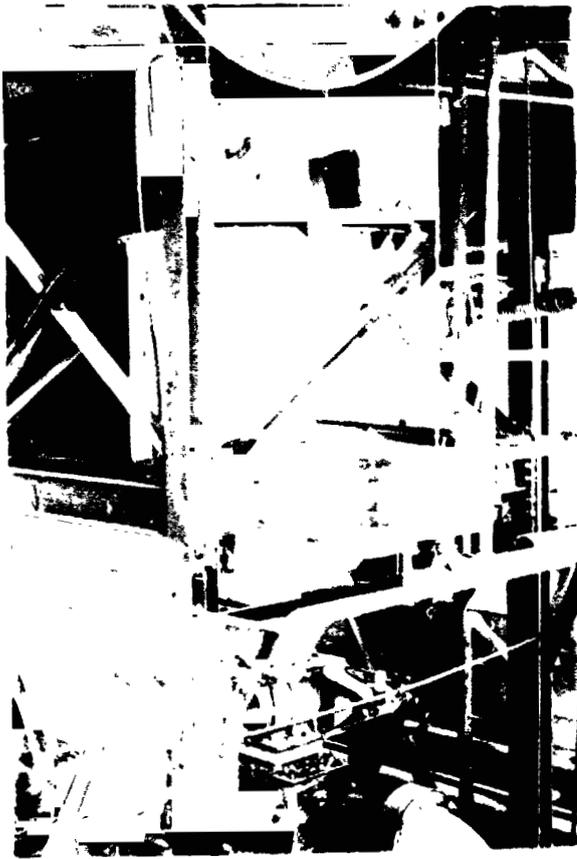
Figure 4 Fuller-Kinyon Pump
Laboratory Test Setup:

a) Feed Material Storage Bin and
Pneumatically Operated Rotary
Valve (opposite)

b) Rotary Valve, 4-Inch Fuller-
Kinyon Pump and Motor, Test
Section, and Top of Receiver
Pressure Vessel (below)
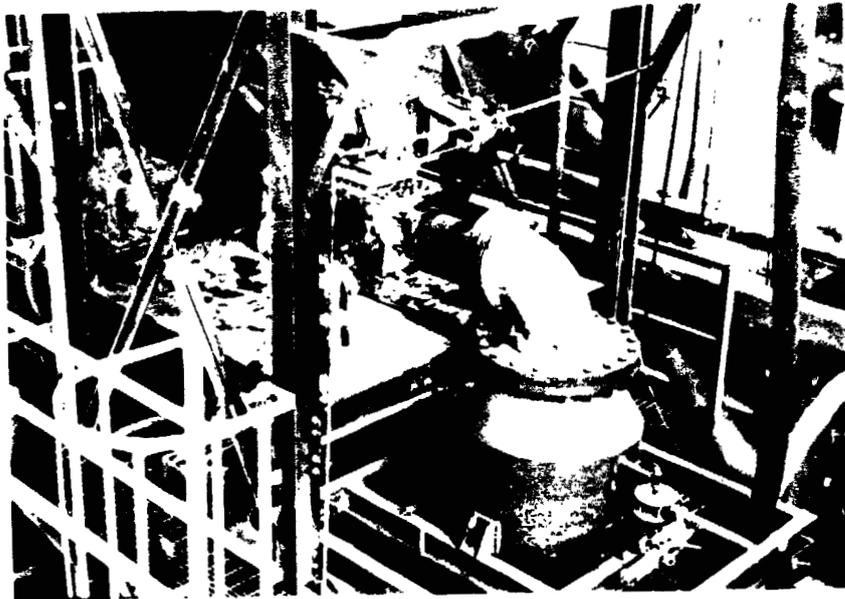
(Figure 4a)
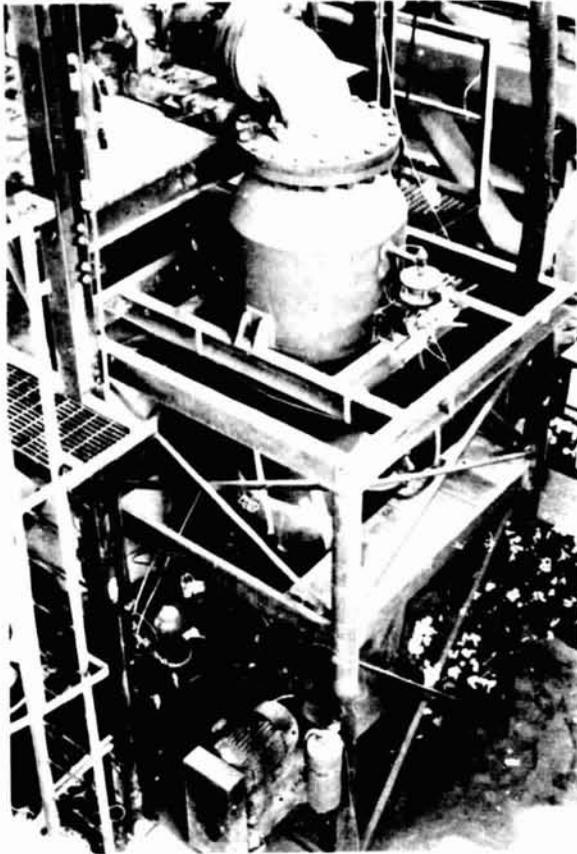
(Figure 4b)

Figure 4 Fuller-Kinyon Pump
Laboratory Test Setup:

c) Receiver Pressure Vessel and
Compact Fuller-Kinyon Pump for
Feed Material Recycling by
Pneumatic Conveying

(Figure 4c)

Figure 5 Divergent Truncated Cone Test Section
Number 1 Installed

selected rotational speed had been reached, the rotary valve isolating the feed material storage bin from the pump was then opened, thus allowing feed material to enter the pump and test section (seal enclosure). Feed material which had been extruded through the system would then accumulate in the receiver vessel.

For tests with non-zero pressure differentials, the receiver vessel was maintained closed and was supplied with a sufficient quantity of inert gas at the start of testing to pressurize the vessel to the desired level. In general, pressurization was performed as quickly as possible after establishment of a steady rate of feed material transfer through the system. (Significant additional pressurization was seldom required to maintain the desired pressure level, with the exception, of course, of those tests during which the seal was not successfully maintained.)

Upon completion of a test and shutdown of the pump, the receiver vessel was vented, if required, and the accumulated feed material was removed through a large butterfly valve at the bottom of the vessel (the feed material would either be scrapped or recycled). At this time, samples were often taken to determine the feed material size distribution and/or moisture content.

In the event of seal loss and rapid depressurization of the receiver vessel (blow-back) during tests with pressure differentials, venting of the system occurred through the pump and feed material storage bin to a dust collector, and for short periods, to a vent in the roof of the laboratory. The dust collector was also used during pneumatic conveying of the feed material to the storage bin prior to testing.

The feed material transfer rate (feed rate) through the system was determined by continuously recording the output from the load cell on a strip chart recorder. The input power to the 40-horsepower pump motor was monitored by observing readings on a power analyzer, while the screw rotational speed was monitored with a tachometer. (Tachometer readings were displayed in digital form on a meter at floor level.) The pressure differential across the pump and seal was monitored by observing readings on a pressure gage mounted on the receiver vessel.

PRELIMINARY TESTS

Preliminary tests with zero gas pressure differential across the pump were conducted with both the 4-1/4" x 4-1/4" pitch screw and the 3-1/4" x 3" pitch screw operating at each of two rotational speeds (1160 rpm and 570 rpm). These tests were conducted with various lengths of smooth straight pipe downstream of the pump to form the seal enclosure for the extruded feed material. Seal lengths tested ranged up to the maximum values which could be obtained with the test setup.

The results of the preliminary tests led to the following observations (a detailed presentation of results can be found in Reference 1):

1) For relatively "new" -1/4" coal and the range of moisture contents observed (less than 5%), the maximum feasible seal length is about 10 inches for both screws and both screw rotational speeds. (This length is considerably shorter than what one might expect on the basis of the results of the materials properties tests.)

2) For both screws and both screw rotational speeds, pump power consumption is relatively similar and in general, varies only moderately with seal length for lengths less than 10 inches (at this length, pump power consumption generally tends to rise rapidly beyond the 40-horsepower rating of the pump motor, necessitating shutdown).

3) The coal feed rate (weight per time) appears to be basically a function of feed material density. Increased moisture content (again in the range of observed values) is associated with de-creased feed material density and decreased feed rates. To a lesser extent, coal recycling is associated with increased feed material density (through the production of finer size distribu-tion) and increased feed rates. It should be noted here that some of the effect of moisture content on feed rate may be attributable to bridging of the coal in the feed hopper of the Fuller-Kinyon pump and resultant restriction in the supply of feed material to the pump itself.

4) For the <u>same</u> feed material, there appears to be little difference between the two screws in terms of the feed rate to be expected at the two screw rotational speeds and various seal lengths.

5) For the <u>same</u> feed material, the feed rate at various seal lengths is somewhat higher for both screws at a screw rotational speed of 1160 rpm.

TESTS WITH PRESSURE DIFFERENTIALS*

Since the data from the preliminary tests provided little if any differentiation between the two screws tested, the 3-1/4" x 3" pitch screw was chosen for tes . with gas pressure differentials across the pump (at least initially), primarily because its smaller pitch and larger number of screw fligh's should enhance whatever pressure sealing effects occur within the screw itself.

Initial tests were conducted at screw rotational speeds of 1160 rpm and 570 rpm with extruded material seals formed in various lengths (9 inches or less) of straight 4-inch diameter pipe downstream of the 4-inch Fuller-Kinyon pump. These tests were conducted with a coal/limestone mixture and were limited by higher than expected pump power consumption to pressure differentials less than 15 psi.

Divergent truncated cone test section number 1 was then designed and fabricated for use in place of the straight pipe as a seal enclosure (see Figure 6 and 7). Tests with this test section were limited by pump power consumption to pressure differentials less than 10 psi for a screw rotational speed of 1160 rpm. (Tests were performed with both the coal/limestone mixture and also -1/4" coal.) However, the comparatively high feed rates observed at zero pressure differential indicated that the test section itself introduced very low resistance to the feed material extrusion process.

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* A detailed presentation of results can be found in Reference 1, with the exception of the results for the 1-1/2" x 1-1/2" pitch screw, which are presented in Reference 2.

Figure 6   Divergent Truncated Cone Test Section Number 1

Screw with terminal taper

TEST SECTION Nº1
(Seal enclosure)

11"

1" Spacers with gaskets

4"

5"

4" Fuller-Kinyon pump barrel

To receiver vessel

12"
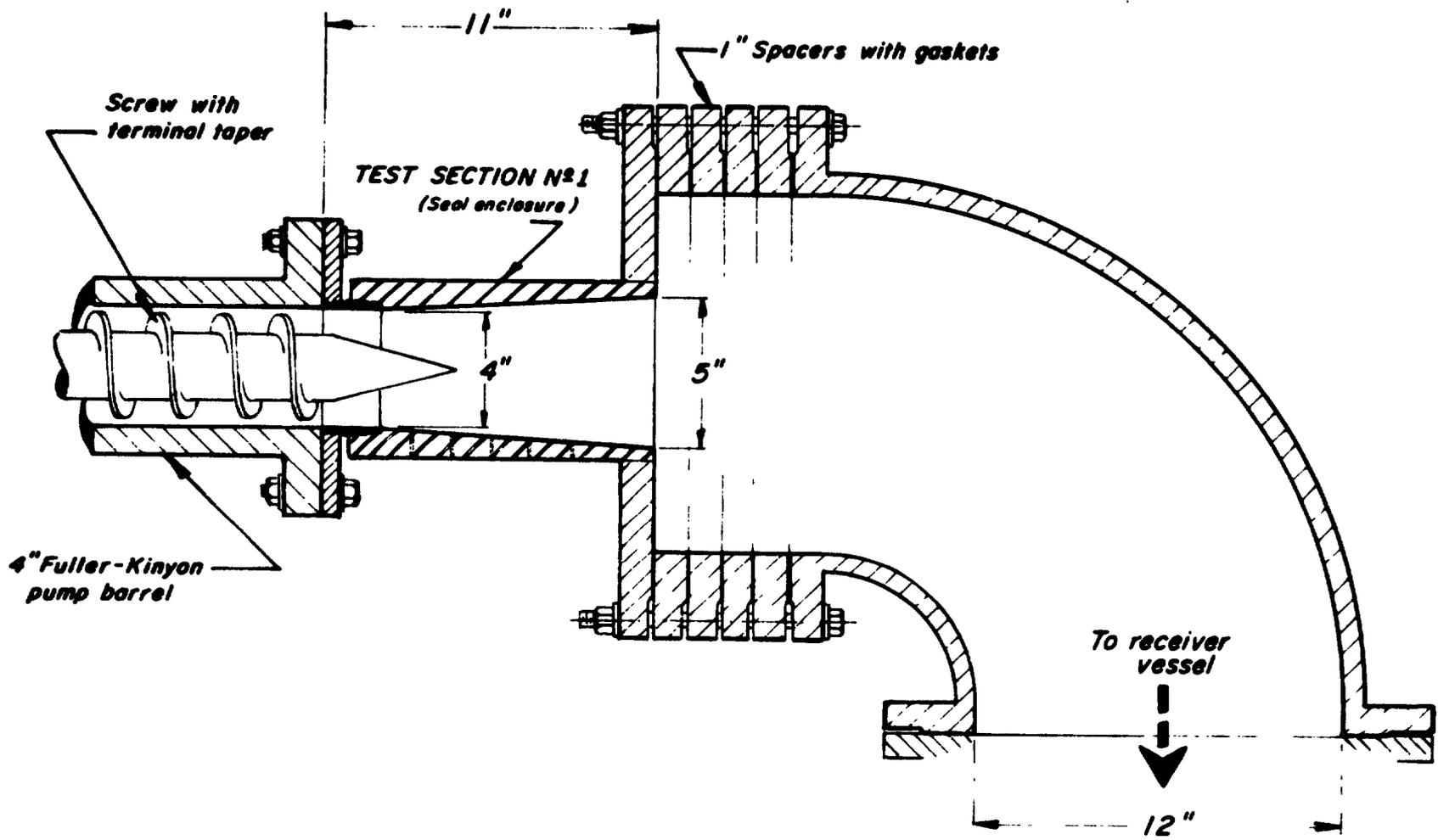
Figure 7   Installed Configuration - Divergent Truncated Cone Test Section Number 1

Further tests (again with test section number 1) were conducted at a screw rotational speed of 570 rpm. These tests were limited by pump power consumption to pressure differentials less than 40 psi in one sequence, 35 psi in another, and somewhat less than 70 psi in a third (in this case blow-back through the pump became a frequent occurrence). Higher than expected pump power consumption was once again observed during these tests together with a dramatic decline in feed rate with increasing pressure differential.

After these tests, the test setup was partially disassembled for in-spection. Significant wear was noted on the terminating flight of the 3-1/4" x 3" pitch screw but not in the barrel liner to the pump or in the divergent truncated cone test section. (The screw wear noted is shown in Figure 8.) This wear was not entirely surprising in light of the fact that the flights of this screw did not have the normal hard-facing. While such wear had not been found on the 4-1/4" x 4-1/4" pitch screw (also without hard-facing), it had also not been used for tests with pressure differentials.

It looked as if the worn profile of the terminating flight of the screw could have led to significant wedging of feed material against the barrel liner and wall of the test section and also could have been conducive to blow-back. For this reason, it was decided to continue testing with the 4-1/4" x 4-1/4" pitch screw while at the same time rebuilding and hard-facing the 3-1/4" x 3" pitch screw for further testing later on.

Ten more tests were then conducted, 4 tests with the 4-1/4" x 4-1/4" pitch screw and 6 tests with the 3-1/4" x 3" pitch screw after it had been rebuilt and hard-faced. Results were very disappointing: tests with both screws were limited to pressure differentials of approximately 30 psi or less, pump power consumption was observed to rise rapidly and almost linearly with increasing pressure differential, and significant heating of the pump housing (barrel) and test section occurred in the vicinity of the terminating flight of each screw. This heating was not significantly affected when, for some of these tests, the barrel was extended to obtain an additional one-inch straight-pipe length between the terminating screw flight and the entrance to the divergent truncated cone test section. Inspection of the 4-1/4" x 4-1/4" pitch screw after testing indicated the beginning of the type of wear pre-viously observed for the 3-1/4" x 3" pitch screw. The latter screw showed no wear after testing, presumably as a result of the hard-facing.

Figure 8   Sequence of Observations Made After Test Number 79 of Wear on Terminating
Flight of 3-1/4" x 3" Pitch Screw

(Note: When viewed from tip, screw rotation is clockwise in this sequence.)

The conclusion of all this was basically that the worn-down version of the 3-1/4" x 3" pitch screw had apparently permitted successful tests at higher pressure differentials. A terminating flight similar to that of a wood screw apparently led to performance superior to that obtained with the standard design.

Because of the evidence of significant loading being transmitted to the terminating flight of the screw during tests involving pressure differentials, two approaches were then taken to alleviate the axial loading transmitted to the screw as a result of seal enclosure wall friction. The first approach involved conducting a number of tests with a modified version of the divergent truncated cone test section used previously. Holes were tapped into the wall of the test section in order to permit jetting of nitrogen into the test section at various positions along its length during testing (see Figure 9). It was thought that the effective cone length could be reduced through downstream gas "lubrication" at the wall and partial fluidization of the extruded feed material. Tests with this arrangement were marginally successful in that some increase was noted in the pressure differentials which could be handled by the system. However, feed rates were observed to decrease significantly with these increasing pressure differentials.

The second approach involved testing with a new divergent truncated cone test section (test section number 2) having a cone angle of about 24° in contrast with the 6° angle of the previous test section. Test section number 2 is shown in Figures 10 and 11.

Results were basically similar to those for the previous test section (test section number 1) in the sense that full pump power consumption was reached at a pressure differential of about 35 psi (again for the 3-1/4" x 3" pitch screw operating at a rotational speed of 570 rpm). Once again a very linear increase in power consumption was noted with increasing pressure differential. The most interesting result was the observation that, for pressure differentials between 35 psi and 40 psi, pump power consumption dropped by about 50% with no apparent change in material flow rate or smoothness of operation. Blow-back occurred on both of these tests at a pressure differential slightly above 40 psi.
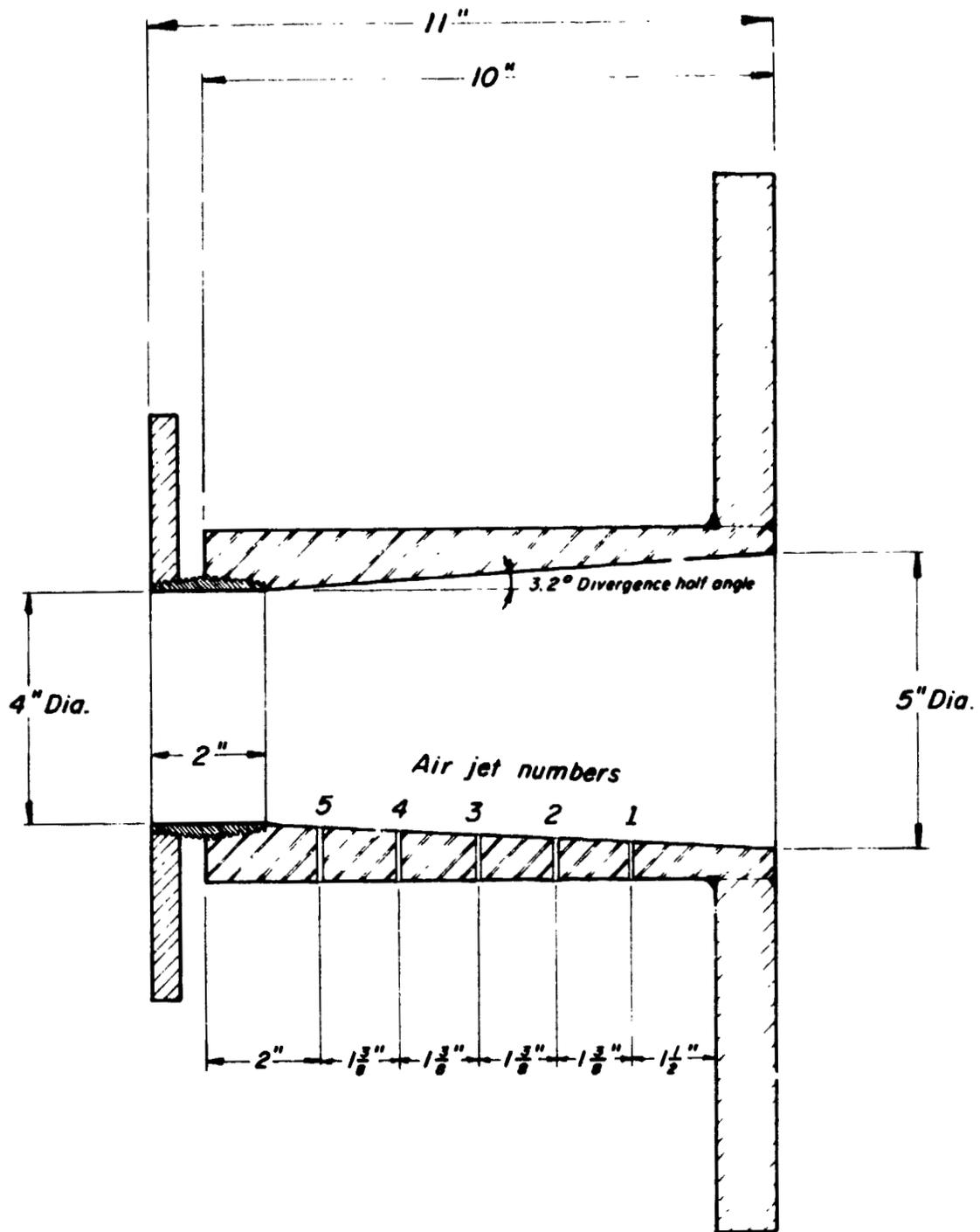
Figure 9   Divergent Truncated Cone Test Section Number 1 -
            Air Jet Locations for Test Numbers 90-106
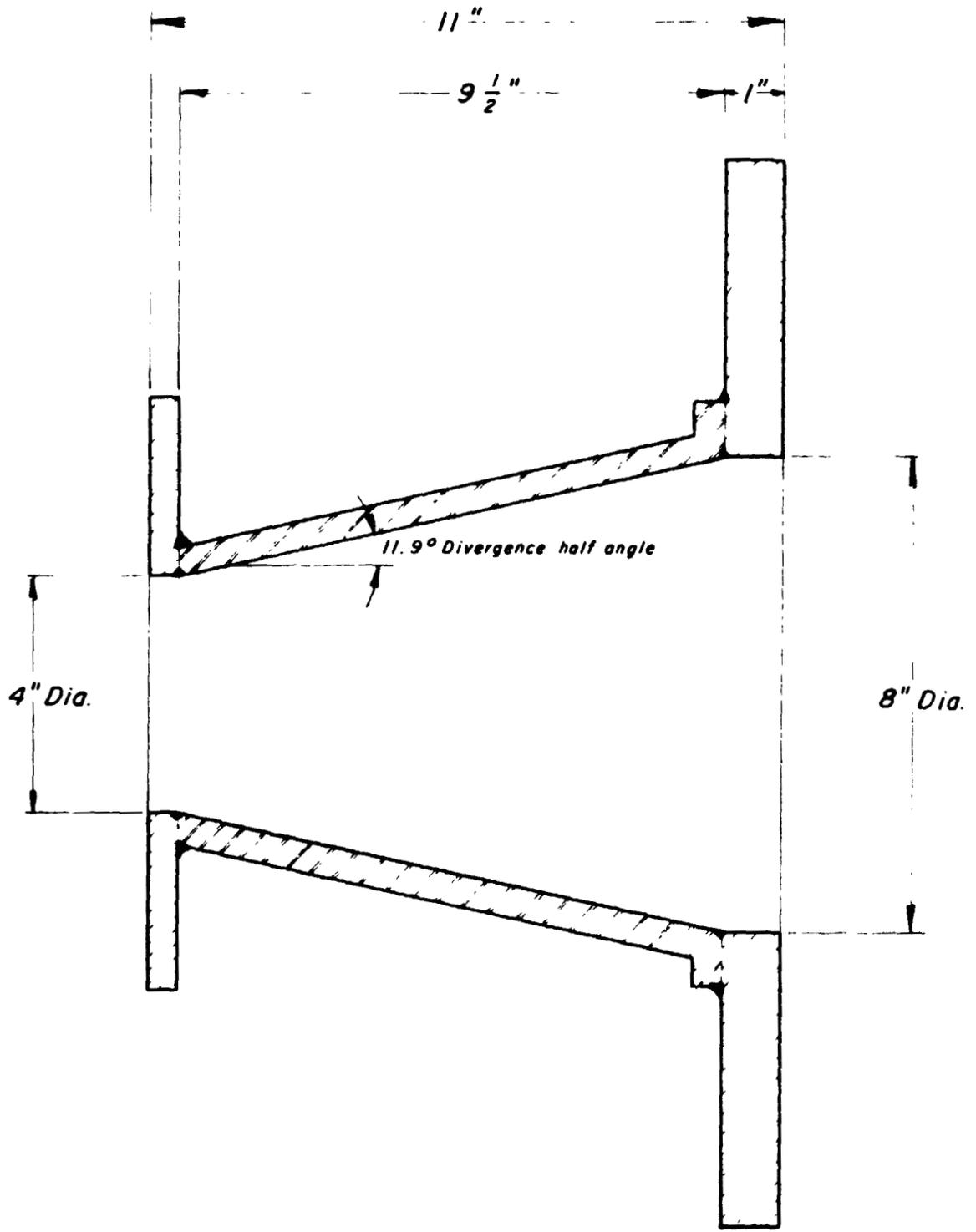
Figure 10 Divergent Truncated Cone Test Section Number 2
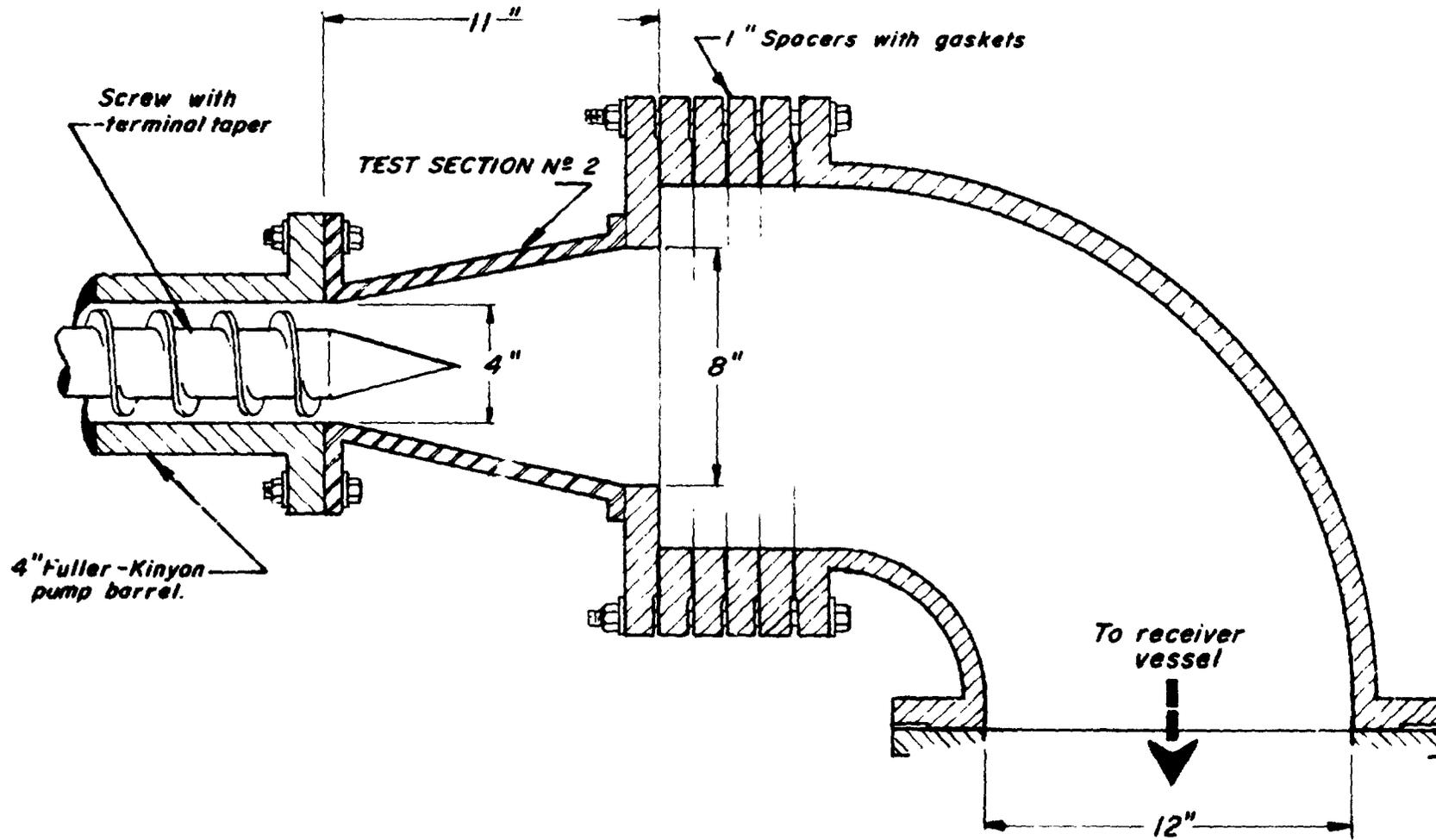
Figure 11 Installed Configuration - Divergent Truncated Cone Test
Section Number 2

The observed drop in power consumption appeared to indicate that leakage back through the screw had led to fluidization of the feed material in the hopper and the pump barrel, thus reducing power consumpti n in this region by reducing material wall friction. The fact that blow-back was observed at a pressure differential slightly above 40 psi tended to support the possibility that a precariously balanced fluidization situation could have existed at slightly reduced pressure differentials.

At this point the tests were stopped for evaluation of the results obtained, prior to further testing. Primarily in an effort to reduce power requirements and at the same time increase the effective seal which could be obtained, two major changes were decided upon. First the pump's normally smooth barrel liner was replaced with a fluted barrel liner (illustrated in Figure 12). It was theorized that this fluted barrel liner would help to constrain the feed material from rotating while enhancing its axial movement, thus increasing the efficiency of the pump. Second, a significartly smaller pitch screw was fabricated, namely a 1-1/2" x 1-1/2" constant-pitch screw. While the displacement rate for this screw would be significantly less than that of the screws previously tested, increased displacement efficiency was expected due to the reduced helix angle for this screw. Furthermore, the potential for blow-back was expected to be reduced due to the increased number of flights for thi\` screw.

The combination of this screw and the fluted barrel liner was used throughout the remainder of the tests. These tests were also conducted with newly obtained coal having much higher moisture contents than the coal used for previous tests. Moisture contents generally varied from 9% to 13%, in comparison with previous moisture contents which were less than 5%.

Tests were first performed with both divergent truncated cone test sections and with straight-pipe sections of various lengths. In comparison with the results obtained previously, these tests yielded relatively similar results (in all respects) for the two divergent truncated cone test sections and somewhat improved results for straight-pipe seals. With respect to the latter, successful operation with a seal length of 10 inches was achieved up to pressure differentials of about 70 psig, at which blow-back became a limiting factor (close to maximum pump power was also required at this pressure differential).
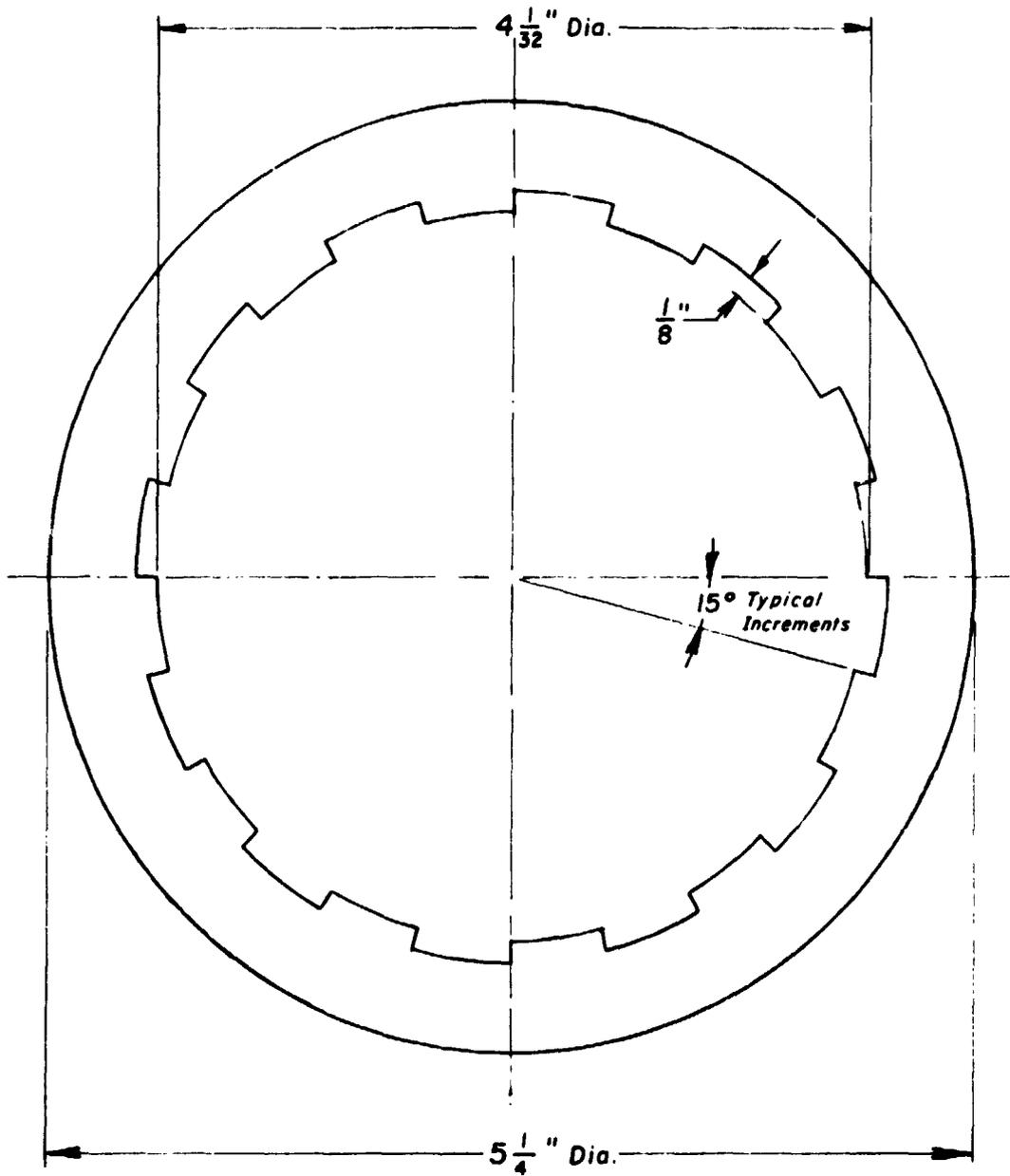
Figure 12   Fluted Barrel Liner

In an effort to extend the seal length beyond 10 inches, final tests were then conducted with seal enclosures consisting of combinations of various lengths of straight pipe and one or the other of the divergent truncated cones. (Feed material would travel through the straight pipe into one of the cones into the receiver vessel.) The majority of these tests were performed with divergent truncated cone test section number 1 and lengths of straight pipe varying from 6 to 7-1/2 to 9 inches. (All of these tests were performed at a screw rotational speed of 570 rpm.)

For the 6 and 7-1/2-inch lengths, tests were limited by blow-back to pressure differentials of approximately 40 psig and 55 psig, respectively. (Pump power requirements, while observed to increase with increasing pressure differential, remained relatively low.) For the 9-inch length (the last test performed), successful pump operation occurred for pressure differentials up to 90 psi, at which the feed rate through the pump began to surge somewhat, indicating that blow-back was most likely imminent. Once again, pump power requirements, while observed to increase with increasing pressure differential, remained relatively low, reaching 33 horsepower at 90 psi. In addition to this result, the feed rate for the pump remained relat ely constant through- out the course of this test (while pressure differentials were increased) and at a value close to that expected for very low pressure differentials.

This last result was considered most important since it appeared to indicate that the substantial overall seal length utilized (19 inches) had made it possible to counteract the phenomenon of decreasing feed rate with increasing pressure differential noted during previous testing. However, since the 90-psi pressure differential reached also appeared to be close to producing blow-back with this seal and close to requiring pump power in excess of that available, it also appeared that further tests at pressure differentials of this order and higher would have to be conducted with some- what longer seals and greater pump power than that currently available. It was also felt that this would be an even stronger requirement for tests with drier coal, such as that used for tests prior to these most recent tests. As a result, testing was stopped at this point.

CONCLUSIONS AND RECOMMENDATIONS

The major results of this project can be summarized as follows:

1) Successful operation of a modified 4-inch diameter Fuller-Kinyon pump was limited, with a few exceptions, to pressure differentials across the pump of 60 psi or less as a result of pump power requirements which exceeded the 40-horsepower rating of the pump motor in the test setup. Furthermore, operation with straight-pipe seal lengths was in general limited ' seal lengths of 10 inches or less for the same reason.

2) If the significant drop in pump power consumption observed just prior to blow-back during some of the tests can be attributed to partial fluidization of the feed material in the pump barrel, then it is likely that a significant portion of the normally observed power consumption also occurred in the barrel.

3) The power required by the modified Fuller-Kinyon pump appeared to increase linearly with increasing pressure differential across the pump.

4) Power requirements were found to be somewhat reduced for a worn version of one of the screws tested.

5) In many cases, unaccountably similar results were obtained for significantly different types of seal configurations, significantly different screws, and significantly different screw rotational speeds. According to Reference 3, explanation of his may be related to rejection of feed material from the pick-up flights of the screw in the pump hopper (due to centrifugal forces) and also to gas infiltration into the feed material in the hopper (due to leakage back through the pump barrel when operating against a gas pressure differential).

6) Increasing the pressure differential across the modified Fuller-Kinyon pump led to a significant and apparently linear decrease in the feed rate for the pump and, for the limited seal lengths which could be tested, an increasing likelihood of blow-back through the system. Explanation of this may also be related to the gas infiltration problem cited previously.

7) Positive modification of many of the above results appears to be possible with longer seals, particularly longer "combination" seals (straight-pipe section combined with a divergent truncated cone).

Most of these results indicate that insufficient power was provided by the motor used to drive the modified Fuller-Kinyon pump (in particular this limited the seal lengths which could be used and therefore the pressure differentials which could be handled). Interestingly, a substantial increase in the power consumed by the system would not violate the guidelines established at the beginning of the project, namely that overall feed system power consumption should be less than 1% of the power eventually generated from the feed material. If we assume for the moment that feed rates of the order of 300 lb/min of -1/4" coal can be realized with the modified 4-inch diameter Fuller-Kinyon pump, then according to the guideline, the overall feed system based on this pump should operate on something less than 240 horsepower (based on power generation of 1 kwhr per pound of coal). Thus, it would be entirely reasonable to increase the power available to operate the modified 4-inch Fuller-Kinyon pump.

It therefore is recommended that further testing be performed with the substantial test setup which is presently available, in conjunction with replacing the 40-horsepower pump motor presently in the test setup with a motor having a significantly higher power rating. (This should in turn allow pump operation with longer seals and at higher pressure differentials than those tested so far.) It is also recommended that if this further testing is successful, additional testing be performed to determine the potential reduction in power requirements which might occur as a result of alterations in screw design and the design of the pump barrel.

ACKNOWLEDGEMENT

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BABCOCK & WILCOX'S EXPERIENCE WITH
TWO-PHASE FLOW MIXTURES OF COAL
AND GAS

A. G. Bolumen

The Babcock & Wilcox Co.
Barberton, Ohio

BABCOCK & WILCOX'S EXPERIENCE WITH TWO-PHASE FLOW MIXTURES OF COAL AND GAS

A. G. Bolumen, Process Engineer
Fossil Power Generation Division
Power Generation Group, Barberton, Ohio

ABSTRACT

This paper describes The Babcock & Wilcox (B&W) Company's general background in conveying crushed and pulverized materials but more specifically addresses the transport and feeding of pulverized coal at moderate pressures in dilute phase.

Based on B&W's background, two pulverized coal injection systems were designed, installed and placed in operation on blast furnaces and have an accumulated operating time of over 15 years. The larger and newer system was placed in operation in 1973 and has achieved essentially 100% availability. This system is capable of injecting 33 tons of coal per hour and operating at pressures up to 95 psig.

INTRODUCTION

Conveying a material pneumatically from one place to another has been done for many years and industry has relied on pneumatic transport for grain, cement, wood chips, and coal In general, the conveying of most materials presents a great difficulty to those knowledgeable in the art. However, certain processes, for example the firing of pulverized coal boilers, require steady non-pulsing feed. Developing technologies such as coal gasification, fluidized bed combustion, and magneto-hydrodynamics, require that pulverized or crushed coal be transported, split into equal flow streams, and fed into the process at high pressure in a controlled manner.

The Babcock & Wilcox Company has been designing and supplying systems for conveying and feeding pulverized and crushed coal for many decades. The following describes some of the Company's background and developmental work in this area.

PULVERIZED COAL FIRING OF BOILERS

Pulverized coal has been used as boiler fuel since the 1920's. In this application pulverized coal infers that the coal has been milled until 70 weight percent will pass through a 200 mesh seive Two principal systems — the bin system and the direct fired system — have been used for processing, distributing, and burning pulverized coal in power plants.

The bin system (see Figure 1) consists of two steps:

1. Coal preparation and storage
2. Coal transport and injection

From Figure 1 it can be seen that raw coal from a bunker is fed by gravity through a feeder to a pulverizer with a single outlet. In the mill, the coal is simultaneously pulverized and dried. Hot air to dry the coal and transport it out of the pulverizer is provided by a fan and an air heater. The temperature of this air may be as hot as 650° F depending on surface moisture. The pulverized coal leaving the pulverizer is separated from the conveying air in a mechanical dust collector where 90 to 95 percent of the coal is collected and discharged by gravity into a storage bin. The remaining 5 to 10 percent is recovered in bag filterhouses, and the clean moisture-laden air is vented to the atmosphere.

From storage, the pulverized coal is generally conveyed by pneumatic transport through pipelines to utilization bins sometimes as far as 5,000 feet from the point of preparation. For this phase of pneumatic transport of the coal, a differential pitch screw pump is often provided. The coal in the transport line is then distributed through a system of two-way valves to any number of bins. From the utilization bins the coal is picked up in an air stream and transported and fired into the boiler furnace as required.

In the direct fired system (see Figure 2), raw coal is fed to the pulverizer where it is dried as well as pulverized. The primary air fan provides hot air from the boiler air heater to the pulverizer for drying and conveying the coal to the boiler furnace. Each pulverizer has multiple outlet lines with each line feeding a burner. The velocity in the pulverizer discharge pipes must be sufficiently high to prevent settling of coal which may lead to burner line fires. At

the burners, the air-coal mixture must be uniform and the velocity suitable for stable combustion.

CRUSHED COAL FIRING OF BOILERS

Crushed coal ($1/4''$ x O) is used as the fuel for the B&W cyclone furnace. Preparation of coal for burning in a cyclone furnace consists of crushing the coal to about $1/4$-inch top size. In some plants, this crushing is done in a crusher, which discharges coal directly into the cyclone furnace with primary air. This is called the direct fired system (see Figures 3 and 4) and a feeder and a crusher are required for each cyclone furnace. This arrangement is advantageous when there is limited plant space or there is an existing coal handling facility.

In other plants, coal is crushed to size before it is delivered to the main bunkers. This is called the indirect system. Economics generally favor the indirect system because it permits more efficient use of larger and fewer crushers.

When burning lignite or other high moisture coals, the direct fired, predrying, bypass system (Figure 5) is used. This is a variation of the direct fired system described above and incorporates a mechanical dust collector between the crusher and the cyclone furnace. Hot drying air is added to the raw coal before the mixture enters the crusher. Mechanical dust collectors separate the crushed coal from the moisture laden drying air leaving the crusher and the drying air is then vented into the secondary boiler furnace. The venting of this moisture laden air to the boiler secondary furnace instead of the cyclone furnace helps to maintain maximum temperature in the cyclone with improved performance and slag tapping characteristics. Hot primary air conveys the crushed coal from the exit of the mechanical dust collector to the cyclones. The pressure drop of the air passing through a cyclone is in the range of 20 to 40 in. wg; thus, the pressure at the coal inlet to the conveying line may be 50 to 70 in. wg.

In 1966, tests were carried out at B&W's Alliance Research Center to investigate transport and distribution of $1/4''$ x O crushed coal by dense phase from a fluidized feed tank. Coal to air ratios ranged from 30 to 173 lb. coal/lb air and pressures at the feed tank varied from 30 to 58 psig. Heavy pulsations and some pluggage were experienced in many runs due to both the inability to uniformly fluidize a material consisting of a wide range of particle sizes (100:1) and to non-uniform distribution of the various size fractions throughout the bed of coal in the feed tank. Several successful runs were conducted but there was insufficient investigation of the test conditions which made these runs different from the ones with pulsations.

DENSE PHASE TRANSPORT OF PULVERIZED COAL

In addition to conveying pulverized coal to steam generators at low pressures (less than 2 psig.), B&W has also designed and built systems to convey pulverized coal at pressures up to 95 psig.

The Company began to look at systems which could be utilized to inject pulverized coal into blast furnaces as a supplementary fuel in the early 1950's. In order to fire pulverized coal into a blast furnace, the coal must be divided into many streams, each stream going to a tuyere.

An attempt to split the coal flow from one pipe into many smaller ones was made by using a horizontal distributor (see Figure 6) similar to the coal distributing equipment being used on several zinc-fuming furnaces since 1942. However, tests indicated that the degree of unbalance between individual transport lines was greater than desired.

As a result of the unsatisfactory performance encountered in the tests made on the horizontal distributor, a new distributor was designed; model tested; and based on the model test results, built and tested in full scale. The vertical distributor, (see Figure 7) after some modifications, performed satisfactorily at coal to air ratios from 33 lb. coal per lb. air to 1 lb. coal per lb. air and system pressures up to 40 psig.

Tests were also conducted at the Company's Alliance Research Center for transporting pulverized coal through $1^{1}/2$ and $2^{1}/2$ inch pipe at various coal rates, velocities, and pressures up to 40 psig.

The information and design data provided by the above tests led to the design and construction of the pulverized coal injection system installed on Armco Steel's Bellefonte blast furnace at Ashland, Kentucky.

DESCRIPTION OF THE BELLEFONTE PULVERIZED COAL INJECTION SYSTEM

The pulverized coal injection system used in conjunction with the Bellefonte furnace (see Figure 8) was the first of its kind to be operated in the industry. It was placed into commercial operation in 1966 and is still operating today. This system is designed to inject up to 25 tons of pulverized coal per hour through the furnace's sixteen tuyeres.

Raw coal is stored in two storage bins on top of the pulverizer building. Starting at this point, the coal is handled by two parallel and independent systems, each of which supplies pulverized coal to eight of the sixteen furnace tuyeres. Coal from the storage bins is

conveyed into lock hoppers (two for each parallel system) where the pressure is increased to 55 psig. The raw coal is then fed to the pulverizer where the coal is dried and pulverized. It is then conveyed in a light phase coal-air mixture to a cyclone separator at the blast furnace. The coal which is separated in the cyclone passes through a rotary valve, is re-entrained in injection air and the dilute phase mixture is delivered to a vertical bottle distributor where it is divided into eight streams. Individual coal transport lines run from the distributor bottle to eight adjacent tuyeres.

DESCRIPTION OF THE AMANDA PULVERIZED COAL INJECTION SYSTEM

A larger second generation pulverized coal injection system has been in service since June, 1973 on Armco's Amanda blast furnace. This system, which is the largest in commercial operation in the United States, can feed a maximum of 33 tons of pulverized coal per hour to the blast furnace. Presently the blast furnace operates at between 25 and 30 psig. However, the pulverized coal injection system was designed for 40 psig operation at the blast furnace for future conditions.

The Amanda furnace's pulverized coal injection system (see Figure 9) is divided into three subsystems, namely:

1) Raw Coal Storage and Pulverization
2) Pulverized Coal Storage and Pressurization
3) Pulverized Coal Transport and Distribution

1) Raw Coal Storage and Pulverization System

This is a relatively conventional bin-type coal pulverizing system operating at atmospheric pressure. There are two raw coal bunkers, two raw coal feeders, two B&W air-swept pulverizers and two sets of peripheral equipment including primary air fans, pulverizer air heaters, cyclone separators and bag filters. The pulverized coal from the cyclone separators and bag filters discharge into one pulverized coal reservoir tank, which operates at atmospheric pressure. Each path is completely independent of the other so an outage of one will not shut down the entire system.

2) Pulverized Coal Storage and Pressurization System

The pulverized coal in the reservoir tank is fluidized with inert gas at atmospheric pressure and transported to the three feed tanks. The inert gas is produced in a gas generator which burns natural gas

in air. The inert gas is dried, compressed, and supplies both the reservoir tank and the pressurized feed tanks. During operation the coal feed tanks are loaded cyclically. While one tank is emptying, one is being filled and one is on standby fully loaded and pressurized to the prevailing feeding pressure to assure continuity of feed to the blast furnace. Load cells supporting each feed tank measure the weight of coal added to or transported from each tank. When a feed tank that is being charged with pulverized coal becomes full, a ball cock between the reservoir tank and feed tank is closed and the tank is pressurized with inert gas. At the proper time, the ball cock between the feed tank and transport line is opened and the fluidized pulverized coal flows from the feed tank to the blast furnace at a rate of flow that is controlled by the inert gas pressure in the feed tank.

As the feeding tank empties, the load cells measure the change in weight of the tank. When the tank reaches minimum weight (low coal level), the ball cock on the next tank, which is full of coal, opens and the ball cock on the first feed tank closes. The pressure in the tank which has just emptied is reduced to atmospheric pressure and the tank is then ready for filling from the reservoir tank. The Amanda system is designed to operate with only two tanks. To assure system reliability a spare feed tank is provided and all three are kept in operation, which increases the probability that the spare will function properly when needed. If a tank malfunctions, it is taken out of service and the system transferred to the next tank automatically.

Except for some automatically actuated valves, the pulverized coal storage and pressurization system is comprised entirely of static equipment. Material flow through the storage and pressurization system is by gravity

3) Pulverized Coal Transport and Distribution System

The pulverized coal transport and distribution system is a completely static pipeline system which conveys the coal pneumatically from the feed tanks to the furnace area and distributes it among the tuyeres.

The transport of pulverized coal to the blast furnace which is located 340 feet from the coal preparation and pressurization system is accomplished by the addition of a small quantity of compressed air to the coal stream leaving the feed tank. A distributor (see Figures 10 and 11) is located at the far end of the 340 feet transport line. The coal-air mixture is divided into 24 streams with each line supplying one blast furnace tuyere.

The rate at which coal is injected into the furnace is normally a function of hot metal output. Therefore,

it is logical to make the injection rate proportional to the blast furnace wind rate since this factor is indicative of furnace productivity.

The injection rate is a function of the differential pressure across the transport and distribution system and the density of the coal-air mixture. However, only the system pressure drop is varied to control the injection rate as the density of the coal-air mixture is held constant. Since the blast furnace operates at constant pressure, the feed rate is determined by controlling feed tank pressure. The density of the mixture is controlled so that the system is always in a stable mode of transport. This is accomplished by controlling the transport air rate.

A simple parallel control assigns a specific system pressure drop and air flow to each desired injection rate. Deviations from the proper relation between the desired injection rate and the controlled variables are nullified by a corrective vernier derived from the actual injection rate.

The actual injection rate is determined by differentiating the weight signals from the feed tank load cells to obtain the rate at which each tank is changing weight. An electromechanical auctioneer that receives its actuating intelligence from the feed tank sequence control always picks the rate signal from the discharging feed tank for control purposes.

TWO — PHASE FLOW FUNDAMENTALS

B&W's approach to understanding two phase flow characteristics is based on a combination of theory and testing. The phenomena of gas-solids flow are studied by postulating theoretical relationships and then either confirming or modifying these relationships based on experimental data. This approach is superior to the empirical methods that are normally employed and can be used for extrapolation with greater confidence.

To properly understand and design a two-phase system, it is essential to be able to develop slip velocity and performance diagrams for both horizontal and vertical flow. The slip velocity is used to calculate the static head or suspension losses, and the losses which result from the frictional resistance between the coal-air mixture and the pipe wall. The performance curve relates pressure loss (exclusive of shock and acceleration losses), superficial gas velocity, spatial density and pipe diameter, and is used to establish regions of stable and unstable transport.

Pressure Drop in Vertical Section

The total pressure drop of a gas solids system consists of the following components:

1) Static head or suspension losses
2) Frictional losses
3) Losses associated with solid accelerations, bends, expansions, etc.

1.) Static Head or Suspension Losses

The slip velocity diagram is essential in predicting static head or suspension losses. A general slip velocity diagram for a vertical flow system is shown in Figure 12. This shows a plot of slip velocity versus pressure gradient for a given particle size. The slip velocity is assumed to be related to the superficial gas velocity and the true particle velocity by:

$$\frac{V_d}{e} = \frac{V_o}{e} - V_s \qquad (1)$$

where

V_d = Rate of slip between fluid and particle assemblage defined as the approach velocity uncorrected for voidage, ft/sec

e = volume fraction of voids in a bed of particles, dimensionless

V_s = average solids velocity, ft/sec

For a transport system, V_s may be calculated as follows:

$$V_s = \frac{R}{\rho_f} = \frac{R}{\rho_s (1-e)} \qquad (2)$$

where

R = Solid feed rate, lb/(ft^2)(sec)

ρ_f = Dispersed density of solids, lb/ft^3

ρ_s = True particle density, lb/ft^3

With reference to Figure 12, line OQ represents the relationship between pressure gradient and slip velocity when a fluid is passed through a bed of particles in which the particles are supported by direct contact with each other and the retaining wall. Whether the bed is moving or in a static condition, the same result is obtainable provided the material in each case is in a similar state of particle arrangement and relative velocity. Under these conditions, the total pressure drop across the bed increases with the velocity of the fluid as shown in Figure 12, but the length, L, of the bed corresponding to a given amount of the solid phase remains constant.

As the fluid velocity through a fixed bed is increased, a point Q is reached at which the particles begin to rearrange themselves into a more open packing. This condition is called the point of quiescence. The particles will yield much more readily to any external force. A further increase in the fluid velocity

will result in an increase in the total pressure drop, Δ P, and in the bed length, L, due to expansion. The approximate net result is operation along the vertical line QF until point F is reached. The latter condition may be called the point of incipient fluidization, at which the weight of the particles is completely borne by the air stream. The individual particles now possess freedom of motion, and the bed behaves like a "boiling" liquid.

Full fluidization is attained at Point F. As the velocity in the system is increased from V_f to V_∞, the value of $\Delta P/L$ decreases, however, ΔP remains essentially constant because of a corresponding decrease in stream density. In pneumatic conveying we are interested in velocities shown between V_f and V_∞ on Figure 12. The curve between points B and C represents the region of dense phase transport or fluidized bed operation. In this area the pressure gradient is described by:

$$\Delta P/L = (\rho_s \cdot \rho) \quad (1 - e) \qquad (3)$$

At the point (C, V_∞), the superficial fluid velocity exceeds the settling velocity of all the particles in the bed. The curve then passes through a maximum at (A, V_M). A region of instability exists between points C and A along the F, V_∞ curve; here pulsations may occur in a pneumatic system. Finally, the pressure gradient becomes zero at the terminal settling velocity of a single particle in an infinite fluid.

Means for predicting curve F, V_∞ are provided by the following relationships (4) and (5):

C   = f (Re), a relationship between a modified drag coefficient and a (4) modified Reynolds number which coincides with the well-known resistance curve for spheres settling in an infinite fluid (see Figure 13).

$$V_d = V_c \left[\frac{1 + KNV_c \times 10^6}{135 \, e} \right]$$

, an empirical   (5)

relationship representing data on gas fluidization (see Figure 14)

where

$$C = (C_0)(Y)$$

$$C_0 = \left[\frac{4}{3} \, \frac{g \, \emptyset_s \, D_p \, (\Delta P/L)}{\rho V_c^2 \, (1 - e)} \right] \left[\frac{2}{K_v \, (K_v + 1)} \right]$$

$$Y = Y'' + (Y' - Y'') \sin \frac{24}{CRe} \frac{\pi}{2}$$

$$Y'' = \frac{1}{1 + \dfrac{9.65 \, (1 - e)}{k}}$$

$$Y' = \frac{1}{1 + \dfrac{9.65 \, (1 - e)}{k} - \dfrac{14.83 \, (1 - e)^2}{k}}$$

$$Re = \frac{Re_p}{K_v}$$

$$Re_p = \frac{\emptyset_s \, D_p \, V_c \, \rho}{\mu}$$

$$K_v = \frac{1}{1 - 1.107 \, (1 - e)^{2/3}}$$

$$K = \frac{\sigma \mu^3}{\left[g \rho \, (\rho_s - \rho) \mu \right]^{4/3}}$$

$$N = \frac{1 - e}{\dfrac{\pi}{6} \, D_p^3}$$

Y   = Generalized dispersion factor as defined above, dimensionless

k   = Ratio of specific heats, C_p/C_v, dimensionless

$\emptyset_s$ = Carman's shape factor for securing the specific surface of non-spherical particles

D_p = Diameter of sphere of the same volume as that of the particle, ft

V_c = Calculated fluid velocity relative to the particle for gas fluidized bed assuming the validity of Figure 13

ρ = Density of fluid, lb/ft^3

K   = A dimensional factor

N   = Particle concentration, number $/ft^3$

Trial and error is now required in the solution of equations (4) and (5).

2.) Frictional Losses

The pressure drop due to wall friction can be determined empirically by starting with the usual friction factor versus Reynold's number correlation, and the Fanning equation by using the viscosity of the air and the density of the mixture.

$$Re_t = \frac{DG}{\mu} (1 + L_x) \tag{6}$$

$$\left(\frac{\Delta P}{L}\right)_{fs} = \frac{4f}{D} \frac{G^2}{2\rho_f g} (1 + L_x)^2 \tag{7}$$

$$\rho_f = \frac{\dfrac{R}{V_o - V_d}}{1 + \dfrac{R}{\rho_s (V_o - V_d)}} \tag{8}$$

Re_t = Pipeline Reynolds Number = $\dfrac{DG}{\mu}$, dimensionless

D = Inside diameter of transport line, ft.

G = Gas flow rate, lb/(ft^2) (sec)

μ = Viscosity of fluid, lb/(ft) (sec)

L_x = Solid to gas loading, lb/lb

$(\Delta P/L)_{fs}$ = Pressure gradient component due to frictional resistance of the mixture with the pipe wall, lb/ft^3

f = Fanning friction factor, dimensionless

g = Local acceleration due to gravity = 32.17 ft/(sec)^2

V_o = Superficial fluid velocity, ft/sec

Equations (6) and (7) are usually adequate for normal engineering applications at low solid to gas loading; but at high loadings the actual pressure drops could be 30 to 40% lower than predicted. Therefore, to improve their accuracy, it is valid to use a linear function of solids loading in correcting the mass rate term. Equations (6) and (7) may then be rewritten as:

$$Re_t = \frac{DG}{\mu} (1 + aL_x) \tag{9}$$

$$\left(\frac{\Delta P}{L}\right)_{fs} = \frac{4f}{D} \frac{G^2}{2\rho_f g} (1 + aL_x)^2 \tag{10}$$

where

a = Dimensionless correction constant applied to solids loading in the modified Fanning relationships for two-phase flow.

3.) Acceleration and Shock Losses

Pressure drop across bends and other fittings have been extensively studied for one-phase flow. A correlation relating the bend losses for two-phase flow to single-phase flow was developed by B&W.

A similar procedure was then used to obtain pressure drops through other types of fittings.

Pressure Drop in Horizontal Sections

Means for the prediction of slip velocities for horizontal flow could not be found in the literature and was therefore developed by the Company from tests performed at the Alliance Research Center. These tests also identified regions of stable and unstable flow.

Performance Diagram

The total line loss is defined by means of the following relationship:

$$(\Delta P/L)_{LL} = M\rho_f + (\Delta P/L)_{fs} \tag{11}$$

where

$$m = \frac{(\Delta P/L)h}{\rho_f}, \text{ lb force/lb mass}$$

$(\Delta P/L)h$ = Pressure gradient component due to slip between the gas and solid phases in horizontal flow, lb/ft^3

The first term accounts for the "suspension" losses while the second term relates to the wall friction whose value was already defined (Equation 10) so that:

$$\left(\frac{\Delta P}{L}\right)_{LL} = m\rho_f + \left(\frac{4f}{D}\right)\left(\frac{G^2}{2g\rho_f}\right)(1 + aL_x)^2 \tag{12}$$

The pressure drop equation derived for line losses (static head and friction) forms the basis for predict-

ing system performance. The performance diagram shown on Figure 15, is typical of those which can be developed quantitatively for pulverized coal. For a fixed pipe diameter, such a diagram shows the relationship between the pressure gradient (line losses only) and superficial gas velocities for parameters of different coal feed rates, W_1, W_2, and W_3.

Starting on the upper left part of the diagram, the pressure gradient decreases with gas velocity. This results from the coal dispersion becoming more dilute, resulting in lower "suspension" losses; but as the velocity increases to higher values, the term which describes the friction of solids against the wall begins to prevail and the curves pass through a minimum. For dilute and light phase transport there is a minimum velocity below which the solids begin to settle in horizontal flow. This velocity is the saltation velocity and is shown by the dotted curve on Figure 15. Also, as the solid loading increases, particles begin to settle out and the system may be expected to exhibit flow instability. In dense phase flow, on the other hand, smooth operation is possible below the saltation velocity because the relative low voidage hinders particle separation and the solids remain in a dispersed condition. But pluggage may take place in dense phase transport if the velocity drops below the minimum velocity required for smooth flow.

Three other curves are also superimposed on Figure 15. These are lines of constant density and pertain to the spatial concentration (ρf) of the solids dispersed inside the pipeline. A high value of spatial concentration is shown at A lb/ft^3, an intermediate value at B lb/ft^3, and a relatively low value at C lb/ft^3.

By means of these constant density lines and the saltation velocity curve we can separate the above chart into five different regions:

Region 1 — stable dense phase
Region 2 — unstable
Region 3 — unstable for horizontal flow
Region 4 — questionable
Region 5 — stable dilute phase

It should be recognized, however, that the apparent stability of Regions 1 and 5 may be upset by the interaction of other variables in the system such as feed tank internals, bends and distribution devices.

By using the techniques described, it is possible to identify the regions of instability which can exist in every pneumatic transport system. Also these regions may vary for different materials depending on the particle characteristics. B&W has defined these regions for a limited application. To be able to design systems capable of achieving smooth flow, and to satisfy the requirements which new processes demand, considerable research and development is needed.

ACKNOWLEDGEMENTS
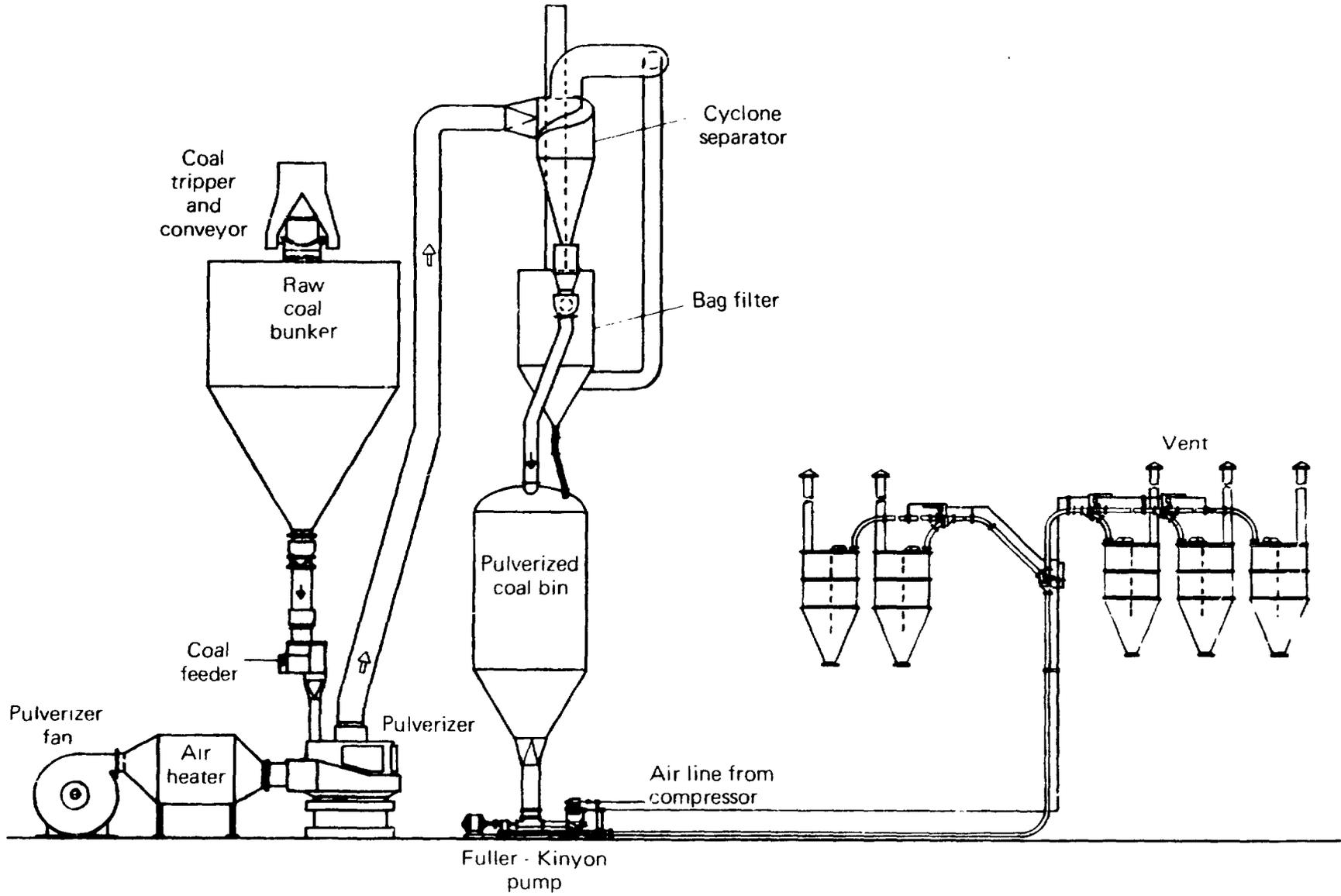
Bin system

Figure 1

Direct firing system
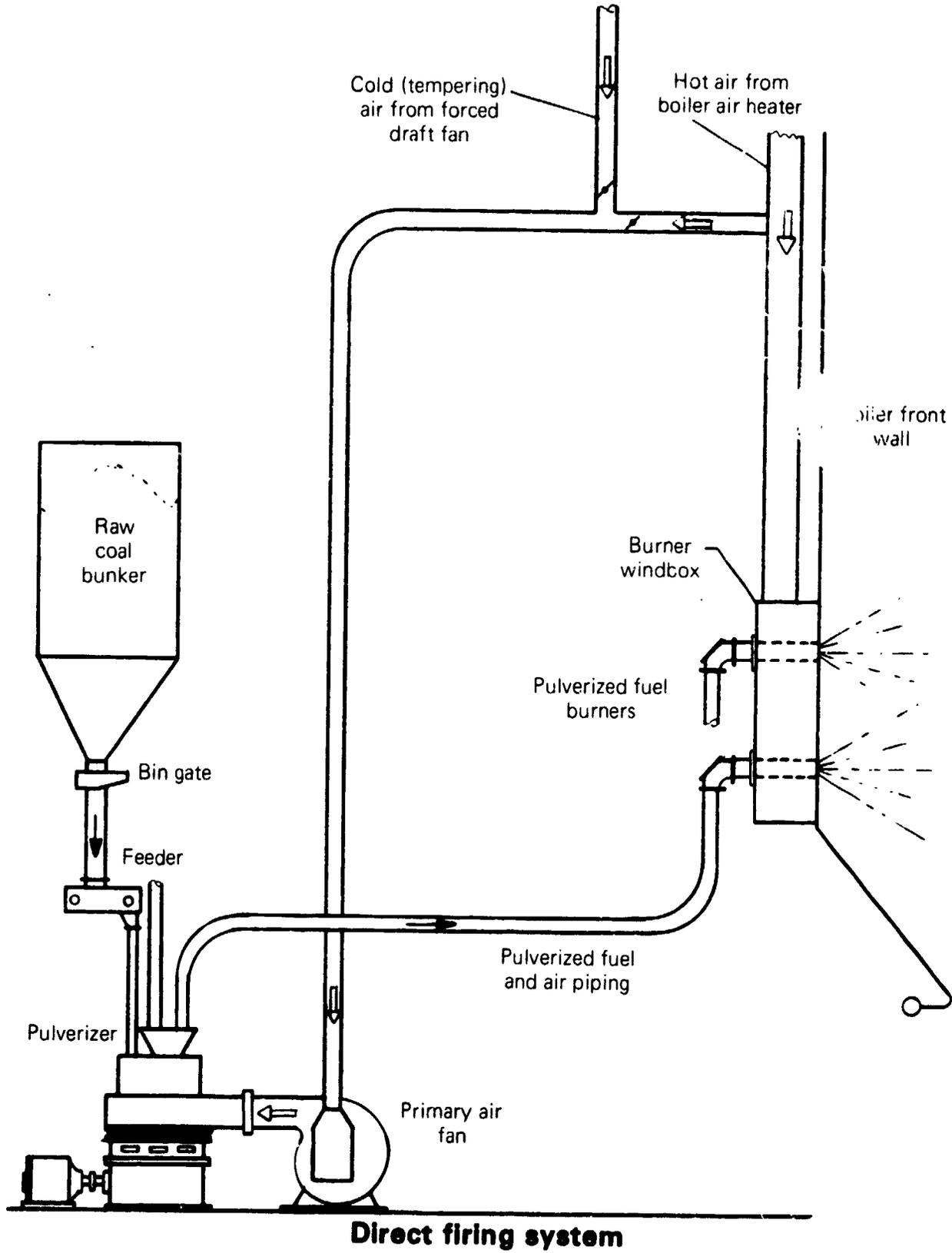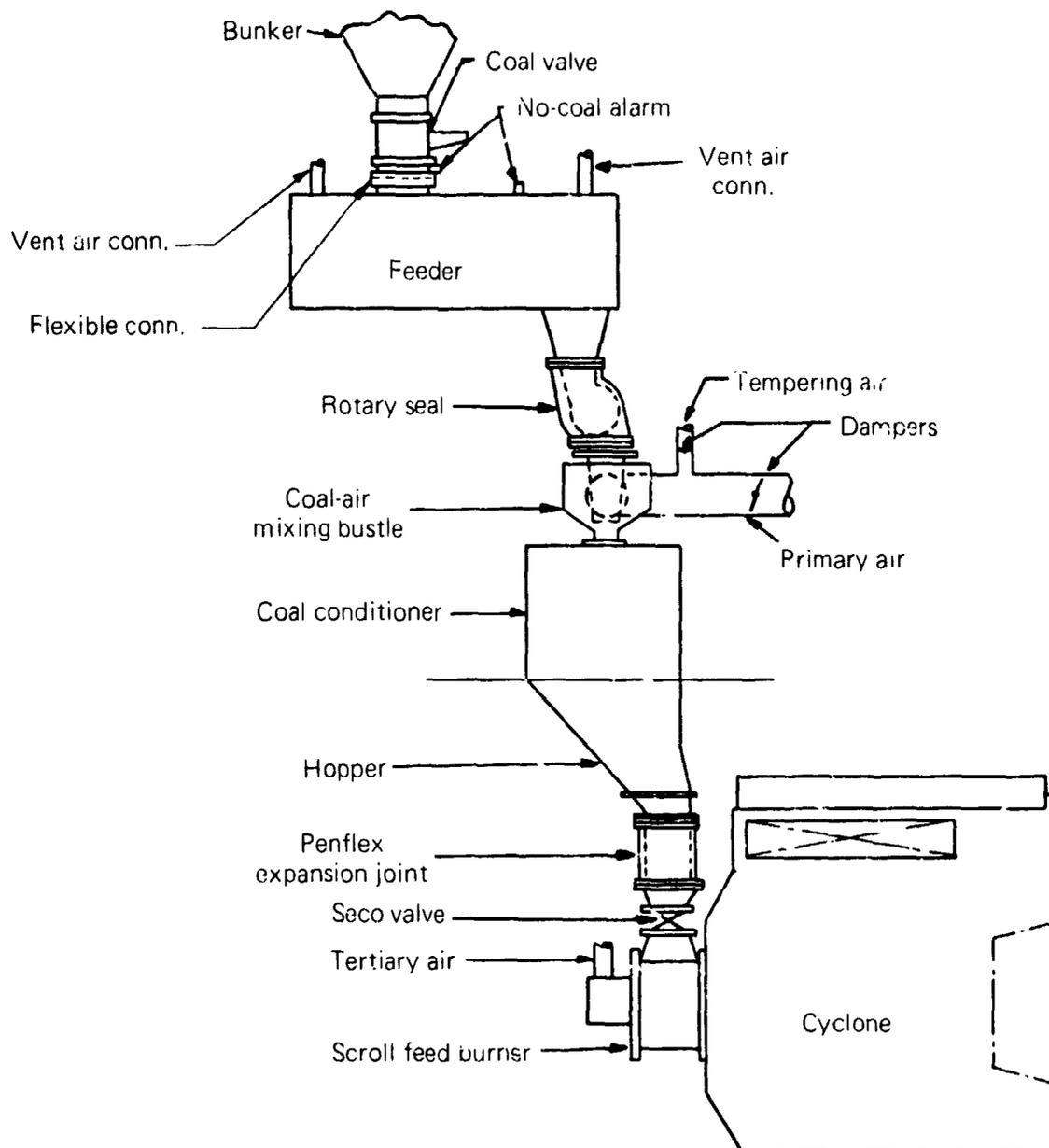
Figure 2

Direct fired gravity system

Figure 3
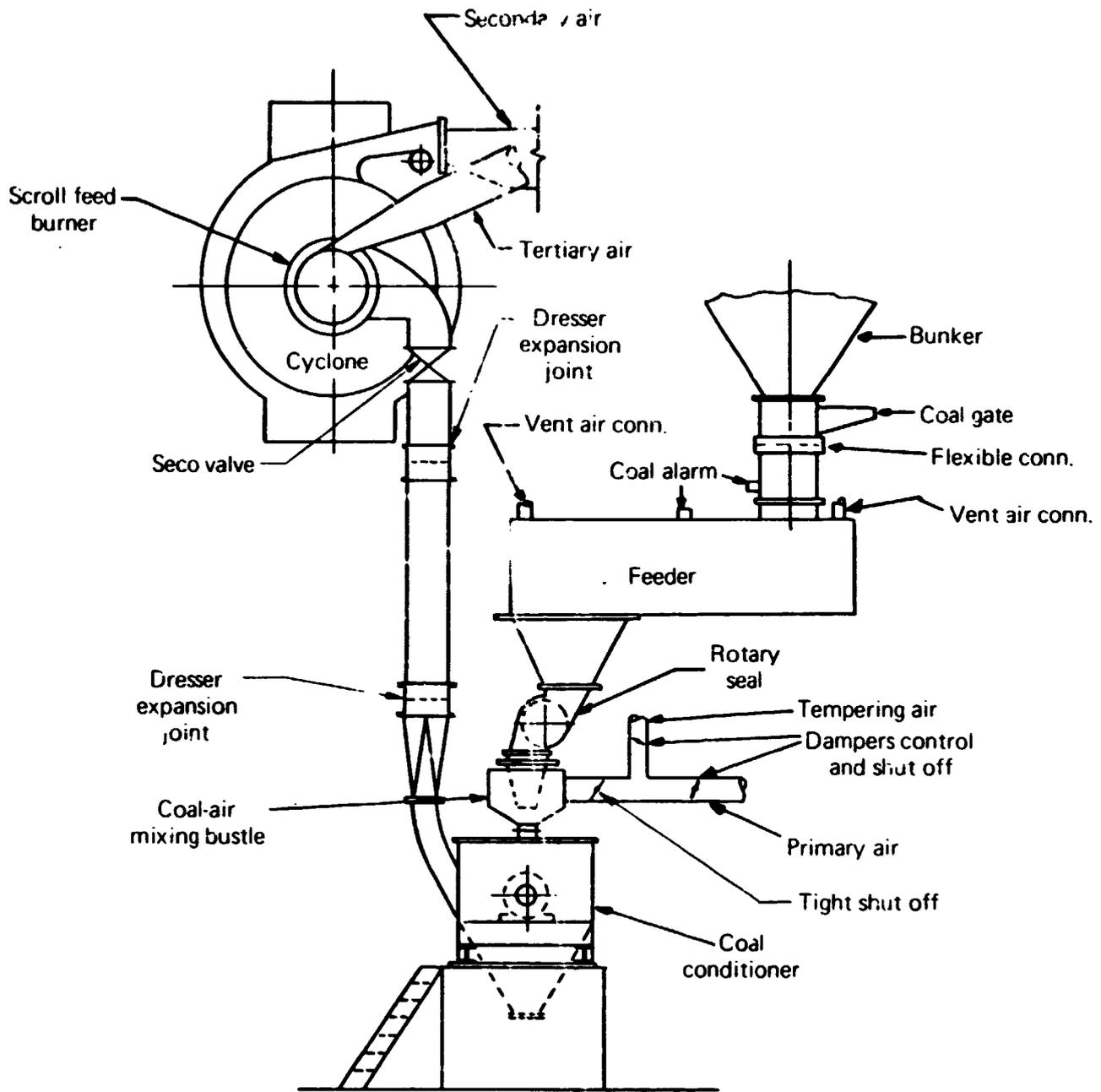
Direct fired ai .ift system

Figure 4
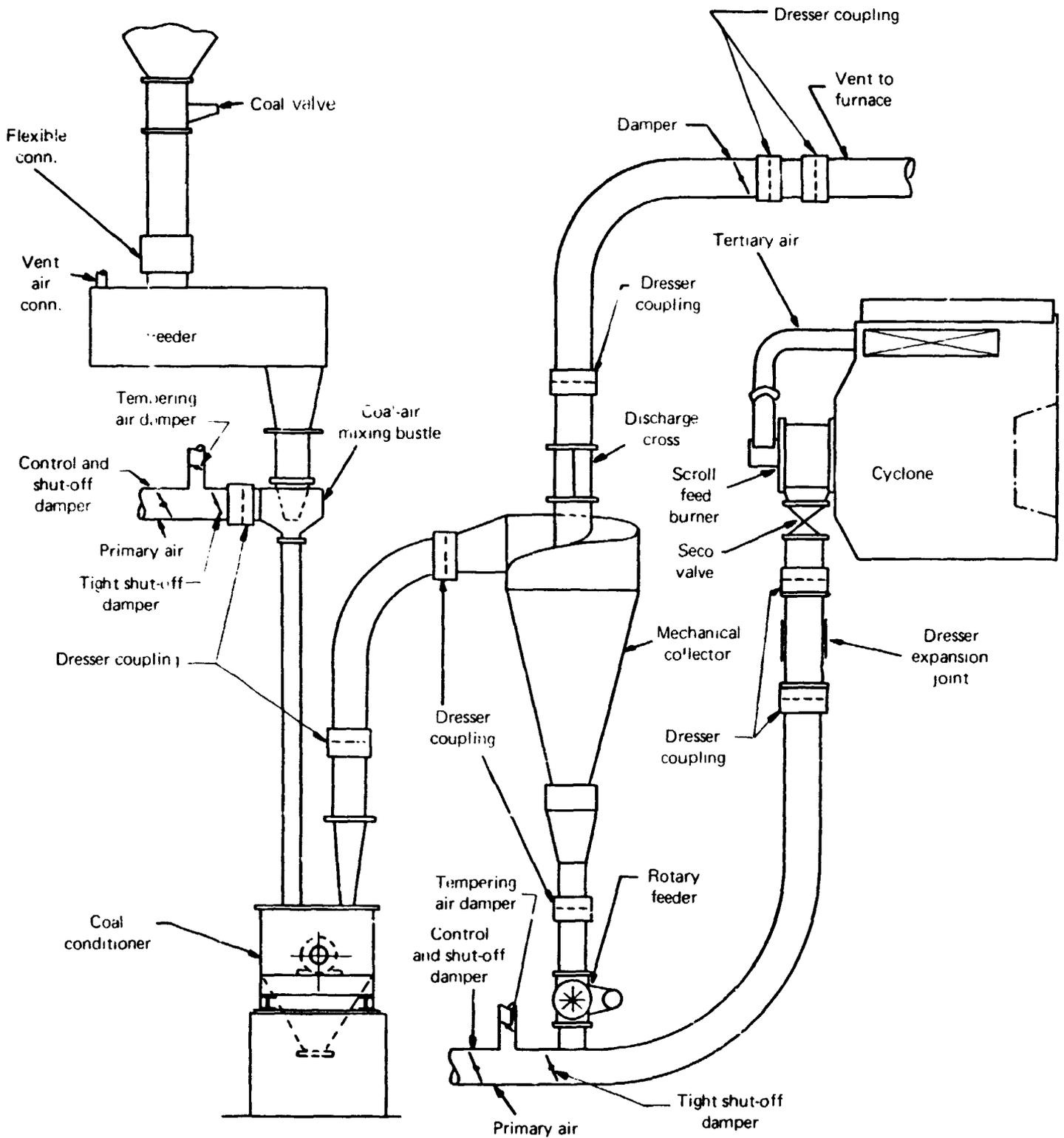
Direct fired, pre-drying by-pass system
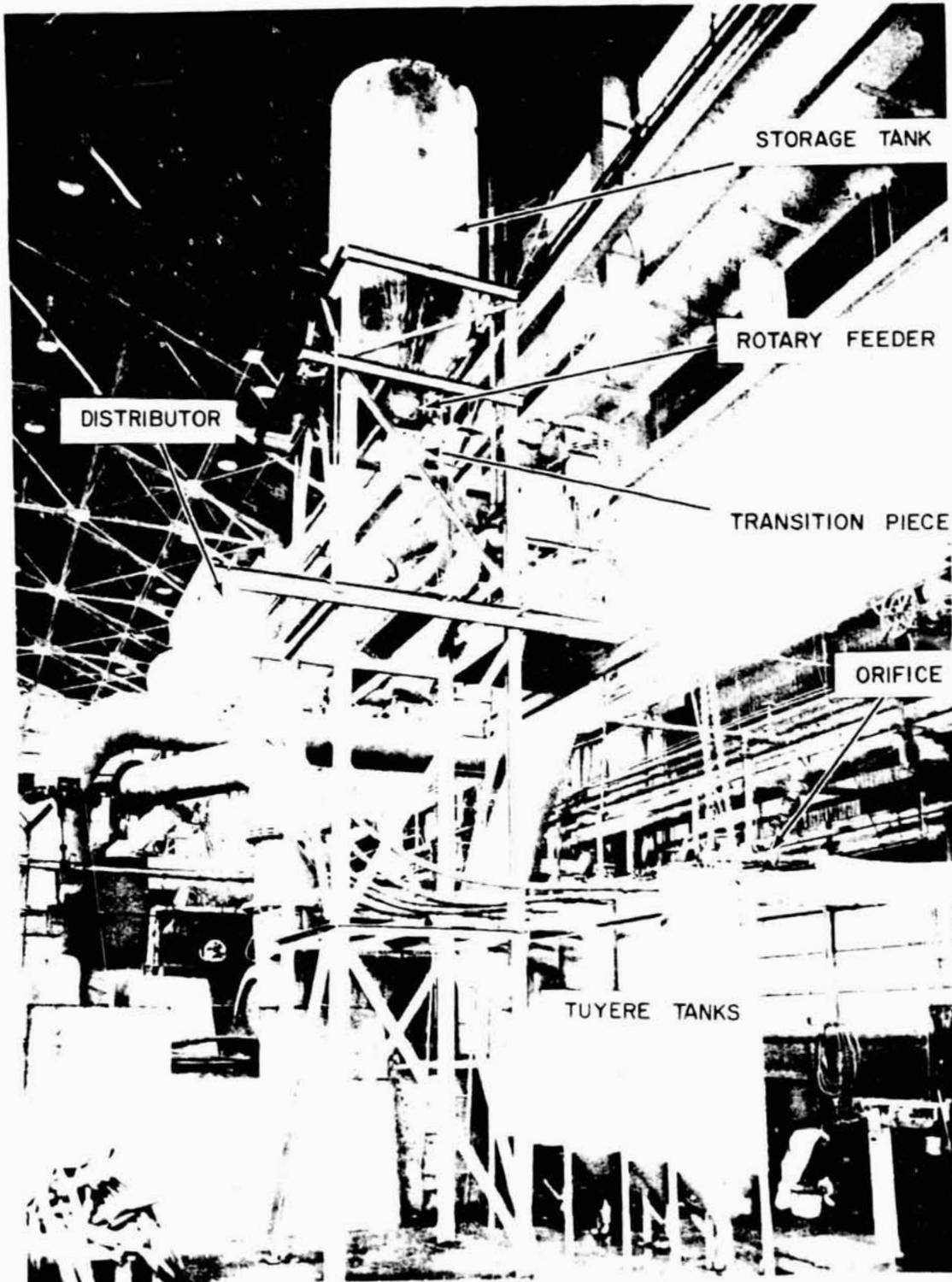
Figure 5

