DEVELOPMENT OF DRY COAL FEEDERS

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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 Braceton, 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:	$350^{O}F$ 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

FLOW INDICATOR
PISTON OPERATED VALVE
DIAPHRAGM TYPE VALVE
SAFETY RELIEF VALVE
PRESSURE TAP
COMPRESSOR
ELECTRIC MOTOR
THERMAL MASS FLOW METER
REGULATOR
MECHANICAL SCREW
MECHANICAL POWER TRANSMISSION

50 SCF/MIN
150 PSI
CO_2 PLANT

HYD. RESERVOIR
40 GAL.

HYD. PUMP/MOTOR
12 GPM, 260 PSI/3 1/2 HP

1 IN. φ

FLOW DIVIDERS

1 IN. φ

1/2 IN. φ TYP

COOLING
TOWER

H_2O RESERVOIR

COOLANT PUMP/MOTOR

HEAT EXCHANGER H_2O PUMP/MOTOR

FLOW DIVIDER

COOLANT RESERVOIR

TO REACTOR

10 HP

(150 PSI)

HIGH PRESSURE
STORAGE BIN

15 HP

3 IN. φ

1/2 IN. φ

PT

1 IN. φ

8 FT O.U. x 10 FT HIGH

3 IN. φ

ATMOSPHERIC
STORAGE BIN

7 1/2 HP

STEAM – O_2

RECIRCULATING   1 IN. φ
MAKE-UP GAS

START-UP GAS   1 IN. φ

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.
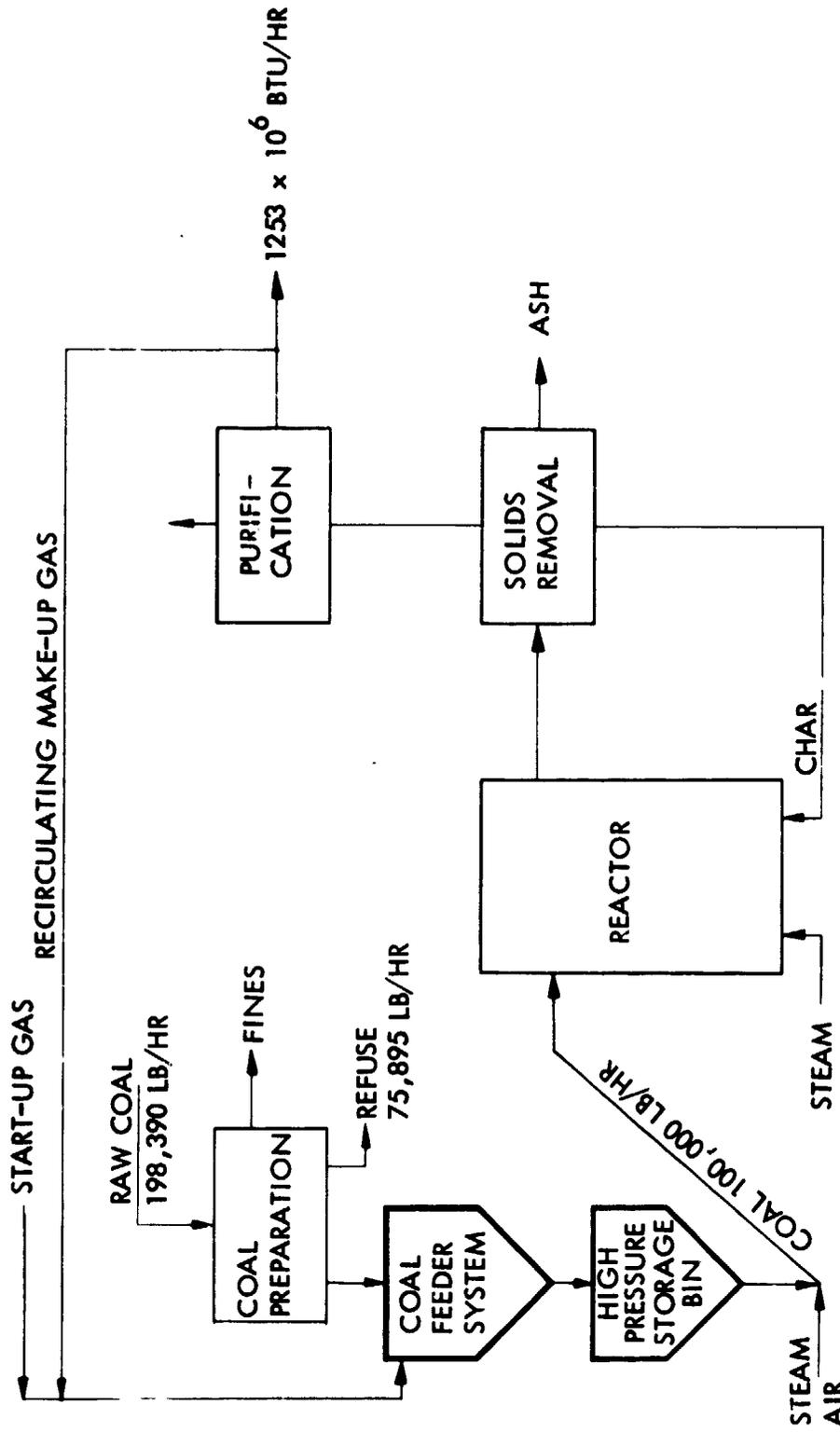
Fig. 2 Coal Gasification Process (Low BTU)
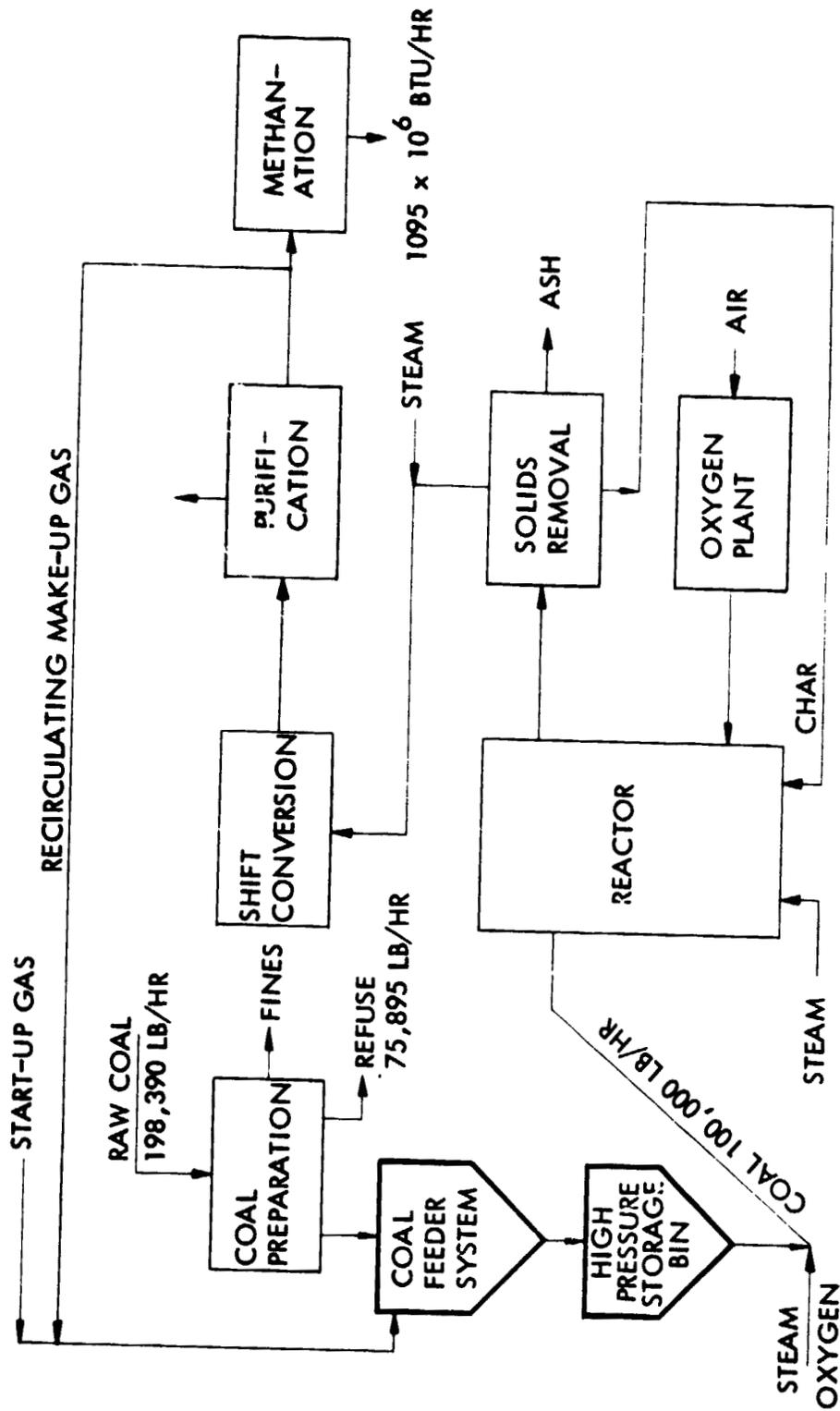
Fig. 3 Coal Gasification Process (High BTU)

