

December 1970

TMC Report S-985

FINAL REPORT

GASEOUS OXYGEN-GASEOUS HYDROGEN
ROCKET ENGINE INJECTOR VALVE TECHNOLOGY

NASA Contract No. NAS9-10086

TMC Project No. 5082

N71 - 17555

FACILITY FORM 602

(ACCESSION NUMBER)

235

(PAGES)

CR-114857

(NASA CR OR TMX OR AD NUMBER)

(THRU)

G3

(CODE)

15

(CATEGORY)

Prepared by

R. A. Lynch
R. A. Lynch

Approved by

H. Wichmann
H. Wichmann

T. Linton

T. Linton, Manager
Rocket Programs

NOTICE TO USERS

Portions of this document have been judged by the Clearinghouse to be of poor reproduction quality and not fully legible. However, in an effort to make as much information as possible available to the public, the Clearinghouse sells this document with the understanding that if the user is not satisfied, the document may be returned for refund....

If you return this document, please include this notice together with the IBM order card (label) to:

Clearinghouse
Attn: 152.12
Springfield, Va. 22151

TABLE OF CONTENTS

<u>SECTION</u>	<u>SUBJECT</u>	<u>PAGE</u>
I	Introduction	1
II	Summary	3
III	Parametric Valve Study	5
III-1	Flow Factor Determination	5
III-2	Parametric Trade-off	9
III-3	State-of-the-Art Review	10
III-4	Analysis of Performance and Characteristics	30
III-4A	Shut-off Device Performance	33
III-4A-1	Linear Motion	33
III-4A-2	Rotary Motion	35
III-4B	Actuator Performance	51
III-4B-1	Solenoid	51
III-4B-2	Pneumatic Cylinder	66
III-4B-3	Hydraulic Cylinder	92
III-4B-4	Torque Motor	93
III-4C	Shut-off Device/Actuator Interface Performance Consideration	112
III-4D	Physical Characteristics	113
III-4D-1	Linear Motion Shut-off Device	113
III-4D-2	Rotary Motion Shut-off Device	118
III-4D-3	Solenoid Actuator	118
III-4D-4	Pneumatic and Hydraulic Cylinder Actuators	126
III-4D-5	Torque Motor Actuator	126
III-4E	Design Layouts and Design Studies	131
III-4F	Trade-off Curves	143
III-4G	Utilizing Parametric Design Curves	150
III-4H	Valve Selection	162
III-5	Detail Design	171
IV	Conclusions	188
V	Recommendations	189
VI	Bibliography	190
Appendix A	APL Computer Program for Fast Acting Valves	
Appendix B	Sample of Design Curve Utilization to Determine a Valve's Characteristics	

LIST OF FIGURES

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
I-1	Valve Study Program Plan - Flow Diagram	2
III-1	Fuel Valve $C_d A_t$ Required vs. Inlet Pressure	7
III-2	Oxidizer Valve $C_d A_t$ Required vs. Inlet Pressure	8
III-3	Weighting Factor Determination - Emphasis Curve Method/Shut-off Device	11
III-4	Stroke per Convolution Capability of CRES Hydrogen Bellows	24
III-5	Free Length (in/conv) Capability - CRES Hydroform Bellows	25
III-6	Span of CRES Hydroform Bellows	26
III-7	Free Length Required - Nesting Ripple Welded Metal Bellows	27
III-8	Stroke per Convolution Capability - Nesting Ripple Welded Metal Bellows	28
III-9	Span of Nesting Ripple Welded Metal Bellows	29
III-10	Flow Diagram for the Valve Analysis Program	32
III-11	Linear Motion Shut-off Device - Schematic Drawing	34
III-12	Linear Motion Shut-off Device Moving Mass vs. Flow Factor ($C_d A_t$)	36
III-13	Linear Motion Shut-off, Valve Force Required vs. Flow Factor, $t = .01$ sec	41
III-14	Linear Motion Shut-off, Valve Force Required vs. Flow Factor, $t = .02$ sec	42
III-15	Linear Motion Shut-off, Valve Force Required vs. Flow Factor, $t = .03$ sec	43
III-16	Linear Motion Shut-off, Valve Force Required vs. Flow Factor, $t = .05$	44
III-17	Linear Motion Shut-off, Valve Force Required vs. Flow Factor, $t = .07$ sec	45
III-18	Linear Motion Shut-off, Valve Force Required vs. Flow Factor, $t = .10$ sec	46
III-19	Linear Motion Shut-off Device Flow Factor ($C_d A_t$) vs. Acceleration for various motion times	47
III-20	Rotary Motion Shut-off Device - Schematic Drawing	48
III-21	Valve Torque vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for Rotary Motion Shut-off Device, $t = .01$ sec	56
III-22	Valve Torque vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for Rotary Motion Shut-off Device, $t = .02$ sec	57
III-23	Valve Torque vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for Rotary Motion Shut-off Device, $t = .03$ sec	58
III-24	Valve Torque vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for Rotary Motion Shut-off Device, $t = .05$ sec	59
III-25	Valve Torque vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for Rotary Motion Shut-off Device, $t = .07$ sec	60
III-26	Valve Torque vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for Rotary Motion Shut-off Device, $t = .10$ sec	61
III-27	Rotary Shut-off Device - Ball Diameter vs. Flow Factor	62

LIST OF FIGURES (Continued)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
III-28	Rotary Motion Shut-off Device - Moving Mass vs. Flow Factor	63
III-29	Solenoid Actuator - Schematic Drawing	64
III-30	Pneumatic Cylinder Actuator - Schematic Drawing	72
III-31	Cylinder Output Force vs. Piston Diameter, $P_1 = 20$ psia	84
III-32	Cylinder Output Force vs. Piston Diameter, $P_1 = 30$ psia	85
III-33	Cylinder Output Force vs. Piston Diameter, $P_1 = 60$ psia	86
III-34	Cylinder Output Force vs. Piston Diameter, $P_1 = 150$ psia	87
III-35	Cylinder Output Force vs. Piston Diameter, $P_1 = 400$ psia	88
III-36	Auxiliary Pressure Pneumatic Cylinder Output Force vs. Piston Diameter	89
III-37	Pneumatic Cylinder Actuator - Pneumatic Cylinder Delay vs. Required Motion Time (O_2 gas media)	90
III-38	Hydraulic Cylinder Output Force vs. Piston Diameter	99
III-39	Hydraulic Cylinder Actuator - Hydraulic Delay vs. Required Motion Time	100
III-40	Torque Motor Actuator - Schematic Drawing	101
III-41	Torque Motor Output vs. Motion Time-Torque Motor Length = .4 inches	106
III-42	Torque Motor Output vs. Motion Time-Torque Motor Length = .6 inches	107
III-43	Torque Motor Output vs. Motion Time-Torque Motor Length = .8 inches	108
III-44	Torque Motor Output vs. Motion Time-Torque Motor Length = 1.0 inches	109
III-45	Torque Motor Output vs. Motion Time-Torque Motor Length = 1.25 inches	110
III-46	Torque Motor Output vs. Motion Time-Torque Motor Length = 1.50 inches	111
III-47	Torque to Linear - Force Required Conversion Factor vs. Acceleration	114
III-48	Actuator Linear Force to Valve Torque Conversion Factor vs. Acceleration	115
III-49	Linear Motion - Shut-off Device - Physical Characteristic Schematic	116
III-50	Linear Motion Shut-off Device Envelope Dimensions vs. Flow Factor	119
III-51	Linear Motion Shut-off Device Weight vs. Flow Factor	120
III-52	Rotary Motion Shut-off Device - Physical Characteristic Schematic	121
III-53	Rotary Shut-off Device Envelope Dimensions vs. Flow Factor	122
III-54	Rotary Motion Shut-off Device Weight vs. Flow Factor	123
III-55	Solenoid Actuator - Physical Characteristics Schematic	124
III-56	Hydraulic or Pneumatic Cylinder Actuator-Physical Characteristics Schematic	127
III-57	Cylinder Actuator Envelope Dimensions vs. Piston Diameter	128
III-58	Line Pressure Operated Linear Cylinder Weight vs. Piston Diameter	129

LIST OF FIGURES (Continued)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
III-59	Auxiliary Pressure Operated Cylinder Actuator - Piston Diameter vs. Actuator Weight	130
III-60	Torque Motor Envelope Dimensions for Various Power Levels	132
III-61	Torque Motor Weight vs. Length at Various Power Levels	133
III-62	Double Acting Poppet Valve	134
III-63	Vall Valve	136
III-64	Coaxial Poppet Valve	137
III-65	3-Way Pilot, Line Pressure Pneumatic Operated Poppet Valve	138
III-66	Dual Piloted, Line Pressure Pneumatic Operated Poppet Valve	140
III-67	Torque Motor Operated Ball Valve	141
III-68	Auxiliary Pressure Pneumatic Operated Ball Valve	142
III-69	Auxiliary Pressure Pneumatic Operated Poppet Valve	144
III-70	Torque Motor, Ball Screw Operated Poppet Valve	145
III-71	Auxiliary Pressure Hydraulic Operated Poppet Valve	146
III-72	Configuration Feasibility Chart for Opening Response less than .020 seconds	147
III-73	Configuration Feasibility Chart for Opening Response less than .050 seconds	148
III-74	Configuration Feasibility Chart for Opening Response less than .100 seconds	149
III-75	Weight vs. Flow Factor at Various Inlet Pressures for a Line Pressure Pneumatic Cylinder Operated Linear Motion Valve, $t_{open} \leq 10$ ms	151
III-76	Weight vs. Flow Factor at Various Inlet Pressures for a Line Pressure Pneumatic Cylinder Operated Rotary Motion Valve, $t_{open} \leq 10$ ms	152
III-77	Weight vs. Flow Factor ($C_d A_t$) at Valve Inlet Pressure from 20 to 400 psia, for an Auxiliary Pressure Cylinder Operated Linear Motion Valve, $t_{open} \leq 10$ ms	153
III-78	Weight vs. Flow Factor ($C_d A_t$) at Valve Inlet Pressures from 20 to 400 psia, for a Hydraulic Cylinder Operated Valve, $t_{open} \leq 10$ ms	154
III-79	Weight of Torque Motor Operated Linear Motion Valve, $t_{open} \leq 10$ ms	155
III-80	Weight vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures for a Line Pressure Pneumatic Cylinder Operated Linear Motion Valve, $T_{open} \leq 80$ ms	156
III-81	Weight vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures, for a Line Pressure Pneumatic Cylinder Operated Rotary Motion Valve, $T_{open} \leq 80$ ms	157
III-82	Weight vs. Flow Factor ($C_d A_t$) at Valve Inlet Pressures from 20 to 400 psia, for an Auxiliary Pressure Pneumatic Cylinder Operated Linear Motion Valve, $t_{open} \leq 80$ ms $P_{aux.} = 1500$ to 3000 psia	158

LIST OF FIGURES (Continued)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
III-83	Weight vs. Flow Factor ($C_d A_t$) at Various Inlet Pressures from 20 to 400 psia, for an Auxiliary Pressure Pneumatic Cylinder Operated Rotary Motion Valve, $P_{aux.} = 1500$ to 3000 psia, $t_{open} \leq 80$ ms	159
III-84	Weight vs. Valve Inlet Pressure at Various Valve Flow Factors and Electrical Power Levels for a Torque Motor Operated Linear Motion Valve, $t_{open} \leq 80$ ms	160
III-85	Weight vs. Valve Inlet Pressure at Various Valve Flow Factors and Electrical Power Levels for a Torque Motor Operated Rotary Motion Valve, $t_{open} \leq 80$ ms	161
III-86	Valve Performance Characteristics - Rating Charts and Limits	169
III-87	Ox Valve $C_d A_t$ required vs. P_1 ($\Delta P = .04 P_1$)	172
III-88	Fuel Valve $C_d A_t$ required vs. P_1 ($\Delta P = .04 P_1$)	173
III-89	O_2/H_2 Pilot Operated Coaxial Shut-off Valve (400 psia)	184
III-90	O_2/H_2 Pilot Operated Coaxial Shut-off Valve (22 psia)	186

LIST OF TABLES

<u>Table No.</u>	<u>Title</u>	<u>Page</u>
III-I	Range of Interest	6
III-II	Valve - Shut-off Device	12
III-III	Valve - Guidance (Linear)	13
III-IV	Valve - Guidance (Rotary)	14
III-V	Valve - Dynamic Seal (Linear Motion)	15
III-VI	Valve - Dynamic Seal (Rotary Motion)	16
III-VII	Valve - Static Seal	17
III-VIII	Valve - Shut-off Seal	18
III-IX	Valve Shut-off Seal Alignment	19
III-X	Actuator Type	20
III-XI	Selected Valve Detail Configuration Candidates	21
III-XII	Valve Element - Motion Characteristics	31
III-XIII	Linear Motion Shut-off Device Data Summary	37
III-XIV	Rotary Motion Shut-off Device Data Summary	52
III-XV	Solenoid Actuator Data Summary	67
III-XVI	Pneumatic Cylinder Actuator Data Summary	76
III-XVII	Pneumatic Media Correction	91
III-XVIII	Hydraulic Cylinder Actuator Data Summary	94
III-XIX	Torque Motor Actuator Data Summary	103
III-XX	Instructions for Utilizing Design Curves to Predict Performance and Physical Characteristics of Valve Configurations	163
III-XXI	Analysis Summary - Fuel Valve, 1000 lb thrust, O/F = 2.0, P_1 = 30 psia	175
III-XXII	Analysis Summary - Ox Valve, 1000 lb thrust, O/F = 2.0, P_1 = 30 psia	177
III-XXIII	Analysis Summary - Fuel Valve, 1000 lb thrust, O/F = 2.0, P_1 = 400 psia	179
III-XXIV	Analysis Summary - Ox Valve, 1000 lb thrust, O/F = 2.0, 400 psia	181
III-XXV	Analysis Summary - Fuel Valve, 1000 lb thrust, O/F = 2.0, P_1 = 22 psia	183

LIST OF ABBREVIATIONS

<u>Abbreviation</u>	<u>Description - Units</u>
A_t	Flow area - in. ²
A_m	Solenoid magnetic area - in. ²
a	Acceleration - g's
B	Magnetic flux density - kilogauss
b	Flange thickness - inches
C_1	Constant, of pneumatic media
C_d	Equivalent orifice coefficient
$C_d A_t$	Flow factor - in. ²
D	Diameter - inches
D_A	Armature diameter - inches
D_B	Diameter of Ball - inches
D_{c_o}	Coil Outside Diameter - inches
D_m	Motor maximum diameter - inches
D_o	Flow Passage maximum diameter - inches
D_p	Piston diameter - inches
D_s	Seat sealing Diameter - inches
d	Coil wire diameter - inches
d_r	Piston rod diameter - inches
E	Young's modulus - psi
F	Force - lb
F_A	Total Actuator force developed - lb
F_V	Shut-off device opening force required - lb
F_{V_c}	Shut-off device closing force required - lb
f	Coefficient of friction
H	Magnetomotive force - OERSTAD
I	Electrical current - amperes
I_{xx}	Moment of inertia - in.-lb-sec ²
k	Specific heat ratio
L	Inductance - Henries
l	Length - inches
l_c	Coil length - inches
l_{iron}	Iron path length - inches
M	Mass - lbf

LIST OF ABBREVIATIONS (Continued)

<u>Abbreviation</u>	<u>Description - Units</u>
M_A	Actuator moving mass - lbf
M_S	Solenoid moving mass - lbf
M_V	Shut-off device moving mass - lbf
N	Number
N_C	Number of turns of wire in solenoid coil
N_I	Ampere turns of solenoid coil
P	Pressure - psia
P_1	Valve inlet pressure - psia
P_2	Valve outlet pressure - psia
$P_{aux.}$	Auxiliary pressurant supply pressure - psia
P_C	Cylinder closing chamber pressure - psia
P_O	Cylinder opening chamber pressure - psia
Q	Volumetric flow rate - in. ³ /sec
R	Gas constant - $\frac{\text{in.} \cdot \text{lb}}{\text{lb} \cdot ^\circ\text{R}}$
R_C	Solenoid coil resistance - ohms
s	Stroke - inches
T	Temperature, - $^\circ\text{R}$
T_V	Torque required - in.-lb
T_T	Total torque developed - in. lb
t	Shut-off device travel time - sec
t_i	Opening delay time - sec
t_2	Closing delay time - sec
W	electrical power - watts
W_t	Weight - lbs
w	Wall thickness - inches
$\dot{w}$	Flow rate - $\frac{\text{lb}}{\text{sec}}$
ρ	Density - $\frac{\text{lb}}{\text{in.}^3}$
θ	Seat half angle - degrees
$\dot{\omega}$	Angular acceleration - $\frac{\text{rad.}}{\text{sec}^2}$

I. INTRODUCTION

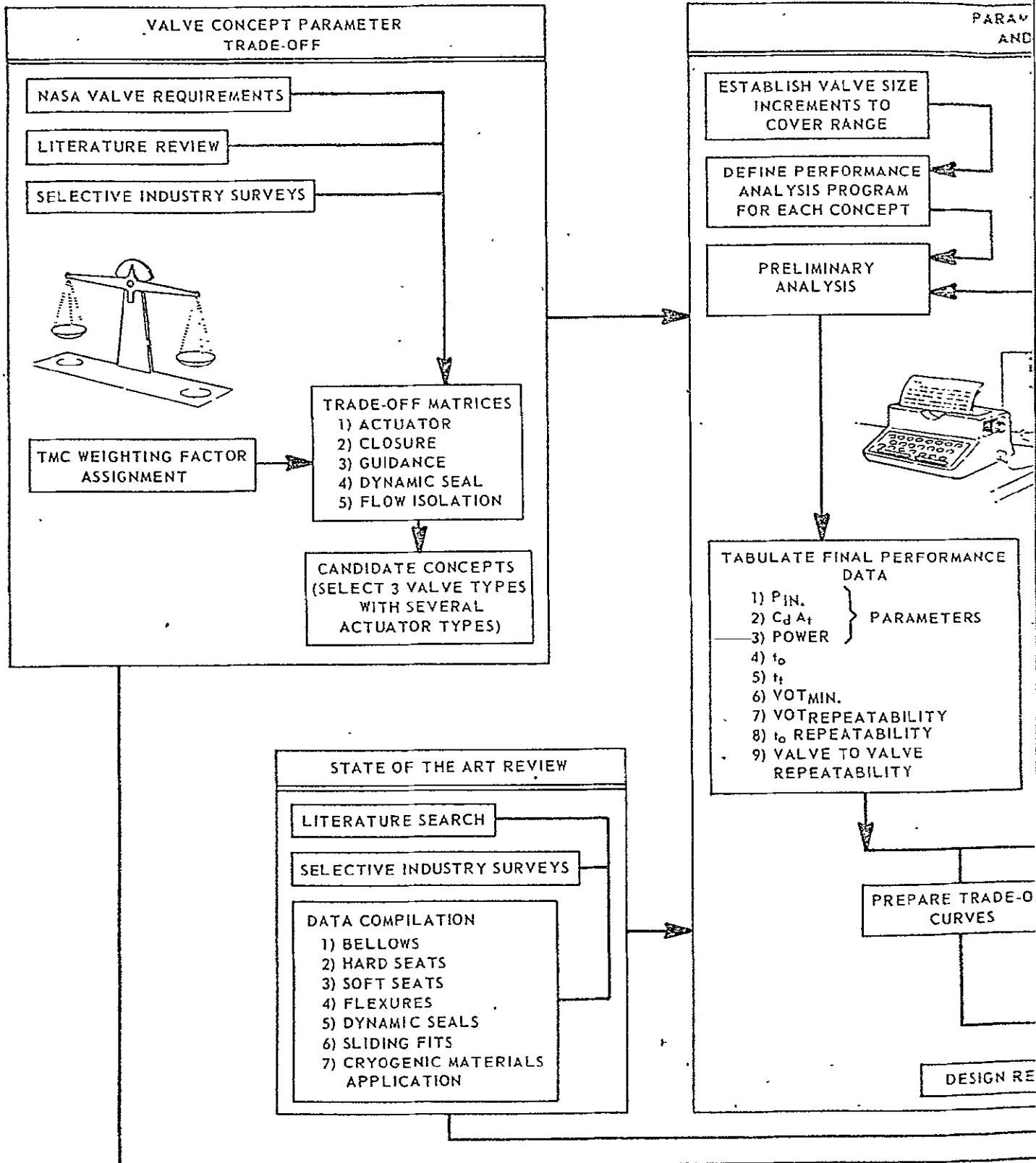
Under NASA Contract NAS9-10886, the work described in this report was performed to generate parametric data on injector valves for controlling gaseous propellants to rocket engines.

The objectives of the program were to develop a comprehensive technology base in the field of high performance rocket engine injector valves for gaseous propellant service in spacecraft reaction control systems. This technology base will provide the basis for sound system selection and configuration decisions on an overall vehicle basis; and will provide assurance that system functional requirements and characteristics represent achievable goals. The total effort establishes a sound technical framework for the expeditious development of engine valves.

The method of accomplishing these objectives is illustrated in the Program Plan Flow Diagram of Figure I-1. This effort has resulted in the accomplishment of the following:

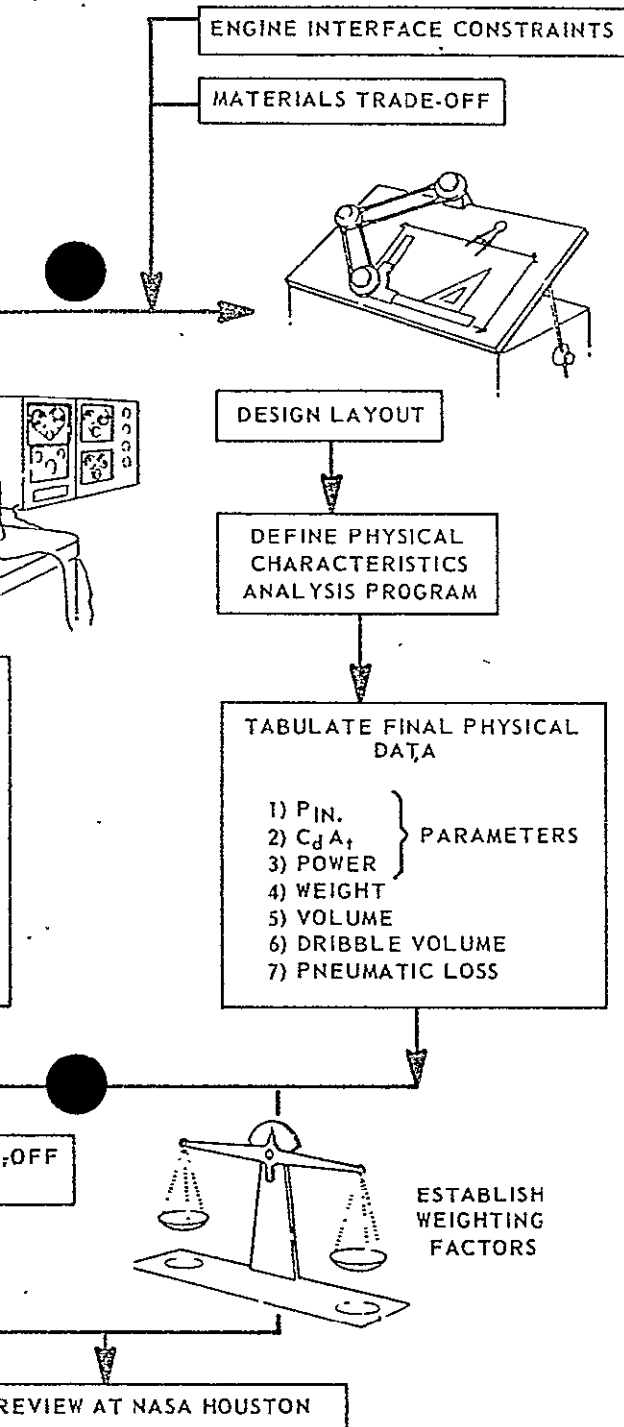
- . A comprehensive technique was developed for systematically performing trade-off studies of valve designs. This technique represents valid decision modeling of valve design features requiring evaluation on a subjective basis.
- . A state-of-the-art survey was completed to assess the current status of metal bellows technology with respect to high performance valve requirements.
- . A comprehensive analysis program was developed for predicting performance and physical characteristics of at least eight valve configurations as a function of rocket engine performance criteria.
- . The analytical program was substantiated by a concurrent design effort to document several sample applications.
- . Parametric design curves were prepared for predicting valve performance and physical characteristics.
- . Trade-off curves of various valve configurations, as a function of performance and physical characteristics, were prepared.
- . A systematic approach for presenting and evaluating valve configurations for a specific application was developed and demonstrated.
- . Detail design layouts were completed of valve configurations for two specific applications. The configurations of these designs were selected using the analysis program and evaluation systems developed during this effort.

VALVE STUDY PROGRAM



AM PLAN - FLOW DIAGRAM

PARAMETRIC ANALYSIS AND TRADE-OFF



POINT DESIGN

SELECT OPTIMUM CONFIGURATIONS

$F_t =$	1000 LB.	1000 LB.
$O F =$	2.0	2.0
$P_{IN} =$	22 PSIA	400 PSIA

ANALYSIS

LAYOUT

MATERIALS TRADE-OFF

RELIABILITY ANALYSIS

DESIGN REVIEW

FINAL REPORT

II. SUMMARY

The development of a comprehensive valve technology base to support gaseous propellant reaction control system design studies, establish a basis for objective trade-offs and selection and support valve design and development programs has been recognized as a critical item in the overall advanced spacecraft development program. As a result, the program described in this report was executed by The Marquardt Company during the period from 6 May 1970 to 15 December 1970, under contract to NASA-MSC, Houston, Texas. As a result of this effort, a parametric valve program has been developed which is a useful tool to the systems designer in establishing realistic valve requirements and estimating the impact of valve characteristics on the total system, and to the valve designer in selecting optimum elements and configurations of valves for a specific application.

To establish a common denominator, applicable to all valves, which would combine several criteria of a rocket engine injector valve, the equivalent orifice of the valve was adopted as a convenient method. This flow factor ($C_d A_t$) and the inlet pressure provide sufficient input to establish the valve's physical and performance characteristics and is readily traceable to application criteria such as thrust, O/F, valve pressure drop, and propellant.

Having established a base from which to analyze valve characteristics, the scope of the analytical effort had to be established within reasonable bounds of a workable number of configurations. A valve concept trade-off study was performed wherein various alternative elements which perform a function essential to the total valve's function, were evaluated on the basis of their capability to demonstrate desirable characteristics. This evaluation was performed in a systematic method comparable to decision modeling since subjective judgments were primarily involved. The result of this trade-off was the selection of a limited number of candidate valve configurations which appeared compatible with the design goals and the range of interest. The trade-off also indicated an optimum breakdown of a valve design for analysis such that the analysis would provide maximum utilization.

A detailed analysis program was developed which produced data on shut-off devices and actuators, and established interface criteria such that various combinations of shut-off device and actuator could be evaluated and performance and physical characteristics generated. This program offers capability to perform analytical studies of eight basic valve configurations and can include several other configurations with minor modifications. The analysis program was employed to compute parametric data for several sample applications and design curves covering the total range of interest were prepared. Employing the data and design curves, sample trade-off curves were prepared which define the operating envelopes of the eight configurations as a function of several typical design criteria. For two specific applications (1000 lb thrust, O/F of 2.0 inlet pressure of 22 psia, and 1000 lb thrust, O/F of 2.0 and inlet pressure of 400 psia) performance and physical characteristics data on the various configurations were calculated and compiled. In lieu of specific system criteria, an arbitrary rating system was developed and the compiled data evaluated to systematically select the optimum valve configuration for each specific application.

Following a design review with the NASA technical monitor, detail design layouts of the selected valve configurations for the two specific applications were prepared. For a 1000 lb thrust engine with a propellant supply pressure of 400 psia, a pneumatic operated linear motion valve configuration was selected as offering the optimum performance and physical characteristics for the application, wherein a propellant inlet pressure of 22 psia is available for a 1000 lb thrust rocket engine. In arriving at these recommended valve designs, the analytical techniques developed during this effort were employed, and the data were evaluated by the methods described herein.

The completion of this program resulted in the development and demonstration of a comprehensive valve evaluation technique and the generation of extensive parametric data applicable to gaseous propellant valves. The systematic approach to the evaluation of valve designs and characteristics, employed throughout this effort, results in valve selection with a maximum of objectivity.

III. PARAMETRIC VALVE STUDY

This section of the report presents the results of work performed with the goal of developing techniques whereby optimum selections of valves for gaseous propellant systems can be made with a maximum of objectivity. The range of interest of this particular effort was for hydrogen/oxygen fueled rocket engines with thrusts from 500 to 2500 lbf, O/F ratios from 2.0 to 8.0 and inlet pressures from 20 to 400 psia (Table III-I). The techniques and methods described herein, however, have universal application.

III-1 Flow Factor Determination

For the range of interest, allowable valve pressure drop was defined as a function of valve inlet pressure only. To facilitate the handling of the variables of fluid, pressure drop and flow rate, a convenient factor had to be found which would easily relate to the valve. For this purpose, the valve was equated to an orifice in the gaseous propellant line. By combining flow rate, inlet pressure, pressure drop and flowing gas properties in the orifice flow equation (eq. (1)) the Flow Factor (product of minimum orifice area and orifice coefficient) for the required valve can be determined.

$$C_d A_t = \frac{\dot{w} \sqrt{RT}}{27.82 P_1 \sqrt{\frac{k}{k-1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1} \right)^{\frac{k+1}{k}} \right]}} \quad (1)$$

where:

C_d = orifice coefficient

A_t = minimum orifice area - in.²

$\dot{w}$ = gas flow rate - lb/sec

R = gas constant for flowing media - $\frac{\text{in.} \cdot \text{lb}}{\text{lb} \cdot ^\circ\text{R}}$

T = gas temperature - $^\circ\text{R}$

k = specific heat ratio of gas

P_1 = inlet pressure - psia

P_2 = outlet pressure = $P_1 - \Delta P$ - psia

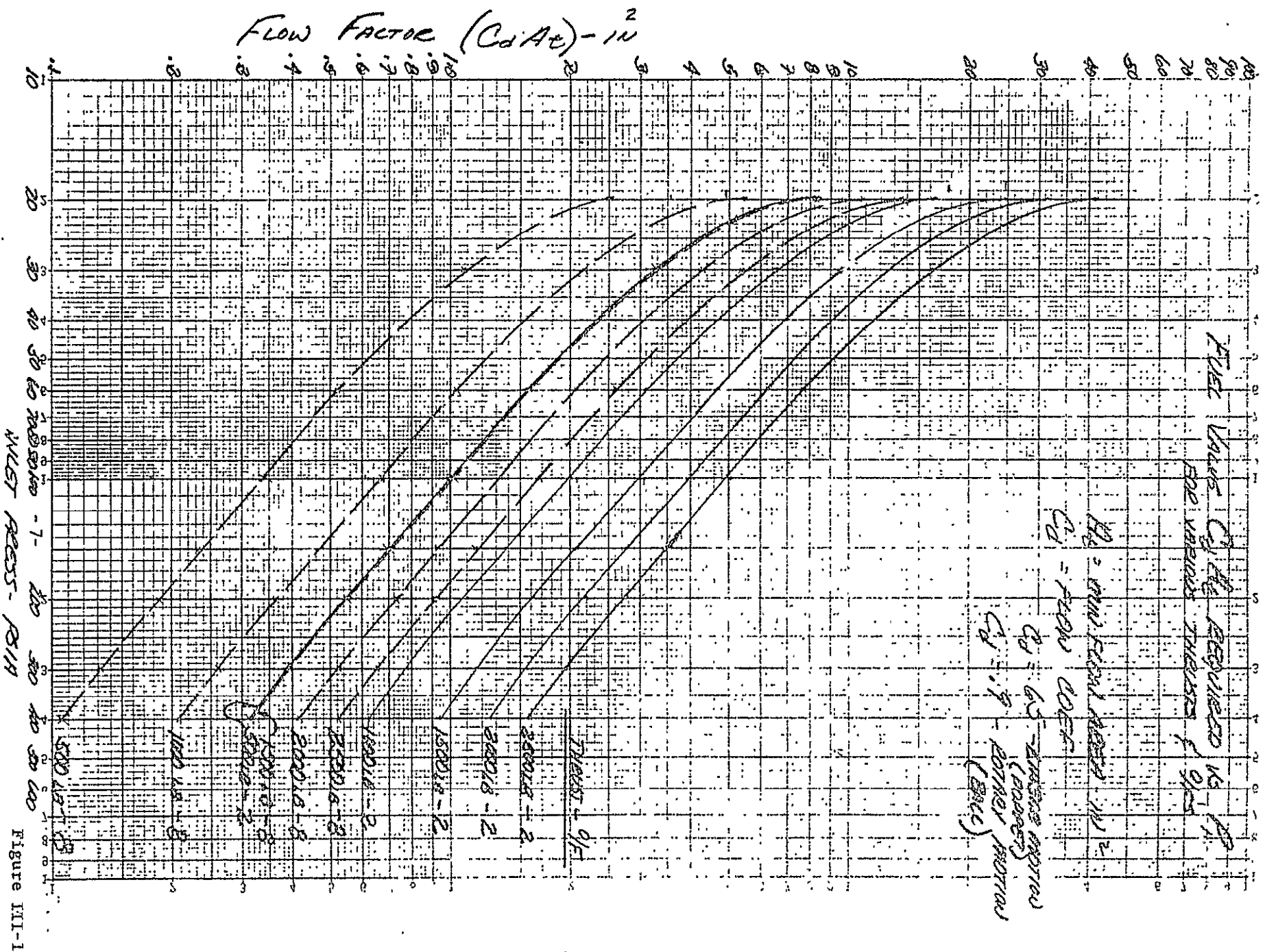
The Flow Factor is also indicative of the valve's physical size. A substantial quantity of data exists which relates various valve types to the equivalent orifice coefficient.

For the range of interest of this effort, Flow Factor versus Inlet Pressure is plotted in Figures II-1 and III-2 for oxygen and hydrogen respectively. Shown on these plots are the locus of specific engine thrust levels and O/F's within the range of interest. Similar type plots can be easily derived for any combination of inlet pressure range, pressure drop characteristic, flow rate and gaseous media.

TABLE III-I

RANGE OF INTEREST

FLUID	GASEOUS HYDROGEN	GASEOUS OXYGEN
SUPPLY PRESSURE	30 TO 400 PSIA	
SPECIFIC IMPULSE	400 SEC	
FLOW RATE	.45 TO 225 PPS AT 30 PSIA .40 TO 2.0 PPS AT 400 PSIA	.92 TO 4.63 PPS AT 30 PSIA .80 TO 4.0 PPS AT 400 PSIA
O/F	2 TO 8	
VOLTAGE	18 TO 32 V D-C	
TEMPERATURE	200 TO 660°R	
THRUST	500 TO 2500 LBF	
PRESSURE DROP	$\frac{3}{20} \times P_{\text{INLET}}$ 0 - 100 PSIA $8 + \left(\frac{7}{100} \right) P_{\text{INLET}}$ 100 - 400 psia	
CYCLE LIFE	200,000 CYCLES	
LEAKAGE	< 200 SCCH AFTER 100,000 CYCLES	



Flow Factor ($C_d A_e$) - in²

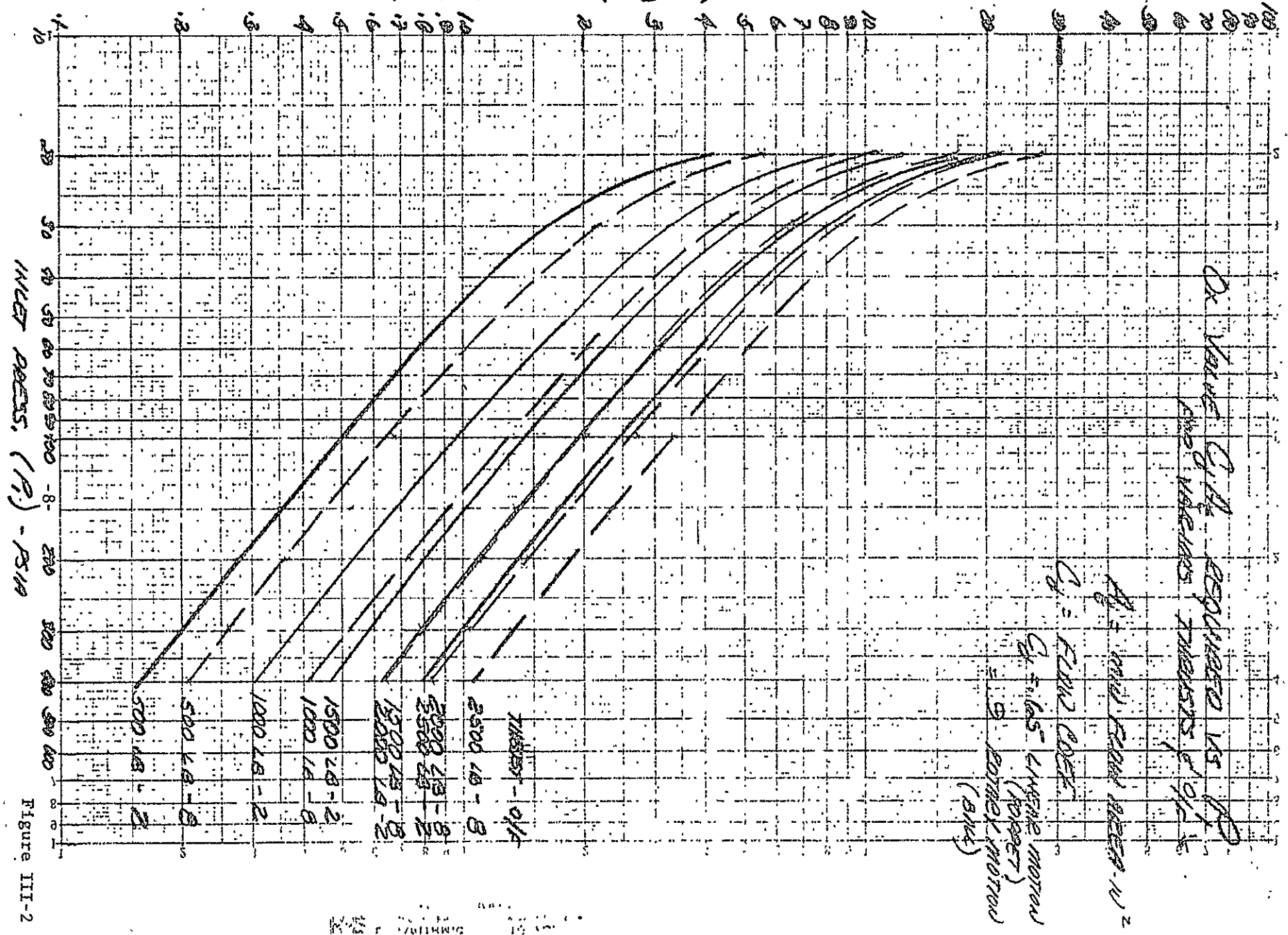


Figure III-2

III-2 Parametric Trade Off

Since many features of a valve design do not readily lend themselves to quantitative analysis, a comparison matrix technique is employed to provide an objective approach to a problem involving judgment. The approach chosen was to examine and segregate various valve designs into the various functional elements which are essential to the over all performance of the valve. For each function, the various alternate mechanisms, devices or configurations were tabulated. This tabulation was compiled with no consideration for limitations which might be imposed by pressure, or valve size, since the purpose of the trade off was general, and over a wide range of pressures and sizes.

Independent of the various alternative methods of performing a function, a tabulation was prepared of the various characteristics of the function. These characteristics were qualities which are essential to the performance of the function which do not lend themselves to objective evaluation. When considering a specific application wherein valve size and operating pressure, and the total system is defined, additional characteristics may be applicable and several of those listed in this report may lend themselves to objective analysis.

Having compiled the characteristics and alternates tabulations, a two dimensional matrix can be constructed for each function. The remaining task, before performing the trade off, was to assign appropriate weighting factors to each characteristic.

To establish weighting factors, the "emphasis curve" method was employed. This method consists of a series of comparisons; the importance of each characteristic for a particular function is compared individually with each of the other characteristics of that function. The weighting factor for each characteristic is determined by the number of times the characteristic is regarded as more important than each of the other characteristics of the function. Figure III-3 illustrates the matrix method whereby this is systematically accomplished. The characteristics are listed sequentially both horizontally and vertically similar to a comparison matrix used in decision modeling. Entering at the line of the first characteristic and proceeding horizontally, characteristic No. 1 is in turn compared to each characteristic of the vertical columns, on a one to one basis. Where the line characteristic is deemed more important than the column characteristic a value of 1 is placed at the intersection. Where the line characteristic is deemed less important than the column characteristic, a zero value is entered at the intersection. When the complete matrix has been evaluated and checked to be sure that the two

intersections for each characteristic pairing (e.g. No. 1 to No. 2 and No. 2 to No. 1) do not contain identical values, the sums of the values of each horizontal line are calculated. These sums represent the weighting factors for each characteristic. The sum of the weighting factors should equal;

$$\frac{N(N-1)}{2} \quad \text{where } N = \text{the number of characteristics within the function}$$

Since this method will always result in one characteristic receiving a weighting factor of zero, an arbitrary value of one is assigned this characteristic.

The evolved weighting factors were employed in the characteristics-alternatives matrix for each of the functions. For each characteristic the alternatives are examined. Each alternative is assigned a weight between zero and the maximum weighting factor, which is a relative judgment of that alternative's ability to possess or demonstrate the characteristic of that function. The more desirable the alternative, the higher the proportion of the maximum weighting factor. When the total matrix has been evaluated, the scores of each alternate are compared by summing the assigned values for each characteristic. The resultant scores represent figures of merit for each alternative, upon which a selection may be made.

Tables III-II through III-X present the trade off matrices for the valves of this study effort. Functions and characteristics were chosen on the basis of being totally independent of valve size, inlet pressure or flowing media to assure applicability to the total range of interest.

On the basis of these trade off matrices a compilation of the highest ranking alternatives was made (Table III-XI) to establish a preliminary estimate of the number of candidate valve configurations and define the scope of the analytical effort. Additionally, specific design details or elements could be identified and assessed relative to the state-of-the-art required to meet the design goals.

With reference to Table III-II, it must be noted that the fluidic shut off which attained the highest figure of merit was not among the selected details of Table III-XI. This exclusion was based upon the inability of a fluidic device to meet the required leakage goal of less than 200 SCCH, when considering the total range of interest (pressure and flow factor)..

III-3 State of the Art Review

With reference to Table III-XI, each candidate detail was examined to assess its current status with respect to reliably performing its function over the range of interest of pressure, temperature, life cycle and size requirements. Specific items, wherein additional investigation was

WEIGHTING FACTOR DETERMINATION - EMPHASIS CURVE METHOD, SHUT OFF DEVICE

<u>Characteristic</u>	<u>Characteristic</u>							<u>Total Figure of Merit</u>
	1	2	3	4	5	6	7	
1. Complexity	-	1	1	-	-	-	-	2
2. Stroke Required	-	-	-	-	-	-	-	0*
3. Force Required	-	1	-	-	-	-	-	1
4. Cycle Life	1	1	1	-	1	1	1	6
5. Dirt Sensitivity	1	1	1	-	-	-	-	3
6. Temperature Sensor	1	1	1	-	1	-	1	5
7. Dyn. Environment Sensitivity	1	1	1	-	1	-	-	4
TOTAL								21

$$N = 7$$

$$N \frac{(N - 1)}{2} = \frac{7 (6)}{2} = 21$$

*Assign an arbitrary value of 1

Figure III-3

TABLE III-II

VALVE - SHUT-OFF DEVICE

Maximum Points	(2)	(1)	(1)	(6)	(3)	(5)	(4)	
Characteristic.								
Alternative	Complexity	"Stroke" Req'd.	Force Req'd.	Cycle Life	Dirt Sensitivity	Temp. Sensitivity	Dynamic Environ. Sensitivity	Figure of Merit
Poppet or Plug	2	1	1	5	2	4	3	18
Gate or Blade	1	0	1	2	2	2	3	11
Spool	2	1	1	4	1	3	1	13
Butterfly	1	1	1	2	2	2	3	12
Ball	2	1	1	4	2	4	3	17
Diaphragm or Boot	1	0	1	0	2	1	3	8
Fluidic	2	1	1	6	3	5	4	22

TABLE III-III

VALVE - GUIDANCE (LINEAR)

Maximum Points Characteristic Alternative	(3)	(6)	(7)	(1)	(4)	(4)	(8)	(2)	(2)	Figure of Merit
	"Stiction"	Stroke	Cycle Life	Side Load Capability	Wear	Temp. Sensitivity	Repeatability	Dirt Sensitivity	Dynamic Environ. Sensitivity	
Sliding Fit	1	6	5	1	2	3	6	0	1	25
Flexure	3	3	6	1	4	4	8	2	1	32
Ball Bushing	2	6	4	1	3	3	7	1	1	28
Linkage	1	3	4	1	2	3	7	1	1	23

TABLE III-IV

VALVE - GUIDANCE (ROTARY)

Maximum Points	(2)	(6)	(7)	(2)	(4)	(5)	(1)	(2)	
Characteristic									
Alternative	"Stiction"	Stroke	Cycle Life	Side Load Capability	Wear	Temp. Sensitivity	Dirt Sensitivity	Dynamic Environ. Sensitivity	Figure of Merit
Ball Bearing	1	6	5	1	3	4	1	1	22
Bushing	0	6	4	0	2	3	0	1	16
Sliding Fit	0	6	5	1	2	4	0	1	19
Flexure	2	2	7	2	4	5	1	2	25

TABLE III-V

VALVE - DYNAMIC SEAL (LINEAR MOTION)

Maximum Points Characteristic Alternative	(6) Effective Dia. Control	(5) Cycle Life	(3) Temp. Sensitivity	(1) Wear	(1) "Stiction"	(2) Dynamic Environ. Sensitivity	(4) Press. Sensitivity	(7) Leakage	Figure of Merit
Bellows	4	4	2	1	1	1	1	7	21
Labyrinth	5	2	2	0	0	2	4	1	16
Piston Ring	3	2	2	0	0	2	4	3	16
"O-Ring"	4	3	2	0	0	2	3	4	18
Diaphragm	3	1	1	1	1	0	0	7	14

TABLE III-VI

VALVE - DYNAMIC SEAL (ROTARY)

Maximum Points Characteristic Alternative	(5) Leakage	(3) Cycle Life	(4) Temp. Sensitivity	(2) Wear	(1) "Stiction"	(1) Dynamic Environ. Sensitivity	Figure of Merit
Bellows	5	1	3	2	1	1	13
Labyrinth	1	3	3	1	0	1	9
Metal Seal Ring	2	3	2	1	0	1	9
"O-Ring"	4	2	3	1	0	1	11
Diaphragm	5	1	2	2	1	0	11
Torque Tube	5	1	2	2	1	0	11

TABLE III-VII

VALVE - STATIC SEAL

Maximum Points	(5)	(4)	(1)	(1)	(3)	(2)	
Characteristic							
Alternative	Leakage	Temp. Sensitivity	Ease of Fabrication	Contamination Sensitivity	Life	Dynamic Environ. Sensitivity	Figure of Merit
Weld	5	4	1	1	3	2	16
Formed Metal	3	3	0	0	3	2	11
O-Ring	4	2	1	0	2	2	11
Labyrinth	1	1	0	0	1	2	5

TABLE III-VIII

VALVE - SHUT-OFF SEAL

Maximum Points Characteristic	(4)	(1)	(3)	(2)	(1)	
Alternative	Seal Dia. Consistency	Life	Temp. Sensitivity	Contamination Sensitivity	Ease of Mfg.	Figure of Merit
<u>Hard</u>						
Flat	4	1	3	1	1	10
Cone	2	1	2	1	0	6
Ball	2	1	2	1	0	6
Labyrinth	1	1	1	2	0	5
Compression	1	0	1	2	1	5
<u>Soft</u>						
Press. Energized	3	1	3	2	1	10
Load Compression Limited	4	1	3	2	0	10
Dimensional Compression Limited	2	0	2	1	1	6

TABLE III-IX

VALVE - SHUT-OFF SEAL ALIGNMENT

Maximum Points	(5)	(4)	(1)	(4)	(1)	(7)	(5)	(2)	
Characteristic	Temp. Sensitivity	Spring Rate	Ease of Mfg.	Wear	"Stiction"	Cycle Life	Dynamic Environ. Sensitivity	Dirt Sensitivity	
Alternative									Figure of Merit
Floating	4	4	1	3	1	6	3	0	22
Flexure	5	2	0	4	1	6	5	2	25
Ball Joint	2	3	0	1	0	5	3	1	15

TABLE III-X

ACTUATOR TYPE

Maximum Points	(4)	(2)	(4)	(4)	(1)	(1)	
Characteristic	Force Output lb of actuator	Temp. Sensitivity	"Stroke"	Speed	Power Consumption	Devel. Status	
Alternative							Figure of Merit
Solenoid	3	1	3	3	0	1	11
Torque Motor	3	1	3	3	0	1	11
Elect. Motor	2	1	4	2	0	1	10
Pneu. Act. Linear	4	2	4	3	1	1	15
Rotary	3	2	4	1	1	0	11
Bi Morph.	3	0	1	4	0	0	8
Piezoelectric	3	0	0	4	0	0	7
Magnetostrictive	3	0	0	4	0	0	7
Electro Thermal	4	0	2	0	0	1	7
Fluidic	2	1	2	3	1	0	9
Stored Energy	3	1	3	2	1	0	10

TABLE III-XI

SELECTED VALVE DETAIL CONFIGURATION CANDIDATES

Function	Selected Alternates			
Valve Shut-Off	POPPET			BALL
Guidance	Flexure	Sliding	Ball Bushing	Flexure Ball Bearing
Dynamic Seal	Bellows		"O-Ring"	Bellows "O-Ring"
Static Seal	Weld			Weld
Shut-Off Seal	(Hard) Flat	(Soft) Load Limited	(Soft) Load Limited	
Shut-Off Seal Alignment	Flexure	Floating	Flexure	Floating
	24 Possible Configurations			8 Possible Configurations

warranted were bellows and "O-ring" type seals. These elements were selected on the basis of a lack of readily available information, covering the entire range of interest, and a need to assess the most current status of development and establish the required development necessary to fulfill the requirements of the range of interest.

The state-of-the-art review was accomplished in two ways; a literature search was conducted for documented experience applicable to the range of interest; a direct request was made of the suppliers to provide full range of capability information, and historical data on their component which would be applicable to the range of interest.

The Marquardt Engineering Library indexes were screened for references pertaining to bellows, seals, leakage and control components. A listing of the technology reports reviewed and which rendered significant impact on this study effort is presented in the Bibliography of this report (Section VI). Numerous trade magazine articles were consulted but in general, lacked sufficient detail to provide a meaningful contribution to the review. Technology reports provided tangible data for assessing development status.

To compliment the information gleaned from the literature search on the subject of bellows, an industry survey letter was prepared and submitted to twelve suppliers of metal bellows to the aerospace industry. This survey requested capability information relative to applications within the range of interest, including design characteristics and variables, and specific historical data to substantiate design capability. The responses to this survey were notably lacking in significant data, being primarily general catalogues and die listings covering the size ranges available. One supplier, Gardner Bellows Company of Van Nuys, California, did provide a listing of aerospace application bellows they had fabricated. This tabulation did include design criteria and traceability to the specific program application. This would have provided sufficient data to evaluate current capability had other suppliers responded to the survey in a like manner. The paucity of historical data can, perhaps, be explained by the usual remote subtler relationship of the bellows supplier to the agency that ultimately tests the component in which the bellows is used. The testing usually does not provide specific bellows performance data other than an indication that over all function of the tested hardware was acceptable or unacceptable. The bellows supplier, therefore, can only assume his product fully conformed to the requirements if in the conduct of the tests, he can ascertain that the required design loads were imposed. Often times, particularly in aerospace applications, detailed test data is not made available to the sub-tier suppliers and confirmation of design can only be assumed.

The responses of the bellows industry survey, though disconcerting, did augment information obtained in the literature search sufficiently to prepare several generalized design curves applicable to a portion of the range of interest. These design curves are presented in Figures III-4 through III-9. Their range of application, based on review of available data, is limited to flow factors of less than 7.5 in^2 and inlet pressures of less than 100 psia. This limit is established as a boundary within which metal bellows may be employed with a high degree of confidence of meeting all requirements of the range of interest without taxing the current state-of-the-art. The use of metal bellows in applications requiring larger flow factors than 7.5 in^2 and/or pressures in excess of 100 psia must be approached with caution and extensive analysis or demonstration testing conducted before committing them to a final design.

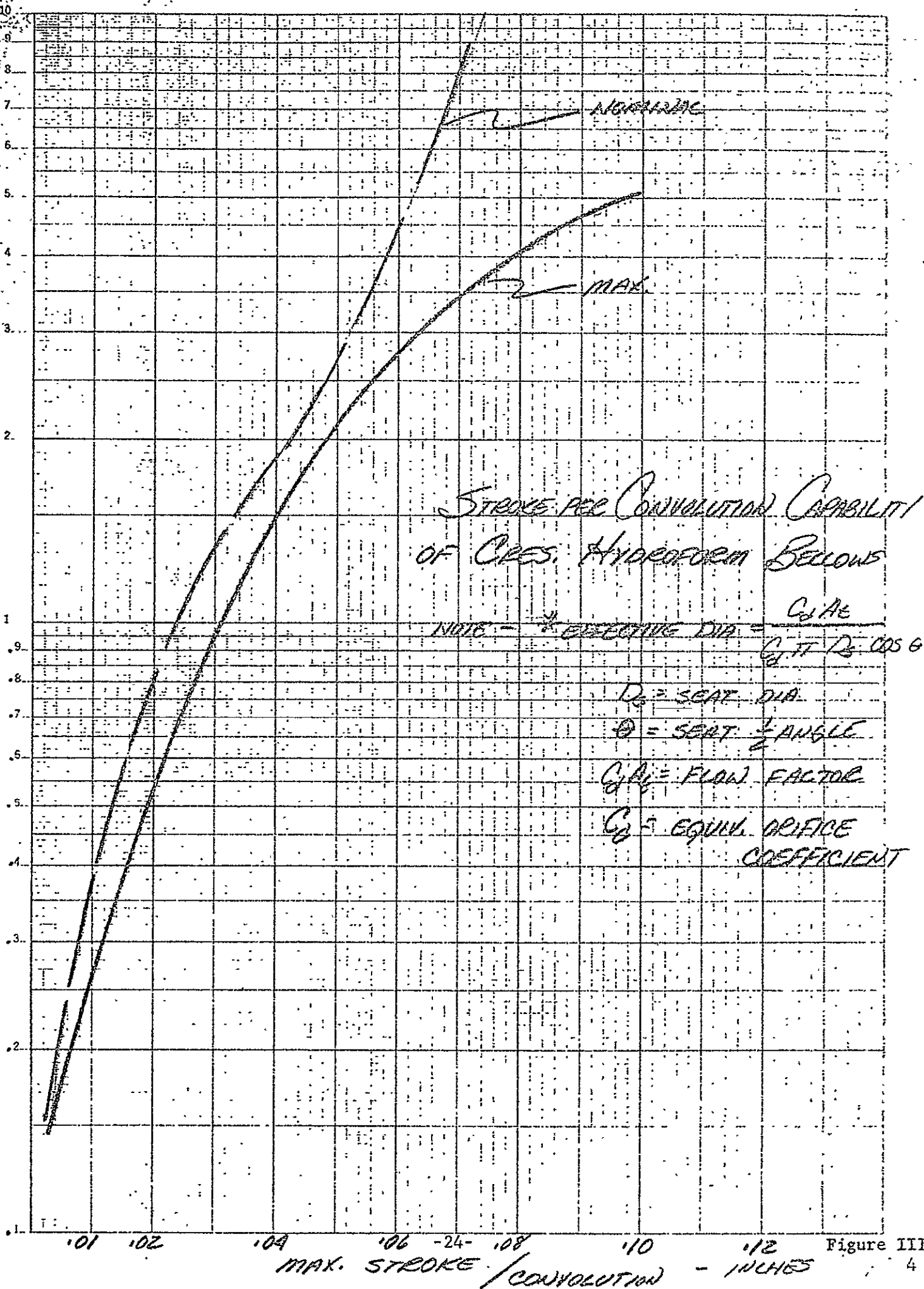
Additional factors which must be analyzed in depth before committing to a metal bellows design are:

- a. The accelerations produced by the rapidly moving elements generate high stress levels, particularly in the convolutions at each end of the bellows, where these accelerations reverse. These high stresses impose severe restrictions on cycle life capability or require design compromises which influence other bellows characteristics (spring rate effective area stability and pressure resistance).
- b. Thermal expansion and contraction of the bellows material or material property changes with temperature reflect on bellows performance characteristics.

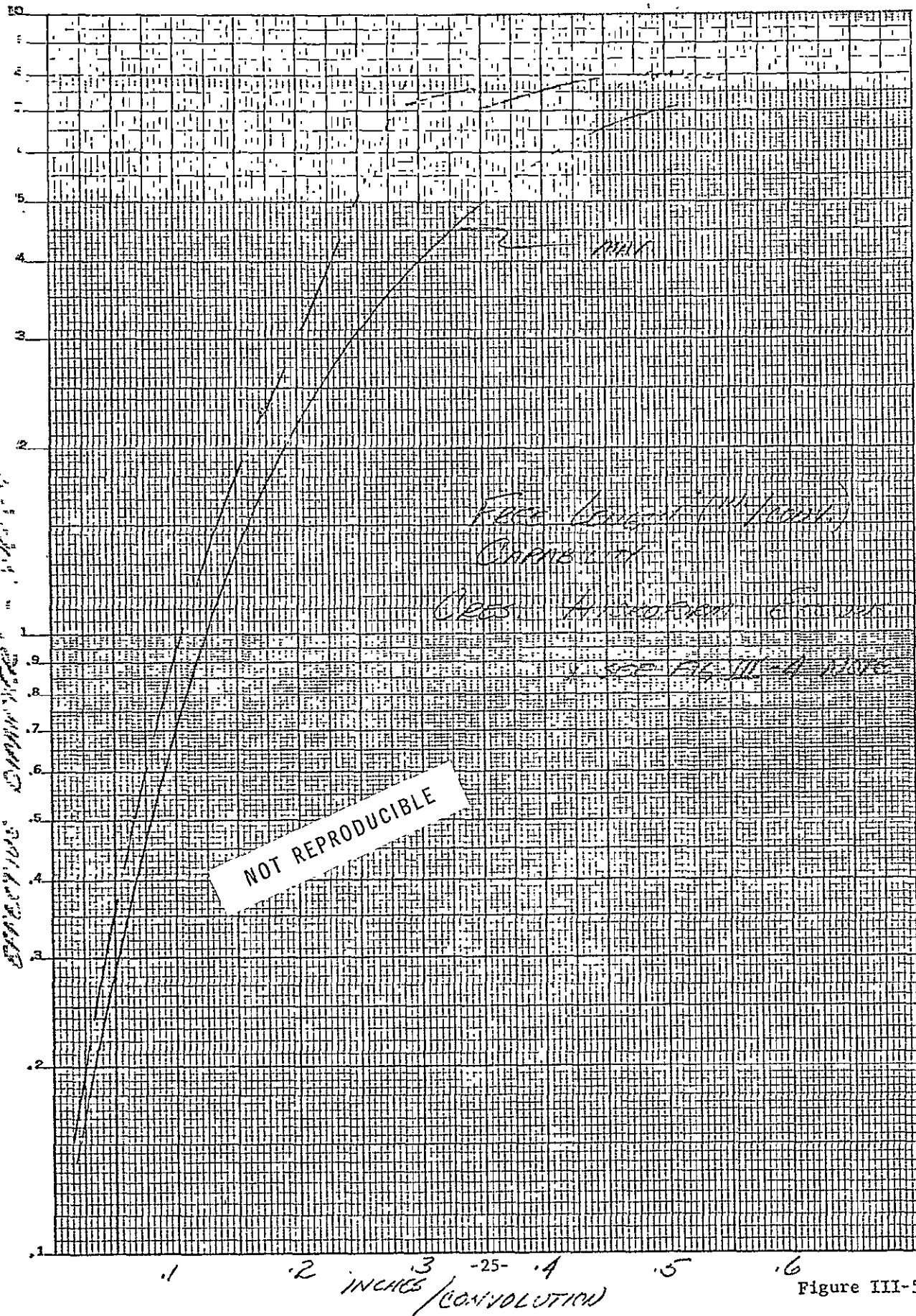
In spite of the extensive analytical methods developed for the design of metal bellows by Battelle Memorial Institute (Reference 11 of Bibliography), the design techniques for metal bellows are as numerous as the suppliers. Each supplier has developed analytical techniques which are based upon their own experience and methodology, relying heavily on empirical data. This lack of a common design approach makes valid design evaluation, by analytical techniques, impractical in most cases, and necessitates extensive demonstration testing.

The literature search for data supporting the assessment of "O-ring" type seal technology yielded substantial theoretical data on interface requirements relative to types of leakage flow. In general, these seals exhibit a common design approach; a formed metal spring is jacketed with a soft material, the spring exerting a preload to force the jacket material into intimate contact with interface surfaces. The major burden of success of any and all seals is placed upon the proper treatment

EFFECTIVE DIAMETER - INCHES

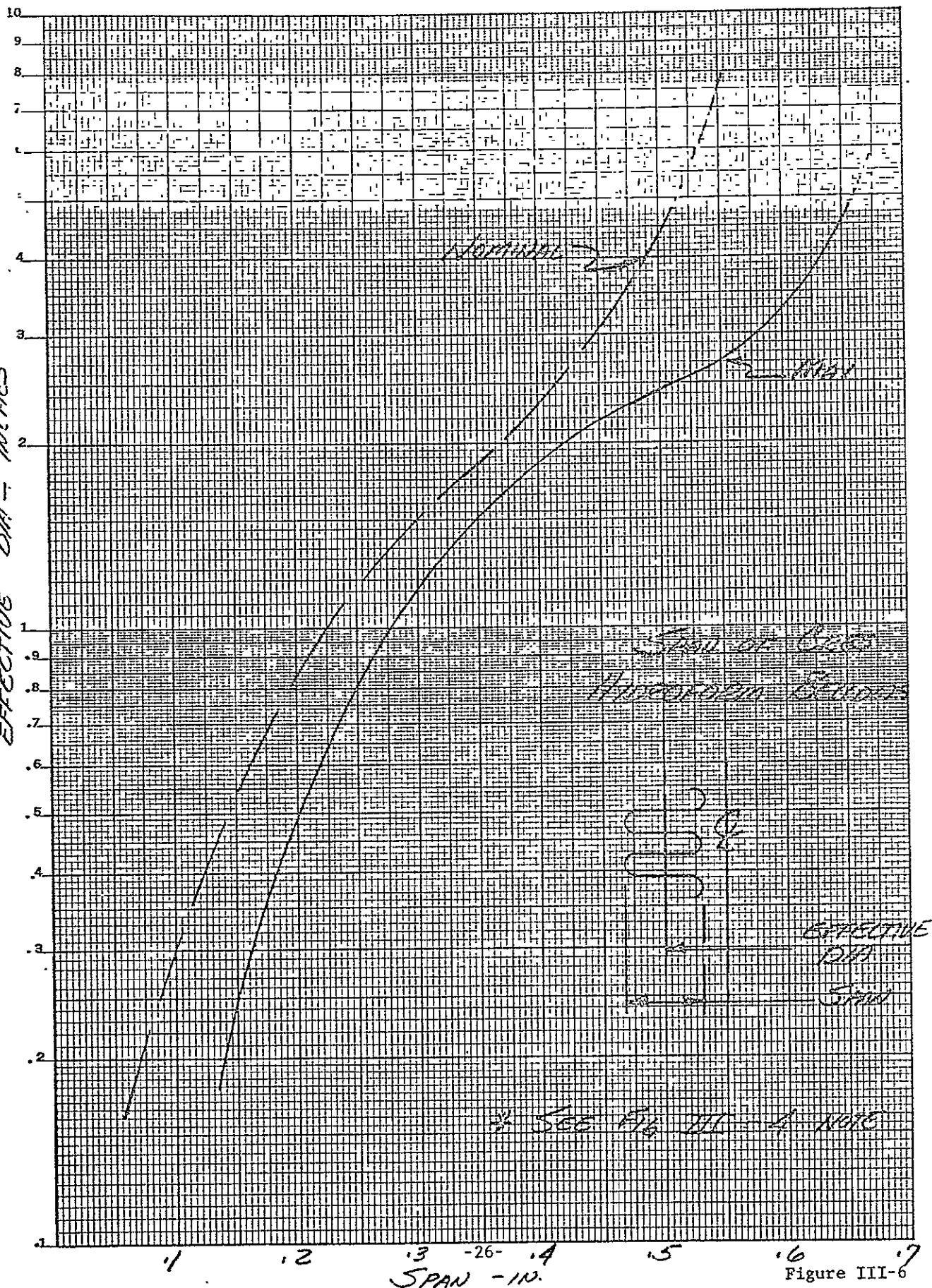


KEUPFEL & ESSER CO



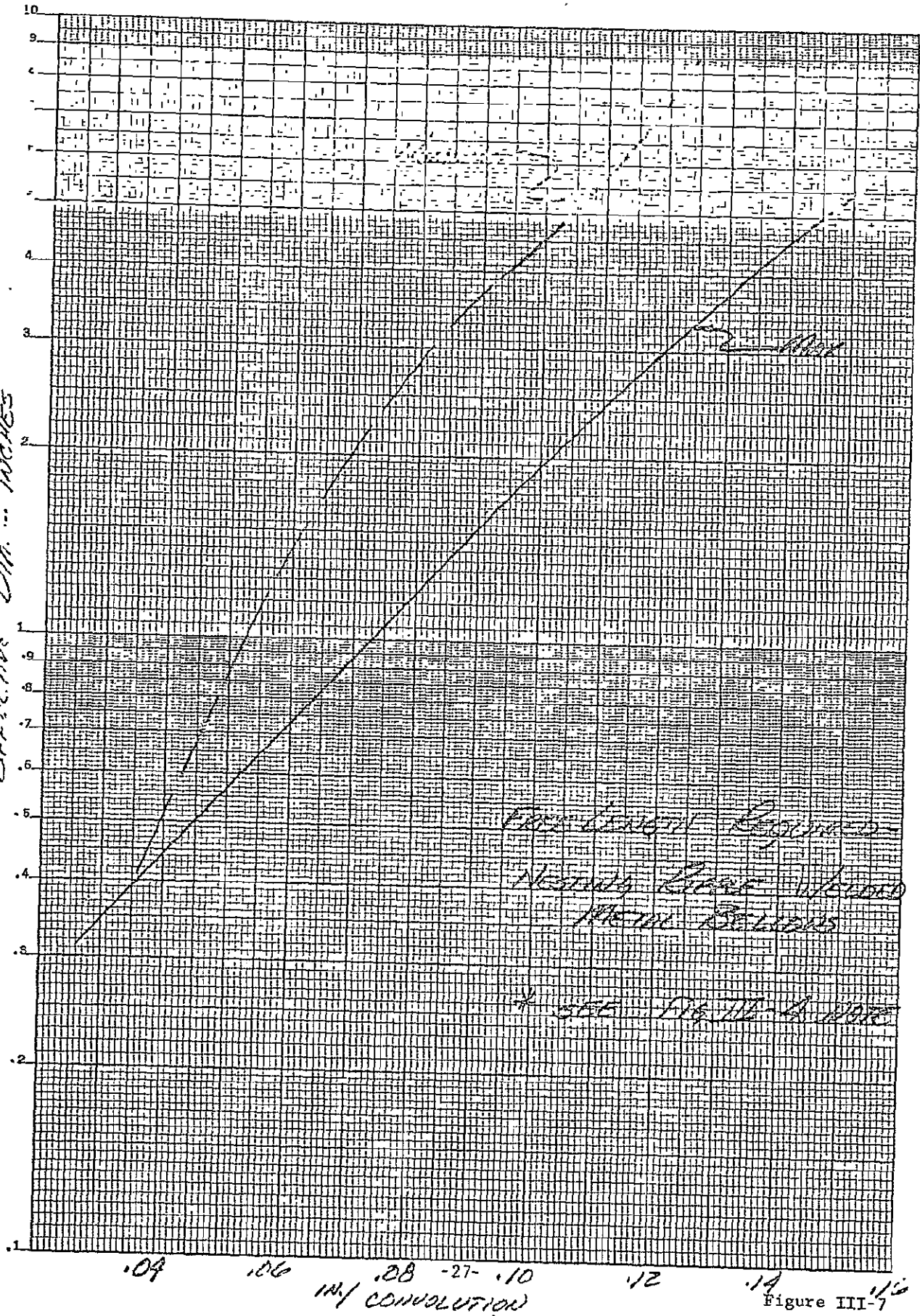
K&W SEMILOGARITHMIC 359-63
KEUFEL & ROSS CO. MADE IN U.S.A.
2 CYCLES X 140 DIVISIONS

EFFECTIVE DRIFT IN INCHES



K&E SEMILOGARITHMIC 359-63
KUTTER & ESSER CO. MADE IN U.S.A.
2 CYCLES X 140 DIVISIONS

EFFECTIVE DIA. IN INCHES



K&E SEMI-LOGARITHMIC 359-61
 KEUFFEL & ESSER CO. MADE IN U.S.A.
 2 CYCLES X 70 DIVISIONS

EFFECTIVE DIA. INCHES

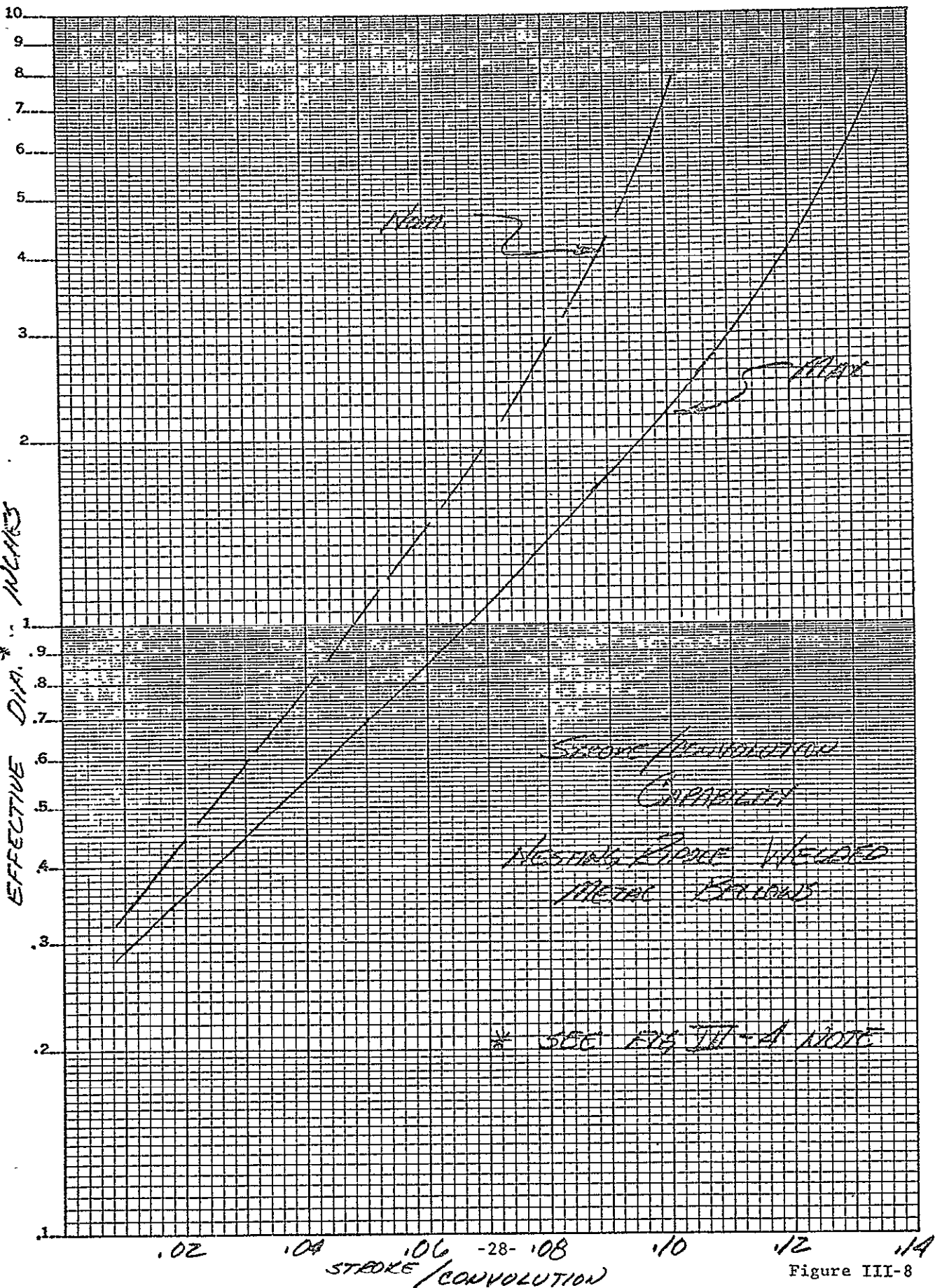
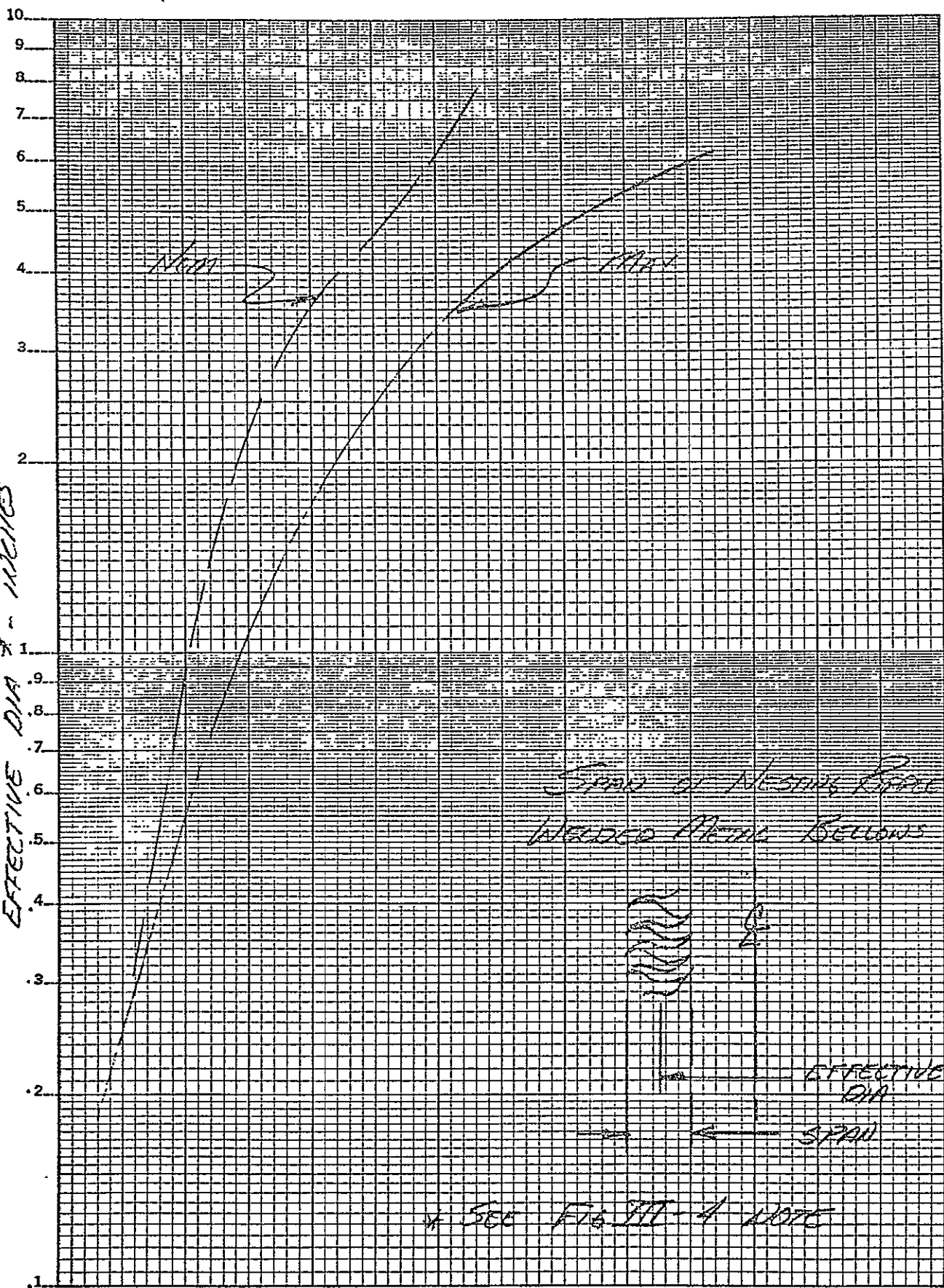


Figure III-8

K-E SEMI-LOGARITHMIC 359-61
KEUFFEL & ESSER CO. MADE IN U.S.A.
2 CYCLES X 70 DIVISIONS

EFFECTIVE DIA. IN INCHES



SPAN - INCHES
Figure III-9

of interface surfaces. A high degree of refinement is essential to prolonged seal life since rms criteria for surface finish requirements are totally inadequate. Particular attention must be paid to surface characteristics such as waviness, lay, crystallinity and grain structure, and cleanliness. In addition, the method of seal installation greatly influences reliability. The major emphasis should be placed on how the design requirements are achieved, once a successful technique has been developed.

III-4 Analysis of Performance and Characteristics

The development of a comprehensive analytical program to confidently predict performance of valve designs and estimate their physical characteristics represents a major portion of this study effort. The approach taken was to create a method which would be of value to both the system designer and the valve designer. To this end, a generalized approach to the analysis program was formulated. Each valve design can be divided into two elements; the shut off device, which is the fluid control element, and the actuator, which provides necessary force/stroke inputs to the moving component(s) of the shut off device. The interface between the shut off device and the actuator can be characterized by force, stroke and time. Shut off device performance is a function of the fluid flow properties desired and actuator performance is established by available power source criteria. A total valve performance prediction can be made when the shut off device force-stroke-time input requirements equal the force-stroke-time output of the actuator. Both shut off devices and actuators can be characterized by the nature of their motion - linear or rotary. This characterization is illustrated in Table III-XII. Having thus reduced the valves to basic elements, the analytical procedure was developed to establish compatible shut off device/actuator interfaces for the various combinations of the elements. A general flow diagram for the analysis program is shown in Figure III-10. The scope of the analytical program, described herein, has been limited to the two types of shut off devices and four types of actuators shown in Figure III-10. This results in capability to analyze a total of eight different valve configurations over the total range of interest.

In depth force balance analyses were developed for each element. To facilitate the preparation of parametric data over the total range of interest, specific assumptions had to be made about each element to yield data. These assumptions are classified as Configuration Criteria (Reference Figure III-10) and reflect specific design practices normally employed in valve design. Specific values used were selected on the basis of past experience, at TMC, in the design of valves for similar type applications. For a specific application, these criteria may be iterated to determine impact on valve performance.

The development of the specific analyses for each element is described in detail in the following subsections.

