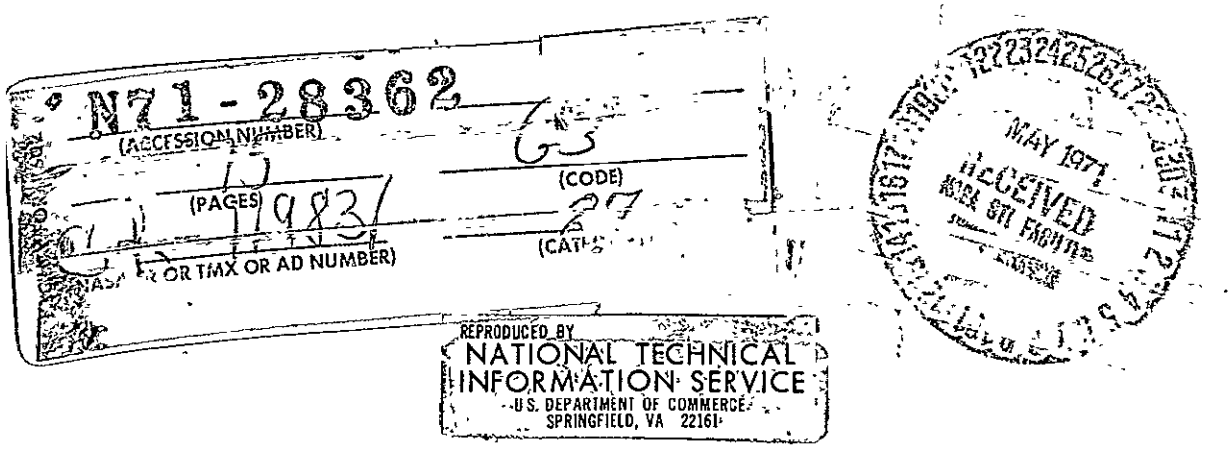
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STUDY OF LOW GRAVITY PROPELLANT TRANSFER

THIRD QUARTERLY PROGRESS REPORT FOR PERIOD
23 DECEMBER 1970 THROUGH 30 APRIL 1971

GENERAL DYNAMICS
Convair Aerospace Division



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10 May 1971

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SECTION 1

INTRODUCTION AND SUMMARY

The basic objective of the present program is to perform an analytical assessment of potential methods for replenishing the auxiliary propulsion, fuel cell and life support cryogens aboard an orbiting space station. This progress report covers the work performed since December 1970.

During the past reporting period a request was made to extend the contract completion date to 30 November 1971. In this connection it was further requested that the reporting period for the third Quarterly Technical Report be extended to 30 April 1971 to allow for the inclusion of more data and provide for a more logical division between this Quarterly and the Final Report. Monthly reports will be provided for the intervening period until work has been completed, and the extension will be accomplishe at no change in estimated cost or fixed fee. A new program schedule is presented in Figure 1-1.

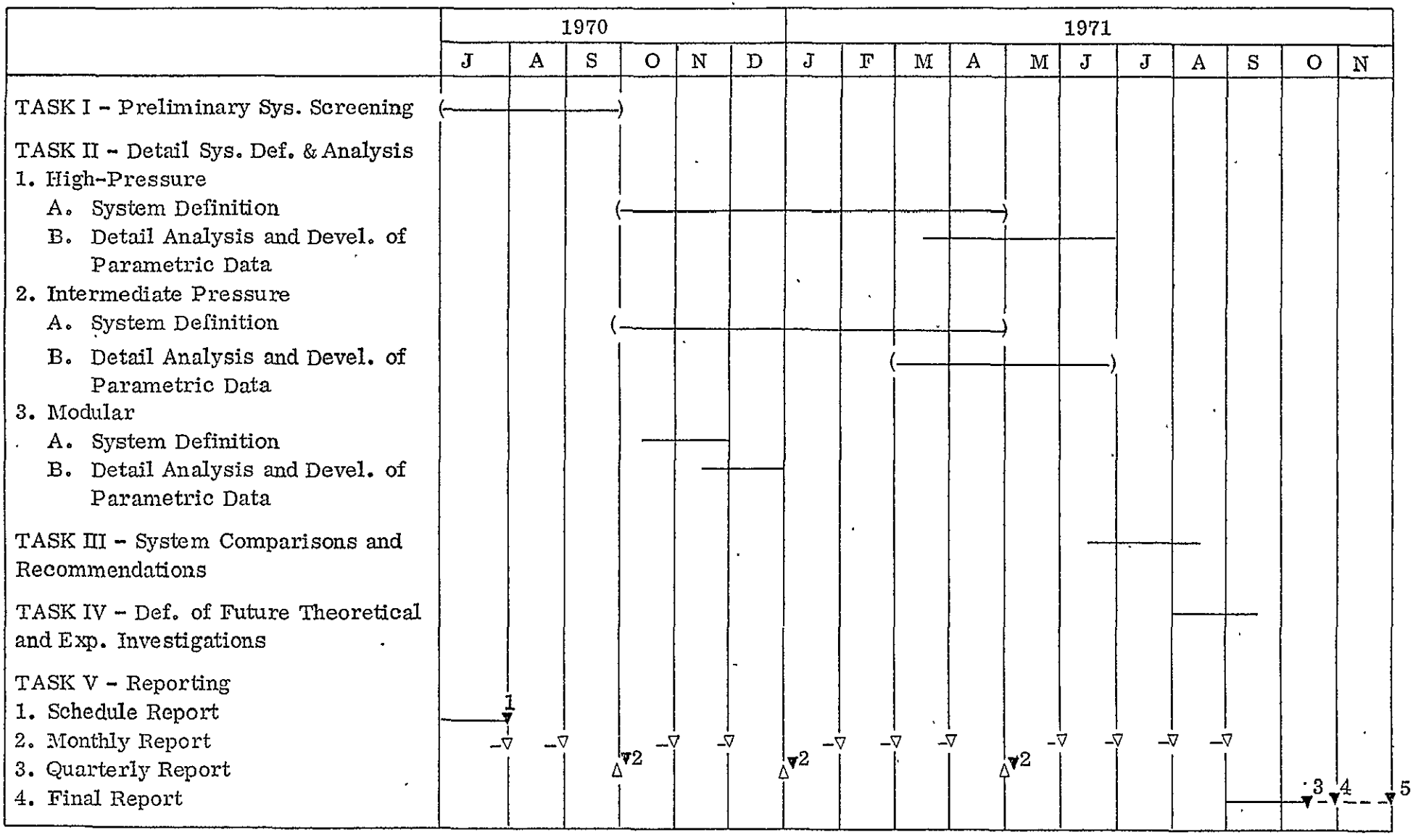
The reasons for the extension request are presented below.

1. Computer facility scheduling problems resulted in a delay in developing the line and receiver tank computer programs.

2. Difficulty has been encountered in finding a suitable high pressure transfer system for comparison with other types of systems. This has resulted in a delay in initiating the detail design task for this system.

3. A contract extension and spreading of tasks at this time allows for a more realistic tie-in between the present program and the Space Station and Space Shuttle programs which it supports.

The following work was performed under the Convair IRAD programs during the past quarter. Hardware design was essentially completed on the bellows, metallic diaphrag and paddle vortex subcritical transfer systems. With respect to the bellows system, a number of different configurations were initially considered. Each was designed to contain a liquid volume of 42.5 ft^3. The overall effects on system weight of bellows length to diameter (L/D) ratio and location of the helium pressurant bottle (within the tank or external to the tank) were determined. The minimum weight system was determined to consist of a bellows with an L/D ratio of 1.5. There was only a small weight difference between locating the helium bottle inside and outside of the propellant tank. The final selection between the two schemes would thus be on the basis of ease of access and cost for a particular application. Thus parametric weight



( ) Planned Convair IRAD Tasks

Milestones:
1. Schedule Report Submitted to NASA/MSFC
2. Quarterly Review as Requested by MSFC
3. Draft of Final Report Submitted
4. Final Review
5. Final Report Published

Figure 1-1. Revised Program Master Schedule

data were generated which are applicable to both concepts. Overall system parametric weight data were generated for the basic L/D = 1.5 configuration over a range of volumes corresponding to spherical tank diameters of 25 to 150 inches and expulsion pressures of 50 to 500 psia.

In the case of the paddle vortex system it was determined that hardware weights and power requirements were competitive with other systems at the nominal 42.5 ft^3 tank volume. However, there is still some uncertainty as to the magnitude of fluid residuals for this system and further work will be accomplished in this area during the next reporting period.

The receiver tank chilldown program was modified to extend its' usefulness over the range of pressures from 1 psia to 1,000 psia and, in the case of H_2 densities, from 0.00033 to 4.55 lb/ft^3 by incorporating additional properties data. Initial runs with a combination line and receiver tank chilldown program indicated that the line chilldown in most cases occurred only over a small fraction of the total transfer time and did not significantly effect the chilling and filling of the receiver tank. Transfer line and receiver tank chilldown were thus investigated further using the line and receiver programs separately.

In order to preform the line chilldown analysis, parametric line weights were defined for both Al Aly and CRES lines over a range of diameters from 0.5 to 3 inches and for operating pressures up to 2,000 psi. The chilldown analysis performed indicates that only at maximum line lengths (200 ft) and minimum inlet pressures or flow rates is the line chilldown likely to become a significant factor.

Analysis performed on receiver tank chilldown shows that the maximum pressure which can occur in a locked-up (no venting) tank is a strong function of the rate of heat transfer between the tank wall and fluid and also the inflow rate. As an example, maximum pressures ranged from 66 psia to 189 psia for the same 42.5 ft^3 receiver tank when changing the wall to fluid heat transfer coefficient from 2 to 20 $Btu/hr\text{-}ft^2\text{-}°F$.

High pressure transfer system definition work was continued. To date six different schemes have been considered. At this point in time a system which transfers supercritical fluid by heating the supply fluid shows the most promise. It appears to be an order of magnitude lighter than its nearest high pressure competitor.

Supercritical fluid transfer methods, including receiver bottle refrigeration, simple blowdown and pumping have proven to be unfeasible. Preliminary analysis of a vortex assisted transfer scheme also indicates this system will be unfeasible when compared to the supply heating method.

During the next month, the high pressure system detail analysis and definition of parametric data tasks will be initiated and the basic analysis will be completed on receiver tank and line chilldown over a range of line lengths and sizes and receiver tank sizes and design pressures.

SECTION 2

BELLOWS SYSTEM DESIGN

The work presented in this section was performed under the Convair 1971 Independent Research and Development (IRAD) program and is reported herein only for reference as it relates to the pertinent subject of cryogenic propellant transfer in space.

The section contains a summary of the design and development of weight data associated with a positive expulsion system using a metallic bellows. The basic concept is illustrated in Figure 3-8 of Reference 2-1. The design tasks, as reported herein, were to define an optimum configuration, including any pressurization storage and thermal control provisions, and then to develop detailed weight and size data over a range of propellant volumes corresponding to spherical tank diameters of 25 to 150 inches and expulsion pressures of 50 to 500 psia. Details are presented in the following paragraphs.

2.1 SYSTEM DEFINITION

As part of the system definition task a number of different configurations were considered in order to determine a reasonably optimum system. Sizes and operating conditions, as defined for the baseline space station resupply and presented in Ref. 2-1, were used. Each bellows system was designed to contain a total liquid volume of 42.5 ft^3 and be capable of expulsion into a receiver tank at 100 psia. Safety factors of 1.33 on allowable and 1.67 on ultimate were used in all designs. Also, since CRES bellows are the best candidates under the present state-of-the-art, the basic pressure vessel was assumed to be 347 CRES for compatibility of required welding processes between the tank and bellows.

Analysis of both pressurization and thermal control aspects were initially required in order to obtain a basis for pressure vessel design and to determine requirements for vacuum jacketing of the basic bellows tank. One of the trade-offs to be considered is whether the pressurant bottle should be located external to the system or packaged as part of the bellows tankage. Also, the effects of overall tank volume efficiency on total weight is important and thus the weight of excess pressurant storage requirements must be included in any weight comparisons. Helium stored at high pressure was assumed to be the pressurant. With respect to pressurization and thermal control requirements, the critical system would be that using liquid hydrogen; therefore, this propellant was chosen for the baseline case.

In developing pressurant requirements, the Epstein correlations, as reported in Ref. 22, were used. The basic equations are presented below.

$$W_P = (CF)\frac{P_t V_t}{R_G T_o} \tag{2-1}$$

$$CF = \left\{\left(\frac{T_o}{T_s} - 1\right)\left[1 - e^{-P_1 C^{P_2}}\right]\left[1 - e^{-P_3 S^{P_4}}\right] + 1\right\} \tag{2-2}$$

where

$$C = \frac{\rho_w C^o_{p_w} t_w T_s R_G}{p^o_G c^o_{p_G} D_T}$$

$$S = \frac{h_c \theta_T T_s R_G}{p^o_G c^o_{p_G} D_T}$$

T_o = pressurant inlet temperature

T_s = initial temperature of propellant

ρ = density

c_p = specific heat at constant pressure

t = tank thickness

R = ideal gas constant of pressurant

p = absolute tank pressure during expulsion

h_c = gas-to-tank wall heat transfer coefficient

θ_T = total propellant outflow time

D_T = tank diameter

P_1 = correlation constant; use 0.33 for H_2 and 0.775 for O_2

P_2 = correlation constant; use 0.281 for H_2 and 0.209 for O_2

P_3 = correlation constant; use 4.26 for H_2 and 3.57 for O_2

P_4 = correlation constant; use 0.857 for H_2 and 0.790 for O_2

Subscripts

G = gas

w = wall

Superscripts

o = variables at a temperature equal to the inlet pressurant temperature and a pressure equal to expulsion pressure

It is noted that in the complete equation given in Reference 2-2, there is another term to account for the effect of external heat transfer. Calculations showed, however, that for the present conditions where superinsulation is used this term is negligible and was therefore not included here.

Using Equation 2-2 and heat transfer coefficients from Reference 2-3, the collapse factor (CF) was determined to be 2.5 for the present case. This assumes a pressurant temperature of 500°R and a propellant temperature of 40°F. The expulsion pressure was taken to be 200 psia. Other data used for input to the equations are presented below.

$C^{o}_{p_G}$ = 1.25 Btu/lb-°F, t_w = 0.2 in.

ρ_w = 0.1 lb/in^3, θ_T = 1,000 sec

$C^{o}_{p_w}$ = 0.211 Btu/lb-°F, D_T = 52 in.

h_c = 23.3 Btu/hr-ft^2-°F

It is noted that the above values are based on a nominal estimate of real tank conditions and are chosen to provide a reasonable input to the overall bellows design. From Equation 2-1 the pressurant requirement is calculated to be 15.8 lb. Then, assuming 3,000 psia helium storage at 500°R and a useful storage bottle pressure decay to 500 psia, a helium bottle volume requirement of 8.8 ft^3 was determined. This size bottle would be applicable to the nominal expulsion of 42.5 ft^3. It is noted that for similar operating conditions, the helium storage volume is essentially proportional to the actual propellant storage volume. This relation was used in evaluating the various design configurations.