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	55,803	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 $450^{\circ}F$. They have a capacity of about 40 ft^3.
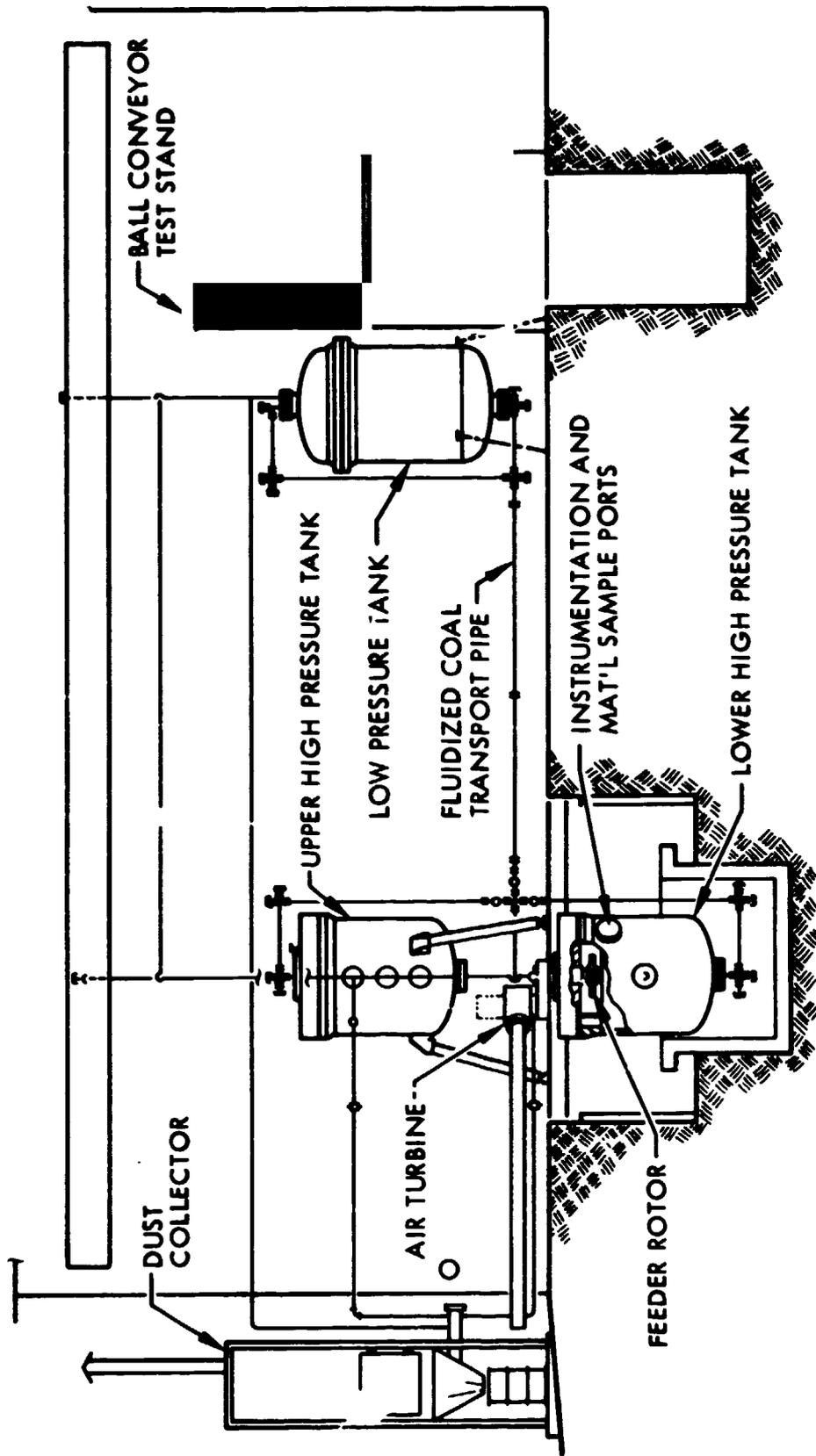
Fig. 4 Facility Layout - Coal Transfer Lines

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.
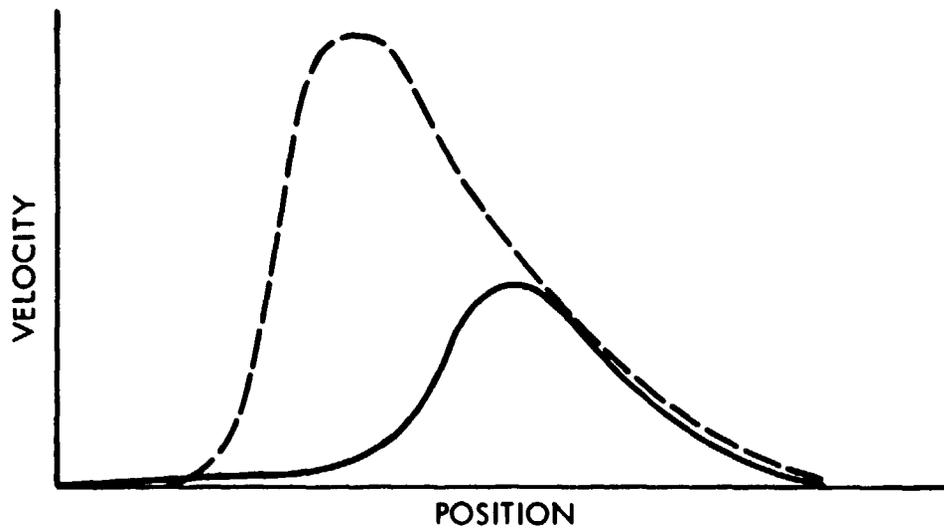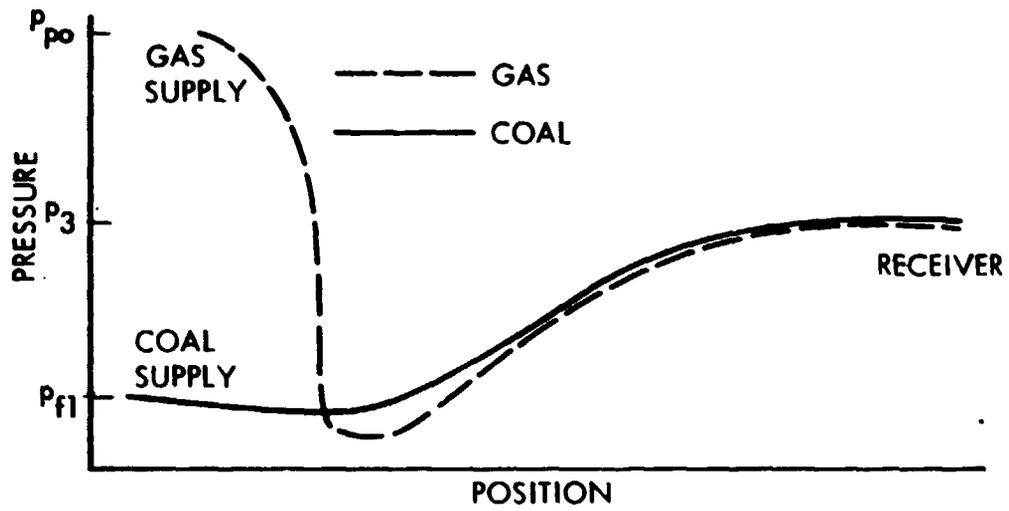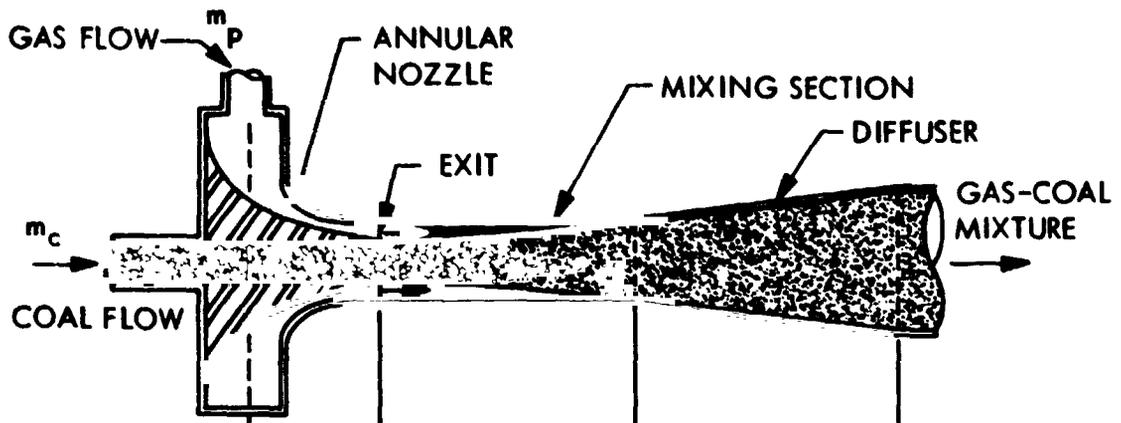
Fig. 5   Ejector Schematic