View of distributor test apparatus

Figure 6

View of coal distributor test apparatus
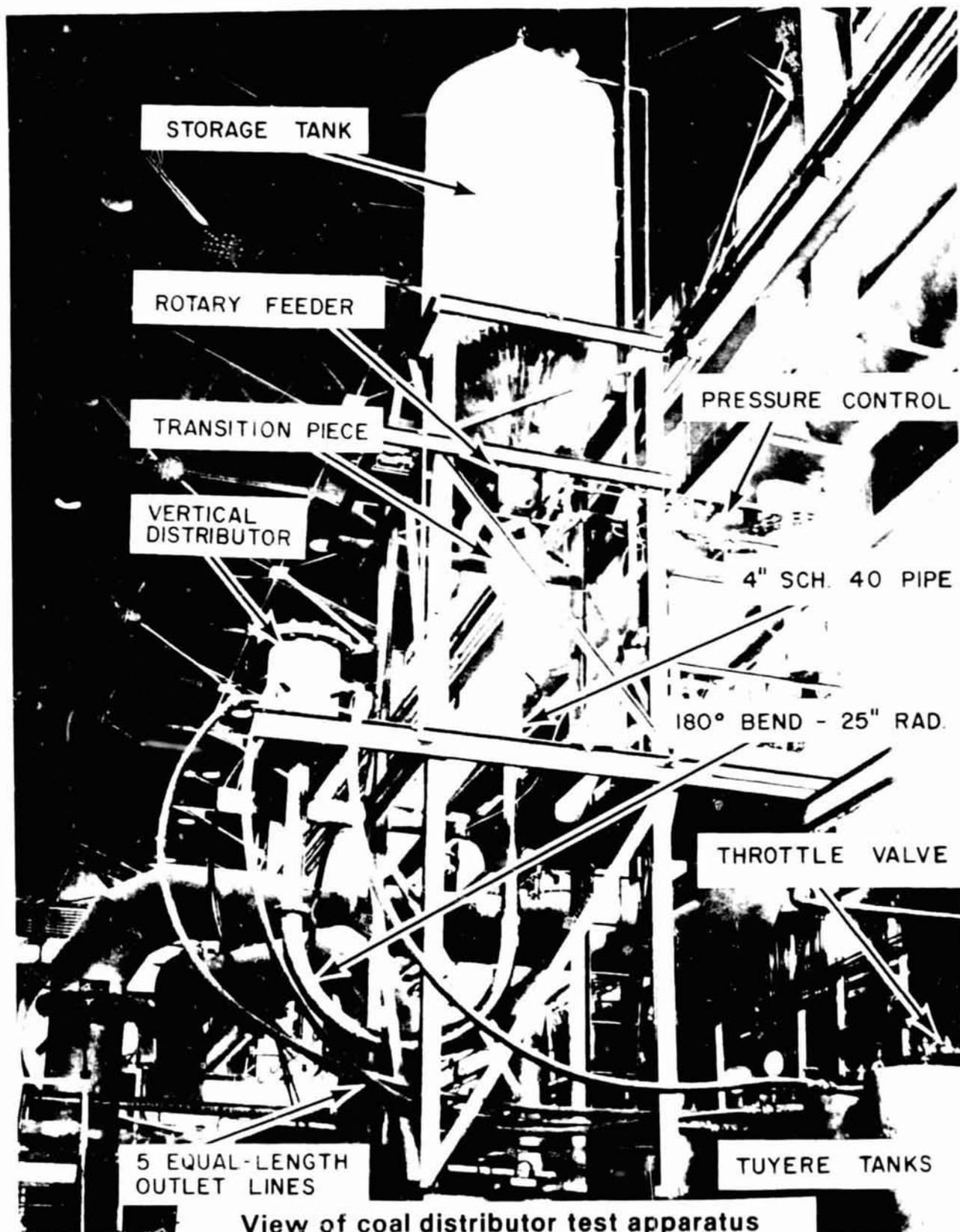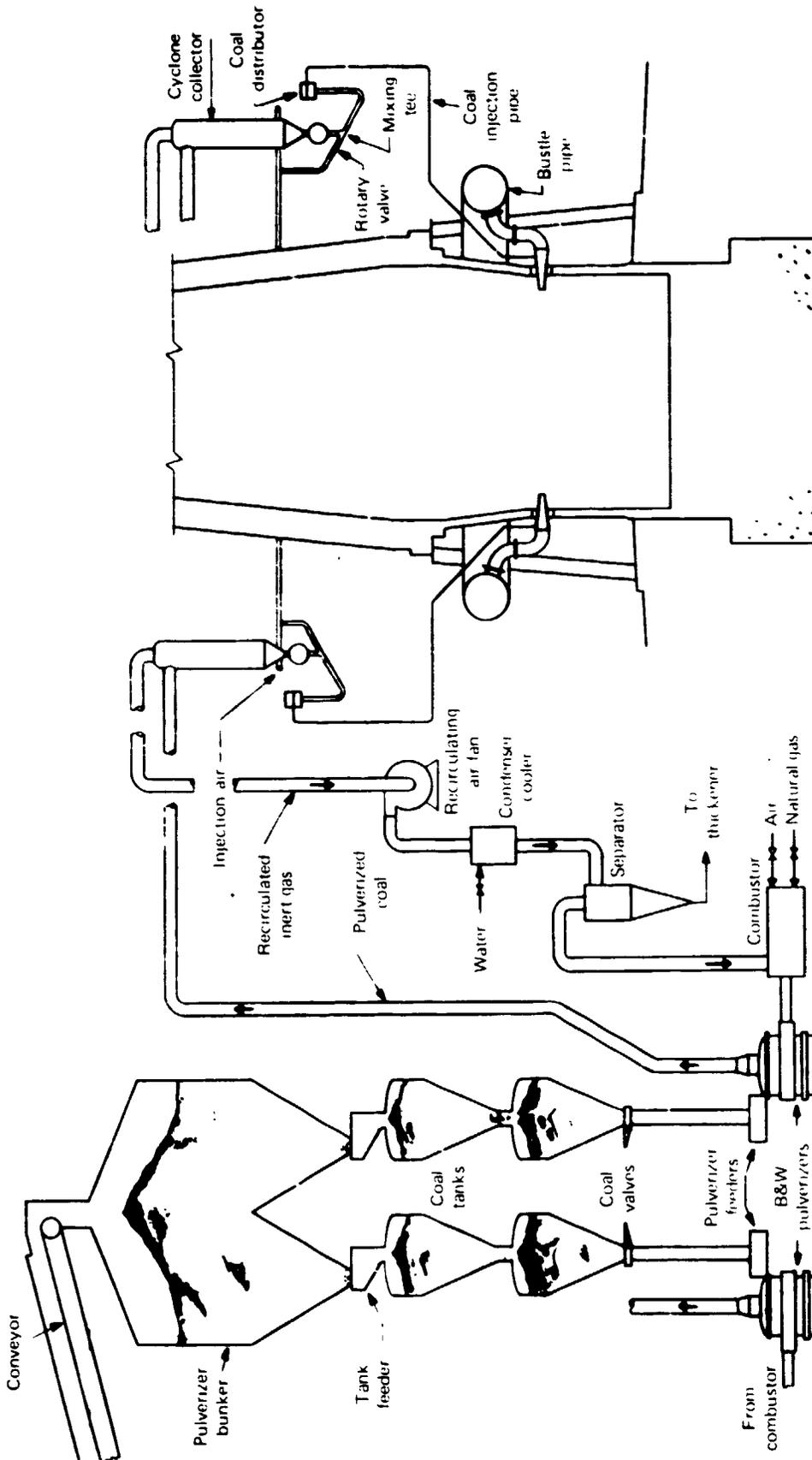
Bellefonte's pulverized-coal-injection system

Figure 8
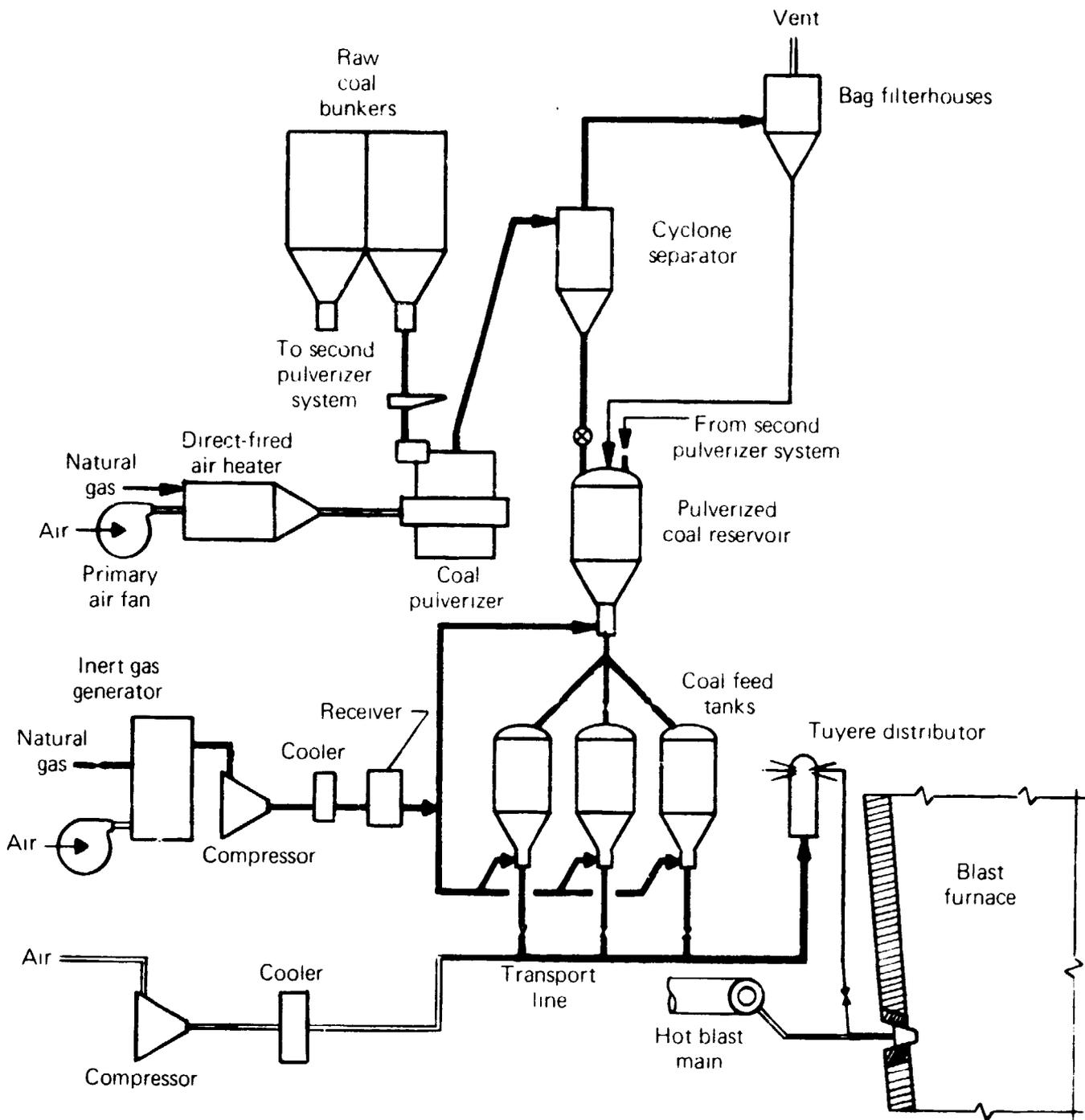
Amanda's pulverized-coal-injection system

Figure 9
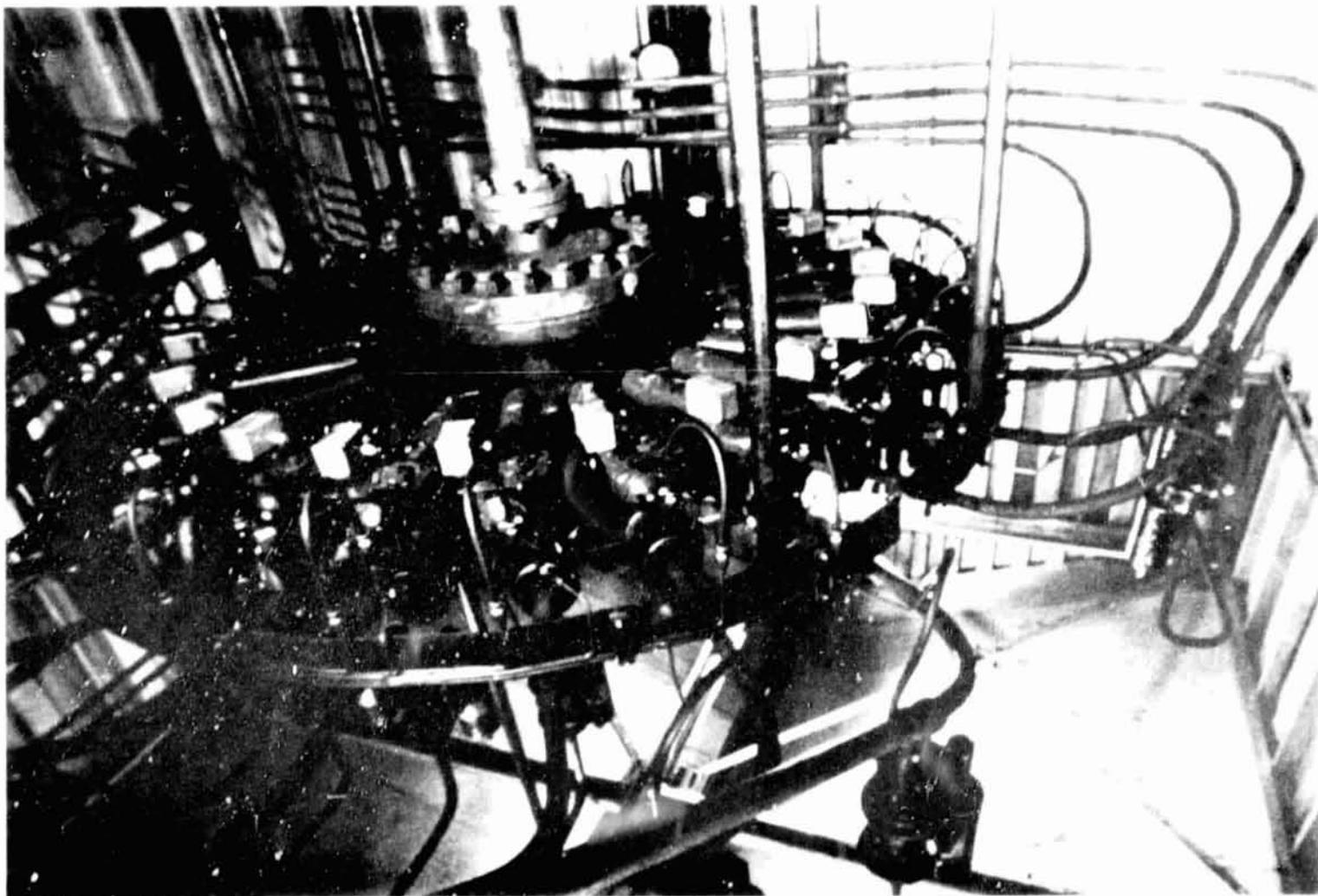
Bottom of distributor bottle

Figure 10
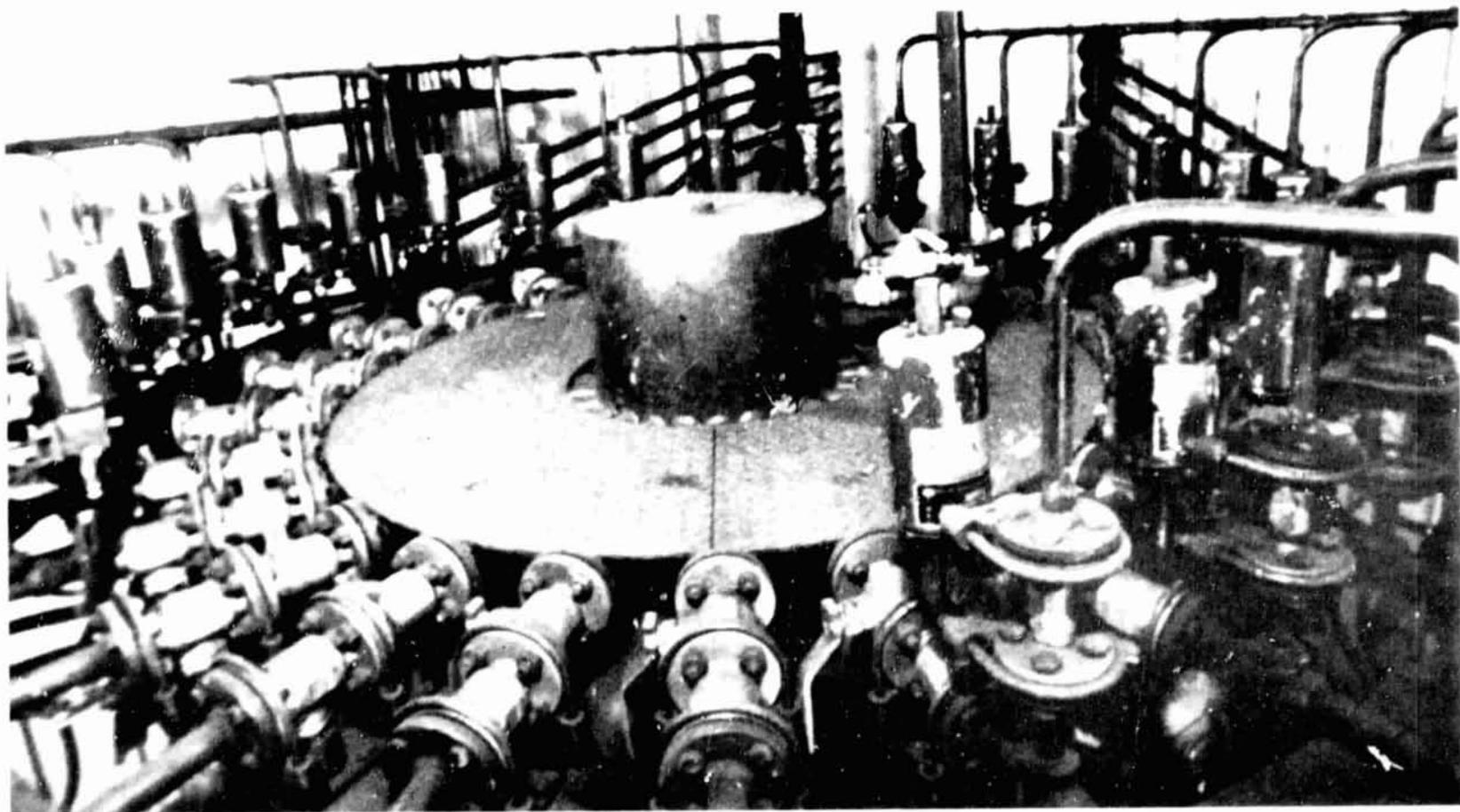
Top of distributor bottle

Figure 11
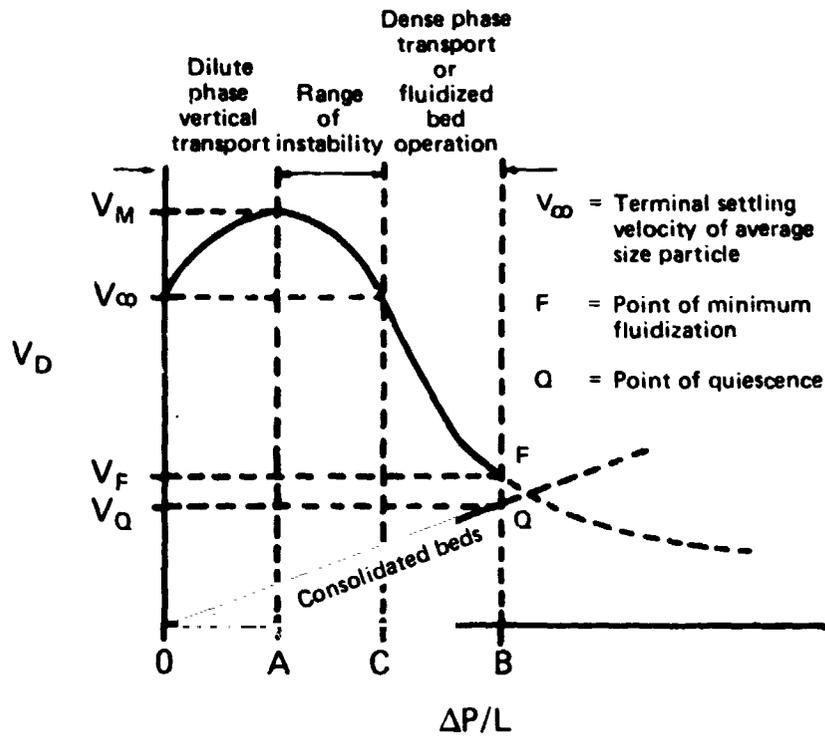
Slip velocity diagram for vertical flow system

Figure 12
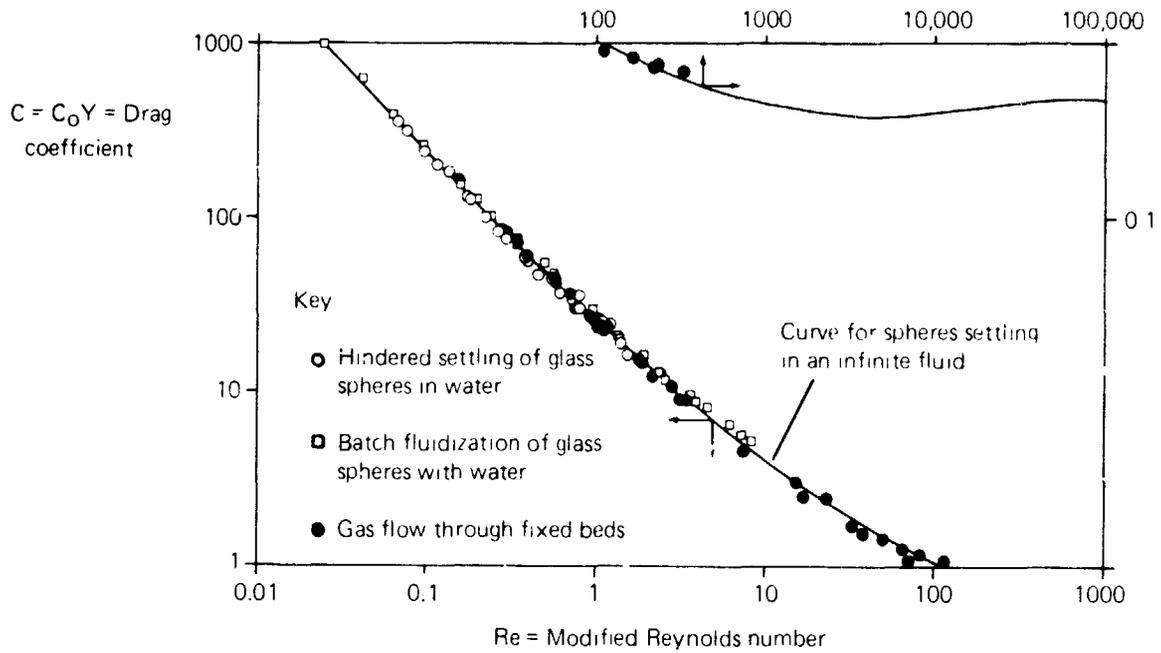
Generalized correlation of data on flow of
fluids through beds of granular solids

Figure 13
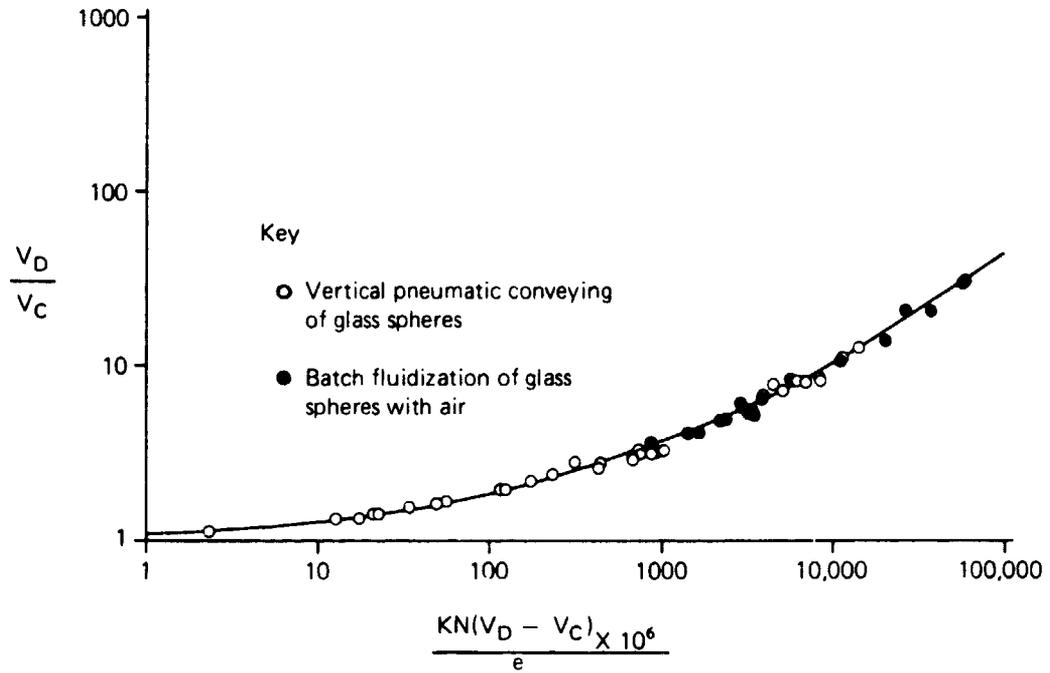
Generalized correlation of data on gas fluidization

Figure 14

V_O = superficial gas velocity
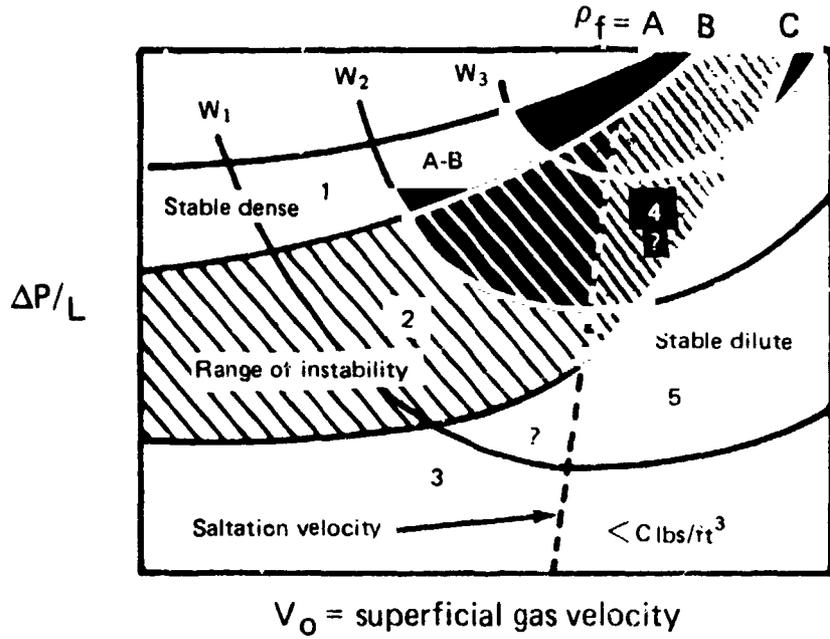
Performance curve for pneumatic transport system

Figure 15

PRESSURIZED FEEDING ON THE GEGAS

SYSTEM

A. H. Furman

General Electric Corporate Research
and Development
Schenectady, New York

ABSTRACT

A continuous process to feed coal directly into a pressurized gasifier is described. Coal fines are heated and mixed with a recycled tar binder and extruded through a novel die system against gasifier pressure. Performance data on a 2" system is given and scale up to a larger 6" system is described.

INTRODUCTION

The fixed bed gasifier has historically been limited
to coal of a restricted size and type. Coal fines below
1/8" were screened out and stockpiled because the fines,
if gasified, tended to plug up the bed, and if the air
flow was high, to be eluted from the bed before gasifica-
tion could occur. However, run of mine coal can contain
as much as 30% - 40% fines and with today's energy prices,
one is forced by economics to utilize this fraction. In
General Electric's gasification work, the approach taken
to utilizing the fines has been to extrude the coal
directly into the gasifier using tar as a binder. This
technique has the advantage of utilizing the gasifier
tar and coal fines, of eliminating the maintenance and gas
handling losses associated with conventional lock hopper
systems, and of enabling continuous feeding to pressurized
reactors.

Success of the operation hinges on the ability of the
extruder to meet the following goals: provide the necessary
sealing capability to withstand gasifier pressure; have the
ability to extrude the coal fines at a competitive cost
and power requirement; and have the size, reliability and
wear resistance to handle large tonnages of coal without
excessive downtime and maintenance.

Work has been done in the past in the area of coal
extrusion, such as the Coal Logs[1] work done in the
late 1940's or the more recent work by the Bureau of
Mines[4], but none of it has focused on the use of the
extruder as a pressure feeder, and this is where its real
advantage lies - in its ability to pump compacted coal
directly into a pressurized system. The present alterna-
tive to direct extrusion is to use commercially available
double roll presses to briquette the coal fines and to use
an additional lock hoppering step with its inherent losses
to inject the briquetted fines into the gasifier.

EXTRUSION PROCESS EQUIPMENT

A general schematic of the coal extrusion process is shown in Fig. 1. Screened coal fines are mixed with a tar binder and fed into the hopper of the extruder. The mix is conveyed down the barrel by the screw and compressed into a solid continuous rod of extrudate in the die area. A chopper downstream of the die is used to break the extrudate off into sized lengths one diameter long and may be used on larger machines to quarter the extrudate. Both the chopper and die are operated at gasifier pressure which in the GEGAS-D machine will be up to 23 atm. The feed hopper is at atmospheric pressure as is the extruder barrel, with the critical gas seal being maintained by the coal pack in the die. With this system, no lock hoppers are needed to feed coal into the gasifier. On the GEGAS machine, an isolation chamber with two valves will be provided between the extruder and the gasifier to permit off-line pressure testing, and to isolate the extruder from the gasifier in the event a leak develops or if system maintenance is required.

We have worked with five extruders in developing this process to the present pilot plant scale. The first machine was a modified one inch plastic extruder capable of 15 lb/hr of extrudate. Our present machine is a six inch capable of 4000 lb/hr. The following is a description of the equipment in more detail and a summary of the operating results.
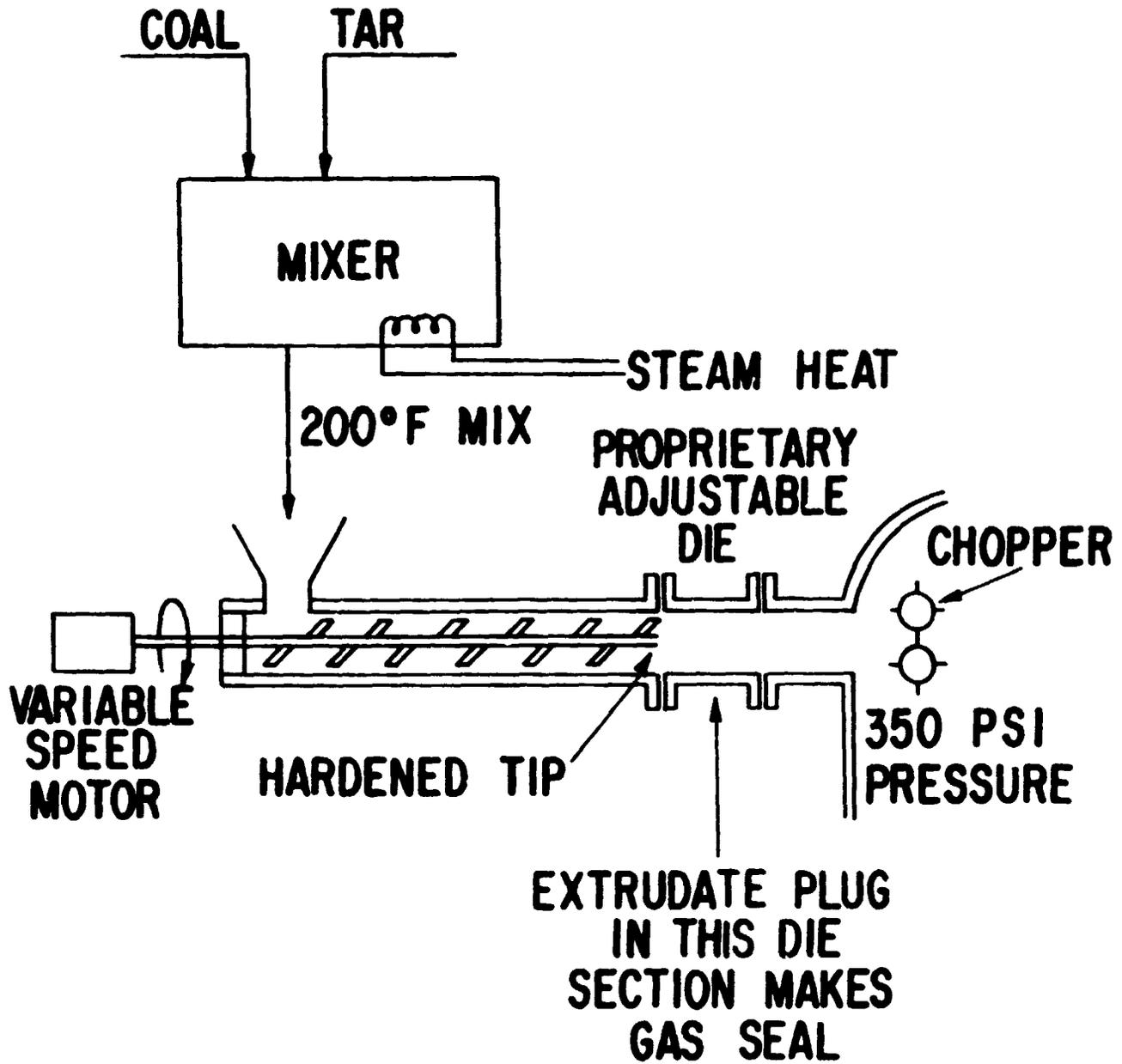
Figure 1.   Pressurized Coal Extrusion Feeding Process

ONE INCH EXTRUDER

Our original idea for coal extrusion was to use the plastic properties of the coal itself by heating it to approximately 750°F, at which point it softens to a semi-liquid which can be extruded. We used a one inch plastic extruder with a 24:1 electrically heated Xaloy barrel and a two stage vented screw made out of chrome plated 4140. This approach was dropped at an early stage, however, due to difficulties in controlling the process. Coal is an extremely variable substance and when heated its properties change rapidly with time. Large amounts of gas are evolved during the softening process and this plus the fact that the coal can change from a powder to a tarry consistency to a rigid solid as a function of time and temperature led us to abandon this in favor of a low temperature extrusion process using supplied tar as a binder.

Modifications were made to the one inch screw and hopper to permit feeding the dry coal tar mixture, and a fixed length die of one inch ID was added to form the extrudate. Using ash as a diluent and with the extruder directly connected to our one ft. laboratory gasifier as shown in Fig. 2, we made our first successful run on highly swelling Pittsburgh #8. Maximum output of the machine was 15 lb/hr at a specific power consumption of 17 lb/kw-hr. Density of the extrudate was on the order of 75 lb/ft^3.

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Figure 2. One-Inch Coal Extruder During On-Line Testing

The variability of coal became evident very early in our work as it was discovered that different coals required different back-pressures in the die to form a coherent rod of coal. Too low a backpressure and the density and strength falls off to the point where the extrudate is not strong enough to withstand the ΔP from gasifier to atmospheric pressure. Too high a back-pressure and the die will jam up. We also found that with binder percentages less than $\sim 14\%$ the coal mix would not flow through a die whose cross section was less than that of the barrel.

Screw wear also proved to be a problem with the one inch machine and as the screw wore the pumping efficiency fell off rapidly. The smooth Xaloy barrel liner while it stood up to the abrasion of the coal very well, also did nothing to help the pumping efficiency due to the low co-efficient of friction between it and the coal.

TWO INCH EXTRUDER

Using the operating experience gained on the one inch, we scaled the process up to a two inch extruder, the basic components for which were supplied by the Bonnot Company, located in Kent, Ohio. The machine has a large feed hopper, a short 6:1 barrel with a heat treated ribbed liner for high pumping efficiencies, and 17-4 ph heat treated stainless steel screw for erosion and corrosion resistance. The barrel is jacketed for 150# steam; the feed hopper is cooled to prevent hang up of material due to softening of the tar binder. On both this and the one inch machine, the coal and pulverized tar are premixed and fed cold to the extruder. Heat from the steam jacket and from frictional work being done on the coal during compaction is sufficient to melt and spread the tar binder. Screw speed can be varied from 10 to 90 rpm; power is supplied via a 3 hp electric drive.

TWO INCH DIE

To control extrudate formation on line using different
coal inputs, an adjustable back-pressure die was developed.
As in any extrusion process, the die is perhaps one of the
most critical components in the system, ours being no
exception, and much development work has gone into the de-
sign and construction of a die which could compensate for
the different operating conditions required of different
mixes. The die has two modes of adjustment - the surface
area in contact with the coal can be varied, and the cross-
sectional discharge area can be adjusted to give the proper
backpressure to the forming coal rods. The die is circular
in cross-section and can be varied from one diameter to
2.5 diameters in length. Adjustment can be made manually
via hydraulic actuators or can be accomplished automatically
using a feed back control system. Die material is stainless
steel.

An off line pressure vessel is used to simulate gasifier
pressure. The vessel is rated at 1200 psi and is bolted up
to the extruder and pressurized to the desired operating
conditions. Water is presently being used in the system for
convenience and for safety reasons in the event a leak
develops back through the extruder. A simple breaker plate
is used inside the vessel to break the extrudate off in
lengths about one to two diameters in length. The extruder,

its control system and feed hopper can be seen connected
to the pressure vessel in Fig. 3.
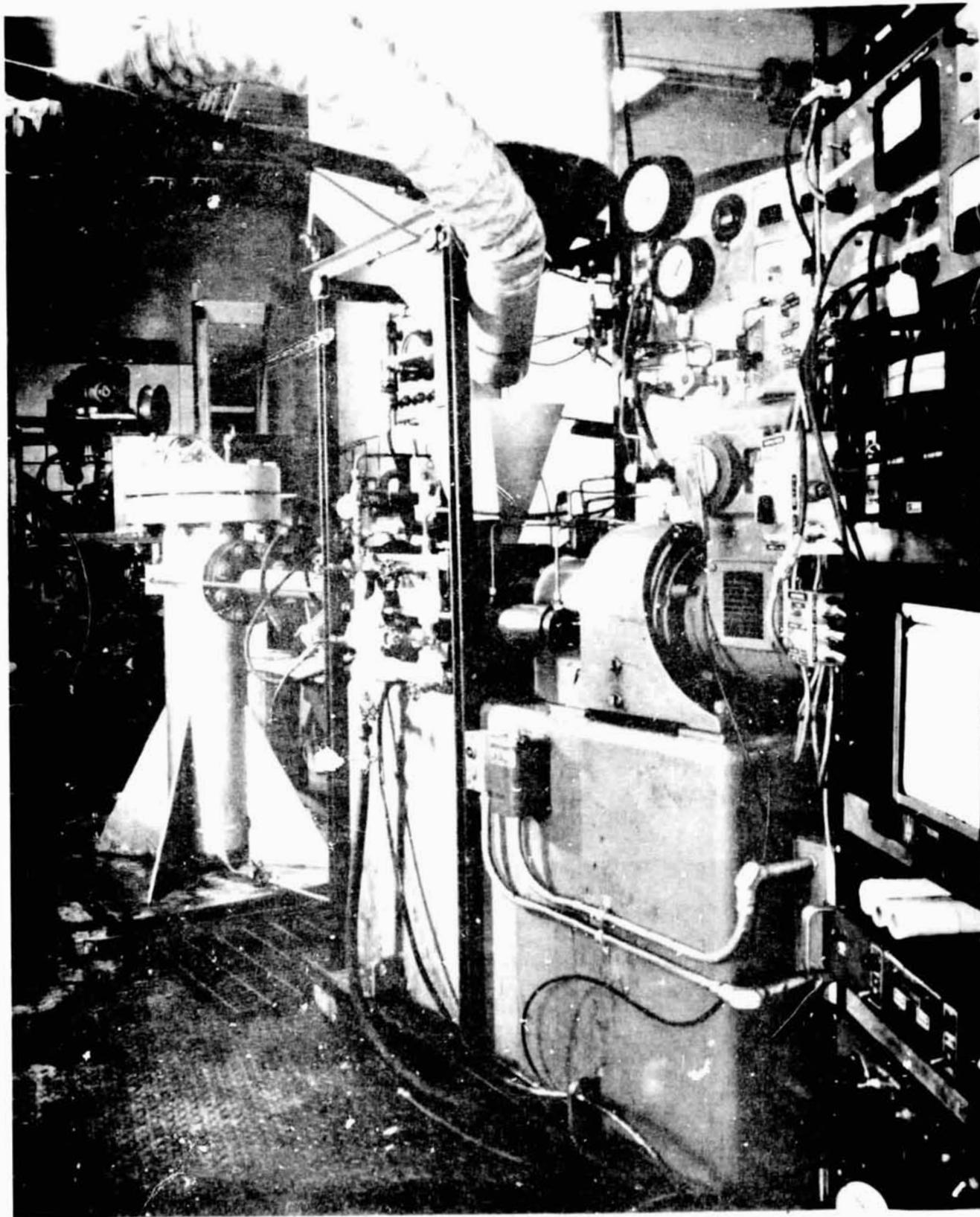
Figure 3. Two-Inch Coal Extruder Feeding Into 1000 PSI Pressure Vessel.

TWO INCH SCREW PERFORMANCE

Table 1 shows our operating results to date. Although
we have done test work on a variety of coals ranging from
lignites to anthrac'te, most of our work has concentrated
on Pittsburgh #8, which has become our reference feedstock.
All of the mixes which these results were obtained on were
90% coal, 10% asphalt pitch and all percentages are on a
weight basis.

Whether one chooses to extrude only part of the coal
feed or all of it depends on the power requirements and on
the cost of the extrusion equipment which, in turn, depends
on the capacity of a given piece of equipment in pounds per
hour/ft^2 barrel.

We have been developing the capability of the extruder
along these lines and have achieved a significant increase
in performance in scaling up from the one inch to the two
inch extruder. This is due to design improvements in the
barrel, screw, and die and also to more favorable surface to
volume ratios obtained with the larger machine. At one atm.,
the specific power consumption of 200 lb/kw-hr and output of
11,000 lb/hr-ft^2 barrel represents a significant improvement
over the one inch system. Higher numbers here are indicative
of better performance and we are aiming for > 200 lb/kw-hr at
a specific output of at least 20,000 lb/hr-ft^2 barrel on our
pilot plant system. To put the power numbers in perspective,

gas losses from a conventional lock hopper system at 23 atm. amount to ∿ 1.5%. The energy requirements to recompress and reinject this gas at 23 atm. is equivalent in energy terms to that required to extrude the coal. Furthermore, if one requires that the coal fines must be utilized in the lock hopper as is being done with the extruder, a briquetting operation will be required; the power requirements for this added to the lock hopper losses gives a rather high specific power consumption of ∿ 130 lb/kw-hr for the combination lock hopper briquetting operation.

We have continually been increasing the pressure to which we can deliver coal by extrusion and have successfully operated against pressures as high as 1100 psi at low delivery rates. Power requirement increases with increasing pressure as indicated by the extruder performance at 20 atm. This increase is greater than that suggested by the theoretical P-V work, however, and by judicious operation of the extruder and die, we feel we can bring these numbers down.

TABLE 1

Improved Performance With Extruder Scale-Up

EXTRUDER	HYDROSTATIC BACK PRESSURE (ATM)	SPECIFIC OUTPUT (lb/hr/ft^{2}bbl)	POWER CONSUMPTION (lbm/KWe - hr)
1"	1	2,700	17
2"	1	11,000	200
2" screw/ram	1	9,500	230
2"	20	9,000	120
6"	1	7,000	310
6"	20	5,00C	170

TWO INCH MACHINE WEAR

Machine wear has been a recurring problem with the process and harder materials have been employed to handle it. Coal is a very abrasive material and where there is sliding contact with the coal occurring under pressure, there is wear. On the two inch machine, both the barrel and die have held up exceptionally well and we are still using the same hardware after two years of test work. The screw is a different story, however.

We generally operate in the range of 3000 to 5000 psi on the coal, this being the pressure that the screw exerts on the coal to force it through the die against 20 atm. backpressure. Since the extrudate mix behaves almost as a solid in the barrel, it is not possible to gradually build up to this pressure over a section of the screw as can be done with a tapered compression zone on a plastics extruder. We are forced, instead, to develop the required pressures with only about the last two flights of the worm, and this is where our wear occurs. The rest of the worm while it does show some wear, is not subjected to nearly as severe conditions as the tip.

To combat the wear, we first tried flame spray coatings but these tended to chip and flake off. Stellite on top of a heat treated 17-4 stainless steel got our wear down to more reasonable limits while providing the necessary strength.

We have recently switched to a machinable titanium carbide
tip which is mechanically fastened to the extruder shank.
This is our second effort using carbide; the first screw
had a solid titanium carbide tip which was vacuum brazed
to the shank. The fact that this tip could not be re-
moved limited its flexibility, and the fact that it broke
under load limited its usefulness.   Wear with the carbide
is down in the 5 ppm range, where we are expressing the lbs
of screw metal lost per one-million lbs of mix extruded.
We would like to see the wear rate below 0.1 ppm and are
continuing to look at advanced materials and coating processes
to accomplish this goal.

Table 2 summarizes our results to date.

TABLE 2

WEAR RATES OBTAINED ON EXTRUDER SCREW

SCREW MATERIAL	WEAR, PPM
Carbon Steel	> 100
4140 Chrome Plated	> 100
Flame Sprayed Coatings	Chipped
17-4 Heat Treated Stainless Steel	30
Stellite® #1	16
Titanium Carbide (Ferrotic® CS-40)	5
Hi Chrome Cast Steel	0.5
Advanced Coatings	?

TWO INCH SCREW/RAM

The ability of the extruder to survive on coal is of paramount importance to the success of the project and we felt that in view of the wear problem with the screw, we should investigate an alternative means of fuel injection which didn't put as much of the burden on the rotating screw. This resulted in the screw-ram concept which combines the best of both a screw and a ram extruder.

We had built a small ram extruder early in our work and while it showed little promise due to low potential throughputs and sealing capability, it did operate with very low erosion rates and was capable of developing high ram pressures.

After extensive design work, the existing two inch screw extruder was modified into what we feel is a very versatile piece of test equipment which can be run as a screw extruder, a ram extruder or a screw/ram. Fig. 4 shows the machine during off-line atmospheric testing.

Operation as a screw/ram is similar in concept to an injection molding machine. On the feed stroke the hydraulically powered screw rotates to feed the coal/tar mix forward while at the same time retracting from the barrel. At a preselected point, the screw stops rotating and hydraulic actuators ram the screw forward to compact

_ae coal in the die area. Hydraulic pressure is then
relieved on the actuators and the screw begins the feed
cycle again. Wear was expected to be substantially
reduced on the screw tip because during conditions of
high face loading, i.e. when the coal is being forced
through the die, the screw is non-rotating and hence
there is no relative velocity between it and the coal.
The adjustable circular die from the two inch machine
is used and the axial force which can be developed on the
coal is only limited by the size of the actuator.
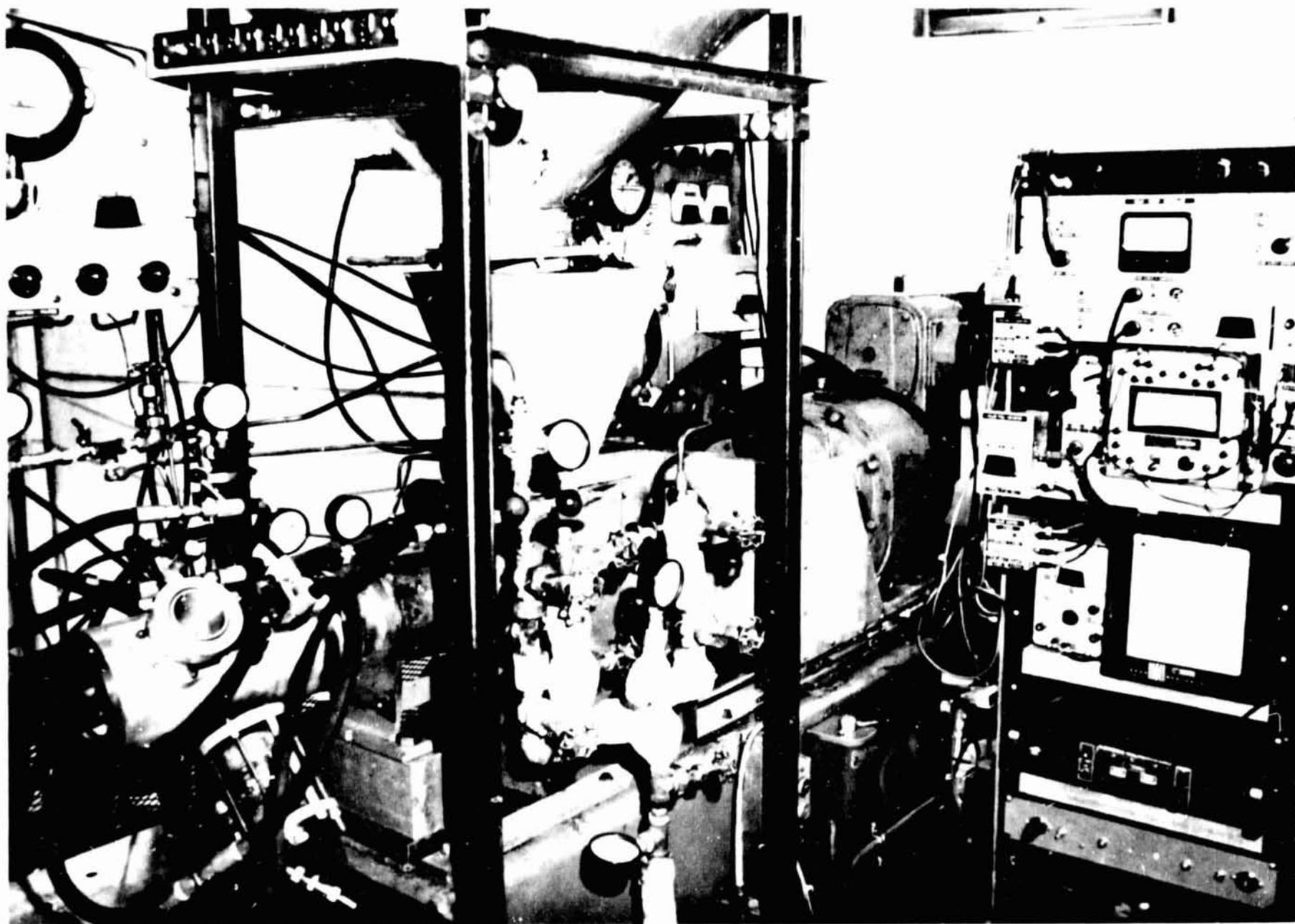
Figure 4.   Two-Inch Screw/Ram Coal Extruder

77-55

SCREW/RAM PERFORMANCE

The performance of the machine at one atm. was better than expected. Smooth steady operation was the rule rather than exception on a sampling of ten different coals run. Screw wear was reduced by almost 30%, power consumption was 230 lb/kw-hr, and specific deliveries approached 10,000 lb/hr per ft^2 of barrel area. While axial pressures as high as 20,000 psi could be exerted on the coal, best results were obtained in the 3000 to 5000 psi range. Pressures much higher than this caused surface degradation of the extrudate, resulting in a weaker rather than stronger briquette.

Operation at pressure with the screw/ram proved to be a problem. We could not reliably maintain a gas seal against backpressures greater than 23 atm. Leakage through or around the coal plug would begin to develop as the screw started its feed stroke. The loosely compacted coal being fea in front of the screw did not provide enough support for the compacted coal plug in the die. A tapered die with a slightly diverging cross-section was tried in an effort to force the coal plug to act more like a cork, but we still could not continually maintain a seal on the back stroke at pressures higher than about 20 to 25 atm.

By restricting the rearward movement of the screw
during feed, however, we were able to seal at pressures
in excess of 40 atm, but because of the additional work
being required of the screw, both power and wear increased
to the point where the advantages of the screw ram concept
over the straight screw began to diminish.

SIX INCH EXTRUDER

General Electric is currently engaged in a joint gasification program with E.P.R.I. as part of a contract to build and operate a stirred fixed bed gasifier capable of processing 2000 lb/hr of coal at 23 atm pressure. The facility is located at the General Electric Research and Development Center in Schenectady, New York.

To feed coal to the gasifier, both a lock hopper system and an extruder will be used. Incoming run of mine coal will be screened to remove the size fraction below 1/8". These fines will be sent to the extruder while the above 1/8" will be lock-hoppered into the gasifier. Both feed systems are sized such that they can each handle the full output of the gasifier independently of each other.

The GEGAS-D extruder is a six inch heavy duty screw type powered by a 60 hp SCR drive. It is nominally rated at 2000 lb/hr but on the basis of our results so far, we feel this is a conservative number. The hopper of the machine is equipped with two counter-current packer wheels which help cram the hot mix into the screw cavity. The screw itself is of the segmented auger type with individual cast sections stacked along a central drive shaft to give a continuous worm. With this design, screw geometry can be varied quite easily and worn areas of the screw can be replaced without removing the entire screw. L/D of the machine is 4:1 and the barrel is made up of two steam jacketed

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sections each equipped with a hardened and ribbed steel barrel liner. The die is a scaled up version of the two inch and is a combination variable area-variable length. Die material is stainless steel.

For off-line pressure testing, a 700 psi chamber is used as an extrudate receiver and is isolated from the extruder with a high speed knife gate valve in the event that the pressure seal is lost in the extruder die. During on-line testing against gasifier pressure, this receiver will be connected to the pressurized coal feed auger on the gasifier and will house a hydraulically powered chopper. Curved teeth on a rotating flywheel are used to break the solid 6" logs into pieces small enough to be fed by the auger. A 10" Everlasting® slide gate valve mounted between the gasifier auger and receiver isolates the extrusion system from the gasifier during start-up and during emergency shutdowns.

As a safety interlock, valve actuation is protected by a logic system which prevents an operator from opening the valve unless the pressure in the receiver and gasifier are equalized. Pressure equalization is handled by a small line using high pressure N_2, allowing the operator to test the extruder seal before exposing it to full operating conditions by opening the valve. Both the Everlasting® valve and the emergency 10" knife gate valve mounted between the extruder die and receiver are designed to close against a column of extrudate. With both these valves closed and the receiver vented,
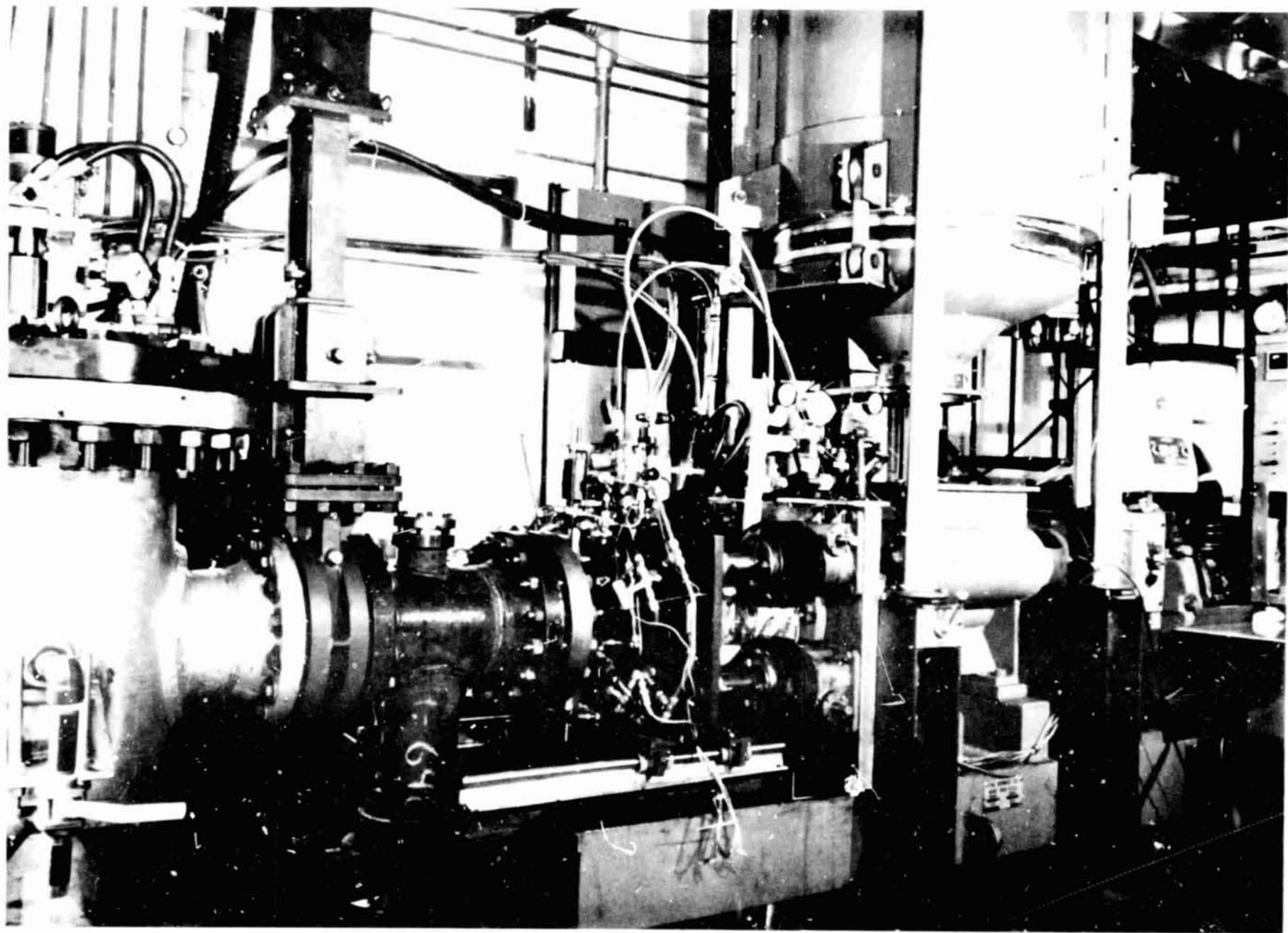
Figure 5. Six-Inch GEGAS-D Coal Extruder Connected to Pressure Receiver

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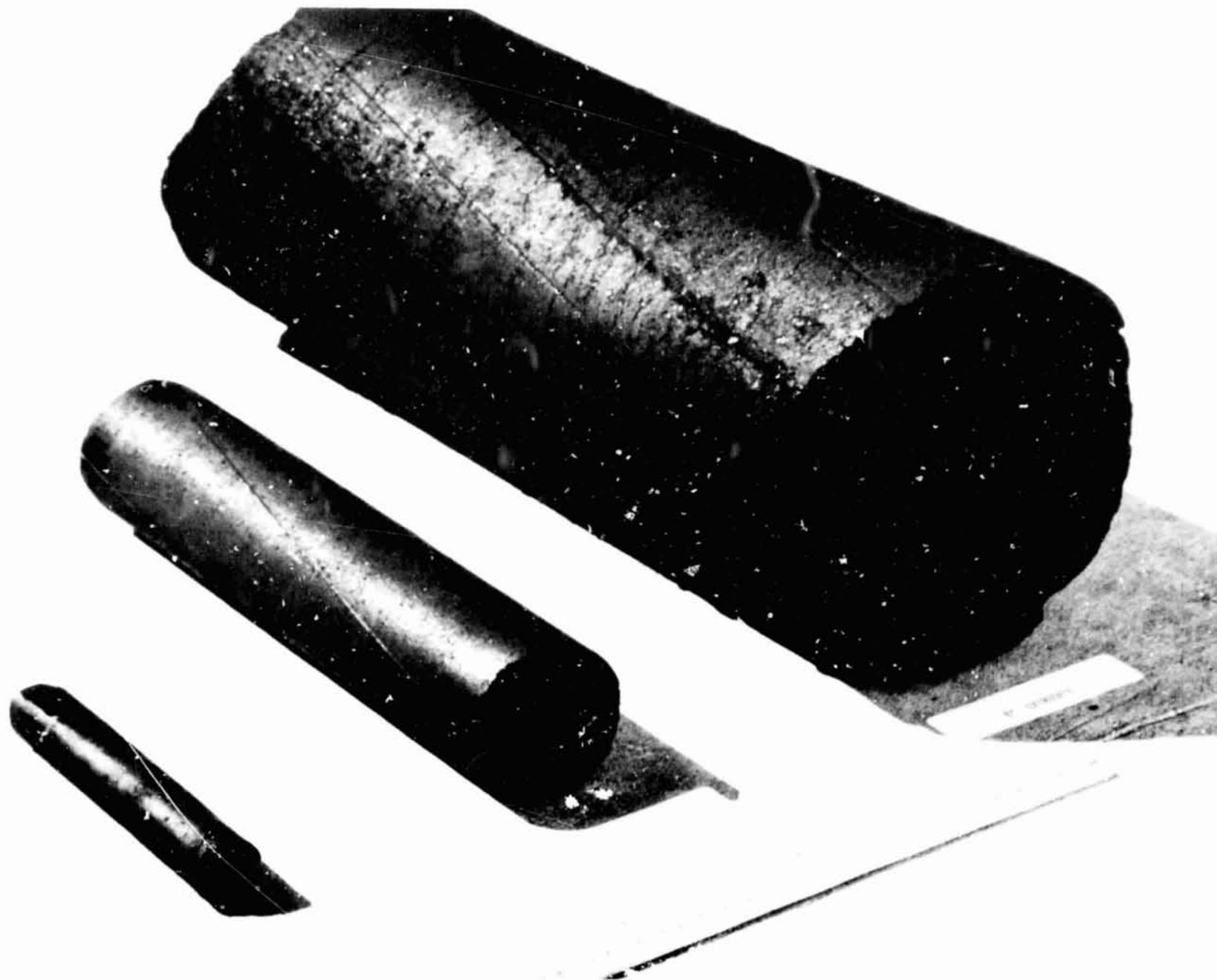
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Figure 6. One-Inch, Two-Inch, Six-Inch Coal Extrudate

the extruder can be isolated at essentially atmospheric conditions while the gasifier remains at operating pressure.

Hot mix is supplied to the extruder via a batch paddle mixer and a conveyor system. Preheated 250°F mix is being used instead of the cold milled tar and coal used previously, because with the larger cross-sectional area of the six inch and the deeper screw flights, the heat transfer between the walls and mix is insufficient to melt and spread the tar binder. Using a separate mix system also gives us more system flexibility and allows us to separate the extrusion problems from the mix prep problems. Tar injection into the mixers is handled hot on a weight percentage basis. Figure 5 shows the extruder connected to the receiver/chopper in preparation for air backpressure tests on the gasifier. Figure 6 shows the comparative size of the extrudate produced from the one inch, two inch and six inch machines.

Six Inch Performance

To date about 30,000 lbs. of mix have been extruded in a series of off-line tests at backpressures ranging from 1 to 33 atm. This new system has gone through check-out runs and operating results so far suggest better than anticipated performance at one atmosphere. Extrudate has been delivered at 1 atm. at a rate of 7000 lb/hr-ft^2 bbl at > 310 lb/kw-hr. power consumption, with a projected maximum delivery in excess of 4000 lb/hr. or 20,000 lb/hr-ft^2 bbl area specific delivery.

By increasing operating rpm of the screws, the output per
machine could be increased still further, but would be limited
ultimately by the point where the feed screw begins to starve
feed.

Several short duration tests have been made off line against
hydrostatic heads up to 33 atmospheres. Although run times have
been short due to the rather limited volume of the receiver/
chopper housing, the operating data does show that the extruder
can deliver extrudate against a 500 psig head. Water was used
as the pressurizing medium in these tests for reasons of con-
venience and safety.

As Table I indicates, specific power consumption at pressure
has been lower on the 6" machine than the 2" extruder (170 lb/kw-
hr. vs 120 lb/kw-hr for the 2"), indicating that scale-up as well
as equipment improvements have again been effective in lowering
the process power requirements. The specific delivery of 5000
lbm/hr-ft^2 hbl shown for the 6" machine is artificially low due
to the low delivery rates dictated by the limited volume of the
off-line chopper housing/receiver. These power figures do not
include the work required to heat and blend the tar-coal mixture.

Test results have indicated that the extruder density has a
direct relation on the sealing capability of the machine. Satis-
factory gas seals have been c' ined against 33 atm. backpressure
with effective coal plug lengths as short as 4.5". Power consump-

tion is also a function of density however, and an operating trade-off has to be made. On the 6" machine this means densities of 73-78 lb/ft^3 to obtain gas seals at 33 atm. backpressure with reasonable power consumption.

Control of extrudate formation using the variable length variable area die scaled from the two-inch machine has generally been acceptable, although there are periods of instability where power and density fluctuations occur. These are most prevalent during start-up and shut-down and hopefully will not present a major problem during long term running.

Six Inch Wear

Wear on the auger parts has been below 0.5 ppm, which represents a ten fold decrease over measured wear on the two-inch screw. Auger material is a high-chrome, cast steel with a R_c hardness of 60-67. This is not a particularly hard or exotic material and the wear reduction obtained is felt to be a combination of both materials and more favorable geometric factors obtained from scale-up. The die has also held up well and the original hardware is still in service with no wear problems as yet. Long term testing involving several hundred tons of extrudate will ultimately be needed, however, to determine wear rate projections.

Current Tests

The extruder is presently connected to the GEGAS fixed bed gasifier. Unfired air tests are in progress now using the gasifier as a backpressure vessel. Based on the success of these tests, several long term runs with the gasifier operating

at 20 atm. on 100% extrudate feed are anticipated. These tests will supply
not only critical data needed to determine the viability of the extrusion
process but will also determine how the extrudate behaves as a gasifier feed
stock. Gasification rates, fines carryover and tar mass balances are parameters
to be measured. Success of these tests would pave the way for development of
a commercial sized ten or twelve inch extruder by supplying the critical relia-
bility and operating cost data needed to compare this approach with other
feeding systems. Based on current data, three of the twelve inch extruders
would supply the 20 tph feed data to a commercial gasifier of twelve foot
internal diameter.

DISCUSSION

Process Variables

Coal's variability even within seams is legendary. This has contributed to the large fraction of art in the business of coal utilization. Each coal behaves differently when burned or gasified and in our extrusion work, we have found that no two coals extrude exactly alike. While the best way to determine how a particular coal will behave is to actually extrude it, there are certain variables which give us a clue to a coal's performance. These are the rank of the coal, the particle size distribution of the coal and the moisture content of the coal. Two other important variables, not a function of the coal type but directly related to the process, are the amount and type of binder used and the addition of other components to the extrudate mix.

It has been our experience that the higher rank coals are generally easier to extrude, anthracite being the easiest and some of the sub-bituminous and lignites being the most difficult. There are most likely exceptions to this rule, but with the coals we have run, this has proven to be the case. As for the reason we don't have a clear cut answer. Electron micrographs taken of the different coals show differences in the surface structures which could account for some of the differences in extrudibility.

The size distribution of the coal mix is an important variable. One would theoretically like a bell shaped distribution curve to give an optimum packing density to the extrudate. Unfortunately, in using screened run-of-mine coal, we take everything below 1/8" and are therefore unable to control our size distribution. This distribution in turn varies with the mining technique used and type of coal being mined. In general, too little fines and a weak extrudate is obtained because of poor density; too high a fines percentage and the binder content must be increased to compensate for the increased surface area.

Moisture content effects the handling of the coal, the strength of the extrudate and the power required to extrude it. We generally like to see the moisture content in the 3 to 5% range. Handling problems with screening and conveying the coal arise with moisture contents above this. Drying to below 3% on the other hand, penalizes the process unnecessarily because power is being wasted to excessively dry the coal. Also, more power is required to extrude the drier mixes. The moisture in the coal has the added advantage of helping to spread the tar binder during the heating stage.

The amount and type of binder used is a critical variable in the process for both economic and physical reasons. Coal tar pitch obtained from the spray quench system of the gasifier will be used in our pilot plant system. This use

of tar as a binder provides a means for recycling what is otherwise a difficult disposal problem. Whether the system will be self sufficient in tar will depend on the type of coal being gasified and the amount of tar required for the extrusion process.

Ideally, we would like to see neither a surplus nor a deficiency of tar. Presently, we are using binder contents in the range of 6-12%, the actual percentage used being a function of the mix being extruded and the type of tar being used. In our lab work, we are presently using asphalt pitch because it is obtained locally, easy to work with, and does not represent a health hazard as do some of the coal tars. Other binders such as bentonite and sulfite liquor have also been used.

With the extrusion process, it is possible to blend coals with other components to obtain extrudate with specific properties. The use of ash as a diluent is a general technique to reduce the swelling tendencies of coal by separating the coal particles and reducing their tendency to stick together. In extruded coal, the ash appears to have a rather specific effect in reducing the measured swelling index of the extrudate and it does so by an amount greater than proportionate to the amount of ash used. Figure 7 shows the effect of added ash on FSI for a Pittsburgh #8 coal. This technique has allowed

us to process highly caking coals in a conventional fixed
bed gasifier without stirring. The ash is, however,
quite abrasive and does put an additional load on the
extruder by requiring higher throughputs due to the dilu-
tion of the mix by the inert ash.
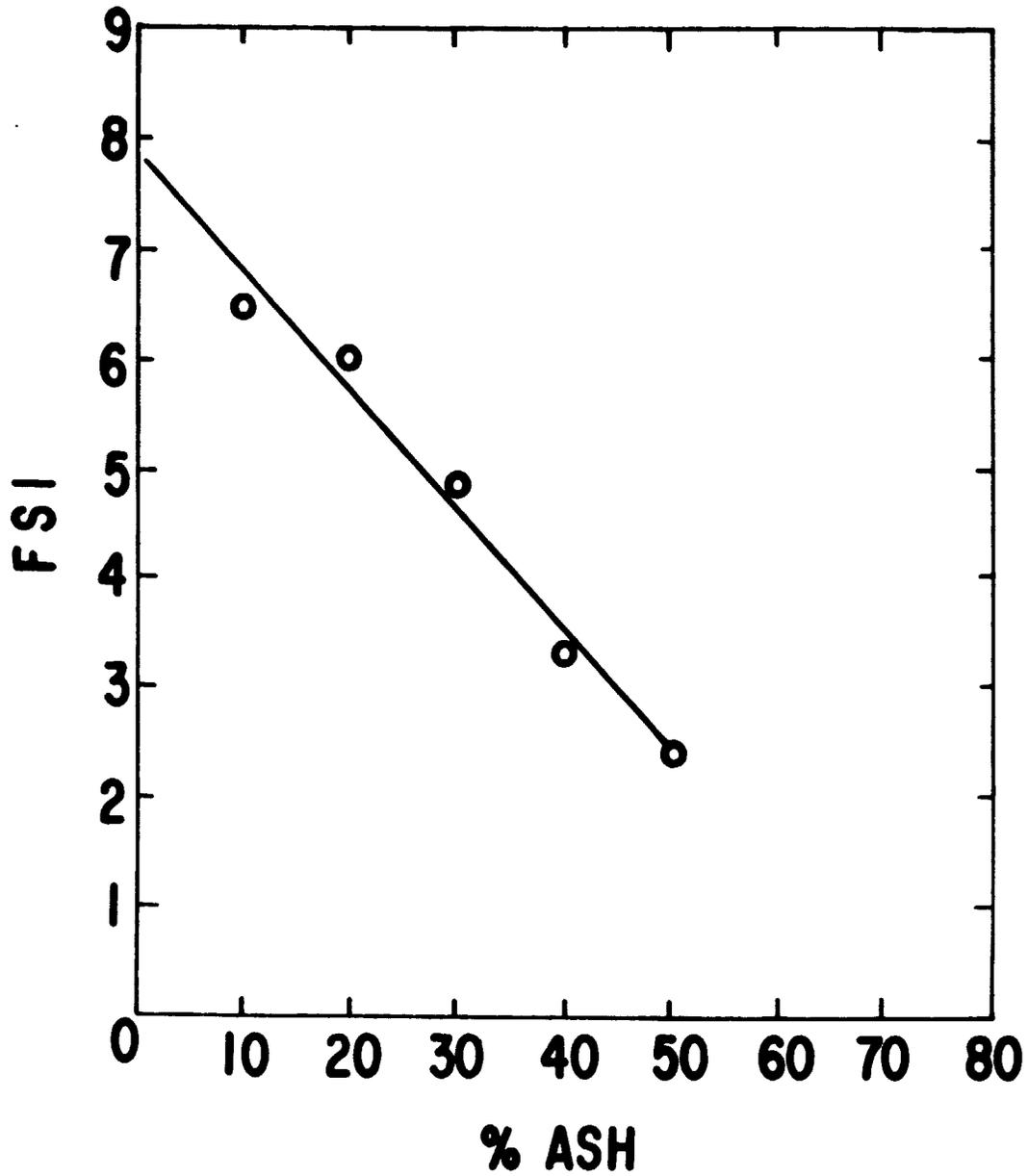
Figure 7.   Effect of Added Ash on Free Swelling Index

TABLE 3

COALS EXTRUDED

COAL	MINE LOCATION
Anthracite	Pennsylvania
HVA Bituminous (Pitt. #8)	Pennsylvania
HVA Bituminous (Ill. #6)	Illinois
Bitumincus, M-V	Missouri
Bituminous, HVB	Kentucky
Bituminous	Japan
Subbituminous	Wyoming
Subbituminous	Japan
Lignite	Utah
Lignite	Texas
Lignite	Canada
Minewaste	Illinois

SUMMARY

In summary, we are developing the coal extrusion process as a means of compacting and feeding coal fines directly into a pressurized gasifier. We chose the "cold" extrusion process over a hot one and a screw machine over other types. We have seen economies of scale in going from smaller to larger machines and have demonstrated gas sealing capability at competitive power consumptions on the 2" machine. Wear and process control are continuing problems, but ones in which progress is being made. The next major hurdle is to demonstrate that the process parameters can be scaled up to the 6" extruder and to measure the four critical parameters of sealing, power, output and wear while operating on an actual pressurized gasifier.

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N78-13256

COAL EXTRUSION IN THE PLASTIC STATE

C. England

P. R. Ryason

Jet Propulsion Laboratory
Pasadena, California

Coal Extrusion in the Plastic State

C. England and P. R. Ryason

ABSTRACT

Continuous feeding of coal in a compressing screw extruder is described as a method of introducing coal into pressurized systems. The method utilizes the property of many bituminous coals of softening at temperatures from 350 to 425°C. Coal is then fed much in the manner of common thermoplastics using screw extruders. Data on the viscosity and extruder parameters for extrusion of Illinois No. 6 coal is presented.

Introduction

This paper describes work on the feeding of coal to pressurized systems utilizing the properties of many coals to become semi-fluid under the action of heat and pressure. Experimental work has centered on modified thermoplastic extruders utilizing a single screw design in which bituminous coals enter as a granular solid and exit as a viscous fluid at elevated pressure.

The technique is based on the property of most bituminous coals to go semi-fluid or "plastic" at temperatures near 400°C and at modest pressure. Most of the previous interest in these properties has been related to the production of coke for metallurgical purposes where select of coals for blending is made partially on the basis of measurements of plasticity. Tests such as ASTM D-1612-69, utilizing the Giessler plastometer, have been developed for this purpose.

Coals can be rendered plastic at high pressure by devices common to the technology of the processing of thermoplastics.[1] One of the simplest of these devices in the screw extruder, shown schematically in Figure 1, in which the material to be extruded is heated and compressed until the desired outlet conditions are reached. The action of the compressing screw on the viscous material can cause very high pressure to occur in the extruder. Depending on the conditions such as flow rate and viscosity, pressures up to 35 MPa (5000 psi) can be obtained.

Measurement of Coal Viscosity

Of the many properties of coal that are of interest in extrusion, the most important are the viscous properties and their relation to temperature, shear and time. No data were available on the actual viscosity of coal, and an experimental program of capillary rheometry[2] was undertaken to measure the viscosity of several

coals in their plastic range. A short capillary was used in these experiments; the apparatus is shown in Figure 2. Separate preliminary experiments showed (by means of a thermocouple embedded in coal in the die) that steady temperatures were reached in the coal in about seven minutes. On the die axis, coal temperatures were consistently lower than the indicated temperatures by about $2^{\circ}C$. To load the die with coal, it was first weighed, coal added, and die plus coal reweighed. As indicated in Figure 2, a close fitting graphite plug was used to seal the upper surface of the coal to retain volatiles. Apparent viscosities were determined as follows. A loaded die and piston were placed in the brass block, thermal contact established by melting solder in the annular gap between die and heater block, and seven minutes allowed to pass. The cross head of the Model IIC Instron testing machine with a compression cage was then set in motion, and plastic coal forced through the capillary of the die. Either constant force of constant shear rate conditions were employed. Force readings were directly obtained from the load cell on the Instron machine.

Shear rates ($\dot{\gamma}$) were computed from

$$\dot{\gamma} \qquad \frac{4\dot{Q}}{\pi R^3} \quad = \quad \frac{4 \; R^2_d \; dl/dt}{R^3}$$

and shear stresses (τ) from

$$\tau \quad = \quad \frac{R\Delta P}{2L}$$

where $\dot{Q}$ is the volumetric flow rate, R_d is the radius of the die, dl/dt is the cross head speed, R is the radius of the capillary, ΔP is the pressure across the capillary, and L is the capillary length. Apparent viscosities were then computed from

$$\eta_{app} \quad = \quad \frac{\tau}{\dot{\gamma}} \qquad .$$

Measurements were made of apparent viscosity vs
temperature and shear rate for Illinois No. 6, Orient No. 3 bituminous coal. The
analysis of this coal is given in Table 1. Figure 3 shows the apparent viscosity as
a function of temperature with the residence time and shear rate fixed. As one would
expect, the coal softened upon approaching the plastic region, then hardened at
temperature associated with the coking properties. These data were quite sensitive
to residence-time effects; these effects are currently under study.

Figure 4 shows the apparent viscosity as a function of shear rate (essentially
the mass flow rate through the capillary). The viscosity is a strong function of
shear rate, and follows the power law relation[4] in a manner similar to pseudoplastics.
Thus, coal may behave much like a conventional plastic in the extruder. The basolute
value of the viscosity varies widely under the different conditions of temperature,
time, and shear rate, but are comparable in some range to conventional plastics.

Operation of the Screw Extruder

The plasticating screw extruder[2] combines the steps of solids transport, melting,
and pumping in one device, making it attractive as a feeder for coal in the plastic
state by virtue of its simplicity. Solid coal is augered into the central parts of
the extruder where it is heated, compressed, vapors expelled, and melting begun.
Heating is both by conduction from the barrel and by mechanical working of the coal.
When the coal becomes plastic, it is pumped by a drag flow mechanism in which the flow
rate is determined approximately by the equation

$$\overset{\circ}{Q} = \alpha N - \beta \frac{\Delta P}{\mu} \tag{1}$$

where Q is the volumetric flow rate, N is the screw speed, ΔP is the pressure across the pumping portion of the screw, μ is the fluid viscosity, and α and β are constants related primarily to the geometry of the extruder.[2] Equation (1) indicates that flow decreases with increasing pressure and increases as the viscosity rises. Maximum flow occurs at zero pressure and maximum pressure occurs at zero flow.

The volumetric flow rate across the die is determined by the Poiseuille equation for the flow of fluids through cylindrical round tubes.

$$Q = \frac{D^4 \Delta P}{128 \mu L} = K \frac{\Delta P}{\mu} \tag{2}$$

From equations (1) and (2), the pressure is found to be[2]

$$P = \frac{\mu \alpha N}{(\beta + K)} \tag{3}$$

Thus, the pressure developed is proportional to the viscosity and the screw speed. These equations assume that the flow through the extruder is governed by the screw pumping, and neglect compression, melting and solid feed considerations. Each of these have major effects on actual extruder performance.

Screw extrusion of Illinois No. 6 coal was demonstrated with a commercial screw extruder which was previously used to extrude polyethylene. The 3.81 cm dia. extruder (Centerline Machinery Co., Santa Ana, CA) was of conventional design. Screw speed was continuously variable up to a maximum speed of 120 rpm. At any given speed setting, screw speed was constant. A 3.7 Kw (5 hp) DC motor turned the screw through a multiple "V" belt drive. The extruder barrel was conventional: L/D = 22, constructed of 4140 steel with an Xaloy liner spun cast in place, with a Vickers-Anderson type flange at the delivery end. The die was a 0.48 cm (0.189 in) bore capillary which was segmented to allow varying lengths of 2.5, 7.6, and 12.7 cm.