Next, the thermal control requirements were defined. Based on storage for up to seven days prior to use, calculations made for spherical bottles, in connection with the surface tension device, indicated that use of a vacuum jacketed superinsulated tank without venting would be optimum. For the nominal 42.5 ft^3 bottle, approximately 6 inches of "Superfloc" (Ref. 2-4) high performance insulation was found to be required in order to limit the tank pressure rise to a maximum of 100 psia. This assumes an initial ullage volume of 5%.

Pressure rise calculations were made using the data from Reference 2-5. The insulation thickness thus determined (6 inches) was then used to define vacuum jacketing required for the baseline bellows system. A sandwich core type construction similar to that reported in Reference 2-6 was used for the vacuum shell.

Other design ground rules and assumptions used in developing the overall bellows system weight data are presented below.

Bellows Material:	347 CRES
Bellows Wall Thickness:	.010 inches
Bellows Type:	Nested-hydroform
Convolution Width:	1.0 inches

The above data were taken primarily from Reference 2-7.

The different bellows system configurations considered are discussed below. The configurations were chosen to determine the overall effects on system weight of bellows and pressurant bottle design and packaging and bellows length to diameter (L/D) ratio.

The first system analyzed is illustrated in Figure 2-1, and a corresponding weight statement is presented in Table 2-1. The bellows L/D was taken to be 2/1, which was considered to be a reasonable maximum value for the present volume requirement. The helium pressurant bottle in this configuration is designed to be part of the CRES pressure vessel and represents a significant part of the overall system weight. It is noted that a false bulkhead with a foam filler is used to minimize the pressurant required. For this case the helium is further assumed to be stored at the LH_2 temperature. Calculations using Equations 2-1 and 2-2 showed that such a storage condition resulted in essentially the same pressurant volume requirement as for the ambient storage case; i.e., the total pressurant required is greater but the storage density is also greater by an equivalent amount than for the higher temperature condition.

Since the weight of the helium storage bottle for this system was quite high, a design was developed where the bottle is separate from the tank and thus could be made from high strength titanium.

This design is presented in Figure 2-2, and a corresponding weight statement is given in Table 2-2. The primary difference between configurations A and B is the location and material associated with the helium storage sphere. Also, elliptical rather than spherical bellows tank bulkheads are used in order to minimize waste volume at the tank ends.

The Table 2-2 data shows an overall weight reduction from that of Configuration A. It is noted that there is some increase in the expulsion chamber weight due to the separation of the bottle from the tank and the addition of a larger false bulkhead.

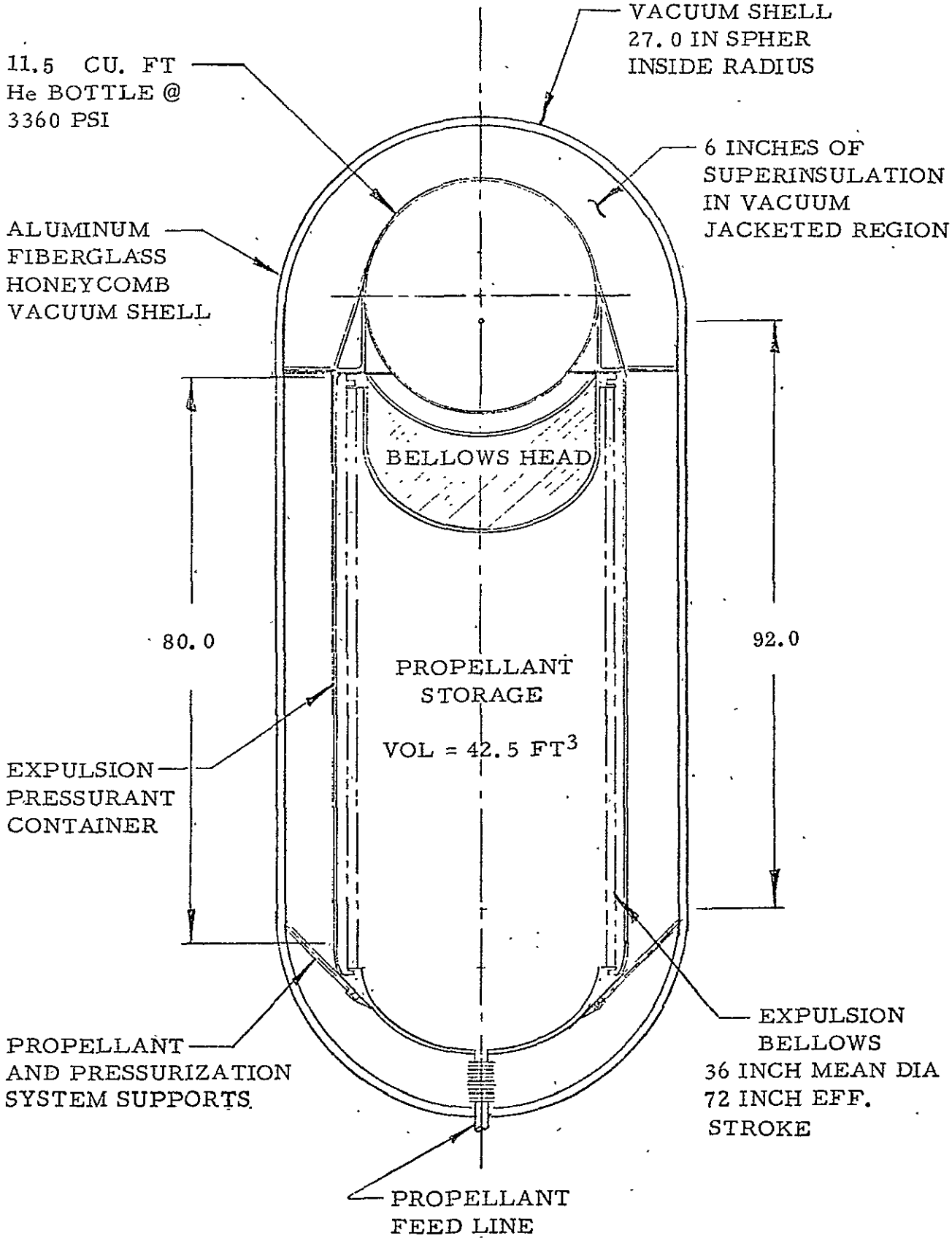


Figure 2-1. Bellows System, Configuration A

Table 2-1. Weights Statement for Configuration A. He Bottle Inside Vacuum Shell W/Sep. Pressure Shell

PRESSURE SHELL		
Cylinder (40" dia, 80" long)	138.0	
Conic Adapter	23.4	
Bulkhead Transition	6.9	
Bellows Stops	6.0	
	173.3	173.3
EXPULSION CHAMBER		
Bellows (36" dia × 72 Eff)	67.5	
Aft Bulkhead	13.4	
Aft Joint Ring	20.1	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	9.2	
Spherical Seg.	6.7	
Elliptical Seg.	13.4	
Foam	59.8	
Forward Joint Ring	10.1	
	203.2	203.2
VACUUM SHELL		
Cylinder (54" dia × 92 " long)	80.6	
Spherical Bulkheads (2)	53.7	
Weld Lands, Rings, etc.	32.6	
Adhesive Weight	31.4	
	198.3	198.3
He PRESSURE BOTTLE (CRES)(34" dia)		394.0
SUPERINSULATION		88.6
	GRAND TOTAL	1057.4

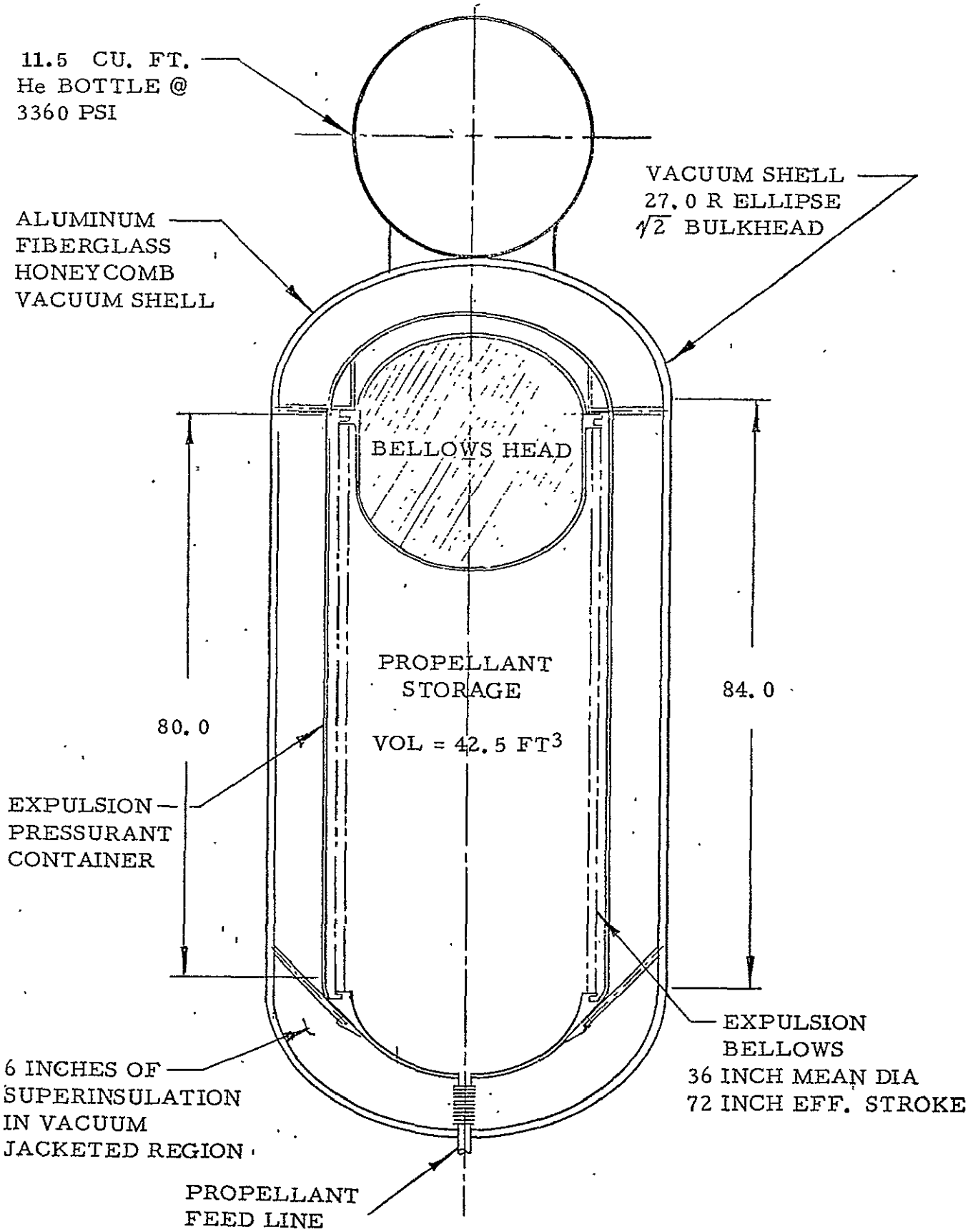


Figure 2-2. Bellows System, Configuration B

Table 2-2. Weights Statement for Configuration B, He Bottle Outside Vacuum Shell W/Sep. Pressure Shell

PRESSURE SHELL		
Cylinder (40" dia × 80" long)	138.0	
Forward Bulkhead	20.4	
Bulkhead Transition	6.9	
Bellows Stops	6.0	
	171.3	171.3
EXPULSION CHAMBER		
Bellows (36" dia × 72" eff)	67.5	
Aft Bulkhead	13.4	
Aft Joint Ring	20.1	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	9.2	
Elliptical Seg. (2)	26.8	
Foam	111.6	
Forward Joint Ring	10.1	
	261.7	261.7
VACUUM SHELL		
Cylinder (54" dia × 84" long)	82.6	
Elliptical Bulkheads (2)	43.6	
Weld Lands, Rings, etc.	32.6	
Adhesive Weight	29.0	
	187.8	187.8
He PRESSURE BOTTLE (Titanium 34" dia)		191.0
SUPERINSULATION		82.9
GRAND TOTAL		894.7

However, this weight increase is more than offset by the significant reduction in helium storage weight.

The next step was to consider a design where the helium bottle would be located inside the overall tank envelope, but would not be part of the tank and could thus be made from titanium. One such system is shown in Figure 2-3, with a corresponding weight statement in Table 2-3. This system illustrates a somewhat unique approach, in that a separate inner shell surrounding the bellows is not employed. This allows the elimination of the weight associated with this pressure shell, but adds additional weight for bellows stops and guides. It is also noted that the vacuum jacket area must now be pressurized with helium which adds considerably to the pressurant storage weight. The overall result is that this system is heavier than previous ones. It would also be a disadvantage to pressurize the area containing

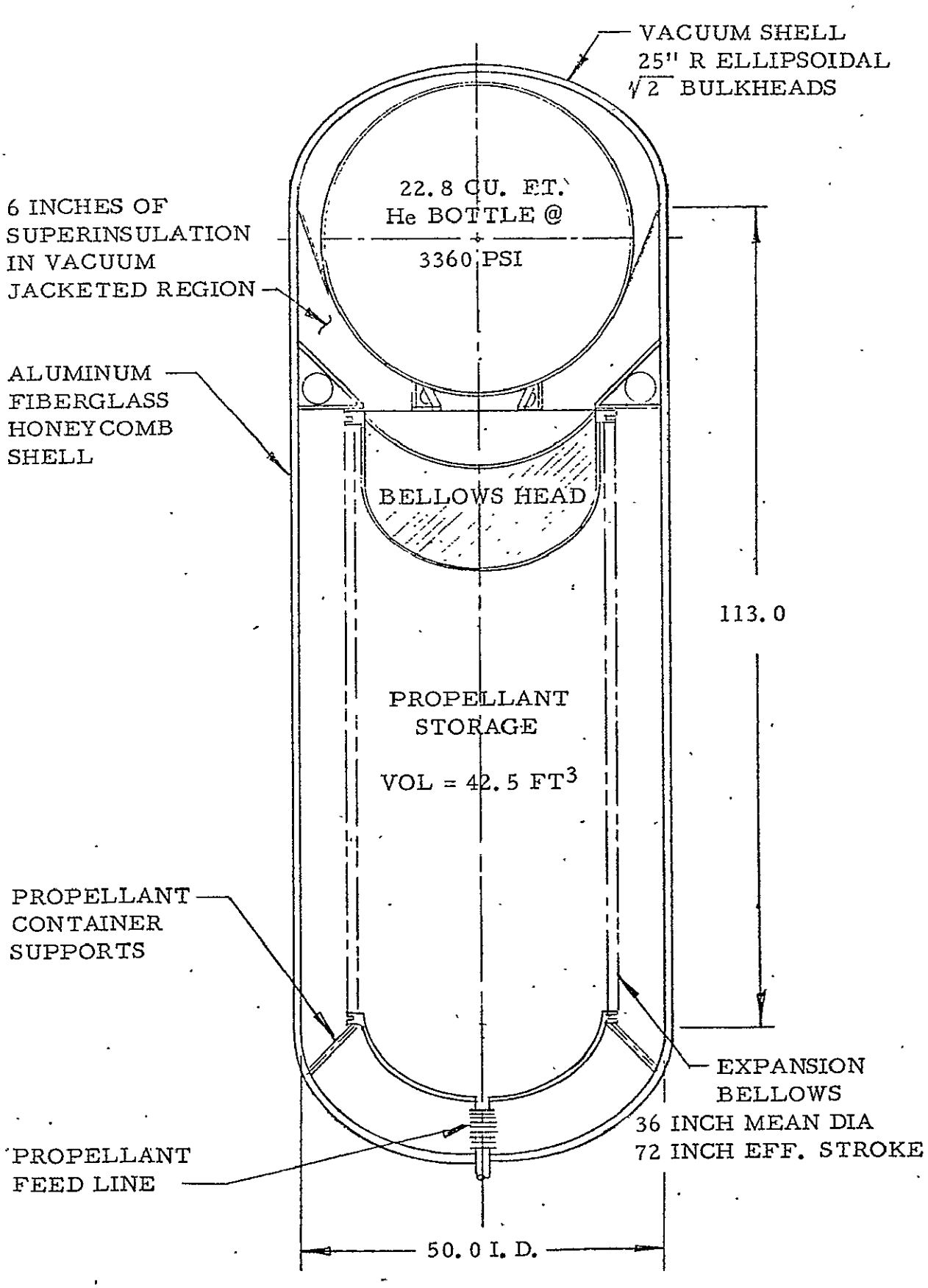


Figure 2-3. Bellows System, Configuration C

Table 2-3. Weights Statement for Configuration C, Combined Vacuum and Pressure Shell, He Bottle Inside

