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WADC Technical Report 54-189 VOLUME II

THEORETICAL INVESTIGATION OF OPTIMUM PRESSURES IN AIRCRAFT HYDRAULIC SYSTEMS

Conrad H. Cooke Eugene Gessner Robert L. Smith

THE GLENN L. MARTIN COMPANY

January 1954

WRIGHT AIR DEVELOPMENT CENTER

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WADC TECHNICAL REPORT 54-189
VOLUME II

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Aircraft Laboratory
Contract No. AF 33(616)-344
RDO No. R-452-495

Wright Air Development Center
Air Research and Development Command
United States Air Force
Wright-Patterson Air Force Base, Ohio

VOLUME 2

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INTRODUCTION

The appendix contains a detail analysis of the weight and space of the elements comprising an aircraft hydraulic system. Other information of interest to the study is also presented.

In the analysis of the elements, many assumptions have been made for the derivation of the formulas for weight and space. Justification of these assumptions where necessary and possible is also presented.

A weight comparison between 3000 psi and 5000 psi for a typical subcircuit has been made by North American Aviation. A presentation of the analysis of Lou Berthelson is presented in Appendix A.

APPENDIX A

DETERMINATION OF THE WEIGHT AND SPACE OF TRANSMISSION LINES

I Derivation of the Theoretical Formula to Determine the Plumbing Weight

The equations which consider fluid weight, tubing weight, and fitting weight have been derived as an approach to express total weight required to install sections of tubing in an airplane -- in one equation; consisting of pertinent variables such as pressure, density of fluid, density of fitting and tubing material, and diameter of tubing. The resulting composite equation is written to show the effect of these variables. Their derivation will follow.

The equations are:

$$W_T = W_o + W_1 + W_f \qquad \text{(I)}$$

$$W_o = \frac{\pi D^2 \rho_o \alpha}{4} \cdot \frac{S_{tl} - GP}{S_{tl} + GP} \qquad \text{(II)}$$

$$W_1 = \frac{\pi D^2 \rho_1}{} \; \frac{2\,GP}{S_{tl} + GP} \qquad \text{(III)}$$

$$W_f = \frac{2.98 \times 10^{-5} P^3 D^2}{36000 + P} + 8.50 \times 10^{-7} DP + 4.87 \times 10^{-3} D \qquad \text{(IV)}$$

Where:

W_T = Combined Average Plumbing Weight Per Inch of Tubing (Lb/In)

W_o = Average Weight of Fluid Per Inch of Tubing (Lb/In)

W_1 = Average Weight of Tubing Per Inch of Tubing (Lb/In)

W_f = Average Weight of Fitting Per Inch of Tubing (LB/In)

D = Outside Diameter of Tubing (In)

P = System Working Pressure (Lb/In)

ρo = Fluid Density (Lb/In^3) at Standard Conditions

α = Density Multiplier Based on Temperature MIL-O-5606

S_{tl} = Ultimate Tensile Stress (LB/IN^2)

$\rho 1$ = Tubing Material Density (Lb/In^3)

ρf = Fitting Material Density (Lb/In^3)

G = Applicable Safety Factor

W = Statistical Average Fitting Weight (Lb.)

K_1, K_2, K_3, Etc. = Constant Values

When the value of W_T has been calculated by the foregoing formulae, a system constant is obtained, which when multiplied by any similar system length yields its plumbing weight.

a. Fluid Weight

The following method was applied in the derivation of formula II, the fluid weight formula.

Fluid weight in the tubing equals its volume times its density.

$$\text{Volume} = \frac{\pi d^2 L}{4}$$

Where:

d = inside diameter of tubing (In)

L = tubing length (In)

Density = $a\rho o$

Where:

a = multiplier (See Ref. 1)

Combining:

$$W_o = \frac{\pi d^2}{4} a\rho o \text{ per inch} \qquad \text{(V)}$$

It is noted that, the expression ($a\rho o$) takes into account the effects of compressibility of fluid due to an increase in pressure.

It is desirable that the inside diameter be converted to outside diameter. A conversion is possible through the use of the thick wall formula: (Ref. 2 and 3)

$$S_t = GP\left[\frac{D^2 + d^2}{D^2 - d^2}\right] \qquad \text{(VI)}$$

When solved for the inside diameter, the equation becomes:

$$d^2 = D^2\left[\frac{S_t - GP}{S_t + GP}\right] \qquad \text{(VII)}$$

Thus Formula V becomes:

$$W_o = \frac{\pi D^2}{4} \rho o a \left[\frac{S_{tL} - GP}{S_{tL} + GP}\right] \qquad \text{(II)}$$

b. Tubing Weight

Formula III, the tubing weight formula, was derived as follows:

Weight of tubing equals the cross-section area times the tubing length times the material density.

$$\text{Cross-section area} = \frac{\pi}{4}(D^2-d^2) \qquad \text{(VII)}$$

Combining:

$$W = \frac{\pi}{4}(D^2-d^2)\rho_L L \qquad \text{(IX)}$$

Again formula VII is utilized to simplify. Accordingly formula IX becomes:

$$W_L = \frac{\pi}{4}\left(\frac{2GP}{S_{tL} + GP}\right)\rho LD^2 \qquad \text{(III)}$$

For one inch of tubing

c. Fitting Weight

The fitting weight formula, consisted of the product of cross-section area, fitting length, fitting material density, and a constant which includes miscellaneous factors such as number of fittings per tubing lengths, , etc.

Most important, at first, is to correctly define the following:

$$W = \frac{wN}{L} \qquad \text{(X)}$$

here:

w = individual fitting weight (lb.)

N = number of fittings considered

L = length of tubing involved (inch)

W = average fitting weight per inch of tubing (Lb./In.)

A theoretical expression for individual fitting weights was first derived on the basis of strength required for internal pressure only, and on the assumption that it was affected by the same variables which affected a section of straight tubing. Thus, its cross-section area becomes:

$$A = K\frac{\pi}{4}(D^2 - d^2)$$

Where K accounts for difference between tubing outside diameter and fitting outside diameter. And if formula VII is introduced to eliminate the variable d (I.D.):

$$A = K \frac{\pi}{4} \left(\frac{2\,GP}{S_{tf} + GP}\right) D^2$$

To obtain a hypothetical length the following relationship was utilized:

Separating Load = Shear Area x Shear Stress

$$\frac{\pi}{4} D^2 P = \pi L' D \times S_{sf}$$

Solving for L':

$$L' = \frac{DP}{4\,S_{sf}}$$

It is assumed that the effective seal between the fitting and the tubing takes place at a point of the inner wall of the "AN" type flare attachment which is approximately equal to the outside diameter of the tubing. In the care of the "ER" type fitting, the seal definitely takes place at the outside diameter of the tube. Fore this reason the separating force is expressed as the internal pressure times the cross-sectional area enclosed by the outside diameter of the fitting.

When area, length, and density are multiplied together:

$$w_1 = K \frac{\pi}{4} \frac{2\,GD^3P^2\rho_f}{S_{sf}\left(\frac{S_{tf}}{G} + P\right)} \qquad \text{(XI)}$$

If the value for the defining material, MIL-S-6758, condition F, or S_{tf} = 180,000 psi is introduced, the factor of safety (G) is set at 5, and all the constant quantities are called K, then formula becomes: (XI)

$$w_1 = \frac{K_1 D^3 P^2}{3600 + P} \qquad \text{(XII)}$$

And this weight is that portion of fitting weight which is required to withstand the effects of internal hydraulic pressure. This is shown more effectively in figure A-1 as being 21.25% of total fitting (for 1/2" fittings at 3000 psi only) weight, and is represented by the darker cross hatched portion.

It is possible to arrive at a value for K, for P = 3000 psig by assuming the diameter and getting the fitting weight from Table A-1.

From Table A-1 it is seen that the statistical average fitting weight (w) equals 0.0251 lbs. for a 1/2 inch diameter fitting.

So:

$$w_1 = .2125\ (0.0251) = 0.00531 \text{ lbs.}$$

and, from formula XII:

$$K_1 = \frac{(36000 + P)w_1}{D^3 P^2} = \frac{(36000 + 3000)(0.00531)}{(.5)^3 (3000)^2} = .0001585$$

Therefore, formula XII becomes:

$$w_1 = \frac{.0001585 D^3 P^2}{(36000 + P)} \qquad \text{(XIII)}$$

The sketch, figure A-1 was drawn for a 1/2 inch diameter fitting since this size and fitting group is most common. It provides a means to estimate the percent of material required to withstand internal hydraulic pressure, and to estimate the percent of material required to withstand torqueing preload which appears lightly cross hatched. The remainder of the material must logically be bulk. Bulk material can be defined as that weight which includes unstressed material found in shoulders, wrench flats, and other spaces resulting from the nature of the design of the fittings.

In an attempt to arrive at a mathematical expression for the extra weight, formula XIII, and actual data, Table A-1, were plotted together for pressure at 3000 psig, figure A-2. It was observed that the difference between these curves was a straight line. This permits us to write:

$$w' = \frac{.0001585 D^3 P^2}{36000 + P} = K'D \qquad \text{(XIV)}$$

Where w' = actual data, Table A-1

In order to break the term K'D down into two expressions, one for the material weight required to withstand torqueing preload, and the other for the material weight which is bulk, figure A-1 was reexamined. The torqueing weight, composed of the cap and union thread engagement region and the cap shoulder region (lightly cross-hatched) is defined as a percent of the total.

This percent of the total is dependent on the diameter of the fitting and on the internal hydraulic pressure tending to separate the fitting.

The remainder which is pure bulk, is a function of fitting diameter, D.

Accordingly we may write:

$$W' - \frac{.0001585 D^3 P^2}{36000 + P} = K'D = K_2 DP + K_3 D \qquad \text{(XIV)}$$

or: $$W' = \frac{.0001585 D^3 P^2}{36000 + P} + K_2 DP + K_3 D$$

Again from TableA1, D = 12 w' = 0.0251

So for torqueing:

$$w_2 = .27(.0251) = 0.00677 \text{ lbs.}$$

For bulk:

$$w_3 = 1 - (.2125 + .27)\ 0.0251 = 0.01295 \text{ lbs.}$$

Since the weight proportion is defined, it is possible to solve for K_2 and K_3 by setting the applicable sections of formula XIV equal to its respective weight. So for K_2:

$$w_2 = K_2DP$$

$$K_2 = \frac{w2}{DP} = \frac{.000677}{(0.5\ (3000)} = .00000452$$

For K_3:

$$w_3 = K_3D$$

$$K_3 = \frac{w_3}{D} \quad \frac{0.01295}{0.5} = .0259$$

Inserting the values of K_2 and K_3 into formula XIVa

$$w = \frac{0.0001585D^3P^2}{36000 + P} + 0.00000452DP + 0.0259D \qquad \text{(XV)}$$

A plot of Equation XV appears in figure A-3 and shows graphically what has gone before.

From figure A-4.

$$\frac{N}{L} = 0.18798$$

Combining formulae X, XV, and XVI, the final formula becomes:

$$W_f = \frac{2.98 \times 10^{-5}P^3D^2}{36000 + P} + 8.50 \times 10^{-7}DP + 4.87 \times 10^{-3}D \qquad \text{(IV)}$$

TABLE A-1

D	W_f	$\frac{N}{L}$	$w = W_f \frac{N}{L}$
.250	.0021840	.2188827	.00997794709
.375	.0032878	.2014873	.01631765376
.500	.0056159	.2239316	.02507864008
.625	.0062842	.0742637	.0846200768
.750	.0119478	.2445254	.0488611817
1.000	.0114928	.2122303	.0541524937
1.250	.0159254	.1405083	.1133413470

TABLE A-2

D	P	DP	D^3	P^2	D^3P^2	36000+P	(d) $\frac{D^3P^2}{36000+P}$	.0001585 (Stress Wt)	.00000452DP (Torque Wt.)	(Bulk) .0259D (Bulk Wt)	(Torque) .00000452DP + .0259 D	FITTING TOTAL
.250	3000	750	.015625	$9(10^6)$	$1.406(10^5)$	3.910^4	$3.605(10^0)$	.00671	.00339	.00647	.00986	.010531
.375	"	1125	.052724	"	$4.74(10^5)$	"	$1.217(10^1)$	.00193	.00509	.00971	.01480	.01673
.500	"	1500	.12500	"	$1.125(10^6)$	"	$2.89\ (10^1)$	.00458	.00679	.01293	.01972	.0243
.625	"	1875	.24414	"	$2.195(10^6)$	"	$5.62\ (10^1)$	.00891	.00849	.01618	.02467	.033580
.750	"	2250	.421875	"	$3.8(10^6)$	"	$9.75\ (10^1)$	.01545	.01018	.0194	.02958	.04503
1.000	"	3000	1.00000	"	$9(10^6)$	"	$2.315(10^2)$	.0367	.01357	.0259	.03947	.07617
1.250	"	3750	1.953125	"	$1.759(10^7)$	"	$4.51\ (10^2)$	.0715	.01696	.0324	.04936	.12086

$$W = W_1 + W_2 + W_3$$

$$= .0001585\ \frac{D^3P^2}{(36000 + P)} + .00000452\ DP + .0259\ D$$

Material: MIL-S-6758 Condition "F"

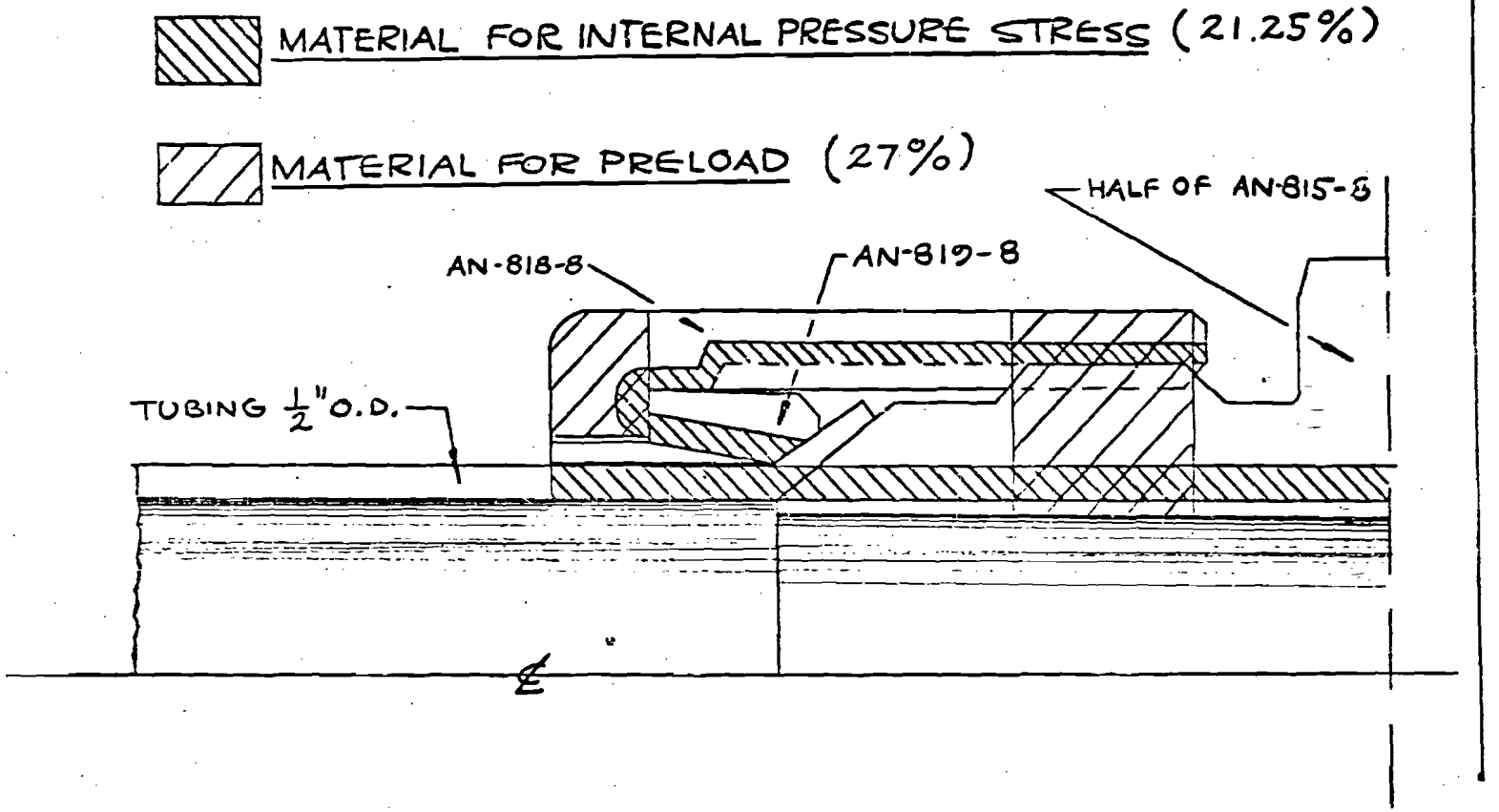
S_t = 180,000 PSI

S_s = 105,000 PSI

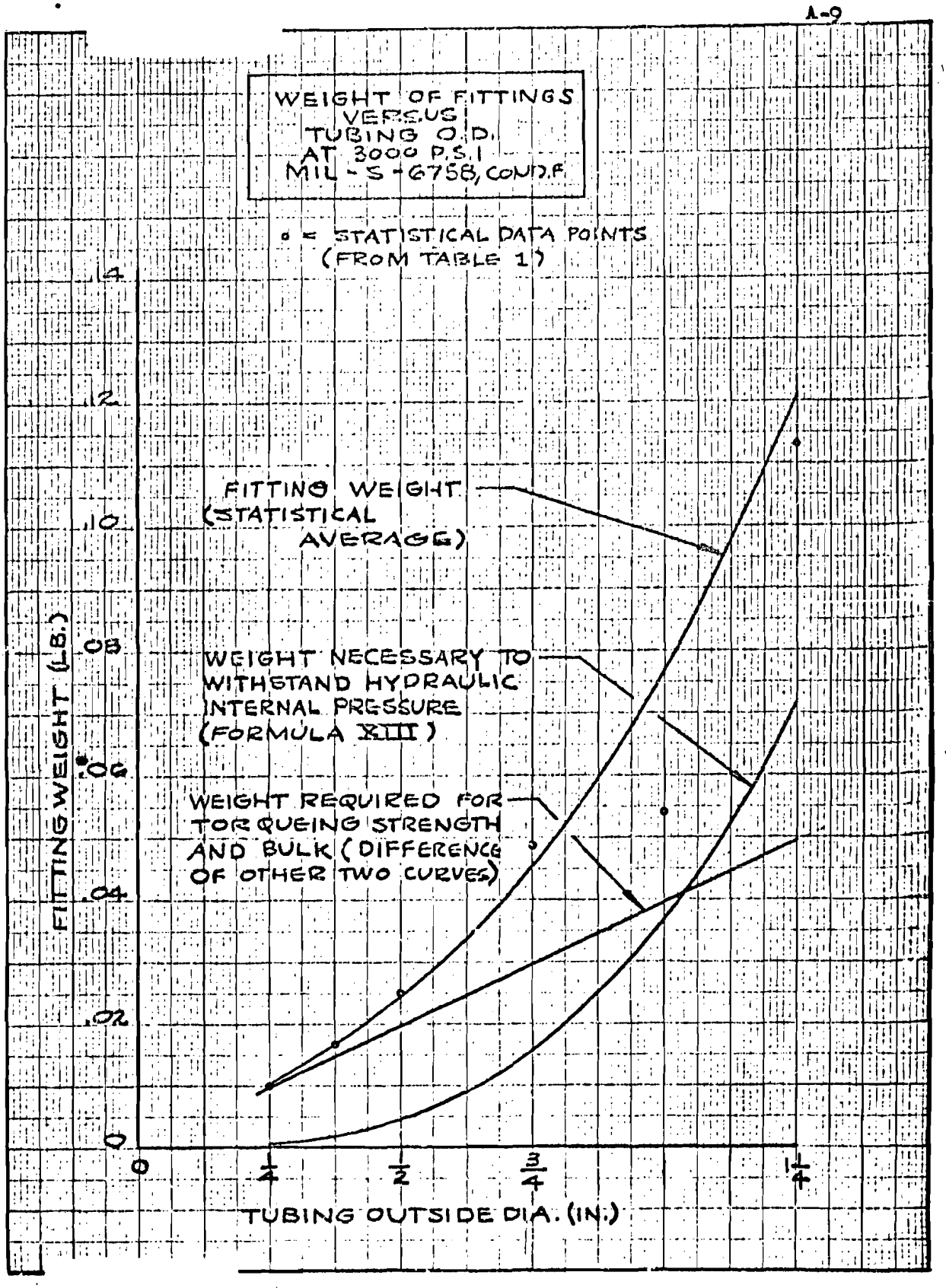
ρ_f = .283 Lb/In^3

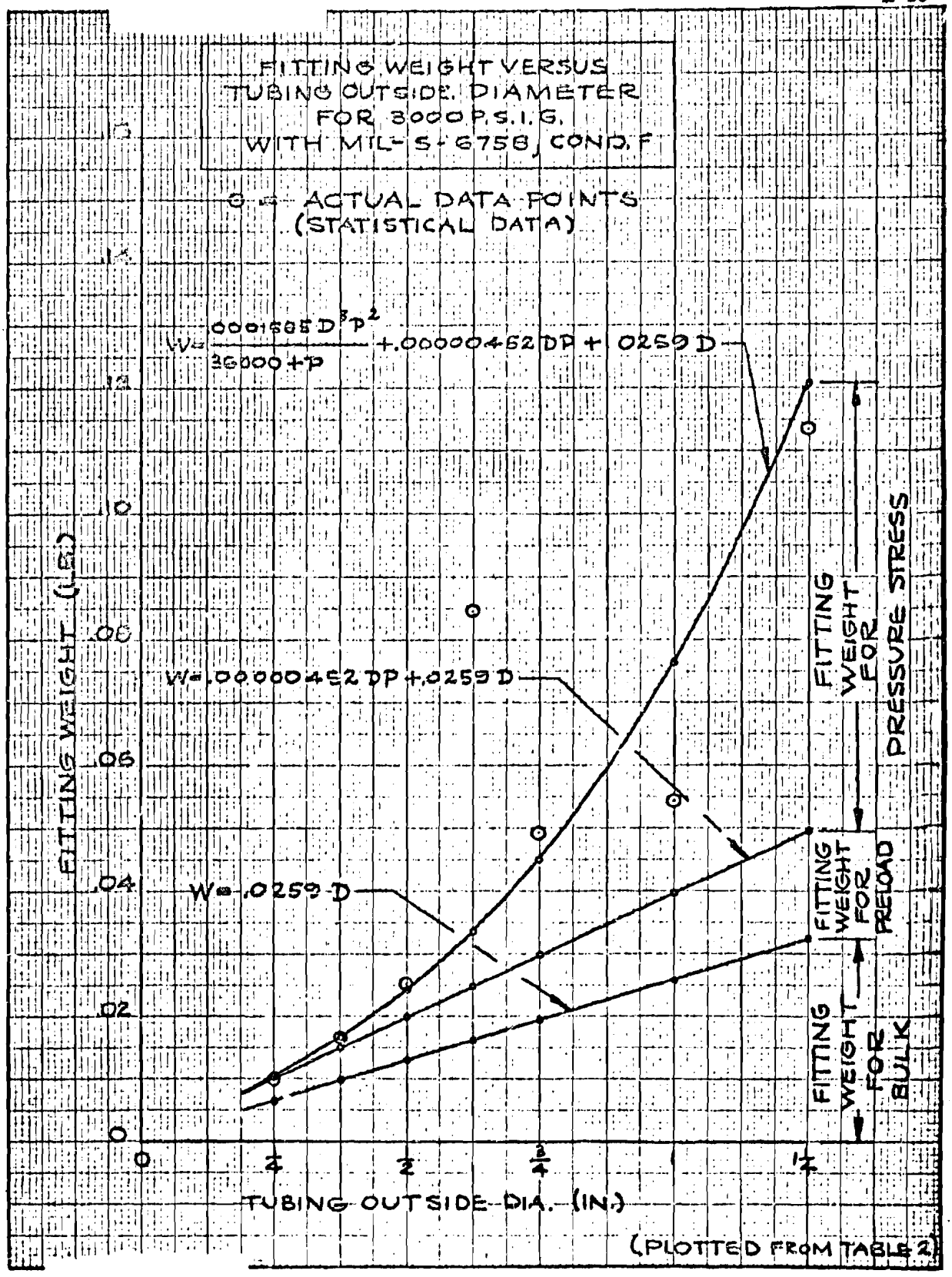
FIG. A-1

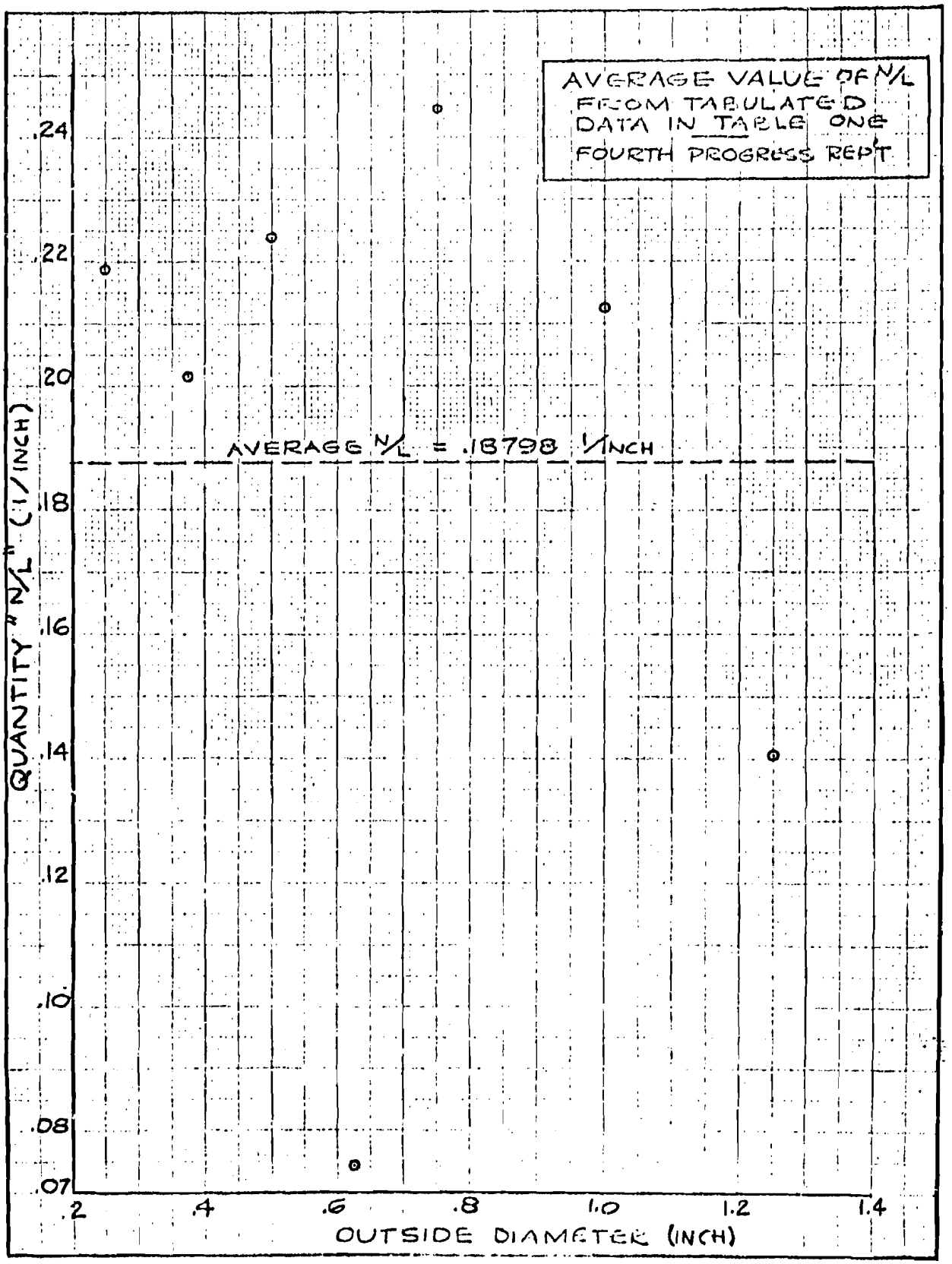
A-8



SKETCH TO DETERMINE UTILIZATION OF FITTING MATERIAL - 5X SIZE


WEIGHT OF FITTINGS
VERSUS
TUBING O.D.
AT 3000 P.S.I.
MIL-S-6758, COND. F
o = STATISTICAL DATA POINTS
(FROM TABLE 1)
FITTING WEIGHT
(STATISTICAL
AVERAGE)
WEIGHT NECESSARY TO
WITHSTAND HYDRAULIC
INTERNAL PRESSURE
(FORMULA XIII)
WEIGHT REQUIRED FOR
TORQUEING STRENGTH
AND BULK (DIFFERENCE
OF OTHER TWO CURVES)
FITTING WEIGHT (LB.)
.14
.12
.10
.08
.06
.04
.02
0
0
1/4
1/2
3/4
1 1/4
TUBING OUTSIDE DIA. (IN.)


FITTING WEIGHT VERSUS
TUBING OUTSIDE DIAMETER
FOR 3000 P.S.I.G.
WITH MIL-S-6758, COND. F
ACTUAL DATA POINTS
(STATISTICAL DATA)
W = .0001505 D³P² / (36000 + P) + .00000452 DP + .0259 D
W = .00000452 DP + .0259 D
W = .0259 D
FITTING WEIGHT (LB.)
.12
.10
.08
.06
.04
.02
0
0
¼
½
¾
1
1¼
TUBING OUTSIDE DIA. (IN.)
FITTING WEIGHT FOR PRESSURE STRESS
FITTING WEIGHT FOR PRELOAD
FITTING WEIGHT FOR BULK
(PLOTTED FROM TABLE 2)


AVERAGE VALUE OF N/L
FROM TABULATED
DATA IN TABLE ONE
FOURTH PROGRESS REPT
AVERAGE N/L = .18798 1/INCH
QUANTITY "N/L" (1/INCH)
.24
.22
.20
.18
.16
.14
.12
.10
.08
.07
.2
.4
.6
.8
1.0
1.2
1.4
OUTSIDE DIAMETER (INCH)

II Determination of the Weight of Transmission Lines

The three types of transmission lines analyzed are pressure, return and supply. The equations evolved in section A-1 apply to all with various assumptions as to the pressure to be used and the line loss to be permitted. A detailed account of the calculation of the weight of pressure tubing follows with the variation of conditions for return and supply lines pointed out.

II.a. Pressure Lines

The total weight of pressure lines has been calculated for 2, 10, 50, and 100 horsepowers. The results have been tabulated in Table A-3 and presented graphically in figure A-5. From these curves it is apparent that the optimum pressure, for pressure lines at all horsepowers and at the conditions given, ranges between 3500 and 4000 psi.

II.a.1. Weight of Pressure Tubing

The weight of the tubing can be calculated from the following equation:

$$W = \frac{\pi D^2 \rho}{4} \left(\frac{2\ GP}{St + GP}\right) \qquad \text{(Appendix A Section I)}$$

Where:

W = Weight of Tubing per Inch Tubing (Lb/In)
D = Outside Diameter of Tubing (In)
ρ = Tubing Material Density (Lb/In^3)
G = Safety Factor
P = Nominal System Pressure

This equation is equivalent to the following equation:

$$W = \frac{\rho\pi}{4} (OD^2 - ID^2)$$

The ID for the various pressures and horsepowers was found by use of figure A-6. The requirements for use of this chart for the flow and the pressure drop per foot.

The flow was obtained from the equation: $Q = \frac{1714 \times HP}{P}$

Where:

Q = Flow (GPM)
HP = Horsepower
P = Nominal Operating Pressure (Lb/In^2)

The average efficiency of the transmission lines was found to be 80%, in a survey of the industry. The average length of tubing in a circuit was found to be 52.25 feet for the M-1, P5M-1, and B-57. The percent pressure drop per foot then becomes:

$$\%\Delta P/ft = \frac{.20}{52.25} = .383\%/ft.$$

With these two values the inside diameter can be found from figure A-6. The interection of the flow and pressure drop is found. If it is below the transition line corresponding to the pressure used to determine the flow and pressure drop, it is in the laminar region. If this is the case a line is drawn parallel to the dark lines (with tube size markings) to the transition line at the pressure used to determine flow. To get the inside reading from the scale given a line must be drawn parallel to the dotted lines to the scale for correct readings. Correct values may be found from the equations:

$$d^5 = \frac{1423 f \rho Q^2}{\pi^2\ \Delta P} \qquad \text{(TURBULENT REGION)}$$

$$d^4 = 5913.8 \quad \rho Q \qquad \text{(LAMINAR REGION)}$$

Where:

- d = inside diameter (In)
- f = friction factor
- ρ = density (lb. sec^2/in^4)
- Q = flow (gal/min)
- V = viscosity (In^2/sec)
- P = pressure drop (lb/In^2/ft)

From these values of the inside diameter the outside diameter can be found from the thick walled formula.

$$OD = ID\sqrt{\frac{St + GP}{St - GP}}$$

Where:

- OD = outside diameter
- ID = inside diameter
- St = ultimate tensile strength of material (105,000 psi)
- G = Safety factor (5)
- P = nominal operating pressure

The weight of the tubing can then be found by:

$$W = \frac{\pi\rho}{4}\,(OD^2 - ID^2)$$

The tabulated values for the weight of tubing at various pressures and 2, 10, 50 and 100 horsepower is presented in Table A-3. The graphical results are presented in figure A-8.

From figure A-8 it is evident that the minimum weight for tubing alone at various horsepowers is slightly less than 3000 psi.

II.a.2. Weight of Fluid in Pressure Lines

The weight of the fluid can be found from the following equation:

$$Wo = \alpha\rho\,\frac{\pi D^2}{4}\,\left(\frac{St - GP}{St + GP}\right) \qquad \text{(Appendix A Section I)}$$

Where:

- Wo = fluid weight per inch of tubing
- D = outside diameter
- S_t = ultimate tensile strength
- G = safety factor
- P = nominal operating pressure
- ρ = density
- α = density correction for pressure (See figure 8)

Or, if the inside diameter is used:

$$W_o = a\rho \frac{\pi d^2}{4}$$

The inside diameter is the same as the one found in the previous section. For this calculation the density at 80° F was used.

The tabulated results are found in Table A-3.

The graphical results are presented in figures A-10A and A-10B. It is clear from these curves that there is a weight saving for the fluid with any increase in pressure.

II.a.3. Fitting Weight

The weight of the fittings was calculated from the equation:

$$\frac{Wt}{In} = \frac{2.98 \times 10^{-5} D^3 P^2}{36000 + P} + 8.5 \times 10^{-7} DP + 4.87 \times 10^{-3} D$$

as outlined in Appendix A Section I.

These weights are tabulated in Table A-3 of the appendix and presented graphically in figure A-11.

For fittings alone the minimum pressure occurs at 2500 psi. A line on figure A-11 is drawn for the fitting weight for 3/16 OD tubing which is a practical limit for fabrication of tubing.

II.b. Determination of the Weight of Return Lines

The weight of return lines has been determined for 2, 10, 50, and 100 horsepower in the same manner as for pressure lines. For return lines a burst pressure of one-half that used in the pressure lines is used to determine wall thickness (See figure A-33). The pressure drop per foot of tubing is assumed to be the same as in the pressure lines and the fitting weight is the same as for pressure lines and the fitting weight is the same as for pressure lines of corresponding outside tube diameters.

The total weight of the return lines is tabulated in Table A-4 and presented graphically in figure A-12. From these curves it is evident that there is a minimum weight per horsepower range from 7000 psi for 100 horsepower to 3000 psi per 2 horsepower.

II.b.1. Return Lines Tubing Weight

The weight of the tubing in the return lines was calculated from the equation:

$$\frac{Wt}{IN} = \rho \frac{\pi}{4} (D^2 - d^2)$$

With the pressure drop and the flow the same for the return lines as the pressure lines, the same inside diameter can be used. The outside diameter was calculated using the thick walled formula with burst pressure in the return lines one-half that of the pressure lines.

$$D = d \sqrt{\frac{St + G\frac{P}{2}}{St - G\frac{P}{2}}}$$

D = Outside diameter
d = Inside diameter
St = Ultimate tensile strength
G = Safety factor
P = System Pressure

The weight of the return lines tubing is tabulated in Table A-4 and presented graphically in figure F-13.

II.b.2. Return Lines Fluid Weight

The fluid weight in the return lines was determined from the equation:

$$\frac{Wt}{IN} = a\rho \frac{\pi}{4} d^2$$

a = Density correction for pressure
ρ = Density at 0 psig and 80° F
d = Inside diameter of tube

a is determined at one-half the system pressure.

The values of the fluid weight are tabulated in Table A-4 and presented graphically in figure A-14.

II.b.3. Return Lines Fitting Weights

The weight of the fittings has been calculated from the same equation used to find the weight of pressure lines fittings.

$$\frac{Wt}{IN} = \frac{2.98 \times 10^{-5} D^3 P^2}{36000 + P} + 8.5 \times 10^{-7} \times OP + 4.87 \times 10^{-3} D$$

D = Outside diameter of tube
P = Nominal system pressure.

The weights of the fittings at 2, 10, 50, and 100 horsepowers and various pressures are tabulated in Table A-4 and presented graphically in figure A-15.

II. C. Determination of the Weight of Supply Lines

The weight of supply lines has been determined on a basis of system horsepower, in the same manner as the weight of pressure and return lines.

The material used for this calculation was 52SO aluminum alloy. A minimum wall thickness of .035 inches and a minimum tubing outside diameter of one-half inch was assumed. The pressure used was the maximum of 80 psia for pump pressurization as outlined in MIL-P-7740 with a safety factor of five.

The flow in the supply lines, for the horsepower chosen, was the same as the flow in the pressure lines for the same system pressure.

$$\text{Flow} = \frac{\text{Horsepower} \times 1714}{\text{System Pressure}}$$

The average length of tubing between the reservoir and the pump was found to be 20.8 feet. The pressure drop in the supply lines was determined as the difference between maximum and minimum pump inlet pressures. The pressure drop per foot of tubing was found as follows:

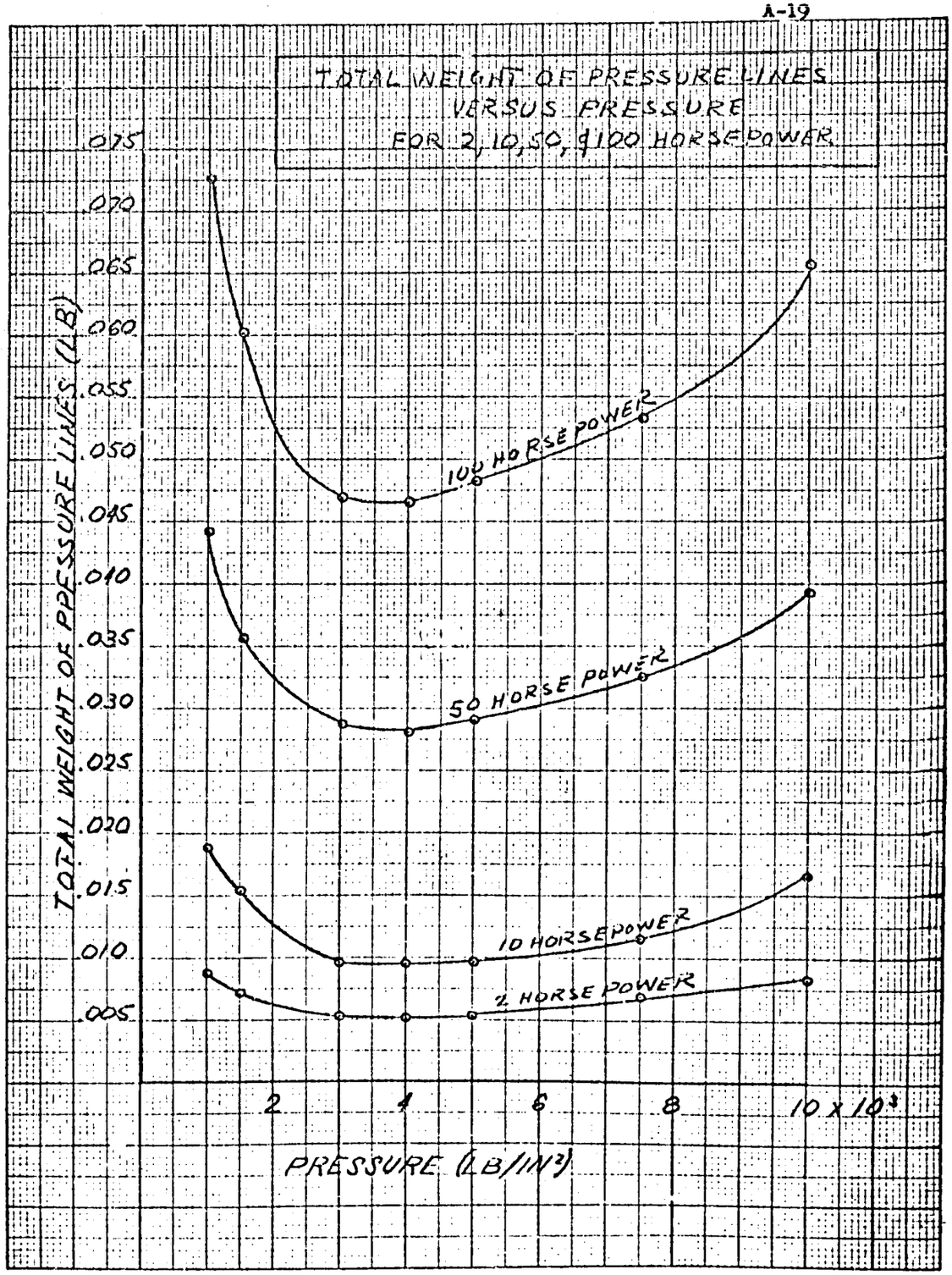
$$\text{Pressure Drop Per Foot} = \frac{\text{Max. Pump Inlet Press} - \text{Min. Pump inlet Press}}{\text{System Pressure}}$$

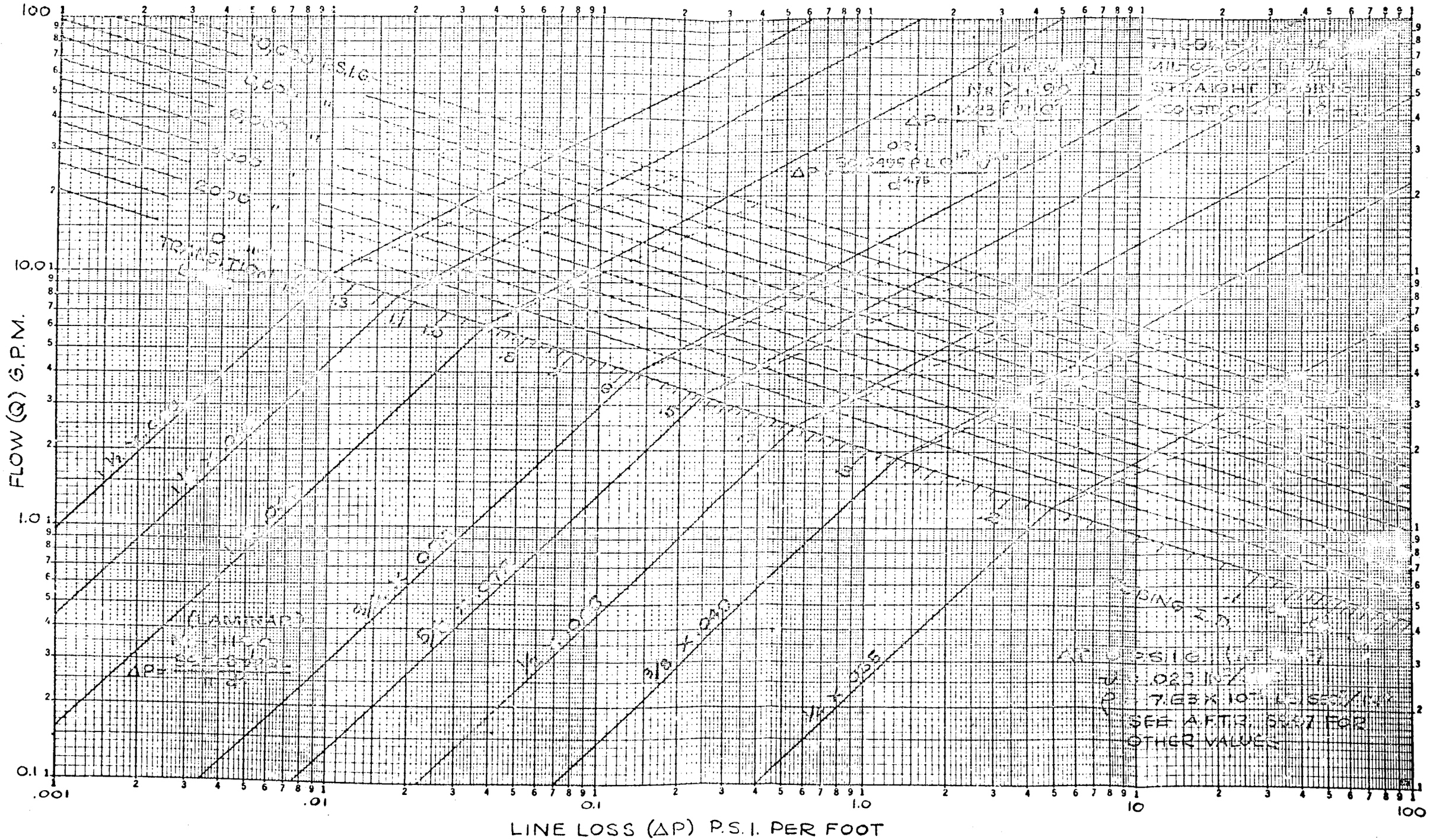
$$\Delta P = \frac{80 - 6.9}{20.8} = 3.52 \text{ psi/ft}$$

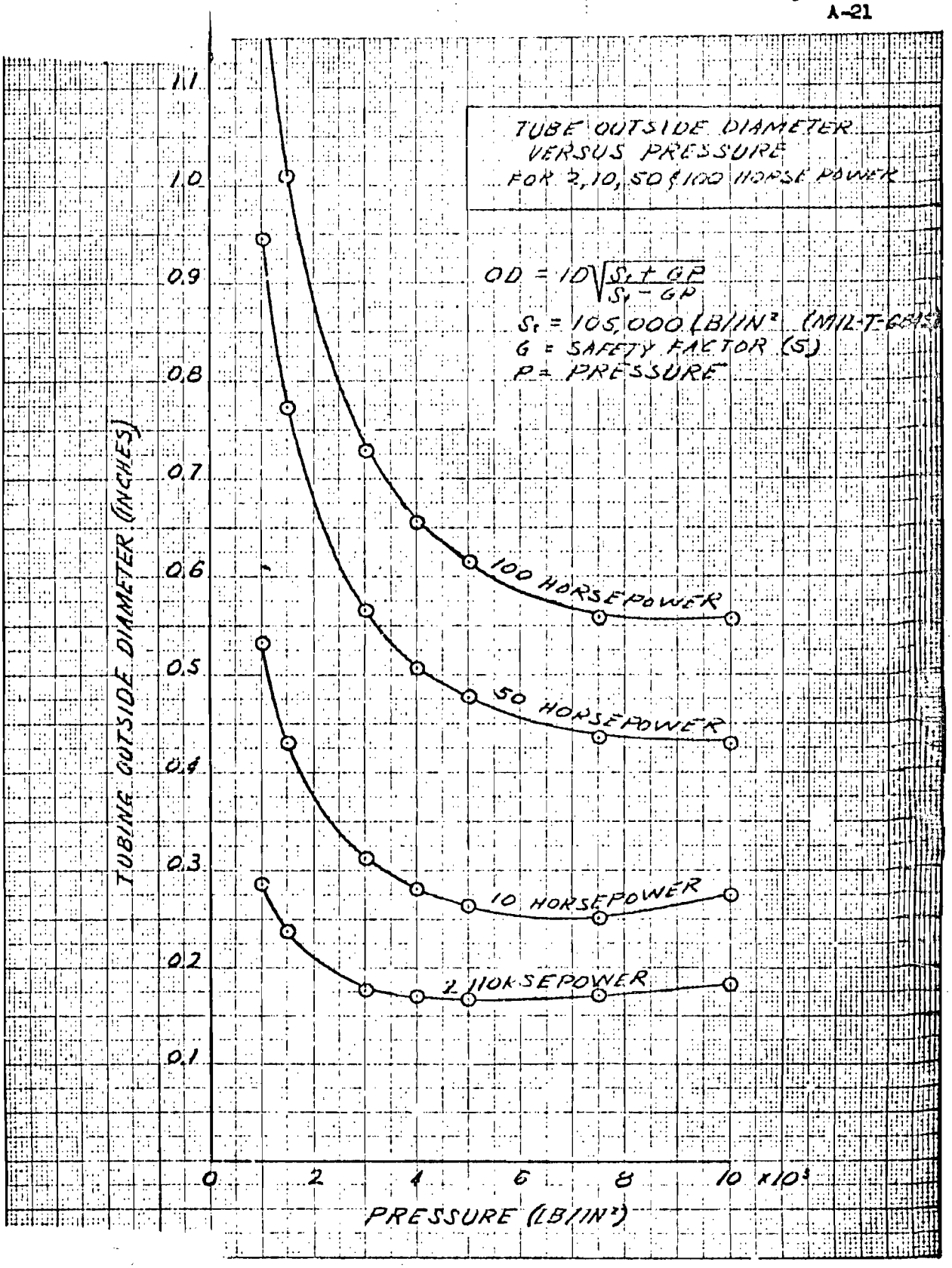
The inside diameter was determined from a graph similar to the one (figure A-6) used for pressure and return lines except that the temperature was assumed to be -20°F. The flows and pressure drop found above are used to find the inside diameter.