The screw was of conventional design with an overall length of 81 cm (31.9 in). The maximum channel depth in the feed section was 0.77 cm, and this section took up the first 40 cm of the screw. The last part consisted of a linear compression section which reduced the flow area by a factor of 2.5. No metering section was used. The end of the screw was conical in shape, allowing about 0.41 cm (0.162") clearance between the cone surface and the interior surface of the die. Additionally, a bar 0.32 cm (0.125" wide) was welded along the surface of the cone to act as a wiper, keeping coal in the die in motion as the screw turned. A clearance of about 0.051 mm (0.002") was maintained between the surface of the wiper and the interior surface of the die.

The barrel was preheated prior to feeding coal by means of electrical heaters clamped onto the barrel. Polyethylene beads were fed slowly into the extruder to help heat the screw. No other means of heating the screw was available. In addition, the coal was preheated to 200°C by electrically-heating the delivery tube of the Vibrascrew powder feeder which was used to meter the coal into the screw extruder. The Vibrascrew bin as well as the entrance port of the screw extruder were blanketed with CO_2 to prevent premature oxidation of the heated coal.

Extrusion of Illinois No. 6 coal was achieved for periods up to 50 minutes before shutdown occurred due to blocking of the die by pieces of coked material. Reliable measurements of the coal viscosity were not obtained from die pressure measurements, and considerable difficulty was experienced with coking, failure of the coal to melt completely, and solids feed problems related to volatilization and to caking phenomena. These problems occurred to a much lesser degree in previously extruded coals (see Ref. 1 on Milburn and Kentucky coals). Rheometry measurements had indicated the potential of rapid coking with Illinois No. 6 coal. Indications of localized coking in the extruder resulting in irregular operation were also seen. These included slow plugging of the die and the production of chunks of coal within the extruder that were sufficiently large to temporarily block the die.

Preparation of the coal varied with the particular test. Most success was found when the coal was dried at 300°F, and sized to a 10/20 mesh. Generally, it appears the problems related to heat transfer and feed are reduced by continually removing fines. Such problems are in part related to the extruder design; it is possible that they are accentuated by the small size of the laboratory extruder.

Power requirements for extrusion of Illinois No. 5 coal varied widely with the particular test and temperature profile in the extruder. Sufficient runs were not made to determine an optimum temperature profile. Table 2 shows typical experimental results from Illinois No. 6, with the energy requirements varying from 68 to 108 kw-hr/ ton. The power appears to be closely related to the detailed temperature profile in the screw, and these relationships will be the subject of further study.

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Table 1. Analysis of Illinois No. 6 (Orient No. 3) Coal

Proximate Analysis:

	As Received	Dry Basis
% Moisture	11.00	
Ash	8.28	9.30
Volatile	32.48	36.50
Fixed Carbon	48.24	54.20
Btu	11,660	13,101

Ultimate Analysis:

	As Received	Dry Basis
% Moisture	11.00	
Carbon	64.73	72.73
Hydrogen	4.39	4.93
Nitrogen	1.33	1.50
Chlorine	0.30	0.34
Sulfur	1.85	2.08
Ash	8.28	9.30
Oxygen (by Difference)	8.12	9.12

Table 2. Experimental Results with 1 1/2 In. Extruder

(Illinois No. 6 Coal)

Coal Preparation

Screen Size	10/20 mesh (Tyler)
Drying	300°F in air

Extruder Temperatures

	Inches from Feed Port	Temperature, °F
Coal Inlet	0	510
Feed Zone	8	520
Compression Zone	15	790
Metering Zone	25	790
Die	31–36	790

Extruder Data

Screw Speed	83 rpm
Coal Feed	9.1 lb/hr
Power	0.89 hp

Calculated Data

Power	10.2 lb/hp-hr
	(248 BTU/lb)
Shear Stress in Die	---
Shear Rate in Die	105 $\sec^{-1}$
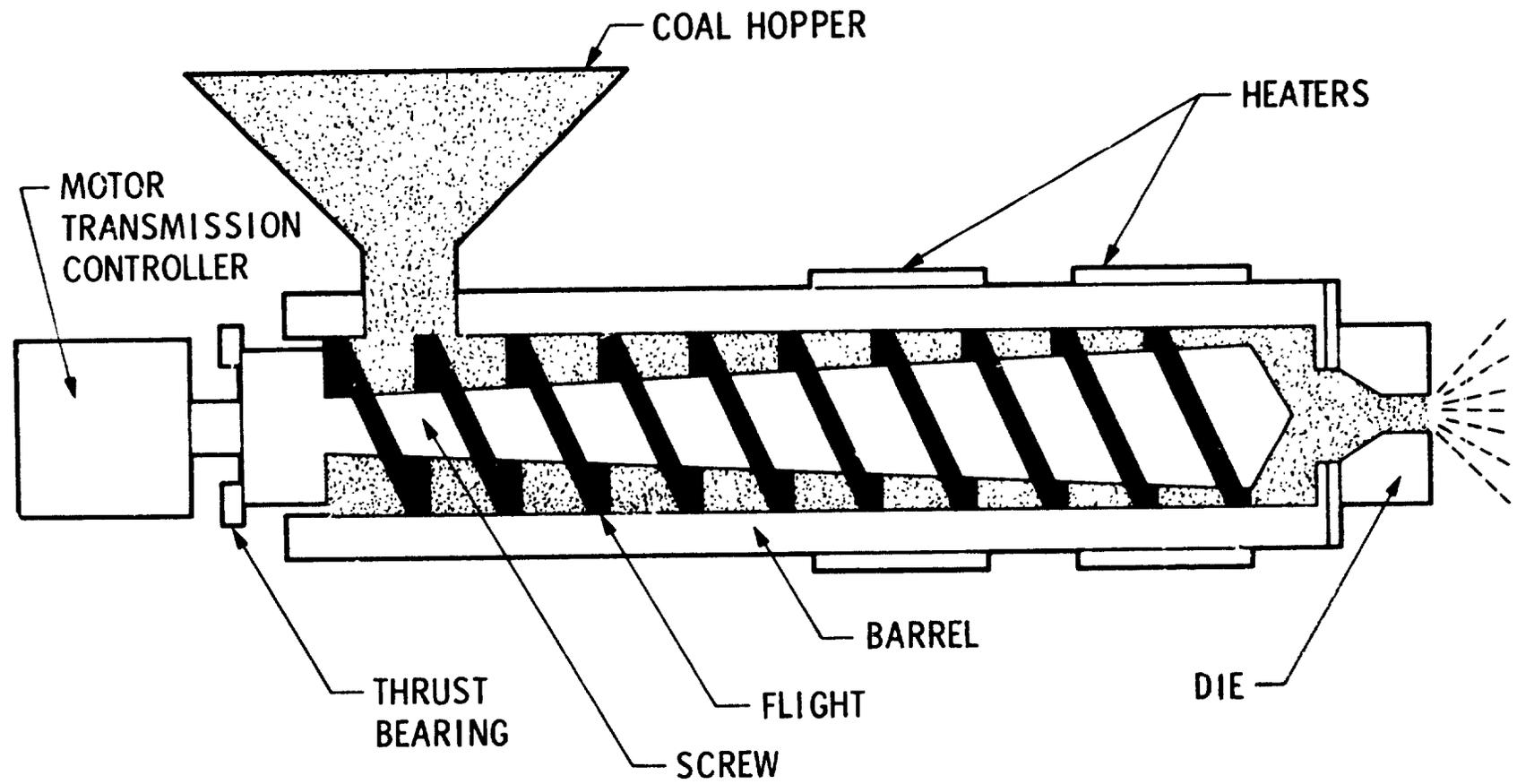
Figure 1. Schematic of a Compressing Screw Extruder
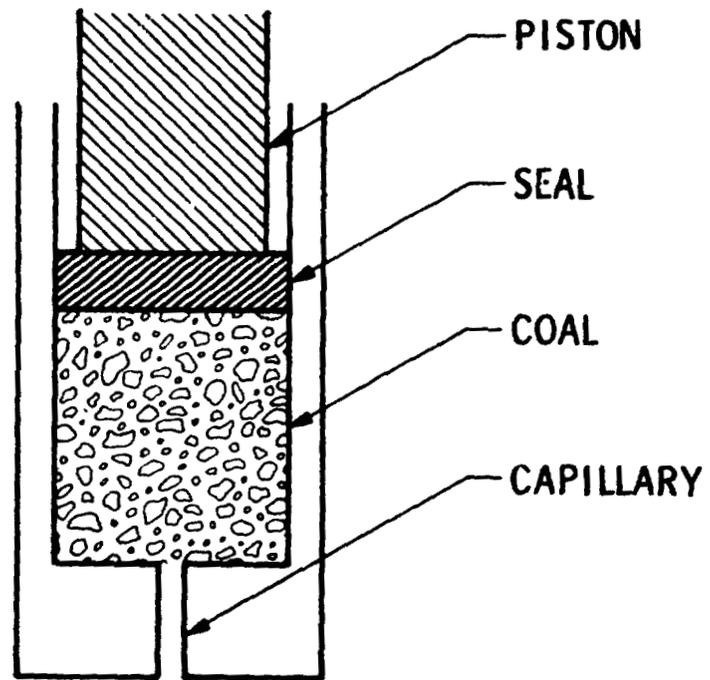
$$Q = \frac{D^4 \Delta P}{128 \mu L}$$

Figure 2. Schematic of a Capillary Rheometer
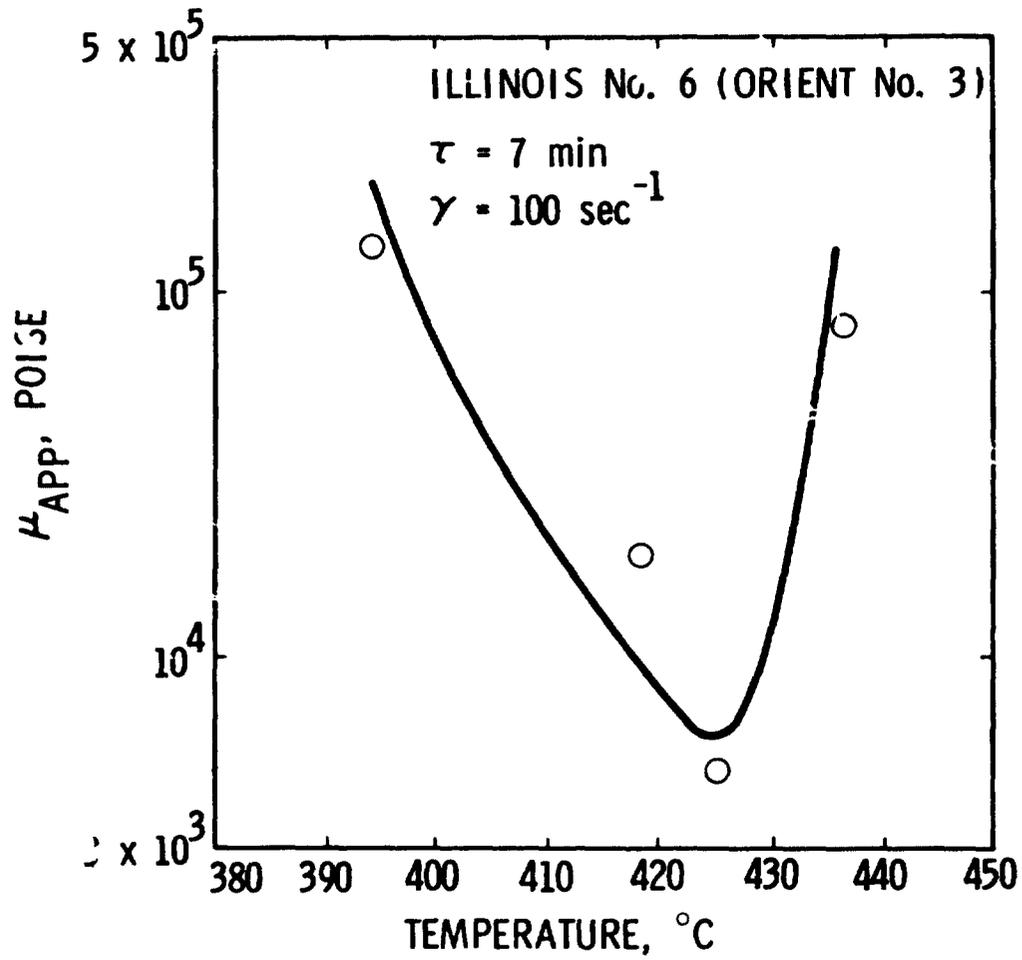
Figure 3. Temperature Dependence of Viscosity of Illinois No.6 Coal
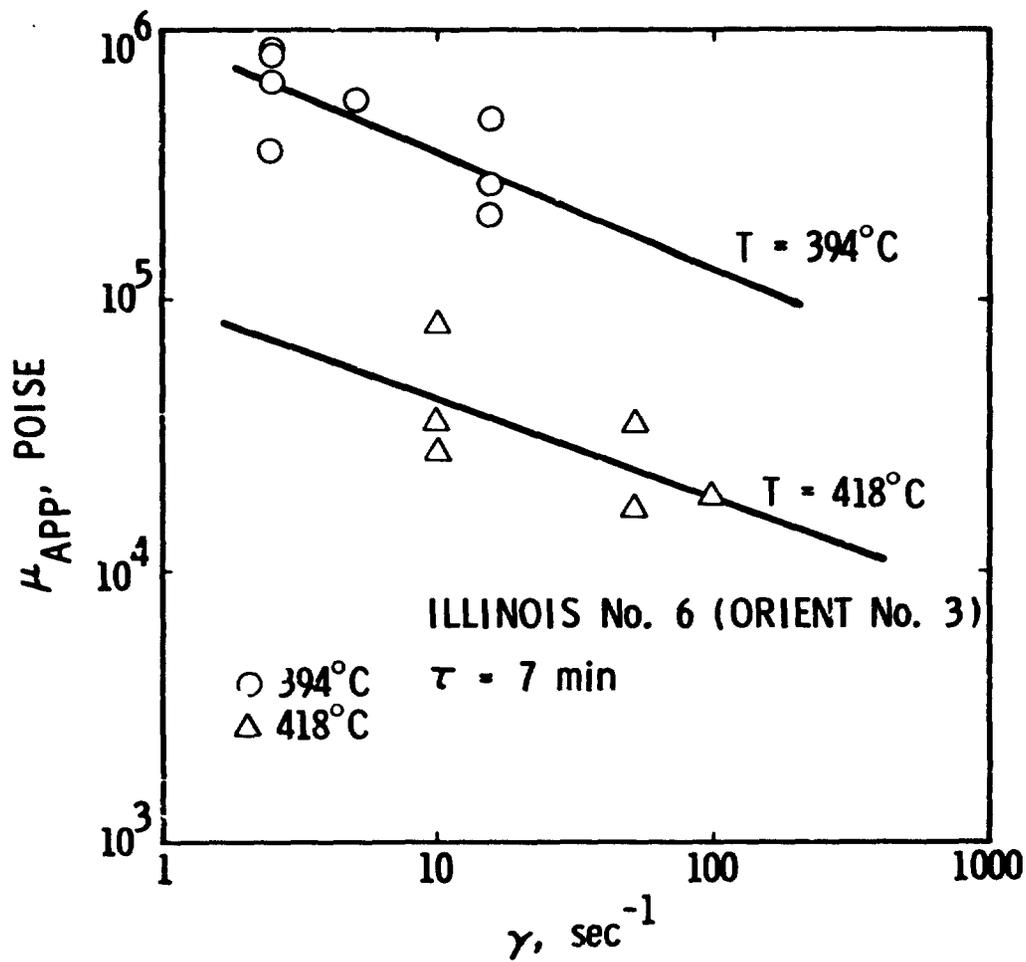
Figure 4. Shear-Rate Dependence of Viscosity of Illinois No.6 Coal

A NOVEL DRY COAL FEEDING CONCEPT
FOR HIGH-PRESSURE GASIFIERS

H. E. Trumbull

H. C. Davis

Batelle, Columbus Laboratories
Columbus, Ohio

N78-13257

ABSTRACT

A novel dry coal feeding concept has been developed for injecting ground coal into high-pressure gasifiers. Significant power savings are projected because the coal is injected directly with a ram and there is no requirement for pumping large volumes of gas or fluid against pressure.

A novel feature of the concept is that a new seal zone is formed between the ram and injection tube each cycle. The seal zone comprises a mixture of a small quantity of finely ground coal and a fluid. To demonstrate the feasibility of the concept, coal was injected into a 1000-psi chamber with an experimental device having a 7-1/2-inch-diameter ram and a 28-inch-long stroke.

A NOVEL DRY COAL FEEDING CONCEPT
FOR HIGH-PRESSURE GASIFIERS*

by

H. E. Trumbull and H. C. Davis
Battelle, Columbus Laboratories

June 8, 1977

BACKGROUND

The development of a dependable system for feeding dry coal into vessels operating at pressures up to 1200 psig is a key element in the development of the second-generation gasification processes. A critical problem confronting any approach for dry-coal feeding is excessive wear and abrasion. In most systems, virtually no abrasion can be tolerated because of the resulting gas leakage, which is especially critical at the pressures contemplated for these second-generation gasification processes.

Battelle initiated a program whose objectives were to conceive and develop a system for dry-coal feeding that would avoid or minimize wear and abrasion and that could be used to feed coal into vessels operating at pressures of 1000 psig or higher. After evaluation of numerous approaches on a small scale (3.5-inch-diameter tube), the concept described in this paper was developed.**

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* Paper prepared for presentation to the Conference on Coal Feeding Systems, sponsored by the U.S. Energy Research and Development Administration (ERDA), at Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California, on June 21-23, 1977.

** Two U.S. patent applications on the concept have been allowed and are expected to issue in 1977.

THE BATTELLE CONCEPT

A new concept of injecting dry coal was conceived and investigated by Battelle under the internally funded Battelle Energy Program. The concept involves filling an inlet tube with coal and pushing the charge of coal into a pressurized gasifier storage chamber with a ram. A seal plug between the ram and tube is formed for each injection cycle by a mixture of graded fine coal and a fluid, e.g., oil. The seal plug may be pre-formed outside the injection tube and placed within the tube after a charge of bulk coal has been loaded. The seal plug accounts for less than 0.5 percent of the total coal volume charged in each injection and the quantity of fluid used to form the plug is less than 0.05 percent of the total volume of coal.

The idea of sealed piston feeding of coal is not new and has been proposed many times before. However, it was not considered feasible because problems of wear and abrasion in such piston feeders become evident long before any practical operating times are accumulated. Wear and abrasion result from the abrasiveness of the coal and the force required to push the coal against the high-pressure differential. The high-pressure differential causes high leak rates to develop where sealing surfaces are abraded.

The Battelle concept minimizes wear and abrasion, while still maintaining sealing effectiveness, by providing a localized seal layer of a coal-fluid mixture. During the exploratory phase of this work, it was found that adding a small amount of fluid to a thin layer of a particulate coal charge in a feed tube resulted in several very important advantages:

(1) Pressure leakage is prevented.

(2) The compression required to prevent leakage with wetted coal is less than that required with unwetted coal.

(3) The thin layer of wetted coal acts as a very effective sealing piston that is movable at far lower pressures than are required for the movement of unwetted coal. The piston moving the coal does not do any sealing when operating with the thin wetted layer of coal.

(4) Wear on the tube wall i. greatly reduced because the
fluid in the wetted layer of coal acts as a lubricant
and coal does not need tc be nearly as compressed
(which contributes to wear) as when operating without
a seal zone. For example, 1-inch chunks of coal were
fed between the seal zones.

(5) Wear that occurs does not result in leakage because
the wetted coal and fluid contained therein flow into
surface irregularities and thus prevent leakage.

On the basis of preliminary estimates, it is believed that this coal-
feeding concept would allow a reduction in installed capital costs and also
savings in power when compared to either lock-hopper or slurry feeders.

Battelle's experimental work has established the feasibility of this
concept. We believe that it is sufficiently premising to warrant further
development.

EXPERIMENTAL PROGRAM

The primary objective of the experimental program was to demonstrate
the feasibility of the Battelle dry-coal feeding concept. A secondary objec-
tive was to assess the applicability of the concept to a commercially sized
gasification system.

The experimental work was undertaken to investigate the influence of
the following variables on the seal time and force required to push the coal
through the injection tube:

(1) Compaction pressure

(2) Viscosity of the sealing fluid

(3) Coal-to-fluid ratios in the seal zone

(4) Seal zone coal mesh sizes

(5) Operating gas pressure levels.

Experimental Equipment

Initial experiments utilizing 2-inch-ID and 3-1/2-inch-ID injection tubes established the feasibility of forming a coal-fluid moving seal against 1000 psig of gas pressure for a time sufficient to reload the tube. Additional larger scale experiments were then performed on a single-cycle basis to analyze the effects of scale-up on the formation of the seal plugs. In this scale-up phase, apparatus with a 7-1/2-inch-ID injection tube equipped with a gate mechanism was fabricated.

A vessel designed to accept a 36-lb charge of coal from the feed tube while pressurized to a maximum 1000 psig was provided. A removable end plate was installed on one end of the vessel to allow removal of the coal after each injection cycle. A sliding gate assembly was provided in the pressure vessel to close off the bottom of the feed tube during the charging and compaction steps of the coal injection cycle. The gate was moved to its position under the injection tube by the action of a sliding wedge on which it rested. The wedge was actuated by a hydraulic cylinder mounted on the outside o' the pressure vessel. This experimental apparatus is shown diagrammatically in Figure 1. A ram with 0.01-inch radial clearance was used to compact the coal used in the injection tube and to push the coal charge into the pressure vessel.

Experimental Procedures

The feeding cycle is shown in Figure 2. In the first step, the ram is withdrawn and the injection tube is filled with bulk coal. In the second step, a new seal zone in a carrier ring is placed above the bulk coal. In the third step, the ram advances to compact the coal. This compaction of the new seal zone forms the seal between the ram and tube. Finally, in the fourth step (1) the gate is opened, (2) the ram injects the lower seal zone and bulk coal, and (3) the gate is closed. The seal zone remaining in the end of the tube and backed up by the ram provides the necessary sealing against pressure until closing of the gate.

Average seal time of all experiments was approximately 40 seconds against 1000 psig, which is more than 10 seconds above the maximum expected pressure exposure time of an operating syst
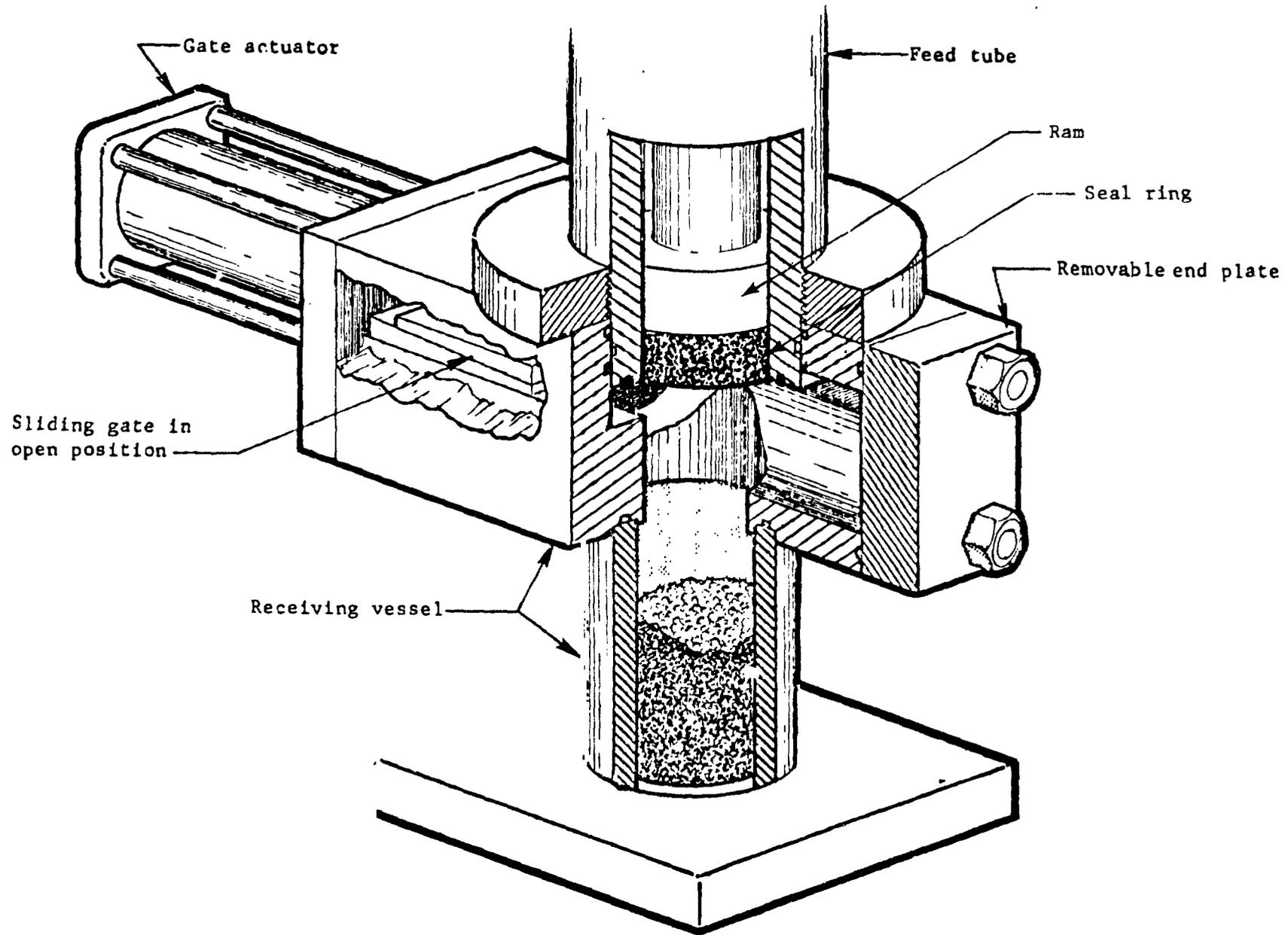
FIGURE 1. EXPERIMENTAL DRY-COAL-FEEDER MECHANISM
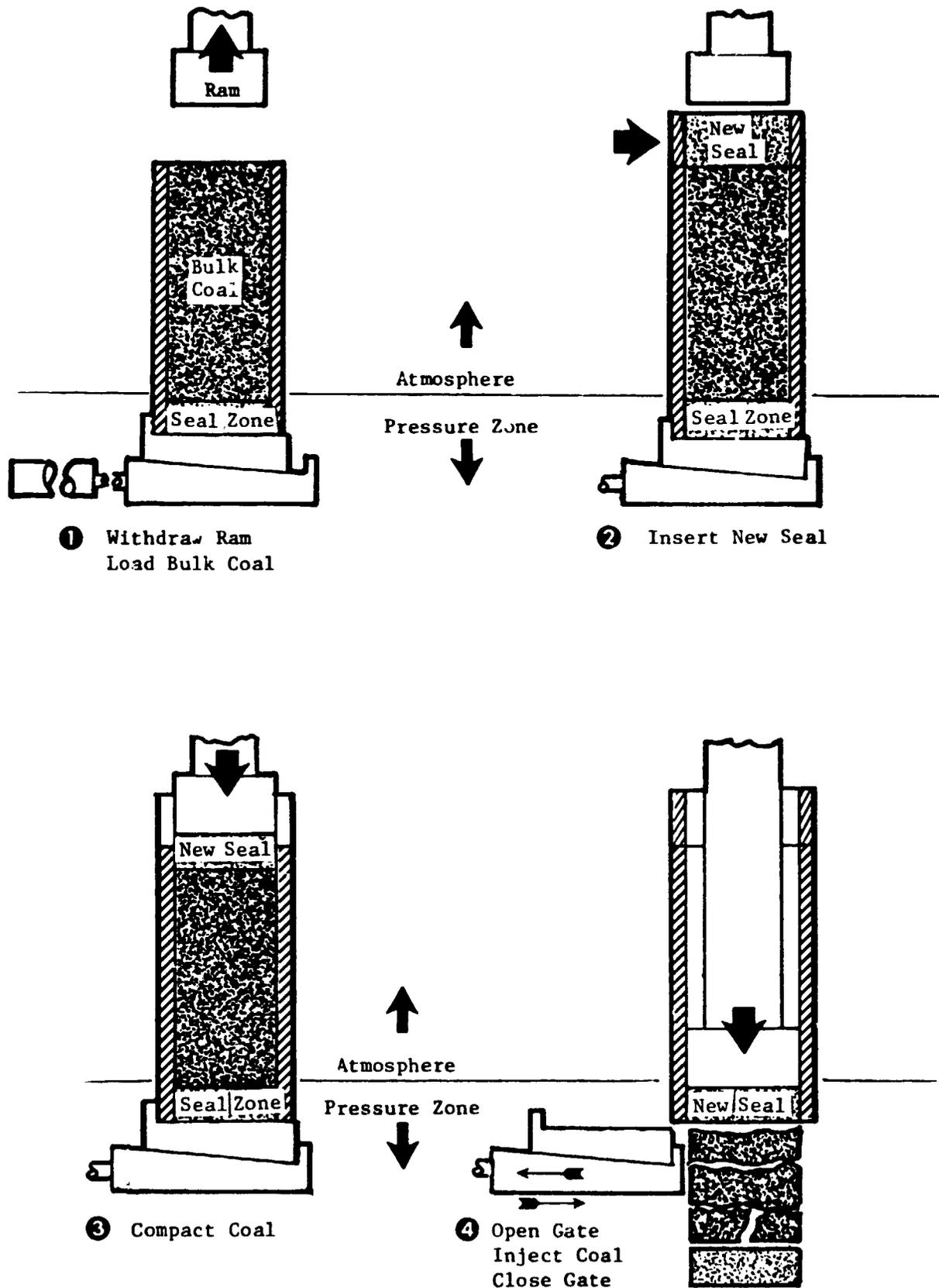
FIGURE 2. COAL FEED CYCLE

Compaction pressures of 2000 psig were required to form seal rings which averaged 36 seconds of seal time. Adequate packing of the fine coal particles occurs at this pressure. Higher pressures tend to distort and crush the coal and increase the injection forces.

A 50-weight motor oil and a 42-ssu-viscosity crude oil were used in the coal paste of the seal ring to determine the influence of viscosity on the seal time. The higher viscosity oil produced seal times that averaged approximately 20 seconds longer than seals produced by crude oil. However, the crude oil produced seals that averaged 27 seconds, which would provide a seal for the length of time required in the proposed full-scale system.

The use of various coal/fluid mixtures for the seal rings was investigated. It was found that an average improvement in seal time of 12 seconds was obtained when using 200-mesh and 20-mesh coal in equal parts rather than 200-mesh coal alone. This suggests a better packing ability, and consequently fewer voids, when using a well-graded mixture of coal rather than a mixture of uniform size. A drier paste mixture also improved the performance of the coal seal by reducing the extrusion of the seal ring around the ram.

Experiments were performed to obtain data concerning the feeding of lump coal, and the effect of different varieties of coal on forming the seal. Illinois Montour No. 4 coal in the size range of 1-inch to 1/4 inch diameter and North Dakota lignite in lumps up to 3 inches in diameter were run at a compaction pressure of 2000 psig. Both lump-coal feeding experiments used seal rings made from the same coal variety as the coal being fed. No differences in the performance of these seal rings could be detected.

Experiments were performed to assess the effect of operating pressure on seal times. The operating pressure in the receiver was increased from 500 to 1000 psig. This resulted in an increase in average seal time from 33 seconds to 44 seconds and is attributed to the increased radial pressure exerted by the seal ring at the higher receiver pressure.

Experimental Findings

From these experiments, we have concluded that

(1) The bulk coal being injected can range in size from 200-mesh to 3-inch lumps (maximum size investigated). With the 2-inch lump coal a moderate amount of crushing occurred (8-1/2 percent passing 1/8-inch mesh).

(2) The type of coal (bituminous and lignite) comprising the seal ring has no apparent effect on forming the seal.

(3) The concept is suited for high-pressure systems of up to 1000 psig and possibly higher.

(4) Seal plugs could be formed separately and transferred into the injection tube after it has been filled with coal.

(5) After feeding to a pressurized reservoir, the compacted bulk coal can be easily restored to its original condition by minimal agitation (such as from a mixing-type mechanism, screw feeder, or even gravity fall).

In summary, Battelle's experimental work has established the feasibility of forming a renewable seal of coal and fluid between a ram and a tube for injecting coal into a high-pressure chamber. Also, it has indicated that this concept has potential application in large scale gasification systems.

COMMERCIAL CONCEPT FOR DRY FEEDING OF COAL INTO A GASIFIER

Several combinations of injection tube diameter, ram stroke, and cycle time can be used to achieve an 80-ton-per-hour feed rate for a full-scale system. Engineering judgments of tradeoffs have led to consideration of a system utilizing two 24-inch-diameter by 10-foot-long injection tubes with the injection ram cycling at a total rate of two injections per minute.

The full-scale feeding apparatus will inject coal into a pressurized storage reservoir unit that would be remote from the gasifier unit. Thus, appears that the coal-feeding unit can be operated at ambient temperature. Also, it is assumed that a satisfactory seal could be obtained in a coal-feeding apparatus with a large diameter injection tube because better seal performance was experienced with the 7-1/2-inch-ID injection tube as compared with the seals obtained with the 3-1/2-inch-ID tube.

The main functional components of the coal-feeding system are:

(1) A press unit to compact the coal and inject it into a receiving chamber having a pressure of 1000 psig. The ram and its cylinder are mounted on a carriage so that the ram can be u ed to inject coal alternately through two injection tubes.

(2) A pair of injection tubes each equipped with a gate to close the bottom end of the tube against the 1000 psig gas pressure in the receiving chamber.

(3) A pair of receiving chambers to receive the coal charge injected by the ram.

(4) A screw conveyor to empty the receiving chambers of the coal charges.

(5) Two bulk coal-loading systems to fill the injection tubes with coal charges.

(6) A seal-ring-assembly system to form the seal ring and convey it to the inlets of the injection tubes.

Figure 3 shows a side and front view of the press feeding unit design which has a height of 33 feet. The unit is approximately 12 feet square and extends ̄ 10 feet below ground level. The press design shown is based on a laminated assembly of steel plates. The total weight of the press and other components is estimated to be 250 tons. A special feature of the press is the hydraulic cylinder which is mounted on a carriage. The carriage moves the cylinder with ram and bulk-coal-loading devices alternately over the two injection tubes so that while the coal is being injected in one tube, the other tube is being loaded. The movable ram also provides the clearance above the tube for the loading of the coal into the tube by the loading unit.
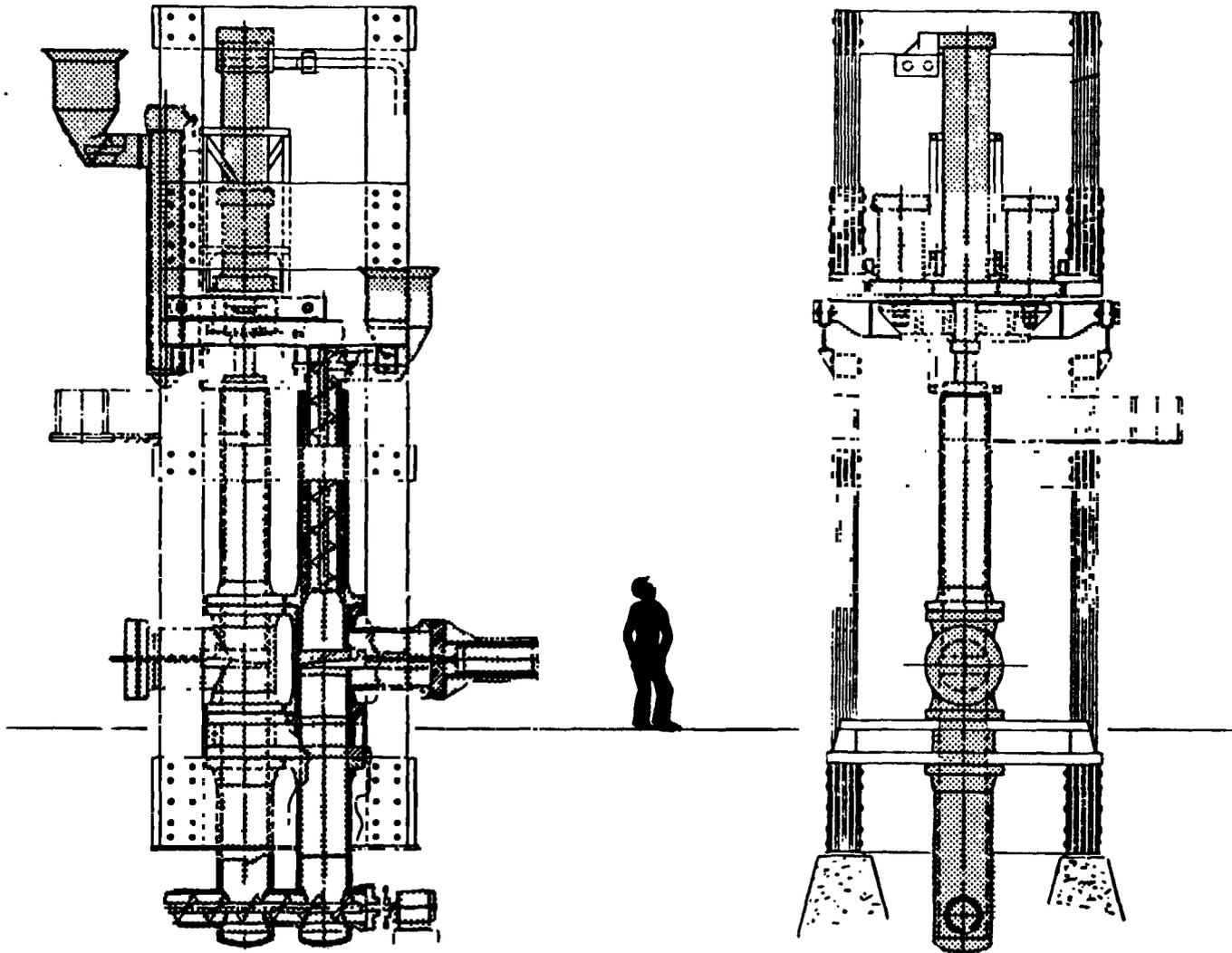
FIGURE 3.   CONCEPT FOR AN 80 TON/HOUR COAL FEEDER

The gate design shown is based on the design used in the experimental apparatus and shows one method of accomplishing the required gating function between the injection tube and the receiving chamber. The gate is opened by withdrawing the lower wedge element partially from under the upper gate element so that it is moved downward and out of contact with the elastomeric seal. Continued movement of the lower wedge carri. the upper element away so that the tube is fully open to the receiving chamber. As the gate opens, the top surface is swept clean by rotating brushes. The debris from this cleaning action is picked up by a small screw to convey the coal particles back into the receiving chamber.

Because of the abrasive action of the coal, a hard wear-resistant surface may be needed on the interior wall of the injection tube. A hardened steel (R_c 42) tube was used in the experimental apparatus. A surface finish of about 4 to 7 microinches is recommended to reduce the force required to push the coal from the tube.

RECOMMENDATIONS FOR DEVELOPMENT

Two additional program phases are envisioned to advance the dry-coal-feed concept from its present experimental position to operation on a gasification pilot 'ant:

Phase II – Continuous Operation, Systems Analysis, and
             Experimental Confirmation of Design Parameters

Phase III – Feeder System Design, Construction, and
             Operation at a Specific Pilot Plant.

The basic task for the proposed Phase II effort would be to convert the present single-cycle coal feeder to continuous operation with a design feed rate between 1 and 2 tons of coal per hour.

Complementary tasks would include:

● Additional experimental work on forming seals. ʳurther
   reduction in compaction pressures and use of fluids
   other than oil would be directed toward obtaining data
   for lowering operating costs of the system.

- Laboratory studies of wear on system components to
  provide some means of assessing the maintenance
  requirements for an operating system.
- Evaluation of mechanical processing of the coal after
  injection so that it is suitable for feeding into a
  gasifier.
- Adaptation of the experimental feeder (or design of a
  scaled-up feeder) for further evaluation of the coal
  feeding concept at a specific pilot plant.

The Phase III effort would be detailed selection of a pilot plant.
Operations under Phase III would use the data from Phase II to design and
construct a feeder system integrated with an existing gasification pilot plant.
Operation of the feeder system should extend over a period sufficient to gain
experience and permit a thorough evaluation of the concept.

FEEDING THE FEEDER

A. L. Kurylchek

Solids Flow Control Corp.
West Caldwell, New Jersey

ABSTRACT:

Too often the equipment used to move "difficult to handle" powdery
material from a hopper to p: cess creates a complex of devices whose end
result falls short on good performance...simply because equipment design, in
many cases, has not kept up with advanced technological concepts in fine powder
handling.

The Feeder, being the "key" to an efficient feed system, must be assured
a continuous flow from the storage tank...without bridging, arching, spasmodic
flow or uncontrolled flushing.

It is the intent of this paper to discuss the causes and effects of flow
problems and also offer solutions based on the combination of theoretical and
practical experience.

CAUSES OF FLOW PROBLEMS

Scores of technical papers have been written on this subject and to elaborate further would not serve any benficial purpose beyond what has already bee established; simply that the problems exist...and how to solve them without rte iting more problems.

It is further generally recognized that the cause of flow problems is related to two (2) major basic areas:

A) Material flow characteristics.   }
B) Bin/hopper design.   } Fig. 1

MATERIAL FLOW CHARACTERISTICS

So as not to over simplify the multitude of catagories of flow characteristics and their relation to flow problems, it can be concluded, for the sake of brevity and better understanding,that flow characteristics can be classified as either cohesive, floodable, or a combination of both.

The flow problems caused when discharging and feeding cohesive or floodable type powders needs no elaboration. However, what may clarify the crux of the problem is to consider and understand how a material can be both cohesive and floodable in nature at the same time.(Refer to Fig. 2)

In short, a cohesive powder becomes floodable in nature when the degree of areation permeates every particle comprising the mass...regardless the particle size. When this occurs, cohesiveness is either non-existant or not enough to cause arching, ratholing or spasmodic flow. It can actually flow ike water!

On the ner hand, when a highly floodable type powder is allowed time to deacrate, i.e., allow the air to escape or free itself from each particle comprising the mass, the particles, regardless their size, tend to adhere to eachother thus causing an adhesive/cohesive characteristic. Needless to say, it is this type of combination situation that merits the most consideration regarding flow problem solutions.

The answer is obvious and possibly over-simplistic. The material should be allowed time .n the storage tank to free itself of air (deaerate) thus eliminating the drastic results of flooding potential. In otherwords, a highly floodable powder can be made relatively cohesive by allowing ample retention time in the storage tank. The flow problems now to be solved deals only with a cohesive type powder.

Cohesive powders can be made to flow from a storage tank by utilizing one : more of the myriad of devices available on the market today. The trick that the Process Lngineer must "pull" is picking the right one.

To simplify this chore, it would pay to consider the design of the storage tank in relation to the material flow characteristics.

BIN/HOPPER DESIGN

Too often, and many times with reasonable justification, the storage tank is designed for capacity requirements and available space...without too much, if any consideration for the material flow characteristics. The end result of actual and potential flow problems needs no elaboration.

If the tank is not properly designed, material hang-ups will occur in the form of arching over the hopper outlet, rat-holing, uncontrolled funnel flow or unpredictable spasmodic flow...each of which can cause results totally unacceptable to efficient operation.

Further, even though ample retention time basically eliminates the flooding nature of the material, once the arch or rathole has been broken, the material cascading down and through the hopper outlet becomes highly aerated (each particle entrapped with air) defying any device to stop the "flood".

It is evident that in order to prevent this phenomena, the hopper opening must be made large enough to avoid a stable arch to form. Also; to avoid funnel flow which in essence can lead to stable ratholes, the slopes of the hopper should be relatively steep to assure flow throughput without material clinging to its sides.

Conical shaped tanks instead of square or rectangular avoid valley angles where the material can build up to a degree that would eventually lead to a completely plugged tank.

Materials of construction and inside "finish" must be taken into consideration to prevent hang-ups due to the inherent sliding friction between the material and the wall it slides on.

One of the most important factors in bin design is the consideration of the POTENTIAL CHEMICAL REACTION of the material. Should the material be hygroscopic in nature (tendency to absorb moisture) and allowed to sit in the tank for prolonged periods of time, the end result would be a tank filled with a "solid" that would defy dynamite to break loose. Needless to say, materials of this nature should be stored in reasonably sized tanks and with the caution of not allowing the material to turn to an unmovable mass, or have its flow characteristics radically change.

Although many Process Engineers have means to consider and avoid the various potential flow problems by utilizing shear-test and other material testing devices for proper bin design, the magnitude and seriousness of the problems caused by improper bin design merits and justifies the services of a Bin design Consultant whose efforts in the past years has proven that storage tanks can be properly designed.

CAUSES OF FEEDING PROBLEMS

If material does not flow properly from a storage tank or cannot be made to flow properly with whatever flow-aid device is used, there is no Feeder on the market that will do an efficient job.

One of the major problems in efficiently feeding powdery materials from a hopper is caused by selecting a Feeder based on volumetric throughput requirements without consideration for the material flow characteristics.

A case in point for example is the selection of a 10" dia. screw feeder (or any similar sized feeding device) which will adequately feed the required flow rates to process. (Refer to Fig. 3) However, should a cohesive material have a tendency to arch over a 10" hopper-bottom opening, the Feeder is useless. If a flow-aid device of sorts is utilized in breaking the arch, material flow to the Feeder (more chances than not) would be inconsistant thereby causing erratic feed, hence erratic accuracy.

Another case in point is that even if the hopper opening is large enough to prevent arching, and gravity flow to the Feeder is not interrupted, the total cross-section of the material in the hopper-bottom must be "kept alive", i.e., discharged at the same time, otherwise dead areas of material are likely to form above the Feeder due to the inherent mass resistance in cohesive powders. (Refer to Fig. 4) Once this situation occurs, density stability in the lower hopper section becomes erratic thereby negating acceptable accuracy. In other words, although volumetic accuracy may be maintained, accuracy by weight would vary erratically due to extreme density variations.

In either case mentioned, and similarly in many others, it becomes evident that a mismatch exists between the Bin and the Feeder.

It can be concluded thus far that a poorly designed bin will be a deterent to good operation regardless the Feeder design...and a poorly designed Feeder cannot enhance the operation even if the Bin design is compatible to the material flow characteristics.

SOLUTIONS

Countless experiments and field experiences corraborate the conclusion that large hopper-bottom openings are required to facilitate uninterrupted flow. The problem that this elimentary conclusion creates is to select and apply a compatible Feeder that can be installed under the hopper yet have the design features to "choke" down the large hopper-opening to a practical sized discharge means. (Refer to Fig. 5)

Another "must" in the Feeder design is to have the ability to "keep alive" (Fig. 5) and discharge every square inch of material in the hopper-bottom (directly over the Feeder) to avoid "dead" areas and excessive compaction.

Conventional bin-dischargers which are designed for installation directly under the bin/hopper bottom utilize the principle of vibration throughout the entire cross-section of the hopper-bottom (and well up into the bin) but the discharging flow pattern is around the periphery of the vibrating (gyrating) element and not throughout the entire cross-section of the hopper-bottom. Further, the majority of these devices do not have the capability to directly vary the flow rates therefore cannot be considered Feeders. Variable speed "take-away" devices are normally used to satisfy the feeding requirement.

THE "SILETTA" LIVE-BOTTOM triple purpose FEEDER (Refer to Fig. 6)

Since its introduction to the market place in 1976, the Siletta LIVE-BOTTOM triple purpose FEEDER has been accepted as a new device offering design features which are compatible to the solution of flow problems when discharging and feeding difricult to handle powdery materials.

Extensive testing and subsequent successful field-installations verify its applicability and performance.

PRINCIPLE OF OPERATION

The unit operates on the principle of controlled vibraticn. A feed-tray component, consisting of narrow slots and sloping plates (similar to a louver) is activated by a high-frequency, low-amplitude precision-tuned vibrator. When stationary ("OFF" position) the material inside the bin/hopper arches over the narrow slots of the sloping plates stopping all flow. Under vibration ("ON" position) the arch is broken and material will flow through the slots with controlled consistancy.

By varying the amplitude (1.5 mm max.) of the feed-tray component (manually, remotely or via an input signal from a process controller) a predictable feed-rate change occurs when necessary to satisfy varying process requirements. Turn-down rates up to 10:1 are possible for many materials thus eliminating the need for a variable speed "take-away" means or any other transfer device, by feeding direct. into process.

In short, the Siletta Live-Bottom triple-purpose Feeder not only serves as a bin-discharger but also performs the functions of an accurate volumetric Feeder and a non-movable cut-off gate...all in one compact machine.

FEEDING THE FEEDER APPLICATIONS

Due to its inherent design features, the Siletta can be applied as a volumetric feeder in lieu of practically any volumetric feeder on the market today...by virtue of its capability to vary the flow rates. However its utilization as a prefeeder makes it ideally suited for those applications requiring extremely high accuracies that only gravimetric feeders can guarentee. (Refer to Fig. 7)

PROBLEM AREAS

When handling non-free flowing and/or fluid-like powders, getting the material out of the hopper and to the relatively small inlet opening of the Weighfeeder creates a problem that many times negates good performance due to material hang-ups in the hopper or uncontrolled flushing when the material arch is broken.

To alleviate the problem of arching or no-flow, a flow-aid device is usually installed in or at the bottom of the hopper. If the material discharges from the flow-aid device (fixed vibration) directly to the feeder variable speed Weigh belt (Fig. 8) a mismatch of flow rates can occur especially when the Weighfeeder is set to deliver a lower flow rate than the bin-discharger is supplying. This causes the material to back up, or flush through the Weighfeeder if the material is floodable in nature. Cycling the bin-discharger (on-off) for lower flow rates sometimes solves the problem but not when the material is fluid-like...which can result in flooding and uncontrolled feed.

To prevent flooding, rotary type vane feeders are usually installed, between the bin-discharger and weighfeeder which have at times solved the problem not-withstanding the maintenance involved.

If the bin-discharger utilizes a variable spe   screw or auger to feed the Weighfeeder (Fig. 9), flooding may also be pre.ented providing the screw or auger does not run empty, otherwise the system will be subject to the same conditions previously mentioned.

A constant speed weigh-belt (Fig. 10) utilizing an adjustable gate to increase or decrease the material throughput per unit of time, is also subject to the same problems when handling highly cohesive, or floodable type powders. The gate will not prevent flushing, and cohesive powders could plug at the gate opening. If the constant speed weighfeeder control gate is eliminated and is fed by a variable speed screw or auger as shown in (Fig. :) the problem of feeding the screw or auger prefeeders still exists because the bin-discharger is supplying a relatively constant flow rate and the excess material has no where to go when low flow rates are required to satisfy process requirements. If the material is highly floodable, it will find its natural outlet and flush through the prefeeder and flood the Weighfeeder.

SOLUTIONS

Weighfeeders operate best when a relatively constant head of material is maintained on the weigh section. When the Weighfeeder increases its speed to deliver more material to satisfy the process (set-point) the bin-discharger must supply this demand by increasing its flow rate accordingly (Fig. 11)....
and visa versa.

In short, predictable flow rate capabilities should be incorporated in the bin-discharger design to assure controlled flow throughput out of the hopper without bridging, rat-holing, flushing or spasmodic flow and at the same time provide the Weighfeeder with the approximate flow rate it requires.

Once this condition prevails, the instantaneous response capabilities of the Weighfeeder to varying material densities and set-point changes will then assure the precise flow rates that it is capable of delivering.

The Siletta Live Bottom triple-function Feeder has been specifically designed to provide predictable flow rates while still having the capabilities to serve as a bin discharger flow-aid device. Its instantaneous "no-flow" feature (non-movable cut-off gate) when the unit is in the "OFF" position, also makes it ideally suited for batching or bag-filling applications.

CONCLUSION

As mentioned in the Abstract, equipment design in many cases, has not kept up with advanced technological concepts of fine powder handling. Although many devices on the market today are "holding their own" regarding performance, it is the combination or sum total of the related equipment in the process line that causes the problems. In other words, the more devices involved, the more chance of problems.

Eliminating the possible "weak-links" in the process train with equipment whose design incorporates the "triple-function" approach merits consideration for its applicability in material flow-feeding systems...which can result in considerable savings in costs, elimination of downtime and time-consuming maintenance.

SOLIDS FLOW CONTROL
CORPORATION

MATERIAL FLOW CHARACTERISTICS

- SLUGGISH:
    - COHESIVE
    - DAMP
    - HYGROSCOPIC
- FLOODABLE
- INTERLOCKS or MATS:
    - FIBERGLASS
    - ASBESTOS, ETC.
- PACKS UNDER PRESSURE
- LIGHT & FLUFFY
- COHESIVE & FLOODABLE

BIN / HOPPER DESIGN

FIG. 1

AIR LAYERS

FLOODABLE

DEAERATED

POTENTIAL ARCH

FIG. 2

FIG. 3 — 10" OPENING — 10" D.

DEAD AREA

FIG. 4 — 10" X 20" OPENING — 10" D.

FIG. 5

~ 36"

"LIVE" DISCHARGE AREA

FEEDER

~ 8" CHOKE

NON-VIBRATING
SUPPORT FRAME

16", 30" or 48"

ROUND or SQUARE
OPENING

FLEXIBLE
DUST-TIGHT SEAL
(ALL AROUND)

SUSPENSION
LINK

ELAS
SPRI

ELECTROMAGNETIC
DRIVE

DISCHARGE
FUNNEL

FEED TRAY LOUVERS

FIG. 6

NON-COHESIVE
POWDERS

COHESIVE
POWDERS

"Siletta"
LIVE-BOTTOM
FEEDER
(VARIABLE RATE)

FIG.7

CONTROL SIGNAL
manual or automatic

BELT SCALE or WEIGHFEEDER

FIG. 8

FIG. 9
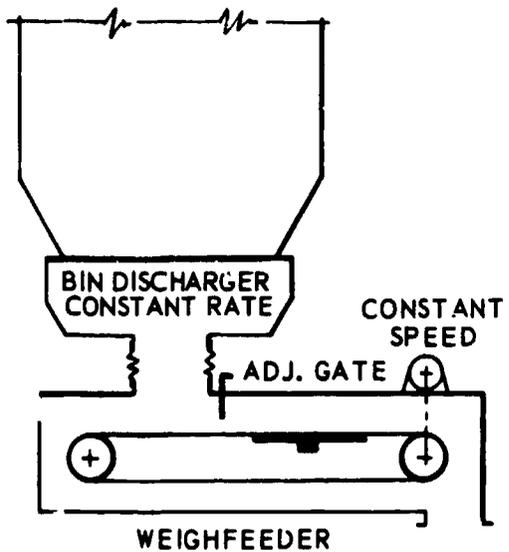
FIG. 10
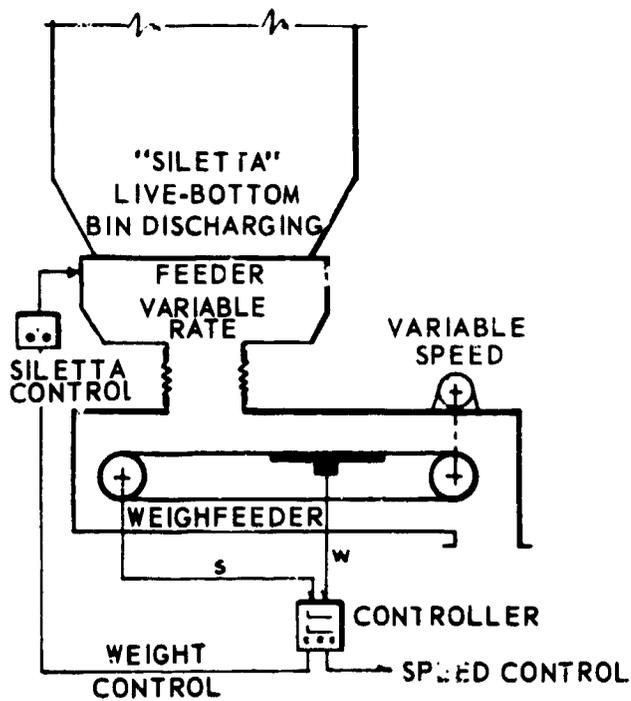
FIG. 11

SAFELY EVACUATING COAL FROM HOPPERS/ SILOS WITH PNEUMATIC BLASTING

J. S. Fischer

Martin Engineering Co.
Neponset, Illinois

ABSTRACT

For years the industry has coped with the problems of evacuating coal and other difficult bulk solids from storage containers. Many methods have been used to manage the problems; manual hammering, vibration, air lances, and vibratory hopper bottoms, to mention a few. Also, mass flow design in storage containers has been an approach to solving the problem. The latter often results in drastically reduced storage capacity and extremely expensive construction. The former methods also present inherent disadvantages of being inefficient, noisy, or expensive to install and operate.

After more than 30 years of being involved in the design and production of flow aid devices, a U.S. manufacturing concern recognized industry's need for an efficient, economical, effective and quiet device for moving coal and other difficult bulk solids. Thus came the advent of the low pressure pneumatic blasting system - a very efficient means of using a small amount of plant air (up to 125 PSI) to eliminate the most troublesome material hang-ups in storage containers. This simple device has one moving part and uses approximately 3% of the air consumed by a pneumatic vibrator on the same job.

The principle of operation is very simple: air stored in the unit's reservoir is expelled directly into the material via a patented quick release valve. The number, size, and placement of the blaster units on the storage vessel is determined by a series of tests to ascertain flowability of the problem material. These tests in conjunction with the hopper or silo configuration determine specification of a low pressure pneumatic blasting system.

This concept has often proven effective in solving flow problems when all other means have failed. A number of case histories in the area of coal handling will be cited where low pressure pneumatic blasting systems have completely solved troublesome flow problems. Further, we will analyze the benefits in each case, including increase in production efficiency and cost savings.

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EVACUATION OF COAL FROM HOPPERS/SILOS

WITH LOW PRESSURE PNEUMATIC BLASTING SYSTEMS

by

John S. Fischer

Martin Engineering Company

Neponset, Illinois

Evacuation of difficult or cohesive bulk particulates from containers has been a very costly problem. It is costly for numerous reasons: production down-time, manual labor to free material blockage, and use of inefficient and costly flow aid devices. Noise emission from certain flow aid devices and physical harm to workers due to numerous injuries, and in some cases, fatalities related to moving stored particulates occur annually,making safety a primary concern of industry today. The pneumatic blasting system alleviates these problems which plague industry since it is a fail safe system of moving the toughest bulk particulates.

Numerous manufacturers have designed and sold many types of flow aid devices. Yet, the vast majority have inherent disadvantages - vibratory hopper bottoms are expensive, and vibrators are relatively inefficient and/or noisy. Several years ago the concept of low pressure pneumatic blasting evolved. The merits of pneumatic blasting include totally unique capabilities such as economy, efficiency, effectiveness, and it is very quiet; thus, safe from a standpoint of preventing loss of hearing due to excessive noise. Technical aspects of the principle of operation will be illustrated and thoroughly explained.

The speaker will present several case histories illustrating a few of the many applications of pneumatic blasting, its truly unique capabilities, and the

degree of its success in solving the flow problem.

The safe and practical concept of pneumatic blasting has been proven and is already well accepted by many industries, though available to American industry for only a few years. In a short period of time, much has been learned in design and application of this concept. As with many truly new ideas, pneumatic blasting was welcomed by many and criticized by an element upon its introduction.

Much of the original theory in developing pneumatic blasting has proven factual; as with most new concepts, some theory has not. We are all in the learning process with regard to pneumatic blasting. Though most of this presentation is based on fact, we cannot ignore theory which must constantly be explored to allow the development of any concept including this revolutionary approach .

Compressed air has long been known as a source of power for many tools and machines, including devices for moving bulk solids. Examples are the piston type vibrator of years ago, and more recently, sophisticated pneumatic vibrators such as the motor driven rotary eccentric type.

Many industrial personnel involved in bulk materials handling are also familiar with direct air application methods of evacuating difficult material, such as the air lance and continuous flow air pads. Several flow aid devices are relatively sophisticated and somewhat more efficient than others. However, all display disadvantages - either noise, structural fatigue or inefficiency, and all consume a relatively high volume of compressed air. This rate of air consumption varies in degrees from tolerable to totally unacceptable by most standards. For instance, air lances will consume 60 to 100 cu.ft. of air per minute. At todays compressed air costs of 12-18¢ per 1000 cu.ft., such usage levels can become prohibitive.

As previously mentioned, another primary safety consideration today is the fact that noise problems exist. Controlling noise at the source is regarded by the Dept. of Labor for OSHA standards as the ideal means of preventing noise induced hearing loss. Realizing that most industrial vibrators could be detrimental to hearing, it became quite obvious that alternate methods of prompting flow of particulate solids had to be found. Pneumatic blasting systems proved to solve the noise problem by preventing noise at the source which is much better than limiting exposure of personnel to excessive noise.

So the pr blem with conventional flow aids was twofold; noise control and the high cost of operating conventional flow aid devices. This situation brought about the advent of a material-moving pneumatic air cannon. The new concept proved to be a solution to industry's dilemma since pneumatic blasting is effective, yet quiet and economical.

This breakthrough occurred in the past three years. It is the product of several years of long, laborious and costly research and development, but it has been well worth it.

Unlike vibration devices, the pneumatic blaster does not move materials exclusively through the reduction of friction. The pneumatic blaster shocks the mass of cohesive material, fracturing it and causing free flow, whether it be bulk solids sticking to the walls of a hopper, silo, chute, or building up under screens, and even flow problems in stockpile storage. While alleviating the noise and cost of operation problems, pneumatic blasters have also proven effective with material-moving problems no other devices could handle. An example of this type of application will be discussed.

In order to pursue discussion regarding pneumatic blasting, we should review

commonly accepted terminology:

slope angle - downward angle of slope measured in degrees from horizontal.

archability - the tendency of a cohesive powdered or granular solid to form an arch or bridge in the hopper or silo.

rathole - the result of material collecting on the wall of the storage container, leaving a hollow core in the center of the storage container

compressibility - a value arrived at by taking the difference between the aerated bulk density and the packed bulk density, and dividing this difference by the packed bulk density.

working bulk density - the working bulk density will equal the packed bulk density, minus the loose bulk density, times the compressibility. This value is added to loose bulk density and equals working bulk density.

angle of repose - the angle between the horizontal and the slope of a heap of material dropped from a specified elevation. For our purposes, it can be defined as the constant angle to the horizontal, assumed by a cone like pile of material.

angle of fall - the angle of repose resulting from a jarring effect.

angle of difference - the value arrived at by noting the difference between angle of repose and the angle of fall.

dispersibility - the direct measure of the ability of a material to flood or be fluidized.

cohesion and uniformity - cohesion and the uniformity coefficient are alternate flow properties used in the flow evaluation. Cohesion is used with powders and very fine particles, or with materials on which an effective cohesion force can be measured. The uniformity coefficient is used for granular and powdered granular materials in which an effective surface cohesion cannot be measured.

surface area – the surface area of a given particle.

hygroscopicity – the tendency of a solid to pick up moisture on its surface from
the ambient atmosphere; to "cake up".

Efficiency of the pneumatic blasting device is relative to a number of factors.
First, the degree of free air flow from the device directly affects the force
output of the unit as well as velocity of the air escaping. The objective in
design is to achieve the optimum degree of velocity and force with minimum air
pressure, and in most cases, minimal volume. In simple terms, the most efficient
design will allow a given volume of air at a given pressure to be released in the
least amount of time. An extremely efficient design – to our knowledge, the most
efficient design – appears in Figure I. This design provides optimum efficiency
since the distance of reservoir opening to discharge opening is a minimal distance
of approximately 8". Thus, the air flow meets very little resistance, allowing
maximum force and velocity output.

Secondly, the air passageway is obstruction free upon activating the unique
patented piston poppet valve. This latter feature significantly increases force
and output in comparison to other designs.

The principle of operation is very simple. Air enters the blaster via a
quick exhaust valve. Air enters chamber (A) and compression causes the piston (B)
to move forward and seat on (C) and air flows through orifice in center of piston,
filling chamber (D). To discharge the blaster, the quick exhaust valve is
activated, releasing air in chamber (A) which allows pressure from chamber (D) to
force piston back into air space (A). Air in reservoir (D) is expelled through
the discharge tube.

This most recent design also affords infinite flexibility in reservoir size

since the valve is external and bolted to the tank. Thus, non-standard ASME code welded reservoir tanks are readily available to meet the user's specific needs.

The quick release valve is activated by any number of control systems, each providing features for various applications: manual pneumatic(Figure II), manual electric (Figure III), or timed electric (Figure IV). The latter two control systems implement solenoid valves to actuate the blaster system. For a completely automatic system, the "timed electric" system is ideal. The entire system can be actuated through a relay connection sensitive to an open gate, operating feeder, or conveyor. In this case, the "timed electric" controls operate the blaster system only upon demand of material.

The more versatile "timed electric" control system is emphasized since it eliminates human error and manual labor; also, the "pneumatic blasting"concept is dependent upon a system installation. Oftentimes, a system installation will require multiple units at various levels on a storage container. With many bulk solids, it is imperative to actuate the units separately or actuate levels of blasters individually. Usually, it is necessary to first evacuate the bulk solids in lower slope section of a container (close to discharge). This is accomplished by firing the first level blaster(s) (Figure V). Once the lower portion of the slope is free of stubborn material, the solids in the upper portion of the container may be broken up into a free-flowing stage (Figure VI) and immediately evacuated through the discharge. Most often, as the density and cohesion factors of material increase, it is necessary to fire the blaster units separately. Conversely, as material density and cohesion decreases, so does the need to actuate blasters separately. Note: This statement is only a "rule of thumb".

Another advantage of "timed electric" is the safety factor. For instance,

not only is there limited and efficient usage of the system since it operates only
on material demand, but when the system operates, certain precautions can be taken
as provided automatically. For instance, when the system is actuated, a hatch on
top of container can be automatically locked, a beeper or siren actuated with
or without a flashing light so personnel are aware.

Specification of a system involves numerous variables. With respect to the
flow characteristics of bulk solids, we must consider the following properties:
particle size, surface area, specific gravity, working bulk density, hygroscop-
icity, moisture content, angles of flow, adhesion, cohesion, and compressibility.
Various combinations of these properties measure basic flow characteristics or
flowability of any given bulk solid. Our concern is with non-free flow or stubborn
solids which require flow aids; thus we confine discussion to these properties.

Massing or caking of materials may be substantially reduced or eliminated
through (1) a modified container design, or (2) a flow aid device or sometimes
additives mixed with the material. All have their advantages and disadvantages.
A common problem in solving the flow problem through container design is keeping
the overlying material weight at a minimum, yet having a slope angle allowing the
material to flow. Generally speaking, mass flow container design for a very
stubborn material results in comparatively low storage capacity at a considerable
expense. A problem may often be solved at much less expense with an effective and
efficient flow aid device and standard container design.

When specifying a pneumatic blasting system, we determine the specific var-
iables or properties which affect the flowability of the problem bulk solid. First,
it is necessary to determine whether the troublesome bulk solid is powder or
granular. For our purposes, the minus 200 mesh size will be powder; plus 200 mesh

size granular. This simplifies specification of a system. Determining whether a material is granular or powder narrows the range of specific properties to consider in determining flowability of the bulk solid. The four absolute properties which will determine the flowability of a powder are: (1) angle of difference, (2) angle of fall, (3) dispersibility, and (4) cohesion. In analyzing the flowability of a granular bulk solid, one needs to consider (1) working bulk density, and (2) surface area of particle.

For the practical discussion of specifying a blaster system, the flow characteristics of the bulk solids must be considered in conjunction with, and equally important, storage vessel size and configuration. It is obvious that the slope angle of a hopper bottom would have a direct bearing on the ability of a given material to evacuate from a hopper/silo, etc.

In summary, sizing, placement, and firing sequence of a blaster system is specified in consideration of the bulk solids flowability, which is contingent on two basic factors: the select properties of the solid and on container size and configuration . . plus, working experience with these systems is very important in specifying a system.

The select properties affecting flowability of a powder or granular substance must be considered absolute, yet one must always be aware of extraneous factors which are not absolute or always existing in a bulk solid. For example, hygroscopicity, moisture content, shape rugosity, temperature, and so on.

Next, what is the real theory behind low pressure pneumatic blasting? Speaking of the design in Figure I, the volume of air is released in approximately .25 sec. at a velocity of 1,198 ft. per sec. at 90 PSI. Assuming the material build-up is

relatively thick and the solid is of typical density (45 to 100 lbs/cu.ft.), the blast of air upon discharge will act as an expanding air pocket, expanding parallel with the wall of the structure, pushing outward at the same time, and ultimately displacing a section of the bulk solid from the wall. Essentially, it is breaking the shear strength of the hung-up or clinging material. As the depth of material build-up and/or density decreases, so does the effective radius of the expelled blast, to an extent (rule of thumb).

It is possible to blast a relatively small hole in the material build-up. When material build-up is not extremely thick and/or dense,the use of directional discharge accessories is necessary. Three basic types of directional accessories are most commonly used: 45 degree el, 90 degree el, or a narrow slotted nozzle directs the blast parallel to the container wall instead of perpendicular and directly into the material.

The pneumatic air cannon can be applied to storage vessels made of concrete, wood, or steel. Also,with pneumatic blasting devices it is possible to pipe the discharge through an extension to remote or inaccessible (exterior) areas within the structure. There are numerous structures where pneumatic blasters are virtually the only flow aid which could perform effectively. One of many such installations exists at a large grain terminal handling soybean meal. The storage facility consists of clustered concrete silos, 30' dia., 90' high, with a cone of 30 degrees off horizontal center. This company intended to keep the meal in storage for up to 6 weeks. Unfortunately, upon opening the discharge gate, most often very little or virtually no material would flow. Consequently, the operator was forced to resort to manual labor - poking and prodding the material free.This can be very dangerous since materials are capable of flooding. The cost of the problem was extremely high for two reasons: manual labor costs, and the production (in this case, transfer) downtime.

This arching problem is typical of soybean meal, yet three factors made this particular problem far worse than what typically exists. A 30 degree slope (off horizontal) is not at all common: most silos are designed with a 45 degree to 70 degree slope. Secondly, as the protein content of soybean meal increases, its flowability decreases. Average soybean meal contains 44% protein; the meal in this particular silo contained 50% protein. Lastly, the clustered silos had only 120 degrees of exposed exterior wall. This proved to compound the problem since it is very important to blast within 18" above or below the intersection of the cone and vertical wall.

This terminal finally found the solution to be a pneumatic blasting system. The flow aid manufacturer decided to tackle this problem with a unique system. Since the manufacturer had already equipped numerous soybean meal silos all over the United States with very successful and cost saving pneumatic blaster systems, the number of blasters and placement of discharge was not difficult to determine. Again, it is very important to blast in two areas or levels; first, in the sloping portion, and a second level within 18" above or below the intersection of cone (or slope sheet) and vertical wall (Figure VIII). This placement and location of units in two levels is found to be most effective on round vessels containing materials which display archability characteristics. The system specified called for three low volume blasters to be mounted on the exposed exterior wall, approximately 40 degrees apart, and 18" above the intersection of cone and vertical wall (FIG.IX) To reach the remaining interior circumference at this junction which is most critical(in this case, inaccessible from the exterior), it was necessary to mount four higher volume units on the uppermost section of the cone. Each of these four units have a discharge extension parallel to the interior wall of the cone, extending 18" above the intersection of the cone and vertical wall (Figure X) with a total length of approximately 12'. Thus the first phase of installation involved

seven blasters, discharges equally spaced 51 degrees apart, 18" above intersection of the cone and vertical wall.

The second phase was very simple, involving the installation of three low volume blaster units, equally spaced 120 degrees apart, 6' up from the discharge on the cone (Figure VIII & IX). The system was put to the ultimate test on 50% protein meal which was left in storage for approximately 8 weeks. Upon twice firing the lower level of blasters, the cone portion was free of material. Next, the upper level of units was fired. freeing the cohesive mass with ease. Upon completely evacuating the structure, the operator was pleased to find 95% of the troublesome material had been removed. To our knowledge, no other flow aid could have been installed. Regardless, it is extremely doubtful that any other conventional flow aid is even capable of evacuating bridged soybean meal from a large storage vessel under any circumstances, and many have been tried.

In the past two years, virtually hundreds of successful pneumatic blasting systems have been installed in numerous industries. To mention a few: wood products, food, chemicals, ores, and plastics (Figure X). Furthermore, the system has been proven effective in moving the most stubborn materials through a wide range of ohesiveness and density. For instance, wood chips at 20 lbs. per cu.ft. (with an extremely high entanglement factor), through very cohesive ore concentrate at 180 lbs. per cu.ft. It is known that blaster systems have worked effectively in promoting flow of at least over 100 different materials of various consistencies.

Finally a proven system has evolved which allows plants to safely move bulk particulates from storage. No longer is it necessary for men to be exposed to

noisy vibrators. Never again should it be necessary for personnel to poke or prod materials out of a container from the top or discharge, or crawl inside a container full of potentially dynamic material. Incidentally, the last two methods have attributed to many deaths and even more injuries, worldwide. Now there's an answer.

In summary, these advantages may be attributed to pneumatic blasting:

1. Minimal air consumption - approximately 3% as much as a pneumatic vibrator.

2. No noise - noise is contained in storage vessel as well as absorbed by the bulk solid.

3. No structural reinforcement or fatigue.

4. Simple - one moving part, no electric motor, shaft, vanes, etc.

5. Ease in mounting - simply cut hole of prescribed diameter (up to 5" dia.), bolt mount plate to concrete, wood or steel structure. No major alteration to existing new structures.

6. No lubrication or filtration.

7. No sparks or flames.

PARTIAL LIST OF MORE COMMON MATERIALS TO WHICH THE BIG BLASTER®AIR CANNON HAS BEEN APPLIED (Figure XI)

Potash	Hay (chopped)
Compacted garbage	Molasses (chopped)
Prepared foundry sand	Sorghum (chopped)
Limestone (powdered)	Copper ore (fine & coarse)
Rice hulls	Copper concentrate
Triple super phosphate	Wood chips
Gypsum (coarse) (dust)	Sawdust
Coffee	Wood bark
Coal	Crackling
Clay (200 mesh)	Nickel ore
Polyester floc	Sugar
Diatomaceous earth	Poultry feed (pellets)
PVC powder	Horse feed (pellets)
Calcite (moist)	Salt (granulated) (rock)
Soybean meal	Filler cake (for animal feed)
Chlorinated trisodium phosphate	Flue dust
Cement	Wheat flour
Meat meal	Iron ore
Bran	Oat flour
Cake flour	Refractory (powder)
Alumina	Foam (ground)
Wheat middlings	Calcium carbonate
Plastic chips	Amonium hydroxide
Lead concentrate	Paper (shredded)
STP-2	
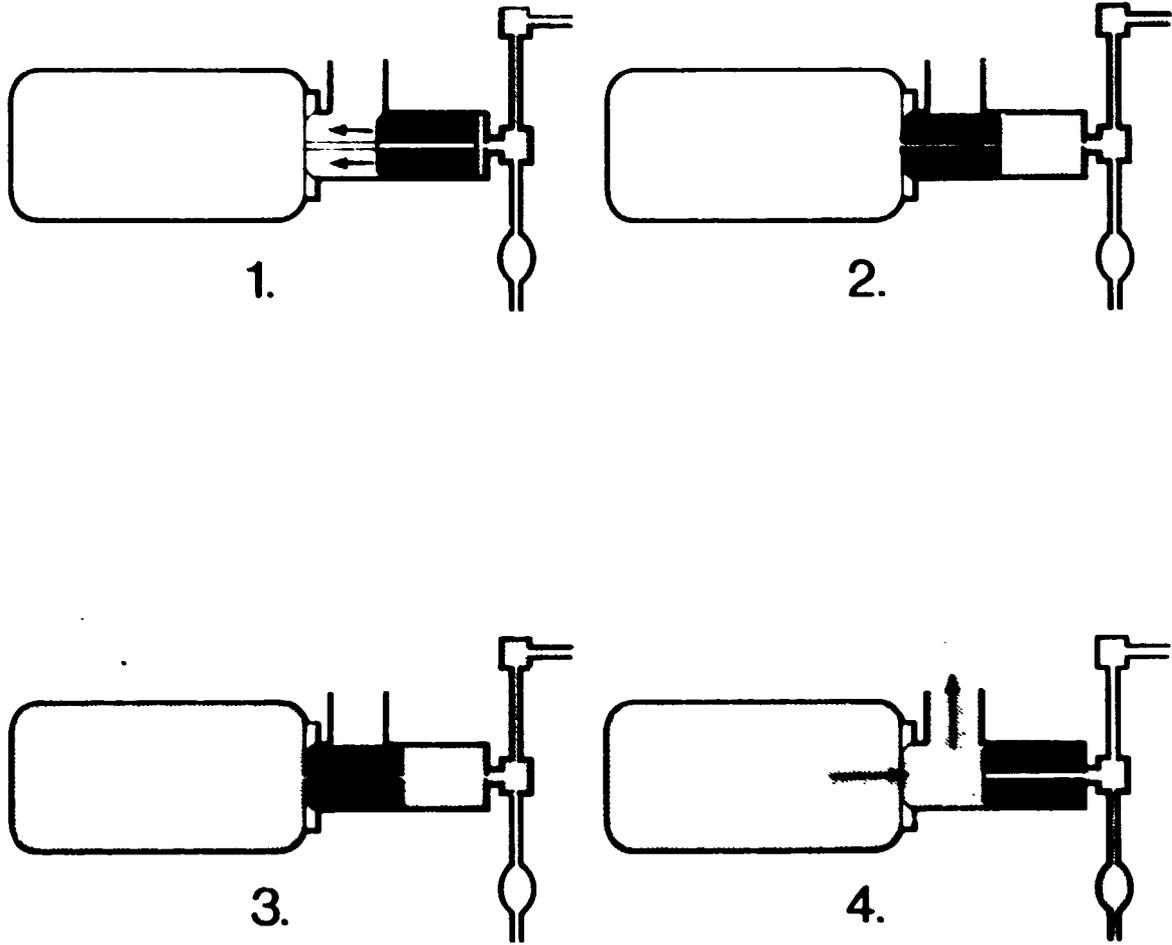
That's all there is to it!