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.
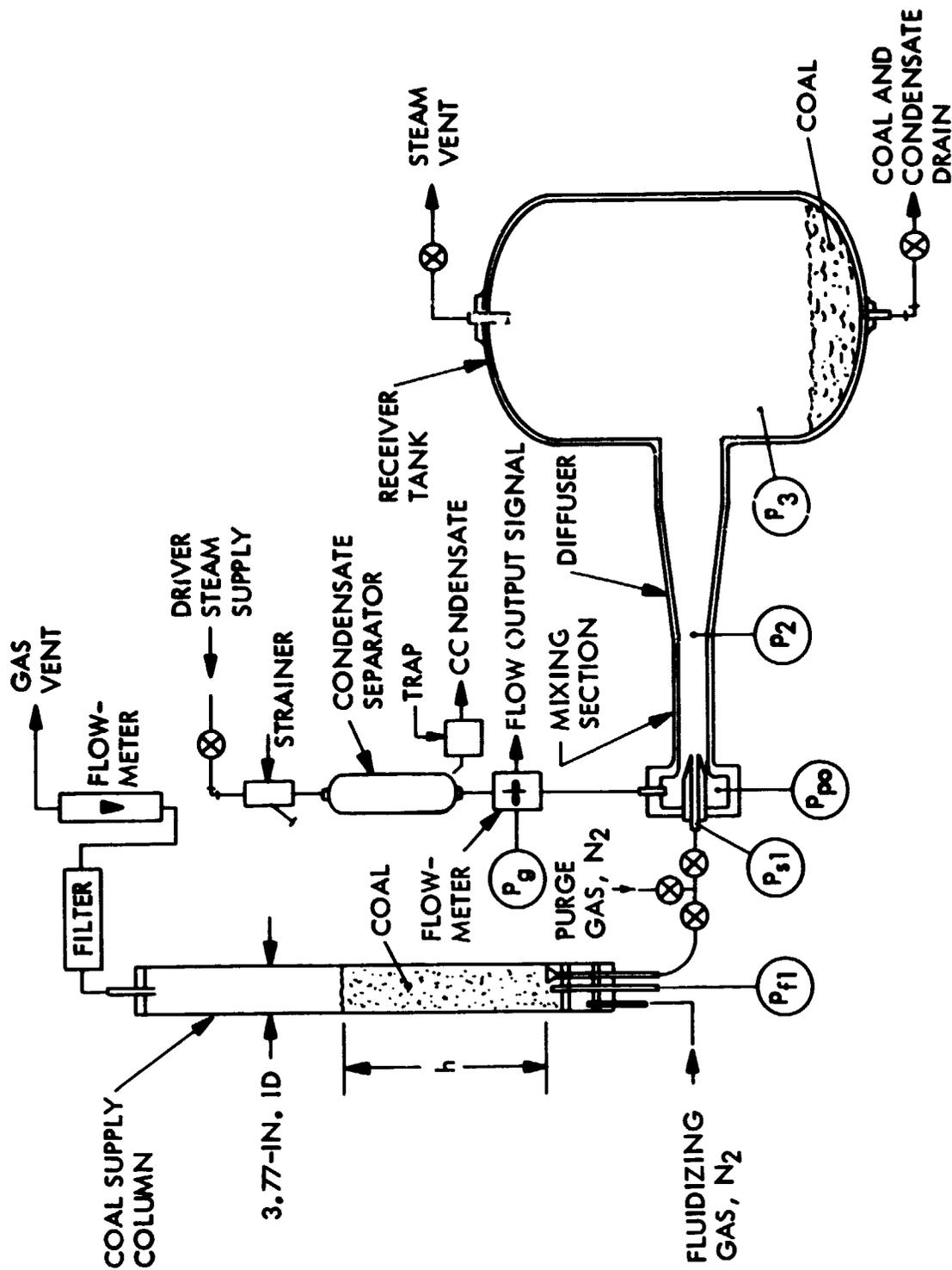
Fig. 6 Laboratory Test Schematic – Steam Driven Ejector
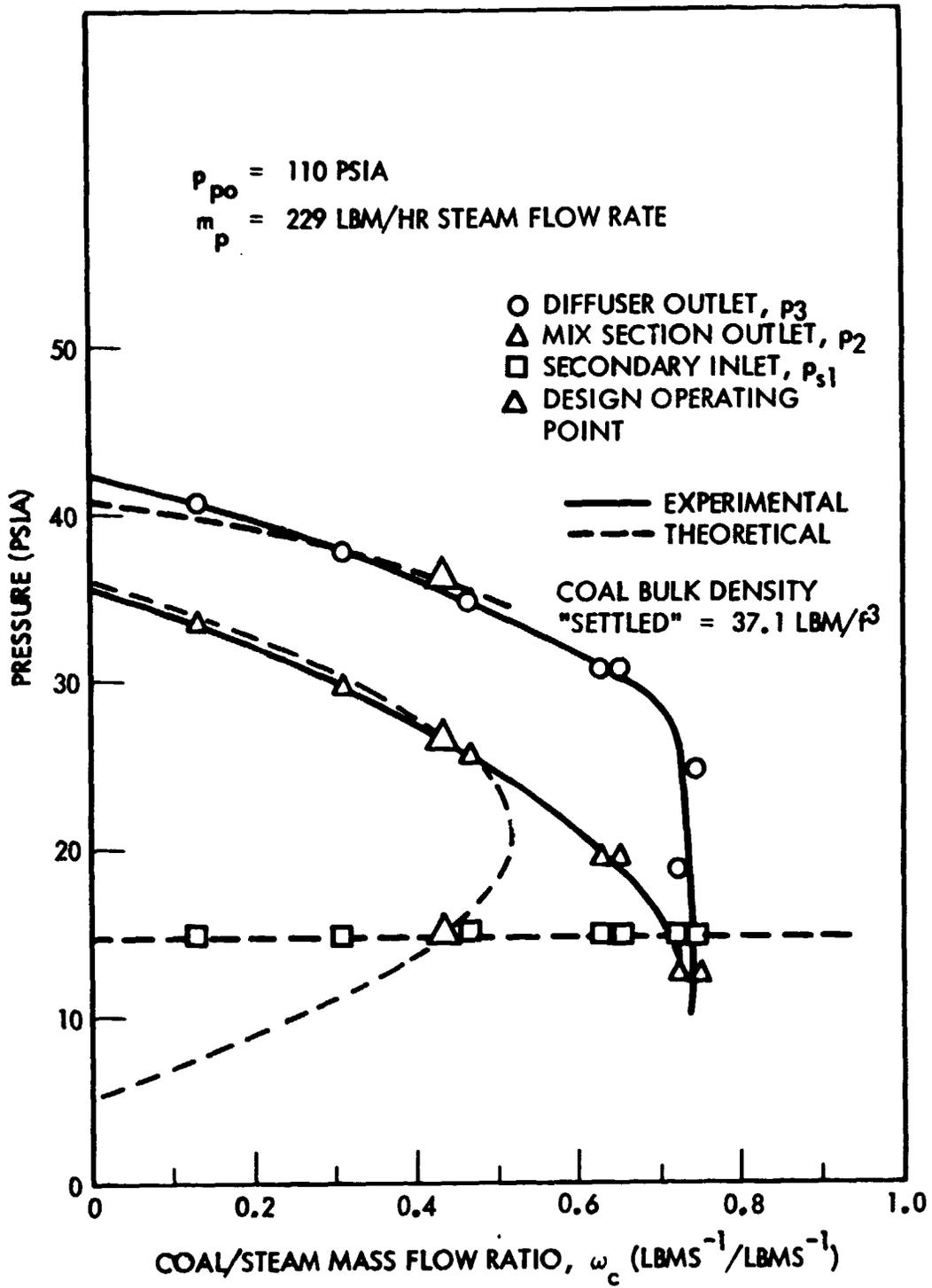
Fig. 7 Ejector Performance with 110-psia Steam Drive

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, $T_{po,} = 135^oF$ 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^oF 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
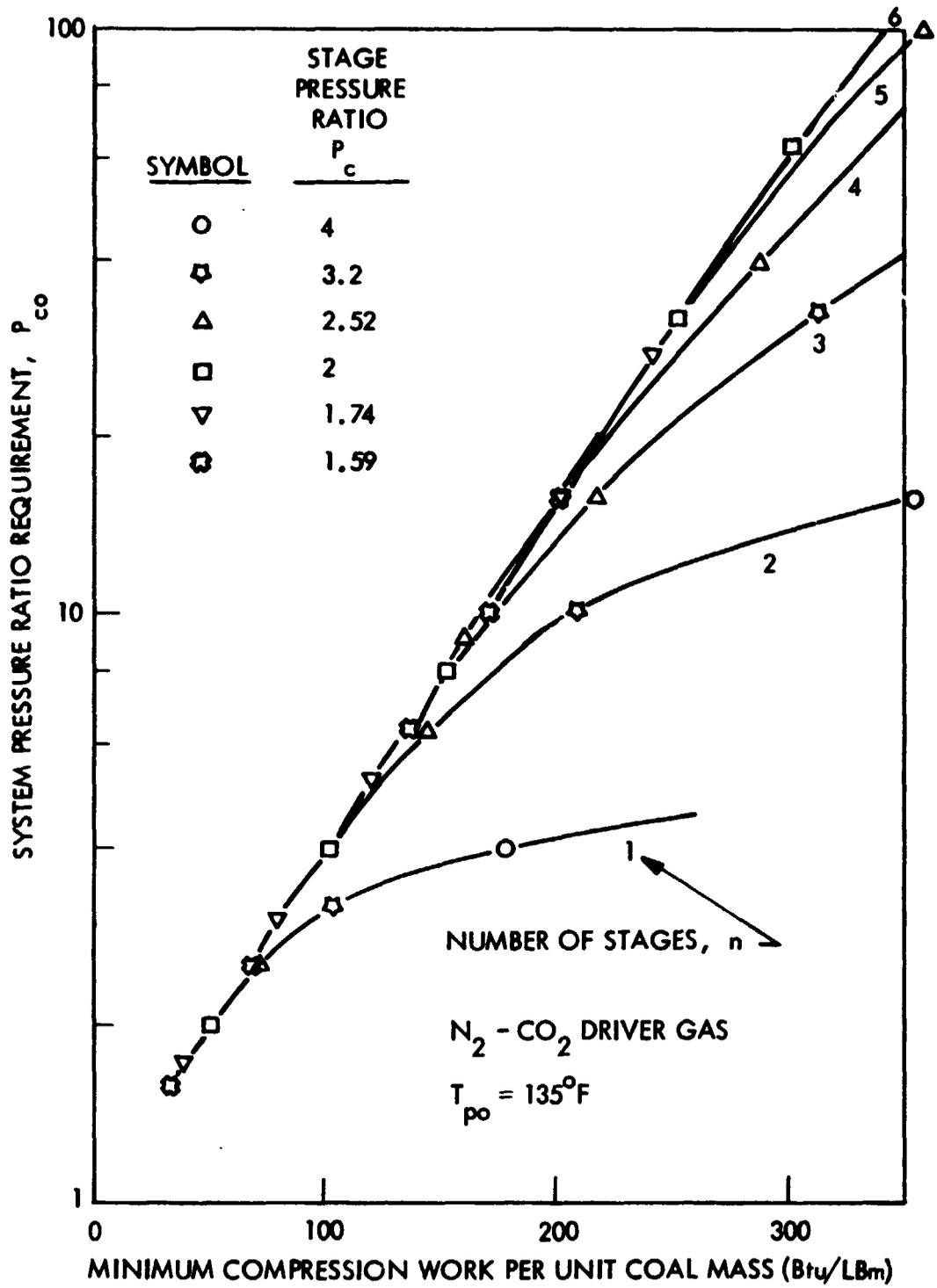
Fig. 8 Multistage Performance with Optimum Stages
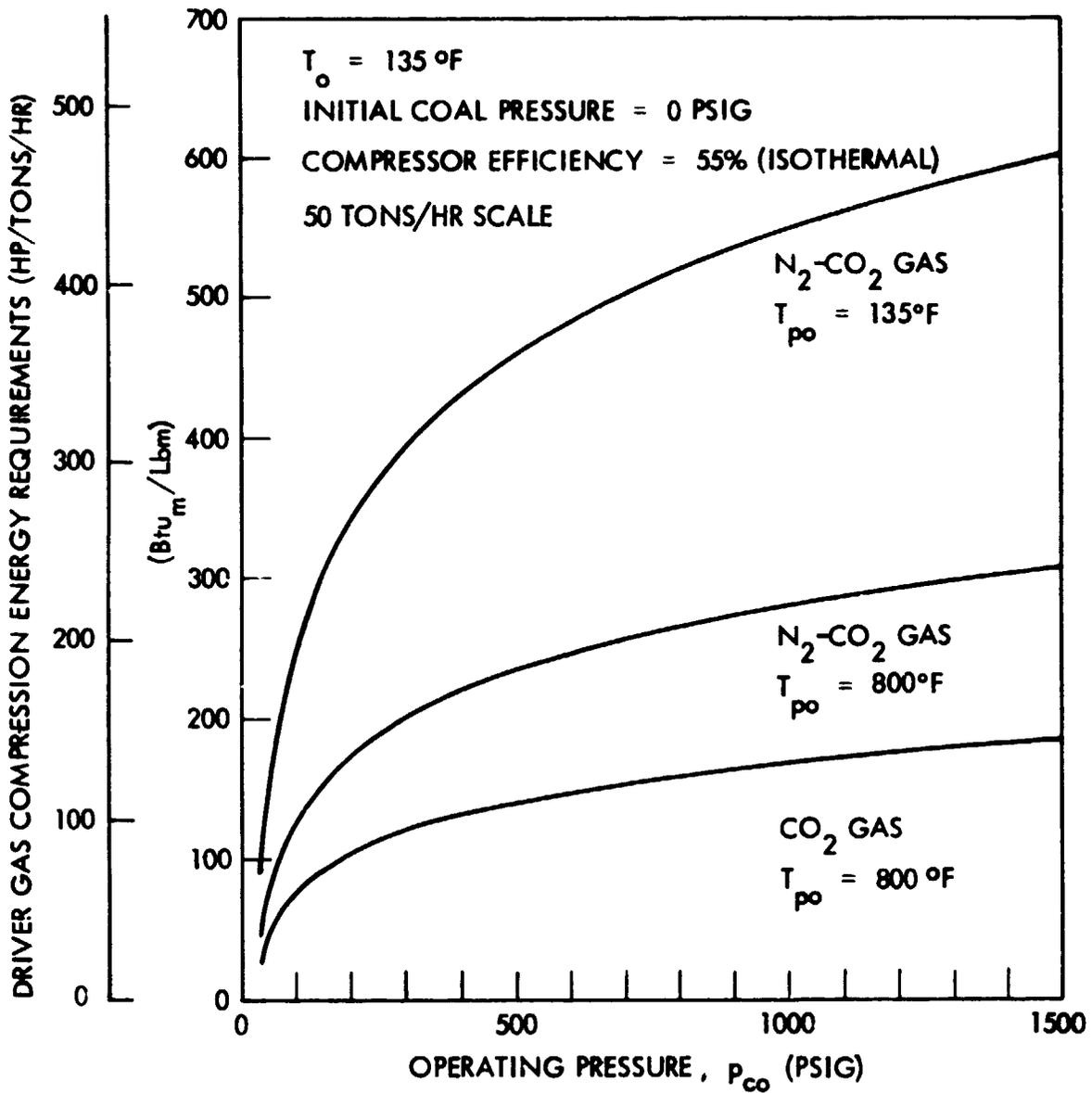
Fig. 9 Ejector System Compression Energy Requirements