VACUUM PRESSURE SHELL		
Cylinder (50" dia × 113" long)	237.3	
Elliptical Bulkhead (2)	86.8	
Weld Lands, Rings, etc.	34.1	
Adhesive Weight	32.3	
	390.5	390.5
EXPULSION CHAMBER (Same as Configuration A)		203.2
ADDITIONAL EXPULSION CHAMBER GUIDES AND STOPS		
Bellows Stops	12.0	
Guides	30.0	
	42.0	42.0
He PRESSURE BOTTLE (Titanium 43" dia)		386.0
SUPERINSULATION		87.8
GRAND TOTAL		1109.5

superinsulation, since some damage to the insulation could result. Therefore, this type of system was eliminated from further consideration.

A relatively conventional configuration with the helium bottle separate but located inside the tank was then analyzed as configuration D. A weight statement is presented in Table 2-4. It is noted that, due to anticipated packaging limitations, a cylindrical helium bottle was used. Due primarily to this factor Configuration D still results in higher weight than Configuration B, which has an external spherical bottle. A check was then made on the potential of using a spherical bottle in a D type configuration, as presented in Figure 2-4. Weight data are presented in Table 2-5 showing a weight reduction. An examination of the data between Tables 2-5 and 2-2, however, shows that the main difference is now the added bulkhead weight for Configuration B. If it is assumed that the tank is prepressurized, such that waste tank space is not a factor, then Configuration B becomes the lower weight system. This is illustrated by the data presented in Table 2-6.

A summary of the results discussed above indicates only a slight difference in weight between systems with and without the pressurant storage inside the propellant tank. Therefore, other considerations such as convenience of access, fabrication

Table 2-4. Weights Statement for Configuration D, Sep. He Bottle Inside Vacuum Shell

PRESSURE SHELL		
Cylinder (40" dia × 87" long)	150.0	
Spherical Bulkhead	17.4	
Bulkhead Transition	6.9	
Bellows Stop	6.0	
	180.3	180.3
EXPULSION CHAMBER		
Bellows (36" dia × 72" eff)	67.5	
Aft Bulkhead	13.4	
Aft Joint Ring	20.1	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Segment	18.4	
Elliptical Segment	26.8	
Forward Joint Ring	10.1	
	159.3	159.3
VACUUM SHELL		
Same as Configuration A		198.3
He PRESSURE BOTTLE		
(Titanium 28" dia w/26.5" cyl. section)		322.0
INSULATION		
Same as Configuration A		88.6
GRAND TOTAL		948.5

and cost will dictate any final selection.

The next step in the tradeoff analysis was to investigate the effects of different bellows L/D ratios. System designs were developed for L/D ratios of 1.5 and 1.0 in order to compare with the L/D = 2 systems previously analyzed. Configurations are presented in Figures 2-5 and 2-6, and weight statements in Tables 2-7 and 2-8. It is seen that the Figure 2-5 system, with an L/D of 1.5, represents the minimum weight. This is primarily due to the overall packaging efficiency of this design and the optimum relation of containing a cylindrical bellows in a tank containing ends which are not flat. That is, a reduction in height with an increase in diameter allows the pressure shell and vacuum jacket to approach a relatively structurally efficient spherical shape, but with an increase in waste volume at the end of the cylindrical bellows.

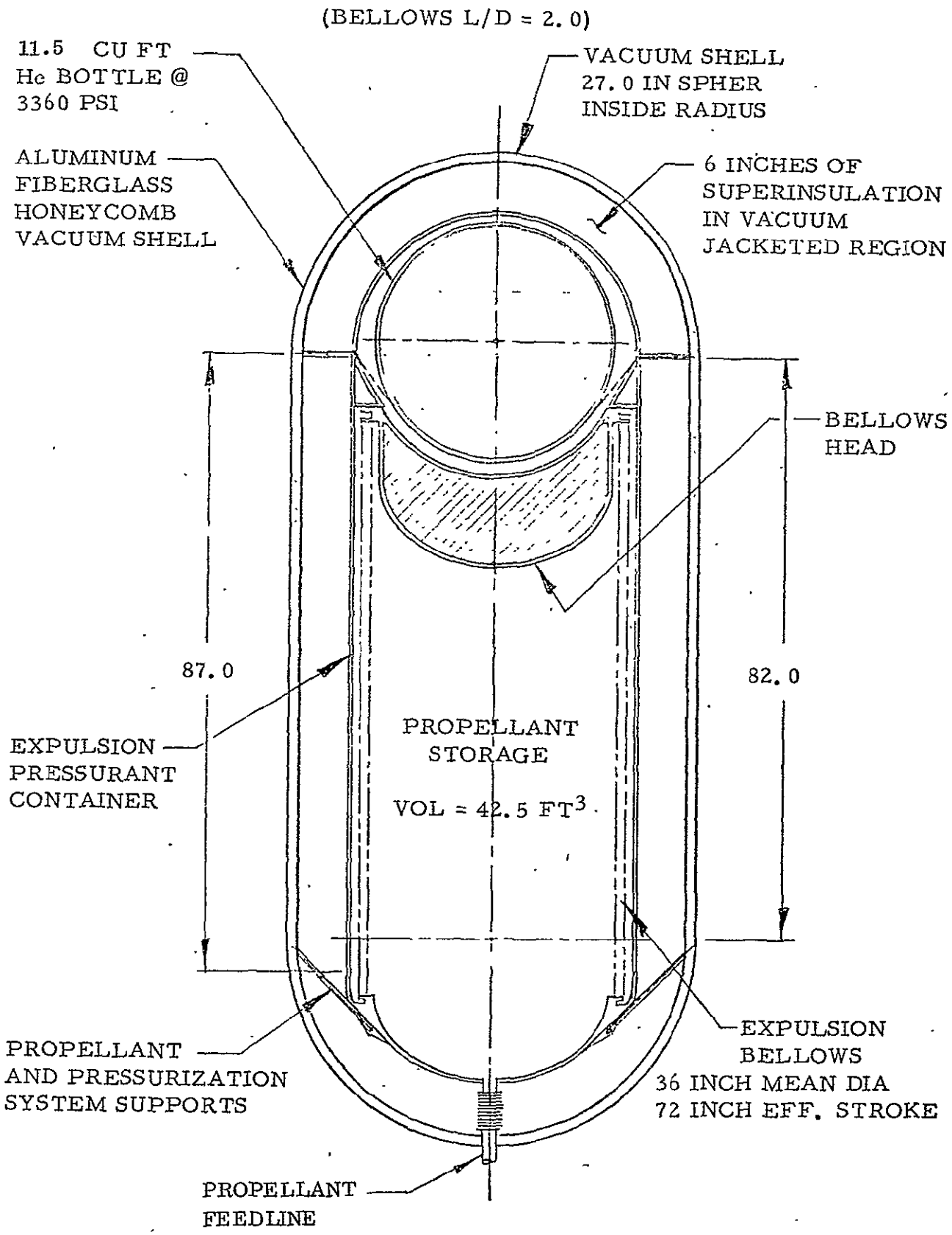


Figure 2-4. Bellows System, Configuration D

Table 2-5. Alt. Configuration D for a 34-Inch He Bottle (Spherical) at 3360 psi

PRESSURE SHELL	(CONF D)	180.3
EXPULSION CHAMBER	(CONF A)	203.2
VACUUM SHELL	(A or D)	198.3
He PRESSURE BOTTLE	(CONF B)	191.0
SUPERINSULATION	(CONF A)	88.6
	GRAND TOTAL	861.4

Table 2-6. Helium Used to Initially Fill the Expulsion Pressurant Chamber to a Pressure Equal to the Propellants, (Foam in the Bellows Head Eliminated)

CONFIGURATION B WEIGHT =	894.7
	-125.0
	769.7 lbs
CONFIGURATION D WEIGHT =	861.4
	-66.5
	794.9 lbs

This (L/D = 1.5) system was thus chosen for further analysis and development of parametric data over the full range of volumes and expulsion pressures required. These parametric data are presented in the following paragraph.

2.2 DETAIL ANALYSIS AND DEVELOPMENT OF PARAMETRIC DATA

Based on the system shown in Figure 2-5, parametric weight data were generated over the range of propellant volumes from 4.7 ft^3 (25 inch dia sphere) to 1020 ft^3 (150 inch dia sphere) and tank pressures from 50 to 500 psi. The analyses presented in the previous paragraph showed only a small difference in system weight between locating the helium bottle inside or outside of the tank. Data were thus generated in a form which is essentially independent of the pressurant location. Also, weights for an intermediate bulkhead are presented separately so that its' use can be independently assessed for any particular design situation. It is noted that the basic weight generation methods are the same as discussed in the previous paragraph. Bellows design and sizing are based on the use of existing fabrication