FIGURE I

77-55

BIG BLASTER®
AIR CANNON

EXHAUST

QUICK EXHAUST VALVE

← AIR SUPPLY

EXHAUST

3-WAY MANUAL VALVE

MANUAL PNEUMATIC

FIGURE II

BIG BLASTER®
AIR CANNON

FIRE-CHARGE
SWITCH

ELECTRIC POWER

EXHAUST

AIR SUPPLY

CHARGE

QUICK EXHAUST VALVE

EXHAUST

3 WAY SOLENOID
OPERATED VALVE

MANUAL ELECTRIC

FIGURE III

BIG BLASTER® AIR CANNON

3 WAY SOLENOID OPERATED VALVE

EXHAUST

AIR SUPPLY

CHARGE

EXHAUST

QUICK EXHAUST VALVE

ELECTRIC POWER

TIMER

ELECTRIC-TIMED
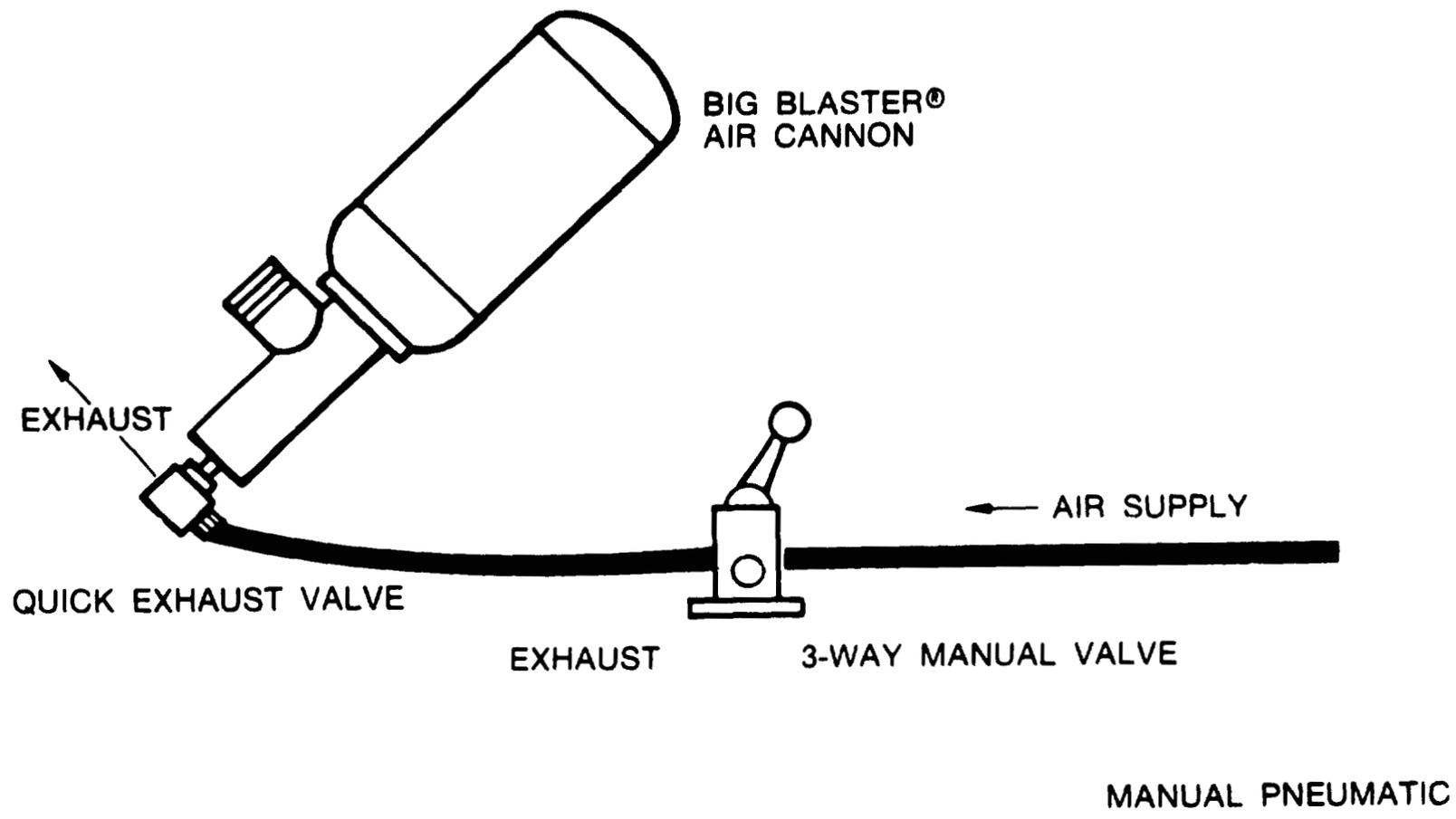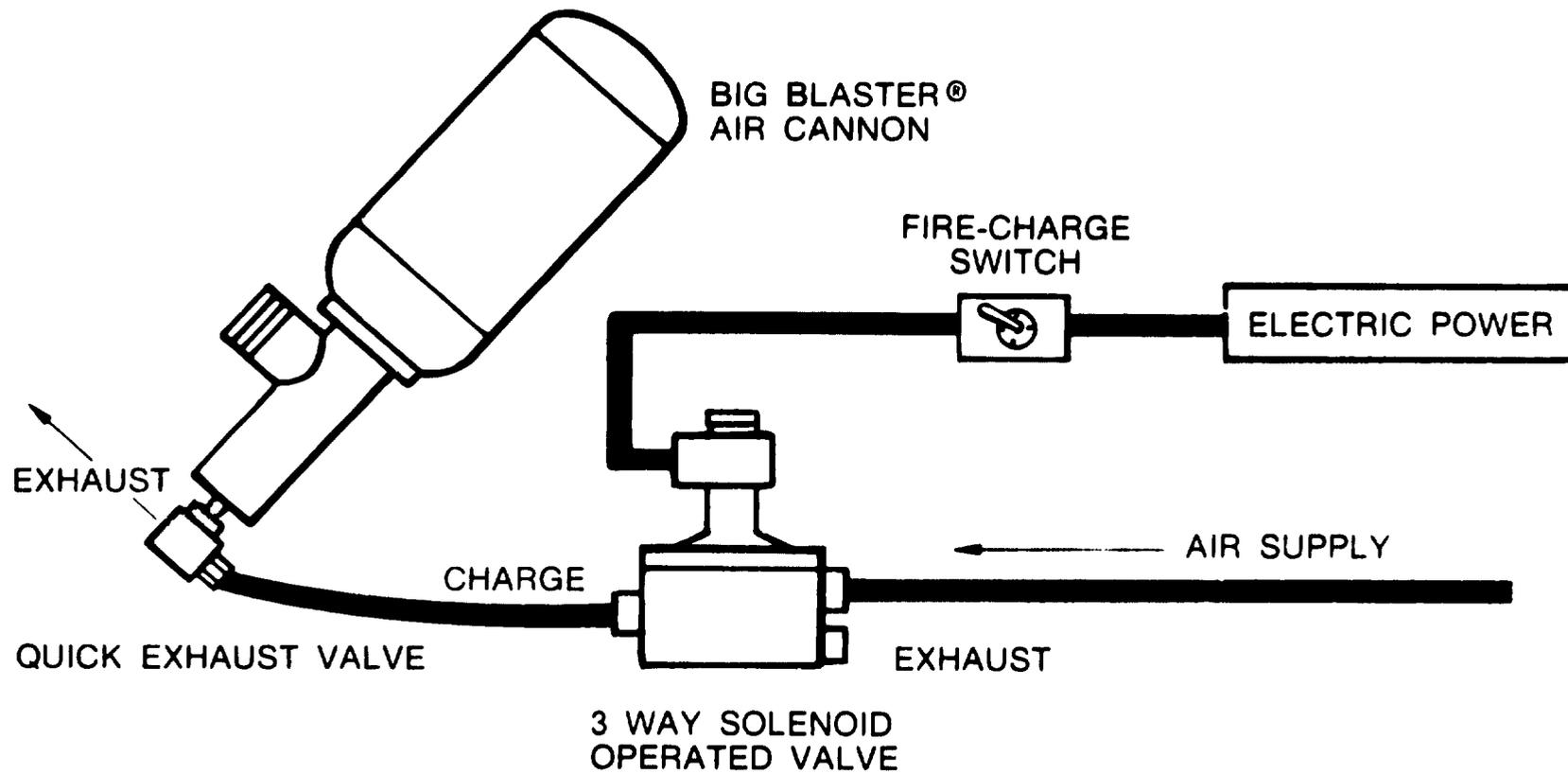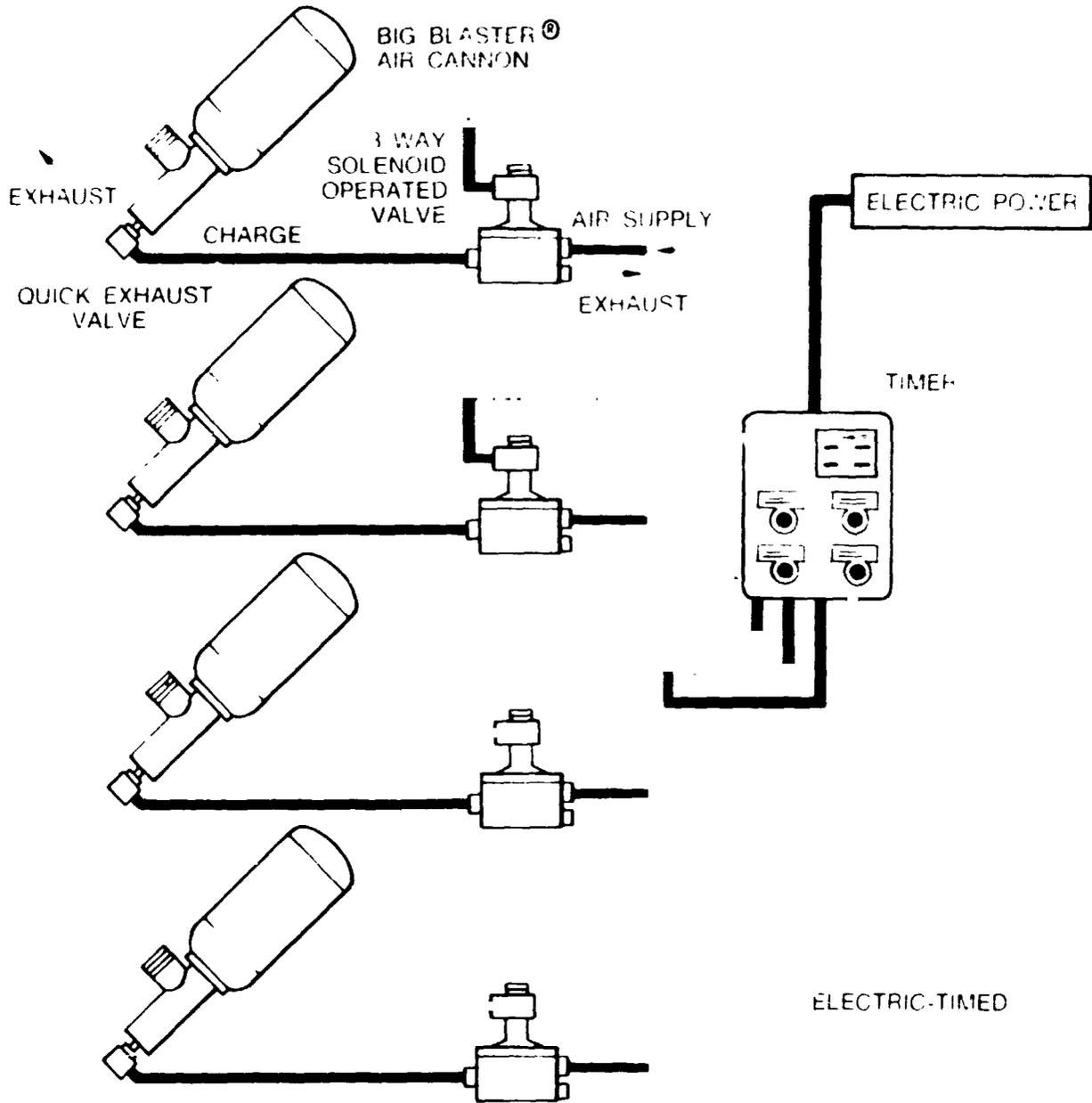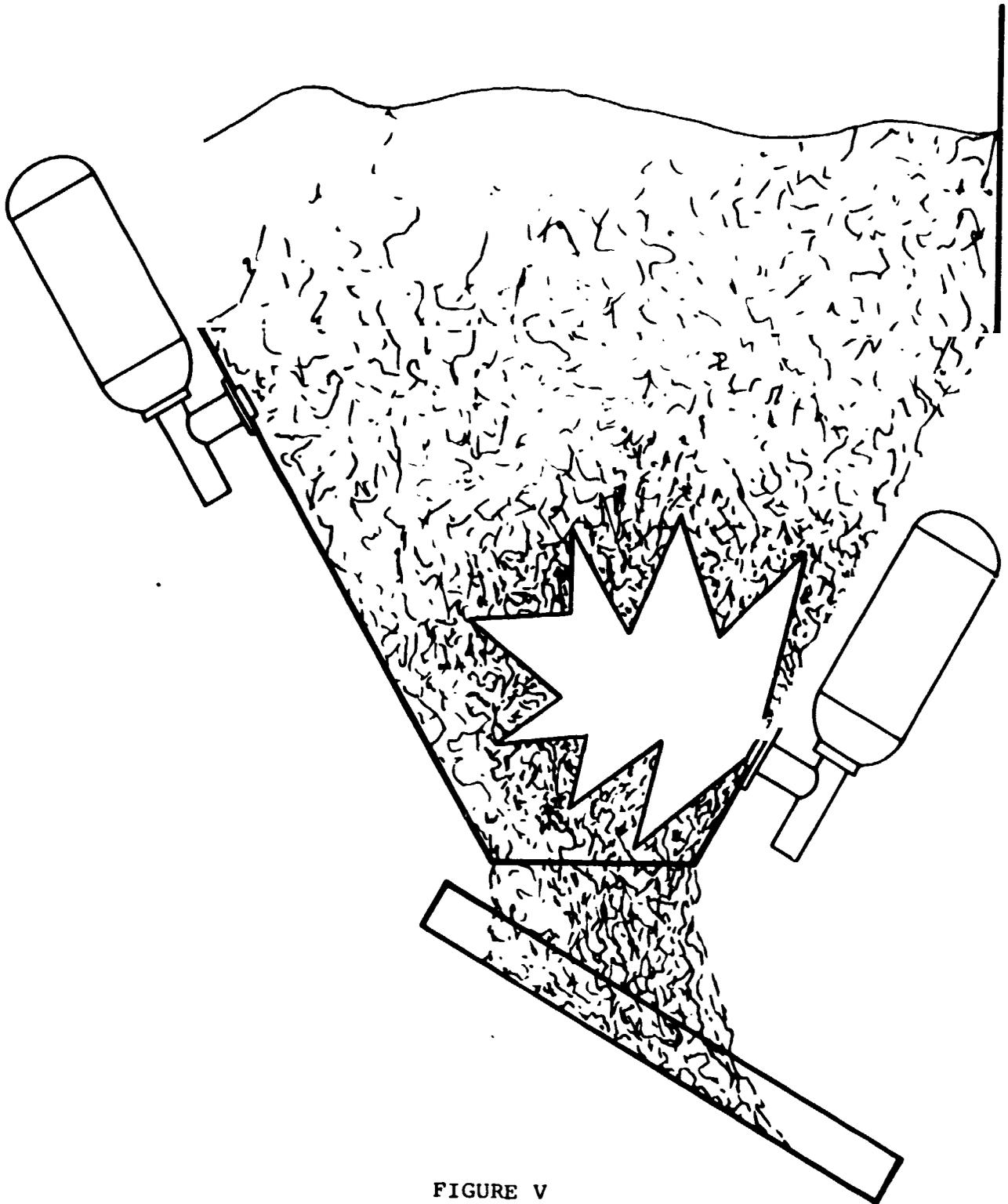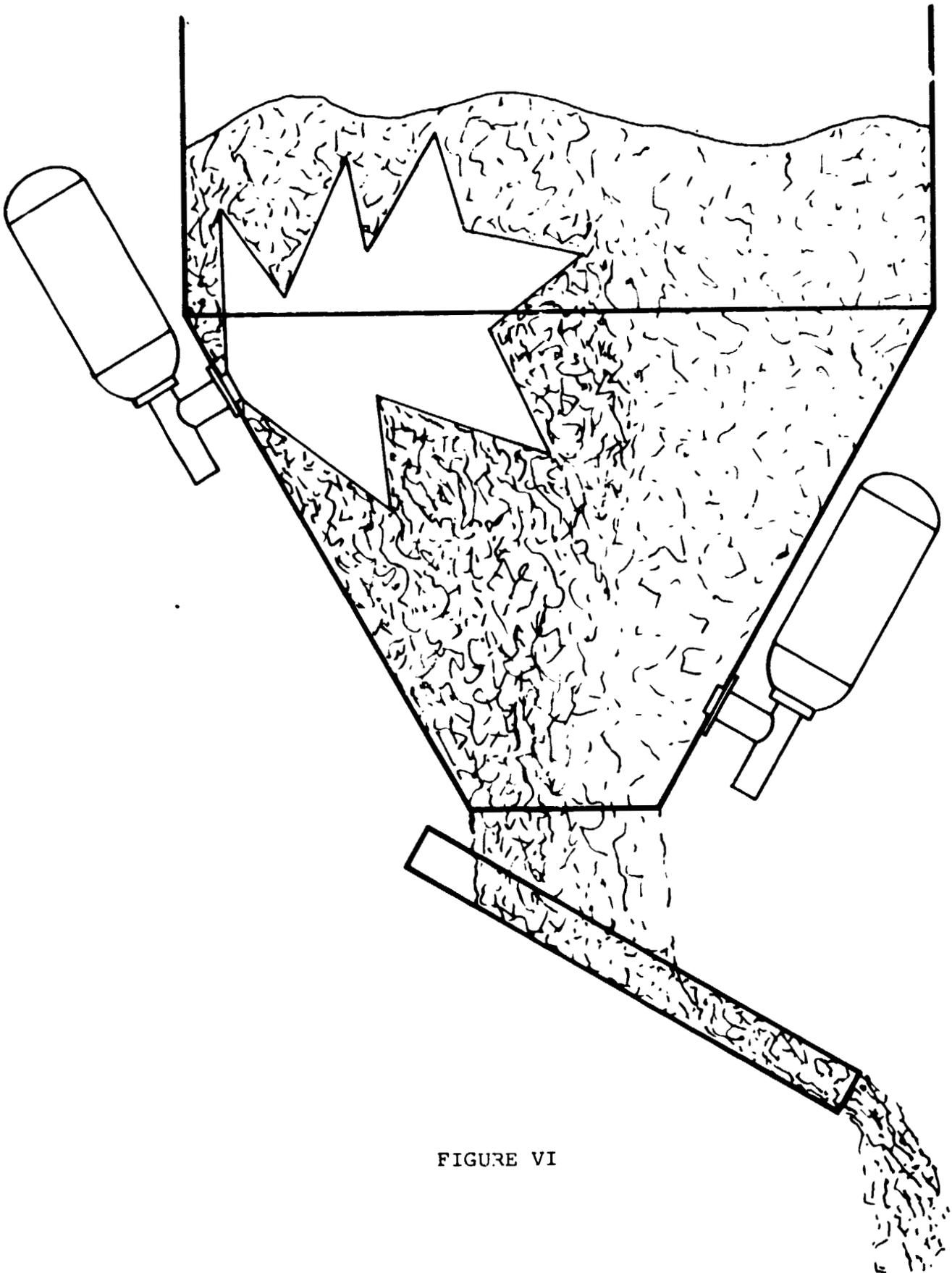
FIGURE IV

FIGURE V

FIGURE VI
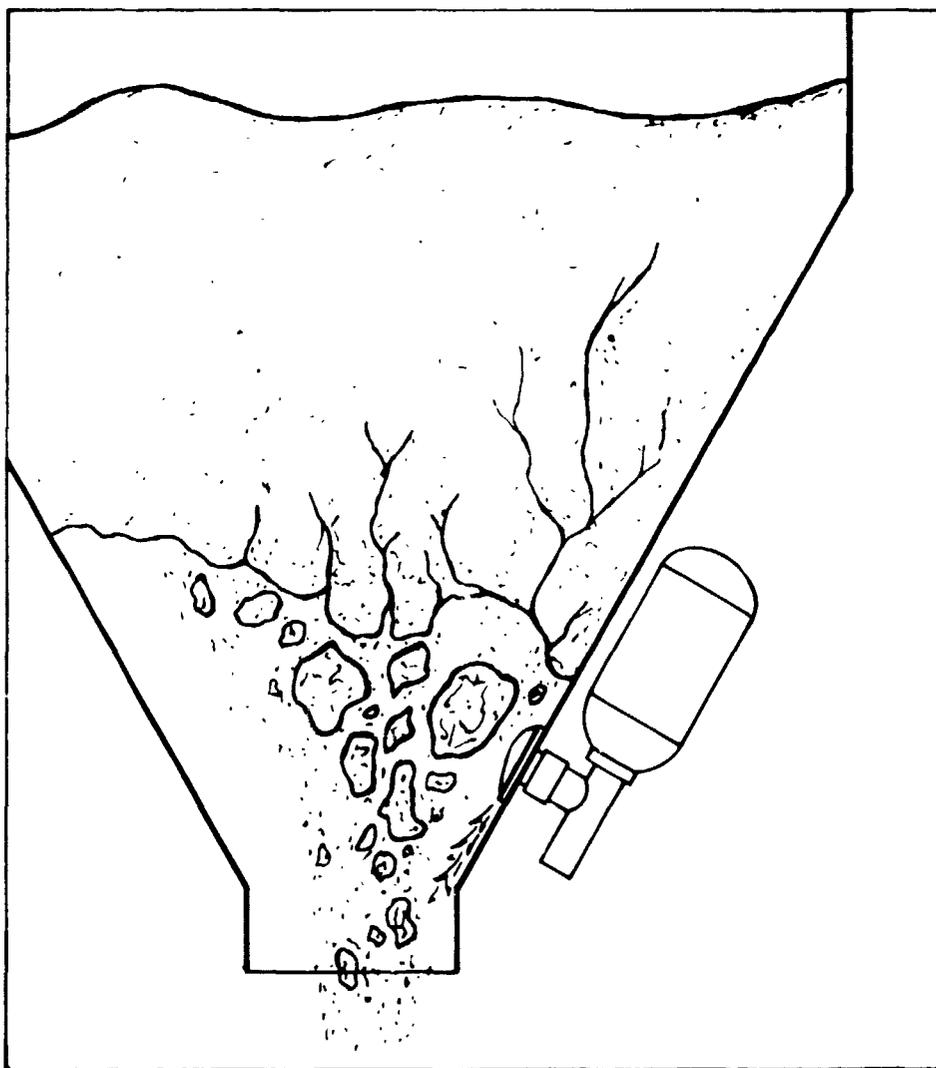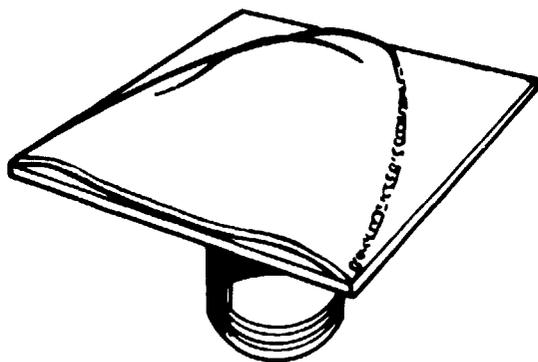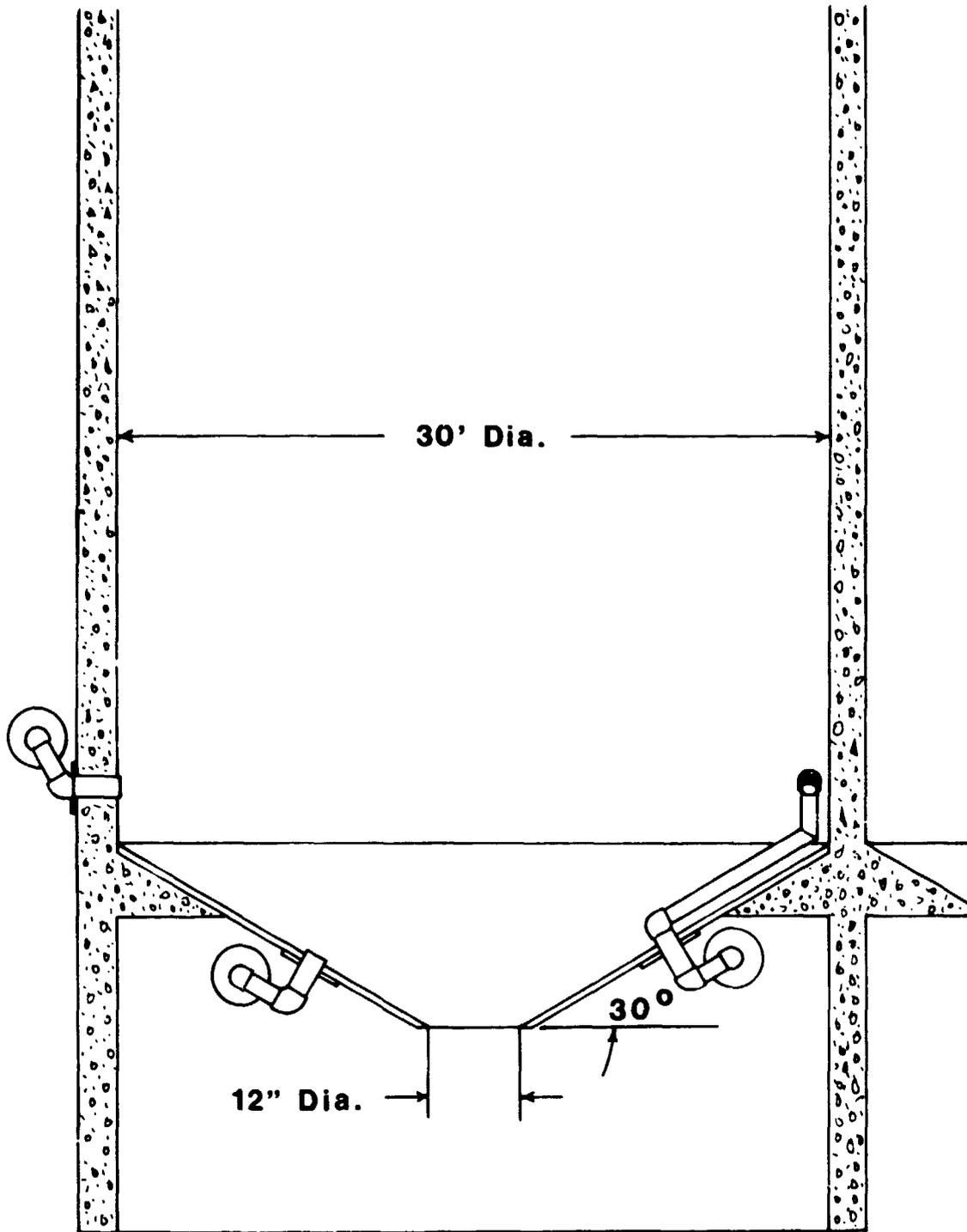
FIGURE VII

FIGURE VIII
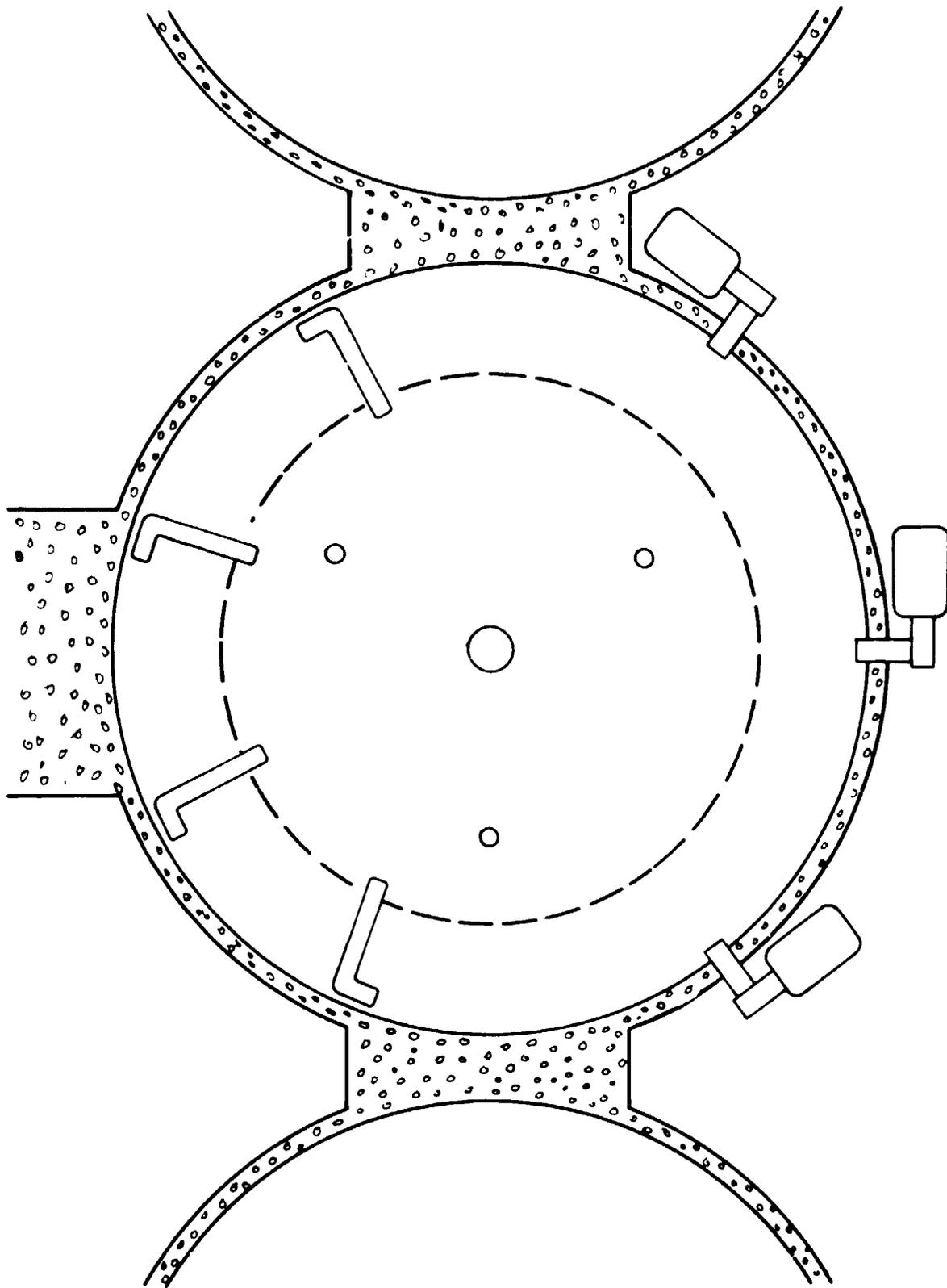
FIGURE IX
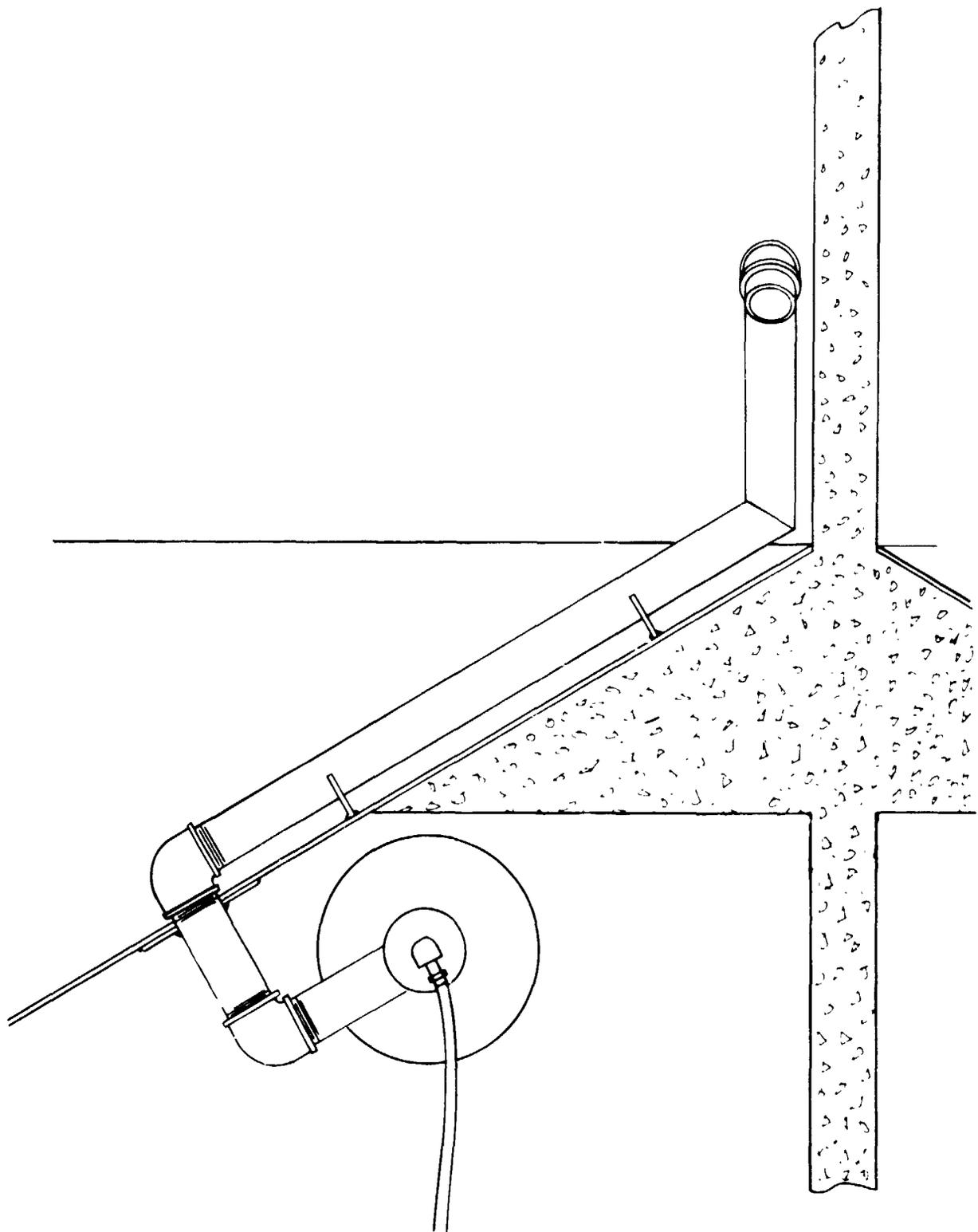
FIGURE X

SESSION IV

ADVANCED CONCEPTS AND ANALYSIS

GRAVITY FLOW RATE OF SOLIDS THROUGH ORIFICES AND PIPES

J. F. Gardner

J. E. Smith

J. M. Hobday

U. S. Energy Research and Development Administration, Morgantown Energy Research Center
Morgantown, West Virginia

ABSTRACT

Lock-hopper systems are the most common means for feeding solids to and from coal conversion reactor vessels. The rate at which crushed solids flow by gravity through the vertical pipes and valves in lock-hopper systems affects the size of pipes and valves needed to meet the solids-handling requirements of the coal conversion process. Methods used to predict flow rates are described and compared with experimental data. Preliminary indications are that solids-handling systems for coal conversion processes are over-designed by a factor of 2 or 3.

Gravity Flow Rate of Solids Through Orifices and Pipes

by

J. F. Gardner[1], J. E. Smith[2], and J. M. Hobday[3]

Lockhopper systems are the most common means for feeding solids to and from coal conversion reactor vessels. The required rates at which these solids must flow affects the size of piping and valves used in the lockhopper systems.

The Morgantown Energy Research Center conducted a literature review on solids feeding to analyze valve size requirements for solids handling service in coal conversion plants. From this literature review, it is apparent that limited investigations have been performed to determine the rate at which crushed solids fl⌐ by gravity from hoppers, pipes and valves.

Most experimental work on gravity flow of solids occurred during the post-WWII years of 1945-1955.[1-6] Gregory[7] in 1952 appraised the problem of gravity flow of solids and drew the following conclusions:

1. The internal pipe diameter should be at least 5 to 7 times as large as the largest particle size encountered in the charge.

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[1] Mechanical Engineer, U.S. ERDA - Morgantown Energy Research Center, Morgantown, WV

[2] Contractor, U.S. ERDA - Morgantown Energy Research Center, Morgantown, WV

[3] Mechanical Engineer, U.S. ERDA - Morgantown Energy Research Center, Morgantown, WV

2. Materials of narrow size range tend to flow more readily. Materials with wide size spectra may exhibit sticking tendencies, as do comparitively small particles (on the order of D_p <0.003 in.).

3. Mild steel encourages particle sticking whereas smoother surfaces of stainless steel and glass are more amendable to unimpeded flow.

4. Flow may be severely affected by surface moisture. Thus a solids surface moisture content in excess of 1 to 2 percent should be avoided.

Gregory's work was a good start on determining the variables which affect gravity flow of solids. Other investigations have determined that the following factors also affect solids flow.

1. Solids Density.

2. Surface properties of the solid.

3. Size and shape of storage reservoir.

4. Length and shape of the standpipe.

5. Temperature of the solids.

6. Velocity effects (transonic effects or Reynolds Number effects.)

7. Electrostatic influences.

Rausch in 1948[5] investigated a wide variety of solids falling through tubes of 3" to 8" diameter. His experimental equipment consisted of a simple hopper with orifices and discharge tubes. He varied the orifice diameters from 1/16" to 2" while varying D_o/D_p from 2.5 to 250. Rausch observed the following effects:

1. Solids head had little effect.

2. The diameter of the discharge duct had little effect on solids flowrate.

3. Particle size had little impact except when considered in combination with orifice diameter.

4. With D_o/D_p <25 the bulk density of the solids passing through the ofifice appeared to be reduced.

5. The angle of approach in the bottom of a delivery hopper (as shown in figure 1) had an influence on flow rate.
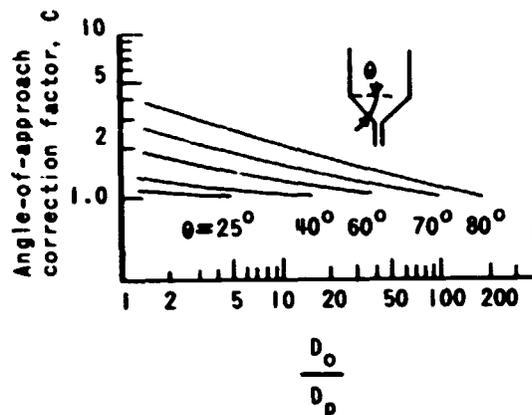
FIGURE 1[5] - Angle of Approach Correction Factor for Gravity Feed Systems

Rausch also developed a correction factor to take into account the pipe wall effect. This is given in Figure 2.
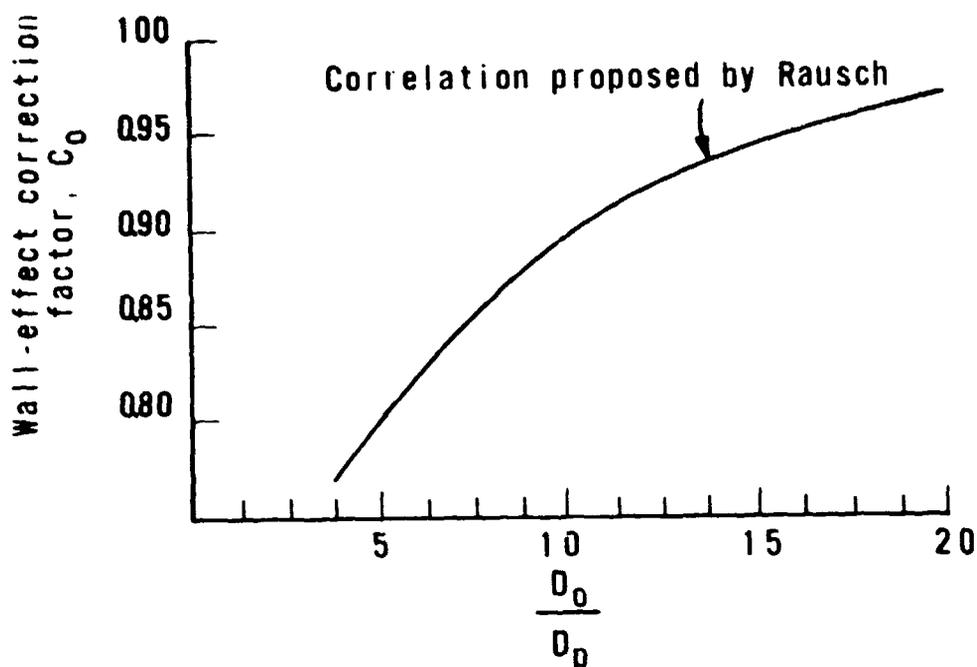
FIGURE 2[5] - Wall Effect Correction Factor for Gravity Flow of Solids

The experimental data of Rausch[5] was correlated and used to develop the following equation:

$$Wa = C \ C_0 \ \frac{(D_0/D_p)^{2.70} g_c^{0.5} \rho_b}{\tan \beta_a^{0.5}} \ D_p^{2.50} . \qquad (1)$$

When rearranged this equation yields:

$$Wa = C \ C_0 \ \frac{\rho_b}{D_p^{0.2}} \ \left[\frac{g_c}{\tan \beta_a}\right]^{0.5} D_0^{2.70} = C_s D_0^{2.70} . \qquad (2)$$

C_s is a constant that is characteristic of the solids involved.

A similar form of this equation has been proposed by a variety of experimenters. These equations differ in the values of C_s and in the exponent of the orifice diameter.[2-8] These differences are listed in Table 1.

Zenz[10] concluded that the Rausch[5] correlation is relatively accurate for a wide range of solids flow conditions. Zenz felt that further tests should be performed to verify and correct the correction factors used in equation 2.

A series of solids flow experiments were performed at the Morgantown Energy Research Center. Limestone solids with the character-istics given in Table 2 flowed from a solids storage bin through a pipe or orifice plate for a specified time into a receiving hopper. The solids were then weighed and a flowrate calculated.

The discharge configurations used in the experiments consisted of 7-foot lengths of 1-1/2, 2, 4, 6, and 8-inch diameter, Schedule 40 carbon steel pipes, and 4, 6, and 8-inch diameter flat plate orifices. For each individual test, the pipe or plate was attached to the bottom of a 25-ton capacity storage hopper having an inverted-frustrum bottom. Additional experiments were also conducted using a variety of types of reducer sections (both conical and concentric in shape) located between the hopper bottom and the discharge orifice.

The results of these experiments are given in Table 3. Photographs of the tests and a typical experimental configuration are shown in Figures 4, 5, and 6.

Observations from these experiments were:

1. A uniform flow of the solids occurred in all tests. The observed flow of the solids (as photographed through a plexi-glass tube) indicated an acceleration of particles and a nearly full tube of solids at the start. The flow stream then tapered with increasing distance from the beginning of the tube.

2. The flow of solids was uniform during the entire period of the tests.

3. A "choked" flow condition did not occur during the tests. A "choked" flow condition, however, could be induced by restricting a small amount of the flow area.

Table 4 compares predicted flowrates with the actual flow rates found in the MERC tests.

From Table 4 it is apparent that flow rates calculated from the equations developed by previous experimenters correlate poorly with the results obtained at MERC. This indicates that good correlations are not available for designing gravity-flow systems for handling crushed solids in coal conversion processes. Further investigation into this problem area thus needs to be done. With accurate prediction methods for solids flow, valves and piping for coal conversion process plants could be accurately sized. For example, the MERC data show that 3885 tons/day of drushed solids can flow under gravity head through an 8" vertical pipeline. This indicates that relatively small valves and piping could be sufficient for commercial-scale coal conversion plants. This would lower costs, and lessen design, fabrication, and operating problems with valves.

The experiments conducted at MERC were not sufficient to fully investigate the effects of pipe and valve size on gravity flow rates of crushed solids. The results indicate the need for further studies. MERC has proposed such studie' to Fossil Energy, ERDA.
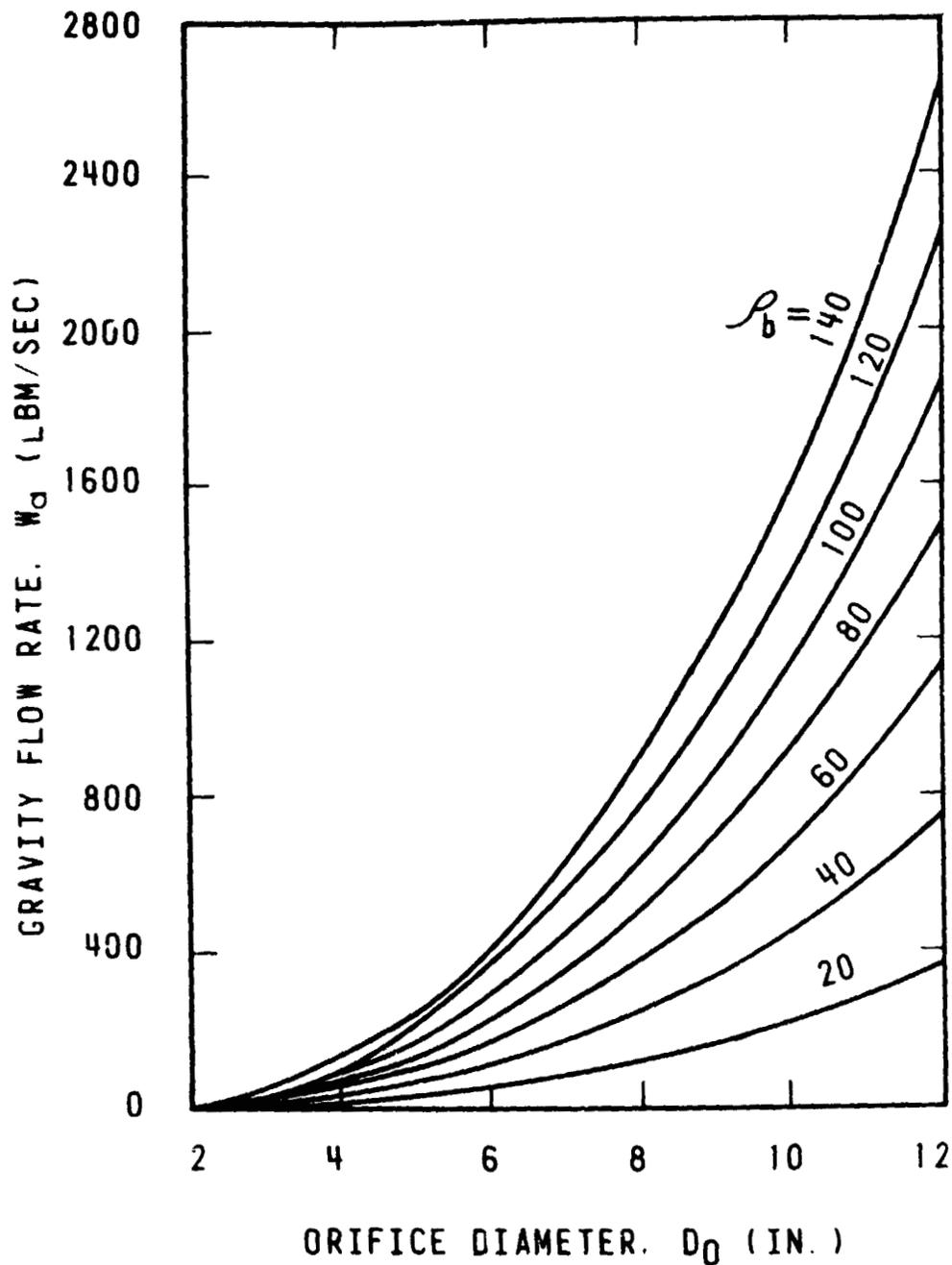
FIGURE 3 - VALUES OF FLOW RATES FOR VARYING BULK
DENSITIES AND ORIFICE DIAMETERS.
BASED ON RAUSCH'S EQUATION [5]
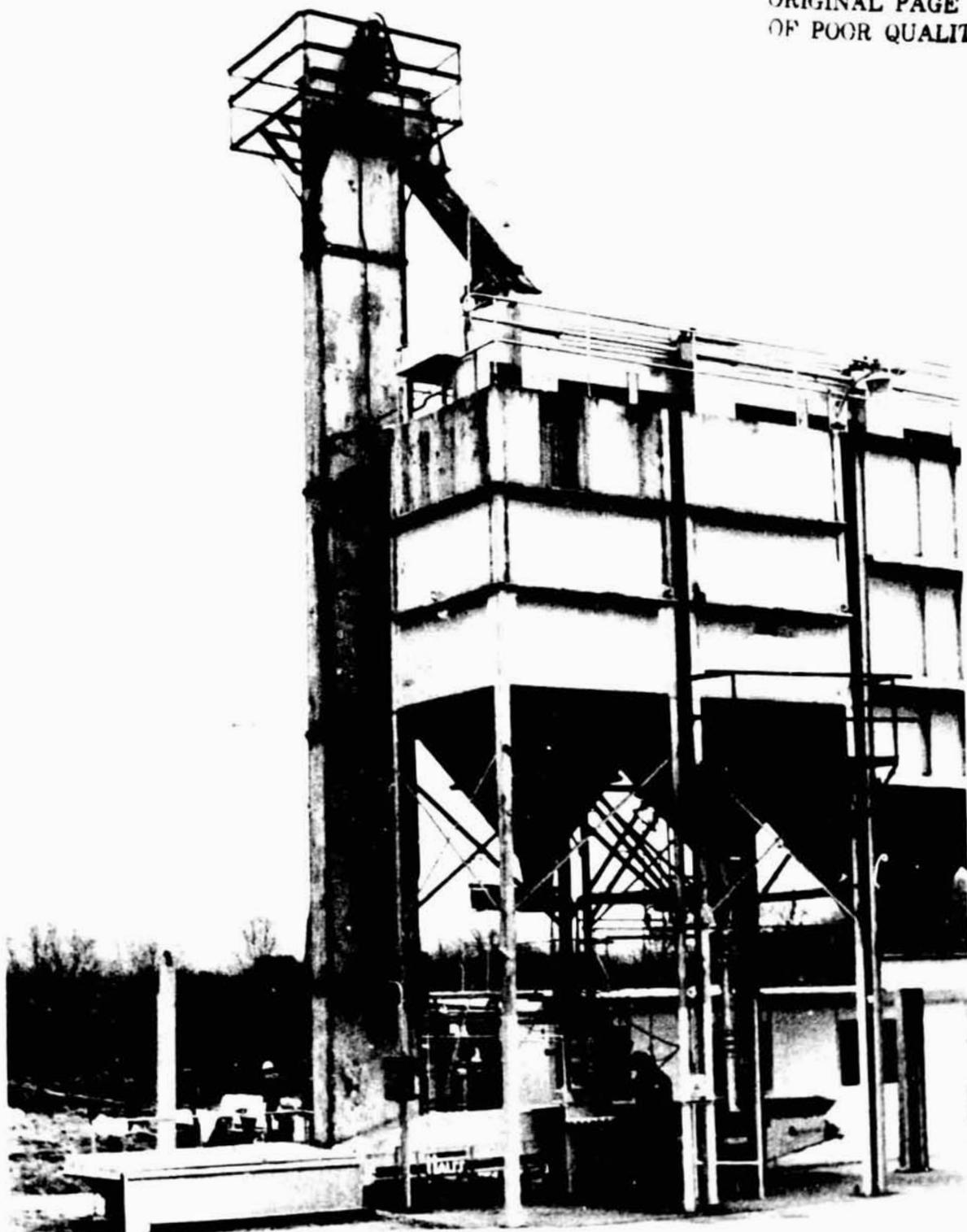
FIGURE 4 - VIEW OF STORAGE HOPPER
USED IN MERC SOLIDS FLOWRATE TESTS
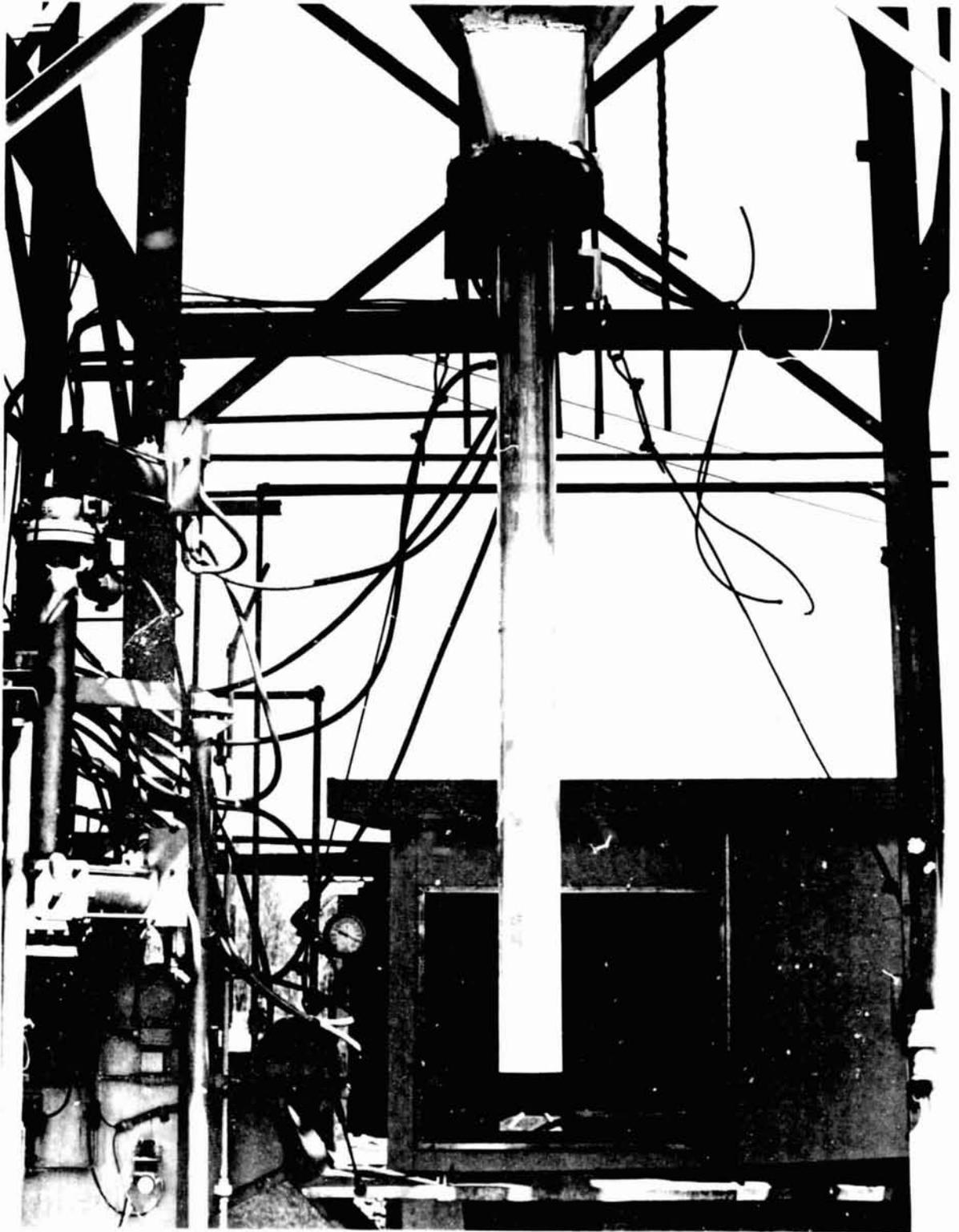
FIGURE 5 - TYPICAL CONFIGURATION
USED IN SOLIDS FLOWRATE TESTS
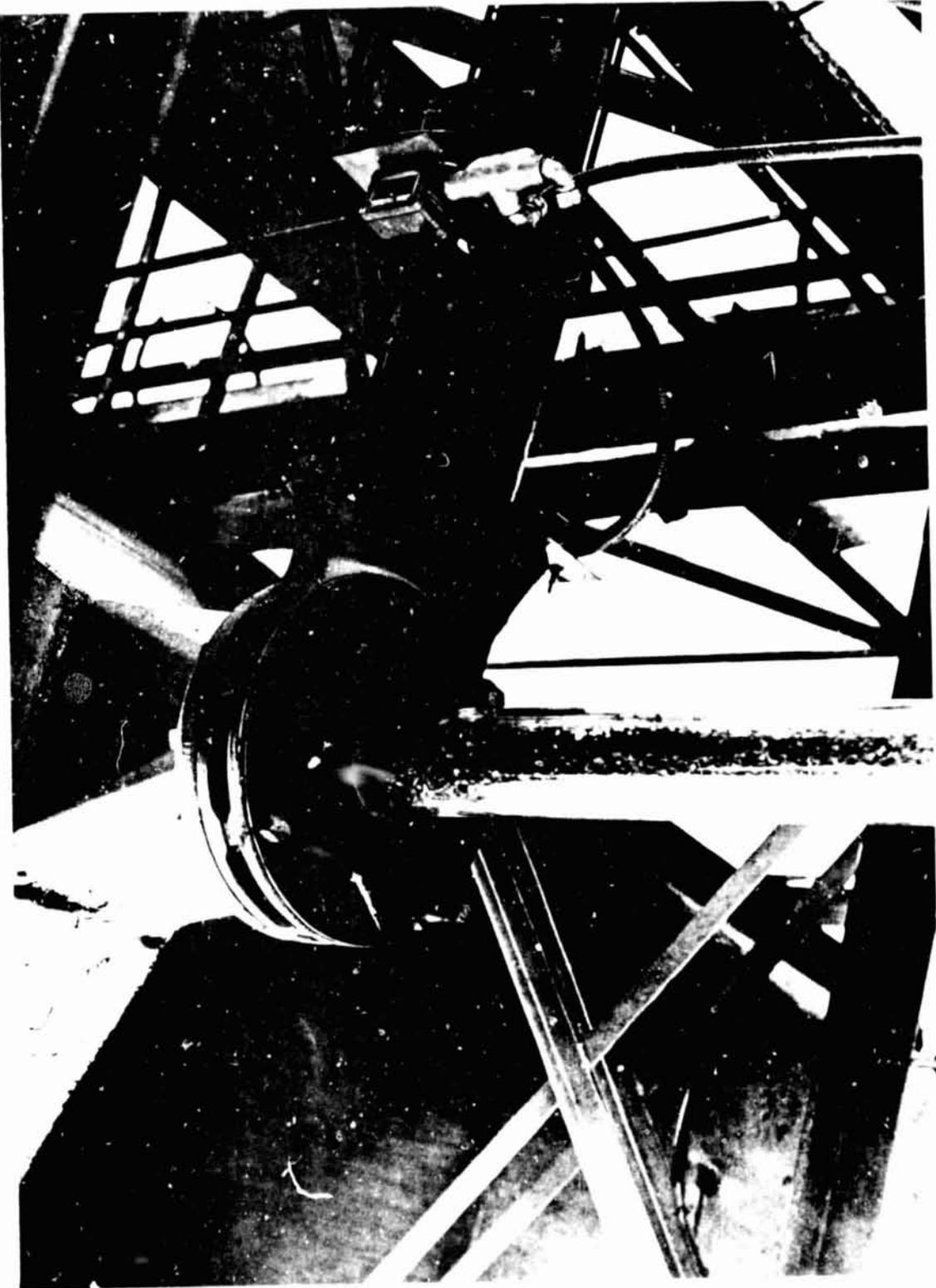
FIGURE 6. - View of Solids Flow During Test

TABLE I. - Reference Values of C_S and the Orifice Diameter Exponent of Various Experimenters

Reference	C_S	Exponent
Gregory [7]	0.28	2.50
Newton et al. [4]	0.14	2.96
Kelly [3]	0.16	2.84
Franklin & Johanson [2]	0.05	2.93
Rausch [5]	0.65	2.70
Kuwai [8]	0.82	2.75
Shirai [6]	0.18	2.50

Table 2 -   Properties of Limestone Solids Used
in Solids Gravity Flow Tests at MERC

Material   -   Limestone

ρ_b = Bulk Density =   88.17 lb_m/ft^3

β_a = Angle of Repose = 38°

D_p = verage Particle diameter = 0.172 in.

<u>Size Analysis:</u>

Screen Size	Weight Percent, %
5/16" x 1/4"	.9
1/4" x 1/8"	82.7
1/8" x 1/16"	7.3
1/16" x 16 mesh	3.0
16 mesh x 30 mesh	0.5
30 mesh x 50 mesh	0.2
50 mesh x 100 mesh	0.3
100 mesh x 200 mesh	0.4
minus 200 mesh	2.7

TABLE 3. - Results of Solids Flowrate Tests
Conducted at MERC

Test Configuration No.	Orifice Size, D_O, in.	Pipe Length L, Ft.	Av. Solid Flowrate W_a lbm/sec	Av. Solids Flowrate* WA, lb/hr (x10^3)
PIPES				
1	1.5	7.0	1.01	3.64
2	2.0	7.0	2.28	8.21
3	4.0	7.0	13.99	50.36
4	6.0	7.0	43.12	155.23
5	8.0	7.0	89.93	323.75
PLATES				
6	4.0	---	13.78	49.61
7	6.0	---	41.74	150.26
8	8.0	---	82.65	297.54

* Solids used as Described in TABLE 2

TABLE 4.- Calculated Flow Rate of Solids W_a (lbm/sec)
Predicted by Various Researchers.

	D_O, in		
	4	6	8
1 M.E.R.C. '*	13.99	43.12	89.93
2 GREGORY [7]	8.96	24.69	50.69
3 NEWTON et al. [4]	8.48	28.15	65.96
4 KELLEY [3]	8.20	25.95	58.73
5 FRANKLIN & JOHANSON [2]	2.90	9.53	22.13
6 RAUSCH [5]	27.45	82.02	178.34
7 KUWAI [8]	37.11	113.17	249.64
8 SHIRAI [6]	5.76	15.87	32.58

** Actual flow rates observed experimentally at MERC, as shown in
Table 3. All other values are calculated.

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Nomenclature

D_o = diameter orifice (in.)
D_p = diameter particle (in.)
WA = flowrate (lbs/hr)
C = correction factor (angle of approach)
Co = correction factor (wall effect)
g_c = gravity (386.5 lbm in/lb$_f$ sec^2)
ρ_b = bulk density (lbs/in^3)
β_a = angle of repose of poured solids (degrees)
L = fluid column height (ft.)
C_s = constant characteristic of the solids involved
Wa = flowrate, lb/sec
De = effective orifice diameter (in.)
h = height of reservoir (ft.)
d = diameter of reservoir (ft.)

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N78-13261

HIGH PRESSURE ROTARY PISTON COAL FEEDER

J. F. Gardner

U. S. Energy Research and Development
Administration, Morgantown Energy Research
Center
Morgantown, West Virginia

H. T. Gencsoy

West Virginia University
Morgantown, West Virginia

D. C. Strimbeck

U.S. Energy and Research and Development
Administration, Morgantown Energy Research
Center
Morgantown, West Virginia

ABSTRACT

This feeder concept uniquely combines the functions of solids feeding, metering, and pressurization into one compact system. Success with the rotary-piston concept would provide a lower-cost alternative to lock-hopper systems.

The rotary-piston coal feeder was conceived at MERC. Initial design of the feeder was accomplished by WVU personnel under contract to ERDA. The design of the feeder is presented, with special emphasis on the diffi-cult problem of seal design. Initial tests will be to check seal performance. Subsequent tests will evaluate solids-feeding ability.

HIGH PRESSURE ROTARY P:: TON COAL FEEDER

by

J. F. Gardner[1], H. T. Gencsoy[2], and D. C. Strimbeck[3]

INTRODUCTION

An important step in all gasification processes is the feeding of coal from atmospheric pressure into gasifiers operating at pressures up to 1500 psig. The ultimate objective of this work is a coal feeder with discharge capabilities of 1500 psig and 350°F. The critical design problems are the rotary-piston mechanism, and lubrication, wear and sealing of feeder components. A two-step design procedure thus was followed.

The basic mechanisms of a rotary feeder were first developed. Consideration then was given to sealing, wear, and lubrication. The initial design is for 100 psig discharge pressure at 300°F, with a feed capacity of 200 to 1000 pounds per hour. Design problems associated with the rotary feeder must be solved at these modest conditions before higher pressures and flows can be achieved.

This design work was done in Phase I, the first year of a three-year ·esearch program supported by ERDA at West Virginia University. The design work is covered in this paper. Phase II consists of fabrication and testing of the prototype pump. Phase III will extend the design to 1500 psig with feed rates of several tons of coal per hour.

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[1] Mechanical Engineer, U.S. ERDA -- Morgantown Energy Research Center, Morgantown, WV.

[2] Professor, Mechanical Engineering & Mechanics, West Virginia University, Morgantown, WV

[3] Chemical Engineer, U.S. ERDA -- Morgantown Energy Research Center, Morgantown, WV.

ROTARY PISTON COAL FEEDER

Feeder Operation

Assembly drawings of the rotary piston coal feeder are shown in Figures 1 & 2. The main parts are an outer casing with an inlet hopper and a discharge opening, a rotating (hollow) disk assembly with a thin rim containing the injection chamber, piston and rod, and a drive shaft assembly. The reciprocating action of the piston is provided by two fixed conjugate cams secured to the outer casing and two roller followers attached to the piston.

The feeder operates as follows. Coal fills the open piston cylinder at top dead center (0°). The piston position then remains constant as the piston and its rotating housing moves clockwise through an angle of 135°. During the next 75° of rotation, the piston is driven radially outward by the contact between the cams and followers. The coal empties by gravity through '· ? discharge. During the next 60° of rotation, the piston dwells at the maximum outward position. The piston then returns to the loading position during the next 70° of rotation. The piston then dwells in this retracte· position for the final 15° of rotation prior to charging with coa.

The rotating disk is driven by an electric motor through a speed reducer at rates up to 30 rpm. This will provide a feed rate of approximately 1200 lb/hr (ρ = 30 lb/ft^3) for finely crushed coal.

Basic Feeder Components

The distinguishing feature of this feeder is a rotating disk containing the injection chamber, and a cam-activated reciprocating piston which provides the pumping action for feeding the coal particles into

a chute against the discharge pressure. The following are brief descriptions of some of the main feeder components.

The Rotating Disk and Piston Assembly

The heart of the rotary-piston coal feeder is the internal rotating disk to which many of the internal parts are supported or connected. The input rotational energy is directly transferred to the disk through the drive shaft, causing the disk to rotate inside the outer casing on two 27-inch outside-diameter ball bearings. With the disk rotating, the cam followers, which are attached to the piston rod, roll along the fixed cam profiles, imparting a reciprocating motion to the piston. This action causes the coal to discharge, and then allows the piston to retract for refilling.

Design of the rotating disk provides sufficient rigidity for proper installation of the large diameter ball bearings, input drive shaft, and other internal parts. The rotating disk is a thin cylindrical shell which is closed at one end. The disk is one piece cast steel construction with all support areas cast integrally with the disk. The interior of the disk is sealed from the high pressure gases at the discharge area.

The rotating disk supports the piston rod at two locations with linear ball bearings. This arrangement reduces friction losses and assures perfect alignment of the piston in the cylinder during operation. A rolling diaphragm at the piston head prevents coal particles from entering the interior of the rotating disk. The diaphragm also seals against high pressure gases when the piston cylinder is discharging coal. For added sealing protection, rod seals and wiper rings are fitted into machined grooves in the area directly below the cylinder bore of the rotating disk.

The Outer Casing

The outer casing consists of two semi-circular cylindrical shells joined at the center with casing bolts, and two circular end plates. The outer casing contains the inlet at the top, and the discharge is attached at the bottom. Wear compensating seals are located around these openings on the outer casing to keep high pressure gas and coal particles from penetrating to the inside of the feeder and contaminating the rotating disk and its feeding mechanism.

Cams and Cam Followers

The reciprocating action of the piston is caused by two fixed conjugate cams attached to the cam side end-plate. Two cam followers secured to the piston rod ride on the cam profile of the rotating disk, providing the forward and backward strokes of the piston in the cylinder. The cam profiles are specially designed to impart a simple harmonic motion to the piston, with a 2-inch stroke and a maximum pressure angle of 30°.

The cam followers are mounted eccentrically on the studs. This provides a preload between the followers and cam surface to ensure constant contact.

Large Diameter Ball Bearings

Two 27-inch O.D. Ball bearings support the rotating disk at both ends in the outer casing. This assures low friction loss, and minimizes deflection and bearing problems associated with the drive shaft. This arrangement also aligns the rotating disk in the outer casing, and thus maintains seal dimensions around the inlet and discharge openings.

Coal Feeder Seals

Sealing problems associated with this feeder design are unusual and challenging. Effective sealing of the piston and rod when facing the high pressure discharge is achieved by the rolling diaphragm in the piston head, and the combination of the seal and wiper rings in the rod, as discussed previously.

Seals between the outer casing and the rotating disk at the inlet and discharge openings had to be specially designed. Commercially available products could not be used. Three types of wear compensating seals were designed for the inlet and discharge openings. The basic ideas behind these designs can be seen in Figures 3 and 4. The principal parts in the wear compensating seals consist of a primary seal nose contacting the rotating disk along the periphery of the discharge opening, and a secondary seal supporting the primary seal with adjustable pre-load elements. A pressure gap is incorporated in all these seals between the seal g. rd and seal nose back surface. The pressure of surrounding gases thus helps to force the seal against the rotating disk and make the contact area leak-proof. Wear is also automatically compensated by the deflection of the seal lip to provide long life for the seal. The seals are also designed for easy accessibility and maintenance.

The three seal assemblies are being built. The three assemblies principally consist of a primary seal nose and a secondary seal. Three different seal materials, i.e. EPDM rubber, polyimide carbon composite and Teflon, bonded onto an epoxy glass fabric laminate, will be used for the primary seals. All three secondary seals will be made from EPDM rubber. The materials selected for the test seals are thought to

be the most compatible with the chemical, thermal and wear requirements for the coal pump. These seals will be tested on a feeder prototype being built at the ERDA, Morgantown Energy Research Center, Morgantown, West Virginia.

A Teflon fiber gasket tape is used as a gasket between the outer casing shell and end plates. The tape has excellent durability and sealing characteristics and is easily installed. The large end bearings are also protected from contamination by a bearing seal strip snap fitted into a groove machined in the outer casing.

SUMMARY

The feeder design consists of a rotating disk in a cylindrical feed chamber containing a reciprocating piston. The piston seals the feed chamber after the coal is dumped into a reactor vessel so that there is no loss of gas from the reactor. These are the only moving parts in the feeder, and their rugged construction should present minimum wear problems.

This feeder is designed for 1200 pounds per hour, operating at 30 rpm. The body of the feeder is 27 inches in diameter and 12 inches wide. The compact design makes it possible to use more than one feeder in parallel to get higher coal feed rates. Higher values of rotational speed could also increase the feed capacity. The drive requirement of a single unit at 30 rpm is 5.3 horsepower.

The coal pump is all-steel construction using commercially-available standard parts wherever possible. The design consists of a minimum number of parts that can be easily manufactured and assembled to provide a reliable unit.

A diaphram seal in the piston head, and a combination of seal and wiper rings in the rod, provide effective sealing of the coal discharge. Seals between the outer casing and the rotating disk at the inlet and discharge openings, however, present challenging design situations requiring special solutions. Three types of wear-compensating seals of various configurations thus were conceived. These seals are being fabricated. They will be tested on a prototype feeder which is currently being made at ERDA facilities in Morgantown, West Virginia.

Success with this coal feed concept will lower the cost and difficulty of pressurizing, metering, and injecting coal into a reactor vessel. The feeder thus could replace lock-hopper feed systems now in general use, including the valves in such systems that are a source of many problems.

If the Phase II tests of the seals succeeds, this work will be extended into Phase III. This phase would be to design a coal feeder for the 1,500 psig pressure required for commercial coal conversion plants.
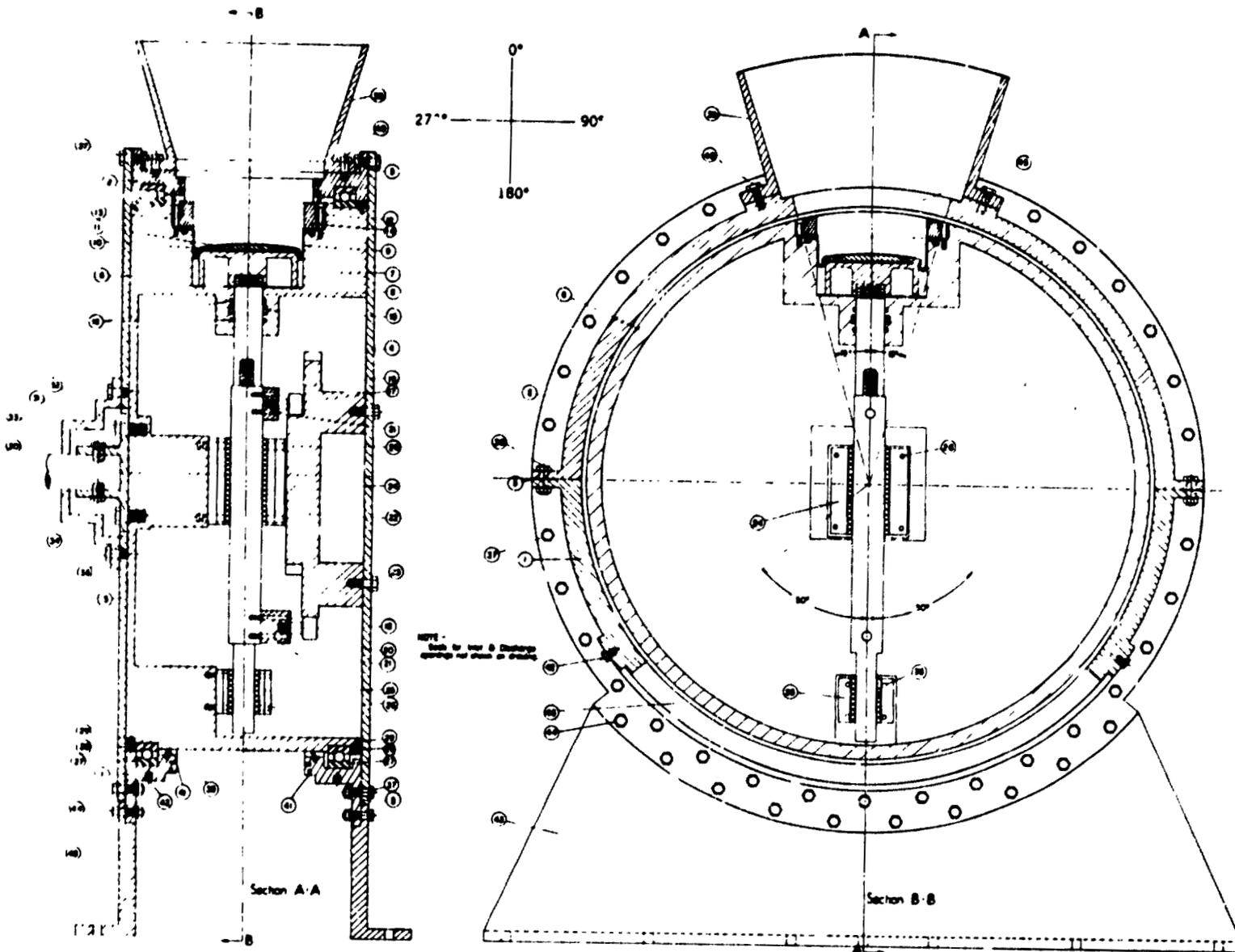
FIGURE 1 · ASSEMBLY OF HIGH PRESSURE ROTARY PISTON COAL FEEDER
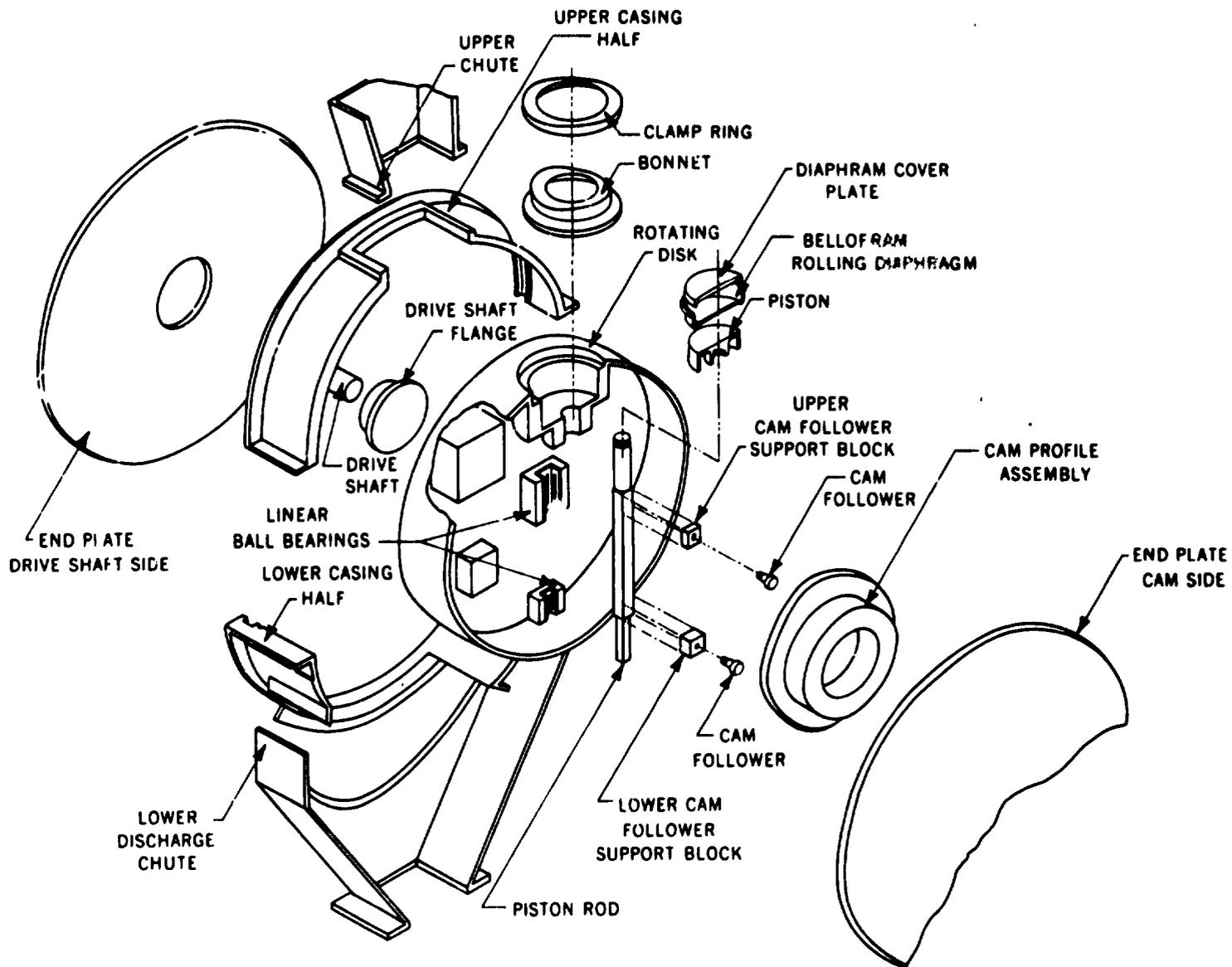
FIGURE 2. - EXPLODED ASSEMBLY OF HIGH PRESSURE ROTARY PISTON COAL FEEDER

HOLES FOR NO. 4 HEX SOC
BUTTON HEAD CAP SCREW

HOLES FOR BOLT PLACED
BETWEEN SPRINGS AT
INTERVALS

HOLES FOR BOLT TO FLANGE
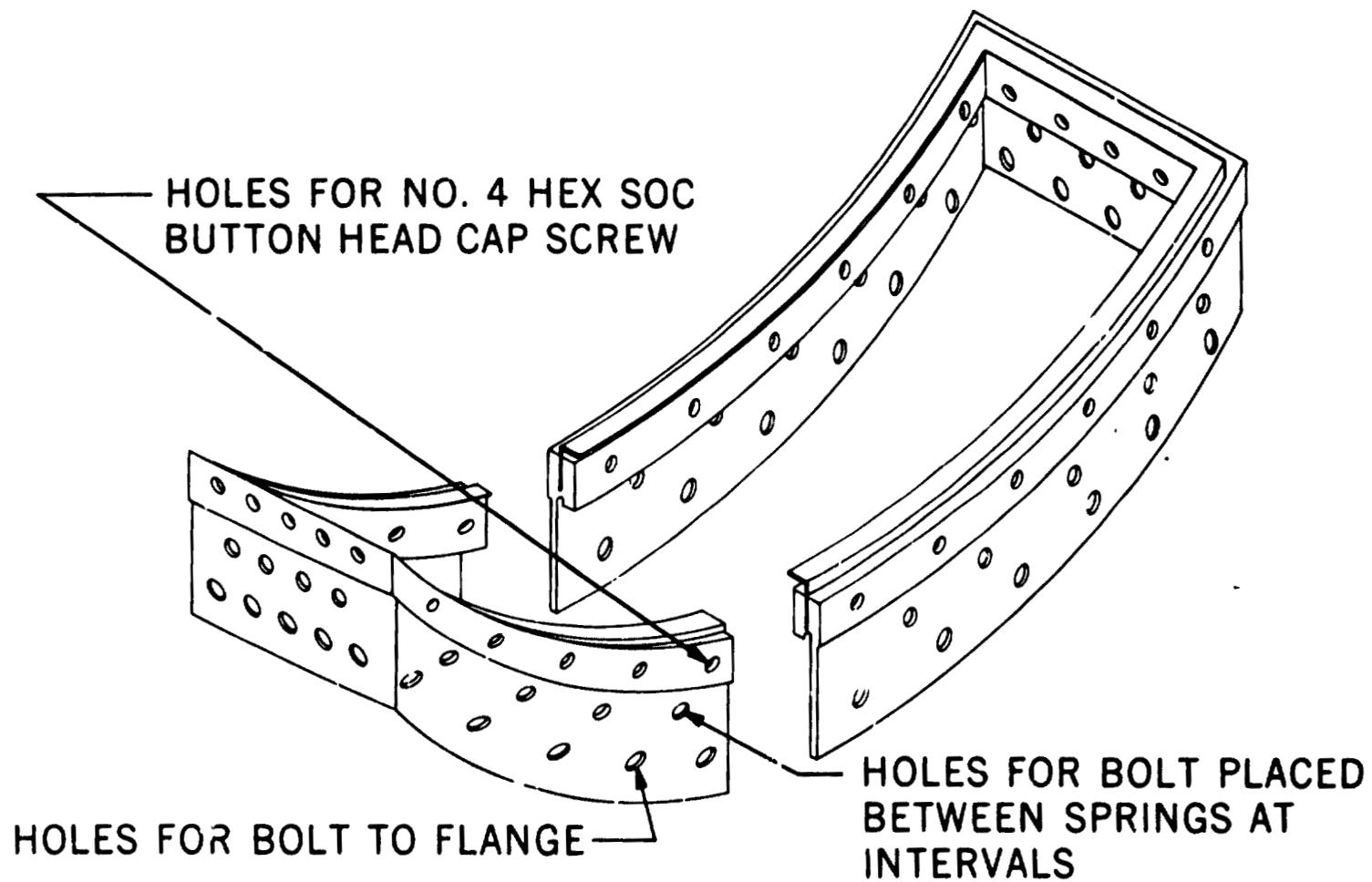
FIGURE 3-SEAL ASSEMBLY FOR HIGH PRESSURE ROTARY PISTON COAL FEEDER

PRIMARY SEAL NOSE

SECONDARY SEAL

NO. 4 HEX NUT

SEAL GUARD

NO. 4 HEX SOC BUTTON HD
CAP SCREW

LOCK NUT

BACK-UP SPRING

FLANGE

BOLT PLACED BETWEEN SPRINGS
AT INTERVALS

BOLT SEAL

SUPPORT PLATE

BOLT TO FLANGE
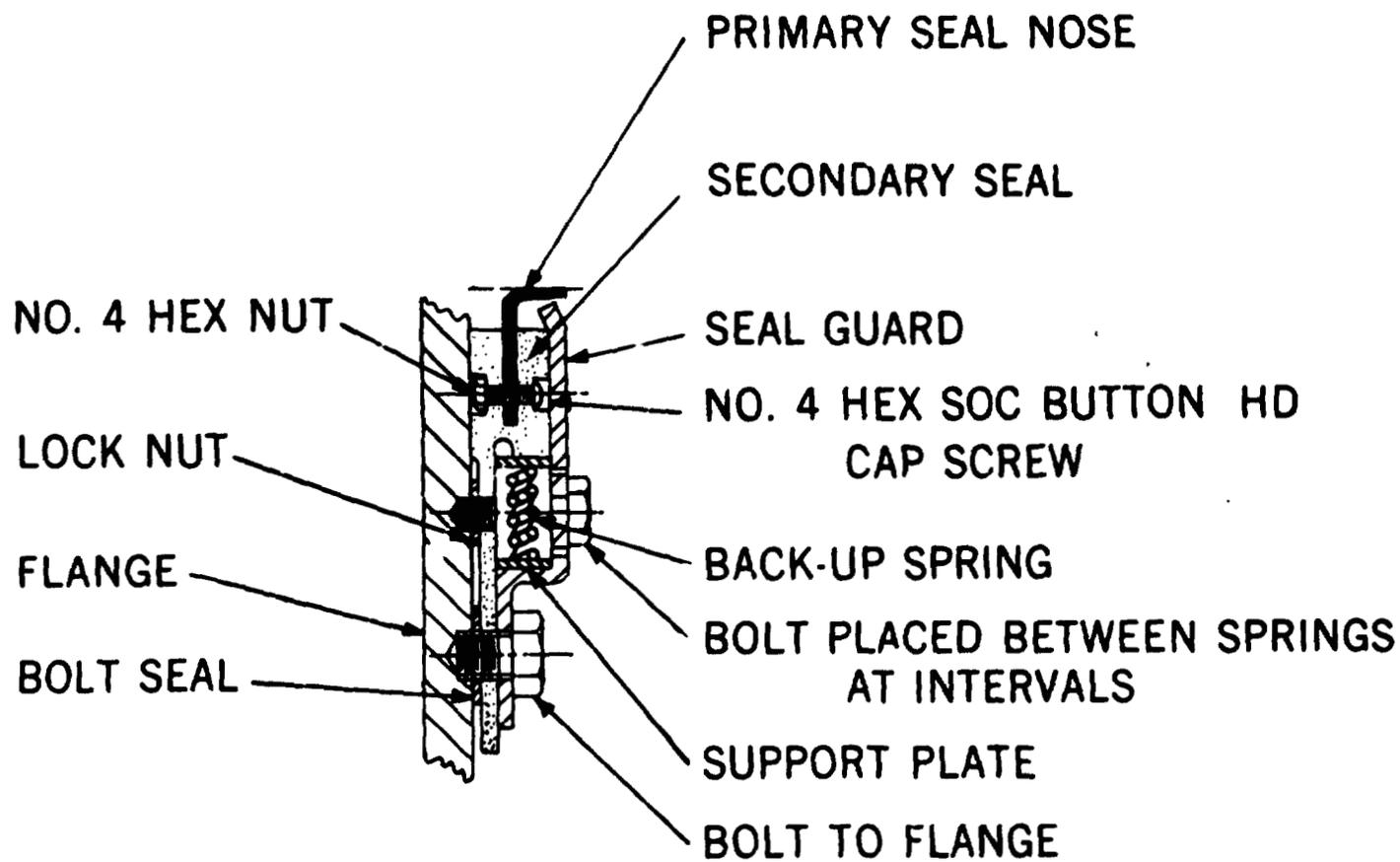
FIGURE 4 - SEAL ASSEMBLY

COAL FEED COMPONENT
TESTING FOR CDIF

C. V. Pearson

B. K. Snyder

T. E. Fornek

Argonne National Laboratory
Argonne, Illinois

COAL FEED COMPONENT TESTING FOR CDIF

by C. Victor Pearson, Burton K. Snyder, Thomas E. Fornek

The significantly higher temperatures, pressures and power densities of magnetohydrodynamic (MHD) combustors, along with the need to utilize low-sulfur, high moisture content coals in very dry form, makes preparation and injection for this type of electric power system quite different from that of current commercial electric power plants. In addition, "seed" material must also be injected at high pressure to enhance electrical conductivity of the high temperature gases.