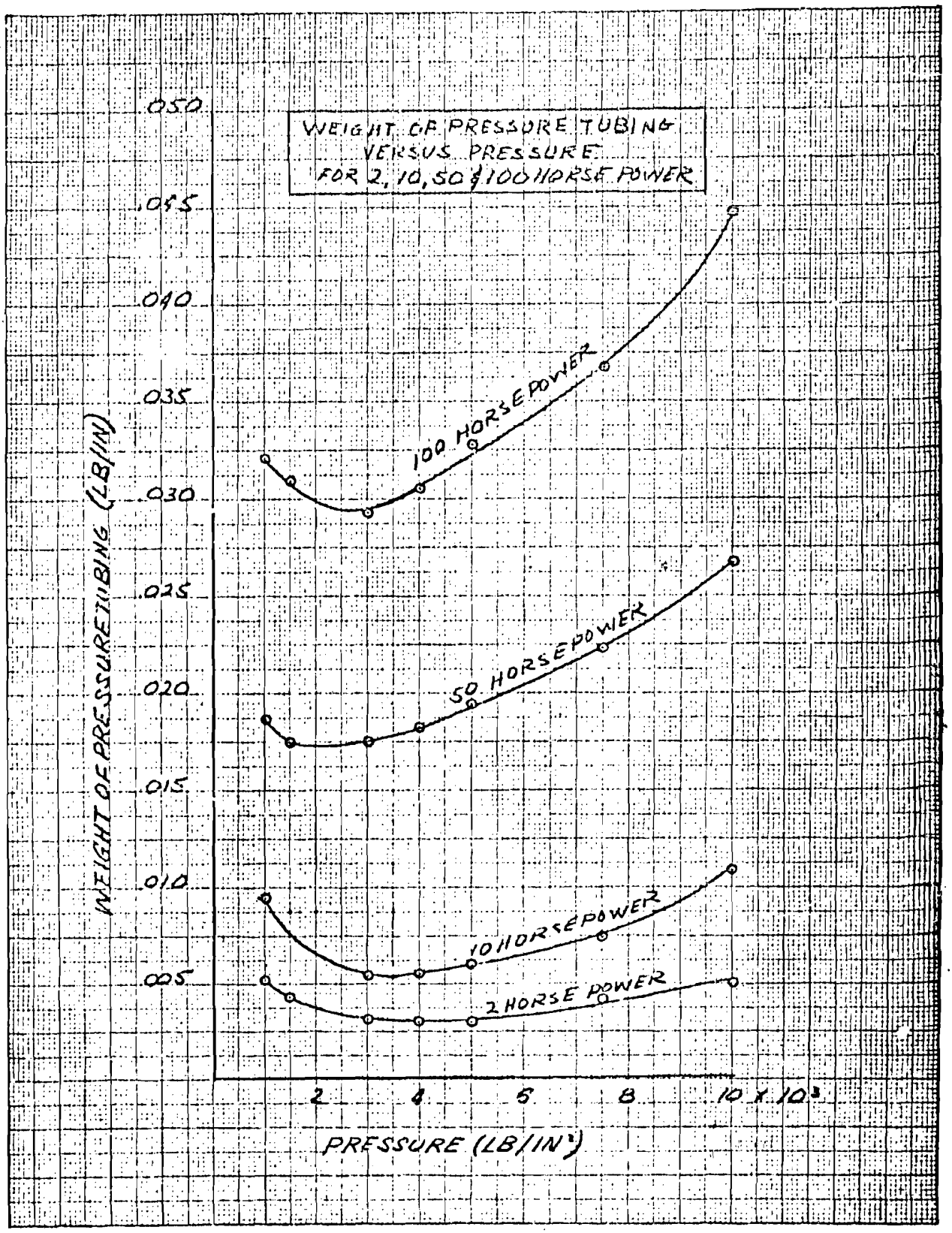
The weight of the supply lines fittings is the same as for pressure lines fittings of the same tubing outside diameters.

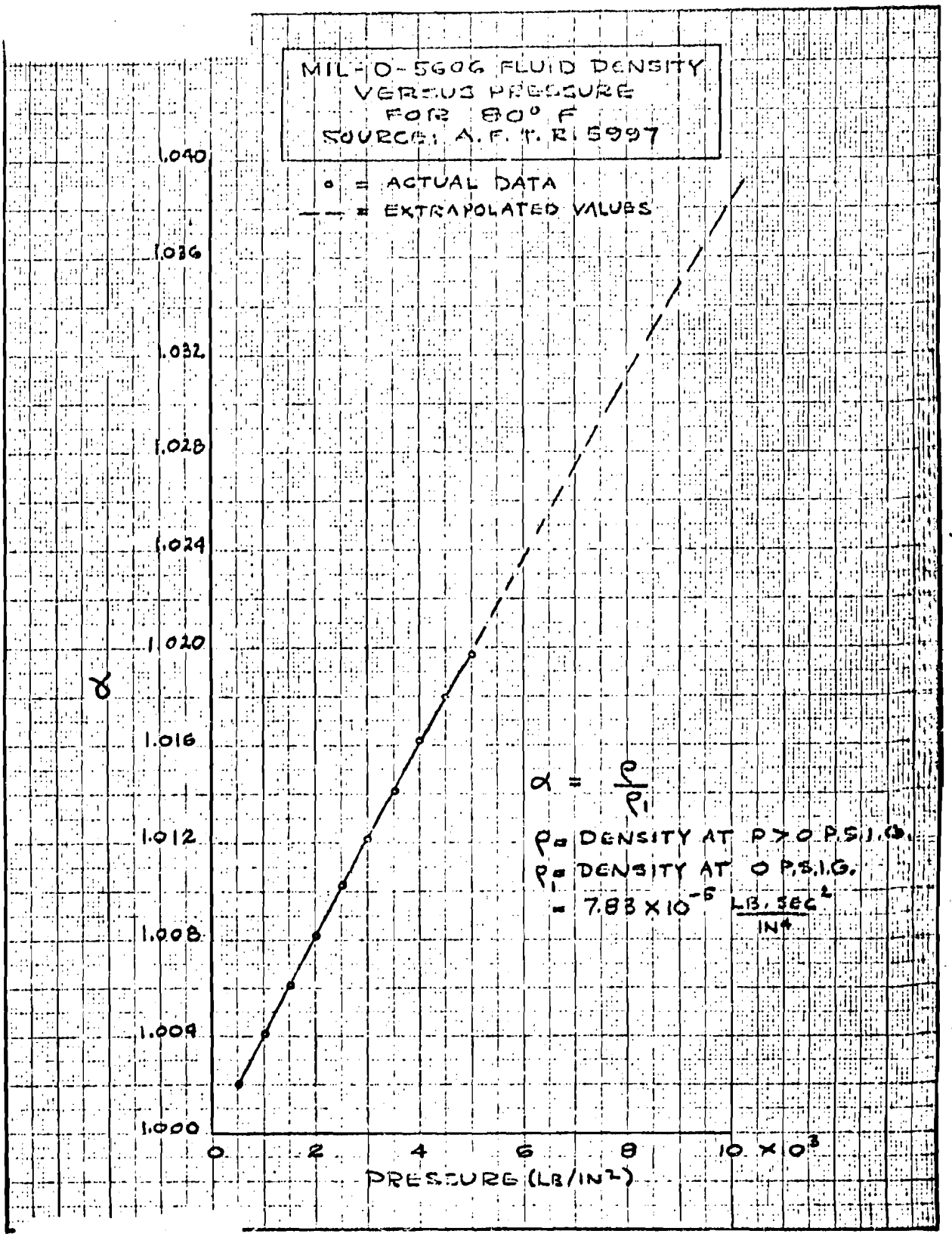
The weights of supply lines are tabulated in Table A-3 and presented graphically in figure A-16. The weights of the tubing, fluid, and fittings are plotted separately in figures A-17, A-18, and A-19.

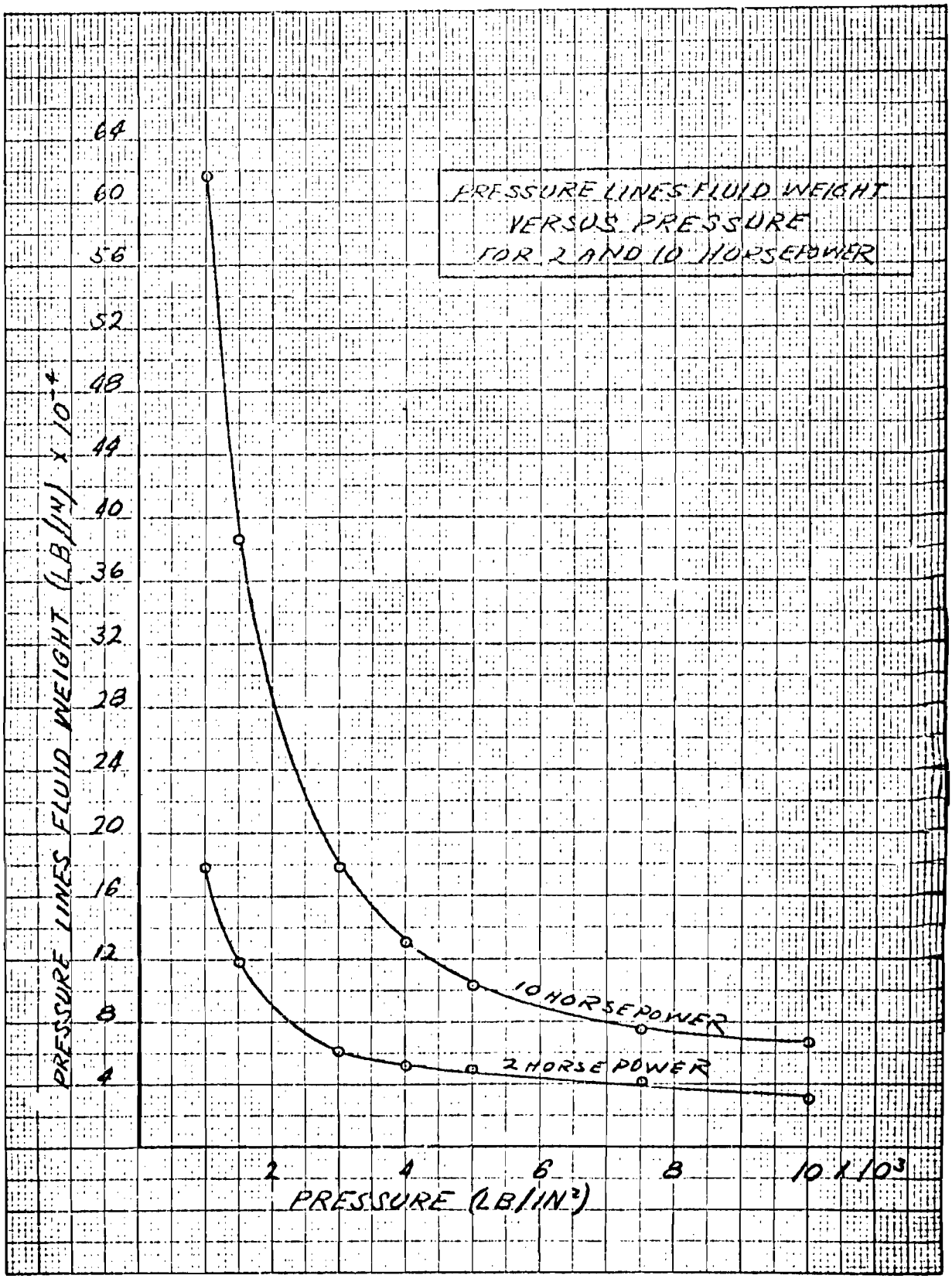

TOTAL WEIGHT OF PRESSURE LINES
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
TOTAL WEIGHT OF PRESSURE LINES (LB)
.075
.070
.065
.060
.055
.050
.045
.040
.035
.030
.025
.020
.015
.010
.005
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x 10³
PRESSURE (LB/IN²)

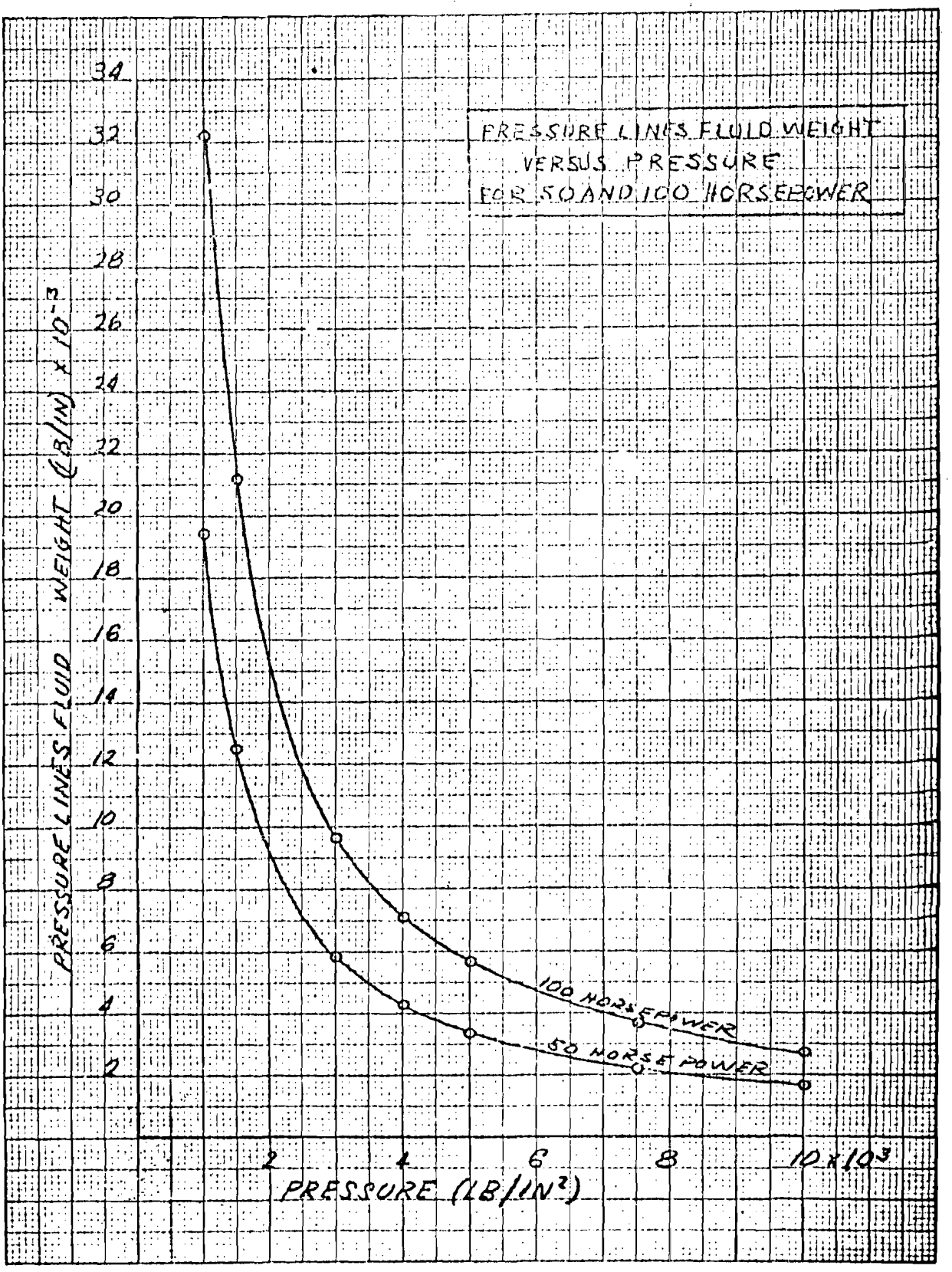

FLOW (Q) G.P.M.
LINE LOSS (ΔP) P.S.I. PER FOOT
100
10.0
1.0
0.1
.001
.01
0.1
1.0
10
100
SEE AFTER SHEET FOR
OTHER VALUES

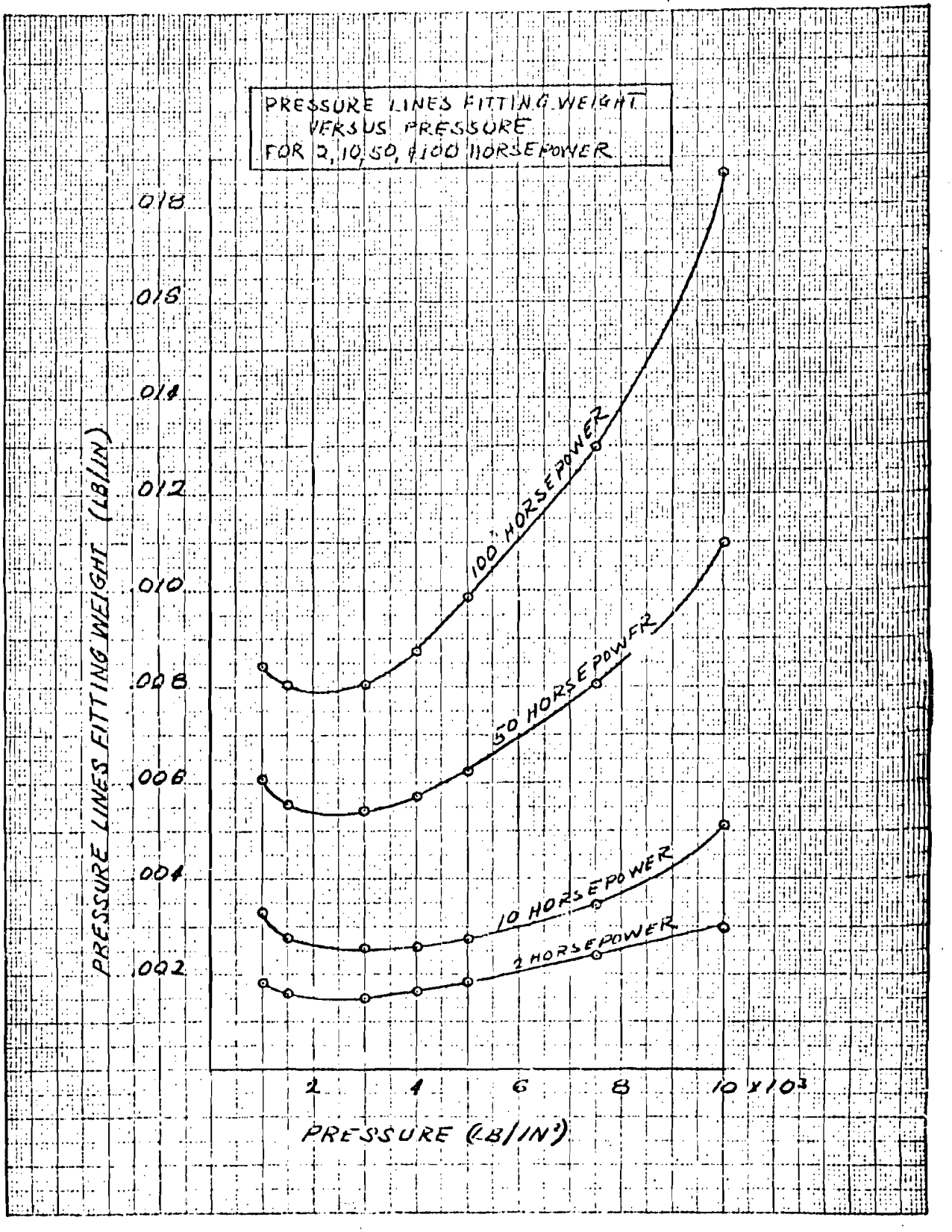

TUBE OUTSIDE DIAMETER
VERSUS PRESSURE
FOR 2, 10, 50 & 100 HORSE POWER
$OD = 10\sqrt{\frac{S_t + GP}{S_t - GP}}$
S_t = 105,000 (LB/IN²) (MIL-T-6845)
G = SAFETY FACTOR (5)
P = PRESSURE
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
TUBING OUTSIDE DIAMETER (INCHES)
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
2
4
6
8
10 x10³
PRESSURE (LB/IN²)

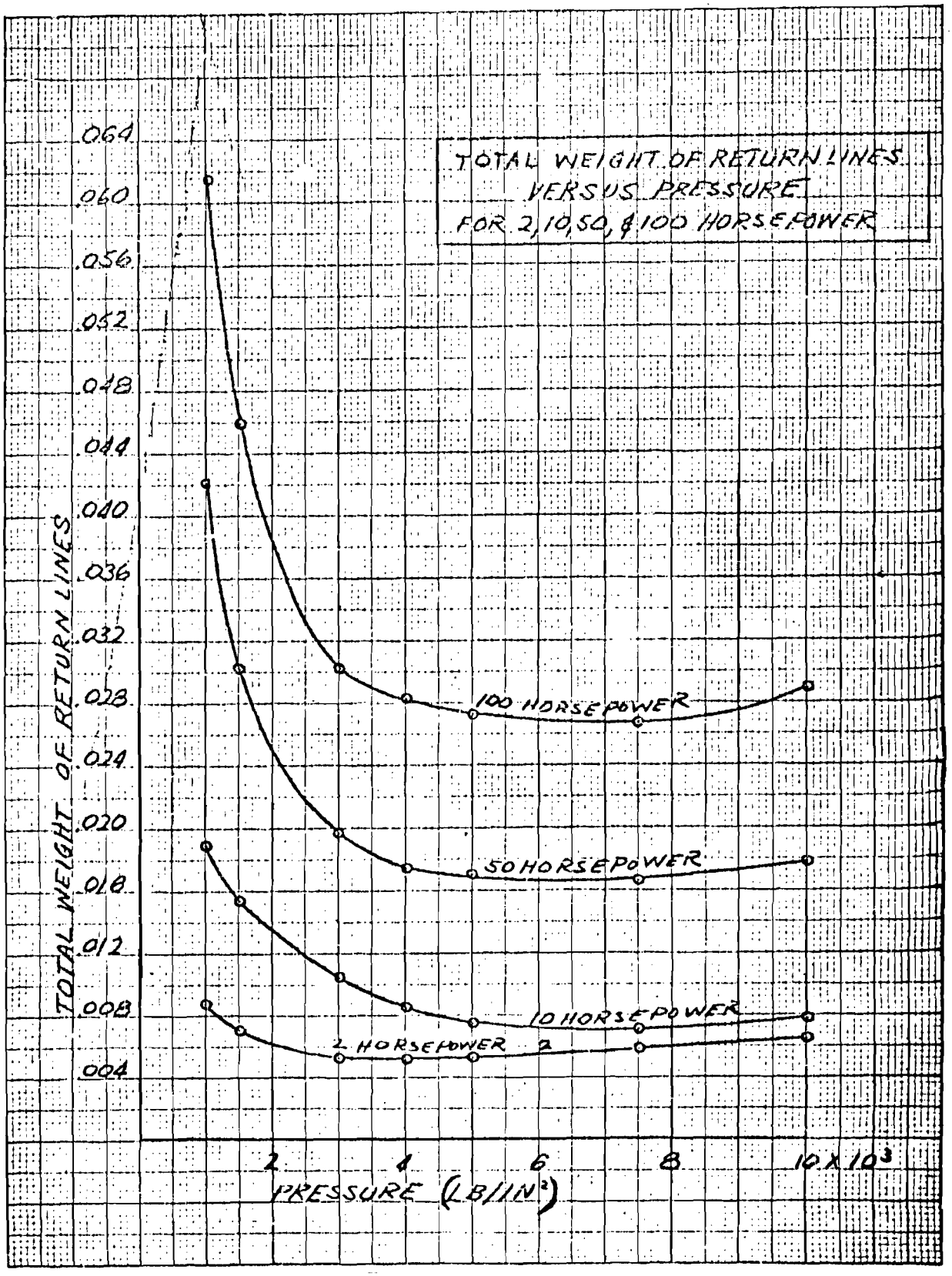

WEIGHT OF PRESSURE TUBING
VERSUS PRESSURE
FOR 2, 10, 50 & 100 HORSE POWER
.050
.045
.040
.035
.030
.025
.020
.015
.010
.005
WEIGHT OF PRESSURE TUBING (LB/IN)
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSE POWER
2
4
6
8
10 X 10³
PRESSURE (LB/IN²)

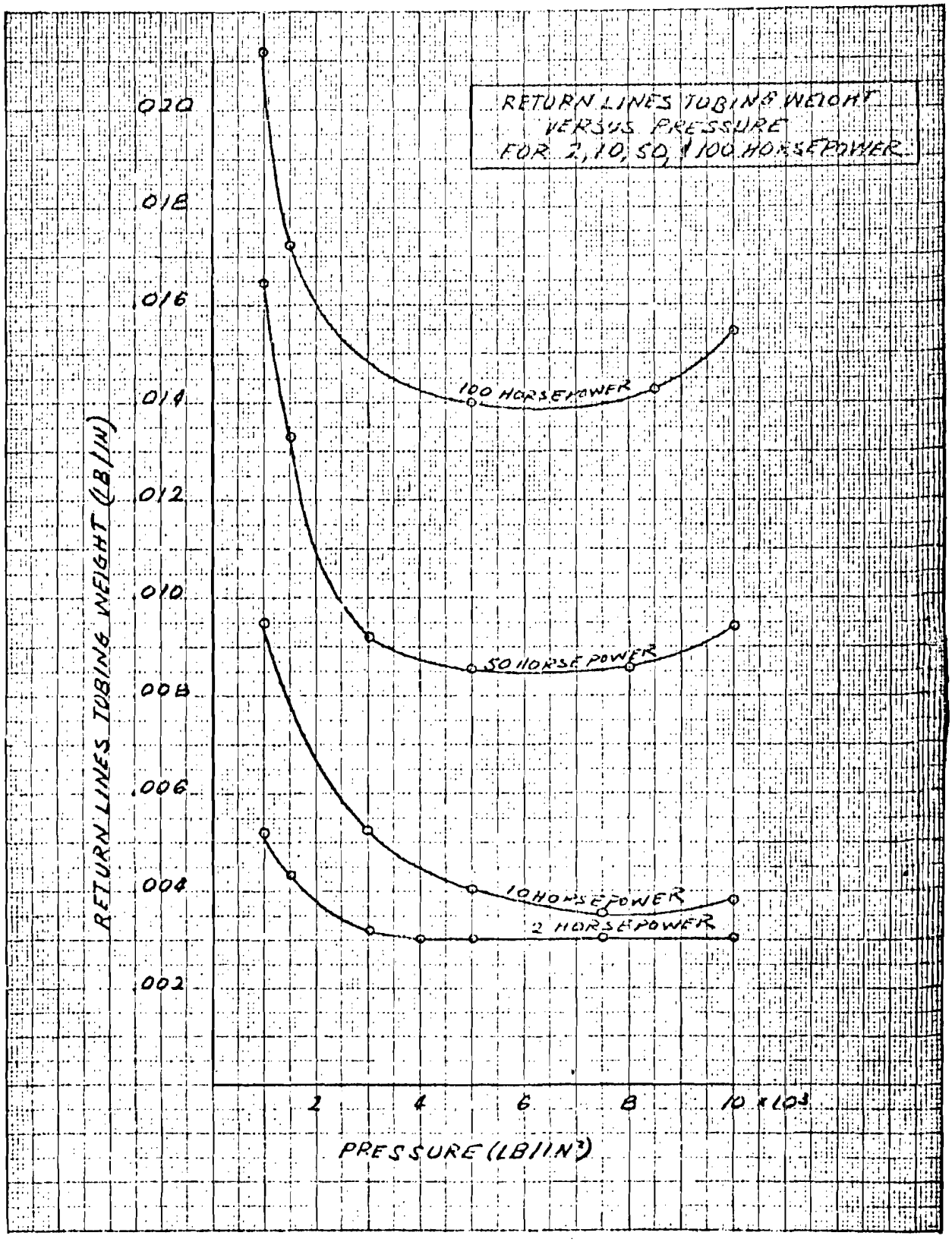

MIL-O-5606 FLUID DENSITY
VERSUS PRESSURE
FOR 80° F
SOURCE: A.F.T.R. 5997
o = ACTUAL DATA
— — = EXTRAPOLATED VALUES
1.040
1.036
1.032
1.028
1.024
1.020
1.016
1.012
1.008
1.004
1.000
α
0
2
4
6
8
10 × 10³
PRESSURE (LB/IN²)
α = ρ/ρ₁
ρ = DENSITY AT P > 0 P.S.I.G.
ρ₁ = DENSITY AT 0 P.S.I.G.
= 7.83 × 10⁻⁵ LB. SEC²/IN⁴

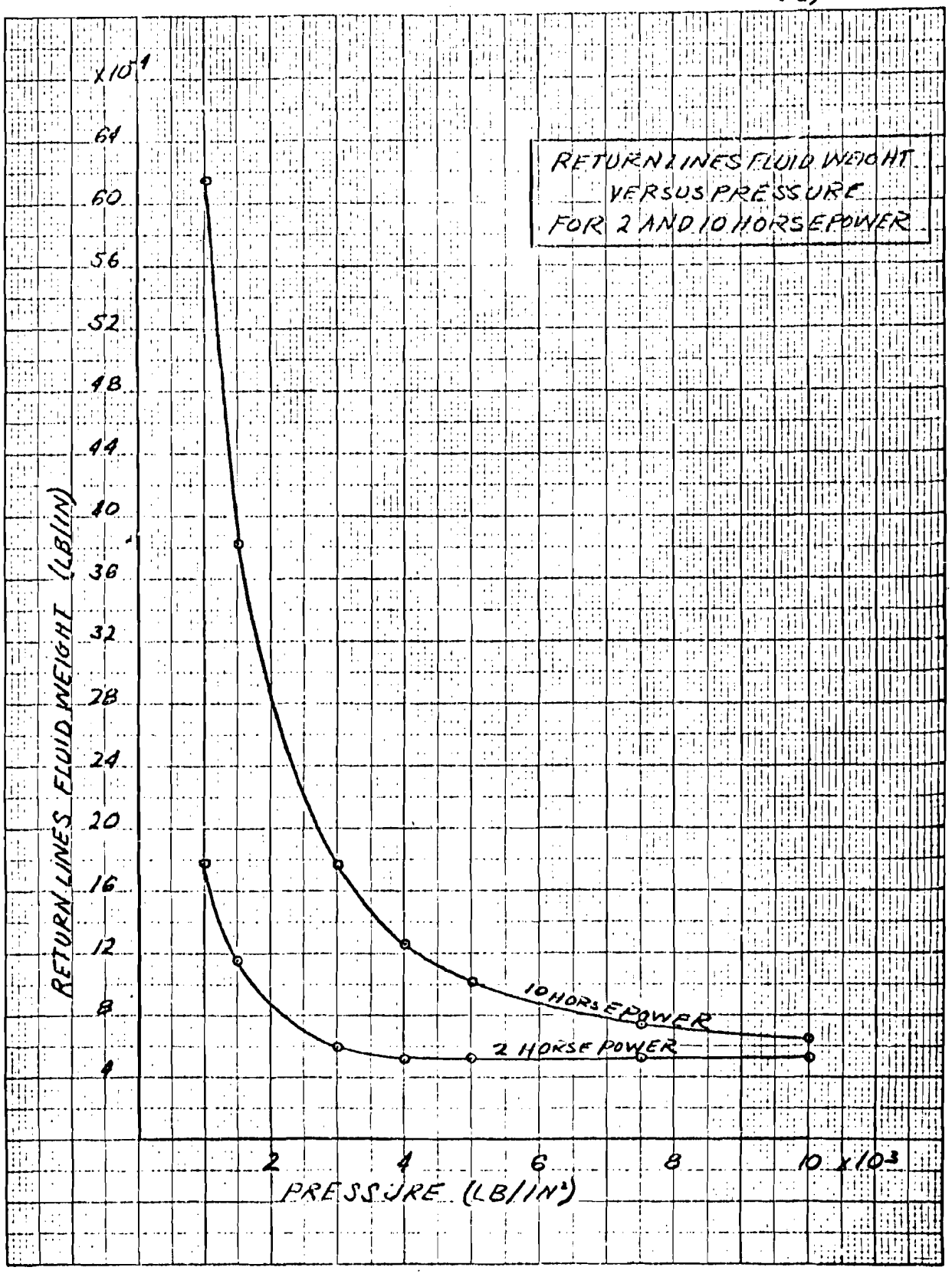

PRESSURE LINES FLUID WEIGHT
VERSUS PRESSURE
FOR 2 AND 10 HORSEPOWER
PRESSURE LINES FLUID WEIGHT (LB/IN) X 10^-4
64
60
56
52
48
44
40
36
32
28
24
20
16
12
8
4
10 HORSEPOWER
2 HORSE POWER
2
4
6
8
10 X 10^3
PRESSURE (LB/IN^2)

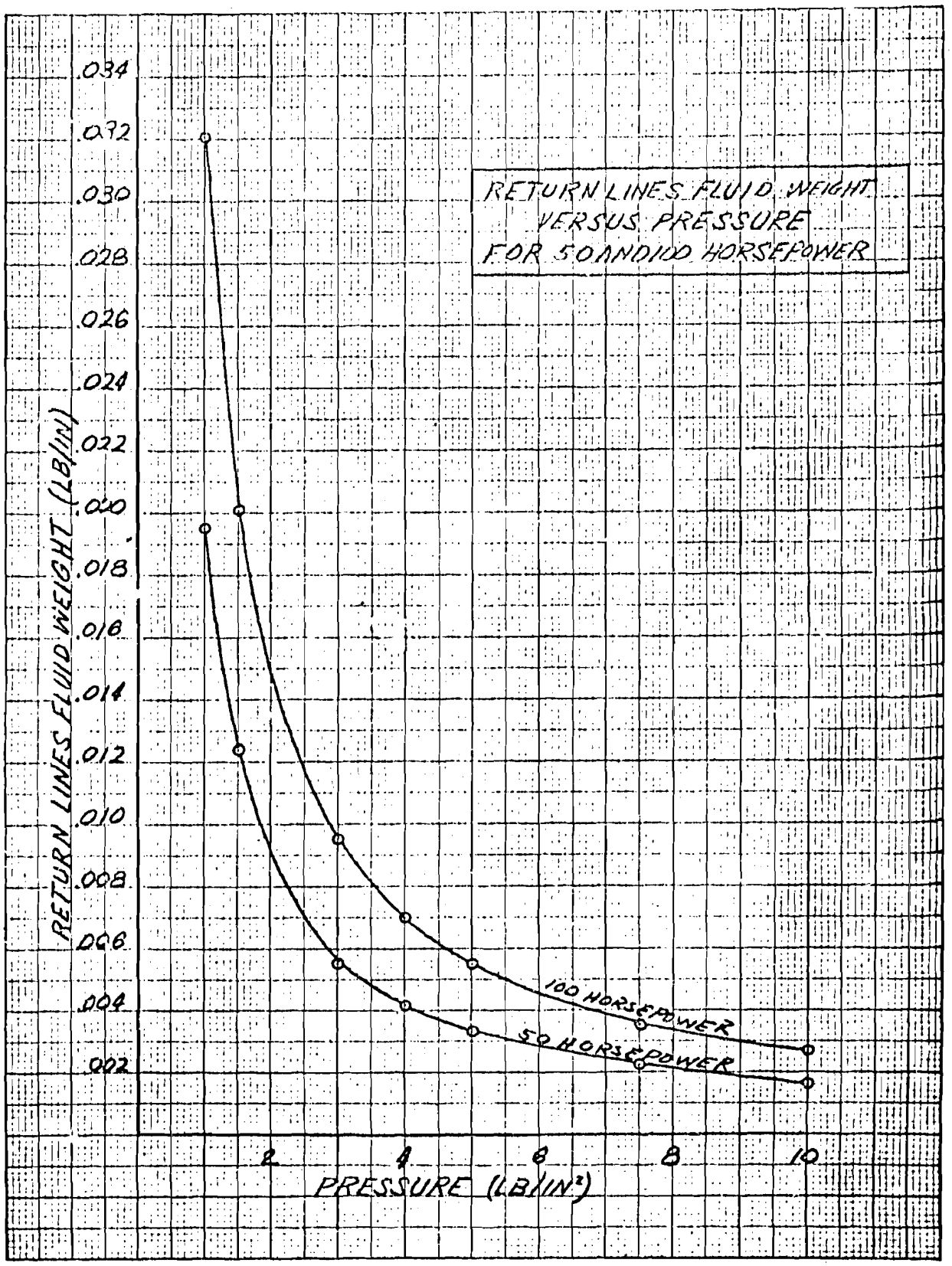

PRESSURE LINES FLUID WEIGHT
VERSUS PRESSURE
FOR 50 AND 100 HORSEPOWER
PRESSURE LINES FLUID WEIGHT (LB/IN) x 10^-3
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
100 HORSEPOWER
50 HORSE POWER
2
4
6
8
10 x 10^3
PRESSURE (LB/IN^2)

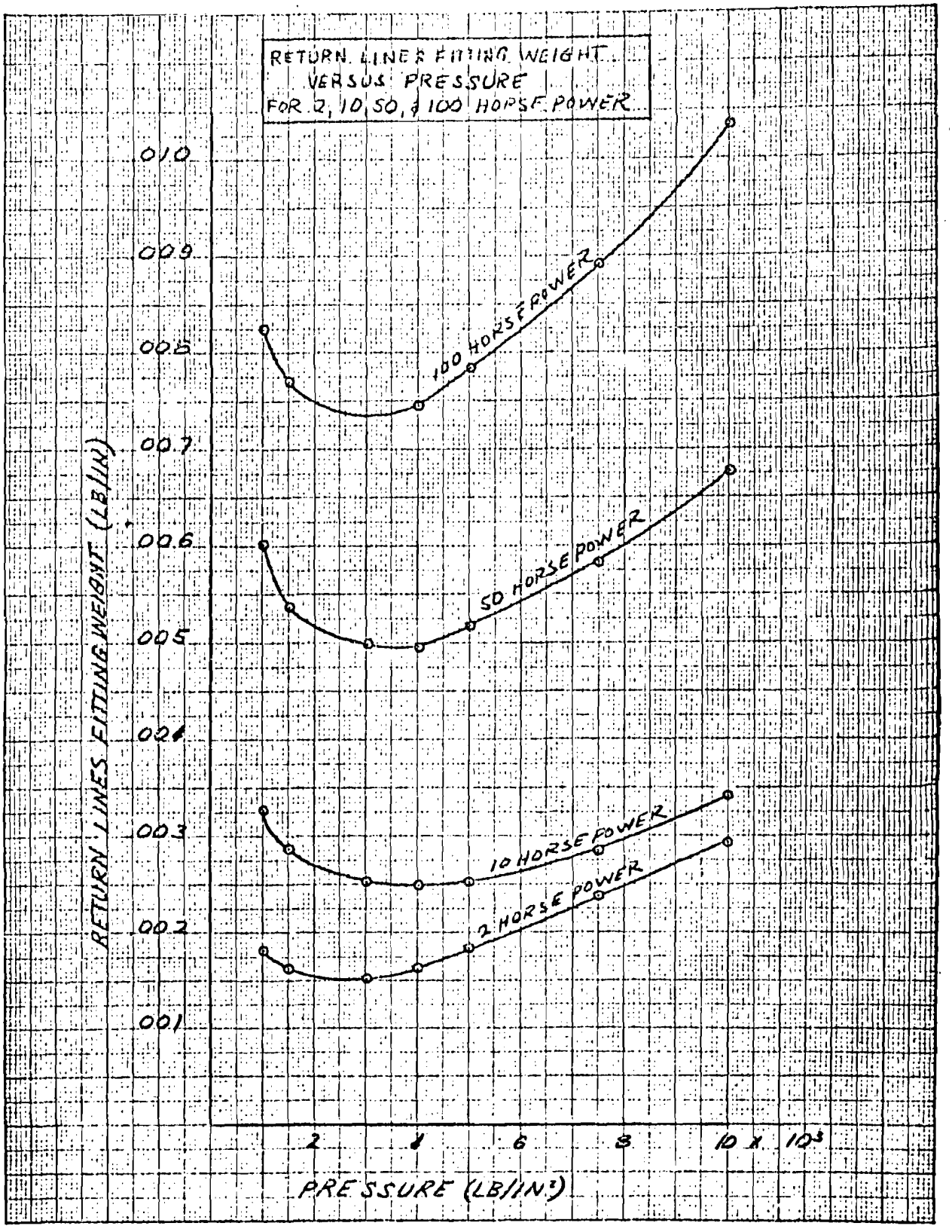

PRESSURE LINES FITTING WEIGHT
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
.018
.016
.014
.012
.010
.008
.006
.004
.002
PRESSURE LINES FITTING WEIGHT (LB/IN)
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X 10³
PRESSURE (LB/IN²)

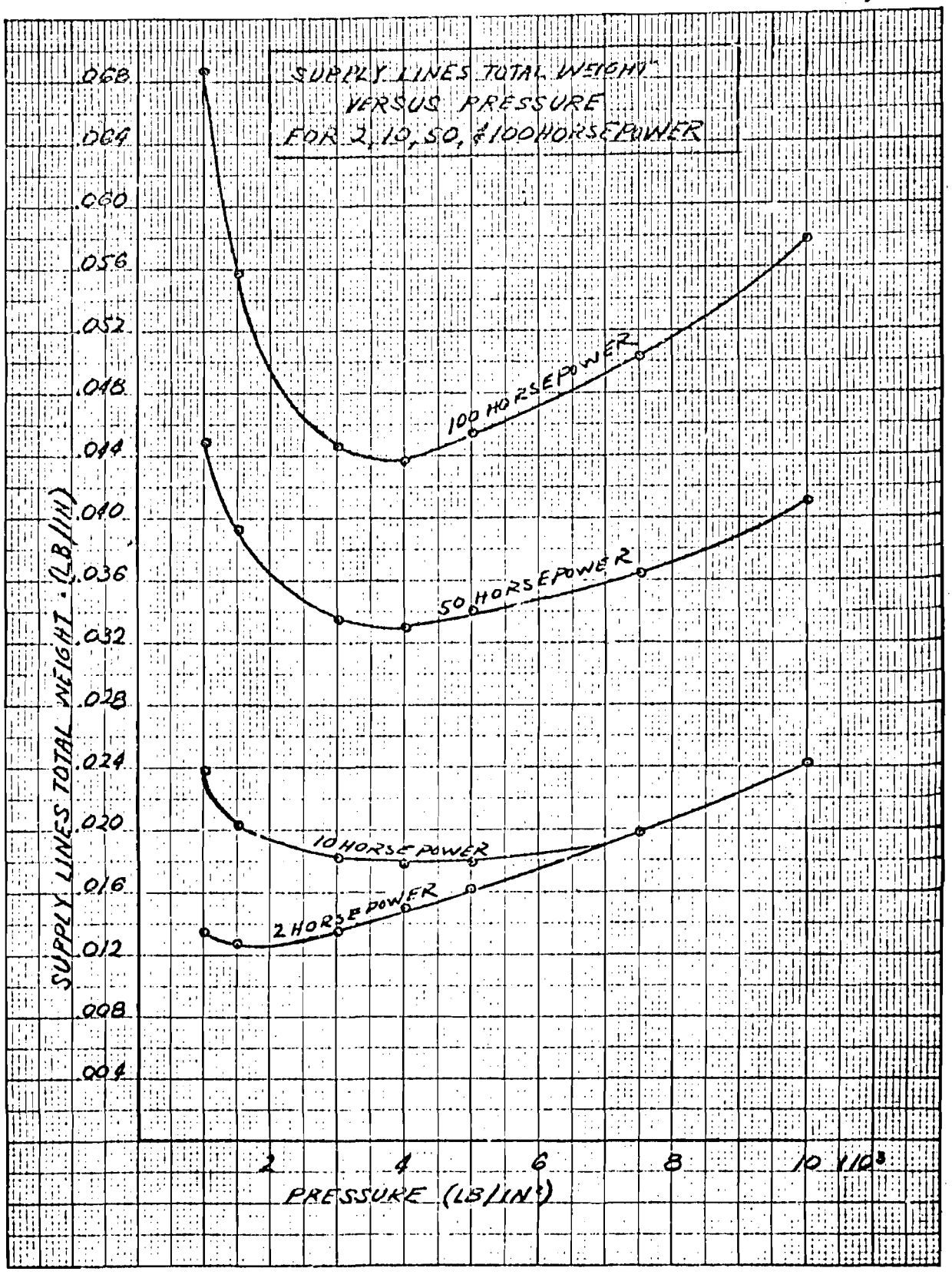

TOTAL WEIGHT OF RETURN LINES
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
TOTAL WEIGHT OF RETURN LINES
.064
.060
.056
.052
.048
.044
.040
.036
.032
.028
.024
.020
.016
.012
.008
.004
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X 10^3
PRESSURE (LB/IN^2)

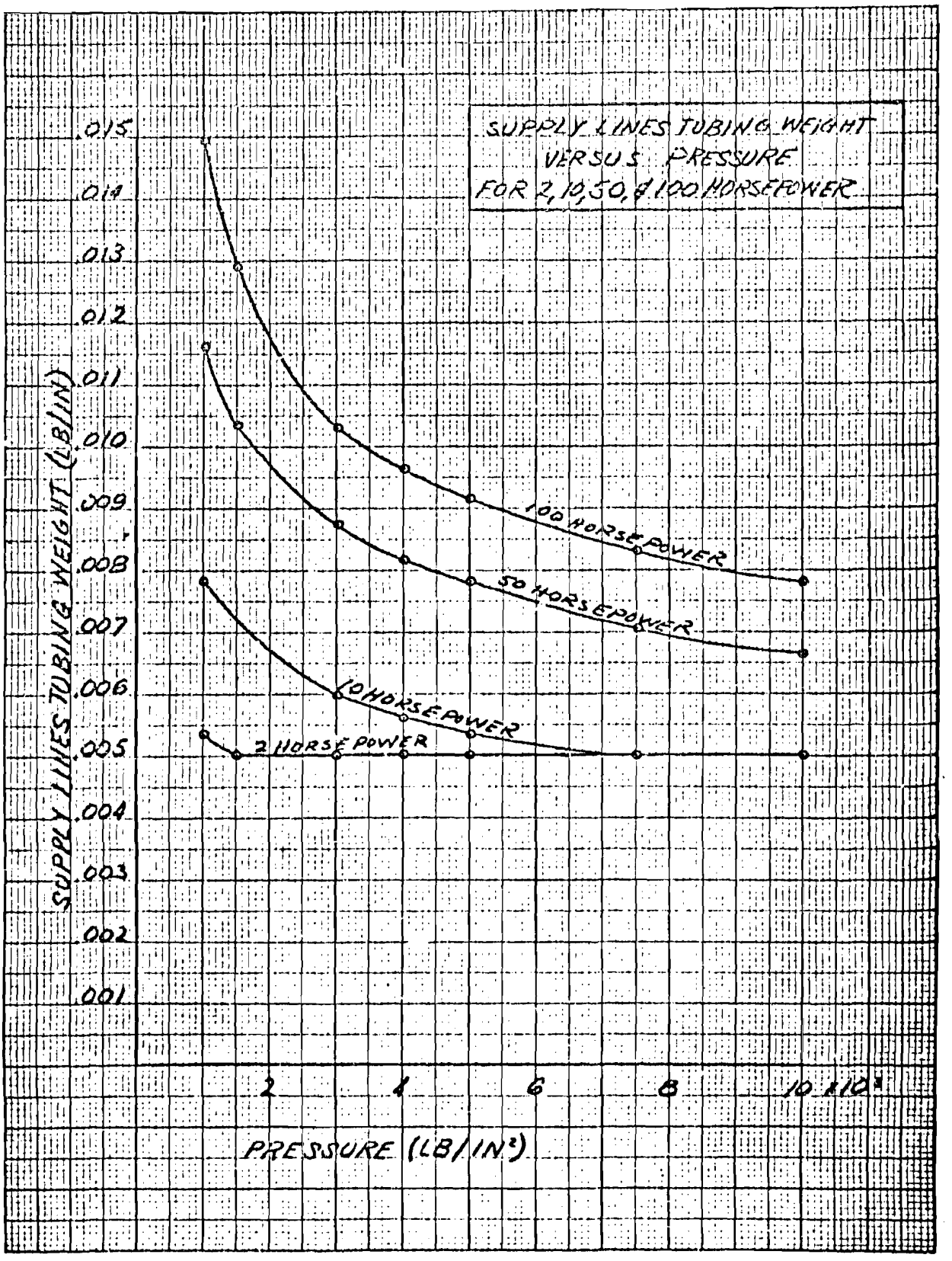

RETURN LINES TUBING WEIGHT
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
RETURN LINES TUBING WEIGHT (LB/IN)
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x 10³
PRESSURE (LB/IN²)

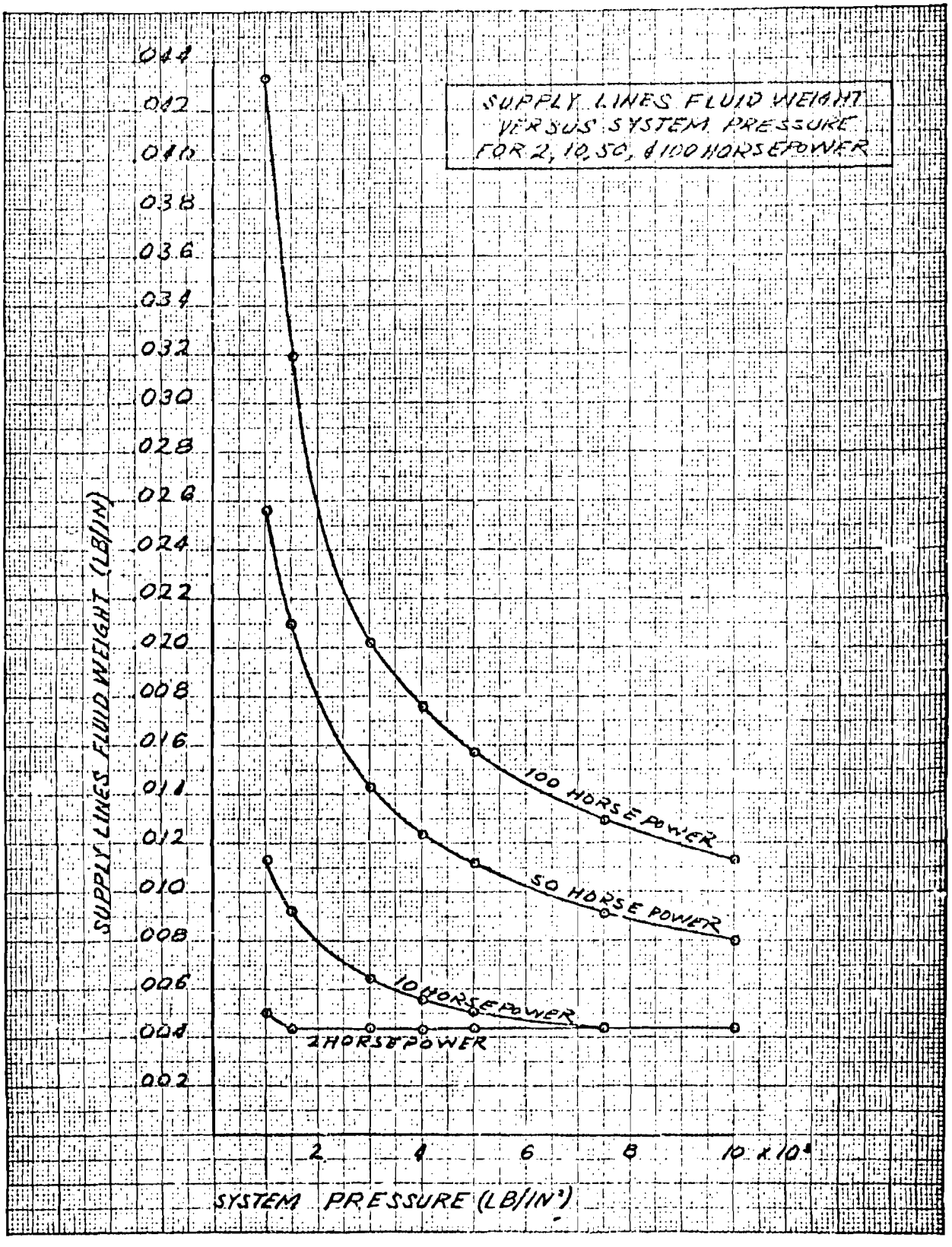

RETURN LINES FLUID WEIGHT
VERSUS PRESSURE
FOR 2 AND 10 HORSEPOWER
RETURN LINES FLUID WEIGHT (LB/IN)
x10^1
64
60
56
52
48
44
40
36
32
28
24
20
16
12
8
4
2
4
6
8
10 x10^3
PRESSURE (LB/IN²)
10 HORSEPOWER
2 HORSEPOWER