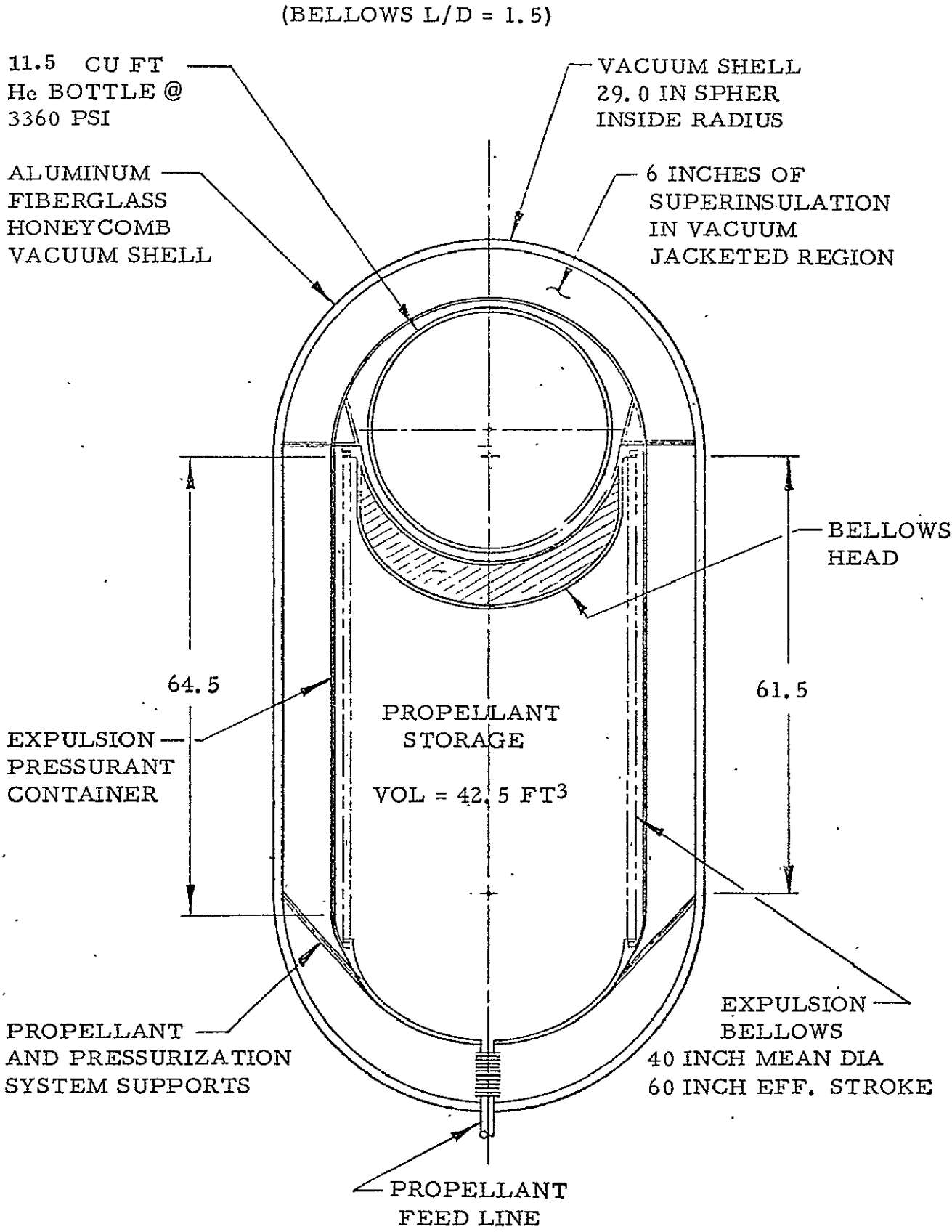


Figure 2-5. Bellows System, Configuration E

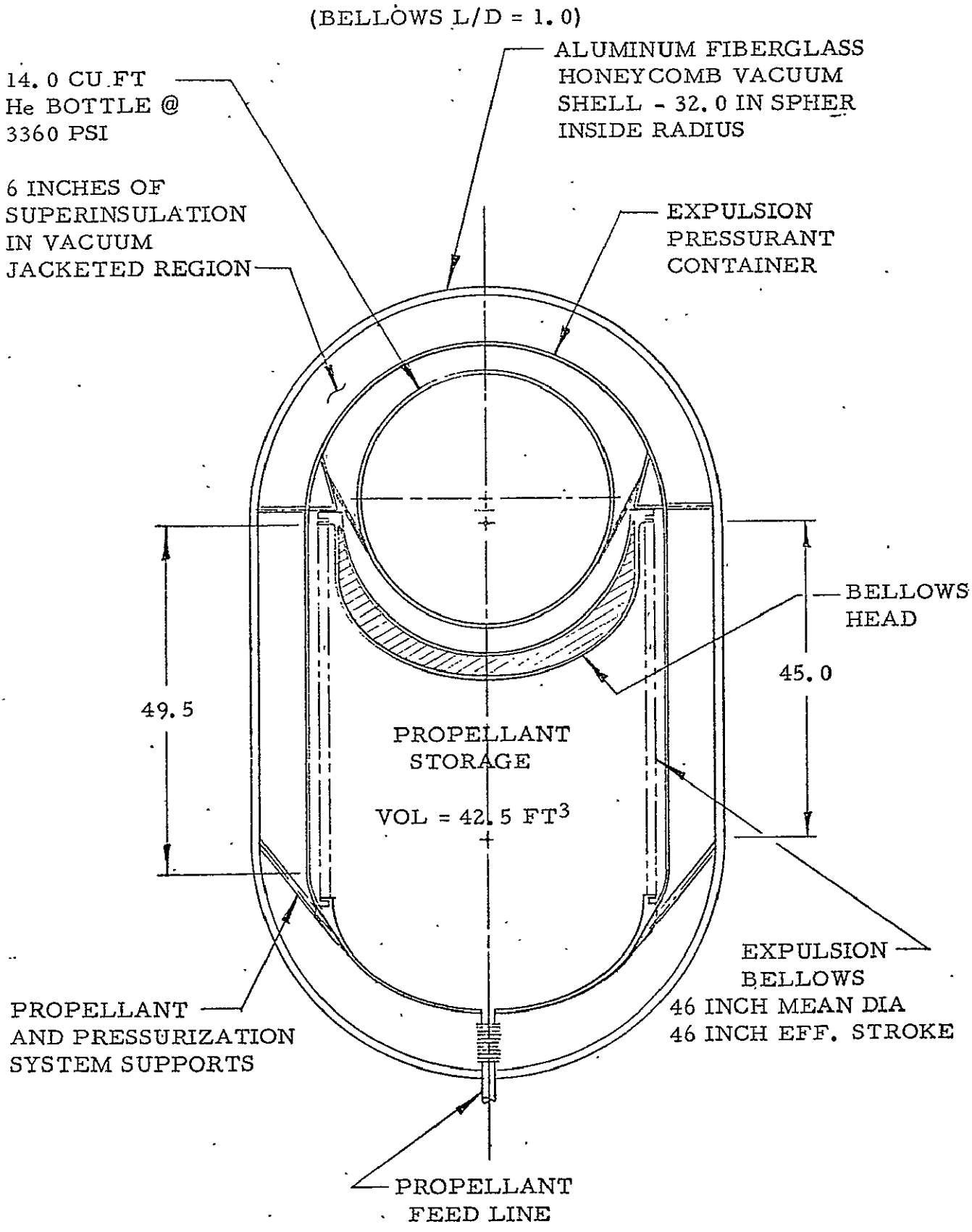


Figure 2-6. Bellows System, Configuration F

Table 2-7. Weights Statement for Configuration E, Bellows L/D = 1.5

PRESSURE SHELL		
Cylinder and Trans.	132.0	
Spherical Bulkhead	20.6	
Bellows Stop	6.0	
	158.6	158.6
EXPULSION CHAMBER		
Bellows (40" dia × 60" eff)	61.4	
Aft Bulkhead	17.4	
Aft Joint Ring	23.0	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	8.5	
Elliptical Seg.	17.4	
Spherical Seg.	12.6	
Foam	35.7	
Forward Joint Ring	11.2	
	190.2	190.2
VACUUM SHELL		
Cylinder	66.4	
Spherical Heads	63.5	
Adhesive	27.3	
Rings, Weld Lands, Brackets, etc.	34.2	
	191.4	191.4
He PRESSURE BOTTLE		191.0
SUPERINSULATION		83.6
GRAND TOTAL		814.8

Table 2-8. Weights Statement for Configuration F, Bellows L/D = 1.0

PRESSURE SHELL		
Cylinder and Trans.	135.0	
Spherical Bulkhead	30.0	
Bellows Stop	6.0	
	171.0	171.0
EXPULSION CHAMBER		
Bellows (46" dia × 46" eff)	54.7	
Aft Bulkhead	26.8	
Aft Joint Ring	28.3	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	10.5	
Elliptical Seg.	26.8	
Spherical Seg.	20.6	
Foam	32.8	
Forward Joint Ring	12.9	
	216.4	216.4
VACUUM SHELL		
Cylinder	54.8	
Spherical Heads	79.0	
Adhesive	29.4	
Rings, Weld Lands, Brackets, etc	36.6	
	199.8	199.8
He PRESSURE BOTTLE		266.5
SUPERINSULATION		83.8
GRAND TOTAL		897.5

procedures. It was felt that even though the actual tooling to fabricate the larger sizes did not presently exist such tooling could be reasonably developed and would show up as a cost item when comparing the bellows with other subcritical transfer concepts.

Weights of the pressure shell and expulsion bellows as an integral system are presented in Figure 2-7. Corresponding vacuum shell weights are presented in Figure 2-8. These are the two main weight components of the overall system, excluding pressurization. For comparison, a curve of spherical vacuum shell weight of the same type of construction as for the cylinder is also shown in Figure 2-8.

It is noted that the jacket weight is presented as a function of its diameter. For a given propellant volume this diameter will be a function of the inner tank radius and the insulation thickness. A curve of inner tank radius versus propellant volume is presented in Figure 2-9.

The weight of a second bulkhead and foam filler is presented in Figure 2-10. Also, for reference the weight of the individual bellows assembly is presented in Figure 2-11.

Insulation volume for a given tank is determined from Figure 2-12 and the weights of titanium helium bottles to be used in the study are presented in Figure 2-13. The total volumes which are available for pressurant storage within the tank are presented in Figure 2-14. The total pressure shell volume is the sum of the propellant and pressurant storage volumes presented in Figure 2-14. The double bulkhead volume is found from the difference between the two curves presented in Figure 2-14.

Following is an example of the use of the data presented in Figures 2-7 through 2-14. It is desired to determine the total weight of a bellows system designed to operate at 200 psia and contain 42.5 ft^3 of fluid. The insulation is to be 6 inches thick, and to allow for weld lands and mounting bosses, the vacuum shell is to be 14 inches larger in diameter than the inner shell. A double bulkhead is to be employed. The resulting weight statement is presented in Table 2-9. Comparing these data with that presented in Table 2-7 shows 605 lb versus 624 lb. Most of this difference is due to a refinement in the assessment of insulation volume for the present calculations.

Table 2-9. Summary Weight Data for Example Case

Item	Size or Pertinent Dimension(s)	Unit Weight, lb	Total Weight, lb	Source of Data
Pressure Shell and Expul. Bellows	42" dia	-	300	Fig. 2-7 & 2-9
Vacuum Jacket	56" dia	-	187	Fig. 2-8
Double Bulkhead	-	-	47	Fig. 2-10
Insulation	54.5 ft^3	1.3 lb/ft^3	71	Fig. 2-12
		TOTAL	605 lb	