Investigations conducted during the conceptual design of the Montana MHD Component Development and Integration Facility (CDIF) identified commercially available processing and feeding equipment potentially suitable for use in a reference design. Tests on sub-scale units of this equipment indicated that they would perform as intended.

Studies on open-cycle MHD topping cycle, steam bottom cycle electric power generating plants indicate such systems have the potential of achieving a 50 percent efficiency (coal-pile-to-busbar) at a competitive cost of electricity.

Accordingly, a national goal for MHD has been established that calls for the "Development of a commercially acceptable system for conversion of coal to electric power by a combined MHD-Steam cycle by 1989."

In order to achieve this goal, two major objectives have been established by ERDA:

1. Near-Term Objective (1985)

Design and test MHD components and sub-systems and to integrate these into system tests to be conducted in the pilot scale Engineering Test Facility (ETF) which could be available as early as 1982.

2. Mid-Term Objective (1990)

To develop and operate a commercial scale demonstration MHD electric power plant before 1990, fueled by coal, in an environmentally acceptable manner.

Continue development, after 1990, of MHD technology to improve the performance, reliability and benefits of commercialization of MHD. This will make more efficient use of coal as commercial plants begin to come on line.

The inherently higher temperatures, pressures and power densities of MHD combustors, however, will require new approaches to coal preparation and injection. Utilization of lower rank, low-sulfur, high-moisture-content coals will influence system design.

The coal-feed system development studies presented in this paper are the result of investigations conducted during the conceptual design of the Montana MHD Component Development and Integration Facility (CDIF), the first of the essential facilities in the MHD program of the Energy Research and Development Administration.

The CDIF located in Montana near the Industrial Park 5 miles south of Butte, occupies approximately 50 acres of a 93-acre site. The facility specifications call for a highly flexible 50 thermal megawatt size, multiple test train facility for (1) MHD component developmental testing and (2) determination of the component and subsystem interactions. Coal and seed processing and injection are an important aspect of the major design development program required to develop MHD to a viable commercial status.

The CDIF conceptual design project established a set of requirements for prepared coal characteristics as a result of several MHD combustor design studies. Existing coal processing and injecting systems for commercial power plants were investigated and found to be inherently inapplicable to MHD requirements.

The coal fired combustor designs proposed for CDIF employed a coal combustion process residence time of the order of 50-70 millisec. Seed vaporization residence time is less than 50 millisec. Variations in feed

rate with respect to time and space in the short residence system will adversely affect combustor performance. The net result is to require that the coal and seed feed systems be dependable, extremely uniform in their dense phase feed injection rates and be able to be closely controlled.

Studies by AVCO, Westinghouse, General Electric, UTSI, and others have shown that optimum cycle efficiency for plants employing near-term technical feasibility require combustor conditions of 5-8 atmospheres combustion pressure, and plasma discharge temperatures greater than 4600°F. If air only is used (i.e. no oxygen additive) the combustor air inlet temperature must exceed 2500°F, (depending upon the type of coal used) in order to achieve the 4600°F flame temperature. In the case of the CDIF, Montana Rosebud, a sub-bituminous coal, is the reference fuel and the inlet temperature of the combustion air must exceed 2900°F.

Figure 1 shows the coal combustor proposed for initial tests in CDIF. Table I lists the characteristics of the Rosebud Coal. Figure 2 shows a block diagram of the subsystems making up the CDIF conceptual system.

Early in the evaluation of the status of technology of dry power feed system it became obvious that mechanical metered feed systems could not be used and that gas conveyance and injection appeared to be the only technically feasible method for feeding the K_2CO_3 seed material, and powdered coal fuel into the plug-flow MHD combustors.

2.5m

AIR INLET PIPE

2nd STAGE COMBUSTOR

4.3m

3.5m

1st STAGE COMBUSTOR

AIR INLET PIPE

4.3m

SLAG QUENCH VESSEL, AND GRINDER

12.8m

FIGURE 1

CDIF COMBUSTOR
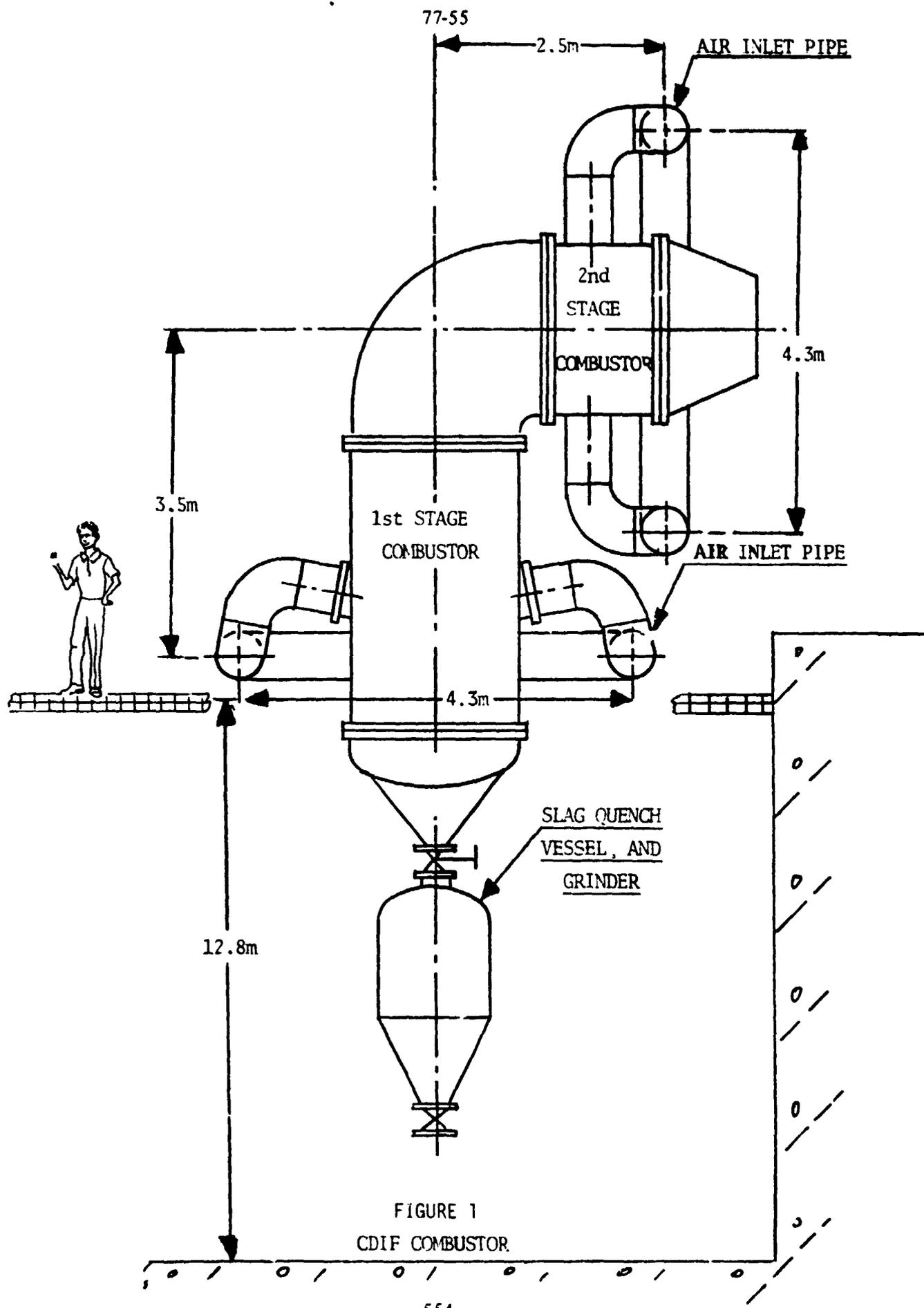
Table 1

CDIF Reference Coals

	Montana Rosebud		Illinois #6	
	Nominal (3)	Range (1, 2, 3)	Nominal (4)	Range (2)
1.0 Ultimate Analysis %, Moisture Free				
Hydrogen	4.6	2.8-6.4	4.8	−
Carbon	65.6	61.8-69.4	68.5	−
Nitrogen	1.0	0.9-1.1	1.3	−
Oxygen	14.4	12.4-16.4	9.2	−
Sulfur	1.1	0.4-5.0	3.6	1.0-6.7
Ash	13.0	6.0-17.0	12.5	7.9-17.5
2.0 Ash Analysis, %				
SiO_2	47.5	22-55	44.7	38-5.
Al_2O_3	21.1	12-25	20.9	13-26
CaO	14.5	5-20	5.8	2-8.5
Fe_2O_3	7.8	2-20	24.1	9-30
MgO	4.6	2.2-7.0	1.8	0.4-2.3
TiO_2	0.8	0.2-1.4	1.0	0.5-1.5
K_2O	0.7	0-1.5	2.3	0.8-2.7
Na_2O	0.4	0-1.2	0.6	0.2-0.8
P_2O_5	0.4	0.1-0.7	0.12	0.06-0.24
3.0 Proximate Analysis, moisture free				
Volatile	37.7	34-42	41.7	30-42
Fixed Carbon	47.6	43-52	45.8	40-52
4.0 Moisture, as received, %	25.5	20-35	9.0	4-9
5.0 Heating Value Dry BTU/lb	11,300	10,650-11,950	12,400	11,700-12,800
6.0 Fusions				
Initial Deformation Temp, °F	2244	1960-2420	1960	1890-2040
Softening Temp, °F	2278	1990-2470	2030	1960-2100
Fluid Temp, °F	2362	2040-2520	2260	2060-2460

References: (1) Reserves and Properties of Rosebud McKay Coal Seams in SE. Montana, D. Brelsford, MERDI, March 1976, ERDA Contract E(49-18)-1811, Task J2.

(2) Coal Conversion Systems Technical Data Book, Institute of Gas Techology, February 1976, ERDA Contract E(49-18)-1730.

(3) Data from (163) full seam, core samples. Colstrip mine, Rosebud seam area C & D, Private communication from Montana Power Company.

(4) Nominal PERC supplied data.
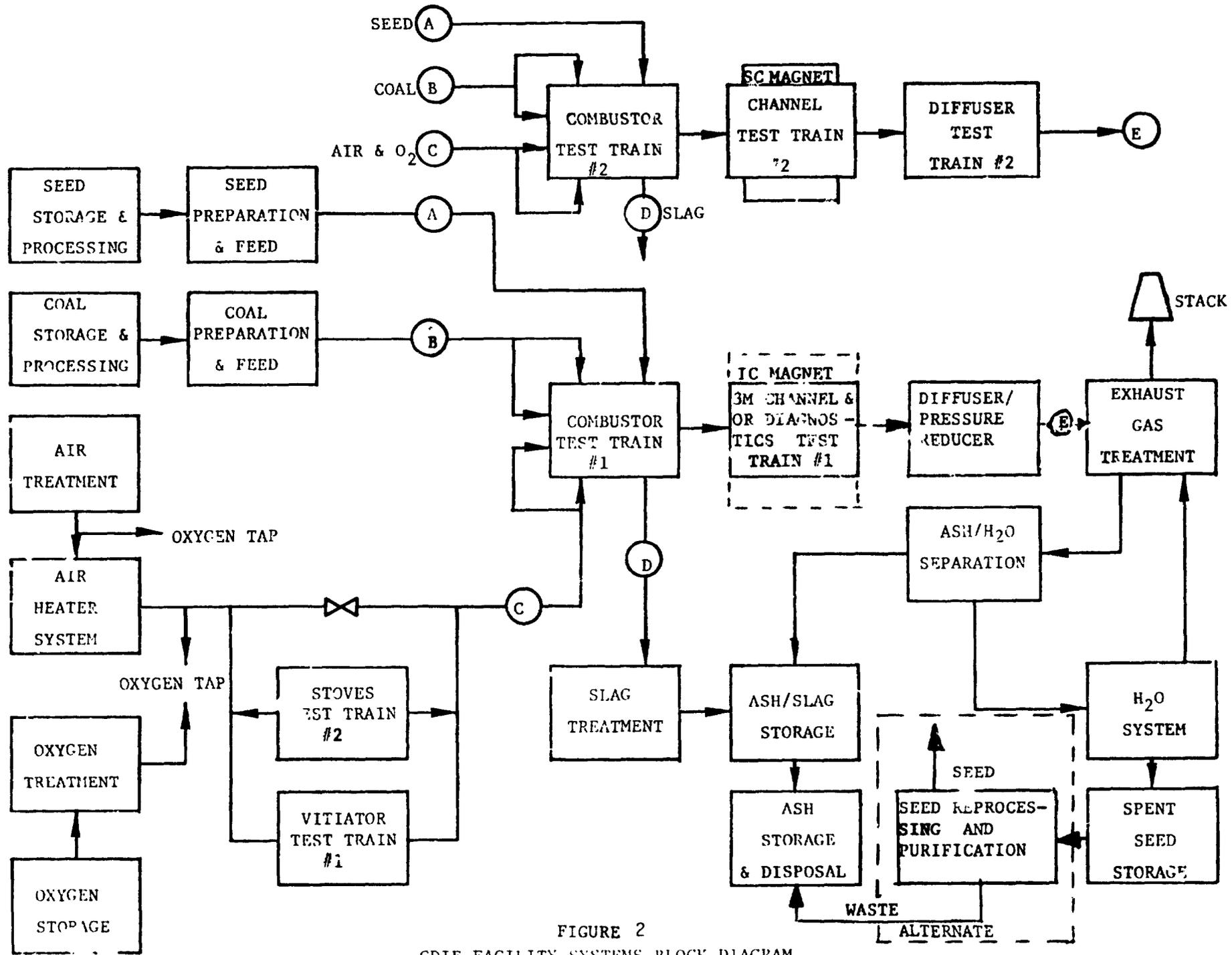
FIGURE 2
CDIF FACILITY SYSTEMS BLOCK DIAGRAM

77-55

The limited study conducted in connection with the CDIF conceptual design included assessment of utility experience with handling and preparation of Montana sub-bituminous coals, survey of existing technology for applicability to the preparation, feeding of sub-bituminous coals under MHD requirements and proof testing to verify design assumptions and commercial vendor assertions. Because of the tight schedule for development of the conceptual design, the program could not afford the luxury of an extensive evaluative program. Instead, due to the expediency of time, reference systems were selected based on proven experience in other applications. The individual components were selected on an engineering judgment basis and then tested to assure that they would work as intended. Those components having large commercial usage under conditions closest to the MHD requirements were given first consideration.

The CDIF conceptual design specified dense phase coal injection (a high as 35 pounds of coal per actual cubic foot of transport gas, or more), at high pressures (up to 10 standard atmospheres). This is considerably different from typical commercial power plant feed practices that use the primary air to convey dilute-phase, pulverized coal directly into the furnace at essentially atmospheric pressure.

In conventional commercial plants moisture is removed from the coal only for the purpose of improving its Hardgrove grindability. The moisture thus removed from the coal ultimately goes back into the furnace with the primary air. Dilute-phase feed and high moisture content fuel are inappropriate for

MHD for several reasons. The need for dense phase coal injection for the sake of achieving extremely high plasma temperatures precludes the use of the relatively cool pulverizer sweep air for coal transport and injection into the furnace. Instead, the MHD system requires separation of the coal from the pulverizer sweep air. This ultimately complicates the coal preparation system by introducing the need for bag filters, cyclones and/or other coal and air separation means.

An additional requirement is imposed by the use of low rank western coals having high moisture contents, (as high as 20% to 30%). In the MHD system moisture imposes a penalty on flame temperature and plasma conductivity, thus requiring the drying of coal to as low as 1% to 2% total moisture content for greatest advantage. Accomplishment of this degree of moisture removal requires higher drying temperatures which, coupled with the high pyrophoricity of sub-bituminous and lignite coals, greatly enhances the probability of explosion and fire and demands the need for oxygen control in the pulverizing and injecting gas systems.

Evaluation of the effects of these commercial application conditions on the MHD combustor performance indicated that certain penalties for less than optimum performance would be incurred. The net effect of inability of the commercially available components to meet all the MHD system requirements was assessed in terms of effect on combustor-power channel performance and total system response to the anomalies generated. In the final analysis, a trade-off was made to determine which off-design conditions created by the use of commercial state-of-the-art feed system components could be tolerated most in this test facility. Of all the off-design conditions, those tending to reduce combustor flame temperature appeared to l the

easiest compensated for (i.e.. add oxygen to raise the flame temperature).
Uniformity of flow and feed rate was considered to be the most critical.
Consequently, all system design features were optimized in order to produce
optimum uniform air, coal and seed flow conditions.

Figures 3 and 4 show the effect of air inlet temperature and coal
moisture content on flame temperature.

The CDIF conceptual design verification tests for the coal and seed feed systems
included processing and feeding Montana Rosebud coal and K_2CO_3 powder through
equipment considered to be best capable of meeting the CDIF requirements.
These tests also gave the manufacturer the necessary experimental data that
in turn could be used to recommend the proper size equipment for the conceptual design.

Ten barrels of coal were obtained by the Montana Energy Research
Development Institute from the Peabody mines and shipped to Williams Patent
Crusher Co., St. Louis to be roller milled to the size and dryness necessary
for safe and dependable storage, handling and feeding in an inert gas storage
and lock hopper feed injection system. Both McKay and Rosebud seam coals
were tested in these preliminary assessments of commercial capability.

A roller type mill was selected because of its ability to be adjusted
to produce uniform size over a wide range of preselected gradations. Since
CDIF is to test different types of combustors, the flexibility of the roller
mill over other types of mill became an important consideration. Tests were
run to determine ability to produce pulverized coal ranging from a coarse
75% - 10, + 100 mesh; and less than 15% fines, down to 95% through - 200 mesh.
Moisture content in the dried coal was to be less than 2% (mine-mouth water
content is normally in excess of 25% for combined surface and bound moisture.)
The tests indicated that with proper adjustment, the coarse coal size range

FIGURE 3
EFFECT OF COMBUSTION
AIR TEMPERATURE ON FLAME
TEMPERATURE
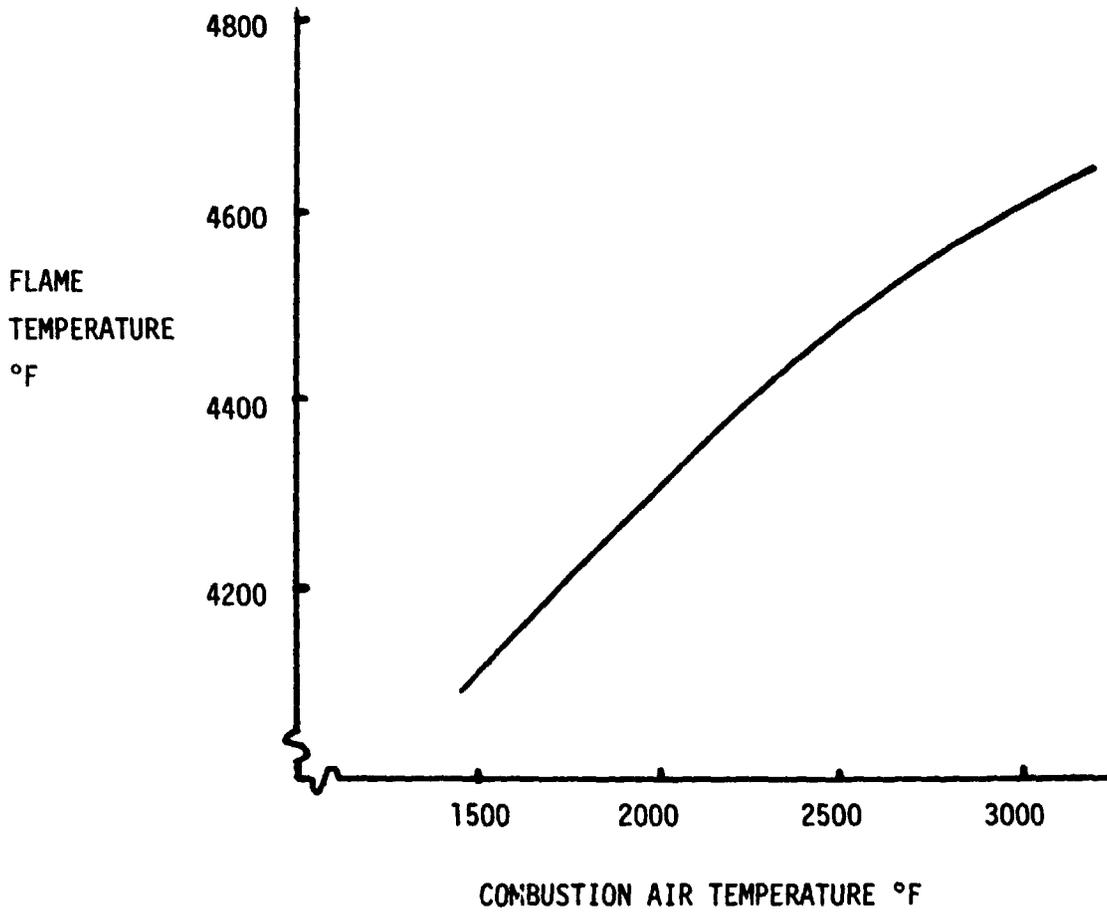
FIGURE 4
EFFECT OF MOISTURE IN ROSEBUD
COAL ON FLAME TEMPERATURE

PERCENT MOISTURE IN COAL
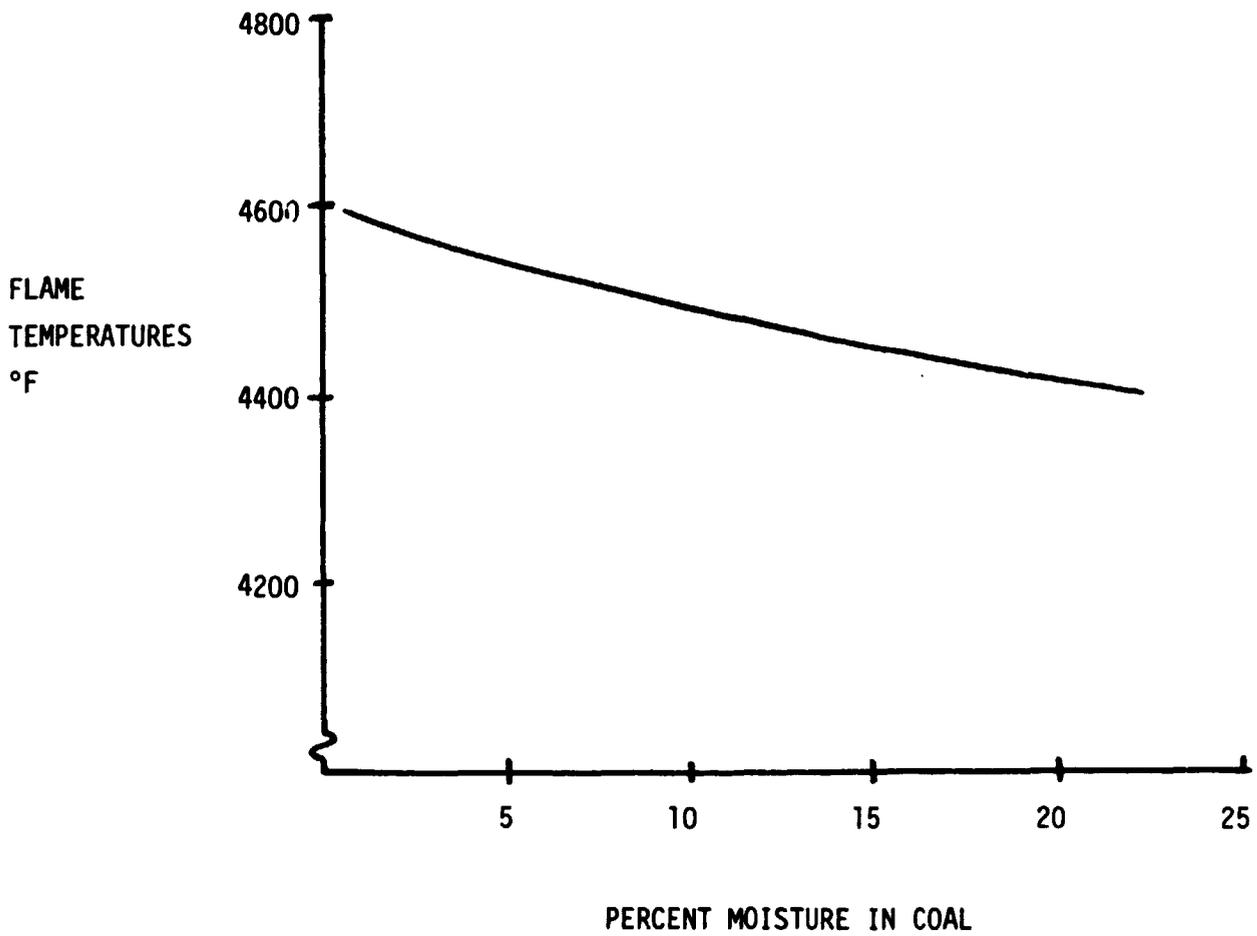
could be dried to 4% total moisture and the fine size range could be dried to 2%, in a single drying stage without using excessive drier temperatures or complicated additional equipment. Based on the tests to determine how the Montana Coal pulverized and dried, the manufacturer determined the recommended size and capacity of the roller mill, separator, air heater, bag house and fan to process the 25 tons/hour of moist coal for CDIF.

The 10 barrels of coal supplied to the Williams Patent Crusher Co. for the processing tests were then reloaded into the barrels and three barrels of 90%-325 mesh pulverized and dried ($\sim$2% moisture) coal was sent to Petrocarb Inc. for feed flow tests. The fine material was used because previous experience in the industry had indicated that fine material was more difficult to feed than coarse material. Also, since one of the CDIF combustor study designers had specified a fineness down to 80%-325 mesh, this particular size was used in the tests.

The Petrocarb tests consisted of two distinct feed flow tests on a special mini-injector. The first series were designed to determine whether a sample of dried Rosebud Coal sized 90% minus 325 mesh could be injected at controlled rates between 15 and 42 pounds per minute when feeding into a receiver pressurized to 100 psig and at atmospheric pressure.

The second series was intended to determine whether calcined, pulverized potassium carbonate could be fed at rates between 5 and 15 pounds per minute when feeding into a receiver pressurized to 100 psig.

The experimental test equipment used included a special "Mini Injector" which has a storage capacity of approximately two cubic feet and which is capable of feeding at rates consistent with a one-half inch Mix-Tee and transport line (or smaller). This unit is normally used by Petrocarb for pre-

liminary evaluations to obtain data on characteristics of materials to be injected. The unit was arranged for feeding through a 25 foot length of 0.5 inch diameter reinforced rubber hose into the receiver maintained at atmospheric pressure.

A special Petrocarb Model 24 Injector that can be equipped to feed from a few pounds to several hundred pounds per minute into a receiving vessel that can be operated at pressures up to 100 psig was used for the high pressure receiver tests.

The test procedure involved manually filling the injectors after screening through a 1/8 inch mesh sieve to remove any tramp material. The receiving vessel was mounted on a platform scale used to measure the weight of material transported from the injector to the receiver. Material was fed for timed intervals designed to demonstrate repeatability and rate control as a function of differential pressure between the two vessels. Dry nitrogen gas was used for pressurizing and transport of the dried coal. The pressure and gas flow rates were manually controlled with regulating valves and monitored by flow indicating rotameters and pressure gages.

The "mix-tee" and transport line each were 1/2 inch diameter. The transport line consisted of 40 feet of horizontal and 21 feet of vertical 1/2 inch Schedule 80 pipe followed by a 180° bend and about 15 feet of 0.88 inch ID hose into the top of a pressurized receiver injector. The outlet vent of the pressurized receiver was equipped with a cotton dust bag.

The test to determine coal flowability and controllability consisted of filling the injector hopper and pressurizing to 15 psig. The coal was observed to flow smoothly and the injector ran completely empty without interruption.

The injector hopper was then refilled and the time for feeding 5 pounds of coal was measured. Each of eight consecutive five-pound batches took 18 seconds which corresponds to 16.6 pounds per minute. This was regarded by Petrocarb as good repeatability.

A series of runs were then made at pressures varying from 5 to 35 psig injector pressure and two different transport gas flow rates. These data are presented in Figure 5. The curves are plotted with pressure as the ordinate and solids rate as the abscissa. The curves illustrate the effect of the volumetric flow rate of transport gas on solids flow rate as well as the pressure in the injector. The preliminary result with the "Mini" Injector fully justified continuing the test with the larger scale equipment to demonstrate the capability of the Petrocarb Injector to feed controllably and uniformly against elevated back pressures.

A second series of tests was made with the Special Model 24 Injector feeding into a receiver which was maintained at atmospheric pressure. A third series of tests were made with the receiver maintained at 100 psig. Three high pressure runs were made using 15.2 SCFM, 12.8 SCFM, and 10.1 SCFM transport gas additions, respectively. The resulting test data are plotted in the set of curves shown in Figure 6. Two of the curves were observed to cross at approximately 40 pounds per minute coal flow rate. This is due to the fact that 10.1 SCFM additional transport gas is slightly inadequate at the higher rates and thus the flow rate dropped off. This indicated that the test was not at the optimum minimum gas flow rate (i.e., the rate which will result in the maximum solids flow for a given injector pressure). This phenomenon will occur even though the solids flow is smooth and reproducible but would normally be avoided in practice by adding slightly more transport gas.

FIGURE 5

TEST WITH PETROCARB MINI INJECTOR

(1/2" MIX-TEE WITH 25' OF 0.5" DIAMETER HOSE - ATMOSPHERIC PRESSURE RECEIVER)
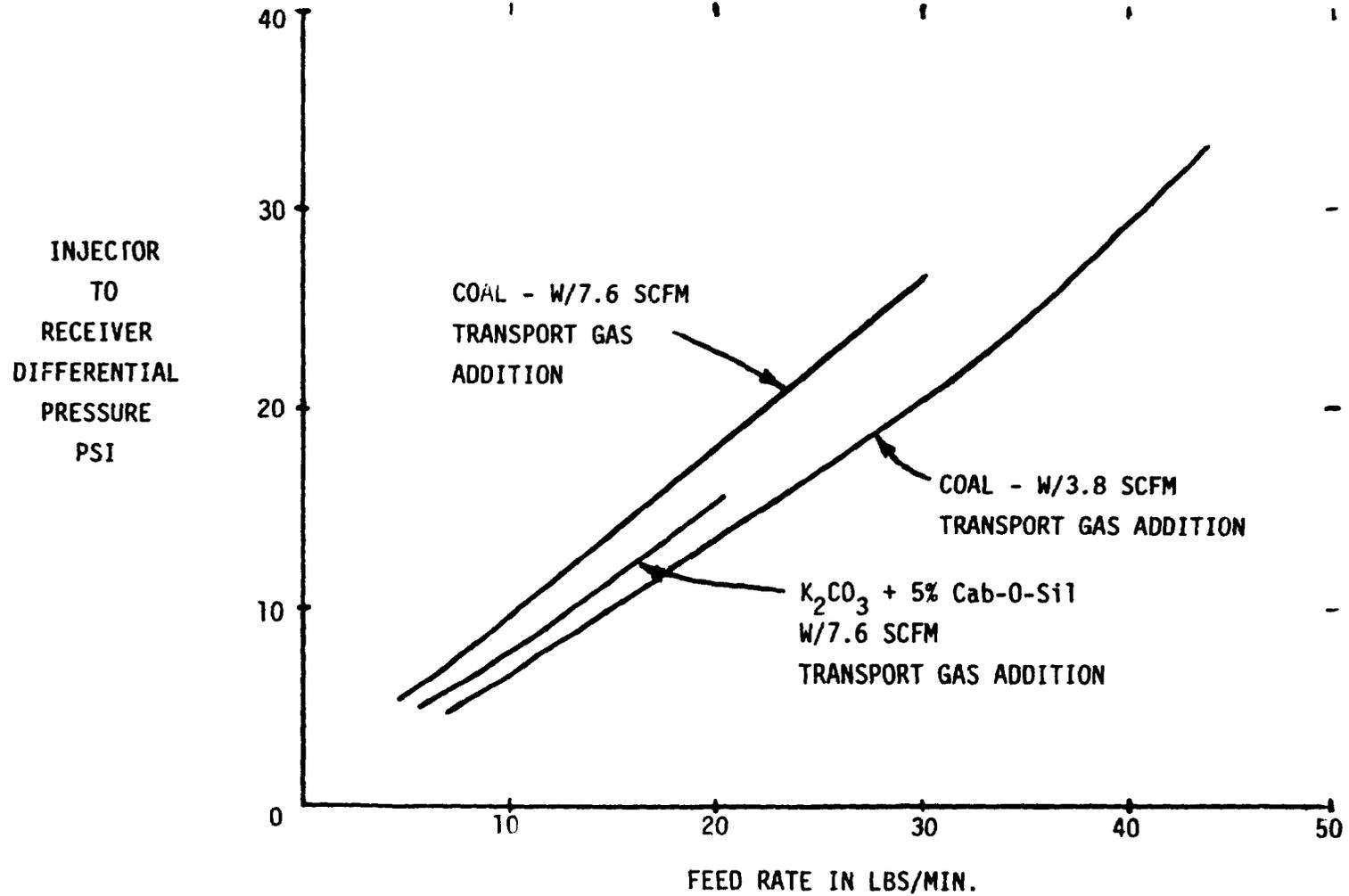
FIGURE 6
TEST DATA FOR PETROCARB
SPECIAL MODEL 24 INJECTOR

ROSEBUD COAL - 90% -325 MESH
1/2" TEE, 40 FT. HORIZONTAL + 21 FT.
VERTICAL - 1/2" SCH. 80 PIPE

INJECTOR
TO
RECEIVER
DIFFERENTIAL
PRESSURE
PSI

5.4 SCFM TRANSPORT
GAS ADDITION W/O PSIG
RECEIVER PRESSURE (ATM)

17.9 SCFM TRANSPORT
GAS ADDITION W/100 PSIG
RECEIVER PRESSURE

10.1 SCFM
TRANSPORT GAS
ADDITION W/100 PSIG
RECEIVER PRESSURE

12.8 SCFM
TRANSPORT GAS
ADDITION W/100 PSIG
RECEIVER PRESSURE
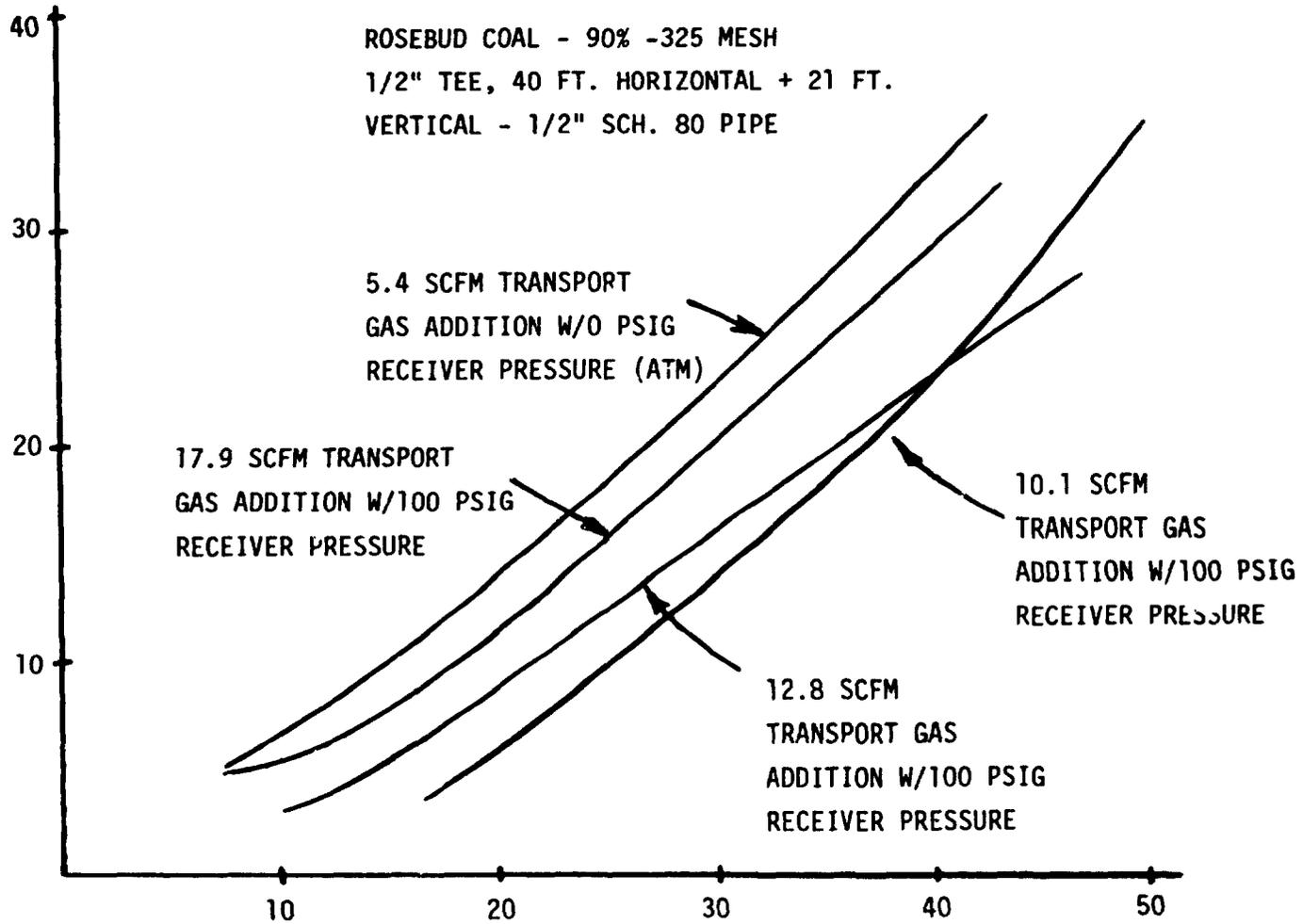
Based on the data obtained in the coal feed tests it was concluded that if the coal sample tested was truly representative, there should be no major problem in designing the system to feed coal uniformly at the flow rates of 15 to 42 pounds per minute for the CDIF in a system utilizing a number of 1/2 inch Schedule 80 pipe feed lines. The low end of the feed rate (15 lb/min) could be reduced further to 10 pounds per minute but the solids/gas ratio would have to be decreased somewhat. The feed mass flow rate ratio of 7 lbs coal/1 lb of air results in some dilution of the net gas inlet temperature, but the resulting decrease of approximately 60°F in the flame temperature was considered acceptable.

Two drums of calcined pulverized potassium carbonate (-325 mesh K_2CO_3 powder)were supplied for the seed feed tests. Even though the containers were sealed and the inner contents protected by plastic bags, the anhydrous solids were noticeably lumpy and agglomerated. The angle of repose was negative and the material very difficult to handle.

An attempt was made to inject this material in the "Mini Injector" but flow was sporadic and non-uniform. The equipment was very difficult to clean up and all valves and fittings required complete dismantling, cleaning, and drying.

Petrocarb recommended that consideration be given to adding a flow promoting reagent such as Cab-O-Sil (a fumed silica) to determine whether the potassium carbonate could be made flowable. However, several other small tests were made to study means of converting the K_2CO_3 to a more flowable form for feeding.

In one test the K_2CO_3 was heated in a small fluidizer to +300°F with the intent that moisture would be driven off and the solids converted into a flowable condition. The capability of feeding hot materials up to a temperature

of about 1000°F were considered not to be unduly difficult if the properties of the solids were improved. However, preliminary tests indicate no improvement at temperatures up to 300°F, so the test was discontinued.

A similar test was made in a fluidizer to study the change in properties caused by additions of Cab-O-Sil. A marked improvement was obtained by the use of the flow agent. However, it was evident that because the carbonate on hand was lumpy, each mass of agglomerate required intimate hand mixing of the two materials. Had the carbonate been freshly pulverized, it is believed that a simple mixing would have sufficed. However, because of the encouraging results of Cab-O-Sil addition, a small test was conducted to prove that the treated carbonate could be injected reliably. About 75 pounds of carbonate was carefully mixed in small batches with 5% by weight of Cab-O-Sil. This work was done by manually rubbing materials together in plastic bags. Obviously, it is not practical to produce large samples for test by this means, but since this was a small batch test conducted in the "Mini Injector", this proved to be a practical expediency and provided a good feed material. Petrocarb indicated they could predict with good reliability extrapolation to large scale equipment based on these small scale tests.

The sample of 5% Cab-O-Sil and carbonate was fed at three different feed rates and the results are included on the plot in Figure 5 with the coal tests. Excellent correlations were developed that were then used as the basis for sizing and selection of a Petrocarb seed injector as a reference design for the CDIF concept. The injection work was thereupon terminated. Further testing will be necessary to establish long-term performance and reliability of this, or other MHD seed feed concepts.

One additional test was made which involved the mixing of incremental quantities of raw carbonate to the sample containing 5% Cab-O-Sil. This indicated that the Cab-O-Sil ratio to K_2CO_3 could be reduced to 2.5% without noticeably changing the flow properties of the mixture. As lesser percentages of Cab-O-Sil addition were made, the flow properties were noticeably poorer.

Conclusion

The coal and seed processing and feeding investigations conducted in connection with the CDIF Conceptual design identified commercially available equipment suitable for use in a reference design and established reference operation conditions for the desired performance of the coal and seed processing and feed systems. This information became a part of the conceptual design description and was provided to the Architect Engineer for his consideration in the preparation of the Title I design. Figure 7 shows the flow schematic of the 50 MW CDIF, coal feed system developed as a result of this preliminary coal feed system design development program for the CDIF.

Since this was a preliminary evaluation, it is recommended that further development be conducted to improve the processing and feed system components design and operation to improve the reliability, uniformity, and quality of performance. Special emphasis should be placed on improving flow measurement instrumentation nd reduction of the conveyance air weight fraction.
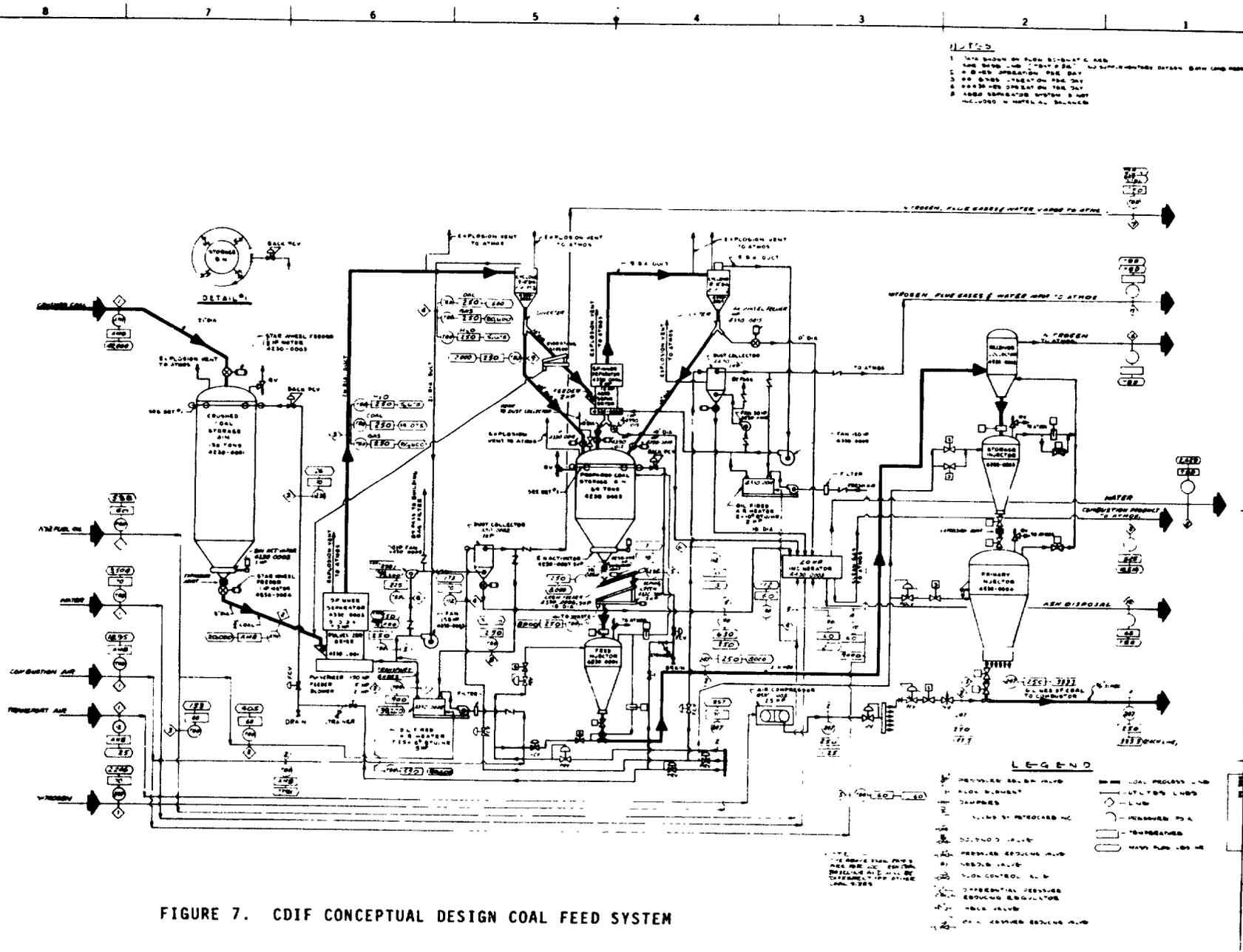
FIGURE 7. CDIF CONCEPTUAL DESIGN COAL FEED SYSTEM

STORAGE AND FEEDING OF COAL

A. W. Jenike

J. W. Carson

Jenike and Johanson, Inc.
No. Billerica, Massachusetts

ABSTRACT

Reliable feeding of coal from storage bins to process
requires the knowledge of the behavior of coal during flow.
Sponsored by AIME Research, in flow of bulk solids was under-
taken in the 1950's by Jenike, and led to the development of
flow ability testing equipment and of the "Mass Flow" concept
of design for reliable flow. The theory has since been
expanded to two-phase, solids-gas system, and has found world
wide application in the design of storage and feeding systems.

STORAGE AND FEEDING OF COAL

by

Andrew W. Jenike and John W. Carson
Jenike and Johanson, Inc.
Consulting Engineers
2 Executive Park Drive
No. Billerica, Massachusetts 01862

HISTORICAL

In 1954 the Coal and Minerals Beneficiation Divisions of AIME

sponsored a study on the "Flow of Bulk Solids" proposed by Dr. Andrew
W. Jenike. The Engineering Foundation provided seed money. The University of Utah, the American Iron and Steel Institute and the National
Science Foundation provided the bulk of the research funds. Under
Jenike s leadership, a Bulk Solids Flow Laboratory was set up at the
University of Utah in 1956; by 1962 a theory of flow of bulk solids
as well as equipment to measure the pertinent solids properties had
been developed and proved in industrial applications to storage bins
and feeders.

The design method has been verified through extensive experimental
work, especially in the United Kingdom [42] and Germany [25], and applied
to hundreds of storage plants and reactor vessels all over the world.

In the past ten years the theory of flow has been extended to two-
phase, solid-gas systems.

FEEDING

Consider coal being fed from a storage bin or hopper to a process.
The feeder is designed to feed at a given rate against a given gas pressure.
But, for the feeder to operate, coal must flow from the hopper into the
feeder uniformly at the given rate. To assure such flow it is necessary
that: (a) the hopper leading to the feeder be sufficiently steep and
smooth, (b) the inlet area into the feeder be sufficiently large, and
(c) the inlet area be fully live and effective. These conditions are
rarely satisfied unless the bin and feeder have been designed on the
basis of the flow theory supported by measurements of the flow properties
of the used coals.

* Numbers in brackets refer to references at end of paper

FLOW PATTERNS

When the hopper, i.e. the converging part of a storage bin, is sufficiently steep and smooth and the hopper outlet larger than critical for the least free-flowing coal in question, coal flows out of the bin by gravity without any stagnant regions in the bin. Such a flow pattern is referred to as "mass-flow." A mass-flow bin is shown, and its properties listed, in Fig. 1.

If the hopper is not sufficie ly steep and smooth, coal flows only in a narrow channel which forms within stagnant coal. This flow pattern is referred to as "funnel flow." The critical outlet dimensions of a funnel flow hopper are always larger than those of a mass-flow hopper, sometimes, several times larger. A funnel-flow bin is shown, and the properties listed, in Fig. 2.

The regions of mass-flow are indicated approximately in Figures 3 and 4 as a function of the hopper slope angle and the friction angle between coal and the wall of a circular cone and a long wedge, respectively.

CRITICAL HOPPER OUTLET

Coal arches across a hopper outlet when the strength of the coal is greater than the stresses which act in the arch. The flow criterion follows from that observation: coal will flow provided its cohesive strength is less than the stresses in a potential obstruction to flow (arch, stable rathole). The strength of a given coal is a function of: (a) the degree of compaction, which is measured by the consolidating pressure applied during compaction, (b) time of application of the consolidating pressure, (c) moisture content, (d) temperature. The function of strength versus consolidating pressure, under given conditions (b), (c) and (d), is referred to as the flow-function FF, Fig. 5. Evidently, the higher the FF - line lies, the more strength a material develops and the less free-flowing it is. The method used to measure the flow-function and the wall friction angle is described in references [2 and 11]. The theory of one-phase, solids only, flow is derived in references [1,3,5, 6,7,8,13,14,27] and summarized in reference [11]. Various aspects of testing materials and proper bin and feeder design are described in references [9,10,12,17,20,22,29,32,33-35].

APPLICATIONS

A typical application of the theory results in a sketch such as shown in Fig. 6. This gives the bin and feeder dimensions, the material and surface finish of the hopper walls [e.g. 304SS-2B finish] and the screw size, rpm and horsepower. Note the design of the screw in which the volumetric flow rate increases from zero to maximum in the direction of coal flow within the length L of the feeder inlet. This is necessary to make the hopper outlet fully effective and permit mass flow.

Other applications are described in the references: loads on bin walls [24,26,37-39]; design of moving bed reactors [30]; briquetting [15,16,28]; flow-corrective inserts [18,19,21]; in-bin blending [31]; bin vibrations [41]; underground mining [4].

TWO-PHASE, SOLID-GAS SYSTEMS

In the past ten years, the theory of single-phase solids flow has been extended to two-phase solids-gas systems. This became necessary in the calculation of flow rates of powders out of hoppers. During the flow of a solid down a vessel, there occur significant bulk density changes, as the solid first compacts under increasing consolidating pressures, then expands toward the hopper outlet. The volume of the pores changes correspondingly giving rise to gas (air) pressure changes within the pores. This leads to significant gas pressure gradients in fine, impermeable materials. These gradients add (or subtract) to the gravity forces which cause flow, and thus significantly affect the flow rate [23,40], as well as the critical outlet dimensions for flow without arching and the rate of settlement of powders [36]. Few analyses and results of applications of the two-phase theory have been published to date, most of the work having been of proprietary nature.

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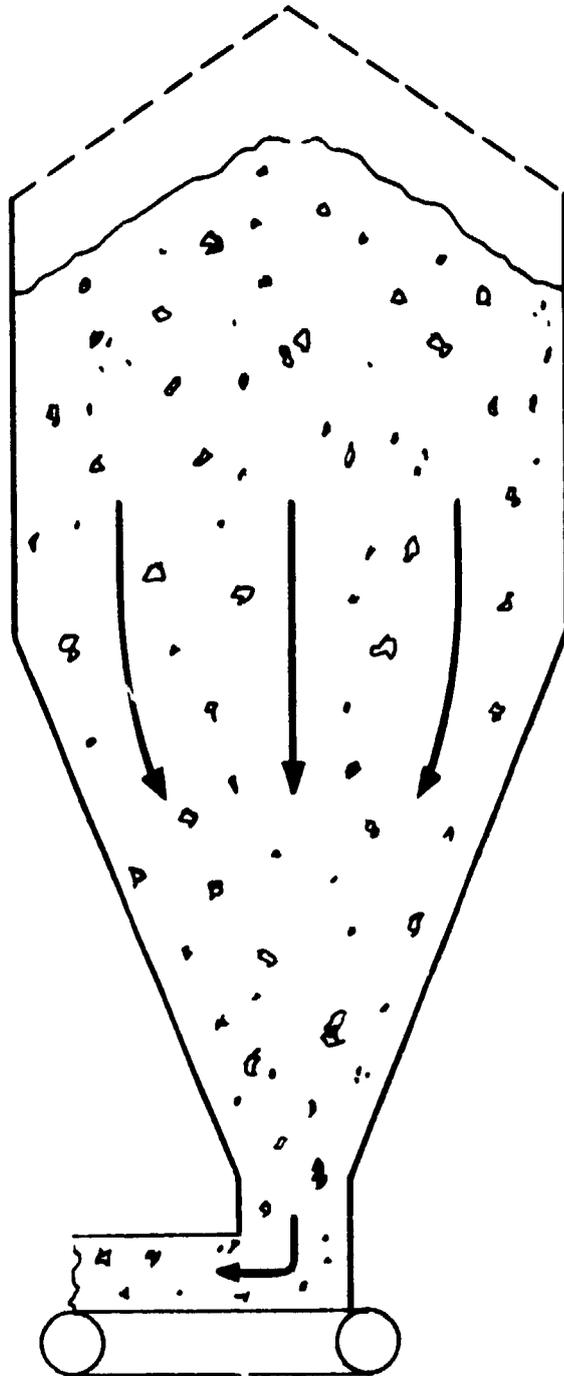
first-in,

first-out,

deaerates,

remixes,

feeds uniformly
at constant density.

Fig. 1   Mass Flow Bin
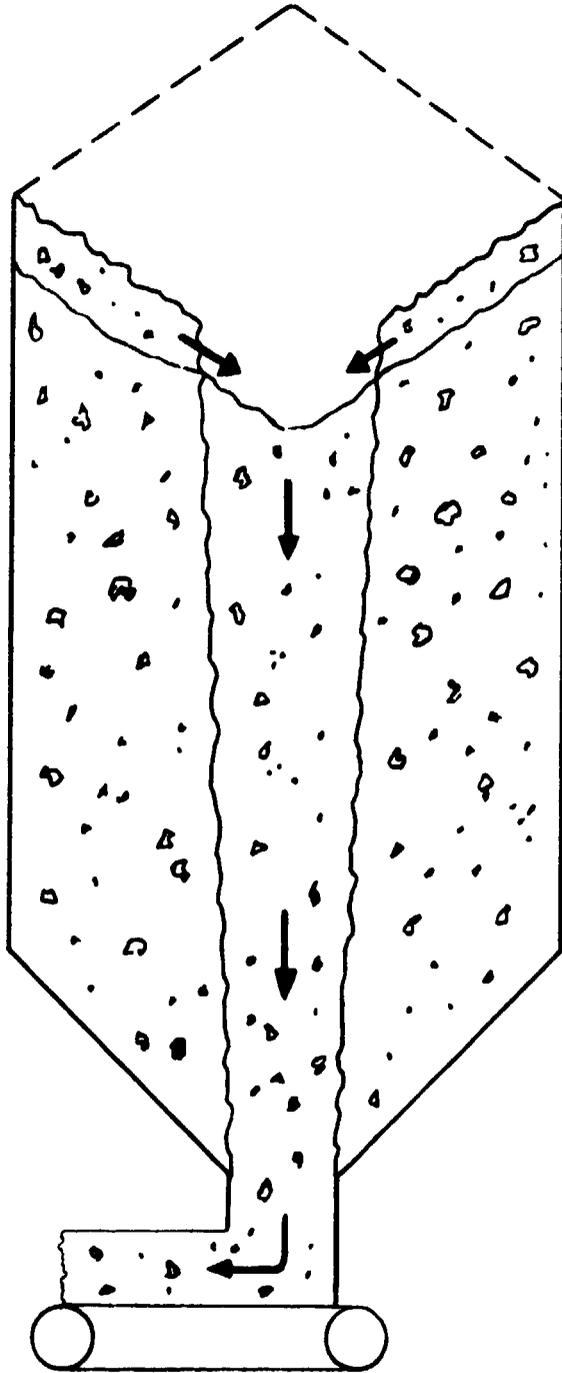
first-in,
last-out,
segregates,
ratholes,
flushes.

Fig. 2   Funnel Flow Bin
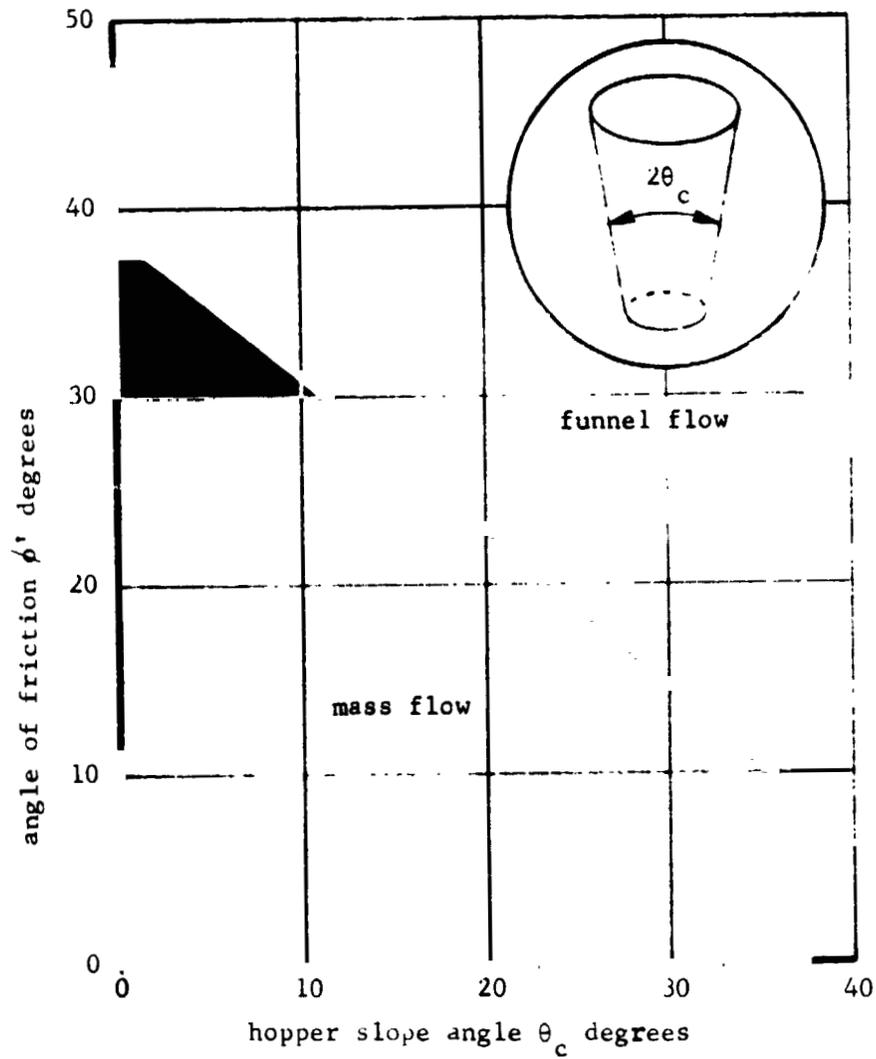
Fig. 3 Ranges of Mass Flow and Funnel Flow in Conical Hoppers
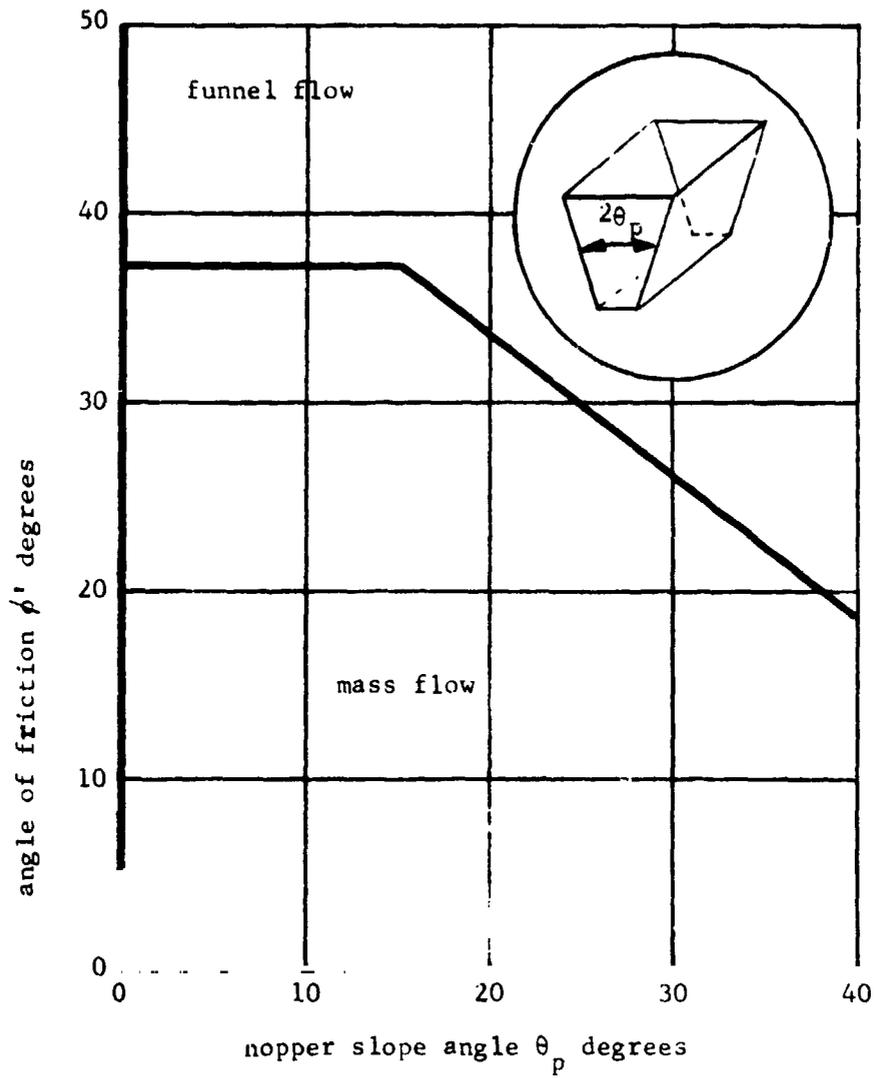
Fig. 4   Ranges of Mass Flow and Funnel Flow in Plane Flow Channels
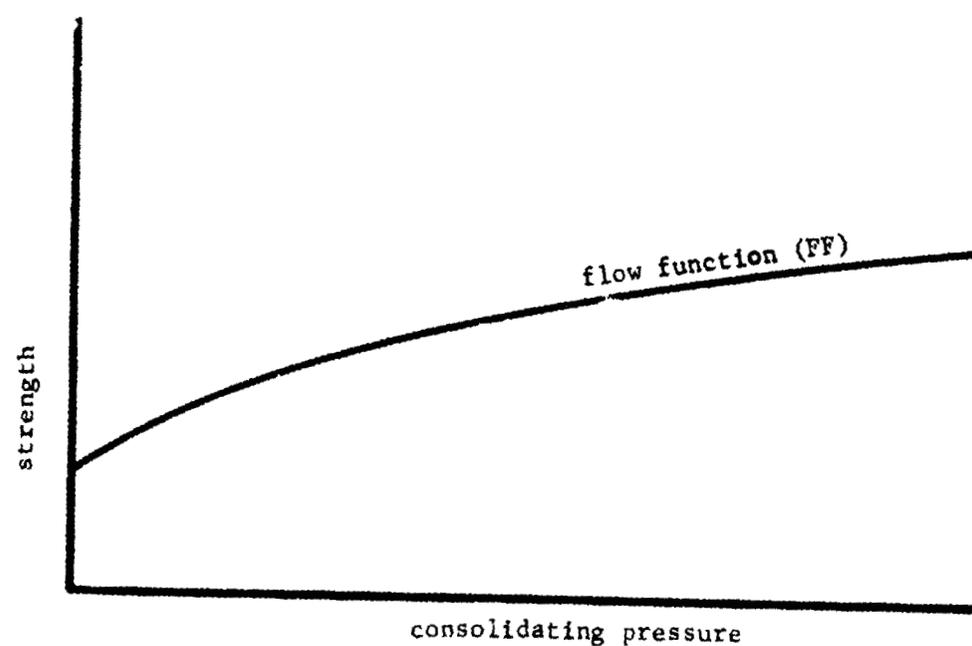
Fig. 5   Relationship Between Strength and Consolidating Pressure
for a Typical Bulk Solid

304SS-2B

θ_c

θ

B

E

screw dia., rpm and HP
specified
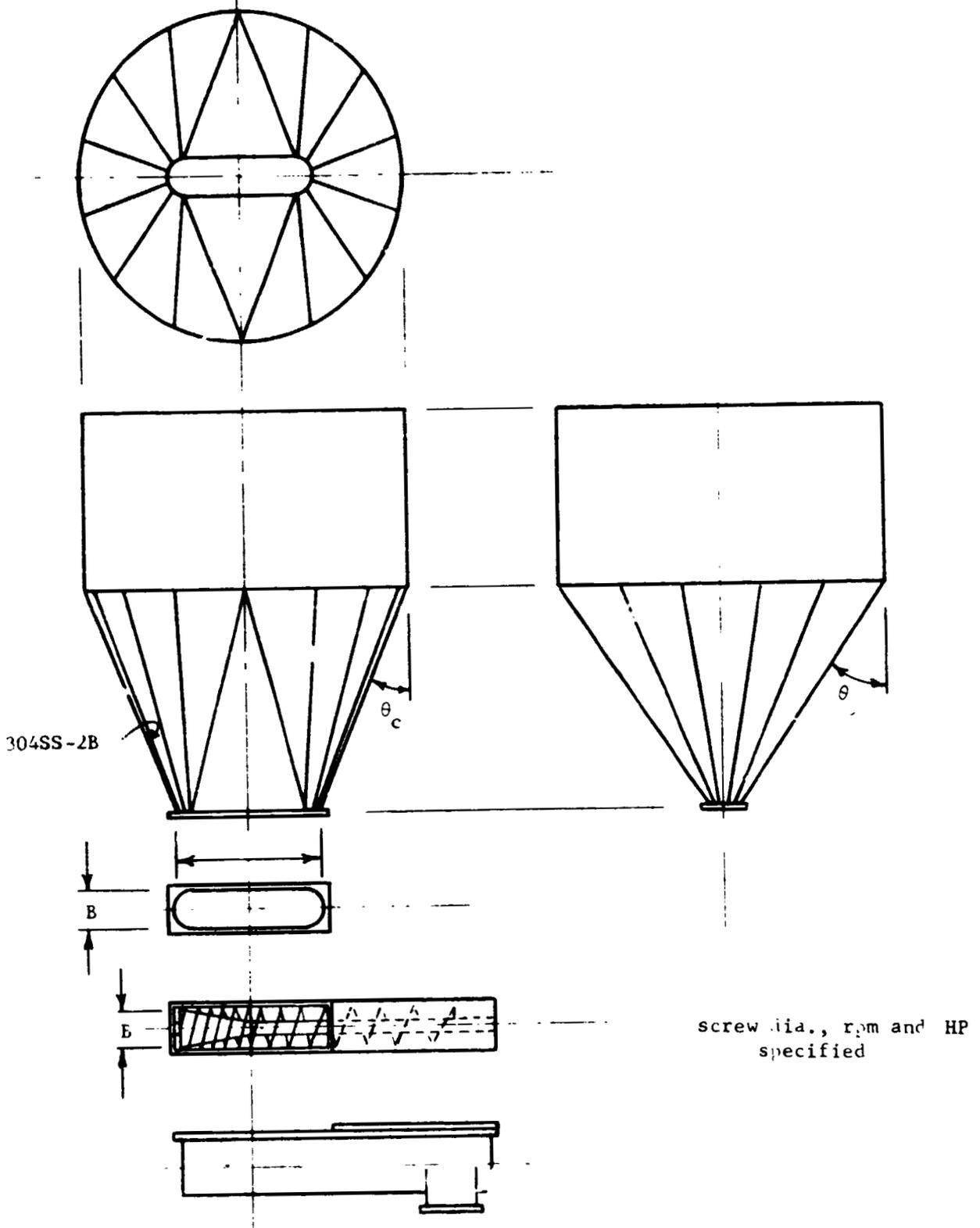
Fig. 6   Typical Mass Flow Bin and Feeder Design

INJECTION OF COAL BY SCREW FEED

R. Fisher

Bechtel, Inc.
San Francisco, California

N78-13264

Injection of Coal by Screw Feed

SUMMARY

The use of the screw feeder for injecting solids through a 20 to 30 psi barrier is common practice in the cement making industry. An analytical extrapolation of that design, accounting for pressure holding characteristics of a column of solids, shows that coal can be fed to zones at several hundred psi with minimal or no loss of gas. A series of curves showing the calculated pressure gradient through a moving column of solids is presented. Mean particle size, solids velocity, and column length are parameters. Further study of this system to evaluate practicality is recommended.

INTRODUCTION

The screw feeder has long been in use as a handy device for moving granular solids through troughs or conduits from one point to another, usually against a distributed resistance in the form of a gradient in elevation or pressure. Desirable features of the system are the continuity and simplicity of operation and the minimal need to depend upon integrity of precisely formed parts. Such features are desirable in meeting the increasingly severe conditions under which solids are handled in the process industry and in the treatment of coal.

A practical example of the above is the use of the screw feeder to pump dry cement solids. The unit loads a system of conduits for the air-driven transport of cement to storage silos. The system is pressurized to perhaps three atmospheres (gauge) at the pump station. The dry solids are pumped into the system by the screw feeder against this pressure head. The rate of solids injection may be as much as 200 tons per hour.

The screw feeder has also been used to inject solids at high pressure, as in the extrusion molding of plastic products. The feeder is of a special configuration designed to develop pressure over a relatively short length of pump. Compression of the solids within a constricted channel and extrusion of this material in a plastic form from this section is the objective. The interplay of frictional forces at the leading edge of the screw and the surface

of the channel to produce forward motion is similar to that in the
cement solids pump.

The foregoing practices suggest that the injection of ground solids into
a pressurized zone should be practical. It is not desired to compact
the solids but to move them through the screw channel in a lightly packed
condition, utilizing the drag of this structure on gas to reduce or over-
come the counterflow of gases.

An analysis of the pressure gradient likely to develop over the length
of the solids pump was performed and is presented below. It is based
simply on the calculation of gas pressure along the length of a uniformly
moving column of solids. Several sets of operating conditions were
studied. The results indicate that solids can be transferred continuously
through a pressure barrier of several hundred psi with a modest velocity
of flow, i.e., from 1 to 5 feet per second.

DISCUSSION

System Configuration

The screw feeder system considered reasonable for coal feeding is shown in Figure 1. It is very similar to the system used to feed ground cement except that the channel containing the screw is much longer, extending to perhaps thirty feet. It is fed from a hopper containing solids to a short section of the pump channel in which the pitch of the screw is considerably larger than in the remaining section, where the pitch is constant. The screw terminates at the high pressure end before a port through which the solids can discharge. The port is covered by a cap, which acts as a check valve against blowout.

The function of the screw section immediately below the feed hopper is to engage the ground solids and maintain a continuous column of lightly packed material flowing into the channel. The average density of the solids/gas mixture in this section is believed to be less than that downstream.

The pressures p_1, p_2, p_3, p_4, p_5 along the channel and p_T, the pressure in the vessel, are progressively higher. Gas movement along the channel can be from or toward the zone of high pressure or can be non-existent as in the results developed below.
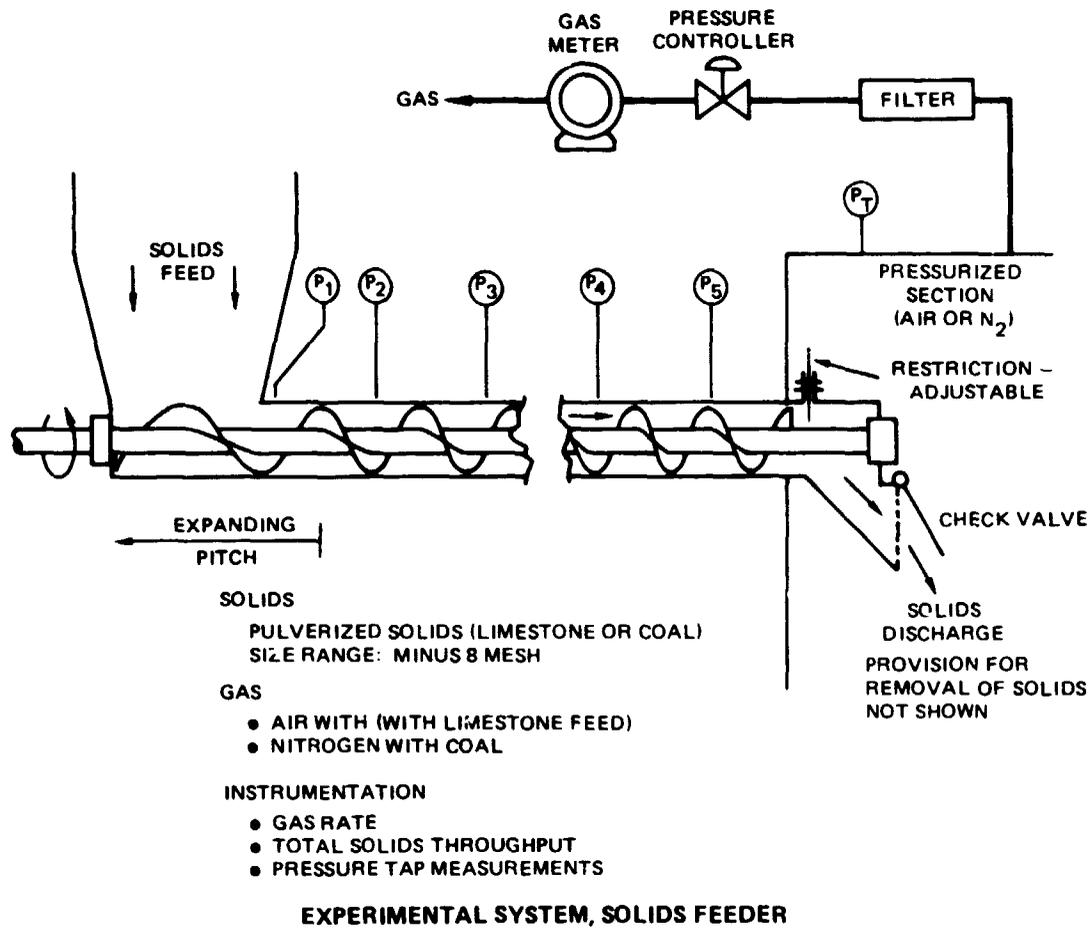
Figure 1. EXPERIMENTAL SYSTEM, SOLIDS FEEDER

Model for Analysis

The physical model conceived for purposes of analysis is a straight circular conduit that is fully occupied by a column of moving solids (Figure 2). The column is homogeneous, representing a plug of solids, lightly packed, moving at a constant velocity. At any station along the pump channel, the conditions are at steady state. The relative movements of gas and solids at the station produce a gas pressure gradient, which is a function not only of the relative velocity but also of the shape and size characteristics of the solids.

For purposes of evaluating the pressure gradient the Ergun equation can be used (1). This relation gives the change in pressure with respect to distance in terms of the character of the solids structure and the velocity of the gas relative to a bed of stationary solids. The relation as applied to a thin transverse section is:

$$
- \frac{dp}{dL} = \left[\frac{150}{144 g_c} \frac{(1-\epsilon)^2}{\epsilon^3} \right] \frac{\mu U_g}{(\phi_s d_p)^2} + \left[\frac{1.75}{144 g_c} \frac{(1-\epsilon)}{\epsilon^3} \right] \frac{\rho_g U_g^2}{\phi_s d_p} \tag{1}
$$

$$
= C_1 (G_g/\rho_g) + C_2 \left[(G_g)^2/\rho_g \right]
$$

where $G = \rho_g U_g = \rho \epsilon u_g$

The assumption is now made that the same relation can be used to derive the pressure gradient at a given station when the bed as a whole moves with a constant velocity u_s.

$$
- \frac{dp}{dL} = C_1 G'_g/\rho_g + C_2 (G'_g)^2/\rho_g \tag{2}
$$
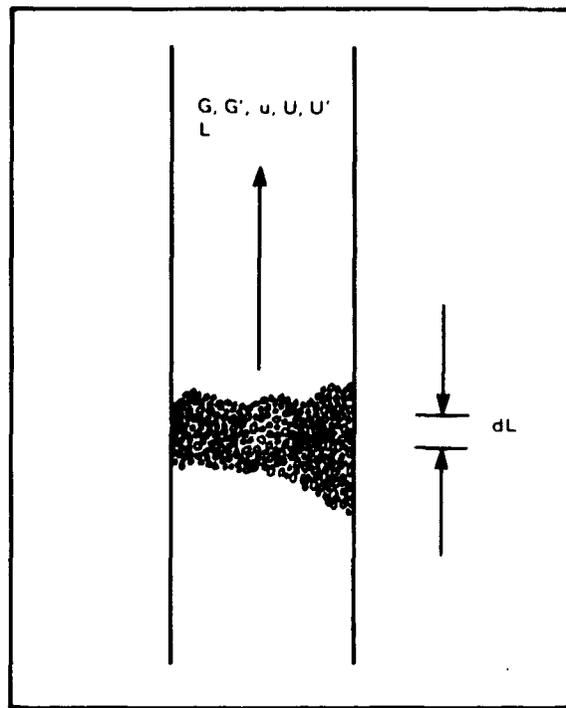
CHANNEL FOR TRANSPORT OF SOLIDS

Figure 2. CHANNEL FOR TRANSPORT OF SOLIDS

The term G' is the mass velocity of gas with respect to the solids, which are in motion with the velocity u_s.

$$G'_g = \rho_g \varepsilon (u_g - u_s) = \rho_g U'_g \qquad \text{See Note} \tag{3}$$

Thus $\quad dp/dL = C_1 U'_g + C_2 (U'_g)^2 \rho_g \tag{4}$

and $\quad U' = \varepsilon(u_g - u_s) \tag{5}$

U' can be regarded as superficial velocity of gas referenced to the solids. If the gas velocity u_g becomes zero, then:

$$U'_g = -\varepsilon u_s, \text{ a constant.} \tag{6}$$

For this special case the gradient

$$- dp/dL = C_4 + C_5 p \tag{7}$$

Integration leads to

$$L_2 - L_1 = (1/C_5) \ln [(p_1 + C_4/C_5)/(p_2 + C_4/C_5)], \tag{8}$$

the length of column necessary to sustain a pressure difference $p_2 - p_1$.

Note: $\quad G'_g = G_g - (\rho_g/\rho_s) [\varepsilon/(1-\varepsilon)] G_s$

RESULTS

Ranges of conditions for several variables were covered in calculating
the performance of the model, the object being to find the length of
column needed to establish a given pressure difference when pumping
solids at practical velocity:

Particle diameter (mass mean): 0.625 to 2 mm or 250 to 9 mesh
                                (Tyler screen)

Solids velocity: 1 to 5 fps

Void fraction: 0.35 to 0.60

Entrance pressure: 15 psia

Discharge pressure: 300 psia

Volumetric pumping efficiency:* 35%

Gas molecular weight: 25

The resulting performances are plotted as several families of curves
in Figure 3, giving the pressure gradients throughout the pumping
channel for the several conditions of mean particle size and voids
fractions. Representative of a practical set of conditions for which
the model performance may be of interest are the following:

Void fraction   0.45

Solids velocity   2 fps

Particle diameter,
mass mean   115 mesh (opening)

---

*Ratio of actual solids velocity to product of screw pitch and rotation
 rate.

PERFORMANCE – PRESSURE VS. LENGTH, VOID FRACTION 0.60

Figure 3a PRESSURE VERSUS LENGTH OF CONDUIT
1 fps, PARTICLE VELOCITY

PARTICLE
DIAMETER: $\bar{d}_p$ = 0.0625 mm (0.000205 ft,
TYLER MESH: 250)

VOID
FRACTION: ϵ = 0.60

Figure 3b PRESSURE VERSUS LENGTH OF CONDUIT

$\bar{d}_p$ = 0.125 mm (0.00041 ft,
TYLER MESH: 115)

ϵ = 0.60

Figure 3c PRESSURE VERSUS LENGTH OF CONDUIT
5 fps

$\bar{d}_p$ = 0.5 mm (0.00164 ft,
TYLER MESH: 32)

ϵ = 0.60

Figure 3d PRESSURE VERSUS LENGTH OF CONDUIT

$\bar{d}_p$ = 2 mm (0.00656 ft
TYLER MESH. 9)

ϵ = 0.60

(50 psia 158')

5 fps

FIGURE 3 A, B, C, D

PERFORMANCE – PRESSURE VS. LENGTH, VOID FRACTION 0.45

Figure 3e PRESSURE VERSUS LENGTH OF CONDUIT

2 1 fps, PARTICLE VELOCITY

PARTICLE
DIAMETER $\bar{d}_p$ = 0.0625 mm (0.000205 ft,
TYLER MESH. 250)
VOID
FRACTION: ϵ = 0.45

PRESSURE psia

CONDUIT LENGTH (ft)

Figure 3f PRESSURE VERSUS LENGTH OF CONDUIT

5 3 2   1 fps

$\bar{d}_p$ = 0.125 mm (0.00041 ft,
TYLER MESH 115)

ϵ = 0.45

PRESSURE psia

CONDUIT LENGTH (ft)

Figure 3g PRESSURE VERSUS LENGTH OF CONDUIT

5 fps   3

2

1

$\bar{d}_p$ = 0.5 mm (0.00164 ft,
TYLER MESH. 32)

ϵ = 0.45

PRESSURE psia

CONDUIT LENGTH (ft)

Figure 3h PRESSURE VERSUS LENGTH OF CONDUIT

$\bar{d}_p$ = 2 mm (0.00656 ft,
TYLER MESH. 9)

ϵ = 0.45

5 fps

PRESSURE psia

CONDUIT LENGTH (ft)

FIGURE 3 E, F, G, H

597

PERFORMANCE – PRESSURE VS. LENGTH, VOID FRACTION 0.35

Figure 3i PRESSURE VERSUS LENGTH OF CONDUIT

5 3 2   1 fps, PARTICLE VELOCITY

PARTICLE
DIAMETER: $\overline{d_p}$ = 0.125 mm (0.00041 ft,
                      TYLER MESH: 115)
VOID
FRACTION: ϵ = 0.35

CONDUIT LENGTH (ft)

ORIGINAL PAGE IS
OF POOR QUALITY

Figure 3j PRESSURE VERSUS LENGTH OF CONDUIT

5 fps   3 fps   2 fps

1 fps

$\overline{d_p}$ = 0.5 mm (0.00164 ft,
              TYLER MESH: 32)

ϵ = 0.35

CONDUIT LENGTH (ft)

Figure 3k PRESSURE VERSUS LENGTH OF CONDUIT

5 fps

3 fps

$\overline{d_p}$ = 2 mm (0.00656 ft,
              TYLER MESH: 9)

ϵ = 0.35

CONDUIT LENGTH (ft)

FIGURE 3 I, J, K

The length of column required to pump solids from 15 psia to 300 psia in this system is 16 feet.