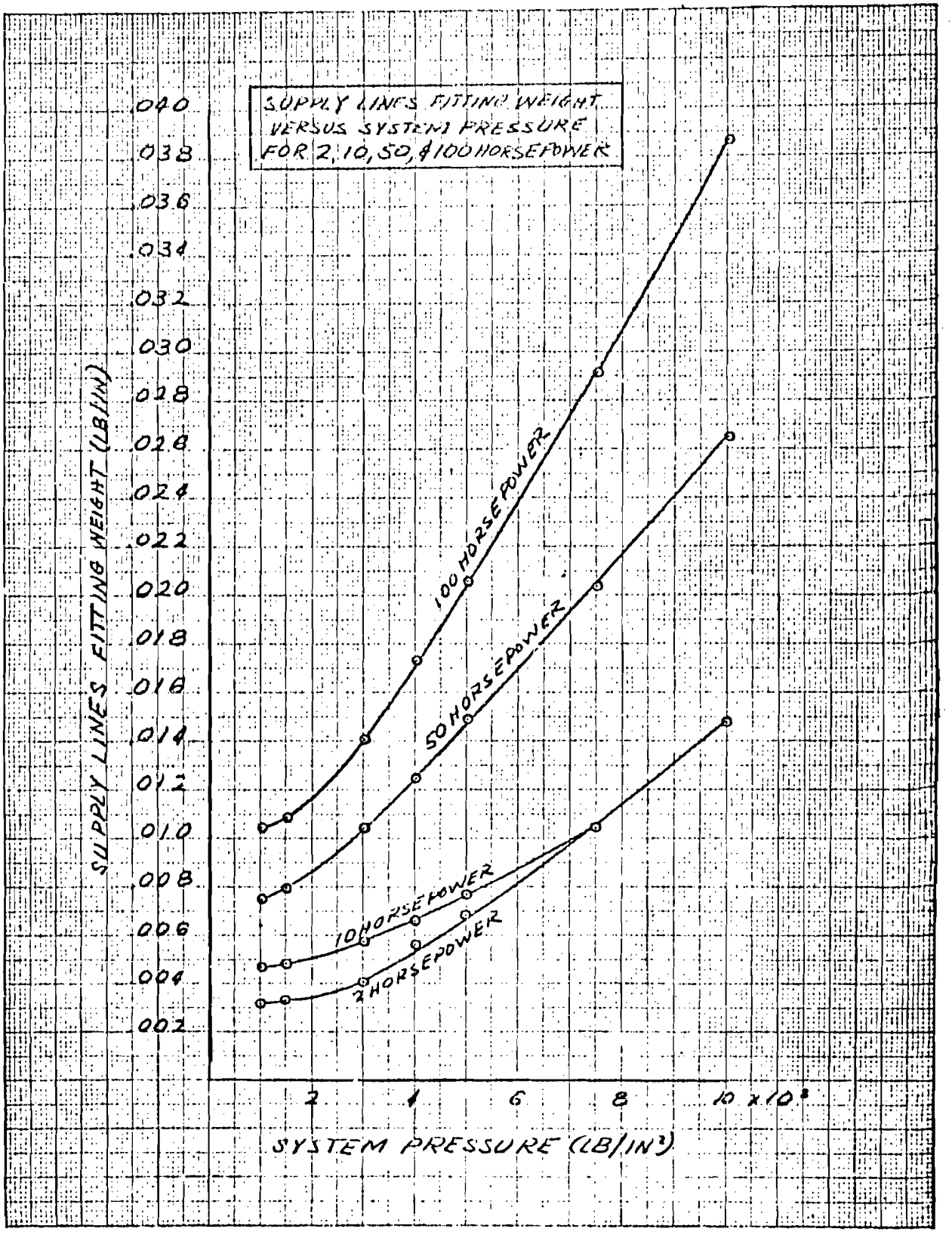

RETURN LINES FLUID WEIGHT VERSUS PRESSURE FOR 50 AND 100 HORSEPOWER
RETURN LINES FLUID WEIGHT (LB/IN)
.034
.032
.030
.028
.026
.024
.022
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
2
4
6
8
10
PRESSURE (LB/IN²)
100 HORSEPOWER
50 HORSEPOWER


RETURN LINES FITTING WEIGHT
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSE POWER
.010
.009
.008
.007
.006
.005
.004
.003
.002
.001
RETURN LINES FITTING WEIGHT (LB/IN)
100 HORSE POWER
50 HORSE POWER
10 HORSE POWER
2 HORSE POWER
2
4
6
8
10 X 10³
PRESSURE (LB/IN²)


SUPPLY LINES TOTAL WEIGHT
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
SUPPLY LINES TOTAL WEIGHT (LB/IN)
.068
.064
.060
.056
.052
.048
.044
.040
.036
.032
.028
.024
.020
.016
.012
.008
.004
2
4
6
8
10
10^3
PRESSURE (LB/IN²)


SUPPLY LINES TUBING WEIGHT
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES TUBING WEIGHT (LB/IN)
.015
.014
.013
.012
.011
.010
.009
.008
.007
.006
.005
.004
.003
.002
.001
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x10³
PRESSURE (LB/IN²)


SUPPLY LINES FLUID WEIGHT
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES FLUID WEIGHT (LB/IN)
.044
.042
.040
.038
.036
.034
.032
.030
.028
.026
.024
.022
.020
.008
.016
.014
.012
.010
.008
.006
.004
.002
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x 10⁴
SYSTEM PRESSURE (LB/IN²)


SUPPLY LINES FITTING WEIGHT
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES FITTING WEIGHT (LB/IN)
.040
.038
.036
.034
.032
.030
.028
.026
.024
.022
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x 10³
SYSTEM PRESSURE (LB/IN²)

TOTAL WEIGHT OF PRESSURE LINES

System Pressure	HP	ID	OD	Tubing WT(#/in)	Fluid WT(#/in)	Fitting WT(#/in)	Total Weight #/in
1000	2	.275	.313	.0052	.001782	.00181	.008792
	10	.508	.548	.00948	.00617	.00327	.01892
	50	.901	.946	.0187	.0194	.00609	.04419
	100	1.160	1.219	.0321	.0322	.00841	.07271
1500	2	.221	.261	.00433	.001103	.00159	.007105
	10	.401	.441	.00876	.00386	.00277	.01539
	50	.721	.773	.01745	.01249	.00557	.03551
	100	.940	1.010	.0309	.0212	.00805	.06015
3000	2	.158	.198	.0032	.000601	.00152	.005321
	10	.272	.313	.00538	.001784	.00253	.009694
	50	.492	.566	.0176	.00581	.00544	.02885
	100	.632	.728	.0293	.00962	.00805	.04697
4000	2	.1475	.1875	.00301	.000523	.00163	.005163
	10	.232	.281	.00565	.001305	.00259	.009545
	50	.418	.506	.01825	.00423	.00572	.02820
	100	.542	.656	.03065	.00711	.00879	.04655
5000	2	.147	.1875	.00304	.000521	.00183	.005391
	10	.206	.263	.00600	.001035	.00273	.009765
	50	.372	.474	.0194	.00336	.00626	.02902
	100	.482	.615	.03275	.00565	.00984	.04824
7500	2	.129	.1875	.00416	.000406	.00238	.006946
	10	.176	.251	.00743	.000755	.00343	.011615
	50	.300	.435	.0223	.002195	.00806	.032555
	100	.385	.558	.0367	.00362	.01297	.05329
10000	2	.112	.1875	.00508	.000309	.00293	.008319
	10	.165	.275	.01087	.000667	.00502	.016557
	50	.260	.432	.0267	.00162	.01099	.03931
	100	.334	.557	.0447	.002735	.01865	.066085

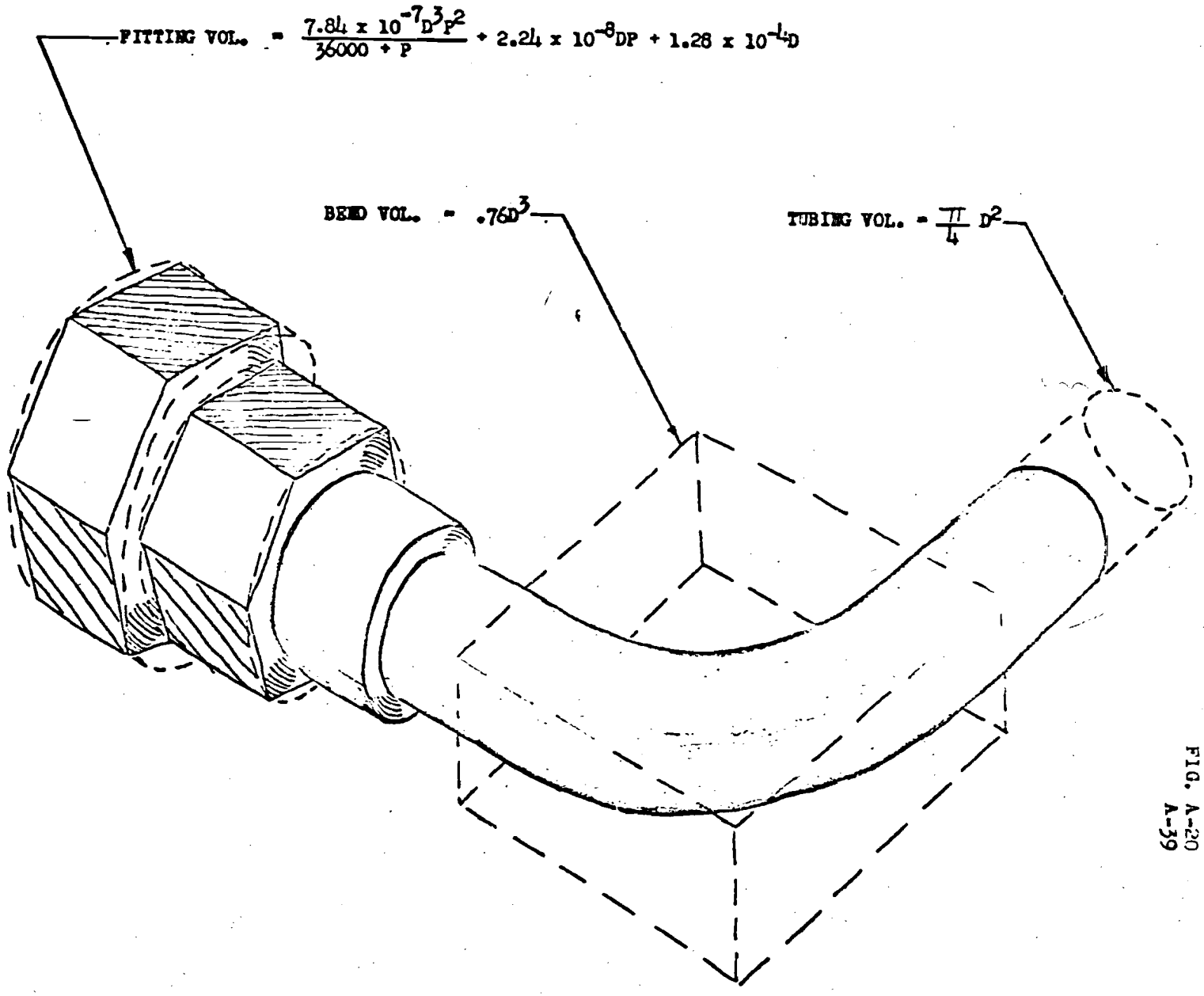
TOTAL WEIGHT OF RETURN LINES

System Pressure	Back Press.	HP	ID	OD	Tubing Wt(#/IN)	Fluid Wt(#/IN)	Fitting Wt(#/IN)	Total Wt. (#/IN)
1000	500	2	.273	.313	.0052	.001775	.001814	.008789
		10	.508	.548	.00948	.00615	.003269	.018899
		50	.901	.941	.01648	.01947	.00606	.04201
		100	1.16	1.20	.0212	.03204	.00325	.06149
1500	750	2	.221	.261	.00433	.001164	.001633	.007027
		10	.401	.441	.00876	.003835	.00286	.015455
		50	.721	.761	.0133	.0124	.005369	.031069
		100	.940	.980	.01723	.03105	.007702	.045982
3000	1500	2	.158	.198	.0032	.000558	.001523	.005281
		10	.272	.312	.00523	.001765	.002525	.009520
		50	.492	.532	.00919	.00554	.004987	.019717
		100	.632	.678	.01353	.00953	.007173	.030233
4000	2000	2	.1475	.1875	.00301	.000521	.001628	.005159
		10	.232	.272	.00474	.00126	.00249	.00849
		50	.418	.460	.00829	.00416	.004964	.017414
		100	.542	.596	.0138	.00700	.007455	.028255
5000	2500	2	.1475	.1875	.00301	.000524	.001841	.005375
		10	.206	.246	.00402	.00101	.002515	.007545
		50	.372	.420	.00854	.00330	.005182	.017022
		100	.482	.542	.0140	.00552	.007845	.027365
7500	3750	2	.1475	.1875	.00301	.000526	.002362	.005898
		10	.176	.216	.00352	.000739	.002946	.007105
		50	.300	.358	.00858	.002245	.005828	.016653
		100	.385	.460	.01423	.00354	.00892	.02669
10000	5000	2	.147	.1875	.00304	.000528	.002932	.006500
		10	.165	.210	.00380	.000649	.003407	.007856
		50	.260	.331	.00943	.00161	.006765	.017805
		100	.334	.426	.01546	.00266	.010735	.028855

TABLE A-5
A-38

TOTAL WEIGHT OF SUPPLY LINES

Pressure	HP	ID	OD	Tubing (#/in^2)	Fluid (#/in^2)	Fittings (#/in^2)	Total (#/in^2)
1000	2	.460	.530	.00534	.00502	.00315	.01349
	10	.690	.760	.00781	.0113	.00470	.02381
	50	1.040	1.110	.0116	.0256	.00626	.04482
	100	1.350	1.420	.01493	.0433	.01042	.06865
1500	2	.430	.500	.00501	.00439	.00330	.01270
	10	.620	.690	.00629	.00913	.00483	.02025
	50	.925	.925	.01035	.021	.00797	.03932
	100	1.160	1.160	.01290	.03195	.01089	.05574
3000	2	.430	.500	.00501	.00439	.00407	.01347
	10	.520	.590	.00598	.00642	.00579	.01819
	50	.777	.847	.00875	.01432	.01047	.03354
	100	.922	.992	.01031	.0202	.01408	.04459
4000	2	.430	.500	.00501	.00439	.00563	.01503
	10	.485	.555	.00560	.00558	.00663	.01781
	50	.723	.793	.00817	.0124	.01251	.03308
	100	.861	.931	.00966	.0176	.01735	.04361
5000	2	.430	.500	.00501	.00439	.00683	.01623
	10	.460	.530	.00535	.00502	.00574	.01791
	50	.690	.760	.00781	.0112	.01493	.03404
	100	.814	.884	.00915	.01573	.02062	.0455
7500	2	.430	.500	.00501	.00439	.01043	.01983
	10	.430	.500	.00501	.00439	.01043	.01983
	50	.620	.690	.00706	.00913	.02040	.03659
	100	.737	.807	.00831	.0129	.02917	.05037
10000	2	.430	.500	.00501	.00439	.01479	.02419
	10	.430	.500	.00501	.00439	.01479	.02419
	50	.580	.650	.00664	.00800	.02649	.04113
	100	.690	.760	.00781	.01130	.03866	.05777



TOTAL PLUMBING SPACE = FITTING VOLUME + BEND VOLUME + TUBING VOLUME

FIG. A-20

III. Determination of the Space Occupied by the Transmission Lines

The total space occupied by the transmission lines is considered to be the effective volume of the fittings, plus the block volume of the bends plus the volume of the tubing as illustrated in figure A-20.

$$\text{Vol/In} = \frac{7.84 \times 10^{-7} D^3 P^2}{26000 + P} + 2.24 \times 10^{-8} DP + 1.28 \times 10^{-4} D + .760 D^3 + \frac{\pi}{4} D^2$$

D = Tubing outside diameter
P = System pressure

All of these values are dependent on the outside diameter of the tubing; therefore, the space occupied by the plumbing will decrease as long as the outside diameter of the tubing decreases. Since the inside diameter of the tubing decreases with any increase in pressure, the outside diameter will decrease until the wall thickness of the tubing increases faster than the inside diameter decreases.

The diameters used for space are the same as those determined for weight.

The space occupied by pressure, return, and supply lines is tabulated in Tables A-6, A-7, and A-8 respectively. The space is presented graphically in figures A-21A, A-21B, A-25, and A-29.

For pressure lines there is no exact minimum space per horsepower. For two horsepower the minimum space is near 4000 psi due to minimum tube size. For ten horsepower the minimum is near 7500 psi while for 50 and 100 horsepower no minimum is reached in the range investigated.

For return and supply lines no minimum is reached except for those horsepowers that become minimum sizes.

III. a. Tubing Space

The space occupied by the tubing, as shown in figure A-20 is the actual volume of the tubing and can be calculated from the equation:

$$\text{Vol/In} = \frac{\pi}{4} D^2$$

D = Outside Diameters of the Tubing

The space occupied by tubing for pressure, return and supply lines is tabulated in Tables A-6, A-7, and A-8. The results are presented graphically in figures A-22, A-26, and A-30.

III. b. Bend Space

The space occupied by the bends is illustrated in figure A-20. The volume of this block can be calculated by the following equation:

$$\text{Vol/In} = (R + \frac{D}{2})^2 \times D \times \frac{N}{L}$$

R = Bend Radius to centerline of tube (IN)
D = Outside Diameter of tube (IN)
N = Number of 90° Bends
L = Length of Tubing (IN)

The minimum bend radius of present tubing was found to be 3.5 times the outside diameter. If this value is substituted in the above equation it becomes:

$$\text{Vol/In} = 16\ D^3 \times \frac{N}{L}$$

Statistical data was collected from several airplanes and it was found that the average number of 90° bends per inch of tubing was .0475. Therefore the equation becomes:

$$\text{Vol/In} = .76\ D^3$$

The bend space of pressure, return, and supply lines is tabulated in Tables A-6, A-7, and A-8 respectively. The results are presented graphically in figures A-23A, A-23B, A-27A, A-27B, and A-31.

III. c. Fitting Space

The space occupied by the fittings, as illustrated in figure A-20, is the effective volume occupied by the fitting.

The volume of the fittings is considered in three sections:

1. The actual volume of the sleeve extending beyond the nut.
2. The volume of the cylinder that would enclose the nut.
3. The volume of a cylinder that would enclose the fitting between the nuts.

The effective volume of the fittings was found for a number of airplanes for the 1/2 inch tubing. Dividing this colume by the length of 1/2 inch tubing we obtain the effective volume per inch of tubing -- .082 IN^3/IN.

The equation for the weight of fittings has already been developed. (Appendix A, Section I)

$$Wt/In = \frac{2.98 \times 10^{-5} D^3 P^2}{36000 + P} + 8.5 \times 10^{-7} DP + 4.87 \times 10^{-3} D$$

The weight divided by the density and multiplied by an appropriate constant equals the volume of the fittings per inch of tubing. This equation is set equal to the volume of fittings for 1/2 inch tubing and 3000 psi, found above, and the new constants evaluated.

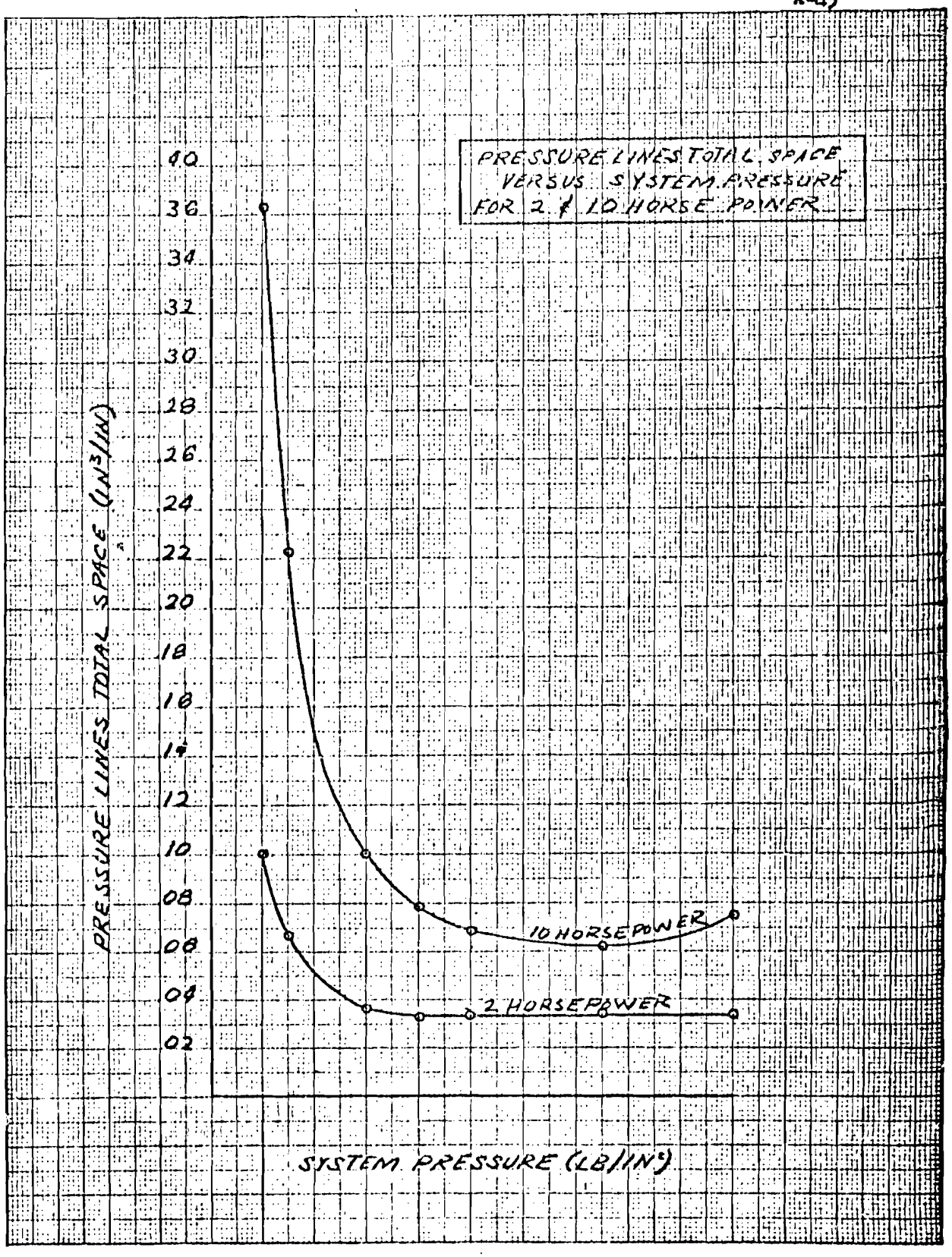
$$Vol/In = .082 = \frac{K}{\rho} \frac{2.98 \times 10^{-5}(.5)^3(3000)^2}{36000 + P} + 8.5 \times 10^{-7}(.5)(3000) + 4.87 \times 10^{-3} D$$

$$\frac{K}{\rho} = .0263$$

Solving for new constants:

$$Vol/In = \frac{7.84 \times 10^{-7} D^3 P^2}{36000 + P} + 2.24 \times 10^{-8} DP + 1.28 \times 10^{-4} D$$

The values for 2, 10, 50, and 100 horsepower pressure, return, and supply lines are tabulated in Tables A-6, A-7, A-8 and plotted in figures A-24, A-28, and A-32.


PRESSURE LINES TOTAL SPACE
VERSUS SYSTEM PRESSURE
FOR 2 & 10 HORSE POWER
PRESSURE LINES TOTAL SPACE (IN³/IN)
40
36
34
32
30
28
26
24
22
20
18
16
14
12
10
08
06
04
02
10 HORSEPOWER
2 HORSEPOWER
SYSTEM PRESSURE (LB/IN²)

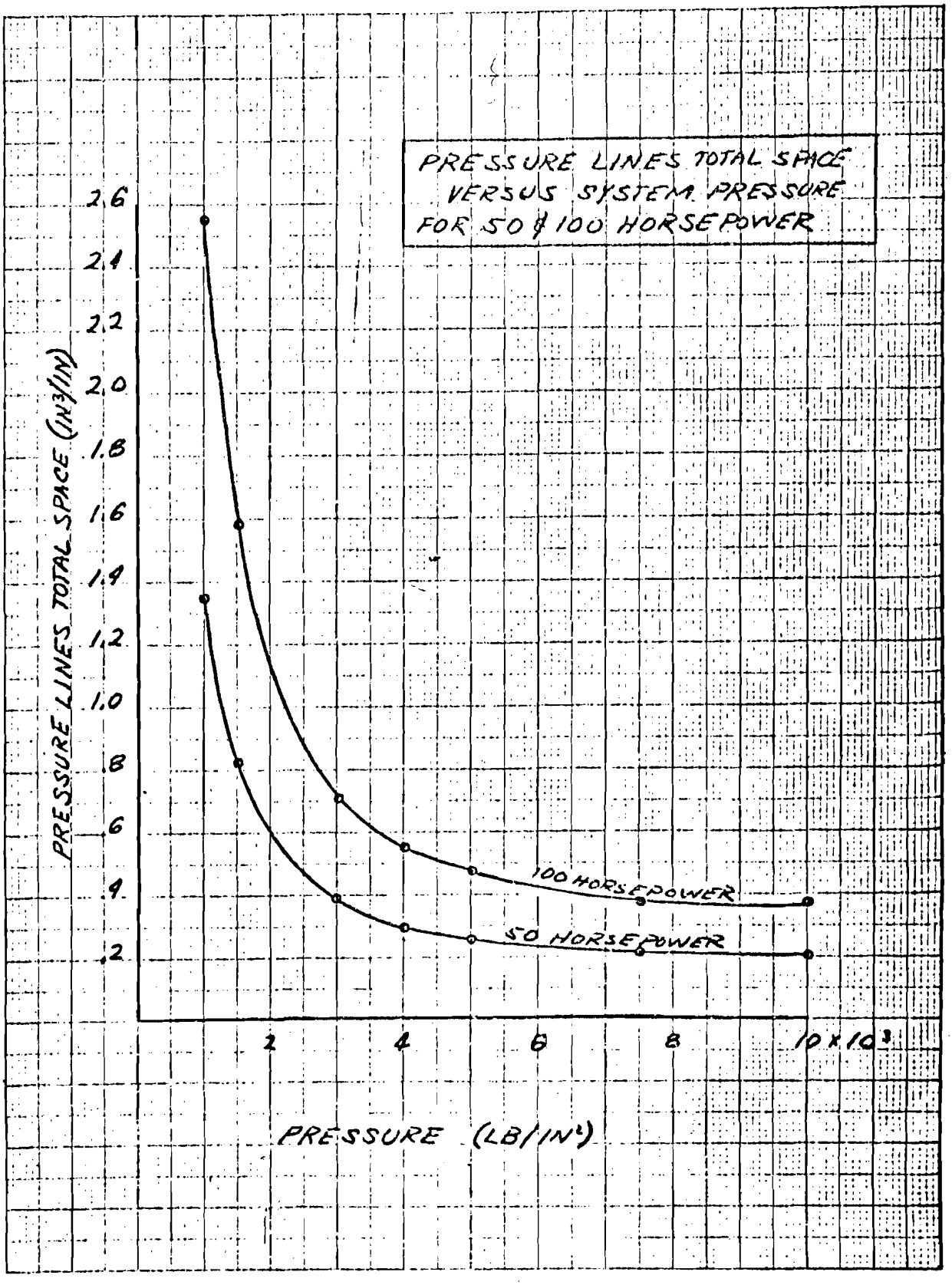

PRESSURE LINES TOTAL SPACE
VERSUS SYSTEM PRESSURE
FOR 50 & 100 HORSEPOWER
PRESSURE LINES TOTAL SPACE (IN³/IN)
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
100 HORSEPOWER
50 HORSEPOWER
2
4
6
8
10 x 10³
PRESSURE (LB/IN²)

FIG. A-22
A-45

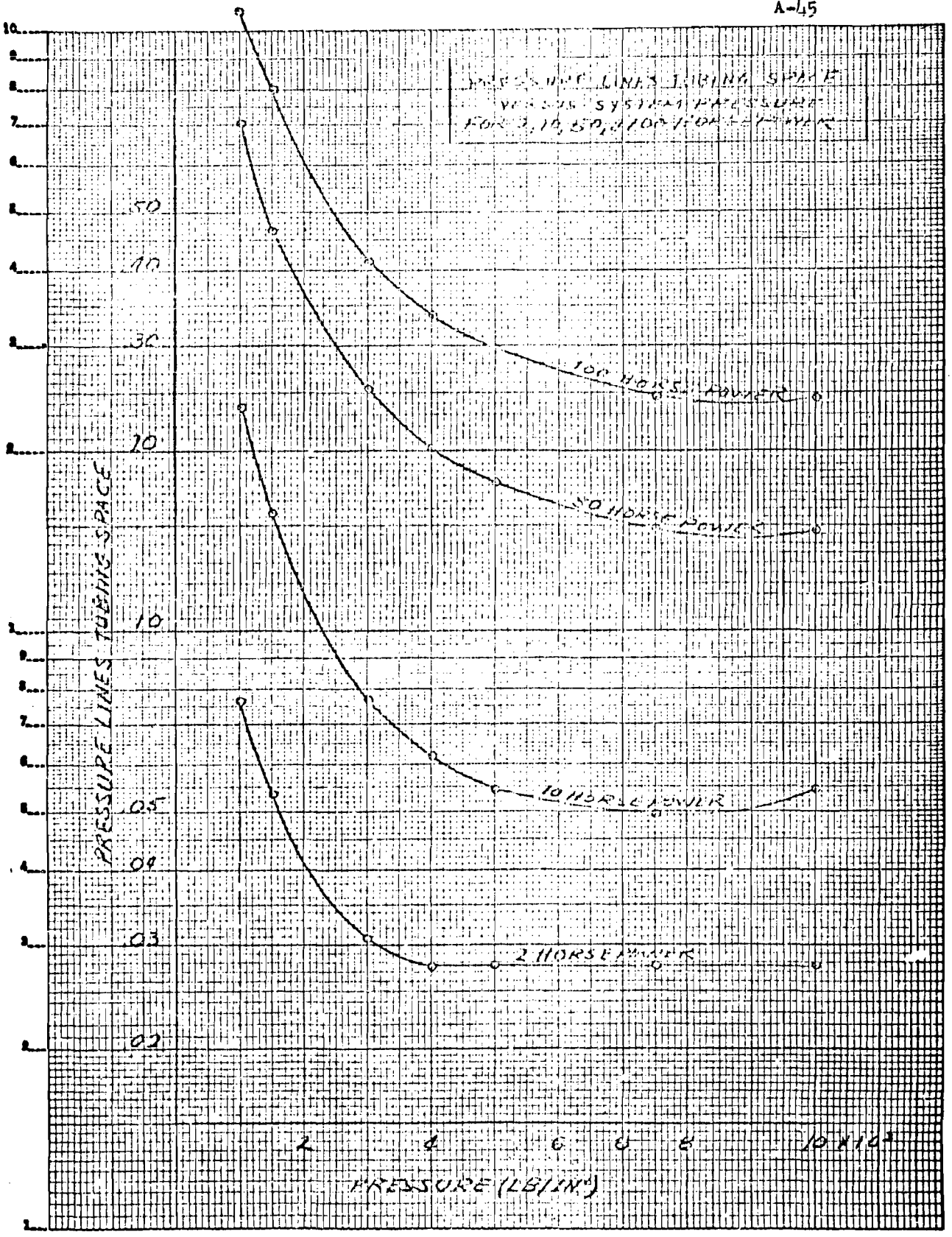
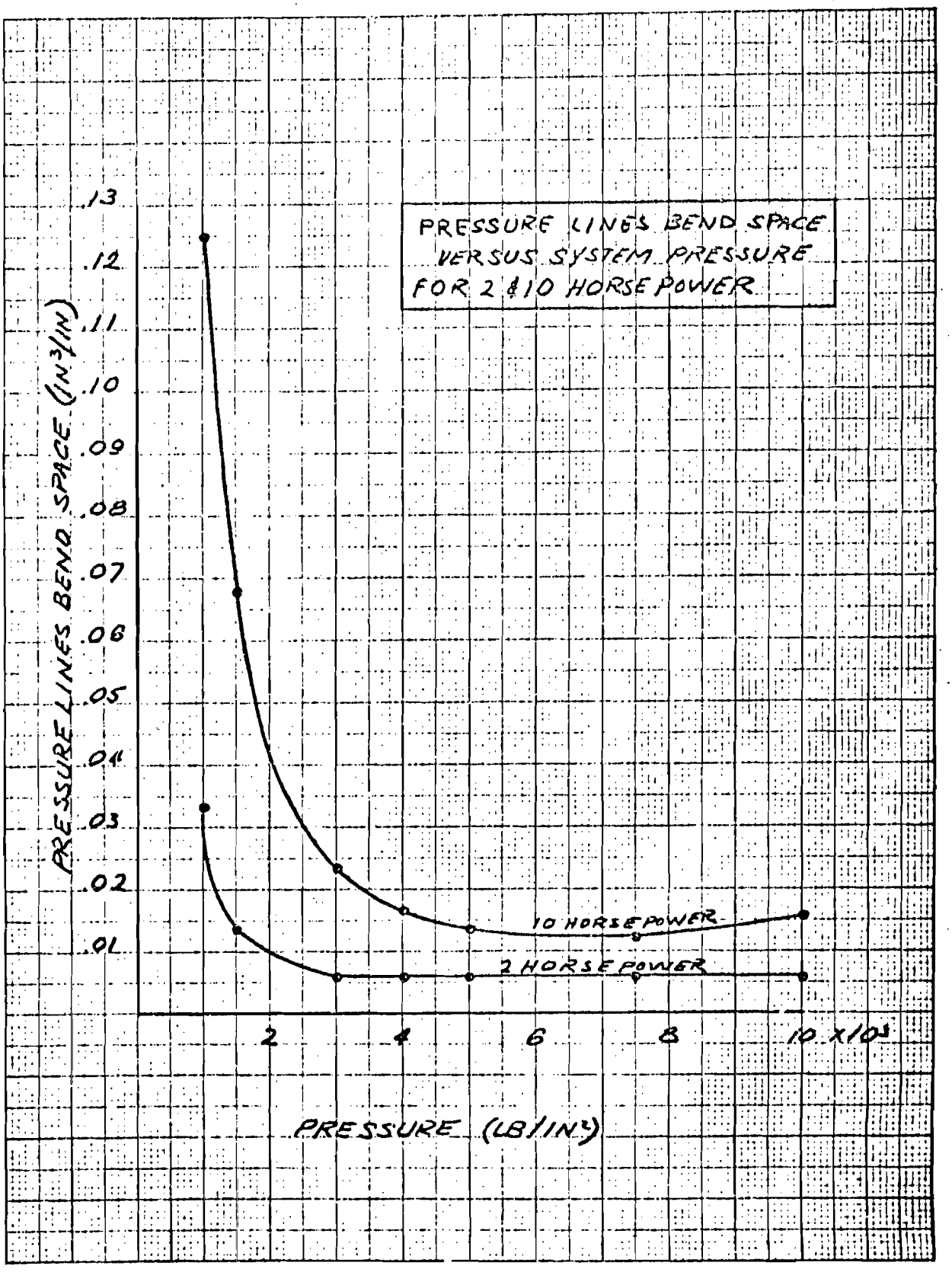
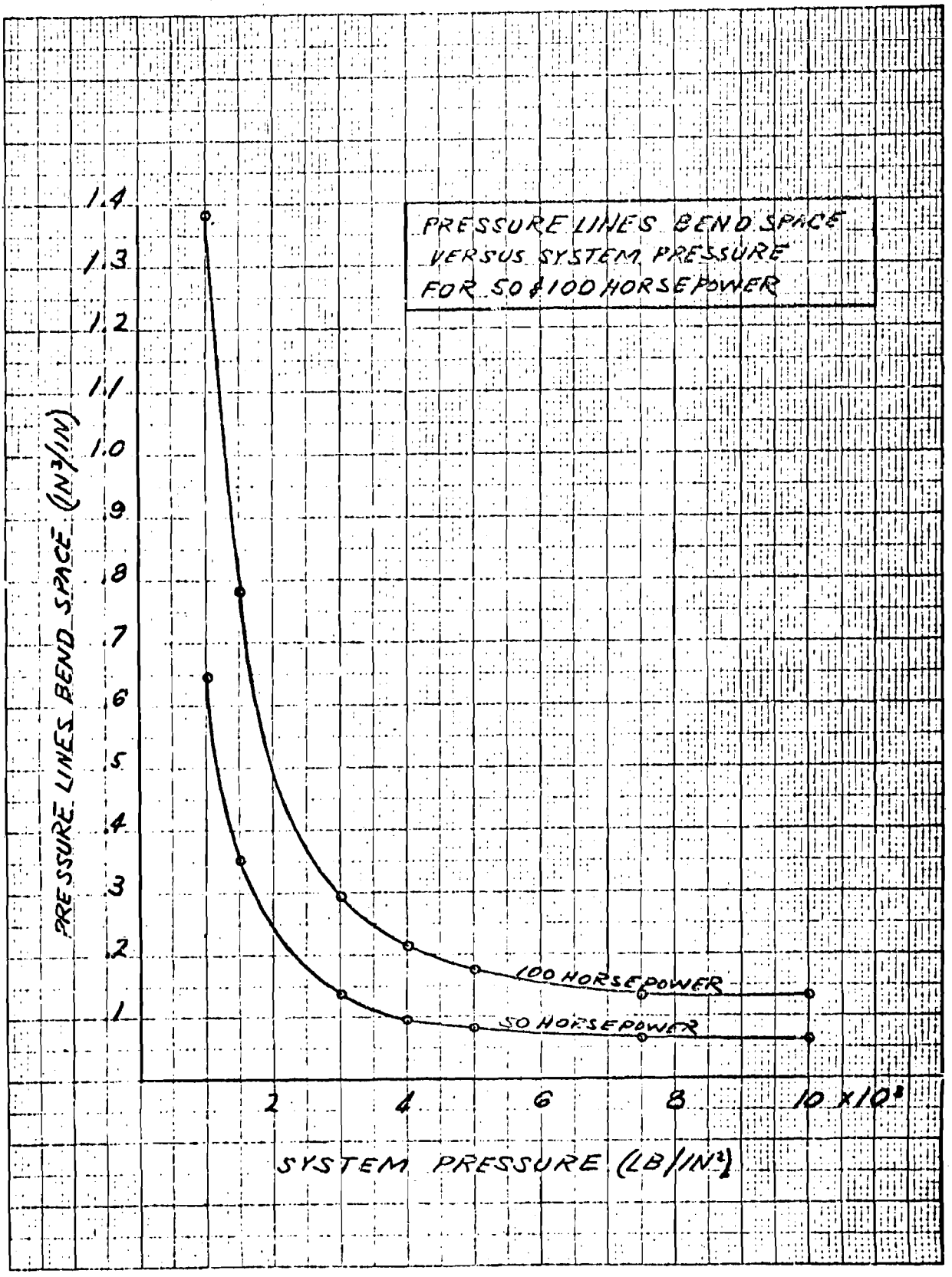
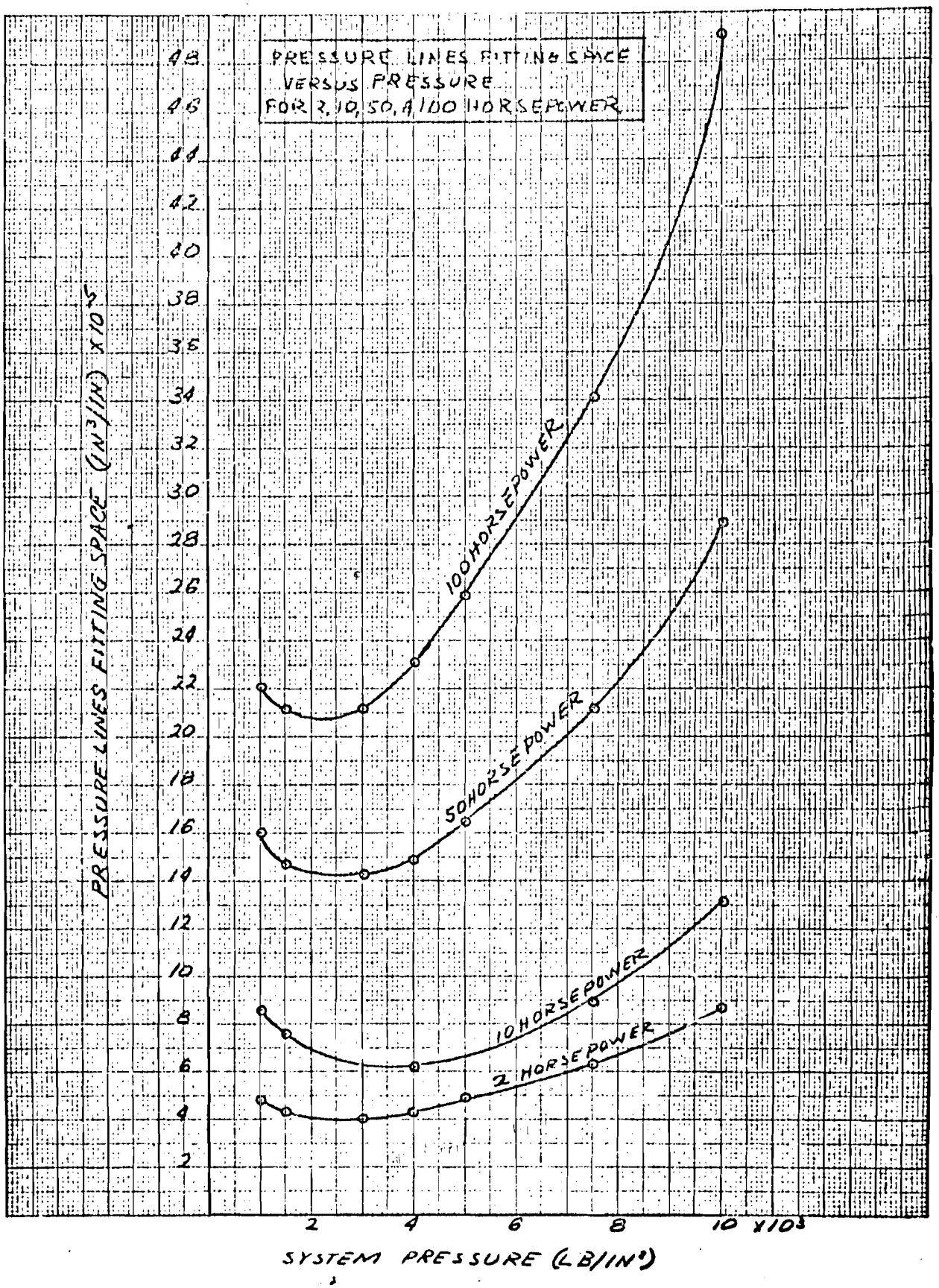

PRESSURE LINES TUBING SPACE
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, 100 HORSEPOWER
100 HORSE POWER
50 HORSE POWER
10 HORSE POWER
2 HORSE POWER
PRESSURE (LB/IN²)
PRESSURE LINES TUBING SPACE
.50
.40
.30
.20
.10
.05
.04
.03
.02
2
4
6
8
10 x 10³

FIG. A-23A
A-46

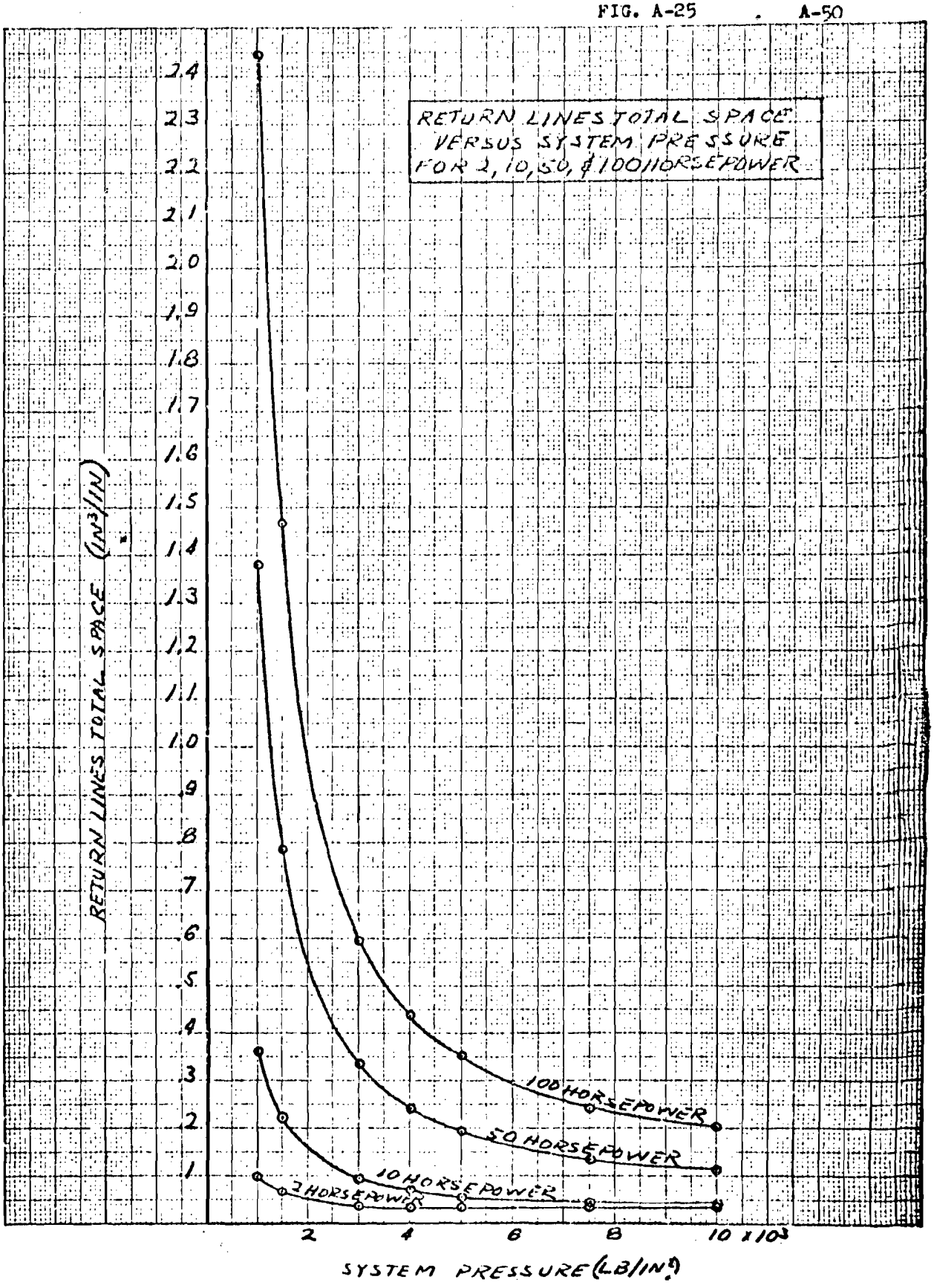

PRESSURE LINES BEND SPACE
VERSUS SYSTEM PRESSURE
FOR 2 & 10 HORSE POWER
PRESSURE LINES BEND SPACE (IN³/IN)
.13
.12
.11
.10
.09
.08
.07
.06
.05
.04
.03
.02
.01
10 HORSE POWER
2 HORSE POWER
2
4
6
8
10 X10³
PRESSURE (LB/IN²)

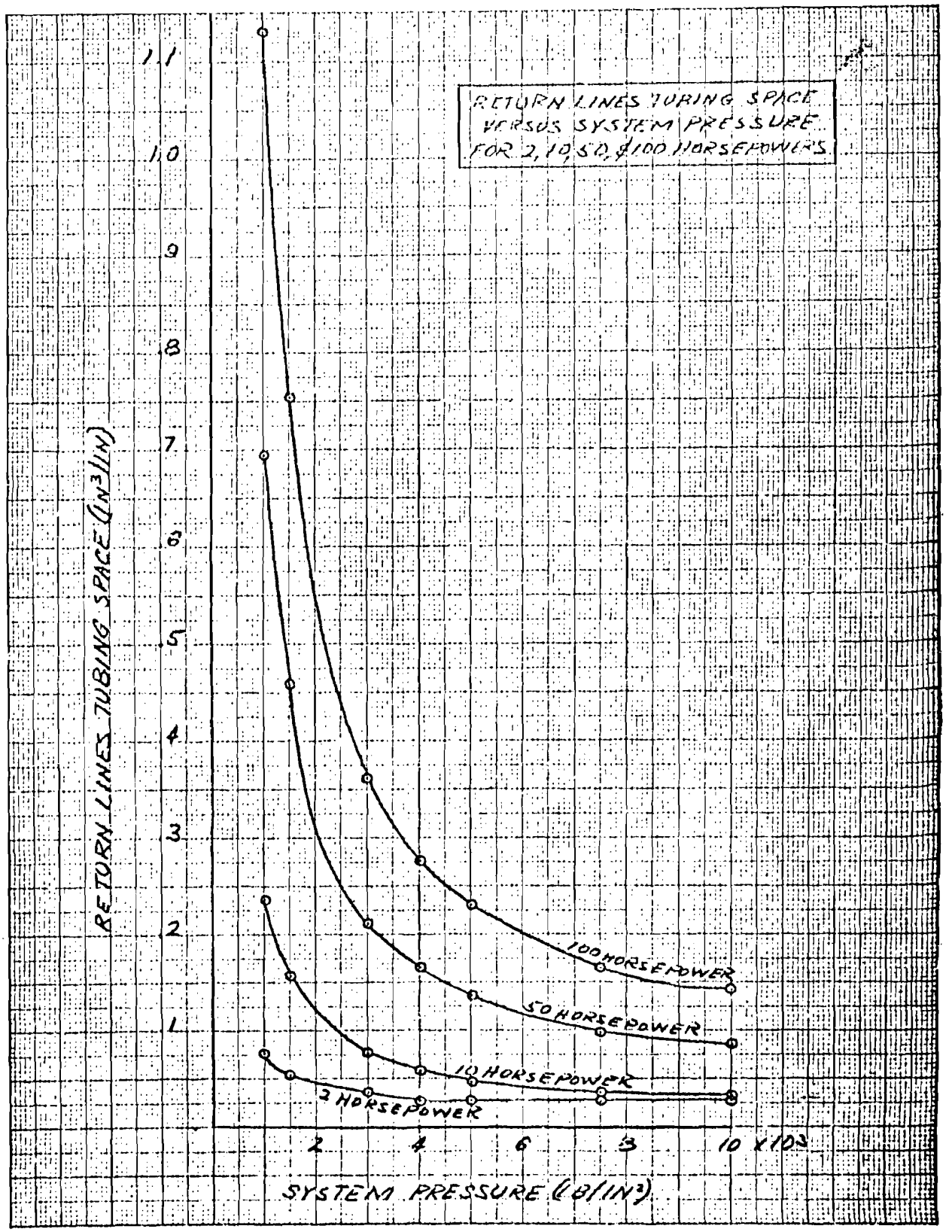

PRESSURE LINES BEND SPACE
VERSUS SYSTEM PRESSURE
FOR 50 & 100 HORSEPOWER
PRESSURE LINES BEND SPACE (IN³/IN)
1.4
1.3
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
100 HORSEPOWER
50 HORSEPOWER
2
4
6
8
10 X10³
SYSTEM PRESSURE (LB/IN²)