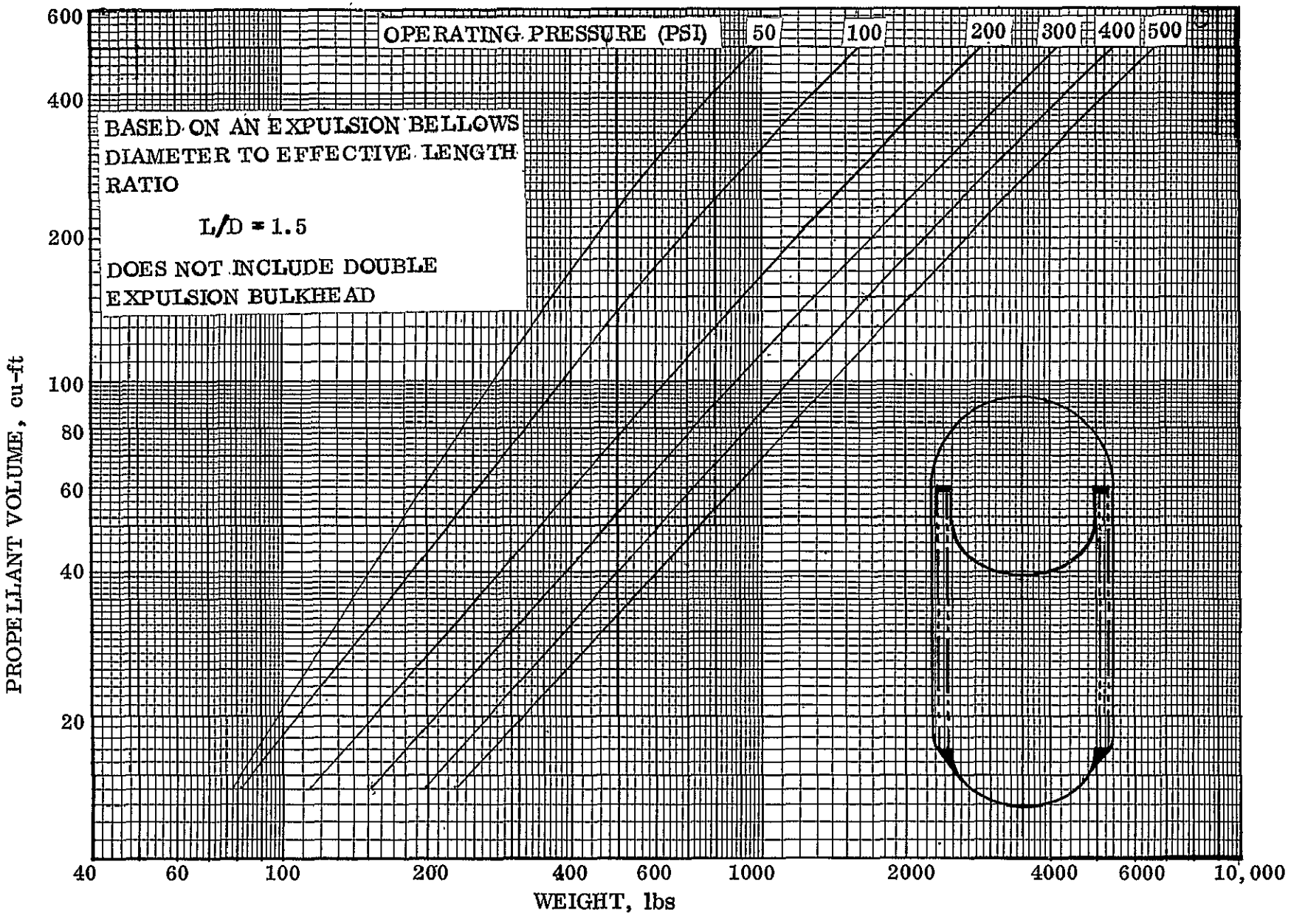


Figure 2-7. Pressure Shell and Expulsion Bellows Weight Data

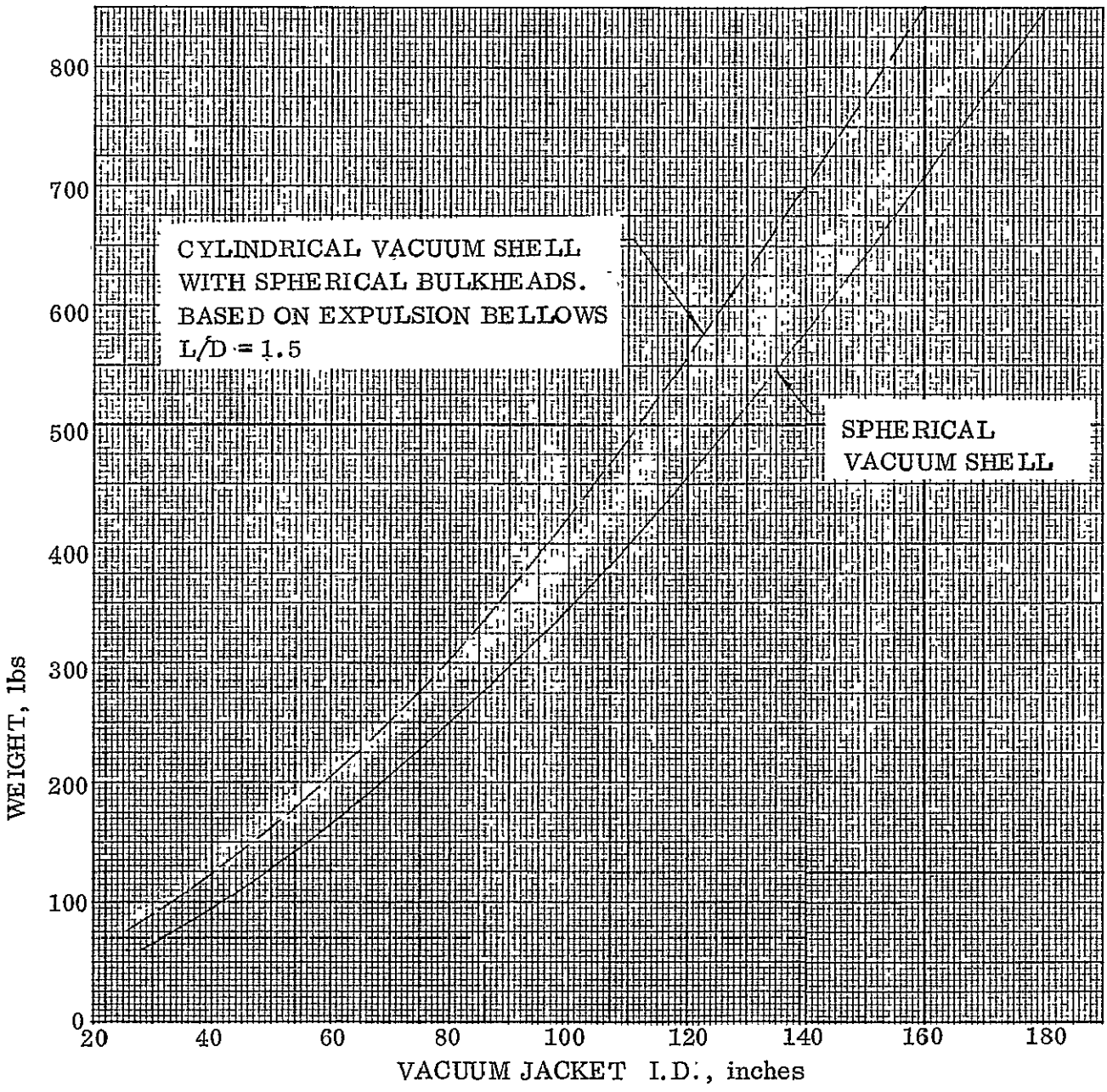


Figure 2-8. Parametric Vacuum Jacket Weights

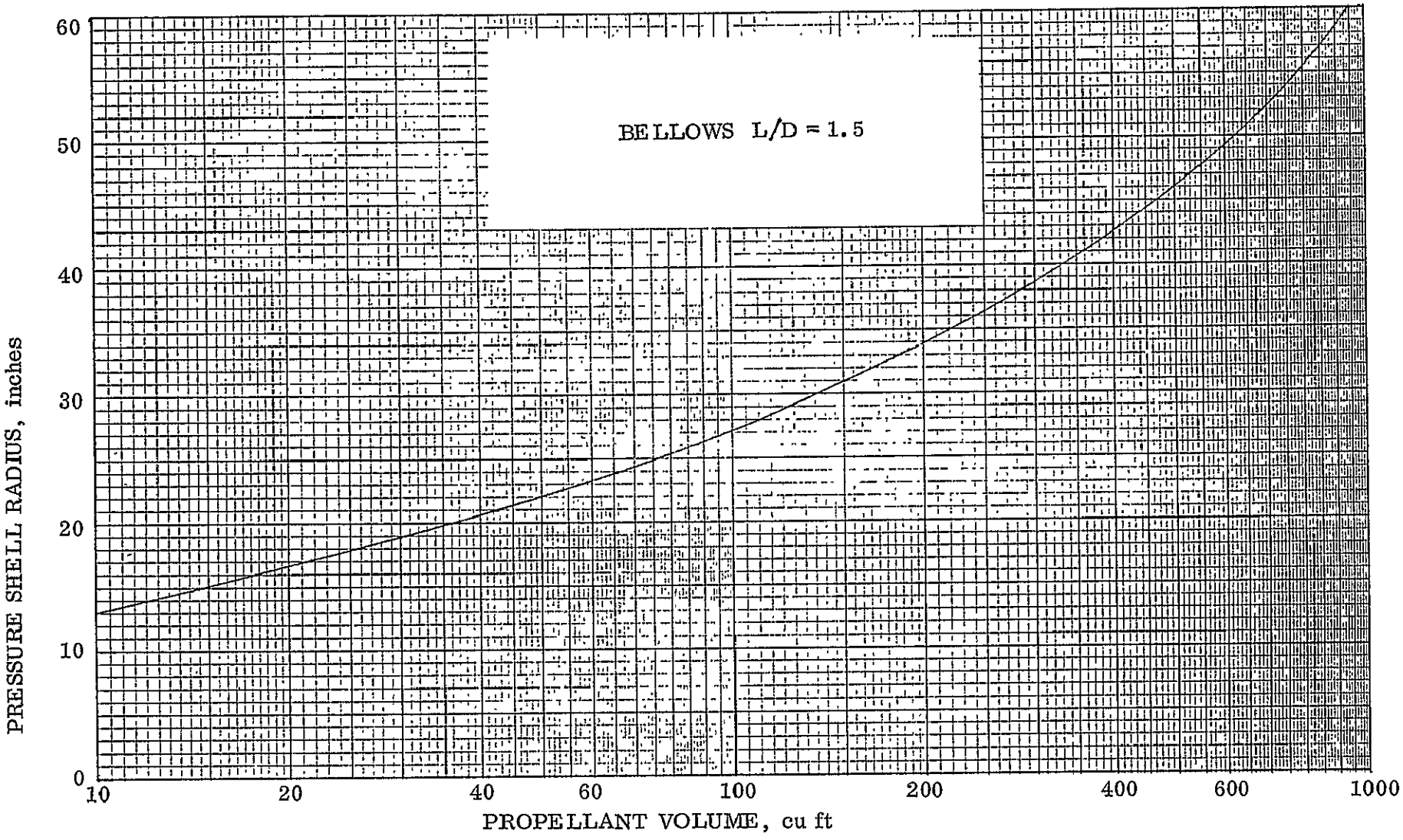


Figure 2-9. Bellows Expulsion System Pressure Shell Radius Vs. Propellant Volume

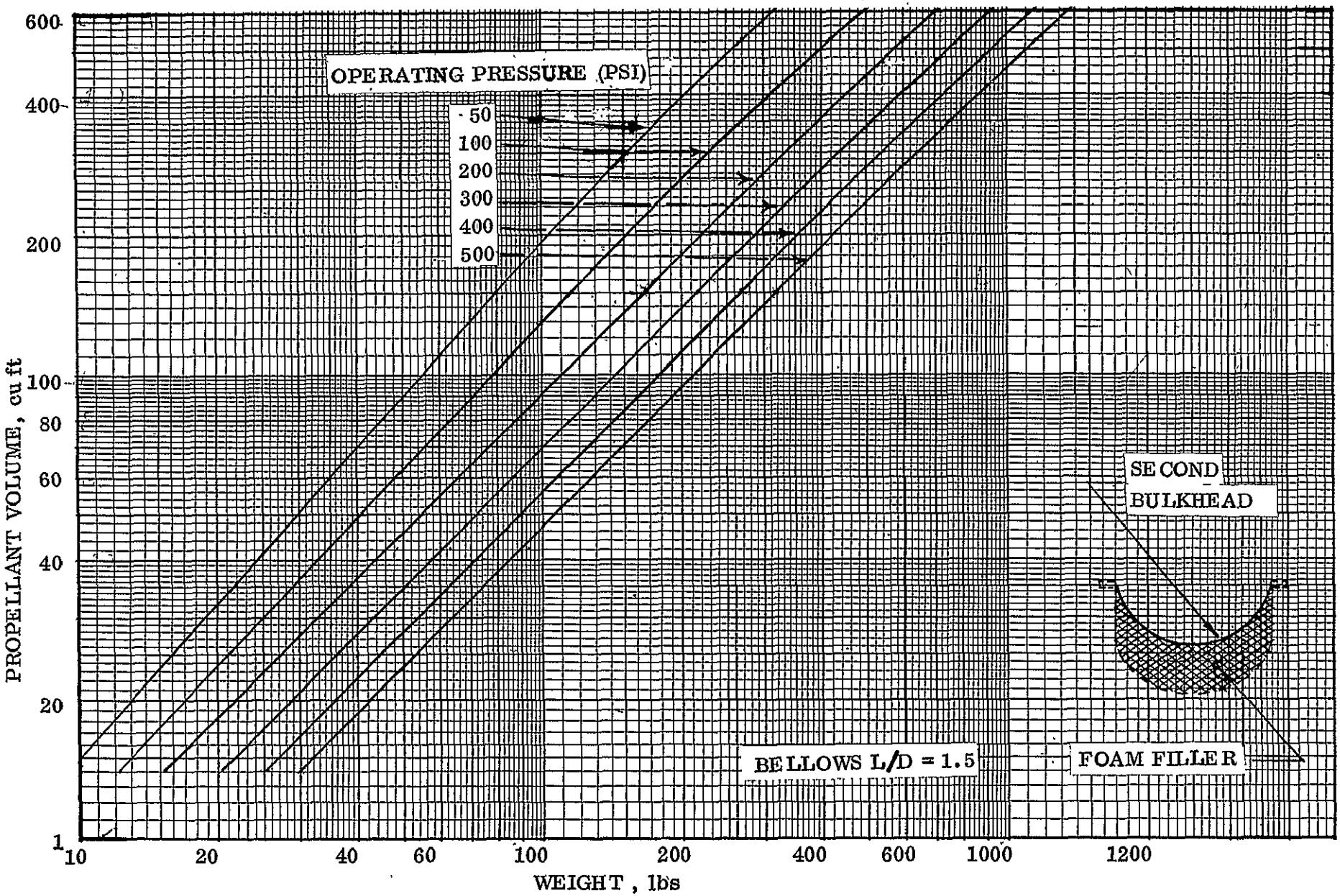


Figure 2-10. Double Expulsion Bulkhead Weight Data

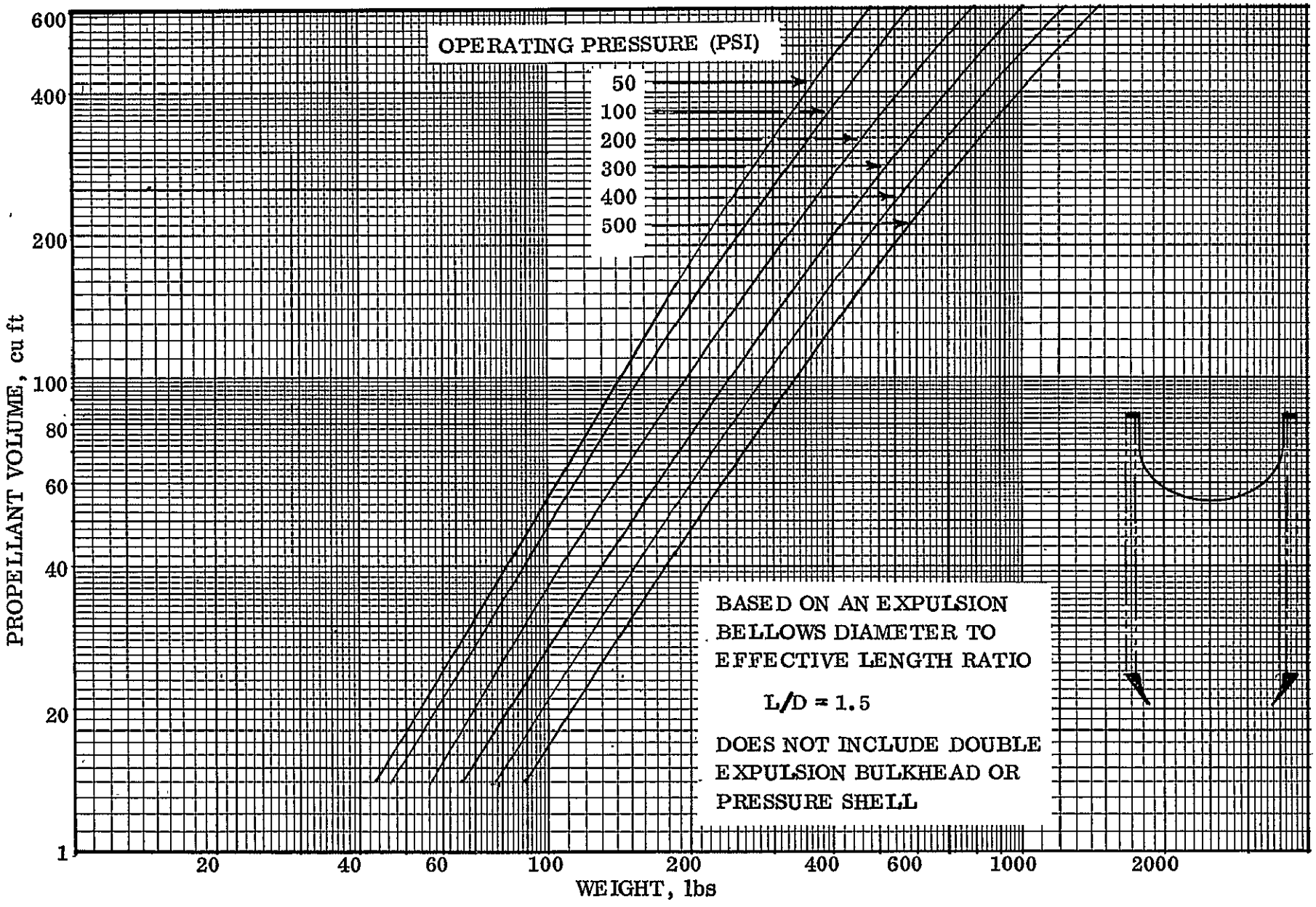


Figure 2-11. Bellows and Expulsion Head Weight Data

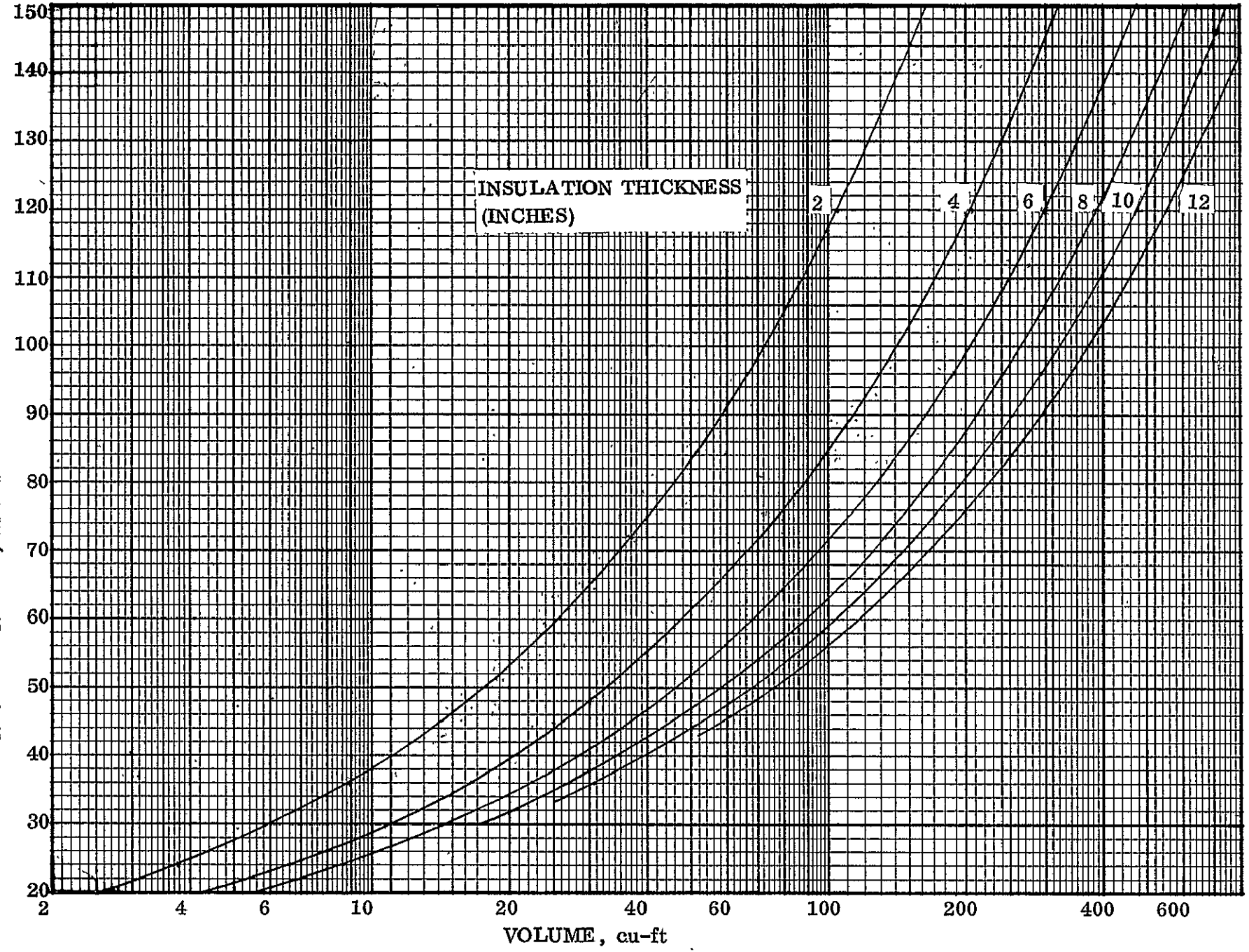


Figure 2-12. Insulation Volume For Bellows Expulsion Systems With L/D = 1.5

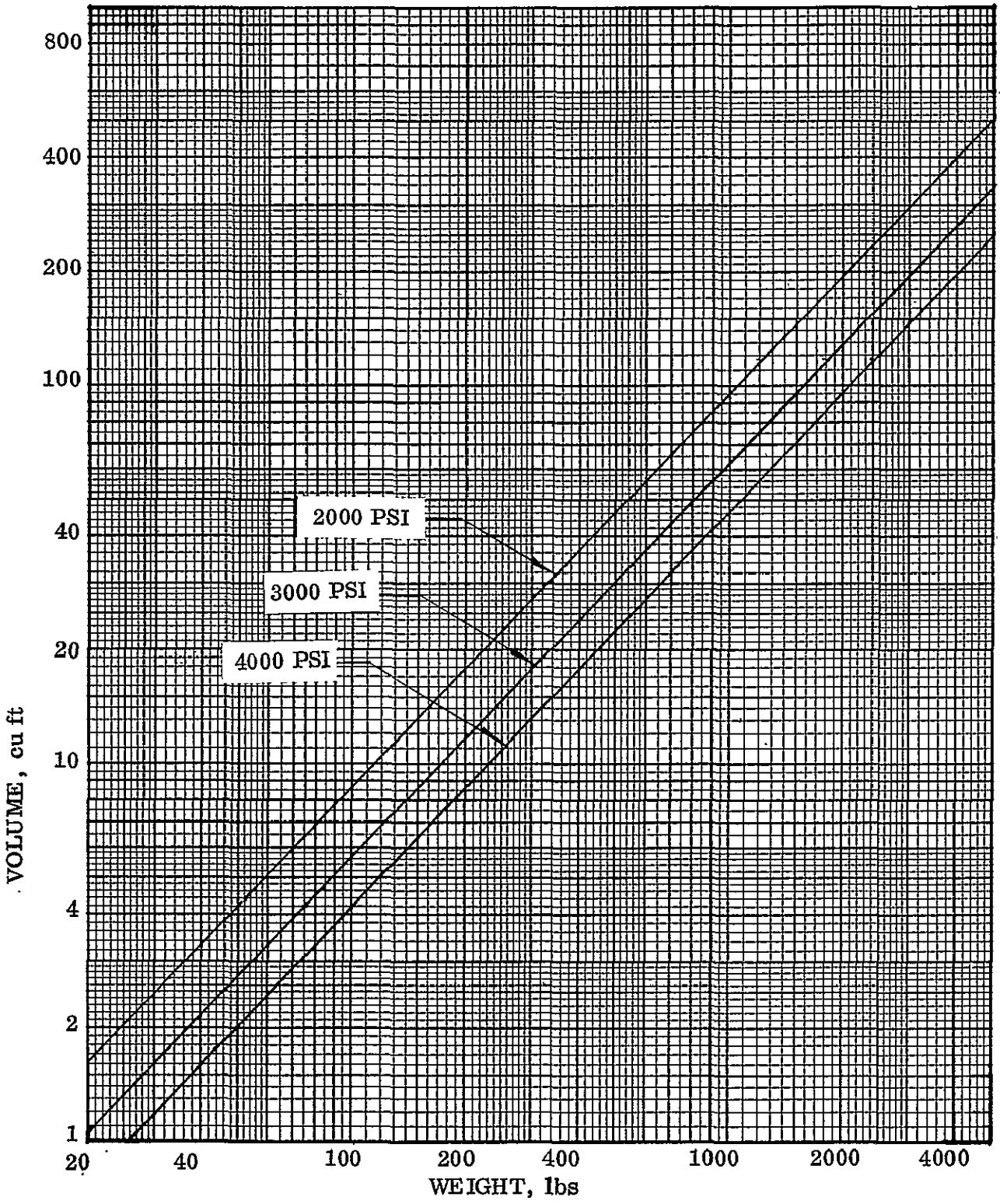


Figure 2-13. Helium Pressure Bottle Weight as a Function of Volume for Titanium Bottles

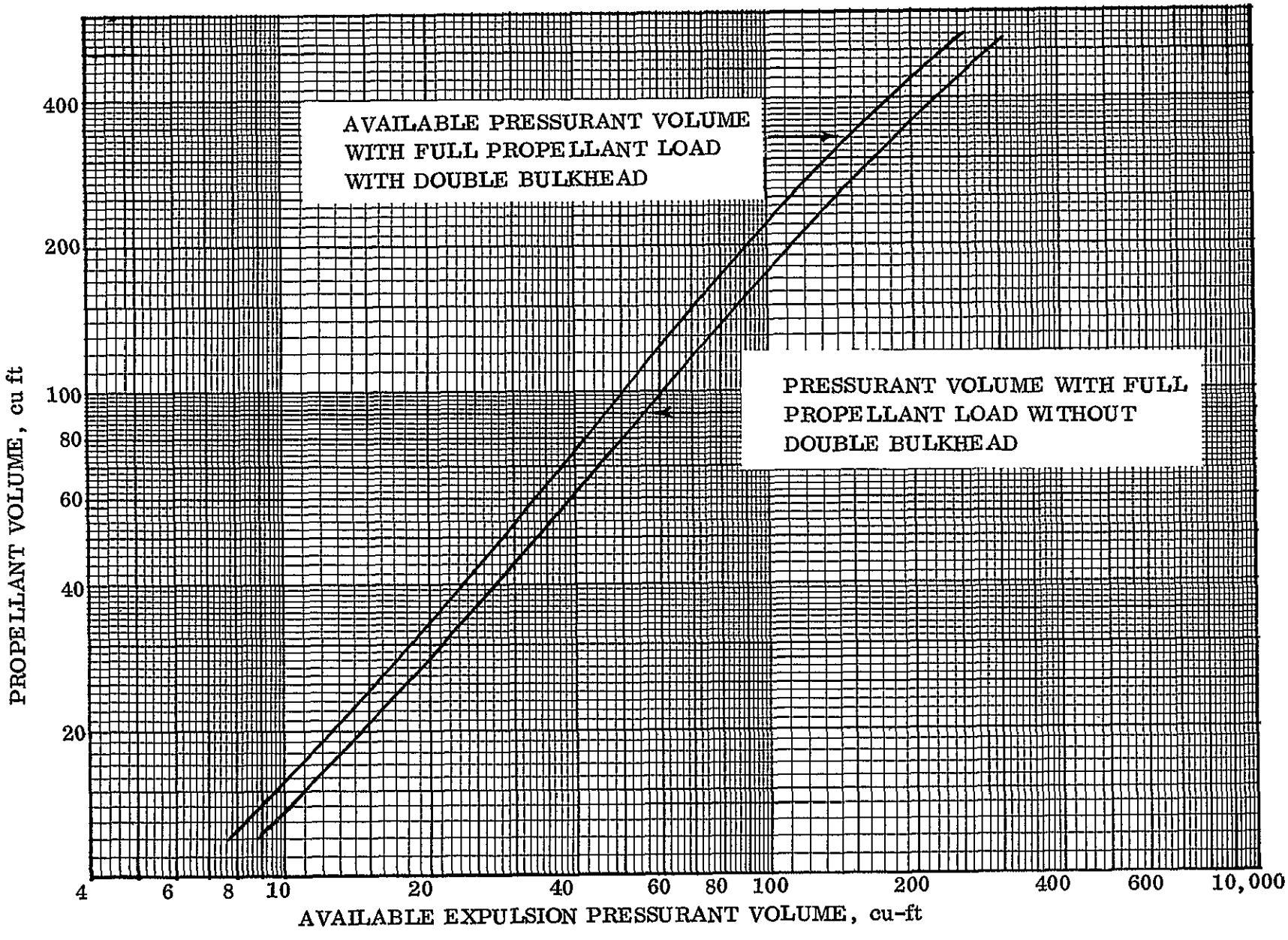


Figure 2-14. Bellows Expulsion System Pressurant Storage Volumes Available

Some difference is also to be found between the vacuum shell weights, primarily due to a slight difference in the space allowed between the jacket and the inner shell.

To determine the helium bottle weight, an iteration is required since the helium requirement depends on the volume to be pressurized and vice versa. For an 11.5 ft^3 helium bottle, assumed to be stored in the tank; from Figure 2-14 the volume remaining to be pressurized is equal to 24.6 ft^3 - 11.5 ft^3 + 42.5 ft^3 = $\underline{55.6\ ft^3}$. Ratioing the helium storage volume from that required to pressurize 42.5 ft^3 results in V_{He} = 8.8 (55.6/42.5) = $\underline{11.5\ ft^3}$, which checks with the initial assumption. Then from Figure 2-13 for the 3,000 psia storage case the bottle weight is 198 lb. The total dry system weight is thus 803 lbs. This same analysis can then be accomplished for other propellant volumes and pressurant storage conditions over the range of data presented in Figures 2-7 through 2-14. It is noted that for each individual case an assessment of the actual helium requirements and storage bottle configuration must be made using Equations 2-1 and 2-2, in conjunction with the above data.