The results are further summarized in Figure 4 showing the length of conduit necessary to overcome 300 psia pressure (feed at 15 psia) for three feeds of different particle size (mass mean diam.) and for solids velocity of 2 fps. The performance can be readily ascertained for the feeds used at the R&D agencies.

Mean Particle Size	Voids Fraction	Feeder Column Length Required	Typical Process
.21 mm, 65 mesh	0.45	40 ft	(Acceptor Process –
	0.35	19	Consolidated Coal)
.17 mm, 80 mesh	0.45	27	(IGT "HYGAS")
	0.35	13	
0.075 mm, 200 mesh	0.45	6	(BCR "BI-GAS," USBM
	0.35	2.7	"SYNTHANE," Koppers-Totzek)

The voids fractions are felt to lie between the two values cited. The larger represents solids of uniform size; the smaller, a representative case for a distribution of sizes. The latter is more likely.

To obtain the feeder column length needed to reach 1,000 psia, the values shown above may be tripled. The use of two or more screw feeders in sequence should be feasible.
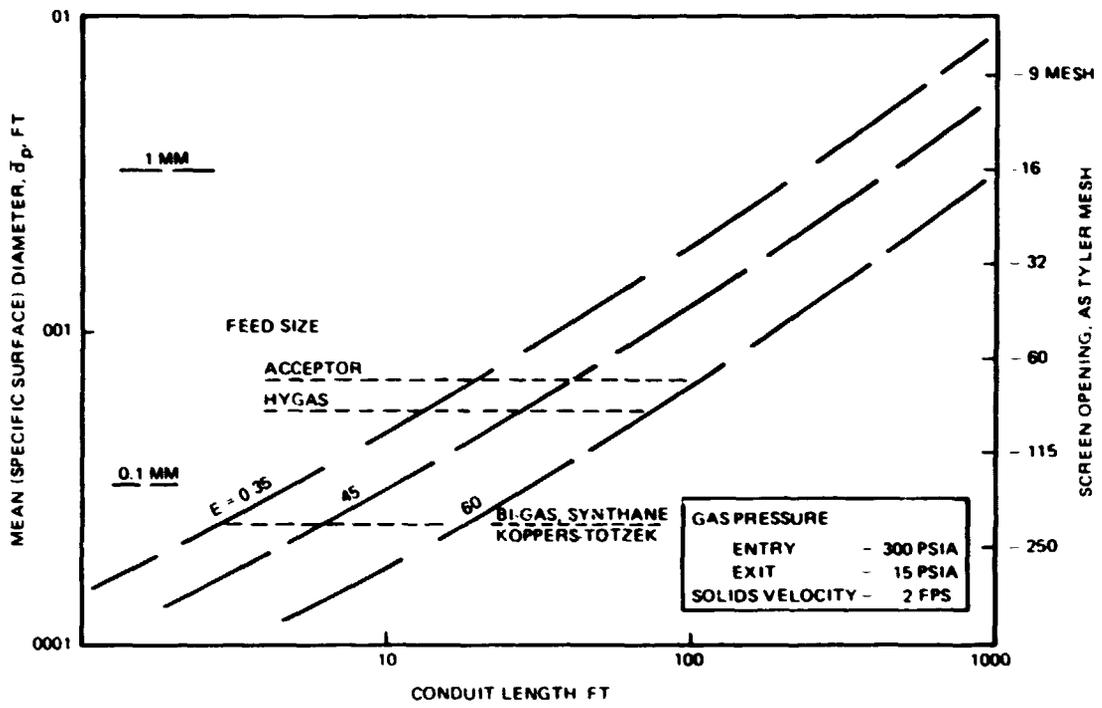
Figure 4. EFFECT OF PARTICLE SIZE ON CONDUIT LENGTH

COMMENT

The application of the screw feeder for charging coal to pressurized gasifiers is regarded as worthy of further study based on the considerations noted below:

1. The major development programs in coal conversion use pulverized feed which is fine enough to be adaptable to screw feeder operation. Solids should be capable of being fed to pressures as high as 1,000 psia, dry. The porosity is that characterizing normally loose-packed solids.

2. Localized malfunction of the column of solids is essentially eliminated by the positive action of the screw along the entire length of the system.

3. Feed is continuous and can operate at zero backflow of gases.

4. The feeder operates without maintenance of close fitting surfaces for sealing.

5. The feeder is an extension of existing technology. Commercial feeders and extruders are presently operating at the pressure gradients (psi/ft) in the range of those applying to the proposed systems. Flow stability has been studied experimentally.

6. The cost of a screw pump is evidently not an outstanding consideration.

NOMENCLATURE

$$C_1 = \frac{150}{144\ g_c} \cdot \frac{(1-\epsilon)^2}{\epsilon^3} \cdot \frac{\mu}{(\phi_s \sigma_p)^2} \ ,$$ a constant representative of

the particulate character of a given stream of solids

$$C_2 = \frac{1.75}{144\ g_c} \cdot \frac{(1-\epsilon)}{\epsilon^3} \cdot \frac{1}{\phi_s \sigma_p} \ ,$$ a constant similar in character

to C_1

$C_3 \ = \ $ constant of integration, corresponding to $L_1 = 0$

$C_4 \ = \ C_1 U_g'$ for the case $u_g = 0$; i.e., $-C_1 \epsilon u_s$

$C_5 \ = \ C_2 (U_g')^2$ M/RT for the case $u_g = 0$; i.e., $C_2 (\epsilon u_s)^2$ M/RT

$\sigma_p \ = \ $ Mean specific surface diameter, ft

$g_c \ = \ $ Conversion constant, 32.2 lb(m) ft/lb(f)-sec^2

$G \ = \ $ Mass velocity lb(m)/ft^2 sec

$L \ = \ $ Distance along channel, ft

$M \ = \ $ Molecular weight, lb(m)/mole

$p \ = \ $ Pressure, absolute, psia

$R \ = \ $ Gas constant, 10.71 psia/mole $-$ $^{\circ}$R

$T \ = \ $ Temperature, $^{\circ}$R

u   = Velocity, referenced to a station at fixed value of L, fps

U   = Superficial velocity, based on cross section of column, fps

ϵ = Voids fraction, dimensionless

ϕ_s = Sphericity, dimensionless

μ = Viscosity, lb(m)/ft-sec)

ρ = Density, lb(m)/ft^3

Superscript:

'   = Prime. Denotes velocity referenced to a station moving with
       the velocity of the solids

Subscripts:

1,2 = Denote stations at the ends of the column

g   = Denotes gas

s   = Denotes solids

<div align="center">REFERENCE</div>

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MATERIAL HANDLING SYSTEMS FOR THE
FLUIDIZED-BED COMBUSTION BOILER
AT RIVESVILLE, W. V.

J. G. Branam
W. W. Rosborough
The MITRE Corporation
METREK Division
McLean, Virginia

ABSTRACT

The 300,000 lbs/hr steam capacity multicell fluidized-bed boiler (MFB) utilizes complex material handling systems. The material handling systems can be divided into the following areas:

- Coal Preparation; Transfer and Delivery

- Limestone Handling System

- Fly-Ash Removal

- Bed Material Handling System

Each of the above systems will be described in detail and some of the potential problem areas will be discussed. A major potential problem that exists is the coal drying system. The coal dryer is designed to use $600^{O}F$ preheated combustion air as drying medium and the dryer effluent is designed to enter a hot electrostatic precipitator ($730^{O}F$) after passage through a cyclone.

Other problem areas to be discussed include the steam generator coal and limestone feed system which may have operating difficulties with wet coal and/or coal fines.

INTRODUCTION

The Monongahela Power Company's Power Station in Rivesville,
West Virginia, provided the facility for the installation and operation
of a fluidized-bed combustion (FBC) steam generator. The space for
the FBC steam generator was available in the Rivesville Power Station's
existing boiler room where four (4) older boilers had been removed.
This plant not only had the space, but also had available concrete and
steel plate bunkers for storage of coal and limestone. The existing coal
handling facilities were also utilized by installing an interconnecting
conveyor to deliver coal into the bunkers. The plant's facilities for
boiler feed water makeup, treatment and storage were made available
for use. The existing fly ash storage silo and ash disposal site were
available and are used for this project.

The installation of connecting headers and valving allows 300,000
lbs/hr of 925°F steam @1250 psig from the fluidized-bed combustion
steam generator to be delivered into an existing 1250 psig, 925°F steam
header. The steam will be delivered to an operating turbine generator
whose full load requirement is 450,000 lbs/hr of steam.

The agreement between Monongahela Power Company and Pope,
Evans and Robbins was signed in June 1973 for the installation and the
later operation of the fluidized-bed steam generator.

RIVESVILLE PLANT EXISTING EQUIPMENT

Existing steam and power generating equipment in the Monongahela
Power Company's Rivesville Plant:

Existing Power Generating Units

Unit No.	Capacity MW Rated	Capacity MW Actual	Year Installed	Type
5	35	50	1943	Condensing
6	65	94	1957	Condensing

Existing Steam Generating Units

Unit No.	Capacity Lbs/hr.	Year Installed	Operating Conditions
7	450,000	1943	1250 psig/925°F
8	900,000	1957	1275 psig/950°F

As of this date, the FBC steam generator has not feed steam into the plant header. Operations thus far have been conducted by venting steam to atmosphere.

COAL PREPARATION, TRANSFER AND DELIVERY

The fluidized-bed combustion steam generator coal preparation and storage system was designed to process 50 tons per hour. Dry coal is purchased at the present time due to potential problems with the coal drying system.

The dry coal is conveyed by Monongahela Power Company's coal handling equipment and deposited in one of the existing steel plate coal bunkers which were renovated for the FBC coal storage system. From the steel plate bunkers, the coal is carried by new Redler conveyors to a surge hopper. From the surge hopper coal can be routed directly to the dry coal bunker if drying, crushing and classifying are not required, or through the dryer, crusher and classifier if required. The dryer can be by-passed to permit the use of the crusher and classifier only if necessary.

The coal dryer is a parallel-flow rotary type which was designed to remove the mositure with 600°F air. The 60,000 lbs/hr drying air supply is provided from the preheated combustion air system.

The coal would normally be discharged from the dryer into a crusher where it would be reduced to 1/4 inch top size. Because the coal is purchased dry and double screened, the existing coal drying and sizing system is by-passed and the coal dryer is not used.

A schematic flow diagram describing this system is presented in Figure No. 1 titled "Coal and Limestone Schematic Flow Diagram."

Coal feeds from the dry coal bunker to the vibrating feeders then to weigh belt feeders. It is then conveyed by bucket elevators to the three coal storage bins. The three bins gravity feed coal at a rate regulated by rotary feeders located directly under the bins. The rotary valves feed into pipes which are designed to discharge 6,000 lbs/hr from the north and south bins serving Cells A, B and C and 3,600 lbs/hr from the east bin which is the Carbon Burnup Cell bin. The discharge rate will vary depending uupon the turbine generator load.

The coal is then mixed in the fuel feed pipe with limestone and is fed to the vibrating table feeders by means of rotary valves. The vibrating table divides the coal and limestone into eight channels which feed the mixture into the 1 1/2" needles (fuel injection pipes). Cells A, B and C are equipped with vibrating feeders located on the north and south side. (See Figure 2). The Carbon Burnup Cell is equipped with

one vibrating table. Air from the auxiliary forced draft fan is introduced at the vibrating feeder to inject the coal into the bottom of the fluidized bed.

Flyash from Cells A, B and C is reinjected into the Carbon Burnup Cell to prevent excessive unburned carbon losses.

LIMESTONE PREPARATION AND STORAGE

Limestone from a nearby quarry is used to remove sulfur during the combustion process. It is delivered to the plant by truck and is unloaded pneumatically. The limestone is prescreened prior to delivery to insure its minus 1/8 inch particle size.

The blower system on the delivery truck pneumatically transfers the limestone through a pipe to a cyclone separator located above the limestone storage bunker. The rate of transfer is 6 to 7 tons per hours. The limestone, after separation from its transport air, is discharged by gravity into the limestone bunker.

The limestone storage bunker is part of an existing concrete coal bunker. It has a storage capacity of approximately 450 tons. The stored limestone is supplied by gravity to the vibrating feeders beneath the bunker as required by load demand. The limestone is fed from the storage bunker by the vibrating feeder to a weigh belt conveyor which transports it by bucket elevators and Redler conveyors to three bins. (See Figure 3).

The three bins are located to the north, south and to the east of the steam generator. The limestone from the north and south bins has three discharge connections with rotary feed valves located on the bottom. The east bin has only one rotary feed valve.

Limestone is mixed with coal after it leaves the rotary feed valves which control the feed rate to be mixed with coal in the coal-limestone feed pipe.

The mixture of coal and limestone is then fed to vibrating feeders. The mixture is divided into eight streams per feeder and is then fed into the 1 1/2" stainless steel injection needles which deliver the coal and stone to the fluidized bed.

FLUE GAS CLEANING AND ASH REINJECTION

The high carbon fly ash from Cells A, B and C is removed from the mechanical cyclones and/or the electrostatic precipitator and is injected into the Carbon Burnup Cell. The flue gas cleaning system was designed to reduce the exhaust gases particulate concentration to 15 pounds per hour at rated load. The exhaust gas cleaning system has

three mechanical dust collectors and an electrostatic precipitato..

Flue gas from the Carbon Burnup Cell is directed to the No. 1 fly ash collector. The 40,000 lbs/hr of flue gas contains 13,120 lbs/hr of solids of which the mechanical collector is designed to remove 12,460 lbs/hr of solids. The remaining 660 lbs/hr is carried over into the flue gas system and fly ash collector No. 2.

The flue gas from the steam generator mechanical collectors with small amounts of particulate matter from dust collectors serving other systems are combined and enter the electrostatic precipitator at a rate of 406,000 lbs/hr. The gases contain 1,440 lbs/hr of solids of which the precipitator collects 1,425 lbs/hr. The difference is 15 lbs/hr of entrained solids in the flue gas.

FLY ASH REMOVAL

The fly ash collector and storage system was designed to receive ash removed from the steam generator exhaust gases.

The fly ash from the No. 2 cyclone and from four (4) of the six (6) electrostatic precipitator hoppers is transferred by a pneumatic system to the plant ash silo. The existing silo is located outside of the plant and receives the fly ash at a rate of 13,885 lbs/hr. This includes the 1,300 lbs/hr of fly ash entering the system from the electrostatic precipitator. The electrostatic precipitator uses a disposal system similar to that of the fly ash No. 2 collector.

Air for ash transport is supplied from two (2) blowers to the pneumatic system at the rate of 530 SCFM at 14 psig.

The pneumatic system transfers fly ash at a rate of 12,460 lbs/hr from the No. 2 collector serving the carbon burnup cell. An air lock assembly discharges the ash to the pneumatic transport line which carries it to the plant ash silo.

The fly ash collected from the No. 1 collector serving cells A, B and C is transferred from the hoppers of this equipment by gravity fed rotary valves or by lock hoppers. The fly ash to be reinjected is pneumatically conveyed by $600^\circ F$ air from the forced draft combustion air to the Carbon Burnup Cell where it is injected at a rate of 15,560 lbs/hr of high carbon fly ash. This feed system by-passes the CBC vibrating feeder. The fly ash to be transferred to the plant ash silo is fed to the transport line by the lock hoppers and carried to the silo by air from the blowers (See Figure 4).

BED MATERIAL HANDLING SYSTEM

The fluidized-bed boiler was designed so that 80,000 lbs/hr of bed material could be removed and separated into undesirable particles or desirable material on the basis of size. The undesirable bed material is discharged into the existing ash silo while the desirable material is lifted by steam jet ejectors and air from the blowers to the bed material storage tank from which it can be reinjected into the fluidized beds. Four (4) rotary feeder valves with an air slide located under the steam generator are used to discharge the bed material from the different cells and transport it to the bed material classifier.

The hot bed material is then separated by the classifier. Material 1/8 inch or less passes through and is discharged by a rotary feeder from the classifier into a vacuum line which transfers the material to the bed material storage bin. The bed material in the storage bin can be returned to the boiler cells as required. The rate of material return is controlled by rotary feed valves and is assisted by air from the forced draft duct.

Material larger than 1/8 of an inch is collected at the base of the screen and discharged by a rotary feeder into a pneumatic system which transfers the material to the ash cooler. After the bed material is cooled, it is transferred by a pneumatic system that discharges the material to an existing ash silo located outside the plant. (See Figure 5). The classifier can be by-passed and all bed material routed through the ash cooler to the silo.

OPERATING EXPERIENCE

Operating History

The initial coal fire was achieved December 7, 1976, in the carbon burnup cell. Since that time, the carbon burnup cell has operated on coal for approximately 343 hours and Cells D (carbon burnup cell) and C have operated in parallel for approximately 23 hours. The first parallel operation of D and C cells occurred on April 5, 1977. Parallel operation of cells D, C and B was attained on April 19 and three (3) cell operation has been repeated three (3) times since then. Four (4) cell operation has not yet been achieved. (June 1977).

The boiler reaches normal operating pressure (1250 psig) with Cells D and C in service, but not normal steam temperature since the primary and secondary superheaters are located on Cells A and B respectively. With Cells D, C and B operating at normal pressure with a shallow bed and near normal temperature, a steam flow of about one-half load (150,000 lbs/hr) can be reached.

The D or carbon burnup cell is the only cell equipped with oil burners. Start-up of adjacent cells is accomplished by opening a slide gate between cells, (Figure 6) then allowing hot bed material to flow into the next cell and igniting any residual carbon in the bed along with the coal being fed. This system has worked relatively well, although there have been some failures to achieve ignition when the slide gate was opened. Transfer of hot bed material has occurred rapidly (well under one minute) in every case except one, when a clinker blocked the slide gate opening.

Coal System Design

Both the original coal bunker and the new coal bins exhibit funnel flow because of their design and because in some cases fine moist coal has been used. As a result, severe problems of arching and ratholing above each outlet have occurred. Consequently, the live storage capacity of each unit is greatly reduced.

In addition to the above problems, flow problems occur in the bins because of their smaller outlets (4" for the CBC bins and 6" for the A, B and C cell bins vs. 2' in the bunker). The cause of this problem is the four (4) small ledges of each outlet where the square outlet interfaces with the round discharge pipe. Since the storage bins receive coal at fewer points as compared to the bunkers, fines accumulate below the bin feed point resulting in undesirable separation of fines and coarser material. Some operational problems occurring in the fluidized-bed boiler may be attributed to the material coming out of the central outlet having more fines than that coming out of the two end outlets. (Figure 7).

The coal dryer has not been operated as the mixture of $500^{\circ}F$ drying air and coal dust is considered somewhat hazardous with the dryer located inside the plant. A further concern is that the coal fines-air mixture entering the electrostatic precipitator from the coal dryer might be ignited by a spark.

Some problems have been experienced with air leakage through the rotary valves above the vibrating feeders. These problems have not been as severe since vents were installed below the rotary valves at the upper end of the feed pipe; however, the vented air was not provided for in the auxiliary fan design. Plugged rotary valves may be caused by feeding material when flow has stopped in the vibrating feeder, and forcing material into the rotary feeders with a rod when attempting to restore coal flow as arching and ratholing occurs. Some material build-up on the rough carbon steel vanes of the rotary feeders has occurred.

Difficulty is experienced in obtaining equal distribution to the feed needles. The variation in flow rates through the several outlets of the vibrating feeders (Figure 8) is on the order of $\pm$ 30%. This appears to be inherent in the design.

Pluggage of the 1 1/2" stainless steel needles which connect the vibrating feeders to the fluid bed has been a frequent cause of shutdown. In some cases, this was caused by failure to establish or maintain sufficient air flow to the needles to prevent them from becoming overheated. When coal flow is initiated into an overheated needle, a coke plug can form and the needle may be rendered inoperative. Other potential causes of needle stoppage are failure to maintain adequate air pressure on the vibrating feeder as the bed height is increased.

The automatic sequencing of the coal and stone vibrating feeders under the bunker, the weigh belts, the bucket elevators and the Redler conveyors (Figure 9) has not yet worked as designed. The signals from the "Bindicators" in the surge bins often provide false information as the material assumes various configurations in the bins. It has been necessary to station personnel at the bins to initiate fill up and shutdown operations.

Oversize and foreign material has caused coal feed stoppages.

Bed Material System

This system (Figure 10) has operated successfully for short periods of time (total operating hours are estimated to be 100). Some difficulty has been encountered in attempting to feed from the bed material storage tank to the boiler cells despite the fact that an aerating ring is provided.

The vibrating air slide which handles bed material being removed, shook loose from its lead anchored bolts and had to be grouted and through bolted.

Some of the specially designed rotary feeders required resetting of clearances to prevent jamming, when handling hot bed material.

Fly Ash Collection, Disposal and Reinjection System

This system has performed relatively well although no reinjection has been attempted yet.

Some initial plugging of ash hoppers was caused by the operation speed of the lock hopper valves (too slow).

A potential problem exists which may become troublesome when larger quantities of calcines limestone dust begin to enter the ash silo and are quenched by water during removal from the silo.

The temperature of the ash to the power company ash silo must be monitored closely in order not to exceed their maximum allowable temperature of $150^{\circ}F$.

Stone Handling System

The limestone is delivered by truck and charged pneumatically into the concrete storage bunker. Before going into the bunker, the material goes through a separator and dust collector removing most of the fines.

The time required to unload a truck is about three (3) hours and this is two (2) times the intended unloading time; however, no availability problems have been experienced as yet. The required lift is 94 feet.

Discharge from the bunker is through two (2) 2' square outlets into vibrating conveyors and then by a bucket elevator and Redler conveyor to one of three limestone bins. Feed from these long rectangular bins is through a transition piece. The transitions have an outlet which is square and which feeds to a circular pipe whose inside diameter is, at most, 1/2" larger than the inside dimension of the square.

PRACTICAL APPLICATIONS OF OPERATING EXPERIENCE

Coal Systems

A temporary solution to the coal feeding problem has been to accept only properly sized (1/4" x 1/2"), dry (3% max. moisture) coal. This involves additional expense and since considerable breakage occurs in the plant coal handling system, a substantial percentage if fines must still be handled.

All welds on the new stainless steel sloping plates in the bins have been polished. These lessen the effects of the shallow corners in each pyramid. Low pressure pulse air supplied to the region between the pyramid and cone would probably promote better flow.

Smooth stainless steel liners were installed in the rotors of two of the rotary feeders, but this reduced the feed rate by 25% and could not be tolerated.

Vibrators have been installed on the bins.

Vents have been installed to avoid the air bubbles which form at the rotary feeder just under the bins. This is a temporary solution and may bleed off more air from the vibrating feeders than the auxiliary F. D. fan can supply, when all four (4) cells are in operation. If this proves to be the case, then a correct differential between vibrating feeder and fluidized bed cannot be maintained and needle pluggage may occur. This problem will be especially troublesome when a maximum bed depth is desired.

The ultimate disposition of the coal dryer problem is not clear at this time.

Some possible solutions are:

o   Move the dryer outside the plant.

o   Use the flue gas as the drying medium instead of heated air

o   Install explosion relief doors with ducts to the outside of the plant.

Limestone Systems

The limestone truck unloading time could be reduced by assisting truck mounted blower with conveying air from another source.

Coal and Limestone Systems

The control arrangement which starts and stops the vibrating feeders, weigh belts, bucket elevators and Redler conveyors fully loaded in response to bin levels could be modified to stop only the vibrating feeders first, then after a suitable time delay, to permit the other elements of the system to unload, the weigh belts, bucket elevators and Redler conveyors could be stopped. This would permit gradual loading and reduce the frequency of shearing pins in the bucket elevator and Redlers.

CONCLUSIONS

Rivesville has a plethora of material handling problems. Some of these problems are common to most material handling systems and can be corrected by available methods and equipment. Other problems which are peculiar to fluidized bed boilers are more difficult, but practical solutions must be found if the fluidized bed boiler is to achieve commercial success.

We believe that the most pressing problem facing this technology is the method of dividing the fuel into the required number of individual

streams. The second most obstinate design hurdle may be the method of conveying the indivudual streams into the fluidized bed, while avoiding such problems as feed stoppage, localized reducing conditions, poor distribution and excessive elutriation of fines.

We have not addressed the questions of combustion efficiency, automatic combustion control and sulfur removal since they are not directly related to materials handling and experience at Rivesville is not sufficient at this time to have conducted definitive evaluations. It appears that if the material handling and fuel feed systems can be made tc function properly the boiler will be stable, efficient, reliable and will meet emission standards.

ACKNOWLEDGEMENT

We wish to thank the Pope, Evans and Robbins, Inc. personnel who have been extremely cooperative and open concerning the hardware performance at Rivesville.
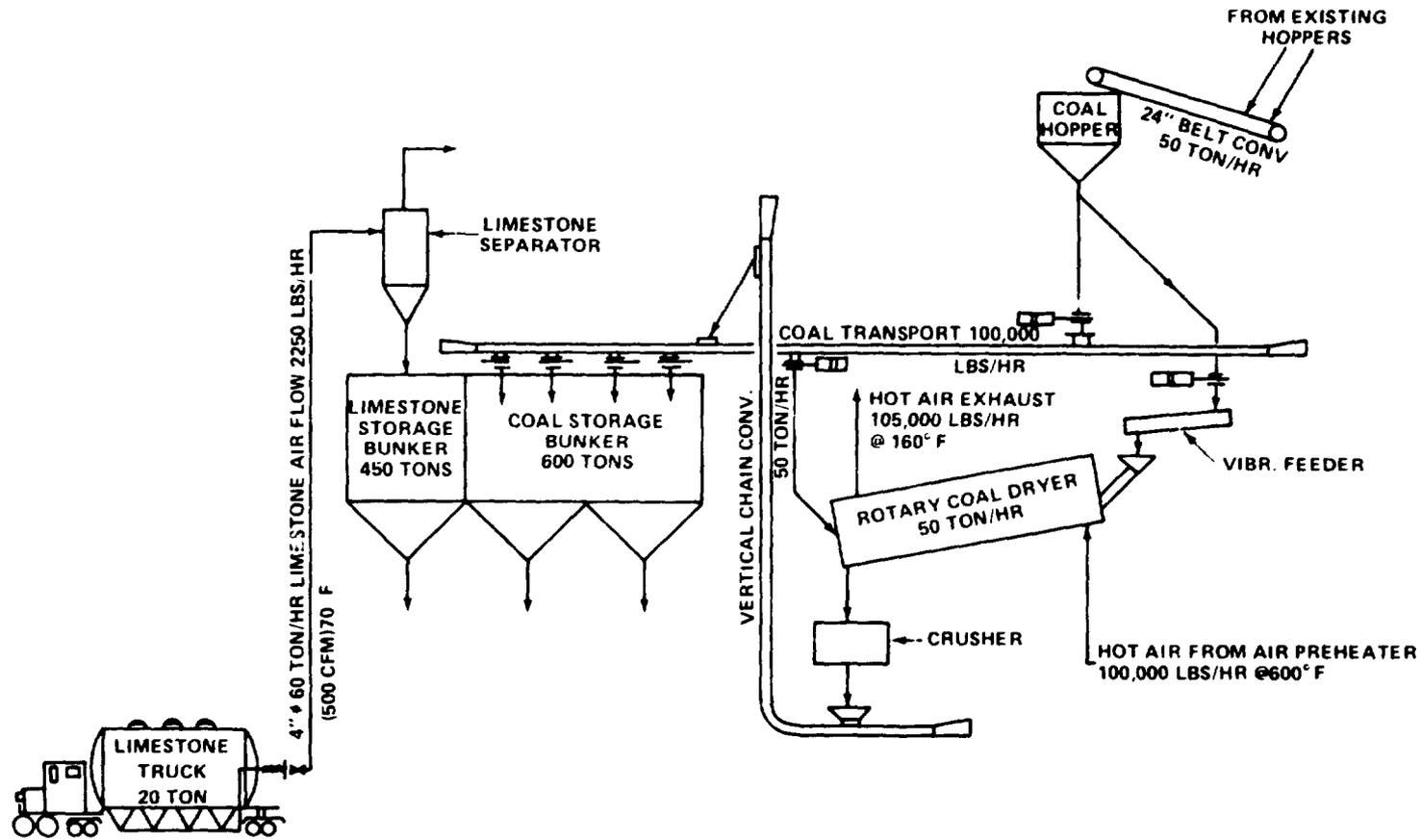
FIGURE 1
COAL & LIMESTONE
HANDLING SYSTEMS
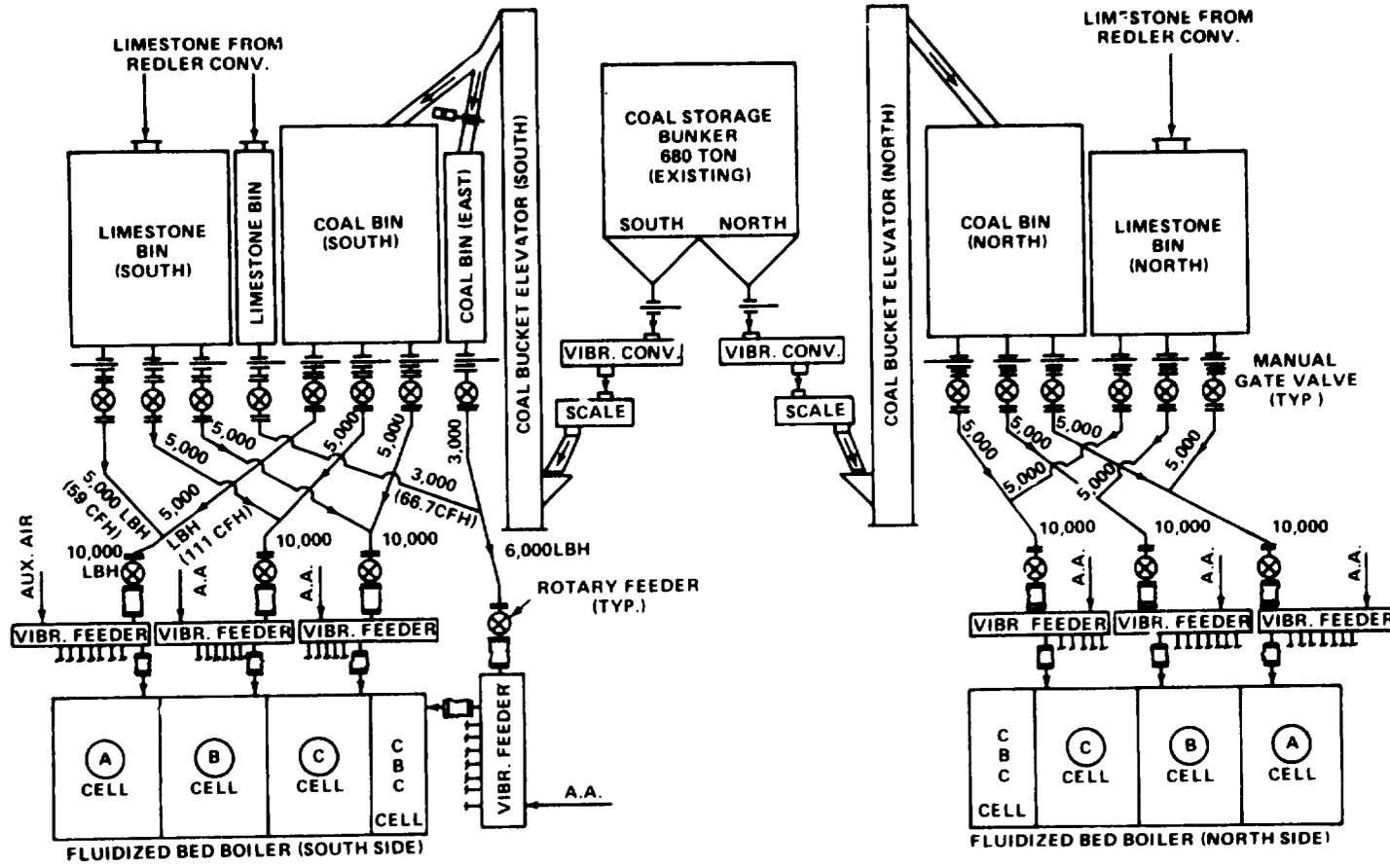
FIGURE 2

FURNACE FEED SYSTEMS

LIMESTONE
STORAGE
BUNKER

LIMESTONE BUCKET ELEVATOR (SOUTH)

REDLER CONV.

REDLER CONV.

LIMESTONE BUCKET ELEVATOR (NORTH)

LIMESTONE
STORAGE
BUNKER

VIBR. CONV.

SCALE

TO
LIMESTONE
BIN
SOUTH

TO
LIMESTONE
BIN
EAST

TO
LIMESTONE
BIN
NORTH

VIBR. CONV.

SCALE
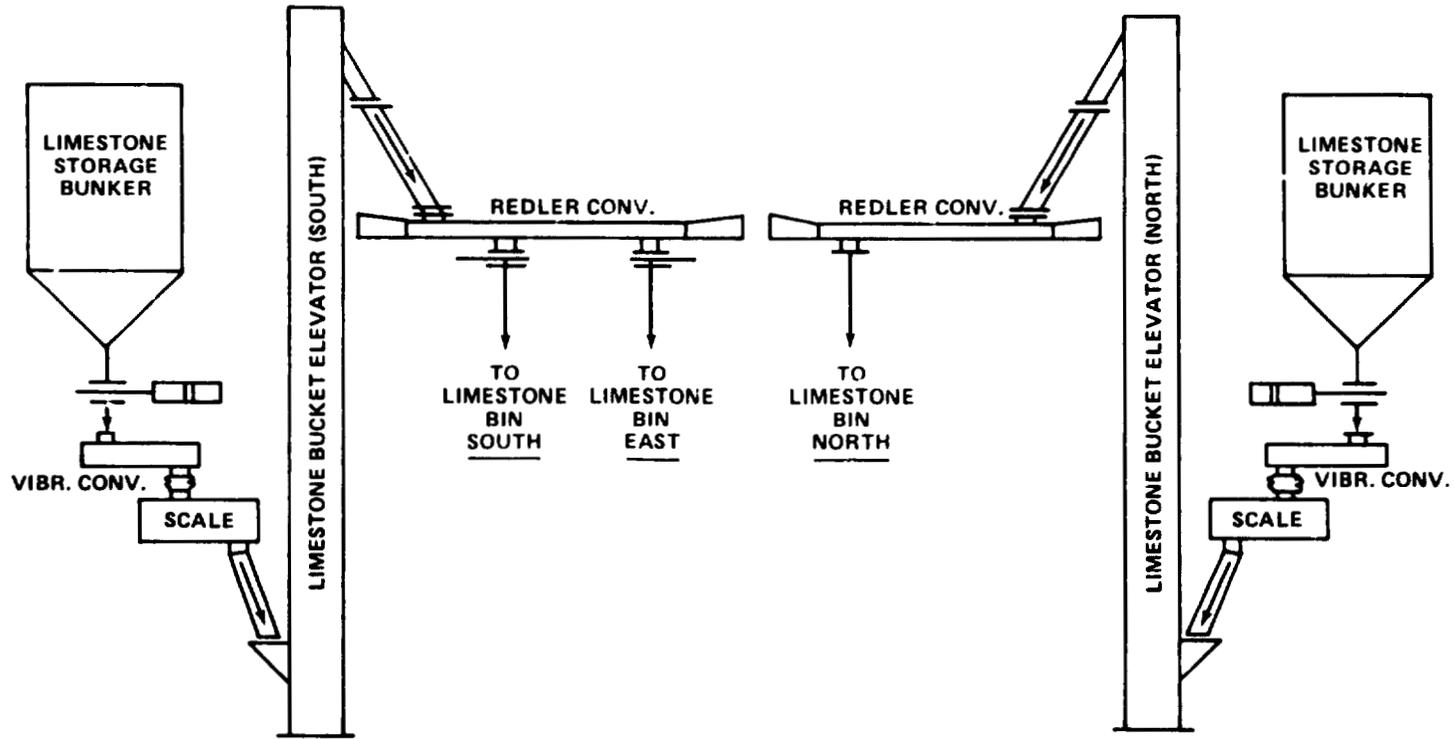
FIGURE 3
LIMESTONE HANDLING SYSTEM
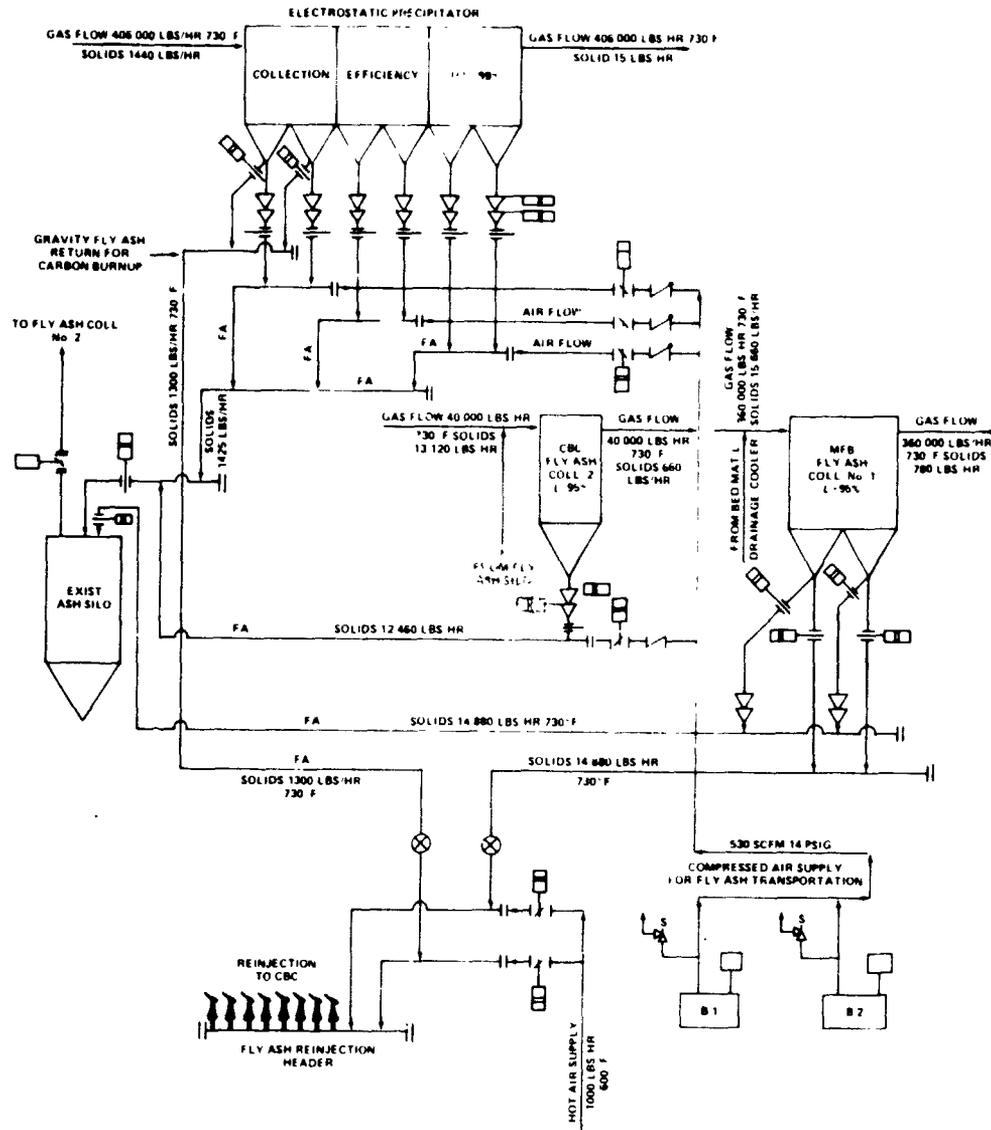
**FIGURE 4
FLY-ASH COLLECTION AND
REINJECTION SYSTEMS**
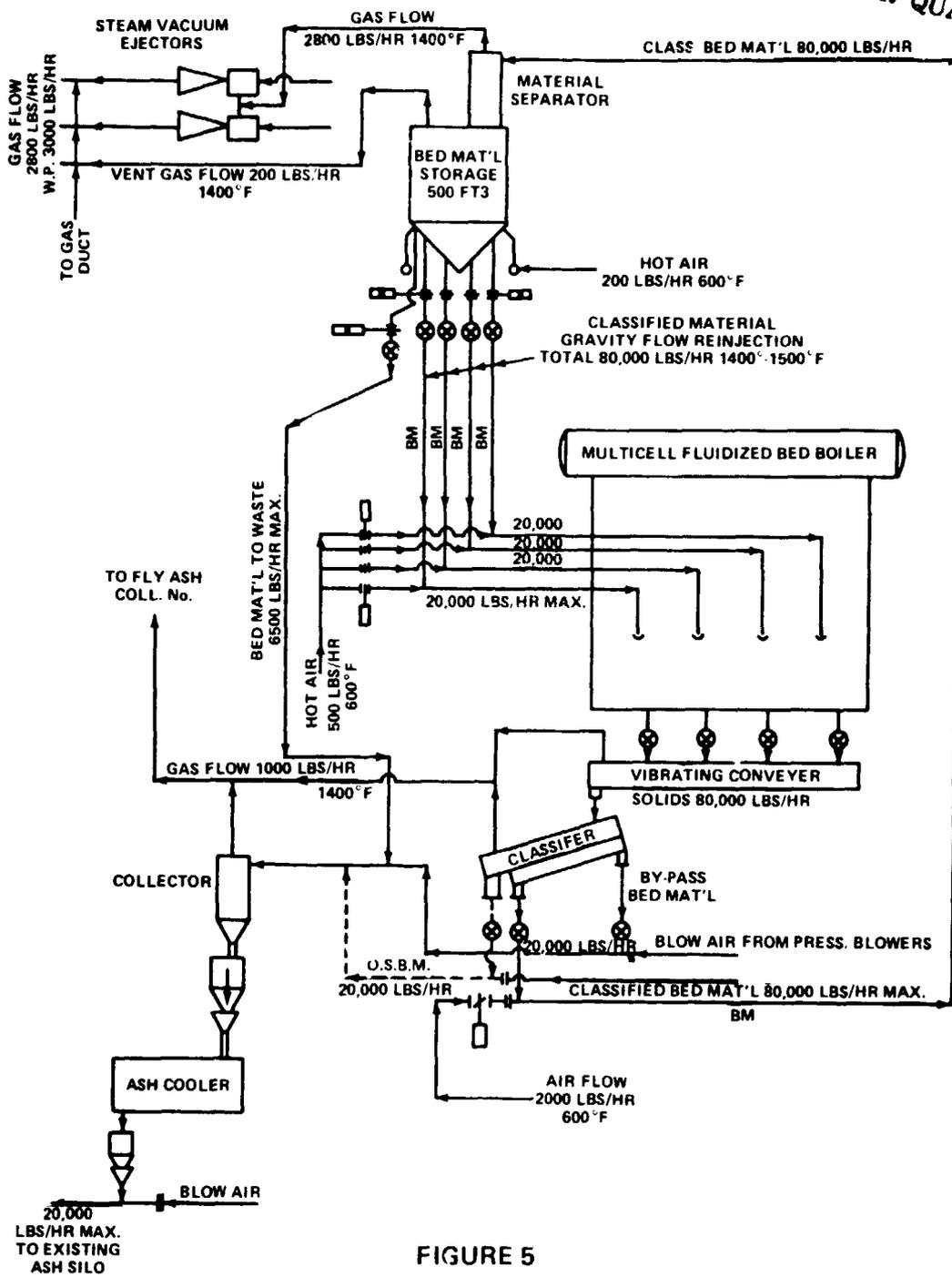
FIGURE 5

BED MAT'L HANDLING
SYSTEM

**FIGURE 6
CARBON BURNUP CELL SHOWING SLIDE GATE**
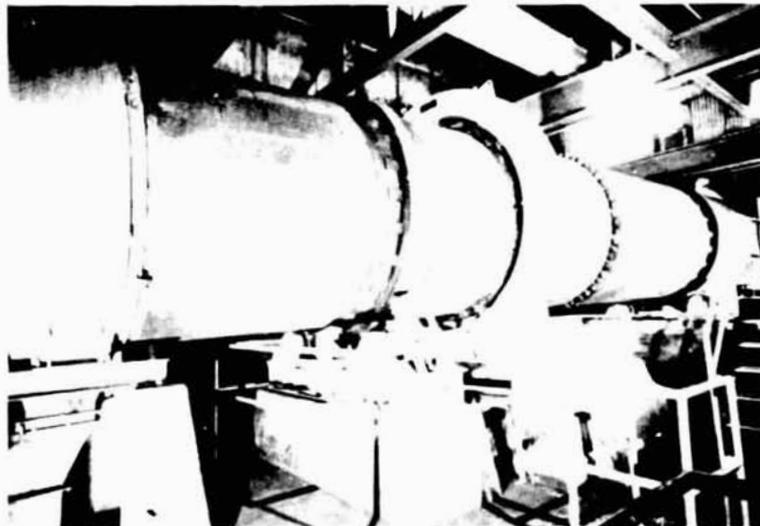
**FIGURE 7
COAL DRYER**

FIGURE 8
FEED NEEDLES
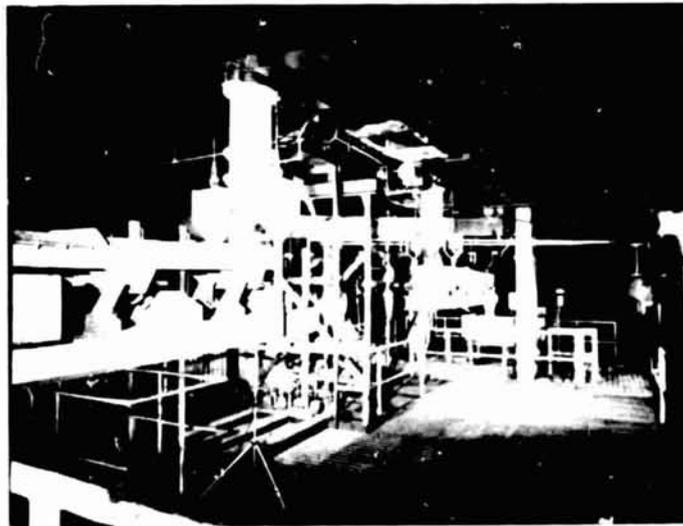
FIGURE 9
VIBRATING FEEDERS LIMESTONE
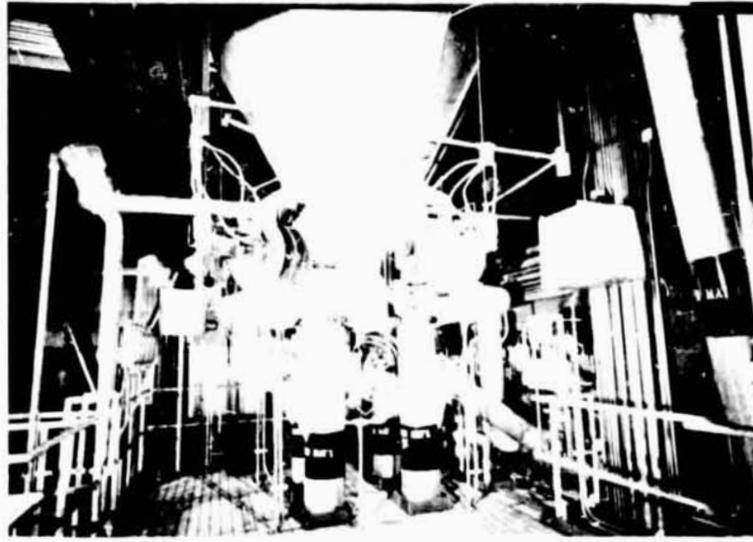
FIGURE 10
BED MATERIAL STORAGE
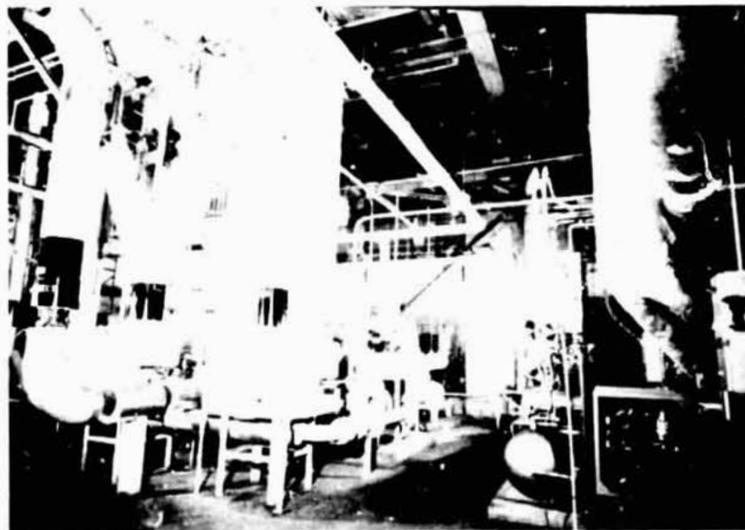
FIGURE 11
FLY ASH REINJECTION AND/OR DISPOSAL

COAL/OIL SLURRY FEED PUMP
DEVELOPMENT

D. E. Davis

G. S. Wong

Rockwell International/Rocketdyne
Division
Canoga Park, California

H. H. Gilman

Electric Power Research Institute
Palo Alto, California

ROCKETDYNE'S ADVANCED COAL SLURRY PUMPING PROGRAM

by

Dr. D. E. Davis*, u. S. Wong,** and H. H. Gilman***

ABSTRACT

The Rocketdyne Division of Rockwell International Corporation is conducting
a program for the engineering, fabrication, and testing of an experimental/
prototype high-capacity, high-pressure centrifugal slurry feed pump for coal
liquefaction purposes. The program is being conducted for the Electric Power
Research Institute (EPRI) of Palo Alto, California.

The abrasion problems in a centrifugal slurry pump are primarily due to the
manner in which the hard, solid particles contained in the slurry are trans-
ported through the hydraulic flow passages within the pump. The abrasive
particles can create scraping, grinding, cutting, and sandblasting effects on
the various exposed parts of the pump. These critical areas involving abra-
sion and impact erosion wear problems in a centrifugal pump are being addressed
by Rocketdyne. The mechanisms of abrasion and erosion are being studied
through hydrodynamic analysis, materials evaluation, and advanced design
concepts.

INTRODUCTION

With the impending shortage and increasing cost of energy resources, the con-
version of coal to alternate usable forms such as liquefaction and gasifica-
tion are increasing in importance. Because of this urgent need, the Fossil

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*   Program Manager   Coal Slurry Pump Program, Rocketdyne Division, Rockwell
International Corporation

**   Project Engineer, Advanced Pump Systems, Rocketdyne

***   Engineering Manager, Advanced Fossil Power Systems, Electric Power
Research Institute

Fuel and Advanced Systems Division of the Electric Power Research Institute
is sponsoring a project for early development of a slurry feed pump for coal
liquefaction. The coal liquefaction processes presently being considered
for commercial development generally require high-pressure coal slurry feed
to be supplied to a preheater and reactor at high capacities.

The operating SRC pilot plants at Wilsonville, Alabama, and Tacoma, Washing-
ton, currently use conventional low-capacity, rec procating feed pumps.
However, the future scaleup of the present type of feed pump to commercial-
size plants would mean using an extremely large number of reciprocating pumps
in parallel with the high capital and maintenance costs associated with mul-
tiple units. Therefore, high-volume centrifugal pumps producing high pres-
sures are considered to be excellent candidates for addressing the above
problems with respect to commercial liquefaction plant feed systems. Although
centrifugal pumps are advantageous for high-volume applications, current
designs are subject to excessive internal wear because of high velocities.

The primary approach being taken in the current EPRI program is to conduct a
comprehensive engineering study of the problems of pumping highly abrasive
coal/oil slurries. A concentrated effort is made to identify the problems by
making maximum use of the data, knowledge, technology, and experience that
exist in industry and at Rocketdyne.

The present test program and design study are continuing with the ultimate
goal of providing a reliable centrifugal coal slurry feed pump designed for
the following conditions:

●	Capacity	5000 gpm
●	Feed Rate	50 to 100%
●	Pressure	3000 psi
●	Temperature	$550^{\circ}F$
●	Coal/Oil Slurry	
	Concentration	50% (by weight)
	Specific Gravity	1.20
	Viscosity	25 to 50 c.p.
	Solid Size	200 mesh

• Operation

Minor Maintenance	6 months
Major Overhaul	1 year

The results and data obtained in the study program on slurry flow, material erosion, and promising high-wear-resistant materials and coatings will be utilized in the prototype slurry pump study to ensure a realizable and practical pump design.

TECHNICAL DISCUSSION

The mechanisms of erosion and abrasion wear, and its relation to velocity and material wear resistance, is now discussed in order to present an understanding of the phenomenon and the design problems it presents. Both hydrodynamic and mechanical design considerations are discussed where abrasion control can be exercised through control of pump internal flow velocities, simple geometry, and generous and continuous bends and curvatures. The effect of pump design parameters on pump internal relative velocities is briefly described. A discussion of promising pump materials, of construction, and hard facing coatings and inserts is also presented.

PUMP ABRASION AND EROSION

Figure 1 illustrates critical areas in a centrifugal pump where abrasion and impact erosion wear problems may be expected. These include abrasion on impeller front shroud and back plate, casing and volute, wear ri   and seals; and erosion of impeller blade leading and trailing edges and the volute cut-water. Although particle size and shape are factors affecting wear, generally abrasive wear in slurries is found to vary as the 2.7 power of slurry velocity. Thus, if slurry flow velocity is doubled, the erosion wear rate is increased sixfold. In an optimum slurry pump design, therefore, local slurry velocities relative to metal surfaces must be kept low to reduce abrasion and high-velocity impingement against metal surfaces. Also, in appropriate parts of the pump, high hardness material and coating should be used to reduce abrasive wear and ductile materials may be used to reduce impact erosive wear.
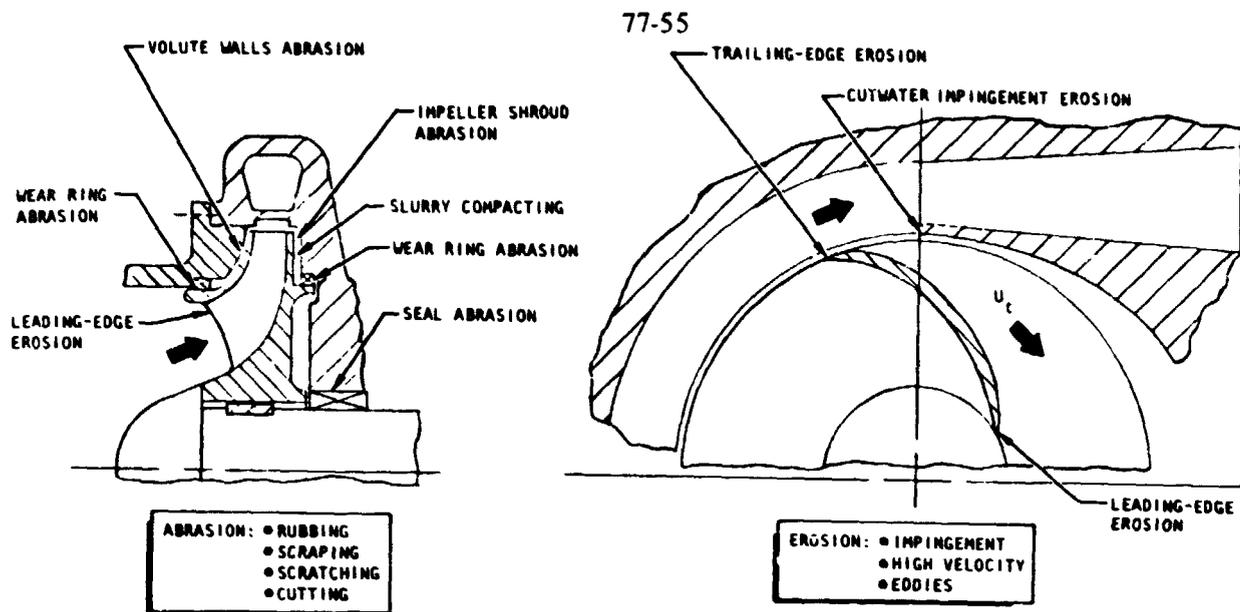
Figure 1. Critical Areas of Wear in a Centrifugal Pump

Despite the high pump tip speed needed to develop high pressures in a minimum number of stages, an understanding of the slurry wear problems will enable the designer to maintain low local slurry velocities near metal surfaces to ensure maximum pump operating life.

HYDRAULIC DESIGN

The maximum relative velocities that occur in pumps are at the impeller inlet eye, at the impeller exit, and at the vol'.e cutwater. Typical velocities as a function of pump parameters flow coefficient (ϕ) and head coefficient (ψ) are shown in Figures 2 and 3. The impeller inlet relative velocity can be minimized by reducing the inlet eye diameter or by introducing inlet whirl in the direction of rotation. Using impeller inlet whirl is desirable since it reduces interstage diffusion requirements in a multistage pump (as the diffuser is not required to remove all the whirl velocity) and permits all impellers to be hydrodynamically identical.

The impeller design affects efficiency directly. The number of blades, entrance angle to the blades, blade thickness and blade contour are all pertinent design lements. Slurry flow velocities are also directly related to
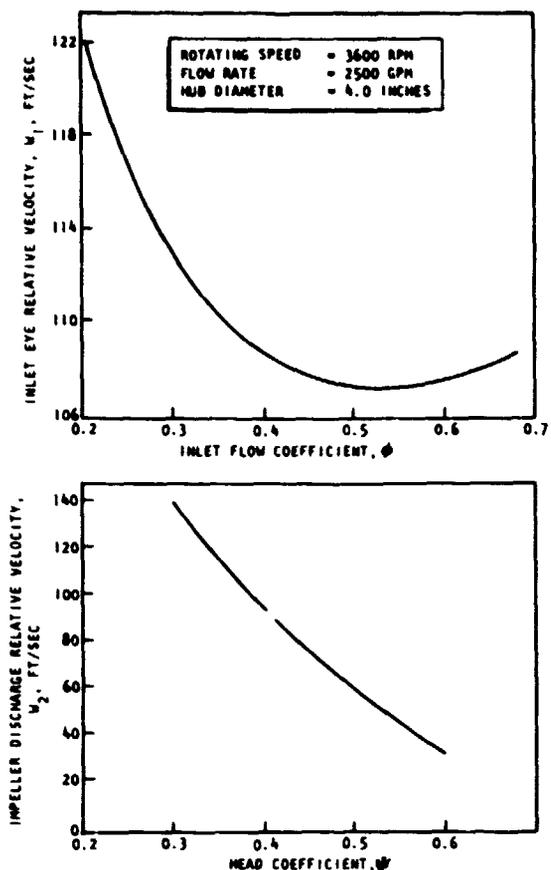
Figure 2. Effect of Pump Design
Parameters on Pump
Relative Velocities
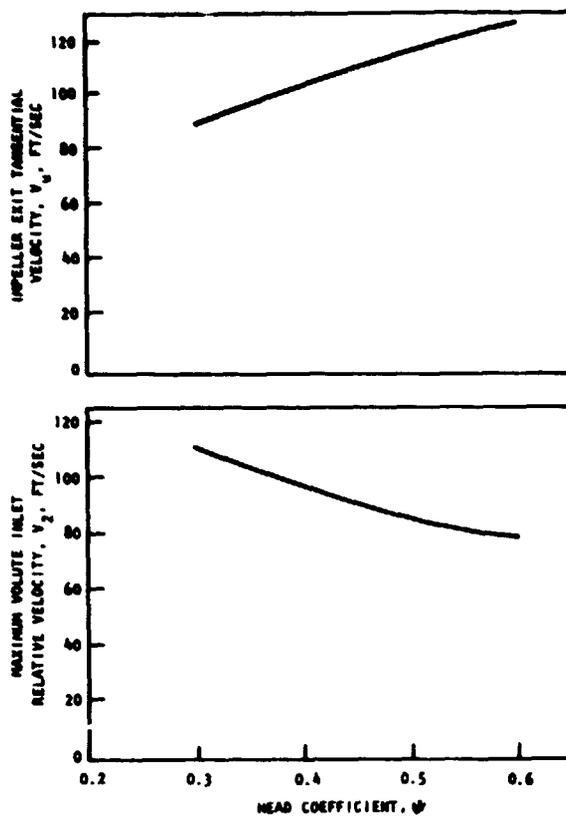
Figure 3. Effect of Pump Head
Coefficient on Volute
Inlet Relative Velocity

these same design elements. Thus, if an impeller is designed for large flow passage areas by using fewer and thicker blades to reduce slurry velocity, it must be traded off with possible flow separation and eddies, causing localized wear, and loss in hydraulic efficiency that may result.

Impeller front shroud and back plate clearances can be used to control internal velocities and close-clearance abrasion to some extent. Increasing the clearances will minimize potential abrasion wear problems; however, this must be evaluated against reduced efficiency, induced recirculation within the clearances, and secondary flows from high to low pressure areas. Secondary flows themselves will cause localized erosion. Using impeller wear rings is a limited solution to this problem.

As high slurry velocity occurs at the impeller exit or volute inlet, increasing the cutwater clearance at the volute tongue will reduce slurry velocity and impact erosion. However, loss in efficiency must be minimized. Operation of the pump at partial capacity (low flow conditions) must also be kept at a minimum, as accentuated "off-design impingement angle" will cause high impact erosion at the impeller blade and volute cutwater.

WEAR MECHANISM

Erosion of a surface by solid particles entrained in a fluid stream is observed in many different applications such as coal-burning turbines, coal hydrogenation process equipment, coal transport pipelines, suction dredges, and coal-burning, furnace-induced fans. In the present application of a slurry pump for a coal liquefaction process, the solids concentration by weight will be as high as 50 percent. In the study of the wear phenomenon in the pump design, this high concentration of solids at high velocities must be carefully considered.

The two basic types of wear mechanisms that occur in a fluid stream containing solid particles are: ductile erosion (abrasion), and brittle erosion (impact). The former type is one in which the material removal is due entirely to the cutting or displacing action of the particle, similar to grinding or single-tool cutting processes; the latter type is one in which the removal of surface material is caused by a stream of impinging solid particles as in sandblasting, or by liquid impact, as in the case of impeller cavitation.

The mechanical properties involved in the erosion mechanism between the solid particles and material surface include: hardness, plastic flow stress, and Young's modulus. The slurry properties involved are: fluid carrier, concentration. velocity, impingement angle, and particle size distribution, shape, and rotation.

Abrasion

The erosive cutting of ductile materials has been studied analytically by Finnie (Ref. 1). The analysis essentially involves the solution of the

equations of motion of the particle into the eroded material under the action of the resisting plastic flow stress of the material. The amount of material the particle removes is then determined from the displacement path of particle in the material. From analyses, the following relationship is obtained for the volume of material removed by particles:

$$Q = C \frac{MV^2}{2} \frac{1}{P} f(\alpha) \tag{1}$$

where

Q =   volume of material removed

M =   mass of impinging particles

V =   particle impinging velocity

P =   plastic flow stress

$f(\alpha)$ =   a function of impinging angle α to the eroding surface

C =   a constant ranging 1/8 to 1/12

Equation (1) indicates that for the erosion of ductile materials or abrasion, the material removal varies linearly with the total quantity of the impinging particles and to the square of the impinging velocity and is independent of the particle size. The only material property that affects the erosion is the plastic flow stress p. It also varies as a function of impinging angle as shown by the dotted curve for aluminum alloy in Figure 4.

The variation of the erosion of ductile material with impinging angle α can be explained as follows. At small values of α, erosion increases
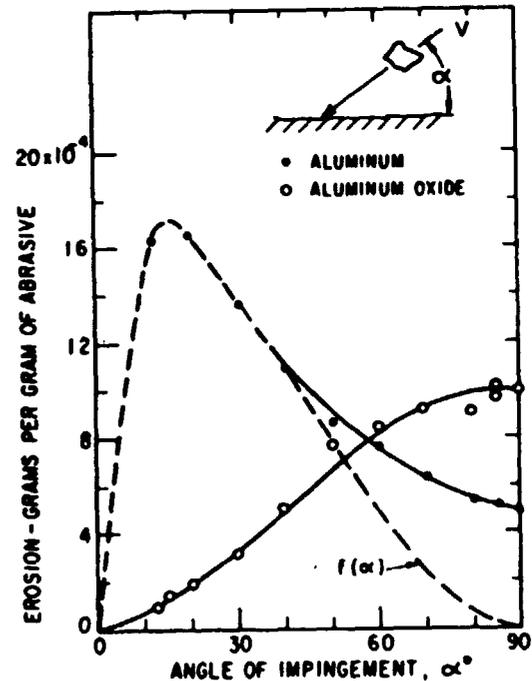
Figure 4. Erosion of Aluminum and High-Density Aluminum Oxide by SiC Particles (120 Mesh and 500 ft/sec)

with α due to increasing normal velocity component and resulting in increasing force of penetration into the material by the particle. At large values of α, the forward cutting velocity component decreases with α, resulting in decreasing forward displacement and erosion. The maximum material removal can be seen to occur at an impinging angle of approximately 20 degrees. The analytical relationship, however, underestimates the material removal at large angles as indicated by the solid curve obtained experimentally. The deviation is attributed generally to the surface roughing and work hardening by the impinging particles at large angles.

Erosion

The study of erosion for brittle materials has been made by Sheldon and Finnie (Ref. 2). The analysis applies the classical Hertz equations to derive the depth of penetration and the resulting stresses in the region around the material indentation and then followed the Weibull theory of mean fracture strength of a material with a volume distribution of flaws to determine the volume of fracture for an impinging angle of 90 degrees. The following relationship for weight removal in cubic centimeter per 10^9 particles was obtained:

$$W = K R^a V^b \tag{2}$$

where

W   = weight removal per 10^9 particles

R   = mean particle radius

V   = impinging velocity with $\alpha = 90$ degrees

K   = erosion parameter in cc per 10^9 abrasive particles

a, b = exponents dependent on materials and shape of particles

The erosion weight loss on brittle materials is dependent on particle size and is a function of larger power of the velocity than the weight loss on ductile material. The variation of brittle material erosion with impinging angle is shown by the solid curve for aluminum oxide in Figure 4. Experimental results indicate the size exponent (a) can vary from 3 to 5 and the velocity exponent (b) is close to 2.7.

CANDIDATE MATERIALS

The optimum slurry pump may be comprised of both ductile and brittle materials, with the latter used in selected high-wear areas where impingement angles are low. Since hardness is often a desirable property in low-impingement areas where wear can be a problem, the list of candidates will reflect mostly materials of high hardness. Producibility is another major factor which must be considered for any list of materials, with casting, plasma-spraying, pack or vapor-diffusion processes, welding, brazing, and mechanical joining as possible fabrication methods.

PROGRAM STATUS

A complete engineering study is being made on several potential concepts for the high-pressure, high-capacity, multistage centrifugal pump for coal/oil slurry. This study is further supported by optimization analysis and experimental evaluation of wear-velocity relationships. Detailed design layouts are being made such that: component hydraulic design; structural stress; critical speed dynamics; thrust, seals and bearing loads; and material processes can all be properly evaluated.

The hydraulic design of pump internals emphasizes optimum velocity distributions and minimum highly localized velocities. Generous flow radius of curvature is used and abrupt surface discontinuity is avoided. The mechanical design of the pump utilizes the most appropriate wear-resistant materials, and is configured for simple fabrication and assembly. Since abrasive and erosive wear eventually occurs, the pump design also incorporates convenient disassembly for inspection and parts replacement during minor maintenance and final overhaul.

TABLE 1. POTENTIAL PUMP MATERIALS

Material	Hardness (DPH) (Approximate)
Ductile:	
Stainless Steel (CA6NM)	320
Cast Steel (ASTM216)	250
Titanium (C.P.)	250
Brittle:	
Ni-Hard Cast Iron	650
400C Steel	650
Manganese Steel	580
White Cast Iron	500
Coatings & Inserts	
Silicon Carbide	3000
Silicon Nitride	3000
Tungsten Carbide	2100
Alumina	2000
Tirbaloy T-800	700
Stellite (1016)	650

REFERENCES

1. Finnie, I. - "Erosion of Surfaces by Solid Particles," Wear, Vol. 3, 1960, p. 87.

2. Sheldon, G. L., and I. Finnie - "The Mechanism of Material Removal in the Erosive Cutting of Brittle Materials," Journal of Engineering for Industry, Trans. ASME, Vol. 88, 1966.

ACTON MASS FLOW SYSTEM
APPLIED TO PFBC FEED

E. Homburg

Acton Corp.
Cleveland, Ohio

N78-13267

ABSTRACT

Dense phase pneumatic conveying and the Acton Mass Flow concept
are defined with emphasis on the specific advantages to the coal and
dolomite feed to the Pressurized Fluidized Bed Combustor. The transport
and feed functions are explored with a comparison of designing the process
for a combined function or for individual functions. The equipment required
to accomplish these functions is described together with a typical example
of sizing and air or gas requirements. A general outline of the control
system required to obtain a uniform feed rate is provided. The condition
of the coal and dolomite and conveying gas as required to obtain reliable
transport and feed will be discussed.

ACTON MASS FLOW SYSTEM APPLIED TO PFBC FEED

ACTON SPECIALIZES IN THE PNEUMATIC TRANSPORT OF DRY MATERIALS. THESE CAN RANGE IN BULK DENSITY FROM 5#/CF TO 250#/CF AND IN PARTICLE SIZE FROM SUB-MICRON TO 3/8".

WE DO SOME WORK IN "DILUTE" PHASE" PNEUMATIC CONVEYING WHERE MATERIAL IS MOVED BY VELOCITY AND PRODUCT TO AIR LOADINGS ARE BELOW A 10:1 RATIO BY WEIGHT.

OUR PRIMARY EFFORTS ARE WITH "DENSE PHASE" SYSTEMS, WHICH CAN OBTAIN LOADINGS RANGING UP TO 100:1.

MOST DENSE PHASE SYSTEMS ARE BEST SUITED TO HANDLING POWDERS. THE ACTON MASS FLOW SYSTEM, SINCE IT DOES NOT DEPEND UPON "FLUIDIZATION" OF THE MATERIAL BEING TRANSPORTED, IS CAPABLE OF HANDLING MATERIALS TO 3/8" GRANULE SIZE.

PNEUMATIC CONVEYING GENERALLY USES GAS VELOCITY TO CONVEY THE MATERIAL, HAVING TO REACH PICK UP VELOCITIES, AND MAINTAIN CARRYING VELOCITIES TO TRANSPORT MATERIAL. THEREFORE THE HEAVIER AND COARSER THE GRANULE, THE HIGHER THE VELOCITY AND CORRESPONDING GAS VOLUME REQUIREMENT.

IN CONTRAST THE ACTON MASS FLOW SYSTEM CREATES A DENSE, LOWLY AERATED MASS OF MATERIAL WHICH IS THEN MOVED THROUGH THE PIPELINE BY GAS PRESSURE. THIS REDUCES LINE VELOCITIES TO A RANGE OF 500 to 2000 FPM INSTEAD OF 3000 TO 8000 FPM WITH A CORRESPONDING REDUCTION IN GAS VOLUME.

THE ACTON MASS FLOW SYSTEM OFFERS SEVERAL FEATURES WHICH CAN BE ADVANTAGEOUS
IN THE FEEDING OF A PFBC.

LOW VELOCITY MINIMIZES SYSTEM WEAR AND SUBSEQUENT MAINTENANCE.

LOW VOLUME INTRODUCES A MINIMUM OF GAS INTO THE COMBUSTION CHAMBER.

LOW VELOCITY MINIMIZES FINES GENERATION WHICH WILL BE REFLECTED IN
THE EXHAUST GAS PARTICULATE CARRY-OVER.

MATERIAL TRANSPORT CAN BE VARIED BY SIMPLE GAS FLOW ADJUSTMENT.

MULTIPLE LINES CAN BE FED FROM A SINGLE GAS PRESSURE SOURCE (ACTON
MASS FLOW PUMP).

CONTINUOUS FLOW CAN BE ACHIEVED BY LOCK-HOPPER FEED.

THE ACTON MASS FLOW SYSTEM IS NON-PLUGGING.

IN THE PFBC APPLICATION THE REQUIREMENT IS TO TRANSFER COAL AND DOLOMITE FROM
A STORAGE AREA TO THE COMBUSTOR AND TO CHARGE THE COMBUSTOR AT A CONTROLLED
RATE. THESE FUNCTIONS MAY BE APPROACHED AS A COMBINED OPERATION OR AS SEPARATE
OPERATIONS. I RECOMMEND DESIGNING FOR TWO (2) SEPARATE FUNCTIONS, THAT IS,
HAVING A TRANSPORT SYSTEM AND A FEED SYSTEM.

THE FEED SYSTEM PUMP WOULD BE SET AS CLOSE TO THE COMBUSTOR AS POSSIBLE THUS
ISOLATING THE FEED FROM SHORT TERM VARIATIONS THAT MAY OCCUR IN A LONG LINE.

THE ACTON APPROACH TO LONG DISTANCE TRANSPORT IS TO CAUSE THE MATERIAL TO FORM
A DENSE MASS IN THE LINE. PRESSURE IS THEN BUILT UP BEHIND THIS MASS AND THIS
PRESSURE DRIVES THE MASS THROUGH THE LINE. THE RESULTING MASS FLOW CAN BE SEEN
BY RUNNING A PRESSURE PROFILE OF THE LINE.

A PRESSURE SENSOR WILL REGISTER DURING PASSAGE OF A PRESSURE-PISTON AND WILL BE BLANKED OFF AS A MATERIAL MASS PASSES. A SIGNAL FROM THE SENSOR, THROUGH AN AMPLIFIER TO A STRIP CHART RECORDER, WILL PRODUCE A MODIFIED SINE WAVE. SEE FIGURE 1.

THIS CHART WILL SHOW THE FREQUENCY OF THE MASS FORMATION.

THIS FREQUENCY WILL VARY WITH DIFFERENT MATERIALS, BUT IS USUALLY IN THE 3 to 5 SECOND RANGE. FROM THIS DATA AND THE MATERIAL DENSITY AND AVERAGE TRANSFER RATE, WE CAN CALCULATE THE SIZE OF THE MATERIAL MASS. THIS WILL RANGE FROM 6 TO 8 FEET FOR MATERIALS SUCH AS OUR COAL AND DOLOMITE. THE PRESSURE PISTON WILL OCCUPY ABOUT 8 TO 12 FOOT OF SPACE.

WE HAVE AN IN-HOUSE TEST FACILITY FITTED WITH ACRYLIC PIPE SECTIONS WHICH PERMITS OBSERVATION OF THIS FLOW PHENOMENA. THE OBSERVATIONS VERIFY THE PRESSURE SENSOR DATA.

THIS CONCEPT AVOIDS DENSE PHASE CONVEYING'S RELIANCE ON AERATION AND THE STRONG PLUGGING TENDENCIES THAT PLACED LIMITS ON ITS ABILITY TO TRANSPORT OVER AN EXTENDED DISTANCE. WITH THE ACTON SYSTEM DISTANCE IS LIMITED ONLY BY THE AVAILABLE PRESSURE.

WITH THIS APPROACH THE PREPARATION AND STORAGE FACILITY CAN BE KEPT AT THE RECEIVING-UNLOADING LOCATION. THE MATERIALS CAN BE TRANSPORTED TO THE COMBUSTOR LOCATION ON A BATCH BASIS AT A RATE COMPATABLE WITH USAGE.

REFER TO FIGURE 2, PFBC TRANSPORT AND FEED SYSTEM FLOW DIAGRAM. THE SYSTEMS FOR COAL AND DOLOMITE ARE IDENTICAL IN CONCEPT AND OPERATION, DIFFERING IN SIZE BECAUSE OF THE USAGE REQUIREMENT. THE COAL SYSTEM WILL BE USED AS AN EXAMPLE. THE DESIGN CONCEPTS WILL ALSO APPLY TO THE DOLOMITE SYSTEM.

THE TRANSPORT SYSTEM CONSISTS OF AN ACTON MASS FLOW PUMP, THE TRANSFER LINE, AND A RECEIVER.

THE FEED SYSTEM CONSISTS OF A LOCK-HOPPER, AN ACTON MASS FLOW PUMP, AND MULTIPLE FEED LINES.

THE TRANSPORT RECEIVER AND FEED LOCK-HOPPER FUNCTIONS ARE PERFORMED BY ONE VESSEL.

IN DESIGNING THE FEED SYSTEM CERTAIN COMBUSTOR REQUIREMENTS, OTHER THAN PROVIDING A GIVEN FLOW OF MATERIAL, MUST BE CONSIDERED.

FIRST, TO OBTAIN MAXIMUM COMBUSTION EFFICIENCY THE FEED MUST BE UNIFORM.

THE MASS FLOW FREQUENCY OF 3 TO 5 SECONDS HAS BEEN DESCRIBED. THIS FREQUENCY MAY BE RAPID ENOUGH TO PROVIDE SATISFACTORY COMBUSTION. HOWEVER THE COMBUSTION MAY BE IMPROVED BY SMOOTHING OUT THESE FLUCTUATIONS.

IF THE FEED PUMP IS LOCATED CLOSE TO THE BASE OF THE COMBUSTOR THE SYSTEM IS REQUIRED MAINLY TO LIFT THE COAL TO THE BED OR BEDS. THIS CONFIGURATION PERMITS ABANDONING THE "MASS FLOW" AND OBTAINING A STEADY CONTINUOUS REGULATED DISCHARGE FROM THE LINE. ALSO, THE CONVEYING GAS FLOW REQUIRED IN A VERTICAL LINE IS LESS THAN HALF THAT REQUIRED IN A HORIZONTAL LINE FOR THE SAME RATE SO LESS CONVEYING GAS IS INTRODUCED INTO THE COMBUSTOR. THIS CHANGE IN FLOW CHARACTERISTIC IS THE MAIN REASON FOR SPLITTING THE TRANSPORT AND FEED FUNCTIONS.

THE FEED LINE SIZE IS DETERMINED BY THE MAXIMUM AND MINIMUM RATE REQUIRED TO BE DELIVERED TO THE COMBUSTOR.

THE MATERIAL FLOW FROM AN ACTON PUMP IS REGULATED BY VARYING THE GAS FLOW TO THE PUMP. WE HAVE BEEN ABLE TO OBTAIN AND MAINTAIN CONSTANT RATES OVER A 4:1 RANGE IN A GIVEN PIPE SIZE. WIDER TURNDOWN RATIOS WILL REQUIRE FURTHER DEVELOPMENT WORK AND POSSIBLY A DUAL FEED LINE ARRANGEMENT.

A DUAL LINE WOULD HAVE A CROSS-OVER RANGE. FOR EXAMPLE, IF THE REQUIRED FEED RANGE WAS FROM 5# TO 30# PER CHARGING POINT WE WOULD USE A 1" LINE FOR RATES OF FROM 5 TO 15#/MINUTE AND A 1-1/1" LINE FOR RATES FROM 10 TO 30#/MINUTE. SYSTEM CONTROL WOULD BE SIMPLIFIED IF THE TURNDOWN COULD BE HELD AT 7 TO 28#/ MINUTE WHICH COULD BE HANDLED IN ONE (1) 1-1/2" LINE.

IN THE FEED SYSTEM DESCRIBED THE LOADING WOULD RUN 30:1 AND THE FEED PUMP WOULD BE OPERATING AT 50 PSIG ABOVE THE COMBUSTOR PRESSURE.

OUR PUMP SIZING IS NORMALLY BASED ON SUPPLYING THE REQUIRED RATE WITH A PUMP OPERATING AT 6 to 8 CYCLES PER HOUR. THE PFBC IS A NON-INTERRUPTABLE PROCESS, THEREFORE THERE SHOULD BE A TIME RESERVE BUILT INTO THE SYSTEM TO PERMIT ROUTINE MAINTENANCE OR CORRECTIONS UPSTREAM. THE TIME RESERVE SHOULD BE AS CLOSE AS POSSIBLE TO THE COMBUSTOR. THE SIZING OF THE FEED PUMP SHOULD SUPPLY THIS RESERVOIR. SELECTING THE FEED PUMP CAPACITY TO STORE ONE (1) HOUR'S USAGE SEEMS REASONABLE.

IF A FEED SYSTEM IS TO BE SIZED FOR A RATE OF 25 TPH THE FEED PUMP WOULD HAVE A MINIMUM CAPACITY OF 1000 CF. THIS SHOULD BE THE MINIMUM RESERVE AVAILABLE DURING NORMAL OPERATION. THE TOTAL FEED PUMP VOLUME WOULD BE 1500 CF. WHEN ITS LEVEL DROPS TO 1000 CF IT WOULD BE RE-CHARGED FROM A 500 CF LOCK-HOPPER. THIS RE-CHARGING WOULD OCCUR AT THIRTY (30) MINUTE INTERVALS.

FOR THE LOCK-HOPPER VALVING WE HAVE SELECTED A SPECIAL MODIFICATION OF THE GEMCO TYPE T SPHERICAL VALVE. THIS VALVE USES A HARD FACED SPHERICAL DISC MATING WITH A HARD FACED GROUND SPHERICAL SEAT TO PROVIDE TIGHT SHUT-OFF AT 200 PSIG WITH TEMPERATURES TO 600° F. THE VALVE BODY IS ECCENTRIC MOUNTED TO PROVIDE CLEARANCE DURING ROTATION AND ADJUSTMENT FOR TIGHT SEATING.

THE LOCK-HOPPER WOULD DISCHARGE TO THE FEED PUMP AND THEN BE RE-CHARGED BY THE TRANSPORT PUMP. TO INSURE A FULL CHARGE, 500 CF, AT THE LOCK-HOPPER FOR THE NEXT CYCLE THE TRANSPORT PUMP SHOULD RE-CHARGE THE LOCK-HOPPER IN TWENTY (20) MINUTES. THEREFORE THE DESIGN RATE FOR THE TRANSPORT SYSTEM BECOMES 1250#/MINUTE.

THE TRANSPOF PUMP CAPACITY COULD BE PICKED UP FROM 125 CF TO 500 CF, REQUIRING FROM 4 TO 1 CYCLE TO FILL THE LOCK-HOPPER. NORMALLY THE SIZE SELECTION IS A COST-TRADE OFF IN THE SYSTEM DESIGN. THE COST DIFFERENTIAL FROM A 125 TO 500 CF PUMP WILL NOT BE A SIGNIFICANT AMOUNT IN THIS OVERALL PROJECT SCOPE; THEREFORE THE 500 CF SIZE SHOULD BE SELECTED SINCE IT OPERATES WIT ESS CYCLING AND WILL PRO-VIDE BETTER LONG TERM OPERATION.

THE LOCK-HOPPER WILL BE VENTED TO ATMOSPHERIC PRESSURE BEFORE IT IS RE-CHARGED SO THE TRANSPORT SYSTEM WILL OPERATE FROM ATMOSPHERIC PRESSURE.