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.
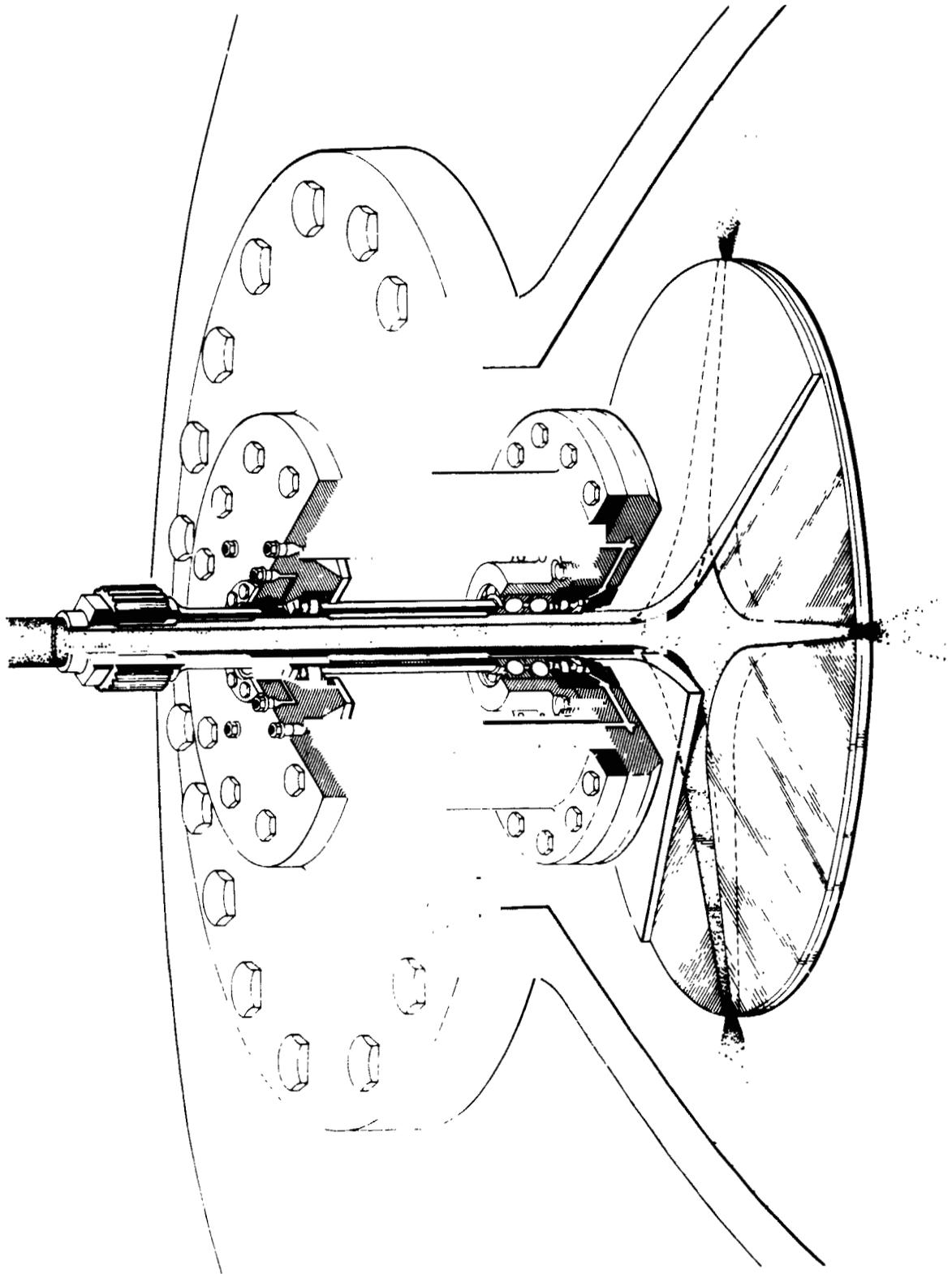
Fig. 10 Kinetic Extruder Concept