SECTION 3

METALLIC DIAPHRAGM SYSTEM DATA

The work presented in this section was performed under the Convair 1971 Independent Research and Development (IRAD) program and is reported herein only for reference as it relates to the pertinent subject of cryogenic propellant transfer in space.

The basic metallic diaphragm system being considered here is essentially the same as presented in Figure 3-9 of Reference 2-1. The data contained in this section is the result of performing a detailed weight analysis of this system. The basic weight data for the diaphragm itself were obtained from point design data on units built by Arde, as presented in References 3-1 through 3-3. The total diaphragm weight consists of that for the thin membrane plus that for the reinforcing wires. From data available, membrane thicknesses and wire diameters were estimated as a function of tank diameter. Examination of the existing data indicated that the spacing between wires, for stability purposes, was at constant angles around the tank. From the above information total diaphragm weights were obtained as a function of tank diameter and are presented in Figure 3-1.

It is noted that these diaphragms are CRES and therefore the inner tank shell should also be CRES. Weight data for such a vessel was developed and is presented in Figure 3-2. This tank configuration is essentially the same as that to be used for other systems requiring a spherical tank. The weld land at the tank circumference, as used for conventional tank construction, will be sufficient for containing the diaphragm flange, and thus an additional weight penalty to incorporate the diaphragm is not incurred. The vacuum jacket is also not significantly effected and thus weight data for a spherical jacket presented in Reference 3-4 is applicable.

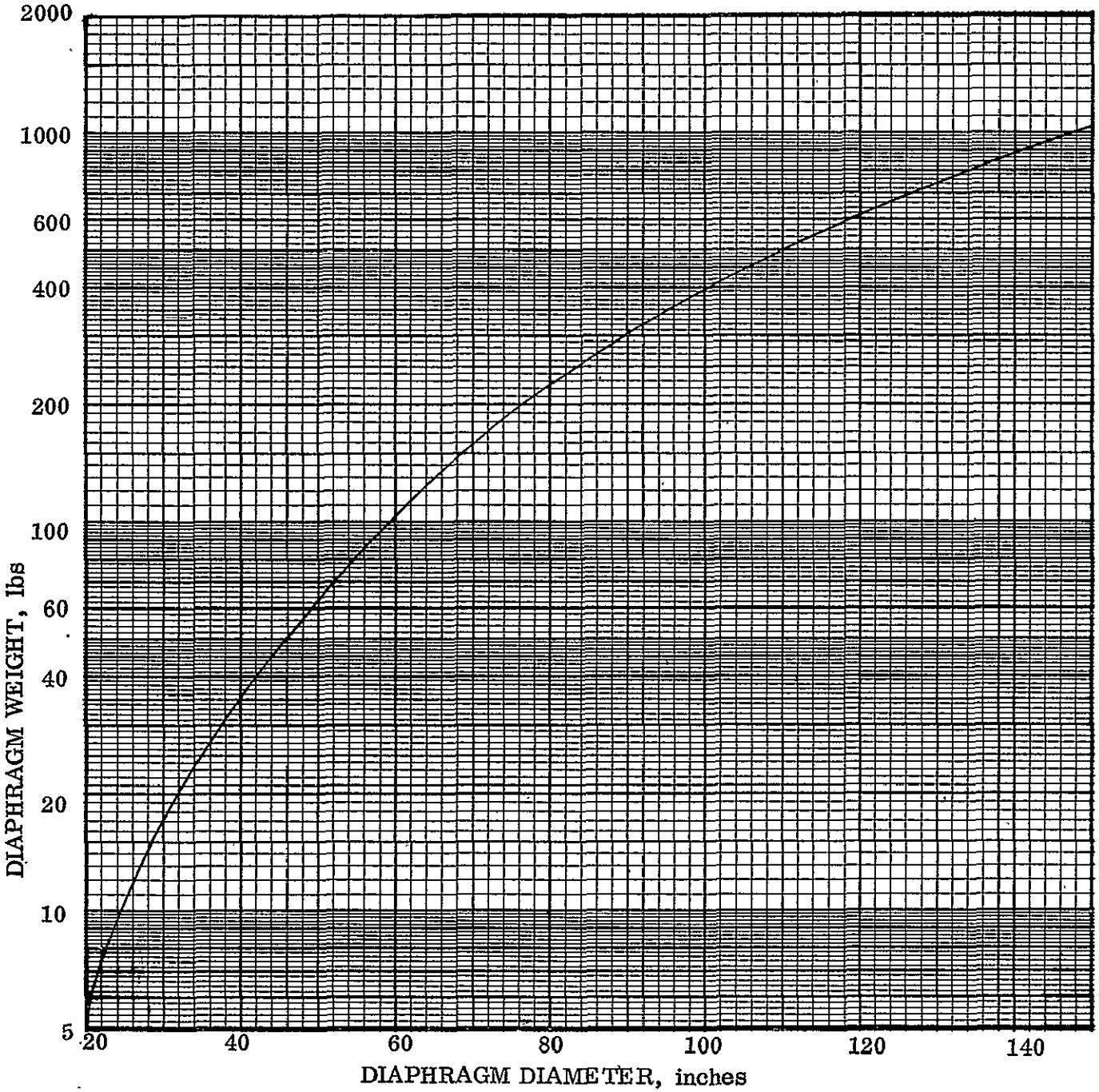


Figure 3-1. Expulsion Diaphragm Weight Based on an Extrapolation of Data for CRES Membranes