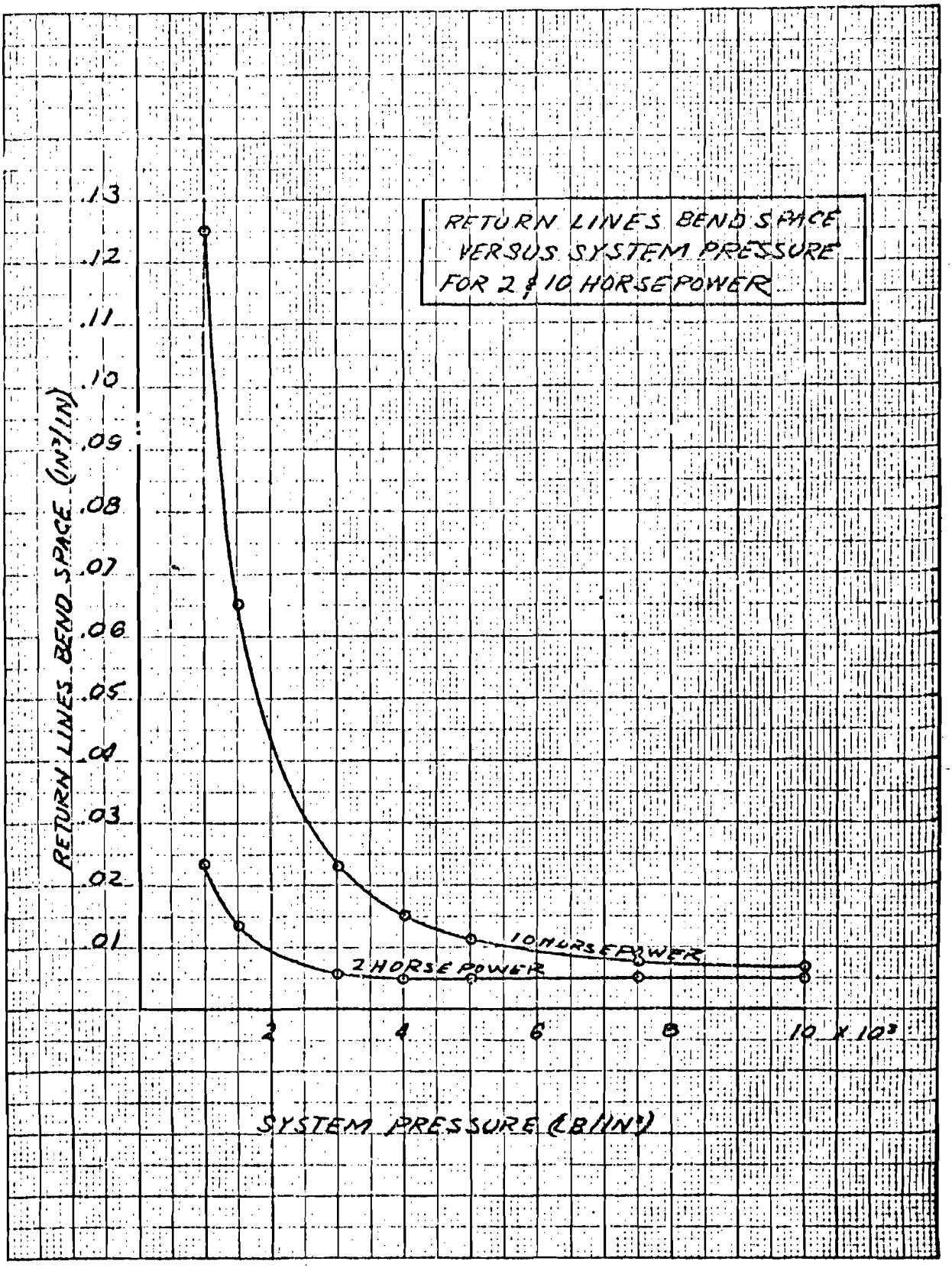

PRESSURE LINES FITTING SPACE
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
PRESSURE LINES FITTING SPACE (IN³/IN) X10⁻⁵
48
46
44
42
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X10³
SYSTEM PRESSURE (LB/IN²)

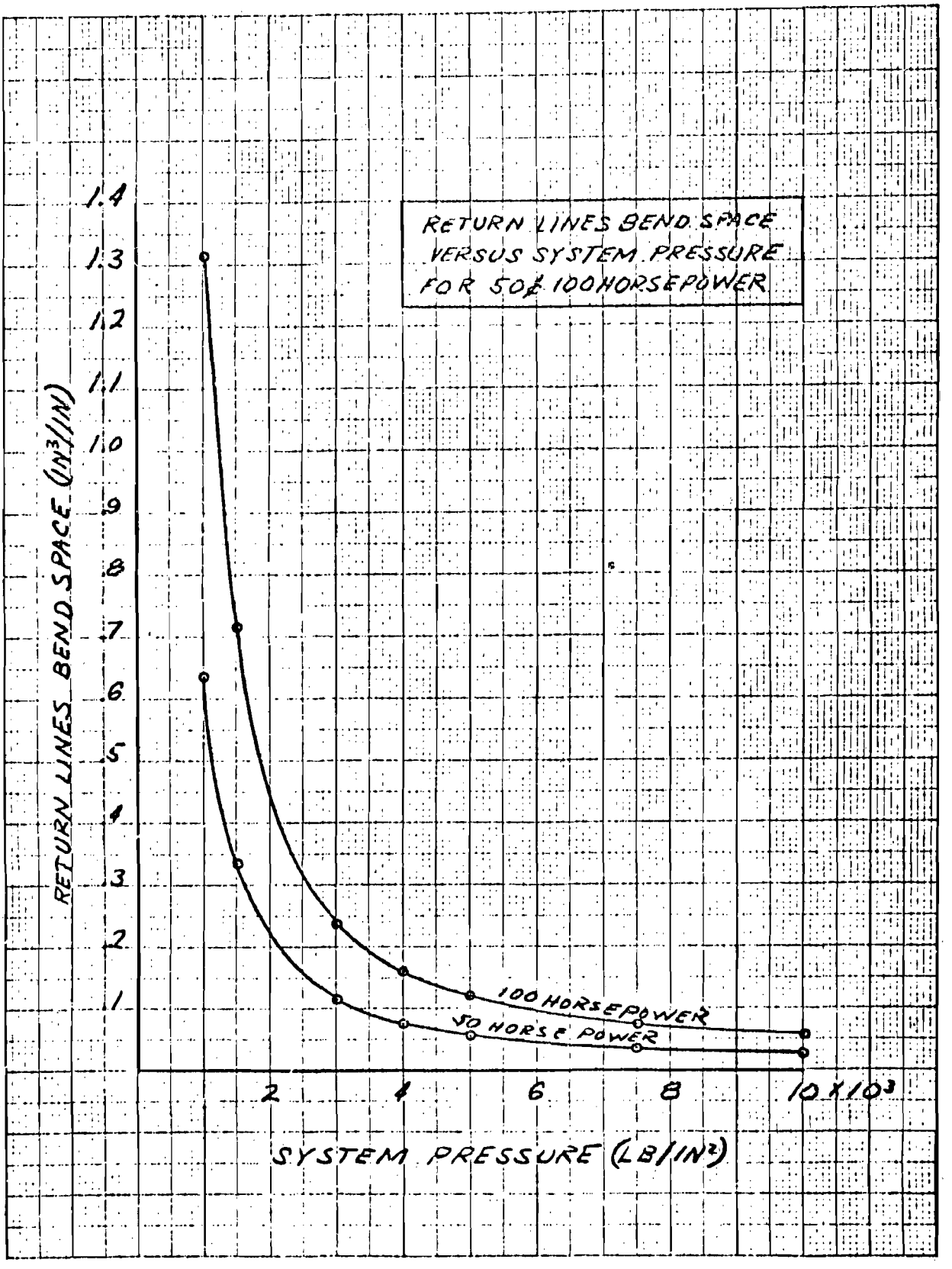
TOTAL SPACE OCCUPIED BY PRESSURE LINES

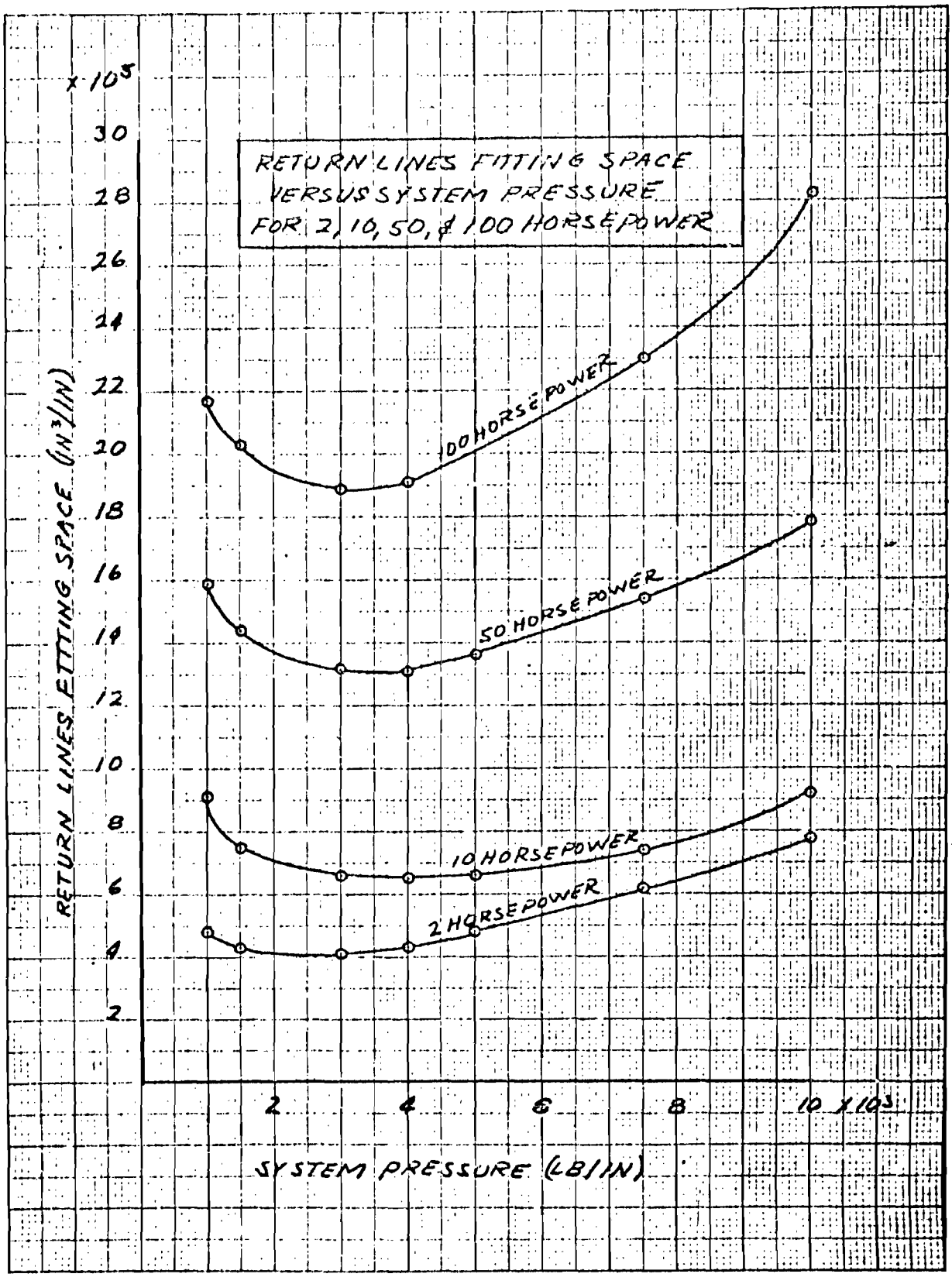
System Pressure	HP	OD (IN)	Tubing (IN^3)	Bends (IN^3)	Fittings (IN^3)	Total (IN^3)
1000	2	.313	.0767	.033	.000048	.100048
	10	.548	.238	.125	.000086	.363086
	50	.946	.702	.644	.00016	1.34616
	100	1.119	1.17	1.38	.00022	2.5522
1500	2	.261	.0536	.0135	.00043	.067143
	10	.447	.157	.0679	.000076	.224976
	50	.773	.469	.3515	.000147	.820647
	100	1.01	.800	.782	.000212	1.582212
3000	2	.198	.0308	.0059	.000040	.036740
	10	.313	.077	.0233	.000067	.100367
	50	.566	.253	.1378	.000143	.390943
	100	.728	.416	.293	.000212	.709212
4000	2	.1875	.027	.0059	.00043	.032943
	10	.281	.062	.01685	.000062	.078912
	50	.506	.201	.0954	.000149	.296549
	100	.656	.338	.2145	.000231	.552731
5000	2	.1875	.0276	.0059	.000049	.033549
	10	.263	.0543	.01382	.000073	.068193
	50	.474	.1765	.0810	.000165	.257665
	100	.615	.297	.1768	.000259	.474059
7500	2	.1875	.276	.0059	.000063	.033563
	10	.251	.0495	.01202	.00090	.061610
	50	.435	.1485	.0625	.000212	.211212
	100	.558	.2145	.132	.000341	.376841
10000	2	.1875	.0276	.0059	.000087	.033587
	10	.275	.0594	.01573	.000132	.075262
	50	.432	.1475	.0613	.000289	.208089
	100	.557	.2440	.1314	.000491	.375891


RETURN LINES TOTAL SPACE VERSUS SYSTEM PRESSURE FOR 2, 10, 50, & 100 HORSEPOWER
RETURN LINES TOTAL SPACE (IN³/IN)
2.4
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x 10³
SYSTEM PRESSURE (LB/IN²)


RETURN LINES TUBING SPACE VERSUS SYSTEM PRESSURE FOR 2, 10, 50, & 100 HORSEPOWERS
RETURN LINES TUBING SPACE (IN³/IN)
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X10³
SYSTEM PRESSURE (LB/IN²)


RETURN LINES BEND SPACE
VERSUS SYSTEM PRESSURE
FOR 2 & 10 HORSEPOWER
RETURN LINES BEND SPACE (IN³/IN)
.13
.12
.11
.10
.09
.08
.07
.06
.05
.04
.03
.02
.01
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X 10³
SYSTEM PRESSURE (LB/IN²)


RETURN LINES BEND SPACE
VERSUS SYSTEM PRESSURE
FOR 50 & 100 HORSEPOWER
RETURN LINES BEND SPACE (IN³/IN)
1.4
1.3
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
100 HORSEPOWER
50 HORSE POWER
2
4
6
8
10 X 10³
SYSTEM PRESSURE (LB/IN²)


RETURN LINES FITTING SPACE
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
RETURN LINES FITTING SPACE (IN³/IN)
x 10⁵
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
2
4
6
8
10 x 10³
SYSTEM PRESSURE (LB/IN)

TOTAL SPACE OCCUPIED BY RETURN LINES

System Pressure	Back Press	HP	OD (IN)	Tubing (IN^3)	Bends (IN^3)	Fittings (IN^3)	Total (IN^3)
1000	500	2	.313	.0767	.02325	.000048	.099998
		10	.584	.236	.125	.000091	.361091
		50	.941	.695	.633	.000159	1.328159
		100	1.20	1.13	1.313	.000217	2.443217
1500	750	2	.261	.0535	.0135	.000043	.067043
		10	.441	.1568	.0651	.000075	.221975
		50	.761	.458	.335	.000144	.789144
		100	.980	.754	.715	.000203	1.469203
3000	1500	2	.198	.0377	.0059	.000041	.036711
		10	.312	.0764	.02303	.000066	.099496
		50	.532	.222	.114	.000131	.336131
		100	.658	.361	.237	.000189	.598189
4000	2000	2	.1875	.0276	.005	.000043	.032643
		10	.272	.0581	.0153	.000065	.073465
		50	.460	.1660	.074	.000131	.240131
		100	.596	.2785	.161	.000191	.439691
5000	2500	2	.1875	.0276	.005	.000048	.032648
		10	.246	.0457	.0113	.000066	.058866
		50	.420	.1365	.0563	.0000136	.192936
		100	.542	.2305	.121	.000213	.351713
7500	3750	2	.1875	.0276	.005	.000062	.032662
		10	.216	.03665	.00766	.000074	.044384
		50	.358	.0983	.03485	.000154	.133304
		100	.460	.166	.074	.00023	.24023
10000	5000	2	.1875	.0276	.005	.000077	.032677
		10	.210	.0346	.00696	.000092	.041652
		50	.331	.086	.0276	.000178	.113778
		100	.426	.1425	.0586	.000282	.201382

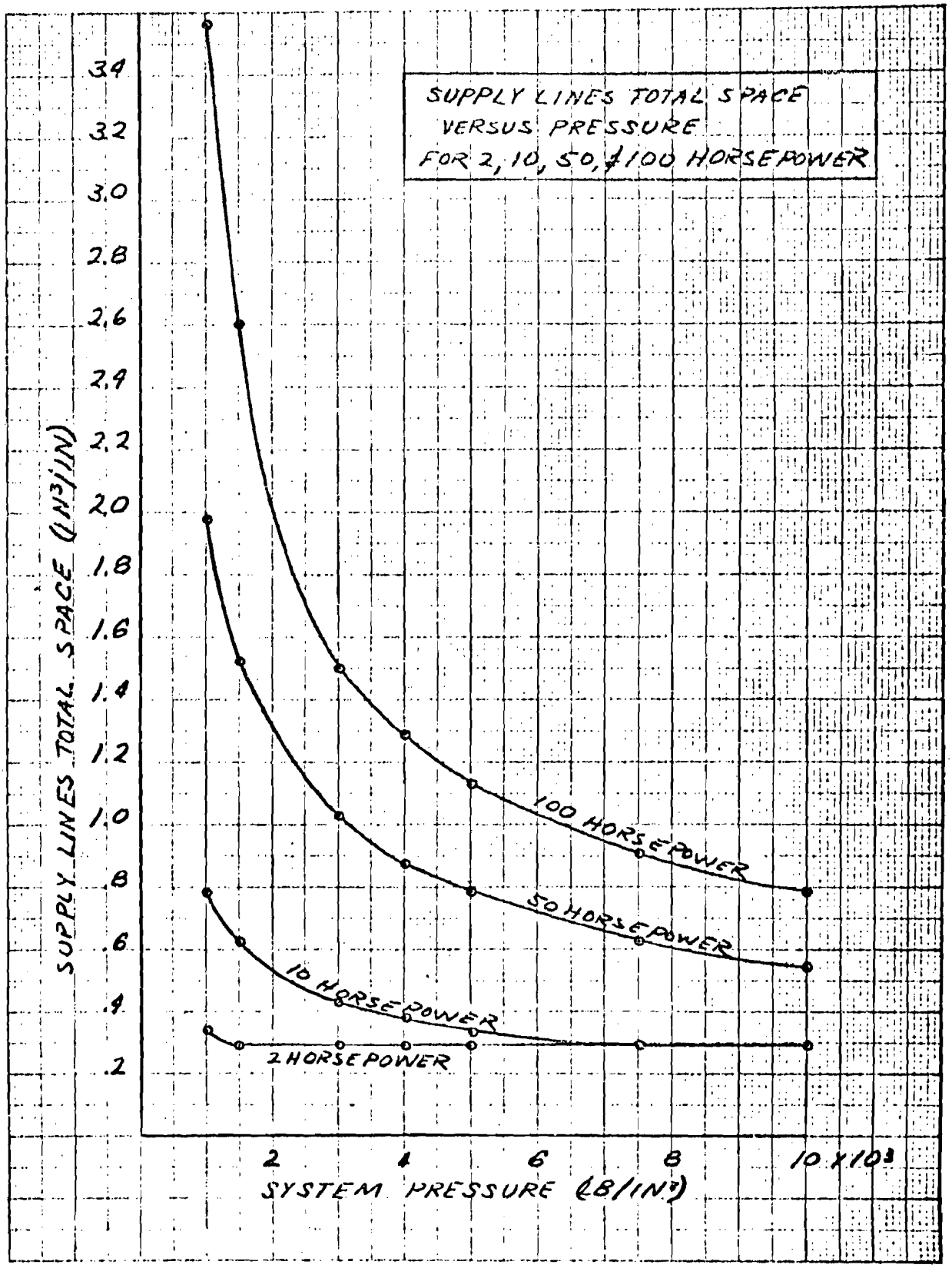
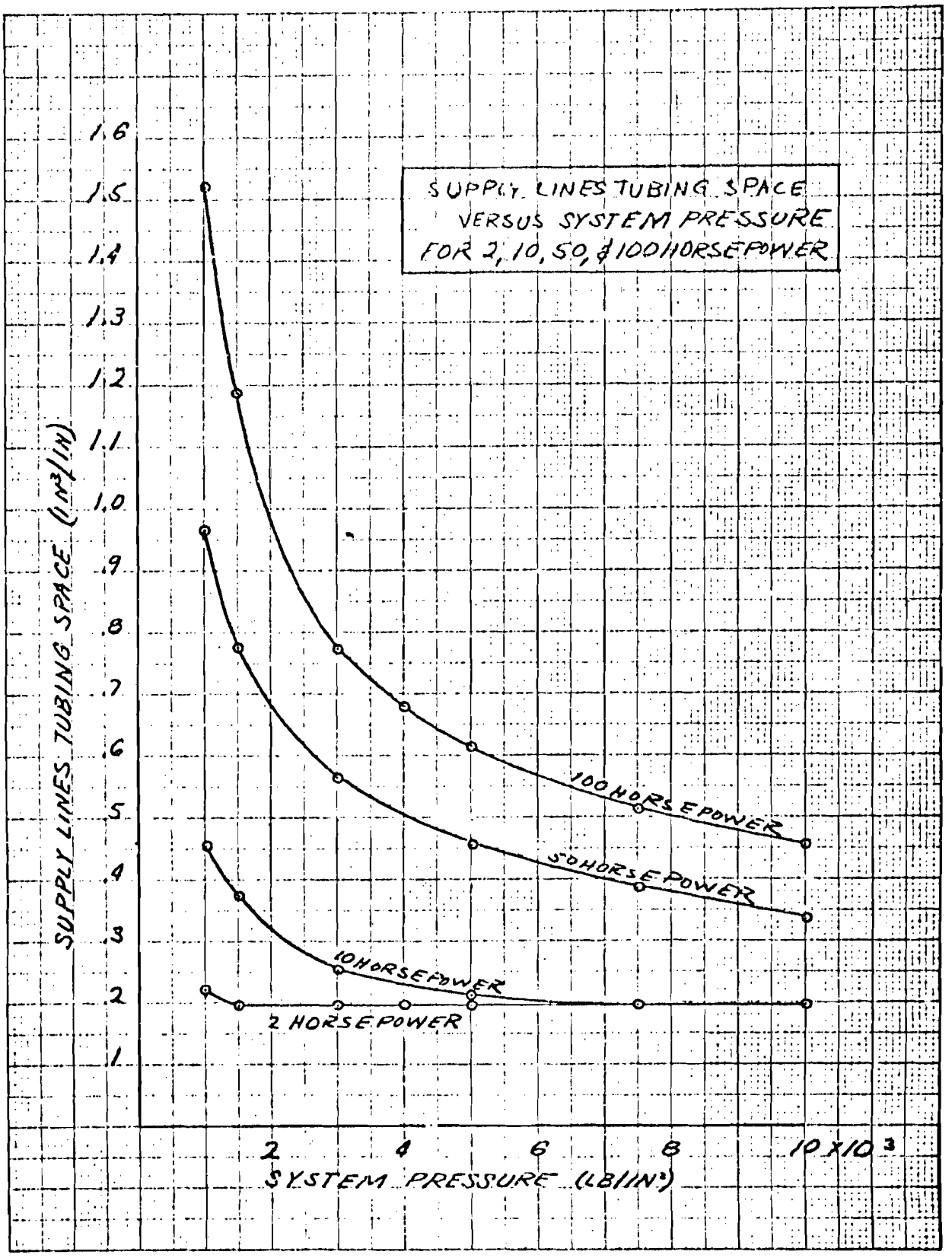
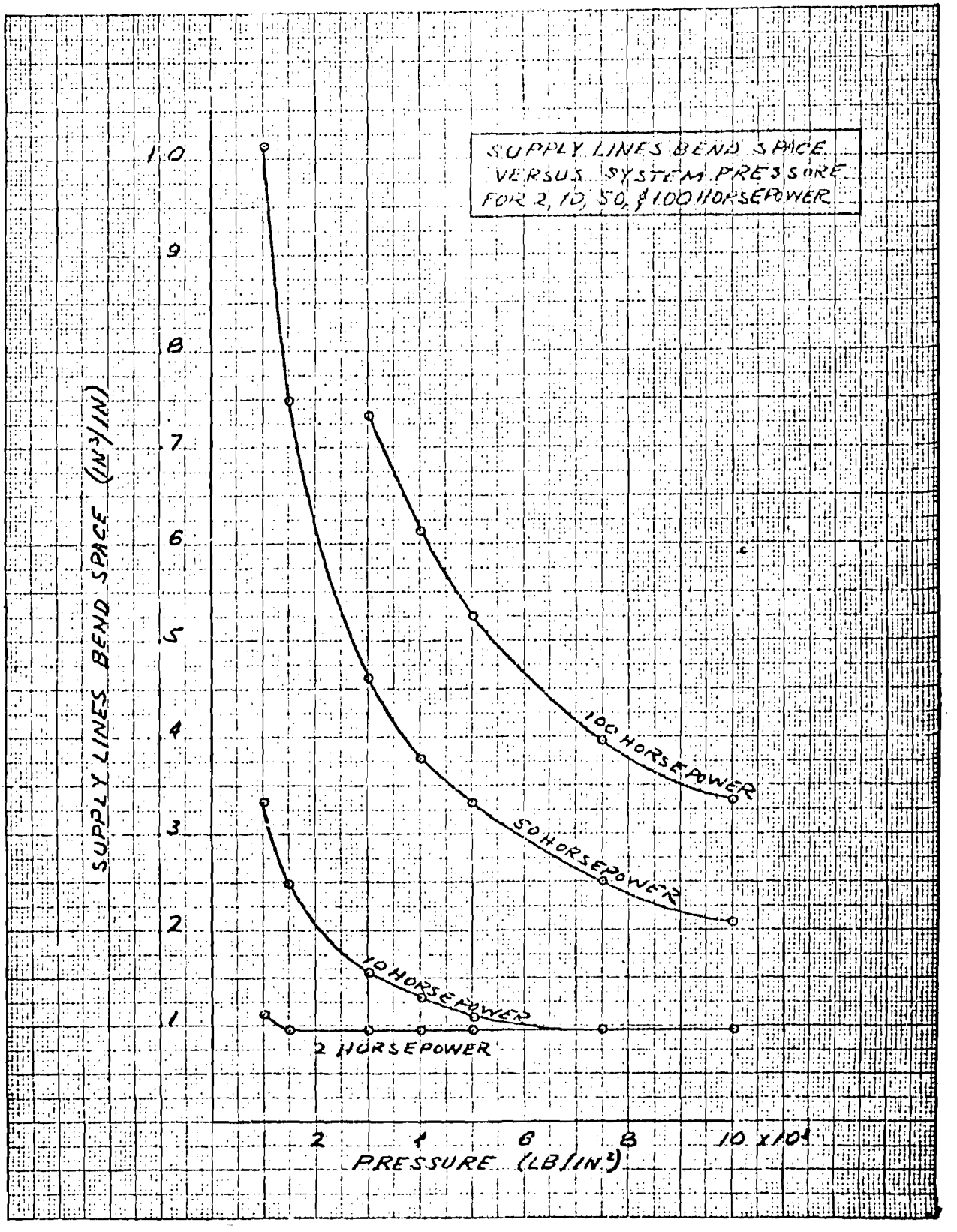

SUPPLY LINES TOTAL SPACE
VERSUS PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES TOTAL SPACE (IN³/IN)
3.4
3.2
3.0
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X10³
SYSTEM PRESSURE (LB/IN²)

FIG. A-30
A-57


SUPPLY LINES TUBING SPACE
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES TUBING SPACE (IN³/IN)
1.6
1.5
1.4
1.3
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 X 10³
SYSTEM PRESSURE (LB/IN²)


SUPPLY LINES BEND SPACE
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES BEND SPACE (IN³/IN)
PRESSURE (LB/IN²)
10 x 10⁴
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER

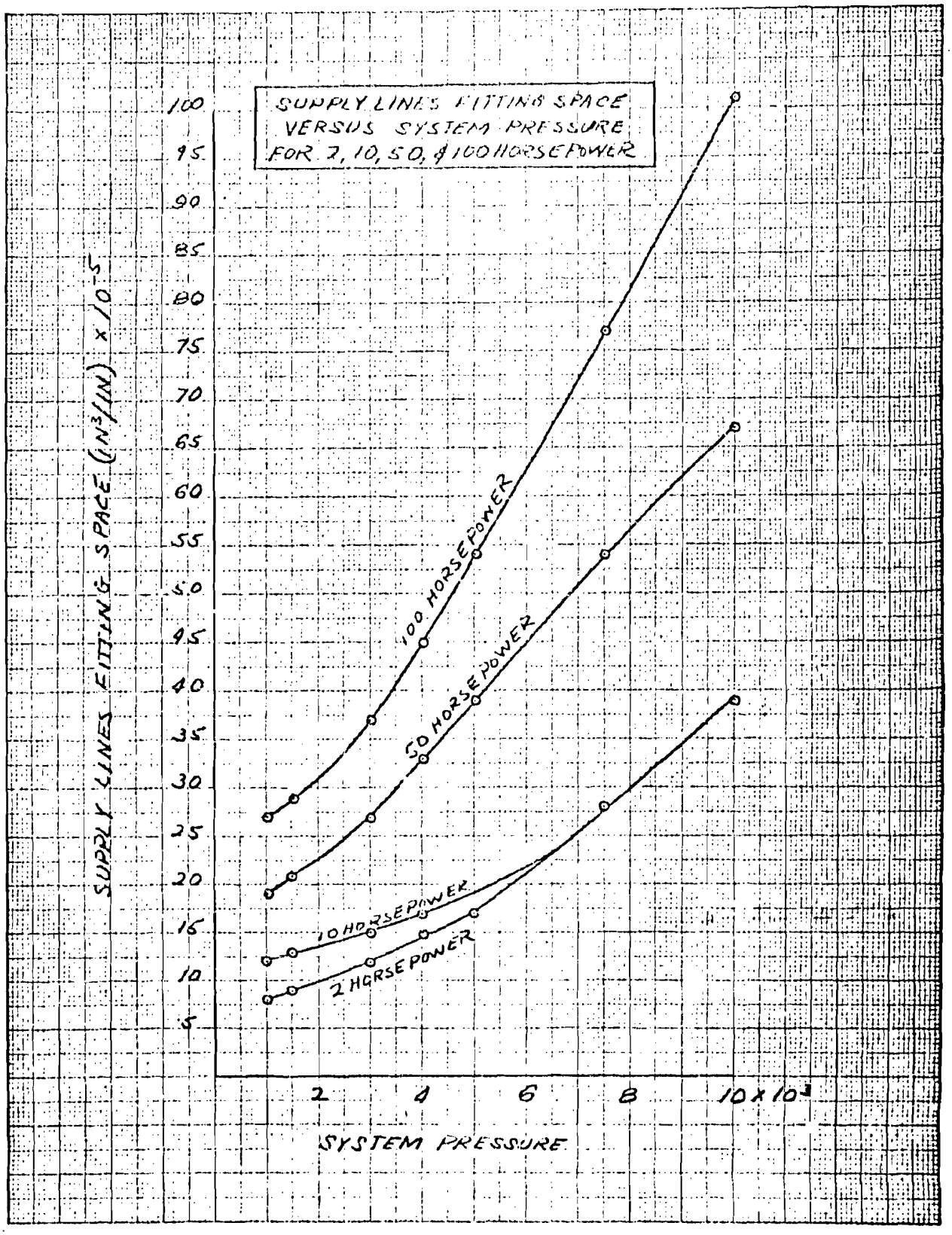
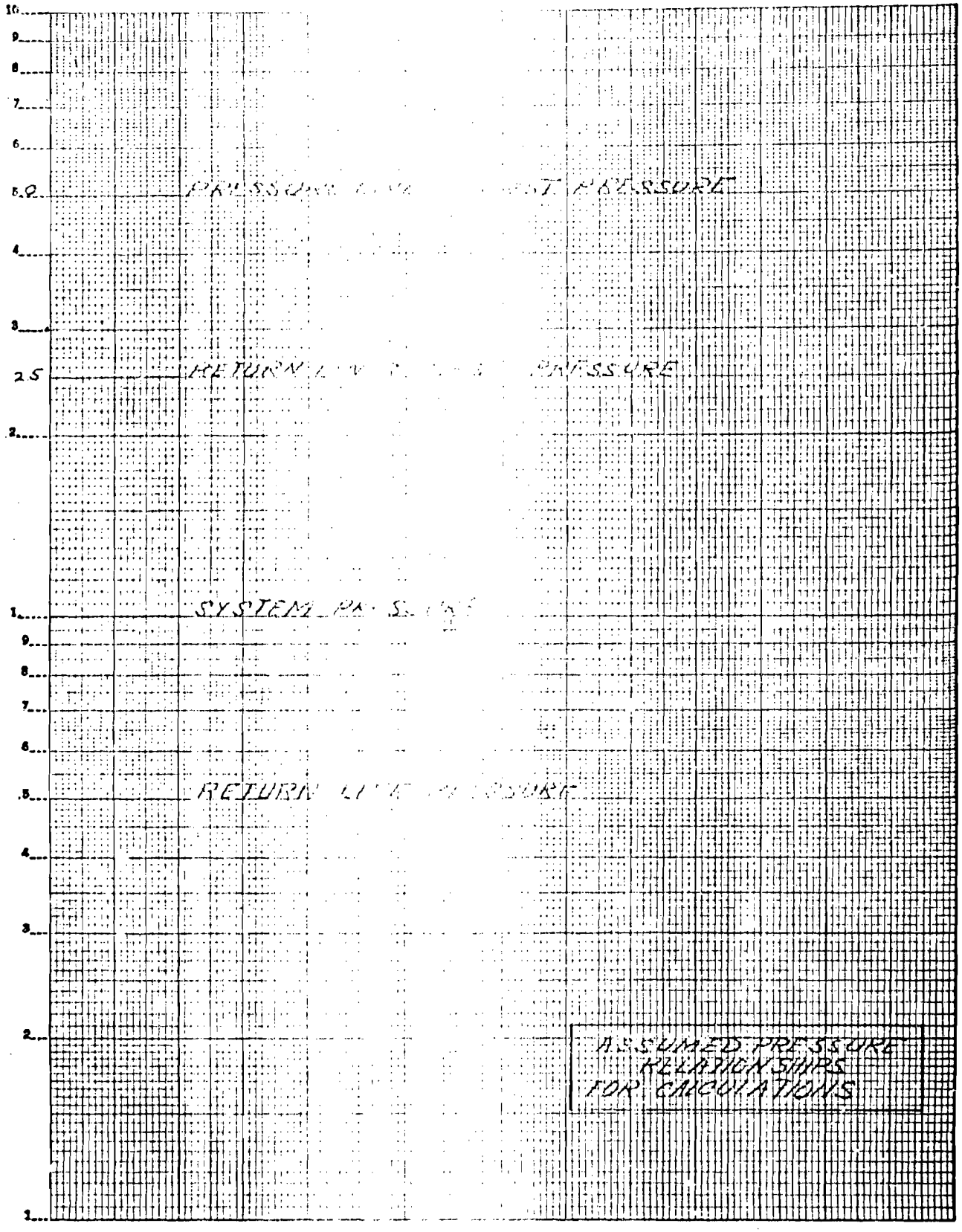

SUPPLY LINES FITTING SPACE
VERSUS SYSTEM PRESSURE
FOR 2, 10, 50, & 100 HORSEPOWER
SUPPLY LINES FITTING SPACE (IN³/IN) x 10⁻⁵
100
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
2
4
6
8
10 x 10³
SYSTEM PRESSURE

FIG. A-33
A-60



TOTAL SPACE OCCUPIED BY SUPPLY LINES

System Pressure	HP	OD	Tubing (IN3)	Bends (IN3)	Fittings (IN3)	Total (IN3)
1000	2	.530	.221	.1129	.00008	.33398
	10	.760	.454	.3337	.00012	.78782
	50	1.110	.967	1.013	.00019	1.98019
	100	1.420	1.52	1.04	.00027	3.56027
1500	2	.500	.1962	.095	.00009	.29129
	10	.690	.374	.2495	.00013	.62363
	50	.995	.7775	.749	.00021	1.52671
	100	1.230	1.188	1.413	.00029	2.60129
3000	2	.500	.1962	.095	.00012	.29132
	10	.555	.252	.456	.00015	.42964
	50	.793	.494	.462	.00027	1.02622
	100	.931	.680	.732	.00037	1.50437
4000	2	.500	.1962	.095	.00015	.29135
	10	.530	.252	.130	.00017	.38217
	50	.760	.494	.379	.00033	.87333
	100	.884	.680	.613	.00045	1.29345
5000	2	.500	.1962	.095	.00017	.29137
	10	.530	.2205	.113	.00020	.3337
	50	.760	.454	.3335	.00039	.78789
	100	.884	.614	.525	.00054	1.13954
7500	2	.500	.1962	.095	.00028	.29148
	10	.500	.1962	.095	.00028	.29148
	50	.690	.374	.250	.00054	.62454
	100	.807	.511	.396	.00077	.90777
10000	2	.500	.1962	.095	.00039	.29159
	10	.500	.1962	.095	.00039	.29159
	50	.650	.332	.209	.00067	.54167
	100	.760	.453	.3335	.00101	.78751

APPENDIX B

THE WEIGHT AND SPACE OF

HYDRAULIC CYLINDERS

AT

VARIOUS PRESSURES

HYDRAULIC CYLINDER WEIGHT STUDY

The following symbols will be used throughout this section.

W_T = Total Hydraulic Cylinder Weight (Lb)

W_B = Unibal Bearing Weight (Lb)

W_w = Web Weight (Lb)

W_D = Disc Weight (Lb)

W_C = Cap Weight (Lb)

W = Barrel Weight (Lb)

W_P = Piston Weight (Lb)

W_R = Rod Weight (Lb)

W_G = Gland Weight (Lb)

W_E = Rod End Weight (Lb)

W_F = Fluid Weight (Lb)

d = Barrel Inside Diameter (In.) (Bore Diameter)

D = Barrel Outside Diameter (In)

P = System Working Pressure P.S.I.

P' = Proof Pressure (1.5 P) (P.S.I.)

P'' = Burst Pressure (2.5 P) (P.S.I.)

S = Stroke (In.)

S_t = Longitudinal Tensile Stress P.S.I.

S_{tt} = Transverse Tensile Stress P.S.I.

S_s = Shear Stress (P.S.I.)

ρ = Material Specific Weight (Lb/In^3)

G = Safety Factor (Number)

D_x = Outside Diameter (In.) Subscript Designates Part, Which Will Agree With Weight Subscript

d_x = Inside Diameter (In.) Subscript Designates Part, Which Will Agree With Weight Subscript

F = Load Through Axis of Cylinder (Lb.)

A = Cross Section Area ($In.^2$)

L_T = Thread Length (In.)

L = Length (In.)

t = Thickness (In.)

D_c = Cylinder Outside Diameter (Space) (In)

D_R = Rod Outside Diameter (Space) (In.)

θ = Quantity Added to Stroke to Yield Cylinder Length (Space) (In)

δ = Quantity Added to Stroke To Yield Rod Length (Space) (In)

P_{CR} = Critical Column Load (Lb.)

a = Cylinder Length (In) (Column)

b = Rod Extension Length at Half Stroke (In.)

L = a + b

E_1, E_2 = Young's Modulus For Cylinder and Rod Material, Respectively PSI

I_1, I_2 = Moment of Inertia Cylinder and Rod, Respectively (In^4)

j_1, j_2 = Stiffness Factors (In.)

$\frac{a}{j_1}$, $\frac{b}{j_2}$ = Radians

It was the object here to obtain the weight equation of each cylinder part in terms of bore diameter, system pressure, and stroke.

For purposes of calculation a cylinder of the type illustrated in figure B-1 was assumed. This choice is based on statistical evidence that this type predominates all other types in current use in the aircraft industry. (See figure B-2) The cylinder was subdivided into regular geometric solids and named for identification.

First is the unibal bearing at head end. A plot, figure B-3 was made of bearing weight versus load and because the points approximated a straight line, the following relation is true:

$$W_B = 1.1 \times 10^{-5} F$$

But:

$$F = \frac{\pi}{4} d^2 P'$$

So:

$$W_B = 1.297\ (10^{-5}) d^2 P \qquad \text{(I)}$$

Next is the web. From data it is logical to assume the web weight to be twice the unibal bearing weight, whereupon there results:

$$W_W = 2.594\ (10^{-5}) d^2 P \qquad \text{(II)}$$

Next, the disc is considered. Basically its weight can be said to be:

$$W_D = \frac{\pi}{4} D^2 t \rho$$

From Lame's formula (Mark's Mechanical Engineering Handbook Ref. 3) it is evident:

$$D^2 = d^2 \left(\frac{St + P''}{St - P''} \right) \qquad \text{(III)}$$

From the equation for stress in a circular plate under uniformly distributed load (Mark's Mechanical Engineering Handbook Ref. 3)

$$t = \sqrt{\frac{0.6\ d^2 P''}{4\ S_t}} \qquad \text{(IIIa)}$$

Also:

$$\rho = 0.1\ \text{Lb/In}^3$$

$$St = 50{,}000\ \text{Lb/In}^2$$

Finally:

$$W_D = .000215\ d^3 P^{1/2} \left(\frac{20,000 + P}{20,000 - P}\right) \qquad \text{(IV)}$$

The basic cap weight equation is:

$$W_C = AL = KAL_T \qquad \text{(V)}$$

For area the following method was used:

Tensile Load = Tensil Area x Tensile Stress

$$\frac{\pi}{4} d^2 P^* = A \times S_t$$

$$A = \frac{\pi\ d^2 P^*}{4\ S_t}$$

And for threa length:

Shear Load = Shear Area x Shear Stress

$$\frac{\pi}{4} d^2 P^* = \pi D L_T \times S_s$$

$$L_T = \frac{d^2 P^*}{4\ S_s D} = \frac{d\ P^*}{4\ S_s} \left(\frac{S_{tt} - P^*}{S_{tt} + P^*}\right)^{1/2} \qquad \text{(VI)}$$

K is assumed to be proportional to pressure and thread length.

So at 3000 psi:

$$L = KL_T = L_T + C \qquad \text{(VII)}$$

From The Glenn L. Martin Company Hydraulic Design Manual it is seen that:

$C = 1.074$ If: $d = 1.5$ In
$P = 3000$ psi
$P^* = 7500$ psi

Also:

$S_s = 37,000$ psi
$S_{tt} = 50,000$ psi

Substituting in equation VI:

$$L_T = \frac{1.5\ (7500)}{4\ (37,000)} \sqrt{\frac{50,000 - 7500}{50,000 + 7500}} = .0635 \text{ In.}$$

From Equation VII:

$$K = \frac{L_T + C}{L_T} = \frac{.0653 + 1.074}{.0653} = 17.45$$

Finally:

$$W_C = \pi \frac{d^2 P^*}{4 S_t} \cdot \frac{dP^*}{4 S_s} \left(\frac{S_{tt} - P^*}{S_{tt} + P^*}\right)^{1/2} \cdot 17.45$$

Or:

$$W_C = 1.157 \, (10^{-8}) \, d^3 P^2 \left(\frac{20{,}000 - P}{20{,}000 + P}\right)^{1/2} \qquad \text{(VIII)}$$

Piston weight was assumed to be:

$$W_P = \frac{\pi}{4} d^2 t \rho$$

To obtain thickness of the piston, a plot was made which is a plot of values from The Glenn L. Martin Company Hydraulic Design Manual, Figure B-4, and from it it was observed that:

$$t = .154d + .569 \qquad \text{(IX)}$$

closely approximated actual conditions.

For steel:

$$\rho = .283 \text{ Lb/In}^3$$

Consequently:

$$W_P = \frac{\pi}{4} d^2 (.154d + .569)(.283)$$

Or:

$$W_P = .0801 \, d^3 + .1266 \, d^2 \qquad \text{(X)}$$

The barrel weight was considered to be:

$$W = \frac{\pi}{4} (D^2 - d^2) L \rho \qquad \text{(XI)}$$

But:

$$D^2 = d^2 \left(\frac{S_{tt} + P^*}{S_{tt} - P^*}\right) \qquad \text{(XII)}$$

And:

$$L = S + \text{Piston Thickness (Equation X)}$$

$$L = S + .54d + .569 \qquad \text{(XIII)}$$

$$= 0.1 \text{ LB/IN}^3 \text{ (Aluminum)} \qquad \text{(XIV)}$$

Combining Equations XII, XIII, and XIV, Equation XI becomes:

$$W = \left[.0121\, d^3 + (.07854 + .0447)\, d^2\right]\left(\frac{P}{10{,}000} - .5P\right) \qquad \text{(XV)}$$

In order to calculate the rod weight it was necessary to establish criteria for rod physical dimensions to obtain comparative values up to 10,000 P.S.I. system pressure.

In order to determine the proportions of diameter and wall thickness of the piston rod, reference is made to figure B-2 which indicates an area ratio of .75. (This means a rod diameter equal to one half the bore diameter). The maximum column strength of a cylinder with such proportions could be obtained by using a solid steel rod, and a plot was made of this case for 3000 P.S.I. However, much greater weight efficiency is achieved without appreciable loss of column strength by use of a hollow steel rod having a wall thickness approximately one eighth of the rod diameter in larger sizes with a minimum wall thickness of .094 inch in smaller sizes as a manufacturing practice limit. A curve representing the column limit of .094 wall thickness rod has been plotted on figure B-5 up to the point where it also equals one eighth the rod diameter, and beyond that point the curve for column limit was plotted with a rod wall equal to one eighth of the rod diameter. It should be observed that the hollow rod column limit is very close to the solid rod column limit at the same pressure. From figure B-6, it should be noted that the majority of cylinders in use today at 3000 P.S.I. fall very close to the column limit. This means that to be fair in the analysis, design of cylinders to a higher pressure necessitates, in many instances, using a shorter stroke and a correspondingly greater load in order to produce the same quantity of work. This same optimum rod design of .094 minimum wall and an increase in thickness equal to one eighth the rod diameter has been used at all pressures.

The graph figure B-5 presented an excellent correlation between bore, stroke, and pressure for each pressure range considered. These values were established and inserted in the weight and space equations.

The critical column equations were calculated by the following equations: *

* "Critical Buckling Loads for Hydraulic Cylinders" by Fred Hoblit, July 1950 issue of Product Engineering magazine

$$P_{CR} = \frac{\left(\frac{3L}{ab}\right) K_1 \; K_2}{K_1 \beta 2 + K_2 \beta 1 + \left(\frac{K_1 \; K_2}{K_C}\right)}$$

$$K_1 = \frac{E_1 \; I_1}{a} \qquad K_2 = \frac{E_2 \; I_2}{b}$$

$$K_C = \frac{E_2 \; I_2}{d}$$

$$\beta 1 = \frac{3 \left[1 - \left(\frac{a}{j1}\right) \mathrm{COT} \left(\frac{a}{j1}\right)\right]}{\left(\frac{a}{j1}\right)^2}$$

$$\beta 2 = \frac{3 \left[1 - \left(\frac{b}{j2}\right) \mathrm{COT} \left(\frac{b}{j2}\right)\right]}{\left(\frac{b}{j2}\right)^2}$$

$$j1 = \frac{E_1 \; I_1}{\mathrm{LOAD}} \qquad j2 = \frac{E_2 \; I_2}{\mathrm{LOAD}}$$

These equations take into account initial offset of the rod as it emerges from the cylinders. This is sometimes referred to as "spring action".

It is evident that two rod equations are required, one for rod wall thickness of one eighth the rod diameter, and another for a constant wall thickness of .094 inch. Further, the former equation then applies if the bore diameter is over an inch and one half, and the latter below this value.

The basic equation for all cases is:

$$W_R = \frac{\pi}{4} \left(D_R^2 - d_R^2\right) L_R \, \rho \qquad \text{(XVI)}$$

Assuming rod wall = 1/8 rod diameter

Or:

$$t = \frac{D_R - d_R}{2} = \frac{D_R}{8}$$

From which:

$$8 \left(D_R - d_R\right) = 2 \, D_R$$

And:

$$4 \, d_R = 3 \, D_R$$

So:

$$d_R = 3/4\ D_R$$

And:

$$d_R^2 = 9/16\ D_R^2 \qquad \text{(XVII)}$$

The rod length is:

$$L_R = S + \text{Gland Length}$$

Gland length is shown in figure B-7 as being $.929\ D_R + .9142$ whereupon:

$$L = S + .929\ D_R + .9142$$

And if length equal to one quarter inch is added for this nut:

$$L = S + .929\ D_R + 1.1642 \qquad \text{(XVIII)}$$

$$\rho = .283\ LB/IN^3 \quad \text{(Steel)} \qquad \text{(XIX)}$$

$$D_R = \frac{d}{2} \qquad \text{(XX)}$$

Combining equations XVII, XVIII, XIX, and XX, Equation XVI becomes:

$$W_R\ (.0244 + .0283)\ d^2 + .0113\ d^3$$

When rod wall = 1/8 (rod diameter)

<u>Assuming rod wall = .094</u>

Or:

$$D_R - d_R = 2\ (.094)$$

Then:

$$d_R = D_R - .188$$

And:

$$d_R^2 = (D_R - .188)^2 \qquad \text{(XXI)}$$

Combining equations XVIII, XIX, XX, and XXI, Equation XVI becomes:

$$W_R = (.0417d - .00784)\ S + .01938d^2 - .003644d + .03946 \qquad \text{(XXII)}$$

<u>When rod wall = .094</u>

Gland weight is expressed as:

$$W_G = \frac{\pi}{4}\ (D_G^2 - d_G^2)\ L\rho \qquad \text{(XXIII)}$$

Referring to figure B-7, it is seen that:

$$D_G = 1.25\ D_R + .700 \qquad \text{(XXIV)}$$

And:

$$L = .929\ D_R + .9142 \qquad \text{(XXV)}$$

Also, it is obvious that:

$$d_G = D_R \qquad \text{(XXVI)}$$

$$\rho = 0.1\ \text{LB/IN}^3\ \text{(Aluminum)} \qquad \text{(XXVII)}$$

Combining equations XXIV, XXV, XXVI, and XXVII, Equation XXIII becomes:

$$W_G = .00265\ d^3 + .03711\ d^2 + .0806\ d + .0352 \qquad \text{(XXVII)}$$

Rod end weight is plotted as a function of the load in figure B-8. Thus the relation below occurs:

$$W_E = 1.293\ (10^{-9})\ (\text{LOAD})^2$$

But load is:

$$\text{LOAD} = \frac{\pi}{4}\ d^2 P$$

Accordingly:

$$W_E = 1.795\ (10^{-9})\ d^4\ P^2 \qquad \text{(XXIII)}$$

Fluid weight equals:

$$W_F = \frac{\pi}{4}\ d^2$$

Or:

$$W_F = .0238\ d^2 \alpha\ S \qquad \text{(XXIV)}$$

And α is a pressure coefficient, see figure A-9.