TABLE III-XII

VALVE ELEMENT - MOTION CHARACTERIZATION

MOTION TYPE	LINEAR	ROTARY
Shut Off Device	Poppet Plug Gate Sliding Spool Blade Diaphragm or Boot	Ball Butterfly Swing Flapper
Actuator	Cylinder Solenoid Piezoelectric Magnetostrictive Thermal Expansion Linear Motor	Torque Motor Electrical Motor Hydraulic or Pneumatic Motor Inertia Wheel

FLOW DIAGRAM FOR THE VALVE ANALYSIS PROGRAM

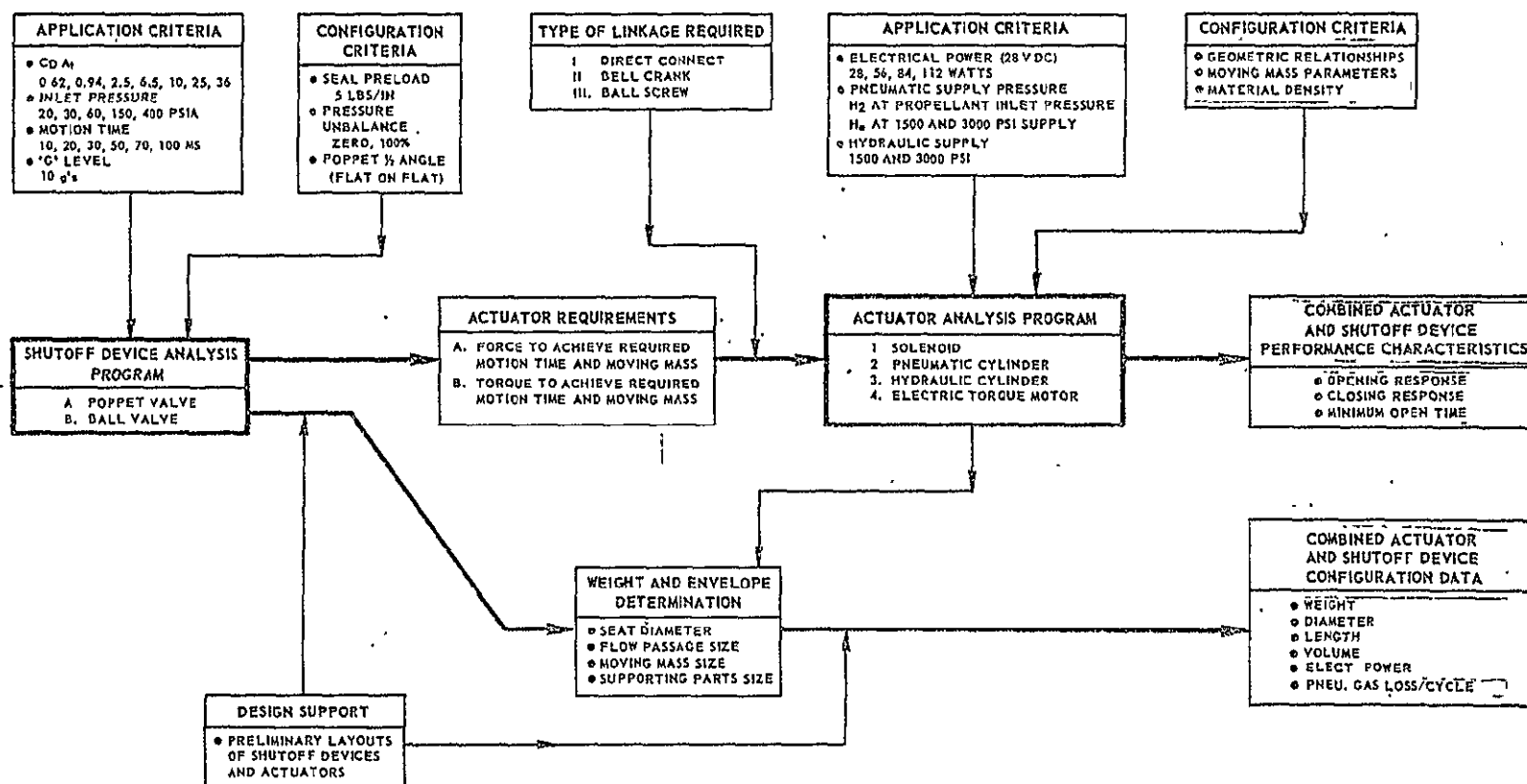


Figure III-10

III-4 A. Shut Off Device Performance

III-4A - 1. Linear Motion - With reference to the schematic drawing of Figure III-11, the linear motion shut off device is analyzed as a balanced poppet valve. The required poppet preload is assumed to be externally applied either by the actuator or by a spring.

For the linear motion shut off exhibiting a flow path similar to that of Figure III-11, an equivalent orifice coefficient (C_d) of .65 is a valid assumption. Applying this coefficient (C_d) to the flow factor ($C_d A_t$), it is possible to calculate A_t , the minimum cross sectional flow area. If this is assumed to occur at the valve seat, then the nominal valve seat can be expressed by equation (2);

$$D_s = \sqrt{\frac{C_d A_t}{.65 \times \frac{\pi}{4}}} \quad (2)$$

When the poppet is withdrawn to open the valve, its stroke(s) must be sufficient to provide an annular flow area across the seat at least equal to A_t . Therefore,

$$\pi D_s \times S \cos \theta = \frac{\pi}{4} D_s^2 \text{ and } S = \frac{D_s}{4 \cos \theta} \quad (3)$$

With the valve in the closed position, some tolerance exists as to what exact seat seal diameter is effective, and also how well pressure balanced the valve remains. From past practical experience a seat land width of .010 is assumed and pressure balancing to a tolerance of $\pm 3\%$ of the seat area is selected. A nominal seat preload of 5 lb/circumferential inch is a practical factor for both soft and hard seat configurations, resulting in a seat bearing stress of 500 psi on a .010 wide seat land.

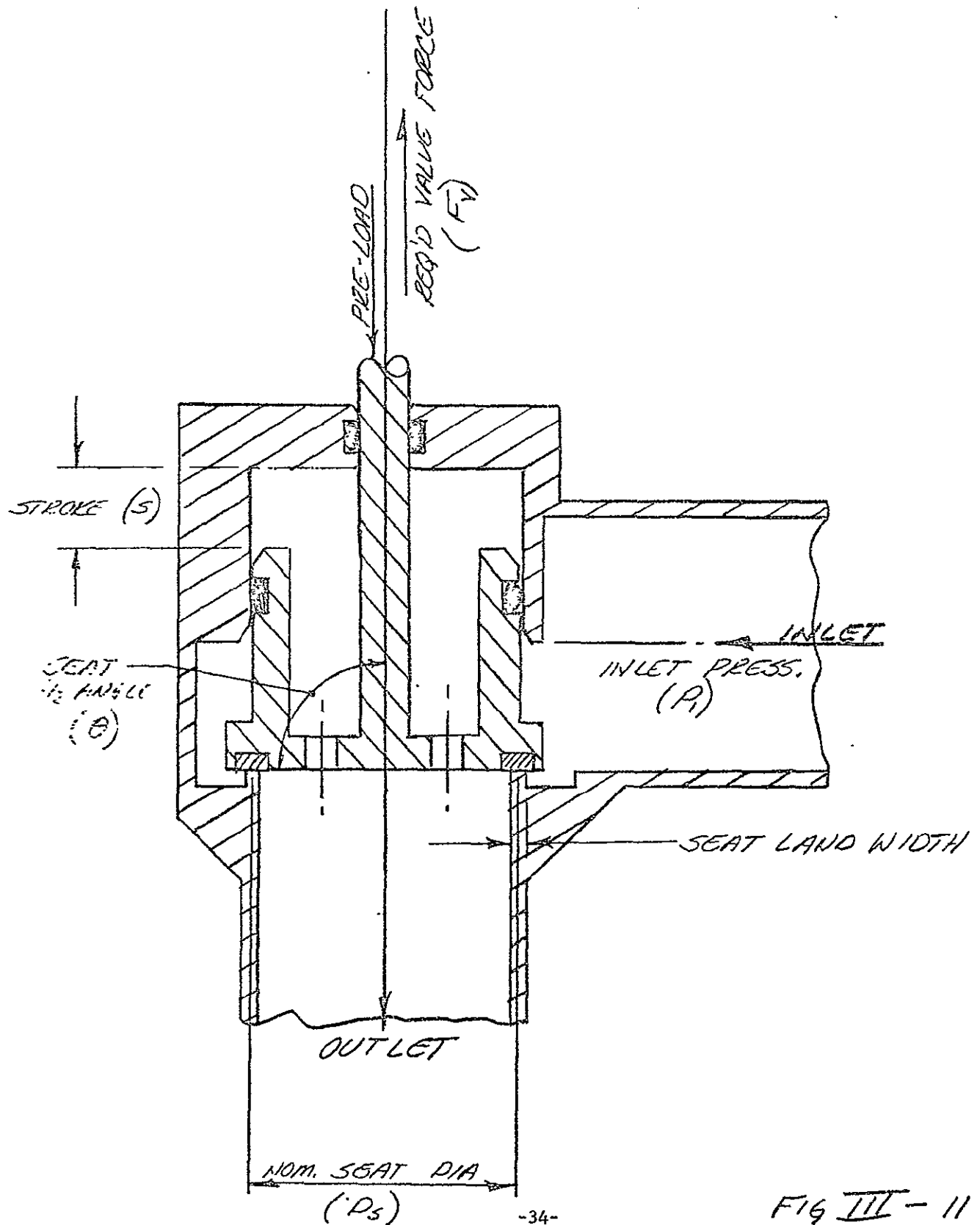
Assuming the valve is to be operated in a maximum acceleration field of 10 g's, the required valve force to effect opening can be defined by the sum of the worst case conditions defined above:

$$F_v = 5 \text{ lb/in. } (\pi D_s) + P_1 \frac{\pi}{4} [(D_s + .010)^2 - .97 D_s^2] + M_v [20 + a] \quad (4)$$

Where F_v = required valve force - lb
 D_s = seat diameter - inches (See Equation 2)
 P_1 = inlet pressure - psia
 M_v = mass of moving component - lbf
 a = acceleration of moving component - g's

Two elements of Equa. (4) must be defined in order to solve for F_v . If it is assumed that poppet stroking is at constant acceleration, F_v remains constant and the acceleration can be expressed as:

LINEAR MOTION SHUT-OFF DEVICE - SCHEMATIC DRAWING



$$a = \frac{2s}{386.4 t^2} \quad g's \quad (5)$$

where s = valve stroke - inches

t = time to travel full stroke(s) - seconds

To complete the solution (M_v) the moving component mass must be defined. To this end, several linear motion shut off device valve designs were reviewed and additional layouts made to cover the range of flow factors ($C_d A_t$). Weight estimates were made for the moving elements based upon stainless steel construction, and plotted as a function of flow factor. A first order fit equation reasonably approximated the characteristic and was adopted as a valid estimate (Reference Figure III-12)

$$M_v = \frac{A_t^{1.66}}{185} \quad (6)$$

Substituting equation (5) and (6) into Equation (4), the required valve force for a linear motion shut off device design can be expressed as a function of flow factor ($C_d A_t$), inlet pressure (P_1) and allowable motion time (t) as:

$$F_v = 5\pi \sqrt{\frac{C_d A_t}{.51}} + \frac{\pi}{4} P_1 \left[\left(\sqrt{\frac{C_d A_t}{.51}} + 0.010 \right)^2 - .97 \frac{C_d A_t}{.51} \right] + \frac{\left(\frac{C_d A_t}{.65} \right)^{1.66}}{185} \left(20 + \frac{\sqrt{\frac{C_d A_t}{.51}}}{2 (\cos \theta) 386.4 t^2} \right) \quad (7)$$

For the purposes of this parametric study a flat seat design ($\theta = 90^\circ$) was assumed and required valve force calculated for 7 flow factors ($C_d A_t$), 5 pressures (P_1) and 6 allowable motion times (t). The calculations, which are tabulated in Table III-XIII were carried out on the APL computer terminal at TMC. A graphic presentation of this data is shown in Figures III-13 through III-18. Figure III-19 graphically presents the relationship between motion time (t), flow factor ($C_d A_t$) and moving component acceleration $\left(\frac{2s}{386.4 t^2} \right)$.

To facilitate the analysis, it is assumed that the calculated F_v , applied constantly, will effect full opening. Increase in preload due to compression of springs is equal to the reduction in pressure unbalance force as down stream pressure builds up, both being proportional to stroke.

III-4A - 2. Rotary Motion - Of the rotary motion valve types, the ball valve represents a valid configuration for the total range of pressures and flow factors of the range of interest. For the purpose of the parametric analysis a ball shut off consisting of a hollow spherical ball requiring 90° rotation was analyzed. This configuration is shown in Figure III-20.

LINEAR MOTION SHUT-OFF DEVICE MOVING MASS VS FLOW FACTOR ($C_d A_e$)

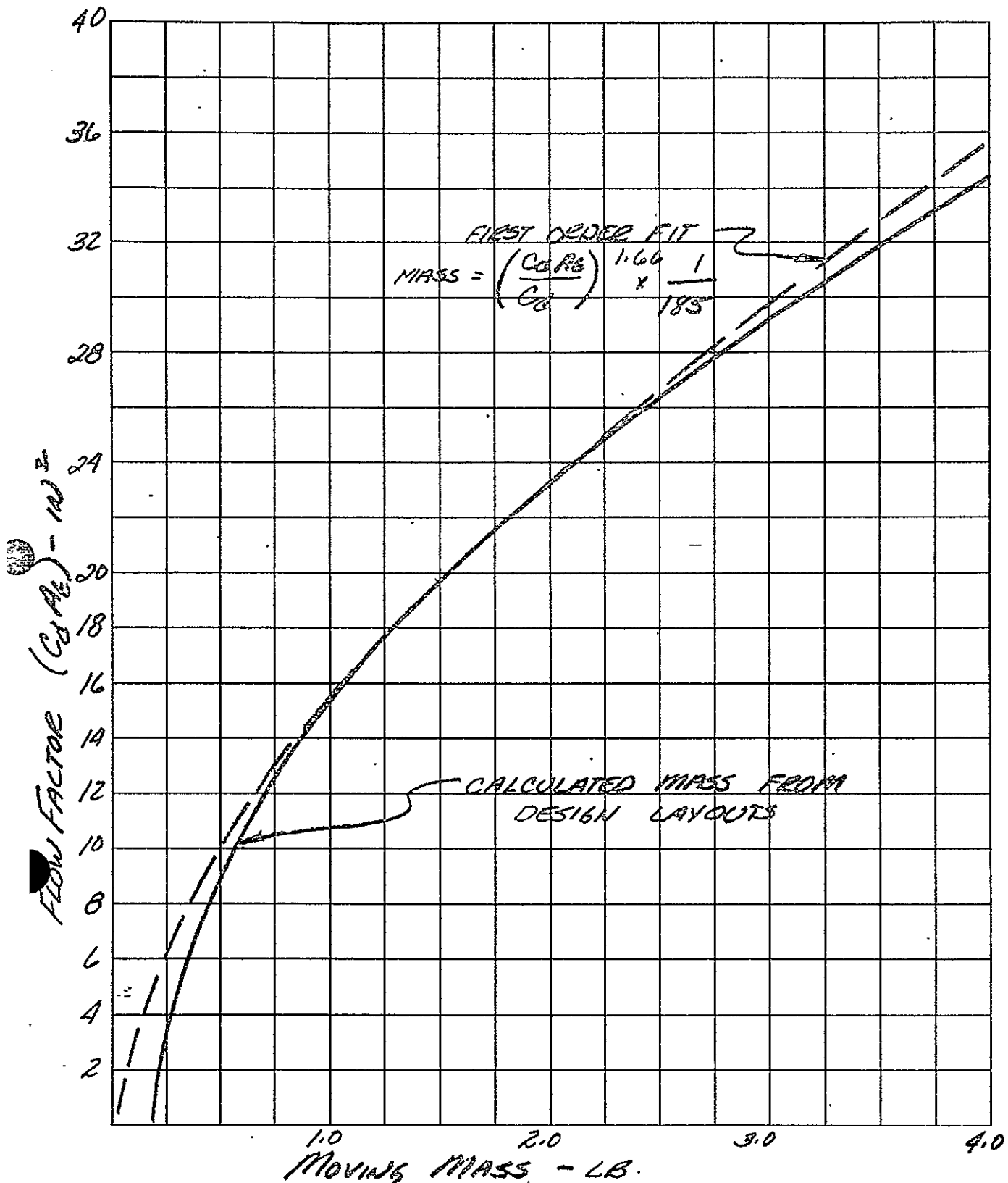


TABLE III-XIII

LINER MOTION SHUT-OFF DEVICE DATA SUMMARY

Flow Factor (W ²) C ₁ R ₂	Inlet Press (PSIA) P ₁	Seat Dia (in) D ₂	Stroke (in) S	Valve Mass (lb) M _V	Ratio T ₁ (sec) t	Ratio Force (lb) F _V		
.62	20	1.104	.28	.005	.01	18.42		
					.02	18.36		
					.03	18.35		
					.05	18.34		
					.07	18.34		
					.10	18.34		
					.01	18.88		
					.02	18.82		
					.03	18.81		
					.05	18.80		
	30				.07	18.80		
					.10	18.80		
					.01	20.26		
					.02	20.20		
					.03	20.19		
					.05	20.18		
					.07	20.18		
					.10	20.18		
					60	.01	24.40	
						.02	24.34	
	.03					24.33		
	.05					24.33		
	.07					24.33		
	.10					24.33		
	150					.01	35.91	
						.02	35.86	
						.03	35.84	
						.05	35.84	
					.07	35.84		
					.10	35.84		
.094		20	1.36	.34	.011	.01	23.01	
						.02	22.87	
						.03	22.84	
						.05	22.83	
	.07					22.83		
	.10					22.82		
	30	.01				23.66		
		.02				23.52		
		.03				23.49		
		.05				23.48		
		.07				23.47		
		.10				23.47		

TABLE III-XIII (Continued)

LUREN MORNS SUR-OT DEPTO DATA JENNINGS

60

Time Falling (min)	Heat Flow (mW)	Sample Dia (mm)	Surface (cm ²)	Volume (cm ³)	Heat Flow (mW)	Heat Flow (mW)
C.R.	P.	D.	S.	V.	T.	F.
.094	60	1.36	.34	.011	.01	25.4
					.02	25.46
					.03	25.49
					.05	25.42
					.07	25.42
					.10	25.42
	150				.01	31.44
					.02	31.30
					.03	31.27
					.05	31.26
					.07	31.25
					.10	31.25
	400				.01	47.65
					.07	47.50
					.03	47.48
					.05	47.47
					.07	47.46
					.10	47.46
2.5	20	2.22	.555	.051	.01	40.35
					.02	39.18
					.03	38.97
					.05	38.86
					.07	38.83
					.10	38.81
	30				.01	41.85
					.02	40.69
					.03	40.47
					.05	40.36
					.07	40.33
					.10	40.32
	60				.01	46.36
					.02	45.20
					.03	44.98
					.05	44.87
					.07	44.84
					.10	44.83
	150				.01	59.89
					.02	58.79
					.03	58.52
					.05	58.41
					.07	58.37
					.10	58.36
	400				.01	87.48
					.07	76.32
					.03	76.10
					.05	75.99
					.07	75.96
					.10	75.95
6.50	20	3.57	.89	.12	.01	80.37
					.02	71.20
					.03	69.51
					.05	68.68
					.07	68.40
					.10	68.27

TABLE III-XIII (Continued)

LINEAR MOTION SHOT-OUT DEGREE DATA SUMMARY

FLYER FORCE (WT)	INERT PRESS (PSI)	SEAT OIA (IN)	SYNCH (IN)	SHOOTING MILES (10)	Ratio PERIOD (SEC)	Ratio FORCE (LB)
Co Re	P ₁	O ₁	S	M ₁	t	F ₁
6.20	30	9.57	.89	.12	.01 .02 .03 .05 .07 .10	70.93 71.77 73.01 73.27 71.86 71.84
	100				.14 .15 .17 .18 .19 .20	71.84 73.46 73.16 73.90 72.66 72.53
	150				.31 .32 .33 .35 .37 .40	72.70 77.54 75.85 74.88 74.74 74.61
	400				.71 .72 .73 .75 .77 .79	75.42 76.66 78.96 78.09 73.85 73.72
10	20	9.43	1.11	.23	.01 .02 .03 .05 .07 .10	121.57 78.04 73.73 71.53 70.92 70.60
	30				.01 .02 .03 .05 .07 .10	126.59 103.35 77.00 75.85 76.74 75.92
	60				.01 .02 .03 .05 .07 .10	142.54 119.30 115.00 112.80 111.18 111.87
	150				.01 .02 .03 .05 .07 .10	180.38 167.15 162.85 160.64 160.04 159.79
	400				.01 .02 .03 .05 .07 .10	323.79 323.06 295.75 293.55 282.84 292.62
25	20	7.05	1.76	1.12	.01 .02 .03 .05 .07 .10	105.71 237.57 226.43 190.48 186.09 183.76

TABLE III-XIII (Continued)

LUNAR MOTION SHOT-OUT DEVICE DATA SUMMARY

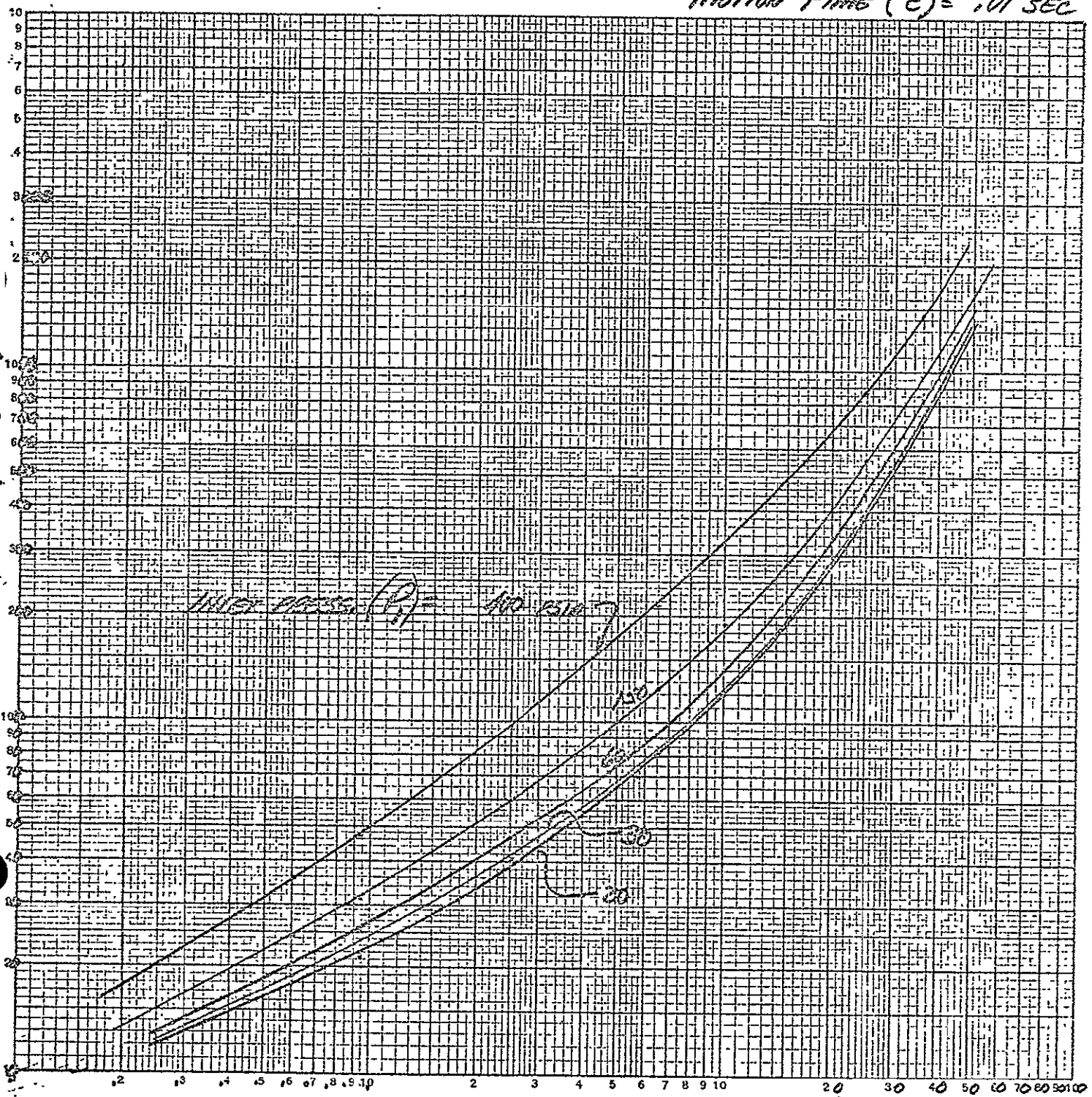
Flow Factor (W ²) C/A ₂	Inlet Press (PSIA) P ₁	Seat Dia (in) D _s	Stroke (in) S	Velocity (ft/sec) V ₁	Ratio V ₂ /V ₁ T	Exit Force (lb) F ₂
25	30	2.05	1.76	1.12	.01 .02 .03 .05 .07 .10 .01 .02 .03 .05 .07 .10	118.36 250.22 219.03 203.14 198.74 196.41 456.31 268.17 257.03 241.07 236.67 229.36
	60				.01 .02 .03 .05 .07 .10	520.17 402.02 370.88 359.94 357.55 348.22
	150				.01 .02 .03 .05 .07 .10	286.88 218.27 207.15 201.21 196.11 191.48
36	30	8.42	2.10	2.08	.01 .02 .03 .05 .07 .10	178.21 325.14 377.21 377.24 262.64 257.51
	60				.01 .02 .03 .05 .07 .10	714.33 393.13 305.29 290.24 280.59 275.46
	150				.01 .02 .03 .05 .07 .10	512.20 447.54 377.15 364.10 334.45 327.32
	400				.01 .02 .03 .05 .07 .10	778.77 609.16 540.72 505.67 496.02 490.89
					.01 .02 .03 .05 .07 .10	1427.59 1057.98 989.53 957.49 944.83 939.70

NOT REPRODUCIBLE

LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS. FLOW FACTOR AT VARIOUS INLET PRESSURES

MOTION TIME (t) = .01 SEC

K&E LOGARITHMIC 46 7402
 3 CYCLES
 LUFFILL & ESSER CO. VALVE FORCE EXP. (Fv) -



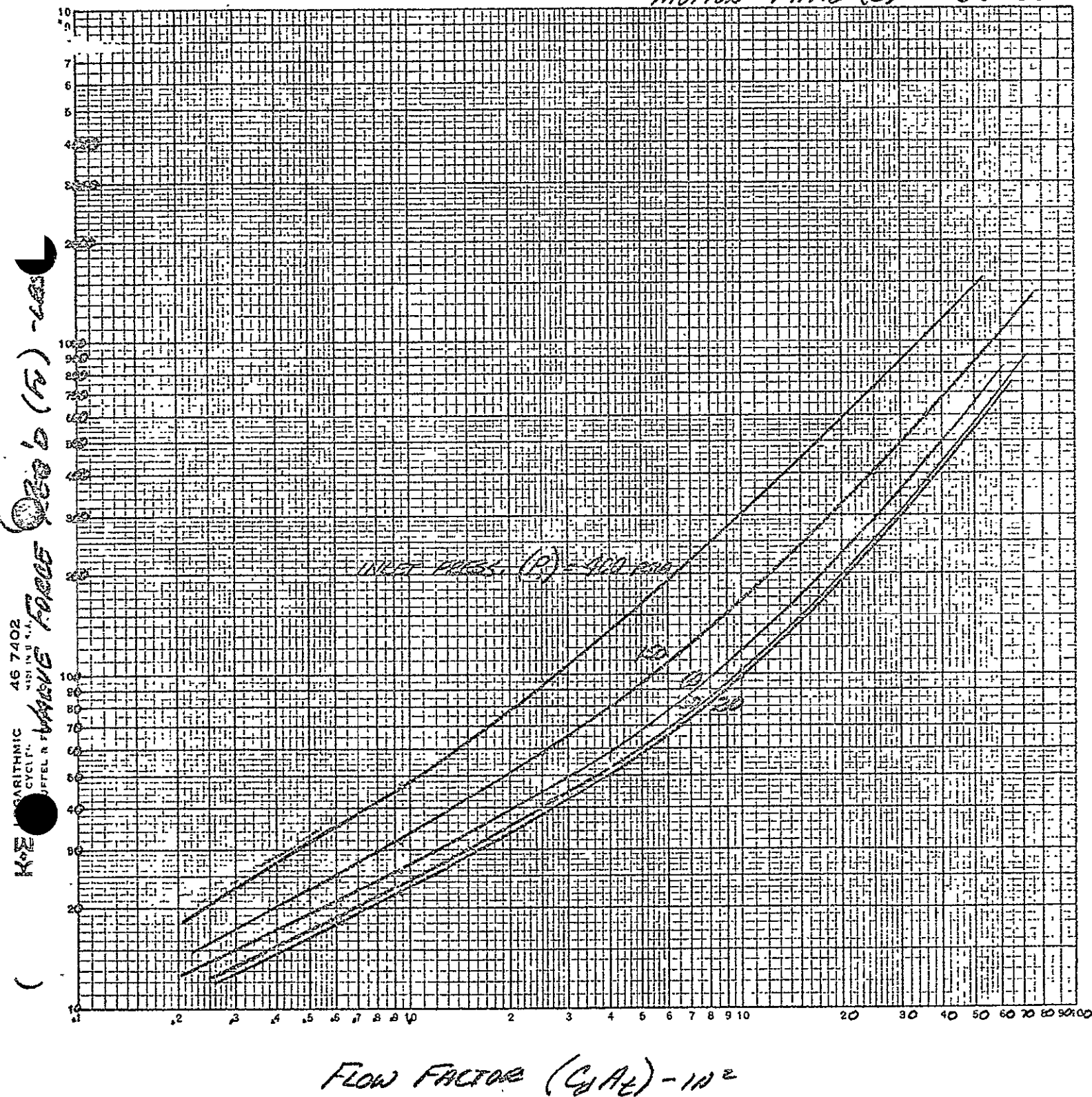
FLOW FACTOR ($C_v A_e$) - IN 2

Figure III-13

LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS FLOW FACTOR

AT VARIOUS INLET PRESSURES

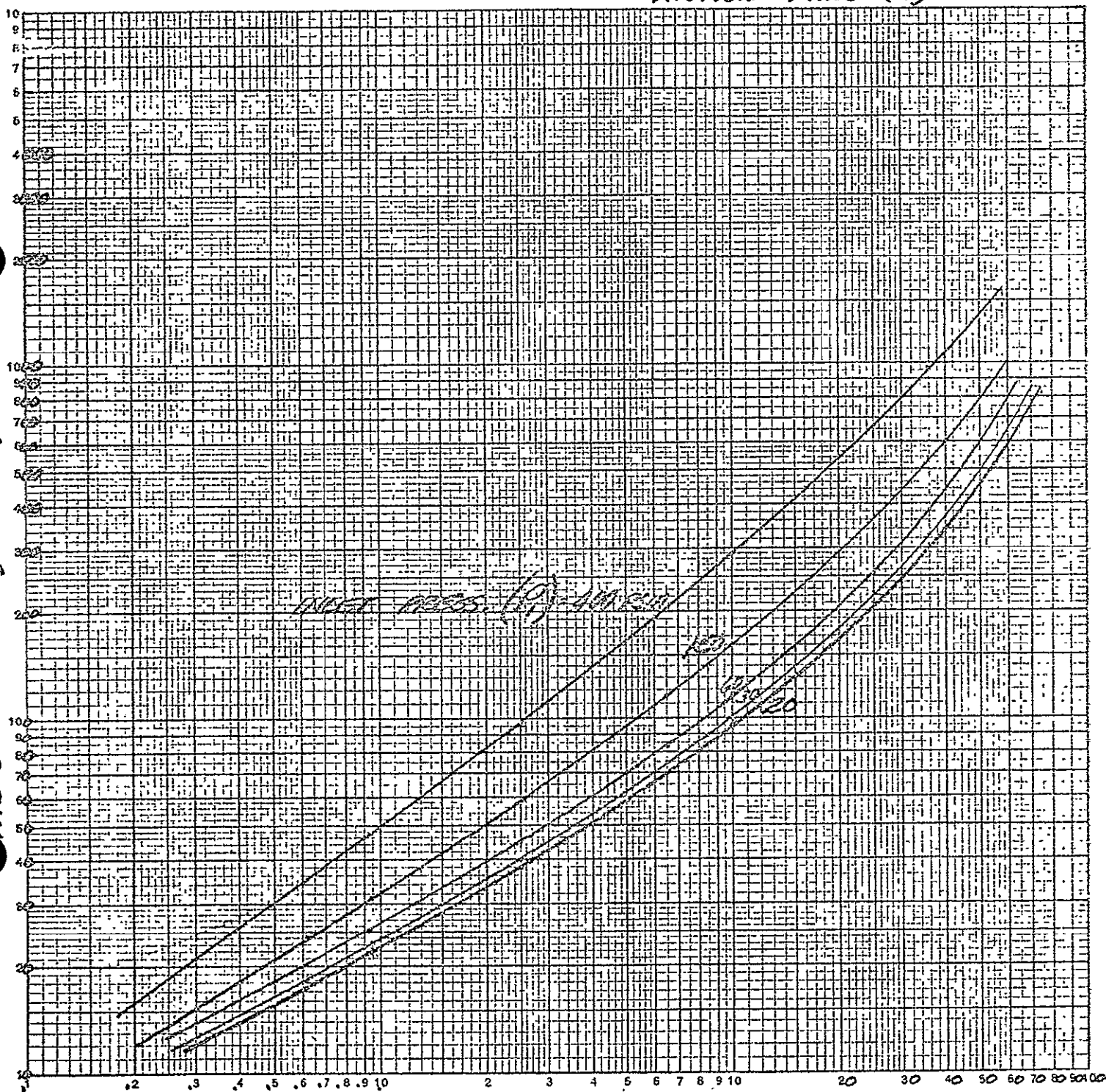
MOTION TIME (t) = .02 SEC.



LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS FLOW FACTOR AT VARIOUS INLET PRESSURES

MOTION TIME (t) = .03 SEC.

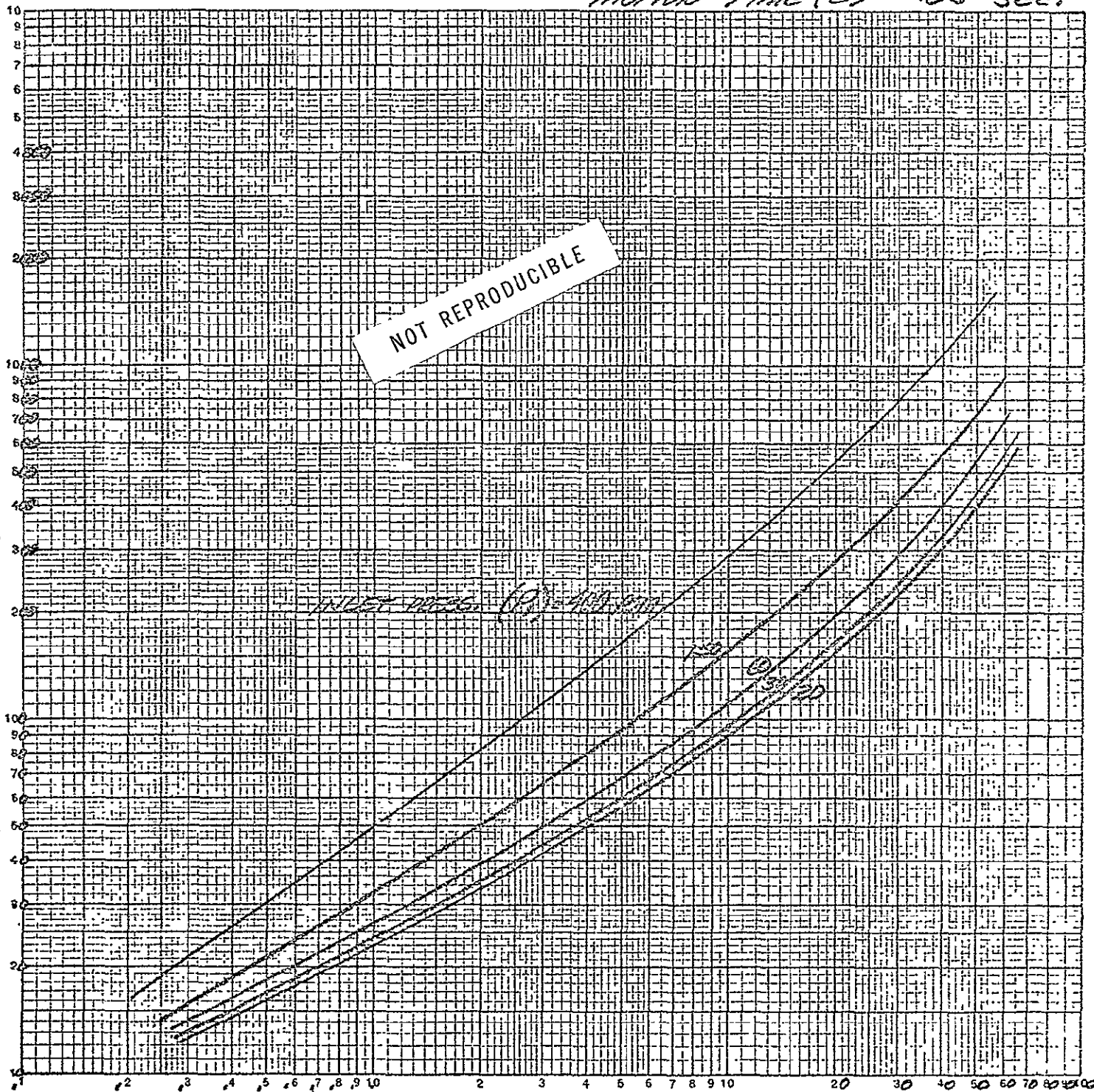
46 7402
LOGARITHMIC
VALVE CO. FORCE (Fv) - L.B.



FLOW FACTOR (Cv A_v) - in²

LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS.
FLOW FACTOR
AT VARIOUS INLET PRESSURES

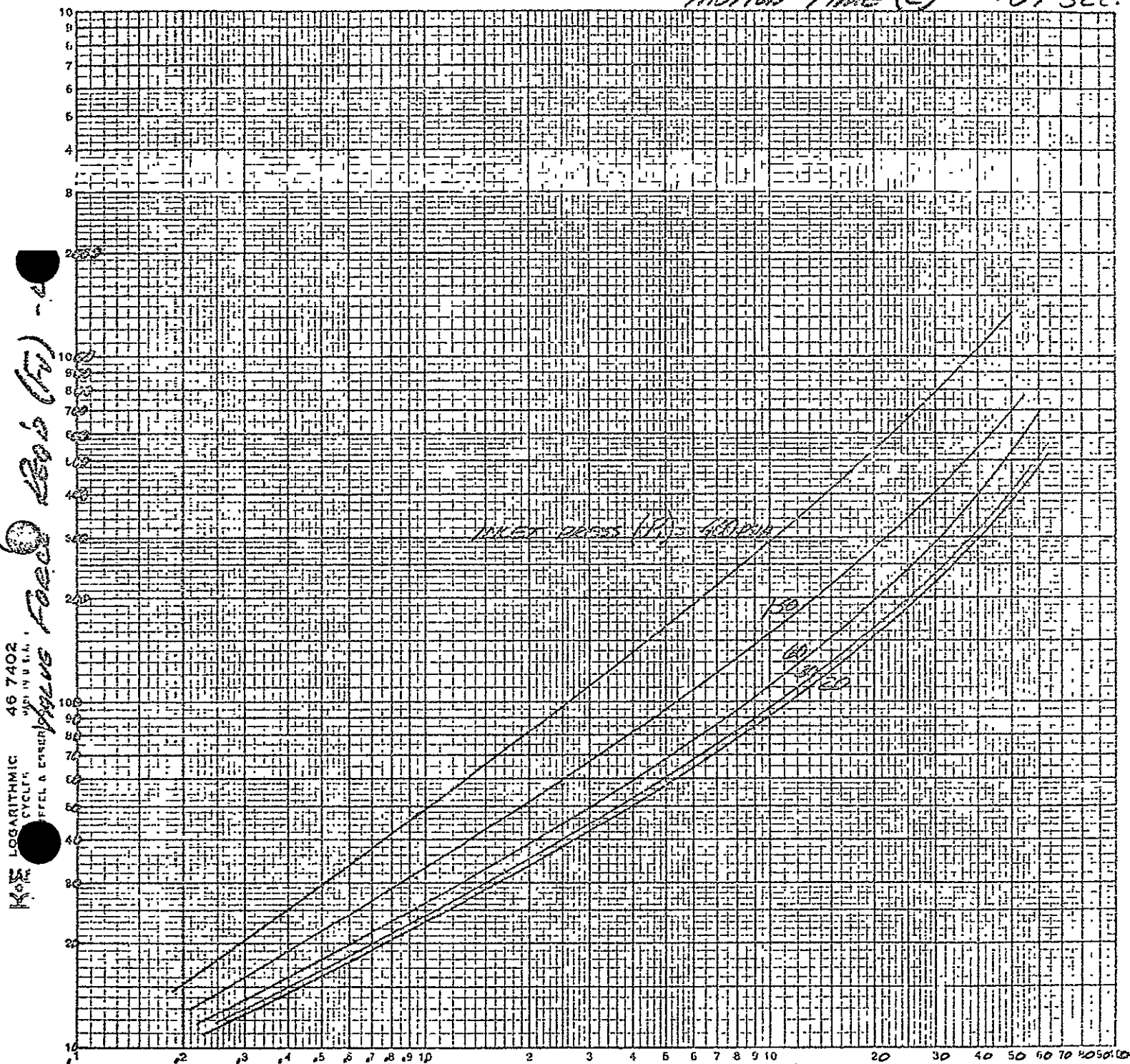
MOTION TIME (t) = .035 sec.



Flow Factor $(C_d A_t) - 10^2$

LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS FLOW FACTOR AT VARIOUS INLET PRESSURES

MOTION TIME (t) = .07 SEC.

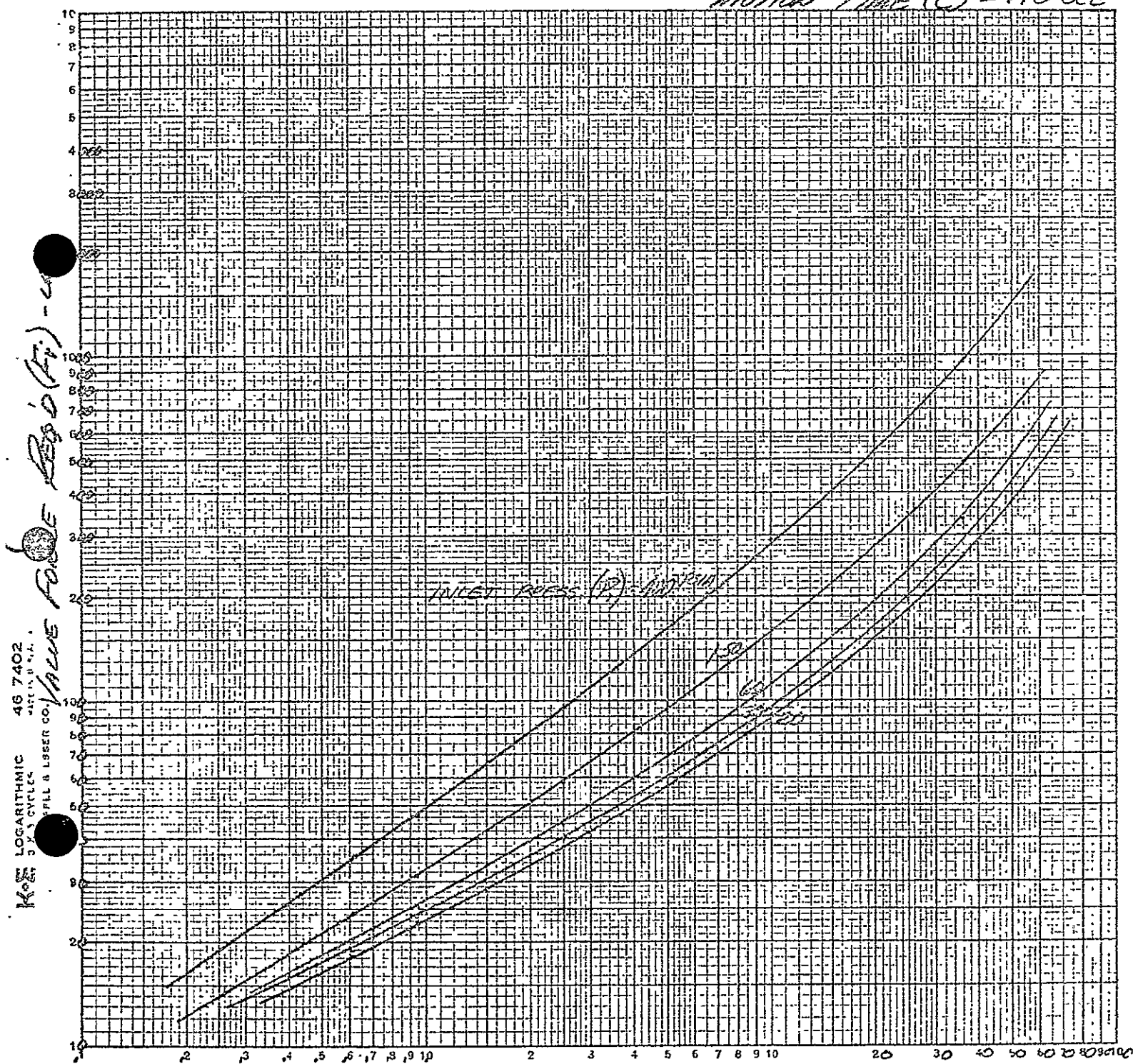


Flow Factor ($C_d A_6$) - in^2

Figure III-17

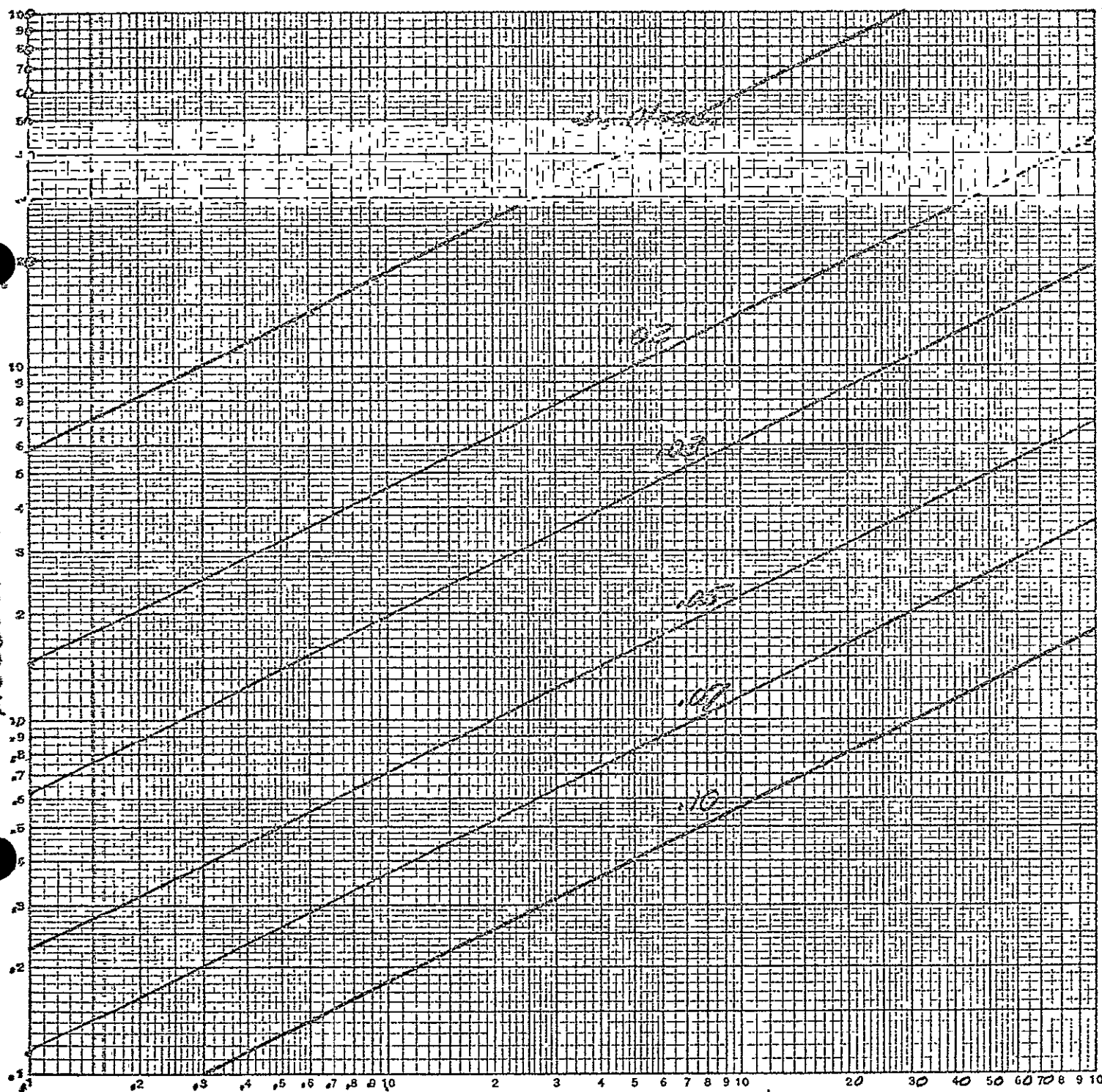
LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS. FLOW FACTOR AT VARIOUS INLET PRESSURES

MOTION TIME (t) = .10 SEC



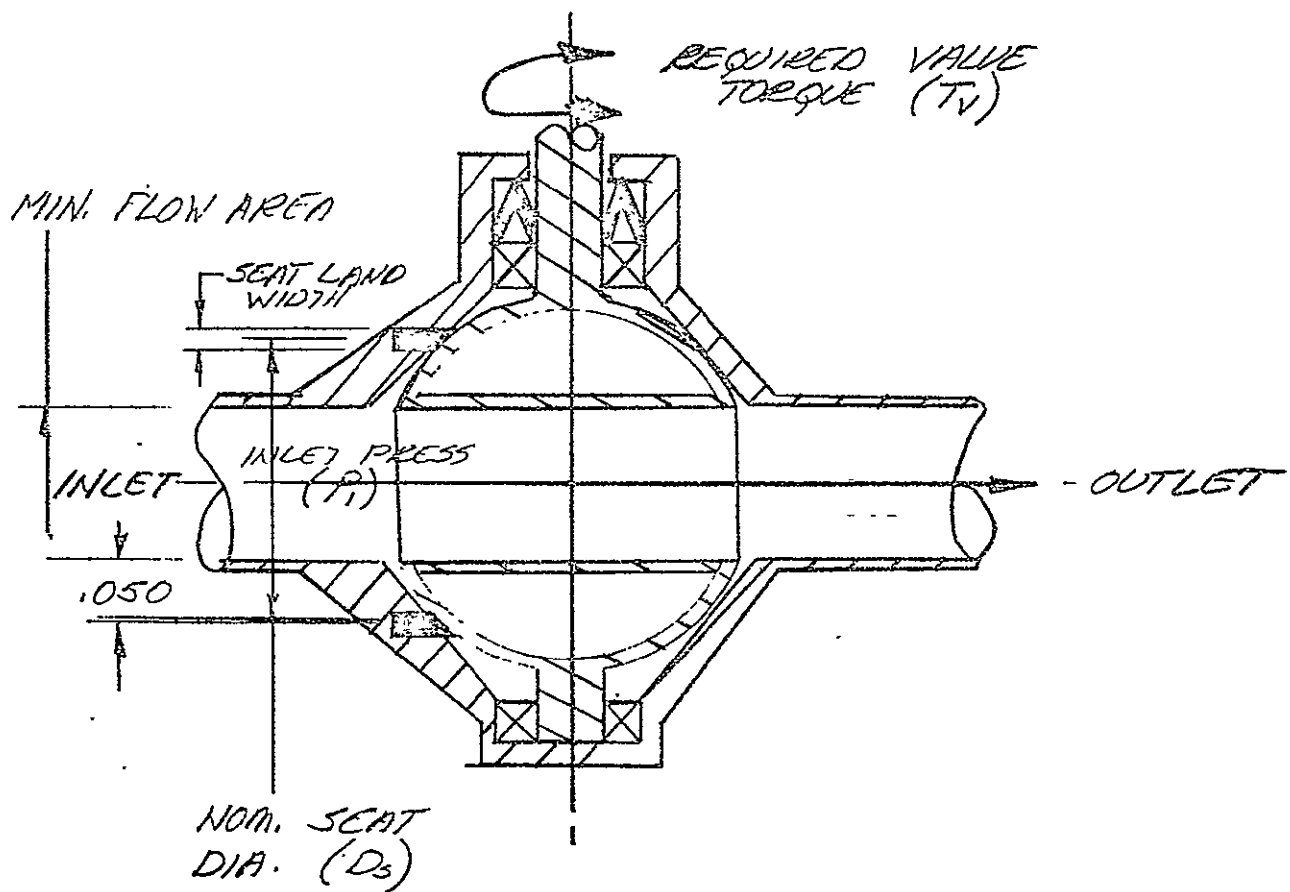
FLOW FACTOR (Cd.Ak) - IN²

LINEAR MOTION SHUT-OFF DEVICE FLOW FACTOR ($C_d A_t$) VS. ACCELERATION FOR VARIOUS MOTION TIMES



Flow Factor ($C_d A_t$) - IN^2

ROTARY MOTION SHUT-OFF DEVICE - SCHEMATIC
DRAWING



For straight through, essentially unobstructed flow passages common in most rotary motion shut off devices, an equivalent orifice flow coefficient (C_d) of .90 is assumed. For any value of flow factor ($C_d A_t$), the minimum cross sectional flow area (A_t) can be calculated.

Assuming a nominal seal preload of 5 lb/circumferential inch, a seat land width of .030, constant angular acceleration, and negligible drag due to the shaft seal and shaft bearings, the required valve torque (T_v) can be expressed as:

$$T_v = \left\{ 5 \pi D_s + P_1 \frac{\pi}{4} [(D_s + .03)^2 - (D_s)^2] f \ell + I_{xx} \dot{\omega} \right\} \quad (8)$$

Where D_s = nominal seat diameter - inches
 P_1 = inlet pressure - psia
 f = coefficient of friction between seat seal and moving component
 ℓ = effective moment arm length seal to axis of rotation - inches
 I_{xx} = moment of inertia of rotating component about axis of rotation - in.-lb-sec²
 $\dot{\omega}$ = angular acceleration of rotating component - radians/sec²

By definition, the flow passage minimum flow area shall be $\frac{C_d A_t}{C_d}$. With reference to Figure III-2, the seat inside diameter is .1 larger than the minimum flow passage diameter and for a .030 wide seat land, the nominal seat diameter can be expressed as:

$$D_s = \sqrt{\frac{C_d A_t}{.9 \times \frac{\pi}{4}}} + .13 \quad (9)$$

The coefficient of friction between the seal material and the ball is estimated at .1, equal to teflon against well finished steel. It is anticipated that seal interface surfaces will be super finished in the actual application.

The effective moment arm length from seat seal interface to axis of rotation (ℓ) and rotating component moment of inertia (I_{xx}) are functions of the physical dimensions of the ball. To assure adequate overlap of the seat seal in the open and closed position, the maximum diameter of the seat seal is defined as $2 \left(\frac{D_B}{2} \cos 45^\circ \right)$. Using previous developed values,

$$D_B = \frac{\sqrt{\frac{C_d A_t}{C_d \frac{\pi}{4}}} + .16}{.707} \quad (10)$$

Where D_B = ball outside diameter - inches

Having established a relationship for the ball diameter (D_B), the effective moment arm from the axis of rotation to the seal nominal diameter can be calculated. To simplify this expression and to add an element of conservatism to the analysis a simplified relationship was chosen where:

$$\ell = \frac{D_B}{2} \quad (11)$$

$$\left[\text{The exact relationship for more refined analysis would be} \right]$$

$$\ell = \sqrt{\left(\frac{D_B}{2}\right)^2 - \left(.707 \frac{D_B}{2} - .015\right)^2}$$

To simplify the determination of the moment of inertia of the rotating component, it is assumed to be a hollow sphere. The contributions of the flow passage liner and the mounting shafts are ignored but a conservative wall thickness estimate is used to cover potential error of this assumption. The assumed wall thickness is defined as:

$$w = .090 + .02 (D_B) \quad (12)$$

Where w = wall thickness - inches
 D_B = ball diameter - inches

Employing equation (12) and previous relationships for dimensions of the ball, and assuming stainless steel construction

$$I_{xx} = .00123 \left[\left(\frac{\sqrt{\frac{C_d A_t}{.707}} + .16}{1.414} \right)^5 - \left(\frac{.48 \sqrt{\frac{C_d A_t}{.707}}}{.707} + .023 \right)^5 \right] \quad (13)$$

Assuming constant angular acceleration and a "stroke" of 90° to cycle from open to closed,

$$\dot{\omega} = \frac{\pi}{t^2} - \frac{\text{RAD}}{\text{sec}^2} \quad (14)$$

where t - allowable motion time - seconds

Combining the functions of Equations (9) through (14) into Equation (8), the required valve torque is defined in terms of flow factor ($C_d A_t$), inlet pressure (P_1) and allowed motion time (t) only:

$$T_v = \frac{\sqrt{\frac{C_d A_t}{.707}} + .16}{1.414} \left[5\pi \left(\sqrt{\frac{C_d A_t}{.707}} + .13 \right) + P_1 \frac{\pi}{4} \left(\sqrt{\frac{C_d A_t}{.707}} + .16 \right)^2 - \right.$$

$$P_1 \frac{\pi}{4} \left(\frac{C_d A_t}{.707} \right) + .00123 \frac{\pi}{t^2} \left[\left(\frac{\sqrt{\frac{C_d A_t}{.707}} + .16}{1.414} \right)^5 - \left(\frac{.48 \sqrt{\frac{C_d A_t}{.707}}}{.707} + .023 \right)^5 \right] \quad (15)$$

A presentation of data resulting from solving Equation (15) for 7 flow factors ($C_d A_t$), 5 pressures (P_1) and 6 allowable motion times (t) is in tabular form in Table III-XIV and graphical form in Figures III-21 through III-26. For reference only, the relationship between flow factor ($C_d A_t$), and ball diameter and ball mass are shown in Figures III-27 and III-28.

III-4 B. Actuator Performance

Of the numerous actuator types available (Reference Table III-IX), the solenoid, torque motor and linear pneumatic cylinder appeared most promising and inherently reliable for the anticipated mission. The potential for an onboard hydraulic system existed and a linear hydraulic cylinder was added as a potential alternate for further analysis. Each actuator type was analyzed on the basis of a single design configuration for each type. The selected design configuration is not necessarily the optimum configuration for the total range of interest.

III-4B 1. Solenoid - Several solenoid design configurations are available which result in force-stroke characteristics similar to those required for the linear motion shut off devices. Of these configurations, a flat face plunger configuration was chosen for analysis. With reference to Figure III-29, the solenoid analysis program is developed.

For a flat faced plunger solenoid using 400 series corrosion resistant steel as the core material, an operating flux density of 8 kilogauss is assumed. The force generated is expressed as:

$$F_A = \frac{\pi/4 D_A^2 B^2}{1.73} \quad (16)$$

Where F_A = total force developed - lbs
 D_A = armature diameter - in.
 B = operating flux density = 8 kilogauss

The total force developed is the sum of the useful output force and the force necessary to accelerate the solenoid's moving mass. Using a plunger of 1.25,

$$F_A = F_V + \rho 1.25 \frac{\pi}{4} D_A^3 \left(\frac{2s}{386.4 t^2} + 20 \right) \quad (17)$$

Where F_A = total force developed - lb
 F_V = useful output force - lb
 ρ = density of moving mass material - .283 lb/in.³
 s = stroke - in.
 t = allowed motion time - seconds

TABLE III-XIV

ROTARY MOUNT SHUT-OFF DEVICE DATA SUMMARY

Flow Factor (W ²)	Inlet Press (PSIA)	Seal Dia (IN)	Base O.D. (IN)	Ratio of Inlet to Seal Dia	Ratio of Inlet to Base O.D.	Ratio of Inlet to Seal Dia	Ratio of Inlet to Base O.D.	Ratio of Inlet to Seal Dia	Ratio of Inlet to Base O.D.	Ratio of Inlet to Seal Dia	Ratio of Inlet to Base O.D.
Ratio	Ratio	Ratio	Ratio	Ratio	Ratio	Ratio	Ratio	Ratio	Ratio	Ratio	Ratio
.62	30	1.07	1.0	.01	7.13	.25					
				.02	3.20						
				.03	2.37						
				.05	1.94						
				.07	1.82						
				.10	1.76						
	30			.01	7.93						
				.02	3.40						
				.03	2.56						
				.05	2.13						
				.07	2.02						
				.10	1.95						
	60			.01	8.33						
				.02	4.11						
				.03	3.16						
				.05	2.73						
				.07	2.61						
				.10	2.55						
	100			.01	11.31						
				.02	5.78						
				.03	4.94						
				.05	4.51						
				.07	4.39						
				.10	4.33						
	400			.01	15.76						
				.02	10.73						
				.03	9.89						
				.05	9.46						
				.07	9.35						
				.10	9.28						
.94	30	1.28	1.0	.01	16.00	.85					
				.02	8.00						
				.03	6.40						
				.05	5.00						
				.07	4.76						
				.10	4.62						
	30			.01	16.79						
				.02	8.40						
				.03	6.72						
				.05	5.24						
				.07	4.96						
				.10	4.83						
	60			.01	17.15						
				.02	8.58						
				.03	6.88						
				.05	5.50						
				.07	5.19						
				.10	5.09						
	100			.01	18.18						
				.02	9.09						
				.03	7.27						
				.05	6.73						
				.07	6.46						
				.10	6.32						
	400			.01	26.93						
				.02	13.47						
				.03	11.19						
				.05	10.32						
				.07	10.00						
				.10	9.80						

TABLE III-XIV (Continued)

ROTARY MOTION SHUT-OFF DEVICE DATA SUMMARY

6

Flow Rate (W)	Inlet Press (PSIA)	Seal Dia (in)	Ball O. D. (in)	Revol Per Min (RPM)	Revol Per Sec (RPS)	Meaning Mass (lb)				
2.50	20	2.01	3	.01 .02 .03 .05 .07 .10	17.77 32.40 16.92 9.92 7.99 6.96	1.00				
	30			.01 .02 .03 .05 .07 .10	105.18 31.31 17.63 10.63 8.70 7.67					
	60			.01 .02 .03 .05 .07 .10	107.31 33.25 19.77 12.76 10.83 9.81					
	150			.01 .02 .03 .05 .07 .10	115.71 39.84 26.17 14.16 17.23 16.21					
	400			.01 .02 .03 .05 .07 .10	131.49 57.62 43.94 36.94 35.01 33.98					
6.50	20	3.18	4.5	.01 .02 .03 .05 .07 .10	770.30 203.63 98.70 44.97 30.16 27.30	3.2				
	30			.01 .02 .03 .05 .07 .10	772.04 205.40 100.46 46.73 31.93 29.07					
	60			.01 .02 .03 .05 .07 .10	777.36 210.70 105.76 52.03 37.23 29.36					
	150			.01 .02 .03 .05 .07 .10	783.25 226.54 121.65 67.92 53.12 45.25					
	400			.01 .02 .03 .05 .07 .10	837.40 270.73 165.79 112.06 97.26 89.40					

TABLE III-XIV (Continued)

ROTARY MOTION SHUT-OFF DEVICE DATA SUMMARY

③

Flow Factor (in ²) C ₀ A ₀	Inlet Press (PSIA) P ₀	Seat Dia (in) D _s	Ball Dia (in) D _b	Ball Motion (sec) t	Rec'd Torque (in lb) T _r	Measured Torque (in lb) T _m				
10	20	3.90	5.6	.01 .02 .03 .05 .07 .10	14.8, 86 508 228.8 100.24 62.07 41.8	5.3				
	30			.01 .02 .03 .05 .07 .10	14.23, 33 512 241.5 102.9 64.74 44.4					
	60			.01 .02 .03 .05 .07 .10	14.81 520 250 110.4 72.8 52.5					
	150			.01 .02 .03 .05 .07 .10	20.05 544 273 135 96.8 76.6					
	400			.01 .02 .03 .05 .07 .10	207.2 611 340 201.4 163.8 143.5					
25	20	6.09	8.7	.01 .02 .03 .05 .07 .10	1540.8 3703 1766 670.5 368.7 208.4	16.7				
	30			.01 .02 .03 .05 .07 .10	1546.7 3712 1773 677.1 375.3 214.4					
	60			.01 .02 .03 .05 .07 .10	1548.6 3732 1792 696 394.9 234.5					
	150			.01 .02 .03 .05 .07 .10	1554.5 3790 1851 755 453.7 243.4					
	400			.01 .02 .03 .05 .07 .10	1570.8 4154 2014 919 617.3 236.8					

PS 300 A

TABLE III-XIV (Continued)

ESTIMATED MARINE SNIT-OFF DENSITIES DATA SUMMARY

④

FLAT FACTOR (W)	INLET PRESS (PSIA)	SEAT DIA (IN)	BACK O.P. (IN)	READ MOTOR PRESS (SCF)	READ TAPPS (W CO)	MOTOR PRESS (LB)				
36	20	7.20	10.4	.01 .02 .03 .05 .07 .10 .101 .102 .103 .105 .107 .110 .101 .102 .103 .105 .107 .110 .101 .102 .103 .105 .107 .110 .101 .102 .103 .105 .107 .110	3577.3 9005 9015 1506 806.4 434.7 3577.6 9018 4055 1575 815.7 444.1 3572.4 9002 4052 1543 843.8 472.1 3570.9 9126 4167 1627 978 536.3 3614.3 9360 4400 1861 1161.9 719.3	27.2				

8-20-68

N. S. SEMILOGARITHMIC 359-91G
 CURTIS & ISSER CO.
 5 CYCLES X 70 DIVISIONS

TORQUE REQ'D AT VALVE - IN. LB.

VALVE TORQUE VS FLOW FACTOR ($C_d A_t$) AT VARIOUS
 INLET PRESSURES FOR ROTARY VALVE SHUT-OFF
 DEVICE

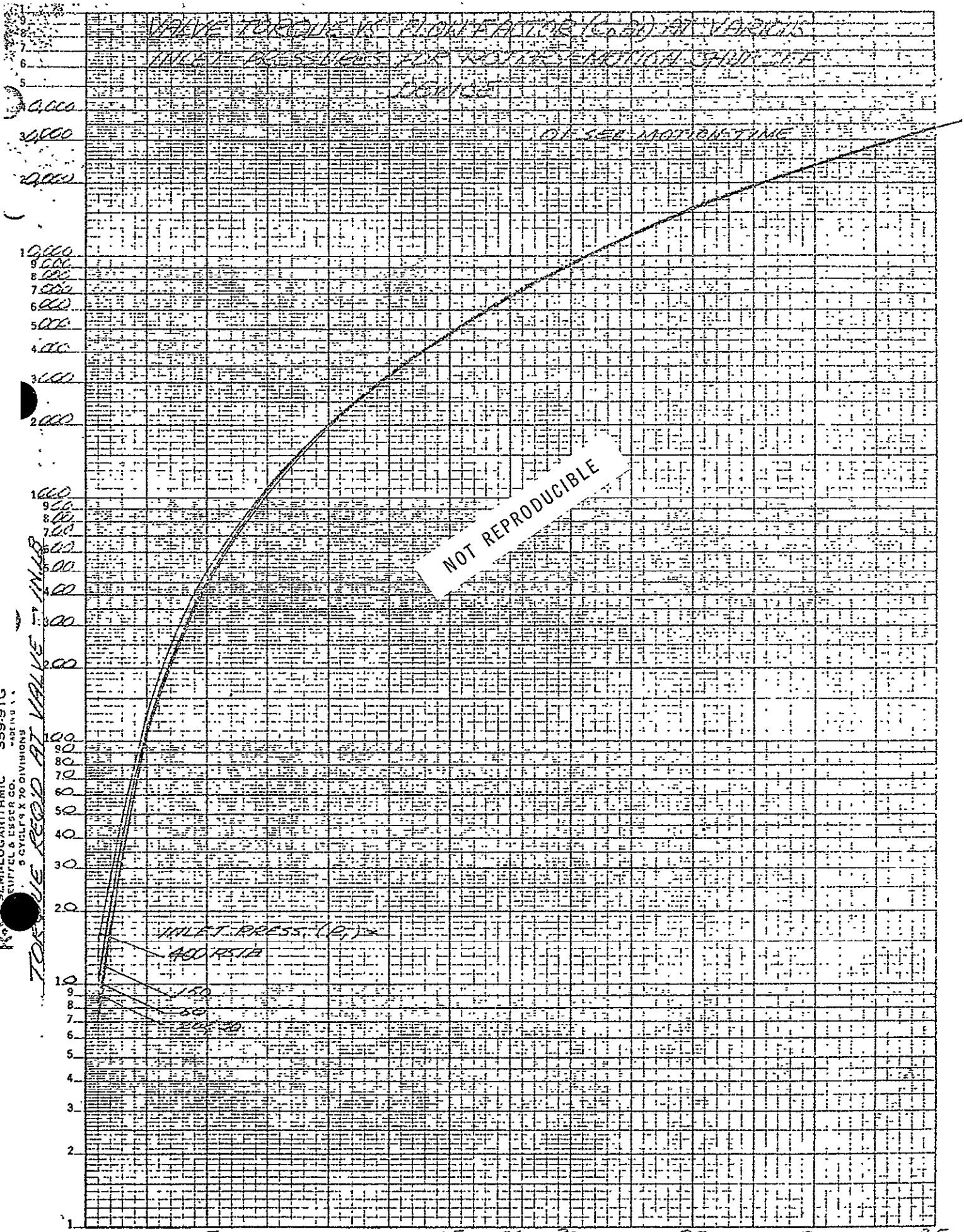
OF SEE-MOTION-TIME

NOT REPRODUCIBLE

INLET PRESS. (P_1) -
 400 PSIA
 150
 50
 20

FLOW FACTOR ($C_d A_t$) - IN.²

Figure III-21



K&E SEMI-LOGARITHMIC 359-91G
 PUFFEL & ESCHER CO. MAY 1961, 14
 5 CYCLES X 70 DIVISIONS

TORQUE REQ'D AT VALVE $\sim$ IN.-LB.
 VALVE TORQUE VS. FLOW FACTOR ($C_d A_t$) AT VARIOUS
 INLET PRESSURES FOR ROTARY MOTION SHUT-OFF
 VALVE

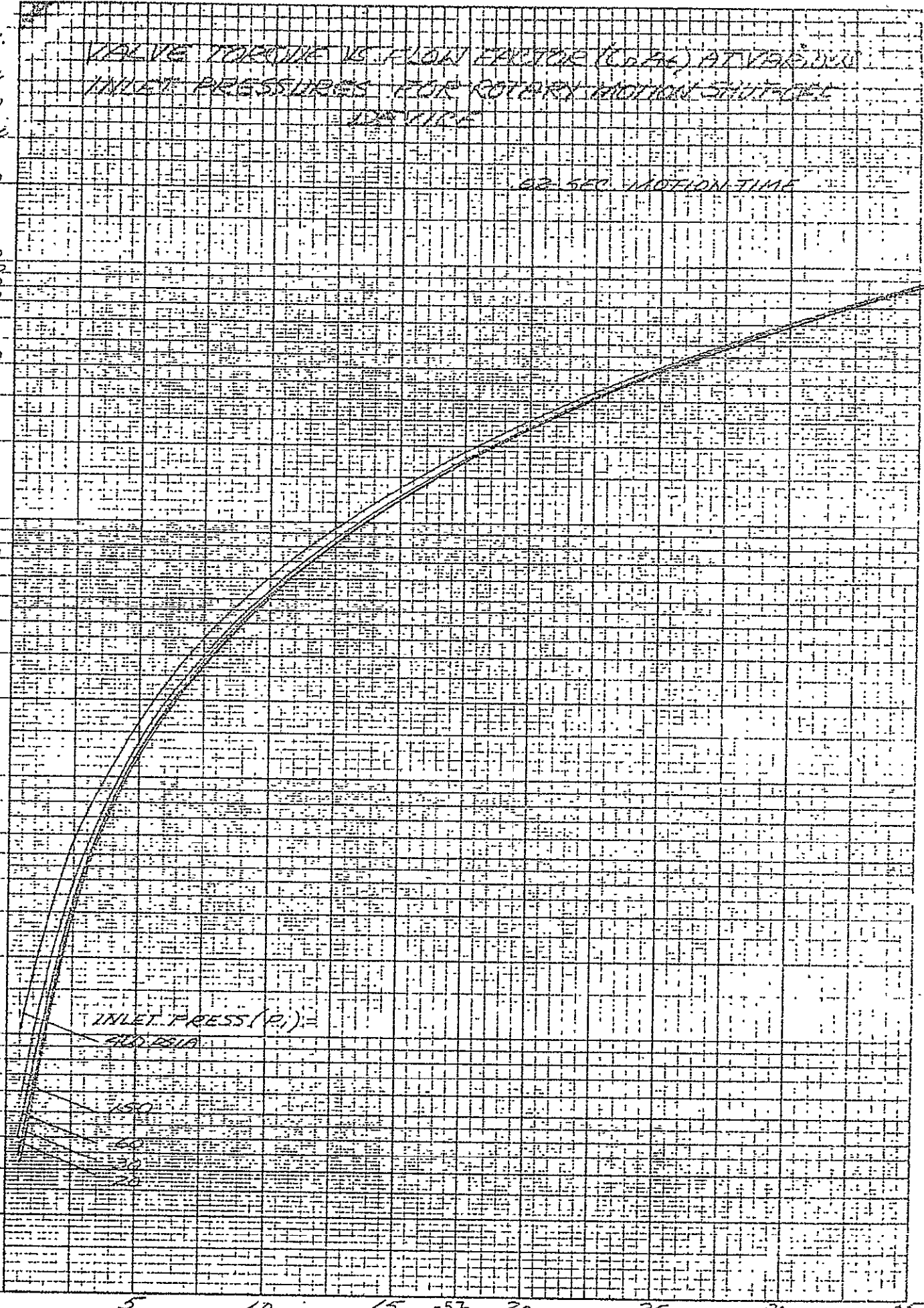
VALVE TORQUE VS. FLOW FACTOR ($C_d A_t$) AT VARIOUS
 INLET PRESSURES FOR ROTARY MOTION SHUT-OFF
 VALVE

62.5 SEC. MOTION TIME

INLET PRESS (P_1) =
 400 PSIA
 150
 60
 30
 20

FLOW FACTOR ($C_d A_t$) $\sim$ IN²

Figure III-22



W. W. SEMI-LOGARITHMIC
 HUFFEL & EAGER CO.
 5 CYCLES X 20 DIVISIONS

100% REQUIRED AT VALUE - IN. C.B. -

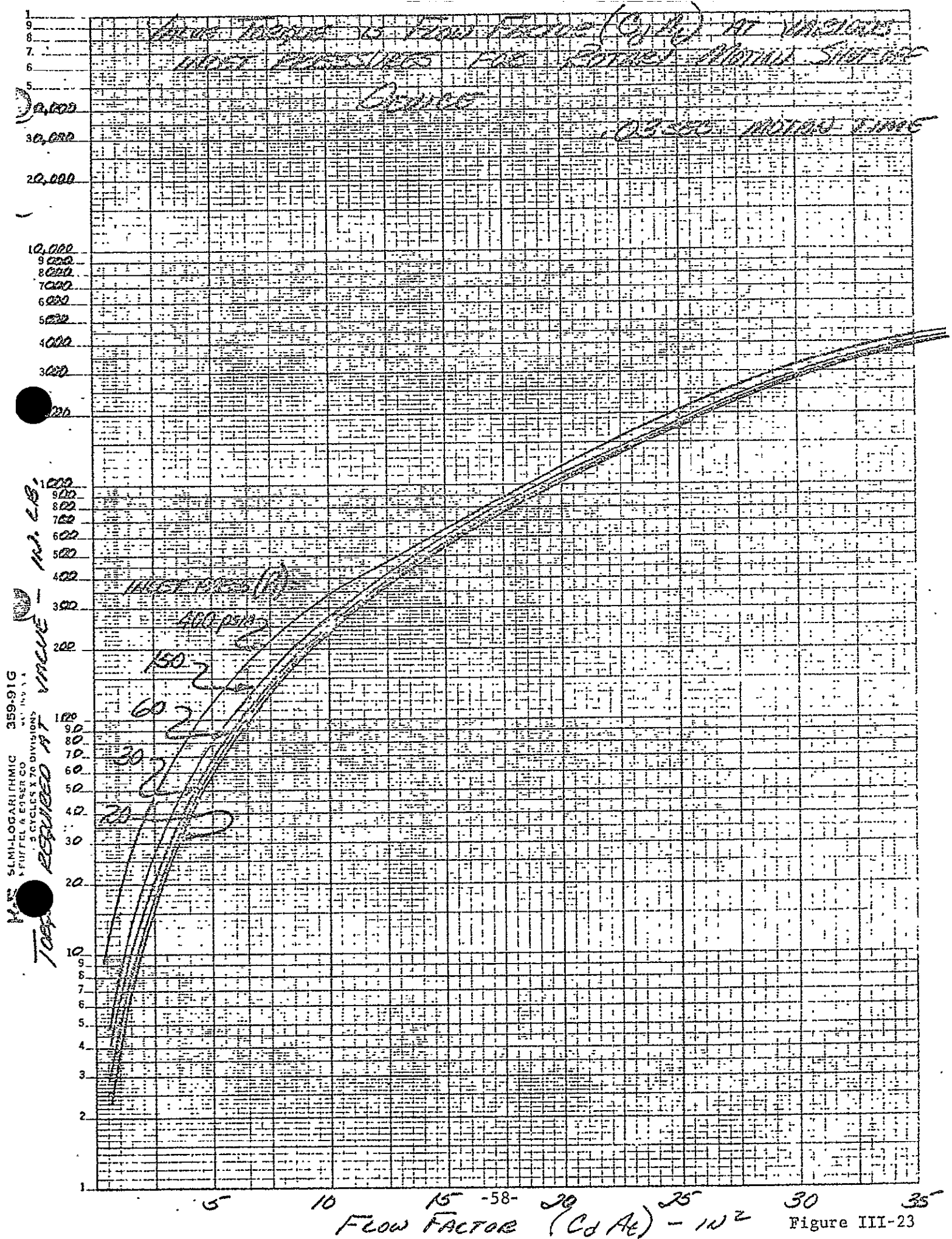


Figure III-23

VALUE TORQUE VS FLOW FACTOR ($C_d A_t$) AT VARIOUS INLET PRESSURES FOR BINARY MOTION SHOCK-EE DEVICE

0.050 SEC. MOTION TIME

359-916
SEMILOGARITHMIC
PAPER
70 DIVISIONS
IN. L.B.
Torque vs $C_d A_t$ at various
inlet pressures

30,000
20,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
10
9
8
7
6
5
4
3
2
1

5 10 15 20 25 30 35
FLOW FACTOR ($C_d A_t$) IN 2

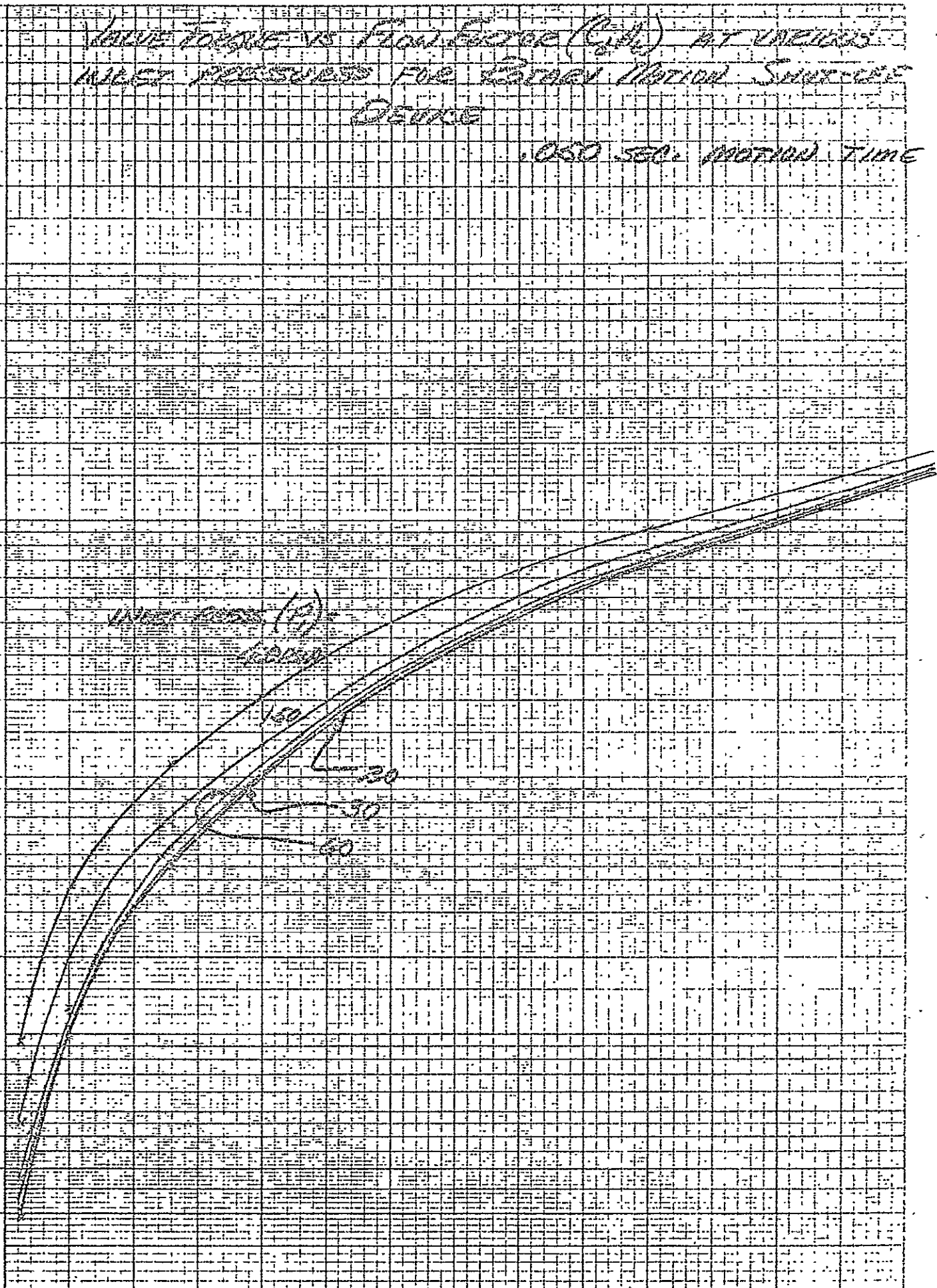


Figure III-24

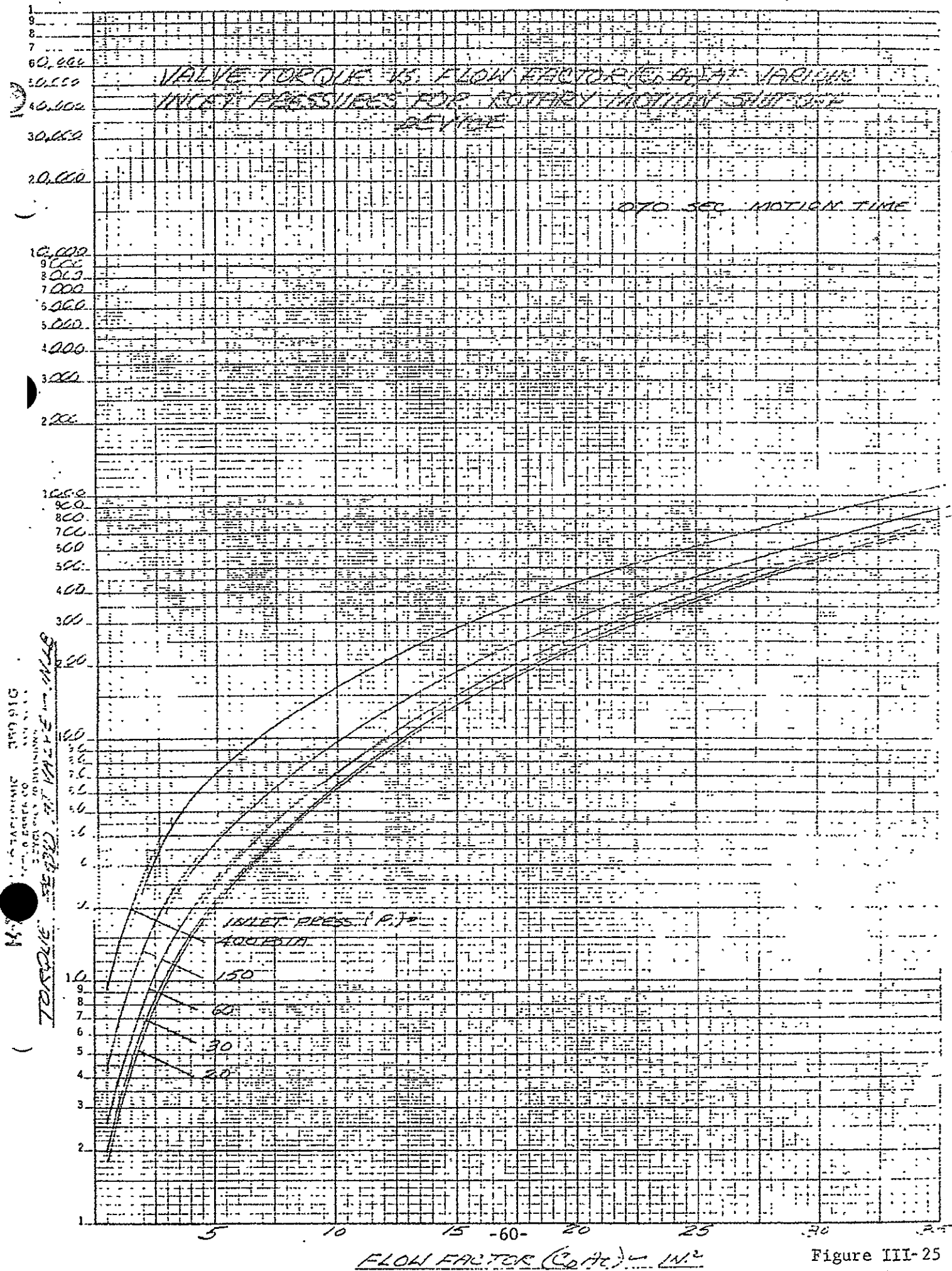


Figure III-25

K-1 SEMI-LOGARITHMIC 359-916
 SUPPLY & DEMAND CO. 1961
 5 CYCLES TO DIVISIONS

TORQUE REQ'D AT VALVE - IN. LB.