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
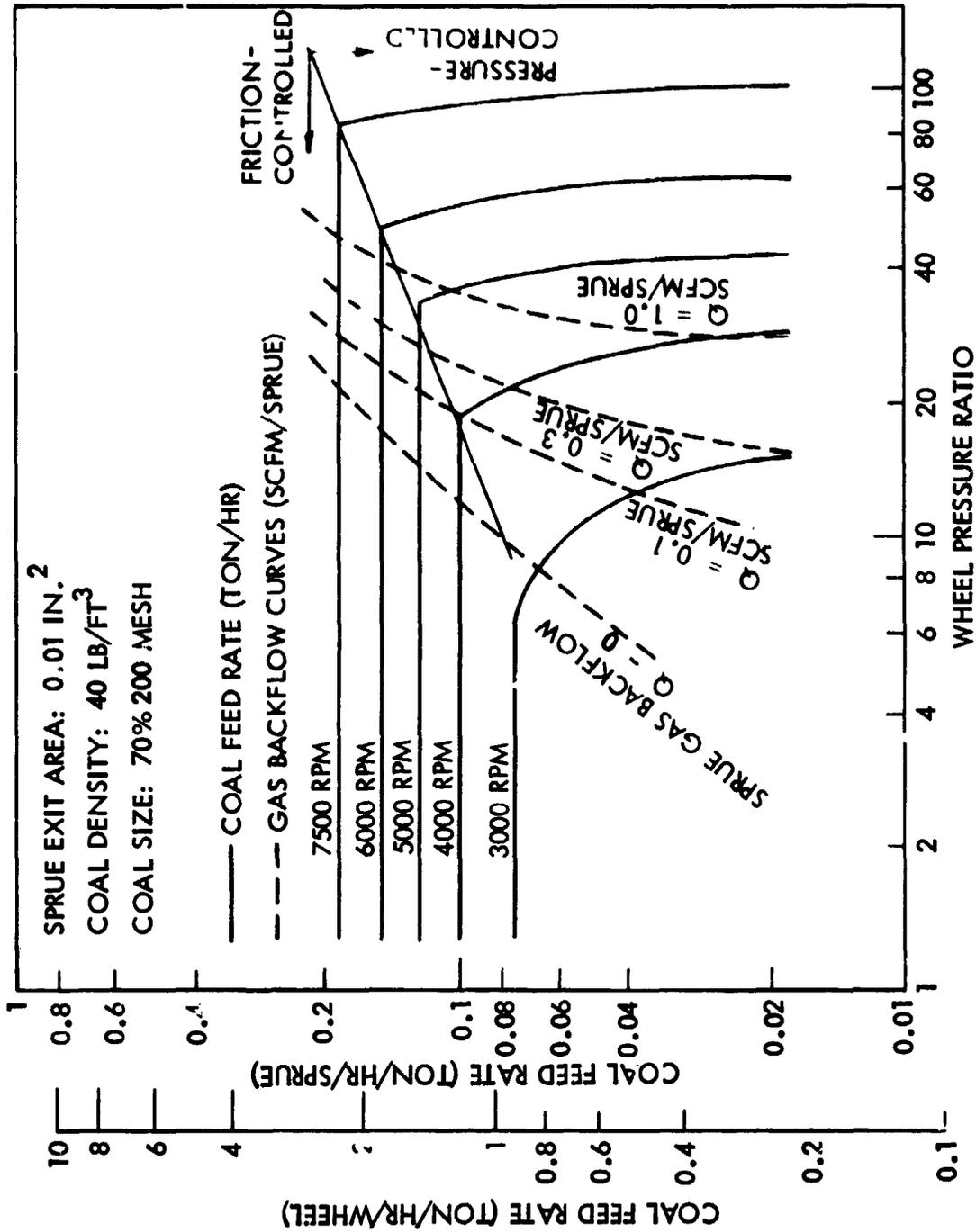
Fig. 11 Performance for Kinetic Extruder No. 2
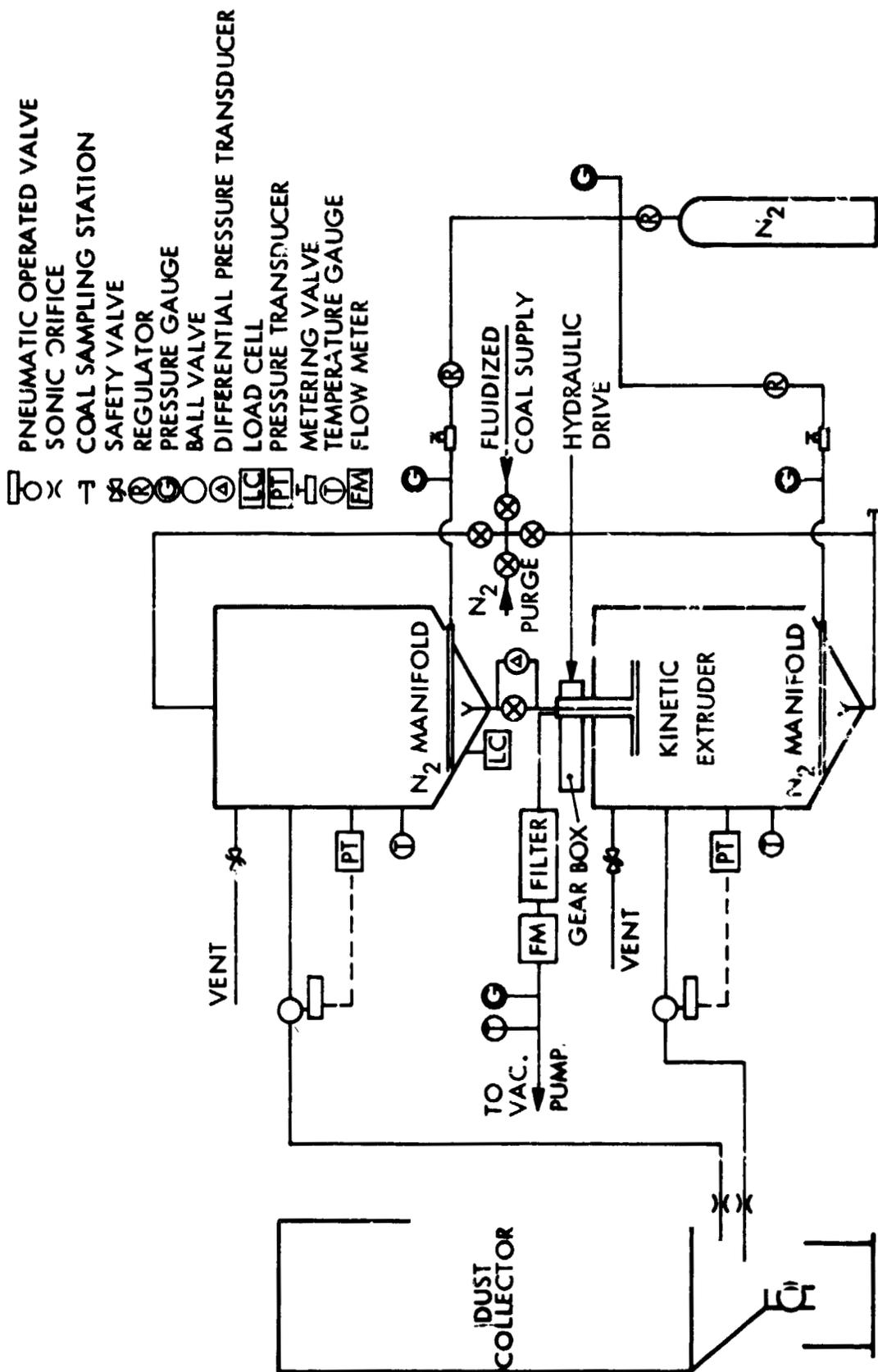
Fig. 12 Kinetic Extruder Test System Schematic
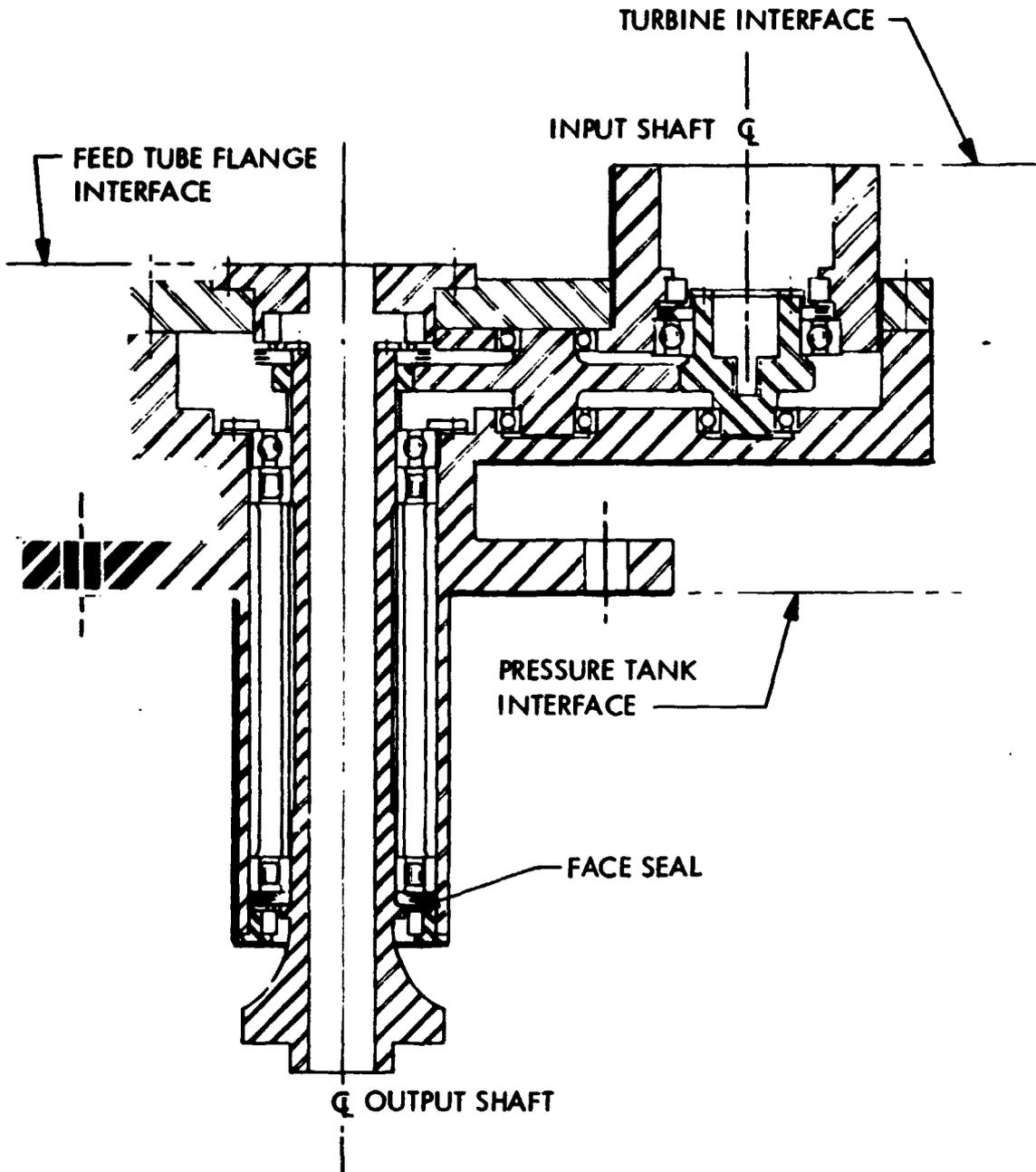
Fig. 13   Rotor Drive System
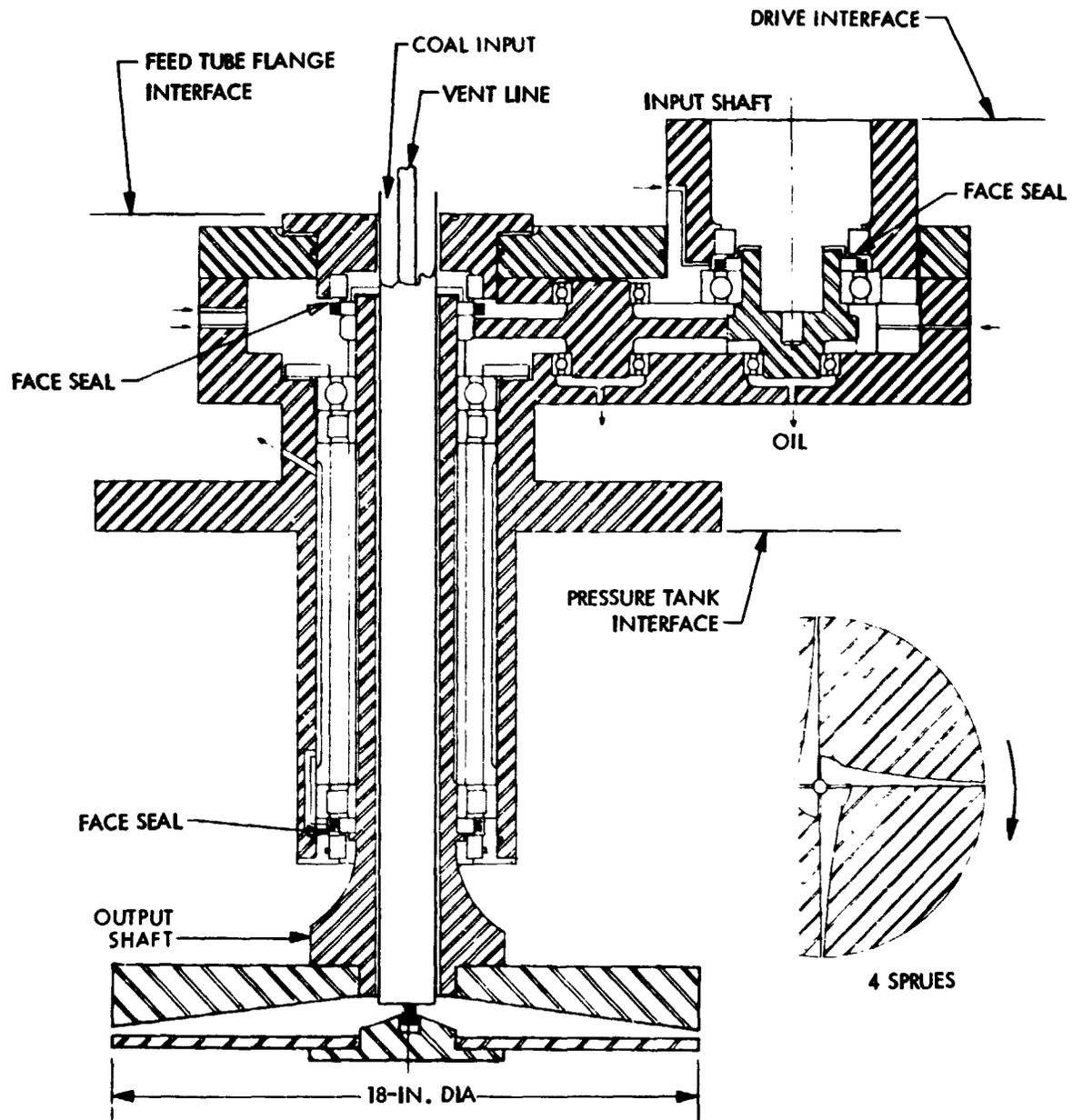
Fig. 14 Kinetic Extruder - Model No. 1 Configuration