All the foregoing equations were checked with data, and it was found that some coefficients had to be amended slightly. The correct equations are listed below.

Final weight equations for hydraulic cylinder parts:

Unibal bearing:

$$W_B = 1.297\ (10^{-5})\ d^2 P$$

Web:

$$W_W = 1.343\ (10^{-5})\ d^2 P$$

Disc:

$$W_D = .000462\ d^3 P^{\frac{1}{2}} \left(\frac{50,000 + 2.5P}{50,000 - 2.5P}\right)$$

Cap:

$$W_C = 1.157\ (10^{-8})\ P^2 d^2 \sqrt{\frac{20,000 - 2.5P}{20,000 + 2.5P}}$$

Piston:

$$W_P = .0248\ d^3 + .0392\ d^2$$

Rod:

1. Rod wall $= \frac{\text{Rod O.D.}}{8}$

$$W_R = (.02625\ S + .03045)d^2 + .01217\ d^3$$

2. Rod wall $= .094$

$$W_R = (.0522d - .00985)\ S + .0243d^2 - .00457d = .0495$$

Barrel:

$$W = \left[.0121d^3 + (.07854\ S + .0447)d^2\right]\left(\frac{5P}{50,000 - 2.5P}\right)$$

Gland:

$$W_G = .0036d^3 + .0436d^2 + .1097d = .0478$$

Rod end:

$$W_E = 3.015\ (10^{-8})\ d^4 P^2$$

Fluid:

$$W_F = .0238\ d^2$$

" " given in figure A-9

The final weight equation is:

$$W_T = W_B + W_w + W_D + 2W_C + W + W_P + W_R + W_G + W_E + W_F$$

and this equation is shown plotted in figures B- 9 and B-10

HYDRAULIC CYLINDER SPACE STUDY

The space occupied by a hydraulic cylinder has been considered to consist of two cylindrical spaces, end to end, on the same longitudinal axis. The larger space is of a length equal to the distance from web end of the hydraulic cylinder to the gland end, (see figure B-11) and with a diameter equal to the outside diameter of the barrel nut. The smaller cylinder has a length equal to the distance from the gland end to the extreme rod end when the rod is extended to full stroke, and its diameter is equal to the largest diameter of the rod end. This arrangement provides for ports, etc.

From the foregoing it is evident that four dimensions are necessary to solve for the space (volume) requirements of a hydraulic cylinder; namely, large cylinder diameter, large cylinder length, small cylinder diameter, and small cylinder length.

To simplify this section of the work a semi-graphical method was utilized. This method consists of solving for general equations and using plots of these equations as the criteria in succeeding volume calculations.

As previously stated, the large cylinder diameter is that equal to the outermost dimension of the barrel lock nut. This is justified, since space enclosed by this dimension is most necessary for port allowances, etc. Referring to figure B-12 it is seen that this dimension is:

$$D_C = d + 2t + \Delta + 2(1.2\ t) + C$$

From The Glenn L. Martin Company Hydraulic Design Manual, data was plotted and from this plot it was seen that:

$$= .0793\ d \quad \frac{50,000 + 2.5P}{50,000 - 2.5P} + .1789$$

Also:

$$C = .385 \text{ for the range from .750 to 4 inch diameter barrel}$$

From the thick wall (Lame') Formula:

$$t = d\left(\tfrac{1}{2}\sqrt{\frac{50,000 + 2.5P}{50,000 - 2.5P}} - \tfrac{1}{2}\right)$$

And this all combines to yield:

$$D_C = d\ (\ 2.273\sqrt{\frac{50,000 + 2.5P}{50,000 - 2.5P}} - 1.2) + .5639 \qquad \text{(XXV)}$$

A plot of which is shown in figure B-13.

The length of the large cylinder was considered to be: (See figure B-14)

$$L_C = 1.5d + t_D + S + L$$

1.5d was considered to be a good allowance for the web length:

From Equation IIIa:

$$t_D = \frac{0.6\ d^2\ (2.5\ P)}{4\ S_t}$$

From Equation XXV:

Finally:

$$L_C = \theta + S \qquad \text{(XXVI)}$$

Where:

$$\theta = (1.9645 + .001583\ P\ d + 1.1642 \qquad \text{(XXVII)}$$

is plotted in figure

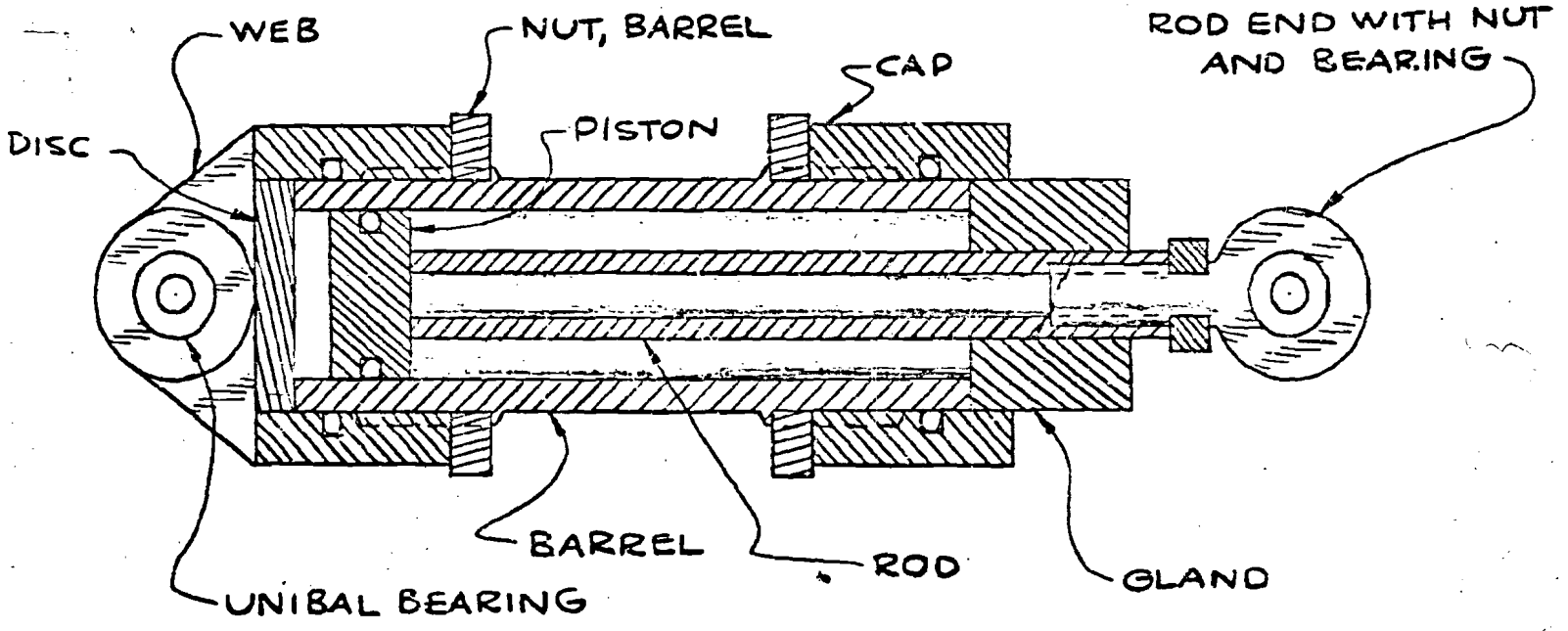
For the smaller cylinder (rod) assumptions were made based on data. A quantity "δ" to be added to the stroke to yield the small cylinder length, was considered to be equal to the bore. The small cylinder diameter was set at 0.85 times the bore diameter. This information is plotted in figure B-15.

For the total volume occupied by a hydraulic cylinder

$$V = \frac{\pi}{4}\left[D_C^2 + S + D_R^2\ (S + S)\right] \qquad \text{(XXVIII)}$$

(The values from the graphs are inserted along with known quantities)

The values for equation were solved and are plotted (figure B-16) to show the minimum pressure.

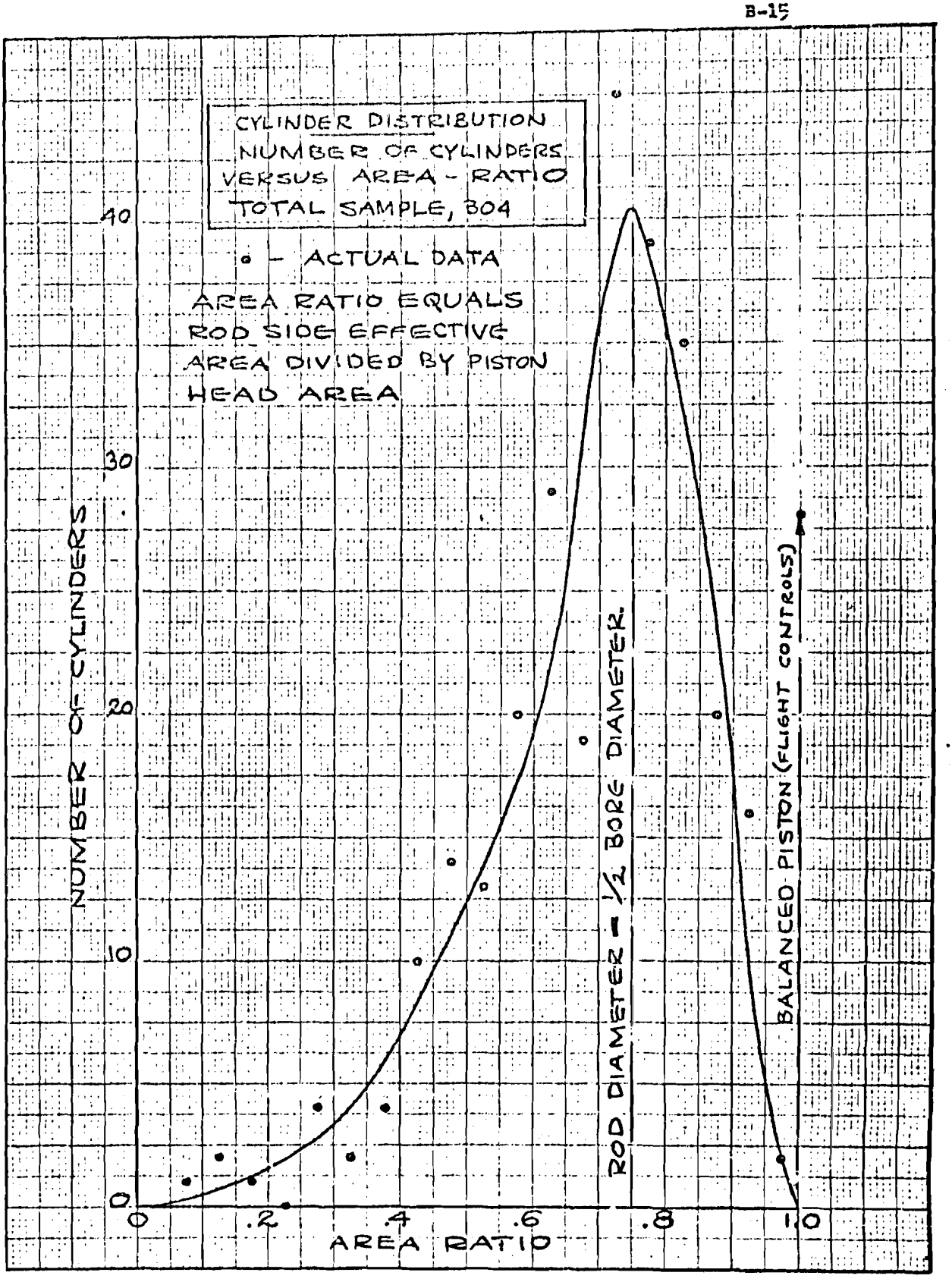


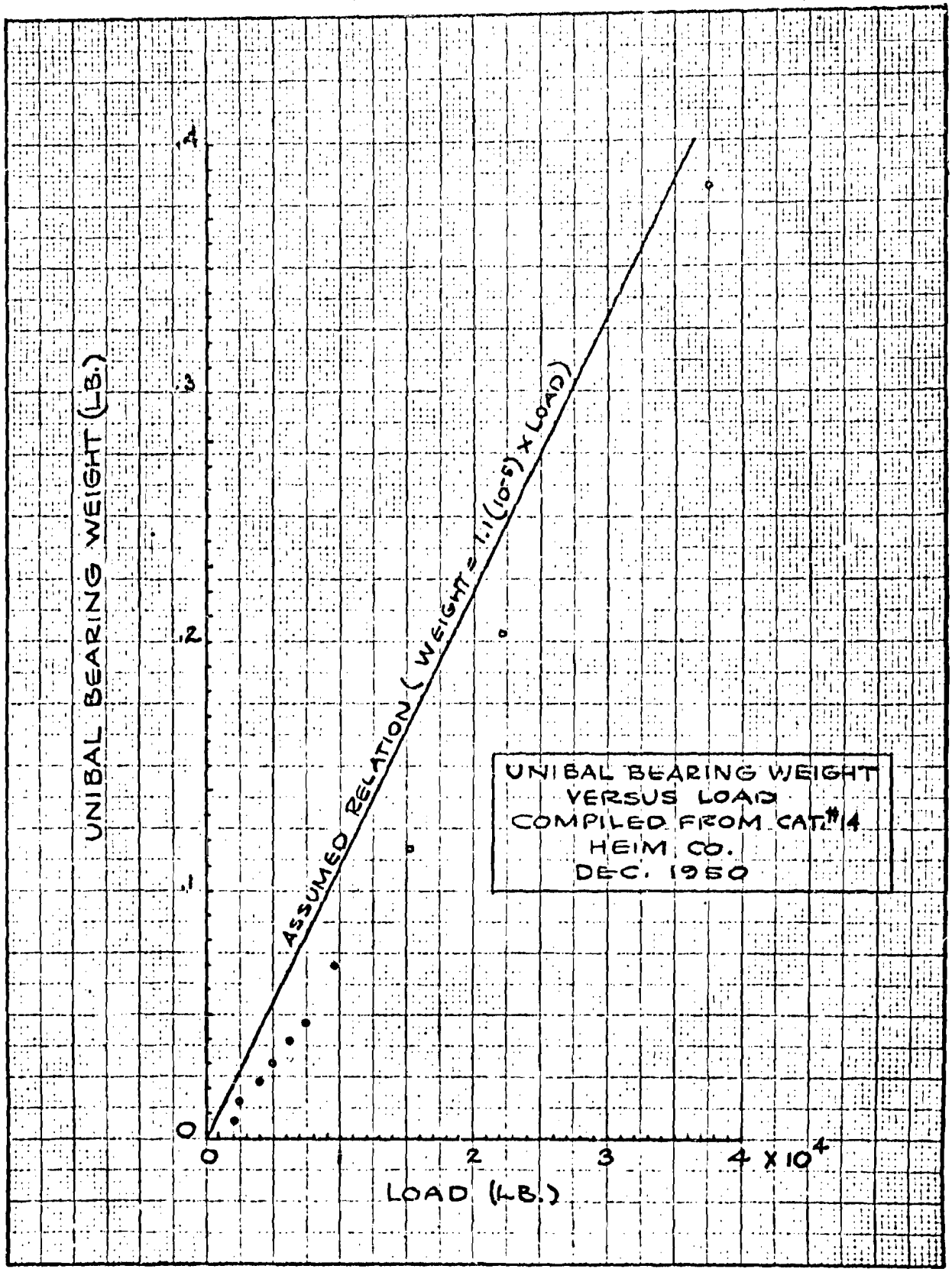
HYDRAULIC CYLINDER

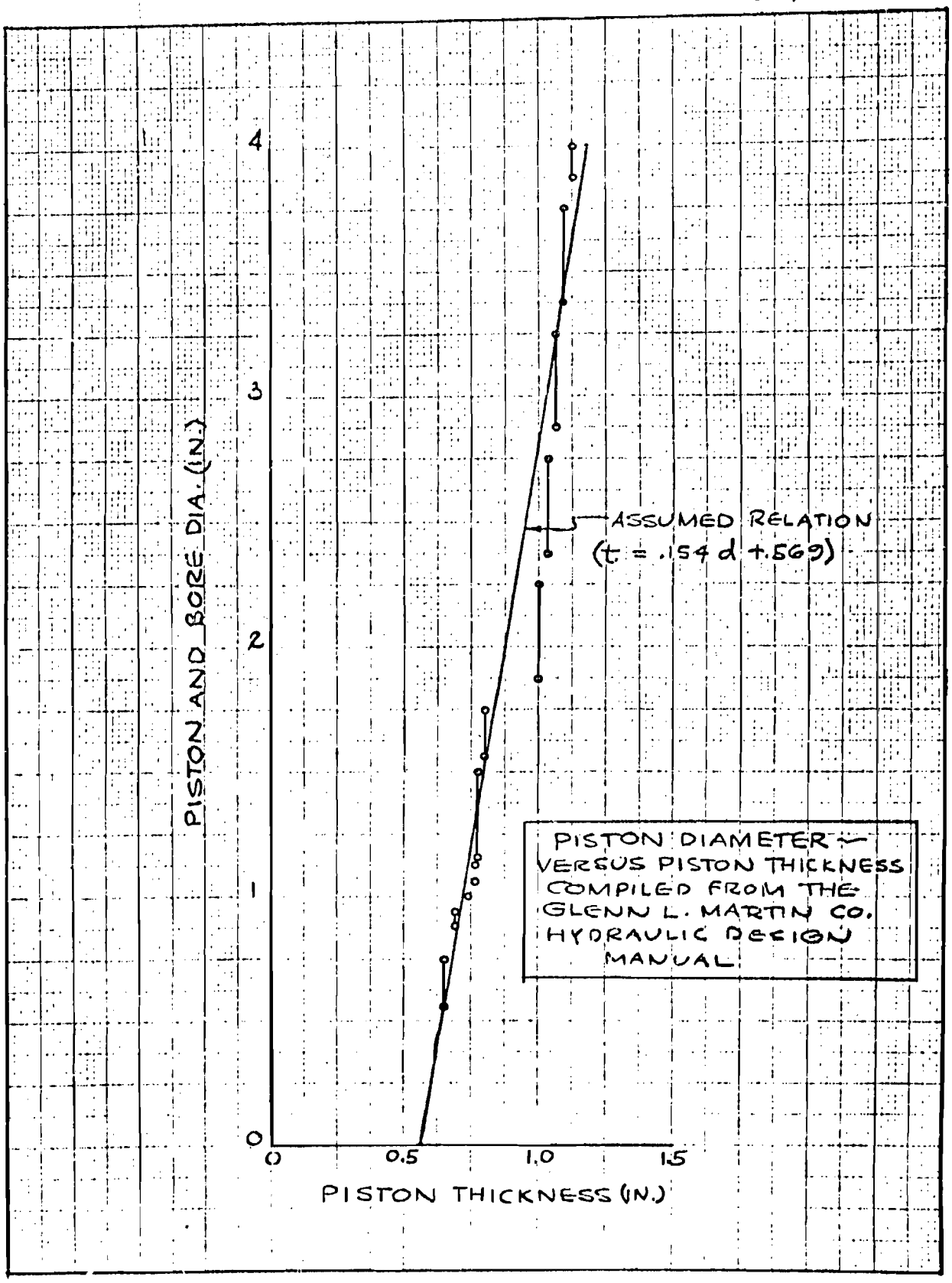
FOR WEIGHT STUDY

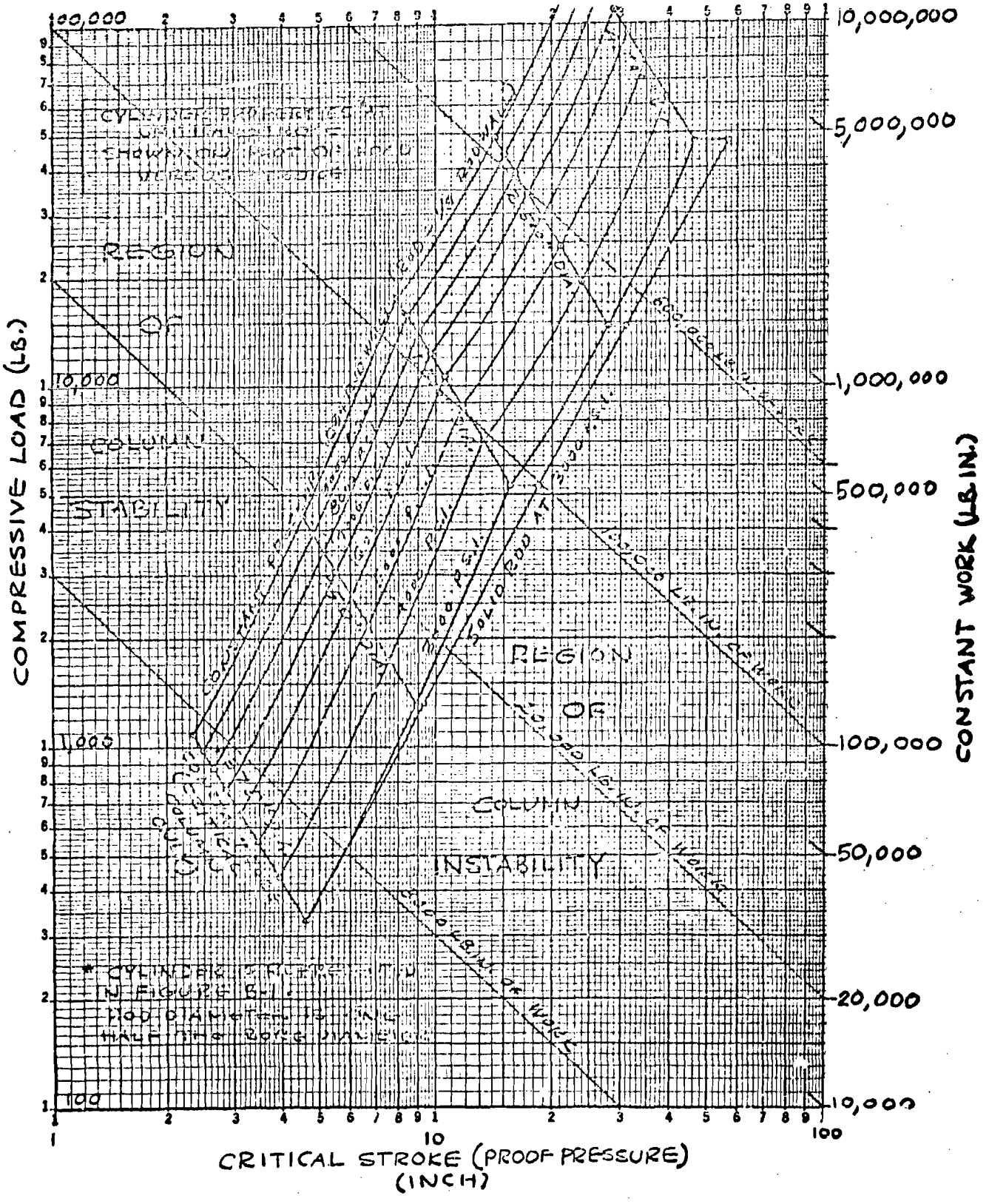
SHOWING BREAKDOWN OF PARTS INTO
ELEMENTS USED FOR ANALYSIS

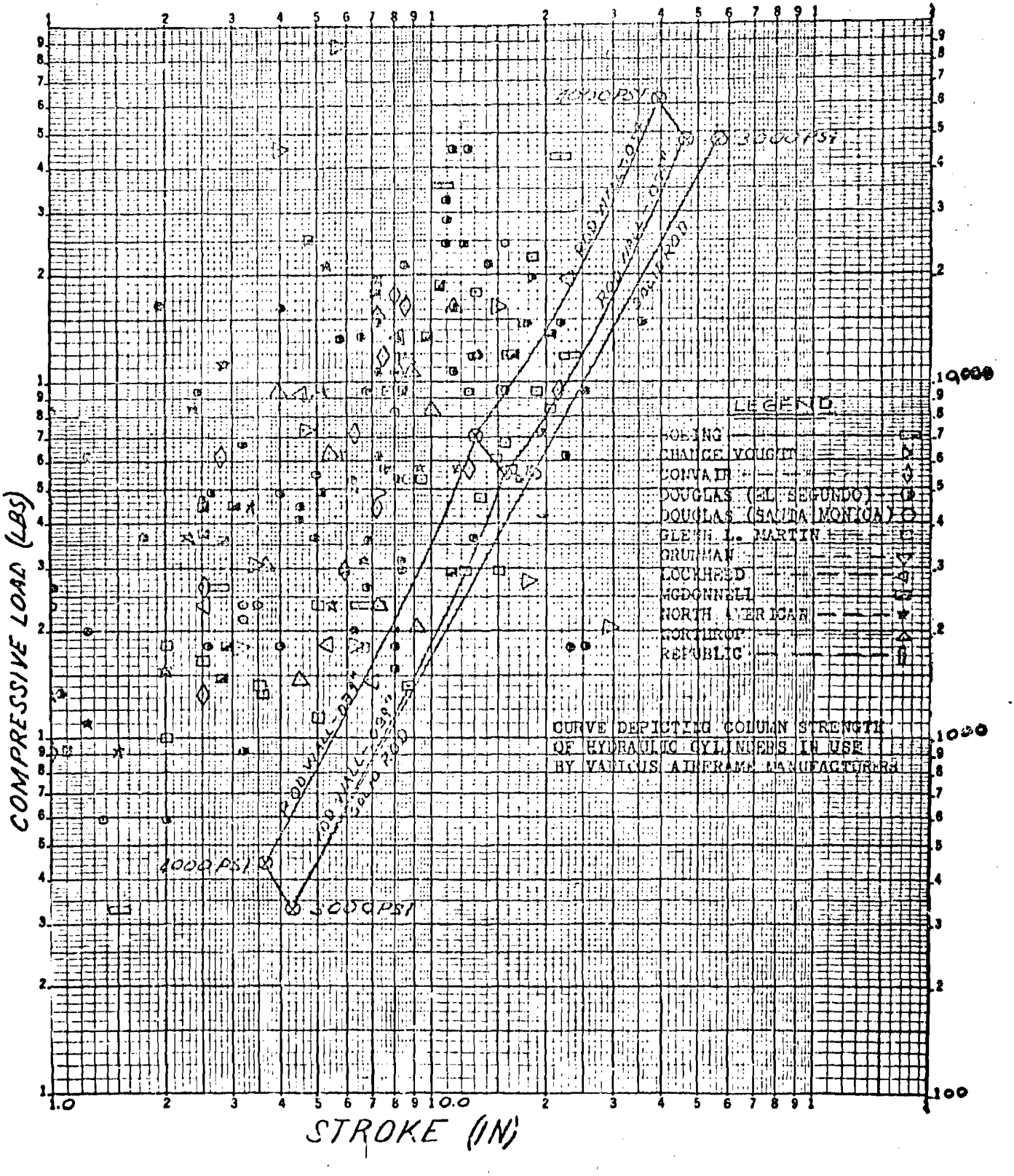
FIG. B-2
B-15

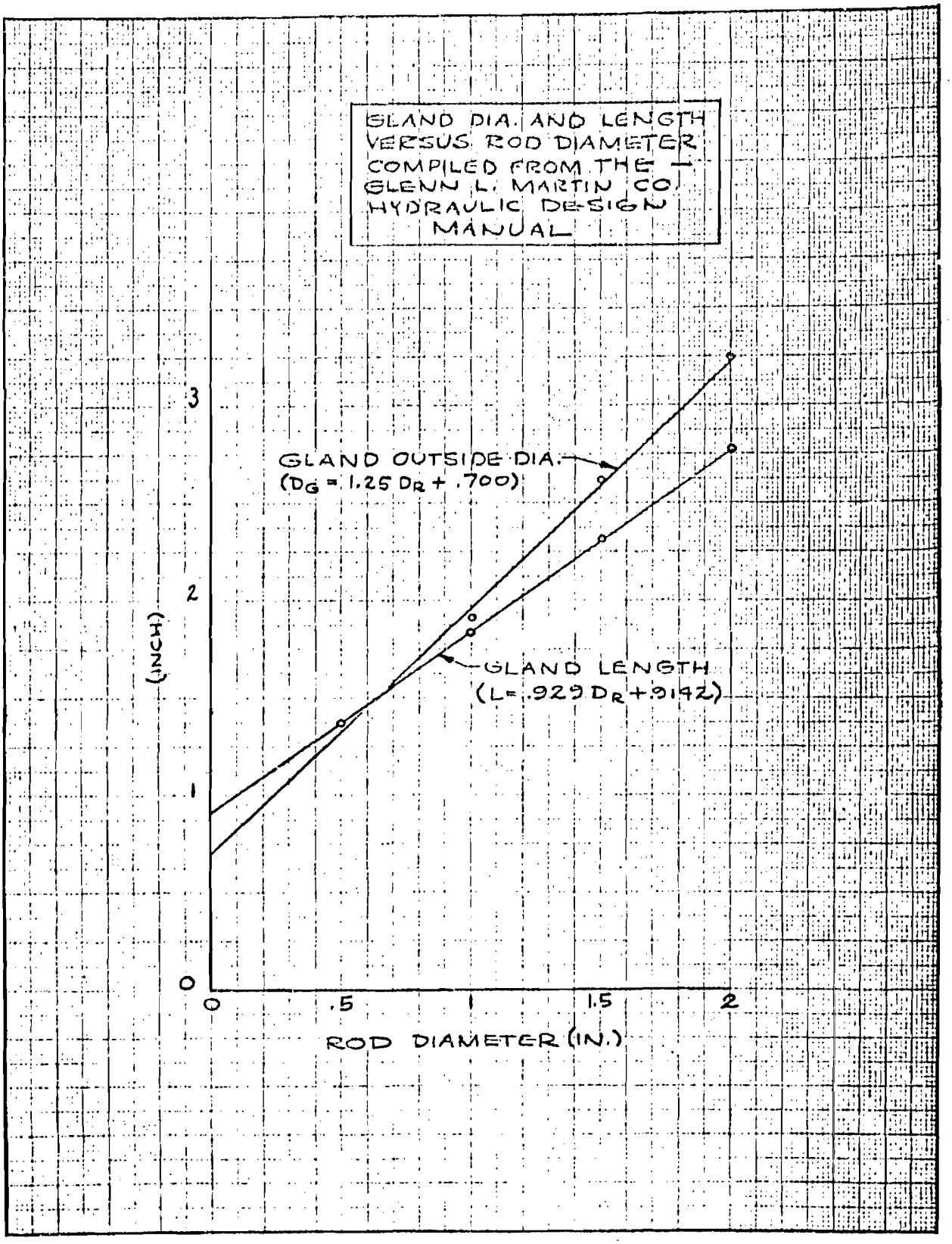

CYLINDER DISTRIBUTION
NUMBER OF CYLINDERS
VERSUS AREA - RATIO
TOTAL SAMPLE, 304
• - ACTUAL DATA
AREA RATIO EQUALS
ROD SIDE EFFECTIVE
AREA DIVIDED BY PISTON
HEAD AREA
NUMBER OF CYLINDERS
0
10
20
30
40
0
.2
.4
.6
.8
1.0
AREA RATIO
ROD DIAMETER = 1/2 BORE DIAMETER
BALANCED PISTON (FLIGHT CONTROLS)

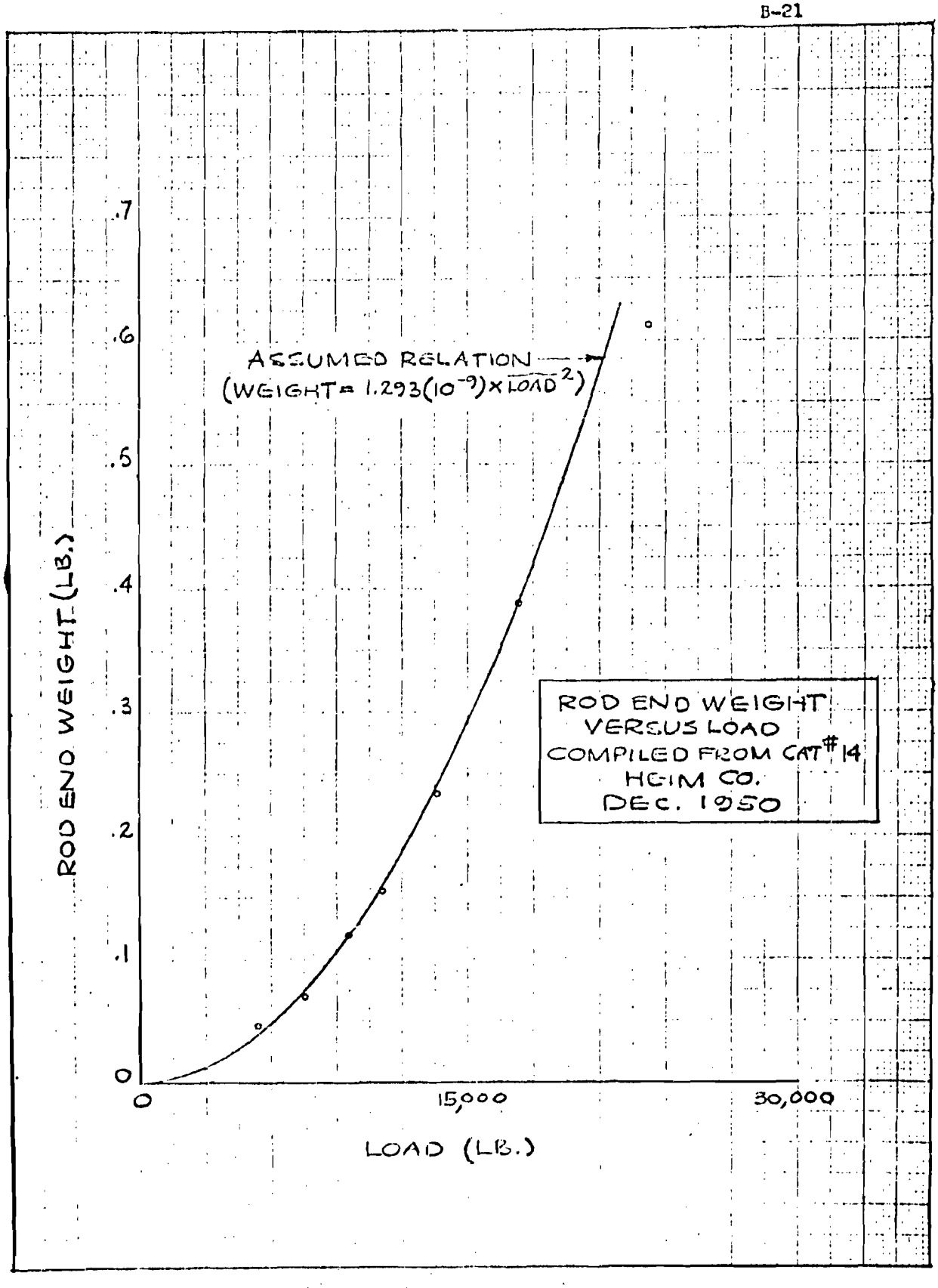

UNIBAL BEARING WEIGHT (LB.)
.4
.3
.2
.1
0
0
2
3
4 X 10^4
LOAD (LB.)
ASSUMED RELATION (WEIGHT = 1.1(10^-5) X LOAD)
UNIBAL BEARING WEIGHT
VERSUS LOAD
COMPILED FROM CAT. #4
HEIM CO.
DEC. 1950

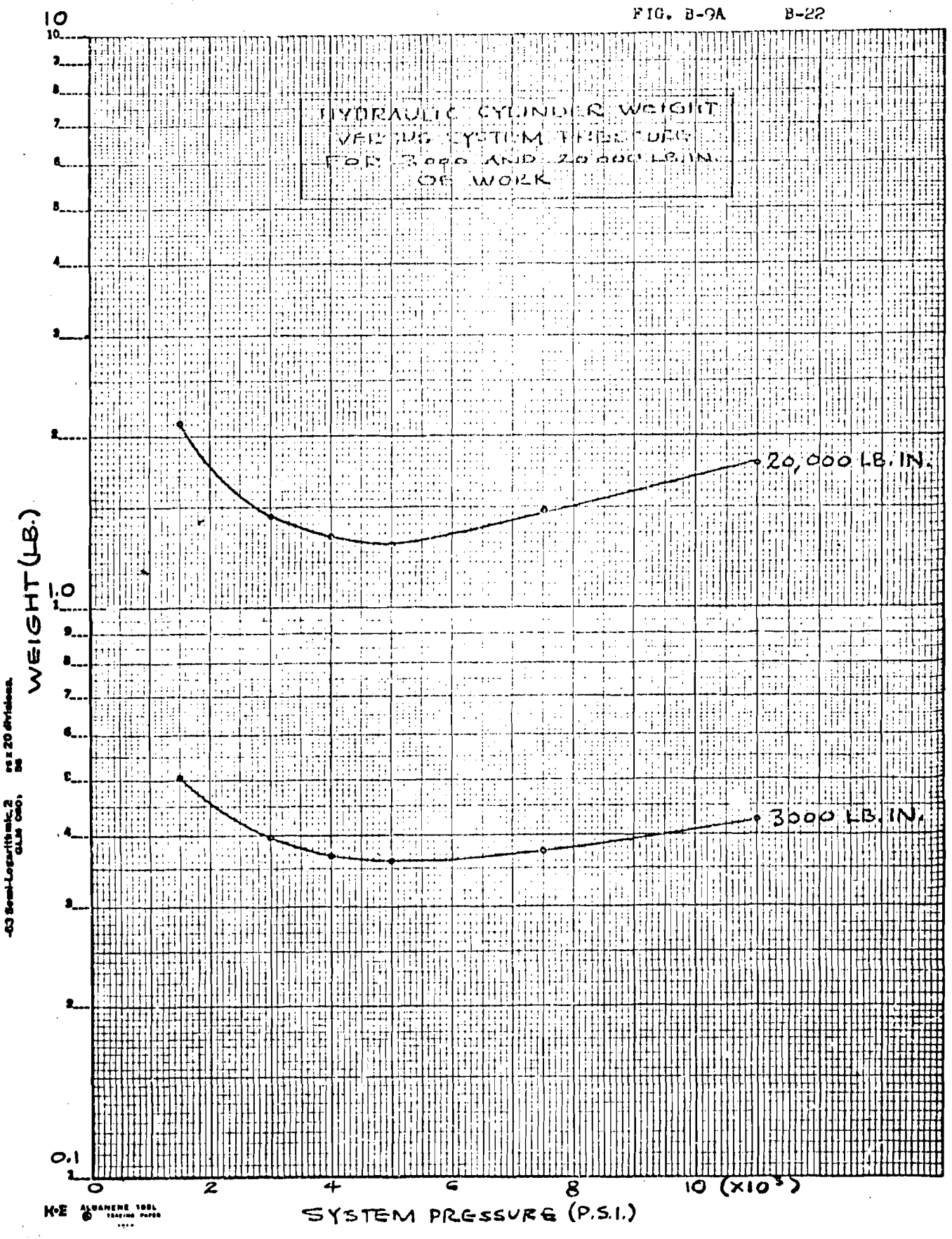

PISTON AND BORE DIA. (IN.)
4
3
2
1
0
0
0.5
1.0
1.5
PISTON THICKNESS (IN.)
ASSUMED RELATION
(t = .154 d + .569)
PISTON DIAMETER ~
VERSUS PISTON THICKNESS
COMPILED FROM THE
GLENN L. MARTIN CO.
HYDRAULIC DESIGN
MANUAL

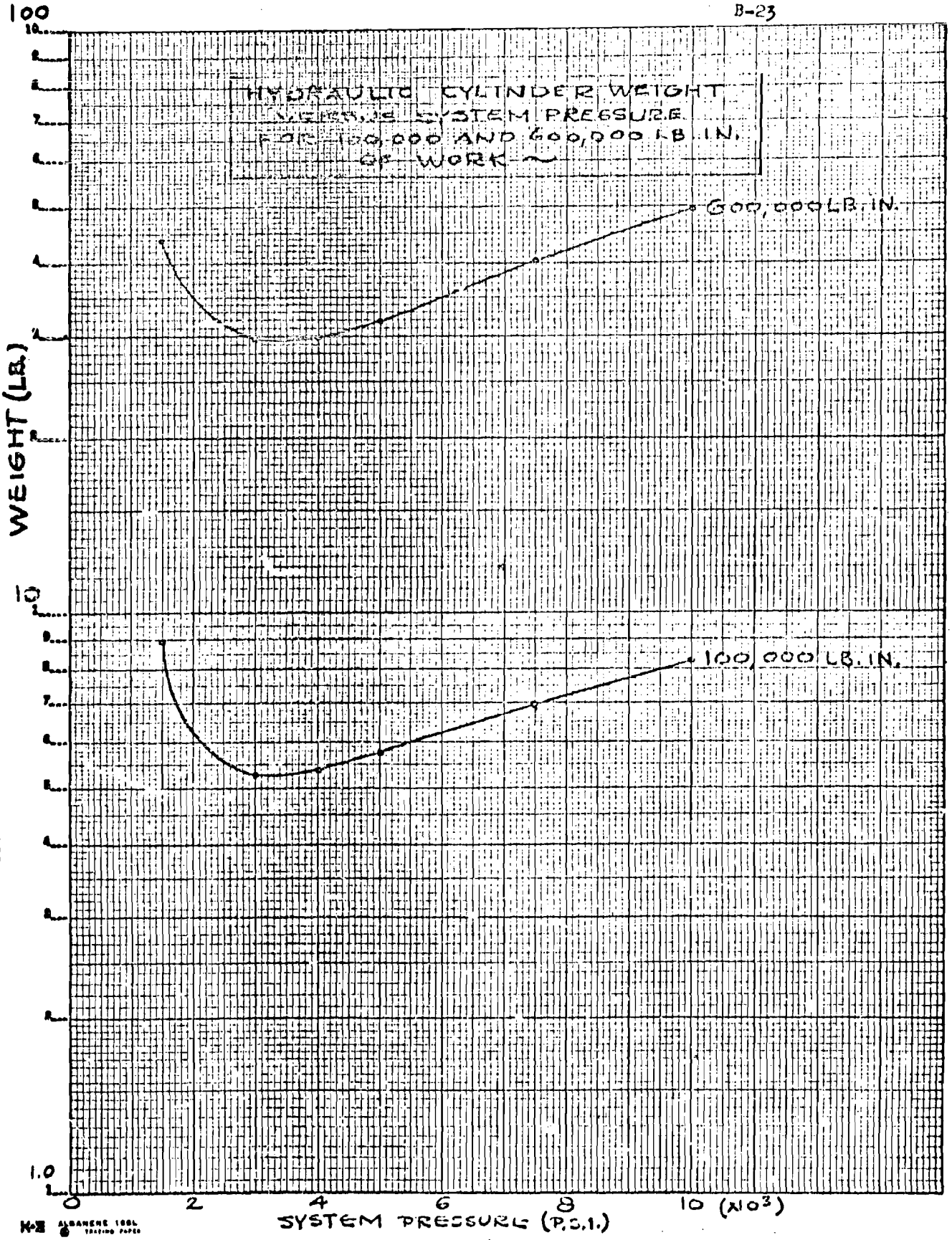

COMPRESSIVE LOAD (LB.)
CONSTANT WORK (LB-IN.)
CRITICAL STROKE (PROOF PRESSURE) (INCH)
REGION OF COLUMN STABILITY
REGION OF COLUMN INSTABILITY
100,000
10,000
1,000
100
10,000,000
5,000,000
1,000,000
500,000
100,000
50,000
20,000
10,000
1
10
100

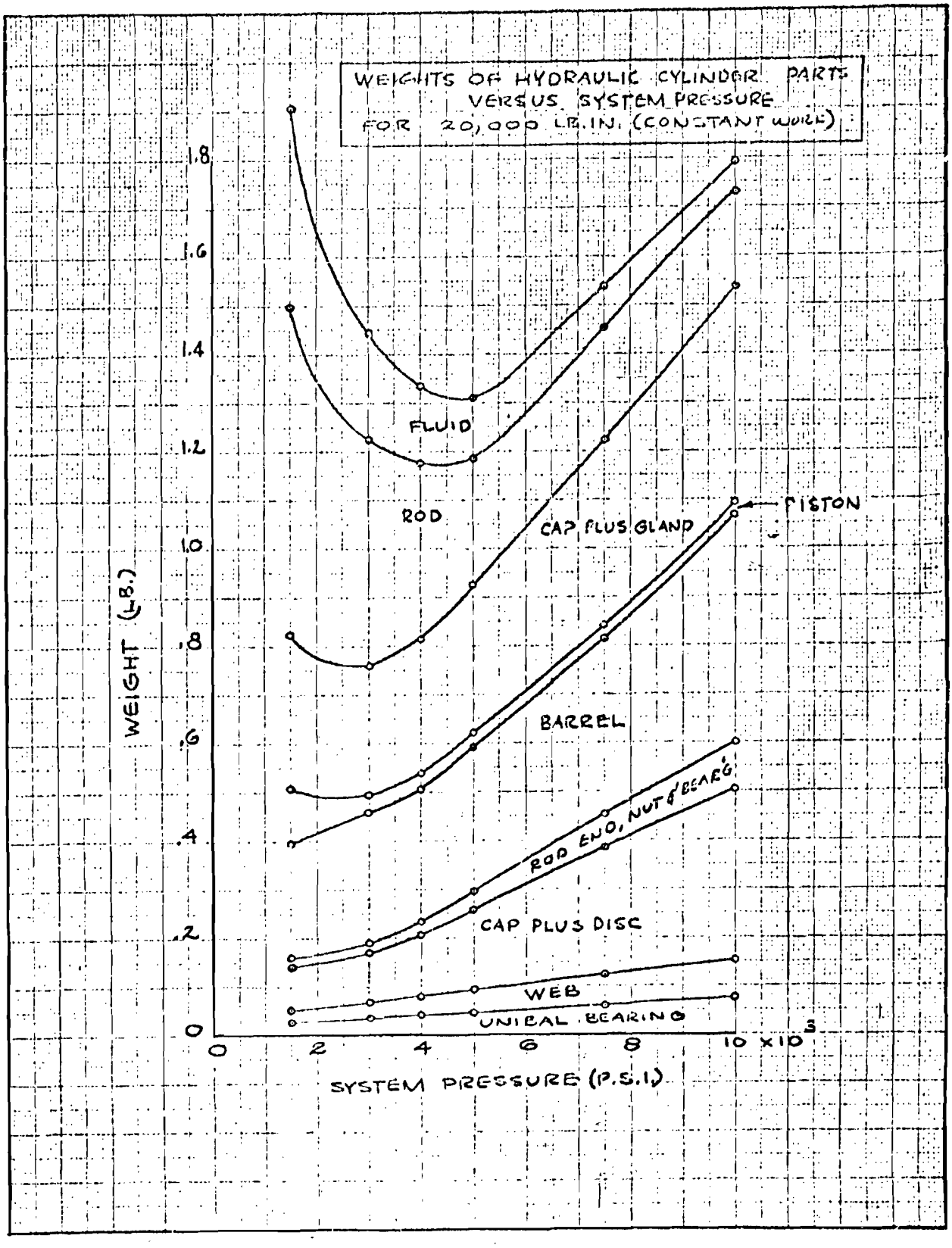


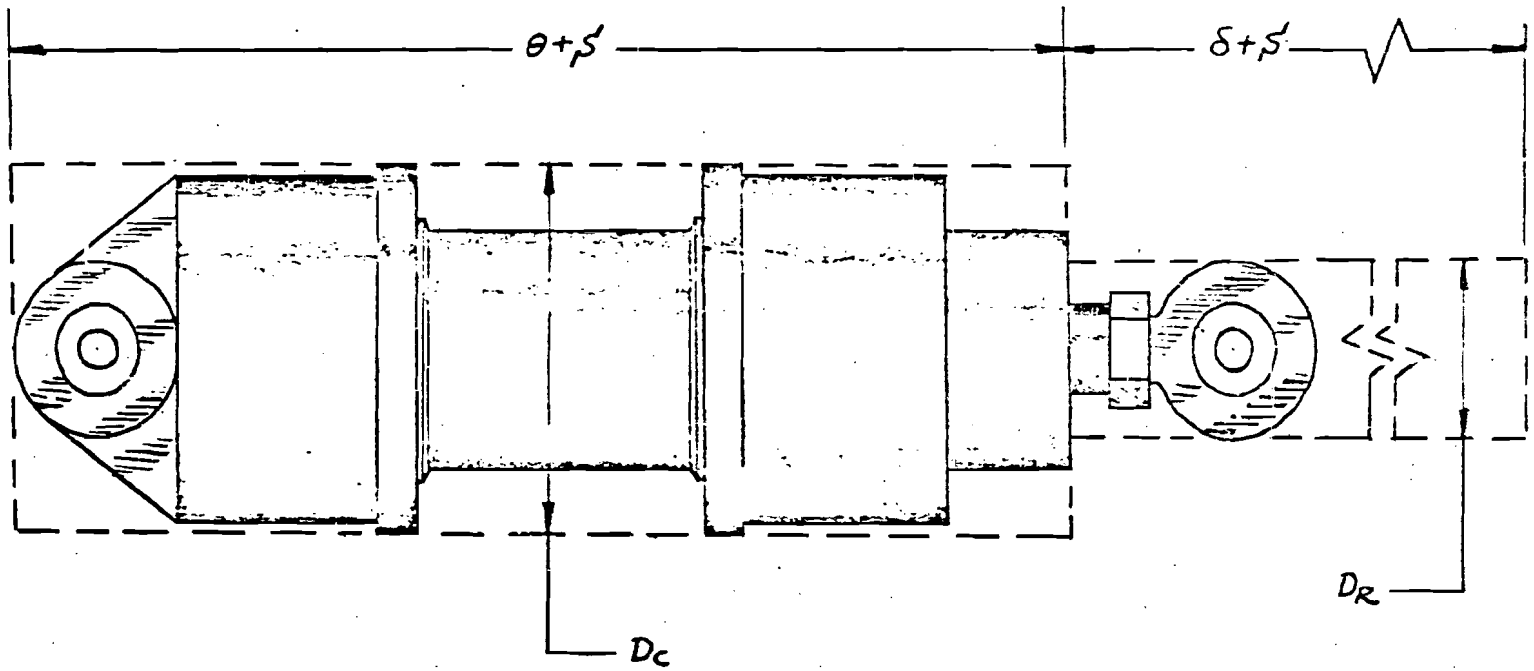

GLAND DIA. AND LENGTH VERSUS ROD DIAMETER COMPILED FROM THE — GLENN L. MARTIN CO. HYDRAULIC DESIGN MANUAL
GLAND OUTSIDE DIA.
($D_G = 1.25\,D_R + .700$)
GLAND LENGTH
($L = .929\,D_R + .9142$)
(INCH)
0
1
2
3
0
.5
1
1.5
2
ROD DIAMETER (IN.)