VALVE TORQUE VS. FLOW FACTOR (C_{DA}) AT VARIOUS
 INLET PRESSURES FOR ROTARY MOTION SHUT-OFF
 DEVICE

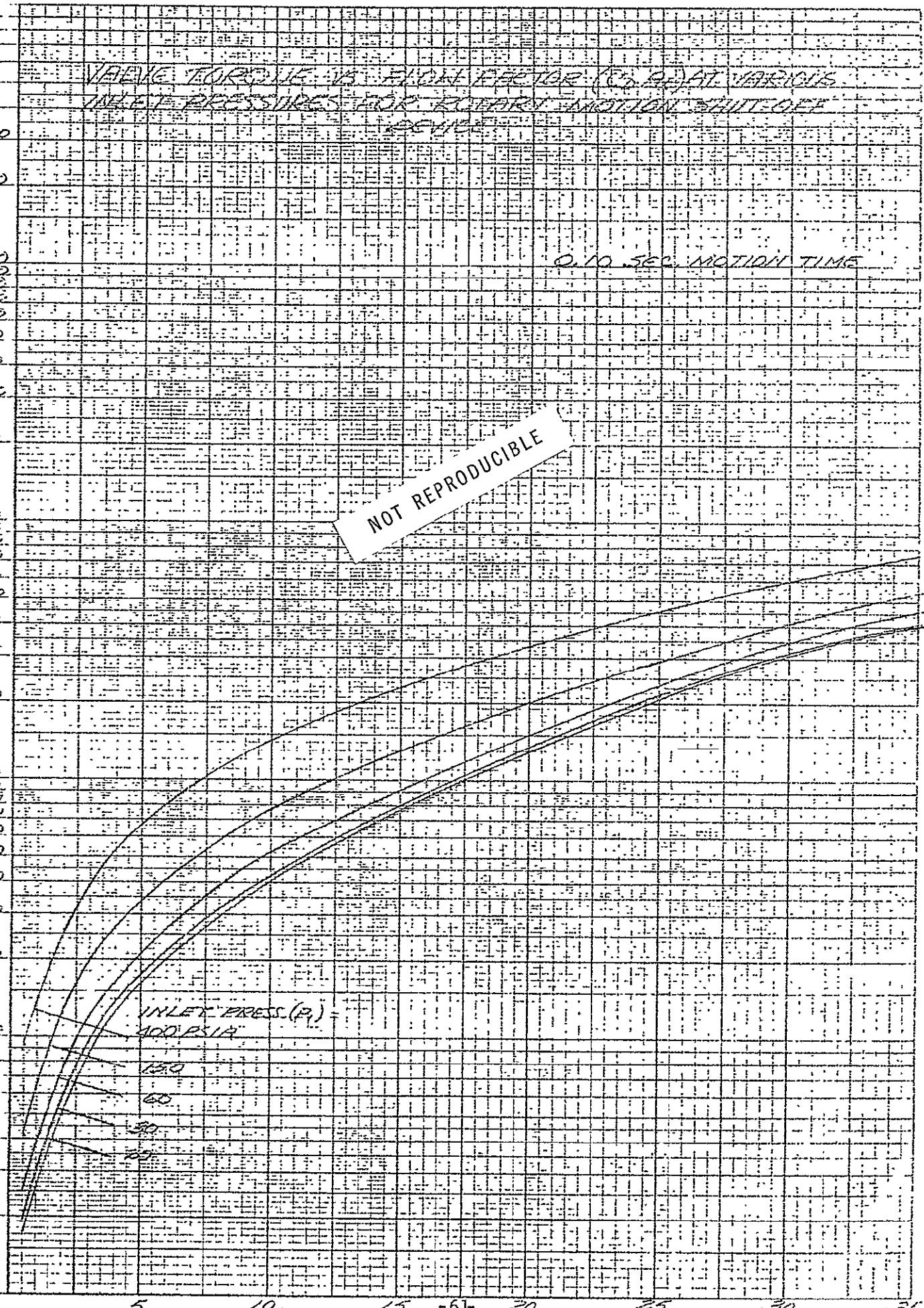
0.10 SEC. MOTION TIME

NOT REPRODUCIBLE

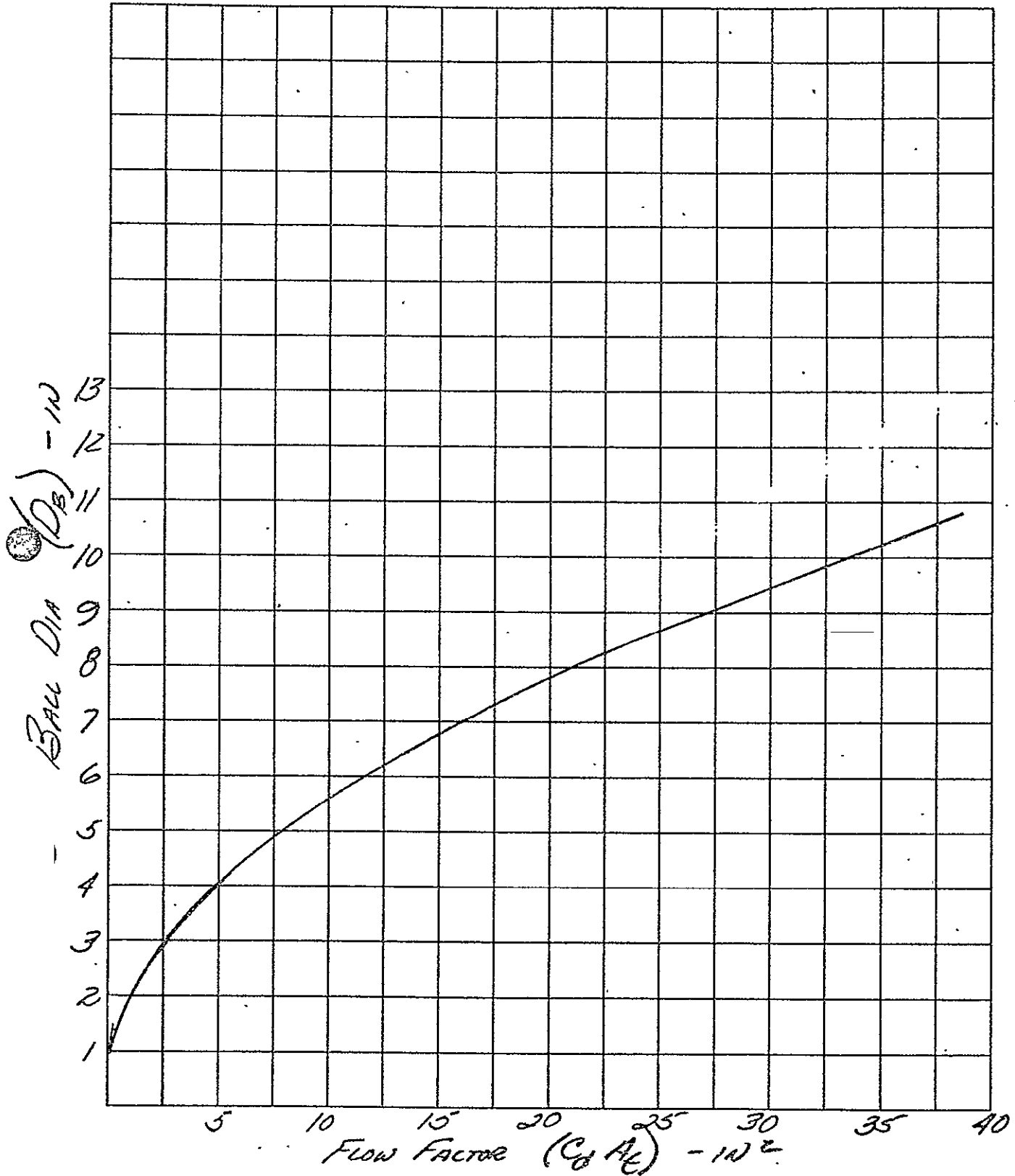
INLET PRESS. (P_1)
 400 PSIA
 150
 60
 50
 20

FLOW FACTOR (C_{DA}) - IN²

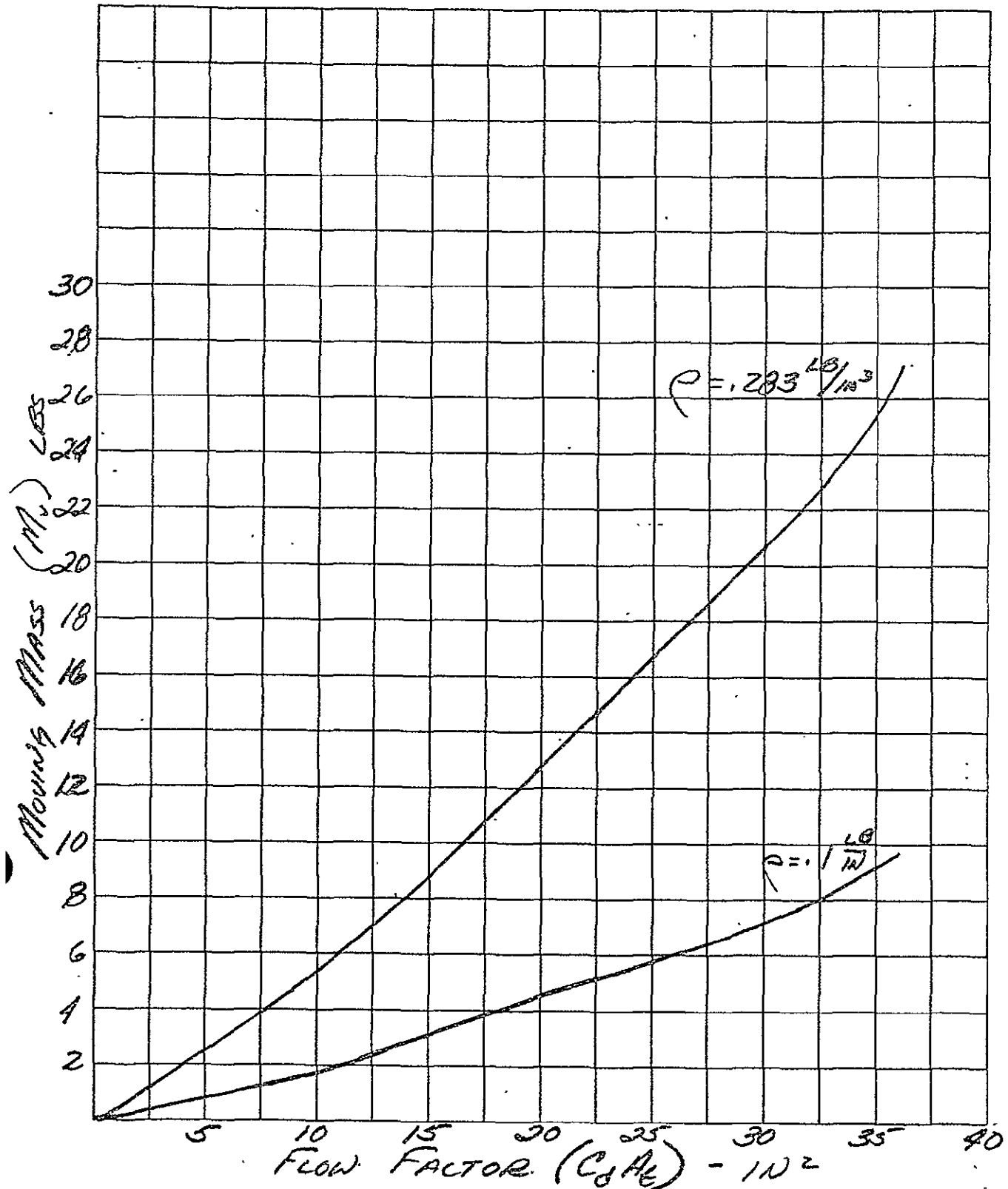
Figure III-26



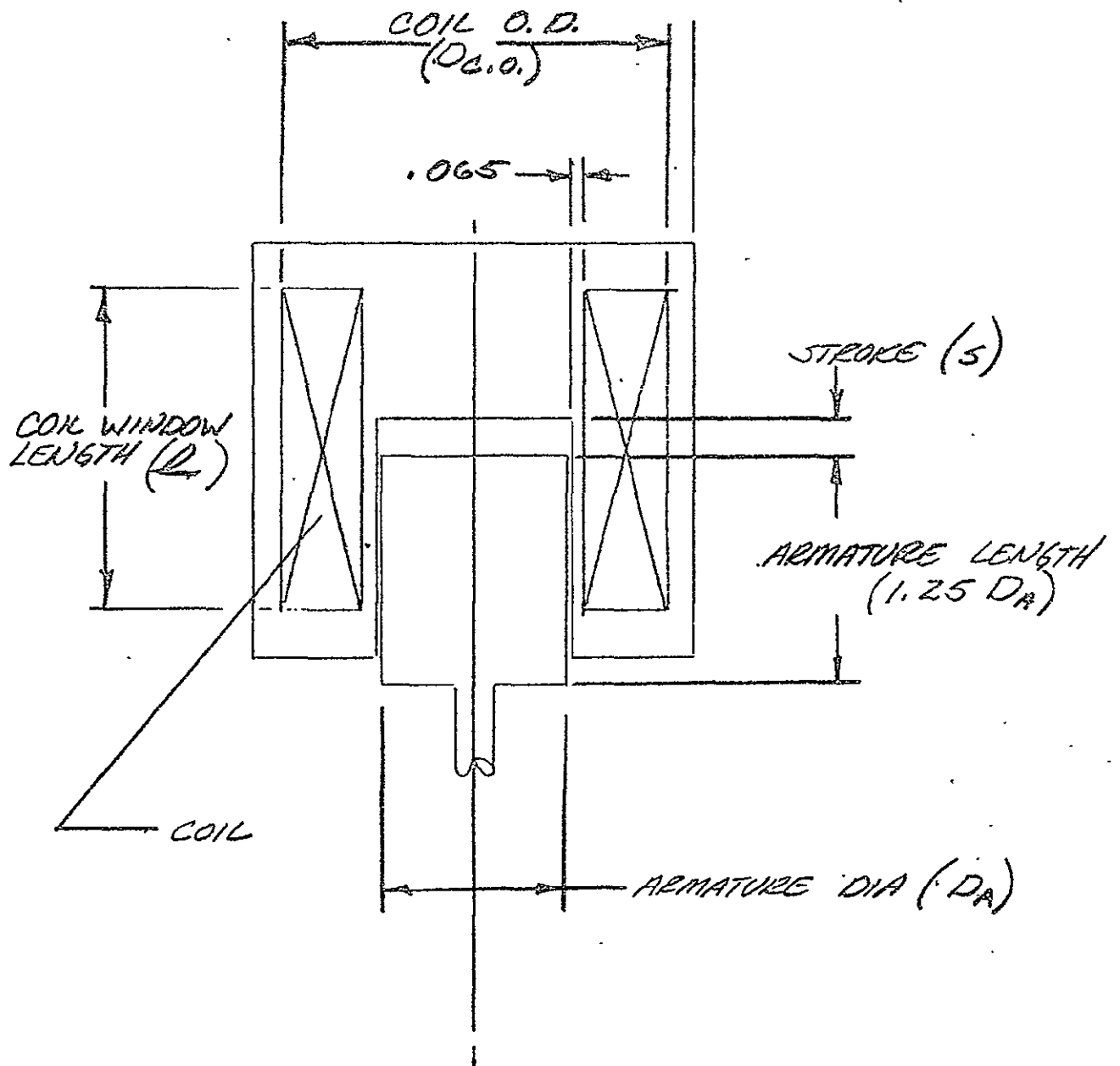
ROTARY SHUT-OFF DEVICE - BALL DIA VS FLOW FACTOR



ROTARY MOTION SHUT-OFF DEVICE - MOVING MASS VS FLOW FACTOR



SOLENOID ACTUATOR - SCHEMATIC DRAWING



Combining equations (16) and (17)

$$\frac{\pi/4 D_A^2 B^2}{1.73} = F_v + \frac{\pi}{4} D_A^3 1.25 \left(\frac{2s}{386.4t^2} + 20 \right) \quad (18)$$

In terms of valve operation, Equation (18) is solved for armature diameter (D_A) for any combination of required valve force (F_v), stroke (s) and allowed motion time (t).

Coil sizing is then estimated for the operating condition by:

$$NI = 1.25 \times 2.02 [H_{\text{iron}} \ell_{\text{iron}} + 10^3 \times B \times (s + .01)] \quad (19)$$

Where NI = ampere turns in coil - ampere turns
H = magneto motive force to drive iron to flux density of B - Oerstad

ℓ_{iron} = length of iron path - inches

B and s are previously defined

A reasonable estimate of the iron path length of $5.5 \times D_A$ is made, and for a 400 series corrosion resistant steel core operating at 8 kilogauss, an H of 10 oerstad is conservative. Equation (19) can then be solved for NI.

For operation at minimum supply voltage and maximum temperature, the pull in current is defined as

$$I_{\text{pull in}} = \frac{18}{R_{\text{c max. temp.}}} \quad (20)$$

Where $I_{\text{pull in}}$ = pull in current - amps
 R_{c} = coil resistance at maximum temperature - ohms

With an operating temperature range of 200°R to 660°R, alloy 30 wire was selected for the coil winding to minimize resistance change with temperature. This will result in more repeatable response over the temperature range. Defining maximum electrical power at ambient temperature (530°R) and 32 V d-c the coil resistance at 530°R is defined as:

$$R_{\text{c 530°R}} = \frac{32^2}{W} \quad (21)$$

where W = maximum electrical power - watts
 $R_{\text{c 530}}$ = coil resistance at 530°R - ohms

For alloy 30 wire the coil resistance at the maximum operating temperature (660°R) is:

$$R_{\text{c 660°R}} = 1.1 \times R_{\text{c 530°R}} \quad (22)$$

and substituting into Equation (19)

$$N_{\text{c}} = \frac{NI}{I_{\text{pull in}}} = \frac{2.02 \times 1.25 [5.5 D_A \times 8000 (s + .01)]}{\frac{18}{1.1 \frac{32^2}{W}}}$$

Where N_c = number of turns in coil windings

The time required to reach the pull in current in an inductive-resistive circuit such as a solenoid can now be defined. The time constant of the circuit can be calculated from the above determined terms as:

$$\left(\frac{L}{R}\right) = \frac{3.19 \times 10^{-3} \frac{\pi}{4} D_A^2 N_c^2}{10^5 (R_c \text{ at temp}) \times 1.25 \text{ s}} \quad (23)$$

Where $\frac{L}{R_c} = \frac{\text{inductance (henries)}}{\text{resistance (ohms)}} = \text{seconds} - \text{time to achieve 63.2\% of steady state current draw}$

Since the current rise-time characteristic of an inductive/resistive circuit is an exponential function of the circuit time constant (L/R) and the steady state current, the time required for the solenoid current to build up to the required pull in current is expressed as:

$$t_1 = - \left(\frac{L}{R_c \text{ at temp}}\right) \ln \left(1 - \frac{I_{\text{pull-in}} \times R_c \text{ at temp}}{E}\right) \quad (24)$$

Where t_1 = time to pull in - seconds
 E = applied voltage - volts d-c

A tabulation of calculated data obtained for 4 flow factors ($C_d A_t$), 4 pressures (P_1), 6 allowable motion times (t) and 4 electrical power levels (W) is shown in Table III- V. It must be noted that significant delay times are predicted for the solenoid actuators for the specific applications analyzed. This delay is the result of the relatively long strokes required, which necessitate large numbers of windings (N_c) to generate sufficient magnetomotive force. Coils with large numbers of turns are characterized by slow rates of current build up, hence, long delays. The magnitude of the delays calculated are generally too long for this application. Therefore, either solenoid actuator strokes must be limited to no greater than .100 inch or an alternate solenoid configuration must be analyzed if reasonably acceptable electrical delays are to be achieved.

III-4B - 2. Pneumatic Cylinder - The performance analysis is based upon a simple double acting cylinder. Cylinder operation is defined, to establish an analytical base, as follows; to open, the opening chamber is pressurized while the closing chamber, which is unpressurized, is vented to ambient; to effect closing, the opening chamber is vented to ambient while the closing chamber is pressurized. Figure III-30 presents a schematic drawing of the pneumatic cylinder actuator and defines specific characteristics employed in the analysis.

The total force developed by the actuator is the sum of the output force (F_v) and the force required to accelerate the moving mass of the actuator. Assuming constant acceleration, consistent with previous analyses, and assuming the required total force is developed with choked flow across the pneumatic supply orifice;

TABLE III-XV (Continued)
SOLENOID ACTUATOR DATA SUMMARY

②

Req'd Force (lb)	Stroke (in)	Req'd Magnet Force (sec)	Total Force Developed (lb)	Magnet Area (in ²)	Actuating Dia (in)	Solenoid Kits, Pkts (lb)	Max Exc. Power (watts)	Ab. Turbul in Coil	Delay to Move (sec)	Flow Rate (in ³ /min)	Inst Recess (in)
Fr	S	t	FA	Am	Ds	MA	W	N/C	t ₁	Q ₁	P ₁
24.33	.276	.10	33.0	.89	1.066	.337	28 56 84 112	13320 6660 4440 3330	1.25 .625 .415 .312	.62	150
35.86		.02	62.0	1.67	1.46	.866	28 56 84 112	13380 6690 4460 3345	2.92 1.46 .973 .730		100
35.84		.03	56.9	1.53	1.398	.760	28 56 84 112	13320 6660 4440 3330	2.36 1.18 .787 .590		
35.84		.05	54.9	1.48	1.375	.724	28 56 84 112	13320 6660 4440 3330	2.13 1.065 .710 .533		
35.84		.07	54.2	1.46	1.366	.710	28 56 84 112	13320 6660 4440 3330	2.08 1.04 .693 .520		
35.84		.10	53.8	1.45	1.360	.700	28 56 84 112	13320 6660 4440 3330	2.04 1.02 .680 .510		
23.52	.339	.02	34.9	.94	1.095	.365	28 56 84 112	15850 7925 5283 3963	1.84 .970 .647 .485	.94	30
23.49		.03	33	.89	1.065	.336	28 56 84 112	15850 7925 5283 3963	1.64 .820 .547 .410		
23.48		.05	31.9	.86	1.047	.319	28 56 84 112	15850 7925 5283 3963	1.50 .750 .500 .375		
23.47		.07	31.9	.86	1.047	.319	28 56 84 112	15850 7925 5283 3963	1.465 .732 .488 .366		
23.47		.10	31.5	.85	1.042	.314	28 56 84 112	15850 7925 5283 3963	1.440 .720 .480 .360		
25.46		.02	39.0	1.05	1.158	.433	28 56 84 112	15890 7945 5297 3973	2.19 1.095 .750 .548		60
25.44		.03	36.4	.98	1.118	.388	28 56 84 112	15890 7945 5297 3923	1.82 .910 .607 .455		

PRECEDING PAGE BLANK NOT FILMED

TABLE III-XV (Continued)

SOLENOID ACTIVATOR DATA SUMMARY

④

Redo Time (hr) (10)	Stroke (in)	Redo Flange (sec) (10)	Total Flange (10)	Pressure Area (in ²) (10)	Pressure Dia (in) (10)	Speed Rate (ft/min) (10)	Max Cur. Power (watts) (10)	No. Turns in Coil (10)	Delay to Arrive (sec) (10)	From Face (in) (10)	From Face (in) (10)
F ₁	S	T	F ₂	A ₁	D ₁	R ₁	W	N ₁	T ₁	F ₂	P ₁
39.18	.554	.02	111	2.99	1.955	2.08	28 56 84 112		13.05 6.53 4.28 3.26	2.5	20
38.97		.03	69.5	1.87	1.545	1.025	28 56 84 112		6.02 3.01 2.01 1.51		
38.86		.05	62.9	1.69	1.470	.885	28 56 84 112		4.76 2.38 1.59 1.19		
38.83		.07	61.4	1.65	1.453	.851	28 56 84 112		4.46 2.23 1.49 1.15		
38.81		.10	60.7	1.63	1.444	.837	28 56 84 112		4.32 2.16 1.44 1.08		
40.47		.03	74.0	1.99	1.595	1.17	28 56 84 112		6.44		30
40.36		.05	66.2	1.78	1.508	.955	28 56 84 112		5.02		
40.33		.07	64.8	1.74	1.492	.925	28 56 84 112		4.72		
40.32		.10	64.0	1.72	1.483	.907	28 56 84 112		4.55		
44.98		.03	89.3	2.40	1.752	1.49	28 56 84 112		7.85 3.93 2.62 1.96		60
44.87		.05	77.8	2.09	1.635	1.22	28 56 84 112		5.92 2.96 1.97 1.48		
44.84		.07	75.5	2.03	1.610	1.16	28 56 84 112		5.50 2.75 1.83 1.38		
44.83		.10	74.4	2.00	1.600	1.14	28 56 84 112		5.31 2.66 1.77 1.33		

DELAY TIMES ARE EXCEED ALLOWABLE RESISTANCE TIME RANGE

PRECEDING PAGE BLANK NOT FILMED

TABLE III-XV (Continued)

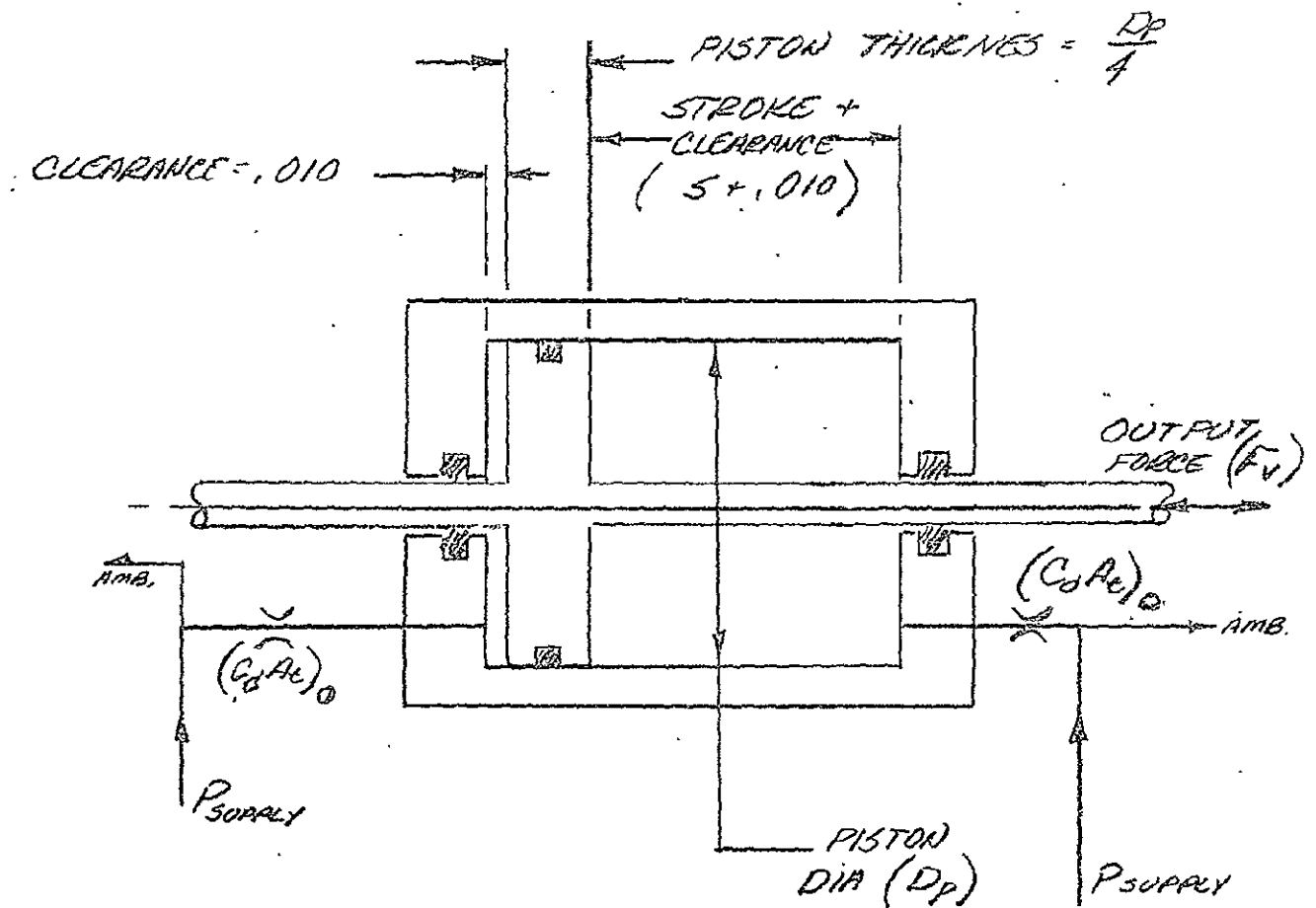
SOLENOID ACTUATOR DATA SUMMARY

③

Rel'd Press (psi)	Stroke (in)	Rel'd Pressure (psi)	Total Force Developed (lb)	Magnetic Area (in ²)	Rel'd Area (in ²)	Solenoid Stroke (in)	Max. Elec. Power (watts)	Ab. Ticks in coil	Delay to Response (sec)	From Rel'd Press	From Rel'd Press
Fr	S	T	FR	AR	DR	MR	W	N	T ₁	Rel'd Press	Rel'd Press
58.41	.554	.05	127.0	3.11	2.09	2.54	28 56 84 112		9.74 4.87 3.25 2.43	2.5	150
58.37		.07	119	3.19	2.02	2.29	28 56 84 112		8.65 4.32 2.88 2.16		
58.36		.01	115.5	3.10	1.99	2.20	28 56 84 112		8.15 4.07 2.72 2.09		
68.27	.89	.10	181	4.85	2.49	4.33	28 56 84 112		20.1 10.05 6.70 5.005	65	20

DELAY TIMES ARE EXCEED
ALLOWABLE RESPONSE TIMES

PNEUMATIC CYLINDER ACTUATOR - SCHEMATIC DRAWING



$$F_A = .52 P_{aux1.} \frac{\pi}{4} D_p^2 = F_v + \rho \frac{\pi}{16} D_p^3 \left(\frac{2s}{386.4t^2} + 20 \right) \quad (26)$$

Where

- F_A = total force developed - lbs
- $P_{aux1.}$ = pneumatic supply pressure - psia
- D_p = piston diameter - in.
- F_v = required output force - lb
- ρ = density of piston material - lb/in.³
- s = stroke - in.
- t = allowable motion time - seconds

This relationship assumes the piston thickness to be $\frac{D_p}{4}$ and assumes that the piston rod areas and masses are negligible. The .52 coefficient of $P_{aux1.}$ is representative of the critical flow pressure ratio for any of the candidate pneumatic gases (hydrogen, oxygen, nitrogen or helium). From this relationship the required piston diameter (D_p) can be calculated for any set of values of the other variables.

For a specific allowed motion time (t), the pneumatic supply orifice must be sized such that the required opening chamber pressure is maintained, resulting in constant force generation. For choked flow through an orifice, the volumetric flow rate is expressed by:

$$Q = \frac{27.82 (C_d A_t)_o P_{supply}}{\rho \frac{P_{aux1.}}{15} \sqrt{RT}} \sqrt{\frac{k}{k-1} \left[\left(\frac{P_o}{P_{aux1.}} \right)^{2/k} - \left(\frac{P_o}{P_{aux1.}} \right)^{\frac{k+1}{k}} \right]} \quad (27)$$

Where

- Q = volumetric flow rate - in.³/sec
- $(C_d A_t)_o$ = flow factor of supply orifice - in.²
- ρ = density of pneumatic media at standard conditions - lb/in.³
- R = gas constant of pneumatic media - in.-lb/lb °R
- T = temperature of gas - °R
- k = specific heat ratio of pneumatic gas
- P_o = pressure down stream of orifice -(cylinder pressure) - psia

Since the cylinder piston moves at constant acceleration the maximum flow rate occurs at the end of the stroke and:

$$Q_{max.} = \frac{\frac{\pi}{4} D_p^2 2s}{t} \quad (28)$$

Combining Equations (27) and (28), the required supply orifice flow factor $(C_d A_t)_o$ can be derived for choked flow

$$(C_d A_t)_o = \frac{\frac{\pi}{4} D_p^2 \times 2s \times \rho \sqrt{RT}}{15 \times t \times 27.82 \sqrt{\frac{k}{k-1} \left[(.52)^{2/k} - (.52)^{\frac{k+1}{k}} \right]}} \quad (29)$$

With the cylinder piston in the valve closed position and assuming an initial opening chamber volume equal to that of .010 head end piston clearance, the time required to charge this volume to the pressure necessary to initiate motion (required output force), can be determined. Since relatively short times are anticipated, adiabatic charging of this volume is assumed and:

$$t_1 = \frac{\frac{\pi}{4} D_p^2 \cdot .010}{k C_1 (C_d A_t)_o R \sqrt{T}} \cdot \frac{P}{P_{auxl.}} \quad (30)$$

where t = time to charge to required pressure - sec
 C_1 = constant of pneumatic media
 P = required cylinder pressure to initiate motion =
 $.52 P_{auxl.}$ - psia

Substituting values derived in Equations (26) and (29) into Equation (30)

$$t_1 = \frac{.2782t \left[F_v + \left(\frac{2s}{386.4t^2} + 20 \right) \frac{\pi}{4} \frac{D_p^3}{4} \right] \sqrt{\frac{k-1}{k} \left[(.52)^{2/k} - (.52)^{\frac{k+1}{k}} \right]}}{k C_1 \frac{\pi}{4} D_p^2 s \rho \sqrt{RT}} \quad (31)$$

For the candidate pneumatic gases, the various constants and properties are tabulated as follows:

Gas	k	R $\frac{\text{in.-lb}}{\text{lb } ^\circ\text{R}}$	C_1	ρ - lb/in. ³
Hydrogen	1.40	9190	.141	3.06×10^{-6}
Oxygen	1.40	577	.560	48.6×10^{-6}
Nitrogen	1.40	662	.523	42.8×10^{-6}
Helium	1.66	4630	.210	5.95×10^{-6}

The determination of the closing pneumatic delay is a more complex task, since simultaneous venting and charging of the cylinder volumes is occurring. The following performance analysis is based upon having equal supply orifice flow factors $(C_d A_t)_o$ for the two cylinder volumes. Also, it is assumed that piston-head clearance is .010 inch. Since the piston diameter (D_p) and supply orifice flow factor $(C_d A_t)_o$ were derived during the opening analysis the force balance at incipient closing motion is expressed by:

$$F_A = \frac{\pi}{4} D_p^2 (P_c - P_o) = F_{vc} + \frac{\pi}{4} \rho \frac{D_p^3}{4} \left(20 + \frac{2s}{386.4 t^2} \right) \quad (32)$$

where P_c = pressure on closing side of piston - psia
 P_o = pressure on opening side of piston - psia
 F_{vc} = required output force to effect closing - lb
 t = allowed motion time for closing - seconds

The required output force to effect closing is derived for the particular application and the pressure differential across the cylinder piston ($P_c - P_o$) can be calculated. Since the charging and venting of the cylinder volumes is simultaneous, the time to charge the closing volume to P_c is equal to the time to vent the opening volume to P_o . Assuming adiabatic charging and venting:

$$t_2 = \frac{\frac{\pi}{4} D_p^2 (s + .010)}{k C_1 (C_d A_t)_o R \sqrt{T}^{\frac{k-1}{2k}}} \left[\left(\frac{P_{supply}}{P_o} \right)^{\frac{k-1}{2k}} - 1 \right] =$$

$$\frac{\frac{\pi}{4} D_p^2 (.010)}{k C_1 (C_d A_t)_o R \sqrt{T}} \frac{P_c}{P_{supply}} \quad (33)$$

Where all terms are as previously defined. From Equation (32) the total force developed (F_A) is known and

$$P_c = P_o + \frac{F_A}{\frac{\pi}{4} D_p^2}$$

Substituting this into Equation 33 and combining terms:

$$\frac{s + .010}{P_{supply}} \left(\frac{F_A}{\frac{\pi}{4} D_p^2} + P_o \right) = \frac{.010}{\frac{k-1}{2k}} \left[\left(\frac{P_{supply}}{P_o} \right)^{\frac{k-1}{2k}} - 1 \right] \quad (34)$$

This relationship can be solved for P_o , for any set of variables for the application. The derived P_o is substituted into Equation (33) and the closing delay time (t_2) solved. The validity of the solution must be checked by solving for P_c and establishing that choked flow at the closing chamber supply orifice exists at the time of incipient closing motion $\left(\frac{P_c}{P_{aux1}} \leq .52 \right)$. If choked flow is not verified a more complex analysis

is required which considers critical and subcritical flow. For the sample data points of this program critical flow was verified and the development of the more complex analysis method was not undertaken.

For the 7 flow factors ($C_d A_t$), 6 allowable motion times (t), and 5 inlet pressures (P_1) a tabulation of pneumatic cylinder actuator data for the linear motion shut off device is presented in Table III-XVI. Data was calculated for pneumatic pressures of 1500 psia and 3000 psia as well as the shut off device inlet pressures, for a pneumatic media of oxygen. Figures III-31 through III-36 present design curves for determining piston diameters required as a function of required output force (F_v), pneumatic pressure (P_{aux1}) and required acceleration ($2s/386.4 t^2$), and Figure III-37 is a design curve for determining opening pneumatic delay (t_1) as a function of acceleration $\left(\frac{2s}{386.4 t^2} \right)$ and allowable motion time (t). Table III-XVII gives correction factors for the pneumatic opening delay time (t_1) for the pneumatic media used.

TABLE III-XVI
Pneumatic Cylinder Actuator Data Summary

①

Req'd Flow Rate (L/B)	Stroke (in)	Motion Time (sec)	Air SAS	Flow Rate (L/min)	Total Flow Rate (L/min)	Port Dn (in)	Supply Pressure (in H ₂ O)	Supply Flow Rate (L/min)	Supply Pressure (in H ₂ O)	Supply Flow Rate (L/min)	Supply Pressure (in H ₂ O)	Supply Flow Rate (L/min)	Supply Pressure (in H ₂ O)
20.26	.276	.01	02	60	21.86	.945	.00577	.00577	.00577	.00577	.00577	.00577	.00577
				1500	20.27	.182	.00019	.00019	.00019	.00019	.00019	.00019	.00019
				3000	20.26	.129	.00009	.00009	.00009	.00009	.00009	.00009	.00009
20.20		.02		60	21.26	.934	.0025	.0025	.0025	.0025	.0025	.0025	.0025
				1500	20.21	.182	.000095	.000095	.000095	.000095	.000095	.000095	.000095
				3000	20.20	.129	.000048	.000048	.000048	.000048	.000048	.000048	.000048
20.19		.03		60	21.15	.929	.00167	.00167	.00167	.00167	.00167	.00167	.00167
				1500	20.20	.182	.000063	.000063	.000063	.000063	.000063	.000063	.000063
				3000	20.19	.129	.000031	.000031	.000031	.000031	.000031	.000031	.000031
20.18		.05		60	21.10	.929	.00099	.00099	.00099	.00099	.00099	.00099	.00099
				1500	20.19	.182	.000038	.000038	.000038	.000038	.000038	.000038	.000038
				3000	20.19	.129	.000019	.000019	.000019	.000019	.000019	.000019	.000019
20.18		.07		60	21.08	.929	.00071	.00071	.00071	.00071	.00071	.00071	.00071
				1500	20.19	.182	.000027	.000027	.000027	.000027	.000027	.000027	.000027
				3000	20.18	.129	.000014	.000014	.000014	.000014	.000014	.000014	.000014
20.18		.10		60	21.08	.929	.00099	.00099	.00099	.00099	.00099	.00099	.00099
				1500	20.19	.182	.000019	.000019	.000019	.000019	.000019	.000019	.000019
				3000	20.18	.129	.0000095	.0000095	.0000095	.0000095	.0000095	.0000095	.0000095
24.40		.01		150	24.40	.638	.0023	.0023	.0023	.0023	.0023	.0023	.0023
				1500	24.42	.201	.00023	.00023	.00023	.00023	.00023	.00023	.00023
				3000	24.41	.142	.00012	.00012	.00012	.00012	.00012	.00012	.00012
24.39		.02		150	24.68	.638	.0012	.0012	.0012	.0012	.0012	.0012	.0012
				1500	24.35	.201	.00012	.00012	.00012	.00012	.00012	.00012	.00012
				3000	24.35	.142	.00006	.00006	.00006	.00006	.00006	.00006	.00006
24.33		.03		150	24.64	.638	.00077	.00077	.00077	.00077	.00077	.00077	.00077
				3000	24.34	.201	.000077	.000077	.000077	.000077	.000077	.000077	.000077
				1500	24.34	.142	.000038	.000038	.000038	.000038	.000038	.000038	.000038
24.33		.05		150	24.62	.638	.00047	.00047	.00047	.00047	.00047	.00047	.00047
				1500	24.34	.201	.000046	.000046	.000046	.000046	.000046	.000046	.000046
				3000	24.33	.142	.000023	.000023	.000023	.000023	.000023	.000023	.000023
24.33		.07		150	24.61	.638	.00033	.00033	.00033	.00033	.00033	.00033	.00033
				1500	24.34	.201	.000033	.000033	.000033	.000033	.000033	.000033	.000033
				3000	24.33	.142	.000016	.000016	.000016	.000016	.000016	.000016	.000016
24.33		.10		150	24.61	.638	.00023	.00023	.00023	.00023	.00023	.00023	.00023
				1500	24.34	.201	.000023	.000023	.000023	.000023	.000023	.000023	.000023
				3000	24.33	.142	.000011	.000011	.000011	.000011	.000011	.000011	.000011
35.91		.01		400	36.11	.471	.00128	.00128	.00128	.00128	.00128	.00128	.00128
				1500	35.94	.242	.00034	.00034	.00034	.00034	.00034	.00034	.00034
				3000	35.92	.171	.00017	.00017	.00017	.00017	.00017	.00017	.00017
35.86		.02		400	35.99	.470	.00064	.00064	.00064	.00064	.00064	.00064	.00064
				1500	35.87	.242	.00017	.00017	.00017	.00017	.00017	.00017	.00017
				3000	35.86	.171	.00008	.00008	.00008	.00008	.00008	.00008	.00008
35.84		.03		400	35.97	.469	.00043	.00043	.00043	.00043	.00043	.00043	.00043
				1500	35.86	.242	.00011	.00011	.00011	.00011	.00011	.00011	.00011
				3000	35.85	.171	.000056	.000056	.000056	.000056	.000056	.000056	.000056
35.84		.05		400	35.96	.469	.00026	.00026	.00026	.00026	.00026	.00026	.00026
				1500	35.86	.242	.000068	.000068	.000068	.000068	.000068	.000068	.000068
				3000	35.84	.171	.000034	.000034	.000034	.000034	.000034	.000034	.000034

TABLE III-XVI (Continued)

Pneumatic Cylinder Retarding Data Summary

Retard Force Curve (LBS)	Stroke (in)	Normal Time (sec)	Pressure Gas	Actual Gas Rate (SCFH)	Time Retard (sec)	Actual Dia (in)	Stroke (in)	Retard Force (LBS)	Retard Time (sec)	Value in Factor (in %)	Value in Factor (in %)
Fv	S	E		PAUX	FA	DA	(LBS/IN)	FA	FA	FA	FA
35.89	.276	.07	02	100	35.95	.469	.00018	.07	.07	.62	100
				1500	35.85	.292	.00005				
				3000	35.89	.171	.000029				
35.89		.10		100	35.95	.469	.00013	.068	.05		
				1500	35.85	.292	.00034				
				3000	35.89	.171	.000017				
23.66	.389	.01		30	—	—	—	—	—	.94	30
				1500	23.67	.197	.00028	.005	.006		
				3000	23.66	.1395	.00014				
23.52		.02		30	—	—	—	—	—		
				1500	23.53	.196	.00014	.01	.01		
				3000	23.52	.139	.00068				
23.49		.03		30	—	—	—	—	—		
				1500	23.50	.196	.00011	.016	.017		
				3000	23.49	.139	.000045				
23.48		.05		30	—	—	—	—	—		
				1500	23.49	.196	.00054	.027	.029		
				3000	23.48	.139	.000027				
23.47		.07		30	—	—	—	—	—		
				1500	23.48	.196	.00039	.038	.040		
				3000	23.48	.139	.000019				
23.47		.10		30	—	—	—	—	—		
				1500	23.48	.196	.000027	.035	.038		
				3000	23.47	.139	.000013				
25.60		.01		60	—	—	—	—	—		60
				1500	25.62	.205	.000299	.029	.032		
				3000	25.61	.145	.000149				
25.46		.02		60	27.03	1.05	.0039	.010	.013		
				1500	25.47	.209	.000148		.011		
				3000	25.47	.144	.000074		.011		
25.49		.03		60	26.83	1.097	.0036	.016	.019		
				1500	25.45	.209	.000099		.017		
				3000	25.44	.144	.000049		.017		
25.42		.05		60	26.73	1.045	.00156	.027	.031		
				1500	25.43	.209	.000059		.029		
				3000	25.43	.144	.000029		.029		
25.42		.07		60	26.71	1.095	.0011	.028	.034		
				1500	25.43	.209	.000042		.040		
				3000	25.42	.144	.000021		.040		
25.42		.10		60	26.69	1.045	.00077	.035	.043		
				1500	25.43	.209	.000029		.058		
				3000	25.43	.144	.000014		.058		
31.44		.01		150	31.23	.727	.00375	.005	.009		150
				1500	31.46	.227	.000281		.005		
				3000	31.45	.161	.000115		.005		
31.30		.02		150	31.8	.722	.00185	.011	.017		
				1500	31.31	.226	.000182		.015		
				3000	31.30	.161	.000091		.015		

TABLE III-XVI (Continued)
Pneumatic Cylinder Actuator Data Summary

③

Req'd Force (lb)	Stroke (in)	Normal Time (sec)	Actu Gas	Actu Gas (psi)	Total Force (lb)	Actu Dia (in)	Stroke Face (in ²)	Delay to Move in (sec)	Value in Face (in ²)	Value Force (psi)
Fv	S	t		Pv	Fa	Dp	(Cp A)	t ₁ t ₂	Cp A	P
31.27	.339	.03	O ₂	150	31.23	.721	.00123	.016 .018	.94	150
				1500	31.28	.526	.000121	↓ .017		
				3000	31.28	.160	.000061	↓ .017		
31.26		.05		150	31.69	.721	.000733	.027 .029		
				1500	31.27	.226	.000023	↓ ↓		
				3000	31.26	.160	.000036	↓ ↓		
31.25		.07		150	31.67	.721	.000507	.028 .041		
				1500	31.27	.226	.000052	↓ .040		
				3000	31.26	.160	.000026	↓ .040		
31.25		.10		150	31.67	.721	.000369	.055 .059		
				1500	31.26	.226	.000056	↓ .058		
				3000	31.26	.160	.000018	↓ .058		
47.65		.01		400	47.98	.592	.002096	.005 .008		400
				1500	47.69	.279	.000036	↓ ↓		
				3000	47.66	.198	.0000278	↓ ↓		
47.50		.02		400	47.70	.592	.00104	.011 .012		
				1500	47.53	.279	.000077	↓ ↓		
				3000	47.51	.198	.000038	↓ ↓		
47.48		.03		400	47.67	.592	.000694	.016 .017		
				1500	47.50	.279	.000084	↓ ↓		
				3000	47.49	.198	.000092	↓ ↓		
47.47		.05		400	47.65	.592	.000416	.027 .029		
				1500	47.49	.279	.000011	↓ ↓		
				3000	47.47	.198	.000053	↓ ↓		
47.46		.07		400	47.64	.592	.000297	.028 .040		
				1500	47.48	.279	.000079	↓ ↓		
				3000	47.47	.198	.000039	↓ ↓		
47.46		.10		400	47.64	.592	.000208	.055 .058		
				1500	47.48	.279	.000055	↓ ↓		
				3000	47.47	.198	.000028	↓ ↓		
40.35	.554	.01		20	—	—	—	—	2.50	20
				1500	40.39	.257	.000767	.039 .077		
				3000	40.36	.182	.000383	↓ .072		
39.18		.02		20	—	—	—	—		
				1500	39.21	.253	.000322	.047 .045		
				3000	39.19	.179	.000186	↓ ↓		
38.97		.03		20	—	—	—	—		
				1500	38.99	.252	.000246	.040 .023		
				3000	38.98	.178	.000123	↓ ↓		
38.86		.05		20	—	—	—	—		
				1500	38.88	.252	.000147	.047 .038		
				3000	38.87	.178	.000074	↓ ↓		
38.83		.07		20	—	—	—	—		
				1500	38.85	.252	.000105	.044 .053		
				3000	38.83	.178	.000053	↓ ↓		
38.81				20	—	—	—	—		
				1500	38.83	.252	.000074	.034 .040		
				3000	38.82	.178	.000037	↓ ↓		

TABLE III-XVI (Continued)
PNEUMATIC CYLINDER ACTUATOR DATA SUMMARY

APPROX FIRE QUANTITY (LB)	STROKE (IN)	APPROX TIME (SEC)	PNEU GAS	PNEU GAS PRESS (PSIA)	TOTAL FIRE QUANTITY (LB)	PNEU DIA (IN)	SEALING FACILITY (IN ²)	DEACTIVATION TIME (SEC)	VALVE FACILITY (IN ²)	VALVE PRESSURE (PSIA)
FV	S	T		PAVS	FA	DP	(C ₁ A ₁)	T ₁ T ₂	C ₂ A ₂	P
41.85	.554	.01	O ₂	30 1500 3000	41.90 41.87	.262 .185	.000796 .000598	.0004 b	.0007 .0006	2.50 30
40.69		.02		30 1500 3000	— 40.71 40.70	— .258 .183	— .000387 .000193	— 1067 b	— .015 b	
40.47		.03		30 1500 3000	— 40.49 40.48	— .257 .182	— .000250 .000128	— .010 b	— .023 b	
40.36		.05		30 1500 3000	— 40.38 40.37	— .257 .182	— .000153 .000077	— .017 b	— .035 b	
40.33		.07		30 1500 3000	— 40.35 40.34	— .257 .192	— .000109 .000055	— .024 b	— .023 b	
40.32		.10		30 1500 3000	— 40.33 40.32	— .257 .182	— .000077 .000038	— .034 b	— .016 b	
46.36		.01		60 1500 3000	— 46.42 46.38	— .275 .195	— .000882 .000491	— 1034 b	— .007 b	60
45.20		.02		60 1500 3000	— 45.23 45.21	— .272 .193	— .000429 .000215	— 1067 b	— .015 b	
44.98		.03		60 1500 3000	— 44.58 45.01 44.94	— 1.41 .271 .192	— .0007691 .000285 .000142	— .010 b	— .034 .023 .023	
44.87		.05		60 1500 3000	— 44.10 44.90 44.88	— 1.405 .271 .192	— .000457 .000171 .000085	— .017 b	— .034 .035 .034	
44.84		.07		60 1500 3000	— 47.48 44.87 44.85	— 1.400 .271 .192	— .0003253 .000122 .000061	— .029 b	— .075 .053 .053	
44.83		.10		60 1500 3000	— 47.91 44.85 44.83	— 1.400 .271 .192	— .000075 .000085 .000043	— .034 b	— .076 .076	
58.89		.01		150 1500 3000	62.69 59.48 59.92	1.014 .313 .222	.01191 .001139 .000509	.0034 b	.0003 .0077 .0077	150
58.73		.02		150 1500 3000	60.20 58.78 58.75	.994 .310 .219	.005719 .000583 .000279	.047 b	.017 .015 .015	
58.52		.03		150 1500 3000	59.76 58.53 58.53	.990 .309 .218	.003784 .000571 .000185	.010 b	.021 .023 .023	
58.41		.05		150 1500 3000	59.53 58.44 58.42	.989 .309 .218	.002262 .000222 .000111	.017 b	.041 .031 .031	

TABLE III-XVI (Continued)

Pneumatic Cylinder Accuracy Data Summary

Req'd Force Class (LB)	Stroke (in)	Portion Time (SEC)	Actg Gas	Actg Gas Press (PSIA)	Total Force Class (LB)	Portion Dia (in)	Sealant Force (in ²)	Dealt to Airline in (SEC)	Valve Fin. Force (in ²)	Valve Force (PSIA)
Fv	S	t		PAV	FA	Da	(C _d A _{th})	t ₁₀	C _d A _{th}	R
80.37	.893	.01	O ₂	20 1500 3000	80.54 80.43	.363 .257	.002467 .001232	.0031 ↓ .0016	6.50	20
71.20		.02		20 1500 3000	71.27 71.23	.341 .242	.001082 .000546	.0012 ↓ .0009		
69.51		.03		20 1500 3000	69.56 69.52	.337 .239	.000710 .000353	.0008 ↓ ↓		
68.64		.05		20 1500 3000	68.68 68.65	.335 .237	.000720 .000710	.010 ↓ ↓		
68.40		.07		20 1500 3000	68.44 68.41	.334 .236	.000589 .000149	.015 ↓ ↓		
68.27		.10		20 1500 3000	68.31 68.29	.334 .236	.000209 .000105	.021 ↓ ↓		
83.93		.01		20 1500 3000	84.12 84.00	.371 .263	.002577 .001287	.0021 ↓ ↓		30
74.77		.02		30 1500 3000	74.84 74.79	.350 .248	.001466 .000522	.0041 ↓ ↓		
73.07		.03		30 1500 3000	73.13 73.09	.346 .245	.000747 .000373	.0063 ↓ ↓		
72.20		.05		30 1500 3000	72.25 72.20	.343 .243	.000443 .000221	.010 ↓ ↓		
71.96		.07		30 1500 3000	72.01 71.98	.343 .243	.000315 .000157	.015 ↓ ↓		
71.84		.10		30 1500 3000	71.88 71.85	.343 .243	.000220 .000110	.021 ↓ ↓		
94.62		.01		60 1500 3000	94.85 94.70	.293 .278	.002906 .001451	.0024 ↓ ↓		60
85.46		.02		60 1500 3000	85.53 85.49	.374 .265	.001310 .000653	.004 ↓ ↓		
83.76		.03		60 1500 3000	83.83 83.79	.370 .262	.000856 .000427	.006 ↓ ↓		
82.90		.05		60 1500 3000	82.96 82.92	.368 .261	.000508 .000254	.010 ↓ ↓		
82.66		.07		60 1500 3000	82.71 82.68	.368 .260	.000362 .000181	.014 ↓ ↓		
82.53		.10		60 1500 3000	82.59 82.53	.367 .260	.000253 .000127	.021 ↓ ↓		

TABLE III-XVI (Continued)
PNEUMATIC CYLINDER ACTUATING DATA SUMMARY

⑥

Fig'd Face Area (in ²)	Stroke (in)	Porting Time (sec)	Press GAS	Press Gas Press (PSIA)	Temp Air/Die (°C)	Porting Dia (in)	Survey Dev Factor (in ²)	Deviation Factor (SEC)	Value from Factor (in ²)	Value from Factor (PSIA)
Fv	S	E		PSIA	Fa	Dp	(C ₁ R ₁)	C ₁ C ₂	C ₁ R ₁	R
121.27	1.107	.01	02	20 1500 3000	121.65 121.40	.445 .315	.004622 .002307	.0016 b	.013 .011	10
98.04		.02		20 1500 3000	98.16 98.08	.400 .283	.001865 .000931	.0053 b	.023 .022	
93.73		.03		20 1500 3000	93.82 93.76	.391 .277	.001189 .000594	.005 b	.031 .032	
91.53		.05		20 1500 3000	91.60 91.56	.387 .274	.000696 .000348	.0084 b	.055 .059	
90.92		.07		20 1500 3000	90.99 90.95	.385 .273	.000494 .002247	.0117 b	.077 .025	
90.60		.10		20 1500 3000	90.67 90.62	.385 .273	.000344 .000172	.017 b	.109 .107	
126.59		.01		30 1500 3000	126.99 126.73	.455 .322	.004825 .002408	.0017 b	.013 .011	30
103.35		.02		30 1500 3000	103.48 103.40	.411 .291	.001966 .000912	.0084 b	.023 .022	
99.05		.03		30 1500 3000	99.14 99.08	.402 .285	.001256 .000627	.005 b	.033 .032	
96.85		.05		30 1500 3000	96.92 96.87	.398 .282	.000737 .000368	.0084 b	.055 .059	
96.24		.07		30 1500 3000	96.31 96.27	.396 .280	.000523 .000261	.012 b	.077 .025	
95.92		.10		30 1500 3000	95.99 95.94	.396 .280	.000365 .000182	.0168 b	.109 .107	
142.54		.01		60 1500 3000	143.02 142.71	.484 .343	.005934 .002711	.0017 b	.013 .011	60
119.30		.02		60 1500 3000	119.47 119.36	.442 .313	.002270 .001134	.0033 b	.023 .022	
115.00		.03		60 1500 3000	115.12 115.04	.434 .308	.001458 .000728	.005 b	.033 .032	
112.80		.05		60 1500 3000	112.89 112.83	.429 .304	.000858 .000429	.008 b	.055 .059	
112.19		.07		60 1500 3000	112.28 112.22	.428 .303	.000609 .000305	.012 b	.077 .025	
111.87		.10		60 1500 3000	111.96 111.90	.427 .302	.000425 .000213	.017 b	.109 .107	

PERMITS CYCLES ACTUARY DATA SUMMARY

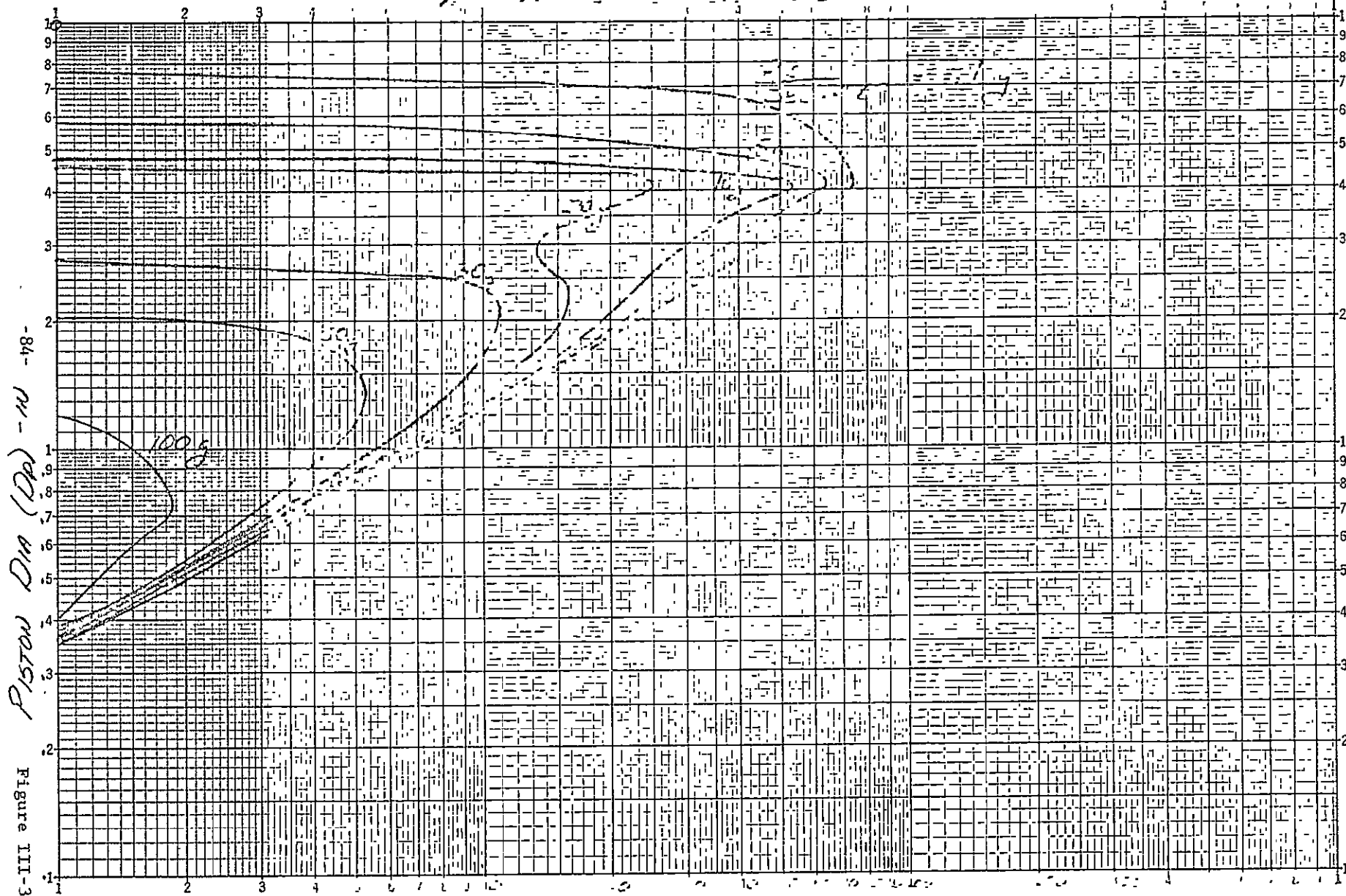
REG'D FIRE QTY (LBS)	STROKE (IN)	PROTON TIME (SEC)	ALSO GAS	PROV GAS (LBS)	TEMP FIRE QTY (LBS)	PROTON DIA (IN)	STROKE FIRE QTY (LBS)	DECAY FIRE QTY (LBS)	VALUE FIRE QTY (LBS)	VALUE FIRE QTY (LBS)
FV	S	E	PROV	FA	DP	(G. R.)	(G. R.)	(G. R.)	(G. R.)	(G. R.)
405.71	1.250	.01	O ₂	20 1500 3000	409.06 406.89	.817 .579	.02458 .01222	.001 ↓	.122 .048	20
237.57		.02		20 1500 3000	238.14 237.77	.623 .441	.007154 .003571	.002 ↓	.064 .035	
206.43		.03		20 1500 3000	206.76 206.54	.580 .411	.004141 .002068	.003 ↓	.041 .016	
190.48		.05		20 1500 3000	190.71 190.57	.538 .396	.002292 .001146	.005 ↓	.091 .079	
186.09		.07		20 1500 3000	186.30 186.16	.552 .390	.001599 .000799	.007 ↓	.123 .082	
183.76		.10		20 1500 3000	183.95 183.83	.548 .388	.001105 .000552	.018 ↓	.123 .144	
418.36		.01		30 1500 3000	421.87 419.59	.829 .587	.02535 .01260	.001 ↓	.122 .048	30
250.22		.02		30 1500 3000	250.84 250.43	.639 .452	.007535 .003761	.002 ↓	.009 .035	
219.08		.03		30 1500 3000	219.44 219.20	.598 .423	.004395 .002195	.003 ↓	.065 .047	
203.14		.05		30 1500 3000	203.39 203.22	.576 .408	.002444 .001221	.005 ↓	.092 .072	
198.74		.07		30 1500 3000	198.97 198.82	.570 .403	.001708 .000853	.007 ↓	.123 .102	
196.41		.10		30 1500 3000	196.62 196.48	.567 .402	.001181 .000590	.010 ↓	.173 .145	
745.39	2.10	.01		20 1500 3000	755.18 748.81	1.110 .785	.05445 .02698	.001 ↓	.353 .142	36
375.78		.02		20 1500 3000	377.05 376.23	.784 .539	.01359 .006781	.002 ↓	.219 .085	20
307.34		.03		20 1500 3000	307.97 307.56	.708 .501	.007401 .003696	.003 ↓	.116 .077	
272.24		.05		20 1500 3000	272.68 272.43	.666 .472	.003932 .001964	.004 ↓	.216 .102	
262.64		.07		20 1500 3000	262.94 262.76	.654 .463	.002709 .001359	.006 ↓	.262 .136	
257.51		.10		20 1500 3000	257.83 257.62	.648 .459	.001859 .000928	.008 ↓	.340 .184	

TABLE III-XVI (Continued)
PNEUMATIC CYLINDER ACTUATING DATA SUMMARY

①

Reg'd Force Curve (L.B.)	Stroke (in)	Normal Time (sec)	Press Gas	Reg'd Gas Press (PSIA)	Total Force Delay (L.B.)	Portion Delay (in)	Safety Valve Pressure (PSI)	Delay to Action at P ₁ (sec)	Value from Pressure (PSIA)	Value from Pressure (PSIA)
FV	S	E	GE	PAUX	FA	DO	(C ₁ A ₁)	T ₁ T ₂	C ₁ A ₁	A
763.35	2.10	.01	02	30 1500 3000	772.49 766.89	1.129 .796	05577 00764	.001 ↓ .162	30	30
393.73		.02		30 1500 3000	395.09 394.21	.802 .568	.01424 .007105	.002 ↓ .080		
325.29		.03		30 1500 3000	325.98 325.53	.728 .516	.007434 .003912	.003 ↓ .077		
290.24		.05		30 1500 3000	290.68 290.40	.688 .488	.009191 .002094	.004 ↓ .102		
280.59		.07		30 1500 3000	280.97 280.72	.678 .474	.002844 .001446	.006 ↓ .130		
275.46		.10		30 1500 3000	275.81 275.58	.671 .475	.001988 .000993	.009 ↓ .189		

CYLINDER OUTPUT FORCE (FV) vs PISTON DIA (Dp) $P_{in} = P_1 = 30 \text{ PSI}$



OUTPUT FORCE (FV) - LBS @ .52 P₁

-84-

PISTON DIA (Dp) - IN

Figure III-31

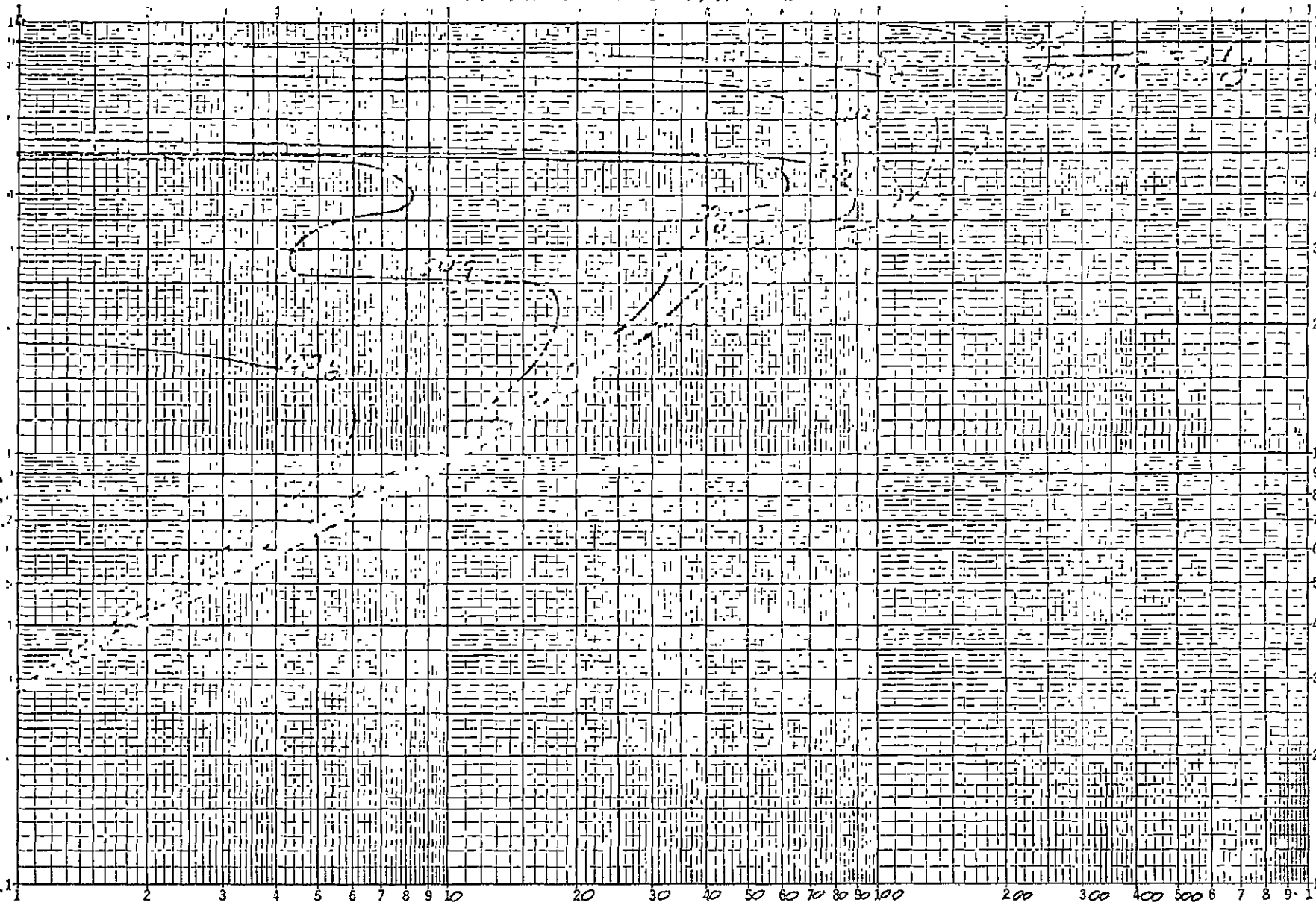
K&E LOGARITHMIC 46 7323
2 X 3 CYCLES
MADE IN U.S.A.
KEUFFEL & ESSER CO.

CHAMBER DIAMETER (Dp) IN vs. OUTPUT FORCE (Fv) LBS

Pressure = 30 PSI

- 85 -
Piston Diameter (Dp) IN

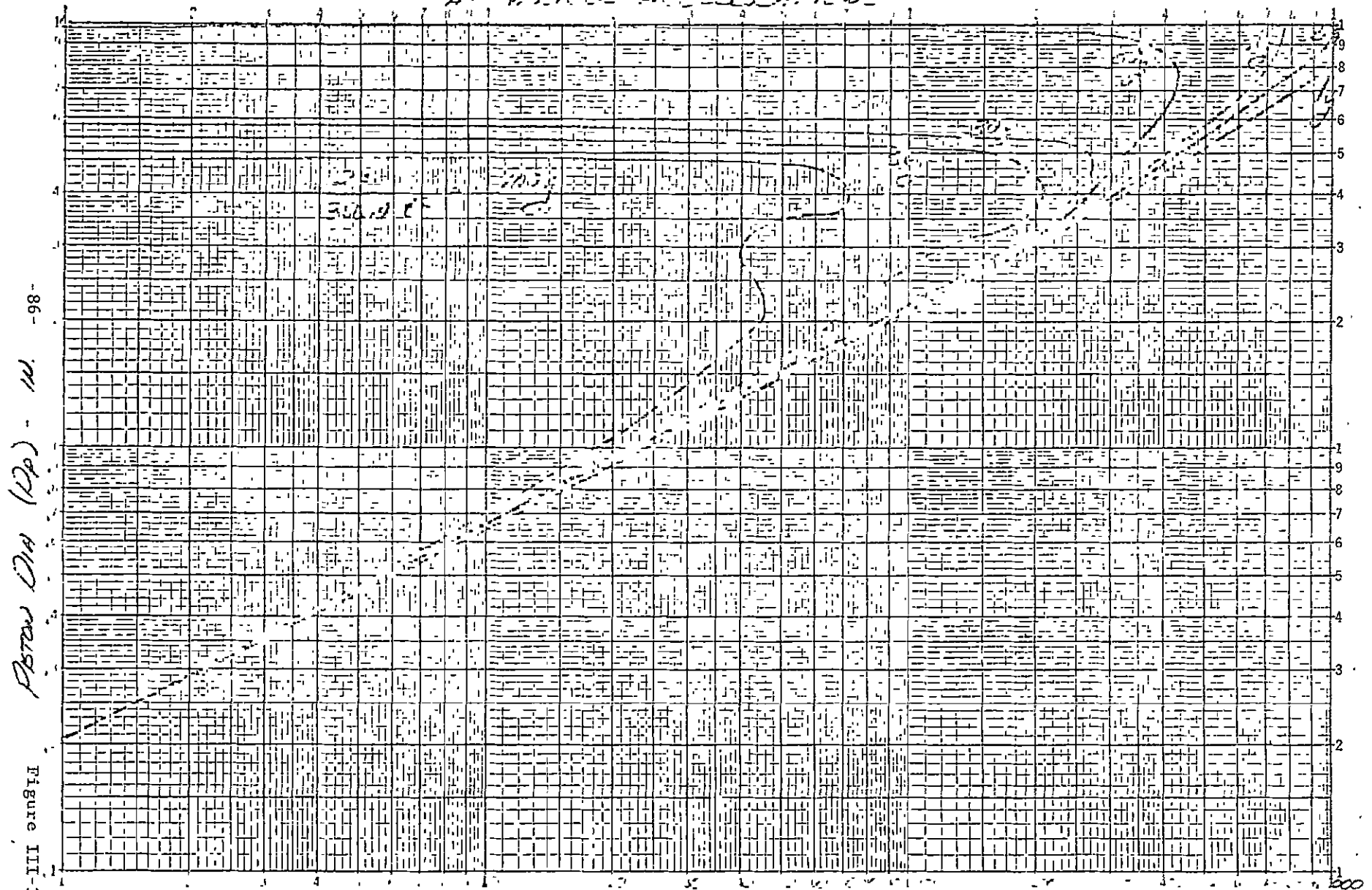
Figure III-32



OUTPUT FORCE (Fv) LBS

K&E LOGARITHMIC 46 7323
2 X 3 CYCLES
MADE IN U.S.A.
KEUFFEL & ESSER CO.