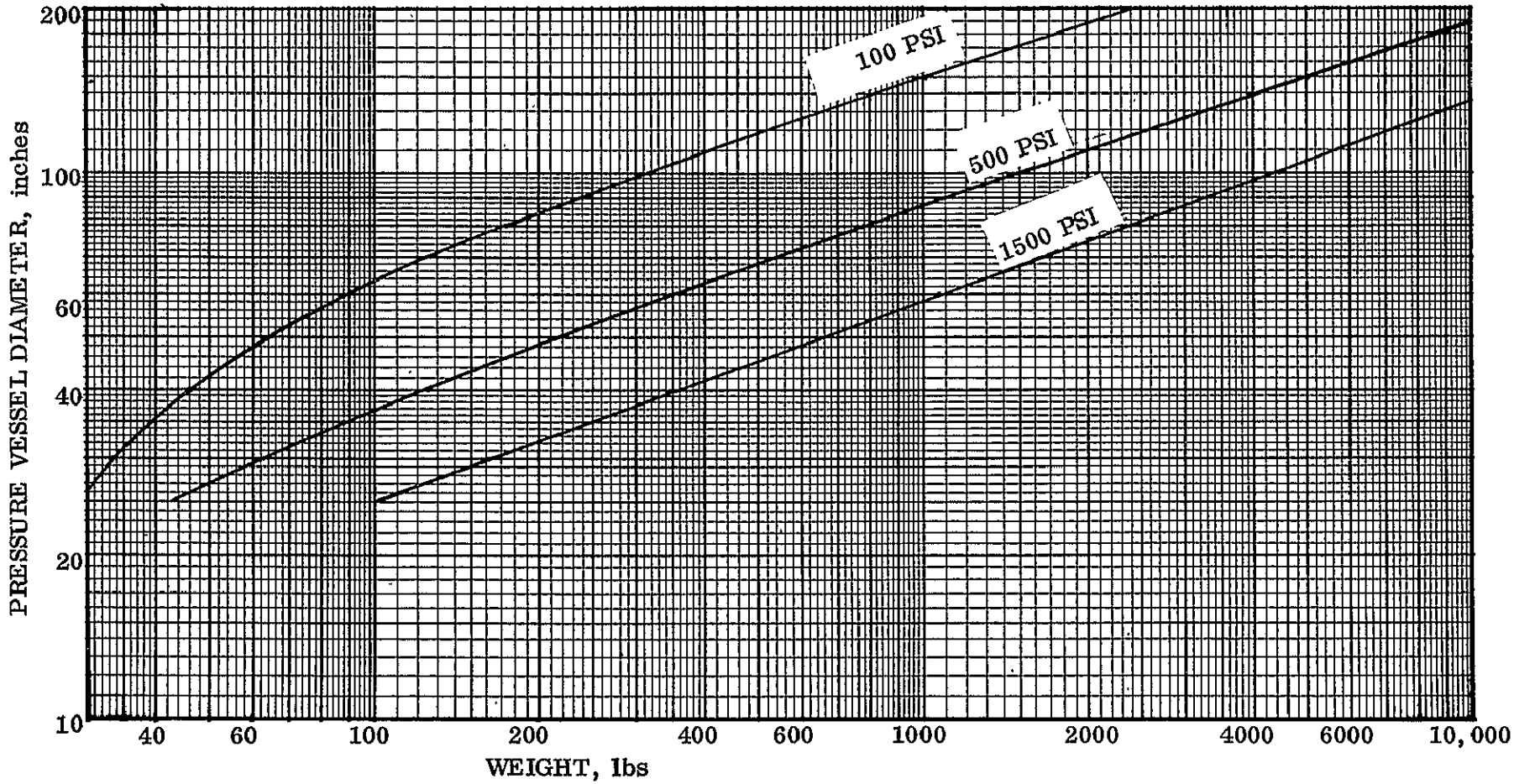


Figure 3-2. CRES Pressure Vessel Weight, Spherical CRES Vessels for use With Expulsion Diaphragms

SECTION 4

PADDLE TYPE VORTEX SYSTEM DESIGN

The work presented in this section was performed under the Convair 1971 Independent Research and Development (IRAD) program and is reported herein only for reference as it relates to the pertinent subject of cryogenic propellant transfer in space.

The basic system concept is the same as that analyzed during the screening analysis and presented in Figure 3-10 of Reference 2-1. A detailed hardware design was accomplished and the resulting configuration is presented in Figures 4-1 and 4-2.

A significant characteristic of this system is that some powered means is required to drive the paddle.

For the present case an electric motor operating through a hermetically sealed flex spline was chosen to have the best potential. Such a system provides a positive seal between the drive motor and tank fluid and thus allows the motor to be external to the tank and separated from the tank fluid. Also, the flex spline system allows a large reduction in speed between the motor and paddle, as required by the present system since paddle rotational rates are desired to be very low.

In determining the required paddle rotation rates the analysis described in Reference 2-1 was used. From this analysis an angular speed of 3 rpm was determined to be applicable for both LH_2 and LO_2 for a spherical tank diameter of 52 inches (42.5 ft^3). An examination of the data shows that for a constant design acceleration, the required rotation is inversely proportional to the square root of the tank diameter; i.e.

$$\omega \approx \frac{1}{D^{1/2}}$$

Based on this assumption paddle weights were determined as a function of tank diameter for the basic configuration shown in Figure 4-1. These weights are presented in Figure 4-3 and are applicable to both LO_2 and LH_2 systems.

The next step in the analysis was to determine power requirements and drive system weights as a function of tank size. Again, based on the Reference 2-1 data, the power was determined to be equal to the drag force times the moment arm times the angular rotation rate; i.e.

$$P = F_D \, r_i \, \omega$$

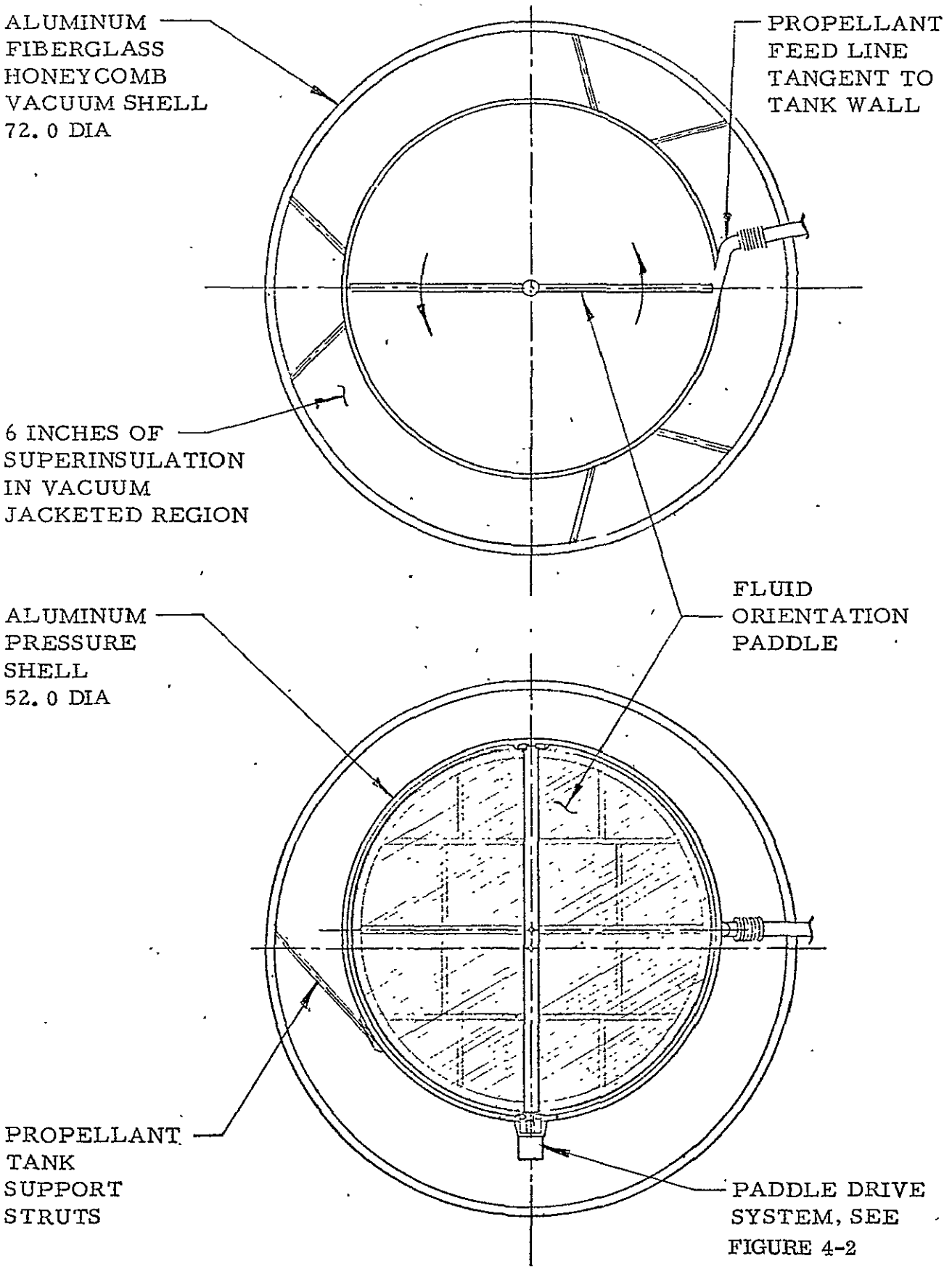


Figure 4-1. Paddle Vortex System

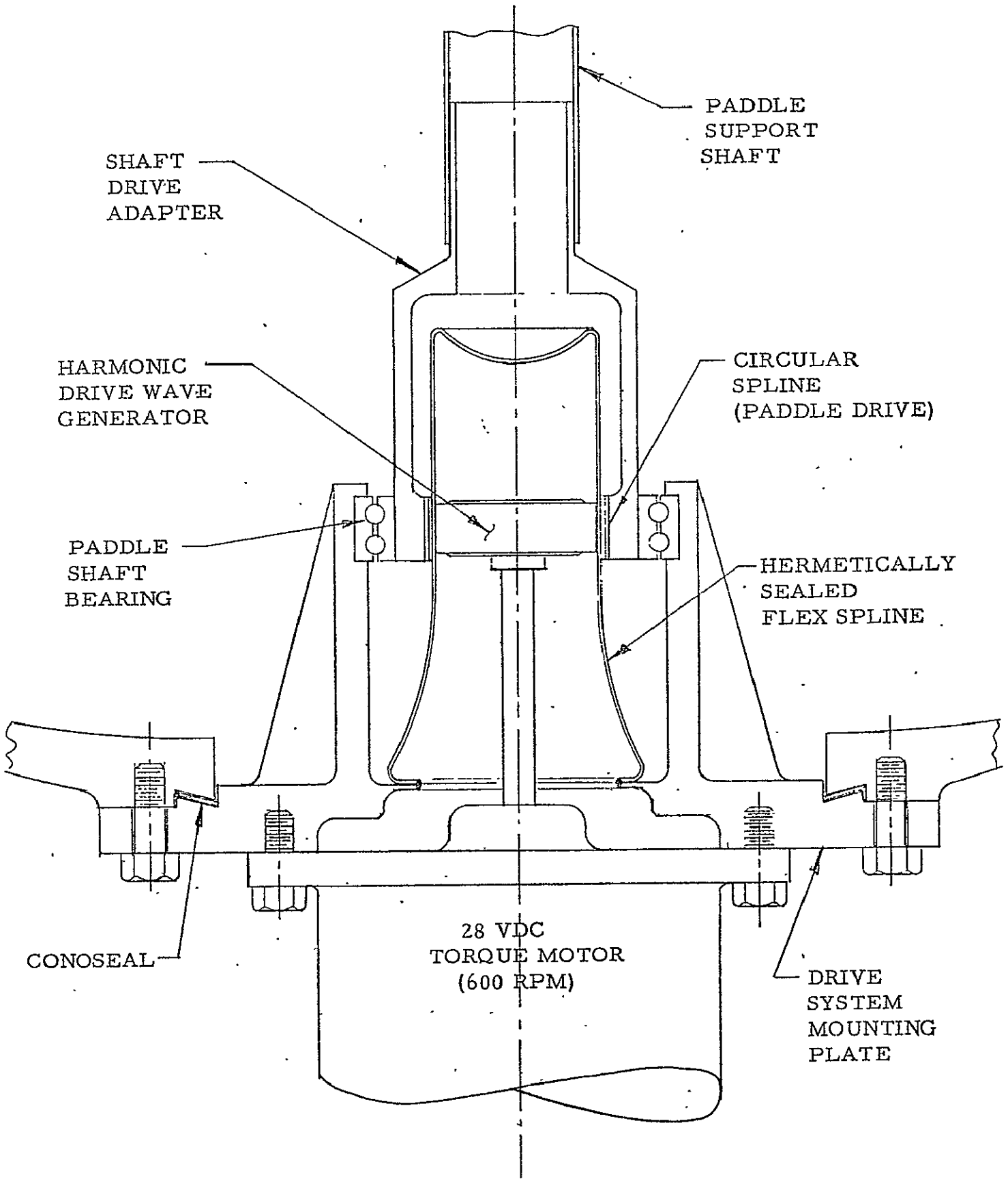


Figure 4-2. Paddle Drive System

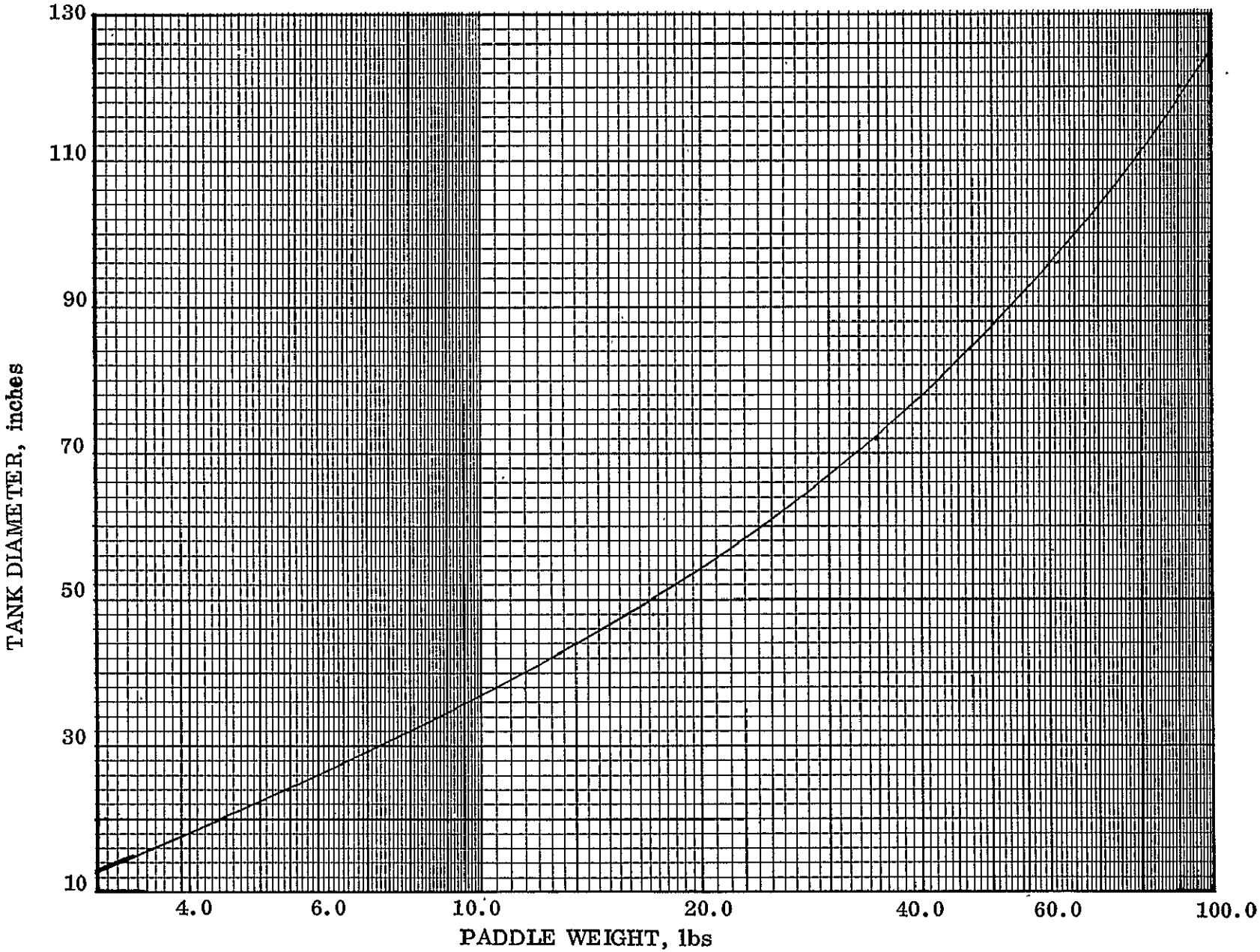


Figure 4-3. Vortex Paddle Weight Versus Tank Diameter

where

$$F_D = C_D \rho A \frac{V^2}{2g}$$

and

$$V = r\omega$$

ρ = fluid density

C_D = drag coefficient

A = paddle area

Then assuming C_D = constant

$$P \approx \omega^3 \rho r^5$$

Substituting the relation for ω as a function of tank radius ($\omega \approx 1/r^{1/2}$) then

$$P \approx \rho r^{3.5} \approx \rho D^{3.5} \qquad (4\text{-}1)$$

Assuming an average motor efficiency of 50% and using the baseline data from Reference 2-1, where P = 14.3 watts motor input for a 52 inch (42.5 ft^3) LO_2 tank, the power requirements at other diameters were determined from the Equation 4-1 relation. These data are presented for both LO_2 and LH_2 in Figure 4-4. Curves, assuming a baseline (42.5 ft^3 tank) paddle rotation rate of 6 rpm, are also given to show the effect on power requirements. At present the optimum rotation requirements are somewhat uncertain due to a lack of information on the effect of speed on actual fluid residuals. It is noted that data are not given for motor powers below .01 horsepower since this was considered a minimum practical motor size to obtain a high reliability for the present application.