ASSUMED RELATION
(WEIGHT = 1.293(10^-9) × LOAD^2)
ROD END WEIGHT VERSUS LOAD
COMPILED FROM CAT # 14
HGIM CO.
DEC. 1950
ROD END WEIGHT (LB.)
.7
.6
.5
.4
.3
.2
.1
0
0
15,000
30,000
LOAD (LB.)


HYDRAULIC CYLINDER WEIGHT
VERSUS SYSTEM PRESSURE
FOR 3000 AND 20,000 LB.IN.
OF WORK
20,000 LB. IN.
3000 LB. IN.
WEIGHT (LB.)
SYSTEM PRESSURE (P.S.I.)
10
1.0
0.1
0
2
4
6
8
10 (x10³)

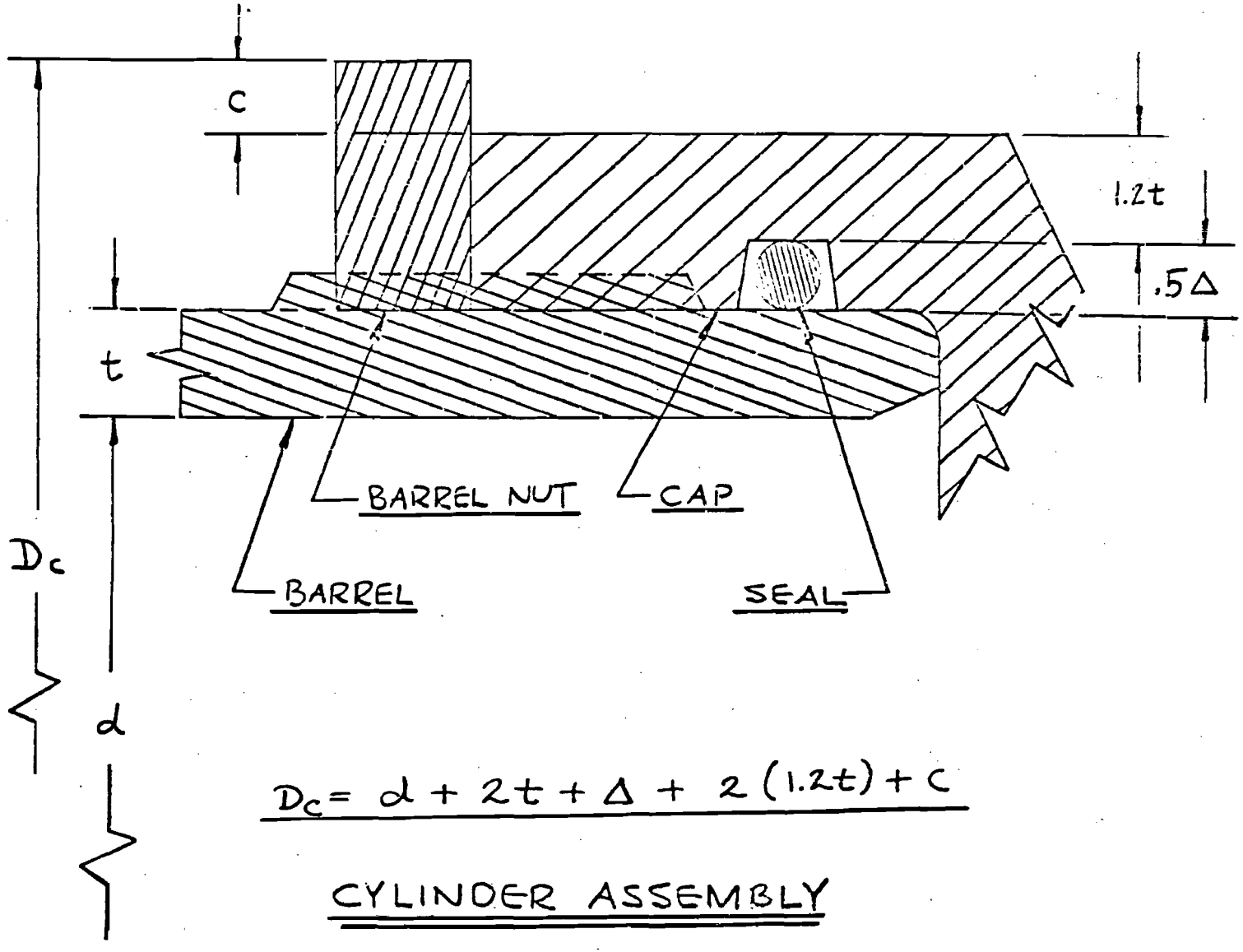

HYDRAULIC CYLINDER WEIGHT
VERSUS SYSTEM PRESSURE
FOR 100,000 AND 600,000 LB. IN.
OF WORK
600,000 LB. IN.
100,000 LB. IN.
WEIGHT (LB.)
100
10
1.0
0
2
4
6
8
10 (X10³)
SYSTEM PRESSURE (P.S.I.)


WEIGHTS OF HYDRAULIC CYLINDER PARTS
VERSUS SYSTEM PRESSURE
FOR 20,000 LB.IN. (CONSTANT WORK)
WEIGHT (LB.)
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
0
FLUID
ROD
CAP PLUS GLAND
PISTON
BARREL
ROD END, NUT & BEARG
CAP PLUS DISC
WEB
UNIBAL BEARING
0
2
4
6
8
10 X10^3
SYSTEM PRESSURE (P.S.I.)



HYDRAULIC CYLINDER
FOR SPACE STUDY

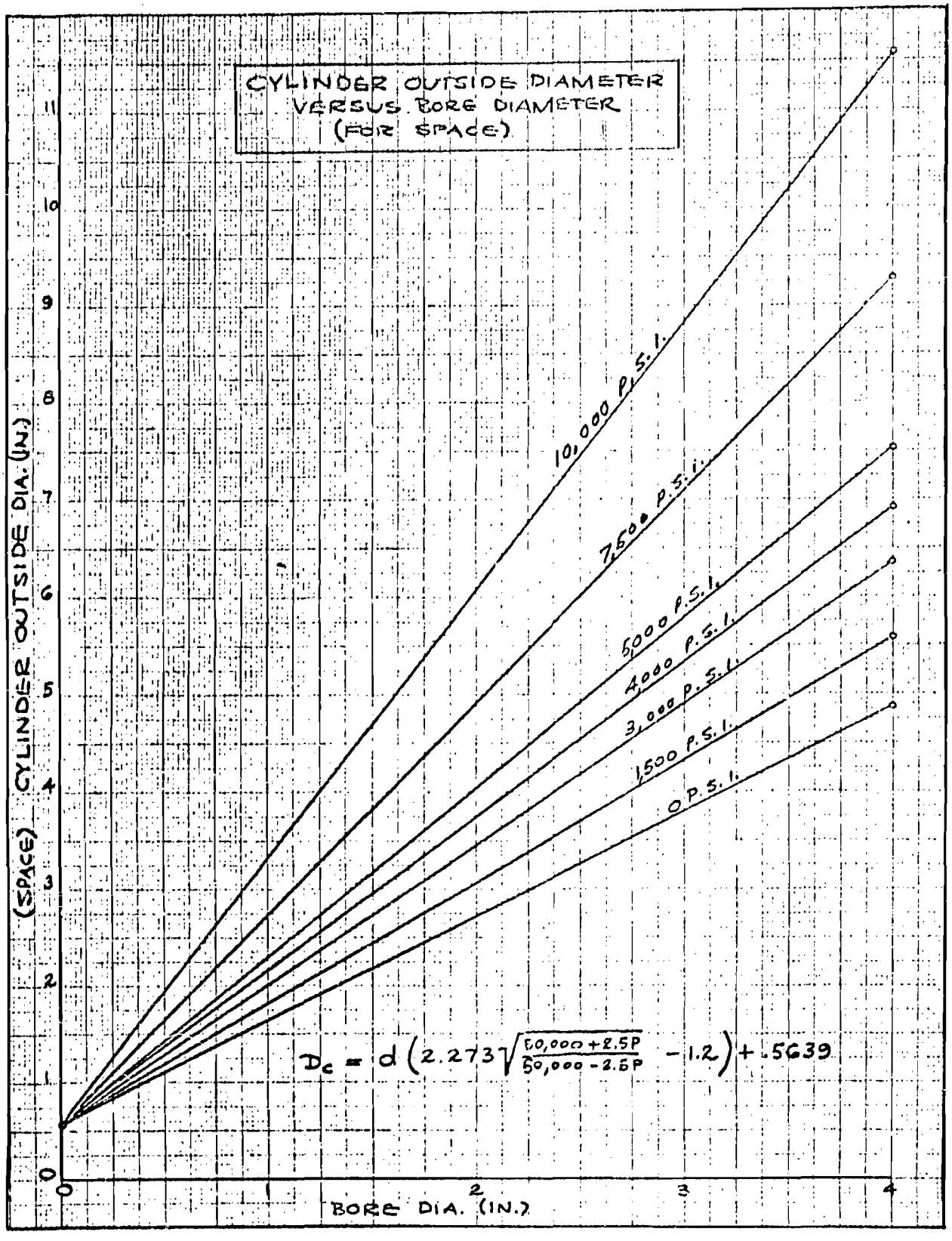
FIG. B-11
B-25

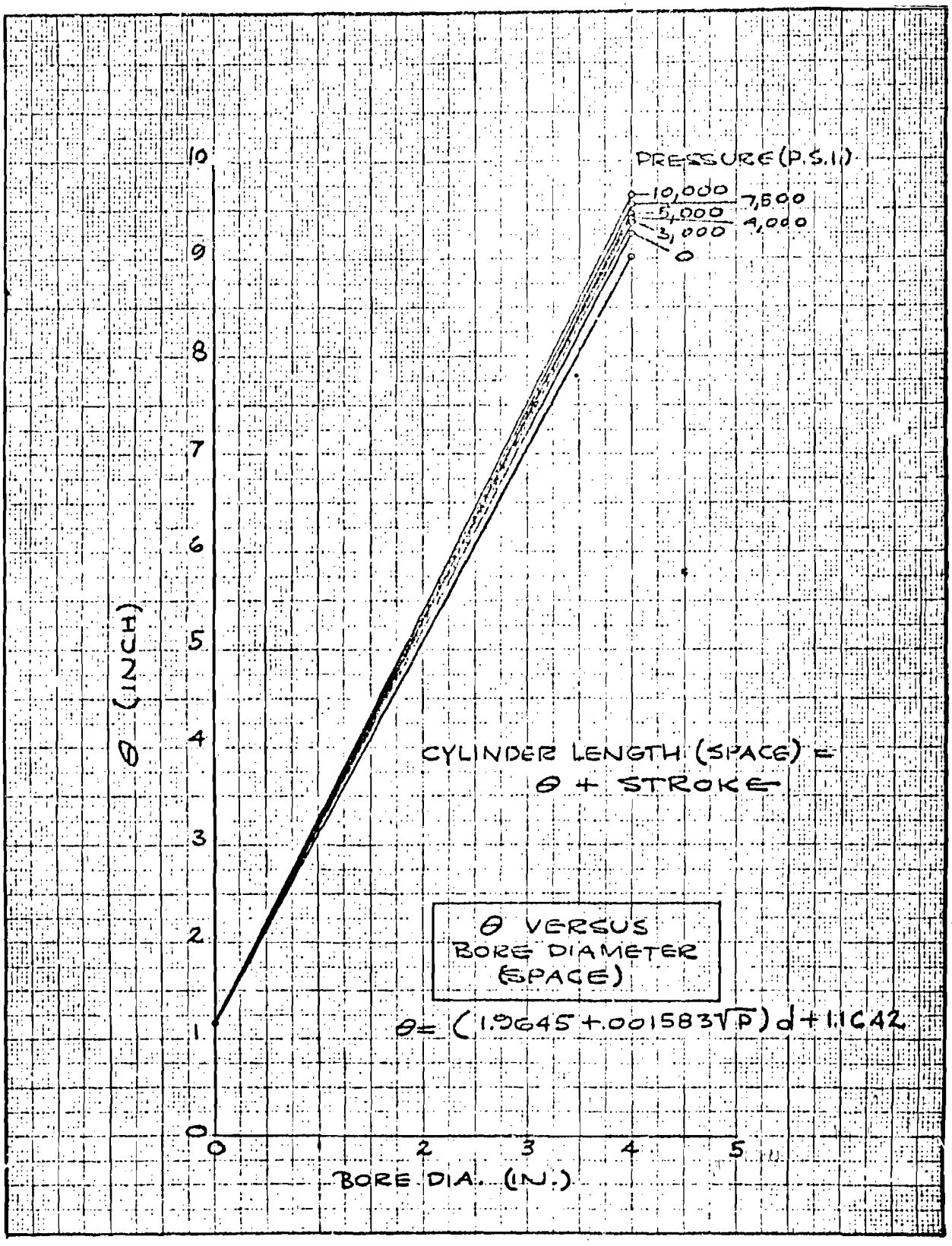


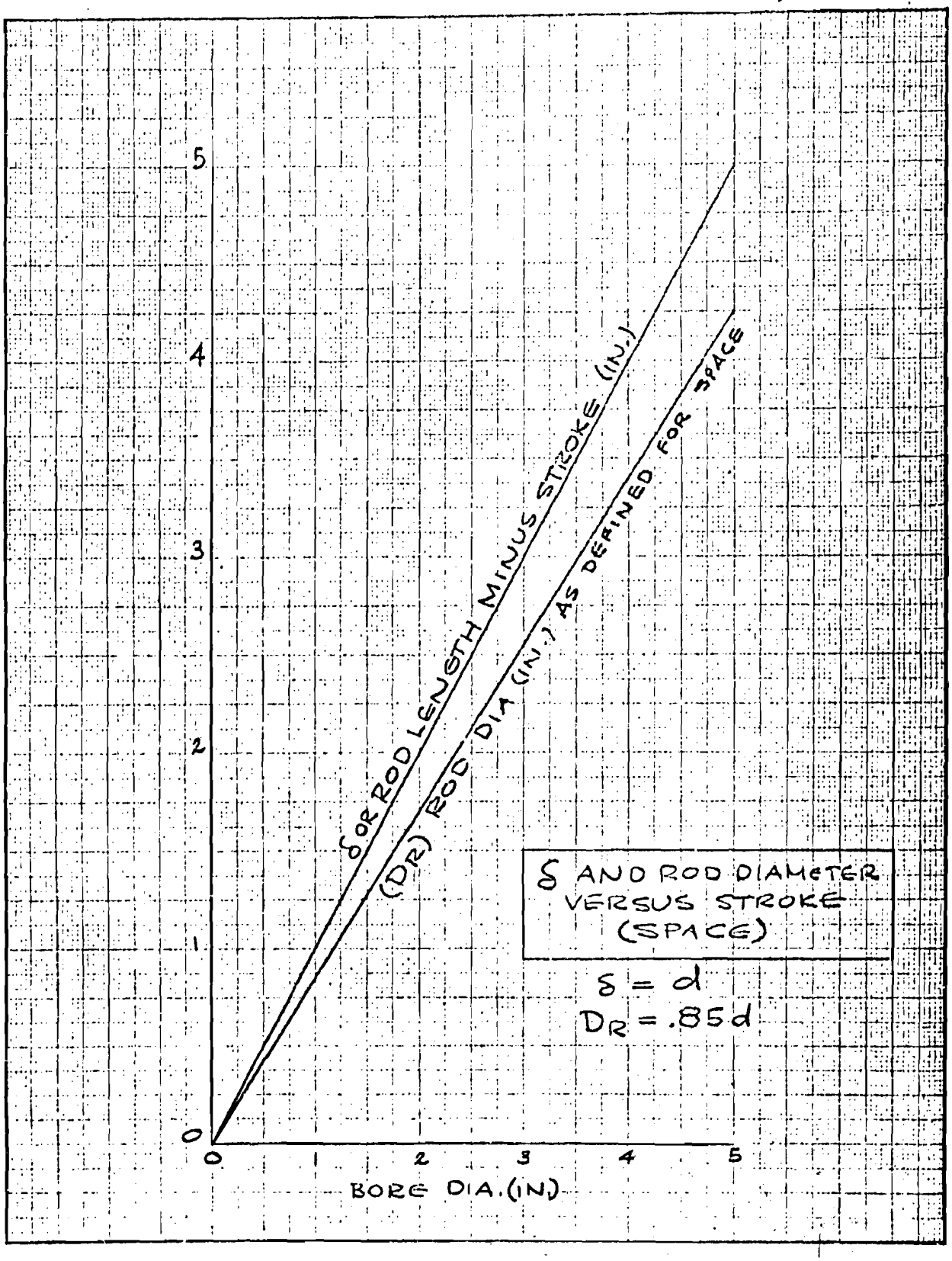
$$D_c = d + 2t + \Delta + 2(1.2t) + c$$

CYLINDER ASSEMBLY

FIG. B-12
B-26


CYLINDER OUTSIDE DIAMETER
VERSUS BORE DIAMETER
(FOR SPACE)
10,000 P.S.I.
7,500 P.S.I.
6000 P.S.I.
4,000 P.S.I.
3,000 P.S.I.
1,500 P.S.I.
0 P.S.I.
$D_c = d\left(2.273\sqrt{\frac{50,000+2.5P}{50,000-2.5P}} - 1.2\right) + .5639$
(SPACE) CYLINDER OUTSIDE DIA. (IN.)
BORE DIA. (IN.)
0
1
2
3
4
5
6
7
8
9
10
11


PRESSURE (P.S.I.)
10,000
7,500
5,000
4,000
3,000
0
CYLINDER LENGTH (SPACE) =
θ + STROKE
θ VERSUS
BORE DIAMETER
(SPACE)
θ = (1.9645 + .001583√P) d + 1.1642
θ (INCH)
BORE DIA. (IN.)
0
1
2
3
4
5
6
7
8
9
10


δ OR ROD LENGTH MINUS STROKE (IN.)
(DR) ROD DIA (IN.) AS DEFINED FOR SPACE
δ AND ROD DIAMETER VERSUS STROKE (SPACE)
δ = d
DR = .85d
0
1
2
3
4
5
BORE DIA. (IN.)

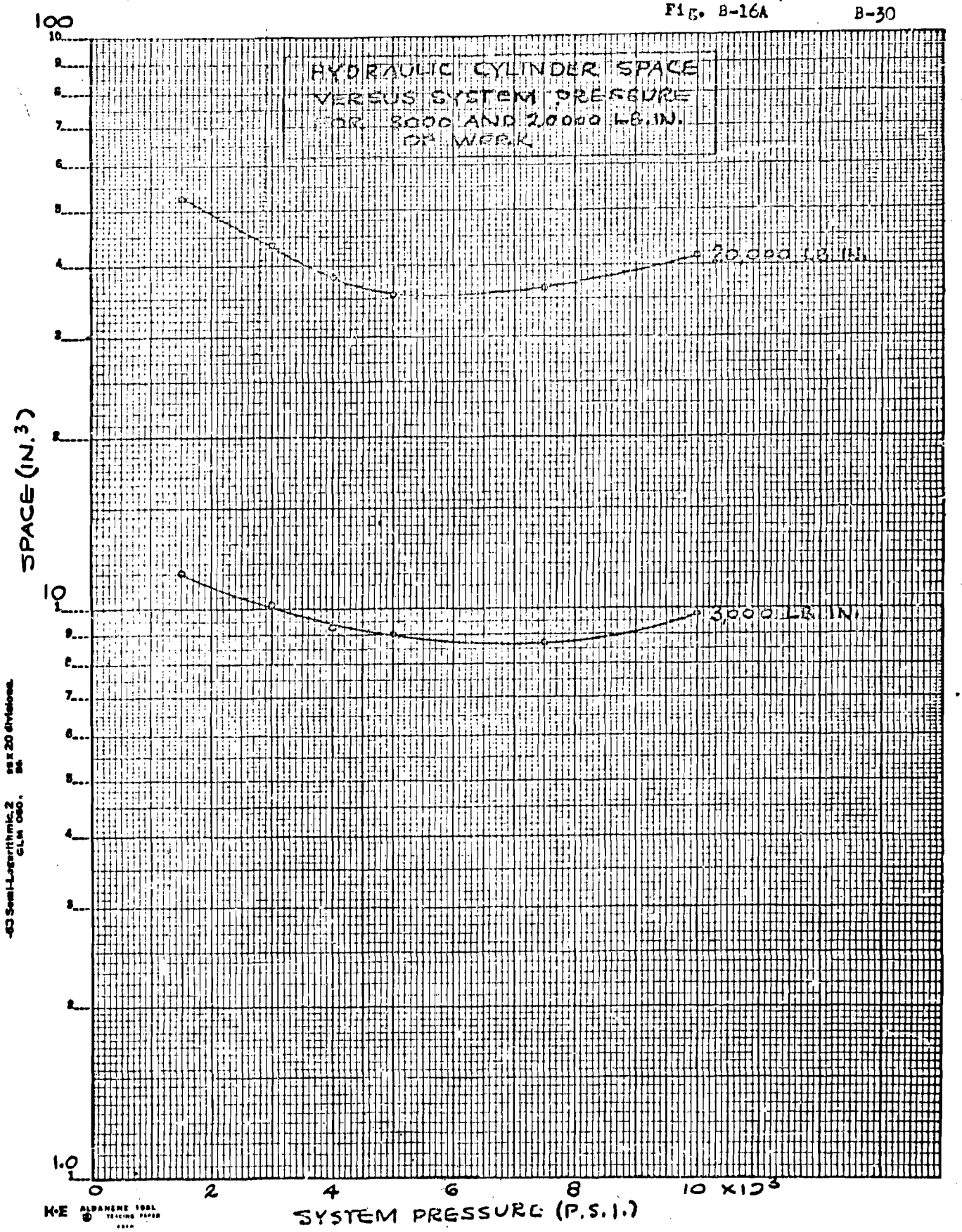
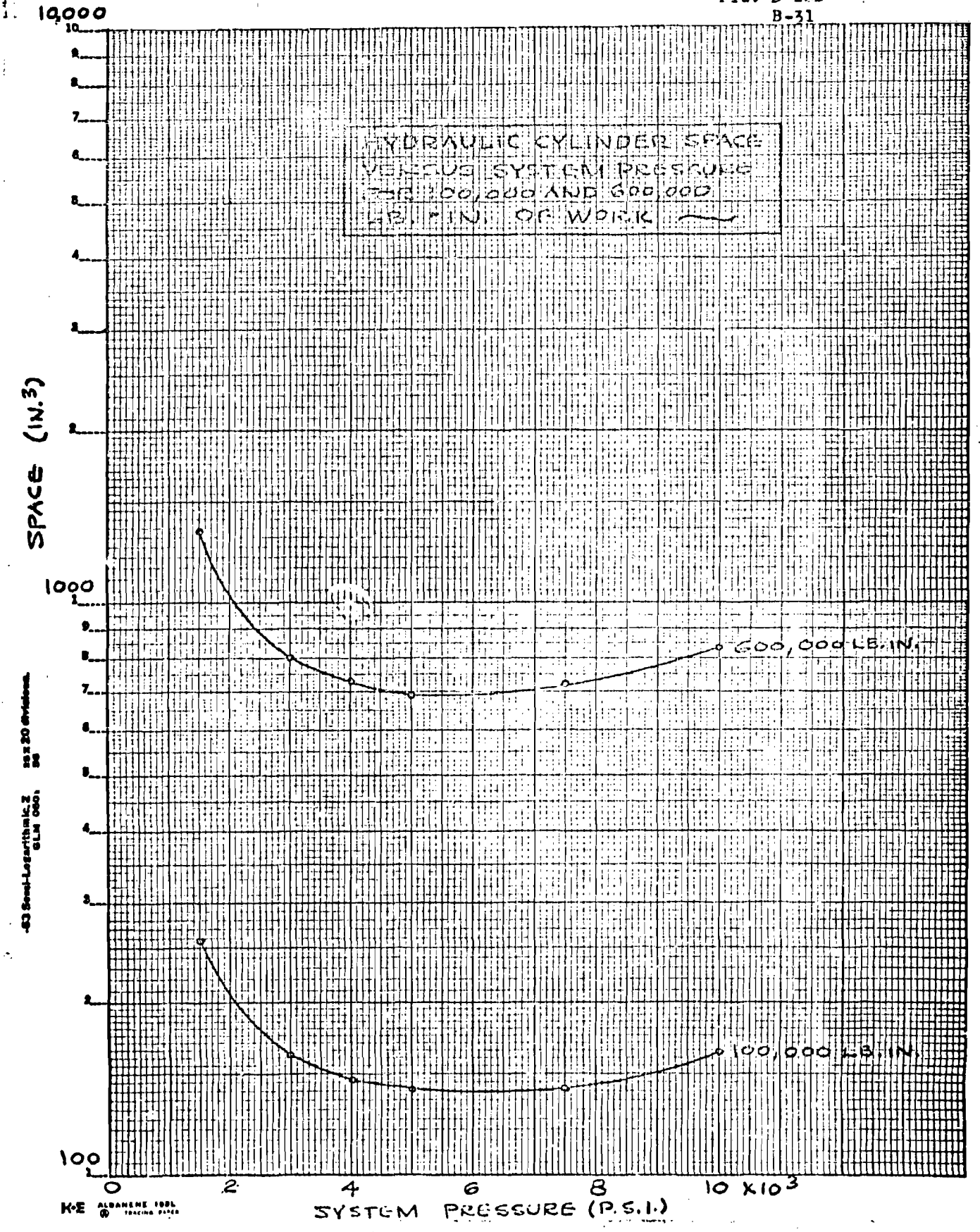


FIG. B-16B


HYDRAULIC CYLINDER SPACE
VERSUS SYSTEM PRESSURE
FOR 100,000 AND 600,000
LB. IN. OF WORK
SPACE (IN.³)
10,000
1000
100
600,000 LB. IN.
100,000 LB. IN.
0
2
4
6
8
10 X10³
SYSTEM PRESSURE (P.S.I.)

CALCULATION OF THE WEIGHT OF HYDRAULIC CYLINDERS

W (WORK)	d	P	S	TOTAL
3,000	.479	3,000	5.54	.39808
	.462	4,000	4.48	.36946
	.448	5,000	3.82	.360776
	.423	7,500	2.85	.384008
	.406	10,000	2.32	.421460
20,000	.935	3,000	10.30	1.444936
	.868	4,000	8.45	1.331672
	.840	5,000	7.22	1.310100
	.792	7,500	5.42	1.540750
	.757	10,000	4.44	1.793504
100.000	2.255	1,500	16.7	8.91810
	1.747	2,500	16.7	5.6007
	1.593	3,000	16.7	5.26971
	1.538	4,000	13.5	5.37050
	1.493	5,000	11.4	5.76279
	1.405	7,500	8.6	6.95838
	1.325	10,000	7.25	8.27974
600,000	4.015	1,500	31.6	43.9658
	3.11	2,500	31.6	31.1106
	2.84	3,000	31.6	29.65390
	2.725	4,000	25.8	29.94380
	2.647	5,000	21.8	31.95175
	2.52	7,500	16.2	40.2212
	2.39	10,000	13.4	49.1201

20,000 INCH LB. OF WORK

d	P	S	UNIBAL BEARING	CYLINDER END	HEAD END CAP	LOLYPOP	BARREL OVER	PISTON
.935	3,000	10.30	3.40 (10^{-2})	3.52 (10^{-2})	.10120	.02074	.2684	.036286
.868	4,000	8.45	3.9 "	4.04 "	.1277	.02735	.2725	.031122
.840	5,000	7.22	4.58 "	4.74 "	.1653	.03755	.2960	.02911
.792	7,500	5.42	6.11 "	6.32 "	.2616	.0668	.3655	.02583
.757	10,000	4.44	7.44 "	7.69 "	.3505	.0989	.4670	.023534

ROD END CAP	ROD	FLUID	TOTAL
.26464	.46747	.217	1.444936
.27696	.36334	.1533	1.331672
.30583	.25931	.1238	1.310100
.38169	.23213	.0829	1.540750
.44781	.19176	.0627	1.793504

W (WORK)	P	d	S	TOTAL SPACE
	$V = \frac{\pi}{4}\left[D_C^2 (\theta + S) + D_R^2 (S + S)\right]$			
3,000	3,000	.479	5.54	10.183
	4,000	.462	4.48	9.288
	5,000	.448	3.82	9.026
	7,000	.423	2.85	8.74
	10,000	.406	2.32	9.773
20,000	3,000	.935	10.30	43.82
	4,000	.868	8.45	37.55
	5,000	.840	7.22	35.86
	7,000	.792	5.42	36.40
	10,000	.757	4.44	41.443
100,000	1,500	2.255	16.7	257.3
	3,000	1.593	16.7	162.1
	4,000	1.538	13.5	147.0
	5,000	1.493	11.4	140.32
	7,500	1.405	8.6	141.72
	10,000	1.325	7.25	164.28
600,000	1,500	4.015	31.6	1,341.5
	3,000	2.840	31.6	801.8
	4,000	2.725	25.8	732.1
	5,000	2.647	21.8	690.1
	7,500	2.520	16.2	720.3
	10,000	2.390	13.4	837.1

CYLINDER DATA
BOEING

NAME	Pist Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod End Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Canopy Latch	1.62	.75	2.78		2.061	1.619	.783			B-47B	6180
Bomb Door			21.50		3.17	2.63	.830				9500
Spoiler	1.75	.88	32.36		2.405	1.796	.746				7220
Aileron	2.00	.75	5.10		2.698	2.698	1.00				8080
Flaperon Power	2.00	.75	2.05		2.698	2.698	1.00				8080
Elev. Power	2.00	.75	3.40		2.698	2.698	1.00				8080
Spoiler			5.92		.99	.39	.394				2970
IFR.			2.5		.89	.44	.494				2670
Rudder Power	2.00	.75	3.40		2.698	2.698	1.00				8080
Hyd. Lock	.75	.5	2.50		.4418	.2455	.555				1325
Aut. Brake	.56	.58			.246	1326	.538			B-47B	738
Main Gear	4.26	1.312	21.93		14.24	12.88	.904			B-52	42700
B. Door Front	2.24	1.612	23.0		3.94	1.92	.482				11950
B. Door Latch			1.5		.11						330
B. Door Rear	2.255	1.620	23.0		3.99	1.92	.482				11950
Steering	3.875	.474	10.50		11.79	10.02	.850				35300
IFR Door	1.000	.695	6.43		.79	.41	.518				2370
Fr. Recp. Toggles	1.058	.748	2.77		.88	.44	.500			B-52	2640

CYLINDER DATA
CHANCE VOUGHT

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
										F7U-1	
Nose Gear	3.25	2.5	17.657		8.296	3.387	.408			1200 psi	1435
Main Gear	3.25	2.5	21.954		8.296	3.387	.408			1200 psi	2750
Main Gear Door	1.122	.499	4.00		.9852	.7696	.801				
Nose Gear Door	1.747	.561	3.56		2.405	2.1578	.897			1200 psi	505
Speed Brake	2.122	.998	16.34		3.53	2.7577	.780			1500 psi	6600
Alevator	2.623	1.373	5.68		3.916	3.916	1.00			3020 psi	26650
Canopy	1.997	1.373	27.44		3.14	1.666	.531			1500 psi	1950
Inboard Slats	1.497	.748	12.490		1.65	1.1561	.701	Lock	Lock	1200 psi	1100
Outboard Slats	1.497	.749	17.84		1.65	1.209	.732	Lock	Lock	1200 psi	1100
Wing Fold	2.50	1.185	15.29		4.909	3.815	.776			2600 psi	9800
Wing Hinge Pin	.998	.499	5.31		.782	.5865	.750			2600 psi	1145
Arrest Gear	1.74	.748	9.80		2.378	1.939	.815		Dash Pot	1050 psi	1650
Gun Charge	1.00		10.0		.785						
Barrier Mast	.625	.249	2.18		.307	.258	.841				792
Ailevator Change	2.38	1.75	1.25		4.49	2.08	.446				13300
Wing C.	4.273	1.373	3.65		14.32	13.85	.967			3000 psi	40500
										F7U-1 F7U-3	
Rudder Yaw Rate	1.125	.686			1.03	.640	.621				1800
Canopy	2.00	1.375			3.14	1.646	.524	3 Int. Locks			2075
40 G Lock and Track Unlock	.500	.311			.1963	.120	.611			1000 psi	25
Nose Gear Door	1.875	1.00			2.75	1.965	.714	Lock			4875
Nose Gear	3.25	2.50			8.30	3.39	.408	Lock	Lock		4500
Arrest Gear	2.125	1.00			3.53	2.745	.778	Internal Dashpot Gage and Reservoir			7320
Wing Fold	3.00	1.50			7.068	5.30	.750	Int. Seq. Valve			16100
Outer Panel Slat	1.50	1.00			1.767	.982	.556	Lock			2740
Power Control	4.50	1.75			15.9	13.49	.848				88170
Main Gear	2.122	1.123			3.53	2.54	.720				6500
Wing Pin Pull	1.25	.624			1.227	.921	.750				1820
Speed Brake	4.50	1.375			15.9	14.43	.908				42700
Main Gear Door	1.25	.500			1.227	1.031	.842				440
Center Sec. Slat	1.50	.750			1.767	1.325	.750				
Rocket Pack Door	1.38	.625			1.496	1.189	.795				3080

CYLINDER DATA

CONVAIR

NAME	Pist Dia	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Rudder (3000 psi)	1.126	.623	4.35		.6804	.6804	1.00	Tandem		XFY-1	2040
Elevon (3000 psi)	2.126	1.125	7.59		2.5778	2.5778	1.00	Tandem		XFY-1	7720
Nose Gear Door	1.00	.75	.720		.785	.343	.437	Ball Lock	Ball Lock	XF-102	2355
Nose Gear	1.562	1.062	10.50		1.911	1.029	.537	Snubber	Snubber		5730
Main Gear	2.00	1.00	12.56		3.14	2.355	.727	Snubber	Snubber		9410
Inboard, Outboard Elevon	2.250	1.062	6.45		3.976	3.084	.777	Tandem			11920
Lower Speed Brake	2.25	1.00	7.45		3.976	3.191	.802	Ball Lock	Ball Lock		11920
Upper Speed Brake	1.75	1.00	6.28		2.405	1.620	.673	Ball Lock	Ball Lock		7210
Rudder	1.625	.875	4.030		2.061	1.460	.708	Tandem		XF-102	6180
Elevon Inboard	2.00	1.125	7.45		2.1548	2.1548	1.00	Tandem		XF2Y-1	6460
Elevon Outboard	1.681	1.00	7.35		1.432	1.432	1.00	Tandem			4300
Engine Air Inlet Duct	1.375	.75	7.2		1.474	1.032	.701				4420
Rudder	1.375	.875	4.26		.873	.873	1.00	Tandem			2620
Ski Actuating Fwd.	2.125	1.000	8.1		3.53	2.745	.756				10600
Ski Actuating Aft	2.625	1.125	8.5		5.39	4.4048	.817				16180
Ski Down Lock	.625	.375	1.00		.3068	.1964	.640				920
Water Rudder, Dive Brake	2.75	2.25	8.00		5.94	3.976	.668			XF2Y-1	17800

CYLINDER DATA

DOUGLAS (EL SEGUNDO)

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Arrest Hook Bumper	.500	.312	2.00	7.562	.1960	.1095	.558	Spring Loaded		A3D	587
Nose Wheel Door Latch	.625	.213	.562	4.25	.3068	.2300	.749			XFLD-1	920
Nose Gear Dwn Latch	.625	.375	3.187	6.937	.3068	.1963	.64			XF3D-1	920
Nose Gear Door Lock	.625	.312	.50	4.25	.3068	.2300	.749			F3D	920
Arrest Hook Centering	.750	.375	1.062	no end bearing	.444	.334	.753			A3D	1330
Nose Gear Link Lock	.875	.50	4.0	3.969*	.6013	.4053	.675	Ext. Spring		A3D	1805
Main Gear Bungee	.875	.437	2.593	15.937	.6013	.4513	.750	Ext. Spring		XFLD-1	1805
Cockpit Enclosure	.875	.750	23.187	31.625	.6013	.1591	.265			XBT2D-1	1805
Landing Flap	.875	.625	8.0	12.281	.6013	.2945	.490			XBT2D-1	1805
Cockpit Enclosure	.875	.750	25.25	33.25	.6013	.1595	.265			AD-2-3	1805
Cockpit Enclosure	.875	.750	24.437	34.0	.6013	.1595	.265			AD-2-3-4	1805
Landing Flap	.812	.625	8.0	12.281	.5185	.2117	.408			XBT2D-1	1555
Main Gear Link Latch	1.0	.50	1.25	17.187	.7854	.5890	.749			XFLD-1	2000
Landing Flap	1.0	.625	8.0	12.25	.7854	.4786	.608			XBT2D-1 AD-1-2	2000
Main Gear Latch	1.00	.500	.312	6.0	.7854	.4784	.608	Spring Loaded		XF3D-1 F3D-1-2	2000
Landing Flap	1.0	.625	6.25	10.562	.7854	.4786	.608			F3D	2000
Main Gear Door Latch	1.062	.500	1.00	5.125	.8866	.6902	.780			XFLD-1,A3D	2660
Landing Flap	1.062	.562	6.75	10.75	.8866	.6381	.720			A2D-1,A3D	2660
Nose Gear Dwn. Latch	1.125	.312	.750	6.0	.9940	.9173	.923	Spring Return		XFLD-1	2980
Emerg. Enclosure	1.125	.875	8.375	15.937	.9940	.3927	.395			AD-5	2980
Nose Gear	1.187	.750	8.250	13.687	1.1075	.6657	.601			F3D-1-2	3320
Nose Wheel Steer	1.250	.562	6.781	no end bearing	1.2272	.9772	.796			A3D	3680
Tail Bumper	1.250	.625	4.937	11.125	1.2272	.9204	.750	Internal Spring		A3D	3680
Tail Wheel	1.250	.875	12.937	20.250	1.2272	.6258	.511			A2D-1	3680
Main Gear Door Latch	1.312	.500	1.750	4.750	1.397	1.356	.866			XA2D-1,F3D	4080
Main Gear Door	1.343	.625	5.50	8.562	1.4180	1.1114	.784			BT2D-1	4250
Main Gear Door	1.375	.625	5.50	7.812	1.4849	1.1781	.747			XBT2D-1	4450
Yaw Damper	1.437	.312	2.50	9.625	1.5462	1.5462	1.00			F3D-3	4630
Yaw Damper	1.437	.375	4.0	12.50	1.6230	1.5125	.931			XFLD-1	4860

CYLINDER DATA (Cont'd)
DOUGLAS (EL SEGUNDO)

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Spoiler	1.437	.50	2.625	6.125	1.623	1.4266	.877			F3D-2	4870
Main Gear Door	1.437	.50	5.125	8.687	1.623	1.4266	.877			F3D-1-2	4870
										XF4D-1	
Speed Brake	1.500	.750	8.125	13.250	1.767	1.3253	.750			XF4D-1	5290
Speed Brake	1.500	1.0	17.0	24.0	1.767	.9817	.556			F3D-1	5290
Main Gear Door	1.500	.625	6.187	10.343	1.767	1.4603	.830			AA2D-1	5290
Main Gear Door	1.531	.500	4.969	8.593	1.8415	1.6451	.892			XF4D-1	5520
Main Gear Door	1.562	.750	7.562	7.562	1.9175	1.4745	.770			AD-2	5750
Main Gear Door	1.562	1.062	18.125	23.750	1.9175	1.0309	.538			A3D	5750
Main Gear Latch	1.625	.500	1.250	10.375	2.0700	1.8740	.906	Ext. Spring		F3D-3	6210
Bomb Bay Fwd. Door	1.625	.750	7.250	14.625	2.0700	1.6260	.785			A3D	6210
Speed Brake	1.625	1.00	20.25	27.25	2.0700	1.284	.620			F3D-1-2	6210
Nose Gear Door Latch	1.687	.375	3.187	8.656	2.2365	2.1260	.955			F3D-1-2-3	6700
Tail Wheel	1.750	1.375	19.375	26.625	2.4053	.9204	.382			XBT2D-1, AD-1-2	7210
Tail Wheel	1.750	1.375	19.375	26.625	2.4053	.9204	.382			AD-3-4	7210
Main Gear Latch	1.875	.625	1.00	10.093	2.7600	2.3532	.888			A3D	8280
Nose Gear	1.875	.875	8.250	18.0	2.7600	2.1687	.783			XF4D-1	8280
Ail. Mech. Advantage Shift	2.00	.562	2.437	11.00	3.1416	2.8966	.920	Finger Lock		F3D-1-2	9410
Nose Gear	2.00	.662	6.781	17.875	3.1416	2.8931	.920			F3D-1-2	9410
Speed Brake	2.00	1.50	25.125	34.0	3.1416	1.3745	.438			F3D	9410
Main Gear	2.125	1.00	8.0	19.312	3.5466	2.7612	.780			XF3D-1	10630
Tail Skid	2.125	1.25	7.250	21.125	3.5466	2.3194	.653			XF3D-1	10630
Main Landing Gear	2.125	1.375	11.375	24.750	3.5466	2.0617	.581			XBT2D-1	10630
Nose Gear	2.250	1.062	8.250	15.125	3.9761	3.0895	.778			A3D	11900
Main Gear	2.250	1.062	8.500	15.50	3.9761	3.0895	.778			F4D-1	11900
Main Gear	2.250	1.375	12.875	19.375	3.9761	2.4912	.628			AD-1	11900
Main Gear	2.250	1.375	12.437	21.750	3.9761	2.9912	.628			XBT2D-1	11900
Main Gear	2.250	1.375	13.156	19.375	3.9761	2.4912	.628			XBT2D-1	11900
										AD-1-2	11900
Main Gear	2.25	1.000	13.156	19.50	3.9761	3.1907	.803			AD-3-4	11900
Wing Fold	2.375	.875	5.750	12.687	4.4301	2.8288	.639			XF4D-1	13300
Wing Flap	2.375	1.00	6.562	14.687	4.4301	3.6447	.823			A3D	13300
Side Dive Brake	2.375	1.750	8.125	17.437	4.4301	2.0248	.456			XBT2D-1	13300
										AD-1-2	13300

CYLINDER DATA (cont'd)
DOUGLAS (EL SEGUNDO)

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Main Gear	2.437	1.062	9.343	19.250	4.6664	3.7798	.810			F3D-1-2	13300
Bomb Bay Rear Door	2.500	1.000	7.250	13.656	4.9087	4.1213	.841			A3D	14730
Lower Dive Brake	2.500	1.625	17.687	24.375	4.9087	2.8348	.576	Lock		XBT2D-1	
										BT2D-1	14730
Lower Dive Brake	2.500	1.625	17.781	24.375	4.9087	2.8348	.576	Lock		AD-	14730
Speed Brake	2.500	1.500	21.750	29.500	4.9087	3.1416	.640	Lock		A2D-1	14730
Main Gear	2.500	1.875	35.562	43.437	4.9087	2.1487	.438			A3D	14730
Escape Chute Door	2.625	1.375	1.937	10.187	5.4119	3.927	.725	Single Acting		F3D-1	16230
Main Landing	2.625	1.875	11.406	16.375	5.4119	2.6507	.490			A2D-1	16230
Spoiler	2.625	2.00	4.00	10.437	5.4119	2.2703	.420			F3D-3	16230
Speed Brake	2.625	1.750	18.250	26.562	6.5000	4.0947	.630			A3D	19500
Fin Fold	3.000	1.250	8.437	17.062	7.0686	5.8414	.828			XBT2D-1	21200
Wing Fold	3.000	1.500	14.000	21.750	7.0686	5.3015	.751			[illegible]T2D-1	21200
Wing Fold	3.250	1.375	12.125	19.250	8.2958	6.8109	.822			F3D	24850
Wing Fold	3.250	1.500	12.125	19.250	8.2958	6.5287	.787			F3D-1-2,A2D-1	24850
Wing Fold	3.250	1.375	10.937	20.437	8.2958	6.8109	.822			XA2D-1	24850
Wing Fold	3.500	1.625	10.937	21.750	9.6211	7.5472	.784			AD	28850
Wing Fold	3.625	1.500	10.812	20.312	10.700	8.9392	.834			F3D-3	32100
Wing Fold	4.375	1.625	12.437	23.750	15.100	13.030	.864			A3D	45300
Wing Fold	4.375	1.875	11.250	21.000	15.100	12.340	.818			A2D-1	45300

CYLINDER DATA

DOUGLAS (SANTA MONICA)

NAME	Pist Dia	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Lower Cargo Door	2.00	1.25	15.5		3.14	1.913	.610			YKC-124B	9410
Main Gear	6.00	2.125			28.27	24.74	.874	Swivel Joints			83500
Nose Gear	3.25	2.50	15.6		8.30	3.39	.408				24900
Nose Gear Stear	4.375	1.375			15.0	13.53	.903				45000
Down Latch	1.000	.623	1.00		.785	.480	.611				2355
Jam Latch	1.000	.625	3.48		.785	.479	.610				2355
Center Latch	.937	.562	10.70		.689	.441	.640				2065
Nose Load Door	2.000	1.25	12.5		3.14	1.913	.609				9410
Ramp Up Latch	.500	.312	1.375		.1963	.1203	.613				589
Nose Load Ret.	1.500	.375	180.0		1.767	1.657	.938				5300
Fwd. Ramp Ext.	1.375	1.250	27.9		1.474	.247	.1675				4340
Ramp Latch Adj.	1.00	.437	3.2		.785	.635	.808			YKC-124B	2355

CYLINDER DATA

THE GLENN L. MARTIN COMPANY

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Flare	Load
Canopy Latch	.750	.500	5.00	11.750	.4417	.2453	.555			B-57B	1135
Hydro Flap Latch	.812	.500	3.50		.5178	.322	.621		2700 psi	PSM-1-2	1400
Nose Gear Latch Pin	.836	.498	2.50	5.375	.5489	.354	.645			PLM-1	1645
Hatch Snubbing	.873	.737	6.6	14.9	.5986	.1716	.1278			PSM-1-2	1795
Sonobouy Door	.875	.500	2.00	8.593	.6013	.405	.623			PSM-1-2	1803
Tail Bumper	1.000	.500		16.42	.785	.589	.750			PLM-1	2355
Pilots Seat	1.000	.500	6.25		.785	.599	.762			XB-51	2355
Aft Gear Door Lock	1.000	.375	5.00		.785	.675	.860			XB-51	2355
Aileron Quad Stop	1.000	.937	.625		.785	.095	.121			PSM-1-2	2355
Rudder Boost	1.063	.750	3.625	15.218	.440	.440	.498			PSM-2	1320
Jet Doors	1.116	.498	15.0	23.4	.985	.790	.801			PLM-1	2950
Nose Gear Door	1.125	.75	8.94	16.21	.9940	.5522	.556			B-57B	1180
Center Wing Flap	1.125	.809	12.25	20.625	.985	.471	.478			PLM-1	2950
Main Gear Door	1.125	.75	9.14	16.21	.9940	.5522	.556			B-57B	800
Elev. Boost	1.183	.625	3.437		.787	.787	1.00			XB-51	1180
Stabilizer	1.312	.4375	4.437	10.912	1.202	1.202	1.00	Balanced		B-61	1800
Aileron Boost	1.375	.685	3.062	15.688	1.106	1.106	1.00			404	1661
Elevator Boost	1.375	.750	4.00	19.00	1.03	1.03	1.00			PSM-2	3090
Demand Assist.	1.438	.799	7.187		1.629	1.188	.730			PEM-5A	2440
Bomb Bay Door	1.500	1.00	13.437	21.218	1.767	.982	.556		2700 psi	PSM-1-2	4750
Canopy	1.500	1.25	31.29	41.56	1.767	.540	.306			B-57B	2860
Aft Ramp	1.5	.745	8.75	14.0	1.767	1.33	.753			404	5300

Rudder Boost	1.563	.687	6.718	14.7	1.5303	1.5303	1.00		680 psi	PSM-1	1050
Elev. Boost	1.563	.687	6.718	14.7	1.5303	1.5303	1.00		1720 psi	PSM-1	2660
Dive Brake	1.625	.75	2.00	22.919	2.0739	1.6321	.787			B-57B	1000
Nose Gear	1.56	1.122	16.187	27.75	1.911	.926	.484			PLM-1	5730
Spoiler	1.563	.688	5.375	24.382	1.5303	1.5305	1.00			XB-51	2870
Nose Gear	1.625	1.00	8.662	24.8	2.0739	1.2285	.621			B-57B	1400
Wing Glap	1.687	1.125	14.797		2.243	1.258	.560		2700 psi	PSM-1-2	6050
Dive Flap	1.687	1.375	16.688		2.243	.769	.312			XB-51	6730
Landing Gear	2.00	.999	18.9	26.74	3.14	2.355	.750	Snubber	Snubber	404	9420
Wing Flap	2.00	.75	6.72	14.156	3.14	2.698	.858			404	9420
Nose Gear	2.00	.999	15.437	23.562	3.14	2.355	.750	Snubber	Snubber	404	9420
Tip Gear	2.00	.810	7.687		3.14	2.625	.836			XB-51	9420
Aft Gear Surge	2.00	1	1.875		3.14					XB-51	9420
Speed Brake	2.125	1.250	20.25	30.25	3.5466	2.319	.654			B-57B	6400
Spoiler Ail	2.248	.747	7.11	14.76	3.538	3.538	1.00			PLM-1	11920
Wing Flap Cyl	2.375	1.250	3.90	18.30	3.2029	3.2029	1.00	Balanced		B-57B	11300
Spoiler Aileron	2.250	.750	8.07	17.37	3.534	3.534	1.00			PSM-1-2	11910
Bomb Door	2.250	1.247	15.8	28.2	3.976	2.749	.693			PLM-1	11910
Hydro-Flap	2.375	1.750		15.00	4.439	2.025	.457			PSM-1-2	13300
Main Gear Down Lock	2.500	1.375	3.844	30.5	4.909	3.435	.700			PBM-5A	7360
Main Gear Uplock	2.500	.625	2.063		4.909	4.602	.938			PBM-5A	7360
Fwd. Gear	2.500	1.25	17.61		4.909	3.682	.750			XB-51	14720
Nose Gear	2.750		8.625		5.940	5.155				PBM-5A	17800

CYLINDER DATA

The Glenn L. Martin Company

NAME	Pist. Dia.	Rod Dia	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Aft Gear	2.750	1.25	13.07		5.940	4.713	.793			XB-51	17800
Main Gear	2.750	1.250	10.73	20.63	5.4396	4.7121	.868			B-57B	8700
Bomb Door	3.00	2.125	18.312		7.069	3.539	.500			XB-51	21200
Bomb Door	3.00	2.125	18.312	34.1875	7.0686	3.522	.498			B-57B	11300
Steer	3.25	1.88	8.375		8.296	5.520	.667			XB-51	24850
Nose Gear Steer	3.38	1.96	4.95		8.973	8.091	.901			404	26900
Nose Steer	4.25	1.0	4.25		14.19	13.315	.939			PBM-5A	21250
Landing Gear	4.375	1.50	17.0		15.0	13.233	.883			PBM-5A	22500

CYLINDER DATA
GRUMMAN

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Canopy	1.50		25.6		1.767	.982	.556			F9F-4-5	2645
Nose Wheel	1.667		5.781		2.237	1.795	.802				3355
Dive Brake	3.125		10.625		7.67	5.902	.770				11650
Inboard Flap	1.313		3.984		1.353	.911	.672	Ball Lock			2030
Outboard Flap	1.750		2.937		2.405	1.804	.749	Ball Lock			3610
Aileron Boost	2.875		4.937		3.36	3.36	1.00	Balanced Rod Each Rod			5040
Wing Lock	1.875		4.925		2.098	1.052	.502				3140
Seat Adjust	1.312		4.00		1.353	.911	.672				2030
Main Gear	1.500		6.063		1.767	.982	.556	Ball Lock			2645
Inboard L.E.	1.125		1.554		1.994	.687	.691	Ball Lock			1495
Main Gear Uplock	1.00		2.625		.785	.635	.809				1178
Main Gear	3.25		10.187		4.750	1.230	.259		Ball Lock		7120
Mech. Aileron Boost	.750		3.00		.441						661
Outboard L.E.	1.125		1.093		.994	.687	.691				1840
Arresting Hook	1.125		3.562		1.227						
Wing Fold	4.875		7.906		18.66	15.52	.832				2800
Nose Wheel	1.375		7.812		1.485	1.042	.701			F9F-4-5	2230

CYLINDER DATA

LOCKHEED

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Jury Strut	.785	.5	7.0		.484	.2877	.593			P2V-7	1454
Nose Gear	2.375	1.375	8.16		4.412	2.938	.666				13200
Spoiler	1.375	.875	4.52		2.75	2.1488	.778				8250
Main Gear Uplock	.375	.375	5.3		.6012	.4812	.80				1803
Main Gear	2.875	1.5	22.37		6.469	4.702	.727				19400
Pod Actuating	1.0	.5	3.5		.785	.5887	.750				2360
Main Gear Jury	1.25	.811	.344		1.227	.7104	.579			P2V-7	3560
Bomb Door Motor											
Wing Flap Motor											

TABLE B-16
B-47

CYLINDER DATA
MCDONNELL

NAME	Pist Dia.	Stroke		Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
			10.00		.765	.434	.552			F101	3530
			1.90		.371	.174	.469				1670
			2.78		1.108	.912	.825				4980
			1.88		1.108	.912	.825				4980
			12.72		8.946	6.541	.730				40200
			13.14		4.43	3.322	.750				19900
			2.1[illegible]		.887	.691	.780				4000
			23.88		7.069	3.522	.499				31900
			9.95		2.237	1.795	.803			F101	10500
Wing Fold	4	1.62	10.50		12.57	10.50	.836			F2H-3	18830
Elev. Feel Damper	----	----	3.00		3.55	3.55	1.00				4500
Aileron Ratio Change	----	----	2.50		.88					F2H-3	3750
Speed Brake	2.375	1.5	20.13		4.430	2.663	.601			F3H-1	13500
Nose Gear	2.25	1.5	16.26		3.976	2.209	.556				11940
Main Gear Uplock	.875	.500	2.90		.601	.405	.673				1803
Aileron	----	----	4.77		4.180	4.180	1.00	Valve in Piston			25086
Elevator	----	----	2.80		.490	.490	1.00	Valve in Piston			1475
Rudder	.735	.368	.92		.422	.331	.749				990 Tension
Rudder Feel	2.00	1.375	8 .44		3.141	1.651	.527				9420
Main Gear	1.125	.625	5.86		.994	.685	.690				2982
Main Gear Door	2.750	1.000	7.08		5.94	5.155	.868				17800
Tail Skid	1.250	.875	11.28		1.227	.626	.511				2900
Flap	1.625	.750	4.75		2.074	1.632	.787				9040
Arrest Gear	2.375	1.250	9.68		4.43	3.203	.723				13250
Wing Fold	1.482	.740	2.85		1.727	.920	.750				3690
Eng. Aux. Air Door	.625	.375	1.10		.309	.198	.641				918
Stab.-Hyd. Motor										F3H-1	14750
Slat-Hyd. Motor											260 in lb.