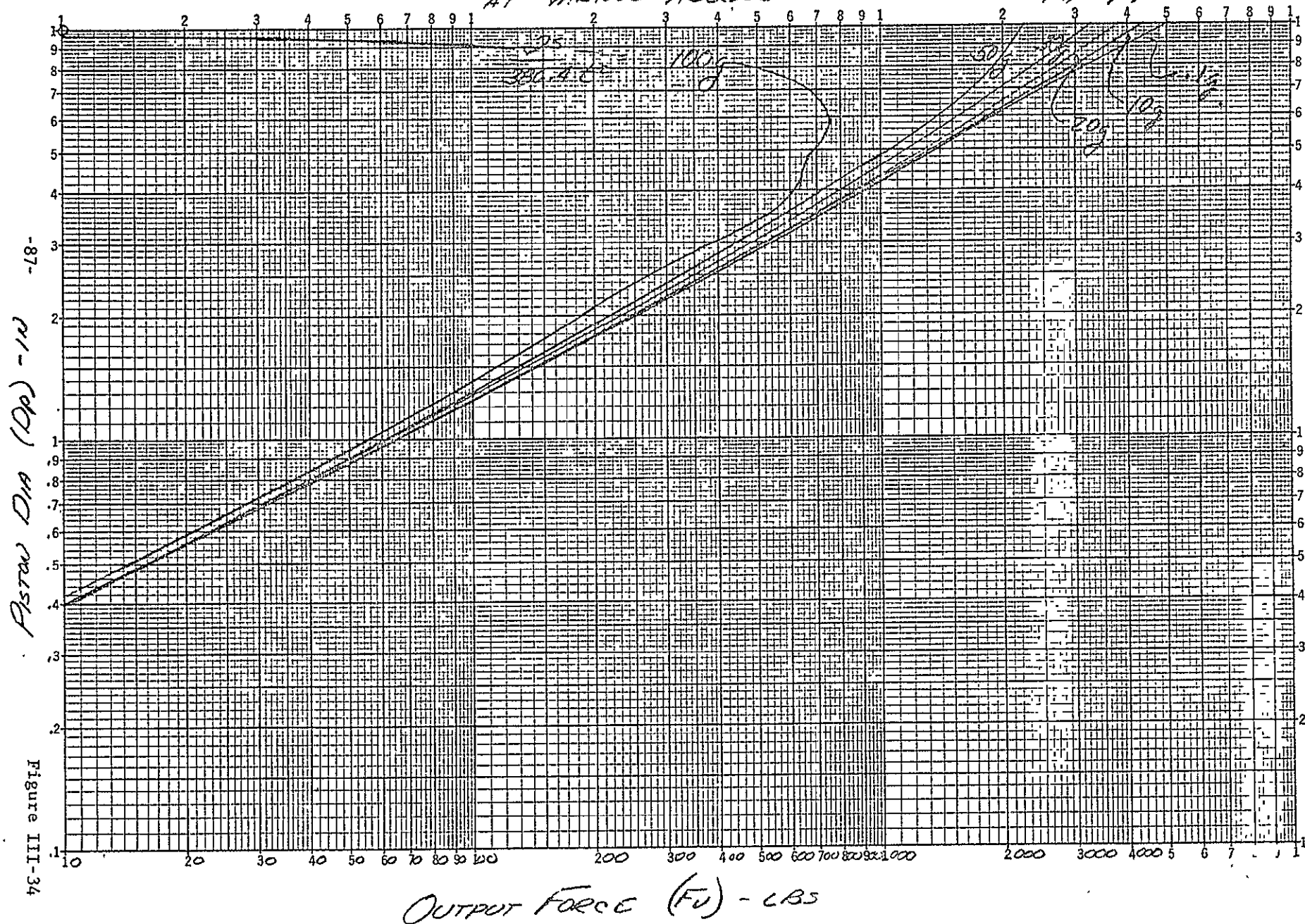
CYLINDER OUTPUT FORCE (F_v) VS PISTON DIA (D_p) $P_{in} = P_1 = 60 \text{ PSIA}$
AT VARIOUS DISPLACEMENTS



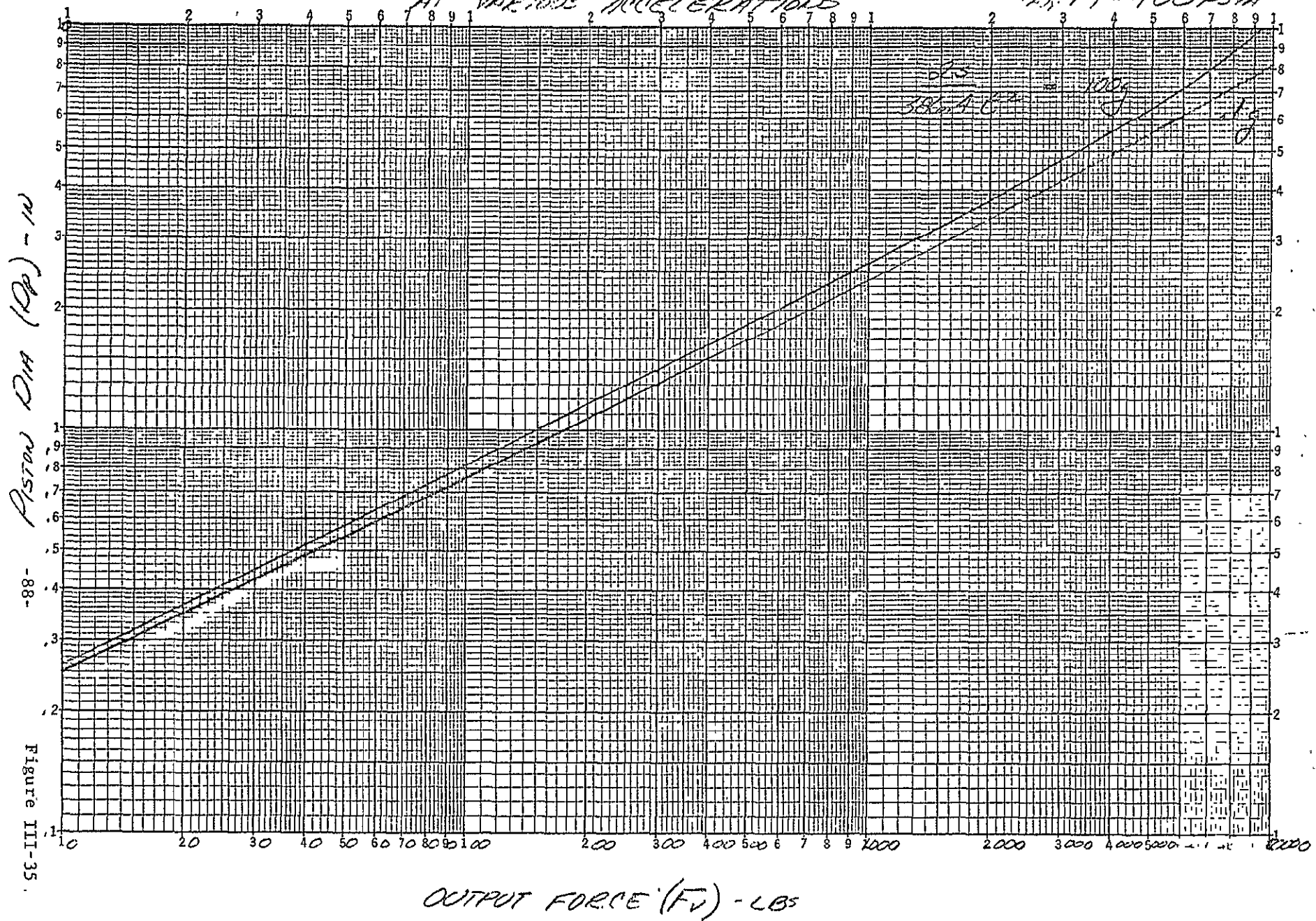
Output Force (F_v) - LBS @ .5 in P_1

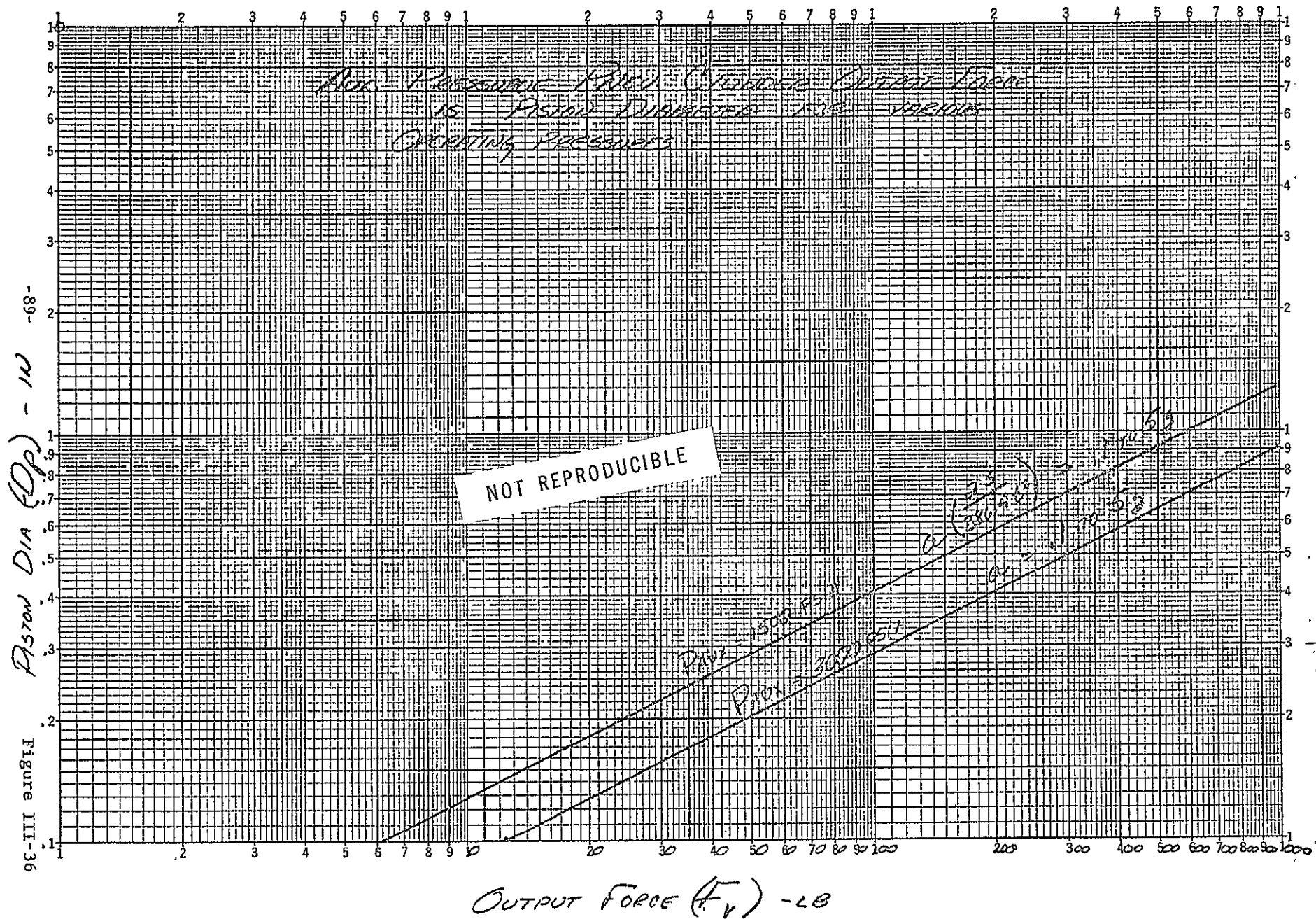
K&E LOGARITHMIC 4G 7323
2 X 3 CYCLES
KEUFFEL & ESSER CO. MADE IN U.S.A.

CYLINDER OUTPUT FORCE (F_u) VS PISTON DIA (D_p)
AT VARIOUS ACCELERATIONS $P_{PIN} = P_i = 150 \text{ PSIA}$



CYLINDER OUTPUT FORCE (FV) VS PISTON DIA (DP) $P_2 = P_1 = 400 \text{ PSIA}$
AT VARIOUS ACCELERATIONS





PNEUMATIC CYLINDER ACTUATOR
PNEUMATIC CYLINDER DELAY
Req'd Motion Time

O₂ GAS PNEUMATIC MEDIA

PNEUMATIC DELAY TIME - SECONDS

10^{-2}

10

10^{-3}

10

10^{-4}

10

10^{-5}

10

10

10^{-4}

.01 .02

.04

.06

.08

.10

.12

.14

-90-

REQUIRED MOTION TIME (t) - SEC. Figure III-37

TABLE III-XVII

PNEUMATIC MEDIA CORRECTION

MEDIA	MULTIPLY t_1	FROM FIGURE	III-37 by
O ₂		1	
H ₂		1.0106	
N ₂		1.1008	
He		.8258	

III-4B 3. Hydraulic Cylinder - A double acting cylinder identical to that assumed for the pneumatic cylinder actuator analysis is the basis for the hydraulic cylinder analysis. All assumptions and nomenclature are identical with the exception of the supply orifice pressure ratio at time of incipient motion. For a hydraulic system, a cylinder pressure of 80% of the supply pressure is assumed at time of motion. For purposes of the data generated, a hydraulic fluid with a specific gravity of 1.0 relative to water was assumed and viscosity affects are considered negligible.

Assuming rod areas and masses are negligible, the force balance at the time of incipient motion is:

$$F_A = F_v + \frac{\pi}{4} \frac{D_p^3}{4} \left(20 + \frac{2s}{386.4 t^2} \right) = .8 P_{aux1} \frac{\pi}{4} D_p^2 \quad (35)$$

Deriving the piston diameter (D_p), the maximum volumetric flow rate at the end of the stroke can be determined by:

$$Q = \frac{\pi}{4} D_p^2 \frac{2s}{t} \frac{\text{in}^3}{\text{sec}} \quad (36)$$

With the maximum volumetric flow rate established, the supply orifice flow factor $(C_d A_t)_o$ can be derived from:

$$(C_d A_t)_o = \frac{\frac{\pi}{4} Q}{6430 \sqrt{.2 \cdot P_{aux1}}} \text{ in.}^2 \quad (37)$$

The hydraulic opening delay is the time required to fill and pressurize the cylinder volume to 80% of the supply pressure. Since an incompressible fluid is assumed, pressurization is instantaneous once the volume has been filled. Assuming a .010 inch piston head clearance and a hydraulic line volume, downstream of the supply orifice equal to .015 inch of piston head clearance, the hydraulic delay time can be calculated from:

$$t_1 = \frac{\left(\frac{\pi}{4} \right)^2 D_p^2 (.025)}{(C_d A_t)_o \times 6430 \sqrt{P_{aux1}}} \quad (38)$$

where $(C_d A_t)_o$ = supply orifice flow factor from Equation (37) - in.²
 t_1 = hydraulic opening delay time - seconds

For an incompressible hydraulic fluid it can be assumed, with reasonable validity, that the closing hydraulic delay will be equal to the opening hydraulic delay ($t_1 = t_2$).

For the sample points within the range of interest (7 flow factors, 5 inlet pressures and 6 allowable motion times) hydraulic actuator data is tabulated in Table III-XVIII for hydraulic supply pressures of 1500 and 3000 psia. Design curves for determining piston diameter (D_p) and hydraulic delay time ($t_1 = t_2$) as a function of required output force (F_v), acceleration ($\frac{2s}{386.4 t^2}$) and allowable motion time (t) are plotted in Figures III-38 and III-39 for the assumptions stated in this discussion.

III-4B - 4. Torque Motor - A torque motor with a rotary output and having a configuration similar to that shown in Figure III-40 was the basis of the analysis described herein. The relationships used to define torque motor performance and characteristics were developed on the basis of data offered by Magnetic Technology, Inc. of Canoga Park, California, a recognized supplier of special application torque motors to the aerospace industry.

The configuration of the torque motor to which this analysis applies is shown in Figure III-40. The peak torque capability of a motor is a function of the physical dimensions and is expressed as:

$$T_{\text{peak}} = 2.375 D_A^2 \times (\ell - .25) \quad (39)$$

where T_{peak} = peak torque - in. lb
 D_A = armature outside diameter - in.
 ℓ = armature length - in.

This peak torque must equal the sum of the required output torque plus the torque necessary to accelerate the rotating element of the motor.

$$\therefore T_{\text{peak}} = T_v + I_{xx} \dot{\omega} = 2.375 D_A^2 (\ell - .25) \quad (40)$$

where T_v = required output torque - in. lb
 I_{xx} = moment of inertia of rotating element - in. lb sec²
 $\dot{\omega}$ = angular acceleration - RAD/sec

From previous torque motor designs fabricated and tested by Magnetic Technology, an armature inside diameter of 80% of the outside diameter is a reasonable proportion. The armature contains the windings and commutator ring and average densities of this element are of the order of .21 lb/in. The moment of inertia of the armature (rotating element) then becomes:

$$I_{xx} = \frac{\pi \ell}{58900} (.59 D_A^4) \quad (41)$$

Assuming constant acceleration and 90° of rotation to cycle from open to closed or closed to open, the expression for peak torque (Equation 41) can be rewritten as:

$$T_{\text{peak}} = 2.375 D_A^2 (\ell - .25) = T_v + \frac{\pi^2}{t^2} 10.03 \times 10^{-6} D_A^4 \ell \quad (42)$$

HYDRAULIC CYLINDER ACTUATOR DATA SUMMARY

0

Req'd Inches (in)	Stroke (in)	Porting Time (sec)	Flow Rate (gpm)	Total Flow (gpm)	Press Drop (psi)	No. Row Rate (in/100)	Stroke Rate (in/100)	Delay to Actuator (sec)	Valve Flow (in)	Valve Press (psi)
F _v	S	t	P _{gpm}	F _t	D _p	Q	C _p	t _i	C _v	P
20.26	276	.01	1500	20.26	.1463	.4633	1.423	.0018	.62	60
			3000	20.26	.1035	.2327	1.624	.0018		
20.20		.02	1500	20.20	.1464	.2321	3.701	.0036		
			3000	20.20	.1035	.1160	1.308	.0036		
20.19		.03	1500	20.19	.1464	.1546	2.466	.0059		
			3000	20.19	.1035	.0773	.871	.0059		
20.19		.05	1500	20.19	.1464	.0827	1.479	.0091		
			3000	20.19	.1035	.0464	.523	.0091		
20.18		.07	1500	20.18	.1464	.0662	1.056	.0127		
			3000	20.18	.1035	.0331	.523	.0127		
20.18		.10	1500	20.18	.1464	.0464	.739	.0181		
			3000	20.18	.1035	.0232	.261	.0181		
24.40		.01	1500	24.41	.161	.5607	8.94	.0018		150
			3000	24.40	.1136	.2803	3.16	.0018		
24.34		.02	1500	24.35	.1607	.2797	4.40	.0036		
			3000	24.35	.1134	.1398	1.577	.0036		
24.33		.03	1500	24.34	.1607	.1864	2.972	.0059		
			3000	24.34	.1134	.0832	1.051	.0059		
24.33		.05	1500	24.33	.1606	.1118	1.783	.0091		
			3000	24.33	.1133	.0559	.630	.0091		
24.33		.07	1500	24.33	.1606	.0798	1.273	.0127		
			3000	24.33	.1133	.0399	.450	.0127		
24.33		.10	1500	24.33	.1606	.0891	.559	.0181		
			3000	24.33	.1133	.0315	.279	.0181		
35.91		.01	1500	35.93	.1952	.8252	13.16	.0018		400
			3000	35.92	.138	.4125	4.65	.0018		
35.86		.02	1500	35.86	.195	.4119	6.569	.0036		
			3000	35.86	.138	.2059	2.322	.0036		
35.84		.03	1500	35.85	.195	.2745	4.378	.0059		
			3000	35.85	.138	.1372	1.598	.0059		
35.84		.05	1500	35.85	.195	.1647	2.626	.0091		
			3000	35.84	.138	.0823	.9284	.0091		
35.84		.07	1500	35.85	.195	.1176	1.876	.0127		
			3000	35.84	.138	.0588	.6631	.0127		
35.84		.10	1500	35.85	.195	.0823	1.313	.0181		
			3000	35.84	.138	.0412	.4642	.0181		
23.66	339	.01	1500	23.67	.1585	.6694	16.67	.0015	.94	30
			3000	23.66	.112	.3346	3.773	.0015		
23.52		.02	1500	23.52	.158	.3326	5.305	.0029		
			3000	23.52	.1115	.1663	1.825	.0029		
23.49		.03	1500	23.50	.158	.2215	3.532	.0044		
			3000	23.49	.1115	.1107	1.249	.0044		
23.48		.05	1500	23.48	.158	.1328	2.118	.0074		
			3000	23.48	.1115	.0664	.7488	.0074		
23.47		.07	1500	23.48	.158	.0949	1.513	.0103		
			3000	23.48	.1115	.0474	.535	.0103		
23.47		.10	1500	23.48	.158	.0664	1.059	.0147		
			3000	23.47	.1115	.0332	.374	.0147		
25.60		.01	1500	25.61	.165	.7244	11.53	.0015		60
			3000	25.61	.1165	.3621	4.083	.0015		
25.46		.02	1500	25.47	.1645	.3602	5.743	.0029		
			3000	25.47	.1164	.1801	2.030	.0029		
25.44		.03	1500	25.44	.1645	.2399	3.825	.0044		
			3000	25.44	.1164	.1199	1.352	.0044		
25.42		.05	1500	25.43	.1645	.1438	2.294	.0074		
			3000	25.42	.1164	.0719	.8109	.0074		
25.42		.07	1500	25.42	.1643	.1027	1.638	.0103		
			3000	25.42	.1164	.0514	.579	.0103		
25.42		.10	1500	25.42	.1643	.0719	1.147	.0147		
			3000	25.42	.1164	.0359	.403	.0147		

NOT REPRODUCIBLE

TABLE III-XVIII (Continued)
HYDRAULIC CYLINDER ACTUATOR DATA SUMMARY

TEST ID (60)	Stroke (in.)	Motor Time (sec.)	Flow Press. (PSIA)	Total Press. Dev. (PSI)	Press. D in (in.)	Flow Rate (in ³ /sec.)	Stroke Rate (in/sec.)	Dev. to Stroke Rate (sec.)	Max. Flow Rate (in ³ /sec.)	Value Press. (PSIA)
F ₁	S	t	P ₁	F ₂	D ₁	Q	(L ₁ A ₁ t ₁)	t ₁	Q ₁	P ₁
31.44	.539	.01	1500	31.45	.182	.583	14.18	.0019	.24	150
			3000	31.44	.129	.446	5.04	.0019		
31.30		.02	1500	31.30	.182	.442	7.00	.0029		
			3000	31.30	.1285	.2213	3.44	.0029		
31.27		.03	1500	31.28	.182	.244	4.23	.0044		
			3000	31.27	.1285	.147	1.66	.0044		
31.26		.05	1500	31.26	.182	.176	2.50	.0074		
			3000	31.26	.1285	.089	.997	.0074		
31.25		.07	1500	31.26	.182	.063	2.01	.0103		
			3000	31.26	.1285	.0631	.0712	.0103		
31.25		.10	1500	31.26	.182	.089	1.410	.0147		
			3000	31.25	.1285	.0442	.0498	.0147		
47.65		.01	1500	47.67	.225	1.348	21.5	.0015		400
			3000	47.65	.159	.673	7.599	.0015		
47.50		.02	1500	47.52	.224	.672	10.72	.0029		
			3000	47.51	.1588	.335	3.78	.0029		
47.48		.03	1500	47.49	.224	.447	7.14	.0044		
			3000	47.48	.1588	.223	2.524	.0044		
47.47		.05	1500	47.48	.224	.268	4.283	.0074		
			3000	47.47	.1588	.1343	1.514	.0074		
47.46		.07	1500	47.47	.224	.191	3.057	.0103		
			3000	47.47	.1588	.089	1.081	.0103		
47.46		.10	1500	47.47	.224	.1343	2.141	.0147		
			3000	47.46	.1588	.0671	.756	.0147		
40.35	.554	.01	1500	40.37	.207	.186	29.7	.0009	2.50	20
			3000	40.36	.146	.9307	10.5	.0009		
39.18		.02	1500	39.20	.204	.909	14.42	.0018		
			3000	39.19	.144	.457	5.096	.0018		
38.97		.03	1500	38.98	.203	.5793	9.557	.0027		
			3000	38.97	.1435	.294	3.378	.0027		
38.86		.05	1500	38.87	.203	.356	5.718	.0045		
			3000	38.86	.1455	.173	2.021	.0045		
38.83		.07	1500	38.84	.203	.255	4.081	.0063		
			3000	38.83	.1455	.127	1.443	.0063		
38.81		.10	1500	38.82	.203	.1791	2.856	.0090		
			3000	38.82	.1435	.0895	1.009	.0090		
41.85		.01	1500	41.88	.211	.193	30.80	.0009		30
			3000	41.86	.149	.965	10.89	.0009		
40.69		.02	1500	40.70	.208	.9387	14.97	.0018		
			3000	40.69	.147	.4692	5.291	.0018		
40.47		.03	1500	40.48	.207	.6225	9.926	.0027		
			3000	40.48	.147	.3112	3.509	.0027		
40.36		.05	1500	40.37	.207	.3724	5.939	.0045		
			3000	40.37	.147	.1862	2.100	.0045		
40.33		.07	1500	40.34	.207	.2658	4.239	.0063		
			3000	40.34	.147	.1329	1.489	.0063		
40.32		.10	1500	40.33	.207	.1860	2.966	.0090		
			3000	40.32	.147	.0929	1.049	.0090		
46.36		.01	1500	46.39	.222	.214	34.12	.0009		60
			3000	46.37	.157	.106	12.06	.0009		
45.20		.02	1500	45.21	.219	.1083	16.63	.0018		
			3000	45.20	.1545	.0213	5.88	.0018		
44.98		.03	1500	44.99	.2185	.0918	11.03	.0027		
			3000	44.98	.154	.3459	3.90	.0027		
44.87		.05	1500	44.89	.218	.091	6.603	.0045		
			3000	44.88	.154	.2070	2.934	.0045		
44.84		.07	1500	44.85	.218	.0856	4.713	.0063		
			3000	44.84	.154	.1478	1.666	.0063		
44.83		.10	1500	44.84	.218	.2068	3.298	.0090		
			3000	44.83	.154	.1034	1.166	.0090		

TABLE III-XVIII (Continued)
HYDRAULIC CYLINDER ACTUATOR DATA SUMMARY

③

Stroke Inch (in)	Stroke (in)	Ported Time (sec)	Hyd. Press. (PSIA)	Total Press. Dev. (PSI)	Press. Dia. (in)	Hyd. Flow Rate (in ³ /min)	Stroke Rate (in/sec)	Leak Rate (in ³ /min)	Leak Rate (in ³ /min)	Leak Rate (in ³ /min)	Leak Rate (in ³ /min)
F _v	S	t	P _{aux}	F _a	D _p	Q	(in ³ /min)	(in ³ /min)	(in ³ /min)	(in ³ /min)	(in ³ /min)
57.89	55.9	.01	1500	57.94	.255	2.765	21.017	.0001	2.50	150	
			3000	57.91	.178	1.382	10.508	.0001			
58.73		.02	1500	58.73	.250	1.355	21.61	.0018			
			3000	58.74	.176	.6773	7.635	.0018			
58.52		.03	1500	58.54	.249	.9000	14.33	.0027			
			3000	58.52	.176	.4499	5.073	.0027			
58.41		.05	1500	58.42	.2485	.5390	8.585	.0045			
			3000	58.41	.176	.2690	3.038	.0045			
58.37		.07	1500	58.39	.2485	.3848	6.130	.0063			
			3000	58.38	.176	.1923	2.169	.0063			
58.36		.10	1500	58.38	.2485	.2693	4.299	.0090			
			3000	58.36	.176	.1346	1.578	.0090			
80.37	89.3	.01	1500	80.46	.292	5.994	45.43	.0006	6.50	20	
			3000	80.40	.206	2.990	23.71	.0006			
71.20		.02	1500	71.24	.275	2.649	42.25	.0011			
			3000	71.22	.194	1.324	14.938	.0011			
69.51		.03	1500	69.53	.272	1.724	27.49	.0017			
			3000	69.52	.192	.8617	9.717	.0017			
68.64		.05	1500	68.66	.270	1.021	16.29	.0028			
			3000	68.65	.191	.5106	5.757	.0028			
68.40		.07	1500	68.42	.269	1.7270	11.59	.0039			
			3000	68.41	.190	.3624	4.098	.0039			
68.27		.10	1500	68.29	.269	.5099	8.100	.0056			
			3000	68.28	.190	.2539	2.863	.0056			
83.93		.01	1500	84.03	.299	6.280	99.66	.0006	30		
			3000	83.96	.211	3.122	35.21	.0006			
74.77		.02	1500	74.81	.282	2.782	44.36	.0011			
			3000	74.78	.199	1.390	15.68	.0011			
73.07		.03	1500	73.10	.2785	1.812	28.90	.0017			
			3000	73.08	.196	.9059	10.22	.0017			
72.20		.05	1500	72.23	.277	1.074	17.13	.0028			
			3000	72.21	.1955	.5371	6.056	.0028			
71.96		.07	1500	71.99	.276	.7649	12.20	.0039			
			3000	71.97	.195	.3823	4.331	.0039			
71.89		.10	1500	71.88	.276	.5345	8.523	.0056			
			3000	71.85	.195	.2672	3.013	.0056			
94.62		.01	1500	94.74	.317	7.046	112.8	.0006	60		
			3000	94.67	.224	3.520	39.70	.0006			
85.46		.02	1500	85.51	.202	3.180	50.71	.0011			
			3000	85.48	.213	1.589	17.93	.0011			
83.76		.03	1500	83.80	.298	2.078	33.13	.0017			
			3000	83.78	.2105	1.038	11.71	.0017			
82.90		.05	1500	82.93	.2975	1.234	19.67	.0028			
			3000	82.91	.210	.6166	6.593	.0028			
82.66		.07	1500	82.69	.297	.8785	14.01	.0039			
			3000	82.67	.210	.4392	4.852	.0039			
82.53		.10	1500	82.56	.297	.6140	9.742	.0056			
			3000	82.54	.210	.3069	3.461	.0056			
121.27	110.7	.01	1500	121.65	.359	11.21	178.7	.0004	10	20	
			3000	121.40	.254	5.597	63.11	.0004			
98.04		.02	1500	98.10	.323	4.525	72.16	.0009			
			3000	98.06	.228	2.262	25.50	.0009			
93.73		.03	1500	93.78	.315	2.889	45.99	.0014			
			3000	93.75	.222	1.441	16.25	.0014			
91.53		.05	1500	91.57	.311	1.689	26.84	.0023			
			3000	91.54	.219	.8445	9.523	.0023			
90.92		.07	1500	90.96	.3105	1.199	19.12	.0031			
			3000	90.94	.219	.5992	6.757	.0031			
90.60		.10	1500	90.63	.310	.8361	13.33	.0045			
			3000	90.61	.219	.4180	4.713	.0045			

NOT REPRODUCIBLE

TABLE III-XVIII (Continued)
HYDRAULIC CYLINDER RETURN DATA SUMMARY

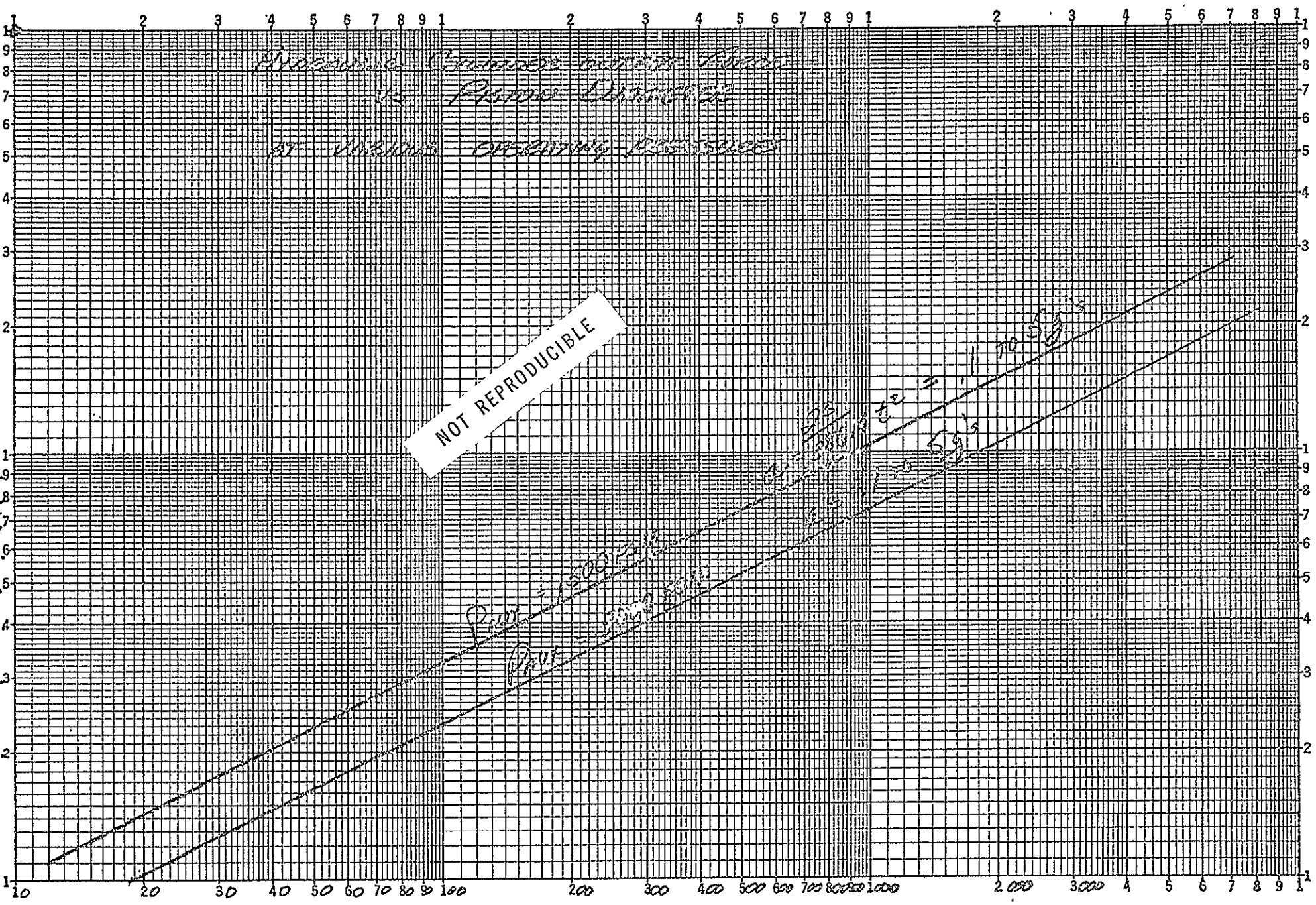
TEST NO. PRESS. CYLINDER (LBS)	STROKES (IN.)	Normal Time (sec)	Hyd. Press. (PSI)	Total Press. Drop (LBS)	Assn. Dia (IN.)	Flow Rate (GPM/SEC)	Stroke Dist. Frac. (IN.)	Delay to Return (SEC)	Valve Open (IN.)	Valve Press. (PSI)
F _v	S	t	P _{aux.}	F _A	D _p	Q	(L ₁ A ₁) ₀	t ₁	C ₁ A ₁	P ₁
126.59	1.107	.01	1500	126.80	.367	11.70	18.31, 18.31	.0005	10	50
			3000	126.66	.259	5.842	6.588	.0005		
103.35		.02	1500	103.42	.383	4.770	7.07	.0009		
			3000	103.38	.235	2.384	2.88	.0009		
99.05		.03	1500	99.10	.324	3.047	4.60	.0014		
			3000	99.07	.209	1.523	1.718	.0014		
96.85		.05	1500	96.89	.320	1.788	2.51	.0023		
			3000	96.86	.226	1.8936	10.08	.0023		
96.24		.07	1500	96.28	.3195	1.269	20.25	.0032		
			3000	96.25	.226	1.6343	7.03	.0032		
95.92		.10	1500	95.95	.319	1.8552	14.12	.0045		
			3000	95.93	.225	4.425	9.99	.0045		
142.54		.01	1500	142.79	.389	13.17	210.1	.0005		60
			3000	142.62	.2745	6.579	74.18	.0005		
119.30		.02	1500	119.39	.3535	5.507	8.82	.0009		
			3000	119.33	.251	2.752	3.03	.0009		
115.00		.03	1500	115.06	.549	3.538	56.42	.0014		
			3000	115.02	.246	1.768	19.94	.0014		
112.80		.05	1500	112.85	.345	2.082	33.20	.0023		
			3000	112.81	.244	1.041	11.74	.0023		
110.19		.07	1500	112.24	.345	1.479	23.59	.0032		
			3000	112.21	.244	1.7394	8.537	.0032		
111.87		.10	1500	111.91	.344	1.032	16.46	.0045		
			3000	111.88	.243	.561	5.819	.0045		
405.71	1.750	.01	1500	407.46	.656	59.43	947.8	.0003	25	20
			3000	406.32	.463	29.63	334.2	.0003		
237.57		.02	1500	237.87	.502	17.35	276.1	.0006		
			3000	237.67	.354	8.667	91.23	.0006		
206.43		.03	1500	206.60	.468	10.05	160.2	.0009		
			3000	206.49	.330	5.020	56.60	.0009		
190.48		.05	1500	190.60	.449	5.500	88.67	.0014		
			3000	190.53	.317	2.779	31.34	.0014		
186.09		.07	1500	186.20	.444	3.880	41.87	.0020		
			3000	186.13	.314	1.939	21.87	.0020		
183.76		.10	1500	183.86	.441	2.682	42.77	.0029		
			3000	183.79	.312	1.340	15.11	.0029		
418.36		.01	1500	420.19	.668	61.29	977.4	.0003		30
			3000	419.00	.472	30.56	344.6	.0003		
250.22		.02	1500	250.54	.515	18.27	291.4	.0006		
			3000	250.33	.364	9.128	102.9	.0006		
219.08		.03	1500	219.44	.452	10.66	170.0	.0009		
			3000	219.20	.340	5.327	60.07	.0009		
203.14		.05	1500	203.27	.464	5.930	94.56	.0014		
			3000	203.18	.328	2.964	33.42	.0014		
198.74		.07	1500	198.86	.459	4.144	66.08	.0020		
			3000	198.78	.324	2.071	23.35	.0020		
196.41		.10	1500	196.52	.456	2.866	45.71	.0029		
			3000	196.45	.322	1.433	16.16	.0029		

HYDRAULIC CYLINDER ACTUATOR DATA SUMMARY

⑤

REQ'D PRESS Q (LBS)	STROKE (IN.)	Normal Time (SEC)	HYD. PRESS. (PSIA)	TOTAL FORCE DEVEL (LBS)	PRESS DIA (IN.)	Flow Rate (GPM/1000)	SAFETY FACTOR (IN ²)	DEVIATION (SEC)	VALVE FLOW (GPM)	VALVE PRESS (PSIA)
F _v	S	t	P _{WH}	F _A	D _p	Q	(G.P.M.)	t ₁	Cy/Sec	P ₁
745.39	2.10	.01	1500	150.97	.892	131.8	2095.1106	.0002	56	20
			3000	747.18	.630	65.39	757.9	.0002		
375.78		.02	1500	376.44	.632	32.95	525.4	.0005		
			3000	376.02	.446	16.45	145.5	.0005		
307.34		.03	1500	307.67	.572	17.95	286.3	.0007		
			3000	307.45	.403	8.968	101.1	.0007		
272.29		.05	1500	272.50	.537	9.539	152.1	.0012		
			3000	272.37	.379	4.767	53.76	.0012		
262.64		.07	1500	262.82	.528	6.572	104.8	.0017		
			3000	262.70	.372	3.284	37.04	.0017		
257.51		.10	1500	257.67	.522	4.510	71.92	.0024		
			3000	257.57	.368	2.254	25.42	.0024		
763.35		.01	1500	768.61	.902	134.5	2145	.0002	36	30
			3000	765.20	.637	66.97	763.1	.0002		
393.73		.02	1500	394.44	.646	34.52	550.5	.0005		
			3000	393.98	.457	17.24	194.4	.0005		
325.29		.03	1500	325.65	.587	19.00	303.0	.0007		
			3000	325.42	.415	9.493	107.0	.0007		
290.24		.05	1500	290.47	.555	10.17	162.2	.0012		
			3000	290.33	.392	5.082	57.30	.0012		
280.59		.07	1500	280.79	.545	7.021	112.0	.0017		
			3000	280.66	.384	3.509	39.57	.0017		
275.46		.10	1500	275.64	.540	4.825	76.94	.0024		
			3000	275.52	.381	2.411	27.19	.0024		

Figure III-38
Piston Diameter (Dp) - in



Output Force (Fv) - LBS

359-91G
 FULL LOGARITHMIC
 5 CYCLES X 70 DIVISIONS

HYDRAULIC DELAY - SECONDS

HYDRAULIC DELAY

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

10

11

12

10⁻⁴

10⁻³

10⁻²

10⁻¹

1

2

3

4

5

6

7

8

9

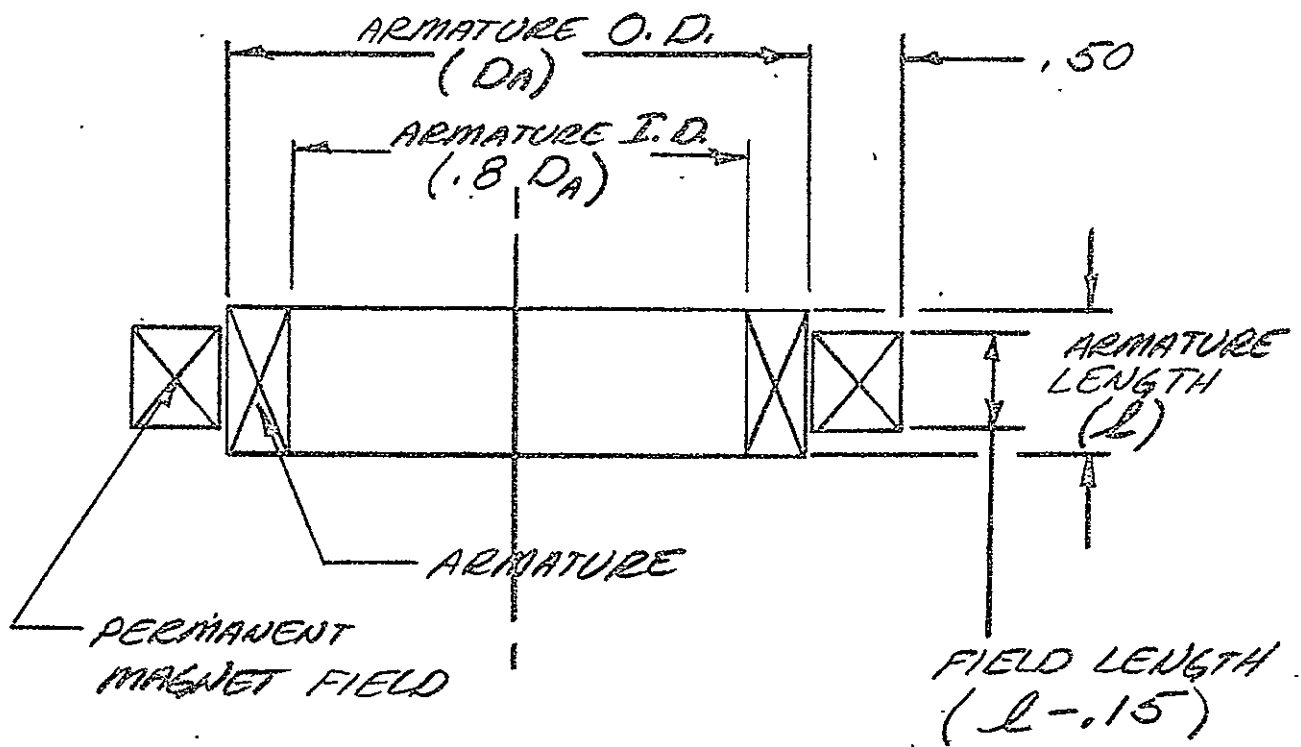
10

11

12

10⁻⁴

TORQUE MOTOR ACTUATOR - SCHEMATIC DRAWING



Power density within the armature coil windings impose another limitation. From previous designs, the relationship between peak torque power and motor physical characteristics is defined as:

$$W = 33 (\ell - .25) + 12 D_A \quad (43)$$

where W = power at peak torque - watts.

Since power is defined at 32 V d-c and 70°F and motor operation is required at 18 V d-c and 200°F. Assuming that the motor armature coil windings are of copper wire, Equation (43), revised to the worst case operating condition becomes:

$$.433 W = 33 (\ell - .25) + 12 D_A \quad (44)$$

For each combination of maximum power (W) and an assumed motor length (ℓ) Equation (44) is employed to calculate armature diameter (D_A). The resultant combination of armature diameter (D_A) and motor length (ℓ) are utilized in Equation (42) to calculate torque output (T_V).

Torque motors, by design, have inherently small electrical time constants. The low self inductance results in the achievement of peak torque within a very short period of time after application of current. For torque motors within the scope of this analysis a rise time to peak torque of 2 milliseconds is assumed. For an application wherein a more refined estimate of response time is desired, a more rigorous analysis should be performed. This more refined analysis would include consideration for optimizing operating flux density and the number of magnetic poles in addition to a more refined output characteristic analysis.

Since the analytical approach is not consistent with the previous actuator analyses, in that the required output (T_V) is calculated from other factors rather than being employed as an input to determine the actuator factors, the data tabulation as shown in Table III-XIX does not relate directly to the sample flow factors ($C_d A_t$) and inlet pressures (P_1) within the range of interest. To facilitate the use of this data and to relate it to a specific application (T_V) Figures III-41 through III-46 present design curves wherein output torque (T_V) versus allowable motion time (t) is plotted as a function of maximum electrical power (W) and motor length (ℓ).

TABLE III-XIX

Torque Motor Actuating Data Summary

Req'd Torque Output (T) IN. LB.	Motor Length (L) IN.	Max. Torque (W) WATTS	Replicate Time (t) SEC.	Measured Dia (D _M) IN.	Motor O.D. IN.	Motor Weight LB.
.077	0.4	28	.01	.598	1.598	.130
.115			.02			
.122			.03			
.126			.05			
.127			.07			
.127			.10			
—			.01			
.261		56	.02	1.608	2.608	.296
.629			.03			
.817			.05			
.869			.07			
.897			.10			
—			.01			
.379			.02			
1.70			.03			
2.07			.05			
2.26			.07			
—		84	.10	2.618	3.618	.509
—			.01			
.379			.02			
1.70			.03			
2.07			.05			
2.26			.07			
—			.10			
—		112	.01	3.629	4.629	.771
—			.02			
1.95			.03			
3.30			.05			
4.01			.07			
.057	0.5	28	.10			
.061			.01	.323	1.323	.131
.061			.02			
.062			.03			
.062			.05			
.062			.07			
—			.10			
.667		56	.01		1.333	2.333
.884			.02			
.995			.03			
1.026			.05			
1.042			.07			
—			.10			
1.609		84	.01		2.343	3.343
2.671			.02			
2.963			.03			
3.118			.05			
—			.07			
—			.10			
—		112	.01		3.354	4.354
4.188			.02			
5.415			.03			
6.066			.05			
—			.07			
—			.10			

TABLE III-XIX (Continued)

Torque Motor Actuator Data Summary

Req'd Torque Capacity (TL) IN. LB.	Motor Length (L) IN.	Max. Power (W) WATTS	Exposure Time (t) SEC.	Mounting DIA (D _m) IN.	Motor O.D. (D _m) IN.	Motor Weight (W _m) LB.
.188	0.6	56	.01	1.058	2.058	.352
.746			.02			
.850			.03			
.903			.05			
.917			.07			
.925			.10			
—			.01			
.845		84	.02	2.068	3.068	.666
2.356			.03			
3.129			.05			
3.342			.07			
3.455			.10			
—			.01			
1.966		112	.02	3.079	4.079	1.053
5.761			.03			
6.807			.05			
7.362			.07			
—			.10			
.285	0.8	56	.01	1.508	1.508	.323
.325			.02			
.332			.03			
.336			.05			
.337			.07			
.338			.10			
—			.01			
1.966		84	.02	1.518	2.518	.693
2.550			.03			
2.850			.05			
2.932			.07			
2.976			.10			
—			.01			
.274		112	.02	2.529	3.529	1.179
4.772			.03			
7.076			.05			
7.710			.07			
8.047			.10			
1.803	1.00	84	.01	1.968	1.968	.610
1.457			.02			
1.577			.03			
1.639			.05			
1.656			.07			
1.666			.10			
—			.01			
3.195		112	.02	1.979	2.979	1.169
3.303			.03			
6.382			.05			
6.680			.07			
6.838			.10			

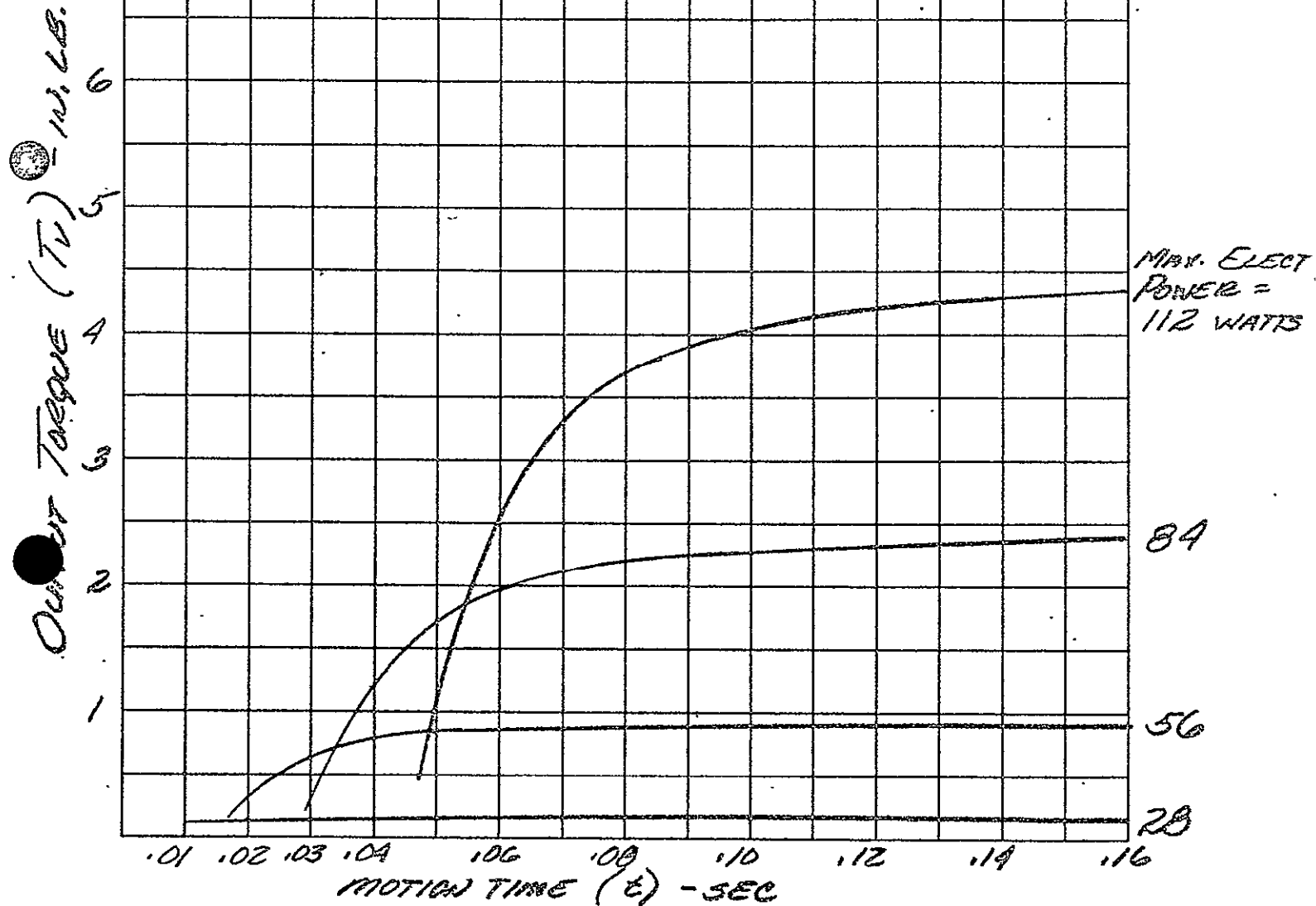
TABLE III-XIX (Continued)

Torque Motor Actuator Data Summary

REQ'D PRESS CURRY (TV) IN. LB.	MOTOR LENGTH (L) IN.	MAX. FORCE (W) POUNDS	REQUIRE TIME (t) SEC.	REMARKS DIA (D _a) IN.	MOTOR O.D. IN.	MOTOR WEIGHT LB.
.180	1.25	84	.01	.281	1.281	.587
.186			.02			
.187			.03			
.188			.05			
.188			.07			
.188			.10			
.528		112	.01	1.291	2.291	.999
3.108			.02			
3.386			.03			
3.831			.05			
3.898			.07			
3.934			.10			
.887	1.50	112	.01	.604	1.604	.695
1.035			.02			
1.063			.03			
1.077			.05			
1.081			.07			
1.083			.10			

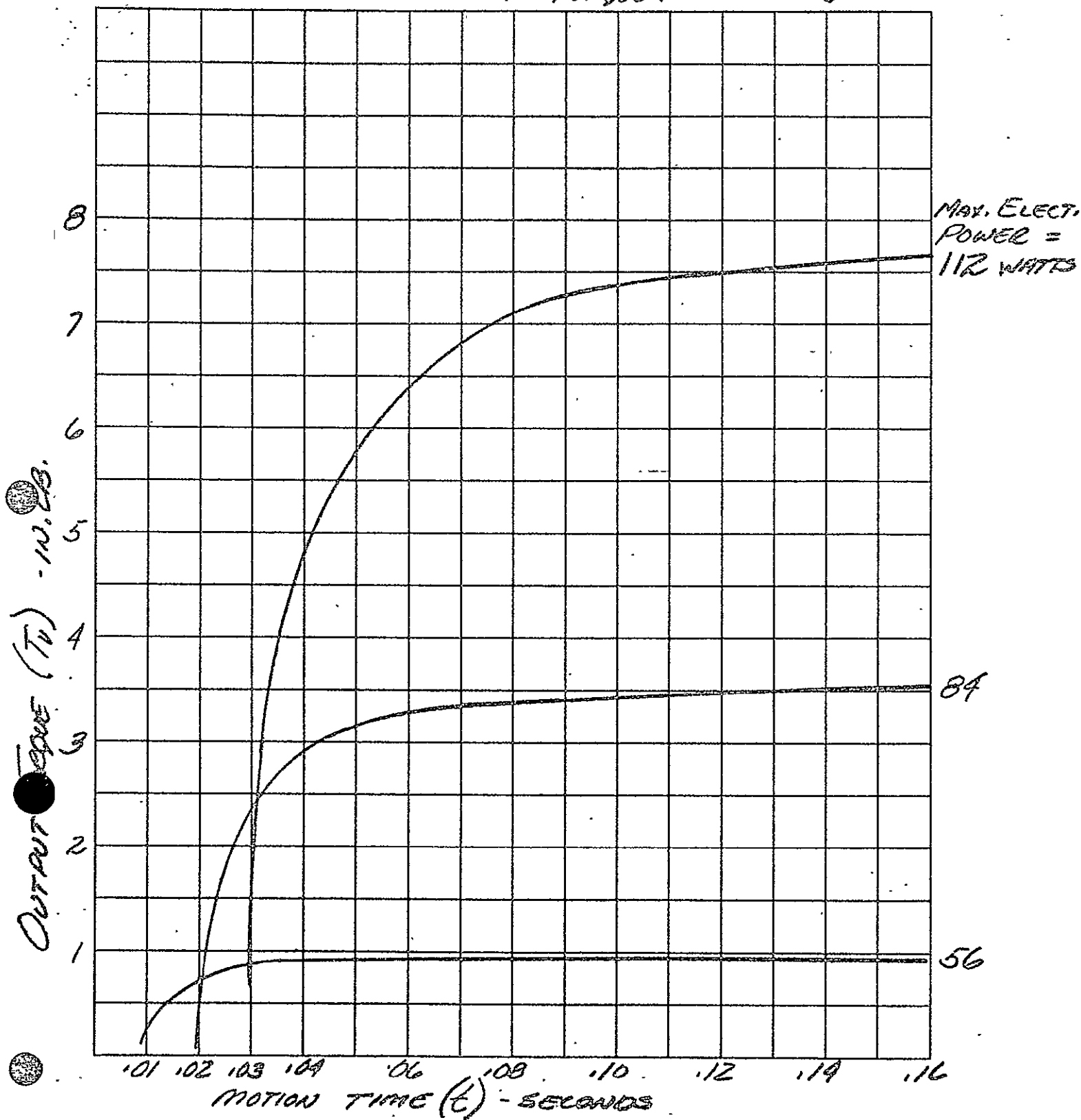
Torque Motor Output vs Motion Time for Various Power Levels

Torque Motor Length = .4 in.



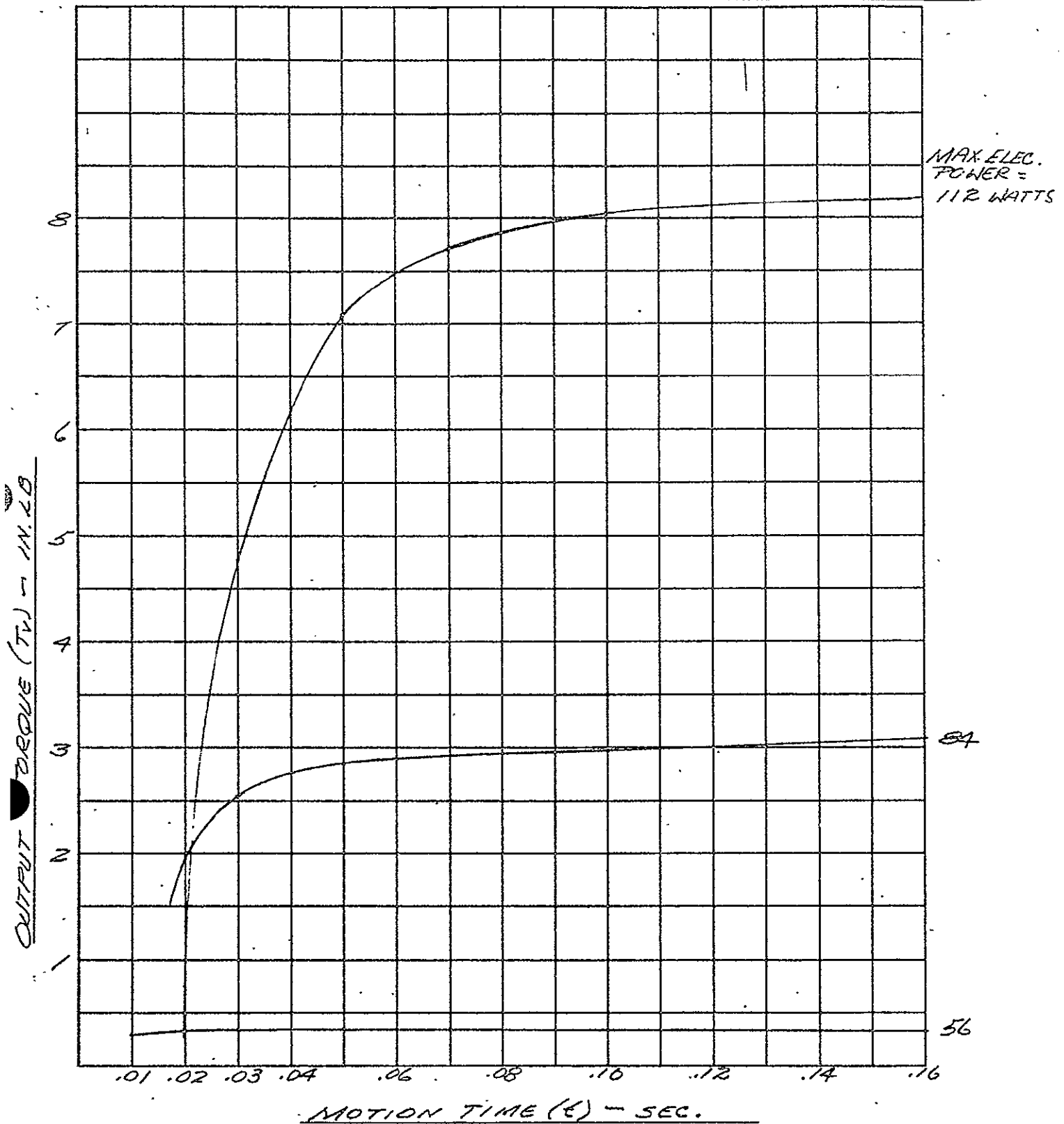
Torque Motor Output vs Motion Time for Various Power Levels

Torque Motor Length = .6 in



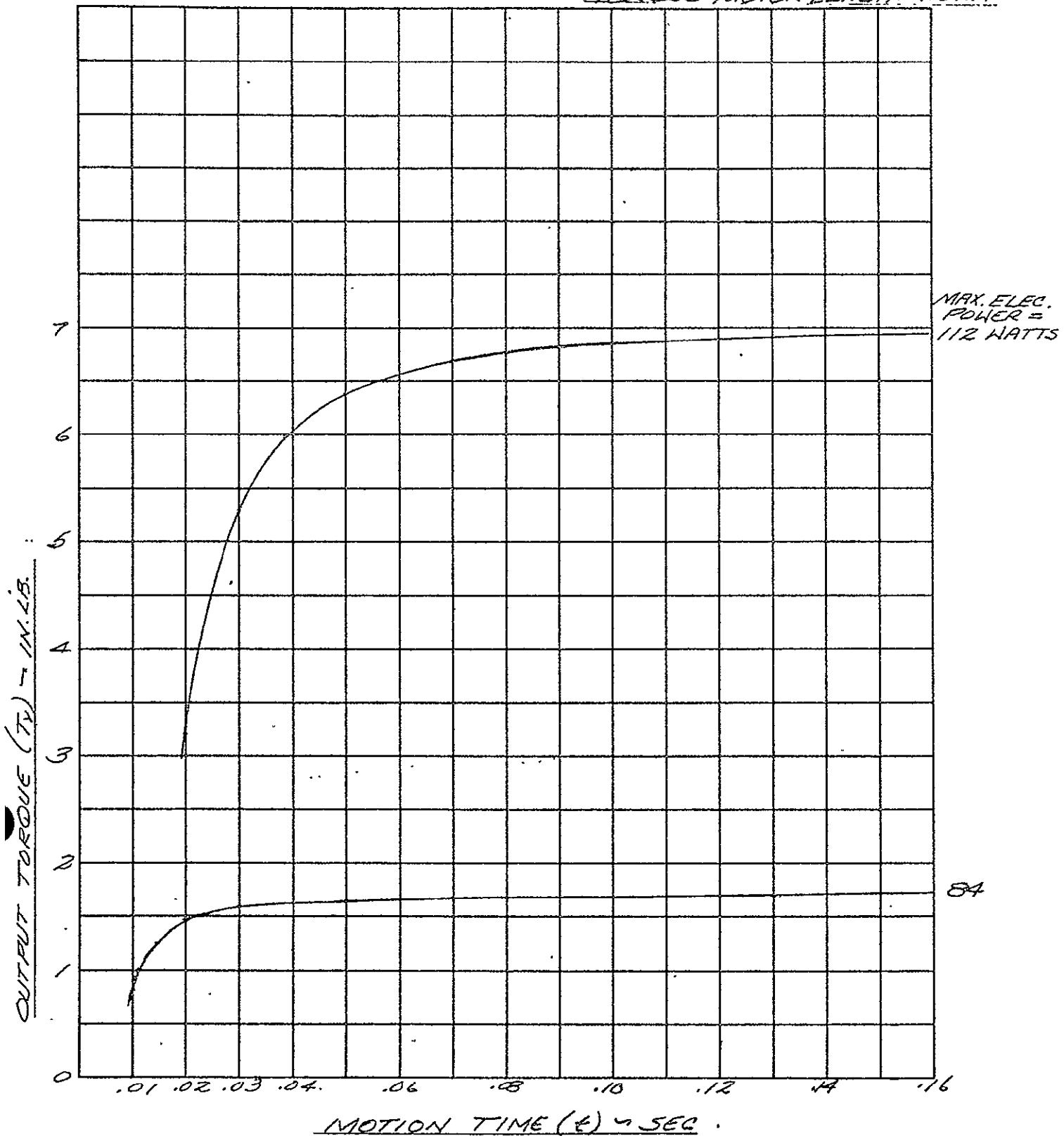
TORQUE MOTOR OUTPUT VS. MOTION TIME FOR VARIOUS
POWER LEVELS

TORQUE MOTOR LENGTH = .8 IN.



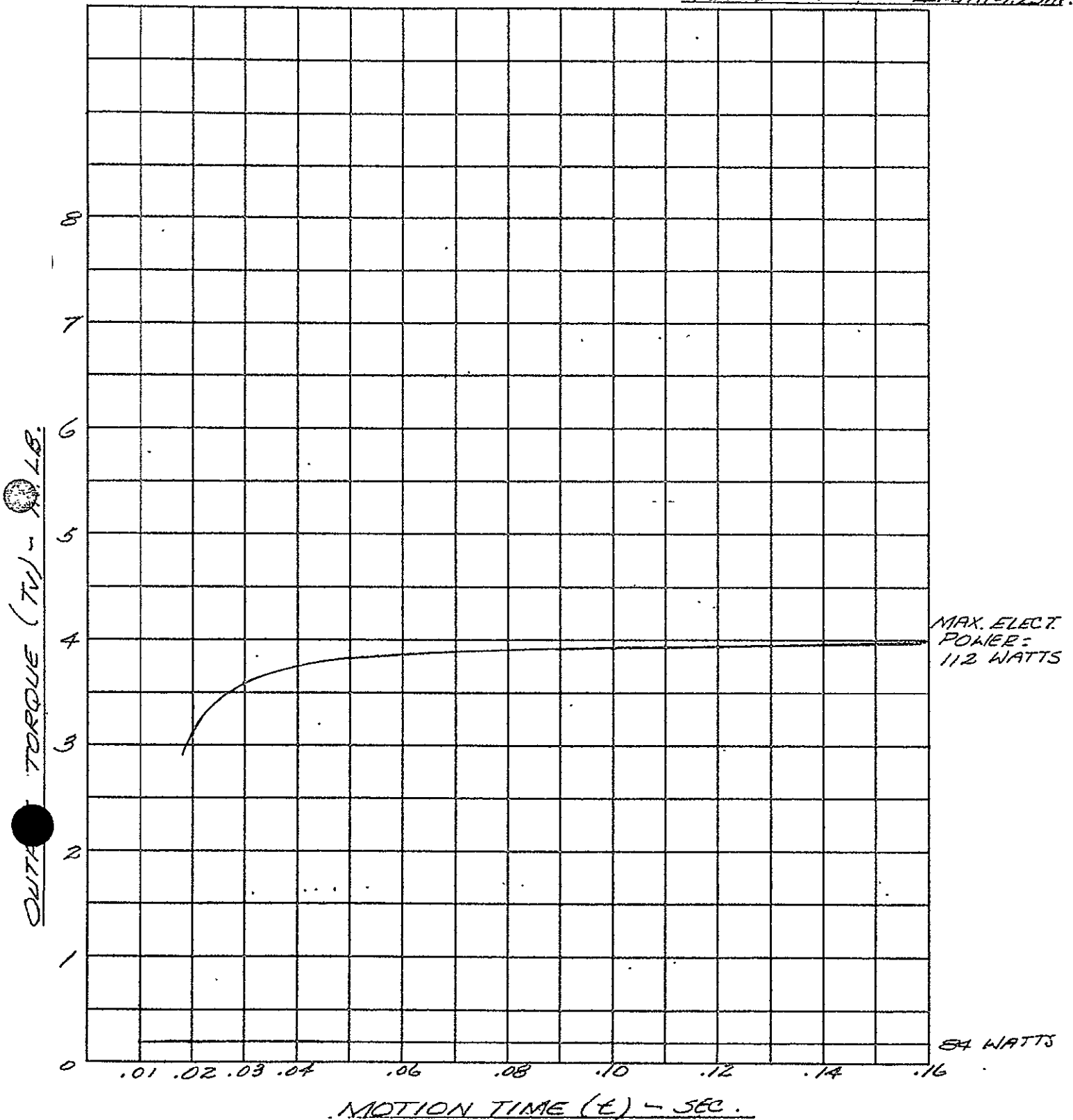
TORQUE MOTOR OUTPUT VS. MOTION TIME FOR VARIOUS
POWER LEVELS

TORQUE MOTOR LENGTH = 1.0 IN.



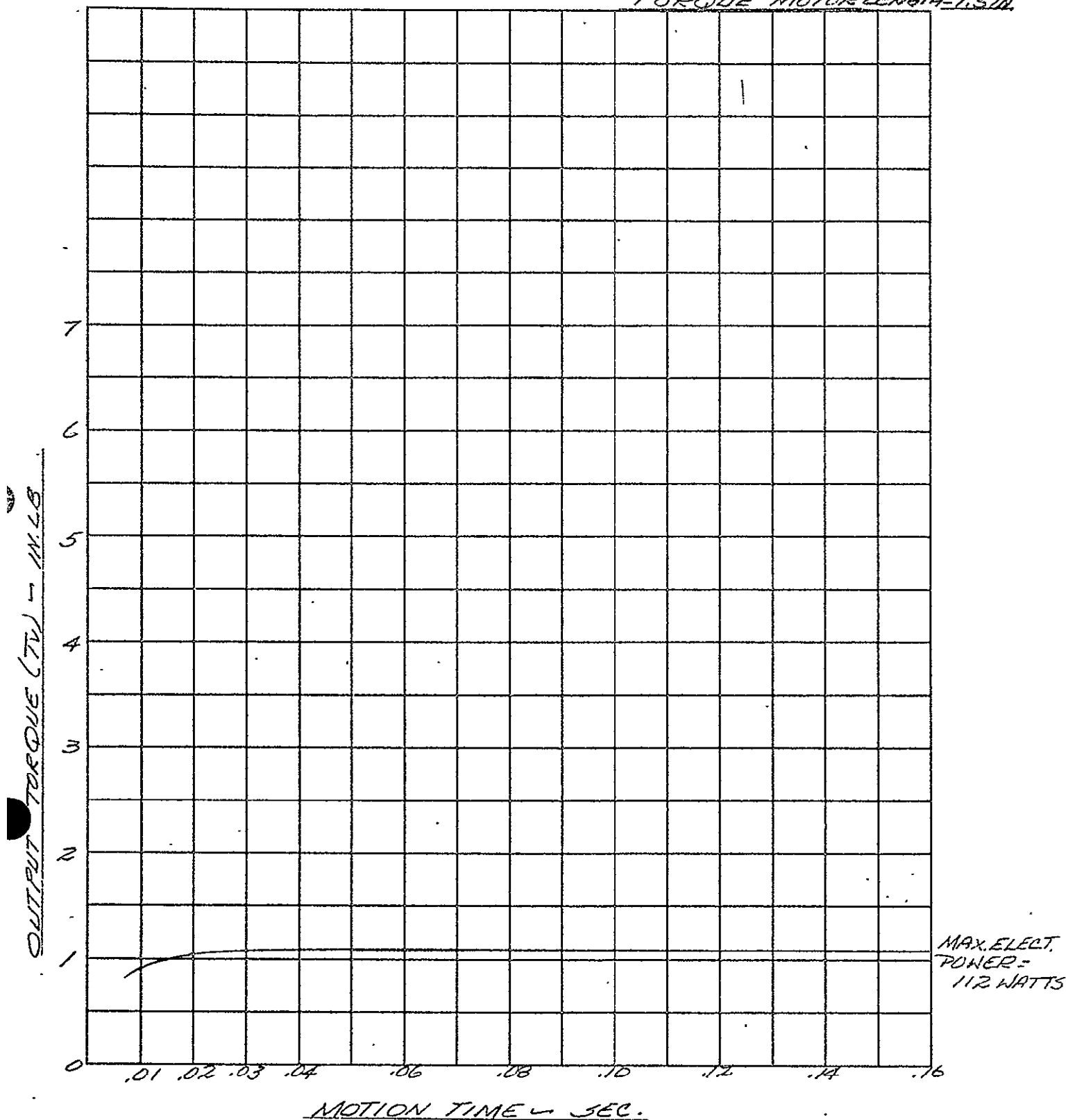
TORQUE MOTOR OUTPUT VS. MOTION TIME FOR VARIOUS
POWER LEVELS

TORQUE MOTOR LENGTH = 1.25 IN.



TORQUE MOTOR OUTPUT VS. MOTION TIME FOR VARIOUS
POWER LEVELS

TORQUE MOTOR LENGTH = 1.5 IN.



III-4C Shut-off Device/Actuator Interface Performance Considerations

Uniting an actuator with a shut-off device to constitute a valve configuration requires some consideration for interface effects on performance. Since the ultimate valve design will integrate these elements into one package it was assumed that where the actuator output characteristic (linear or rotary) is identical with the required input characteristic to the shut-off device (linear or rotary), 100% efficiency is accomplished at the interface. In a design wherein motion conversion is required (linear to rotary or vice versa) to accomplish total valve function, 90% efficiency results. Mathematically, the relationship is:

$$T_v = .90 (F_v \frac{s}{2}) \quad (45)$$

where T_v = required torque to operate rotary motion shut-off device--in.lb.

F_v = output force of linear motion actuator -- in.

s = stroke of linear motion actuator -- in.

and

$$.9(T_v) = F_v \frac{s}{2} \quad (46)$$

where T_v = torque output of rotary motion actuator -- in.lb.

F_v = required force to operate linear motion shut-off device -- lbs

s = stroke of linear motion shut-off device -- in.

These relationships are consistent with the previous assumptions of 90° rotary motion to effect opening or closing.

To include a consideration for motion time (t) and facilitate the preparation of useful design curves, the linear motion element stroke and allowable motion time are combined into the dimensionless quantity for acceleration.

$$\text{acceleration (a)} = \frac{2s}{386.4 t^2} = \frac{\pi}{t^2}$$

Substituting this quantity into Equations (45) and (46) results in:

$$T_v = F_v \times 86.94 a t^2 \quad (47)$$

for a rotary motion shut-off (T_v) operated by a linear motion actuator (F_v) and

$$T_v = F_v \times 107.3 t^2 a \quad (48)$$

for a linear motion shut-off (F_v) operated by a rotary motion (T_v) actuator. Design curves of these relationships are presented in Figures III-47 and III-48.

III-4D Physical Characteristics

During the development of the analyses to predict performance of the shut-off devices and actuators specific dimensional characteristics of each element were determined and employed as transfer functions in solving for performance parameters. To make reasonable estimates of the weight and volume characteristics, the dimensions were employed as well as considerations for the assumed configurations. Additional supportive data was generated by a concurrent conceptual design effort. This conceptual design effort was done without the benefit of the developed parametric performance analysis program. Rather, specific valve concepts were developed for selected flow factors ($C_d A_t$) and inlet pressures (P_1) within the range of interest to document several unique packaging concepts. These layouts also served as a basis for determining the relationship between flow factor ($C_d A_t$) and moving mass. These designs and discussion of them is found in Section III-4E of this report.

As in the development of the analysis program for performance prediction, the approach taken to develop physical characteristic parametric data was to consider a valve to consist of a shut-off device and an actuator, and develop analysis programs for each element.