COAL INPUT

FACE SEAL

VENT

PURGE

FACE SEAL

FACE SEAL

OIL

FACE SEAL

PURGE

FACE SEAL

PURGE

PURGE

BEARING

12 SPRUES

28 IN.
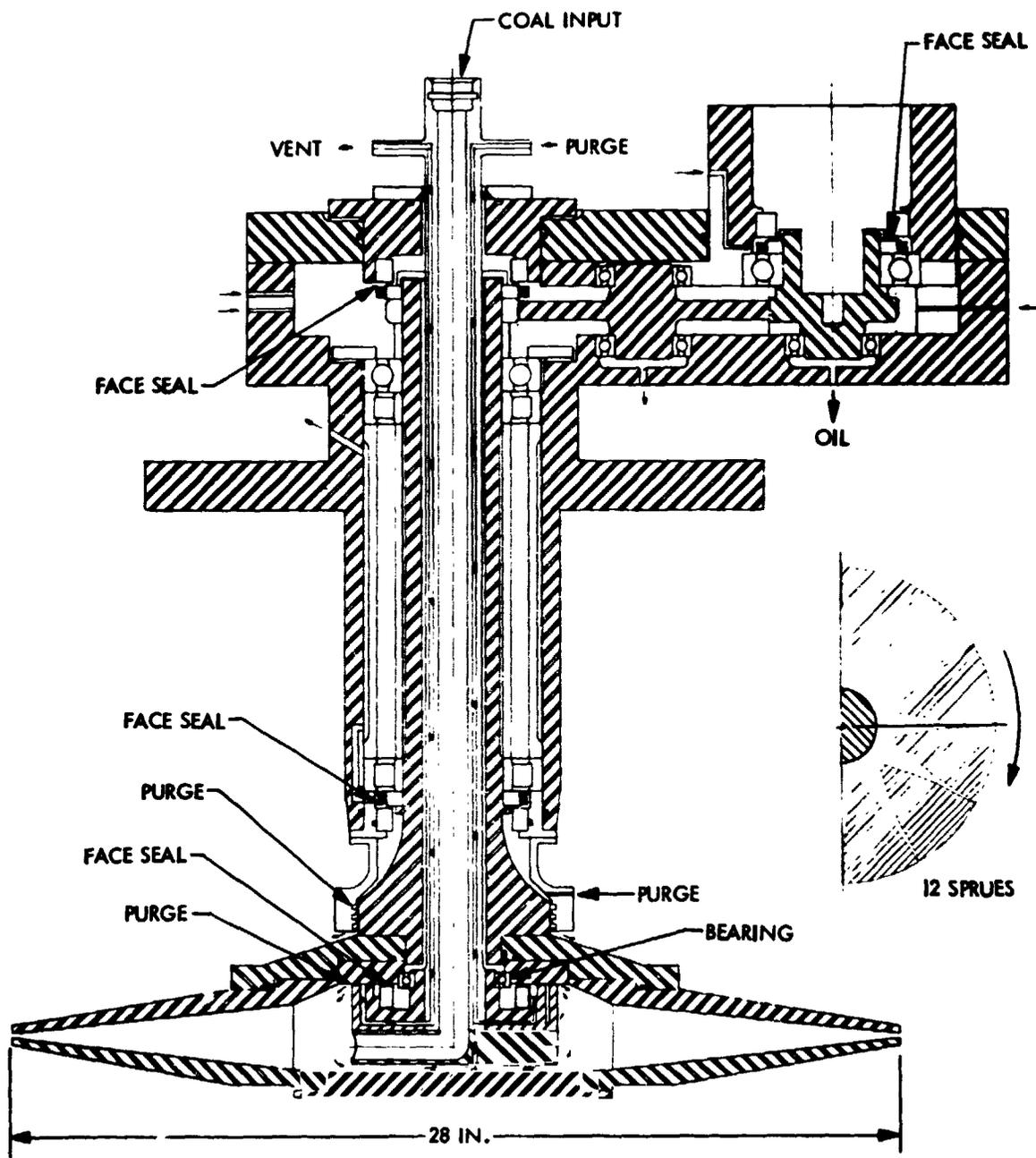
Fig. 15 Kinetic Extruder – Model No. 2 Configuration

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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 performance 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 recommendations 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
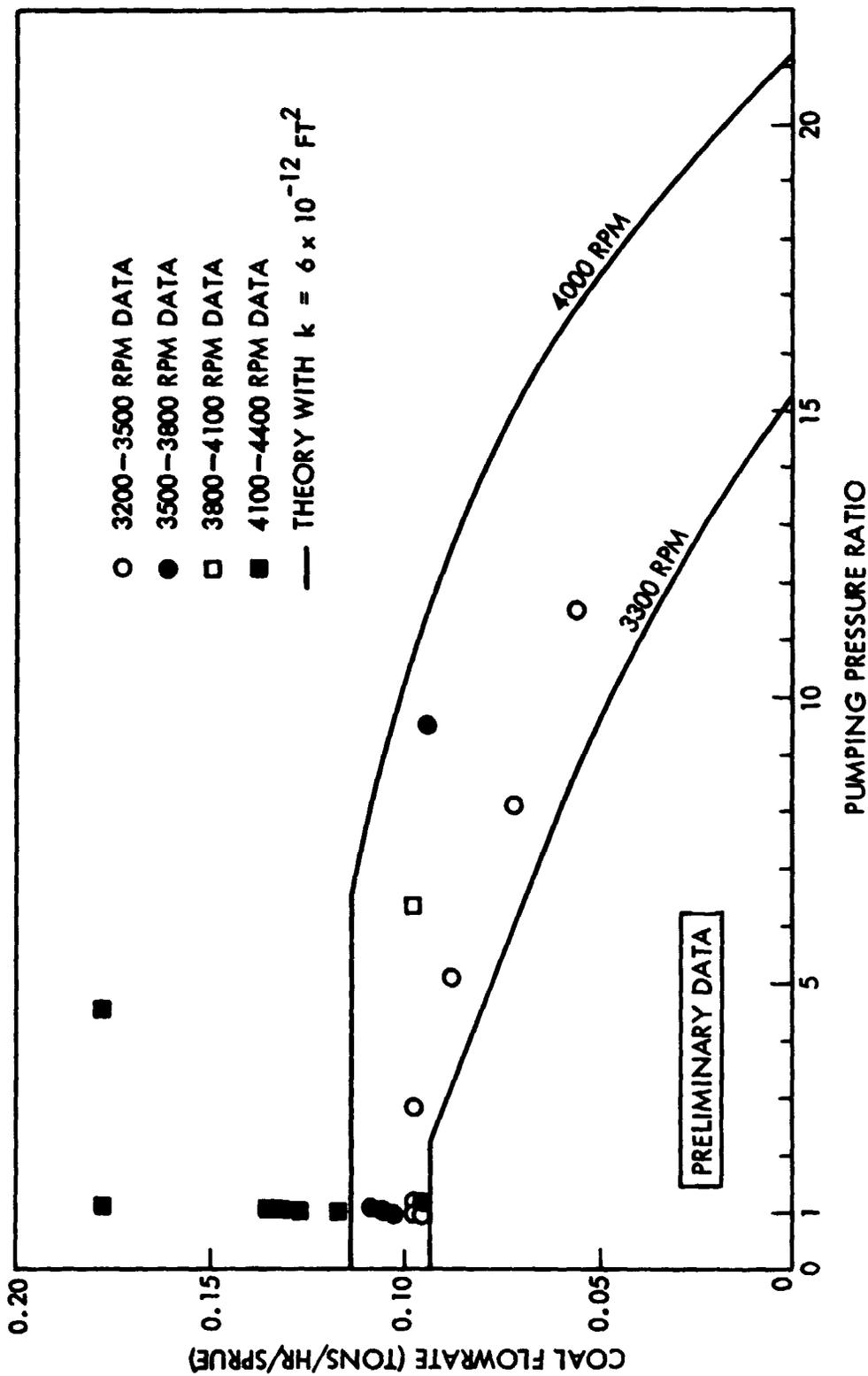
Fig. 16 Kinetic Extruder Model No. 2 Test Data. Inlet pressure 14.7 psia, fine cgal, outermost aeration port open, and sprues modified with epoxy filler, sprue exit area 0.006 in².

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
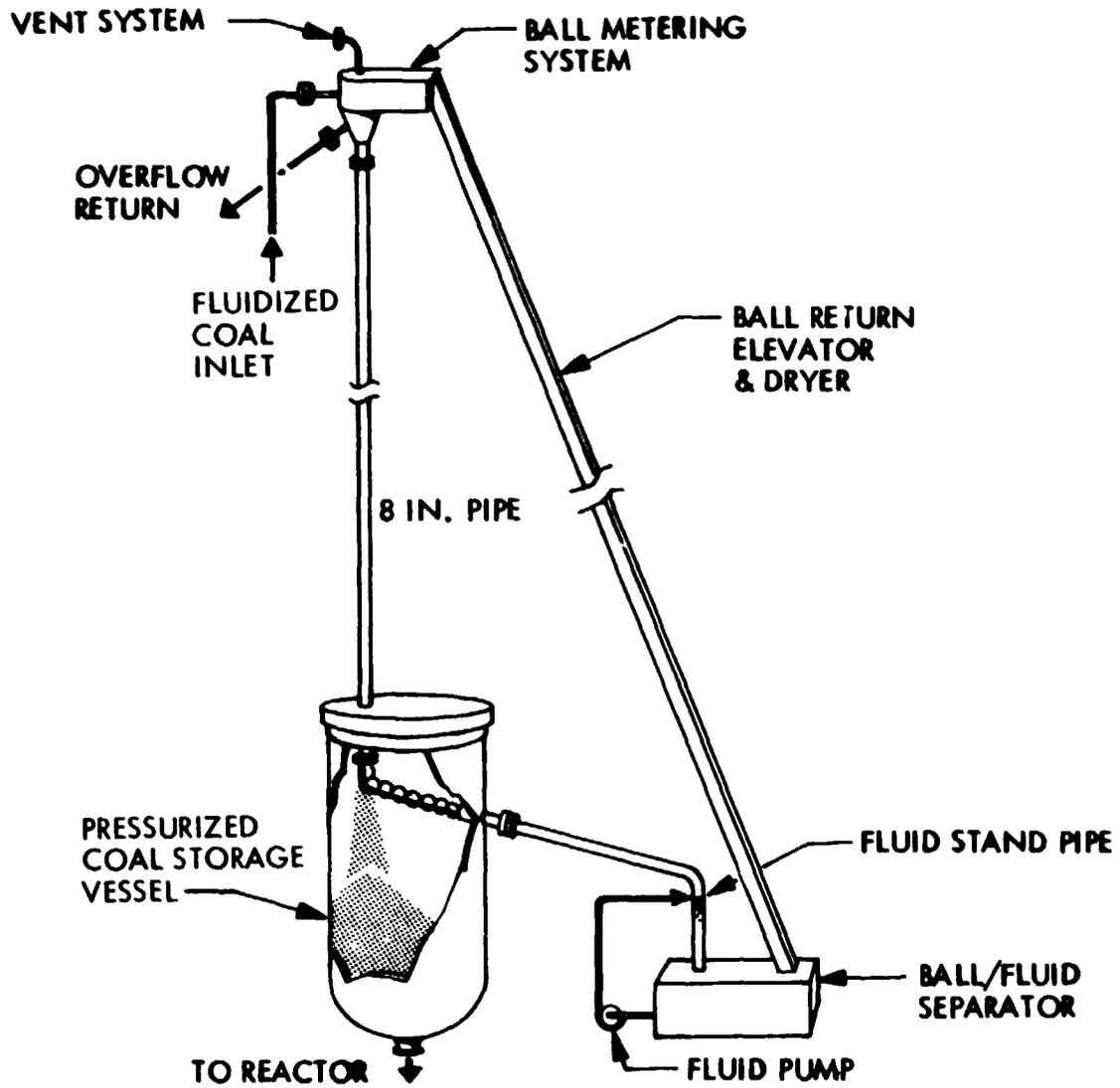
Fig. 17   Ball Conveyor

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.

TO BAG HOUSE
(PLUG FOR FLOW MEASUREMENT)

P.T.

FLOWMETER

18 LB STEEL BALLS

6 IN. O.D. STEEL TUBE

PRESSURE TRANSDUCER (P.T.) TAPS
AT 5 IN. INTERVALS ON ENTIRE
LENGTH

LOAD CELL

N_2 IN

N_2 OUT

RELIEF VALVE

LOAD CELL
CABLE

P.T.