CYLINDER DATA
NORTH AMERICAN

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Aux G.D. Fairing Lock	.625	.375	1.875		.3068	.1968	.641			FJ-2	920
Aux G.D. Fairing	1.25	.625	5.187		1.227	.9202	.751			1000 psi	1227
Aux Gear	2.00	1.000	6.212		3.14	2.355	.750				9410
Aux Gear Lock	1.562	.500	.938		1.911	1.715	.897				5730
Main Gear Fairing Up Lock	.625	.375	.750		.3068	.1968	.641				920
Main Gear Fairing Door	1.188	.688	6.688		1.112	.740	.665				3340
Main Gear	3.00	1.250	5.220		7.068	5.84	.827				21200
Main Gear Up, Down Lock	1.75	.875	1.828		2.405	1.804	.765				7210
Tail Bumper	1.00	.620	5.500		.785	.483	.616				2355
Arresting Hook	2.25	1.500	8.875		3.976	2.21	.556				11930
Aux. Gear Strut											
Speed Brake	1.562	1.000	11.90		1.911	1.126	.588				5730
Wing Fold	2.875	1.187	7.250		6.47	5.358	.828				19400
Wing Lock Pin	1.875	1.062	2.375		2.75	1.868	.679				8250
Power Master Brake	1.250	.375	2.30		1.227	1.117	.911				3680
Gun Bay Purge Door	.688	.500	2.875		3.72	3.535	.950			FJ-2	1150
Rudder Cyl	1.479	.997	4.499		.939	.939	1.00	Tandem		XA2J-1	2810
Stabilizer	2.438	1.497	9.736		2.932	2.932	1.00	Tandem			8790
Aileron	1.70	1.059	3.078		1.388	1.388	1.00	Tandem			4160
Outboard Flap	2.001	.997	7.470		2.349	2.349	1.00	Balanced Piston		XA2J-1	7050
Aileron Cylinder	2.144	1.122	1.448		1.81	1.81	1.00			F-86H	5430
Stabilizer	2.362	.997	3.607		1.00	1.00	1.00				3000
Main Gear Door	1.312	.687	5.313		1.355	.983	.725				5070
Main Gear Uplock	.75	.375	1.00		.44	.33	.750				1320
Main Gear	1.625	.812	5.779		2.075	1.555	.750				6220
Main Gear Down Lock	.687	.312	1.250		.368	.296	.804	Seq. Poppet			1105
Aux. Alight Gear Down Lock	.625	.375	1.500		.307	.20	.651				920
Aux. Alight Gear	1.562	.809	7.784		1.915	1.402	.733				5740
Aux. Alight Gear Lock	.872	.375	2.00		.52	.41	.789				1560
Aux. Gear Door	1.375	.687	3.375		1.49	1.118	.750				4470
Dive Brake	1.75	1.125	9.078		2.41	1.412	.786			F-86H	7230

CYLINDER DATA
NORTHROP

NAME	Pist. Dia.	Rod Dia.	Stroke	Retract Length	Head Area	Rod End Area	Area Ratio	Access Head	Access Rod End	Plane	Loads
Main Gear	2.626	1.25	7.27		5.43	4.203	.775			F-89D	16300
Main Gear Door	1.626	.75	4.4		2.09	1.648	.778				6270
Nose Gear	2.126	1.5	11.52		3.56	1.793	.504				10680
Rudder	1.126	.81	4.50		.488	.488	1.00	Balanced			1463
Elevator	1.688	1.25	3.68		1.016	1.016	1.00	Balanced			3045
Aileron	1.876	1.00	3.03		1.991	1.991	1.00	Balanced			5980
Dive Brake	3.000	1.12	5.98		6.083	6.083	1.00	Balanced			18230
Engine Hoist	1.876	.75	10.0		2.776	2.334	.841				8320
Seat Adjust	.9375	.62	7.2		.6896	.3876	.562	Spring Return			2065
Main Gear Bungee	2.001	1.00	3.93		3.14	2.355	.750	Single Acting			9410
Enclosure Jetis Assist.	2.126	.62	8.87		3.56	3.258	.914				10680
Radar Scope and Console	.937	.62	9.91		.6896	.3876	.562			F-89D	2065

CYLINDER DATA
REPUBLIC

TABLE B-19
B-50

NAME	Pist. Dia.	Rod Dia.	Stroke	Head Area	Rod Side Area	Area Ratio	Access Head	Access Rod End	Plane	Load
Nose Gear Lock	1.50		.604	1.765			Single Acting		F84-F	
Main Gear Retract	1.13	.875	7.92	1.00	.402	.402				1500
Main Gear Uplock	1.00	.500	4.56	.785	.589	.750	Ball Lock and Switch			
Main Gear Downlock	1.50		1.00	1.765			Built in Check Sequence			2645
Main Gear Out Lock	1.25	.500	1.30	1.225	1.029	.840	Single Acting			1838
Wing Slat	1.00	.375	8.58	.785	.675	.859				
Wing Flap	2.88	2.00	2.25	6.51	3.37	.517				9760
Nose Gear Uplock	1.00		.512	.785						1178
Ifr. Nozzle Latch	1.25	.500	2.25	1.225	.840	.919				1838
Nose Gear	2.75	.875	3.92	5.94	5.25	.883				8900
Duct Screen	1.00	.750	3.50	.785	.437	.557				1178
Aileron Power	2.125	.750	4.33	3.55	3.118	1.00				5340
Jato Release	.875	.312	2.62	.601	.525	.673				901
Elevator Power	1.904	.750	3.39	2.85	.845	1.00				4240
Rudder Power	1.211	.625	2.02	1.163	1.163	1.00				2205
Speed Brake	2.125	1.250	9.87	3.55	3.35	.654				5340
Ifr. Release	.937	.500	2.98	.689	.493	.716			F84-F	1034

APPENDIX C

VALVE AND FILTER WEIGHT AND SPACE

APPENDIX C

VALVE WEIGHT AND SPACE

In this section, mathematical expressions for check valve weight and space were obtained. Other valves have been considered as multiples of the check valve weight and space for equivalent horsepower rating.

The check valve weight was obtained by considering, as in other sections, that the part in question consisted of an assembly of several simpler parts. The ends of the check valve appear as being quite similar in design and construction to an "AN-815" union. The barrel part is essentially a section of tubing. The internal parts like poppet and spring are also similar in shape and weight to line fittings. The weight equation became basically:

$$W_{CV} = K_1 W_F + K_2 W_{TUBING} \qquad (C1)$$

Where:

W_{CV} = check valve weight (LB)

W_F = Fitting weight (LB)

W_{TUBING} = Tubing weight (LB)

K_1 = Ratio of theoretical fitting weight to actual fitting weight

K_2 = Ratio of theoretical tubing weight to actual barrel weight

The fitting weight has been expressed in Appendix "A" as:

$$W_F = \frac{2.98\ (10^{-5})D^3P^2}{36000 + P} + 8.5(10^{-7})DP + 4.87(10^{-3})D \qquad (C2)$$

A check valve was disassembled and weighed and from this, it was established that the fitting portion which consists of poppet, spring, and both fitting ends, weighs 0.3461 lbs. For the one half inch tube size, used for 3000 psi, and the barrel weight which was actually a straight tube, was found to be 0.1411 lbs.

If the criteria that D = 0.5 in. and P = 3000 psi is introduced in equation C1, W_F becomes equal to 0.004569 lb. Accordingly:

$$K_1 = \frac{W_{CV} - W_{BARREL}}{0.004569} = \frac{.3461}{0.004569} = 75.7 \qquad (C3)$$

To obtain a value for K_2, it was necessary to find an equation for the check valve body weight. Since the body is essentially a piece of straight tubing, its basic equation is:

$$W_{TUBING} = \frac{\pi}{4} (D_{CV}^2 - d_{cv}^2) L \rho \qquad (C4)$$

Where:

D_{CV} = Outside barrel diameter (IN)

d_{cv} = Inside body diameter (IN)

L = Barrel length (IN)

ρ = Material weight (LB/IN^3)

It was observed from a plot of "L" versus "D_{CV}," for all sizes of check valves:

$$L = 2.00\ D_{CV} \qquad (C5)$$

From Lame's relationship for thick wall cylinders:

$$d_{cv}^2 = D_{CV}^2 \left(\frac{S_t - P}{S_t + P}\right) \qquad (C6)$$

From a plot of "D_{CV}" versus "D": (Tubo Size)

$$D_{CV} = 1.463\ D + .3244 \qquad (C7)$$

The use of material according to Specification MIL-S-6758, Condition F, was assumed and so:

$$\rho = .283\ LB/IN^3 \qquad (C8)$$

$$S_t = 180{,}000\ LB/IN^2 \qquad (C9)$$

When equations C5 through C9 are introduced into equation C4:

$$W_{TUBING} = .0278\ (4.69D + 1)^3 \left(\frac{P}{36000 + P}\right) \qquad (C10)$$

which includes a safety factor of 5.

When D = .500 inch and P = 3000 psi, then W_{BARREL} (by equation C10) is equal to 0.0804 LB.

Actually, as was stated earlier, the barrel was 0.1411 LB, accordingly,

$$K_2 = \frac{0.1411}{0.0804} = 1.755 \qquad (C11)$$

When equations C2, C3, C10, and C11 are inserted in equation C1, the total check valve weight becomes:

$$W_{CV} = \frac{.002256D^3P^2}{36000 + P} + .0000643DP + .369D + .0488(4.69D + 1)^3 + \frac{P}{36000 + P} \qquad (C12)$$

Where:

W_{CV} = Check valve weight (LB)

D = Tubing size (IN)

P = System Pressure (LB/IN^2)

For the space relationship the check valve was considered to be a cylindrical can of an outside diameter equal to the dimension across corners of the hexagon ends, and of a length equal to the check valve length less the threaded ends. (Since these ends are assembled in tubing during use).

By equation C5 the check valve length is stated as:

$$L = 2.09 \; D_{CV} \tag{C13}$$

The relation between the tubing diameter (D) and the dimension across corners (D_{BS}) is:

$$D_{BS} = 1.69 \; D + .375 \tag{C14}$$

Basically, the space volume is equal to:

$$V_S = \frac{\pi}{4} \; D_{BS}^2 \; L \tag{C15}$$

When equation C13 and C14 are substituted into equation C15, the final result is:

$$V_S = 6.9 \; (D + .222)^3 \tag{C16}$$

Where:

V_S = Space volume (IN^3)

D = Tubing size (IN)

Equations C12 and C16 appear plotted in figures C1 and C2 respectively.

Other valves which were considered were, relief valves, servo valves, shuttle valves, solenoid valves, and disconnects. (Both bulkhead station and line to line connection types). The following method was considered advisable since all valves can be considered as being composed of fittings and pressure containers.

For valves which are of the standard "AN" type, such as relief valves, shuttle valves, and disconnects, weight data as listed in "AN" standards was utilized by obtaining a ratio between the respective valve weight and a check valve weight at the same horsepower. This ratio was termed a "multiplier," and they are listed in Table C3. Space was treated similarly except that space was considered to be the volume of the smallest sized box which could contain the valve.

For the more complicated and less conventional type valves such as servo and solenoid control valves, many actual valves from current airplanes were investigated and the observations were tabulated. (See Table C2). The respective weights and space volumes were averaged and those averages were used in a similar manner as the previous types.

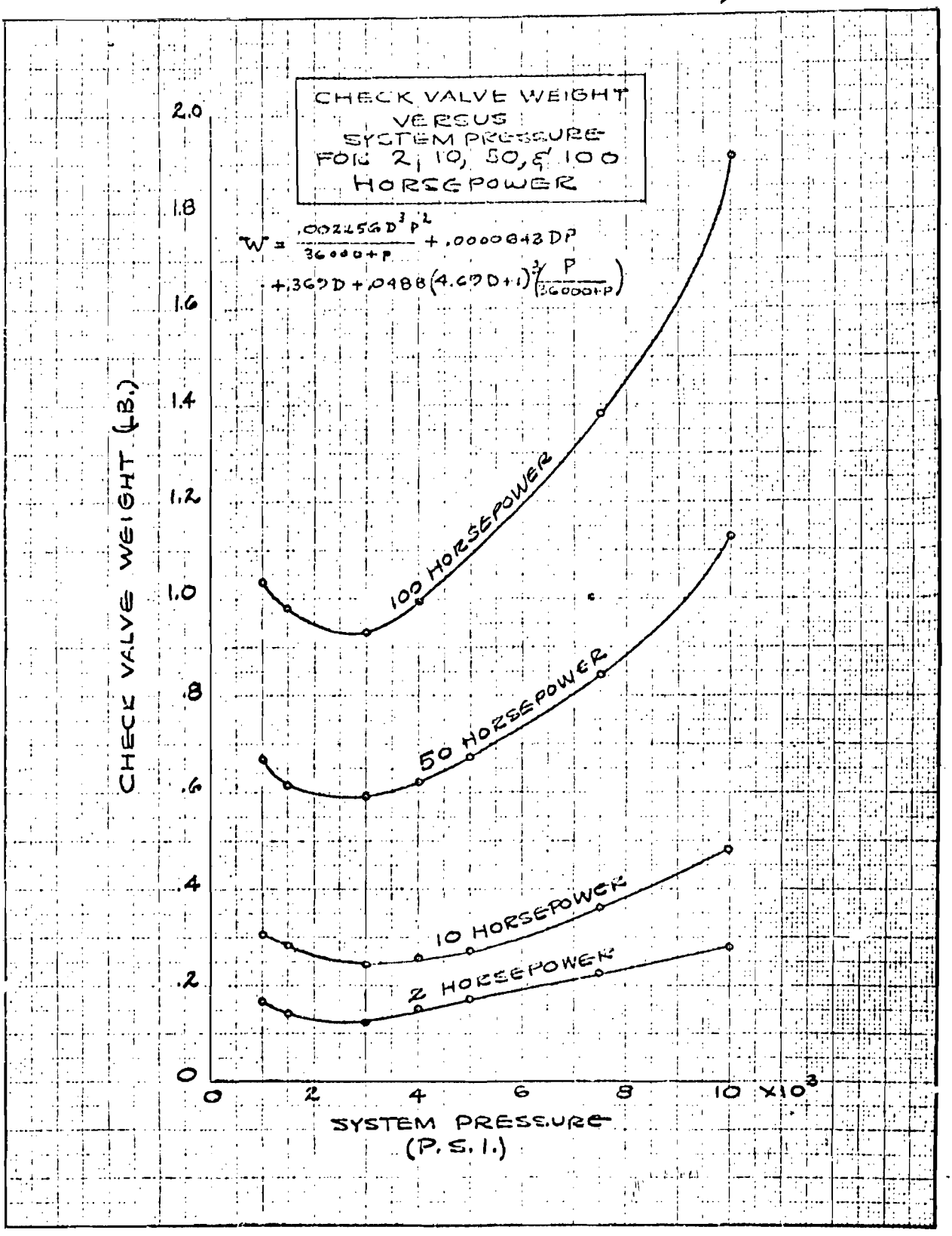
P	HP	$D \sqrt[d]{\frac{S_t + P}{S_t - P}}$	CHECK VALVE WEIGHT	CHECK VALVE SPACE
1000	2	.313	.1699	1.056
	10	.548	.3073	3.16
	50	.946	.6708	11.05
	100	1.219	1.0379	20.20
1500	2	.216	.1446	.780
	10	.441	.2731	2.01
	50	.773	.6157	6.80
	100	1.01	.9804	12.82
3000	2	.198	.1235	.511
	10	.313	.2483	1.056
	50	.566	.5933	3.38
	100	.728	.9332	5.92
4000	2	.1875	.1512	.475
	10	.281	.2565	.876
	50	.506	.6206	2.66
	100	.656	.9972	4.67
5000	2	.1875	.1724	.475
	10	.263	.2733	.786
	50	.474	.6744	2.32
	100	.615	.9758	4.04
7500	2	.1875	.2263	.475
	10	.251	.3614	.731
	50	.435	.8449	1.96
	100	.558	1.3815	3.28
10000	2	.1875	.2817	.475
	10	.275	.4909	.859
	50	.432	1.1295	1.859
	100	.557	1.9145	3.26

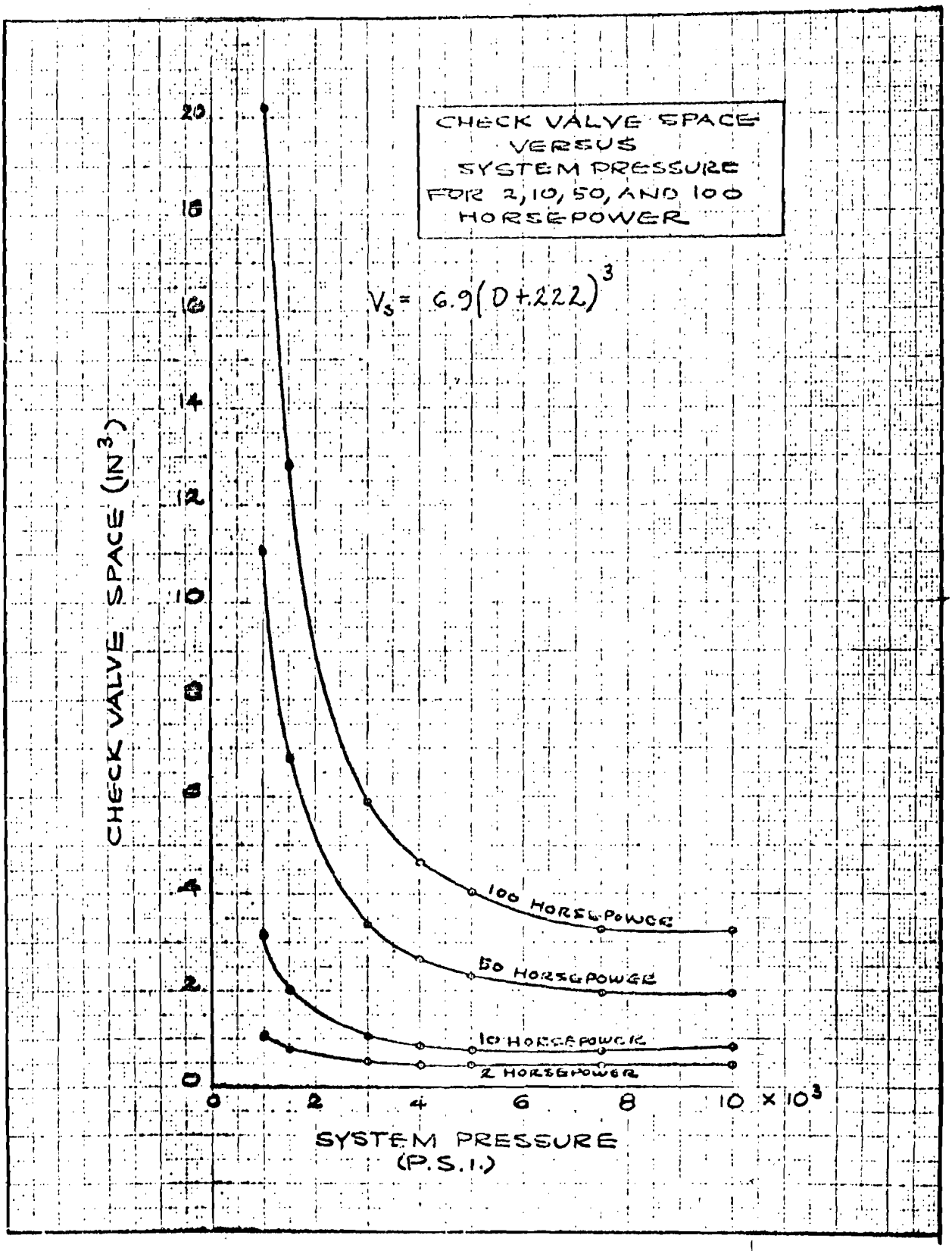
CO.	MFGR. NO.	TYPE	SOLENOID	WAY	POSN	AIRPLANE	USE	WT. LB	FLOW GPM	PRESS PSI	VOL IN3	HP	CHECK VALVE WT LB	CHECK VALVE SPACE IN3	MULT WT	MULT SPACE
Bertea	MFC-6-01	Servo	---	---	---	404	Aileron	2.8	1.73	1500	36.9	1.51	.130	.70	2.15	52.7
Bertea	11800	Selector	Yes	---	---	---	-------	1	3	3000	11.1	5.25	.225	.75	4.45	14.8
G. C. *	AV14B-1128D	Selector	Yes	4	2	---	-------	2	6	3000	133.5	10.5	.285	1.10	7.02	121
G. C.	AV13A-1124	Selector	---	3	2	XP6M-1	Spoiler	1.5	3.5	3000	83.5	6.12	.230	.80	6.52	104
G. C.	AV14C-1153	Selector	---	4	3	XP6M-1	Nose Door	6	34	3000	350.0	59.5	.690	3.90	8.7	89.8
Bendix	548570	Selector	Yes	4	---	---	-------	1.9	6	3000	49.5	10.5	.285	1.10	6.67	45
Hyd-Aire	4952	Selector	Yes	3	---	XP6M-1	Pump Discharge	2.3	6	3000	67.2	10.5	.285	1.10	8.07	60.6
Adel	22982	Selector	Yes	4	3	---	-------	2.1	12	3000	52.9	21	.385	1.70	5.45	3.61
Adel	25036	Selector	Yes	3	2	---	-------	1.5	1	100	24.9	.584	---	---	---	---
G. C.	AV11D-1146	Selector	Yes	3	---	XP6M-1	Pump By-Pass	1.5	1	100	33.4	.584	---	---	---	---
Bendix	550630	Selector	Yes	3	2	XP6M-1	Stabilizer Shut-Off	1.6	13	3000	46.9	22.7	.400	1.80	4	26.1
Hyd-Lowr	XS-180100	Selector	Yes	4	---	XP6M-1	Utility Door	---	50	3000	109.5	87.5	.880	5.35	---	20.5
Hyd-Lowr	XS-18100	Selector	Yes	4	---	XP6M-1	Utility Door	---	20	3000	142.4	35.0	.510	2.60	---	54.8
Bertea	MFC-8-01	Servo	---	---	---	XP6M-1	Hydroflap	3.1	12	3000	81.7	21.0	.385	1.70	8.05	48.1
Bertea	MFC-6-03	Servo	---	---	---	XP6M-1	Rudder Control	1.5	1.5	3000	32.9	2.63	.170	.60	5.7	54.8
Weston	11920	Servo	---	---	---	XP6M-1	Stabilizer	4.85	24	3000	107.0	42.0	.560	2.90	8.66	36.9
Weston	11930	Servo	---	---	---	XP6M-1	Spoiler	3.00	60	3000	240.0	105	1.000	6.20	3	38.7
Weston	11940	Servo	---	---	---	XP6M-1	Hydroflap	2.05	24	3000	89.1	42.0	.560	2.90	3.66	30.7
Weston	11750	Selector	---	4	---	XP6M-1	Wing Flap	2	1.3	3000	48.4	2.27	.160	.55	12.5	88.1
Weston	8440	Selector	Yes	4	---	XP6M-1	Bomb Door	2	20	3000	71.6	35.0	.510	2.60	3.92	27.5
Bendix	551590	Selector	Yes	4	---	---	-------	3.39	16	3000	88.4	28.0	.450	2.20	7.54	40.2
Adel	21700	Selector	Yes	4	---	P5M-2	Flaps-Bomb Door	2.1	20	3000	75.6	35.0	.510	2.60	4.12	29.1
Bertea	MFC-6-03	Servo	---	---	---	P5M-2	Boost	1.438	1.5	3000	32.9	2.63	.170	.60	8.45	54.8
Adel	23386	Selector	Yes	4	3	P5M-2	Hydroflap	2.1	6	3000	75.4	10.5	.285	1.10	7.37	68.5
G. C.	AV14B-1122	Selector	Yes	4	3	---	-------	1.7	3.5	3000	89.6	6.12	.230	.80	7.39	112
G. C.	AV14B-1124	Selector	Yes	4	3	---	-------	1.95	6	3000	116.5	10.5	.285	1.10	6.84	106
G. C.	AV14C-1138	Selector	Yes	4	3	---	-------	1.95	6	3000	116.5	10.5	.285	1.10	6.84	106

*General Controls

TABLE OF MULTIPLIERS

	Weight	Space
Disconnect, in-line	1.538	2.5
Disconnect, Bulkhead	1.338	2.3
Relief Valve	6.5	26
Servo Valve	5.5	45
Shuttle Valve	1.5	9
Solenoid Valve	6.7	64


CHECK VALVE WEIGHT
VERSUS
SYSTEM PRESSURE
FOR 2, 10, 50, & 100
HORSEPOWER
$W = \frac{.00245GD^3P^2}{36000+P} + .0000843DP + .369D + .0488(4.69D+1)^{\frac{3}{2}}\left(\frac{P}{36000+P}\right)$
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
CHECK VALVE WEIGHT (LB.)
2.0
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
0
0
2
4
6
8
10 X10³
SYSTEM PRESSURE
(P.S.I.)


CHECK VALVE SPACE
VERSUS
SYSTEM PRESSURE
FOR 2, 10, 50, AND 100
HORSEPOWER
$V_s = 6.9(D + .222)^3$
CHECK VALVE SPACE (IN^3)
20
18
16
14
12
10
8
6
4
2
0
100 HORSEPOWER
50 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER
0
2
4
6
8
10 × 10^3
SYSTEM PRESSURE
(P.S.I.)

II. Determination of the Weight and Space of Filters

The weight of filters was assumed to be similar to the weight of accumulators; with different constants. (See Appendix D) This equation is of the form:

$$Wt = .00131\, d^2 P^{\frac{1}{2}} \left(\frac{20000 + P}{20000 - P}\right) + 3.67\, d^3 + \frac{.358\, d^2 P}{12000 - P}$$

$$+ .0158\, d^3 + .025\, d^2 + .007\,(\text{Air Volume}) + 1.3$$

In the calculation of the weight of accumulators it was noted that for small work levels (figure D-2) the weight of the accumulators was nearly constant for all pressures.

Therefore a plot of filter weight versus horsepower was made for 1500 psi, 3000 psi, and 5000 psi (figure C-3). From this it was evident that the weight of the filters was constant at all pressures for the same horsepower level. An average of points was drawn resulting in the following equation:

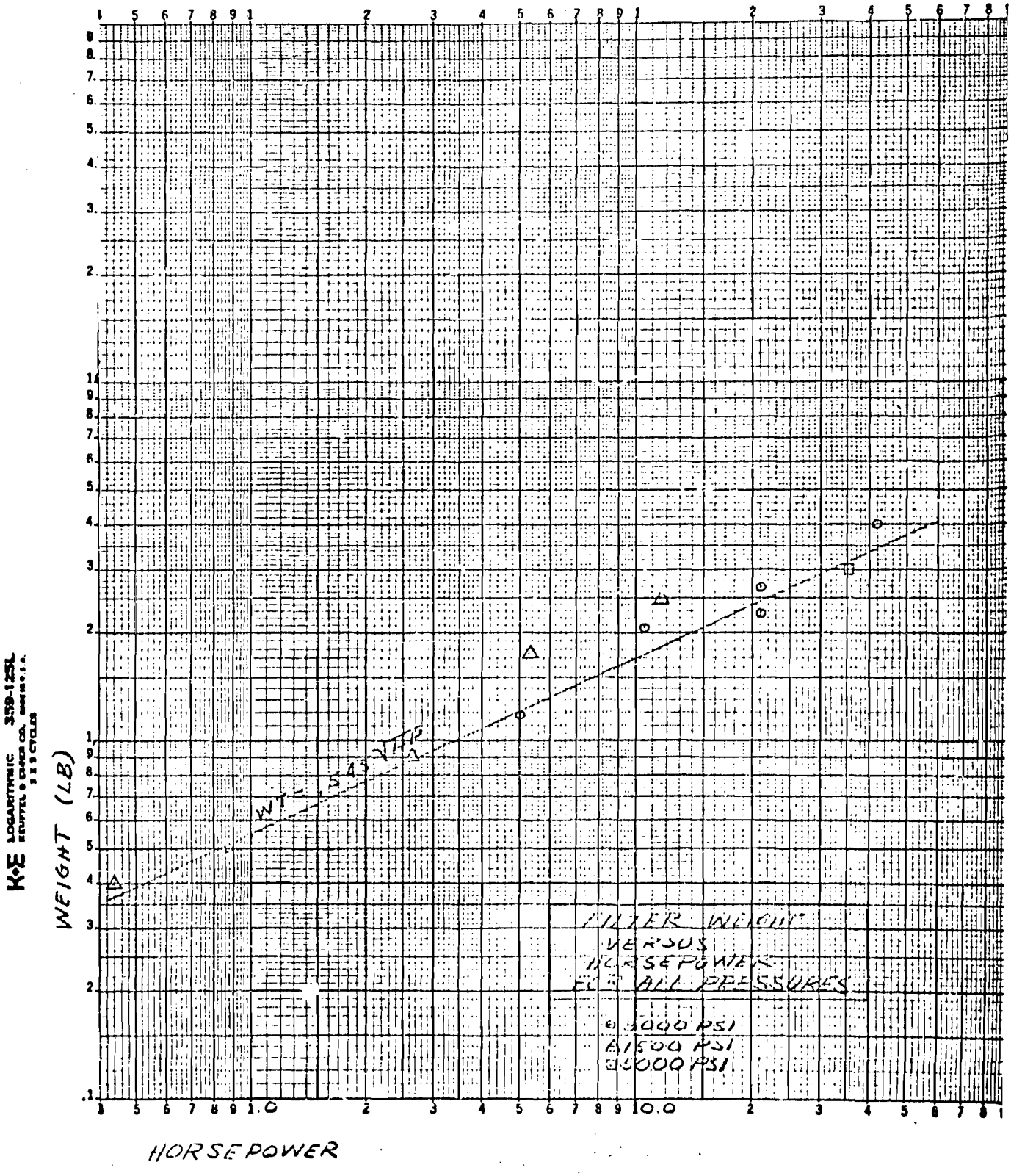
$$Wt = .545\,(HP)^{\frac{1}{3}}$$

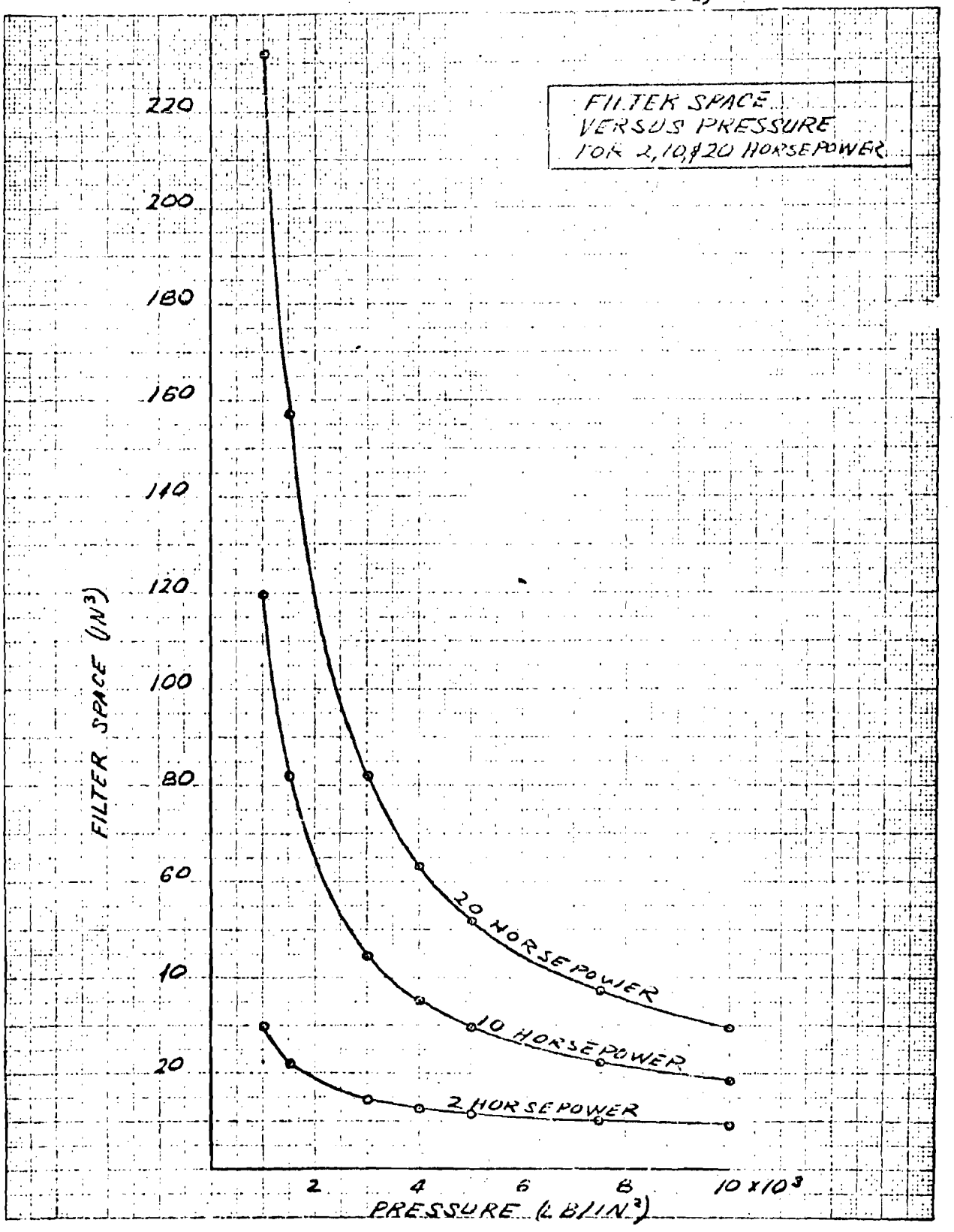
The space of actual filter was calculated and plotted. From this plot an equation was determined for the space of filters at various pressures and horsepowers.

$$\text{Space} = \frac{1.125 \times 10^4\,(HP)}{\text{System Pressure}} + 7$$

The space occupied by filters at 2, 10, and 20 horsepower was calculated and is presented graphically in figure C-4.

FIG. C-3
C-12


WEIGHT (LB)
HORSEPOWER
WT = 5.4 √HP
PUMP WEIGHT
VERSUS
HORSEPOWER
FOR ALL PRESSURES
3000 PSI
6000 PSI
9000 PSI
K+E LOGARITHMIC 359-125L


FILTER SPACE
VERSUS PRESSURE
FOR 2, 10 & 20 HORSEPOWER
FILTER SPACE (IN³)
220
200
180
160
140
120
100
80
60
40
20
2
4
6
8
10 x10³
PRESSURE (LB/IN²)
20 HORSEPOWER
10 HORSEPOWER
2 HORSEPOWER

APPENDIX D

CALCULATION OF THE WEIGHT AND SPACE OF CYLINDRICAL ACCUMULATORS

I The Weight of Accumulators

The final equation for the weight of accumulators is as follows:

Weight = Weight of End Caps + weight of Barrel + Weight of Piston + Correction factor for bulk

$$\text{Weight of End Caps} = .00131\, d^2 P^{\frac{1}{2}} \left(\frac{20000 + P}{20000 - P}\right)$$

$$\text{Weight of Barrel} = 5.67d^3 + .558d^2 \frac{P}{42000 - P}$$

$$\text{Weight of Piston} = .0158d^3 + .025d^2$$

Correction Factor = K = 1.3 + .007 (Air Volume)

where d = Accumulator Bore

P = System Pressure

It is noted that the form of the equations above is the same as the equations found in Appendix B for the calculation of the weight of various sections of the cylinder. The changes have been in the constants to conform to the actual weights of the cylinders and, in the case of the barrel, changes in material strength and density and the replacement of the stroke in terms of the bore.

As a basis for comparison -- three work levels were chosen: 1,200,000 IN-LB; 300,000 IN-LB; and 75,000 IN-LB. For convient calculation this value was divided by the pressure to obtain the air volume of the accumulator. (This assumes 100% efficiency.)

The bore and stroke relationship was obtained for the 100 cubic inch accumulator at 3000 psi. This ratio being: $\frac{S}{d} = 5.68$

A fluid volume of 90% of the air volume was found to be average for 3000 psi accumulators so the equation for finding the bore of the accumulators was:

$$.9 \times \text{Air Volume} = \frac{\pi}{4} d^2 \times 5.68d$$

$$d^3 = \frac{\text{Air Volume}}{4.95}$$

Figure F-1 is an illustration of the weight of accumulators versus air volume for 3000 and 1500 psi accumulators. Two solid lines represent the average weights at these pressures. A dotted line represents the calculated weight using the equations without the correction factor with a phantom line representing the correction factor. The correction factor is the difference between the dotted line and the 3000 psi line.

From the final equation -- page F-1, the weights of accumulators at various pressures and work levels were calculated. The results are presented in Table D-1 and figure D-2. From these curves it is evident there is a minimum weight for any work level at 4000 psi. It is also noted that there is little saving above 3000 psi as the curve is nearly flat between 3000 and 5000 psi.

II Space Occupied by Cylindrical Accumulators

The space occupied by accumulators was considered to be the volume of a cylinder that is the envelope of the accumulator.

$$\text{Space} = \frac{\pi}{4} D^2 \times L$$

Where D = Outside diameter of cylindrical envelope
L = Total length of accumulator

Since the stroke was considered as a multiple of the bore for weight, the length is considered as multiple of the outside diameter. The difference between the bore and the outside diameter and the overall length was also considered to be a constant multiple.

The actual envelope for several accumulators at 3000 psi and various volumes was calculated. With the assumptions as above:

$$\text{Space} = 9.21 \times d^3$$

d = Accumulator bore

The space occupied by accumulators was determined and is tabulated in Table D-2 and presented graphically in figure D-3.

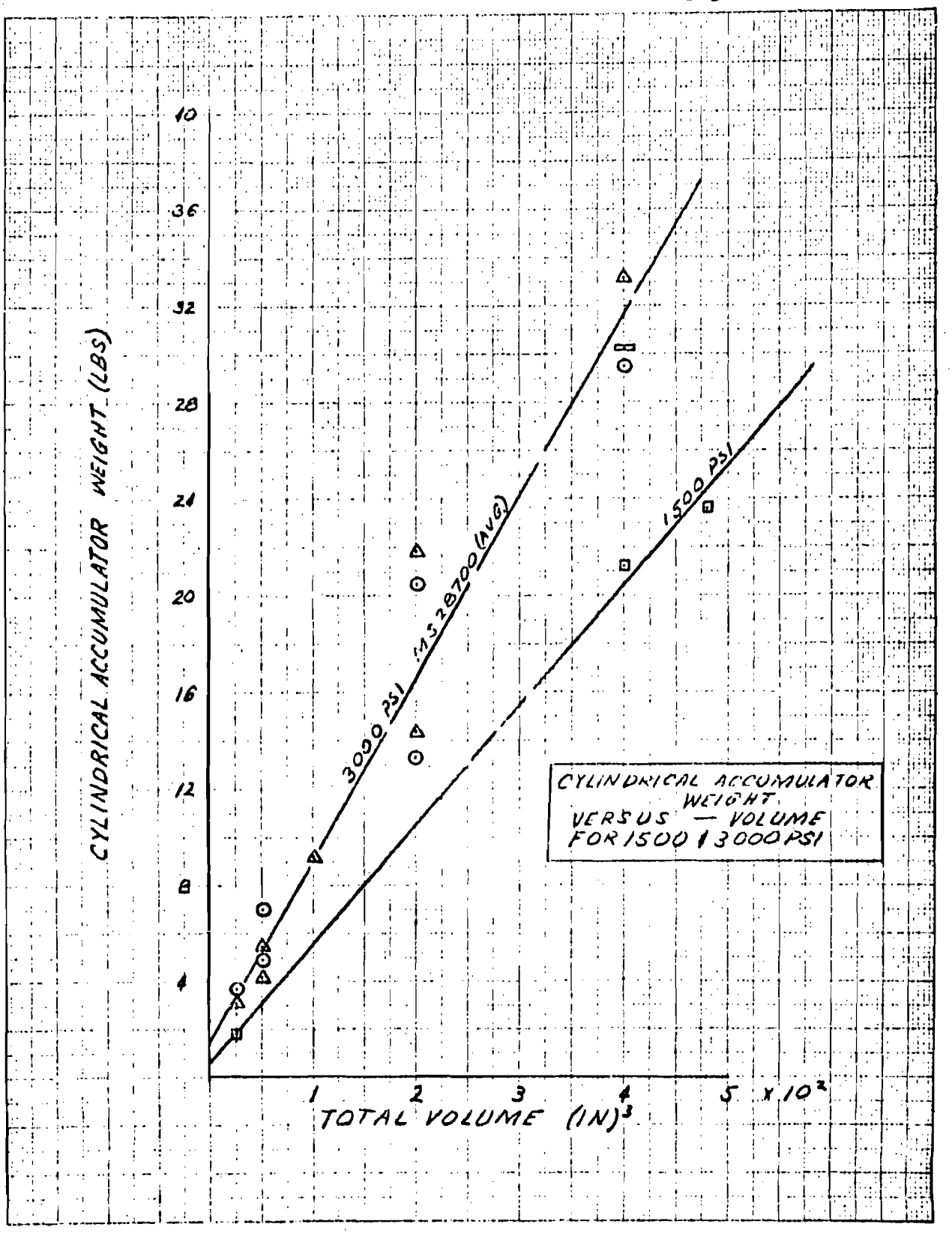
WEIGHT OF ACCUMULATORS AT VARIOUS
PRESSURES AND WORK LEVELS

PRESSURE	WORK	VOL	d^3	d^2	d	END CAPS	BARREL	PISTON	K	WEIGHT
1000	1,200,000	1200	243.5	36.5	6.05	1.538	22.1	4.700	9.7	38.096
	300,000	300	61.0	15.62	3.94	.656	5.14	1.3545	3.4	10.5505
	75,000	75	15.23	6.22	2.496	.261	1.305	.3965	1.825	3.7875
1500	1,200,000	800	162.5	29.8	5.49	1.76	22.45	3.310	6.9	34.420
	300,000	200	40.6	12.81	3.58	.756	5.69	.961	2.7	10.107
	75,000	50	10.15	4.69	2.165	.2765	1.44	.2775	1.69	3.6440
3000	1,200,000	400	81.25	18.7	4.32	1.817	23.4	1.7505	4.1	31.0675
	300,000	100	20.3	7.46	2.73	.725	5.95	.5975	2.0	9.1825
	75,000	25	5.08	2.92	1.719	.2835	1.518	.1531	1.475	3.4296
4000	1,200,000	300	61.0	15.92	3.94	1.98	24.2	1.362	3.4	30.942
	300,000	75	15.23	6.22	2.396	.774	6.14	.396	1.825	9.1355
	75,000	18.75	3.81	2.44	1.562	.3035	1.563	.1212	1.4313	3.419
5000	1,200,000	240	48.75	13.32	3.65	2.055	24.8	1.103	2.95	30.938
	300,000	60	12.2	5.31	2.302	.82	6.32	.326	1.72	9.186
	75,000	15	3.03	2.09	1.447	.3225	1.604	.10015	1.405	3.43165
7500	1,200,000	180	36.55	11.12	3.32	2.78	26.35	.855	2.56	34.545
	300,000	45	9.15	4.375	2.09	1.094	7.22	.2539	1.615	10.1829
	75,000	10	2.03	1.602	1.266	.40	1.65	.0722	1.370	3.4922
10000	1,200,000	120	24.4	8.41	2.9	3.31	28.9	.5962	2.14	34.9462
	300,000	30	6.1	3.34	1.828	1.314	7.37	.1799	1.51	10.3439
	75,000	7.5	1.525	1.329	1.152	.522	1.900	.0571	1.3525	3.8316

SPACE OCCUPIED BY ACCUMULATORS

SPACE = 9.21 d^3

PRESSURE	WORK	d	d^3	SPACE
1000	1,200,000	6.05	243.5	2240
	300,000	3.94	61.0	561
	75,000	2.496	15.23	140.3
1500	1,200,000	5.49	162.5	1497
	300,000	3.58	40.6	374
	75,000	2.165	10.15	93.6
3000	1,200,000	4.32	81.25	749
	300,000	2.72	20.3	187
	75,000	1.718	5.08	46.8
4000	1,200,000	3.94	61.0	561
	300,000	2.496	15.23	140.3
	75,000	1.562	3.81	34.2
5000	1,200,000	3.65	48.75	444.9
	300,000	2.302	12.2	112.3
	75,000	1.447	3.03	27.9
7500	1,200,000	3.32	36.55	336.5
	300,000	2.09	9.15	84.3
	75,000	1.266	2.03	18.7
10000	1,200,000	2.9	24.4	224.5
	300,000	1.828	6.1	56.2
	75,000	1.152	1.525	14.05


CYLINDRICAL ACCUMULATOR WEIGHT (LBS)
10
36
32
28
24
20
16
12
8
4
1
2
3
4
5 X 10²
TOTAL VOLUME (IN)³
3000 PSI
MS 28700 (AVG)
1500 PSI
CYLINDRICAL ACCUMULATOR WEIGHT
VERSUS — VOLUME
FOR 1500 & 3000 PSI