III-4 Linear Motion Shut-off Device

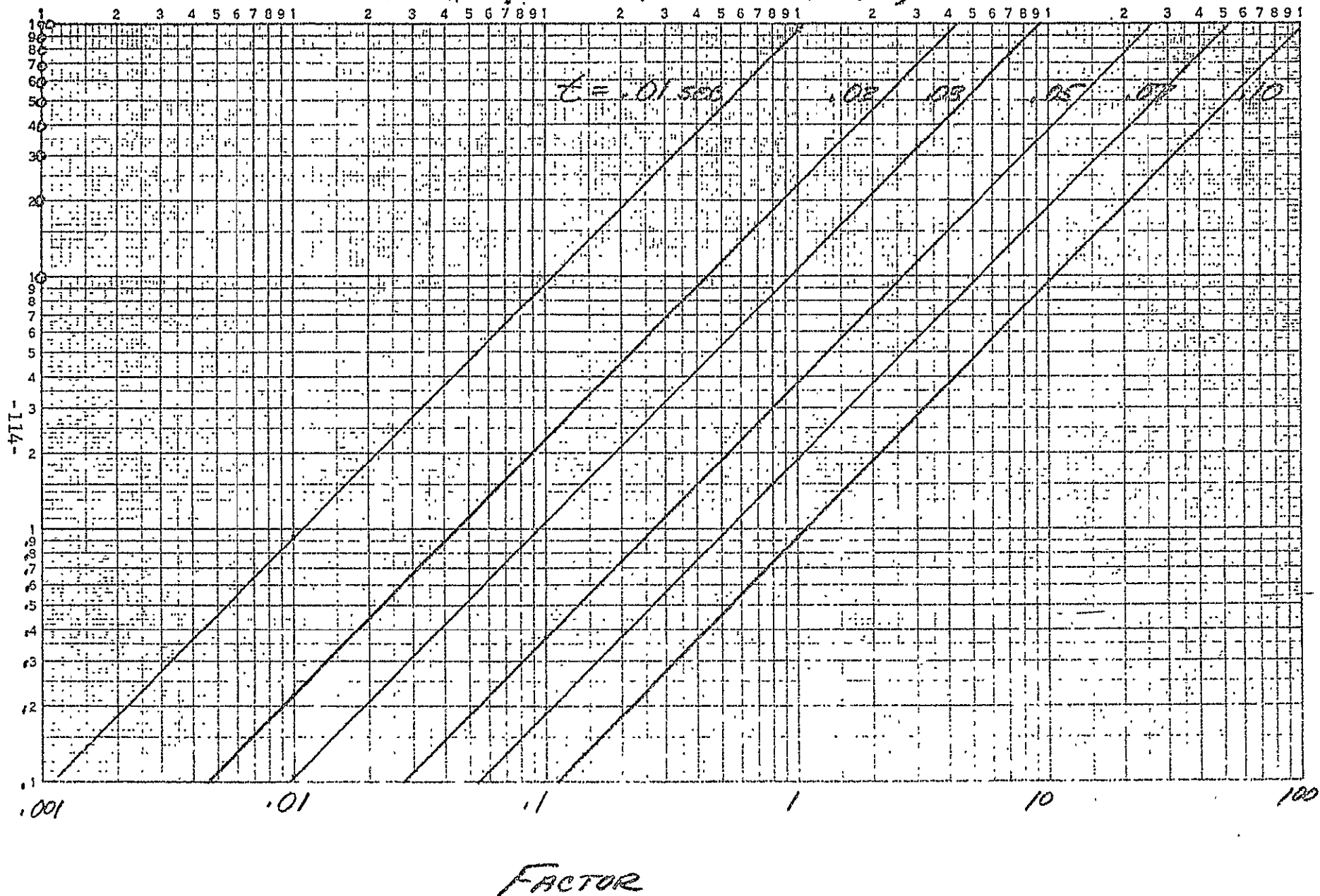
D-1

With reference to Figure III-49, the linear motion shut-off device was assumed to be a coaxial flow balanced poppet configuration. For the coaxial configuration the annular flow area must be at least equal to the minimum flow area allowed in the valve. Allowing a 10% margin on this area, relative to the minimum, the maximum flow passage diameter can be defined in terms of the seat diameter (D_s) or the flow factor ($C_d A_t$) as:

$$D_o = \sqrt{2.1 D_s^2} = \sqrt{2.1 \frac{C_d A_t}{\frac{\pi}{4} C_d}} \quad (49)$$

where D_o = maximum flow passage diameter -- in.
 D_s = nom. seat diameter -- in.
 C_d = equiv. orifice coef. = .65 for linear motion shut-off
 $C_d A_t$ = flow factor -- in²

TORQUE TO LINEAR FORCE REQUIRED CONVERSION FACTOR VS ACCELERATION FOR VARIOUS MOTION TIMES ACTUATOR TORQUE = VALUE FORCE REQ'D (F₀) X FACTOR



ACTUATOR LINEAR FORCE TO VALVE TORQUE CONVERSION FACTOR VS. ACCELERATION
 $T_V = F_V \times (\text{FACTOR})$
 $T_V = \text{VALVE TORQUE}$
 $F_V = \text{ACTUATOR LINEAR FORCE}$

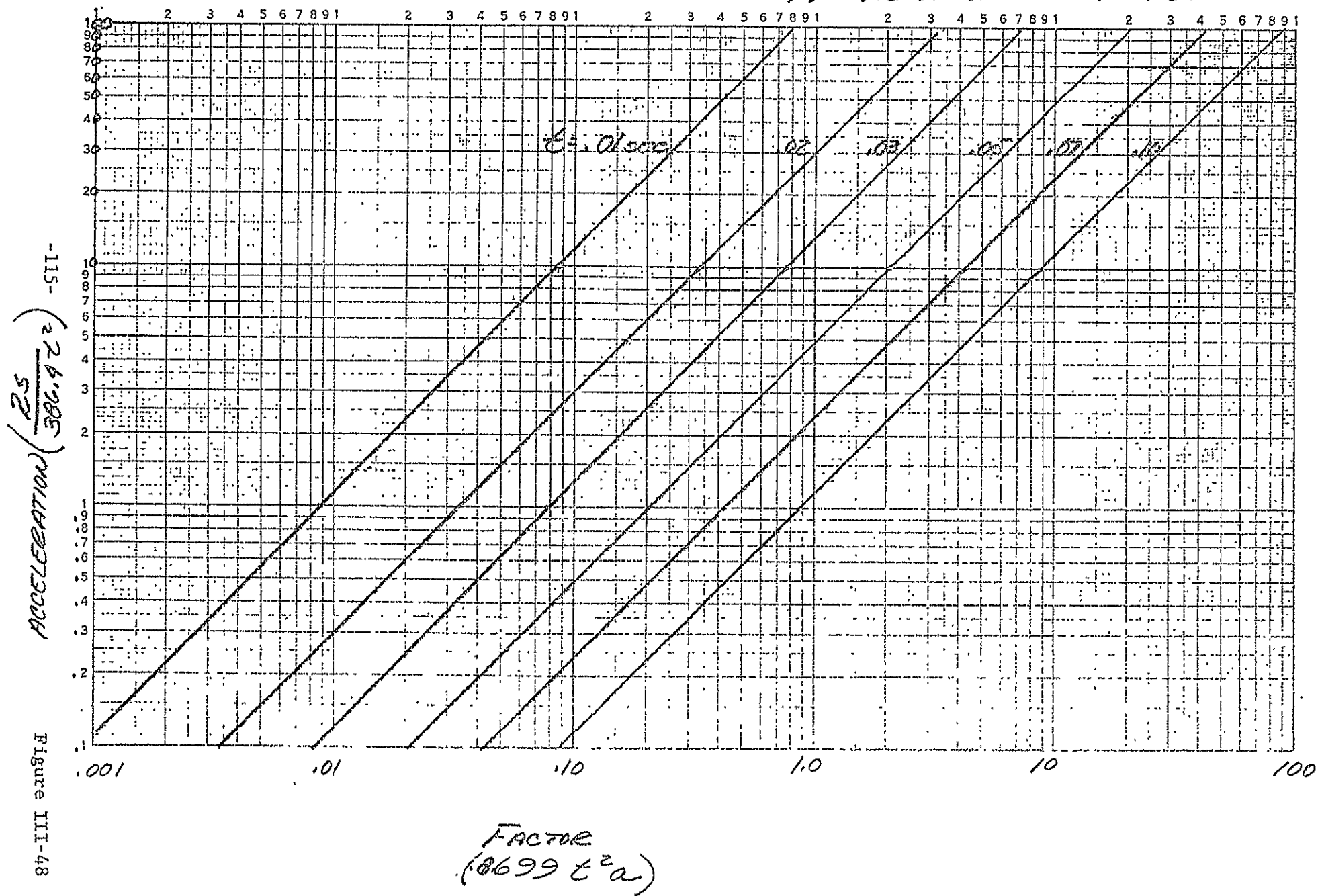
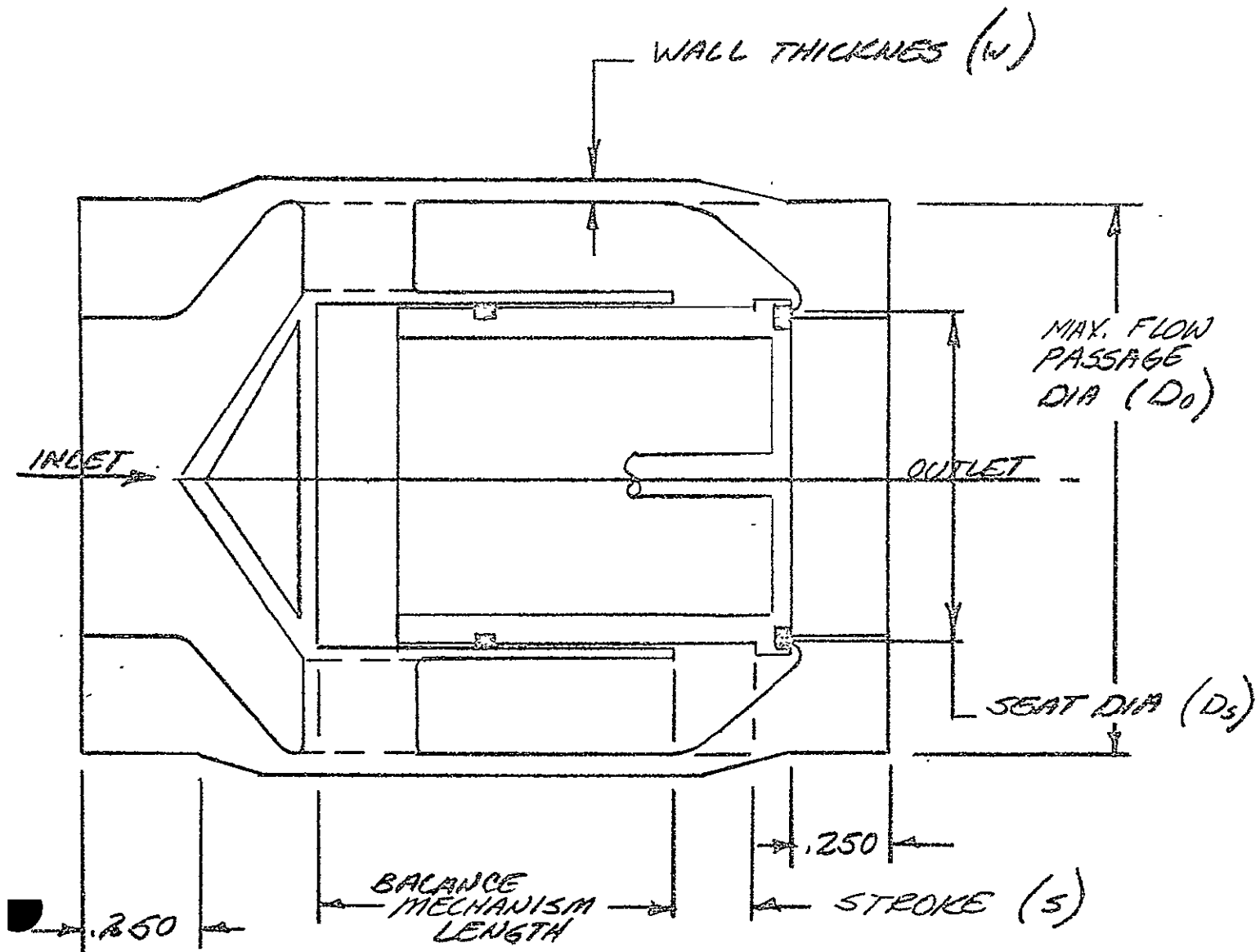


Figure III-48

LINEAR MOTION SHUT-OFF DEVICE - PHYSICAL CHARACTERISTIC
SCHEMATIC



Having established the inside diameter of the pressure vessel (D_o), the wall thickness (w) is determined by the operating pressure range, and stress limits of the material of construction. Based on hoop stress at 2.5 times the inlet pressure, wall thicknesses (w) were determined for several flow factors ($C_d A_t$) and inlet pressures (P_1) at the extremes of the range of interest. This analysis showed that required wall thicknesses ranged from .003 to .030 inches. In the interest of conservative prediction, a constant wall thickness of the pressure vessel of .075 inches was selected. Pressure vessel O.D. hence maximum envelope diameter thus became a function of seat diameter (D_s) or flow factor ($C_d A_t$) only.

$$\text{Max. Envelope Diameter} = \left(\sqrt{2.1 D_s^2} \right) + .15 = \left(\sqrt{2.1 \frac{C_d A_t}{\frac{\pi}{4} C_d}} \right) + .15 \quad (50)$$

Overall length was broken down into elements related to seat diameter (D_s) or flow factor ($C_d A_t$) as shown in Figure III-49. A length to diameter ratio for the balance mechanism of 1.25 was assumed adequate to assure alignment. An allowance of .250 inches for the inlet and outlet interfaces was selected and additional length allowed for the moving element stroke. The .250 inch inlet and outlet interface allowances were judged to be sufficiently generous to accommodate the contribution of the seat seal holder and inlet diffuser length and the overall envelope length relationship was defined as:

$$\ell = .50 + 1.3125 D_s = .50 + 1.3125 \sqrt{\frac{C_d A_t}{\frac{\pi}{4} C_d}} \quad (51)$$

To predict element weight, the configuration geometry was simplified. The pressure vessel was considered to be a tubular section of length, ℓ , (Equation 51) outside diameter of the maximum envelope diameter (Eq. 50) and inside diameter D_o (Eq. 49). The internal elements volume of material were considered to be two times the volume of the moving element. From prior analysis, the moving element mass, hence material volume, relationship to flow factor ($C_d A_t$) had been defined. The weight prediction thus becomes:

$$\text{WT.} = \rho \left[\frac{\pi}{4} \left(\sqrt{2.1 \frac{C_d A_t}{\frac{\pi}{4} C_d}} + .15 \right)^2 - \left(2.1 \frac{C_d A_t}{\frac{\pi}{4} C_d A_t} \right) \right] \times \left[.50 + 1.3125 \sqrt{\frac{C_d A_t}{\frac{\pi}{4} C_d}} \right] + 2 \rho \left(\frac{C_d A_t}{C_d} \right)^{1.66} \frac{1}{52.72} \quad (52)$$

where ρ = density of material of construction -- lb/in³.

Design curves presenting envelope dimensions and element weight as a function of flow factor ($C_d A_t$) are plotted in Figures III-50 and III-51, respectively. The weight prediction is based upon 100% use of the material of construction noted.

III-4D-2 Rotary Motion Shut-Off Device

The ball valve configuration is considered representative of the rotary motion shut-off device. Figure III-52 defines the basic configuration and notation employed in the physical characteristics analysis.

Assume the basic housing, exclusive of inlet and outlet fittings and shaft bearing housings is equivalent to a spherical shell surrounding the ball, with a 0.15 inch wall thickness and a 0.100 diametral clearance with the rotating element. End fittings are assumed to be 0.25 thick flanges of a diameter equal to the ball diameter. The bearing housings have 0.15 walls; have a sum total length of $0.1 \times$ ball diameter and have $ID = 0.1 D_B + 0.2$ and $OD = 0.1 D_B + 0.5$. The sum total of the volumes of the assumed elements of the housing is expressed as:

$$\begin{aligned} \text{Vol.} = & \frac{\pi}{6} \left[(D_B + 0.25)^3 - (D_B + 0.1)^3 \right] + 0.5 (D_B^2 - D_S^2) \frac{\pi}{4} \\ & + \left[(0.1 D_B + 0.5)^2 - (0.1 D_B + 0.2)^2 \right] \frac{\pi}{4} 0.1 D_B \end{aligned} \quad (53)$$

where D_B = ball diameter (equa. 10) - in.

D_S = seal diameter (equa. 9) - in.

Vol. - volume of material in housing - in.³

Multiplying the volume of equation 53 by the density of the material of construction yields housing weight. From Figure III-28 the moving mass may be obtained and the sum of housing weight and moving mass is equal to shut-off device weight. The envelope dimensions are obtained by properly summing the assumed relationships of the various elements which constitute the housing.

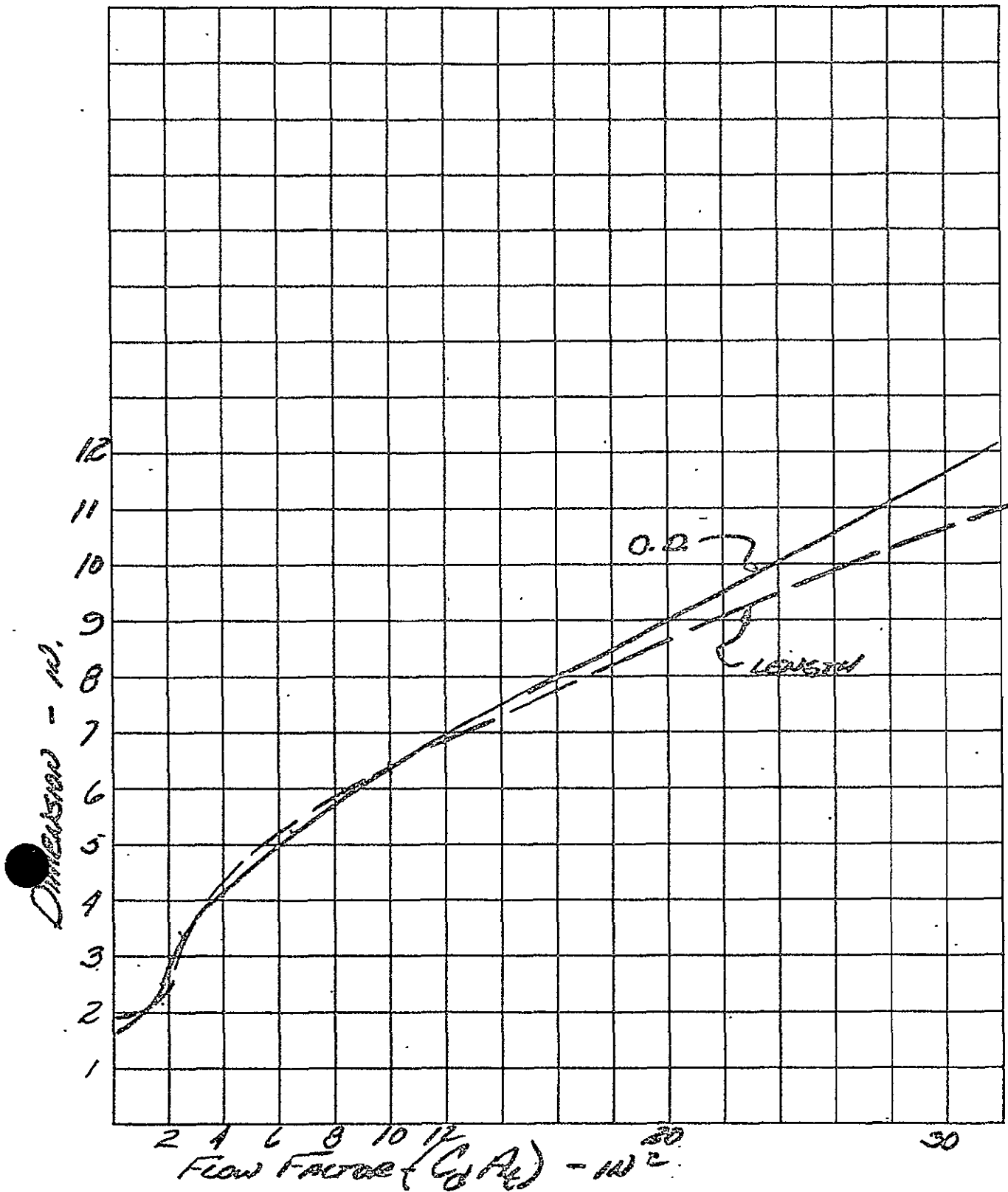
Design curves presenting envelope dimensions as a function of flow factor ($C_d A_t$) are presented in Figure III-53 for the assumed configuration. Figure III-54 plots rotary motion shut-off device weight as a function of flow factor ($C_d A_t$) for all aluminum construction and all stainless steel construction.

III-4D-3 Solenoid Actuator

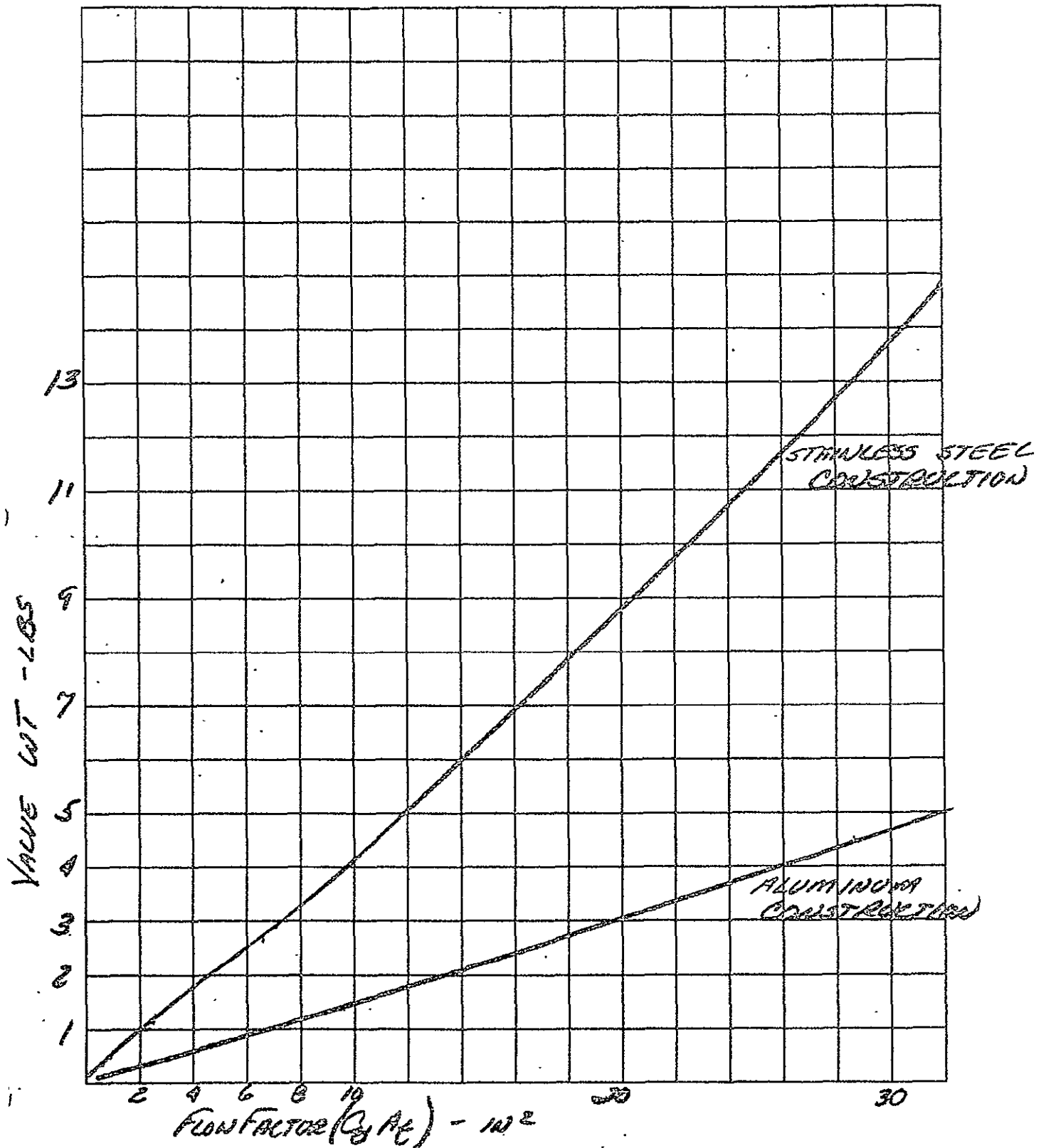
Nomenclature and configuration of the flat faced plunger type solenoid actuator are shown in Figure III-55. For reasonable proportions, the coil length (L_c) is assumed equal to the 2.25 times the armature diameter (D_A). From the performance analysis, the nominal coil resistance at 530°R and the number of turns in the coil have been derived. For alloy 30 wire, the resistance per foot of coil wire can be expressed as a function of the wire resistivity ($30 \frac{\text{ohms}}{\text{circular MIL-ft}}$) and also in terms of the coil dimensions and number of turns (N_c)

$$\frac{\text{ohms}}{\text{ft}} = \frac{30}{10^6 d^2} = \frac{\left(\frac{(32)}{W}\right)^2}{\frac{\pi}{12} N_c \frac{D_{C.O.} + (D_A + 0.13)}{2}} \quad (54)$$

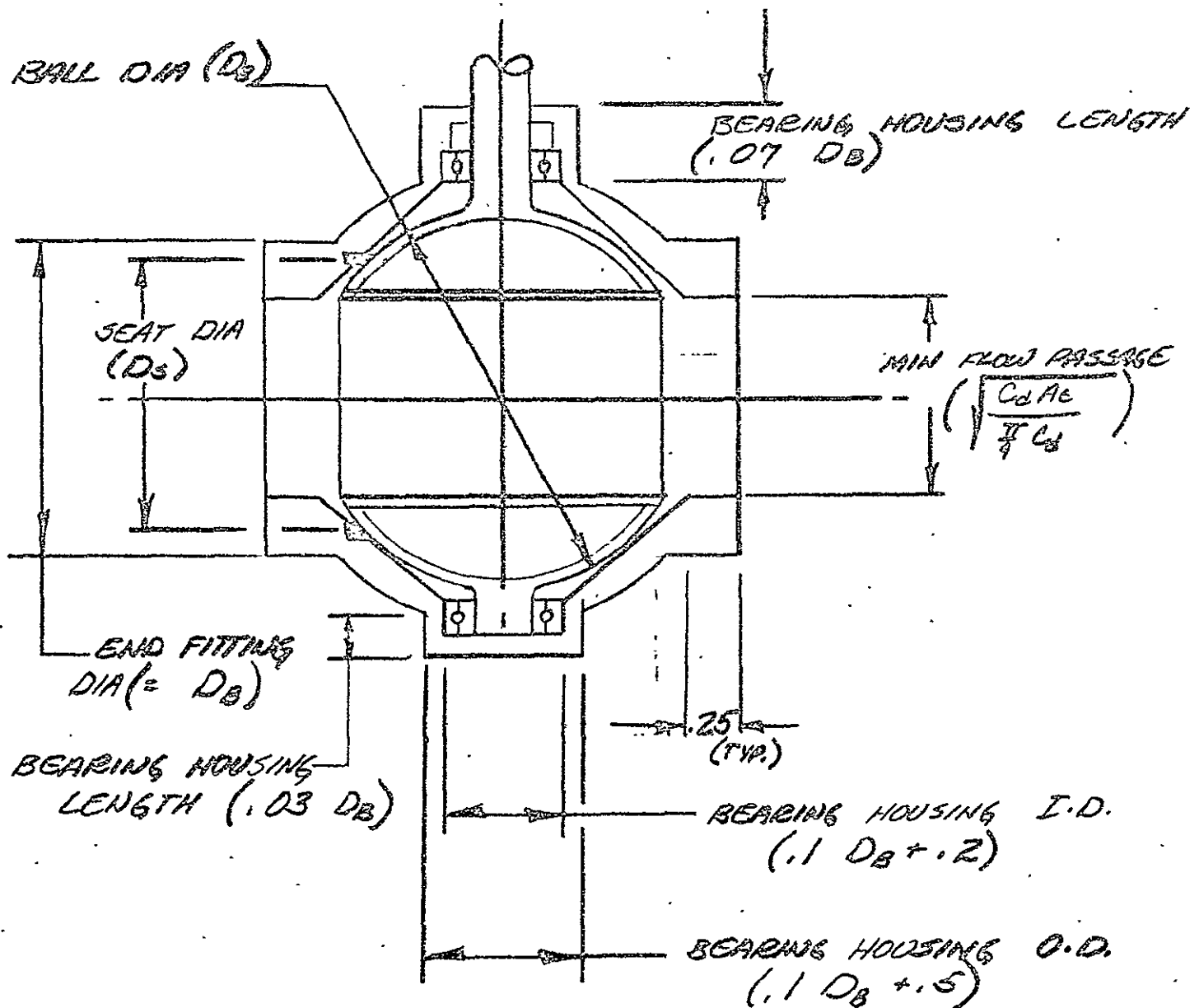
LINEAR MOTION SHUT-OFF DEVICE ENVELOPE DIMENSIONS VS FLOW FACTOR



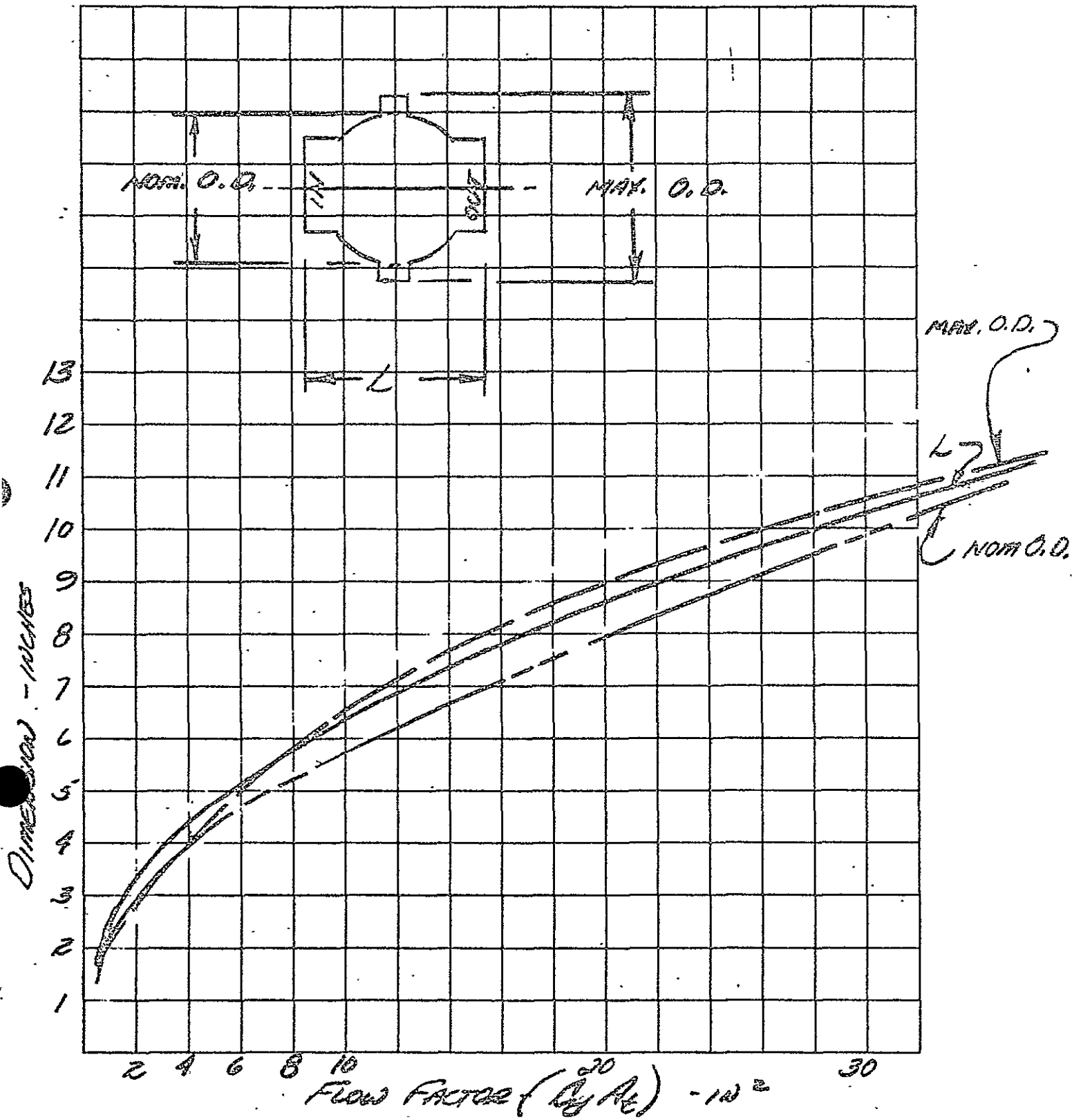
LINEAR MOTION SHOT-OFF DEVICE WEIGHT VS. FLOW FACTOR



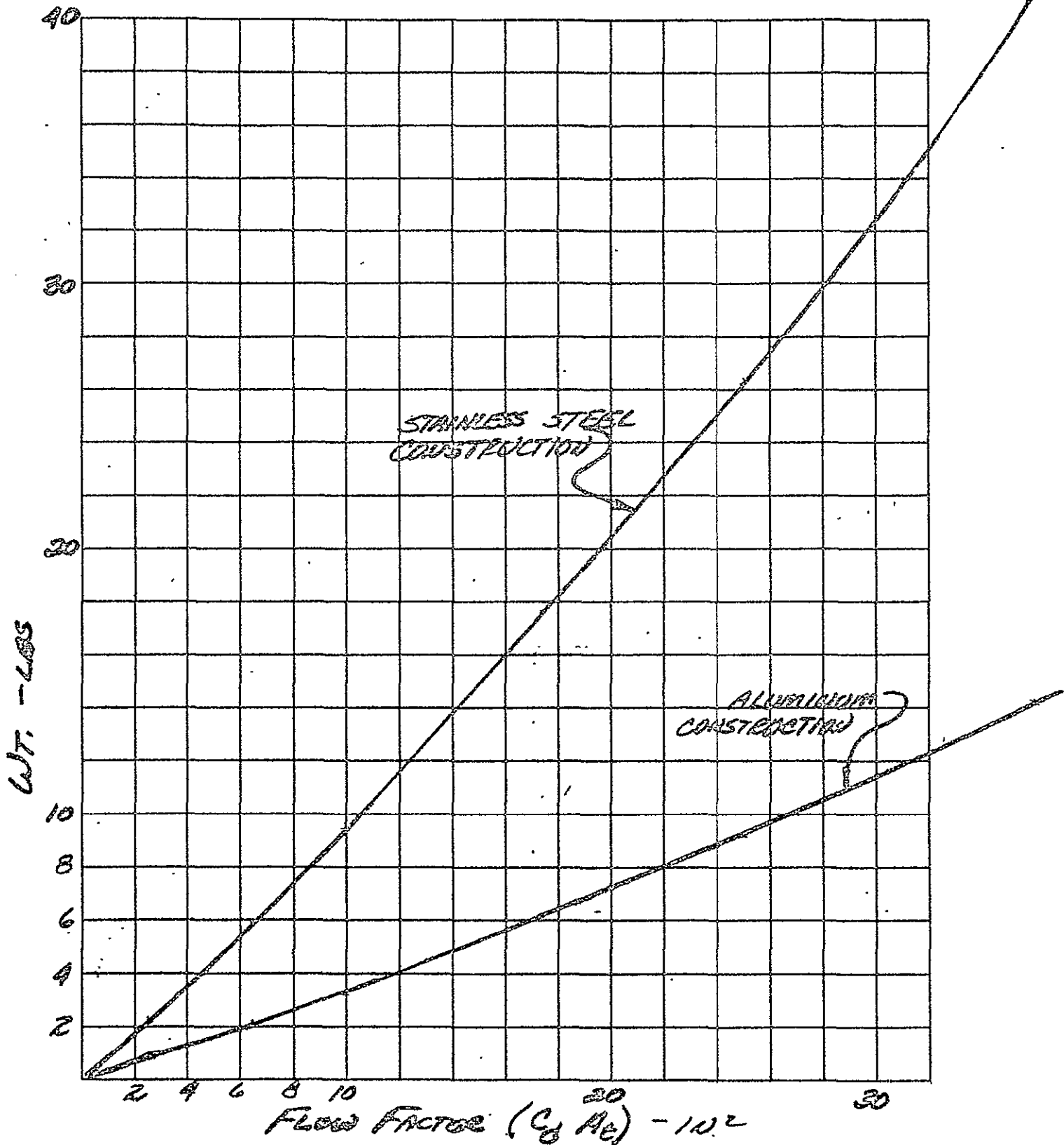
ROTARY MOTION SHUT-OFF DEVICE - PHYSICAL CHARACTERISTIC SCHEMATIC



ROTARY SHUT-OFF DEVICE ENVELOPE DIMENSIONS VS. FLOW FACTOR

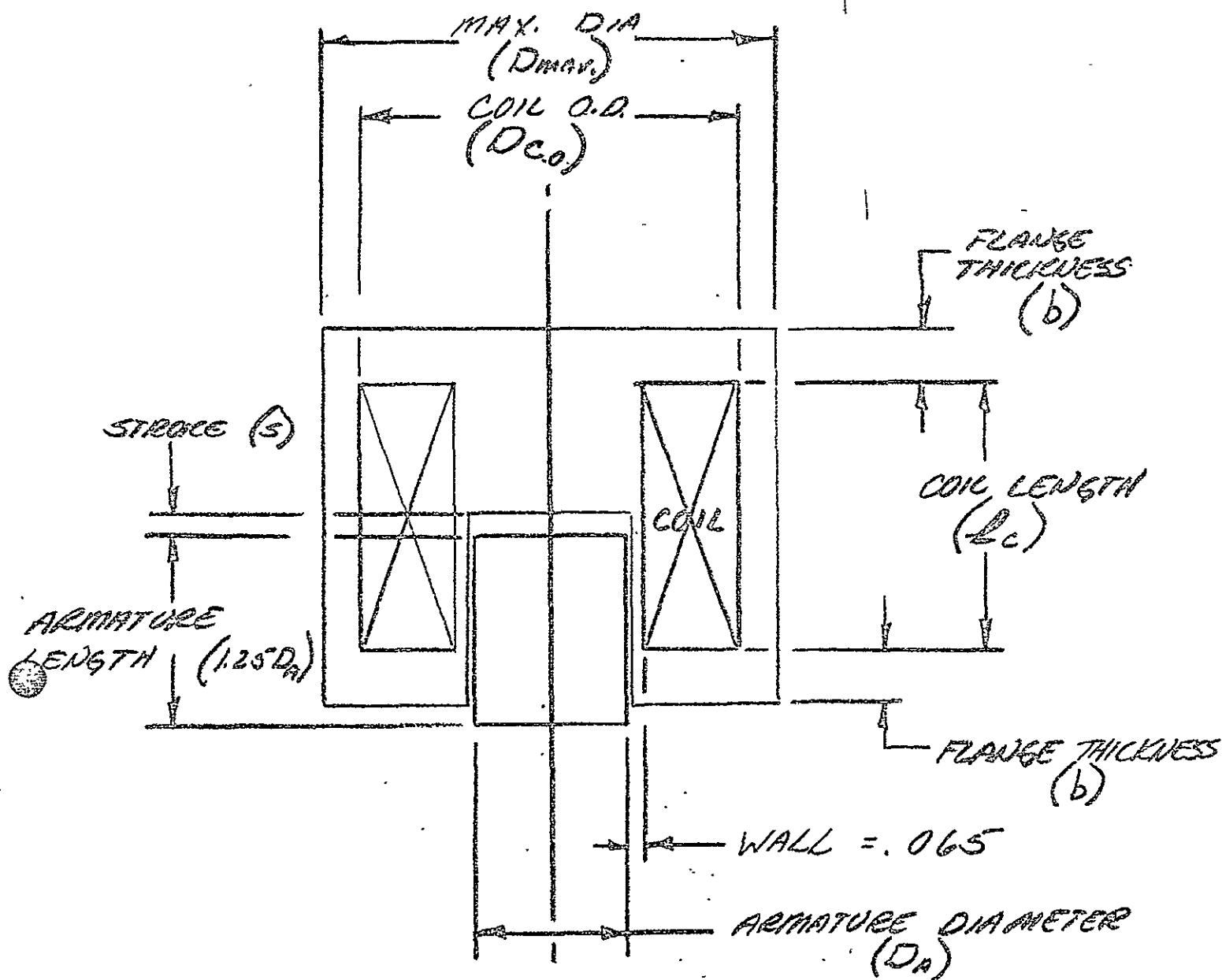


Rotary Motion Shut-off Device Weight vs Flow Factor



Solenoid Actuator - Physical Characteristics

SCHEMATIC



where d = wire diameter - inches
 W = max power at 530°F and 32 vdc - watts
 N_c = number of turns of wire on coil - turns
 $D_{c.o.}$ = coil outside diameter - inches
 $(D_A + 0.13)$ = coil inside diameter - inches

The cross-sectional area of the coil window can be expressed in similar terms:

$$N_c d^2 = \mathcal{L}_c \left(\frac{D_{c.o.}^2 - (D_A + 0.13)^2}{4} \right) \quad (55)$$

where $\mathcal{L}_c$ = length of coil = $2.25 D_A$ - inches

Combining equations 54 and 55, a relationship is established which can be solved for $D_{c.o.}$

$$\frac{D_{c.o.} - (D_A + 0.13)}{D_{c.o.} + (D_A + 0.13)} = \frac{30 \pi N_c^2 W}{12 \times 10^6 \times 32^2 \times \mathcal{L}_c} \quad (56)$$

Should a negative solution to $D_{c.o.}$ in equation (56) result, the feasibility of a coil of alloy 30 wire is non-existent. Prior analytical efforts have demonstrated that changing to a copper wire coil is not a practical alternative in such cases since the thermal coefficient of resistance is predominant. Where negative solutions for $D_{c.o.}$ result, the solenoid actuator is not a feasible alternate.

Since the cross-sectional area of the magnetic core must be constant ($A_m = \frac{\pi}{4} D_A^2$) the solenoid maximum diameter and flange thickness can be determined;

$$\frac{\pi}{4} D_A^2 = \frac{\pi}{4} (D_{max}^2 - D_{c.o.}^2) \quad (57)$$

where D_{max} = maximum diameter - inches

$$\text{and } \frac{\pi}{4} D_A^2 = \pi (D_A + 0.13) b \quad (58)$$

where b = flange thickness

Sufficient dimensional data is now available to determine solenoid envelope and weight - based upon corrosion resistant steel construction ($0.285 \frac{lb}{in^3}$).

For the sample data points within the range of interest, no positive solution for $D_{c.o.}$ was obtained, and solenoid actuators for these applications were not feasible as a direct acting actuator. The long strokes required (>0.100) were the predominant factor which ruled out this type of actuator.

III-4D-4 Pneumatic and Hydraulic Cylinder Actuators

Since both pneumatic and hydraulic cylinder actuators assumed identical configurations, the analytical method of establishing envelope and weight estimates are common. Cylinder wall thickness as a function of piston diameter, internal pressure and hoop stress were determined for the range and a wall thickness of 0.075 inches was selected as adequate and practical for aluminum or stainless steel construction. The configuration and nomenclature employed in the following analysis is shown in Figure III-56. The cylinder and piston material volume can be defined in terms of the piston diameter (D_p).

$$V = \frac{\pi}{16} (D_p)^3 + \frac{\pi}{4} \left[(D_p + 0.15)^2 - D_p^2 \right] \left[s + \frac{D_p}{4} + 0.47 \right] + \frac{\pi}{4} D_p^2 (0.47) \quad (59)$$

where V = material volume of cylinder and piston - in.³
 D_p = piston diameter - in.
 s = stroke - in.

The remaining element that constitutes material volume is the piston rod. Rod diameter is determined by the critical buckling of the rod in compression when 2 x the maximum load is applied. The analysis is based upon the rod being treated as a column with one end fixed and the other end free, and of length equal to the stroke (s). Rod diameter may be expressed as

$$d_r = \sqrt[4]{\frac{128 P_l D_p^2 s^2}{\pi^2 E}} \quad (60)$$

where d_r = rod diameter - inches
 P_l = maximum cylinder pressure - psia
 D_p = piston diameter - inches
 s = stroke - inches
 E = Young's modulus for material of construction - psi

Rod material volume then becomes;

$$\frac{\pi}{4} d_r^2 \left(s + \frac{D_p}{4} + 0.47 \right) \quad (61)$$

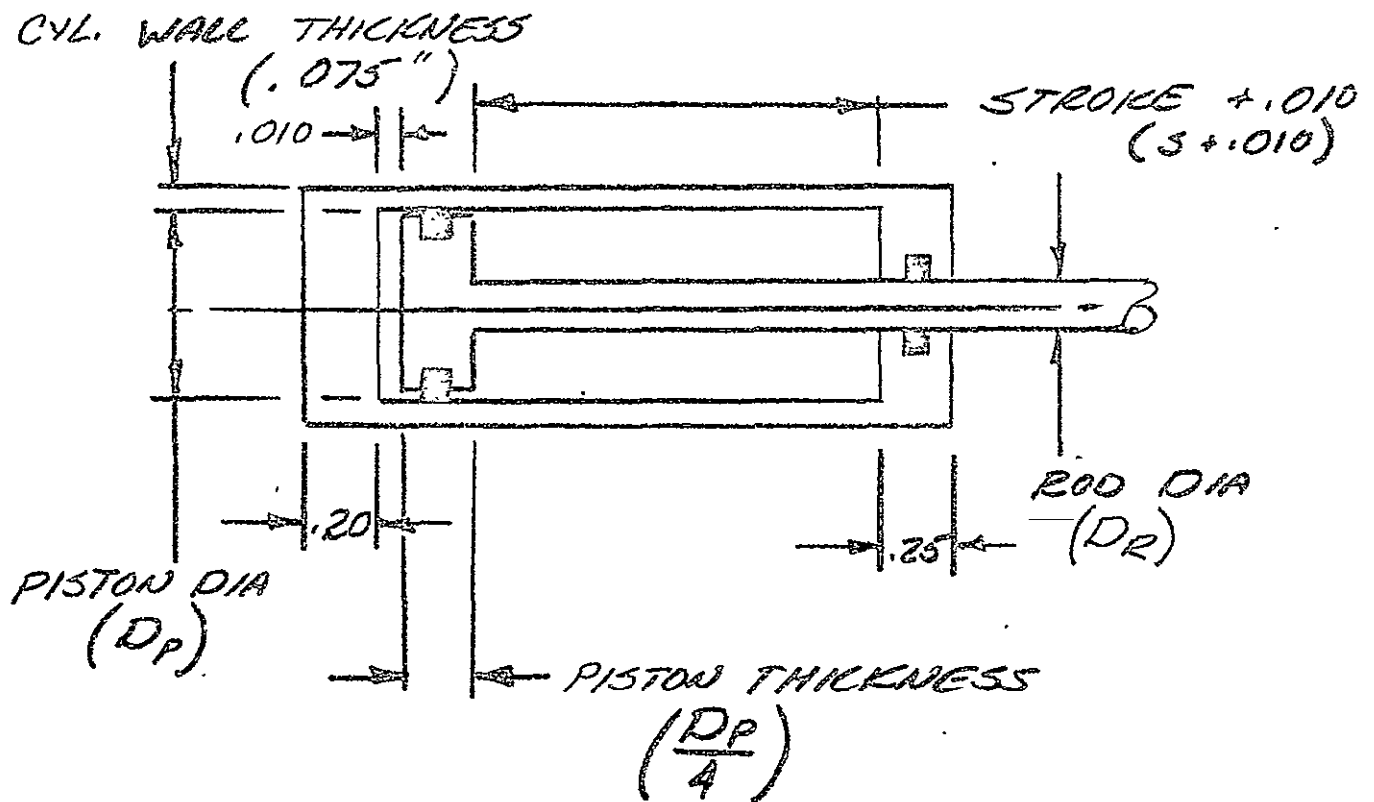
Summing equations 59 and 61 results in the calculated volume of material for the actuator configuration.

A design curve for predicting cylinder actuator envelope dimensions as a function of piston diameter and stroke is plotted in Figure III-57. Weight design curves as a function of piston diameter and material of construction are plotted in Figure III-58 for cylinder pressures up to 400 psia and Figure III-59 for cylinder pressures from 400 to 3000 psia.

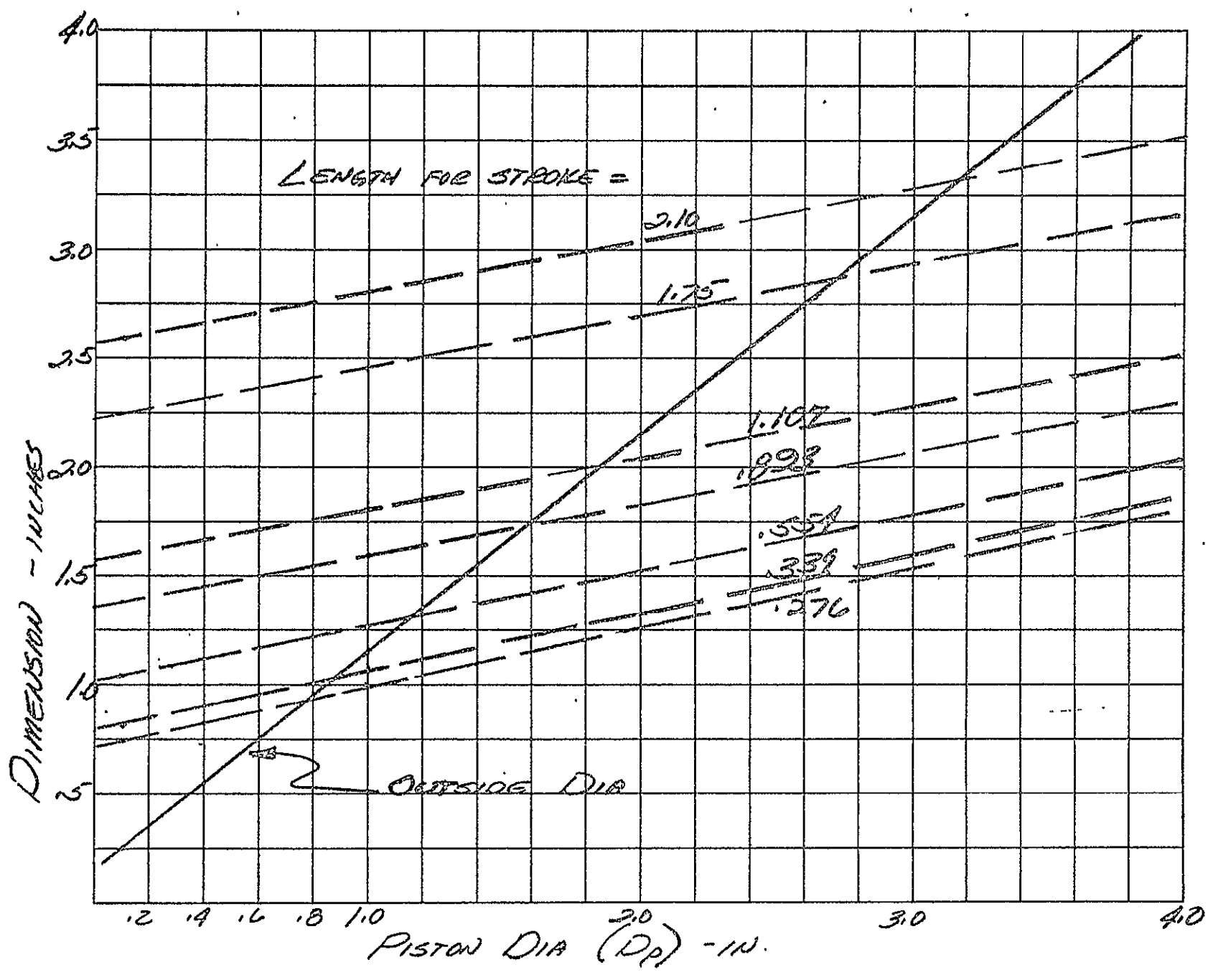
III-4 D-5 Torque Motor Actuator

The performance analysis program of paragraph III-4B-4 generated sufficient dimensional data to establish the actuator envelope and weight

HYDRAULIC OR PNEUMATIC CYLINDER ACTUATOR - PHYSICAL CHARACTERISTIC SCHEMATIC



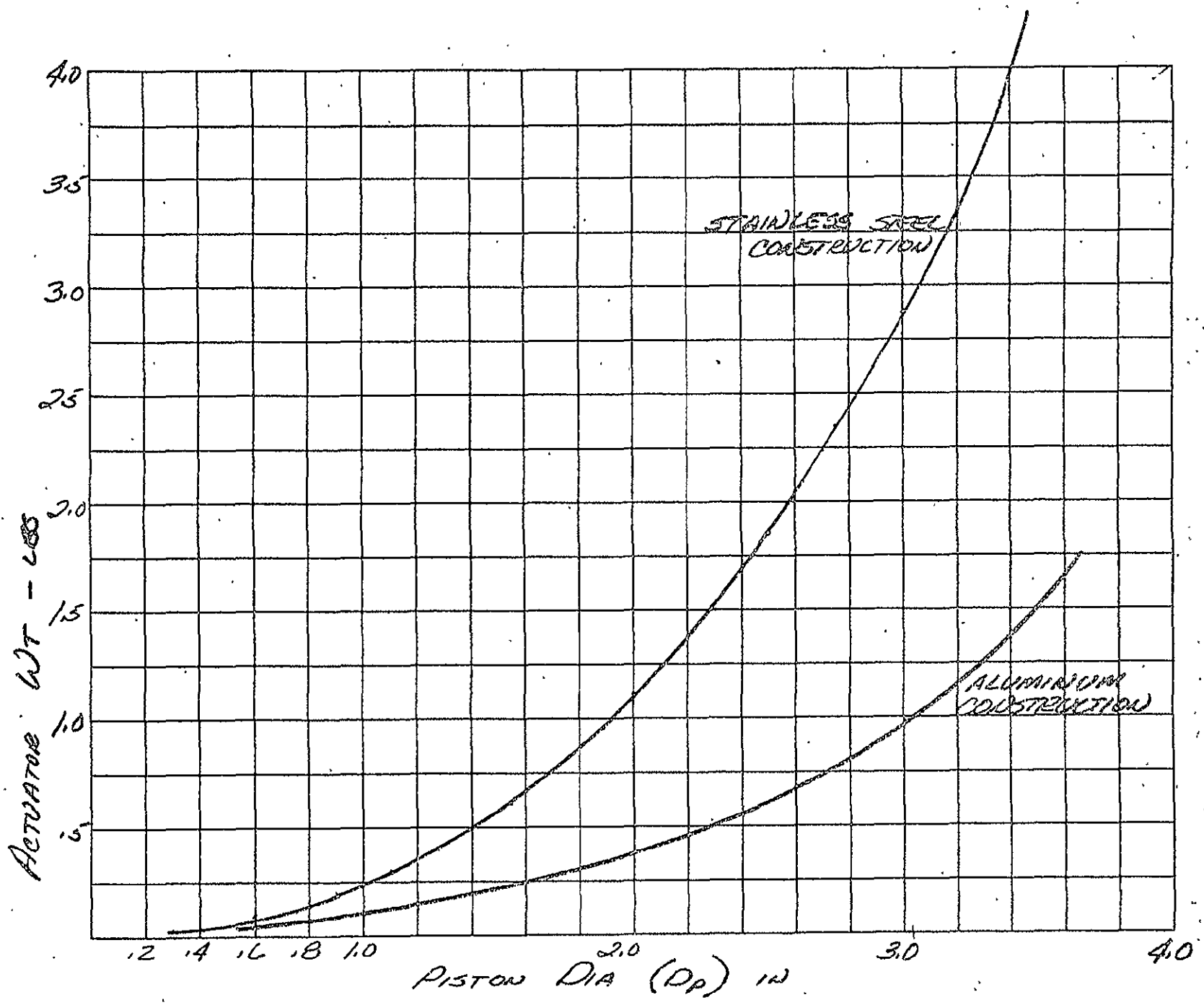
*Cylinders Accurate Envelope Dimensions
vs Piston Diameter*



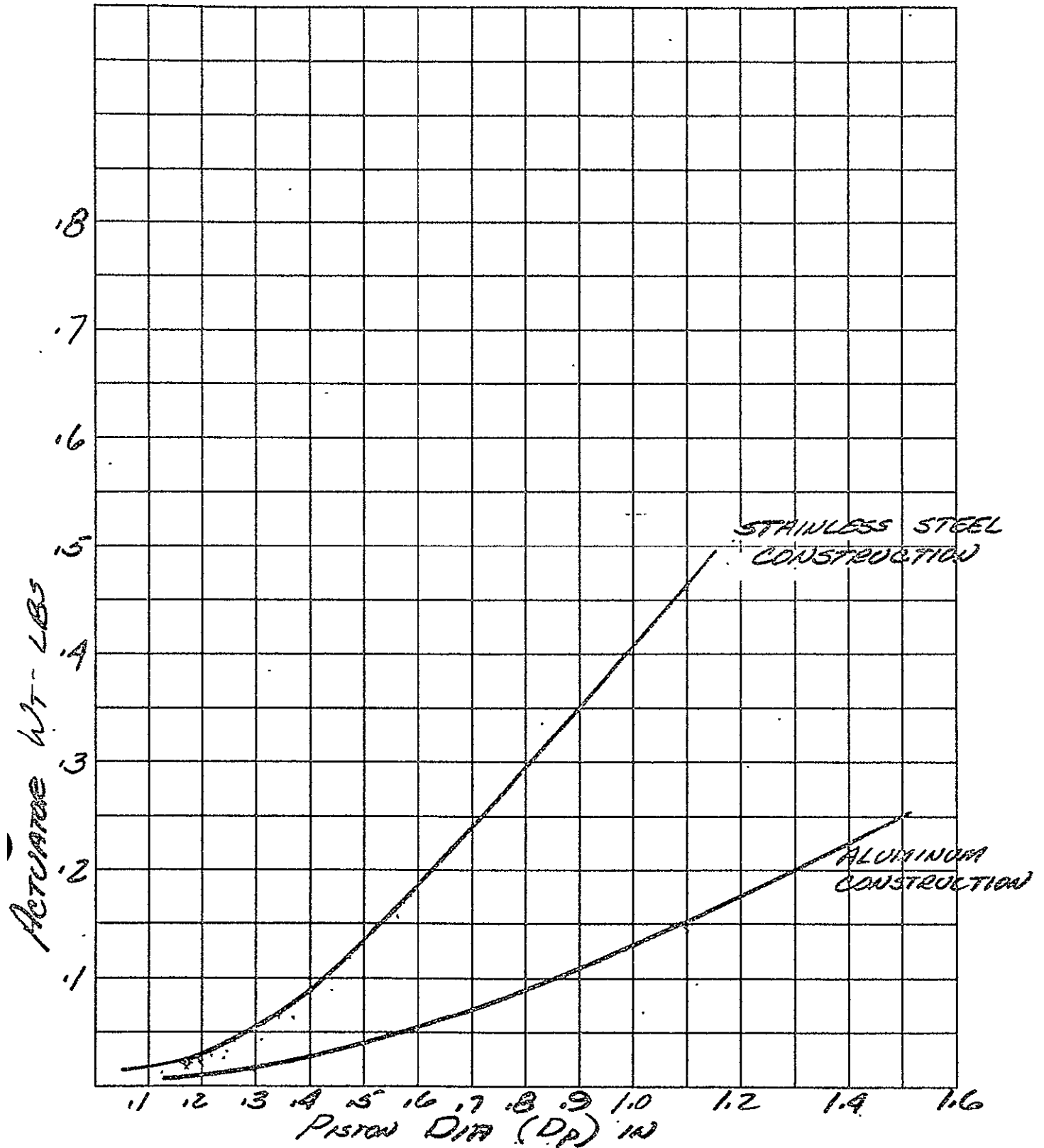
-128-

Figure III-57

*LINE PRESSURE OPERATED LINING CHAMBER WEIGHT
 VS PISTON DIAMETER*



AUXILIARY PRESS. OPERATED CYLINDER ACTUATOR PISTON DIA VS ACTUATOR WEIGHT



(Ref. Figure III-40). Torque motor maximum diameter is expressed as

$$D_m = D_A + 1.0 \quad (62)$$

where D_m = motor maximum diameter - inches
 D_A = armature diameter - inches

Torque motor length is determined directly from the performance analysis or read from the design curves (Figure III-41 through Figure III-46). Motor weight is calculated by applying appropriate material density factors to the available envelope and derived detail dimensions;

$$W_t = 0.283 \frac{\pi}{4} (\mathcal{L} - 0.15) \left[(D_A + 1.0)^2 - D_A^2 \right] + 0.21 \frac{\pi}{4} \mathcal{L} \left[D_A^2 - (0.8 D_A)^2 \right] \quad (63)$$

Design curves for motor envelope and weight as a function of motor length ($\mathcal{L}$) and maximum electrical power (W) are plotted in Figures III-60 and III-61.

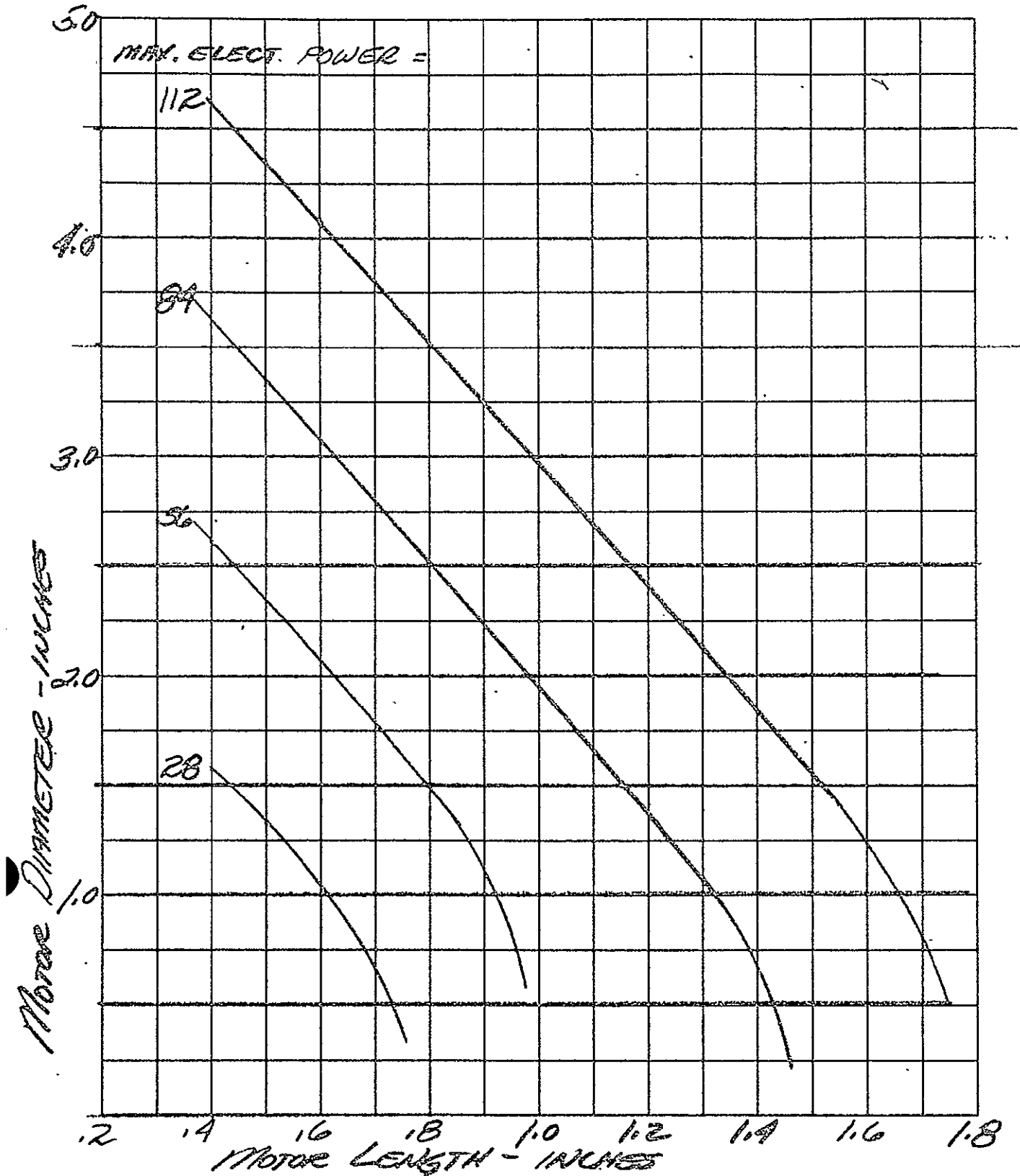
III-4E Design Layouts and Design Studies

Concurrent with the effort to develop the analytical program, a design effort was initiated. The initial objectives of this effort were document several unique design concepts which appeared feasible, establish a basis for developing a relationship between moving mass and some characteristic common to all valve designs, and to determine the validity of scaling factors to overall valve design. As the development of the analytical program progressed, the design effort objectives were modified to include verification of the detail design assumptions used in the analysis and to demonstrate the capacity of the analysis program to provide comprehensive design criteria which will expedite the design effort.

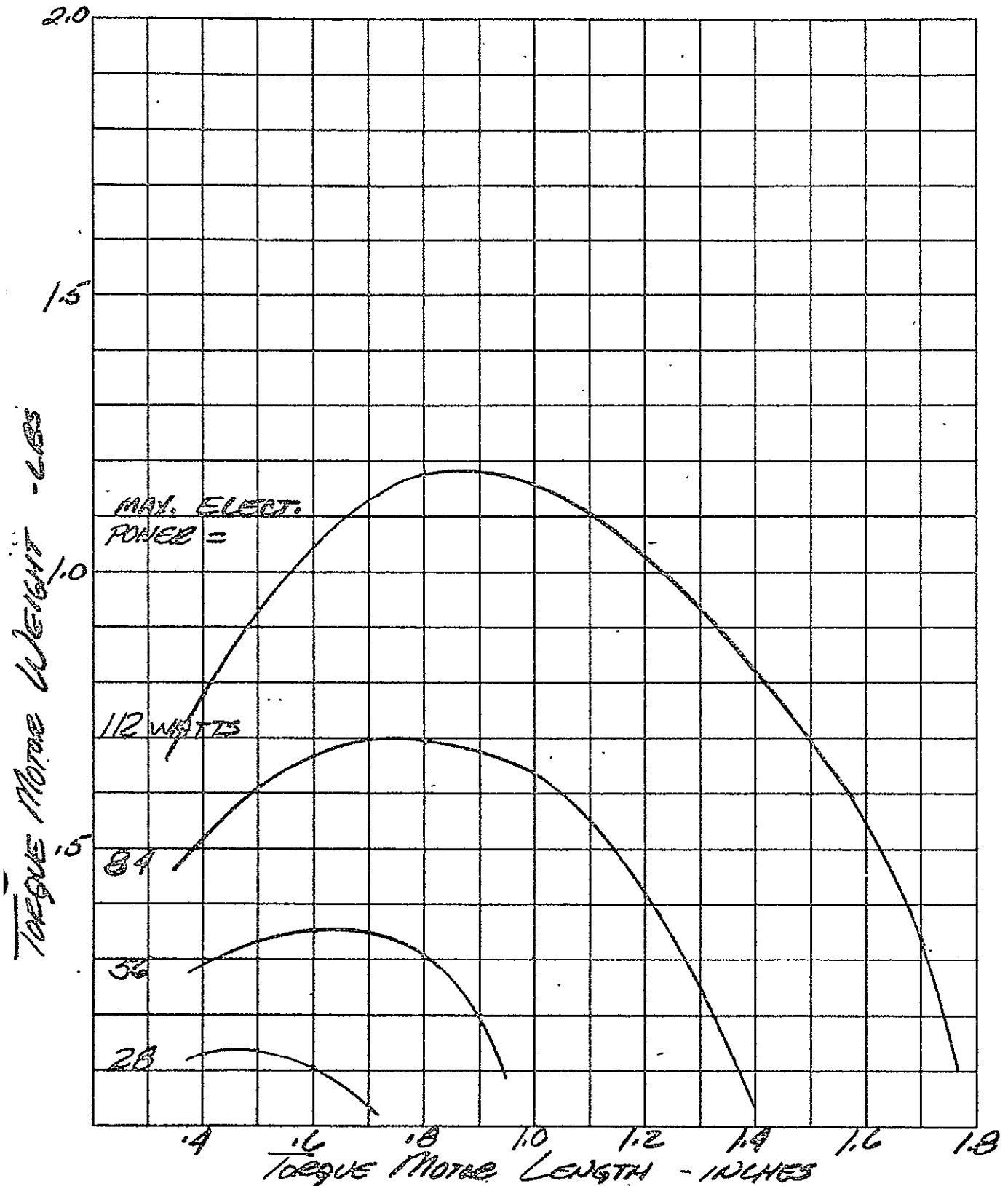
The conceptual designs of Figures III-62, III-63, and III-64, are representative of the initial design layout effort. The design criteria for these layouts was flow factor and inlet pressure only, as development of the analysis program had not progressed sufficiently at that time. The configurations were representative of concepts which would ultimately be consistent with the scope of the analysis and embodied features which offered solutions to anticipated problems of the application.

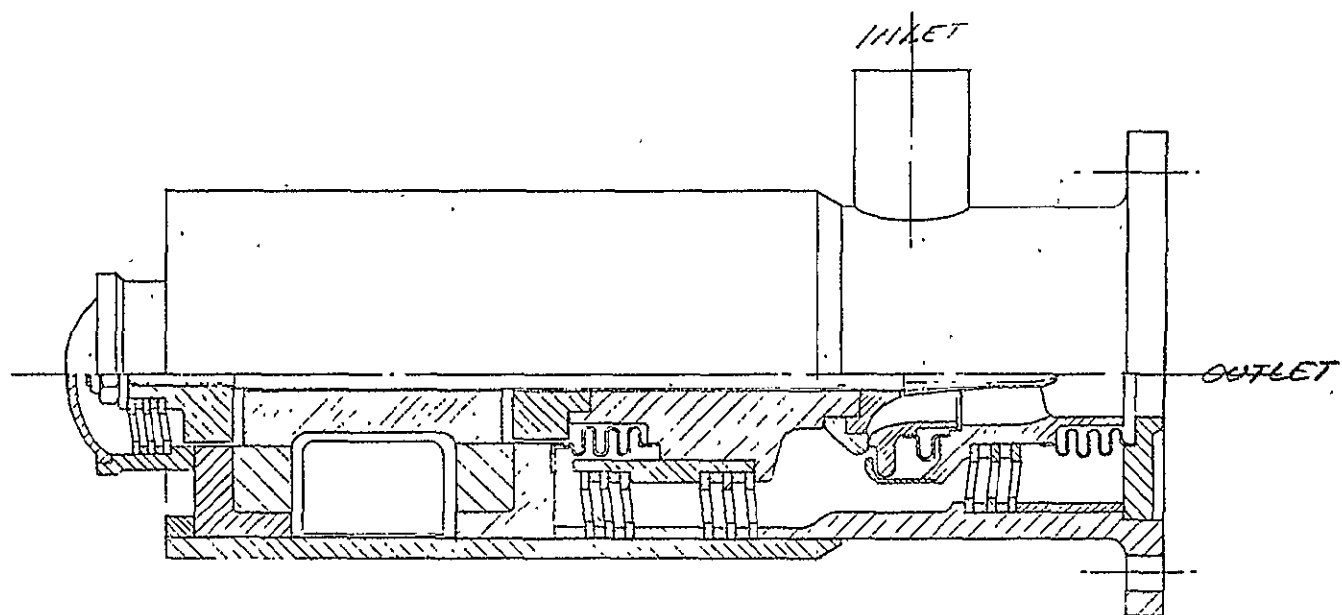
The double acting solenoid operated valve₂ of Figure III-62 was initially sized for a flow factor ($C_d A_t$) of 0.45 in² and an inlet pressure of 400 psia. This concept incorporates several unique features. The double acting configuration (both seat and poppet move) wherein the sum of the motion of the seat and the poppet results in the appropriate flow area across the valve seat allows the use of flexures to provide friction free absolute guidance of the moving elements. The type of flexure anticipated has been developed and used in TMC valve designs but is limited in stroke capability. The double action effectively doubles the range of applicability of flexure guides. Dividing the moving mass into two separate elements, magnetically linked, also proves to be advantageous considering the force stroke characteristic of the solenoid. A reduction in solenoid force requirement is achieved since each moving mass moves only through 1/2 of the total stroke (1/2 the acceleration). The reduced stroke

TORQUE MOTOR ENVELOPE DIMENSIONS FOR VARIOUS POWER LEVELS



TORQUE MOTOR WEIGHT VS LENGTH AT VARIOUS POWER LEVELS





DOUBLE ACTING, PRESSURE-SEALING VALVE
STARK SEAT, HIGH PRESS., SOLENOID OPERATED

of each element also favors metal bellows as the method of accomplishing dynamic sealing at the balance area and required seat preloads. Though the analysis program ultimately indicated that a solenoid actuator for this application was not feasible, the configuration provided a basis for developing a relationship to predict shut-off device moving mass and documented several unique design concepts applicable to other configurations.

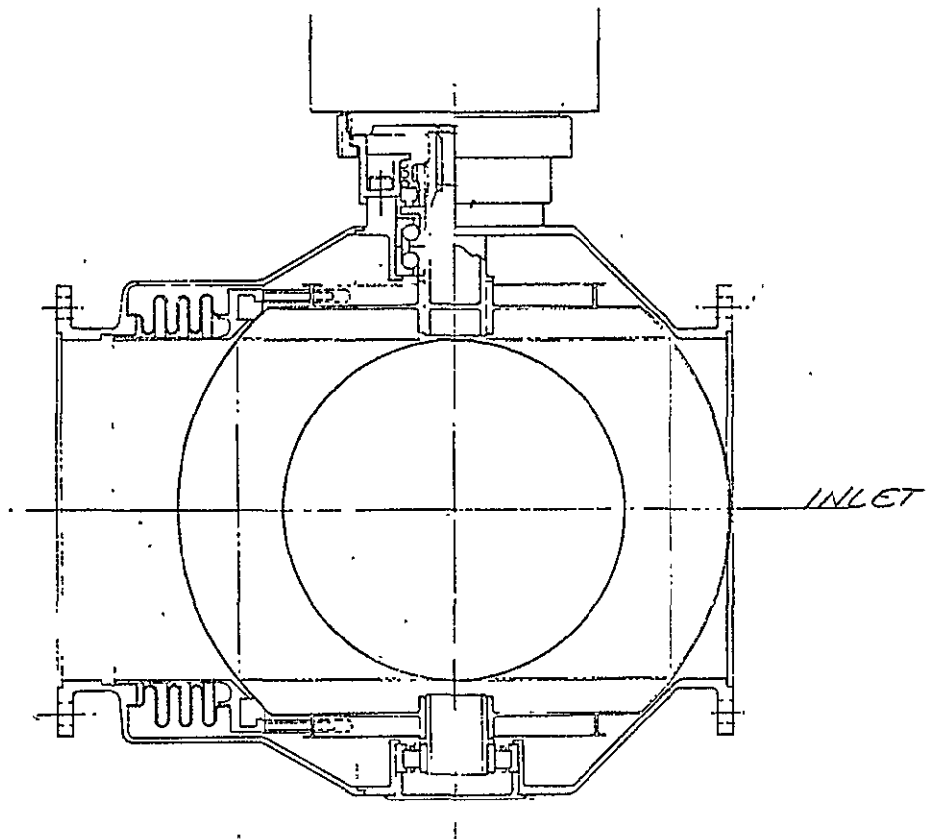
Figure III-63 defines a rotary motion shut-off (ball) and torque motor actuator configuration. Original design criteria was for a flow factor ($C_d A_t$) of 12 in.² and an inlet pressure of 30 psia. Of interest is the cam actuation to retract the seal on opening, thereby reducing the required torque to effect opening and minimizing scrubbing of the seal. The seal is bellows mounted such that the seal is pressure energized against the ball, in the closed position, in addition to the preload resulting from initial compression of the bellows. "Wetted" ball bearing mounting of the rotating element was selected and a spring loaded shaft seal was incorporated. Performance analysis ultimately invalidated a torque motor actuator for this particular application. The concept documents the basic design of a rotary motion shut-off device assumed in the analysis program and substantiates envelope and weight estimates.

The coaxial, pressure operated valve of Figure III-64 offers perhaps the highest density packaging concept. For the design criteria of 17 in.² flow factor ($C_d A_t$) and 30 psia inlet pressure, the configuration incorporates a balanced poppet with metal bellows dynamic seal. The actuator section operates off line pressure and also incorporates a metal bellows dynamic seal. Pressure control of the actuator is accomplished by a 3-way solenoid operated valve. The pneumatic cylinder actuator is a single acting cylinder in that it is pressurized to effect opening, while closing forces are exerted by the inherent spring characteristic of the metal bellows when the cylinder is vented. The analysis program substantiated the validity of this concept for this particular application, though specific dimensional relationships shown would have to be altered to agree with the data generated by the program.

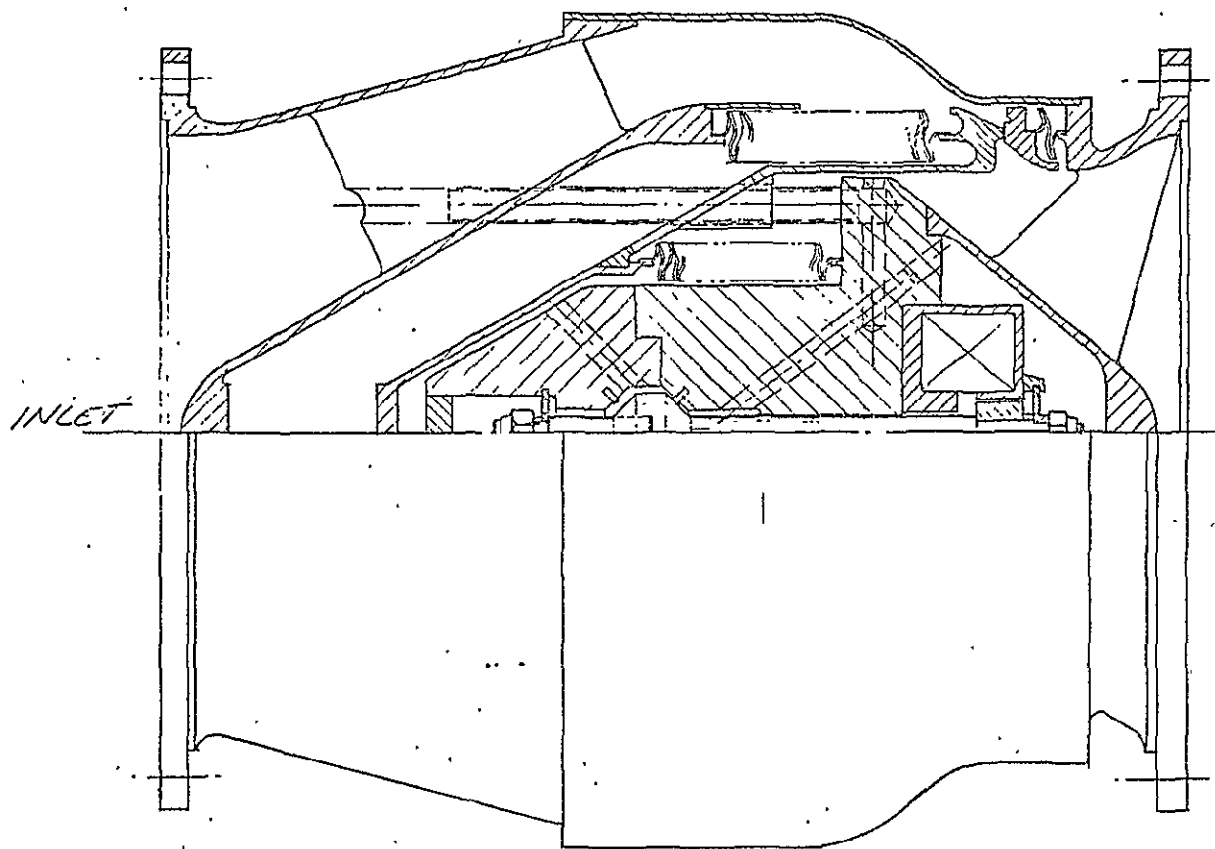
As the analysis program developed, the design effort was revised to utilize the analytical data generated, to the fullest extent. Design criteria became more explicit and the configurations were limited to those which were compatible with the analysis program. Figures III-65 through III-71 present design layouts of valve configurations which reflect design criteria generated by the analysis program.

The line pressure pneumatic operated poppet valve of Figure III-65 is a characteristic vented-pilot operated valve configuration. This concept represents a slight variation of the basic configuration assumed for the analysis program, but is readily analyzed with only slight modification to the analytical techniques described in the previous sections.