TO OBTAIN THE SELECTED 1250#/MINUTE RATE THE ACTON TRANSPORT SYSTEM WILL REQUIRE A GAS FLOW OF 700 SCFM. THE SYSTEM OPERATING PRESSURE WILL BE DETERMINED BY THE TRANSPORT DISTANCE AND THE CHANGES IN ELEVATION. GAS PRESSURE TO THE VESSEL SHOULD BE IN THE 100 TO 125 PSIG RANGE. THE OPERATING PRESSURE CAN BE ESTIMATED BY ALLOWING 40 PSIG FOR THE FIRST 200 FT. AND ADDING 5 PSIG FOR EACH ADDITIONAL 100 FT., ASSUMING LIFT IS LESS THAN 30 FEET.

PROVISION MUST BE MADE TO EXHAUST THE CONVEYING GAS FROM THE LOCK-HOPPER. THIS GAS MUST BE FILTERED BEFORE IT IS EITHER EXHAUSTED TO ATMOSHPERE OR RE-CYCLED.

THE LOGIC FOLLOWED IN SIZING VESSELS FORCED THIS DESCRIPTION TO FLOW FROM THE FEED SYSTEM TO THE TRANSPORT SYSTEM.

THE TRANSPORT SYSTEM HAS BEEN COMPLETELY DEFINED NOW AND THE SIZING EXPLAINED FOR A SELECTED 25 TPH FEED RATE.

TO SUMMARIZE, THE TRANSPORT SYSTEM REQUIRES AN ACTON MASS FLOW PUMP (500 CF CAPA-CITY) LOCATED DIRECTLY UNDER THE DISCHARGE OF THE PREPARED COAL HOPPERS, AN ACTON CONVEYING LINE (DUAL 5" PIPE) FROM THE PUMP TO THE LOCK-HOPPER, THE LOCK-HOPPER-RECEIVER (500 CF CAPACITY) AND AN EXHAUST MEANS INCLUDING A BAG TYPE VENT FILTER.

FOR THE FEED SYSTEM ONLY THE FEED PUMP HAS BEEN SELECTED ALONG WITH THE COMBINATION LOCK-HOPPER-RECEIVER.

THE FEED LINE OR LINES MUST BE SELECTED TO COMPLETE THE FEED SYSTEM DESCRIPTION.

COMBUSTOR DESIGN CONCEPTS INCLUDE SINGLE BED AND MULTIPLE BED DESIGNS. IN A SINGLE BED DESIGN ALL FEED LINES SHOULD EXTEND FROM ONE (1) FEED PUMP. IN A MULTIPLE BED DESIGN I WOULD RECOMMEND KEEPING EACH BED ISOLATED BY HAVING A FEED PUMP FOR EACH BED

THE TOTAL FEED REQUIRED BY A Bc WOULD BE DISTRIBUTED EQUALLY BETWEEN THE SEVERAL LINES AND THE LINES SIZED AS ILLUSTRATED IN THE TURN-DOWN DISCUSSION.

THIS IS A GENERAL DISCUSSION WHICH LAYS OUT THE GUIDELINES FOR SYSTEM DESIGN.
EACH APPLICATION MUST BE EXPLORED TO DETERMINE THE OPTIMUM DESIGN. LOCATION
OF THE STORAGE AREA RELATIVE TO THE COMBUSTOR, COMBUSTOR DESIGN (SINGLE VERSUS
MULTIPLE BEDS) COMBUSTOR SIZE ARE ALL ACTORS AFFECTING TRANSPORT-FEED SYSTEM
DESIGN.

SHORT DISTANCE TRANSFER AND/OR LOW RATES MIGHT MAKE A COMBINED SYSTEM ATTRACTIVE.
FIGURE 3 PRESENTS A FLOW DIAGRAM FOR THIS CONFIGURATION.

ANOTHER ARRANGEMENT, SEE FIGURE 4, WOULD COMBINE THE LOCK-HOPPER AND TRANSPORT
PUMP AND VALVE THE TRANSPORT LINE TO PERMIT DIRECT TRANSPORT INTO THE FEED PUMP
WHILE IT IS PRESSURIZED.

THESE VARIATIONS ARE SUGGESTED TO EMPHASIZE THE NEED TO CAREFULLY EXAMINE THE
COMBUSTOR REQUIREMENTS AND FACILITY LAYOUT BEFORE SPECIFYING A TRANSFER-FEED
SYSTEM.

THE DISCUSSION TO THIS POINT HAS BEEN CONFINED TO THE METHOD OF MOVING THE COAL
AND DOLOMITE AND THE MECHANICAL AND PNEUMATIC EQUIPMENT REQUIRED. THE OTHER IMPOR-
TANT ASPECT OF THIS APPLICATION IS THE RATE CONTROL.

IN THE SPLIT SYSTEM CONCEPT THE RATE CONTROL OF THE TRANSPORT SYSTEM IS SIMPLE.
A FIXED AMOUNT OF GAS IS ADMITTED INTO THE ACTON PUMP AND LINE AND THE COAL
TRANSFERS AT A CONSTANT RATE. DEVIATIONS FROM THIS RATE ARE NOT CRITICAL.
TRANSPORT PUMP CYCLES CAN BE VOLUME CONTROLLED.

THE FEED SYSTEM PRESENTS AN ENTIRELY DIFFERENT PROBLEM. PROPER COMBUSTION CONTROL
REQUIRES THAT THE BED BE FED AT A CONSTANT RATE WITHIN A NARROW TOLERANCE BAND
AND THAT THIS RATE BE VARIABLE WITHIN THE TURN-DOWN RANGE.

THE ACTON SYSTEM IS ADAPTABLE TO THIS SERVICE SINCE WE CAN VARY THE RATE OF DISCHARGE OF THE ACTON PUMP BY CONTROLLING THE FLOW OF SUPPLY GAS.

THE CONTROL LOOP MUST SENSE THE QUANTITY (WEIGHT) OF MATERIAL FED IN A GIVEN TIME SPAN AND PROVIDE FEED BACK TO TRIM THE GAS FLOW TO HOLD THE RATE CONSTANT AT THE SET VALUE.

THE FEED PUMP WOULD BE MOUNTED ON LOAD CELL SYSTEM. THIS CAN BE EITHER HYDRAULIC OR ELECTRONIC. THE LOAD CELL OUTPUT IS FED INTO A RATE CONTROLLER WHICH LOOKS AT THE CELL OUT PUT AT A PRESET TIME INTERVAL, SAY 10 SECONDS, CALCULATES THE AMOUNT FED AND COMPARES THAT TO A PRESET SIGNAL. DEVIATIONS ARE COMPENSATED BY FEEDBACK TO OPERATE THE GAS FLOW CONTROL VALVE.

THE VARIATION IN AIR FLOW WILL TRIM THE COAL DISCHARGE RATE TO BRING THE LOOP INTO BALANCE WITHIN THE PRESCRIBED TURNDOWN RANGE.

WEIGHT AND RATE PRINT OUT AND READOUT CAN BE PROVIDED AS REQUIRED.

THE CONVEYING CAPABILITY OF THE SYSTEM IS AFFECTED BY THE BULK DENSITY, PARTICLE SIZE AND DISTRIBUTION, AND MOISTURE CONDITION OF THE COAL AND DOLOMITE.

THE BULK DENSITIES OF THESE MATERIALS, 45 TO 55#/CF FOR COAL AND 90#/CF FOR DOLOMITE ARE WITHIN THE NOMINAL RANGE OF THE ACTON SYSTEM.

THE PARTICLE SIZE OF MINUS 1/4" FOR COAL AND MINUS 1/8" FOR DOLOMITE ARE SAFELY BELOW THE ACTON UPPER LIMIT OF 3/8".

THERE SHOULD BE A LOWER LIMIT OR AT LEAST A QUANTITATIVE LOWER LIMIT APPLIED
TO THE PARTICLE SIZE RANGE.

IF EXCESSIVE FINES ARE PRESENT, A SUBSTANTIAL PERCENTAGE SUCH AS 20% MINUS
325 MESH, THE NATURAL FLOWABILITY OF THE MATERIALS WILL BE RETARDED. MORE GAS
WOULD BE REQUIRED IN THE PUMP TO PROMOTE FLOW AND MAINTAIN RATE. THIS CONDITION
WOULD REQUIRE REFINING THE CONTROL SYSTEM TO ADJUST NOT JUST THE TOTAL FLOW, BUT
ALSO TO ADJUST THE GAS FLOW SPLIT BETWEEN THE PUMP AND THE LINE. THIS COMPLICA-
TION IS ELIMINATED IF A LOWER FINES LIMIT IS MAINTAINED

AGAIN BECAUSE OF ITS DETRIMENTAL AFFECT ON MATERIAL FLOWABILITY, THE MOISTURE
CONTENT OF THE COAL AND DOLOMITE MUST BE KEPT TO A MINIMUM. PERHAPS IT WOULD
BE MORE APPROPRIATE TO SAY IT SHOULD BE MAINTAINED AT A CONSTANT MINIMUM MOISTURE
CONTENT. AS WITH VARYING FINES CONTENT, VARIATIONS IN MOISTURE CONTENT CAN DRIVE
THE CONTROL LOOP WILD.

THE ACTON SYSTEM IS CAPABLE OF TRANSPORTING VERY "STICKY" LOW FLOWABILITY
MATERIALS SUCH AS PREPARED FOUNDRY SANDS, CAKE MIXES, TITANIUM DIOXIDE AND MANY
METALLIC OXIDES; HOWEVER THESE SYSTEMS ARE SET UP TO HANDLE THIS TYPE OF MATERIAL.
THEY DO NOT SEE A VARIAT. IN FROM FREE-FLOWING TO EXTREMELY SLUGGISH AS COULD
HAPPEN IN THIS APPLICATION IF THESE VARIABLES ARE NOT CONTROLLED.

ALSO, IN MOST INDUSTRIAL INSTALLATIONS, SMALL VARIATIONS IN TRANSFER RATES ARE
NOT CRITICAL. IN THIS APPLICATION WE MUST MAINTAIN A CONSTANT RATE.

COAL FROM OUTSIDE STORAGE OR HAULING MUST HAVE THE SURFACE MOISTURE REMOVED
PRIOR TO ENTERING THE TRANSPORT-FEED SYSTEM. THIS CAN BE ACCOMPLISHED IN AN
AIR SUSPENSION TYPE DRIER. WE WOULD RECOMMEND CONSIDERING OUR ENTON DRYER
WHICH PRESENTS AN ECONOMICAL AND EFFICIENT MEANS OF REMOVING MOISTURE.

THE ACTON MASS FLOW SYSTEM CAN PROVIDE A RELI/PLE, LOW MAINTENANCE, LOW POWER
CONSUMPTION MEANS OF OBTAINING A CONTROLLED FEED RATE TO THE PFBu. ITS OPERATION
CAN BE OPTIMIZED BY USING A PAIR OF SYSTEMS IN SERIES; ONE TO TRANSPORT THE
MATERIAL AND ONE TO FEED IT TO THE COMBUSTOR BY CONTROLLING ThE FINES CONTENT
OF THE FEED TO 10% AND BY REMOVING SURFACE MOISTURE FROM THE FEED.
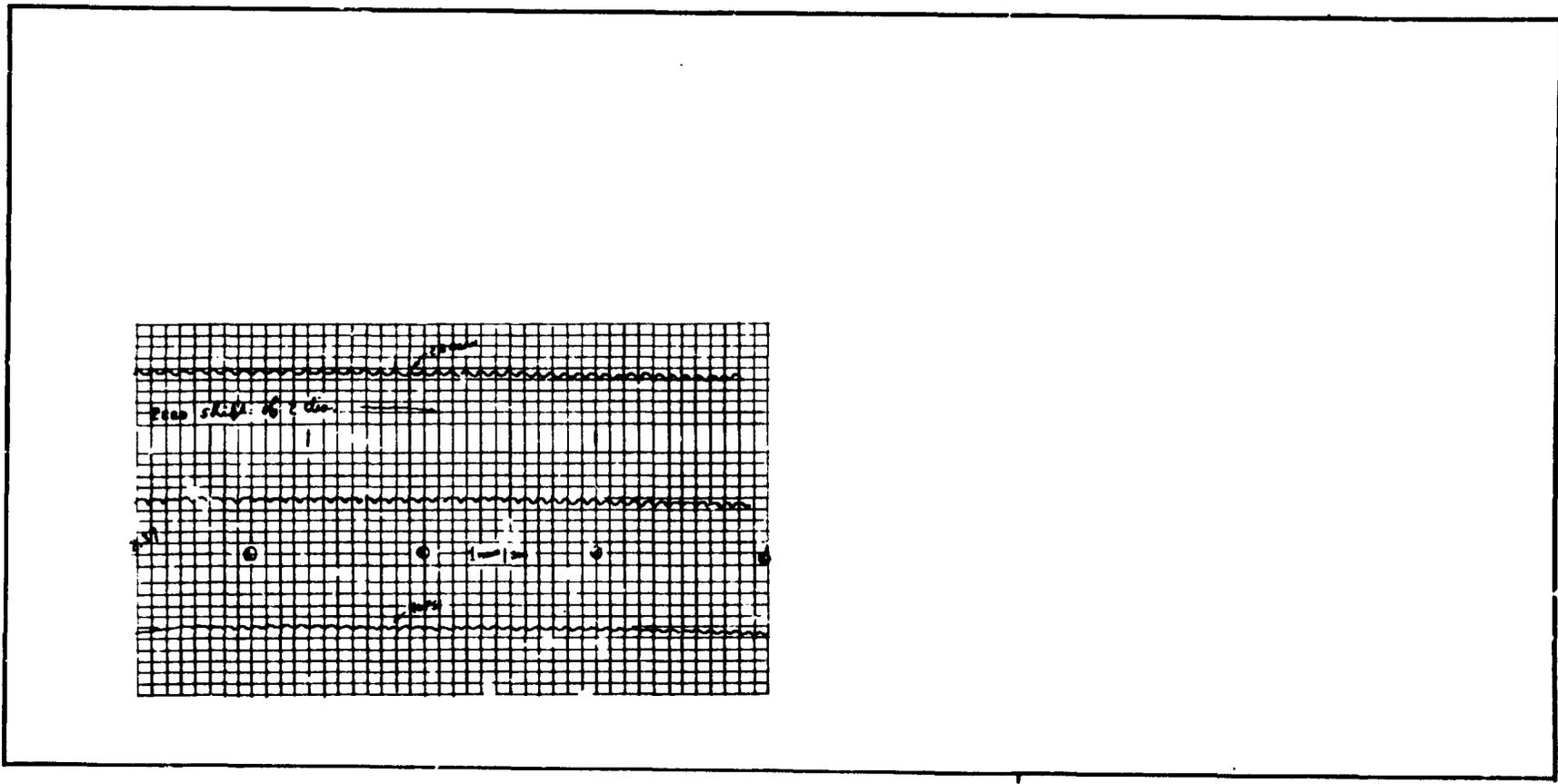
FIG. 1   PRESSURE TRACE
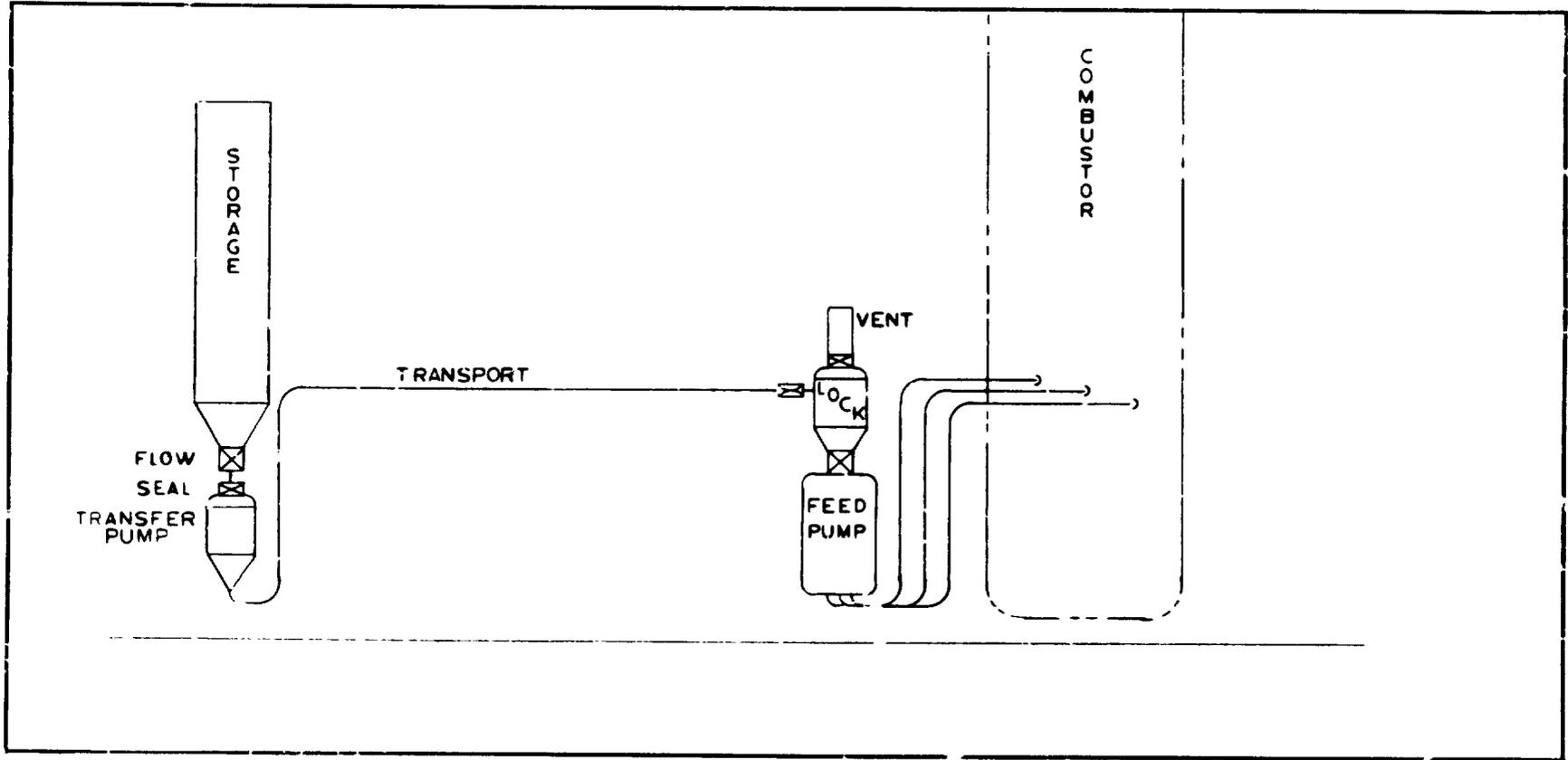
FIG. 2 . FBC TRANSFER & FEED SYSTEM
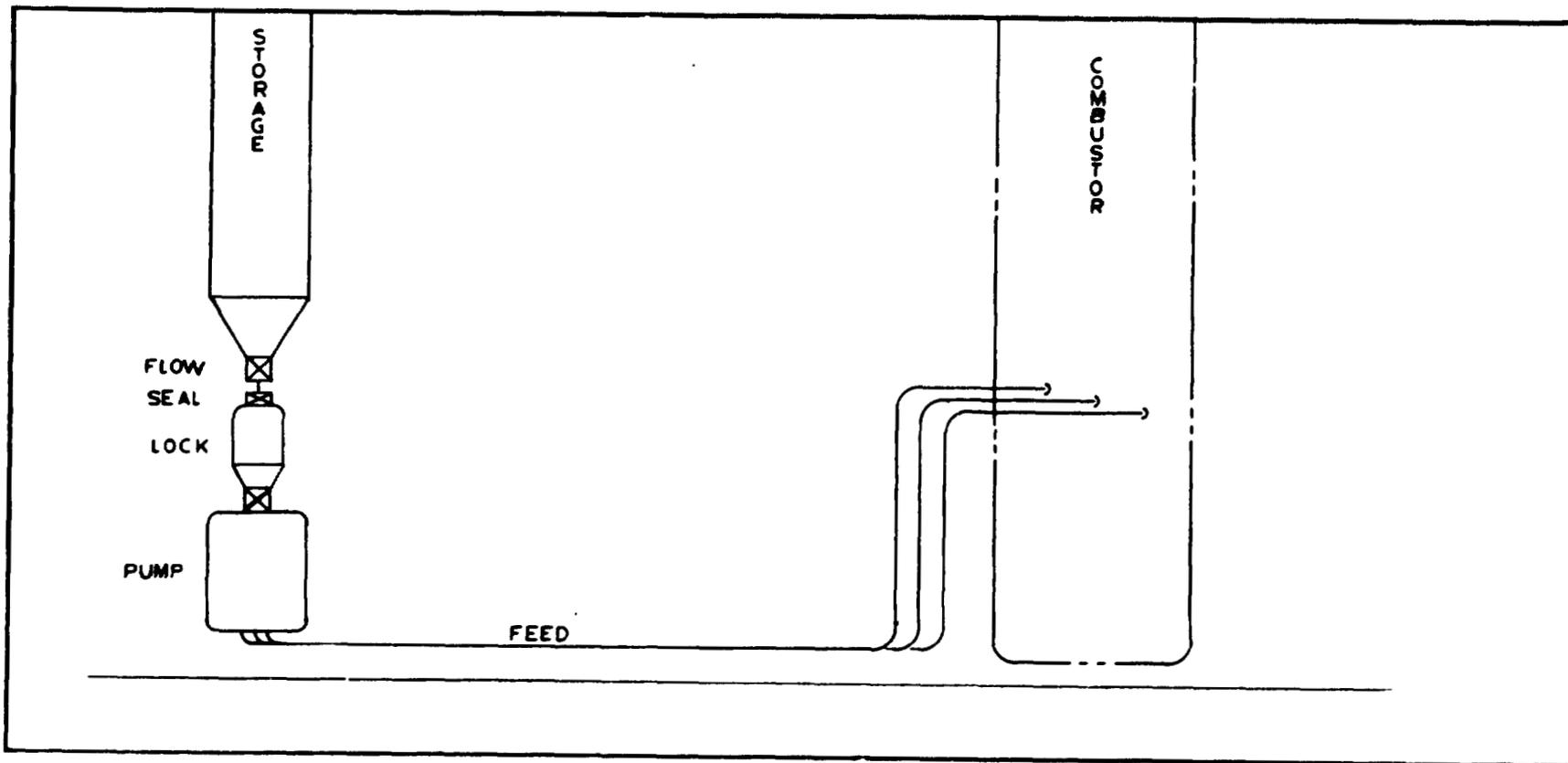
FIG. 3 PFBC FEED SYSTEM
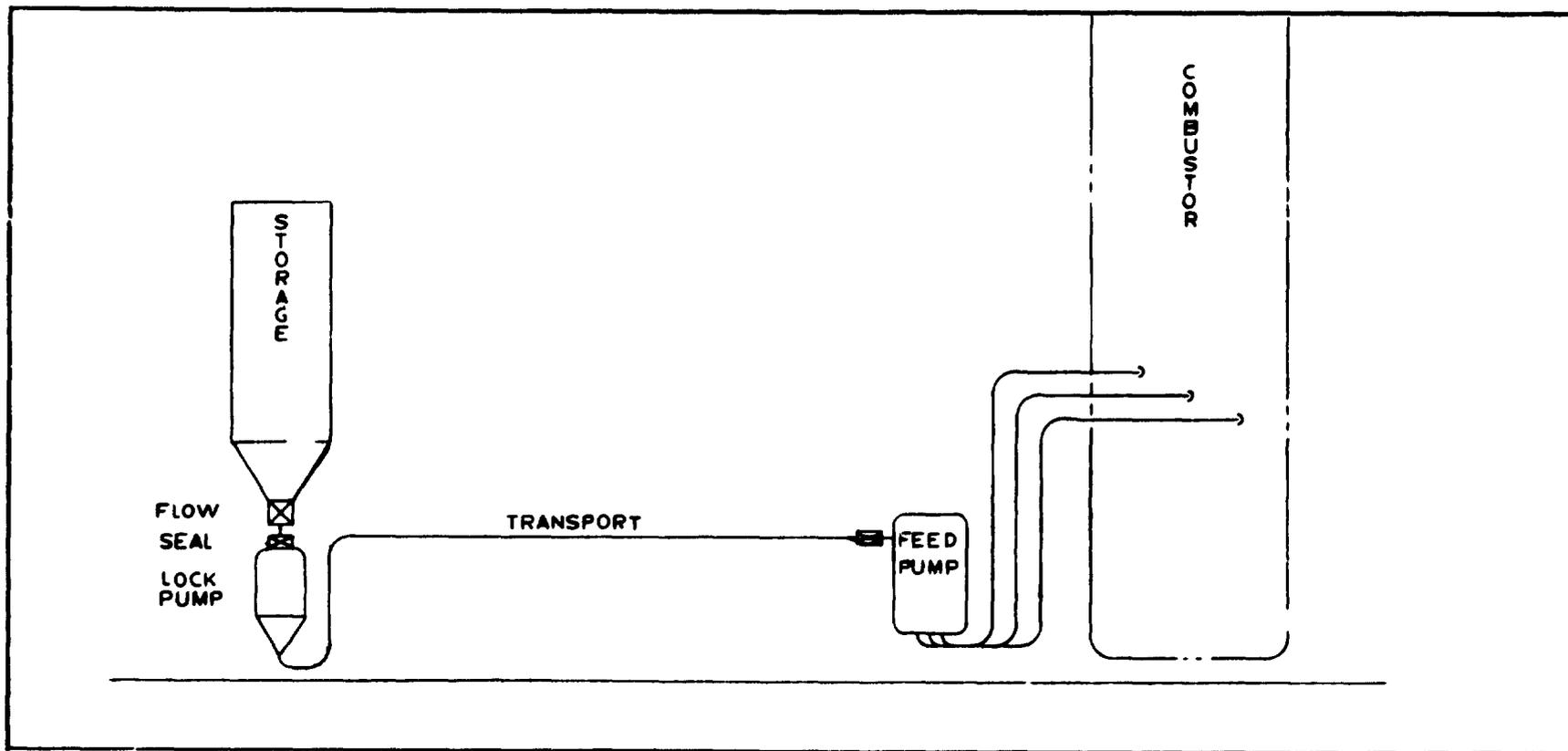
FIG. 4 PFBC LOCK FEED SYSTEM

SOME DEVELOPMENTS IN THE FEEDING OF
COAL TO FLUDIZED BED COMBUSTORS

L. G. Caldwell

Ducon Fluid Transport Division
U. S. Filter Corp.
King of Prussia, Pennsylvania

ABSTRACT

Currently, there is intense interest and considerable funding for the development of fluid bed combustors for high sulphur coal, using limestone or dolomite in the bed for removal of the sulphur. Operating units to date have proven the inadequacies of available material handling techniques for introduction and control of the coal and adsorbent to the beds. Larger units now being contemplated will pose formidable problems in this area. This paper illustrates and describes some of the techniques which have been developed for the existing pilot units and novel ideas under consideration for future, large production units.

Ducon Fluid Transport, with whom I am associated, is a unit
of the U. S. Filter Corporation. We have been engaged in
several aspects of the materials handling associated with fluid
bed combustors, have provided designs and concepts, and have
supplied some equipment in connection with units now being
built. Although our prime field of interest is pneumatic con-
veying, we have developed, or are developing, ideas and devices
which in some cases depart from this field.

Firms working in fluid bed combustion have come to us because
of certain proprietary hardware which we had developed over many
years in other abrasive applications such as mining, petroleum
refining, fly ash, etc. The two principal hardware items we_e
the FLUO/veyor pressure tank high density pneumatic conveying
system, and the PERMA/flo abrasion-resistant rotary air lock.
The former has been used for the high pressure injection of
catalysts into fluid bed reactors and the latter for feeding,
or as an air lock, in the handling of crushed ores, crushed lime-
stone, and crushed coal in a variety of applications. With the
experiences we had with these products, as a starting point, we
became involved in fluid bed combustor activities, and we should
like to share with you now several of the new ideas we have
developed and then to consider a theoretical future utility-sized
power generation boiler and to show how our current technology might
be applied to the feeding of coal and adsorbent.

The first development is the "stream splitter" (fig. 1). The
stream splitter accomplishes the dividing of a single pneumati-
cally conveyed stream of solids into four streams, with each
stream 25% of the original, plus or minus a very small tolerance.
Experience to date shows that tolerances of less than 1% can be
achieved and maintained.

Basically, the stream splitter is a system of branched pipes,
with decreasing diameters to control velocity. The branches,
however, are sharp tees rather than long radius direction changes,
which have been traditional in pneumatic conveying. The use of
sharp tees was pioneered by DUCON in catalyst operations many
years ago. Properly designed and applied, they perform fully
as well as long radius bends.

Figure 2 shows a typical long radius bend. Although the conveying
air is presumed to generally follow the curvature of the pipe, we
know that the solids follow a straight line path, impinging on
a primary, secondary, and sometimes a tertiary point. Incidentally,
we are discussing here conveying in the "stream flow" mode
(usually referred to as "dilute phase"), and this is an essential
requirement for operation of the stream splitter.

In the sharp tee (fig. 3) the approaching stream of material impacts into a pocket and thereafter impinges on itself instead of metal. It makes the direction change with no further impinge- ments, and, surprisingly, with no more pressure drop than the long radius bend.

We experimented with a sharp cross (fig. 4) to see if a stream could be divided equally after a pocket impact and found that the division of flow was very sensitive to the location of the point of impact. We learned that the heaviest flow went to the outlet nearest the point of impact (fig. 5 and 6). Thus, if one could "steer" the approaching stream, one could adjust the split until he had a precise 50-50 division. We accomplished this by the use of flexible pipe joints and a means of changing the pipe geometry and locking its position. We have since added features for thermal expansion.

We found that, once set, the functioning of the device continued without change. In addition, we found that if each of the two legs is again branched, it is possible to achieve a 4-way split with the same close tolerance.

This 4-way splitter was supplied for the San Benito, Texas, fluid bed coal gasifier for the return of fines to the carbon burning cell. It is being designed into several other units on the drawing boards. Although we have not yet done the testing, we believe that the device can be further split into 8 and possibly 16 outlets with good results. The limitation would be that with

decreasing pipe diameter one will reach a point where no further reduction is possible (at about 3 times the diameter of the largest coal particle). We have been highly successful with this device on a 2:1 mixture of 1/4" coal and 1/8" limestone with surface moisture approaching 10%.

The second idea we should like to describe is a coal or coal/limestone feeder capable of introducing a continuous, precisely metered stream into a high pressure combustor or reactor. Fig. 7 shows the arrangement of one version using 2 pressure tanks side by side. These tanks are not simply lock hoppers which dump a charge into the combustor, but each is equipped with a rotary feeder driven by a controllable speed drive responsive to the demands of the process. Since the rotary feeder is completely housed in the pressure envelope, it feels no pressure differential across it. Its drive shaft is brought out through the envelope. Since this feeder must resist abrasion and be totally dependable, it would be well to look more closely at its construction. Fig. 8 is the PERMA/flo feeder as used for many years in highly abrasive applications. Fig. 9 shows a simplified section through the unit where we see the rotor and shoe construction. These two parts are of cast alloy iron with extremely high hardness and arranged so that they can be simply adjusted to one another without disassembly, to compensate for wear. We have taken these two essential parts and a number of others from the PERMA/flo air lock and integrated them into the bottom of a pressure tank.

Our experience over many years with our FLUO/vey r pressure tank conveying system has given us the necessary background to integrate the tank, the appropriate valving, controls, and weighing system into this arrangement that you see.

One tank is always discharging. Its discharge rate is controlled by the demands of the process. The tank is on load cells and we employ a "loss of weight" electronic weighing system which seeks the precise scale discharge rate which the process requires.

The other tank is filling from the atmospheric feed hopper above. When the tank is full, the material cut off valve is closed, the pressure seal valve is closed, and the line from the process vessel is opened to bring the pressure into balance with it. When the first tank is low, but not empty, its bottom discharge valve will be shut and its rotary reeder stopped and at the same instant the bottom discharge valve of the second tank will be opened and its feeder will start. Control of feed will transfer to the second tank.

Before the first tank can be refilled, its vent line must be opened to a dust collector and its pressure reduced to atmospheric. The cycle repeats.

It is possible to accomplish the same result with two pressure tanks, one above the other. In such an arrangement only the lower tank would have the rotary feeder and it would operate continuously. Only the lower tank would be mounted on a "loss of weight" scale. The upper tank would be a pressure vessel which is used to lock in a volumetrically measured batch whenever the lower tank is near empty and needs refilling. The weighing system we would use has a feature whereby the process feed rate is temporarily frozen during the approximately 1/4 minute that the lower tank is being refilled.

This system has been offered for a fluid bed gasifier being used to drive a gas turbine. The gasifier pressure is 90 PSIG.

The third development in which we are engaged is what we might call a "pulsed chute" feeder for fluid bed combustors (fig. 10). Assuming a regulated feed of coal/limestone mixture to the vicinity of the cell, we direct the feed into a small surge hopper which feeds an inclined pipe which terminates, after passing through the cell wall, in the fluid bed. The angle and diameter of the pipe are such that the pipe will remain choked with material, which will not flow without aid. Aid is provided in the form of one or more air e, ;tors through which bursts of high pressure air of extremely short duration and controllable frequency are discharged. Each air pulse discharges almost flat along the surface of the inner pipe wall and completely around the periphery,

surrounding the column of solids with a friction-breaking film of air for an instant. The column moves for a short distance downward. The rate of downward movement of the column is established by the duration and frequency of the air pulses, both of which are electronically controlled. Control of column movement is in response to the material level condition in the surge hopper. High and low level controls or a continuous electronic level control are used to provide the signals to the control system which are converted to the compressed air pulses.

We should like now to propose the possible feeding equipment requirements for a projected 600 megawatt utility sized, fluidized bed boiler such as is currently being studied. We have been asked to consider several possible configurations, but we shall consider one which, it is proposed, would have 35 cells, 7 cells long by 5 cells high (fig. 11). Each cell would be 12 feet wide by 18 feet long. We have been asked to provide for 24 feed pipes for a coal/limestone mixture and it is proposed that half of them feed mixture above the diffuser plate and half of them feed mixture through the diffuser plate from below (fig. 12). Each cell would require a maximum of approximately 11 tons per hour of mixture, made up of approximately 7 tons per hour of coal and 4 tons per hour of limestone.

We propose that at each end of each cell we provide 6 feed pipes from above and 6 feed pipes from below. We propose that each group of 6 pipes be fed from its own surge hopper grouping. For the 6 feeding above the diffuser plate we would use "pulsed chutes" as previously described and these wou. require 6 surge hoppers with level controls and the associated electricals. For the 6 feeding from below the diffuser, we propose the same except that they would terminate in 6 upwardly inclined pipes which pneumatically convey the feed material into the bed with air provided by small blowers.

We propose that each group of 6 surge hoppers in fact be a single 6 compartment circular hopper (fig. 13), which receives a regulated stream of mixture from a disengaging chamber and distributes it uniformly among the 6 compartments by means of a rotary distributor spout.

For each ce l, therefore, there would be 4 disengaging chambers, each receiving 1/4 of the cell feed requirement.

The prepared mixture would be pneumatically conveyed from ground level, in the area of the coal and limestone bunkers, and would be lifted through a single pipeline to the elevation of the cell it is to feed (fig. 14). Using our "stream splitter" we would produce 4 streams to the disengaging chambers, each c which would be 1/4 of the total feed.

It is our belief that control requirements will dictate the need for a separate weighing, proportioning, and mixing unit for each cell. Such a unit might look like fig. 15. Individual bunkers would be provided for coal and limestone. Under each bunker a feed gate would control the flow of material into a scale hopper below. The scale hopper would be refilled whenever it was nearly empty, but would never be permitted to completely empty. A feeder under each scale, with controllable speed drive, would establish the solids feed rate required for the cell. The coal rate would be determined by boiler load and the limestone rate by stack gas composition. The scales would be "loss of weight" electronic with feed rate frozen momentarily during re-filling.

The coal feeder would be a rotary and the limestone feeder a screw, in order to bring both discharging streams close together for feed into a ribbon blender. The ribbon blender would be relatively short, only long enough to provide the desired coal/limestone blending. The discharge of the blender would be into an abrasion-resistant rotary air lock and thence into the conveying pipeline.

Since the arrangement proposed would require separate bunkers for each cell, we have considered what the bunker layout might be (fig. 16). We show an area 90 feet by 114 feet and a configuration under which the previously described weighing, proportioning, and mixing units would be arranged and the 35 pneumatic conveying systems would originate.

The selection of equipment and systems shown here takes into account the critical importance of reliable, uninterrupted feed, minimizing segregation of the mixture components, minimizing attrition of the coal, and minimizing wear of equipment.
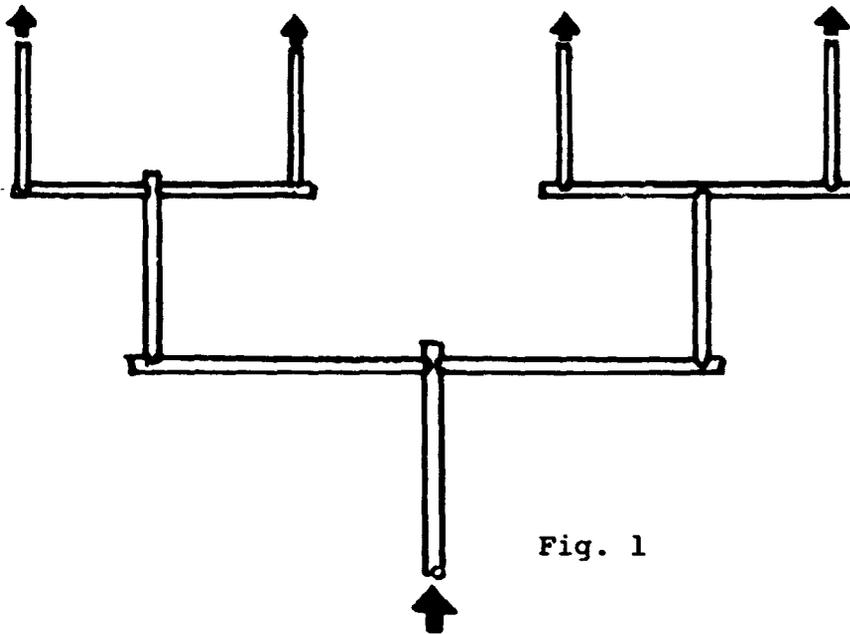
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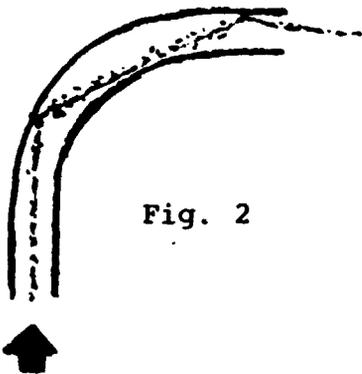
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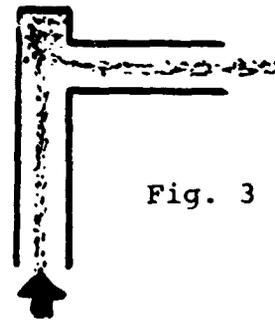
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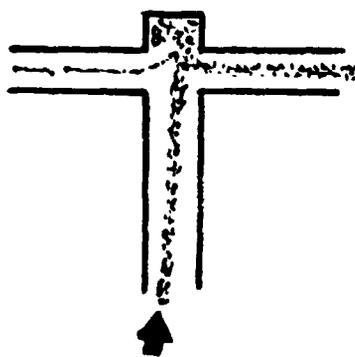
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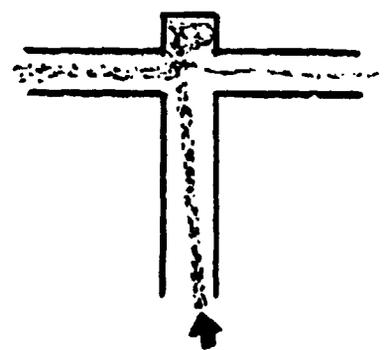
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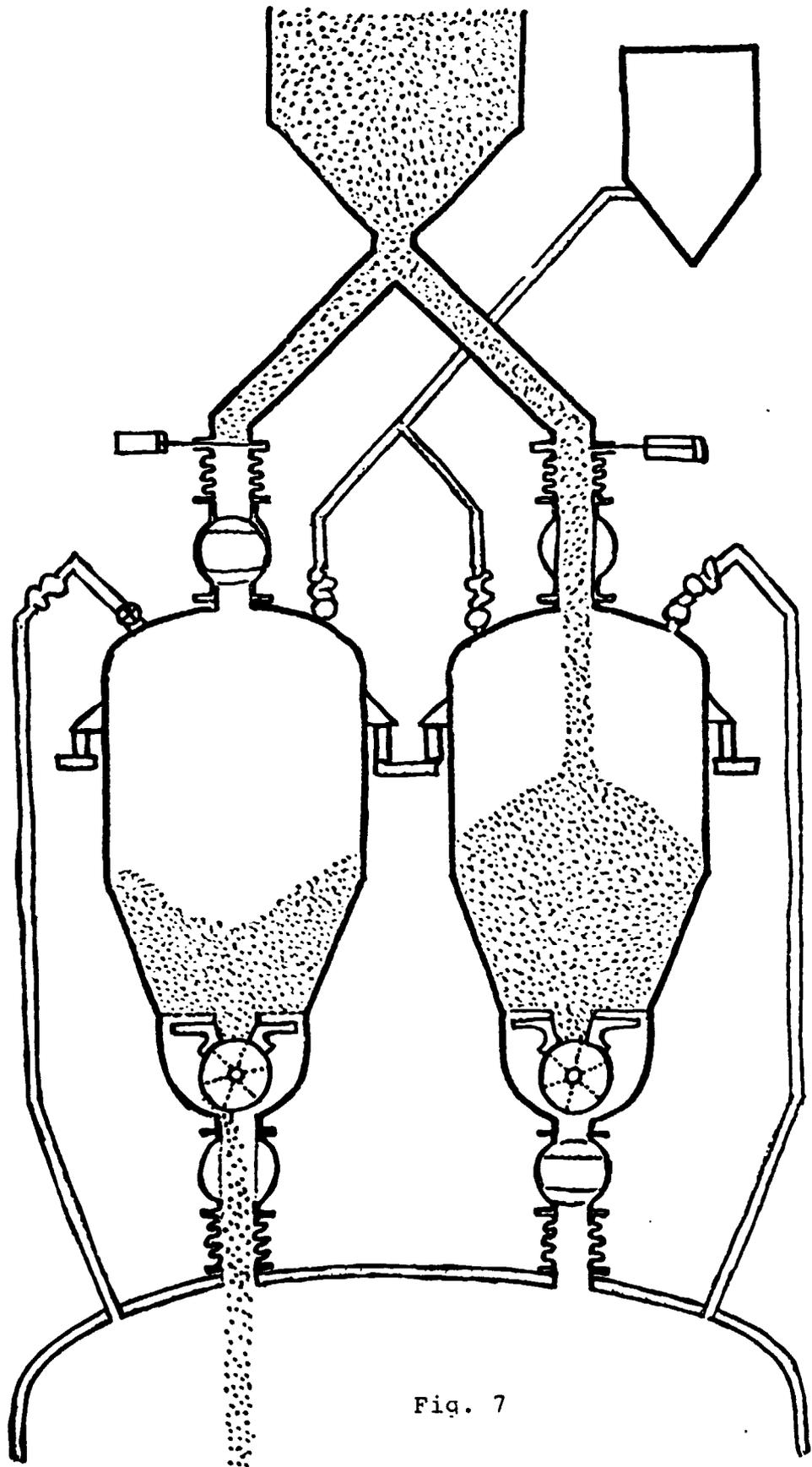
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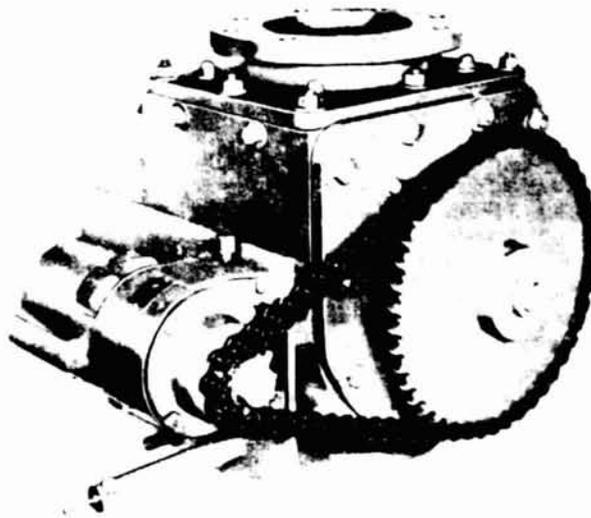
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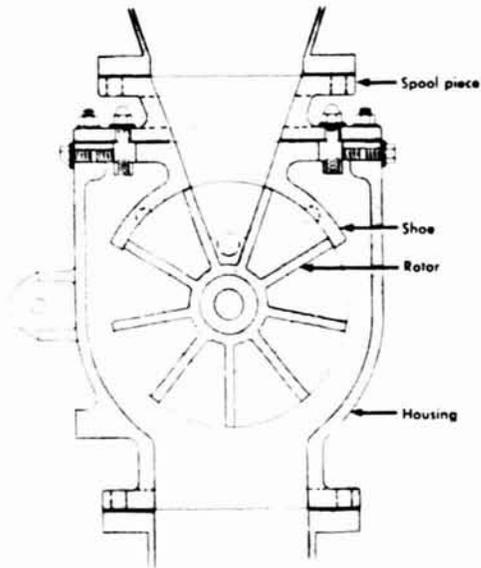
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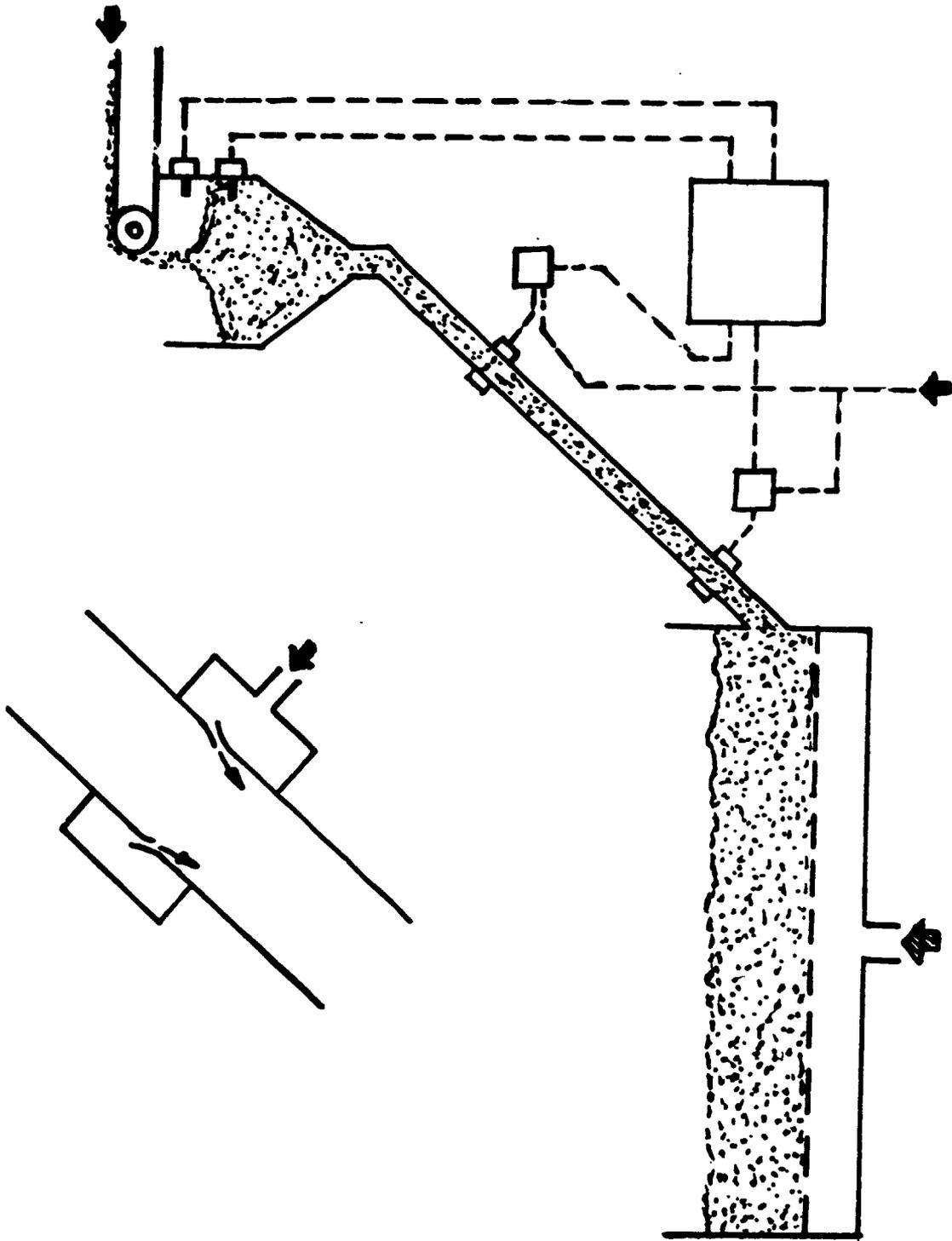
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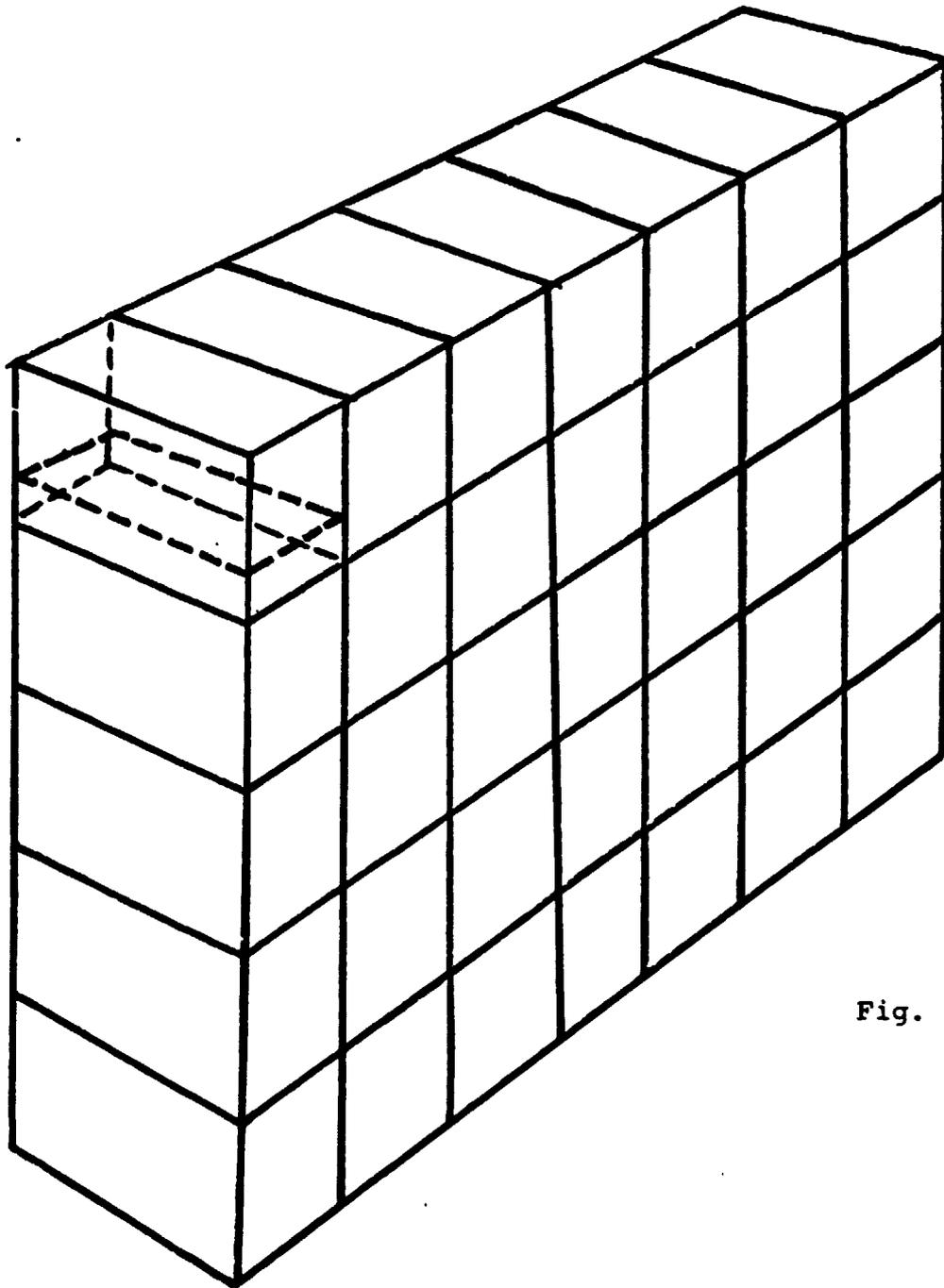
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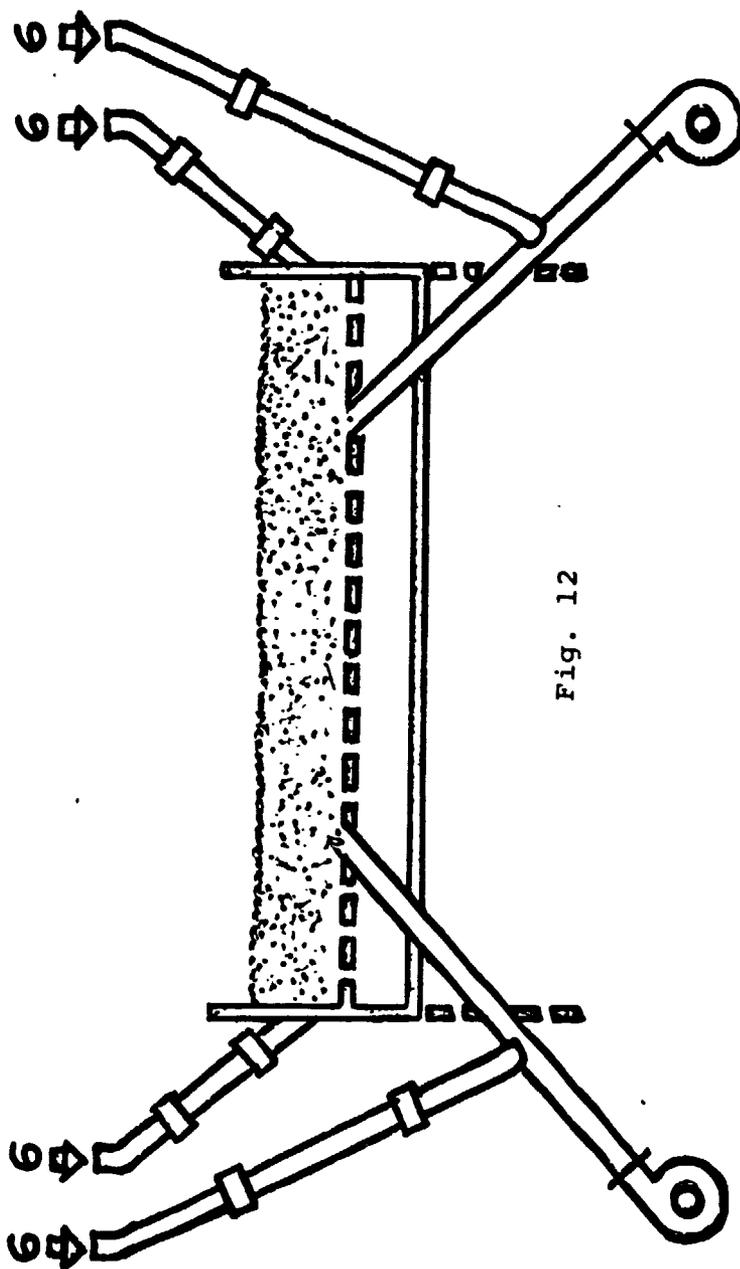
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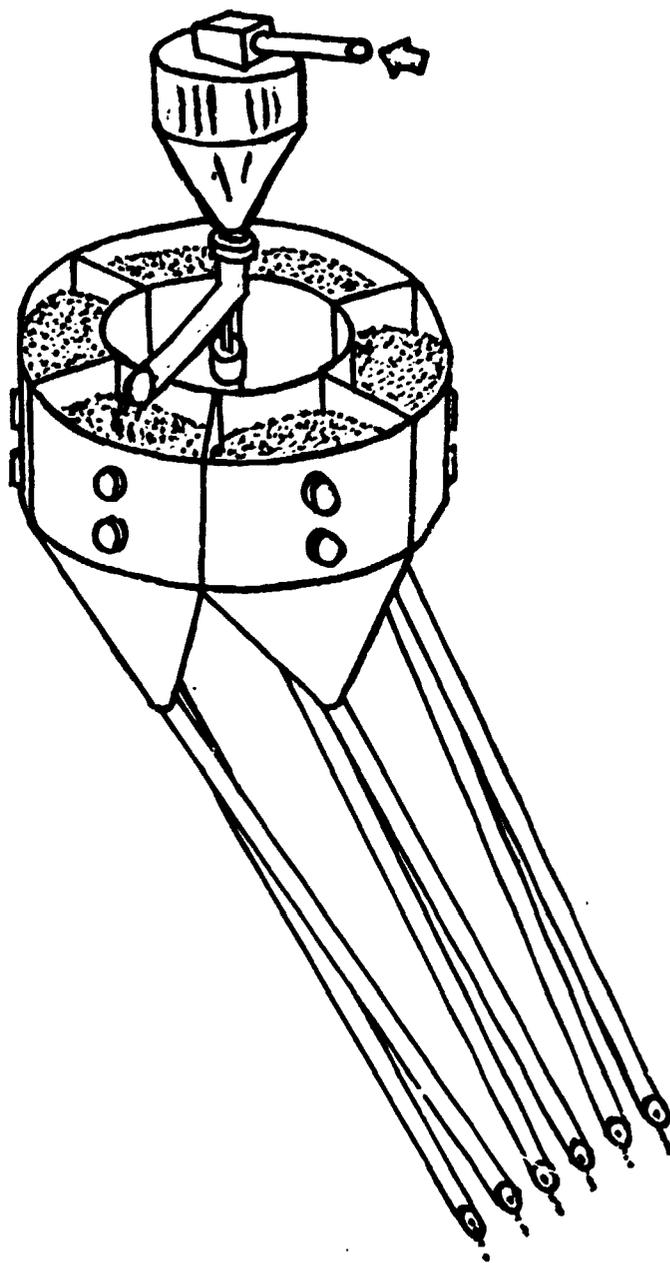
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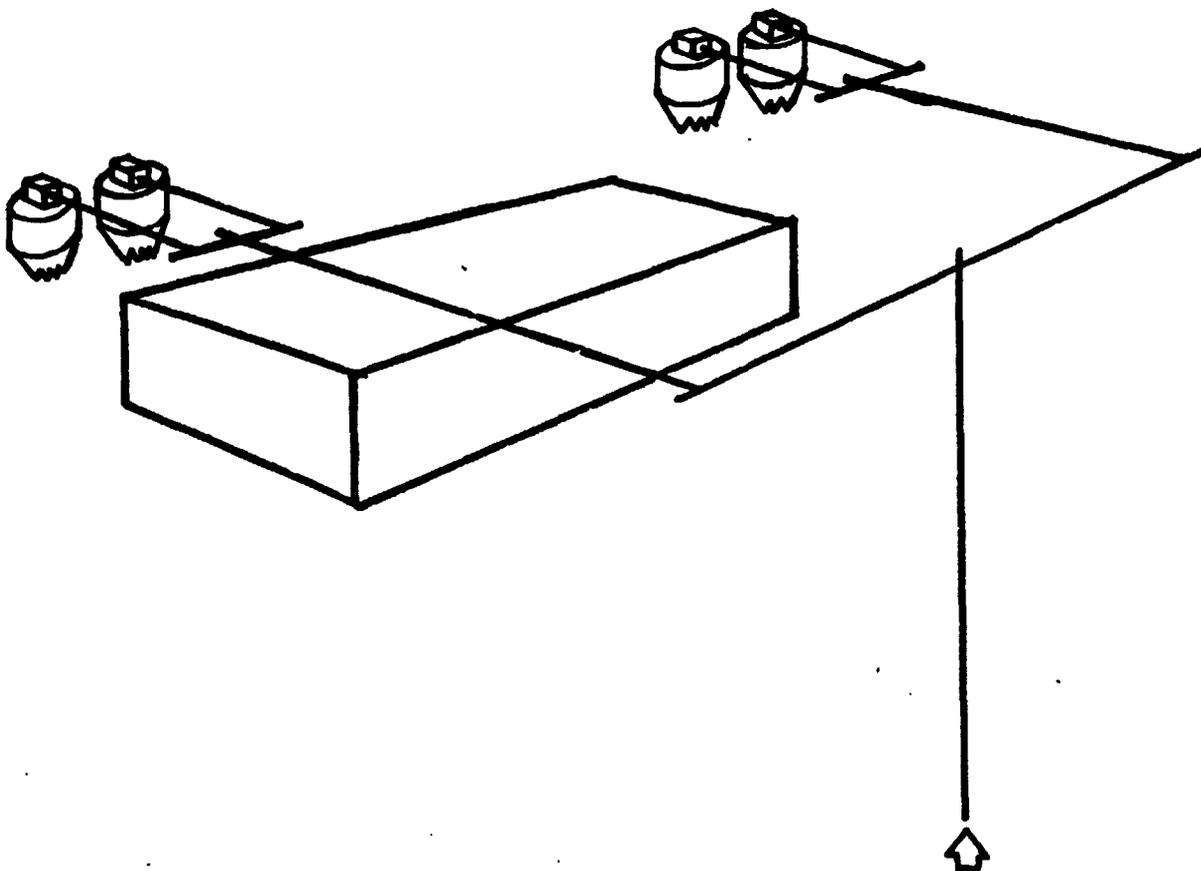
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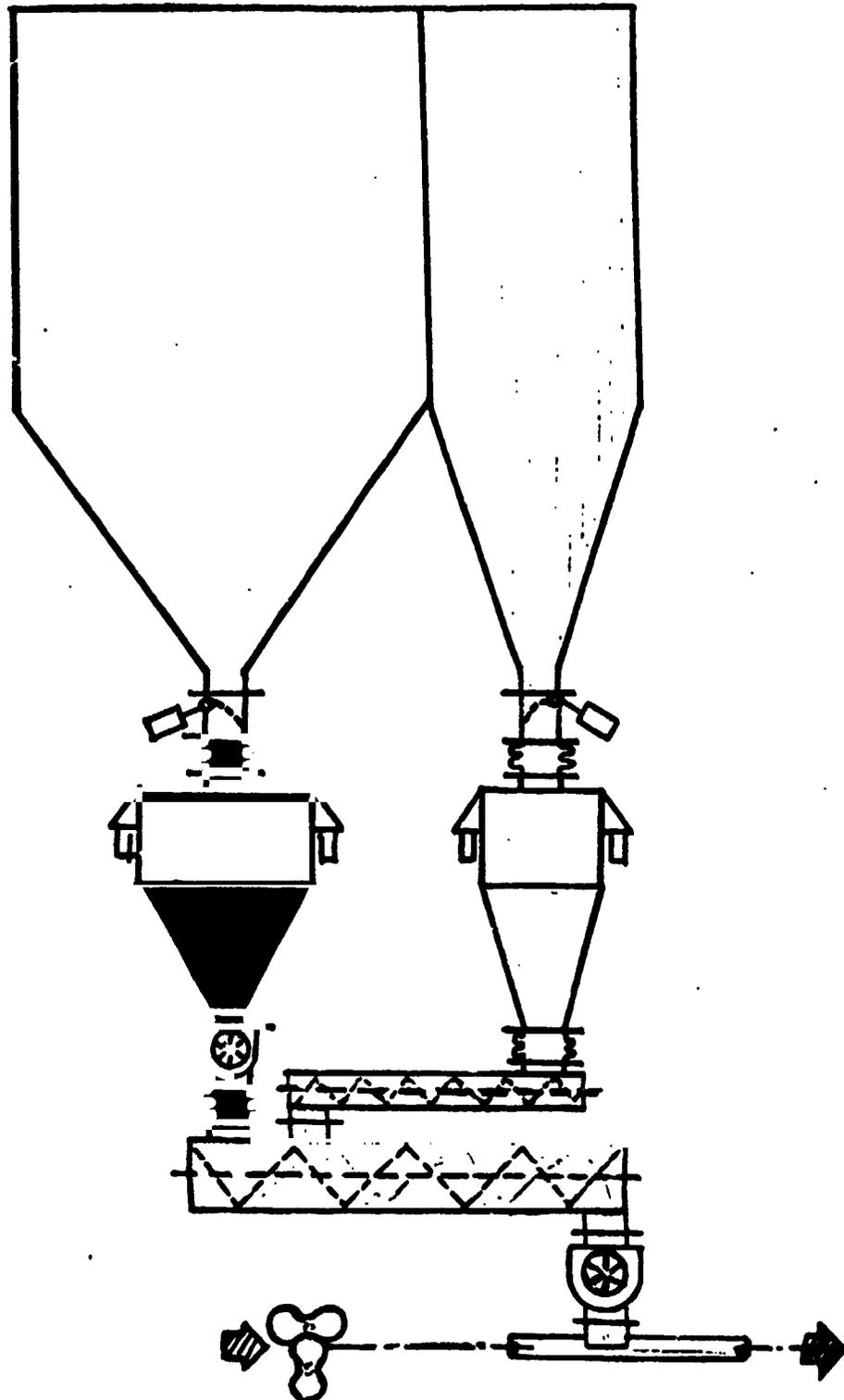
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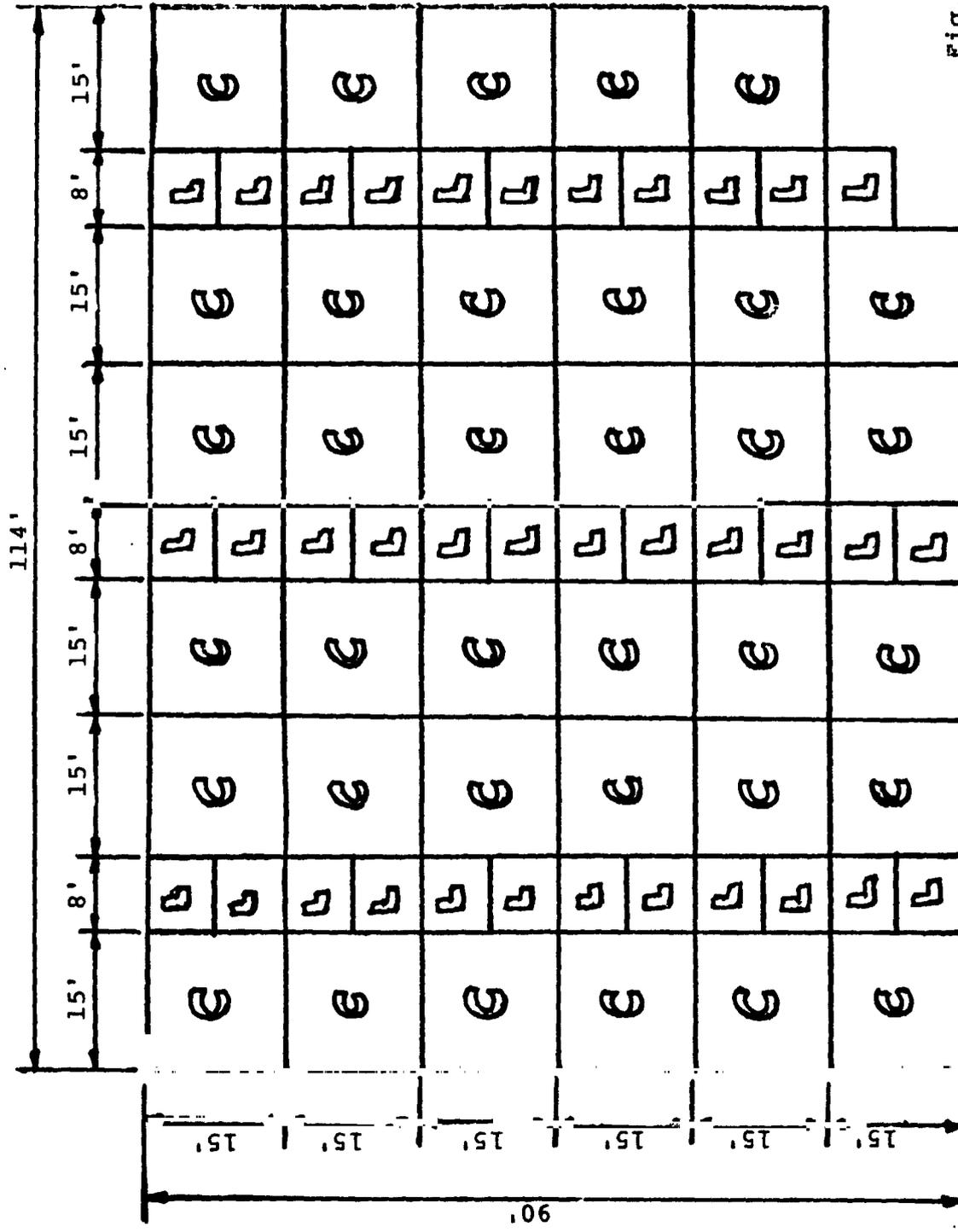
Fig. 16

SESSION V

PANEL REPORTS AND SUMMARY

77-55

P. M. YAVORSKY, ERDA

PANEL I

First of all, I want to thank the members of my panel, who submitted a lot of suggestions concerning this session. I won't be able to cover them all in five minutes, but I will try to gloss over them rapidly. People on my panel were Don Davis of Rockwell International and Mr. H. Gilman of EPRI. In general, I was very pleased with the way Session I turned out. We certainly had a lot of interest. I think the papers, each and every one of them, were most appropriate to this particular meeting. We did not have any papers that were irrelevant. Of course, I appreciate the help from the people here in organizing such a powerful session. The papers also reflect a great amount of work on the part of the authors. I certainly want to express my gratitude for that because, as I know from bitter experience, most of the time you spend writing papers is a spare time job and not something you do at work—you have to do it elsewhere. This is highly appreciated. But of course, there is always room for a little bit more improvement; no matter how well an airplane flies, you can always make a better one the next day. In that sense, let me say if there is any way of improving the presentations of the papers we had here in Session I, I would only ask for a little bit more technical detail in general from all the papers. That has to be taken with a grain of salt because, if you're restricted to 20 minutes or half an hour, you cannot present a lot of tables. But it might be worthwhile, especially, where transcripts are assembled after a particular meeting, to submit a rather detailed paper with tables, data, and photographs of critical components; and then just briefly indicate in the talk

what you have put in the paper. Later, those interested can study the paper. This is the kind of comments I received from some people, and I pass them along to you. It's not my personal viewpoint alone. I think it reflects the thinking of most technical people. Being technical people, they like to look at technical data. So if we could stress that type of input to future meetings, I think it would be appreciated.

I did detail some comments on each specific papers, but I will not go over those now unless questions come up later. Let me say they were all very good papers; there were some outstanding revelations, at least revelations to me. I was very interested in the Kamyr type feeder; it fed 4-inch coal and 10-inch coal into pipelines—this was very interesting. I think it was the highlight of the presentation. Now if that kind of device can be adapted to feeding coal into high-pressure units, it will be wonderful. If there is a way to further develop that device, I would say, it would have to be in the direction of seeing if it can be adapted to feed fine coal. It may be impossible, but it would be at least an interesting research exercise.

The practical presentations by Dick Santore of what kind of problems the feeder that they are using on their gasification unit, which is the one that Mr. Rientjes described—the Petrocarb unit—was also very interesting. He showed the kind of troubles you can have with that feeder and how they can be attacked. Same with the IGT feeder; we had some insight on the kind of further development they needed there. Certainly, the

first unit they put in for slurry feeding wasn't the best. Feeder use and concurrent development led to better and better ones. This is the normal procedure in a pilot plant development.

Here, again, I want to emphasize a statement that I have had to put forth many times. Too many people believe that when you build a pilot plant that is it, by God; you put it down on a drawing on a piece of paper, and you go in and throw a switch and everything runs the next day. Don't forget, the only reason you're building a pilot plant is that you don't know how to make it run. If you did, you wouldn't need to build it. You could go on to commercialization. Pilot plants are still development-stage devices that often contain novel units that must be perfected yet. Thus, a report on progress on overcoming operations problems is the kind of true report you like to see. Because you are putting novel units in a pilot plant, tell us what kind of trouble they have and how the problems are being attacked. We need more of that-- an honest revelation of the problems with novel units in pilot plants. Some of that came out in Session I; if anything, we need more of that. The representative from Morgantown (M. Hobday) also did a fine job in the direction of pointing out the problems they had with units they bought that were provided by commercial companies; yet still demanded considerable more upgrading and improvement before they operated adequately, especially the valve section.

Well, I think 5 minutes are just about up. So I'll cut myself off before the chairman hits me with that bell.

R. L. PHEN, JPL

PANEL II

I would like to thank my panel members, Doug Crawford and John Gardner, for their suggestions.

In summary, the work that our panel presented was the following: The ERDA Coal Feeder Development Program was initiated about three years ago, with the selection of three contractors to develop new coal feed systems. These efforts have progressed through conceptual and component design phases, in which a number of concepts were considered. We now have about four to six concepts moving into the pilot stage of the development. Significant progress has already been made in the development of these concepts, as evidenced by the data presented by our panel.

The primary criticism of our panel's efforts was directed at the presentation I made of feed system evaluation. I would like to say that we welcome criticism of our evaluation and the work that we are doing. We will be continuing to evaluate the concepts now under development by the three-contractors, and we will be assisting ERDA in evaluating any new concepts presented to them. So, if you have better ways to evaluate systems, we would like to hear about it.

There are a couple of points regarding our evaluation that I would like to clarify. They are in regard to two terms we used in our analysis, which I think were confusing in my presentation. One was a term which we labeled "R," which was a term that represented the probability of a development being successful. I said, in response to a question asked of me, that we made estimates of that function. When an engineer says he makes estimates, you have to assume that those are engineering estimates; our estimates were based upon the team of people that we had working on this project, which represents 100 years of engineering experience in working with developmental systems. The term, R, incidentally, was used primarily to weight the projected savings that a feeder system might have relative to a lockhopper system. We downgraded the developmental systems relative to lockhoppers using that term.   The term was also used in leading us to the recommendation concerning how many parallel developments we want to continue with at this point in time in our program.

The other term that was confused with the term R was "reliability." Reliability was estimated separately by Kaman Sciences through the analysis of equipment similar to the coal feeders failure rates. Those failure-rate data were used to determine

the standby equipment needed in a commercial plant, the spares required, and down-times; which we then converted to costs in our analysis. I hope that these comments may clarify our analysis. My paper for this conference will have details of the analysis described.

The results of our analysis were that four feeders had the promise of economic advantage over lockhoppers. These were the centrifugal or kinetic extruder, particularly for high-pressure pulverized coal; the rotary piston feeder, for a variety of sizes and pressures of coal; the linear pocket feeder, for a variety of sizes of coal and low-pressure processing; and the screw feeder, for a wide range of coal size and process pressures, but particularly for the higher-pressure pulverized coal.

I would like to say, too, reporting on the ERDA program which sponsored this conference, that there was much interest by ERDA to conduct this conference and to provide an exchange of information between the workers in this field. The conference itself will cause a bubbling up of new ideas, and I think we have seen a good interchange here the last few days. It also accelerates ongoing activities, and it causes existing equipment to be improved through the spirit of competition.

The future efforts of the ERDA program will be to continue the pilot scale feeder developments, begin development of feed system aspects of the effort, and begin to identify specific applications of the feeders with processes. The program strategy will include keeping open options by continuing with several feeder concept developments; we will begin to involve decision makers, the people who will be selecting the feeders in the future; we will begin demonstrating the feeders in pilot plants where we can; and the future beyond that could involve testing in a demonstration-size test facility, and in the demonstration plants themselves.

J. M. Hobday
Panel III

It's a pleasure to be in attendance at this conference. It is something I have advocated for several years and I am happy to see it a reality. JPL should be complimented for their handling of the conference. The people who have participated are to be commended for responding very quickly on a very short-scheduled conference.

Our particular session was very good. The papers were varied. I think one of the things that came out of our session was, that not everybody who has a coal feeding problem can't find a solution. There have been some solutions presented here. In some cases we may have presented solutions that don't have a problem. Adaption may be required. Coal feeding development in support of coal conversion processes may require the process to be modified slightly. This may have an effect on the overall plant we are looking at in the future. The process and feeding the process need to be considered as a system.

During our session, we had seven papers presented. Three of them were on extrusion type feeders. The presentation on the Fuller-Kenyon pump, a dry type feeder, demonstrated the experience and reliability of this piece of equipment and outlines some needs for future work. We had papers on two other types of extruders; the paper by General Electric on their extrusion work and the work done by JPL. We had a paper describing flow; techniques; the unique piston feeder from Bechtel and two papers on things that are here today and can be used. In some cases, they have direct process application and in other cases they have application where we don't get the opportunity to move into a new plant. When we can't move into a new plant, with a new design just as we would like; then we need these kinds of corrective measures

to get us a plant that will operate. Sometimes these devices get us out of trouble after we have used the best information we have and make a few mistakes. Then it's nice to have somebody who can offer a fixed solut'on where we can't back up and do it over.

I want to thank the members on my panel, Jack Smith from the Washington office of ERDA and R. Manvi, who chaired this conference, and all the speakers who participated. They did a fine job.

Since my session was the most recent, it is fresh in our minds, and the panel comments are mostly related to what wasn't in the session.

Most of the papers were in the solids flow area. There was quite a bit of interest in the audience on this and I was sorry we couldn't have more questions due to the length of the program. One of the things that is clear is that there is need for more data on solids flow such as Mr. Gardner is trying to obtain. Dr. Jenike and Dr. Carson mentioned there was some, but I believe much of that is proprietary. For those of us who are trying to build solids flow systems, it is very difficult to find design criteria and work on models without this kind of data. We suggest that data, especially on coal-limestone mixtures, be taken and published in open literature. Also, one of the things you notice when you see a picture of commercial coal handling systems is that most of them are very big, and they are big because of the low density of the coal, usually when it is fluidized.

In pressurized systems, as you know, big is bad. We consider it important to emphasize denser phase solid coal flow in pressurized systems by higher solids loading in pneumatic systems, or possibly by compacting with a Fuller Kenyon pump, or something like that. We didn't have a paper on coal flow measurements. It was scheduled but not given. Certainly there need to be better and more reliable ways of metering solids flows in all kinds of systems.

I mentioned in my comments from the Chair the problem of feedback and oscillation. This is extremely important when the feeder is at the front end of a billion dollar coal processing plant where any ripples you might have in the feed system roll through the whole plant. We would like to see attention paid to feedback and

oscillation, especially with respect to the dilute phase feeders such as the kinetic extruder. We were very pleased that Mr. Branam and Mr. Davis could talk about the operating problems of coal handling. The pictures and the concepts of the big fluidized bed combustors were quite enlightening to many of us, and the gating problems, the valving problems, and the lance problems are quite unique. The Rocketdyne people had a much longer talk prepared; we cut off about half of their slides, and they had a lot more information to give us. I hope that the complete paper will be available for the proceedings.

There is considerable interest in the wear of rotating parts. Again, we would like to see this kind of information published in the open literature. We didn't talk too much about instrumentation, but many of us who work with coal systems realize that the more you know about things, the better off you are. Many people guess at what's going on inside their pipes and inside their gasifiers and near their valves. Real data is desperately needed, and instrumentation systems themselves are needed. So we hope people will consider this area. It's an important area in solids feeding and flow.

We didn't have too much information on modeling. We had Mr. Fischer's presentation on screw feeders. Information like this, whether you believe the model or whatever, gives a lot of confidence in projecting what the equipment might look like and evaluating whether you should go to full-scale equipment or whether you should develop something and try to scale it up. We suggest additional work in the modeling area.

I would also like to thank Dr. Manvi for putting together a very wonderful program on very short order. It probably took him three or four days to organize.

PANEL REPORTS, AND SUMMARY

J. POWELL, ERDA

Gentlemen, we are approaching the conclusion of a successful
operation here at this conference. You have certainly been most generous
with your time, comments, and thoughts, and we want to thank you for that
because that's what we were seeking. Now we will summarize, give each
chairman approximately five minutes, and hope they will hold it close to
that. They will each summarize the panel activities. At the end of that,
we will have a question and answer period, and you may address your questions
to any of the panel chairmen, or if you think I can help, you can give me
a few. At the end of the question and answer period, we will close the meeting.
I want to thank each and all of you for being so attentive and cooperative.
Thank you very much.

PAPERS NOT PRESENTED

AT THE CONFERENCE

N78-13269

THE USE OF TWIN SCREW EXTRUDERS
FOR FEEDING COAL AGAINST PRESSURES
OF UP TO 1500 PSI

W. Wiedmann

Werner & Pfleiderer
Stuttgart, West Germany

W. A. Mack

Werner & Pfleiderer Corp.
Waldwick, New Jersey

To date, oil and natural gas have served most of the world's energy needs. Diminished supplies, however, mean that a return to coal will be essential to fulfill the energy needs of the coming decades. Gasification of coal provides a ready and economic solution to our immediate energy needs.

The processes for gasification of coal were first developed in the mid-thirties and successfully used throughout World War II. Since then, however, very little development was undertaken to improve production methods. Consequently, considerable work must be done to adapt the processes to today's standards.

One process step which has never been satisfactorily resolved is the feeding of the coal against the pressure in a gasification reactor. A variety of devices had been tested, however, none provided an effective commercial solution.

Single-screw extruders have been used for feeding coal, but basic disadvantages have been observed, such as: bridging of the coal in feed section, a high degree of surging in the extruder, very high energy consumption (up to 10 times higher than twin-screw equipment). In addition, it is difficult to maintain steady state pressure build-up conditions, especially with the larger diameter (single-screw) extruders required for commercial size plants.

Also, the pumping efficiency of a single-screw is dependent on tight clearances between the screw crest and barrel. As a result of the high conditioning irvolved with processing coal, these clearances are lost after very short operative time, leading to frequent shutdowns for repair. This is especially true for processes where an absolute minimum of water is needed for efficient operation of the gasification reaction.

Twin-screw mechanisms have been known to provide significantly improved coal conveying capabilities. They can also develop the required pressures for steady state conditions even with larger machine sizes. However, the initial development work performed with counter-rotation twin-screw mechanisms also produced negative results. The primary reason is that counter-rotating screws act as a grinder and change the particle size distribution of the coal considerably. As a side effect, a high amount of layering in the screws was observed.

This paper will describe recent tests with a twin-screw, co-rotating extruder which was successfully used to convey and feed coal against pressures of up to 1500 psi. Intermeshing and self-wiping, co-rotating twin-screws give greatly improved conveying and pressure built-up capabilities and avoid hangup and eventual decomposition of coal particles in the screw flights.

The conveying action of intermeshing, self-wiping, co-rotating extruder systems approaches that of a positive displacement pump. With this feature, it is possible to maintain very accurate control over all aspects of product conveyance in the extruder, i.e., intake, conveyance and pressure buildup.