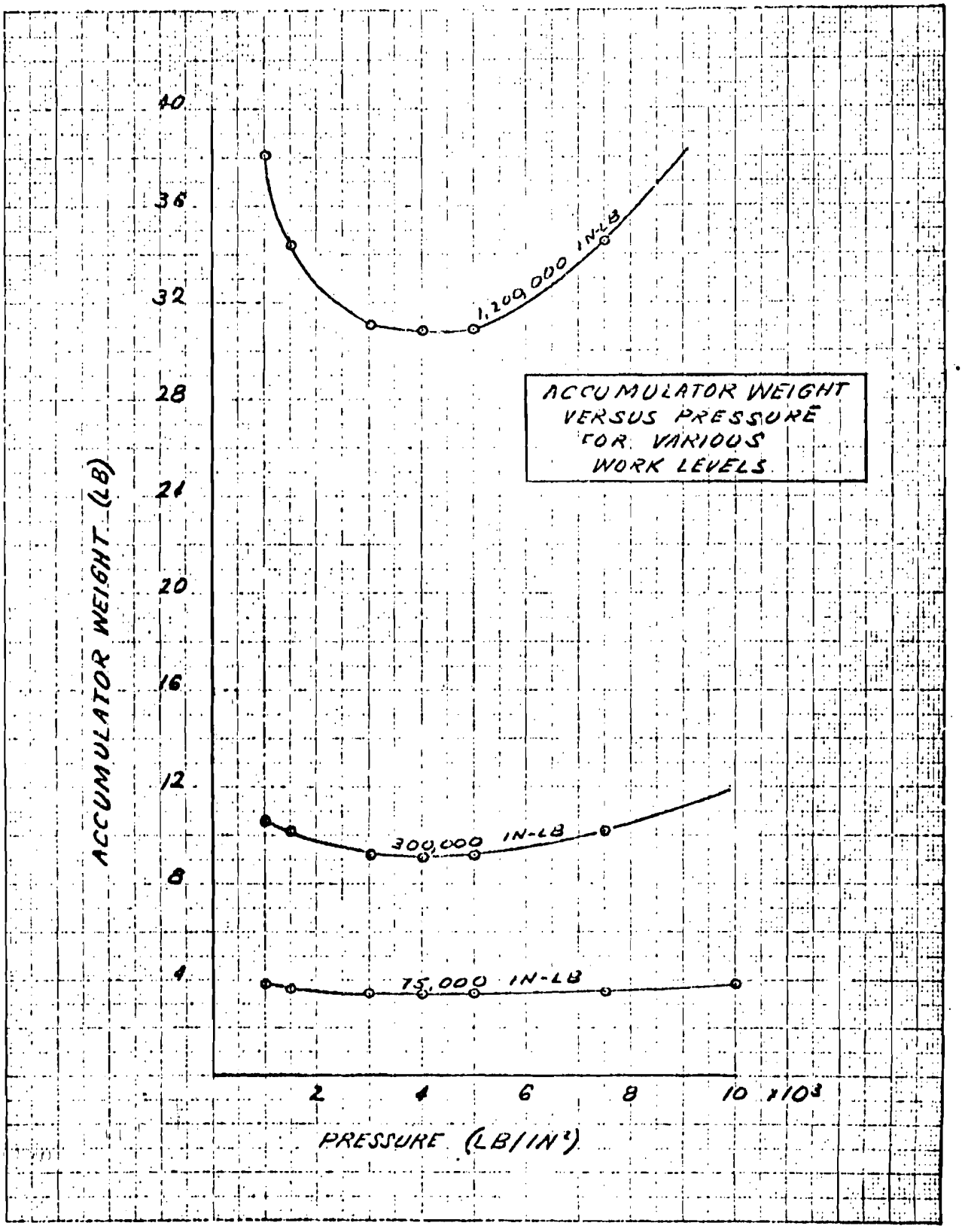
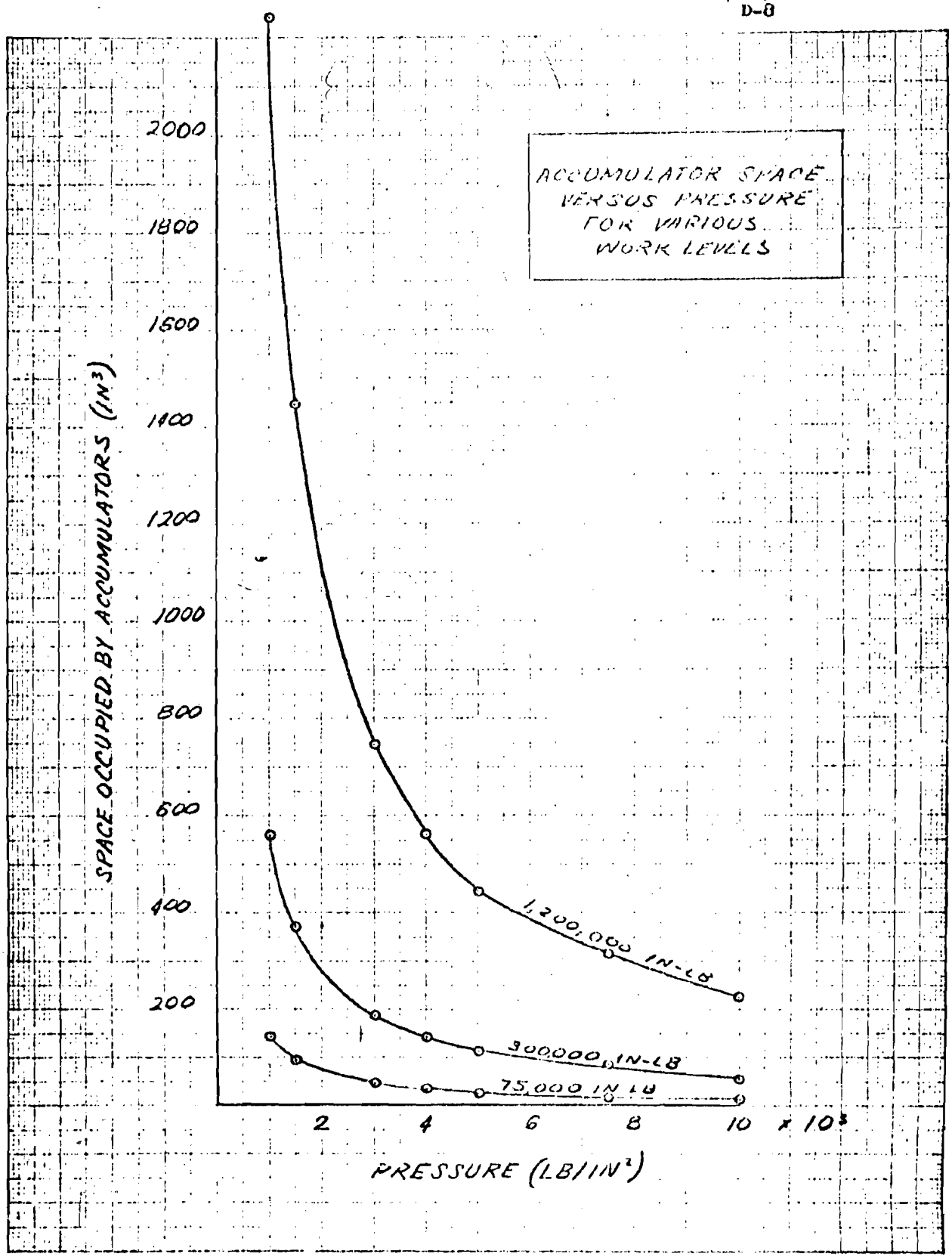

ACCUMULATOR WEIGHT VERSUS PRESSURE FOR VARIOUS WORK LEVELS
ACCUMULATOR WEIGHT (LB)
40
36
32
28
24
20
16
12
8
4
1,200,000 IN-LB
300,000 IN-LB
75,000 IN-LB
2
4
6
8
10 x10³
PRESSURE (LB/IN²)

FIG. D-3
D-8


ACCUMULATOR SPACE VERSUS PRESSURE FOR VARIOUS WORK LEVELS
SPACE OCCUPIED BY ACCUMULATORS (IN³)
2000
1800
1600
1400
1200
1000
800
600
400
200
1,200,000 IN-LB
300,000 IN-LB
75,000 IN-LB
2
4
6
8
10 × 10³
PRESSURE (LB/IN²)

APPENDIX E

DETERMINATION OF THE WEIGHT AND SPACE OF HYDRAULIC PUMPS AT VARIOUS PRESSURES AND HORSEPOWERS

Determination of the Weight and Space of

Hydraulic Pumps at Various Pressures and Horsepowers

It appears logical to assume that the weight of the pump is a direct function of horsepower and independent of working pressure, on the basis that the pump is the optimum design for that pressure.

Examining the cross section drawings of the New York Air Brake 66WA300 or Vickers PV3911, it is evident that the majority of the weight of the package is in the construction of the mechanical parts and joints, and the casing - most of which surrounds the chamber pressurized by inlet or by-pass pressures only.

At a given horsepower this equipment must remain substantially the same. Increasing system pressure can be accomplished by reduction of stroke or bore; and, it is assumed, the reduction in the size of the cylinders will be compensated by increase in wall thickness of the cylinder and pressure chambers.

Since that part of the pump affected by these variations is a relatively small percentage of the total pump weight and space, any error in the above assumption will be negligable within the range of the study.

Therefore, the equations for weight and space for all pressures are based on the most recently developed production pumps for 3000 psi. Any advance in the design of pumps, such as high speed development, would be of equal advantage to all pressures.

In figure E-1 is a plot of weight versus horsepower for some recently developed 3000 psi pumps. Through these points an average is drawn which represents the weight of pumps for any horsepower at all pressures. The equation for this line is:

$$Wt = 3.4 + .72\ (HP)$$

Figure E-2 is a plot of the pump space versus horsepower of 3000 psi pumps. The line drawn is the average of these points. The equation of this line is:

$$Space = 80 + 15.7\ (HP)$$

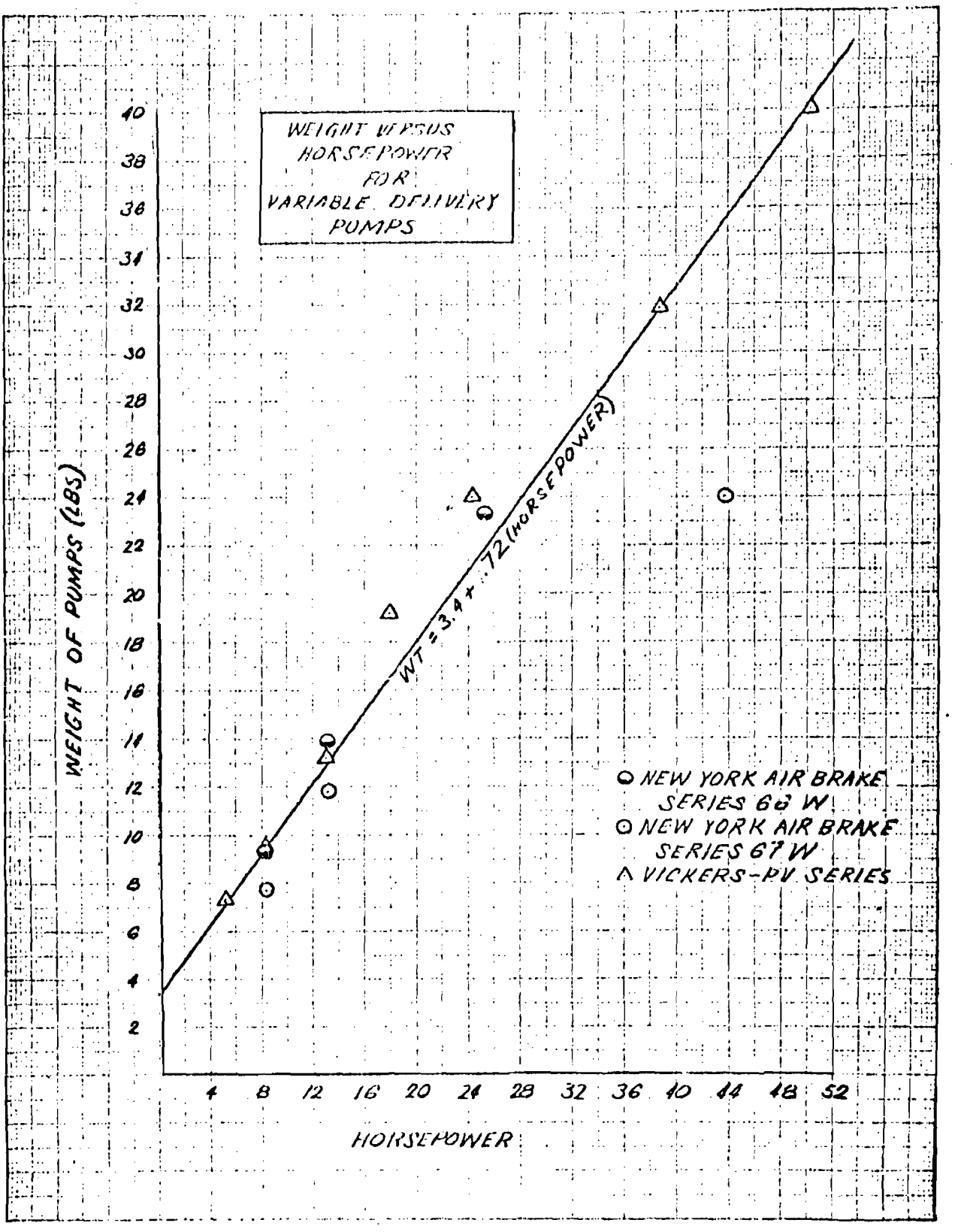
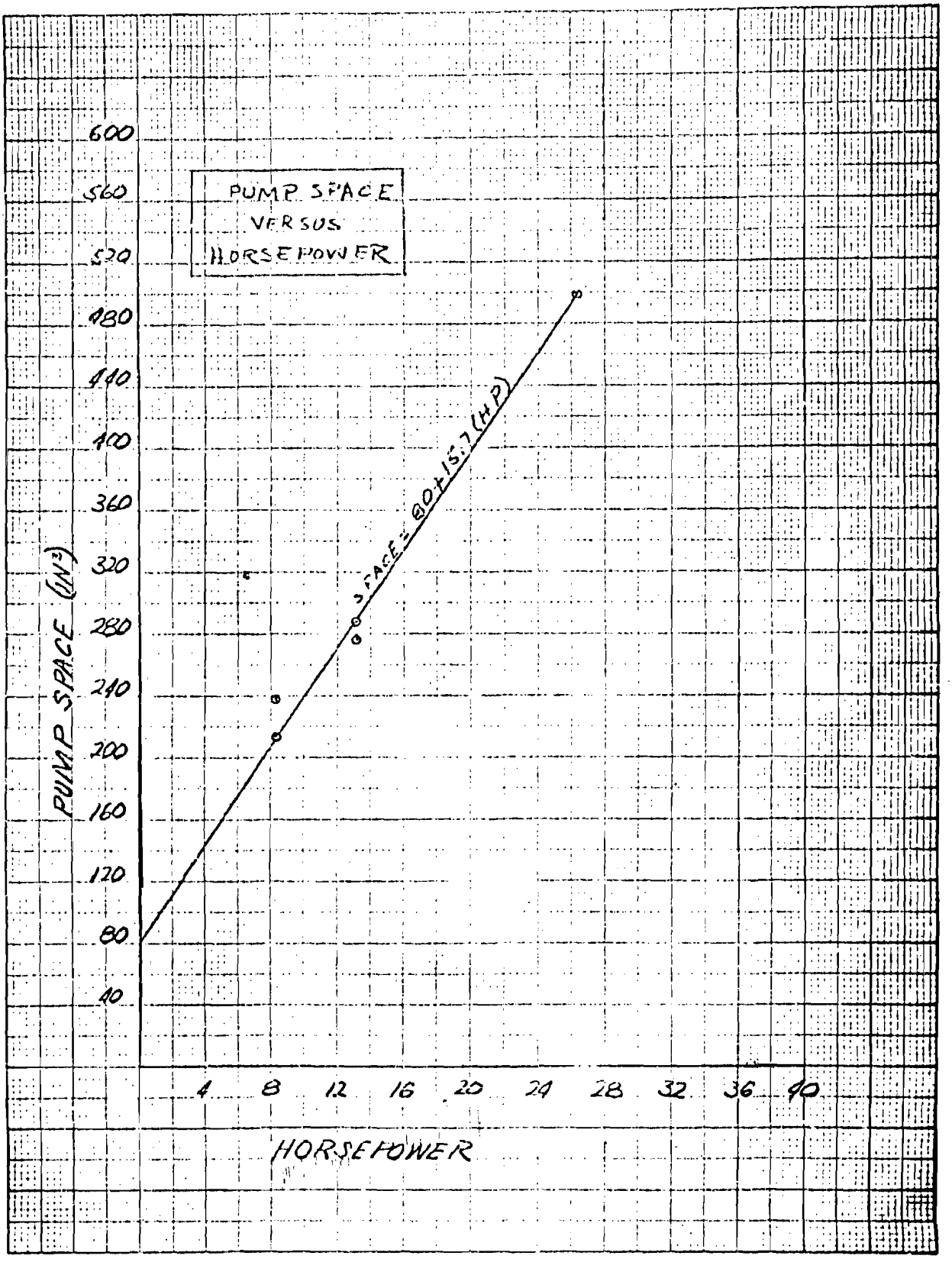

WEIGHT VERSUS
HORSEPOWER
FOR
VARIABLE DELIVERY
PUMPS
WEIGHT OF PUMPS (LBS)
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
WT = 3.4 + .72 (HORSEPOWER)
NEW YORK AIR BRAKE
SERIES 66 W
NEW YORK AIR BRAKE
SERIES 67 W
VICKERS-PV SERIES
4
8
12
16
20
24
28
32
36
40
44
48
52
HORSEPOWER

FIG. E-2
E-4


PUMP SPACE
VERSUS
HORSEPOWER
SPACE = 80 + 15.7 (HP)
PUMP SPACE (IN³)
600
560
520
480
440
400
360
320
280
240
200
160
120
80
40
4
8
12
16
20
24
28
32
36
40
HORSEPOWER

APPENDIX F

THE WEIGHT AND SPACE OF RESERVOIRS AT VARIOUS PRESSURES

APPENDIX F

THE WEIGHT AND SPACE OF RESERVOIRS AT VARIOUS PRESSURES

(This study conforms to Specification MIL-R-5520 A in every detail)

The following symbols were used throughout this section:

W_R = Total weight of reservoir including fluid (LB)

S_R = Total external space utilized by reservoir (IN^3)

V_R = Total internal volume of reservoir (IN^3)

V_{RG} = Total internal volume of reservoir (GAL.)

δ_R = Minimum "full" design level volume of fluid

δa = Air volume in reservoir (IN^3)

δ_e = Emergency fluid volume in reservoir (IN^3)

δ_C = Volume of fluid in reservoir allowed for net depletion from cylinder volumetric changes (IN^3)

δ_A = Fluid volume allowed for accumulators (IN^3)

δ_{LK} = Fluid volume allowed for leakage (IN^3)

δ_S = Total system fluid volume, including the reservoir volume (IN^3)

δ_F = Volume of fluid in the reservoir made unavailable by the action of the largest capacity quantity measuring type hydraulic fuse. (IN^3)

δ_g = Fluid volume allowed to insure that pump suction head is maintained for all altitudes and accelerations within the reservoir design limits. i.e. fluid quantity for negative "g" loads. (IN^3)

δ_T = Fluid volume allowed to equal the maximum thermal contraction (Taken between the temperature limits of +70°F and -40°F) (IN^3)

U_C = Total work of all the cylinders in the system (LB-IN)

U_{AL} = Work of the largest accumulator in the system (LB-IN)

U_A = Total work of all accumulators, including the largest, in the system (LB IN)

U_{CMAX} = Work of cylinders in the largest fused system (LB IN)

U_{EM} = Sum of work of actuators in the emergency system (LB IN)

L_L = Length of line at any one horsepower level (IN)

HP = Horsepower -- used as a work level measure

P = System Hydraulic Pressure (LB/IN^2)

K_1 = Geometrical construction constant

V_C = Total fluid volume of all operating cylinders within the system (IN^3)

V_{AL} = Total air volume in the largest accumulator (IN^3)

V_A = Total air volume of all accumulators, including the largest, in the system (IN^3)

V_{EM} = Actual fluid volume required to complete the program emergency operation. (IN^3)

V_L = Line fluid volume plus fitting fluid volume (IN^3)

V_{FL} = Filter fluid volume (IN^3)

V_V = Valve fluid volume (IN^3)

V_P = Pump or motor fluid volume (IN^3)

Va = Total air space volume (IN^3)

$$L_L \cdot (HP)^{.7065} = L_{L1} \cdot (HP_1)^{.7065} + L_{L2} (HP_2)^{.7065} + \ldots$$

$$\ldots + L_{LN} (HP_N)^{.7065}$$

W_{TANK} = Weight of empty reservoir (LB)

W_{FLUID} = Weight of fluid (LB)

Below are listed, by paragraph numbers, what appears in Specification MIL-R-5520A when restated as mathematical expressions.

3.2.2 $V_R = \delta_R + \delta a + \delta_e$ (F1)

3.2.3 $\delta_R = \delta C + \delta_A + \delta_{LK} + \delta_F + \delta_E + \delta_T$ (F2)

3.2.3.a $\delta_C = 1.5\ (0.25) V_C = .375 V_C$ (F3)

3.2.3.b $\delta_A = V_{AL} + .6\ (V_A - V_{AL})$ (F4)

3.2.3.c $\delta_{LK} = .05\ \delta_S$ (F5)

3.2.3.d $\delta_F = \delta_F$ (F6)

3.2.3.e $\delta_E = K_1 V_R$ (F7)

K_1 = Ratio of allowed fluid to total volume of reservoir and it is dependent on the geometrical configuration.

3.2.3.f $\delta_T = (70 + 40)\ (.000432)\ \delta_S = .0475 \delta_S$ (F8)

3.2.4 $\delta a = 0.1 \delta_R + 0.1 \delta e$ (F9)

3.2.5 $\delta_e = 1.25\ V_{EM}$ (F10)

The items included in the system is defined by:

$$\delta_S = V_R + V_C + V_A + V_L + V_{FL} + V_V + V_P \quad \text{(F11)}$$

Combining equations F1 through F11 and solving for V_R:

$$\begin{aligned}
V_R &= \delta_R + \delta a + \delta_e \\
&= \delta_R + (0.1\ \delta_R + 0.1\ \delta_e) + 1.25\ V_{EM} \\
&= 1.1\ \delta_R + 1.375\ V_{EM} \\
&= 1.1\ \ \delta_C + \delta_A + \delta_{LK} + \delta_F + \delta_E + \delta_T \quad + 1.375\ V_{EM} \\
&= 1.1\ \ .375\ V_C + V_{AL} + .6\ (V_A - V_{AL}) + .05\ \delta_S + \delta_F + 1.1\ K1\ V_R + 1.375\ V_{EM} \\
&= .4125\ V_C + .44\ V_{AL} + .66\ V_A + .1072\ \delta_S + 1.1 \delta_F + 1.1 K_1 V_R + 1.375\ V_{EM} \\
&= .4125\ V_C + .44\ V_{AL} + .66\ V_A + .1072\ \ V_R + V_C + V_A + V_L + V_{FL} + V_V + V_P \\
&\quad + 1.1\ \delta_F + 1.1 K_1\ V_R + 1.375\ V_{EM} \\
&= .5197\ V_C + .44\ V_{AL} + .7672\ V_A + (.1072 + 1.1 K_1)\ V_R + 1.1\ \delta_F + 1.375\ V_{EM} \\
&\quad + .1072\ \ V_L + V_{FL} + V_V
\end{aligned}$$

Finally:

$$V_R = \frac{1}{.8928 - 1.1\,K_1} \quad .5197\,V_C + .44\,V_A + .7672\,V_{AL} + 1.16_F + 1.375\,V_{EM}$$

$$+ .1072\,V_L + V_{FL} + V_V \qquad \text{(F11)}$$

To render this equation more readily useable it was logically assumed from data that:

$$V_V = .1\,V_L \qquad \text{(F13)}$$

$$V_{FL} = .05\,V_L \qquad \text{(F14)}$$

$$\delta_F = 1.5\,V_{CMAX} = 1.5\,\frac{U_{CMAX}}{P} \qquad \text{(F15)}$$

$$V_{EM} = \frac{U_{EM}}{P} \qquad \text{(F16)}$$

$$V_C = \frac{U_C}{P} \qquad \text{(F17)}$$

$$V_A = \frac{U_A}{P} \qquad \text{(F18)}$$

$$V_{AL} = \frac{UAL}{P} \qquad \text{(F19)}$$

Further, when line volume is plotted versus pressure at various horsepower, it was found that the following relation obtains:

$$V_L = \frac{35.85\,(HP)^{.7065}}{P} \qquad \text{(F20)}$$

The quantity $\frac{1}{.8928 - 1.1K_1}$ which appears in equation F11 can be thought of as a multiplier dependent on value of the geometrical construction constant K_1. To obtain values for K_1, the negative "g" load fluid allowances for the utility and surface control systems of the Martin XP6M-1 were divided by their respective total volumes.

Accordingly: for the tank type reservoir:

$$K_1 = 38.3\% = .383$$

And:

$$\frac{1}{.8928 - 1.1K_1} = 2.12 \qquad \text{(F21)}$$

For the piston type reservoir:

$K_1 = 0$ and so:

$$\frac{1}{.8928 - 1.1K_1} = 1.12 \qquad \text{(F22)}$$

Utilizing the relations shown by equations F13 through F20 :

$$V_R = \frac{1}{.6928 - 1.1K_1} \left(\frac{1}{P}\right) .5197\ U_C + .44\ U_A + .7672\ U_{AL} + 1.5\ U_{CMAX} + 1.375\ U_{EM} + 4.42\ L_L \cdot (HP)^{.7065} \quad \text{(F23)}$$

To obtain the volume in a tank type reservoir, equation F21 was utilized:

$$V_R = \frac{1}{P}\ 1.102\ U_C + .933\ U_A + 1.628\ U_{AL} + 3.18\ U_{CMAX} + 2.915\ U_{EM} + 9.37\ L_L\ (HP)^{.7065} \quad \text{(F24)}$$

For the piston type:

$$V_R = \frac{1}{P}\ .582\ U_C + .493\ U_A + .859\ U_{AL} + 1.68\ U_{CMAX} + 1.54\ U_{EM} + 4.95\ L_L \cdot (HP)^{.7065} \quad \text{(F25)}$$

The reservoir weight can be expressed as:

$$W_R = W_{TANK} + W_{FLUID} \quad \text{(F26)}$$

From figure F1 for tank type reservoirs:

$$W_{TANK} = 6.7 + 1.22\ V_{RG} \quad \text{(F27)}$$

Since the fluid (MIL-O-5606) weighs .03085 LB/IN^3 at 70°F

$$W_{FLUID} = .03085\ (V_R - Va) \quad \text{(F28)}$$

To obtain an expression for Va in terms of V_R the following two definitions were solved simultaneously:

$$V_R = 110\% \text{ of } V_{FLUID}$$

$$V_R = V/ + V_{FLUID}$$

And so:

$$Va = .091\ V_R \quad \text{(F29)}$$

Therefore equation F28 becomes:

$$W_{FLUID} = .03085\ (V_R - .091V_R)$$

$$= .03085\ (.909)\ V_R$$

$$= .0281\ V_R \quad \text{(F30)}$$

By definition:

$$V_R = 231\ V_{RG} \quad \text{(F31)}$$

Inserting equations F27, F30, and F31 into F26:

$$W_R = 6.7 + 1.22\ V_{RG} + .0281\ V_R$$

$$= 6.7 + .00528\ V_R + .0281\ V_R$$

$$W_R = 6.7 + .03338\ V_R \qquad \text{(F32)}$$

Using the expression for V_R, equation F24:

$$W_R = 6.7 + \frac{1}{P}\ .0367\ U_C + .0311\ U_A + .0542\ U_{AL}$$

$$+ .106\ U_{CMAX} + .0915\ U_{EM} + .3124\ L_L \cdot (HP)^{.7065}$$

and is the reservoir weight with fluid for the tank type.

From figure F1 for the piston type reservoir:

$$W_{TANK} = 6.32 + 5.89\ V_{RG} \qquad \text{(F33)}$$

Since no air space found in the piston type:

$$W_{FLUID} = .03085\ V_R \qquad \text{(F34)}$$

Combining equations F26, F31, F33, and F34:

$$W_R = 6.32 + 5.89\ V_{RG} + .03085\ V_R$$

$$= 6.32 + .0255\ V_R + .03085\ V_R$$

$$W_R = 6.32 + .05635\ V_R \qquad \text{(F35)}$$

Using the expression for V_R, equation F25:

$$W_R = 6.32 + \frac{1}{P}\ .0328\ U_C + .02775\ U_A + .0483\ U_{AL}$$

$$+ .0945\ U_{CMAX} + .0867\ U_{EM} + .279\ L_L \cdot (HP)^{.7065} \qquad \text{(F36)}$$

and is the weight of piston type reservoir with fluid.

From figure F2 for the tank type reservoir:

$$S_R = 598\ V_{RG}^{.728}$$

$$= (6534\ V_{RG})$$

$$= (28.3\ V_R)^{.728} \qquad \text{(F37)}$$

Using equation F24 for V_R

$$S_R = \frac{1}{P}\ 31.2\ U_C + 26.4\ U_A + 46.1\ U_{AL} + 90\ U_{CMAX}$$

$$+ 82.5\ U_{EM} + 265\ L_L \cdot (HP)^{.7065}\ ^{.728} \qquad \text{(F38)}$$

and this is the equation for space of the tank type reservoir.

From figure F2 for the piston type reservoir:

$$S_R = 300 + 650\ V_{RG}$$

$$= 300 + 2.81\ V_R \qquad \text{(F39)}$$

Using equation F25 for V_R:

$$S_R = 300 + \frac{1}{P}\ 1.636\ U_C + 1.385\ U_A + 2.41\ U_{AL}$$

$$+ 4.72\ U_{CMAX} + 4.33\ U_{EM} + 13.91\ L_L \cdot (HP)^{.7065} \qquad \text{(F40)}$$

Which is the equation for space utilized by the piston type reservoir.

FIG. F-1

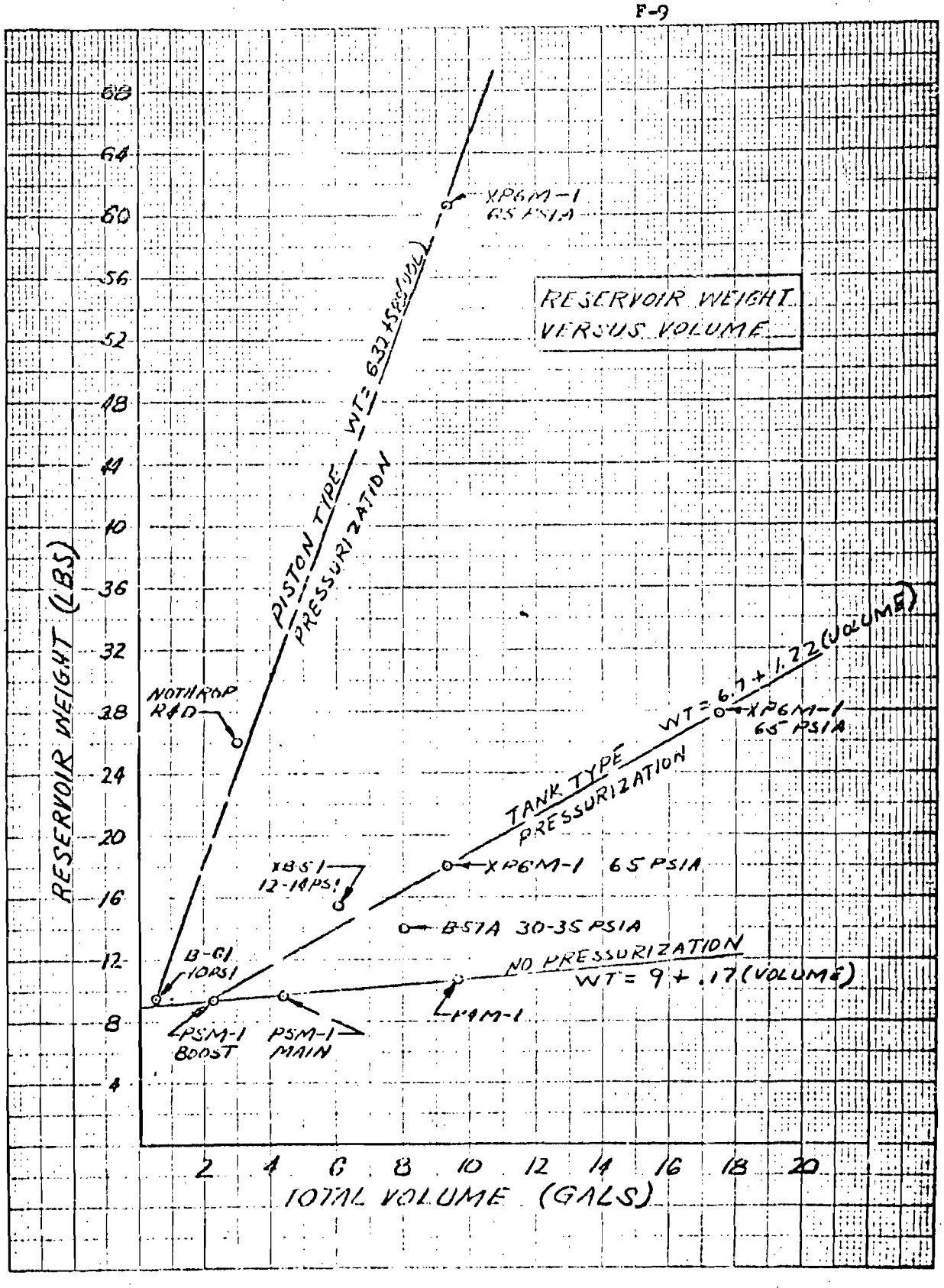
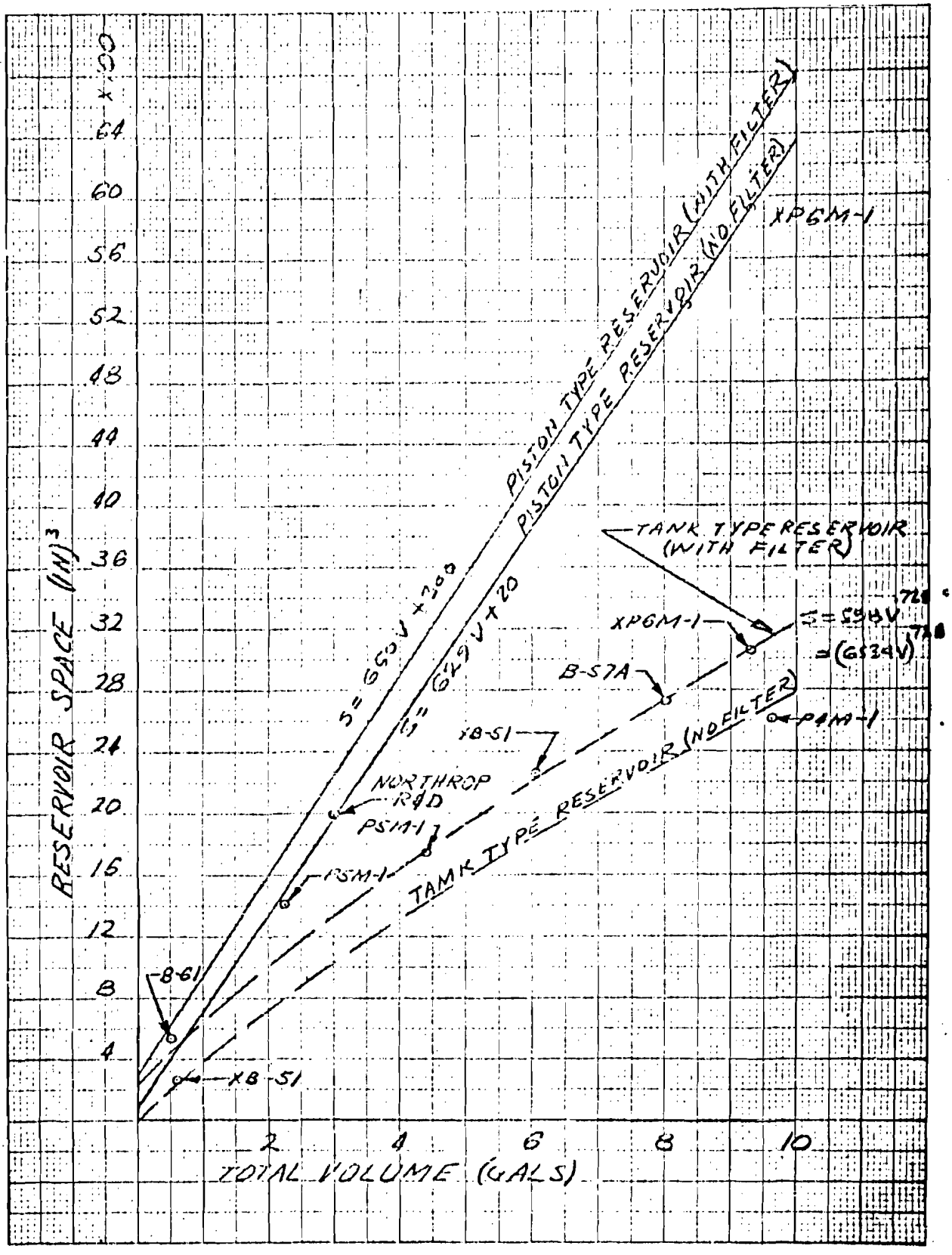

RESERVOIR WEIGHT VERSUS VOLUME
RESERVOIR WEIGHT (LBS)
TOTAL VOLUME (GALS)
PISTON TYPE PRESSURIZATION
WT = 6.32 + 2.25 (VOL)
XP6M-1 65 PSIA
NOTHROP R&D
TANK TYPE PRESSURIZATION
WT = 6.7 + 1.22 (VOLUME)
XP6M-1 65 PSIA
XB51 12-14 PSI
B-61 10 PSI
B51A 30-35 PSIA
NO PRESSURIZATION
WT = 9 + .17 (VOLUME)
PSM-1 BOOST
PSM-1 MAIN
P4M-1

FIG. F-2


RESERVOIR SPACE (IN)³
TOTAL VOLUME (GALS)
PISTON TYPE RESERVOIR (WITH FILTER)
PISTON TYPE RESERVOIR (NO FILTER)
TANK TYPE RESERVOIR (WITH FILTER)
TANK TYPE RESERVOIR (NO FILTER)
XPGM-1
XPGM-1
B-57A
XB-51
PAM-1
NORTHROP R&D
PSM-1
PSM-1
B-61
XB-51

APPENDIX G

ACKNOWLEDGMENTS

In our effort to make the results of this study applicable to the entire industry, an attempt was made to get everyone interested to contribute. Excellent cooperation was obtained from this contacted.

The following companies and organizations willingly contributed information that appeared to be valuable in the optimum pressure study:

1. Aircraft Laboratory - Mechanical Branch
   Wright Air Development Center
2. Bureau of Aeronautics - Airborne Equipment Section
3. Boeing - Seattle
4. Chance Vought - Dallas
5. Convair - San Diego
6. Douglas - El Segundo, Long Beach, Santa Monica
7. Grumman - Bethpage
8. McDonnell - St. Louis
9. North American - Downey, Inglewood
10. Northrop - Hawthorne
11. Republic - Farmingdale

APPENDIX H

NORTH AMERICAN AVIATION

OPTIMUM PRESSURE STUDY

NORTH AMERICAN AVIATION

Optimum Pressure Study

Lou Berthelson

Purpose: Compare weight of utility system at 3000, 5000 psi and other weights considering actuators, lines and fittings. Space saving is evident in all three.

Actuators D Brakes

Weights	5000 psi	3000 psi
Cylinder	.818	.907
Piston	1.803	2.083
Cylinder Nuts	.212	.252
End	.740	.766
End Cap	.801	.865
TOTAL	4.374	4.873

$$\frac{4.374}{.500} = 11.5\% \text{ saving}$$

Assume qualitative conclusion of 10% saving in general run of actuators.

Lines: $Fu = \frac{PbD}{2t}$ Burst Pressure = 4 p

Pressure Drop: $h = f \frac{2}{d} \frac{V2}{2g}$; $\Delta P = h\, P_f$ $g = \frac{f}{d} \frac{Pf}{2} V2$ z

f = friction factor
V = fluid velocity
z = line length
d = ID
= abs viscosity

h = head loss
Pf = fluid density
Nr = Reynolds No.
V = kinematic viscosity

$$f = \frac{64}{Nr} \qquad f = 1190 \qquad Nr\ \frac{pVd}{M} = \frac{Vd}{v}$$

$$\Delta P = \frac{64}{Nr} \frac{1}{d} \frac{(PV^2)}{2} = \frac{64}{\frac{Vd}{v}} \frac{1}{d} \frac{(PV^2)z}{2}$$

$$= \frac{8\ NQz}{A^2}$$

Line and Fluid Weight:

D = O.D. Tube
Pm = Specific Weight Metal
pf = Specific Weight Fluid

$$W = pf \frac{\pi}{4} (D - 2t)^2 + pm \frac{\pi}{4} D^2 - (D - 2t)^2$$

$$= \frac{\pi D^2}{4} \left(1 - \frac{4p}{Fu}\right)^2 (Pf - p_m) + P_m$$

Power = constant PaQn = PoQo
Power Loss = constant PnQn = PoQo

$$\frac{(8nM\ QnZ)}{An} \quad Qn = \frac{(8nM\ QoZ)}{Ao^2} Qo$$

$$\frac{Qn^2}{An^2} = \frac{Qo^2}{Ao^2} ; \quad \frac{Qn}{An} = \frac{Qo}{Ao} \qquad An = \frac{Qn\ Ao}{Qo}$$

$$An = \frac{Po}{Pn} \times \frac{Qo}{Qo} \times Ao = \frac{Po}{Pn} Ao$$

$$\frac{\pi}{4} (Dn - 2tn)^2 = \frac{Po}{Pn} (Do - 2\ to)^2 \frac{\pi}{4}$$

$$Dn^2 \left(1 - 4 \frac{Pn}{Fn}\right)^2 = Do^2 \frac{Po}{Pn} \left(1 - 4 \frac{Po}{Fo}\right)^2$$

$$\frac{Wn}{Wo} = \frac{\left[\frac{\pi}{4} Do^2 \frac{Po}{Pn} \frac{\left(1-4 \frac{Po}{Fo}\right)^2}{\left(1-4 \frac{Pn}{Fn}\right)} (pf-p_{mn}) \left(1- \frac{4Pn}{Fn}\right)^2 - Pmn\right]}{\left[\frac{\pi Do^2}{4} (pf-p_{mo}) \left(1 - \frac{4Po}{Fo}\right)^2 + Pmo\right]}$$

$$\frac{Wn}{Wo} = \frac{\left[\frac{Po}{Pn} \frac{Pmn}{\left(1 - 4 \frac{Pn}{Fn}\right)^2} - (p_{mn} - pf)\right]}{\left[\frac{P_{mo}}{(1 - 4Po^2)} - p_{mo} - pf)\right]}$$

See Curves

Return lines using Al. Alloy

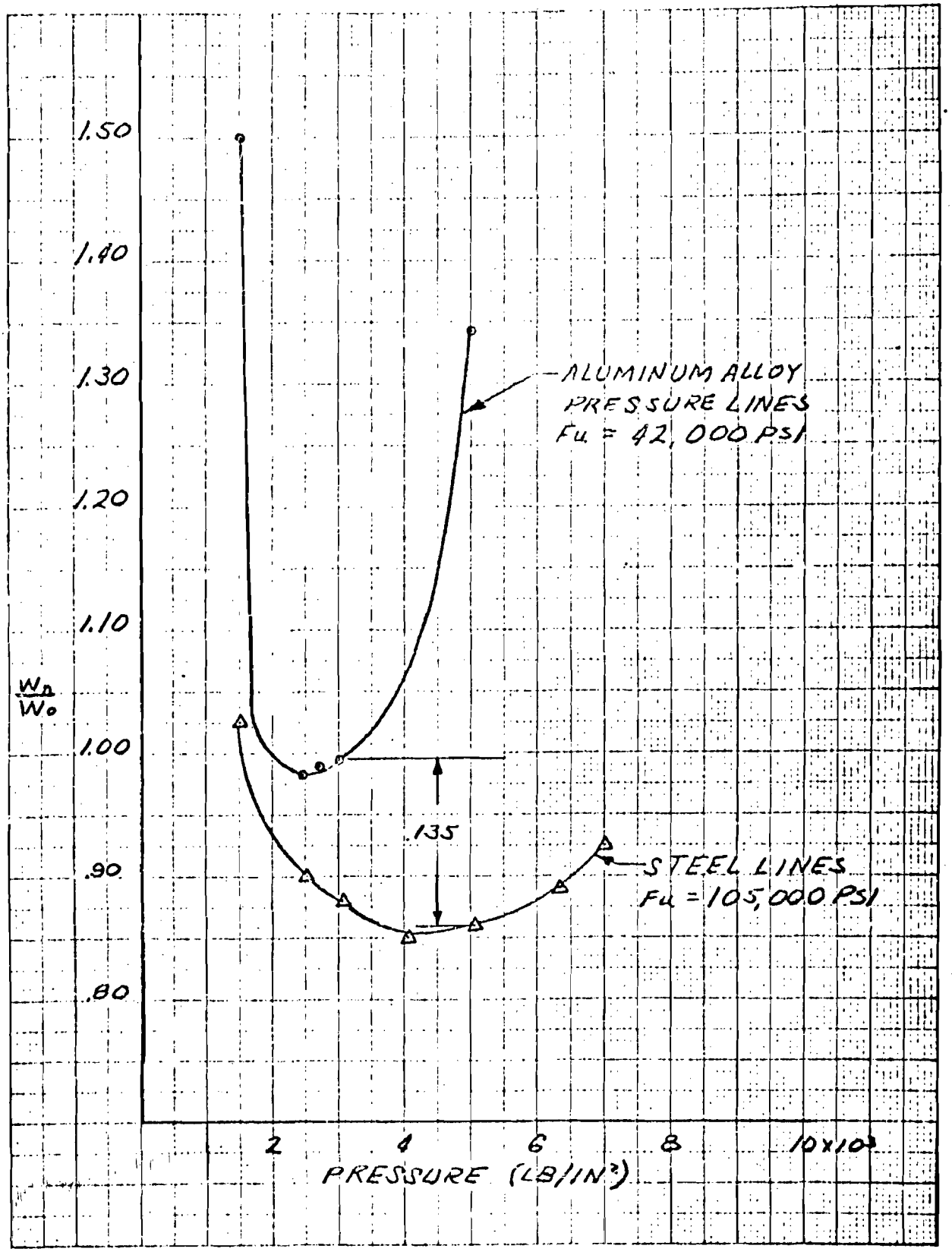
Spec. drop .10(5000) = 500 psi
Burst Factor of 7/1
Burst Pressure 7 x 500 = 3500 psi

$$An = \frac{3500}{2\ Fn} Dn$$

$$Dn^2 \left(1 - \frac{3500}{Fn}\right)^2 = Do^2 \left(1 - 4\ \frac{Po}{Fo}\right)^2 \quad \frac{Po}{Pn}$$

$$\frac{Wn}{Wo} = \frac{\left[\frac{Po}{5000} \quad \frac{Pmn}{\left(1 - \frac{3500}{Fn}\right)^2} - (Pmn - pf)\right]}{\left[\frac{pmo}{\left(1 - \frac{4\ Po}{Fo}\right)^2} - (Pmo - pf)\right]}$$

If Po = 3000 and Pn = 5000; $\frac{Wn}{Wo}$ = .24


1.50
1.40
1.30
1.20
1.10
1.00
.90
.80
Wn/Wo
ALUMINUM ALLOY
PRESSURE LINES
Fu = 42,000 PSI
.135
STEEL LINES
Fu = 105,000 PSI
2
4
6
8
10x10³
PRESSURE (LB/IN²)

USING CONVERSION SUCH 3/8 x .049 to 1/4 x .018

3000 PSI SYSTEM		5000 PSI SYSTEM		$\frac{Wn}{Wo}$
SIZE	WT (Lines Only)	SIZE	WT	
3/8 x .049	.060	1/4 x .035	.029	.48
1/2 x .065	.107	3/8 x .035	.044	.41
5/8 x .083	.174	1/2 x .042	.072	.41
3/4 x .095	.230	5/8 x .049	.107	.46
			AVERAGE	.44
			SAVING	.56

LINES SUMMARY

Assuming 70% pressure lines
and 30% return lines

$$\begin{array}{rcl} .7 \times .135 & = & .095 \\ .3 \times .56 & = & .168 \\ \hline & & .253 \end{array}$$

25% wt. saving in lines by converting 3000 to 5000 psi

FITTING CONVERSION CHART

RETURN LINES						PRESSURE LINES					
3000 PSI Type, Size		Wt = Wo (Al - Al	5000 PSI SIZE	Wt = Wn ALAL	Wn / Wo	3000 PSI Type, Size		Wt = Wo ALAL	5000 PSI Eq Size	Wt = Wn St. Steel	Wn / Wo
	3/8	.056	1/4	.037	.66		3/8	.56	1/4	.129	2.3
815	1/2	.102	3/8	.056	.55	815	1/2	.102	3/8	.187	1.8
	5/8	.161	1/2	.102	.64		5/8	.161	1/2	.333	2.1
	3/8	.076	1/4	.050	.66		3/8	.076	1/4	.141	1.9
833	1/2	.133	3/8	.076	.57	833	1/2	.133	3/8	.211	1.6
	5/8	.214	1/2	.133			5/8	.214	1/2	.405	2.0
	3/8	.107	1/4	.072	.67		3/8	.107	1/4	.214	2.0
834	1/2	.200	3/8	.107	.54	334	1/2	.200	3/8	.313	1.5
	5/8	.246	1/2	.200	.81		5/8	.246	1/2	.595	2.4
				TOTAL	5.10					TOTAL	17.6
				AVERAGE	.57					AVERAGE	2

Fittings and Piping = 81.0 lb.
Assume .35 fraction of weight for fittings (ALAL)
$.35L + L = 81$ $L = 60$ lb. lines 21 lb. fittings

Assume .30 as amount of return lines $.3 \times 21 = 6.3$ lbs.
$.57\ (6.3) = 3.6$ lbs. Return lines at 5000 psi

$2.00\ (21 - 6.3) = \frac{29.4}{33.0 \text{ lb.}}$ Pressure lines at 5000 psi

Fitting loss = 12 lb. = 12/21 = 57%

WEIGHT COMPARISON 3000 and 5000

ITEM	3000 PSI WEIGHT	5000 PSI WEIGHT
Lines	60	48
Fittings	21	33
Cylinders	44	41
TOTAL	125	122

If a better fitting were developed, considerable weight and space saving could be accomplished. Lacking this and considering approximation of this analysis there is no justification of designing to a higher pressure.

NORTH AMERICAN AVIATION

Utility System Weight Breakdown

Typical Airplane

1. Power System		
Main Pumps	14	
Reservoir	6	
Press and Fitting Equipment	2	
Oil and Air Filters	4	
Accumulators	18	
	44	.164
2. Valves		
Relief	1	
Restrictor	1	
Check	3	
Speed Brake	7	
Landing Gear and Brake	13	
	25	.093
3. Actuators		
Landing Gear	25	
Speed Brake	19	
	44	.164
4. Tubing and Fittings		
Power System	26	
Landing Gear and Brake	38	
Speed Brake	17	
	81	.300
5. Fluid		
9.9 Gal.	59	.22
6. Miscellaneous		
Supports Power	1	
Clips and Miscellaneous	13	
	14	.05
	268	

APPENDIX J

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