With the valve in the closed position, as shown, the seat is preloaded closed by the coil spring and inlet pressure acts to hold the poppet closed. Upon opening command (electrical signal) the solenoid causes the pilot valve poppet to move to a position where the bleed tube from the inlet manifold to the cavity within the main valve poppet is closed and the main poppet cavity is vented through the pilot valve. As cavity pressure decays, the differential pressure across the poppet creates opening forces which cause the poppet to move to the full open position. Upon termination of the command signal the pilot poppet moves to effect closing of the vent passage and opens the bleed tube to permit the poppet cavity to be pressurized by inlet pressure, negating



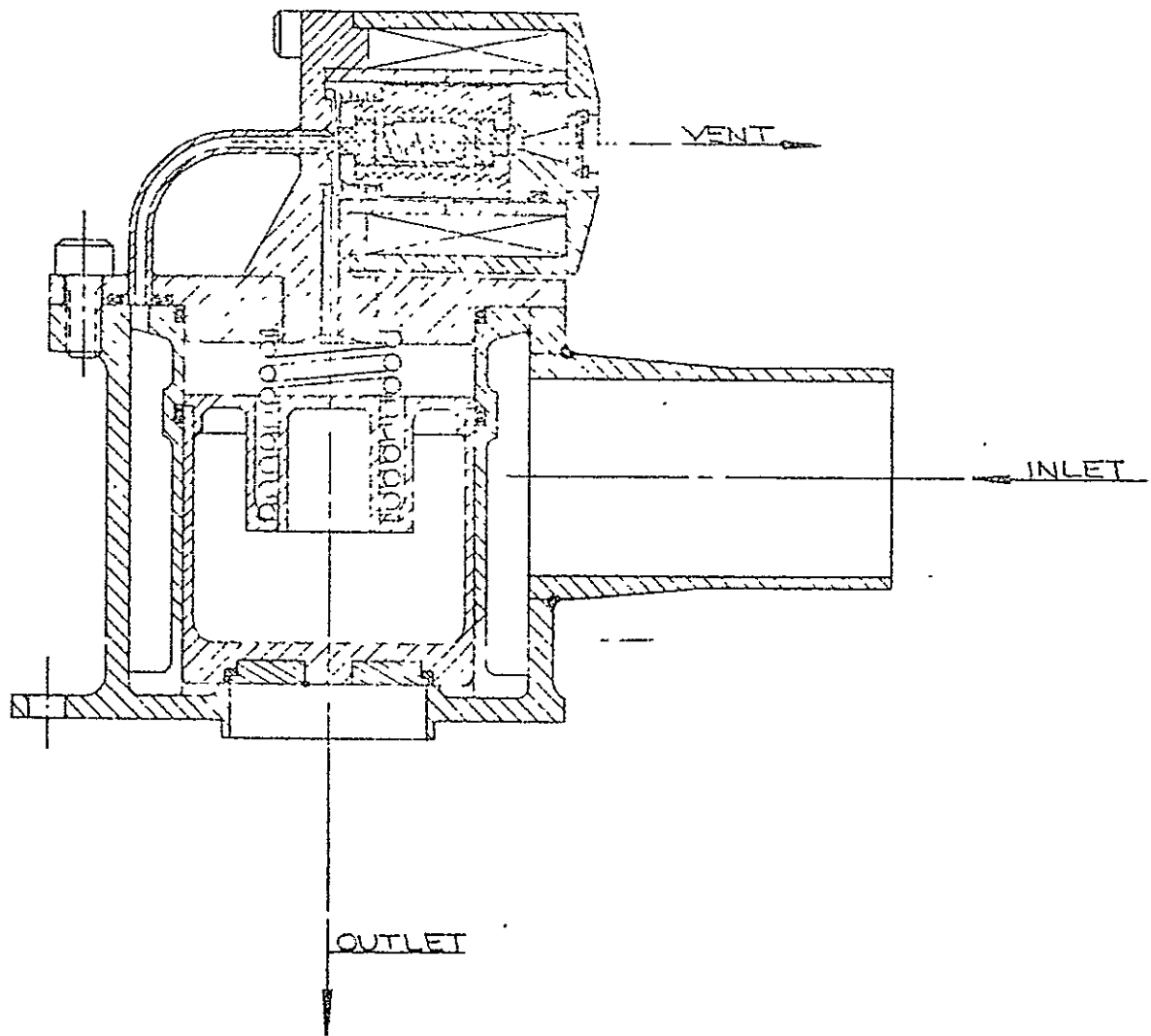
BALL VALVE
SOFT SEAT



COAXIAL POPPET VALVE
PRESSURE ACTUATED, LOW MANIFOLD, HARD SEAT.

REVISIONS				
ZONE	LTR	DESCRIPTION	DATE	APPROVED

QTY REQD	CODE IDENT	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR FINISH NO
LIST OF MATERIALS OR PARTS LIST							
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES XX XXX ANGLES ± .03 ± .010 ± 0° 30'			DRAWN <i>2</i> DATE <i>10-1-70</i> CHECKED <i>2</i>	<i>The Marquardt Company</i> CCL Aerospace Corporation, Van Nuys, California			
DO NOT SCALE THIS DRAWING SURFACE TEXTURE PER USAS 845.1-62 <i>✓</i> BREAK SHARP EDGES			APPROVED <i>11/1/70</i> APPROVED <i>11/1/70</i>	POPPET VALVE - 3 WAY - PILOT LINE PRESSURE - PNEUMATIC OPERATED			
NEXT ASSY USED ON APPLICATION			IDENTIFY PER MPS-800 CLASS		SIZE <i>D</i> CODE IDENT NO <i>86845</i> DRAWING NO <i>L4661</i>	SCALE <i>2/1</i> HT	



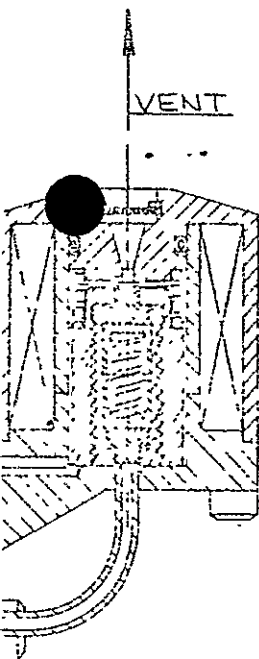
opening pressure forces until the spring moves the main poppet to the closed position. This configuration requires line pressure to effect operation and also exhibits a fail-safe characteristic in that the valve will close in the event of electrical failure or decrease in inlet pressure below a predetermined minimum pressure. Sizing of the valve was based upon a flow factor ($C_d A_t$) of 0.62 in.² and an inlet pressure of 400 psia.

Figure III-66 defines a line pressure pneumatic operated poppet valve which embodies all characteristics of the assumed configuration of the analysis program. Sized for a flow factor ($C_d A_t$) of 0.62 in.² and an inlet pressure (P_1) of 400 psia, valve actuation is accomplished by a double acting cylinder. The packaging concept is unique in that minimum height and width of the package is achieved and the seat seal is the only potential internal leak path with the valve in the closed position. Three-way solenoid operated pilot valves are used to control the pneumatic cylinder actuator. With slight modification, the pilot valves could be combined into a single 5-way valve. Valve seat preloads are exerted by coil springs to assure a fail/safe design.

A rotary motion shut-off, torque motor actuated valve configuration is shown in Figure III-67. Designed for a flow factor ($C_d A_t$) of 0.94 in.² and 60 psia, the configuration is totally consistent with the analysis program and embodies several unique features. Spherical surfaces of the rotating ball are minimal by virtue of the large diameter trunions. Minimum weight is achieved by the thin shell design and assembly is simplified by the design approach. As shown, the bearings are dry, being isolated from the flow media by shaft seals, however, the design is such that leakage of the shaft seals is tolerable since the total housing is a pressure vessel capable of being welded or bolted to contain the media. The valve seal is located at the inlet side of the rotating element, at some minor increase in dribble volume, but minimizing potential leak paths when the valve is in the closed position. The seat is a preloaded soft seal configuration employing metal bellows as a spring loading element and a positive dynamic seal to allow the seal to "float" to achieve the seat seal. As the valve is opened, a cam profile on the ball trunion mechanically retracts the seal to minimize scrubbing and reduce torque requirements to effect rapid opening.

For a flow factor ($C_d A_t$) of 10 in.² and a valve inlet pressure of 30 psi, a rotary motion auxiliary pressure pneumatic cylinder actuated valve design is shown in Figure III-68. Actuator sizing is based upon the availability of a 1500 psia auxiliary pressure source.

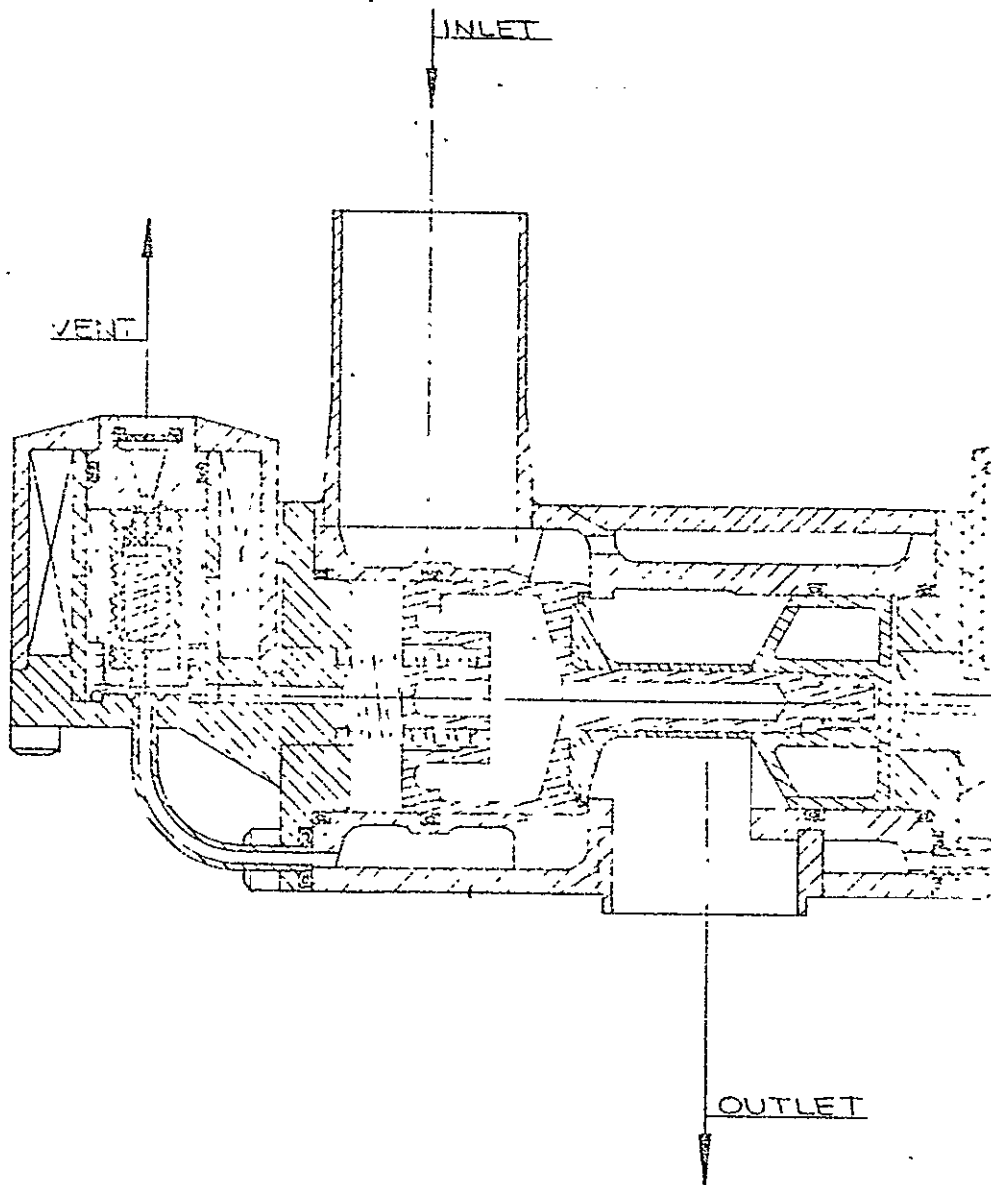
The rotating element is a trunion mounted thin wall spherical element. Shaft seals are eliminated by utilizing the cylinder piston seals to contain line pressure. The actuator is a double acting cylinder with actuator force output being converted to torque by a rack and pinion drive. Actuator pressure control is performed by three-way solenoid valves. As in the previous ball-valve design, cam action retraction of the seat seal is incorporated to minimize seat scrubbing. Metal bellows mounting of the seat seal is employed to achieve pressure energizing of the seal and allow "float" to enhance sealing capability with minimum preload.



REVISIONS				
ZONE	LTR	DESCRIPTION	DATE	APPROVED

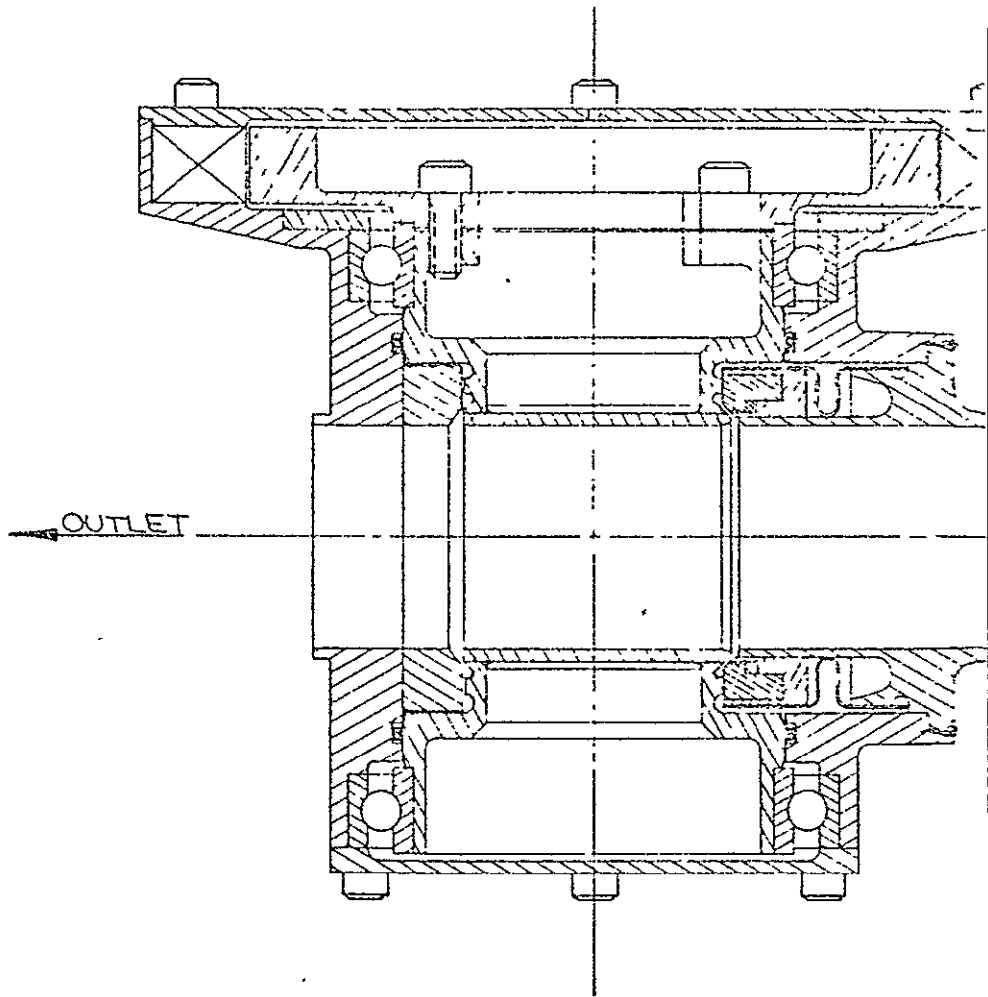
FOLDOUT FRAME

LAYOUT	QTY REQD	CODE IDENT	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL OR NOTE	DATE	ITEM OR NO
LIST OF MATERIALS OR PARTS LIST								
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES				DRAWN: <i>Qu...</i> DATE: <i>12-2-70</i>		THE Marquardt		
TOLERANCES				CHECKED: <i>Qu...</i>		ALL Dimensions Contained in this Drawing		
XX ± .03				XX ± .010		ANGLES ± 0° 30'		
DO NOT SCALE THIS DRAWING				APPROVED: <i>Qu...</i>		POPPET VALVE-DUAL		
SURFACE TEXTURE PER USAS B46.1-62 ✓				APPROVED: <i>Qu...</i>		PILOTED LINE PRESS RE		
BREAK SHARP EDGES				APPROVED: <i>Qu...</i>		PNEUMATIC OPERATED		
NEXT ASSY USED ON APPLICATION				IDENTIFY PEP MPS-803 CLAS		SIZE CODE IDENT NO DRAWING NO		
						D 86845 L 4000		
						SCALE 2/1		



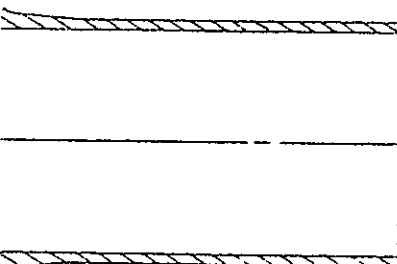
FOLDOUT FRAME

2



FOLDOUT FRAME

ZONE	TH	DESCRIPTION	DATE	APPROVED



INLET

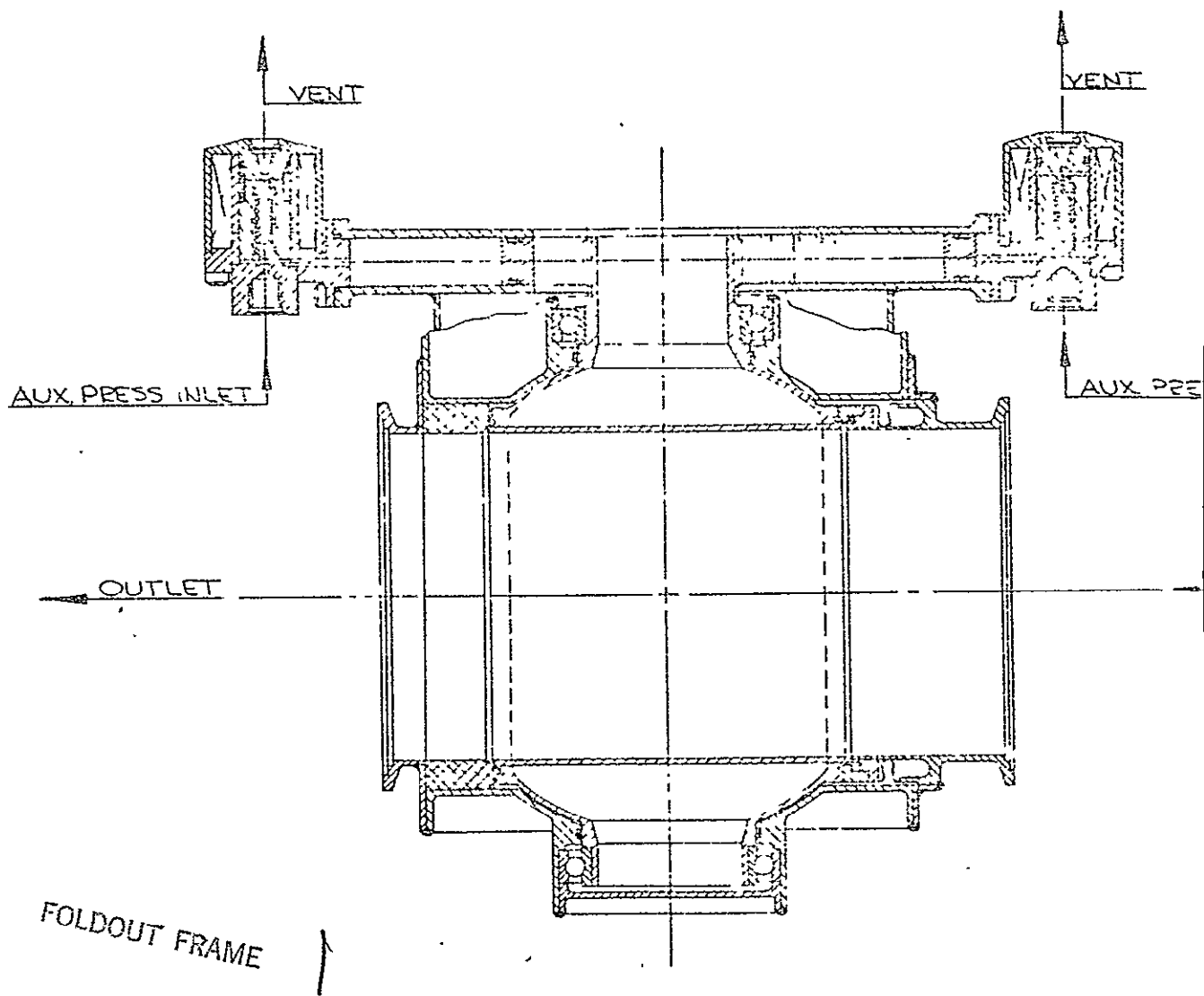
FOI DOUT FRAME

2

QTY	CODE	PART OR	NOVENCATURE OR	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR
REDO	IDENT	IDENTIFYING NO	DESCRIPTION				FIND NO
LAYOUT							
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES							
XX	XXX	ANGLES					
± .03	± .010	± 0° 30'					
DO NOT SCALE THIS DRAWING							
SURFACE TEXTURE PER USAS B46.1-62							
BREAK SHARP EDGES							
IDENTIFY PER MPS-800 CLASS.							
NEXT ASSY USED ON APPLICATION							
LIST OF MATERIALS OR PARTS LIST							
DRAWN CHECKED APPROVED APPROVED <i>[Signature]</i> <i>[Signature]</i> <i>[Signature]</i>							
<i>[Signature]</i> <i>[Signature]</i> <i>[Signature]</i> <i>[Signature]</i>							
Murguade CCI Aerospace Corporation, Van Nuys, California BALL VALVE-TORQUE MOTOR OPERATED							
SIZE D 86845 EDGE IDENT NO L 4663							
SCALE 2/1 WT							
SHEET 1 OF 1 DRAWING NO.							

11-11-67

THE END OF THE



REVISIONS				
ZONE	LTR	DESCRIPTION	DATE	APPROVED

D

C

B

A

PRESS INLET

INLET

FOLDOUT FRAME 2

QTY REQD	CODE IDEAT	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR FIND NO
LAYOUT							
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES							
TOLERANCES							
XX	XXX	ANGLES					
± .03	± .010	± 0° 30'					
DO NOT SCALE THIS DRAWING							
SURFACE TEXTURE PER USAS B46.1-2 ✓							
BREAK SHARP EDGES							
IDENTIFY PER MPS-800 CLASS							
DRAWN CHECKED APPROVED APPROVED DATE <i>Charles J. J.</i> <i>J. L. J.</i> <i>Marquardt</i> CEL Aerospace Corporation, Van Nuys, California							
BALL VALVE-AUX. PRESS. PNEUMATIC OPERATED							
SIZE CODE IDENT NO DRAWING NO.							
D 86845 L 4664							
SCALE 1/1 WT. SHEET 1 OF 1							
NEXT ASSY USED ON APPLICATION							

2-

An auxiliary pressure pneumatic cylinder actuator is employed to operate a linear motion shut-off in the valve design shown in Figure III-69. The poppet is totally balanced in all positions with seat preload exerted by the coil spring, which also assures fail/safe operation. Actuator control, by the 3-way solenoid valves provides rapid response opening and closing.

The configuration of Figure III-70 utilizes a torque motor actuator to drive a linear motion shut-off through a ball-screw linkage. By the unique employment of the ball-screw the poppet rotates 90° as it is retracted. This type of motion imparts a self-cleaning capability or high dirt tolerance to the valve seat seal. The poppet is fully balanced to minimize operating force requirements. The ball screw drive mechanism provides greater than 90% efficiency power transmission as well as imparting the unusual poppet motion. Design criteria for the configurations were determined by the analysis program for a flow factor ($C_d A_t$) of 0.31 in.^2 and an inlet pressure of 400 psia.

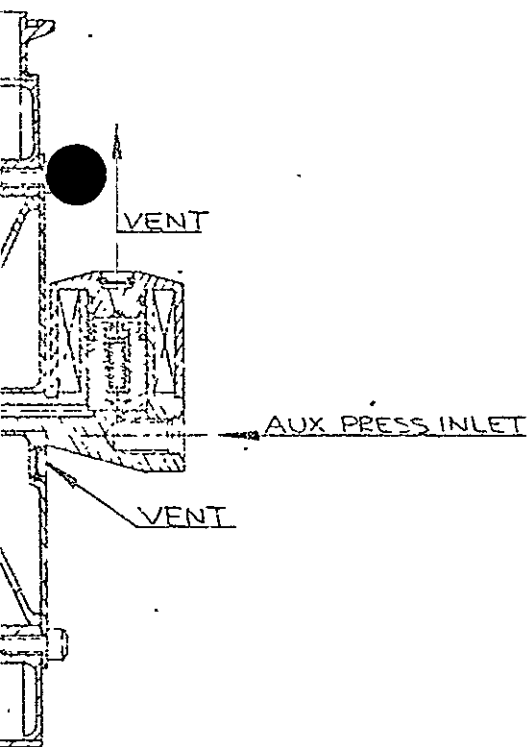
Hydraulic cylinder actuation of a linear motion shut-off device is depicted by the design of Figure III-71. For the design criteria of a flow factor ($C_d A_t$) of 3.3 in.^2 and an inlet pressure (P_1) of 30 psia, metal bellows were selected for the dynamic seals in the propellant wetted volumes. Hydraulic cylinder pressure control is performed by 3-way solenoid operated valves. Valve seat preload is exerted by the coil spring to assure fail/safe operation, and moving element guidance is by sliding fits located outside the propellant cavities. Hydraulic fluid wetted volumes have been isolated from the propellant cavities and located such that appropriate thermal control of the hydraulic fluid may be incorporated, if required, without compromising the design.

III-4F Trade-Off Curves

The sample calculations, performed in accordance with the developed analytical techniques produced a significant quantity of data relative to several valve configurations at operating points within the range of interest. For a specific application, and imposing selected boundaries such as a maximum response time, and maximum weight, it would be advantageous to have available a simple method of determining what valve configuration or configurations are feasible. With such a tool, both the systems designer and the valve designer can more efficiently direct their efforts, concentrating on only feasible configurations.

The analysis program provided detail data on six basic valve configurations, and the design curves facilitate the prediction of performance and physical characteristics data for any of these configurations for any flow factor ($C_d A_t$) and inlet pressure (P_1) within the range of interest. Utilizing this data, several examples of trade-off curves were prepared to demonstrate the capability which had been developed.

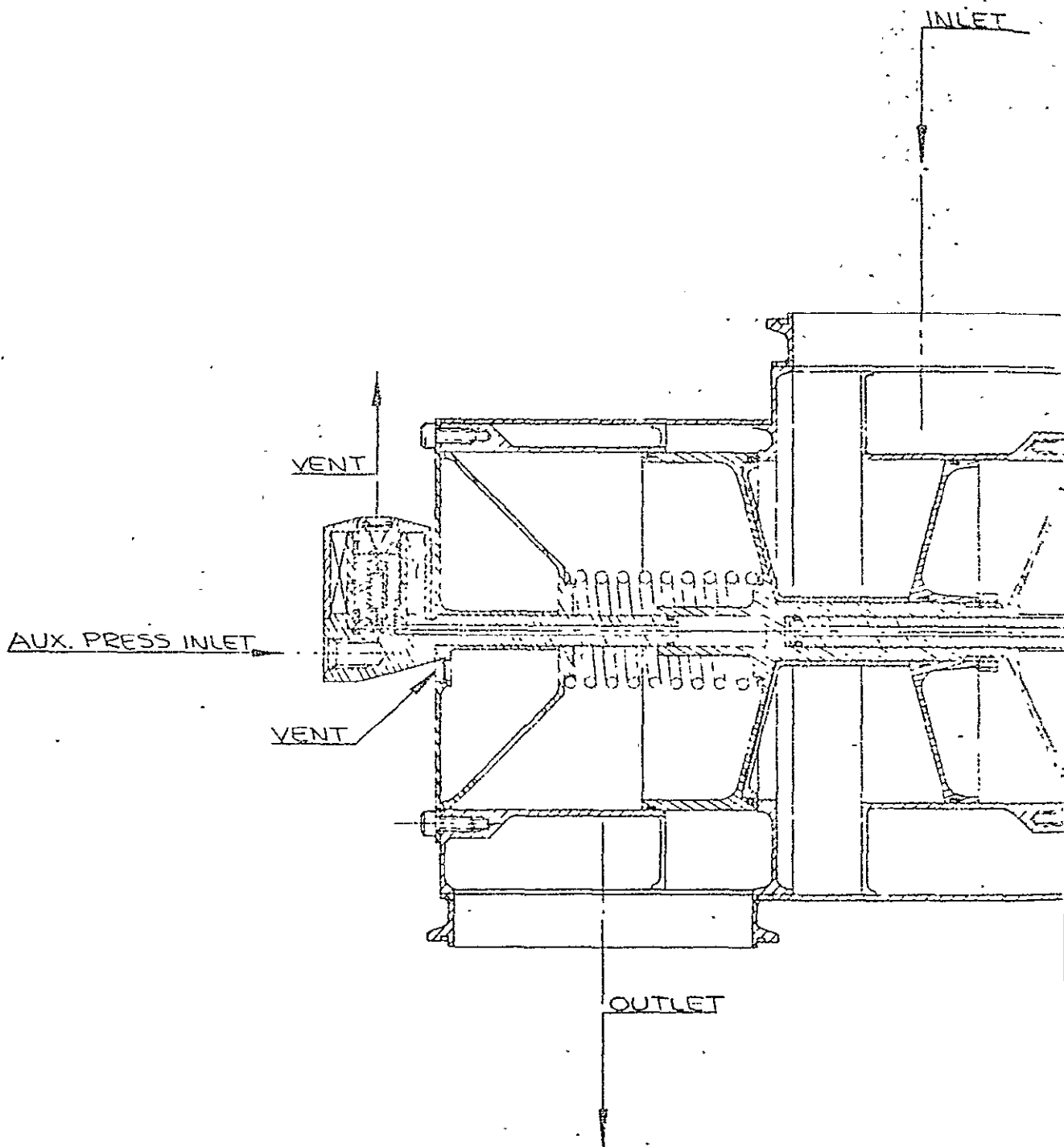
If opening response (at 28 vdc and 530°R in this case) is a prime criteria of the application, operating envelopes of the various configurations can be defined. For a specific maximum allowable opening response time, the analysis program and design curves were employed to define the flow factors ($C_d A_t$) and inlet pressures (P_1) at which the opening response time was equal to the maximum allowed. Figures III-72, III-73 and III-74 present the operating



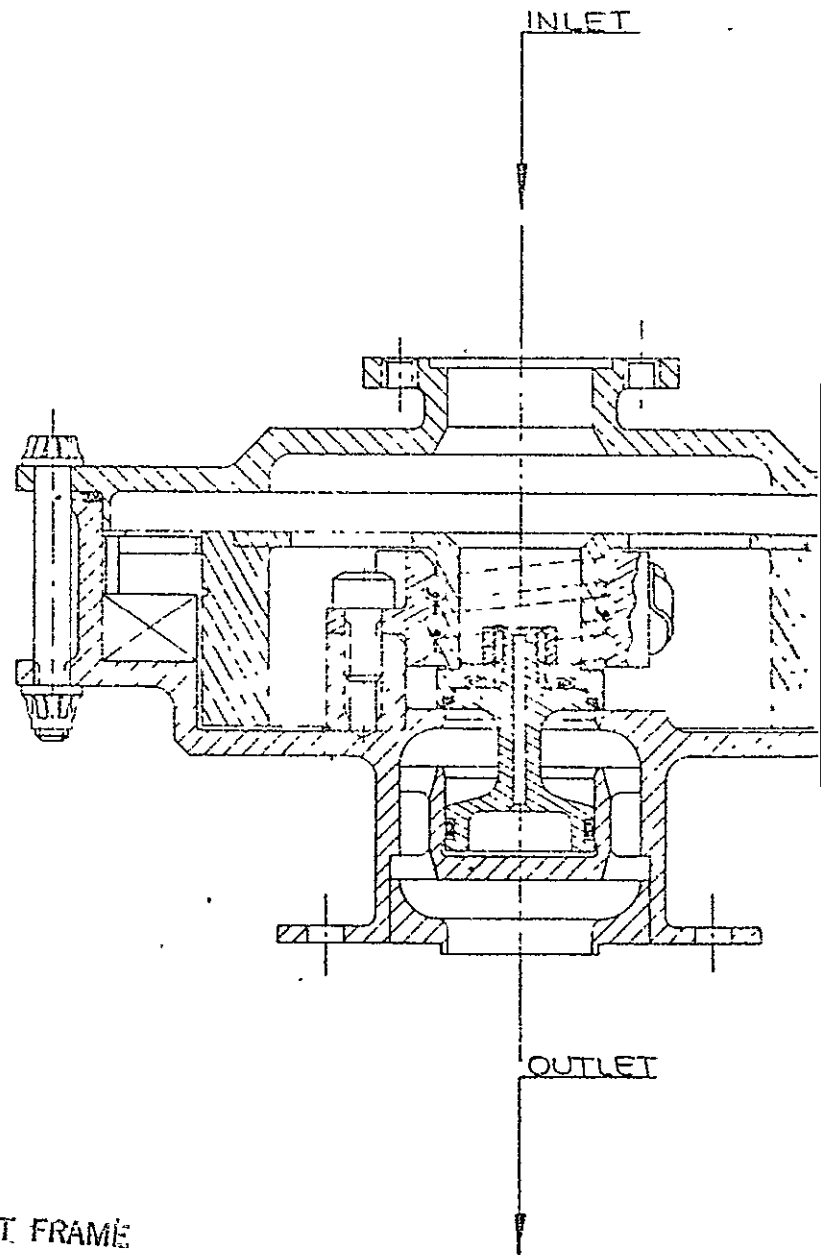
NOT REPRODUCIBLE

FOLDOUT FRAME

QTY		CODE	PART OR	NOMENCLATURE OR	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR
REQD	IDENT		IDENTIFYING NO	DESCRIPTION				FIND NO.
LAYOUT								
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES				DATE				
TOLERANCES:				CHECKED				
XX	XXX	ANGLES		APPROVED				
± .03	± .010	± 0° 30'		APPROVED				
DO NOT SCALE THIS DRAWING				DRAWN				
SURFACE TEXTURE PER USAS B46.1-62				DATE				
BREAK SHARP EDGES				Marquardt				
IDENTIFY PER MPS-800 CLASS				CEL Aerospace Corporation, Van Nuys, California				
NEXT ASSY				FOPPET VALVE-AUX PRESS				
USED ON				PNEUMATIC OPERATED				
APPLICATION				SIZE				
				CODE IDENT NO.				
				DRAWING NO				
				D 86845 L 4665				
				SCALE				
				WT.				
				SHEET 1 OF 1				

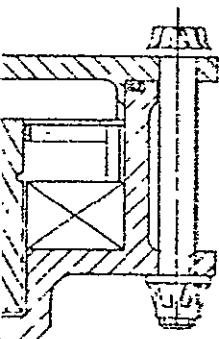


FOLDOUT FRAME
2



FOLDOUT FRAME

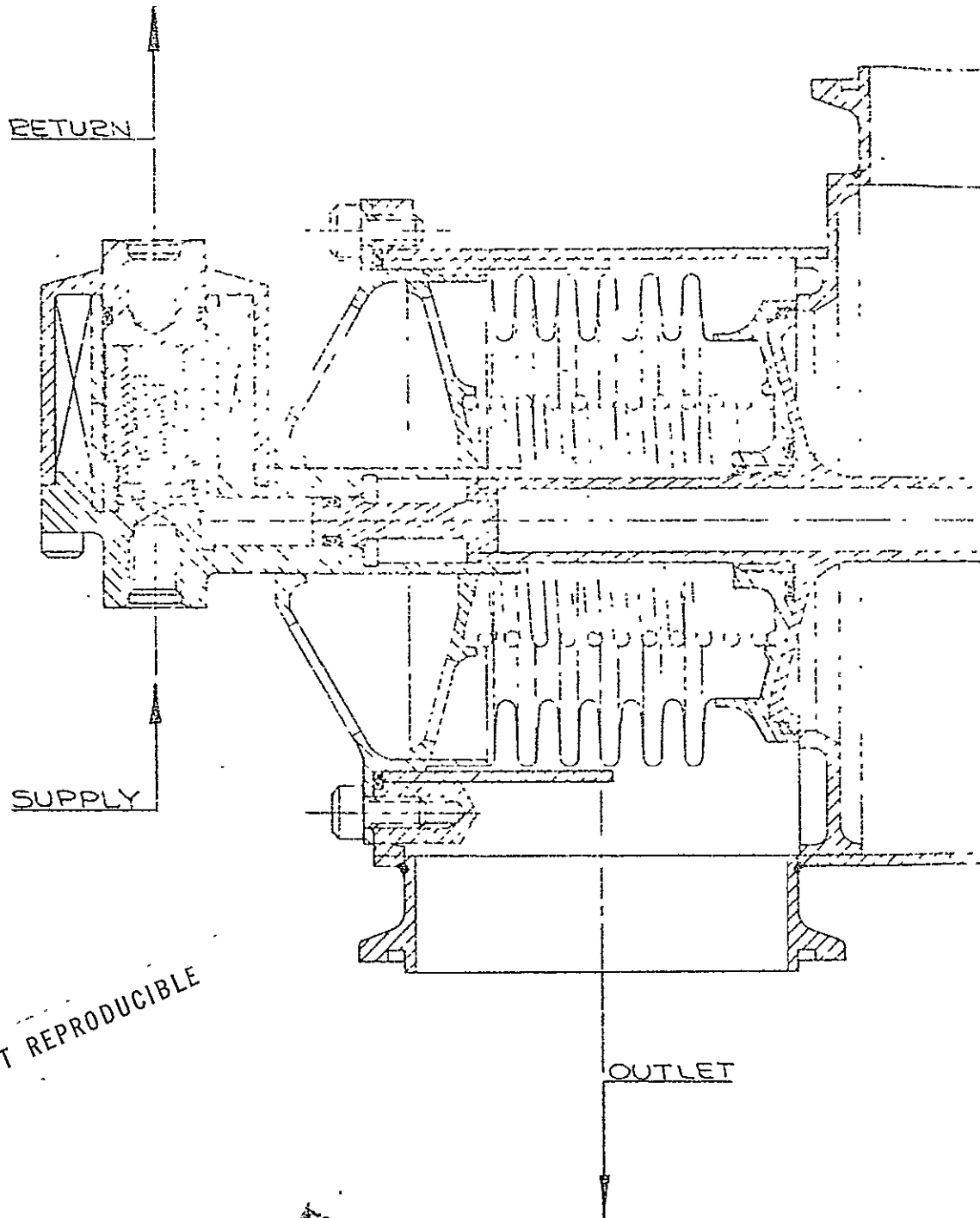
1



REVISIONS			
DATE	DESCRIPTION	DATE	APPROVED

FOLDOUT FRAME 2

QTY REQD	CODE IDENT	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR FIND NO
LIST OF MATERIALS OR PARTS LIST							
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES			DRAWN: <i>[Signature]</i> DATE: <i>6/25/58</i> CHECKED: <i>[Signature]</i> APPROVED: <i>[Signature]</i> APPROVED: <i>[Signature]</i>				
JOX ± .03 JOOX ± .010 ANGLES ± 0° 30' DO NOT SCALE THIS DRAWING SURFACE TEXTURE PER USAS B46.1-62 <i>[Checkmark]</i> BREAK SHARP EDGES			<i>Marquardt</i> (CI Aerospace Corporation Van Nuys, California) POPPET VALVE-TORQUE MOTOR, BALL SCREW OPERATED				
NEXT ASSY USED ON APPLICATION			SIZE CODE IDENT NO DRAWING NO D 86845 L4666 SCALE 2/1 WT SHEET 1 OF 1				



NOT REPRODUCIBLE

EOLDOUT FRAME

7

INLET

RETURN

SUPPLY

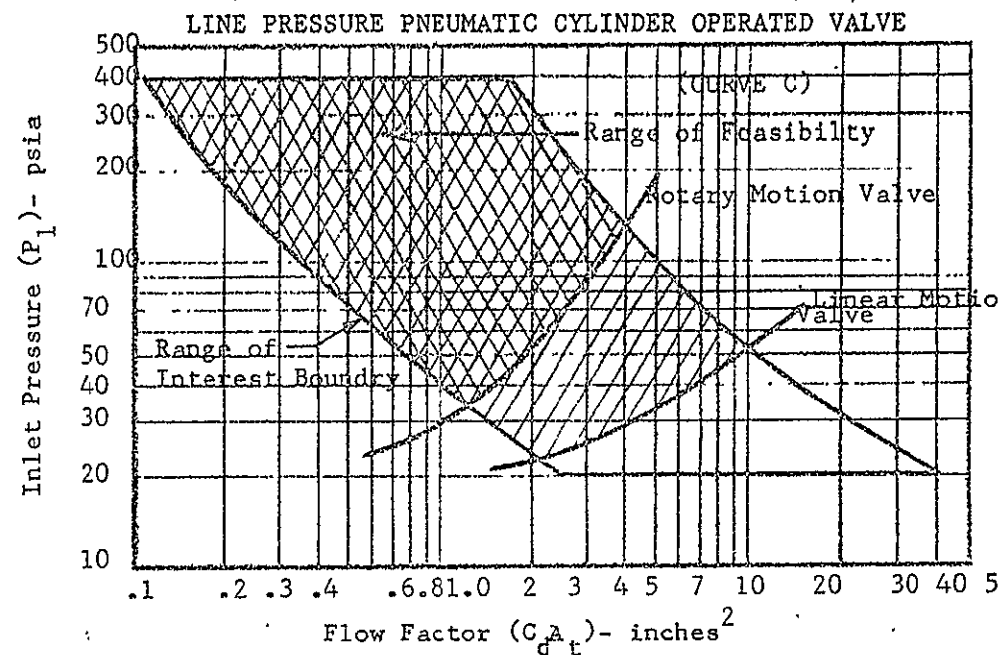
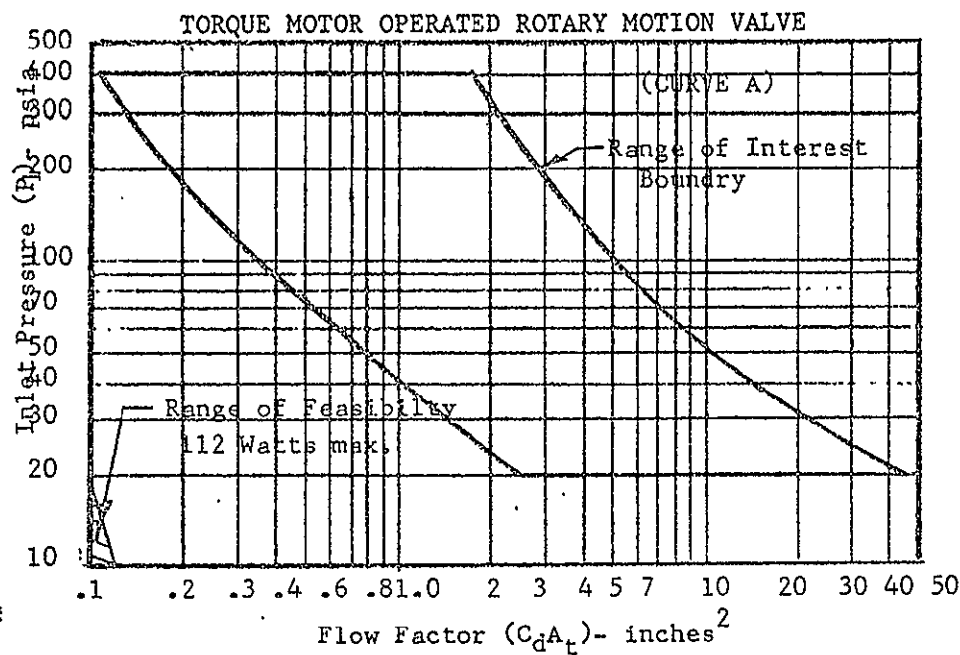
NOT REPRODUCIBLE

FOLDOUT FRAME

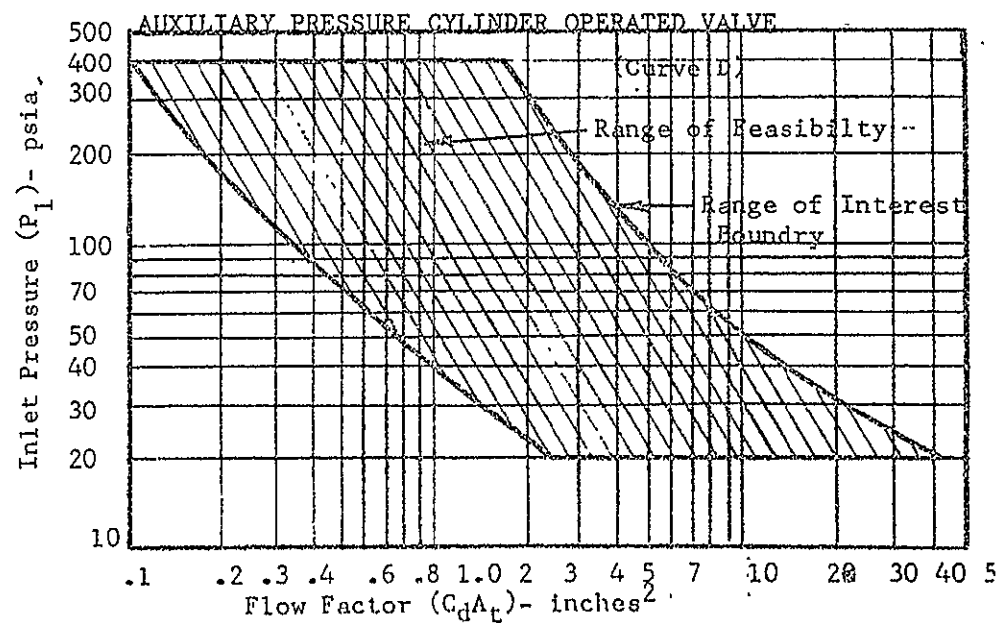
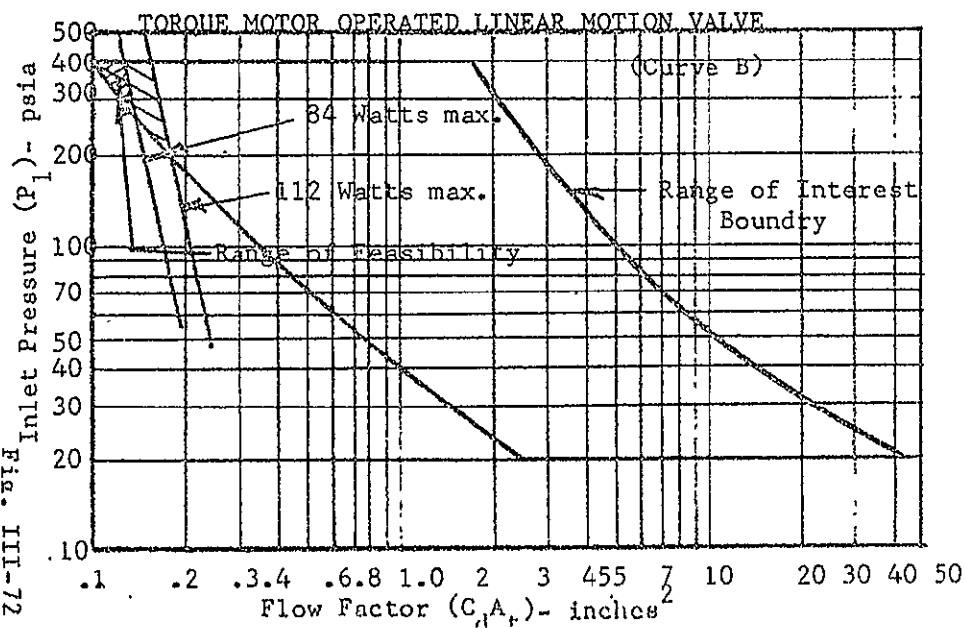
QTY REQD	CODE IDENT	PART OR IDENTIFYING NO	REV. OR REPLACEMENT DATE	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR FINISH NO
LAYOUT							
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES			CHECKED		DATE		
TOLERANCES			APPROVED		DATE		
XXX ± .005			XXX ± .010		XXX ± .015		
DO NOT SCALE THIS DRAWING			SURFACE TEXTURE		PER USAS B-112		
BRAIN SHARP EDGES			IDENTIFY PER VPS-800 CLASS		DRAWING NO.		
NEXT ASSY			USED ON		APPLICATION		
DRAWING NO.			DATE		BY		
DRAWING NO.			DATE		BY		

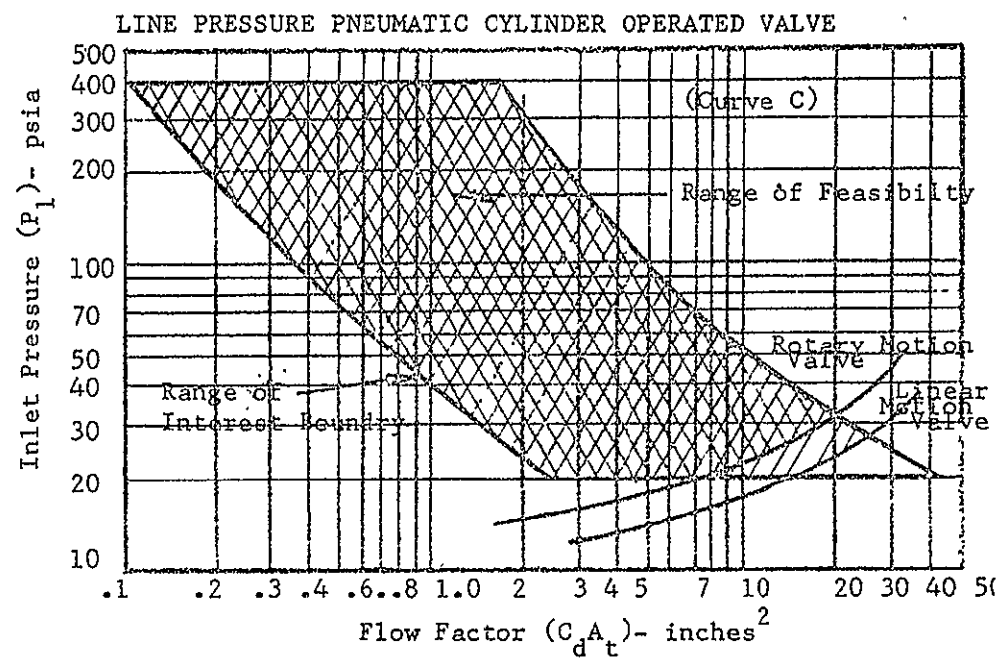
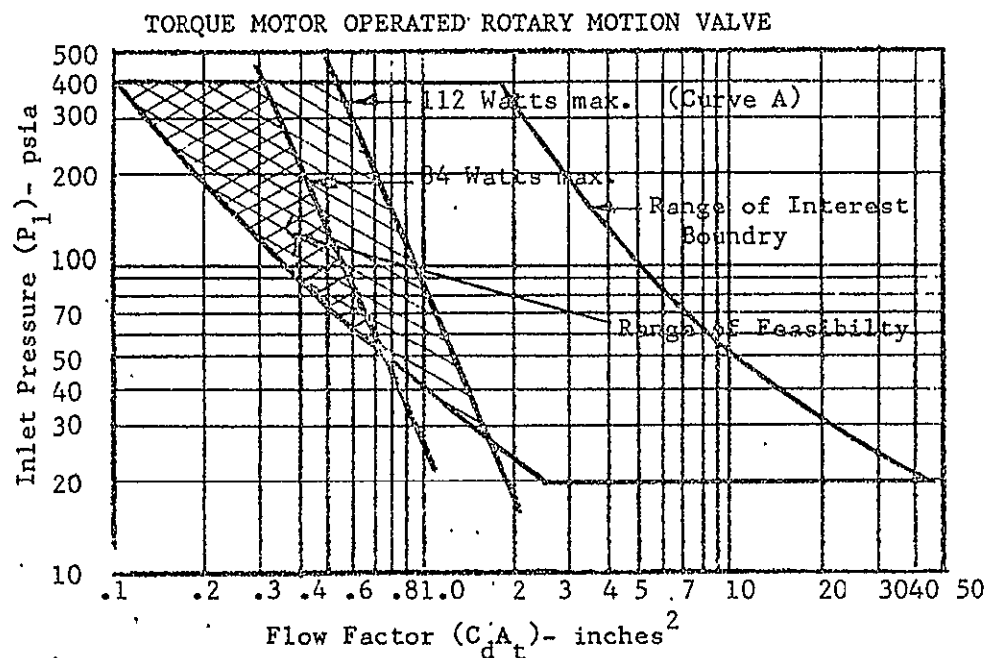
POPFET VALVE, AUX PRESS
HYDRAULIC OPERATED

D 868451 L4667

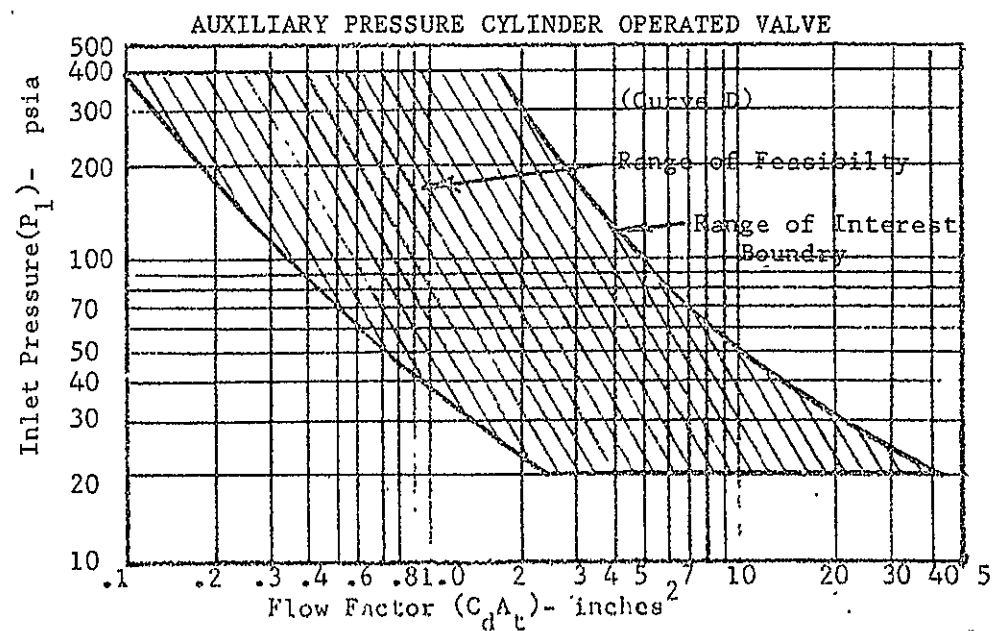
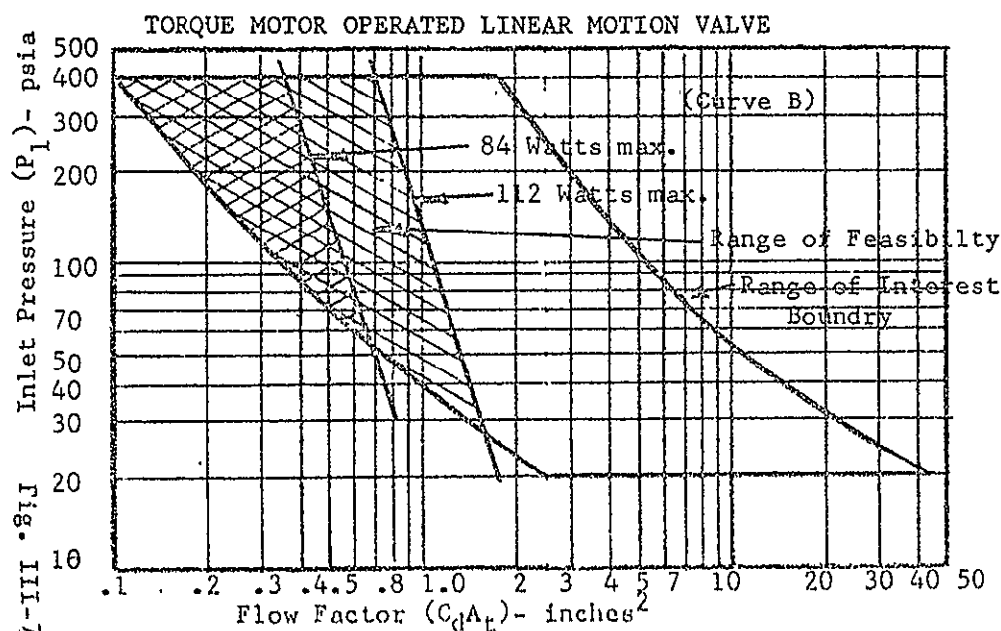


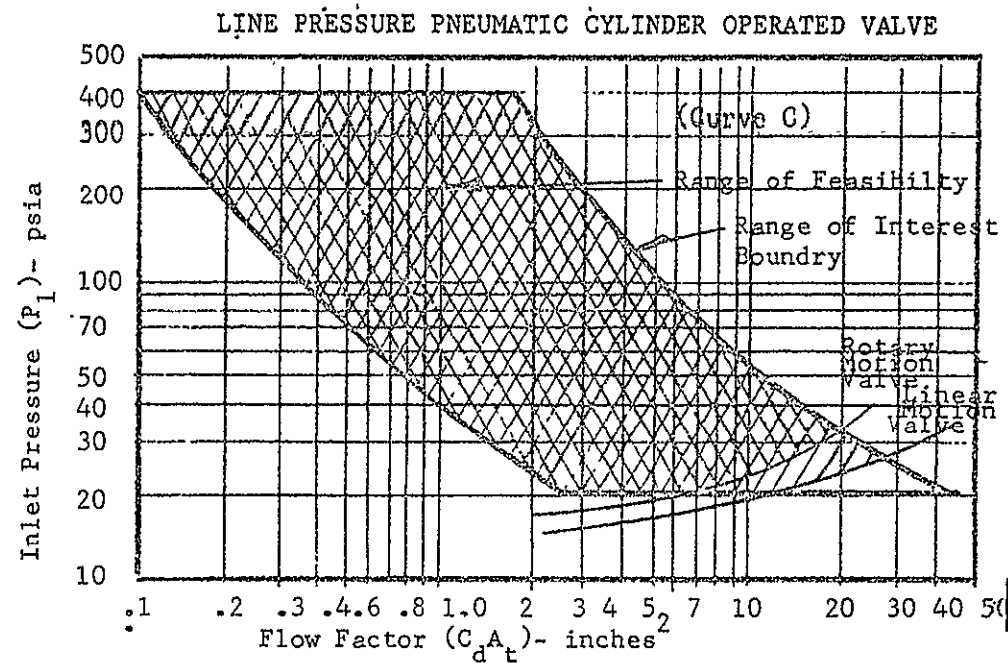
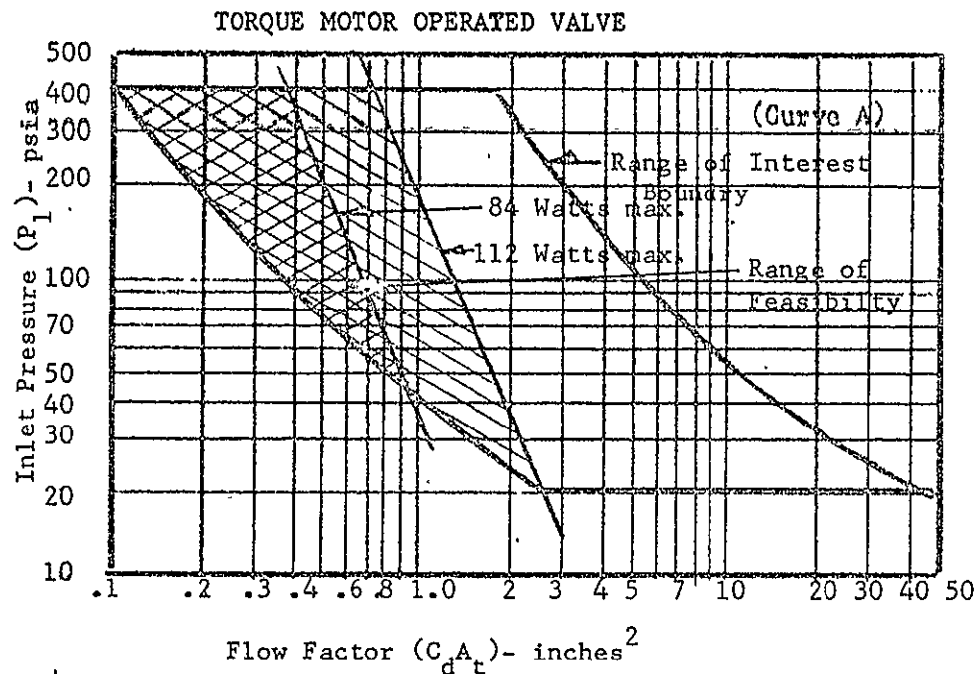
CONFIGURATION FEASIBILITY CHART for OPENING RESPONSE LESS THAN .020 SECONDS



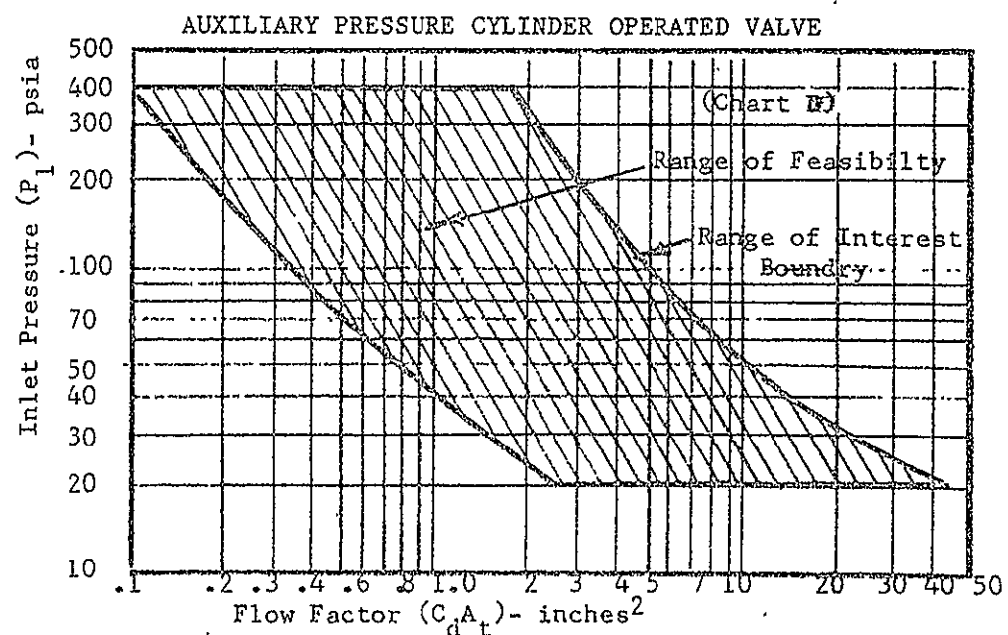
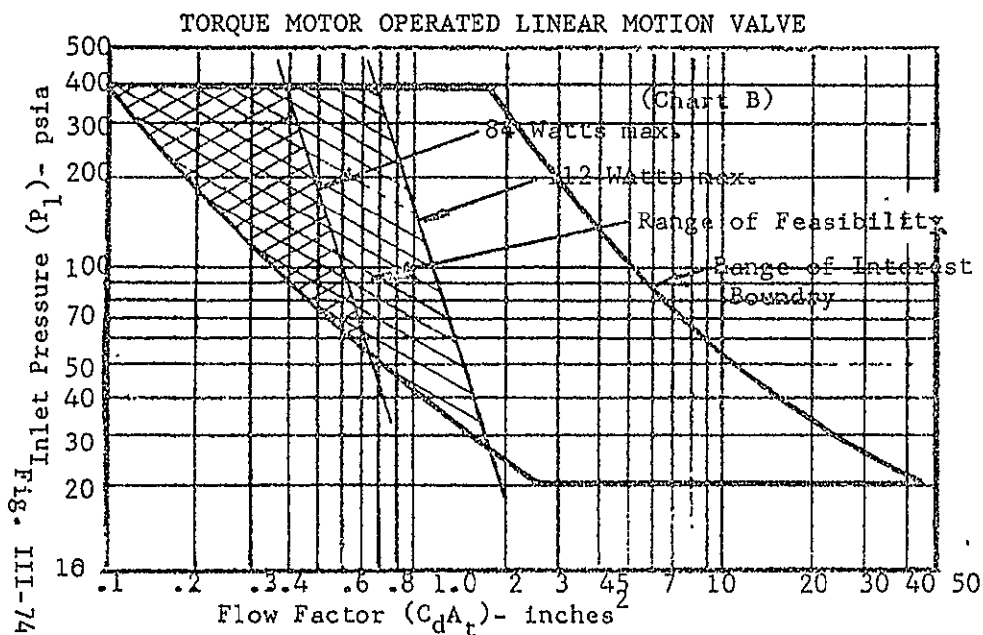


CONFIGURATION FEASIBILITY CHART for OPENING RESPONSE LESS THAN .050 SECONDS





CONFIGURATION FEASIBILITY CHART for OPENING RESPONSE LESS THAN .100 SECONDS



envelopes of the six basic configurations when the maximum allowable opening response is 0.02 seconds, 0.05 seconds, and 0.10 seconds, respectively. Each figure includes four curves to clearly document the operating envelopes of the six basic configurations. Curve A defines the operating envelope of a torque motor actuated rotary motion shut-off, the cross-hatched area representing the range wherein this configuration will exhibit opening response equal to or less than the noted maximum. Two boundaries are defined, as a function of the maximum electrical power limit. Curve B establishes similar operating envelopes for torque motor operated linear shut-off device valves. In curve C, line pressure pneumatic cylinder operated shut-off valve operating envelopes are defined for the allowable opening response times. Curve D established the operating envelope of auxiliary pressure cylinder operated valves, the actuator utilizing either pneumatic or hydraulic fluid at 1500 to 3000 psia supply pressure. Where the intersection of an applications flow factor ($C_d A_t$) and inlet pressure (P_1) falls within the operating envelope for a particular valve configuration, that configuration is capable of opening response equal to, or less than the stated limit.

Where valve weight and opening response are prime considerations, a more useful format of trade-off curve, for each configuration, is shown in Figures III-75 through III-85. Since valve weight will generally increase with decreasing response time, each curve represents the maximum weight of a valid candidate valve configuration which will have an opening response equal to or greater than the value indicated (Ref. Figure III-75 through III-79) for valve opening response times ≥ 0.010 seconds. If slower opening response times are acceptable, reduced valve weights will result and the operating envelope for each configuration will be modified (range of flow factor and inlet pressure where configuration is feasible). This change is illustrated by comparing Figures III-80 through III-85 with Figures III-75 through III-79 for identical configurations. In each figure the extent of each curve is determined by the operating envelope of the configuration at the specified inlet pressure (P_1), or the range of interest, whichever is most restrictive.

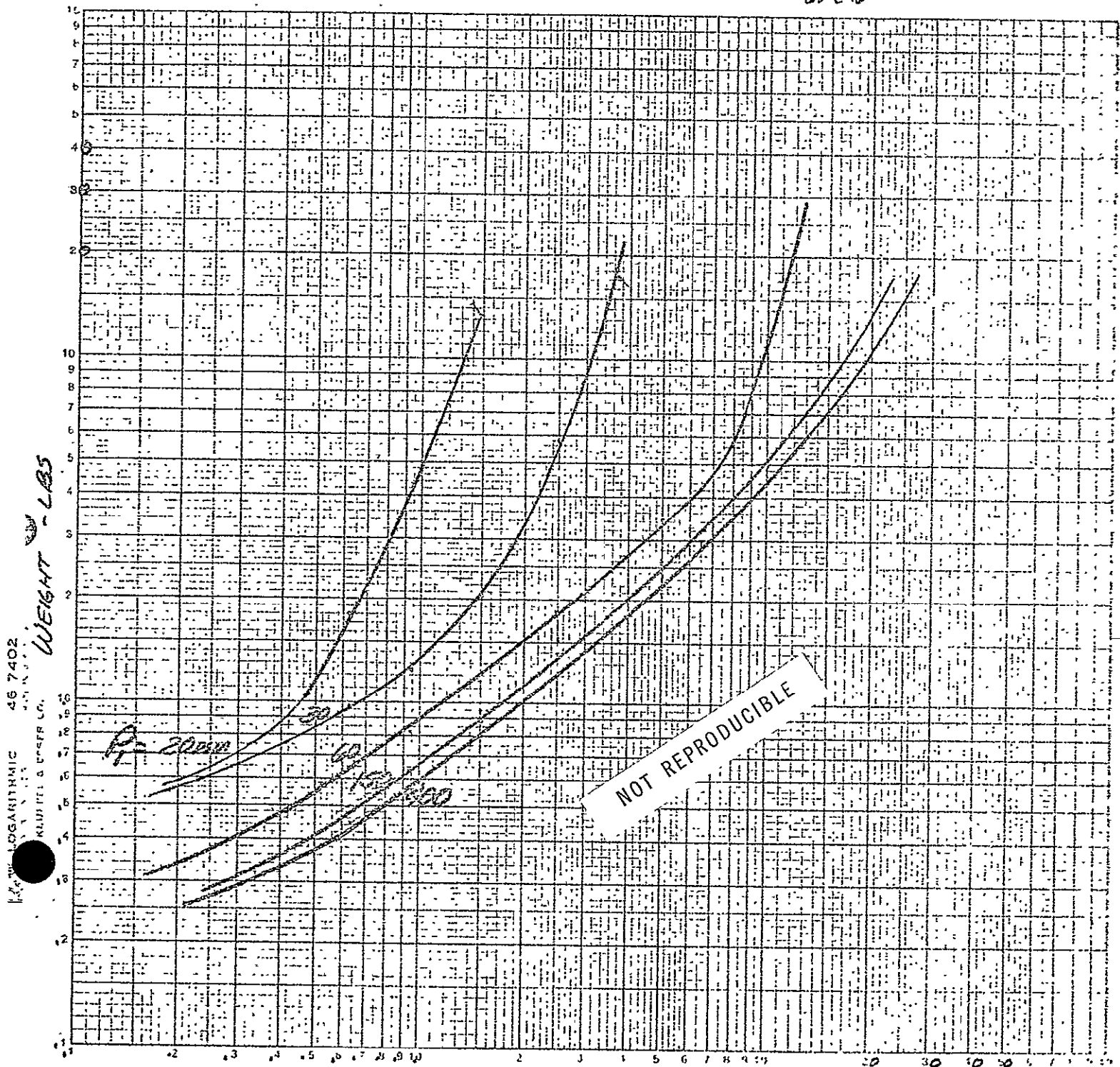
These figures are representative of the type of trade-off curves which may be prepared utilizing the analysis program developed during this program. For other characteristics, such as electrical power, envelope dimensions, or minimum valve open time, similar trade-off curves may be prepared. The number of variables and the influence each exerted on valve performance and/or physical characteristics precluded the preparation of a comprehensive, generalized trade-off curve, but access to an IBM 360 computer, or equivalent, and utilizing the analysis program described herein, the total spectrum of variables can be considered and trade-off curves prepared for the desired increments. Appendix A presents a TMC internal report on the APL programming employed to generate data presented in this report.

III-4G Utilizing Parametric Design Curves

Employing the design curves generated by the analysis program, provides a method whereby the system designer and/or the valve designer may readily compile a significant quantity of data about a valve, for a specific application, without performing extensive analysis and design studies. For the systems designer, rapid predictions can be made of feasible candidates and data generated to appraise valve contributions to system weight and envelope and validate

WEIGHT VS FLOW FACTOR ($C_d A_t$) AT VARIOUS INLET PRESSURES FOR A LINE PRESSURE PNEUMATIC CYLINDER OPERATED LINEAR MOTION VALVE

$t_{OPEN} \geq 10 \text{ ms}$

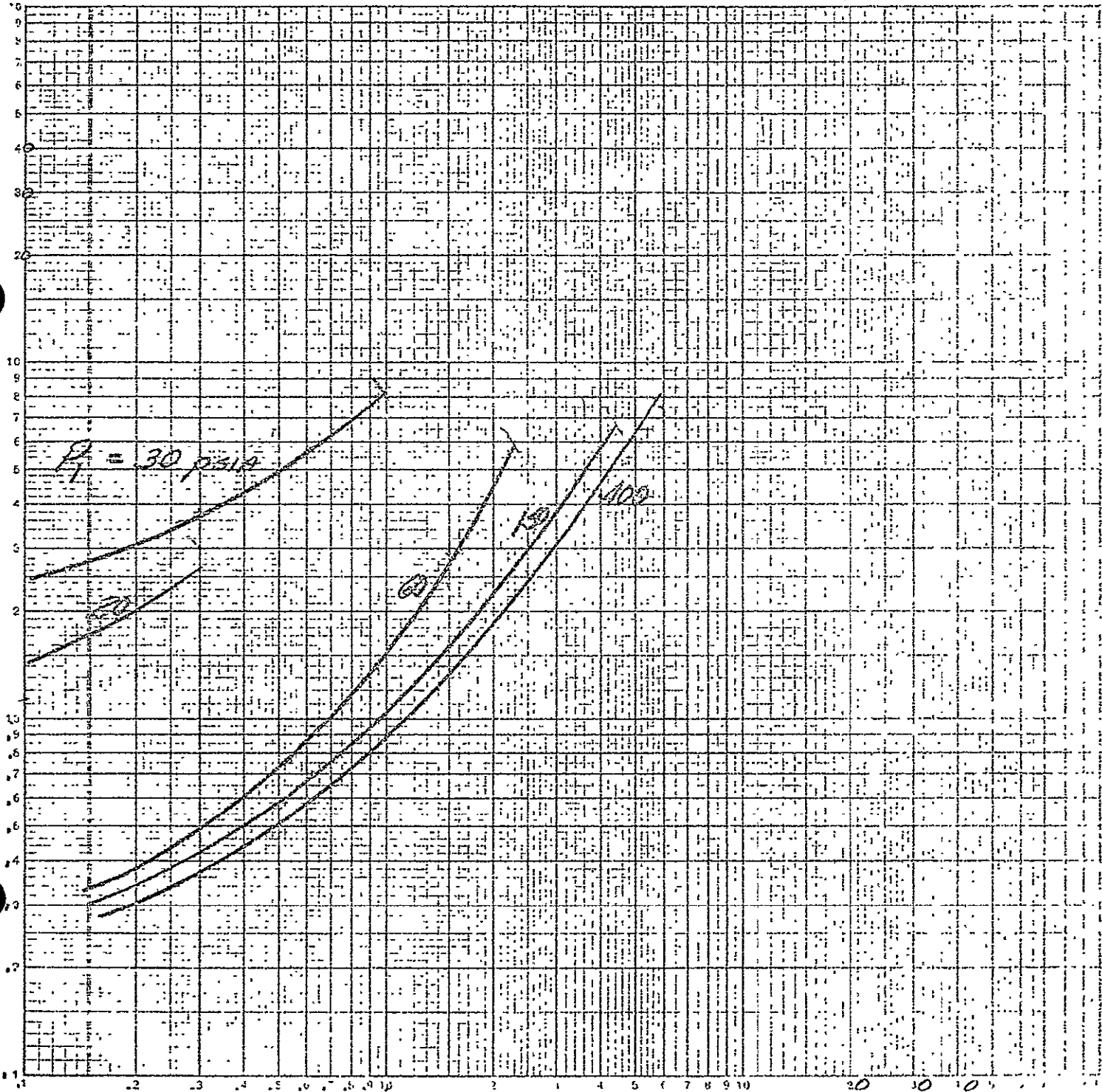


WEIGHT VS FLOW FACTOR ($C_d A_t$) AT VARIOUS INLET PRESSURES FOR A LINE PRESSURE PNEUMATIC CYLINDER OPERATED ROTARY MOTION VALVE

$t_{open} \geq 10 \text{ ms}$

WEIGHT

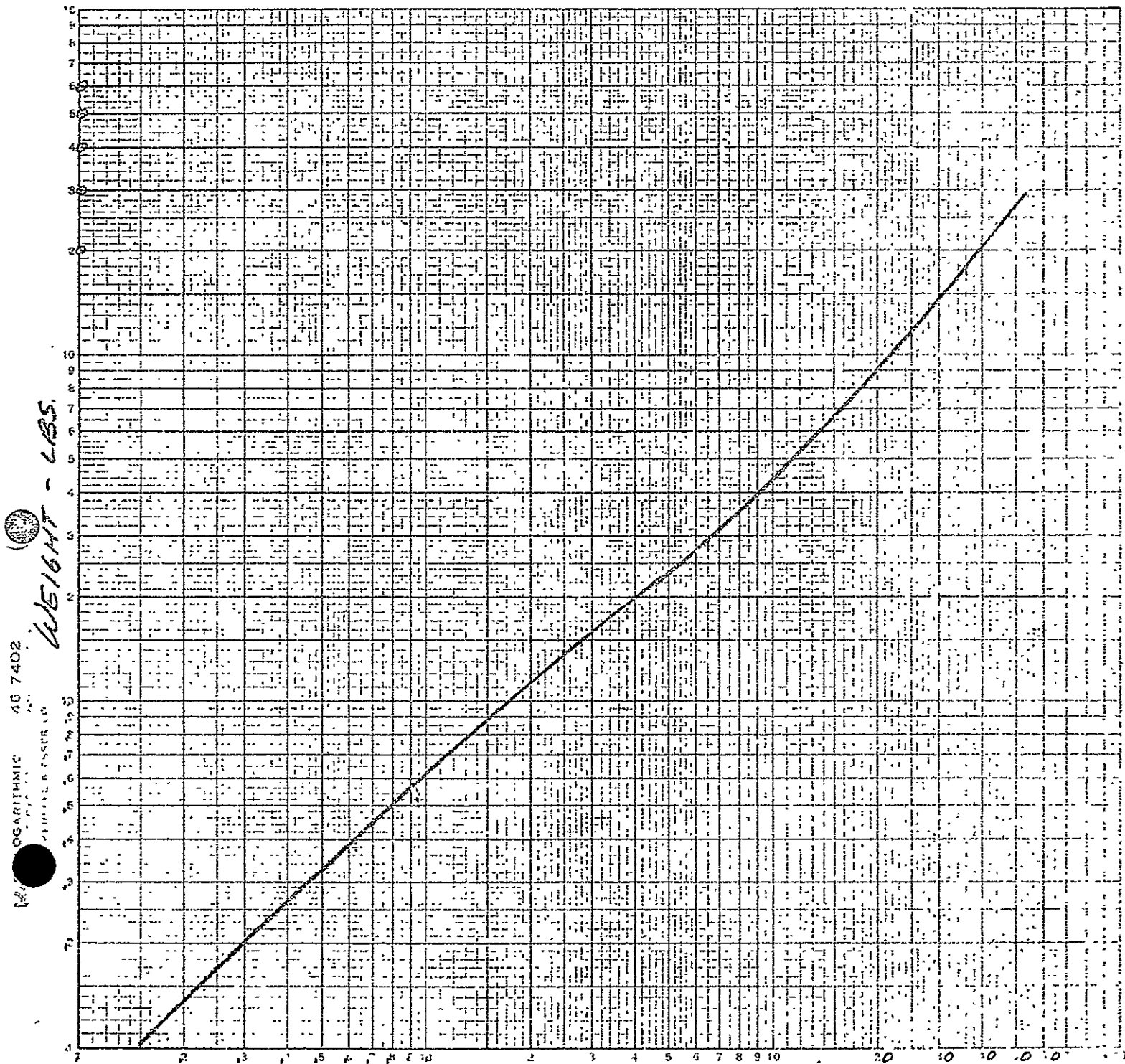
LOGARITHMIC
46 7402
KLIPPEL & BECKER CO.