In the co-rotating systems, the product is moved from one screw to the other in the form of an 8-shaped path and downstream, depending on screw pitch. Very little, if any, material passes through the clearance between the screws because the two screws have opposing directions in the area of intermesh. Very little product pressure exists that would create leak flow through this area. All of the material is, therefore, transferred from one screw to the other at each revolution. The 8-shaped path itself is a very long one, allowing ample opportunity for heat exchange between the material and the barrel walls. In effect, a great variety of different coals can be processed with little or no water addition required to achieve conveyance and pressure buildup.

The improved conveying capabilities of twin-screw mechanisms allow generation of very high extrusion pressures with a very short backup length required. For example, in a well designed twin-screw system,

extrusion pressures of 3,000 to 5,000 psi can be easily generated over a screw length of 2 to 3 D. The pressures can be adjusted without drastically affecting the throughput. This provides added processing flexibility.

The basic advantage of twin-screw mechanisms, therefore, results from superior conveying capabilities, which in turn greatly help in maintaining control over energy input into the material and retention time distribution.

TEST SET-UP

Figure 1 shows the test set-up used to convey the coal with a co-rotating twin-screw system. The coal is metered via a weigh belt feeder into the extruder. Two 58mm (screw diameter) screws with an L/D ratio of 15 were used. Water was metered into the second barrel section. This was done to provide the proper proportion of water for the reaction and also so that the water could be used as a lubricant in the process. Special kneading elements are installed at Sections 2 and 3 to intensively mix the water and coal. Two different coals were tested, lignite coal (particle size 0-6mm, 59% capillary water) see Figure 2, and bituminous coal (particle size 0-.2 mm, no water content) see Figure 3.

TEST RESULTS

In processing lignite coal with the 58 mm extruder, a throughput rate of 170 kg/hr. was achieved. Successful runs were made against 59 atmospheres of pressure. A continuous flow through a transfer pipe of 10 foot length was achieved, thus stimulating actual delivery condition in large gasification reactors. The appearance of the coal strand was very regular.

The bituminous coal was processed with up to 3 percent capillary water. The addition of the water was required to generate a gas-tight paste in the extruder. Special kneading and mixing elements were required in order to homogeneously incorporate the water into the coal. Most of the required energy for mixing was introduced by mechanical energy through the motor. The energy consumption is approximately .03 KW per Kg coal.

Figure 4 shows mechanical energy consumption versus reactor gas pressure. As expected, the energy consumption increases with rising reactor gas pressure. The water in the coal acts as a lubricant, consequently, the pressure decreases with increased water content, see Figure 5.

In addition to conveying of coal through a pipe against pressure, a series of tests were conducted to study the behavior of the coal

at discharge. To economically operate a coal gasifier, it is necessary to obtain uniform particle size distribution of the coal in the gasifier. Figure 6 shows a thermal disintegration of the coal leaving the pipe following extrusion. In order to convey against 1500 psi, the temperature of the coal must reach at least 310°C. Usually, gasification temperatures are mugh higher. This type of disintegration of coal can be applied in fluidized bed reactors.

In the tests, throughput rates of up to 250 Kg/hr. were obtained. The tests demonstrated that co-rotating, twin-screw extruders can be successfully used to convey and feed coal against pressures of up to 100 atmospheres.

Requirements for commercial coal gasification reactors are between 100 and 200 tons per hour. In the plastics industry, today's commercial size extruders have throughput rates of up to 25 metric tons an hour. In order to scale up to commercial requirements, more work must be done to study the behavior of coal conveyed against pressure. The next test series will be carried out on a 120 mm twin-screw extruder which has a throughput rate of up to 1½ metric tons an hour. In this test the extruder will be mounted directly on a gasification reactor. The results of this second phase of scale-up will provide a definite basis regarding the feasibility of a commercial size operation.
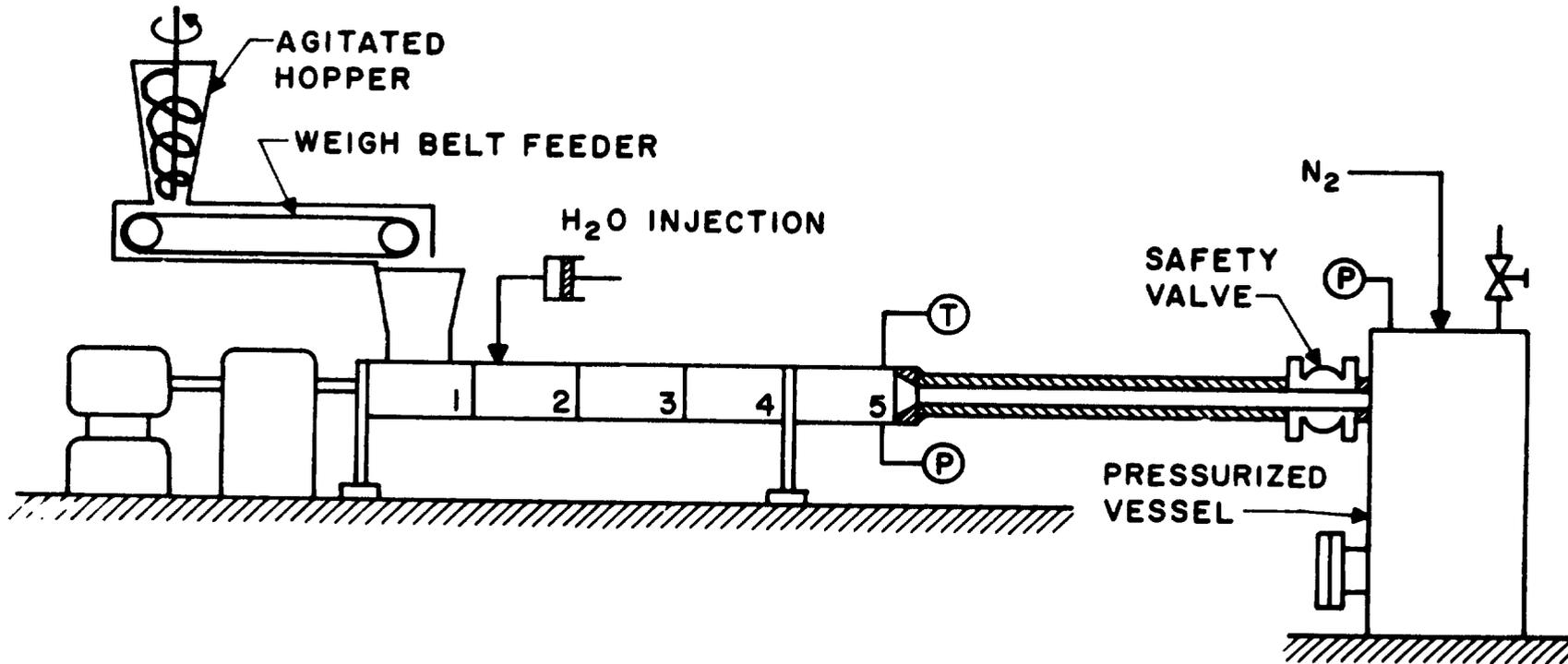
CONVEYING OF COAL AGAINST PRESSURE IN A CO-ROTATING
TWIN-SCREW EXTRUDER

FIGURE I

0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15

Figure 2
Typical lignite moisture

Mined at Hambacher Forst
Water Content   59%
Particle Size   0-6 mm
Ash Content   2%

Figure 3
Bituminous Coal

Particle Size   IKO - N: 0,2
Ash Content   3 %
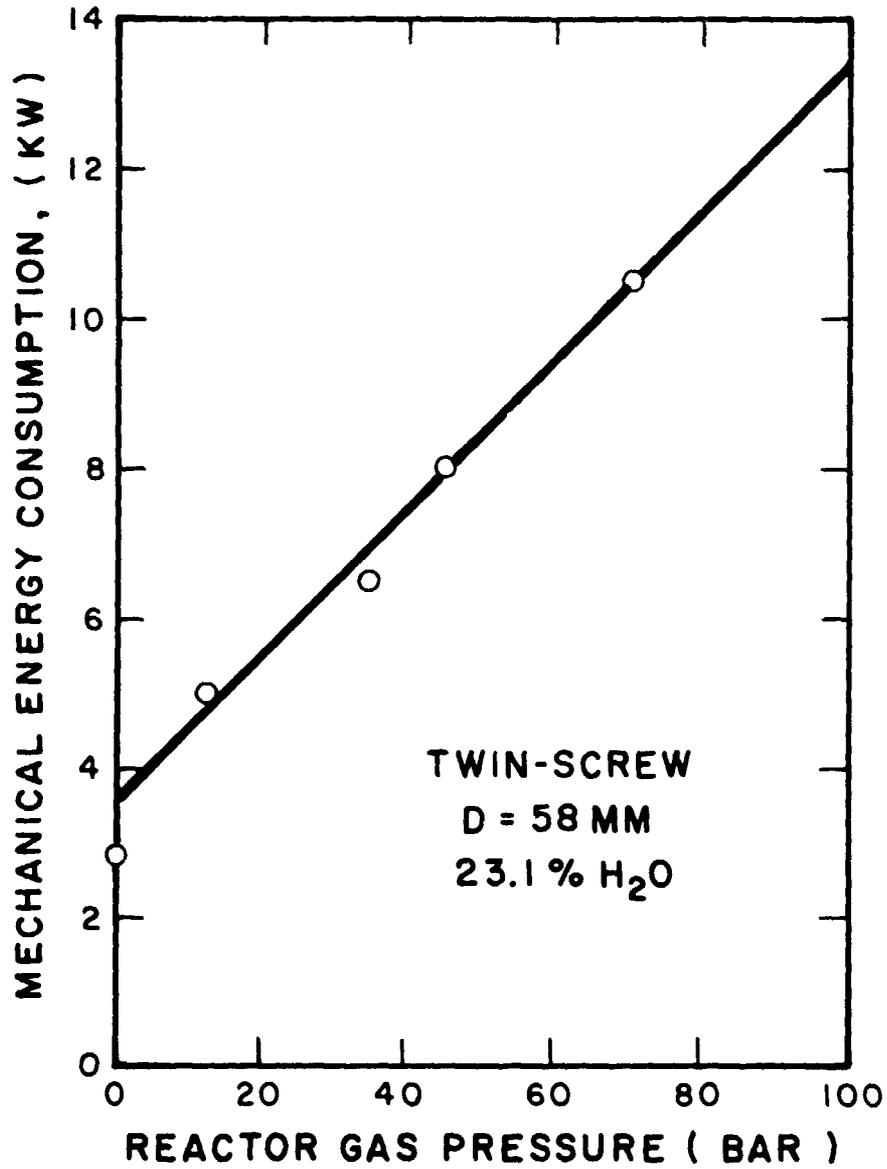
ENERGY CONSUMPTION VS REACTOR PRESSURE
FOR HIGH VOLATILE BITUMINUS COAL

FIGURE 4
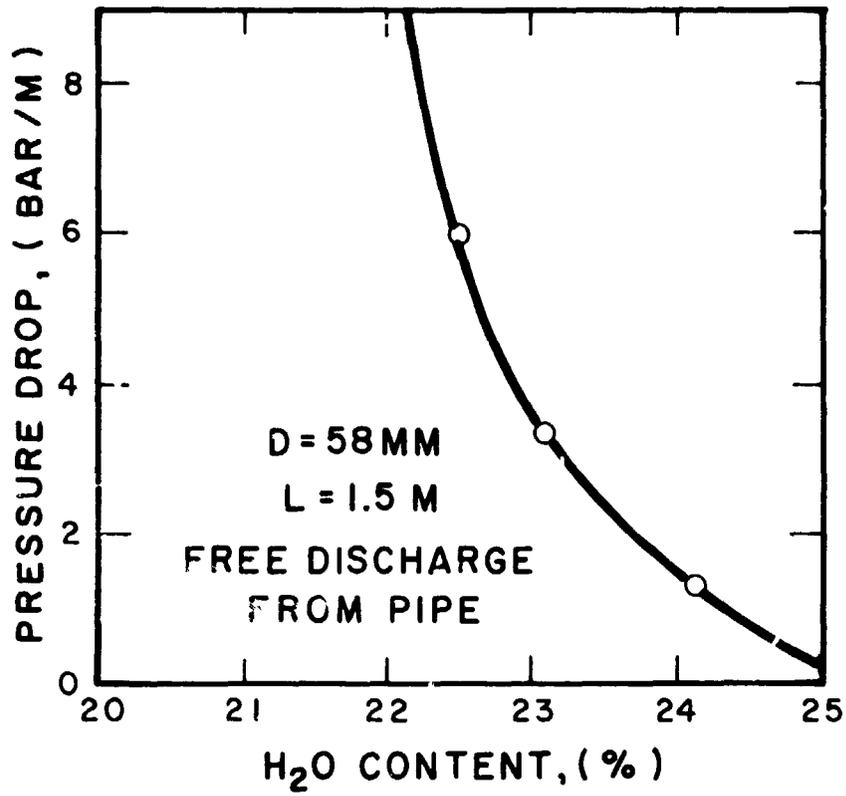
PRESSURE DROP VS H_2O CONTENT

FIGURE 5
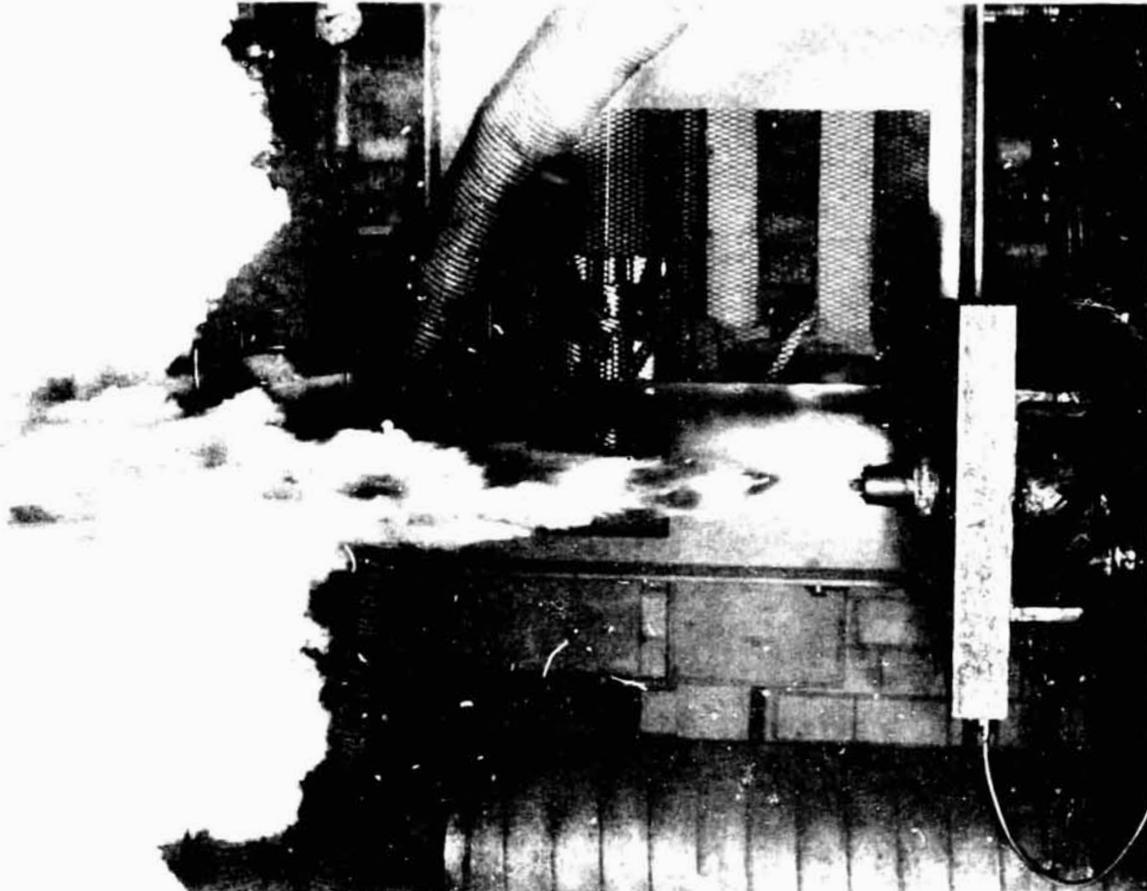
Figure 6
Disintegration of the coal and water mixture
at the discharge end of a Continua
twin-screw machine.

LOCK HOPPER VALVES FOR COAL
GASIFICATION PLANT SERVICF

E. F. Schoeneweis

Rockwell International/Flow Control Division
Pittsburgh, Pennsylvania

LOCK HOPPER VALVES FOR COAL GASIFICATION PLANT SERVICE

* *

By

E. Frederick Schoeneweis
Supervisory Development Engineer

Flow Control Division
Rockwell International
400 N. Lexington Avenue
Pittsburgh, PA 15208

* *

One complication common to most coal gasification processes presently under development or at the pilot plant stage is the injection of coal solids into a reactor vessel which is under significant pressure (generally in the range of 400 to 1600 psi or 28 to 110 bar). Similarly, spent solids, some of which are valuable by-products of the conversion process, must be removed from the high-pressure reactor vessel to some lower pressure for further processing or disposal Among the variety of schemes presently used for accomplishment of these tasks, the lock hopper concept is probably the most widely accepted and simplest form

Although details and equipment requirements of the lock hopper concept may differ from plant to plant, the operating principle generally involves use of pressure equalization and gravity movement of the solids in question A schematic showing one example of the operating principle is in Figure 1

Coal, or other reaction material, is loaded into a holding bin, typically referred to as a weigh hopper, at atmospheric pressure by conveyor or other transport means Valves A and B are in the closed position and the lock hopper between the valves is at atmospheric pressure Valve A is opened and the solids fall from the weigh hopper to the lock hopper Upon depletion of the solids from the weigh hopper, Valve A is closed and the lock hopper is pressurized to the reactor vessel pressure using inert gas Valve B is opened and the solids flow into the reactor vessel Valve B is closed and the lock hopper is depressurized for the subsequent feed cycle Cycling continues according to the charging requirements of the reaction process

Removal of spent solids from the reactor vessel follows a reversal of the feed cycle With Valves C and D closed, the lock hopper between them is pressurized to the reactor level Valve C is opened and the spent solids fall from the reactor to the lock hopper Valve C is closed and the lock hopper is depressurized The spent solids are exhausted to atmosphere or a transport system by opening Valve D Typical systems involve some type of quenching of the char before the lock hopper valves, but char discharge service inherently involves higher temperatures than coal feeding It may be understood that there are many detail variations in design of lock hopper systems Common features are the need for valves which must withstand high pressures when closed but must operate (open

and close) only at a zero or relatively low differential pressure In each valve the flow of solids is ir one direction only, but .t is the direction of increasing pressure in coal feed valves while in the direction of decreasing ,.ressure in char discharge valves.

Although the operating principle of the lock hopper system can be seen to be extremely simple, valve applicatio·s involving this service for coal gasification plants are likewise extremely difficu't. The difficulties center on the requirem.:.it of handling highly erosive pulverized coal or char (either in dry or slurry form) combined with the requirement of providing tight sealing against high-pressure (possibly very hot) gas. Operating pressures and temperatures in these applications typically range up to 160ú psi (110 bar) and 600F (316C), with certain process requirements going even higher. In addition, and of primary concern, is the need for reliable operation over long service periods with the provision for practical and economical maintenance. Currently available data indicate the requirement for something in the order of 20.000 to 30,000 open-close cycles per year and a desire to operate at least that long without valve failure.

Rockwell Approach

The Rockwell International interest in new valve developments for coal gasification plants was stirred in the early part of 1972 by inquiries concerning valves for coal and char lock hopper service. Since these inquiries involved applications in relatively small pilot plants, many of the requirements were found to have possible satisfaction by incorporation of customizing features into existing product lines. However, when the size range of valves required for full scale plants was recognized, it became clear that many valves considered suitable for pilot plants would not scale up practically. Consequently, evaluation of the fundamental needs for full scale plant valves was undertaken.

It was recognized that certain fundamental elements of similarity existed between problems involved in this application and those encountered in the development of primary coolant valves for the Fast Flux Test Facility (FFTF). The hostile internal environment (molten sodium) inspired a new concept in gate valve design which provided for rotary transport of the closure element to and from the seat to minimize the need for sliding of functional components against one another. These valves, like the lock hopper valves, have the requirement to provide a clear through-port in the open posi-

COAL SUPPLY

ATMOSPHERIC PRESSURE | WEIGH HOPPER

VALVE A

COAL LOCK HOPPER

VALVE B

REACTOR VESSEL

HIGH PRESSURE

VALVE C

CHAR LOCK HOPPER

VALVE D

ATMOSPHERIC PRESSURE

Figure 1:
Schematic of Lock Hopper system.

tion and tight sealing at high temperatures in the closed position.,

It was believed that operating requirements for high-pressure lock hopper valves would yield more readily to this type of approach than to any attempt at adaptation of conventional valve concepts which are generally characterized by their reliance on substantial relative motion of contacting or closely proximate operating parts.

Basic Concepts

Basic concepts in Rockwell s lock hopper valve development program had their beginning with the approach shown in Figure 2 Problem solving efforts were focused primarily on the pulverized coal feed system, in which flow of material is in the direction of gate opening and applied pressure differential is in the direction of gate closing Valving for other systems in the gasification process, in which material flow and pressure differential are in the same direction, were de-emphasized temporarily with the view that their solution hinged upon this primary effort.

It should be pointed out that certain practical considerations are inherent in the Rockwell concept approach Opening of the valve is assumed to occur with zero or very low gate differential pressure. Closing of the valve is assumed to occur in the absence of falling coal. Neither of these requirements are believed to pose serious system design problems The latter, which is possibly the more difficult to achieve, can generally be handled with proper valve sequencing. For example, if Valve A in Figure 1 is closed only after the weigh hopper is empty, there is no problem. However, assurance of meeting this requirement can be accomplished by inclusion of a "dirty valve" preceding the lock hopper valve for the purpose of shutting off the solids stream. This component could be of relatively simple design, since it would not be subject to differential seat pressure.

Recognizing that the high cycle life demands on the valve require extreme attention to all areas of potential wear, the reference design attempted to illustrate in a simplified ma. ·ier, methods of possible solution. The potential wear areas can be considered in three categories: (1) seating surfaces; (2) internal closure element transport mechanism, and (3) stem seal surfaces.

To protect seating surfaces, the reference design provides straight lifting and lowering of the gate from and to the seat. Additional protection
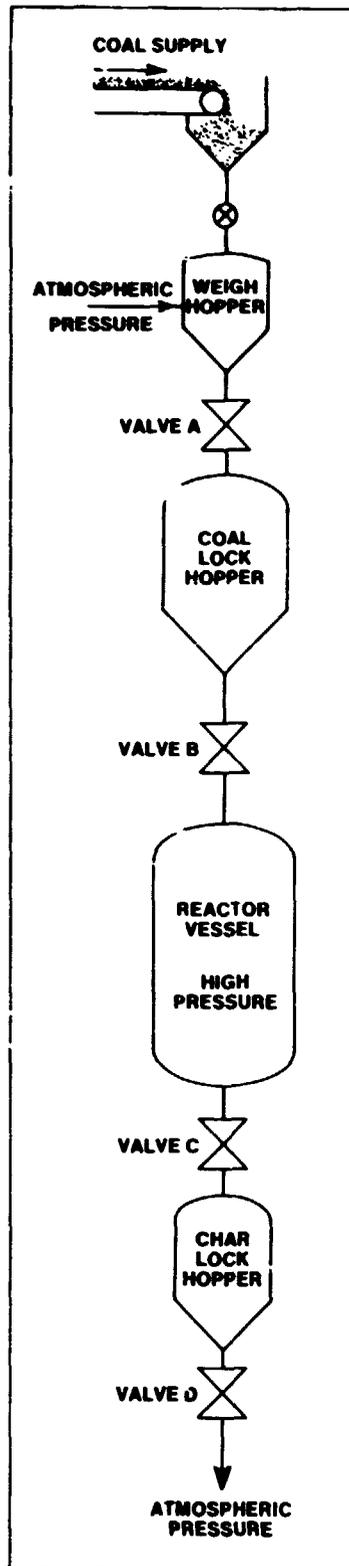
against damage is supplied with the provision for a clean gas purge of the local seating surfaces during closure prior to the instant of contact. Possibilities of local distortion from thermal expansion or mechanical loadings are minimized by use of maximum center-line symmetry in the design of both the seat and gate. The principal seating force is provided by contained pressure and by local design of the seating surface, therefore, optimization of seat contact stress is possible for every size and pressure.

To provide a clear through-port in the open position, it is necessary to transport the closure element from its seated position to an alternate position outside of the flow stream. By use of rotation rather than translation, the reference concept minimizes the force-distance parameter of the required bearing surfaces, and makes it convenient to provide substantial protection against the contamination of the bearing interface by coal or char particles. The provision for positive clean gas purging of the journal bearings maximizes the effectiveness of this protection.

Since sealing force is provided primarily by the pressure differential on the gate, mechanical loading on the journal bearings is limited basically to the weight of the hinged parts, plus a moderate margin to insure tight seating when differential pressure is low. This makes it possible to provide for generously proportioned bearing surfaces, with relatively low contact loading. Since service temperatures

preclude the use of organic lubricants, a dry bearing of suitable material was contemplated to provide adequate wear life.

In the matter of stem seals, it is apparent that the problem of preserving a satisfactory seal over an extreme cycle life can be more easily handled with a fractional turn rotating stem than with a long-travel translating stem. Rockwell's experience and comprehensive experimental programs involving performance of packing materials at elevated temperatures provided confidence in obtaining a solution to this problem area.

A key observation, with regard to the reference design of Figure 2, is that the concept lends itself to quantifiable scaling between prototypical small valves and larger valves, which may ultimately be used in full scale coal gasification plants. It follows, therefore, that experience gained with any reasonably-sized valve in a pilot plant application could be extrapolated to apply to much larger valves in full scale plants.

IGT Valve Test Program

During the latter part of 1973, Rockwell proposed a cooperative valve test program with the Institute of Gas Technology (IGT) in Chicago, Illinois, to demonstrate the working principles involved in the above conceptual approach to coal gasification lock hopper valving.

IGT is a research and educational organization affiliated with the Illinois Institute of Technology. It specializes

in development of energy systems, economic evaluation of comparative systems, and dissemination of this information to a diversified group of industry, international and government clients. IGT has one of the largest total programs underway in the world for conversion of coal to synthetic pipeline gas via three alternate HYGAS processes.

An agreement was reached to test two size 3 "scale model" valves in a simulated coal feed lock hopper system operating between atmospheric and 1000 psig (69.0 bar) pressures and ambient temperature. Rockwell agreed to furnish the two valves mounted in a lock hopper simulator complete with controls for sequencing the operation and pressure cycling. IGT agreed to furnish a coal feed mechanism to the lock hopper entrance valve, the pressurizing medium source, a dummy reactor vessel and technicians for operating and monitoring the tests. The IGT test program was viewed as an important "ground breaking" effort in providing valuable technical and marketing support in the event of future demands for lock hopper valves.

Work was begun immediately in an effort to deliver and have the valves ready for testing in the midpart of 1974. The valves were designed and detailed at the Rockwell Valve Engineering and Research Headquarters in Pittsburgh and fabricated at its Kearney, Nebraska, plant. While the valves were being fabricated, the bearing/seal arrangement utilized in the valves was cycle tested in a laboratory mockup under simulated operating conditions, and the control system for the lock hopper simulator was designed and fabricated. Upon valve delivery in Pittsburgh, the lock hopper simulator was completed and the operating system debugged. The assembly was then subjected to operational tests on coal at IGT which were closely monitored by IGT and Rockwell personnel.

From Concept to Design

The size 3 prototype lock hopper valve design shown in Figure 3 was the result of an engineering study based on foreseen requirements of future full scale coal conversion plants in which typical lock hopper valves may be required in size 24 or larger. Basic features of the valve were developed from the "reference" design discussed earlier. Essentially, the size 3 valve is a "scale model" of a larger valve. Deviations from the "scale model" ideal were necessary in the valve body closure area because of the practical necessity of handling internal parts manually at assembly, however, all functional
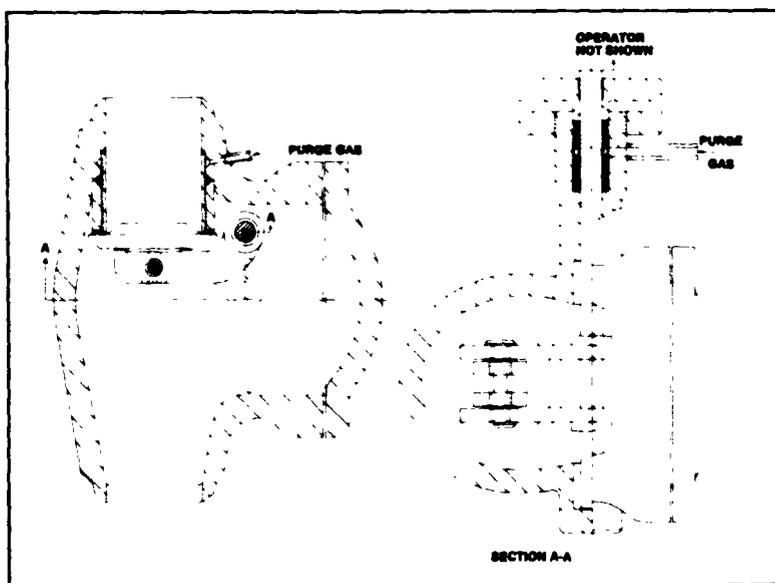
Figure 2:
Rockwell conceptual approach to Lock Hopper valving.

parts were to scale.

No specifications were issued for the valve other than requirements of size and ANSI rating of the body and end flanges Class 900 was designated for conservatism because of the abrasive internal environment and the long term pressure cycling to which the valves would be subject. Actual operating conditions for the valve called for pressures of atmospheric and 1000 psig (69.0 bar) and ambient temperature

The materials of the valve and trim construction are in most instances carbon steel. A 17-4 PH stainless steel was selected for the stem to utilize its high yield strength in obtaining a smaller diameter in addition to its corrosion resistance Selection of 316 stainless steel for the stuffing box spacer and lantern rings was also based on corrosion resistance qualities. Cobalt-based stellite hardfacing was chosen for the seating areas as well as the gate/pin bearing because of its high strength. wear resistance and machinability

Operating requirements imposed on actual plant valves were of primary importance in selection of stem seal and bearing materials A carbon-graphite material was selected for the bearings because of its high-temperature resistance. self-lubricating properties, low coefficient of friction, and good load carrying qualities. In a similar manner, an all-graphite material was chosen for the stem packing because of its wear qualities. high temperature range, and exceptional sealing characteristics

Each valve was furnished with a pneumatic, quarter-turn operator. The only direct force acting against opening or closing the valve is friction developed by the stem packing and bearings and inherent friction in the operator itself. The valve was designed to open
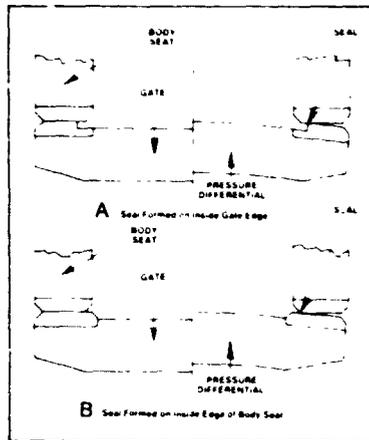
with no pressure differential across the gate and close in the absence of falling coal. In order to achieve initial gate sealing at pressures below the design level, the gate is loaded through the operator to achieve a predetermined seating force. Through proper design of operator control valving, quick opening was accomplished so that gate contact with falling coal was essentially eliminated. Slow closing was employed to insure proper timing for the seat purge cycle and to avoid unnecessary impacting of the seating surfaces

The basic seating arrangement employed in the valve is a "flat" seat. This feature, more than any other, was a source of questioning and discussion by proponents of a "line" contact seal. It is concurred that a line seal would be more gas-tight in a clean gas medium or where only a few sealing/unsealing cycles are required (as in a gasket); however, durability of such a seal in a frequently operated valve in an extremely dirty environment is question-

able As designed, the sealing surface incorporated in the valve does in fact tend towards line contact under elastic deflection of the gate upon application of pressure This type of seal has been demonstrated to be more resistant to degradation in service than a seal depending on an initially highly stressed "knife edge."

Two different arrangements were employed in the seat design as shown in Figure 4. Figure 4A produces an immediately higher stressed seating by the reduction in area In operation, the seal is formed on the inside edge of the gate seat Figure 4B has slightly more seating area and thus produces a lesser contact stress and sealing is accomplished on the inside edge of the body seat Both arrangements exhibited good sealing qualities during preoperational and operating tests Tests on air at 1000 psig (69.0 bar) gave leakage

rates of well under 1 scfh (2.83 x 10^2 scmh) on the "as built" valves

An important feature contributing to success of the valve seating arrangement is the method of attachment of the gate to the pivot arm. The gate is provided 360° of freedom in the plane of seating This is accomplished through the use of a modified ball and socket joint which permits the gate to "find" the body seat in the event of misalignment.

The stuffing box arrangement of the valve is somewhat unconventional and deserves attention At first glance, cost savings appear possible by deadending the stem internally rather than providing a seal arrangement for both ends. However, symmetry in body design was considered as utmost in importance in preventing unbalanced elastic and thermal deformations during valve operation In addition, the design produces a pressure balanced stem and eliminates the need for lock-

ing devices and thrust bearings. Further, the symmetrical body affords the possibility that either "right hand" or "left hand" operators might be furnished as a convenience in production valves.
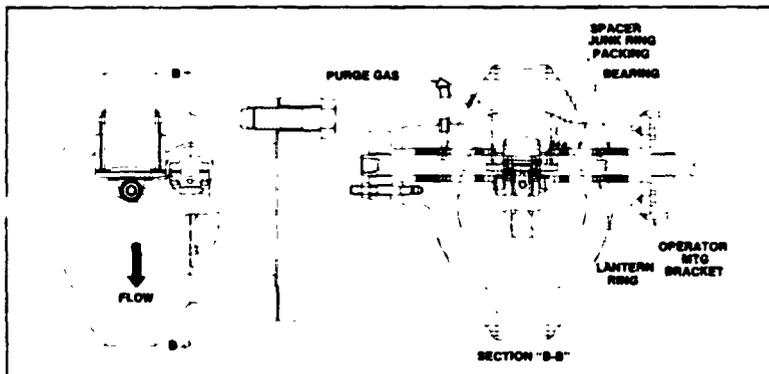
Two internal packing rings act as dirt seals for the bearings. The bearing compartment between the divided packing arrangement is pressurized at some level higher than the valve internal pressure. Any leakage that may occur past the dirt seal packing, therefore, is towards the inside of the valve which further acts in keeping the bearings clean. Junk rings, like the bearings, are a carbon-graphite composition and spacer rings are provided to eliminate shear loads on the junk rings.

Bearing/Seal Testing

Prior to complete valve fabrication, the effectiveness and durability of the stem bearing and seal design was tested in the laboratory mockup configuration shown in Figure 5. The arrangement and dimensions of all working parts were made identical to those used in the actual valve.

Pulverized coal (acquired from the U.S. Bureau of Mines at Bruceton, Pennsylvania—presently ERDA) was loaded into the fixture and kept "stirred-up" during cycling by a paddle arm arrangement to simulate an operating plant valve internal environment. Torque and bending loads were imposed on the stem through the oper-

ator and a stop which engaged the paddle arm at a location equivalent to the moment arm distance between the stem and body seat of the actual valve. Loads applied were equivalent to the foreseen operating loads of the size 3 valve.

A 20,000 cycle test was performed on the mockup. External gland leakage throughout the test was negligible, and the bearings remained free of coal dust particles and showed no evidence of developing excess clearances. Final inspection of the stem, packing and bearings revealed no unusual signs of wear or damage. The test verified the effectiveness of the bearing purge in keeping the bearings clean of coal particles and demonstrated the durability of the bearing arrangement and stem packing over a long service period of cyclic loading.

Lock Hopper Simulator

The lock hopper simulator consisted of the two size 3 valves mounted in series with the space between the two valves being used to represent a lock hopper at the coal feed side of a typical gasification plant. The two valves were operated and sequenced through a pneumatic control system which was part of the total assembly supplied by Rockwell. The simulator was designed for installation between a dry coal feed unit operating at atmospheric pressure and a dummy reactor collection vessel operating at a constant 1000 psig (69.0 bar). These latter

units were supplied by IGT. Coal travel was accomplished by gravity feed and the valves were installed with their flow passages vertical.

Figure 6 shows valve working pressures as a function of lock hopper cycle events. Body pressure of the top valve (Valve "A") fluctuated from atmospheric to 1000 psig (69 0 bar) which was the reactor pressure. Bearing purge pressure of Valve "A" always remained higher than the body pressure. Seat purge pressure remained at atmospheric except during Valve "A" closing.

Body pressure of the bottom valve (Valve "B") remained constant at the reactor level (1000 psig or 69.0 bar). Bearing purge pressure of Valve "B" remained constant at a predetermined level above the reactor pressure. Seat purge pressure followed the body pressure of Valve "A", except during Valve "B" closing.

Basically, operation of the system was quite simple. While the top valve was at atmospheric pressure, it could be opened to allow coal to enter the lock hopper compartment. Upon closing, the compartment was pressurized to the same level as the reactor vessel. The lock hopper outlet valve could then be opened to allow coal to enter the simulated reactor vessel. Opening and closing of both valves was, therefore, accomplished in the absence of gate pressure differential.

Figures 7 and 8 show the lock hopper simulator which was built by Rockwell and tested at IGT.

Functional Testing

During the latter part of 1974, final installation and initial dry run cycling of the assembly described above were performed at IGT. Although the control system had been previously operated and adjusted at Rockwell's Pittsburgh location, additional runs were necessary to insure proper system sequencing and to train personnel on operation procedure. More than 600 cycles were made on the assembly at design pressures but without coal. Some problems were intermittently experienced with system hardware, but these were generally minor in nature and easily corrected.

The assembly was subjected to cycling with coal over a period of several months. Operations generally proceeded eight hours a day, five days a week, with interspersed downtimes for leak rate testing, system hardware adjustment, and various other reasons generally unrelated to the test valve performance.

Figure 6
Valve working pressures as a function of Lock Hopper valve cycle events

Figure 7:
Control panel view of the Lock Hopper simulator ready for shipment to IGT.

Valve seat leakage tests were accomplished in the following manner. All flow passages upstream (to the direction of coal flow) of the seat to be leak tested were sealed to the atmosphere, and a flow meter was connected directly to the area above the seat through the seat purge port. Valve body pressure was increased slowly while leak measurement readings from the flow meter were recorded. Data was collected at various pressure levels up to the 1000 psig (69.0 bar) design pressure.

Performance of both valves during the functional tests with coal were closely monitored. Operation was continuou y smooth and trouble-free. Stem packing and seat leakage tests were made at prescribed intervals with results which were very encouraging. Stem leakage remained negligible and packing gland adjustment was required only once—on the top valve at the operator side. Bottom valve seat leakage remained constantly moderate and top valve seat leakage was undetectable ·r most of the test span. After approximately 3600 coal feed cycles, bottom valve seat leakage was measured at 21.8 scfh (0.62 scmh) at 1000 psig (69.0 bar) seat differential.

The test was stopped after 4400 coal feed cycles (actual operating cycles on each valve exceeded 5000) when the seat leakage rate of the lower valve (valve at entrance to reactor simulator) became excessive (over 200 scfh or 5.7 scmh). The higher seat leakage rate of this valve was later determined caused by an inadvertent error
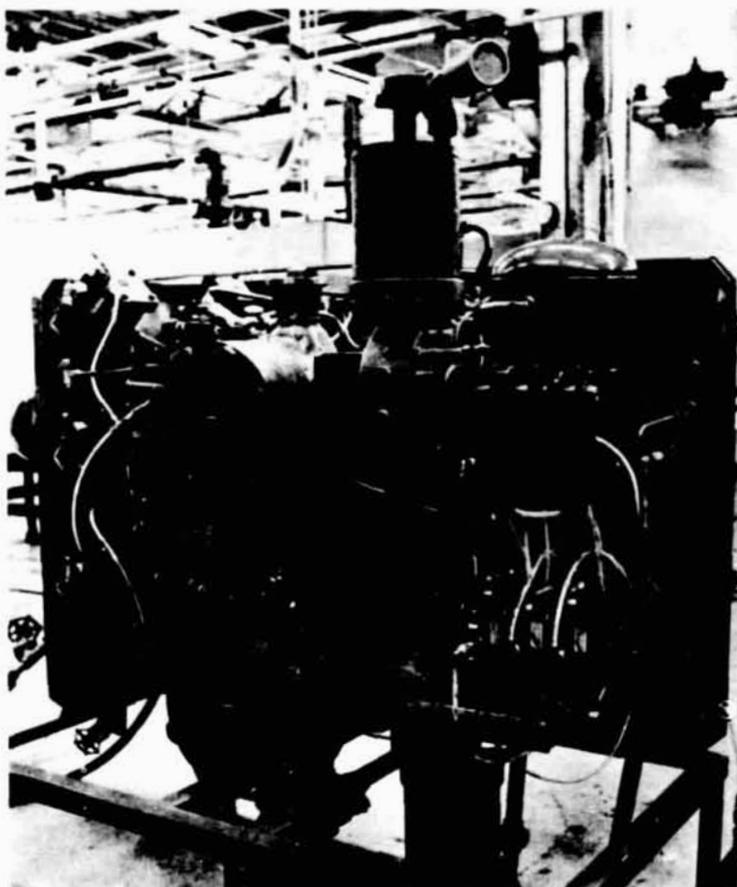
Figure 8:
Control system view of the Lock Hopper simulator showing the two valves used in IGT tests.
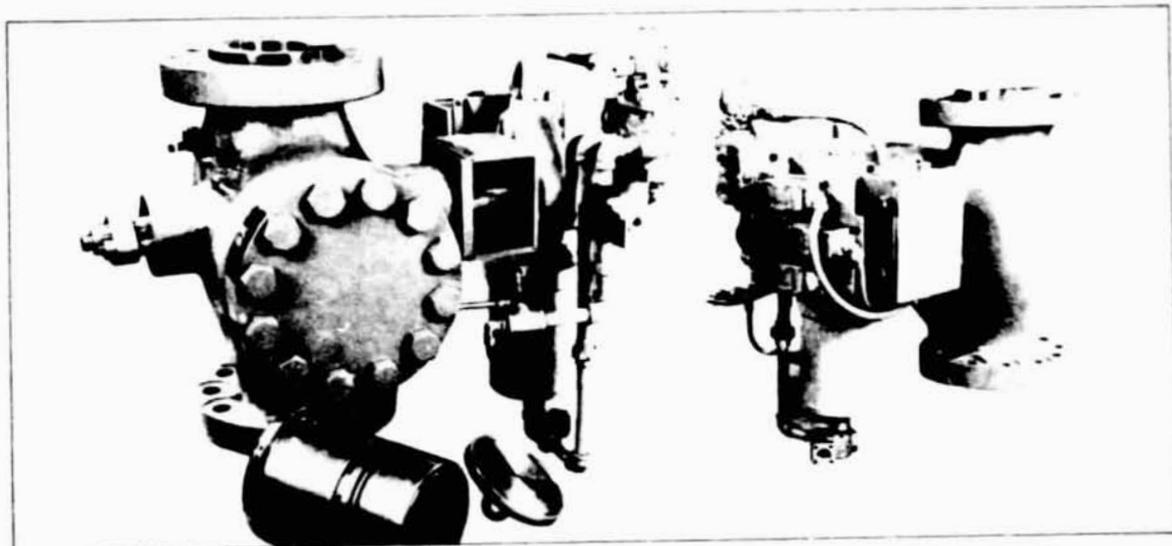
Figure 9.
Size 6, Class 900 Lock Hopper valves.

in setting the seat purge pressure which allowed coal particles to become trapped between the seating surfaces at the time of the leak test.

The valves were disassembled for seat and gate inspection. (This was readily accomplished by access through the valve maintenance cover.) The seating areas, particularly of the top valve, contained numerous indentations and wear areas. The surface edges (refer back to Figure 4), which perform the actual sealing function, were in fair condition. It was decided to have the valves returned to Rockwell's Pittsburgh location for further test and evaluation.

Seat leakage tests were again performed on the valves in the "as received" condition with results showing reasonable leakage rates. Both valves exhibited seat leakage rates of less than 20 scfh (0.57 scmh) at full differential pressure. The valves were completely disassembled and thorough inspection was made of the stem, bearings, packing, etc. All working parts of the valve were shown to be in good working order with no unusual signs of wear or damage, but seat reconditioning was obviously required.

Although original plans called for return of the valves to IGT for further cycling tests, the small scale of the size 3 valves frustrated attempts to lap the seats to their original finish and the effort was abandoned. It was decided instead that data collected over the span of the test program was adequate for evaluation of the design principles employed and had established solid groundwork from which improvements could be generated.

Conclusion

The basic concepts employed in the two size 3 valves which were cycle tested at IGT were successful, even though testing fell short of the 20,000 cycle goal. The excessive leakage rate which resulted in discontinuation of the test was determined to have been caused by inadvertent control system failure related to one of the basic features of the design—that of effective seat purging. The valves otherwise demonstrated good performance and durability, but showed need for improved maintainability. Reliability of the shaft bearings and seals was further demonstrated through successful long-term laboratory tests conducted at Rockwell's Valve Engineering and Research Center in Pittsburgh.

Data collected from the tests conducted at IGT has resulted in certain design improvements addressed mainly to valve maintainability. Cited improvements in design are as follows:

- The seat is on a separate removable sleeve permitting easier maintenance of the Stellite seating surface than was possible with the integral body seat in the size 3 valve. In addition, valve maintainability is improved because a spare seat and disk can be put in a valve to permit it to be returned to service quickly.

- The gate shape has been modified to permit lapping on a flat plate.

- The outlet end of the seat purge annulus has been modified to provide more effective seat purging.

Two size 6, Class 900 lock hopper valves employing the basic features of the size 3 valves and the above cited improvements have been fabricated by Rockwell and are scheduled for installation in the coal feed system of the Energy Research and Development Administration's (ERDA's) Synthane Pilot Plant at Bruceton, Pennsylvania. Photos of these valves before shipment to ERDA can be seen in Figure 9. Like the size 3 valves tested at IGT, these valves are actually "scale models" of larger valves and the design concepts employed are based on requirements for full scale plant valves. Design conditions for the valves have been set at 1000 psig (69.0 bar) pressure and 250F (121C) maximum temperature. Laboratory seat leak rate tests on the "as-built" valves showed an average leak rate of 1.3 scfh (0.03 scmh) with design differential pressure across the gate. It is expected that additional valuable information on lock hopper valving will be made available by this important pilot plant application.

Dr. Sayeed Akhtar
ERDA
Pittsburgh Energy Research Ctr.
4800 Forbes Avenue
Pittsburgh, Pa. 15213

J. W. Alexander
Bullen & Partners
Grosvenor House
125 High Street
Croydon, CR9 1PT
United Kingdom
Phone: 01 686-2622

Dr. R. K. Anand
General Electric Co.
Bldg. #2 - 525
1 River Road
Schnectady, New York 12345

Mr. Frank Au Miller
Sunstrand Corporation
4747 Harrison Ave.
Rockford, Illinois 61101

Dr. M. M. Avedesian
Energy Research Laboratories
CANMET
Department of Energy Mines & Resources
555 Booth Street
Ottawa
Ontario
Canada KIA OGol

Mr. W. Bagley
Airesearch Manufacturing Co.
402 S. 36th Street
Phoenix, Arizona 85010
Phone: (602) 267-2352

Mr. Mark Barrett
Kamyr, Inc.
Glens Falls
New York 12801
Phone: (518) 793-5111

Mr. Al Baturay
EIMCO BSP Division
One Davis Drive
Belmont, California 94002
Phone: (415) 592-4060

Mr. Andrew Bela
The Ralph M. Parsons Company
100 West Walnut Street
Pasadena, Calif. 91124
Phone: (213) 440-2338

Mr. Robert Beale
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-4060

Mr. M.S. Bhat
Bldg. 62
Lawrence Berkeley Lab
University of Calif.
Berkeley, Ca. 94720

Mr. Yesh Bhide
Dept. 3361
Allis-Chalmers Corp.
Box 512
Milwaukee, Wi. 53201
Phone: (414) 475-4702

Mr. John Biel
Stock Equipment Co.
16490 Chillicothe Rd.
Chagrin Falls, Ohio 44022
Phone: (216) 543-8211

Mr. Bimal Biswas
Foster Wheeler Energy Corp.
110 South Orange Ave.
Livington, New Jersey 07039
Phone: (201) 533-3301

Mr. J. Boland
Trane Co.
Engineering Bldg.
Box 1091
La Crosse, Wisconsin 54601

Mr. A. Bolumen
Babcock & Wilcox -20 Van Buren Ave.
Power Generation Group
Barberton, Ohio 44203

Mr. John Bonin
Lockheed Missiles & Space Co., Inc.
Coal Feeder Programs
3251 Hanover Street
Palo Alto, Ca. 94304
Phone: (408) 742-3880

Mr. W. H. Bowman
Amoco Chemicals Corporation
P.O. Box 400
Naperville, Illinois 60540
Phone: (312) 420-5581

Mr. C. L. Bradshaw
Flow Control Division
Rockwell International
400 North Lexington Ave.
Pittsburgh, Penn. 15208
Phone: (412) 247-3305

Mr. John Branam
The MITRE Corporation
Metrek Division
1820 Dolley Madison Blvd.
McLean, "irginia 22101
Phone: (703) 790-6000

Mr. Richard J. Brennan
Stone & Webster Engineering Corp.
245 Summer Street
Boston, Mass. 02107
Phone: (617) 973-5010

Mr. George M. Bretz
Research Dept.
Koppers Co., Inc.
15 Plum St.,
Verona, Pa. 15147
Phone: (412) 391-3300

Mr. Thomas R. Bump
Argonne National Laboratory
9700 South Cass Avenue
Argonne, Illinois 60439

Dr. E. Burke
Lockheed-Palo Alto Research Lab
Orgn. 52-30 B/201
3251 Hanover Street
Palo Alto, Ca. 94304

Mr. Lawrence G. Caldwell
Ducon Company Inc.
840 First Avenue
King of Prussia, Pa. 19406
Phone: (215) 337-3770

Mr. Michael Cambon
40 Pullman Kellogg
1300 3 Greenway Plaza E.
Houston, Tx. 77046

Mr. Dalton E. Cantey
Lockheed Missiles & Space Co.
3251 Hanover Street
Palo Alto, Ca. 94304
Phone: (415) 493-4411

Mr. J. Carson
Jenike & Johanson, Inc.
2 Executive Park Dr.
No. Billerica, Ma. 01862
Phone: (617) 667-5136

Mr. T. Caspary
Envirotech BSP Division
One Davis Drive
Belmont, Ca. 94002

Mr. R. Catena
Texaco
Montebello Research Laboratory
P.O. Box 400
Montebello, Ca. 90640
Phone: (213) 699-0948

Mr. R. Chandrasekhar
Foster Miller Associates
135 Second Ave.
Waltham, Ma. 02154
Phone: (617) 890-3200

Mr. T. Chen
Ingersoll-Rand
Box 301
Princeton, N.J. 08540
Phone: (609) 921-9103

Mr. Christian Clamser
Foster Wheeler Energy Corp.
110 South Orange Ave.
Livingston, New Jersey 07039
Phone: (201) 533-3751

Mr. C. E. Clapp, Jr.
General Resource Corp.
201 South Third St.
Hopkins, Minnesota 55343
Phone: (612) 933-7474

Mr. George T. Claypoole
Pope, Evans and Robbins
1133 Avenue of the Americas
New York, N.Y. 10036
Phone: (212) 730-5492

Mr. Frank M. Condos
Energy Systems Group of TRW Inc.
7600 Colshire Drive - #w2-7700
McLean, Virginia 22101
Phone: (703) 893-2000

Mr. George H. Conover
Ford, Bacon & Davis Utah Inc.
375 Chipeta Circle
Salt Lake City, Utah 84108
Phone: (801) 583-3773

Mr. R. A. Cordiale
C.F. Braun Eng.
1000 Fremont - Room 4205
Alhambra, Calif. 91802

Mr. Edward Cosgrove
Burns & Roe, Inc.
185 Crossways Park Dr.
Woodbury, N.Y. 11797
Phone: (516) 677-2233

Mr. C. Crawford
Fluor Engineers and Constructors, Inc.
3333 Michelson Drive
Irvine, Calif. 92715
Phone: (714) 975-4452

Mr. Douglas Crawford
C.F. Braun Co.
1000 Fremont
Alhambra, Ca. 91802
Phone: (213) 570-1000

Mr. R. Cruce
Procon Inc.
411 S. Boylston Street
L.A., Ca. 90017
Phone: (213) 481-0900

Mr. Cedric Daksla
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103

Mr. Arnold Daniel
Lockheed
3251 Hanover St.
Palo Alto, Ca. 94304
Phone: (408) 742-4321 X 22017

Dr. D. Davis
Rockwell/Rocketdyne Div.
6633 Canoga Ave.
Canoga Park, Ca. 91304

Mr. Harold Davis
Batelle Columbus
505 King Avenue
Columbus, Ohio 43201
Phone: (614) 424-7840

Dr. Samuel Dobner
Union Carbide
1523 54th St.
Brooklyn, New Jersey 11219

Dr. K. Durai-Swamy
Occidental Research Corporation
1855 Carrion Road
La Verne, Ca. 91750
Phone: (213) 967-5225

Dr. R. Durrfeld
Bergbau-Forschung GMBH
D-4300 Essen 1
West Germany

Mr. S. Dzierlenga
Radian Corporation
8500 Shoal Creek Blvd.
Austin, Texas 78766
Phone: (512) 454-4797

Mr. Kurt Eise
Werner & Pfleiderer Corp.
160 Hopper Avenue
Waldwick, New Jersey 07463
Phone: (201) 652-8600

Dr. Christopher England
Jet Propulsion Laboratory
4800 Oak Grove Dr.
Pasadena, Ca. 91103
Phone: (213) 354-4554

Mr. F. J. Enright
Dept. 3361 Coal Gasification
  Systems Operation
Allis Chalmers Corporation
Box 512
Milwaukee, Wisconsin 53201

Mr. Ray Epperly
Jeffrey Dresser Incorporated
1237 Dublin Road
Columbus, Ohio 43215
Phone: (614) 421-3069

Dr. A. Y. Falk
Rocketdyne Division
Rockwell International
6633 Canoga Avenue
Canoga Park, Ca. 91304
Phone: (213) 884-4000 ext. 5471

Mr. Nathandale (Bud) Farris
Mining Research Division
Continental Oil Co.
P.O. Box 1267
Ponca City, Oklahoma 74601
Phone: (405) 762-3456 ext. 8696

Mr. N. Ferry
The Ralph M. Parsons Co.
100 West Walnut Street
Pasadena, Calif. 91124
Phone: (213) 440-2280

Mr. Donald M. Field
Illinois Coal Gasification Group
122 South Michigan Ave.
Chicago, Illinois 60603
(312) 431-4934

Dr. S. Fields
GARD Inc.
7449 North Natchez Ave.
Niles, Illinois 60648
Phone: (312) 647-9000

Dr. B. Firnhaber
Krupp-Koppers GMBH
R & D Dept.
Postfach 8
4300 Essen 1
West Germany

Mr. John Fischer
Koppers Co., Inc./Logan Street
Sprout-Waldron Metal Products Div.
Muncy, Pa. 17756
Phone: (717) 546-8211

Mr. John Fischer
Martin Engineering Co.
U.S. Route 34
Neponset, Illinois 61345
Phone: (309) 594-2384

Dr. C. Fisher
University of Tennessee
Engineering Experiments Station
Knoxville, Tenn. 37916

Mr. R. Fisher
Bechtel Corp. 50/16/A-19 Coal Tech.
50 Beale Street
San Francisco, Ca. 94105
Phone: (415) 768-4661

Mr. R. R. Fleischbein, P.E.
Major Facilities Project
  Management Division
ERDA
20 Massachusetts Avenue, N.W.
Washington D.C. 20545
Phone: 8-376-9263

Mr. Ed Fochtman
IIT Research Institute
10 West 35th Street
Chicago, Illinois 60616
Phone: (312) 567-4270

Mr. Leonard L. Frauen
Dravo Corp.
One Oliver Plaza
Pittsburgh, Pa. 15222
Phone: (412) 566-3783

Mr. Erwin Funk
Kamyr, Inc.
Glens Falls
New York 12801
Phone: (518) 793-5111

Dr. Tony Furman
G.E. Research & Developmt. Ctr.
Comb. Bldg., K-1
Schenectady, N.Y. 12305

Mr. George G. Gale
Occidental Research Corp.
1855 Carrion Road
La Verne, Ca. 91750
Phone: (714) 957-7054

Mr. Robert Gamble
Foster Wheeler Energy Corp.
110 South Orange Ave.
Livingston, New Jersey 07039
Phone: (201) 5332789

Mr. John Gardner
U.S. ERDA-MERC
P.O. Box 880
Morgantown, West Virginia 26505
Phone: (304) 599-7125

Mr. Donald Garner
Suite 125, Grant Bldg.
1651 Old Meadow Rd.
McLean, Va. 22101
Phone: (703) 821-8866

Mr. H. Tahsin Gencsoy
West Virginia University
Morgantown, W. Va. 26506

Mr. E. E. Geraci
Leeds & Northrup Co.
Technical Center-Sumneytown Pike
North Wales, Pa. 19454
Phone: (215) 643-2000

Mr. H. H. Gilman
EPRI
3412 Hillview Ave.
Palo Alto, Ca. 94303
Phone: (415) 493-4800

Mr. V. Giroux
Phillips Petroleum
128 R.B. 5
Bartlesville, Oklahoma 74004

Mr. Frank Glenn
The Ralph M. Parsons Company
100 West Walnut Street
Pasadena, California 91124
Phone: (213) 440-3275

Mr. George Goebel
Envirotech Corporation
3000 Sand Hill Road
Menlo Park, California 94025
Phone: (415) 854-2000

Mr. Phillip Gold
California State University
5151 State University Dr.
Los Angeles, Ca. 90032
Phone: (213) 224-2479

Mr. R. J. Grace
Bituminous Coal Research Inc.
350 Hochberg Road
Monroeville, Pa. 15146
Phone: (412) 327-1600

Ms. Catherine Graves
Atlantic-Richfield
408 So. 3rd Ave.
Arcadia, Ca. 91006

Mr. W. Griffiths
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103

Mr. S. Griffin
Neill & Associates
2550 North Loop West, Suite 730
Houston, Texas 77092
Phone: (713) 688-8996

Dr. Vernon Griffiths
Montana College of Mineral
  Science & Technology
Butte, Montana 59701

Mr. R. Gugger
Curtiss-Wright Corp.
One Passaic St.
Wood-Ridge, N.J. 07075
Phone: (201) 777-2900

Mr. Adi R. Guzdar
Foster-Miller Associates Inc.
135 Second Ave.
Waltham, Ma. 02154
Phone: (617) 890-3200

Mr. Thomas G. Halvorson
Union Carbide Corporation
61 East Park Drive
Tonawanda, New York 14150
Phone: (716) 877-1600

Mr. J. Robert Hamm
Westinghouse Electric
  Research Laboratories
1310 Beulah Road
Pittsburgh, Pa. 15235

Dr. Richard W. Hanks
Brigham Young University
Provo, Utah 84602
Phone: 374-1211 ext. 2587

Dr. J. Hannoosh
Foster-Miller Associates Inc.
135 Second Ave.
Waltham, Ma. 02154
Phone: (617) 890-3200

Dr. John Harding
Foster-Miller Associates Inc.
135 Second Ave.
Waltham, Ma. 02154

Dr. R. Hartung
Lockheed Missiles
3251 Hanover Street
Palo Alto, Ca. 95051
Phone: (415) 493-4411 X 45195

Mr. Andrew C. Harvey
Foster-Miller Associates Inc.
135 Second Ave.
Waltham, Ma. 02154
Phone: (617) 890-3200

Mr. T. Hathaway
Ingersoll-Rand Inc.
Box 301
Princeton, N.J. 08540
Phone: (609) 921-9103

Mr. J. Hnat
G.E. Space Division
Rm. L 9517, Bldg. 100
P.O. Box 8555
Philadelphia, Pa. 19101

Mr. Henry Ho
Fluor Corp.
3333 Michelson Dr.
Irving, Ca.
Phone: (714) 975-3575

Mr. John Mark Hobday
ERDA-MERC
P.O. Box 880
Morgantown, West Virginia 26505
Phone: (304) 599-7746

Mr. S. Holmberg
Flui-dyne Engineering Corp.
5900 Olson Memorial Hwy.
Minneapolis, Minnesota 55422
Phone: (612) 544-2721

Mr. E. Homburg
Acton Corporation
29085 Solon Road
Cleveland, Ohio 44139
Phone: (216) 248-8214

Dr. Keh-Hsien Hsiao
Pullman Kellogg
Research & Development Center
16200 Park Row, Industrial
  Park Ten
Houston, Texas 77084
Phone: (713) 492-2500

Mr. M.A. Huso
Southern Pacific Pipe Lines
610 South Main Street
Los Angeles, Ca. 90014
Phone: (213) 629-6633

Mr. S. Iglesias
Foster Wheeler Energy Corp.
110 South Orange Ave.
Livingston, New Jersey 07039
Phone: (201) 533-1100

Mr. L. Isenberg
Fairchild-Stratos
1800 Rosecrans Ave.
Manhattan Beach, Ca. 90266
Phone: (213) 675-9111

Mr. Sridhar K. Iya
Union Carbide Corp.
P.O. Box 44
Tonawanda, New York 14150
Phone: (716) 877-1600

Dr. J. Jackson
University of Wisconsin
Institute of Mineral & Solids
  Fuel Technology
P.O. Box 413
Milwaukee, Wisconsin 53201

Dr. E. Jacobi
TRW Energy Systems
R4/2050 One Space Park
Redondo Beach, Ca. 90278

Mr. S. Jakopin
Werner & Pfleiderer Corp.
160 Hopper Avenue
Waldwick, New Jersey 07463
Phone: (201) 652-8600

Dr. A. Jenike
Jenike & Johanson, Inc.
2 Executive Park Drive
No. Billerica, Ma. 01862
Phone: (617) 667-5136

Mr. N. L. Jensen Jr.
C.F. Braun & Co.
Power Division/1000 Fremont St.
Alhambra, Ca. 91802
Phone: (213) 570-4318

Mr. Bruce Johnson
ERDA - Box 8213 University Station
The Grand Forks Energy Research Ctr.
Grand Forks, North Dakota 58202
Phone: (701) 775-4207

Mr. J. Johnson
EIMCO BSP Division -Envirotech
One Davis Drive
Belmont, Ca. 94002
Phone: (415) 592-4060

Mr. James Jones
Allis-Chalmers Corp.
Reduction Systems Division/Box 512
Milwaukee, Wisconsin 53201
Phone: (414) 475-4865

Mr. E. Kane
K-Tron Corporation
20 Warrick Ave.
Glassboro, N.J. 08028
Phone: (609) 881-6500

Mr. Marty Kane
K-Tron Corporation
20 Warrick Ave.
Glassboro, N.J. 08028

Dr. Victor Kevorkian
Ingersoll-Rand Company
Box 301
Princeton, New Jersey 08540
Phone: (609) 921-9103

Mr. T. Knowlton
IGT
4201 W. 36th Street
Chicago, Illinois
Phone: (312) 542-7088

Mr. P. Kindersley
Kamyr Valves Inc.
Ridge Center
Glens Falls, N.Y. 12801

Mr. Arthur Kohl
Atomics International Division
8900 De Soto Avenue
Canoga Park, Ca. 91304
Phone: (213) 341-1000

Mr. Richard H. Komai
Southern California Gas Co.
810 So. Flower St.
Los Angeles, Calif. 90051

Mr. Ed Kowalski
The Babcock & Wilcox Company
20 S. Van Buren
Barberton, Ohio 44203
Phone: (216) 753-4511

Mr. Stephen Kramer
The Ralph M. Parsons Company
100 Walnut Street-West
Pasadena, Ca. 91124

Mr. Frederick Kronmiller
Fluor Engineers & Constructors
3333 Michelson Drive
Irvine, Calif. 92715
Phone: (714) 975-3048

Mr. R. Kuiper
Lockheed Missiles
3251 Hanover St.
Palo Alto, Ca. 94304

Mr. J. Kurtz
Union Carbide
P.O. Box Y
Bldg. 9201-3
Oak Ridge, Tenn.

Mr. Andrew Kurylchek
Solids Flow Control Corp.
4 Fairfield Crescent
West Caldwe᾿, New Jersey 07006
Phone: (201) 575-7288

Dr. R. Kushida
Jet Propulsion Laboratory
Bldg. 125-224
4800 Oak Grove Drive
Pasadena, Calif. 91103
Phone: (213) 354-6079

Mr. J. Lacey
ERDA
4800 Forbes Ave.
Pittsburgh, Pa. 15213

Mr. M. Lackey
Union Carbide
Nuclear Division
P.O. Box Y, Bldg. 9201-3
Oak Ridge, Tenn.

Mr. A. W. Larson
Amoco Research Center/ Standard Oil
Sta. H-6
Warrenville Rd.
Naperville, Illinois 60540

Dr. N. J. Lavingia
The Ralph M. Parsons Company
100 West Walnut Street
Pasadena, Calif. 91124
Phone: (213) 440-3339

Mr. John F. Lenc
Argonne National Lab
9700 S. Cass Ave. Bldg. 205
Argonne, Illinois 60439

Mr. J. A. Leppard
Liquid Metal Engineering Center
Rockwell Corp.
P.O. Box 1449
Canoga Park, Ca. 91304
Phone: (213) 341-1000

Mr. John Leszczynski
Envirotech Corp. BSP Division
One Davis Drive
Belmont, Ca. 94002
Phone: (415) 592-4060

Mr. Alan Levy
Lawrence Berkeley Laboratory
Bldg. 62 - Room 351
1 Cyclotran Drive
Berkeley, Ca. 94720

Mr. D. Lewis
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Calif. 91103
Phone: (213) 354-2864

Dr. Han Liu
Burns & Roe, Inc.
185 Crossways Park Dr.
Woodbury, N. Y. 11797
Phone: (516) 677-2240

Mr. Howard Lownes, Sr.
Allen-Sherman-Hoff Co.
One Country View Rd.
Great Valley Center
Malvern, Pa. 19355
Phone: (215) 647-9900

Dr. Bill Luckow
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-4060

Mr. Ming D. Luh
DR DAR-CLE
Chemical Systems Laboratory
Aberdeen Proving Ground
Md. 21010

Mr. Ed Lynch
Center-Line Machinery
13071 Malena Drive
Santa Ana, Ca. 92705
Phone: (714) 835-8800

Mr. R. Mac Pherson
Union Carbide
Nuclear Div.
P.O. Box Y, Bldg. 9201-3
Oak Ridge, Tenn.
Phone: (615) 483-8611 ext. 37129

Mr. R. Madenburg
Morrison-Knudsen
P.O. Box 7808
Boise, Idaho 83729
Phone: (208) 245-5000 Ext. 5069

Dr. Ram Manvi
Jet Propu'sion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-4489

Mr. Dan Marlowe
Envirotech Corp.-BSP Division
One Davis Drive
Belmont, Ca. 94002
Phone: (415) 592-4060

Mr. Joseph Marro
Product Development
Entoleter Inc.
251 Welton St.
Hamden, Conn. 06510

Mr. James Martin
University of Tennessee
Tullahoma, Tenn 37388

Mr. James Matousek
Argonne National Laboratory
9700 South Cass Avenue
Argonne, Illinois 60439
Phone: (312) 739-7711 ext. 5532

Mr. Lawrence Mattson
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-7068

Mr. N. J. Mazzocco
U.S. Energy Research & Development Administration
Pittsburgh Energy Research Division
4400 Forbes Ave.
Pittsburgh, Pa. 15219

Mr. R. McAleer
Foster Miller Associates
3621 Columbia Pike
Arlington, Va. 22204
Phone: (703) 979-7150

Mr. J. McCabe
Mechanical Technology Inc.
968 Albany - Shaker Road
Lathan, New York 12110

Mr. Wayne A. McCurdy
ERDA
20 Massachusetts Ave., N.W.
Washington, D.C. 20545

Mr. Ian M. McKechnie
Design Scientific Design Co., Inc.
Two Park Avenue, New York 10016
Phone: (212) 689-3000

Mr. Gerald McNamee
The Ralph M. Parsons Company
100 West Walnut Street
Pasadena, California 91124
Phone: (213) 440-4668

Mr. Steven D. Mehas
Foster Wheeler Energy Corp.
110 S. Orange Ave.
Livingston, New Jersey 07039

Mr. George Meixel
Washington Corporate Office
Ingersoll-Rand Co.
1666 K. Street, N.W.
Washington, D.C. 20006

Mr. Wesley Menard
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-5153

Dr. John W. Meyer
Lockheed Missiles & Space Co., Inc
Fluid Mechanics Lab 0/52-13 B/201
3251 Hanover Street
Palo Alto, Ca. 94304
Phone: (415) 493-4411

Mr. Matthew Miller
Fairchild Stratos Division
1800 Rosecrans Avenue
Manhattan Beach, Ca. 90266
Phone: (13) 675-9111

Mr. Wayne Miller
Union Oil Company
Research Dept.
P.O. Box 76
Brea, Ca. 92621
Phone: (714) 528-7201

Mr. Dilip Mistry
Ingersoll-Rand Research, Inc.
Box 301
Princeton, New Jersey 08540
Phone: (609) 921-9103

Mr. Seymour Moskowitz
Curtiss-Wright Corporation
One Passaic St.
Wood-Ridge, N.J. 07075
Phone: (201) 777-2900

Mr. Thomas Munro
Westinghouse Energy Systems
Waltz Mill Site-Gate C
P.O. Box 158
Madison, Pa. 15663

Mr. Arthur Murphy
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-2324

Mr. Frank Murphy
Williams Brothers
6600 So. Yale
Tulsa, Okla. 74136

Dr. Bruce Murray
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-3405

Mr. Paul Musser
ERDA
20 Massachusetts Ave.
Washington D.C. 20545
Phone: 8- 3769334

Mr. E. T. Myskowski
Fairchild Stratos Division
1800 Rosecrans Avenue
Manhattan Beach, Ca. 90266
Phone: (213) 675-9111

Mr. Mohammad Nawaz
Gilbert Associates, Inc./ 525 Lancaster Ave.
Reading, Pennsylvania 19603
Phone: (215) 775-2600

Mr. James Nestlerode
Rockwell International
6633 Canoga Ave.
Canoga Park, Ca. 91304

Mr. Floyd Newburn
Rockwell
6633 Canoga Ave.
Canoga Park, Ca. 91304
Phone: (213) 884-3037

Mr. Lawrence K. Nordell
The Ralph M. Parsons Company
100 West Walnut Street
Pasadena, Calif. 91124
Phone: (213) 440-4513

Mr. R. Norman
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-3971

Mr. D. Otth
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 795-2547

Mr. Sal Panico
Pope, Evans and Robbins
1133 Avenue of the Americas
New York, New York 10036
Phone: (212) 730-5884

Mr. N. Patel
Union Carbide Corp.
61 East Park Drive
Tonawanda, New York 14150
Phone: (716) 877-1600

Mr. Doug Paulsrude
Morrison-Knudson
P.O. Box 7808
Boise, Idaho 83729
Phone: (208) 345-5000

Mr. C. Victor Pearson
Argonne National Laboratory
9700 South Cass Avenue
Argonne, Illinois 60439
Phone: (312) 739-7711 ext. 4055

Mr. Gerry Perkins
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-6707

Mr. Douglas Peterson-Conspray
111 West Dyer Road
Unit B Santa Ana, Ca. 92707
Phone: (714) 540-1274

Mr. G. Peterson
Union Carbide
Y-12 Plant, Bldg. 9202
Oak Ridge, Tenn. 37830
Phone: (615) 483-8611 ext. 37030

Mr. Robert Phen
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-6793

Mr. J. Porter
Energy Resources
185 Alewife Brook Pkwy.
Cambridge, Ma. 02138
Phone: (617) 661-3111

Mr. J. Powell
ERDA
20 Massachusetts Ave., N.W.
Washington D. C. 20545
Phone: 376-9259

Mr. Harvey Proctor
So. Calif. Gas Co.
P.O. Box 3249
Terminal Annex
Los Angeles, Ca. 90051
Phone: (213) 689-2867

Mr. B. Quist
Koninklijke/Shell-Laboratorium,
  Amsterdam
Badhuisweg 3, Amsterdam-N
Phone: 020-209111

Mr. Surendra D. Rao
The Ralph M. Parsons Co.
100 West Walnut Street
Pasadena, Ca. 91124

Mr. Joel M. Reid
Milltronics
2409 Ave. J. Suite D.
Arlington, Texas 76011
Phone: (817) 649-8540

Dr. Reimert
Lurgi Mineraloeltechnik GMBH
6000 Frankfurt (Main)
Bockenheimer Landstr. 42
Lurgi-01 Haus, Germany Postfach 119181
Phone: (06 11) 71191

Mr. Harold Reintjes
Petrocarb Inc.
250 Broadway
New York, N.Y. 10007
Phone: (212) 766-1500

Mr. John A. Richards
Conspray Construction Sys. Inc.
111 West Dyer Road
Unit B
Santa Ana, Ca. 92707
Phone: (714) 540-1274

Mr. E. Richardson
Continental
Emsco Co.
P.O. Box 1489
Houston, Texas 77001
Phone: (713) 645-3361 Ext. 407

Dr. H. Richardson
Foster Miller
135 Second Avenue
Waltham, Ma. 02154

Mr. P. Roberts
Chevron Research Co.
576 Standard Ave.
Richmond, Ca. 94802
Phone: (415) 237-4411

Dr. T. Roberts
FMC Corporation
1185 Coleman Ave.
San Jose, Calif.

Mr. Roy Robinson
California Pellet Mill Co.
1114 E. Wabash Ave.
Crawfordsville, Indiana 47933
Phone: (317) 362-2600

Dr. D. Roe
Cameron Engineers
1315 South Clarkson Street
Denver, Colorado 80210
Phone: (303) 777-2525

Dr. Ward C. Roman
United Technologies Research
East Hartford, Conn 06108
Phone· (203) 565-4596

Mr. William Rosborough
Mitre Corporation/Metrek Division
1820 Dolley Madison Blvd.
McLean, Virginia 22101
Phone: (703) 790-6000

Mr. William D. Ross II
Fuller Company
2966 E. Victoria St.
Compton, Ca. 90224
Phone: (213) 636-9821

Mr. W. I. Rudnicki
Power & Energy Systems
Aerojet Liquid Rocket Co.
P.O. Box 13222
Sacramento, Ca. 95813
Phone: (916) 355-3518

Dr. P. Ryason
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-6308

Mr. Richard R. Santore
Synthane Pilot Plant
ERDA
4800 Forbes Ave.
Pittsburgh, Pa. 15213
Phone: (412) 892-2400

Mr. R. Sattler
Phillip Petroleum Co.
Bldg. 85-E PRC
Bartlesville, Okla. 74004

Mr. William Schatz
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-7191

Mr. Milt Schindel
Weighing & Feeding Systems
Milltronics/Process Controls
730 The Kingsway
Peterborough, Canada K9J 6W6

Mr. D. Schneiderman
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-3390

Mr. H. Schurmeier
Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, Ca. 91103
Phone: (213) 354-7517

Mr. R. M. Seidl
Dept. 3361 - GE
Allis-Chalmers Corp.
P.O. Box 512
Milwaukee, Wisconsin 53201

Mr. James Seneker
Union Carbide Corp./Nuclear Div.
P.O. Box Y
Bldg. 9201-3 MS8
Oak Ridge, Tenn. 37830

Mr. Raj D. Shah
Jet Propulsion Laboratory
4800 Oak Grove Dr.
Pasadena, Ca. 91103
Phone: (213) 354-4014

Mr. Wayne E. Shannon
Energy Programs O/52-01, B/201
3251 Hanover Street
Palo Alto, Ca. 94304
Phone: (415) 493-4411 ext. 45827

Mr. S. E. Shields
Amoco Oil Co.
Box 400
Naperville, Illinois 60540

Mr. Morton Shorr
Bell Aero   e-T xtron
Niagara F: .er Operations
Post Office ..x One
Buffalo, New York 14240
Phone: (716) 297-1000

Mr. James C. Short
Fuller Co.
124 Bridge St.
Catasauqua, Pa. 18032
Phone: (215) 264-6561

Mr. Thomas W. Shults
Foster Wheeler Development Corp.
John Blizard Research Center
Applied Thermodynamics/Process Dept.
12 Peach Tree Hill Road
Livingston, N.J. 07039

Mr. J. Sims
TRW Energy Systems Group
7600 Colshire Drive Room W2/7700
McLean, Virginia 22101
Phone: (703) 893-2000 ext. 2428

Mr. M. Slegers
Fairchild Stratos Division
1800 Rosecrans
Manhattan Beach, Ca.

Mr. Art Smith
Envirotech
EIMCO BSP Division
One Davis Drive
Belmont, Calif. 94002
Phone: (415) 592-4060

Mr. Carl B. Smith
Union Carbide
Nuclear Division
P.O. Box X
Oak Ridge, Tenn. 37830
Phone: (615) 483-8611 ext. 3-1761

Mr. Jack Smith
ERDA-FE Room 4203
20 Massachusetts Ave.
Washington D.C.

Mr. Kenneth Smith
NASA-Lewis Research Center
Cleveland, Ohio 44135
   Mail Stop: 500/202

Dr. Richard Smith
Combustion Power Company, Inc.
1346 Willow Road
Menlo Park, Ca. 94025
Phone: (415) 324-4744

Mr. Paul Solt
Fuller Company
124 Bridge St.
Catasauqua, Pa. 18032
Phone: (215) 264-6271

Mr. Sid Sorani
Davy Power Gas Co.
P.O. Box 5000
Lakeland, Florida 33800
Phone: (813) 646-7960

Mr. Charles Space
Reynolds, Smith, Hills Co.
4019 Blvd. Center Drive
Jacksonville, Florida 32201

Mr. M. Spence
Sedco Construction
2807 Buffalo Speedway
 Huston, Texas 77098
Phone: (713) 626-4420

Mr. W. Spurgeon
Gilbert Associates
525 Lancaster Ave.
Reading, Penn.
Phone: (215) 775-2600

Mr. R. Stahura
Martin Engineering Co.
U.S. Route 34
Neponset, Illinois 61345
Phone: (309) 594-2384

Mr. Donald Stein
K-Tron Corporation
20 Warrick Ave.
Glassboro, New Jersey 08028
Phone: (609) 881-6500

Ms. Judith Stelian
Beaumont-Birch Co.
1505 Race St.
Philadelphia, Pa. 19102
Phone: (215) 564-3570

Mr. Richard C. Stone
Environmental Engineering Division
Stone & Webster Engineering Corp.
245 Summer Street
Boston, Mass. 02107
Phone: (617) 973-2800

Mr. H. R. Strop
Anderson IBEC
19699 Progress Drive
Strongsville, O. 44136
Phone: (216) 238-5800

Mr. Stock
Stock Equipment Co.
16490 Chillicothe Road
Chagrin Falls, Ohio 44022
Phone: (216) 543-8211

Mr. Paul Tarman
IGT
IIT Center
3424 South State Street
Chicago, Illinois 60616

Mr. Paul Thibeault
Combustion Engineering
1000 Prospect Hill Rd.
Windsor, Conn. 06095

Mr. Ed Thomas
Air Products & Chemicals, Inc.
P.O. Box 538 .
Allentown, Pa. 18105
Phone: (215) 398-7888

Mr. Andrew Tierney
Fairchild Stratos Division
1800 Rosecrans Ave.
Manhattan Beach, Ca.
Phone: (213) 675-9111 ext 343

Mr. H. Trumbell
Batelle Columbus
Engineering Dept.
505 King Avenue
Columbus, Ohio 43201

Mr. V. Underkoffer
Gilbert Associates, Inc.
525 Lancaster Ave.
Reading, Penn. 19603
Phone: (215) 775-2600

Mr. Garrett Van Dyk
Dorr-Oliver, Inc.
77 Havemeyer Lane
Stamford, Ct. 06901

Mr. Van Dyke
Monsanto Research Inc.
P.O. Box 5555
Miamisburg, Ohio

Mr. Samuel Van Grouw
Combustion Power Co. Inc.
1346 Willow Road
Menlo Park, Ca. 94025
Phone: (415) 324-4744

Dr. Velling
Rheinische Braunkohler::erke
Aktiengesellschaft
Fostfach 41 08 40
5000 Koln 41

Mr. E. J. Vidt
Westinghouse Inc.
R & D Center / 1310 Beulah Rd.
Pittsburgh, Penn. 15235

Mr. V. Viitanen
FMC Corp.
1185 Coleman Ave.
San Jose, Calif.
Phone: 289-2521

Mr. Kenneth E. Walker
Phillips Petroleum Co.
P.O. Box 25
Homer City, Penn. 15748
Phone: (412) 479-9081

Mr. Lloyd W. Ward
Curtiss-Wright Corp.
Power Systems
905 Sixteenth Street, N.W.
Washington, D.C. 20006
Phone: (202) 638-2926

Mr. H. Weisenfeld
ERDA
Major Facilities Project
  Management Division
20 Massachusetts Ave., N.W.
Washington, D.C.

Mr. T. G. Wells, Jr.
Stone & Webster Engineering Corp.
245 Summer Street
Boston, Mass. 02107
Phone: (617) 973-2046

Dr. David Wesley
Institute for Mining
P.O. Box 13015
Lexington, Ky.
Phone: (606) 252-5535

Mr. Allan West
Scientific Design Co., Inc.
Two Park Avenue
New York 10016
Phone: (212) 689-3000

Mr. G. Weth
ERDA
20 Massachusetts Ave., N.W.
Washington, D.C. 20545

Dr. Don White
University of Arizona
Tucson, Arizona 85721
Phone: (602) 884-0111

Mr. Robert Whitson
ERDA-ORO
P.O. Box E
Oak Ridge, Tenn. 37830

Mr. H. Wiedmann
Werner & Pfleiderer Corp.
160 Hopper Ave.
Waldwick, New Jersey 07463

Mr. Reg. Wintrell
Arthur McKee Co.
6200 Oak Tree Blvd.
Cleveland, Ohio 44131

Mr. H. Withers, Jr.
Tenn. Valley Authority
1360 Commerce Union Bank Building
Chattanooga, Tenn. 37401

Mr. Brian A. Wolfe
Babcock & Wilcox
P.O. Box 835
Alliance, Ohio 44601
Phone: (216) 821-9110

Mr. G. S. Wong
Rockwell International
Rocketdyne Division
6633 Canoga Ave.
Canoga Park, Ca. 91304
Phone: (213) 884-3374

Dr. Paul Yavorsky
ERDA
4800 Forbes Avenue
Pittsburgh, Pa. 15213
Phone: FTS 8-726-2000

Mr. Alan Ziobrowski
Babcock Wilcox
Alliance Research Center
1562 Beeson Street
Alliance, Ohio 44601
Phone: (216) 821-9110 Ext. 521