DYNAMIC GAS
SEAL

HYDRAULIC CYLINDER

1 FT
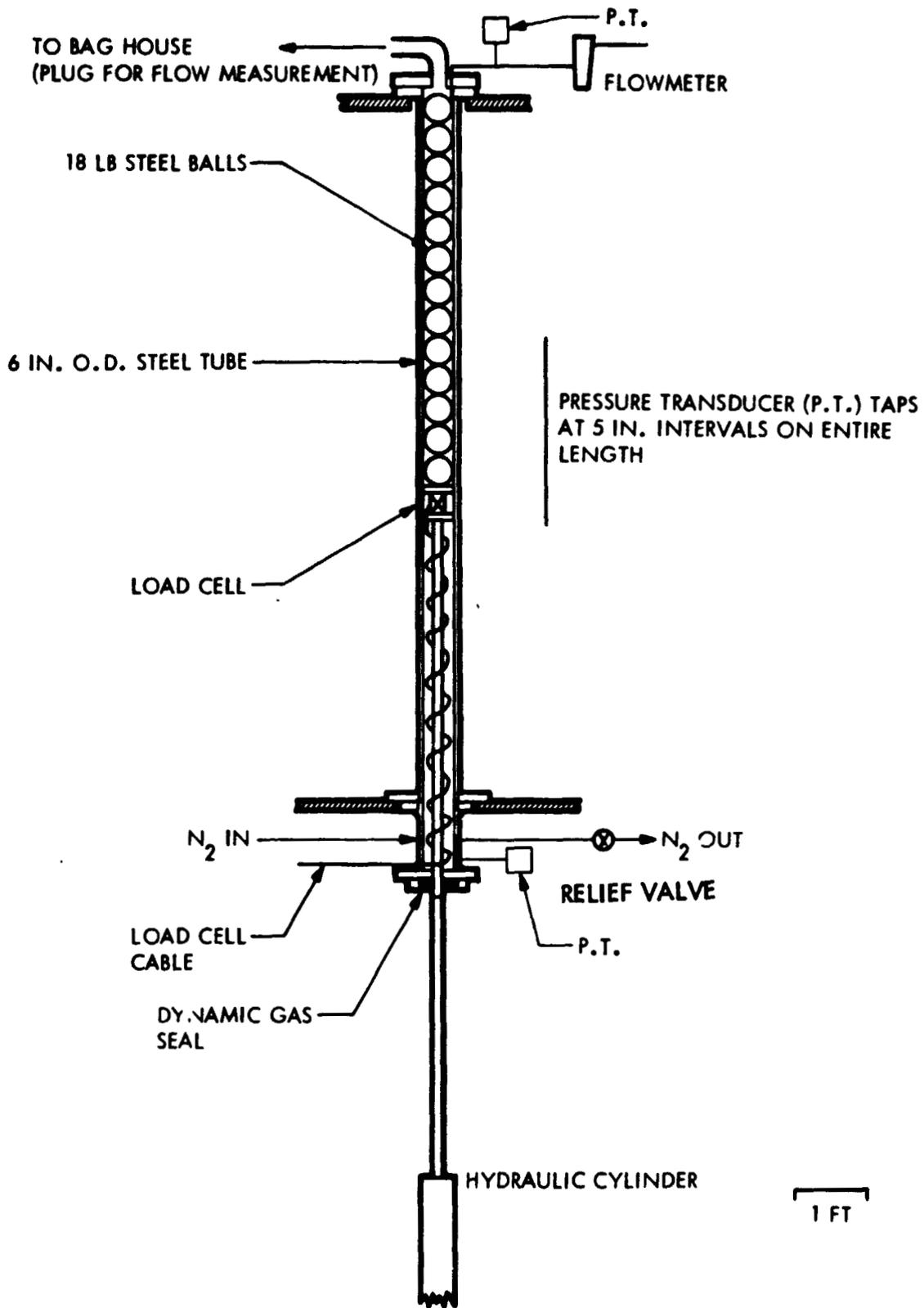
Fig. 18   Ball Conveyor Test Rig

LEGEND

O 563 g/BALL, 1/2 ft/s, FINE COAL

● 520 g/BALL, 1/2 ft/s, FINE COAL

◑ 550 g/BALL, 3/4 ft/s, FINE COAL

△ 580 g/BALL, 1/2 ft/s, COARSE COAL
SCREENED THROUGH NO 16 SIEVE

12 BALL COLUMN
5.25 IN. TUBE

LIMIT LINE

$\dfrac{\text{COLUMN FRICTION}}{\text{COLUMN WEIGHT}}$

$\dfrac{\text{COLUMN BASE PRESSURE}}{\text{COLUMN HEIGHT}} \left(\dfrac{\text{PSI}}{\text{FT}}\right)$
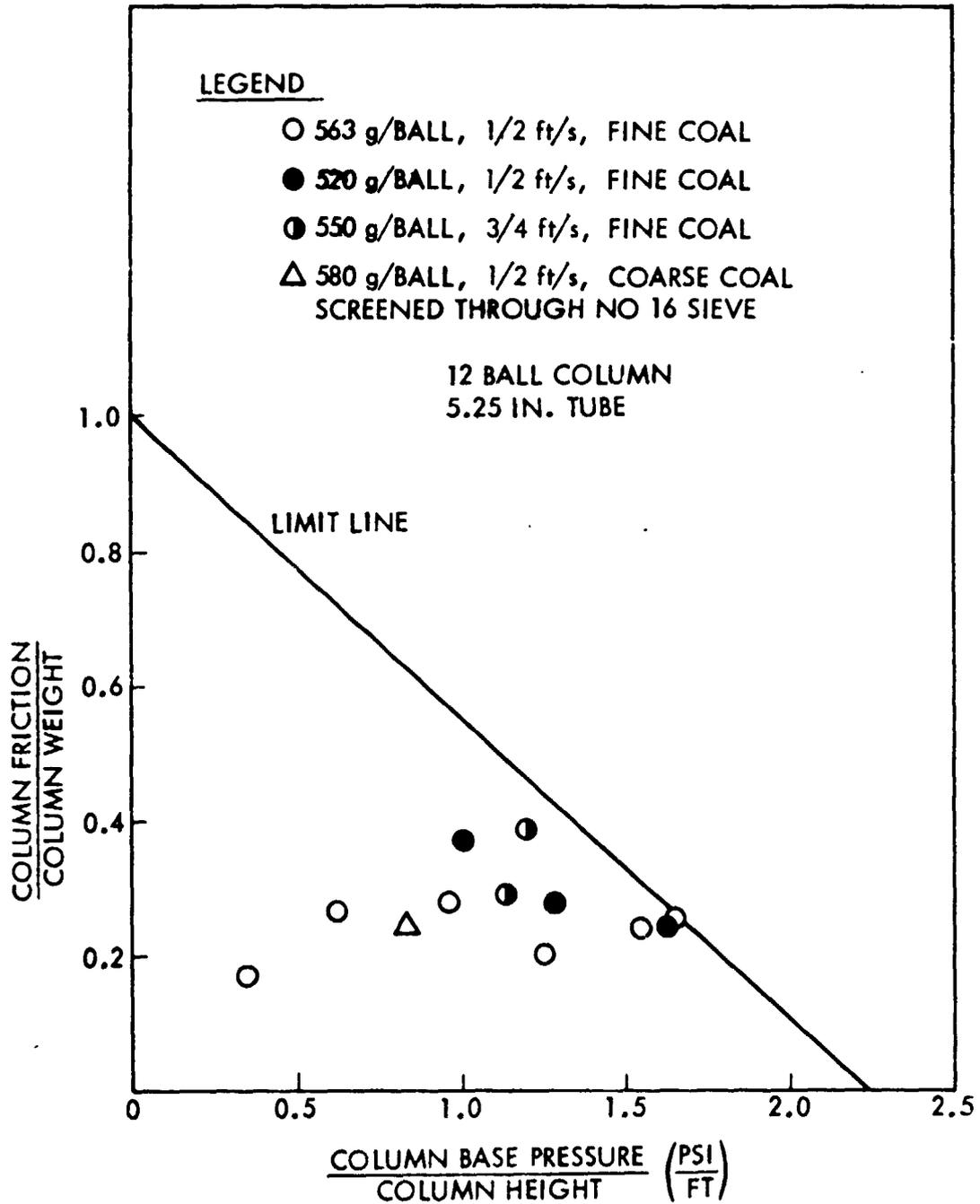
Fig. 19 Ball Conveyor Test Results
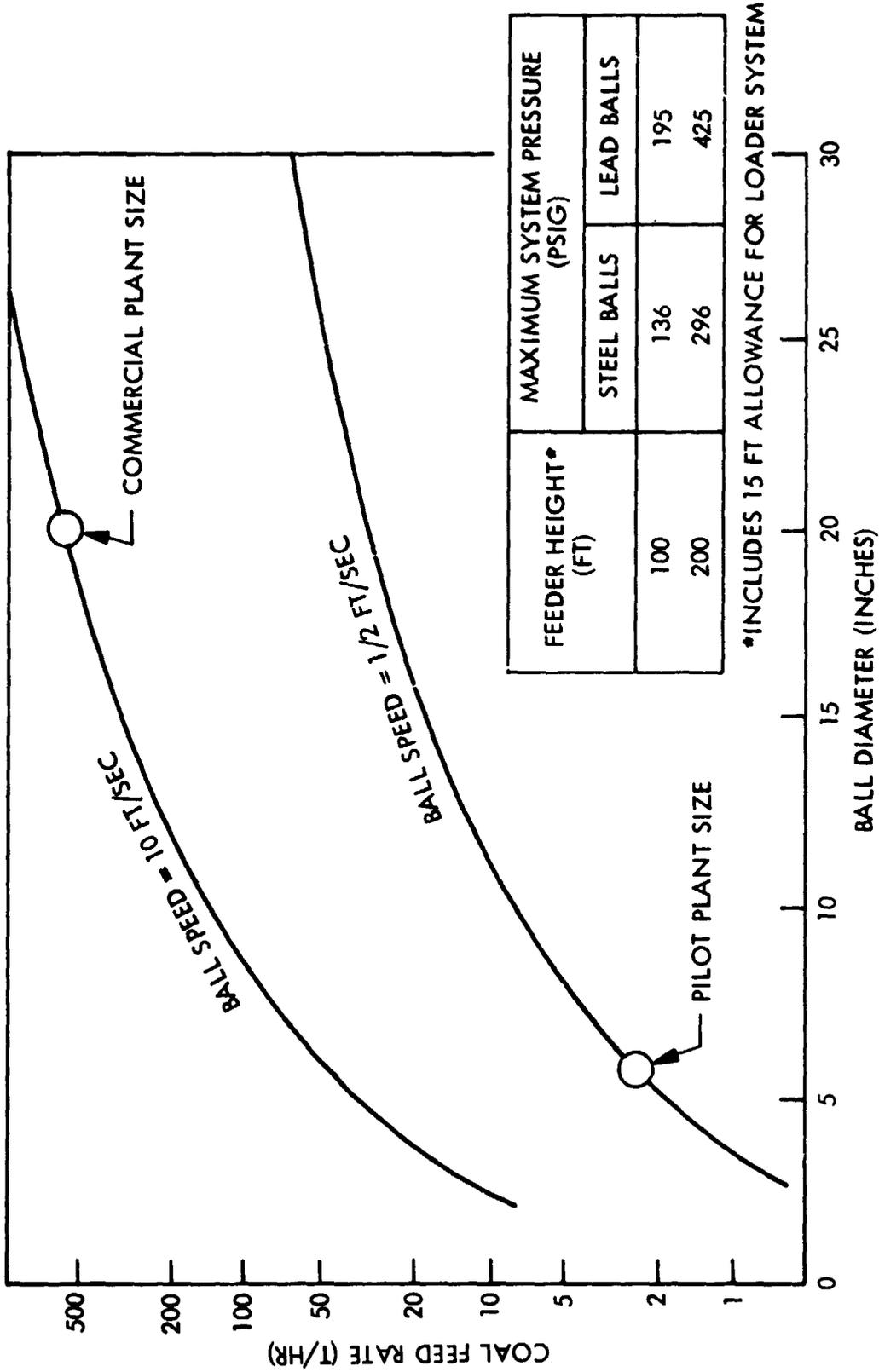
Fig. 20 Ball Conveyor Feed Rates
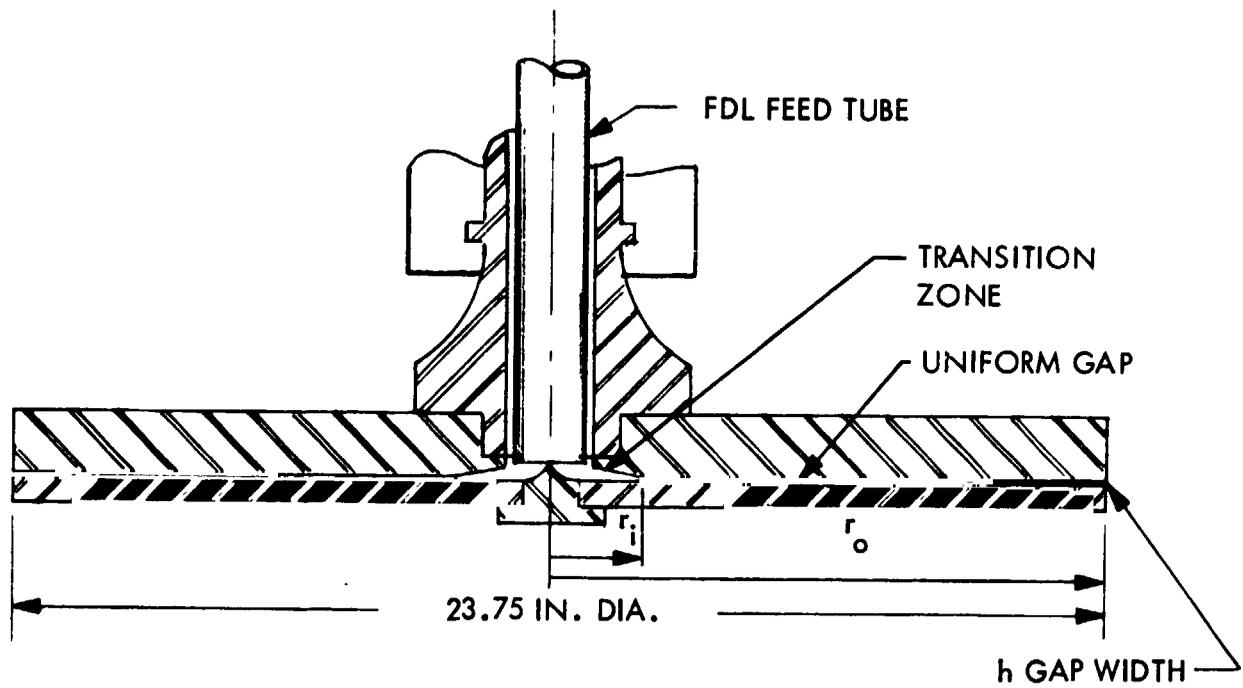
Fig. 21 Fluid Dynamic Lock — Rotor Assembly