Flow Factor ($C_d A_t$) - in^2

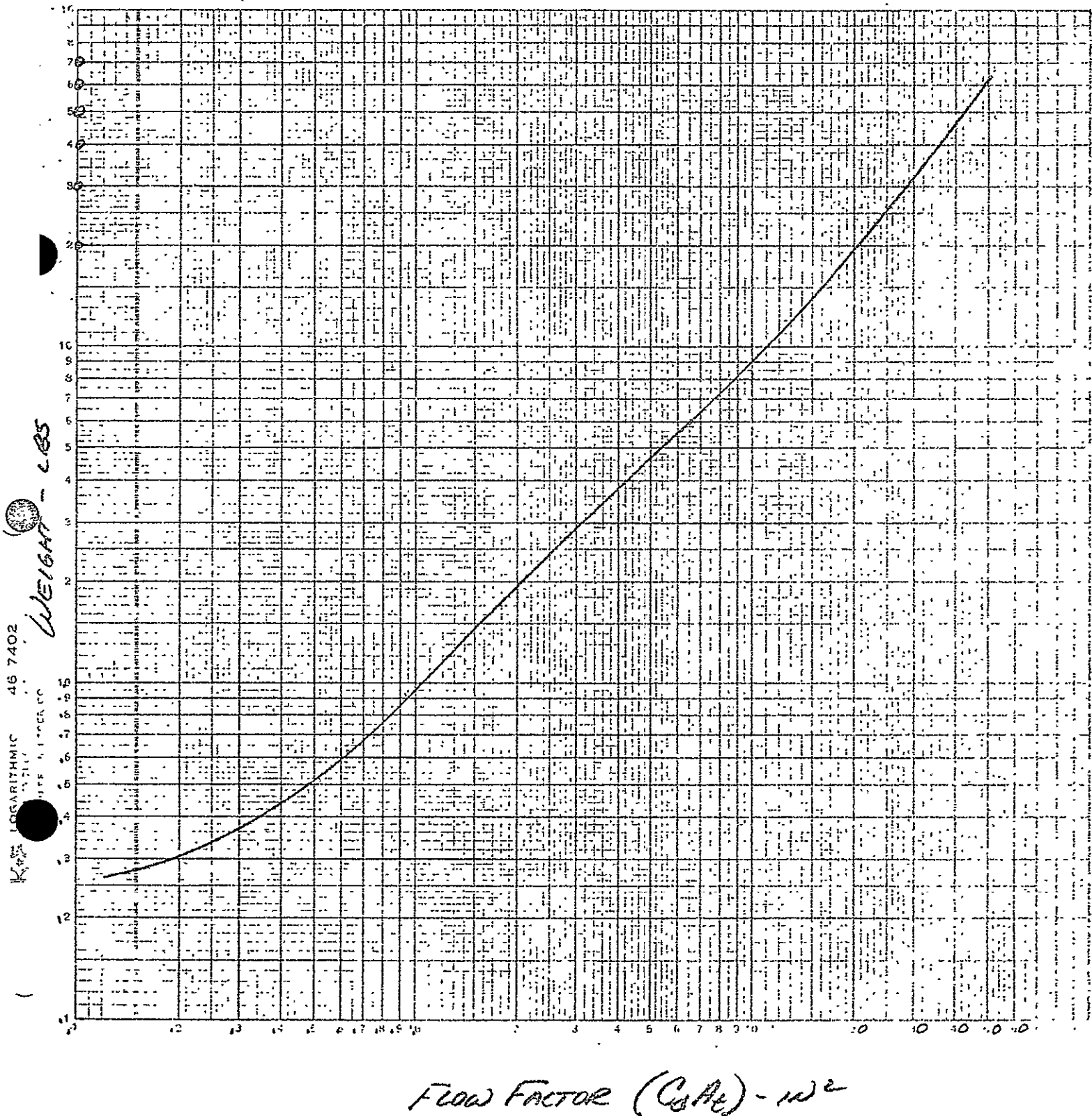
WEIGHT VS FLOW FACTOR ($C_d A_e$) AT VALVE INLET PRESSURES
FROM 20 TO 400 PSIA, FOR AN AUXILIARY PRESSURE
CYLINDER OPERATED LINEAR MOTION VALVE

$t_{open} \geq 10 \text{ ms}$



FLOW FACTOR ($C_d A_e$) - in²

WEIGHT VS. FLOW FACTOR ($C_d A_e$), AT VALVE INLET PRESSURES
FROM 20 TO 400 PSIA, FOR A HYDRAULIC CYLINDER
OPERATED VALVE
($P_{AUX} = 1500 - 3000 \text{ PSIA}$) $t_{OPEN} \geq 10 \text{ MS}$



WT OF TORQUE MOTOR OPERATED LINEAR MOTION VALVE

$$t_{\text{open}} \geq 10 \text{ ms}$$

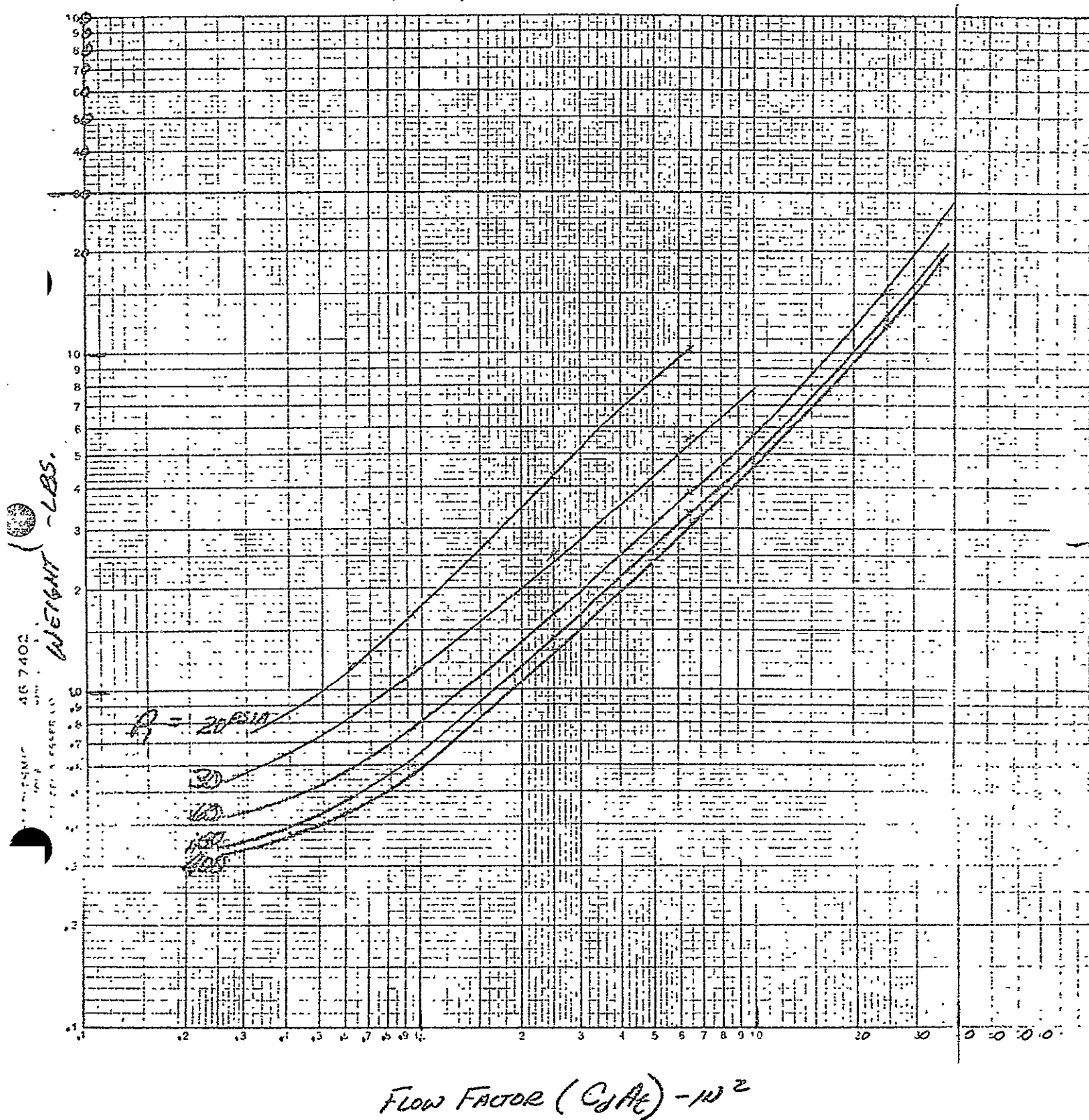
W
84 WATTS

WT
.84 LBS

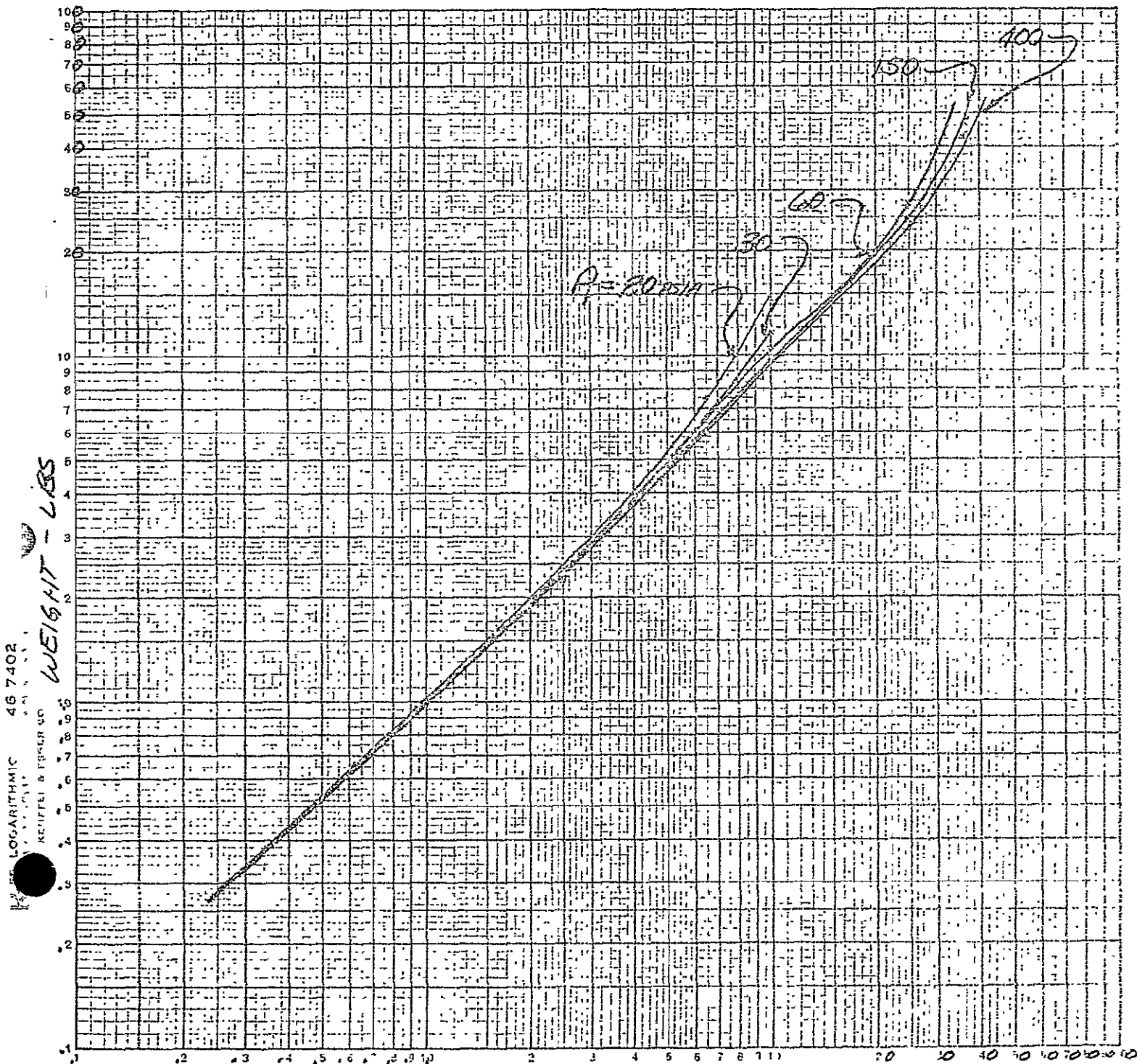
112

.90 LBS

WEIGHT VS FLOW FACTOR ($C_d A_e$) AT VARIOUS INLET
PRESSURES FOR A LINE PRESS. PNEU. CYLINDER
OPERATED LINEAR MOTION VALVE $t_{open} \geq 80ms$

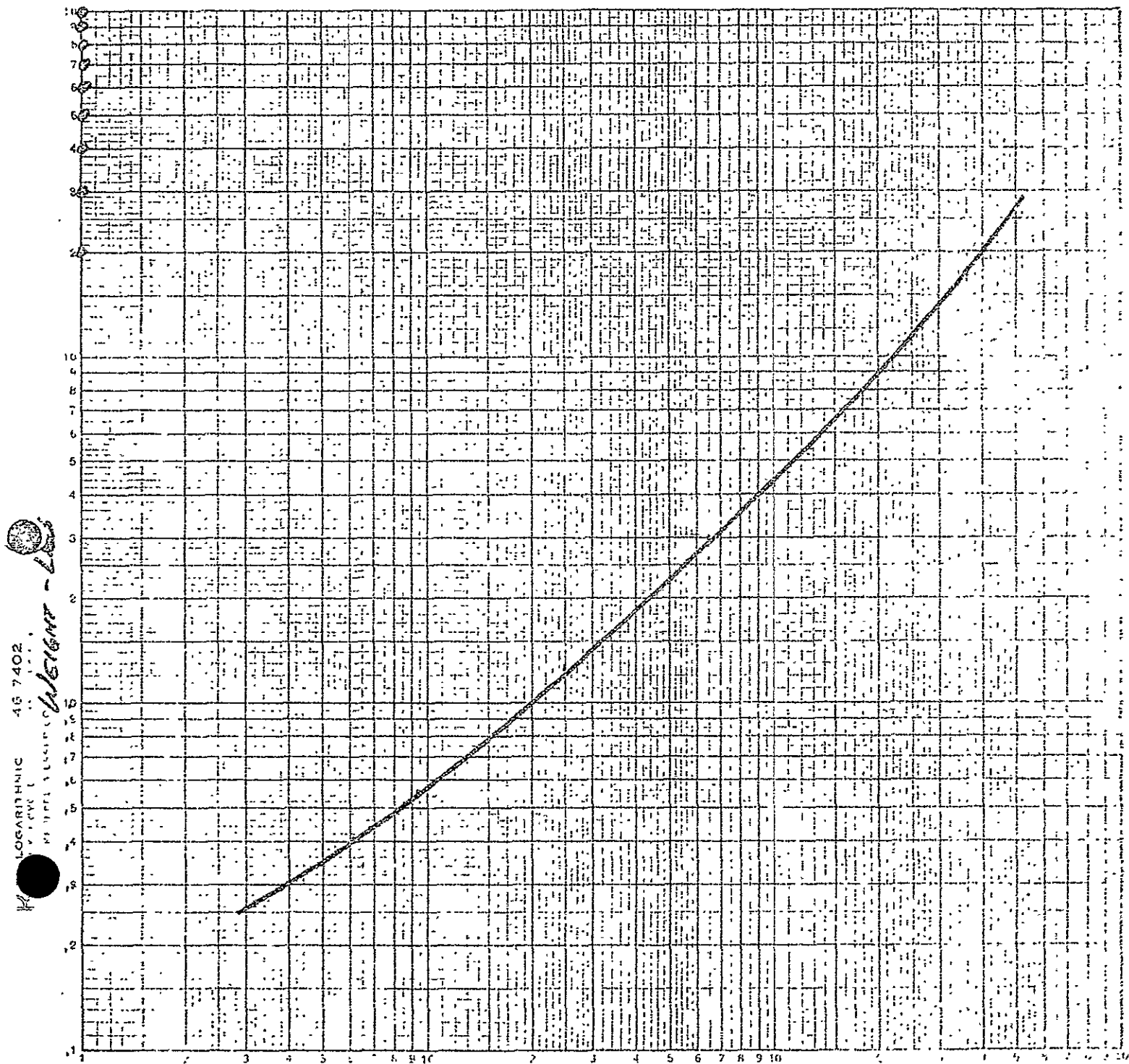


WEIGHT VS FLOW FACTOR ($C_d A_t$) AT VARIOUS INLET
PRESSURES FOR A LINE PRESS. PNEU. CYLINDER
OPERATED ROTARY MOTION VALVE $t_{OPEN} \leq 80 \text{ ms}$



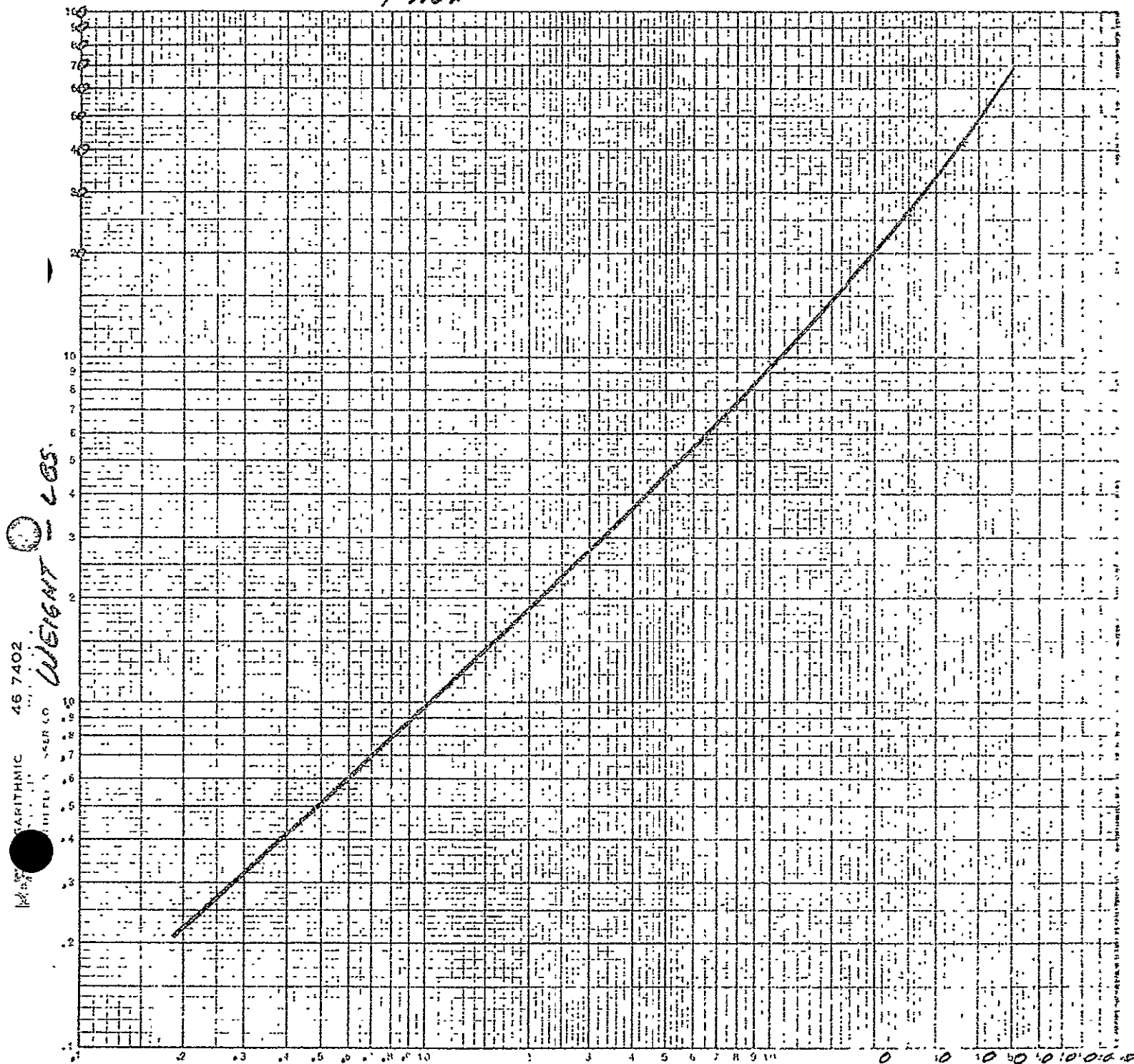
FLOW FACTOR ($C_d A_t$) - in^2

WEIGHT VS. FLOW FACTOR ($C_d A_e$) AT VALVE INLET
 PRESSURES FROM 20 TO 400 PSIA, FOR AN AUXILIARY
 PRESSURE PNEU. CYLINDER OPERATED LINEAR MOTION VALVE
 $t_{OPEN} \approx 80 \text{ ms}$



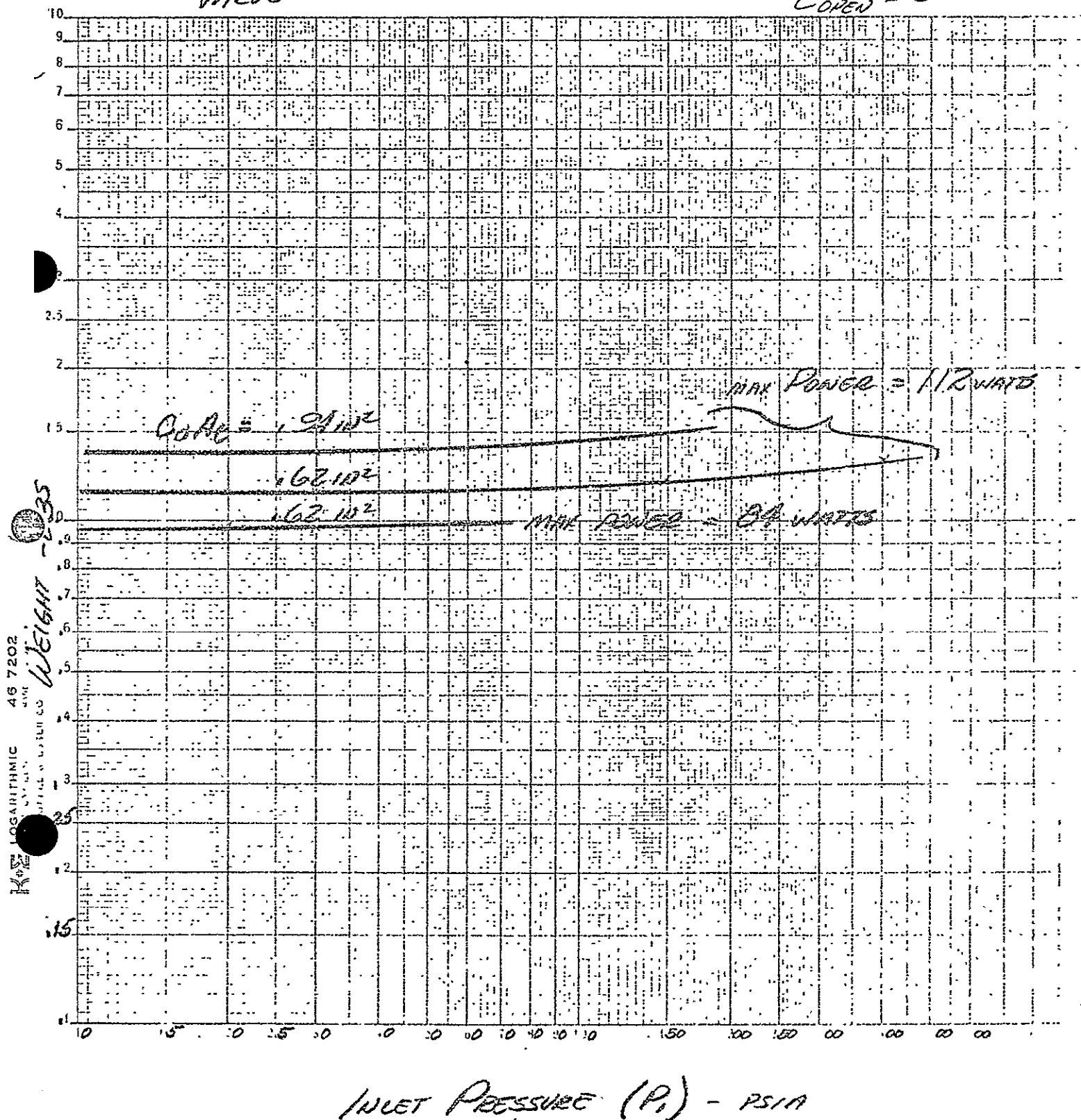
FLOW FACTOR ($C_d A_e$) 10²

WEIGHT VS. FLOW FACTOR ($C_d A_e$), AT VALVE INLET
 PRESSURES FROM 20 TO 40 PSIA, FOR AN AUXILIARY
 PRESSURE PNEU. CYLINDER OPERATED ROTARY
 MOTION VALVE
 $P_{AUX} = 1500 - 3000 \text{ PSI}$ $t_{OPEN} \geq 80 \text{ MSEC}$



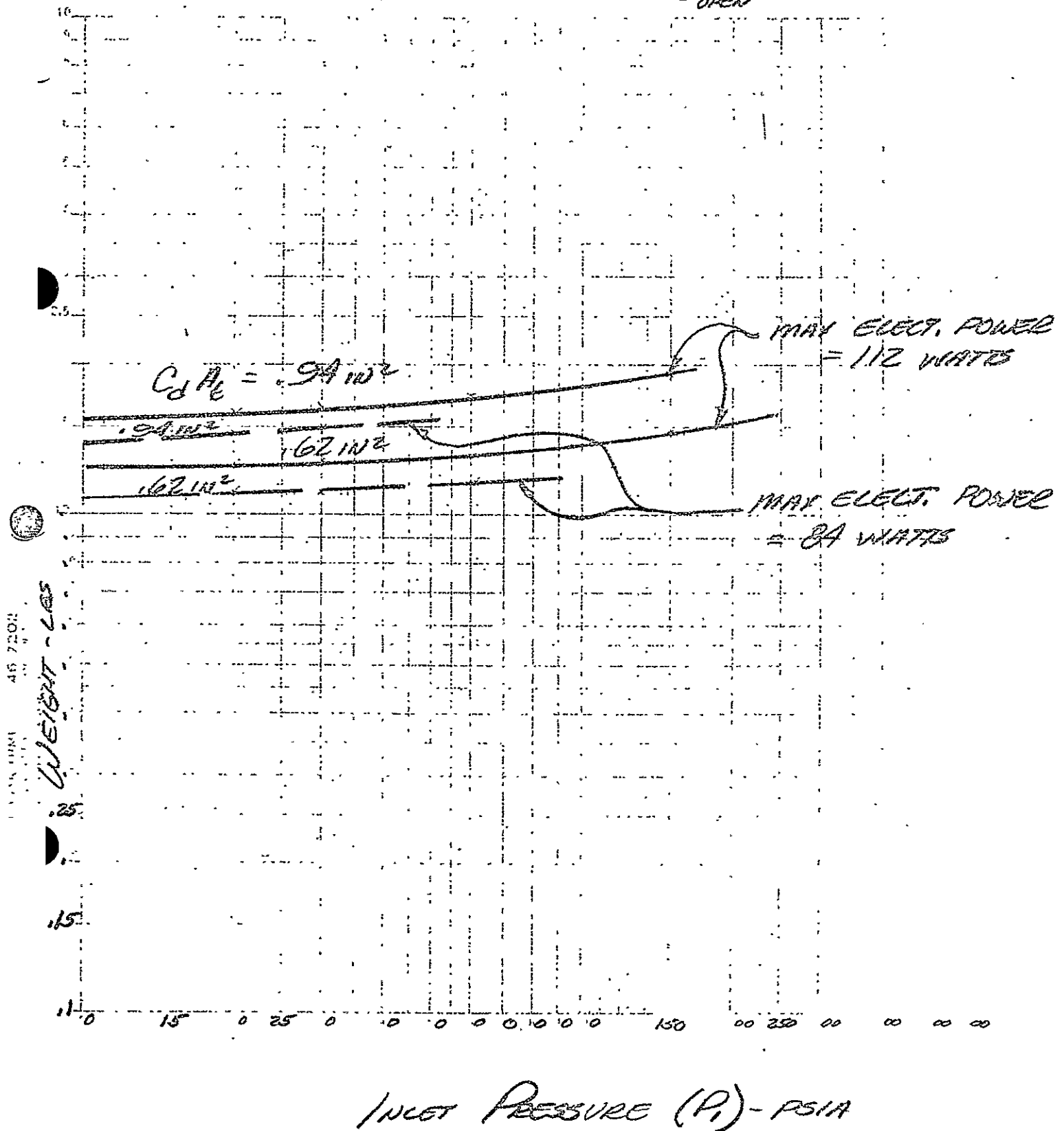
FLOW FACTOR ($C_d A_e$) - IN²

WEIGHT VS VALVE INLET PRESSURE AT VARIOUS
VALVE FLOW FACTORS AND ELECT. POWER LEVELS
FOR A TORQUE MOTOR OPERATED LINEAR MOTION
VALVE $t_{OPEN} \approx 80 \text{ ms}$



WEIGHT VS VALVE INLET PRESSURE VALUE FLOW FACTORS AND ELECT. POWER LEVELS FOR A TORQUE MOTOR OPERATED ROTARY MOTION VALVE

$$t_{OPEN} \geq 80 \text{ ms.}$$



system performance and design criteria. The valve designer may utilize the curves to establish the optimum configuration for a specific application.

Table III-XX presents step by step instructions for utilizing the design curves presented in this report. Following the sequence presented, in its entirety, enables the user to determine performance and physical characteristics data for up to six candidate valve configurations for a specific application wherein the thrust level, O/F, and inlet pressure for an oxygen/hydrogen fueled rocket engine are the input factors. Further refinement of the pneumatic and hydraulic actuated valve configurations may be accomplished by employing the shut-off device analysis and solenoid analysis to appraise the impact of the actuator control valves.

As an example, Appendix B presents the appropriate design curves, marked to indicate their use in determining the valve characteristics of an application where thrust level is 1000 lb, O/F is 2.0 and inlet pressure is 30 psia. The particular valve configuration selected for the example is a line pressure pneumatic cylinder operated linear motion shut-off valve.

III-4H Valve Selection

Having developed an analytical tool that facilitates the generation of specific data for a number of valve configurations, the evaluation of this data relative to the design goals of a particular system must be accomplished in the most objective manner possible. This evaluation will result in the selection of the optimum valve configuration. To accomplish this evaluation, a comprehensive rating system may be developed which assigns objective figures of merit to each of the valve's characteristics. Since the requirements of each particular valve application are the criteria for assigning weighting factors to each characteristic, a generalized rating system is not meaningful. A rating system for each application may be developed by the method described in paragraph III-2. In lieu of a specific application and to illustrate the composition of a rating system, Figure III-86 presents an arbitrary rating system shown to illustrate a method whereby objective assignment of figures of merit to each valve candidate characteristics may be accomplished.

With a valid rating system available, the analysis program and/or design curves are employed to generate performance and physical characteristics data for likely candidate valve configurations. The data is tabulated in some workable fashion, the rating system utilized to assign a figure of merit to each data point, and the figures of merit for each valve configuration summed to yield a total score which is compared with the scores achieved by other candidate valves. If the rating system is truly valid, the highest scoring candidate is the most optimum choice. The highest scoring three or four candidates should then be examined on an intuitive basis to see whether the apparent optimum choice is a reasonable selection from the standpoint of overall engineering judgment. If some doubt still exists as to the validity of the selection, sensitivity analysis of the rating system may be performed to assess the impact of changes in the rating system on the final scores of the various candidate valve configurations. If the results of the scoring are highly sensitive to small changes in the rating system, effort should be directed toward improving the validity of the rating system. Should the scoring show little reaction to small changes in the rating system, the indicated optimum choice may be selected with confidence.

TABLE III-XX

INSTRUCTIONS FOR UTILIZING DESIGN CURVES TO PREDICT PERFORMANCE AND PHYSICAL CHARACTERISTICS OF VALVE CONFIGURATIONS

Given: O_2H_2 Engine Thrust, O/F and Inlet Pressure

1. From Figures III-1 and III-2, determine flow factor ($C_d A_t$) for fuel valve and oxidizer valve.
2. For each value of flow factor ($C_d A_t$), inlet pressure (P_1), and assumed motion time, t , determine
 - A. Linear motion shutoff device force required (F_v) (Figures III-13 through III-18 and Acceleration (Figure III-19)
 - B. Rotary motion shutoff device torque required (T_v) (Figures III-21 through III-26.
3. Design curves are available for predicting performance and characteristics of 4 actuator types.

TABLE III-XX (Continued)

SHUTOFF DEVICE	ACTUATOR	PROCEDURE
LINEAR MOTION	LINE PRESSURE OPERATED PNEUMATIC CYLINDER ACTUATOR	A. FOR VALUES OF F_v, P_1 AND ACCELERATION, DETERMINE PISTON DIAMETER (D_p) FROM FIGURES III-31 THROUGH III-35. B. FOR VALUES OF MOTION TIME (t) AND ACCELERATION, DETERMINE OPENING PNEUMATIC DELAY FROM FIGURE III-37 (CORRECT FOR APPROPRIATE PNEUMATIC MEDIA PER TABLE III-XVIII). C. THE CLOSING PNEUMATIC DELAY CAN BE CALCULATED BY SOLVING A COMPLEX FUNCTION, BUT DOES NOT LEND ITSELF TO A TWO DIMENSIONAL PLOT D. DETERMINE SHUTOFF DEVICE ENVELOPE DIMENSIONS AND WEIGHT FROM FIGURES III-50 and III-51, RESPECTIVELY, FOR VALUE OF FLOW FACTOR ($C_d A_t$) E. DETERMINE ACTUATOR ENVELOPE DIMENSIONS AND WEIGHT FROM FIGURES III-57 & 58, RESPECTIVELY, FOR VALUE OF PISTON DIAMETER (D_p) F. OPENING RESPONSE = MOTION TIME (t) + PNEUMATIC DELAY + COMMAND VALVE RESPONSE (MAY BE KNOWN OR CAN BE COMPUTED BY THIS ANALYTICAL PROGRAM) G. UNIT WEIGHT = SHUTOFF DEVICE WEIGHT + ACTUATOR WEIGHT

TABLE III-XX (Continued)

		H. UNIT ENVELOPE = SHUTOFF DEVICE ENVELOPE + ACTUATOR ENVELOPE (DESIGN CONSIDERATIONS MAY ALLOW BOTH ELEMENTS TO BE CONTAINED WITHIN SHUTOFF DEVICE ENVELOPE - SUBJECTIVE JUDGMENT) I. THE ANALYTICAL PROGRAM MAY BE EMPLOYED TO DETERMINE: 1. VALVE STROKE 2. SEAT DIAMETER 3. VALVE MOVING MASS 4. ACTUATOR SUPPLY VALVE FLOW FACTOR REQUIRED 5. VOLUME OF GAS USED PER VALVE CYCLE 6. ACTUATOR PRESSURES AT TIME OF CLOSING MOTION INITIATION
LINEAR MOTION	AUXILIARY PRESSURE OPERATED PNEUMATIC CYLINDER ACTUATOR	A. FOR VALVES OF F_v, ACCELERATION, AND THE SELECTED AUXILIARY PRESSURE, DETERMINE PISTON DIAMETER (D_p) FROM FIGURE III-36 B. FOR BALANCE OF PERFORMANCE AND CHARACTERISTICS, PROCEED AS IN (B) THROUGH (I) ABOVE <u>EXCEPT</u> - ACTUATOR WEIGHT IS DETERMINED BY FIGURE III-59 (RATHER THAN FIGURE III-58.
LINEAR MOTION	HYDRAULIC OPERATED CYLINDER ACTUATOR	A. PROCEED AS IN (A) AND (B) ABOVE (AUXILIARY PRESSURE OPERATED PNEUMATIC CYLINDER) EXCEPT, USE FIGURES III-38 & 39 FOR PISTON DIAMETER DETERMINATION AND HYDRAULIC DELAY, RESPECTIVELY
LINEAR MOTION	TORQUE MOTOR ACTUATOR	A. FOR VALUES OF ACCELERATION AND MOTION TIME, DETERMINE TORQUE FACTOR FROM FIGURE III-47 CALCULATE REQUIRED TORQUE (T_v)

TABLE III-XX (Continued)

ROTARY MOTION

LINE PRESSURE
OPERATED
PNEUMATIC
CYLINDER
ACTUATOR

- B. DETERMINE MOTOR LENGTH AND ELECTRICAL POWER FROM FIGURES III-41 THROUGH III-46
- C. TORQUE MOTOR DELAY IS $< .002$ SEC BY DESIGN LIMIT. THEREFORE OPENING AND CLOSING RESPONSE = $(t + .002)$ SEC
- D. FROM MOTOR LENGTH AND POWER DETERMINE ENVELOPE DIMENSIONS FROM FIG. III-60 AND WEIGHT FROM FIGURE III-61.
- E. ADD SHUTOFF DEVICE WEIGHT AND ENVELOPE VALUES TO ACTUATOR WEIGHT AND ENVELOPE VALUES TO OBTAIN TOTAL UNIT CHARACTERISTICS
- A. FOR VALUE OF TORQUE (T_v) AND MOTION TIME (t) CALCULATE REQUIRED ACTUATOR OUTPUT FORCE (F_v) UTILIZING FACTOR SELECTED FROM FIG. III-48 AT A SELECTED ACTUATOR ACCELERATION VALUE
- B. FOR VALUES OF F_v , P_1 AND ACCELERATION, DETERMINE PISTON DIAMETER (D_p) FROM FIGURES III-31 THROUGH III-35.
- C. FOR VALUES OF MOTION TIME (t) AND ACCELERATION, DETERMINE OPENING PNEUMATIC DELAY FROM FIGURE III-37.
APPROPRIATE PNEUMATIC MEDIA PER TABLE III-XVIII.
- D. THE CLOSING PNEUMATIC DELAY CAN BE CALCULATED BY SOLVING A COMPLEX FUNCTION, BUT DOES NOT LEND ITSELF TO A TWO DIMENSIONAL PLOT

(3)

E. DETERMINE SHUTOFF DEVICE
ENVELOPE DIMENSIONS AND WEIGHT
FROM FIGURES III-53 & III-54, RE-
SPECTIVELY, FOR VALUE OF FLOW
FACTOR ($C_d A_t$)

F. DETERMINE ACTUATOR ENVELOPE
DIMENSIONS AND WEIGHT FROM
FIGURES III-57 & 58, RESPECTIVELY
FOR VALUE OF PISTON DIAMETER
(D_p)

G. OPENING RESPONSE = MOTION TIME (t)
+ PNEUMATIC DELAY + COMMAND VALVE
RESPONSE (MAY BE KNOWN OR CAN BE
COMPUTED BY THIS ANALYTICAL
PROGRAM)

H. UNIT WEIGHT - SHUTOFF DEVICE
WEIGHT + ACTUATOR WEIGHT

I. UNIT ENVELOPE = SHUTOFF DEVICE
ENVELOPE + ACTUATOR ENVELOPE
(DESIGN CONSIDERATIONS MAY ALLOW
BOTH ELEMENTS TO BE CONTAINED
WITHIN SHUTOFF DEVICE ENVELOPE -
SUBJECTIVE JUDGMENT)

J. THE ANALYTICAL PROGRAM MAY BE
EMPLOYED TO DETERMINE:

1. ACTUATOR STROKE
2. SEAT DIAMETER
3. VALVE MOVING MASS
4. ACTUATOR SUPPLY VALVE FLOW
FACTOR REQUIRED
5. VOLUME OF GAS USED PER VALVE
CYCLE
6. ACTUATOR PRESSURES AT TIME OF
CLOSING MOTION INITIATION

ROTARY MOTION

AUXILIARY
PRESSURE
OPERATED
PNEUMATIC
CYLINDER
ACTUATOR

A. PROCEED AS IN (A) THROUGH (J) ABOVE
EXCEPT PISTON DIAMETER (D_p) SHALL
BE DETERMINED FROM FIGURE III-36 AT THE
SELECTED AUXILIARY PRESSURE, AND
ACTUATOR WEIGHT SHALL BE DETERMINED
FROM FIGURE III-59.

ROTARY MOTION

HYDRUALIC
OPERATED

CYLINDER
ACTUATOR

A. PROCEED AS IN (A) ABOVE (AUXILIARY PRESSURE OPERATED PNEUMATIC CYLINDER) EXCEPT USE FIGURES III-38 AND III-39 FOR PISTON DIAMETER DETERMINATION AND HYDRAULIC DELAY RESPECTIVELY

ROTARY MOTION

TORQUE MOTOR

- A. SELECT DESIRED MOTION TIME (t) AND ELECTRICAL POWER LIMIT
- B. DETERMINE MOTOR LENGTH AND ELECTRICAL POWER FROM FIGURES III-41 THROUGH III-46.
- C. TORQUE MOTOR DELAY IS $< .002$ SEC BY DESIGN LIMITATION. THEREFORE OPENING AND CLOSING RESPONSE = $(t + .002)$ SEC
- D. FROM MOTOR LENGTH AND POWER DETERMINE ENVELOPE DIMENSIONS FROM FIGURE III-60 AND WEIGHT FROM FIGURE III-61.
- E. ADD SHUTOFF DEVICE WEIGHT AND ENVELOPE VALUES TO ACTUATOR WEIGHT AND ENVELOPE VALUES OBTAIN TOTAL UNIT CHARACTERISTIC

VALVE PERFORMANCE CHARACTERISTICS RATING CHARTS AND LIMITS

PERFORMANCE PARAMETER	STUDY LIMIT	MAX. NO. ATTAINABLE POINTS	DISTRIBUTION
NOMINAL ELECTRICAL POWER @ 70°F, 32 vdc	112 WATTS	15	POWER - WATTS RATING POINTS
RESPONSE A. NOMINAL OPENING RESPONSE @ 28 vdc, 70°F	250 ms	20	NOM. OPENING RESPONSE - ms. RATING POINTS
B. MINIMUM OPEN TIME (= NOM. CLOSING TIME)	100 ms	20	MIN. OPEN TIME - ms. RATING POINTS
VALVE WEIGHT	$\frac{\text{THRUST}^*}{50}$ (0 TO 99 PSIA) $\frac{\text{THRUST}}{500}$ (100 TO 400 PSIA)	25	% OF LIMIT WEIGHT - LBS RATING POINTS

* THRUST = ENGINE THRUST OF APPLICATION

VALVE PERFORMANCE CHARACTERISTICS

RATING CHARTS AND LIMITS

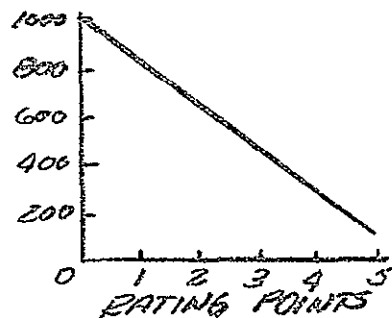
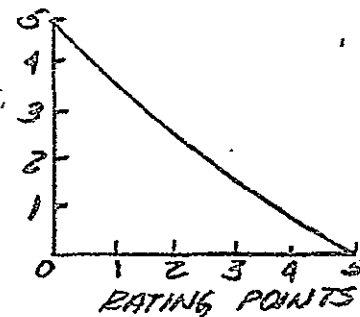
PERFORMANCE PARAMETER	STUDY LIMIT	MAX. NO. ATTAINABLE POINTS	DISTRIBUTION
VALVE VOLUME	1000 IN ³	5	
ACTUATOR PNEUMATIC SUPPLY		10	NONE REQUIRED - 10 OPERATING FLUID - 7 AUXILIARY SUPPLY - 0
ACTUATOR PNEUMATIC LOSS PER CYCLE	EQUIV. OF 5 MS OF VALVE RATED FLOW	5	

Figure III-86
(Continued)

To illustrate valve selection by this method and present a matrix format which provides ready reference to data items and to scoring, Tables III-XXI and III-XXII have been prepared for the hydrogen and oxygen valves, respectively, for a 1000 lb thrust, O/F of 2.0 and inlet pressure of 30 psia application. Tables III-XXIII and III-XXIV present similar data for the same application, except that inlet pressure is increased to 400 psia. The performance rating portion is based upon the sample rating system defined in Figure III-86. The data presented in these tables reflects the capability of the analysis program to predict detail performance data on the basis of the most elementary design criteria.

III-5 Detail Design

At this point in the program a design review was held with the NASA Houston technical monitor, for the purpose of reviewing overall program effort, and to finalize the selection of candidate valves, for two applications, for further design effort. The criteria established at this time required a revision in the inlet pressure, for the low pressure engine application, from 30 psia to 22 psia. Also, the pressure drop characteristic was revised such that it satisfied the relationship $\Delta P = .04 \times P_1$. This revised pressure drop characteristic is reflected in Figures III-87 and III-88 which present flow factor vs. inlet pressure relationships for various thrust and O/F levels. The rating system weighting factors were revised to more severely penalize a design requiring auxiliary pressurant since this additional requirement would add complexity to the total vehicle system design. In addition, the commonality of design of the fuel and oxidizer valves for a particular application was deemed essential.

These changes resulted in no change in the selected configuration for the high pressure (400 psia) application as shown in Tables III-XIII and III-XIV and the agreed upon configuration would be a line pressure pneumatic cylinder actuated linear motion shut-off valve. The impact on the low pressure application (P_1 changed from 30 to 22 psia) was significant and the analysis program was employed to generate new predicted characteristics. At this time, more in-depth analysis was performed to refine the characteristics predictions and the rating system was modified to impose a severe penalty on designs requiring auxiliary pressure. A summary of this analysis and the subsequent application of the revised rating system to the data is presented in Table III-XXV. Also included in this table, are data for aluminum construction configurations and a line pressure pneumatic operated poppet valve incorporating a novel sliding seat seal which would significantly reduce force requirements. The final selection of the candidate configuration for detail design was based on only those configurations for which rating points are shown, and is a line pressure pneumatic cylinder actuated poppet valve.

Detail design of the two selected configurations commenced with an analysis effort employing the analysis program applicable to the configuration but with several iterations to refine specific criteria. Minor iterations of detail element design criteria were performed to optimize the design and establish design margins. The design criteria were then issued for preparation of drawings.

The detail design layout for the hydrogen valve for a 1000 lb thrust, O/F of 2.0 and inlet pressure of 400 psia application is shown in Figure III-89 (TMC drawing L4687). The coaxial configuration of the valve results in the highest density packaging. All stainless steel construction was selected since minimal weight savings can be achieved by utilizing aluminum for sections which do not require magnetic properties (the solenoid operated pilot valve accounts for approximately 60% of the material volume).

90 April
MONTANA (1944)

丁巳年

500 LB 43

100-25-1-2

Figure III-87

CONFIGURATION CODE

(Reference TABLES III-XXI through III-XXV)

SHUTOFF

I = LINEAR MOTION

II = ROTARY MOTION

ACTUATOR

A = SOLENOID

B = PNEUMATIC CYLINDER

C = HYDRAULIC CYLINDER

D = TORQUE MOTOR (ROTARY)

PERFORMANCE RATING

9/F = 2.0)

UNMATED 16HT	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELECT. POWER	NOM. OPENING RESPONSE	NOM. MIN. OPEN TIME	VALVE WEIGHT	VALVE VOLUME	AUX. PRESS.	PNEU. LOSS/CYCLE	TOTAL
25.	IN.	STD IN. ³	IN. ³								
2	5.2 DIA x 8.6	34.4			15	6	16	2	7	3	49
5	5.2 DIA x 8.45	25.2			13	2	18	3	7	3	46
7	5.2 DIA x 5.4	22.8			9	0	19	3	7	4	42
1	1	22.8			7	0	19	3	7	4	40
1	1	22.8			4	0	19	3	7	4	37
5	5.2 DIA x 5.4	19.2			17	11	20	3	0	4	55
		17.12			15	8	20	3	0	4	50
		16.76			13	2	20	3	0	4	42
		16.50			9	0	20	3	0	4	36
		16.50			7	0	20	3	0	4	34
		16.50			4	0	20	3	0	4	31
	5.2 DIA x 5.4	19.2			17	11	21	3	0	4	56
		17.12			15	8	21	3	0	4	51
		16.76			13	2	21	3	0	4	43
		16.50			9	0	21	3	0	4	37
		16.50			7	0	21	3	0	4	35
		16.50			4	0	21	3	0	4	32
	5.2 DIA x 5.4		.1250		18	15	20	3	7	5	58
			.1110		16	11	20	3	7	5	52
			.1086		14	8	20	3	7	5	57
			.1074		11	3	20	3	7	5	49
			.1066		8	1	20	3	7	5	44
			.1066		6	0	20	3	7	5	41
	5.2 DIA x 5.4		.0624		18	15	21	3	7	5	59
			.0555		16	11	21	3	7	5	53
			.0538		14	8	21	3	7	5	58
			.0533		11	3	21	3	7	5	50
			.0532		8	1	21	3	7	5	45
			.0532		6	0	21	3	7	5	42
	4.8 x 9.2 x 5.3	90.0			14	7	12	2	7	1	43
	4.8 x 7.95 x 5.3	38.0			11	3	15	3	7	3	42
	4.8 x 7.95 x 5.3	34.3			11	3	15	3	7	3	42
	4.8 x 7.40 x 5.3	25.0			9	1	16	3	7	3	39
	4.8 x 7.05 x 5.3	22.0			9	1	16	3	7	4	40
	4.8 x 7.0 x 5.3	15.8			6	0	17	3	7	4	37
	4.0 x 7.2 x 5.3	16.8			6	0	17	3	7	4	37
	4.8 x 6.57 x 5.3	370			18	13	17	3	0	0	51
	4.8 x 6.02 x 5.3	107			16	9	17	3	0	1	46
	4.8 x 5.85 x 5.3	52.9			14	6	17	3	0	2	42
	4.8 x 5.75 x 5.3	24.7			11	2	17	4	0	3	37
	4.8 x 5.70 x 5.3	17.1			9	1	17	4	0	4	35
	4.8 x 5.67 x 5.3	10.4			6	0	17	4	0	4	31
	4.8 x 6.2 x 5.3	371			18	13	17	3	0	0	51
	4.8 x 5.85 x 5.3	106			16	9	17	3	0	1	46
	4.8 x 5.70 x 5.3	51.8			14	6	17	4	0	2	43
	4.8 x 5.66 x 5.3	24.2			11	2	17	4	0	3	37
	4.8 x 5.62 x 5.3	15.9			9	1	17	4	0	4	33
	4.8 x 5.60 x 5.3	12.3			6	0	17	4	0	4	33

ANALYSIS SUMMARY

TABLE

(FUEL VALVE - 1000 LB THRUST)

CONFIG. CODE		FLOW FACTOR C _d A _c IN ²	INLET PRESS. P _i PSIA	ELECT POWER W WATTS	ACTUATOR PNEUMATIC PRESS. PAUX. PSIA	ACTUATOR HYDRAULIC PRESS. PAUX. PSIA	NOMINAL OPENING TIME t _o MIS.	NOMINAL CLOSING TIME t _c MIS.	ESTIM. WEIGHT LBS
PT.	ACT								
I	B	6.5	30		30		25.0	40.0	7.22
							36.4	59.0	5.55
							63.0	99.0	5.30
							89.0	138	5.30
							120	197	5.30
					1500		12.1	20.0	5.45
							24.1	30.0	
							36.3	59.0	
							60.0	99.0	
							85.0	138	
							121	197	
					3000		12.1	20	5.13
							24.1	30	
							36.3	59	
							60.0	99	
							85.0	138	
							121	197	
C	6.5	30				1500	10.6	10.6	5.52
							21.1	21.1	
							31.7	31.7	
							52.8	52.8	
							73.9	73.9	
							105.6	105.6	
					3000		10.6	10.6	3.05
							21.1	21.1	
							31.7	31.7	
							52.8	52.8	
							73.9	73.9	
							105.6	105.6	
I	B	6.5	30		30		30.0	34.5	11.80
							50.1	57.0	7.50
							50.1	57.0	7.30
							70.3	81.0	6.85
							70.3	81.0	6.90
							100.2	123	6.65
							100.2	123	6.45
					1500		10.0	13.5	6.44
							20.0	25.0	6.18
							30.0	37.0	6.09
							50.1	59.0	6.06
							70.3	81.0	6.04
							100.2	117	6.03
					3000		10.0	13.5	6.27
							20.0	25.0	6.09
							30.0	37.0	6.05
							50.1	59.0	6.03
							70.3	81.0	6.03
							100.2	117	6.02

PERFORMANCE RATING

TESTED SHT	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELECT. POWER	NOM. OPENING RESPONSE	NOM. MIN. OPEN TIME	VALVE WEIGHT	VALVE VOLUME	AVG. PRESS.	PNEU. LOSS/CYCLE				TOTAL
S.	IN.	STD IN ³	IN ³											
	4.8x6.5x5.3		2.62		18	15	17	3	7					65
	4.8x6.02x5.3		1.695		16	11	17	3	7					59
	4.8x5.85x5.3		1.337		14	8	17	3	7					54
	4.8x5.75x5.3		1.169		11	3	17	3	7					47
	4.8x5.70x5.3		1.113		9	1	17	4	7					43
	4.8x5.67x5.3		1.092		6	2	17	4	7					41
	4.8x6.20x5.3		1.31		18	15	17	3	7					65
	4.8x5.85x5.3		1.346		16	11	17	3	7					59
	4.8x5.70x5.3		1.168		14	8	17	4	7					53
	4.8x5.66x5.3		1.085		11	3	17	4	7					47
	4.8x5.62x5.3		1.056		9	1	17	4	7					43
	4.8x5.60x5.3		1.045		6	2	17	4	7					41

ANALYSIS SUMMARY

FIG. NO.	ACT	FLOW FACTOR $C_d A_t$ IN^2	INLET PRESS. P_1 PSIA	ELECT. POWER W WATTS	ACTUATOR PNEUMATIC PRESS. $P_{AUX.}$ PSIA	ACTUATOR HYDRAULIC PRESS. $P_{AUX.}$ PSIA	NOMINAL OPENING TIME t_o ms.	NOMINAL CLOSING TIME t_c ms.	ESTIM. WEIGHT W_{E16} LBS.
C		6.5	30			1500	10.2	10.2	6.44
							20.5	20.5	6.18
							30.7	30.7	6.09
							51.4	51.4	6.06
							72.0	72	6.04
							103	103	6.23
						3000	10.2	10.2	6.27
							20.5	20.5	6.09
							30.7	30.7	6.05
							51.4	51.4	6.03
							72.0	72.0	6.03
							103	103	6.02

TABLE III-XXII

O/F = 2.0)

PERFORMANCE RATING

TESTED SMT	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELECT. POWER	NOM. OPERATING TIME	NOM. MIN. OPER. TIME	VALVE LIFT/HAT	VALVE VELOCITY	PLUN. PRESS.	PLUN. LOSS/CYCLE	TOTAL
S.	IN.	STD IN ³	IN ³								
5	4.15 DIA x 6	31.6			18	12	7	4	7	1	53
5	3.75 DIA x 6	12.89			15	7	15	5	7	1	54
5	3.75 DIA x 6	11.90			14	4	15	5	7	1	50
5	3.75 DIA x 3.75	10.50			11	1	16	5	7	1	45
	↓	↓			9	0	16	5	7	1	42
	↓	↓			6	0	16	5	7	1	39
	3.75 DIA x 3.75	8.32			18	12	19	5	5	5	59
	↓	7.75			15	7	19	5	0	5	51
	↓	7.75			14	4	11	5	0	5	47
	↓	7.60			11	1	19	5	0	5	41
	↓	7.60			9	0	19	5	0	5	38
	↓	7.60			6	0	19	5	0	5	35
	3.75 DIA x 3.75	8.32			18	12	20	5	0	5	60
	↓	7.75			15	7	20	5	0	5	52
	↓	7.75			14	4	20	5	0	5	47
	↓	7.60			11	1	20	5	0	5	42
	↓	7.60			9	0	20	5	0	5	39
	↓	7.60			6	0	20	5	0	5	36
	3.75 DIA x 3.75		.0525		18	14	19	5	7	5	68
	↓		.0480		15	11	19	5	7	5	62
	↓		.0480		14	7	19	5	7	5	57
	↓		.0470		10	3	19	5	7	5	49
	↓		.0470		7	1	19	5	7	5	44
	↓		.0470		5	0	19	5	7	5	41
	3.75 DIA x 3.75		.0263		18	14	20	5	7	5	69
	↓		.0290		15	11	20	5	7	5	63
	↓		.0290		14	7	20	5	7	5	58
	↓		.0230		10	3	20	5	7	5	50
	↓		.0230		7	1	20	5	7	5	45
	↓		.0230		5	0	20	5	7	5	42
	4.05 x 7.55 x 4.0	66.4			15	6	5	4	7	2	39
	3.8 x 6.25 x 4.0	26.5			14	3	11	5	7	9	44
	3.8 x 5.40 x 4.0	14.4			11	1	14	5	7	7	43
	3.8 x 5.05 x 4.0	9.2			9	0	15	5	7	5	41
	3.8 x 4.7 x 4.0	5.2			6	0	15	5	7	5	38
	3.8 x 4.70 x 4.0	108			18	0	13	5	0	1	39
	3.8 x 4.32 x 4.0	34.6			15	5	16	5	0	4	45
	3.8 x 4.20 x 4.0	16.6			14	2	16	5	0	5	42
	3.8 x 4.15 x 4.0	10.6			11	0	16	5	0	5	37
	3.8 x 4.10 x 4.0	7.6			9	0	16	5	0	5	35
	3.8 x 4.10 x 4.0	6.6									

NOT REPRODUCIBLE

ANALYSIS SUMMARY

(OR VALVE - 1000 LB THRU)

FIG. NO.	ACT	FLOW FACTOR $C_d A_c$ IN ²	INLET PRESS. P_1 PSIA	ELECT POWER W WATTS	ACTUATOR PNEUMATIC PRESS. PAUX. PSIA	ACTUATOR HYDRAULIC PRESS. PAUX. PSIA	NOMINAL OPENING TIME t_o MS.	NOMINAL CLOSING TIME t_c MS.	
B		3.25	30		30		10.0	16.3	2.25
							20.1	32.5	2.40
							30.2	48.8	3.32
							50.3	81.4	3.10
							70.6	113.9	3.10
							101	163	2.15
					1500		10.0	16.0	1.60
							20.1	32.0	
							30.2	48.7	
							50.3	81.3	
							70.6	110.8	
							101	162	
					3000		10.0	16.0	1.57
							20.1	12.0	
							30.2	48.7	
							50.3	81.3	
							70.6	110.8	
							101	162	
C		3.25	30		1500		10.8	10.8	1.59
							21.5	21.5	
							33.0	33.0	
							54.0	54.0	
							77.0	77.0	
							115	110	
					3000		10.8	10.8	1.56
							21.5	21.5	
							33.0	33.0	
							54.0	54.0	
							77.0	77.0	
							115	115	
B					30		20.1	37	9.80
							30.2	54	5.00
							51.5	80	3.80
							72.0	91	3.50
							102	131	3.30
					1500		10.0	143	3.31
							20.0	45	3.12
							30.1	61	3.05
							50.2	91	3.04
							70.3	113	3.03
							101	167	3.02

TABLE III-XXII (Continued)

PERFORMANCE RATING													
UNLIMITED 16 HT	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELECT. POWER	NOM. OPENING TIME	NOM. MIN. OPEN TIME	VALVE WEIGHT	VALVE VOLUME	AUX. PRESS.	PNEU. LOSS/CYCLE			
35.	IN.	STD IN ³	IN ³										TOTAL
8	3.8x4.48x4.0	52.7			18	2	16	5	0	3			44
7	3.8x4.23x4.0	17.1			15	5	16	5	0	5			46
4	3.8x4.13x4.0	8.3			14	2	16	5	0	5			42
3	3.8x4.08x4.0	5.1			11	0	16	5	0	5			37
2	3.8x4.06x4.0	3.8			9	0	16	5	0	5			35
1	3.8x4.05x4.0	3.16			6	0	16	5	0	5			32
3	3.8x4.57x4.0		.704		18	15	15	5	7	5			65
2	3.8x4.26x4.0		.203		15	11	16	5	7	5			59
5	3.8x4.16x4.0		.106		14	8	16	5	7	5			55
2.5	3.8x4.03x4.0		.026		11	3	16	5	7	5			47
2	3.8x4.01x4.0		.019		8	1	16	5	7	5			43
2	3.8x4.00x4.0		.016		6	0	16	5	7	5			39
3	3.8x4.38x4.0		.360		18	15	16	5	7	5			66
1	3.8x4.17x4.0		.114		15	11	16	5	7	5			59
5	3.8x4.09x4.0		.057		14	8	16	5	7	5			55
	3.8x4.0x4.0		.016		11	3	16	5	7	5			47
	3.8x4.0x4.0		.016		9	1	16	5	7	5			43
	3.8x4.0x4.0		.016		6	0	16	5	7	5			39

ANALYSIS SUMMARY

FIG. CODE		FLOW FACTOR	INLET PRESS.	ELECT POWER	ACTUATOR PNEUMATIC PRESS.	ACTUATOR HYDRAULIC PRESS.	NOMINAL OPENING TIME	NOMINAL CLOSING TIME	ESTIM. WEIGHT
UT. F	ACT	$C_d A_t$ IN ²	P_1 PSIA	W. WATTS	$P_{AUX.}$ PSIA	$P_{AUX.}$ PSIA	t_o MS.	t_c MS.	LEBS
I	B	3.25	30		3000		10	62	3.18
							20	44	3.07
							30.1	61	3.04
							50.2	91	3.03
							70.3	113	3.02
							101	162	3.02
I	-	3.25	30			1500	10.5	10.5	3.23
							21.2	21.2	3.08
							31.7	31.7	3.05
							53.0	53.0	3.02
							74.0	74.0	3.02
							110	110	3.02
							10.5	10.5	3.13
							21.2	21.2	3.05
							31.7	31.7	3.03
							53.0	53.0	3.02
							74.0	74.0	3.02
							110	110	3.02

PERFORMANCE RATING

ITEM NO.	ENVELOPE IN.	PNEU. GAS USED/CYCLE STD IN ³	HYD. FLUID USED/CYCLE IN ³	ELECT. POWER	NOM. OPENING TIME	NOM. MIN. OPEN TIME	VALVE WEIGHT	VALVE VOLUME	AUX. PRESS.	PNEU. LOSS/CYCLE	TOTAL
7	4.10m x 2.6				11	4	2	5	10	5	37
7	3.6 DIA x 2.8				12	6	0	5	10	5	38
7	3.0 DIA x 3				13	7	0	5	10	5	40
1	1.8 x 3.0 x 1.8	.750			17	12	7	5	7	5	55
2	1.8 x 2.9 x 1.8	.421			14	7	10	5	7	5	48
		.360			12	3	10	5	7	5	42
		.330			9	0	10	5	7	5	36
		.300			6	0	10	5	7	5	33
		.280			2	0	10	5	7	5	29
1	1.8 x 2.65 x 1.8	.750			17	12	10	5	0	5	49
7	1.8 x 2.6 x 1.8	.421			14	7	10	5	0	5	41
		.360			12	3	10	5	0	5	35
		.330			9	0	10	5	0	5	29
		.300			6	0	10	5	0	5	26
		.280			2	0	10	5	0	5	22
5	1.8 x 2.57 x 1.8	.750			17	12	10	5	0	5	49
7	1.8 x 2.54 x 1.8	.421			14	7	11	5	0	5	42
		.360			12	3	11	5	0	5	36
		.330			9	0	11	5	0	5	30
		.300			6	0	11	5	0	5	27
		.280			2	0	11	5	0	5	23
	1.8 x 2.65 x 1.8		.0165		17	14	10	5	0	5	51
	1.8 x 2.6 x 1.8				15	10	10	5	0	5	45
					13	8	10	5	0	5	41
					10	3	10	5	0	5	33
					8	1	10	5	0	5	29
					5	0	10	5	0	5	25
	1.8 x 2.57 x 1.8		.0082		17	14	10	5	0	5	51
	1.8 x 2.54 x 1.8				15	10	11	5	0	5	46
					13	8	11	5	0	5	42
					10	3	11	5	0	5	34
					8	1	11	5	0	5	30
					5	0	11	5	0	5	26

ANALYSIS SUMMARY

TAB. 1

Fig. OF	FLOW FACTOR $C_d A_c$ IN ²	INLET PRESS. P_1 PSIA	ELECT POWER W WATTS	ACTUATOR PNEUMATIC PRESS. P_{aux} PSIA	ACTUATOR HYDRAULIC PRESS. P_{aux} PSIA	NOMINAL OPENING TIME t_o MS.	NOMINAL CLOSING TIME t_c MS.	ESTIMATE WEIGHT LBS.
ACT								
D	.62	400	112			50 40 33	50 40 33	1.47 1.77 1.74
B	.62	400		400 (.554 ACT. STROKES)		13.4 26.7 40.0 67.0 94.0 134	17.7 35.0 53.0 88.0 123 176	.655 .620 ↓
				1500		13.4 26.7 40.0 67.0 94.0 134	17.7 35.0 53.0 88.0 123 176	.610 .595 ↓
				3000		13.4 26.7 40.0 67.0 94.0 134	17.7 35.0 53.0 88.0 123 176	.585 .580 ↓
C					1500	10.9 21.8 32.7 54.5 76.3 109	10.9 21.8 32.7 54.5 76.3 109	.610 .595 ↓
					3000	10.9 21.8 32.7 54.5 76.3 109	10.9 21.8 32.7 54.5 76.3 109	.585 .580 ↓

TABLE III-XXIII (Continued)

				PERFORMANCE RATING									
TEST, $O/F = 2.0$	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELECT. POWER	NOM. OPENING TIME	NOM. PNEU. OPEN TIME	VALVE WEIGHT	VALVE VOLUME	NO. PRESS.	PNEU. GAS/CYCLE			TOTAL
	IN.	STD IN ³	IN ³										
3	1.8 DIA x Z	2.55			16	13	12	5	7	5			58
		2.53			13	8	12	5	7	5			50
		2.53			11	5	12	5	7	5			45
		2.53			7	1	12	5	7	5			37
		2.53			1	0	12	5	7	5			33
		2.53			2	0	12	5	7	5			31
	1.8 DIA x Z	2.50			16	13	13	5	0	5			52
		2.50			13	8	13	5	0	5			44
		2.50			11	5	13	5	0	5			39
		2.50			7	1	13	5	0	5			31
		2.50			4	0	13	5	0	5			27
		2.50			2	0	13	5	0	5			25
	1.8 DIA x Z	2.52			16	13	13	5	0	5			52
		2.52			13	8	13	5	0	5			44
		2.52			11	5	13	5	0	5			39
		2.52			7	1	13	5	0	5			31
		2.52			4	0	13	5	0	5			27
		2.52			2	0	13	5	0	5			25
	1.8 DIA x Z				17	14	13	5	0	5			54
					15	10	13	5	0	5			48
					13	7	13	5	0	5			43
					9	2	13	5	0	5			34
					7	1	13	5	0	5			31
					5	0	13	5	0	5			28
	1.8 DIA x Z				17	14	13	5	0	5			54
					15	10	13	5	0	5			48
					13	7	13	5	0	5			43
					9	2	13	5	0	5			34
					7	1	13	5	0	5			31
					5	0	13	5	0	5			28

ANALYSIS SUMMARY

(FUEL VALVE - 1000 LB. THRU)

FIG. NO.	FLOW FACTOR C _d A _t IN ²	INLET PRESS. P _i PSIA	ELECT POWER W WATTS	ACTUATOR PNEUMATIC PRESS. P _{aux} PSIA	ACTUATOR HYDRAULIC PRESS. P _{aux} PSIA	NOMINAL OPENING TIME t _o MS.	NOMINAL CLOSING TIME t _c MS.	ESTIMATED WEIGHT LBS.
B	.62	400		400		16.7	15.0	.48
						33.0	30.0	
						50.0	45.0	
						84.0	75.0	
						117	105	
						168	150	
				1500		16.7	15.0	.46
						33.0	30.0	
						50.0	45.0	
						84.0	75.0	
						117	105	
						168	150	
				3000		16.7	15.0	.45
						33.0	30.0	
						50.0	45.0	
						84.0	75.0	
						117	105	
						168	150	
C	.62	400			1500	11.8	11.8	.45
						23.6	23.6	
						35.4	35.4	
						59.1	59.1	
						82.7	82.7	
						118	118	
					3000	11.8	11.8	.44
						23.6	23.6	
						35.4	35.4	
						59.1	59.1	
						82.7	82.7	
						118	118	

- 181 -

114

114

114

TABLE III-XXIV (Continued)

PERFORMANCE RATING											
IMATED 16 HT	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELECT. POWER	NOM. DISCHARGE RESPONSE	NOM. MIN. OPEN TIME	VALVE WEIGHT	VALVE VOLUME	AUX. PRESS.	PNEU. LOSS/CYCLE	TOTAL
PS.	IN.	STD IN. ³	IN. ³								
3	1.75x2.2x1.75	7.25 6.15 5.80 5.13 ↓			18 16 14 11 8 6	6 4 2 1 0 0	24 24 24 24 24 24	5 5 5 5 5 5	7 7 7 7 7 7	5 5 5 5 5 5	65 61 57 53 49 47
	1.75x2.10x1.75	6.77 5.42 5.42 4.80 ↓			18 16 14 11 8 6	6 4 2 1 0 0	24 24 24 24 24 24	5 5 5 5 5 5	0 0 0 0 0 0	5 5 5 5 5 5	58 54 50 46 42 40
	1.75x2.05x1.75	6.54 5.40 5.40 4.96 ↓			18 16 14 11 8 6	6 4 2 1 0 0	24 24 24 24 24 24	5 5 5 5 5 5	0 0 0 0 0 0	5 5 5 5 5 5	58 54 50 46 42 40
	1.75x2.05x1.75		.042 .034 ↓		18 16 13 10 8 5	7 5 4 2 1 0	24 24 24 24 24 24	5 5 5 5 5 5	7 7 7 7 7 7	5 5 5 5 5 5	66 62 58 53 50 46
	1.75x2.05x1.75		.023 .019 ↓		18 16 13 10 8 5	7 5 4 2 1 0	24 24 24 24 24 24	5 5 5 5 5 5	7 7 7 7 7 7	5 5 5 5 5 5	66 62 58 53 50 46
	2.8x2.8x2.85			5	6	0	21	5	10	5	52
	3.5x3.5x2.55				10	2	19	5	10	5	51
	3.5x3.5x2.55				8	1	19	5	10	5	48
	4.1x4.1x2.35				6	0	19	5	10	5	45

ANALYSIS SUMMARY

FIG. CODE		FLOW FACTOR	INLET PRESS.	ELECT POWER	ACTUATOR PNEUMATIC PRESS.	ACTUATOR HYDRAULIC PRESS.	NOMINAL OPENING TIME	NOMINAL CLOSING TIME	ESTIM. WGT.
IT	ACT	$C_d A_t$ IN ²	P_1 PSIA	W WATTS	P_{AUX} PSIA	P_{AUX} PSIA	t_o MS.	t_c MS.	LBS
I	B	.31	400		400		10	16.2	.43
							20	32.5	
							30.1	48.8	
							50.2	81.0	
							70.4	113	
							101	160	
					1500		10	16.2	.43
							20	32.5	
							30.1	48.8	
					3000		50.2	81.0	.43
							70.4	113	
							101	160	
II	C	.31	400		1500		10.4	10.4	4.3
							21.0	21.0	
							31.5	31.5	
							53.0	53.0	
							74.0	74.0	
							112	112	
					3000		10.4	10.4	4.3
							21.0	21.0	
							31.5	31.5	
							53.0	53.0	
							74.0	74.0	
							112	112	
	D	.31	400	84			102	102	1.10
				112			52.0	52.0	1.58
							72.0	72.0	1.58
							102	102	1.45

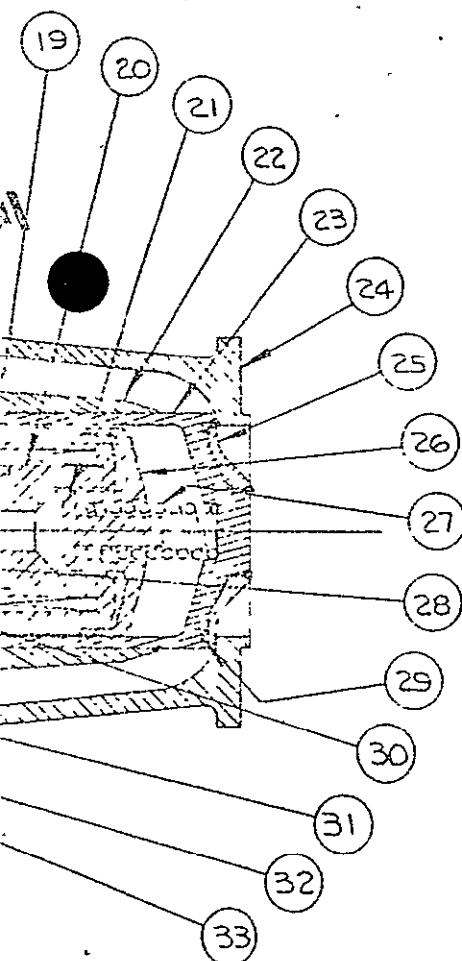
PERFORMANCE RATING

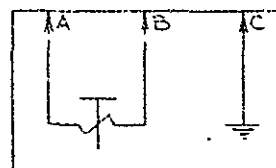
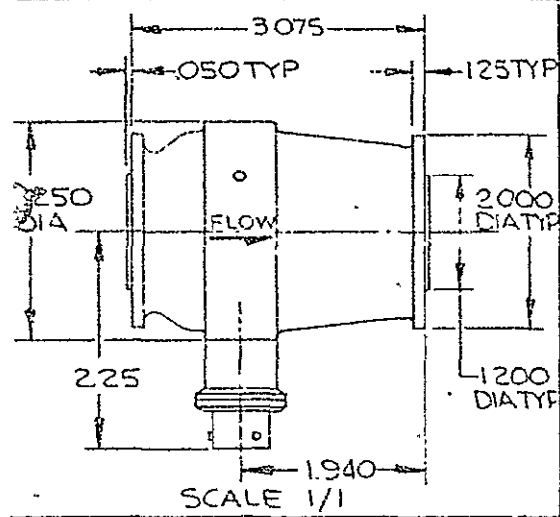
TESTED SHT	ENVELOPE	PNEU. GAS USED/CYCLE	HYD. FLUID USED/CYCLE	ELCT. POWER	NUM. DOCKING TIME	NUM. MIN. DRY TIME	VACUUM CURTAIN	VACUUM CURTAIN	VACUUM CURTAIN	FLUX PRESS.	FLUX LOSS/CYCLE	TOTAL
17	9.4 DIA x 9.4			12	4	9	3	3	15			46
28				8	4	10	4	3	15			44
34				0	4	11	4	3	15			37
	8.35 DIA x 8.1				17	0	8	2	0			27
					16	1	8	2	0			26
					14	0	8	2	0			24
					11	0	8	2	0			21
					9	0	8	2	0			19
					6	0	8	2	0			16
	8.5 x 8.1 x 7.4				9		0	1	15			
	12.5 x 8.1 x 7.4				6		0	1	15			
7	11.8 x 8.1 x 7.4				16	0	0	1	0			17
	9.9 x 8.1 x 7.4				14	0	1	2	0			17
	9.5 x 8.1 x 7.4				11	0	1	2	0			14
	9.2 x 8.1 x 7.4				9	0	1	2	0			12
	9.1 x 8.1 x 7.4				6	0	1	2	0			9
7	9.4 x 12 x 7.4			0	4	0	0	1	25	5		35
	9.4 x 12 x 7.4											
2	9.4 DIA x 9.4											
3												
9												
17	9.4 DIA x 9.4											
28												
34												

ANALYSIS SUMMARY

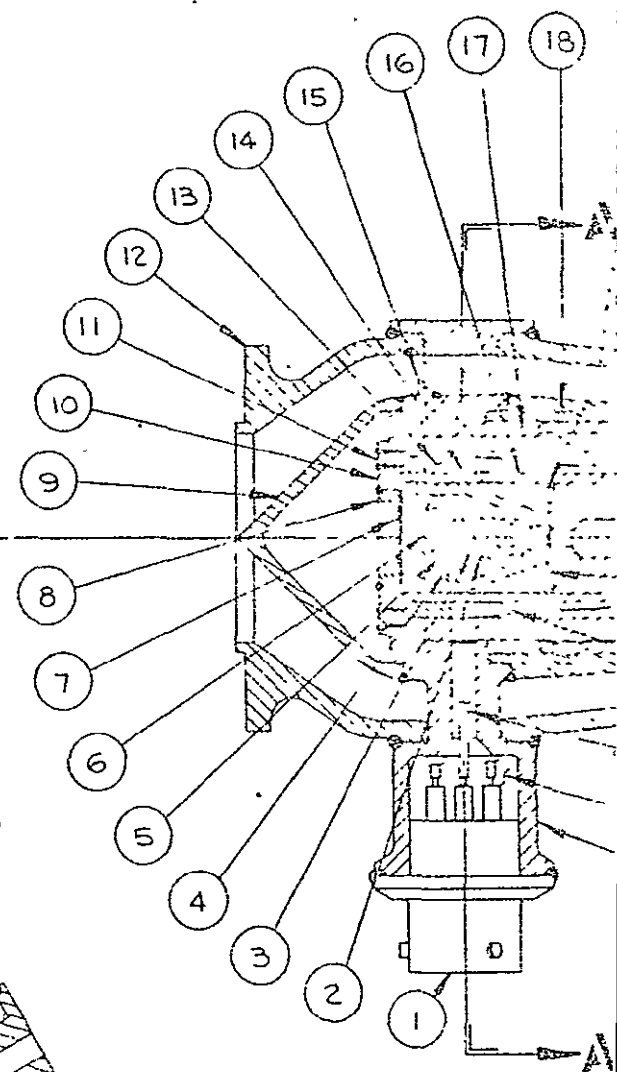
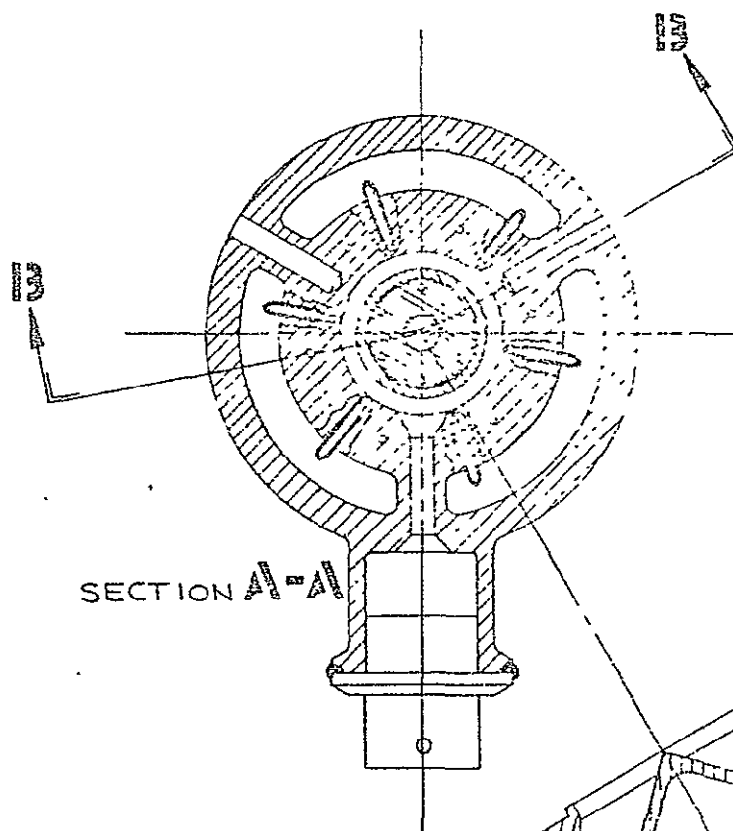
CONFIG. CODE		FLOW FACTOR $C_d A_c$ IN^2	INLET PRESS. P_1 PSIA	ELECT POWER W WATTS	ACTUATOR PNEUMATIC PRESS. $P_{AUX.}$ PSIA	ACTUATOR HYDRAULIC PRESS. $P_{AUX.}$ PSIA	NOMINAL OPENING TIME t_o MIS.	NOMINAL CLOSING TIME t_c MIS.	ESTIM. WEIGHT LBS.
I	B	17.35 CRES. CONST.	22	56	22		144.5	28.3	14.1
				84			138.3	22.2	13.0
				112			135.7	19.6	12.8
							10.1	132	7.91
							20.1	85	7.85
							30.1	95	7.84
							50.2	145	7.83
							70.4	195	7.82
							100.9	270	7.82
I	B	CRES. CONST.	22		1500		70.4		26.6
							100.9		23.1
							20.1	< 300	20.7
							30.1	< 300	18.2
							50.2	265	18.0
							70.2	193	17.8
I	D	CRES. CONST.	270				165	165	22.9
I	D	ALUM. CONST.	270				158	158	11.6
I	B	ALUM. CONST.		56			144.5	28.3	6.7
				84			138.3	22.2	5.63
				112			135.7	19.6	5.39
I	B	CRES. CONST. (SLIDE SEAL)		56			59.4	120	14.1
				84			53.3	103.4	13.0
				112			50.7	100.8	12.8

WATTS AT 32 V.D.C AND 70°F)
REMOTE INTERFACE IF REQUIRED.

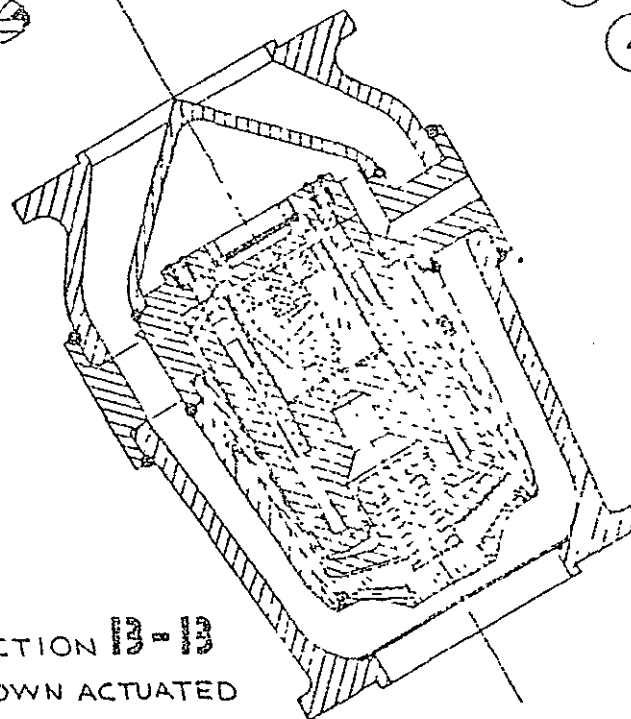
[illegible][illegible]



- NOTES UNLESS OTHERWISE SPECIFIED
1. WELD TYPE FLANGES ARE SHOWN. WELDED, BOLTED OR THREADED INTO
 2. ELECTRICAL RECEPTACLE SHOWN
 3. TEMPERATURE RANGE 200°-660°
 4. INLET PRESSURE 400 PSIA.
 5. ENGINE THRUST 1000 LBS. (O/F = 7)
 6. VOLTAGE RANGE 18-32 V.D.C. (28 V.D.C.)
 7. VENT PORT MAY BE PLUMED TO REE



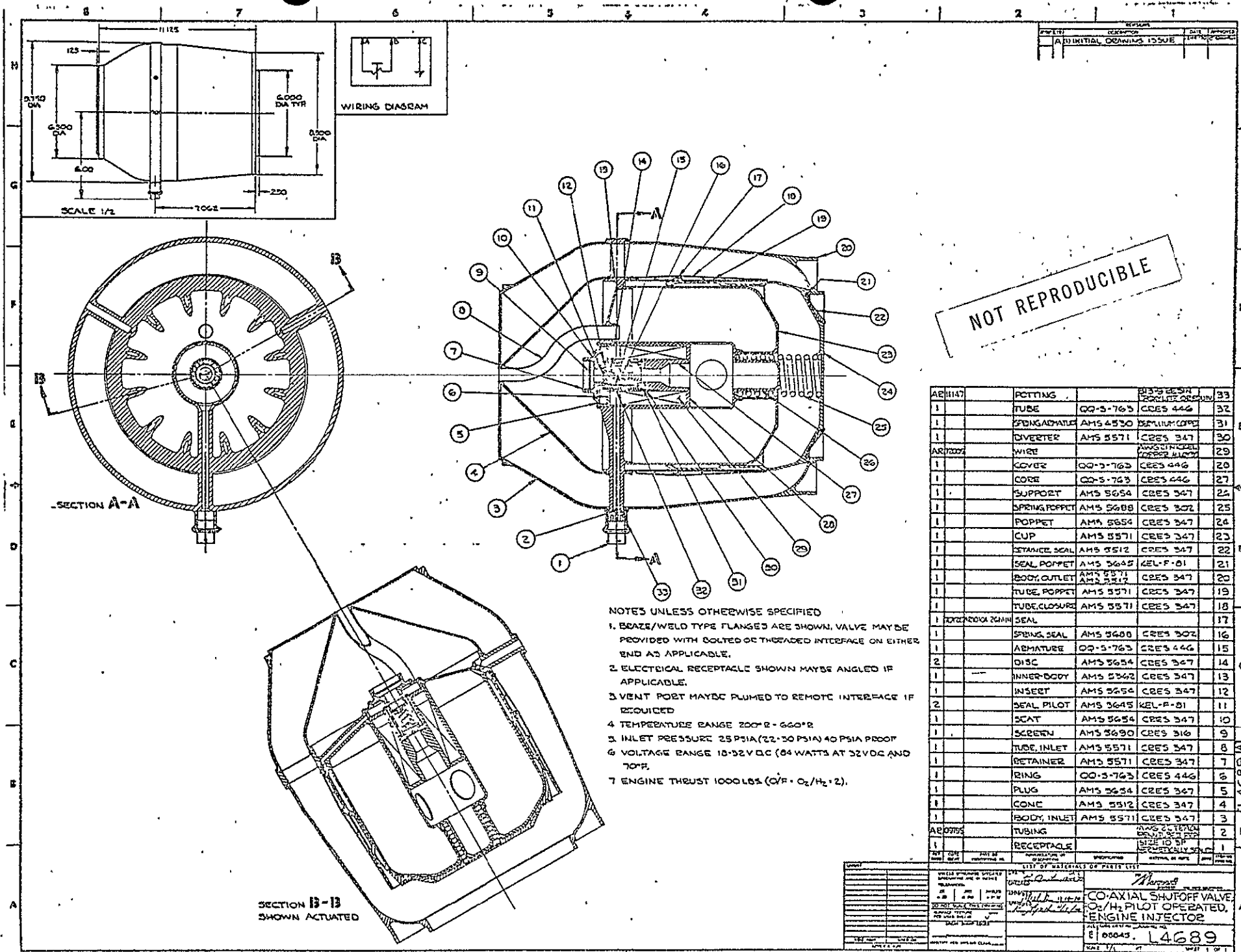
SECTION B-B
 SHOWN ACTUATED



As shown, the main valve seat and the pilot valve seats are KEL-F, in contact with defined lands, and have controlled compression by virtue of metal to metal stops. The seats are maintained in their normal position by spring preloads which assure adequate seat stress to preclude leakage at all pressures up to operating and resistance to dynamic environments up to 10 g's. In the closed position, inlet pressure acts to pressure unbalance the seats in the normally closed position, proportional to the magnitude of the pressure differential across the seat. Inlet pressure fills all internal cavities, except the centerbody inlet cone when the valve is in the normally close, deenergized position. Pressure to the volume within the poppet enters through tubes, projecting into the main flow annulus, passes through the normally open seat of the coaxial pilot valve and into the volume within the main poppet through manifold passages. Energizing the solenoid coil creates tractive magnetic forces which move the pilot valve armature such that the passage from the main flow annulus is blocked and the manifold from the volume within the main poppet is opened to the volume within the inlet cone (the inlet cone interior volume is open to ambient pressure via drilled passages in the struts which support the centerbody). As pressure within the main poppet volume bleeds down inlet pressure, acting on the annular area between the seat sealing diameter and the seal at the major diameter of the poppet, creates force which ultimately exceeds the main poppet spring force and causes the main poppet to move to the open position. So long as the pilot solenoid remains energized to vent the main poppet interior volume to ambient, pressure forces act to hold the main poppet in the open position. Upon termination of the electrical signal to the pilot solenoid coil, the pilot valve armature is returned to its normal position by spring force, such that the interior volume of the main poppet is open to inlet pressure. As pressure in this volume builds up, pressure forces holding the main poppet open, diminish until the spring causes the poppet to move to its normally closed position. For the design shown, the nominal (70°F, 28 vdc) opening and closing response times are 0.028 seconds and 0.27 seconds, respectively at an inlet pressure of 400 psia.

The materials and design concepts shown, all reflect current state-of-the-art capability. The "O-ring" dynamic seal at the main poppet maximum diameter was selected on the basis of the seal experiencing pressure differential only during valve open condition and the ability to accommodate some leakage past this seal without jeopardizing the valve's operational capability, although some external leakage will result through the vent line should leakage of this seal occur. Sliding fit guidance of moving elements, together with soft seal seats is a workable concept even over the wide temperature range and prolonged cycle life. Sliding surfaces will be treated with hard surface coatings exhibiting excellent friction coefficients and the thin sections, required to provide structural stability over the temperature and pressure range, assure minimal thermal gradients within the valve that could jeopardize valve function.

The detail design for the low pressure (inlet pressure = 22 psia) application is shown in Figure III-90 (TMC drawing L4688). This design embodies the same principles of operation as the high pressure valves, but specific geometry and geometric relationships were required to accommodate the large flow factor ($C_d A_t$) required to meet a pressure drop of 1 psi at rated flow. In this design, structural stability and minimum weight became a predominant consideration. Thin wall ribbed construction of the main poppet was selected.



to minimize the mass of the moving element and assure rigidity of sealing and guidance surfaces. The relatively low operating pressure requires substantially large pilot valve flow passages to effect valve operation. As shown, the valve is predicted to have an opening response at 70°F, 28 vdc and 22 psia inlet pressure, of 0.143 seconds and closing response is predicted to be 0.022 seconds. The wide divergence in response time is related to the significant change in volume, within the main poppet, which occurs in opening and closing. Further refinement of the analysis and additional iterations to optimize specific design criteria should result in convergence of these response times.

Materials of construction of the designs have been selected on the basis of compatibility with the propellants and the total operating environment. Joint design of the 400 series stainless steel magnetic circuits consider changes in structural allowables resulting from the grain growth which occurs when this material is exposed to temperatures of welding or brazing.

KEL-F 81 (3-M Company polytrifluorochloroethylene) was selected for the soft seat material on the basis of compatibility with the propellants, desirable structural and thermal properties and retention of resilience at low temperatures. Vespel (DuPont Co. polyimide) is an alternate soft seat material possessing desirable properties. This relatively new material has not been used extensively in similar applications, therefore available information will have to be augmented by development test data before committing to this new material.

IV. CONCLUSIONS

As a result of the effort described in this report, a comprehensive program to predict valve performance and characteristics has been developed and demonstrated. The analysis program developed is of a generalized nature and is not refined to the fullest extent possible in that only limited configurations can be accommodated. In its present status, the analysis program is a valuable tool to both the systems designer and the valve designer, in that rapid analysis and comparison of several valve configurations may be rapidly and conveniently performed.

Employing the data generated by the analysis program to select the optimum valve configuration for a specific application requires the development of a valid rating system. The rating system must properly weigh each characteristic of the valves and proper figures of merit must be assigned to each item of data. The rating system must consider the total impact of each characteristic on the system, and relation to each other characteristic. The systematic method of developing a rating system, described in this report, should be employed to formulate an objective rating system for each application.

V. RECOMMENDATIONS

The eight month study effort described in this report resulted in the development of methods and techniques for predicting valve performance and physical characteristics for a wide range of valve sizes and operating pressures. As a result of this study, the following recommendations are offered as a logical extension of the accomplished effort:

1. Convert the analysis program to batch processing computer to preclude bit storage capacity limitation and realize more economical computer utilization.
2. Expand the analysis program to include additional actuator concepts and unique shut-off concepts.
3. Incorporate optimizing routines into the basic analysis programs for establishing the most favorable geometric relationships within each configuration as a function of performance goals.
4. Conduct systems integration studies to resolve optimum interface criteria with propellant supply system, vehicle power system, command control logic and thruster.
5. Expand the state-of-the-art evaluation of metal bellows, employing available NASA resources, with emphasis on characteristics peculiar to valve applications.
6. Continue detail design studies of selected valve configurations for specific application. Fabricate development hardware to demonstrate design concepts and substantiate analytical program. Conduct a comprehensive test program to demonstrate valve capability and document subelement performance and life characteristics.
7. Employing methods and techniques developed, expand capability to include such other gaseous propellant system components as isolation valves, pressure regulators and mass flow regulators.

VI. BIBLIOGRAPHY

1. "Aerospace Fluid Component Designers Handbook", Volumes I and II, Rev. A, by Glen W. Howell, TRW Systems, Contract No. AF04(611)-8385
2. "Advanced Valve Technology", Interim Reports and Final Report, TRW Systems, Contract No. NAS7-436
3. "Poppet and Seat Design Data for Aerospace Valves", Technical Report AFRPL-TR-66-147, July 1966, Rocketdyne Division of NAA, Contract No. AF04(611)-9712
4. "NASA Contributions to Advanced Valve Technology", Report No. NASA SP-5019, by L.C. Burmeister et al, Midwest Research Institute, 1967
5. "A Bibliography of Aerospace Valve and Fluid Component Technology", Technical Report AFRPL-TR-65-239, by G. Howell, TRW Systems, Contract No. AF04(611)-8385
6. "Advanced ACS Valve Development Program", Technical Report AFRPL-TR-69-250, by G. Pond and H. Wichmann, The Marquardt Company, Contract No. FO4611-68-C-0074
7. "Design and Fabrication of a Multiple Function Hydrogen Control Valve", Marquardt Report No. 6026, (Confidential), October 1963
8. "Bistable Actuator", Report No. MP-122A, by H. Wichmann, The Marquardt Corporation, Contract No. NAS9-1677, 1964
9. "Final Report - Two Way, Latching, D.C. Solenoid Control Valve", Report No. S-479, by R. A. Lynch, The Marquardt Corporation, Contract No. NAS9-4069, 1965
10. "Analysis, Design and Development of High Flow Helium Pressure Regulator, Volume II, by W. Thompson, S. Magee, A. Kelleher and D. Clark, Robertshaw-Fulton Controls Co., AFBMD-TR-60-72(II) Contract AF04(647)-161
11. "Final Report on the Development of Analytical Techniques for Bellows and Diaphragm Design," by T. M. Trainee, L. E. Hulbert, J. F. Lestingi, R. E. Keith, et al, Battelle Memorial Institute, AFRPL-TR-68-22 Contract AF04(611)-10532

BIBLIOGRAPHY(Continued)

12. "State of the Art Survey of Metallic Bellows and Diaphragms for Aerospace Applications" by L. E. Hulbert, R. E. Keith and T. M. Trainer, Battelle Memorial Institute, AFRPL-TR-65-215, Contract AF04(611)-10532
13. "Basic Criteria and Definitions for Zero Fluid Leakage", by R. W. Weiner, JPL, Technical Report No. 32-926, Contract NAS7-100
14. "Leakage Testing Handbook", by J. W. Marc, General Electric, NASA CR-952, Contract NAS7-396
15. "Development and Demonstration of Criteria for Liquid Fluorine Feed System Components", by Advanced Propulsion Staff - McDonnell Douglas Corporation, NASA CR72063, (DAC-60599), Contract NAS w 1351
16. "Flow of Fluids through Valves, Fittings, and Pipe" - Crane Co. Technical Paper 409

APPENDIX A

APL COMPUTER PROGRAM

FOR

FAST ACTING VALVES

MIR # 382

APL COMPUTER PROGRAM FOR FAST ACTING VALVES

September 18, 1970

This paper documents the APL computer program for the performance parameters of fast acting valves for gaseous propellants. The project was 5082: Parametric Valve Studies.

PREPARED BY:

D. Berndt
D. Berndt

APPROVED BY:

R. C. Stechman
R. C. Stechman, Manager
Development Engineering

APL Computer Program for Fast Acting Valves

LMSOD and RMSOD

18 September 1970

A. General Description

Two APL computer programs were written for the actuation parameters of a Linear Motion Shutoff Device (LMSOD) and a Rotary Motion Shutoff Device (RMSOD). The programs were designed to handle a large matrix of input variables; however, individual cases can be input also. The programs were written by D. Berndt and K. Drenning for Bob Lynch.

B. Input Procedure

The names of the programs required are:

LMSOD	:	Main program
DFT	:	Library sub program
EFT	:	Library sub program
QUBIC	:	Special sub program for LMSOD
<u>TITLE</u>	:	Special sub program for LMSOD
RMSOD	:	Main program
DFT	:	Library sub program

A description of the input variables are listed below:

LMSOD
 ENTER 1 TO PRINT VALUES OF LOCAL VARIABLES. OTHERWISE ENTER 0.

?: Printout of local variables is only required for program checkout.

0
 ENTER 1 FOR SHORT TITLE. OTHERWISE ENTER 0

?: The short title lists the name of the pneumatic gas and the variable names. The long title lists the above plus the definitions of variables and their units.

ENTER 1 TO PRINT OUT ANSWERS FOR HYDRAULIC ACTUATOR. OTHERWISE ENTER 0

?: 1
 ARE NEW DATA REQUIRED? ANSWER 1 FOR YES AND 0 FOR NO.

?: 1 New data means new values for the input variables. The input variables are global and are available from the previous use of the program. If "0" is entered, this is the last input.

INPUT CDAT VALUES
 ? Input positive numbers larger than zero. Sample input:

.62 .94 2.5 5.5 10 25 36
 INPUT P1 VALUES IN MATRIX FORM [I;J]

?: The rows of the matrix are the pressure values to be input. The columns correspond to the various CDAT values. The length of the columns has to equal the length of the CDAT input. Sample input:

0	0	60	150	400
0	30	60	150	400
20	30	60	150	0
20	30	60	0	0
20	30	60	0	0
20	30	0	0	0
20	30	0	0	0

Note: It is convenient to form a matrix and set it equal to some global variable (PX) before LMSOD is executed. Then for P1 input, enter PX. Wherever there is a zero in the matrix, the calculations are skipped over.

Continued next page

INPUT T VALUES

□: Input positive numbers larger than zero. Sample input:

.01 .02 .03 .05 .07 .1

INPUT POWER VALUES

□: Input positive numbers larger than zero. Sample input:

28 56 84 112

INPUT AUX PRESSURE VALUES

□: Input positive numbers larger than zero or input zero. If zero is the input, some of the calculations and printouts are skipped over. This is the only variable that can accept an input of zero. Sample input:

1500 3000

SELECT PNEUMATIC GAS : H₂ = 1 , O₂ = 2 , N₂ = 3 , HF = 4 . ENTER ONLY ONE NUMBER

□: 2 Only one pneumatic gas can be used for each execution of the program. Sample input:

This is all of the input. The printout of the title is next.

NOT REPRODUCIBLE

C. Output

The title (short or long) is the first printout. The numerical values of the variables is printed out after each loop of the calculations. If zero is printed out for a variable, it means there was no solution for that variable, or it means that one of the calculated inputs was zero. The solution to the equation for magnetic area AM produces either two roots or none. The actuator time TA is calculated directly from AM; therefore, TA has the same form as AM. When AM is zero, the 2nd, 34d -- nth line for TA and W are not printed out.

If the program is interrupted before the printout is completed, it can be completed by the proper branch. If it is interrupted in one of the LMSOD steps, branch to LAB2 (enter $\rightarrow$ LAB2). If it is interrupted in one of the sub programs, branch to zero (enter $\rightarrow$ 0). The data from one case may be lost.

The program ends by printing out *****.

D. Auxiliary Program

An auxiliary program named INPUT was written to conveniently list the input parameters. This program has no effect on the LMSOD program but does use the same global variables. The INPUT program lists the current values of the input parameters including:

Options on local variables, title, new data, and pneumatic gas.

CDAT	:	Valve flow factor
P1	:	Pressure
T	:	Time
W	:	Power

E. Nomenclature

→ GLOBAL ←

	<u>APL</u>	<u>ALGEBRAIC</u>	<u>NAME</u>	<u>UNITS</u>
<u>INPUT</u>	D10	-	HYDRAULIC CALC OPTION	-
	D9	-	LOCAL VARIABLE OPTION	-
	DB	-	SHORT TITLE OPTION	-
	D1	-	NEW DATA OPTION	-
	CDAT	$C_d A_t$	VALVE FLOW FACTOR	IN^2
	P1	P1	UPSTREAM PRESSURE (MATRIX)	PSIA
	T	t	VALVE MOTION TIME	SEC
	W	W	POWER	WATTS
	PAUX	-	AUXILIARY PRESSURE	PSIA
	GN	-	GAS ID 1=H ₂ 2=O ₂ 3=N ₂ 4=HE	-
	RHO	ρ_g	DENSITY OF GAS	-
	RG	R	GAS CONSTANT	-
	KG	$k = \gamma$	RATIO OF GAS HEAT CAPACITIES	-
	KGRP	-	CONSTANT FOR GAS	-
	CG1	C1	" " "	-
	TG	Tg	GAS TEMPERATURE	°R
<u>SOLENOID</u>	FO	$F_o - F_v$	OPENING FORCE	LBS
	S	S	STROKE	IN
	AM	AM	MAGNETIC AREA	IN^2
	TA	t_a	ACTUATOR TIME	SEC
	PS	P _s	P1 or PAUX	PSIA
	FA	FA	ACTUATOR FORCE	LB
	CDAO	$C_d A_o$	INLET ORIFICE AREA	IN^2
	TAO	T_{ao}	ACTUATOR TIME	SEC
	(CLOSE)			
	PO	P _o	OPENING CHAMBER PRESS	PSIA
<u>PNEUMATIC</u> (OPEN)	PO1	-	PRELIMINARY CALC OF PO	PSIA
	PO2	-	" " " "	PSIA
	PO3	-	FIRST GUESS OF PO	PSIA
	TC	t_c	ACTUATOR TIME	SEC
<u>HYDRAULIC</u>	FAH	FA	ACTUATOR FORCE	LB
	FAHO	-	PRELIM CALC OF FAH	LB
	FAHCK	-	CHECK CALC OF FAH	LB
	QH	Q	HYDRAULIC FLOW RATE	IN^3/SEC
	CDAOH	$C_d A_o$	INLET ORIFICE AREA	IN^2
	TAH	TA	ACTUATOR TIME	SEC

NOMENCLATURE

→ LOCAL ←

APL	ALGEBRAIC	NAME	UNITS
G		COUNTER FOR LOOP L11 (PO)	—
H		INDEX FOR PS AND LOOP L7	—
I		INDEX FOR CDAT	—
J		INDEX FOR P1 ROWS	—
K		INDEX FOR T	—
L		INDEX FOR W AND LOOP L4	—
M		INDEX FOR FO	—
D2	$= \sqrt{\frac{CDAT}{.51}}$	GROUP OF TERMS	—
D3		OPTION USED IN 'TITLE' PROG	—
D4		GROUP OF TERMS IN EQ FOR FO	—
D6		" " " " " " TA	—
D7		" " " " " " "	—
C1	$\frac{k-1}{2K}$	" " " "	—
C2		" " " " " " AM	—
C3		" " " " " " PO	—
C4		" " " " " " "	—
C5		" " " " " " AM	—
C7		" " " " " " "	—
C8		" " " " " " "	—
C9		" " " " " " FA	—
C10		" " " " " " FAH	—
C11		STORAGE FOR FAH, CDAOH, TAH, QH	—
AMO		PRIM CALC OF AM	IN ²
AMCK		CHECK CALC OF AM	IN ²
X1		RESULT OF QUDIC SUBPROG	—
X		1 AMO	—
FAO		PRIM CALC OF FA	LB
FACK		CHECK CALC OF FA	LB
GAS		GAS PROPERTIES	—
VO	VO	GROUP OF TERMS IN EQ FOR TC	—
FAC	FAC	FA USED IN CLOSING EQ	LB

F. Equations

LINEAR MOTION SHUT OFF DEVICE

SOLENOID

Eq No.

$$\textcircled{1} F_0 = 5\pi \sqrt{\frac{C_d A_t}{.51}} + P_1 \frac{\pi}{4} \left[\left(\sqrt{\frac{C_d A_t}{.51}} + .01 \right)^2 - .97 \frac{C_d A_t}{.51} \right] +$$

$$+ \frac{\left(\frac{C_d A_t}{.65} \right)^{1.66}}{185} \left(20 + \frac{\sqrt{\frac{C_d A_t}{.51}}}{722 t^2} \right)$$

$$\textcircled{2} S = \frac{1}{4} \sqrt{\frac{C_d A_t}{.51}}$$

$$\textcircled{3} A_M = \frac{1.73}{64} \left[F_0 + 1.25 \left(\frac{A_M}{.785} \right)^{3/2} \times .283 \left(\frac{S}{143.2 t^2} + 20 \right) \right]$$

$$\textcircled{4} t_a = - \frac{-.0863}{10^8} \times \frac{F_0 + .981 \left(\frac{A_M}{.785} \right)^{3/2} \left(\frac{-.00385}{t^2} + 5.66 \right)}{\frac{32^2}{W} \times 1.25 S}$$

$$\times \left[\frac{2.525 \left(55 \frac{A_M}{.785} + 8000 S + 90 \right)}{.01598 W} \right]^2 \times \ln .585$$

NOT REPRODUCIBLE

LMSOD

PNEUMATIC OPENING:

$$\textcircled{5} \quad F_A = F_0 + \frac{.293 \pi}{16} \left(\frac{F_A}{\frac{\pi}{4} .52 P_S} \right)^{3/2} \times \left(\frac{\sqrt{\frac{C_d A_T}{8.16}}}{193.2 t^2 + 20} \right)$$

$$\textcircled{6} \quad P_S = P_1 \text{ or } 1500 \text{ or } 3000$$

$$\textcircled{7} \quad C_d A_0 = \frac{F_A \sqrt{\frac{C_d A_T}{8.16}} P_S \sqrt{RT}}{7.25 P_S t 15 \sqrt{\frac{k}{k-1} [.52^{2/k} - .52^{k+1/k}]}}$$

$$\textcircled{8} \quad t_{a_0} = \frac{F_A / P_S}{k \times C_1 (C_d A_0) R \sqrt{T}}$$

Table: GAS	ρ	R	k	$\sqrt{\frac{k}{k-1} [.52^{2/k} - .52^{k+1/k}]}$	C ₁	T
	lb/in ³					°R
H ₂	3.06×10^{-6}	9190	1.4	.480	.141	530
O ₂	48.6×10^{-6}	577	1.4	.1180	.560	530
N ₂	42.8×10^{-6}	662	1.4	.1180	.573	530
He	5.95×10^{-6}	4630	1.66	.480	.210	530

LM50D

PNEUMATIC CLOSING

$$(10) F_{Ac} = F_A - \left(5\pi \sqrt{\frac{CdA^2}{.51}} + P_1 \frac{\pi}{4} \left[\left(\sqrt{\frac{CdA^2}{.51}} + .01 \right)^2 - .97 \frac{CdA^2}{.51} \right] \right)$$

$$(11) \frac{S + .010}{P_s} \left(\frac{F_{Ac}}{\frac{\pi}{4} D_p^2} + P_0 \right) = \frac{.010}{\frac{k-1}{2k}} \left[\left(\frac{P_s}{P_0} \right)^{\frac{k-1}{2k}} - 1 \right]$$

solve for P_0

$$(12) V_0 = \frac{F_A (S + .010)}{.52 P_s}$$

$$(13) t_c = \frac{V_0}{k C_1 (CdA_0) R \sqrt{T} \times \frac{k-1}{2k}} \left(\left(\frac{P_s}{P_0} \right)^{\frac{k-1}{2k}} - 1 \right)$$

LMSOD

HYDRAULIC

$$(14) F_A = F_0 + \left(\frac{\sqrt{\frac{C_d A^2}{8.16}}}{193.2 t^2} + 20 \right) \times \left[\frac{.283}{16} \pi \left(\frac{F_A}{\frac{\pi}{4} \cdot 8 P_s} \right)^{3/2} \right]$$

$$(15) Q = \frac{\pi}{4} \frac{F_A}{.785 \times .8 \times P_s} \times \frac{S}{t}$$

$$(16) C_d A_0 = \frac{\frac{\pi}{4} Q \times 60}{29.81 \times 7.48 \times 1728 \sqrt{\frac{.8 P_s}{\text{Sp. Gr.}}}} \quad \text{Sp. Gr.} = 1$$

$$(17) t_a = \frac{\frac{\pi}{4} \frac{F_A}{.785 (.8 P_s)} \times .05}{Q}$$

RMSOD

$$(18) T_0 = \frac{\sqrt{\frac{C_d A^2}{.707}} + .16}{14.14} \left[5\pi \left(\sqrt{\frac{C_d A^2}{.707}} + .13 \right) + P_1 \frac{\pi}{4} \left(\sqrt{\frac{C_d A^2}{.707}} + .16 \right)^2 - P_1 \frac{\pi}{4} \left(\frac{C_d A^2}{.707} \right) \right] +$$

$$+ .00123 \frac{\pi}{t^2} \left[\left(\frac{\sqrt{\frac{C_d A^2}{.707}} + .16}{1.414} \right)^5 - \left(\frac{.48 \sqrt{\frac{C_d A^2}{.707}}}{.707} + .023 \right)^5 \right]$$

Some of the equations were simplified to the following form before being programmed:

Eq No

⑤

$$F_A = F_0 + C_9 F_A^{3/2}$$

$$C_9 = \frac{.21292}{P_s^{3/2}} \left(\frac{\sqrt{C_d A t}}{551.87 t^2} + 20 \right)$$

⑩

$$F_{AC} = F_A - \left[22.0 \sqrt{C_d A t} + P_1 (.0462 C_d A t + .02199 \sqrt{C_d A t} + .0000785) \right]$$

⑪

$$C_3 P_0^{C_1} + (S + .010) P_0^{(C_1+1)} = C_4$$

$$C_1 = \frac{k-1}{2k}$$

$$C_3 = \frac{(S + .010) F_{AC}}{.7854} + \frac{.010}{C_1}$$

$$C_4 = \frac{.010 P_s^{(C_1+1)}}{C_1}$$

⑭

$$F_A = F_v + C_{10} F_A^{3/2}$$

$$C_{10} = C_9 \times .524$$

$$\left(\frac{.52}{.8} \right)^{3/2} = .524$$

⑮

$$Q = \frac{F_A S}{.8 t P_s}$$

⑯

$$C_d A_0 = .00061726 Q / \sqrt{P_s}$$

⑰

$$t_a = \frac{.06250 F_A}{Q P_s}$$

G. APL Programs and Sample Output

▽ LMSOD;G;H;I;J;K;L;M;D2;D3;D4;D6;D7;C1;C2;C3;C4;C5;C7;C8;C9;C10;C11;AMO;AMCK;X1;X;PACK;PAO;CAS;VO;F

[1] 'ENTER 1 TO PRINT VALUES OF LOCAL VARIABLES. OTHERWISE ENTER 0.'

[2] D9←□

[3] 'ENTER 1 FOR SHORT TITLE. OTHERWISE ENTER 0 '

[4] D3←0 D8←□

[5] 'ENTER 1 TO PRINT OUT ANSWERS FOR HYDRAULIC ACTUATOR. OTHERWISE ENTER 0 .'

[6] D10←□

[7] 'ARE NEW DATA REQUIRED? ANSWER 1 FOR YES AND 0 FOR NO.'

[8] D1←□

[9] →(D1=0)/LAB1

[10] 'INPUT CDAT VALUES'

[11] CDAT←,□

[12] 'INPUT P1 VALUES IN MATRIX FORM [J;J]'

[13] P1←□

[14] 'INPUT T VALUES'

[15] T←,□

[16] 'INPUT POWER VALUES'

[17] P←,□

[18] 'INPUT AUX PRESSURE VALUES'

[19] PAUX←(PAUX=0)/PAUX←,□

[20] 'SELECT PNEUMATIC GAS : H2 = 1 , O2 = 2 , N2 = 3 , HE = 4 . ENTER ONLY ONE NUMBER'

[21] GN←□

[22] LAB1:2p' ←

[23] TITLE

[24] 2p' '

[25] GAS←((3.06E-6 9190 1.4 0.48 0.141)×(GN=1))+((4.86E-5 577 1.4 0.48 0.56)×(GN=2))

[26] GAS←GAS+((4.28E-5 622 1.4 0.48 0.523)×(GN=3))+((5.95E-5 4630 1.66 0.485 0.21)×(GN=4))

[27] GAS←(RHO+GAS[1]),(PG+GAS[2]),(KG+GAS[3]),(KCPP+GAS[4]),(CG1+GAS[5]),TG+530

[28] H←J←J←K←L←I'←1

[29] LAB2:→(P1[I;J]=0)/LAB2

[30] D4←(5×01×D2)+((P1[I;J]×01÷4)×(((D2+0.01)×2)-0.97×((D2+(CDAT[I]÷0.51)×0.5)×2)))

[31] FO[M]←D4+(((CDAT[J]÷0.65)×1.66)÷185)×20÷D2÷722×T[K]×2

[32] S←D2×÷4

[33] C2←1.25×(÷0.785×1.5)×0.283×20+S÷193.2×T[K]×2

[34] C5←÷÷0.02703125×C2

[35] C7←((C5÷3)×3)-(FO[M]÷C2)+C5×(C5÷3)×2

[36] C8←(2×C5×(C5÷3))-3×(C5÷3)×2

[37] D6←D7←X1←TA←AM←0×13

[38] X1←C8 OUBJC C7

[39] X←X1-(C5÷3)×X1÷0

NOT REPRODUCIBLE



***** LINEAR MOTION SHUT OFF DEVICE *****

INPUT: W = POWER ~ WATTS
P1 = UPSTREAM PRESSURE ~ PSIA
PS = P1 OR PAUX ~ PSIA
SOLENOID ACTUATOR: FO = OPENING FORCE ~ LB
AM = MAGNETIC AREA ~ SQ IN
PNEUMATIC ACTUATOR: FA = ACTUATOR FORCE ~ LB
(OPENING): TAO = ACTUATOR TIME ~ SEC
(CLOSING): PO = OPENING CHAMBER PRESS ~ PSIA
HYDRAULIC ACTUATOR: FA = ACTUATOR FORCE ~ LB
TAO = ACTUATOR TIME ~ SEC
THE PNEUMATIC GAS IS :
--> OXYGEN <--

CDAT = VALVE FLOW FACTOR SQ IN
T = REQUIRED MOTION TIME ~ SEC
PAUX = AUXILIARY PRESSURE ~ PSIA
S = STROKE ~ INCH
TA = ACTUATOR TIME ~ SEC
CD*AO = INLET ORIFICE AREA ~ SQ IN
TC = ACTUATOR TIME ~ SEC
CD*AO = INLET ORIFICE AREA ~ SQ IN
Q = FLOW RATE ~ CUBIC IN/SEC

	FO	PS	S	FA	AM	CD*AO	TAO	TA	PO	TA	TC	W	CDAT	Q	P1	T
SOL	20.26		0.276		3.02	1.07		1.618E00		4.307E-01		28	0.62		60	0.01
								8.090E-01		2.153E-01		56				
								5.393E-01		1.436E-01		84				
								4.045E-01		1.077E-01		112				
PNEUMC		60		21.86		5.171E-03	6.766E-03	1.536E01		5.592E-03						
PNEUMC		1500		20.27		1.918E-04	6.766E-03	4.378E02		5.004E-03						
PNEUMC		3000		20.26		9.585E-05	6.766E-03	8.761E02		5.002E-03						
HYDRLC		1500		20.26		7.423E-06	1.814E-03						4.655E-01			
HYDRLC		3000		20.26		2.624E-06	1.814E-03						2.327E-01			
SOL	20.20		0.276		8.32	0.76		2.959E00		1.996E-01		28	0.62		60	0.02
								1.479E00		9.978E-02		56				
								9.863E-01		6.652E-02		84				
								7.397E-01		4.989E-02		112				
PNEUMC		60		21.26		2.514E-03	1.353E-02	1.602E01		1.030E-02						
PNEUMC		1500		20.21		9.560E-05	1.353E-02	4.379E02		1.001E-02						
PNEUMC		3000		20.20		4.779E-05	1.353E-02	8.761E02		1.000E-02						
HYDRLC		1500		20.20		3.701E-06	3.628E-03						2.321E-01			
HYDRLC		3000		20.20		1.308E-06	3.628E-03						1.160E-01			

LMSOD. SAMPLE OUTPUT

NOT REPRODUCIBLE

VRMSOD[]V

```

V RMSOD;I;J;K;M;D1;D2;D3;D4
[1] 'ARE NEW DATA REQUIRED? ANSWER 1 FOR YES AND 0 FOR NO.'
[2] D1←[]
[3] →(D1=0)/LAB1
[4] 'INPUT CDAT VALUES'
[5] CDAT←[]
[6] 'INPUT P1 VALUES'
[7] P1←[]
[8] 'INPUT T VALUES'
[9] T←[]
[10] LAB1: '      OPENING      CDAT      UPSTREAM      OPENING'
[11] '      TORQUE      PRESSURE      TIME'
[12] 2p ' '
[13] I←J←K←M←1
[14] D4←TO+0×1(pCDAT)×(pP1)×pT
[15] L2:D4[M]←((D2+0.16)÷14.14)×(5×01×D2+0.13)+(D3×((D2+0.16)×2))-(D3←P1[J]×01÷4)×(D2+(CDAT[I]÷0.707)*
0.5)*2
[16] TO[M]←D4[M]+(0.00123×01×T[K]*2)×(((D2+0.16)÷1.414)*5)-(((0.48×D2)÷0.707)+0.023)*5
[17] 10 2 13 2 15 0 16 2 DFT TO[M],CDAT[I],P1[J],T[Y]
[18] M←M+1
[19] I←I+1
[20] →(I≤pCDAT)/L2
[21] I←1
[22] J←J+1
[23] →(J≤pP1)/L2
[24] J←1
[25] K←K+1
[26] →(K≤pT)/L2
V

```

.RMSOD

∇ QUBIC[0]∇

```

∇ R←A QUBIC B;P;Q;C;D;U
[1] R←0×13
[2] P←A÷3
[3] Q←B÷2
[4] D←(Q×2)-P×3
[5] →(D>0)/L1
[6] →(D=0)/L2
[7] →(D<0)/L3
[8] L1:R[1]←((×C)×(|C←Q+D×0.5)*÷3)+(×C)×(|C←Q-D×0.5)*÷3
[9] →0
[10] L2:R[1]←2×(×C)×(|C←Q)*÷3
[11] R[2]←-(×C)×(|C)*÷3
[12] →0
[13] L3:U←-20Q+P×P×0.5
[14] R[1]←2×(P×0.5)×20U÷3
[15] R[2]←2×(P×0.5)×20(U÷3)+01×120÷180
[16] R[3]←2×(P×0.5)×20(U÷3)+01×240÷180

```

∇

QUBIC'

VTITLE[0]V

```

V TITLE
[1]  →D3/I2
[2]  →(D8=1)/I1
[3]  '
[4]  '
[5]  'INPUT:          W = POWER ~ WATTS          CDAT = VALVE FLOW FACTOR   SQ IN '
[6]  '                P1 = UPSTREAM PRESSURE ~ PSIA      T = REQUIRED MOTION TIME ~ SEC'
[7]  '                PS = P1 OR PAUX ~ PSIA             PAUX = AUXILARY PRESSURE ~ PSIA'
[8]  'SOLENOID ACTUATOR: FO = OPENING FORCE ~ LB        S = STROKE ~ INCH'
[9]  '                AM = MAGNETIC AREA ~ SQ IN         TA = ACTUATOR TIME ~ SEC'
[10] 'PNEUMATIC ACTUATOR: FA = ACTUATOR FORCE ~ LB      CD×AO = INLET ORIFICE AREA ~ SQ IN'
[11] '                (OPENING) TAO = ACTUATOR TIME ~ SEC'
[12] '                (CLOSING): PO = OPENING CHAMBER PRESS~PSIA TC = ACTUATOR TIME ~ SEC'
[13] 'HYDRAULIC ACTUATOR: FA = ACTUATOR FORCE ~ LB      CD×AO = INLET ORIFICE AREA ~ SQ IN'
[14] '                TAO = ACTUATOR TIME ~ SEC          Q = FLOW RATE ~ CUBIC IN/SEC'
[15] 'THE PNEUMATIC GAS IS :
[16] I1: ' →→ ' ;((GN=1)/'HYDROGEN'),((GN=2)/'OXYGEN'),((GN=3)/'NITROGEN'),((GN=4)/'HELIUM'); ' ←←'
[17] '
[18] I2: '          FO      S      AM      AM      TA      TA      W      CDAT      P1  T
[19] '          PS      FA      CD×AO      TAO      PO      TC      Q'

```

V

TITLE

```

      ▽ INPUT
[1]  ' '
[2]  'D0 = ';D9;' = ';(D9=0)/'DO NOT';' PRINT LOCAL VARIABLES'
[3]  'D8 = ';D8;' = ';(D8/'SHORT'),(D8=0)/'LONG';' TITLE'
[4]  'D10 = ';D10;' = ';(D10=0)/'DO NOT';' PRINT ANSWERS FOR HYDPAULIC ACTUATOR'
[5]  'D1 = ';D1;' = 'NEI' DATA APP ';(D1=0)/'NOT';' REQUIRED'
[6]  'GN = ';GN;' (1= H2, 2= O2, 3= H2, 4=HE)'
[7]  ' '
[8]  'CDAT = '; 10 2 DFT CDAT
[9]  'P1 = ';(17 0 ,(((OP1[1;])-1)*2)*P(10 0))) DFT P1
[10] ' '
[11] 'T = '; 10 2 DFT T
[12] 'V = '; 10 0 DFT V
[13] 'PAUX = '; 10 0 DFT PAUX
[14] ' '
[15] 'NUMBER OF CASES = ';((P,P1)-+/,P1<0)*PT
[16] 'MAX NUMBER OF LINES/CASE = ';1+((1+D10)*P*PAUX)+PW
[17] ' --'
      ▽

```

NOT REPRODUCIBLE

INPUT

INPUT

D9 = 0 = DO NOT PRINT LOCAL VARIABLES
D8 = 0 = LONG TITLE
D10 = 1 = PRINT ANSWERS FOR HYDRAULIC ACTUATOR
D1 = 1 = NEW DATA ARE REQUIRED
GN = 2 (1= H2, 2= O2, 3= N2, 4=HE)

CDAT =	0.62	0.94	2.50	6.50	10.00	25.00	36.00
P1 =							
	0	0	60	150	400		
	0	30	60	150	400		
	20	30	60	150	0		
	20	30	60	0	0		
	20	30	60	0	0		
	20	30	0	0	0		
	20	30	0	0	0		
T =	0.01	0.02	0.03	0.05	0.07	0.10	
W =	28	56	84	112			
PAUX =	1500	3000					

NUMBER OF CASES = 126
MAX NUMBER OF LINES/CASE = 9

INPUT SAMPLE OUTPUT

APPENDIX B

SAMPLE OF DESIGN CURVE UTILIZATION TO
DETERMINE A VALVE'S CHARACTERISTICS
FOR AN APPLICATION WHERE:

Thrust = 1000 lb

O/F = 2.0 (Oxygen/Hydrogen)

Inlet Pressure = 30 psia

t = 0.03

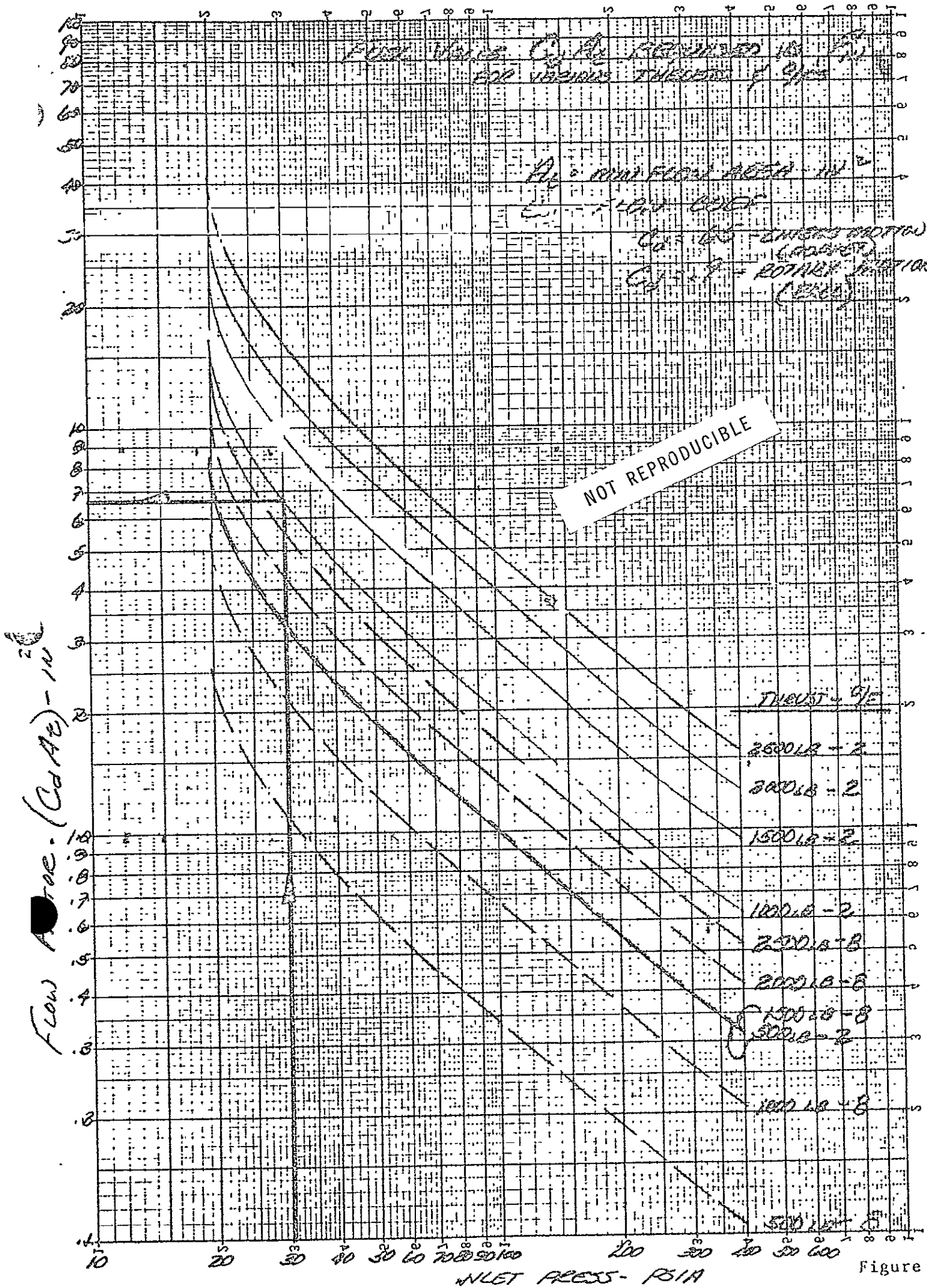


Figure LJI-4

51MHTBIRALIC
 210722 C X S
 20 1953
 K&E
 KEULEY & EBER CO
 WYOMING

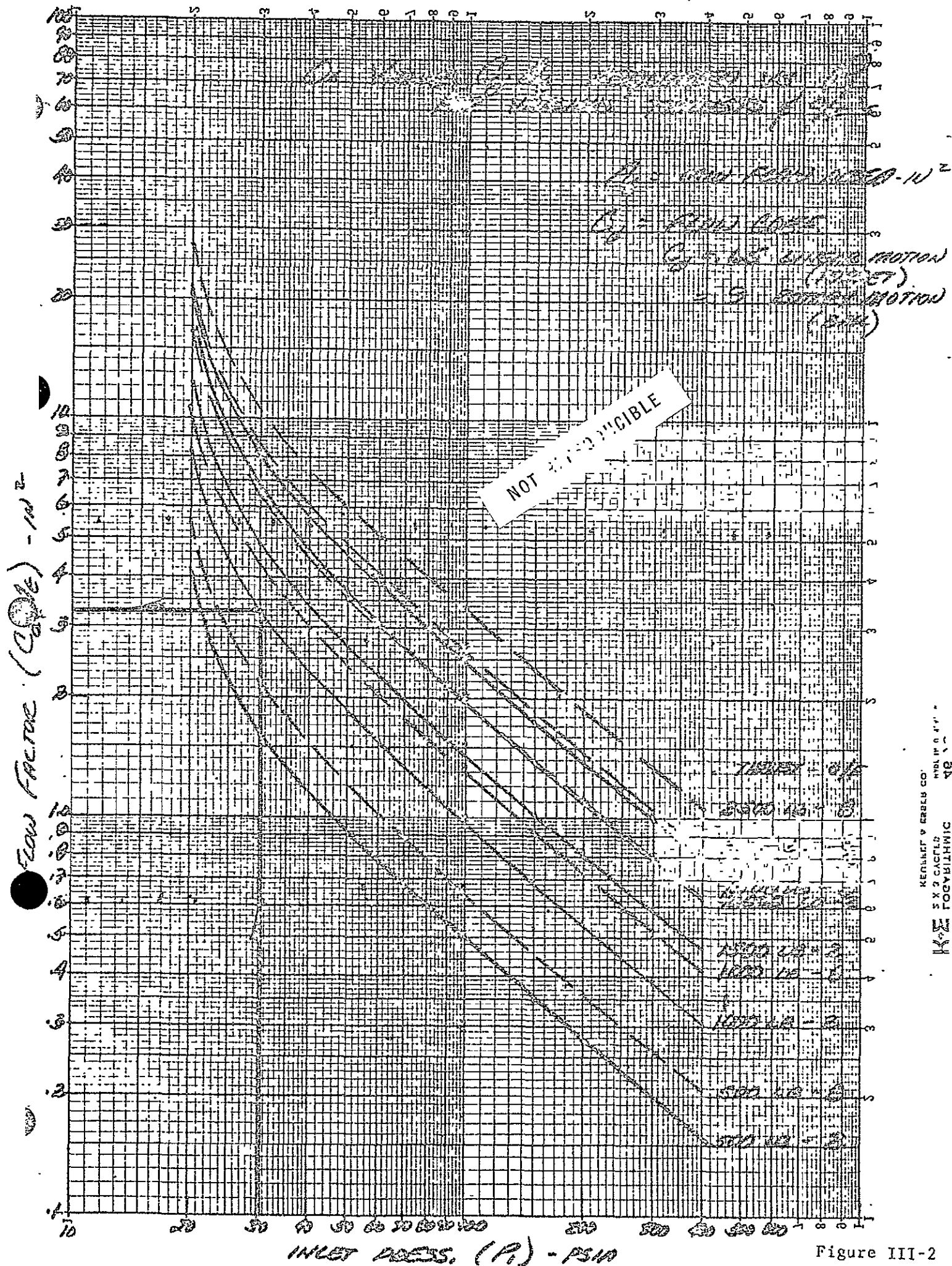
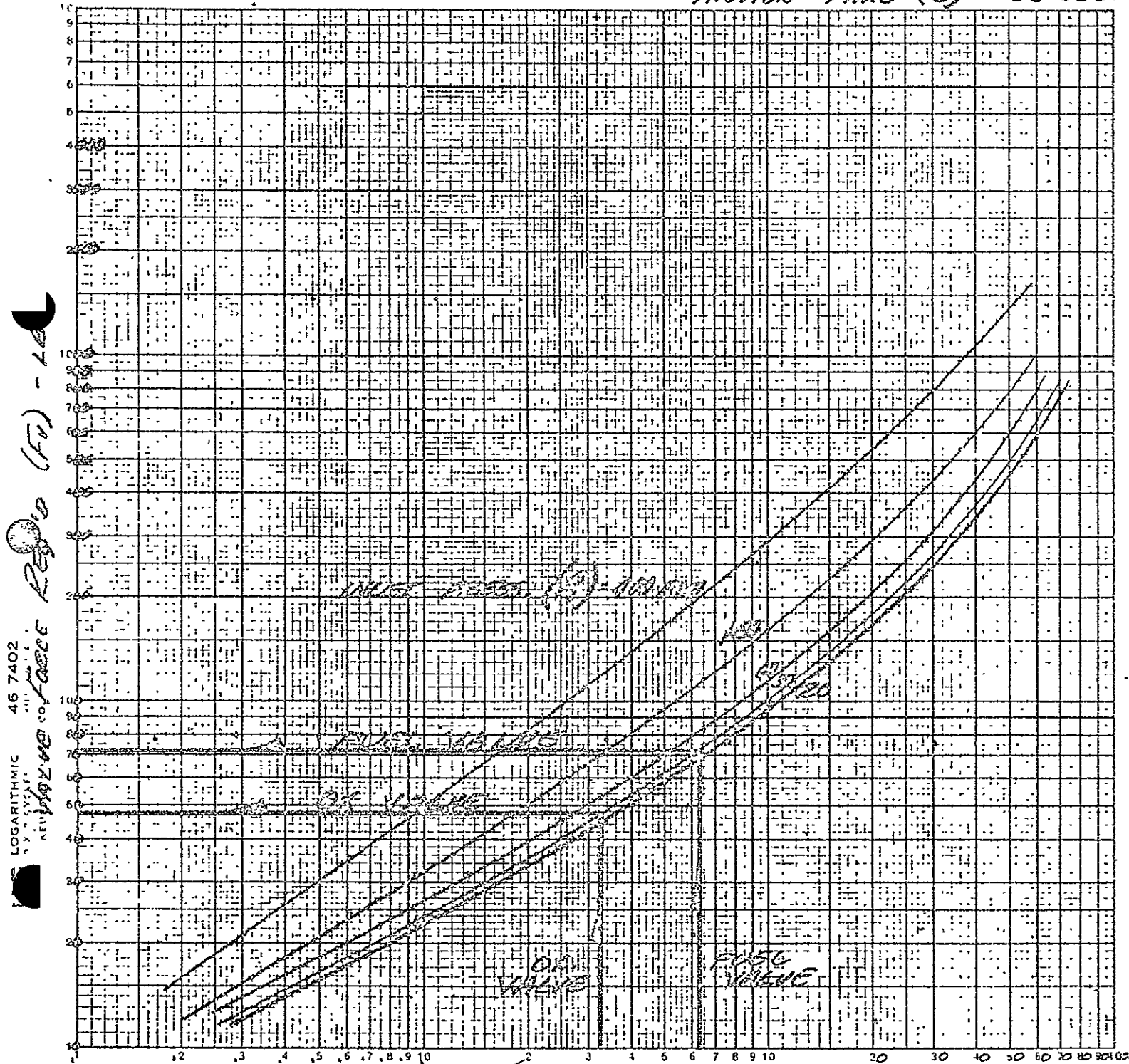


Figure III-2

LINEAR MOTION SHUT-OFF, VALVE FORCE REQUIRED VS FLOW FACTOR AT VARIOUS INLET PRESSURES

MOTION TIME (t) = .03 SEC.



Flow Factor (Ca A) - IN²

LINEAR MOTION SHUT-OFF DEVICE FLOW FACTOR ($C_d A_t$) VS. ACCELERATION FOR VARIOUS MOTION TIMES

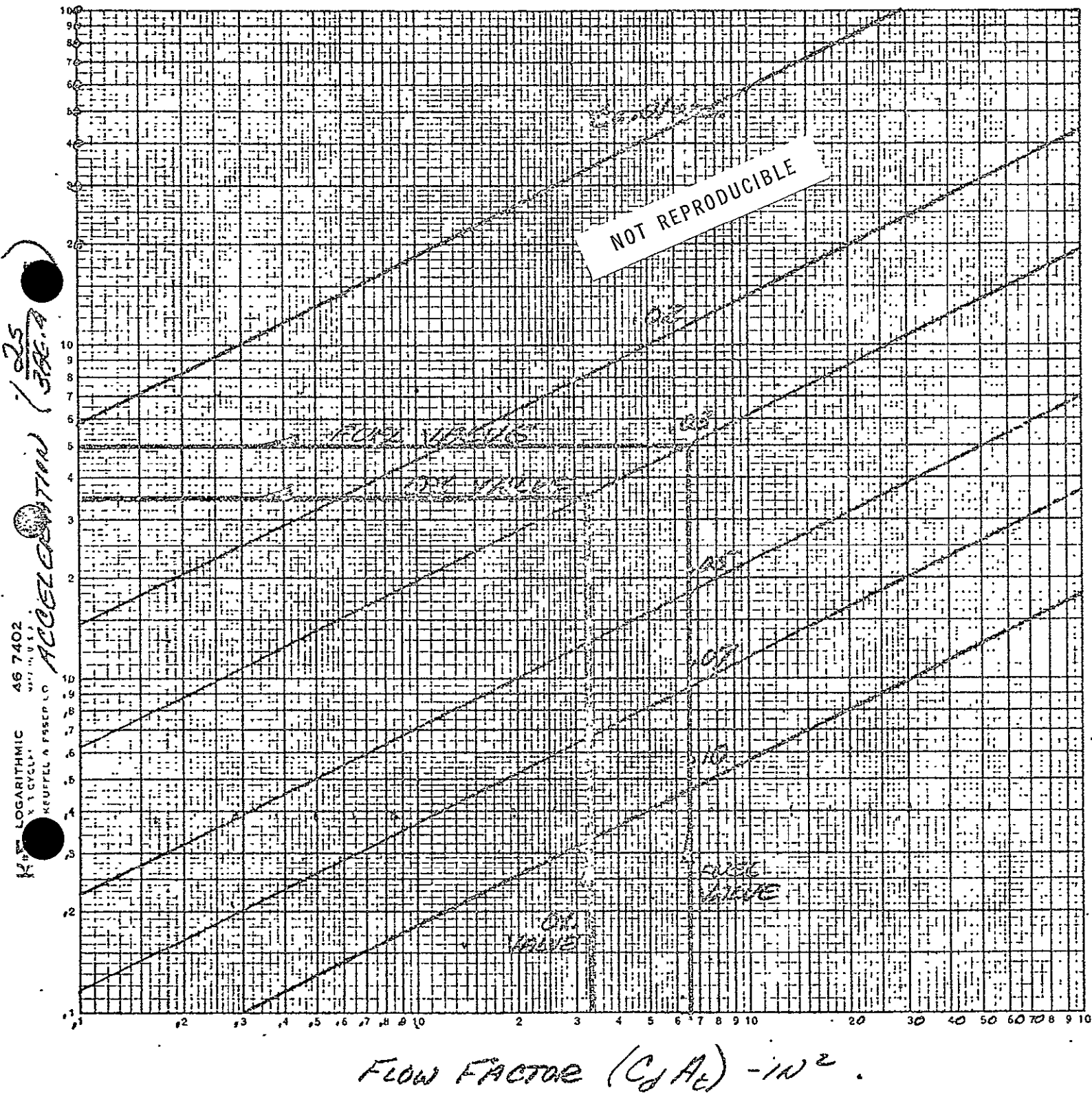
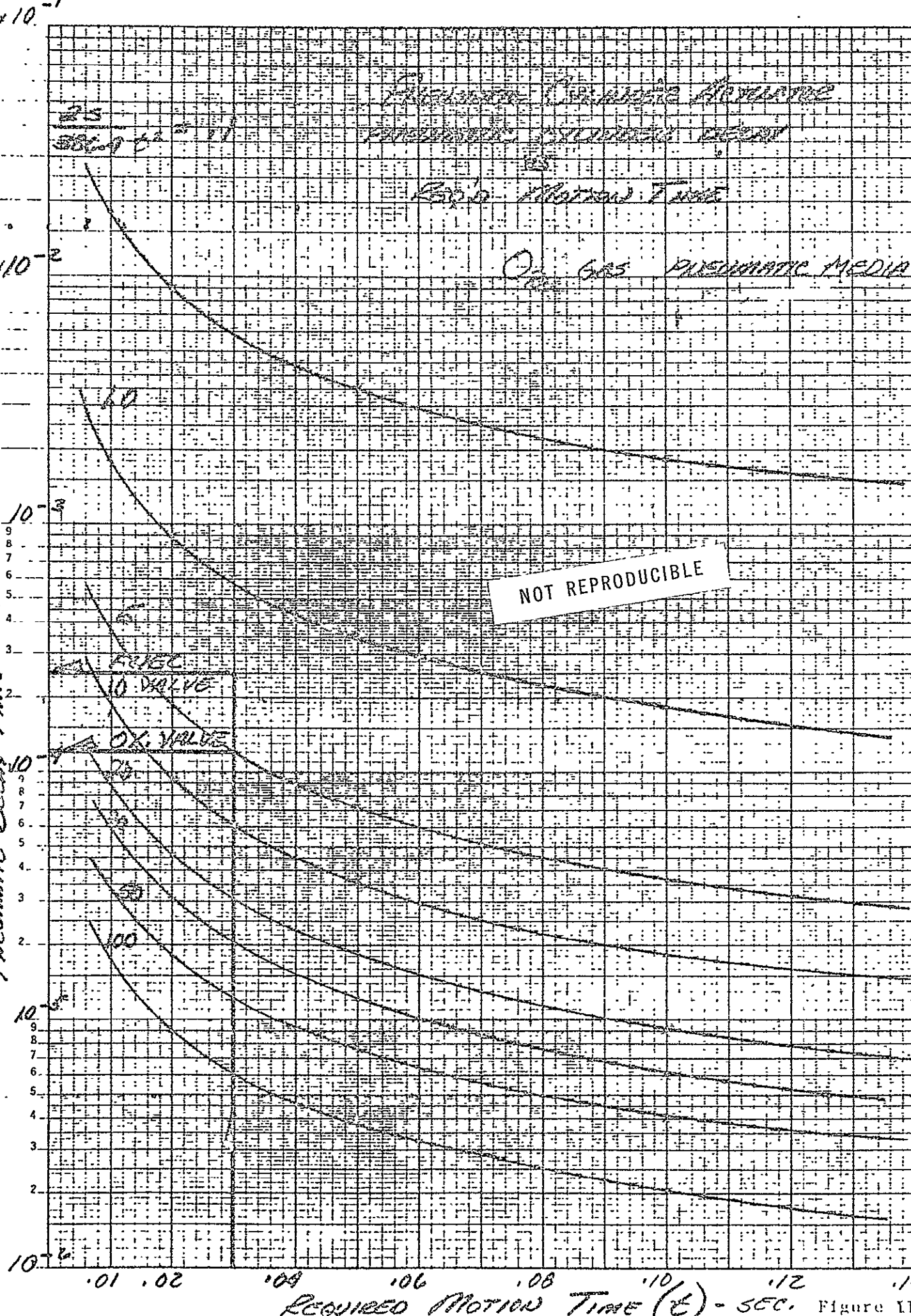


Figure III-19



Figure III-12.

SECONDS
 Pneumatic Delay Time



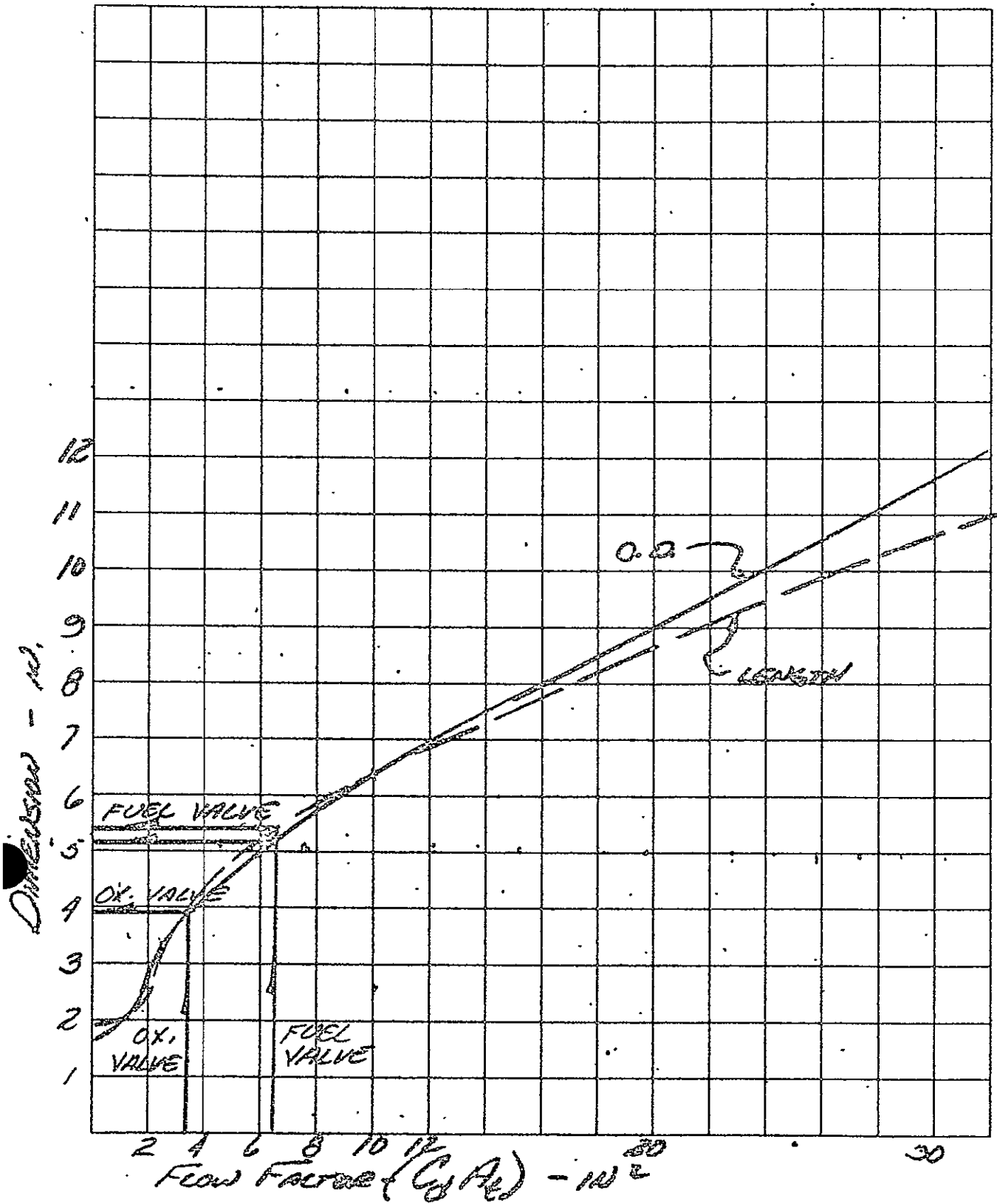
REQUIRED MOTION TIME (t) - SEC. Figure III-37

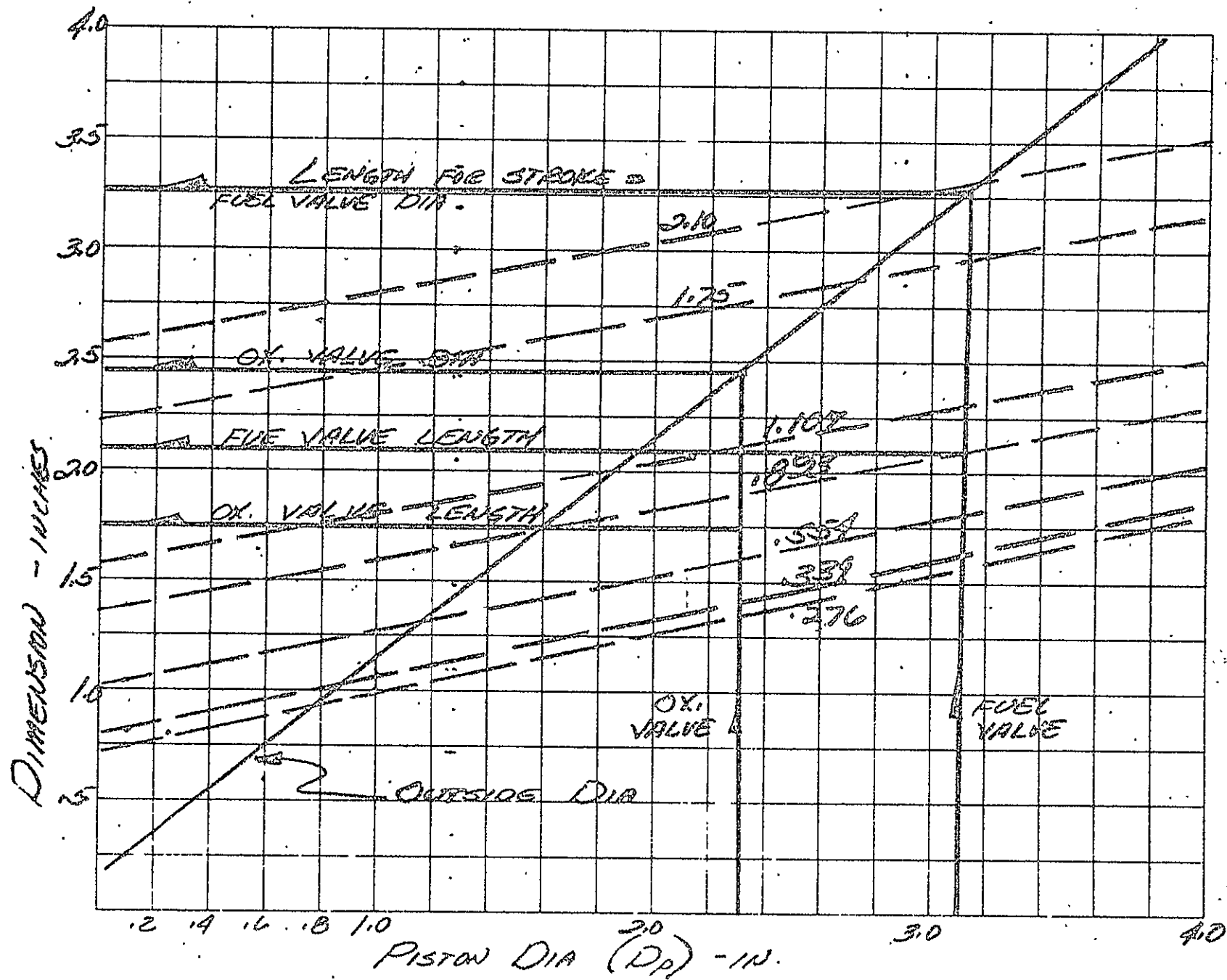
TABLE III-XVII

PNEUMATIC MEDIA CORRECTION

MEDIA	MULTIPLY t	FROM FIGURE	III-37 by
O ₂	1	←	OX. VALUE
H ₂	1.0106	←	FUEL VALUE
N ₂	1.1008		
He	.8258		

LINEAR MOTION START-OFF DEVICE ENVELOPE DIMENSIONS VS FLOW FACTOR





THE
Marquardt
 CORPORATION
 VAN NUYS, CALIFORNIA
 CYCLOPUMP AIRFRAME ENGINE DIMENSIONS
 15 PISTON DIAMETERS

PREPARED BY _____
 DATE _____

LINEAR MOTION SHUT-OFF DEVICE WEIGHT VS FLOW FACTOR

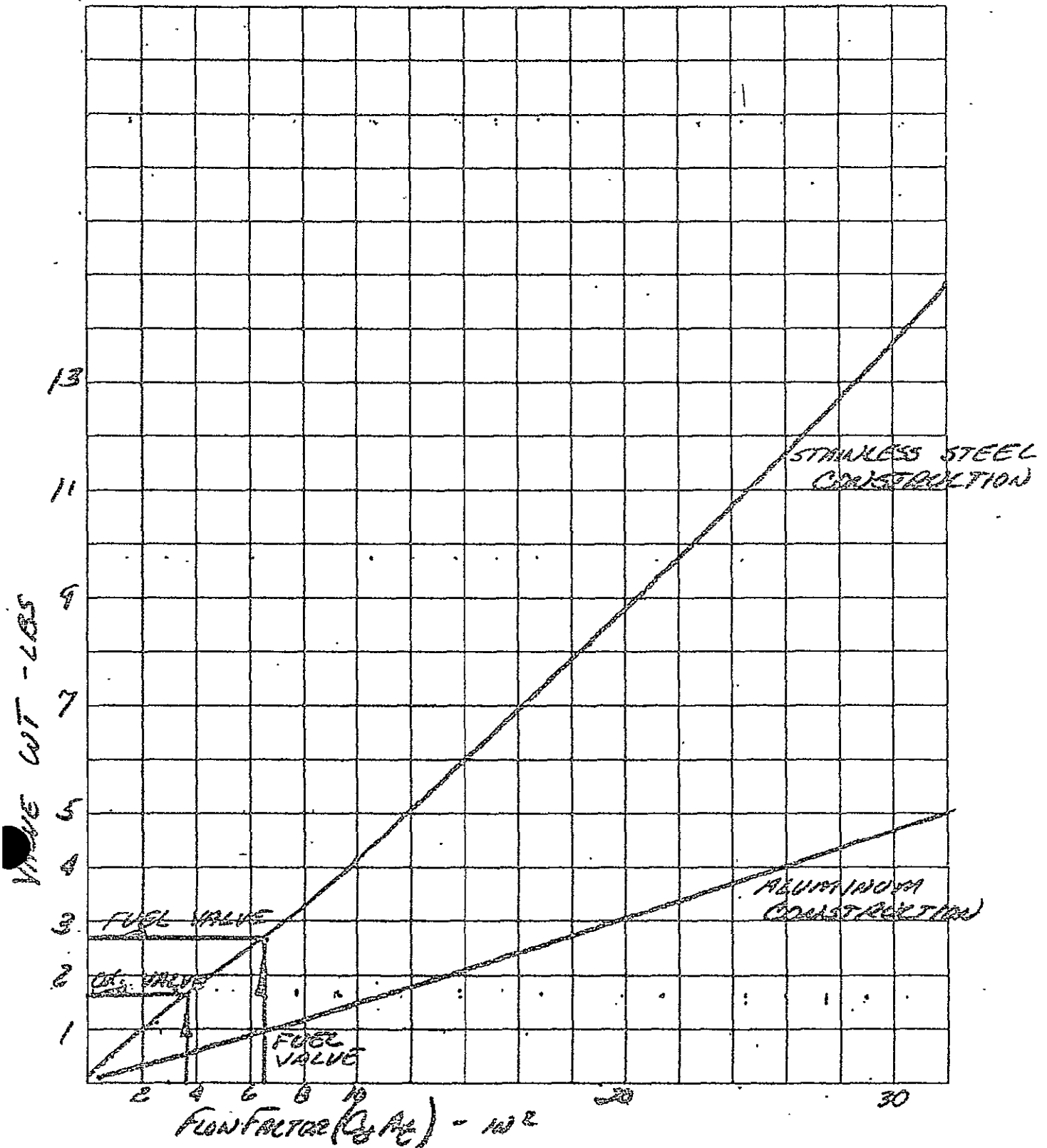
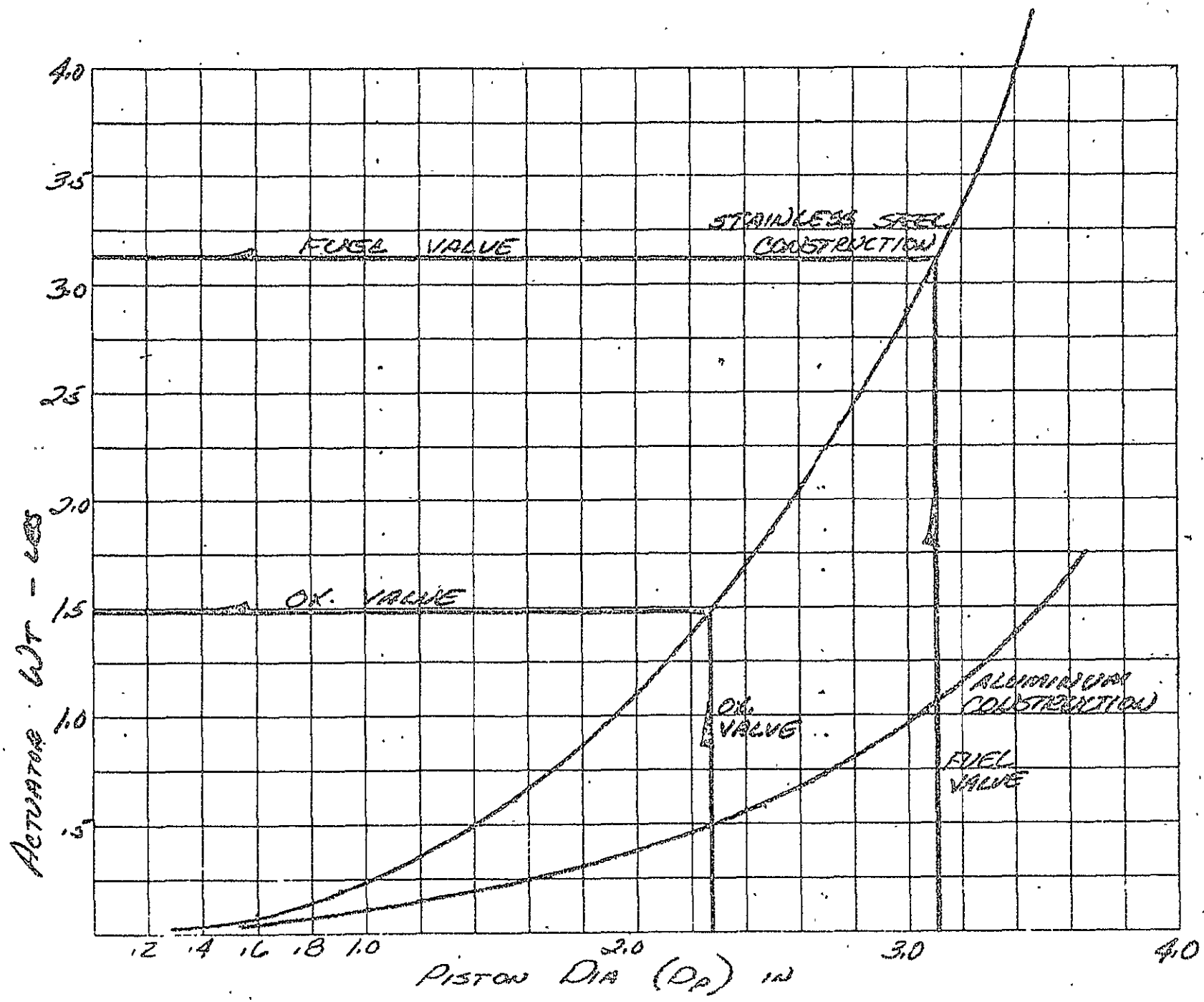


Figure 111

1235



THE
Marquardt
CORPORATION VAN NUYS, CALIFORNIA

Live Valves Compared Lineare Change Weight
vs Piston Diameter

PREPARED BY
DATE