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 uniformly 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 loading and the spacing between the disks.

A fluid dynamic lock was designed and built which is interchangeable with the kinetic extruder model No. 1. During the test runs, the same mechanical difficulties were encountered as with the kinetic extruder. 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 tests which permit the evaluation of the performance potential of the fluid dynamic lock in Pilot Plant use. Based on studies 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 coal 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.
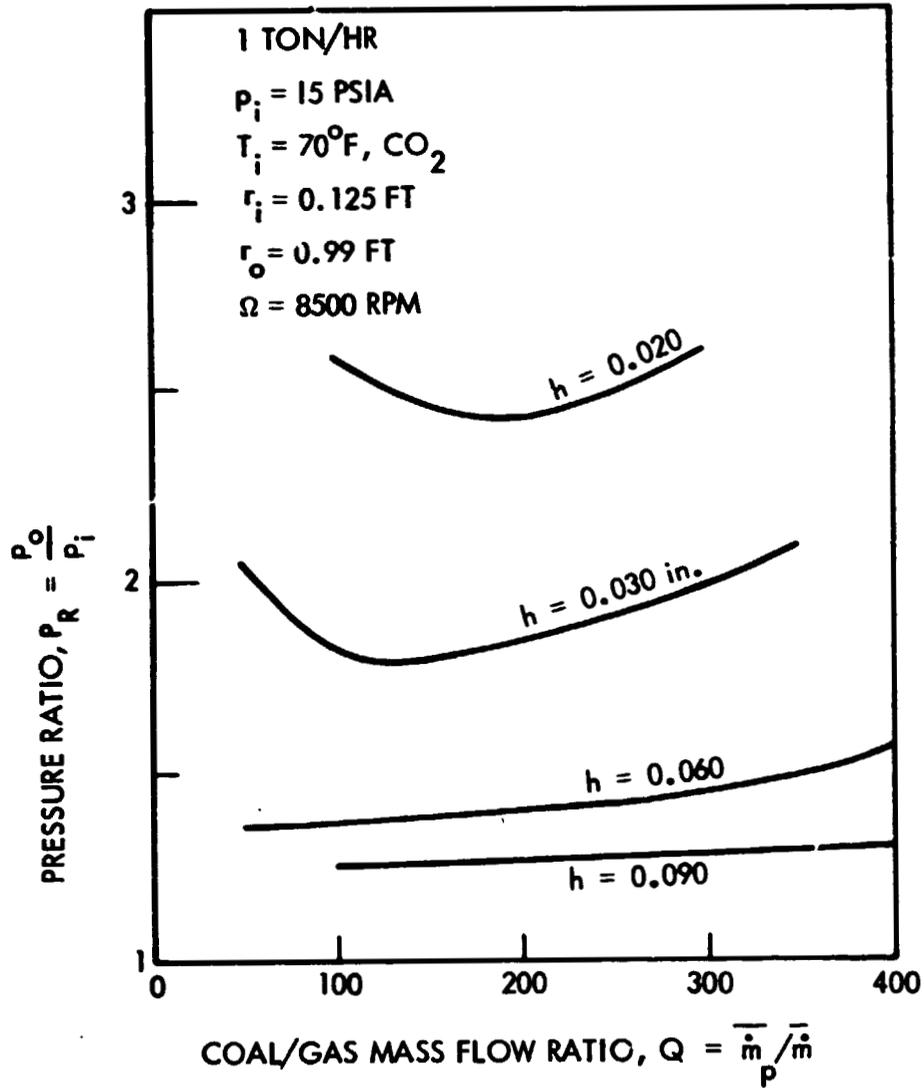
Fig. 22   FDL Performance Predictions.   Pressure Ratio As a Function of
Coal/Gas Flow Rate Ratio and Disk Spacing.
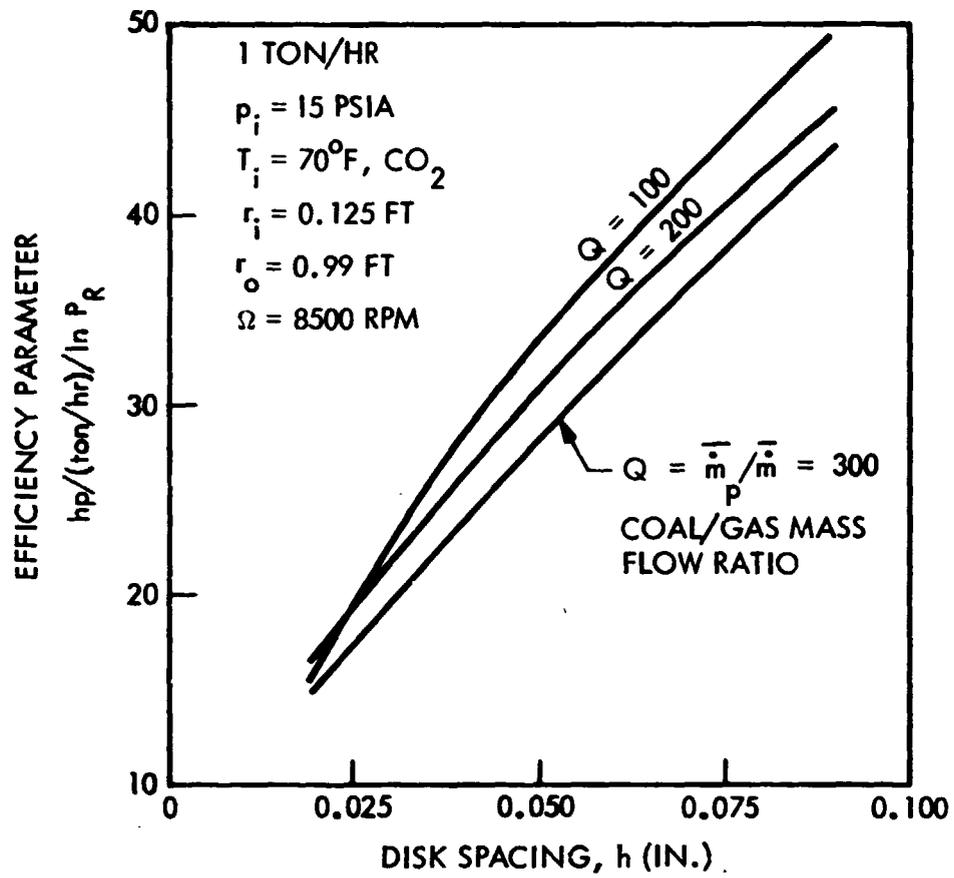
Fig. 23 FDL Performance Predictions. Efficiency Parameter as a Function of Coal/Gas Flow Rate Ratio and Disk Spacing

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