Based on a perusal of available vendor motor and drive data, the weight curves presented in Figure 4-5 were developed. Motor weight data are presented as a function of horsepower for 600 rpm DC motors and drive weights are presented for both 100:1 and 200:1 speed reduction systems. Totals are also given. Data for an 11,000 rpm motor are presented for reference.

Also for reference, the weights of a power supply system to provide 24 hours of operation are presented in Figure 4-6. This is based on the use of fuel cells with a weight assessment, as presented in Reference 2-1 of 94 lb/kw plus 2.9 lb/kw-hr.

Weights presented in Figure 4-6 for 24 hours should represent a maximum since actual transfer times would be considerably less than this. Actual times would depend on receiver and line chilldown considerations and supply tank residuals as affected by outflow rate.

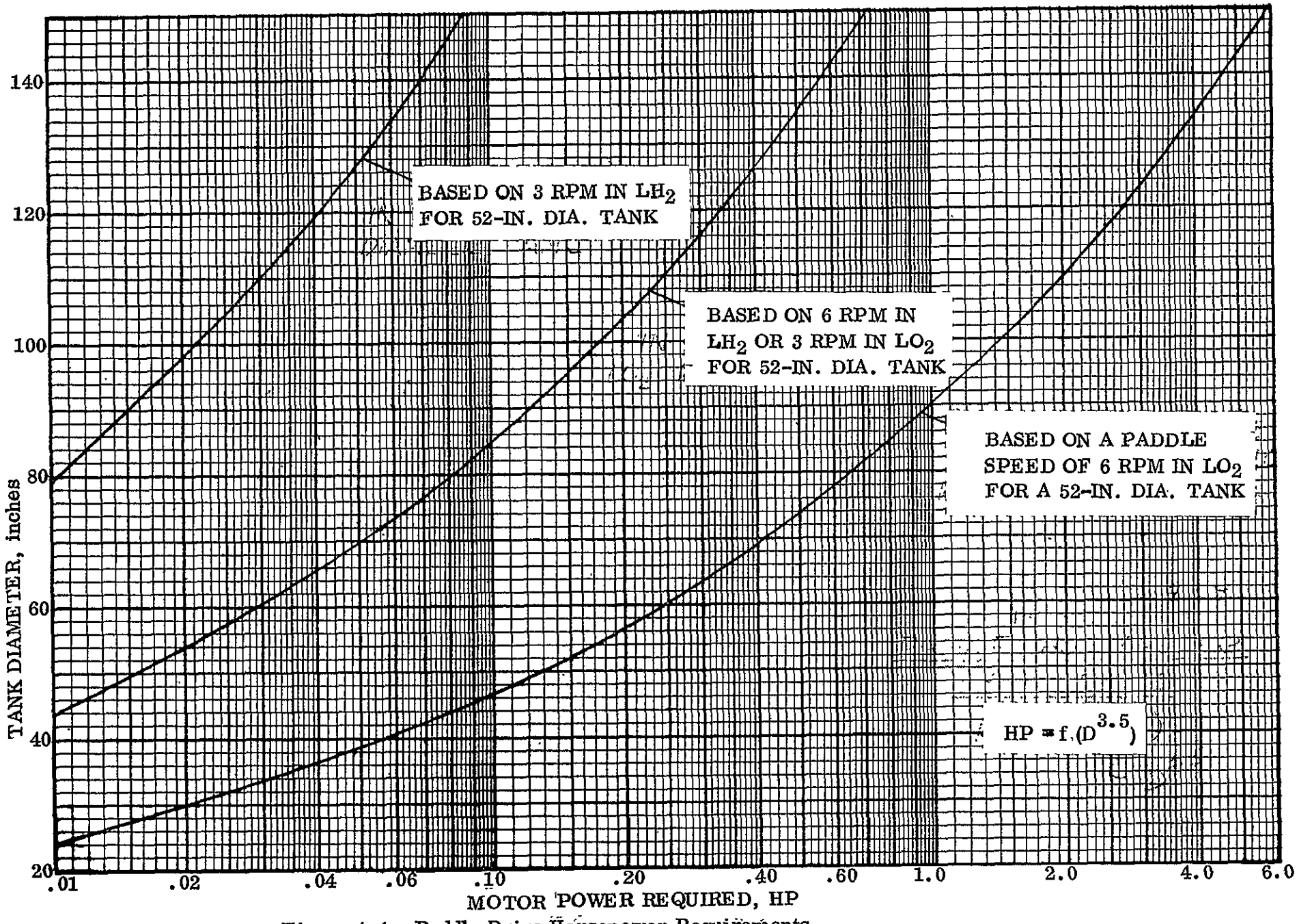


Figure 4-4. Paddle Drive Horsepower Requirements

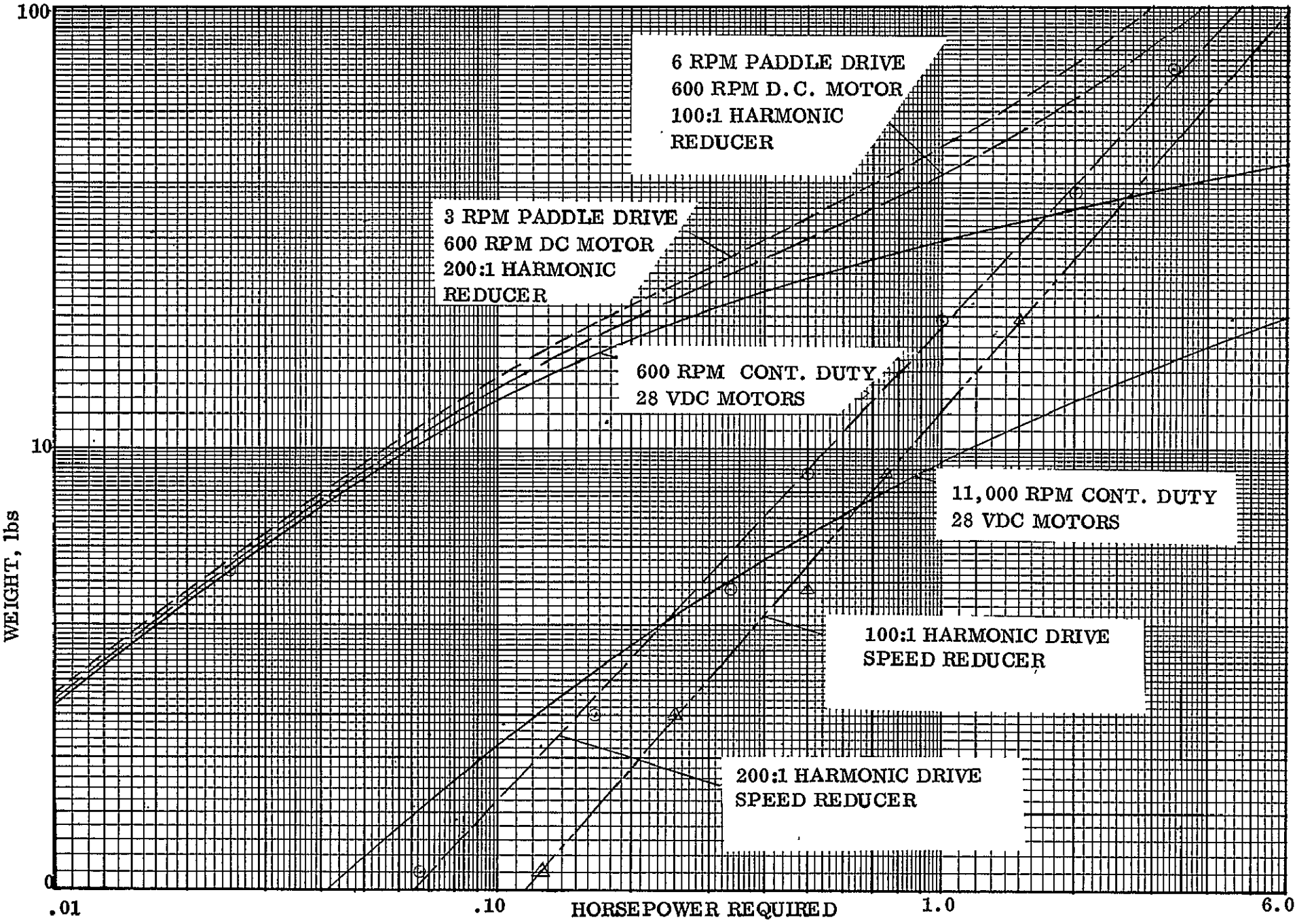


Figure 4-5. Paddle Drive System Weight

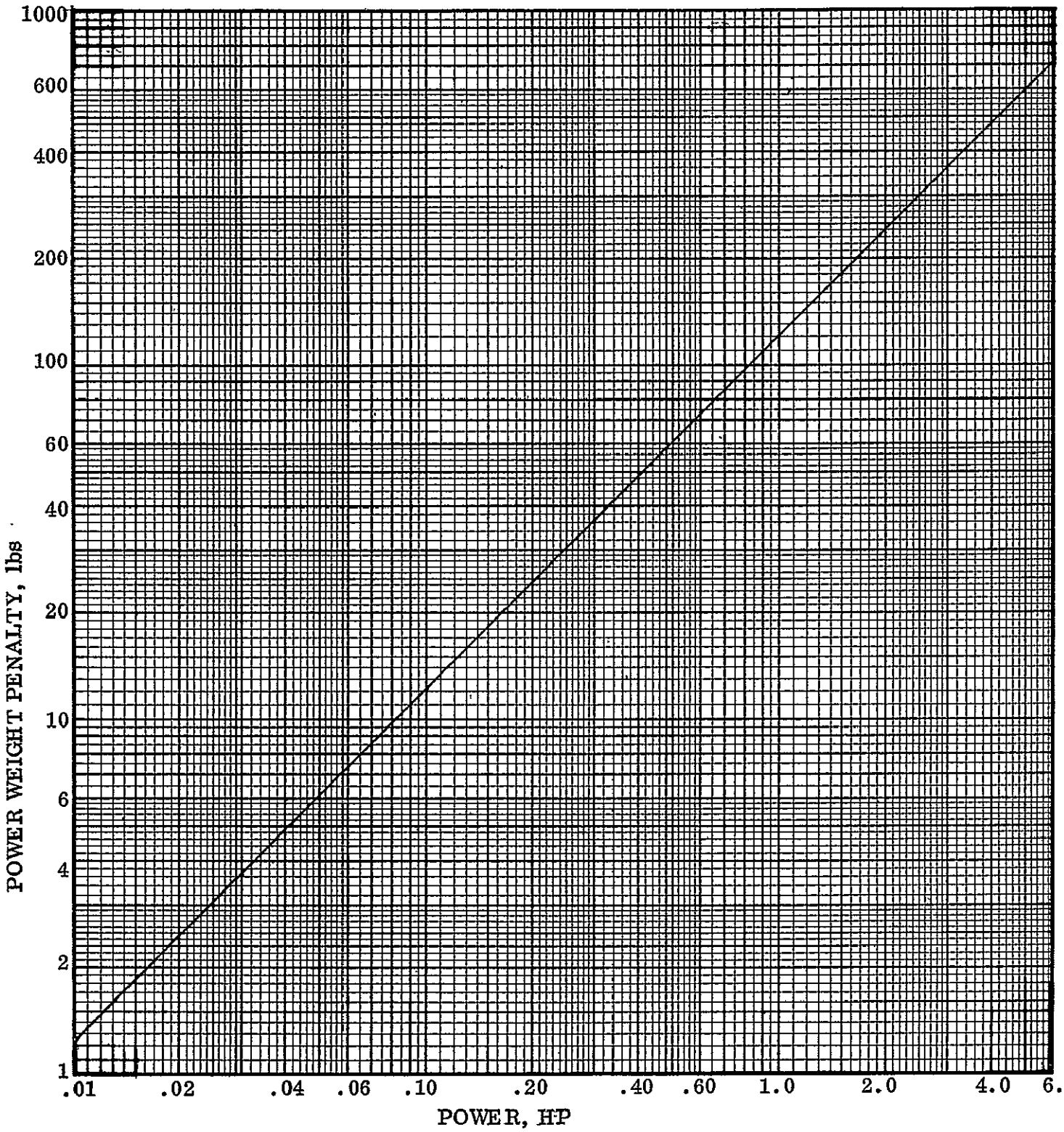


Figure 4-6. Power Weight Penalty Versus Horsepower for a 24-Hour Propellant Transfer

Following is an example of the use of the data presented. Assuming the requirement to transfer LO_2 from a 52 inch diameter tank, where the nominal baseline rotation rate is taken as 3 rpm, a paddle weight of 18.5 lb is found from Figure 4-3. From Figure 4-4 the motor power required is 0.018 HP and thus, from Figure 4-5, the drive plus motor weight is 4.4 lb. From Figure 4-6 the power supply weight penalty is 2.25 lb. The total paddle system weight, excluding tankage, is thus found to be 25.15 lb, which is competitive with other systems at this tank size.

One of the main considerations important to the analysis of this system is the magnitude of the propellant residuals. Further work will be expended in this area during the next reporting period.

SECTION 5

LINE AND RECEIVER TANK CHILLDOWN

The work presented in this section was performed under the Convair 1971 Independent Research and Development (IRAD) program and is reported herein only for reference as it relates to the pertinent subject of cryogenic propellant transfer in space.

Initial runs with a combination line and receiver tank chilldown program, as described in Reference 3-4, indicated that the line chilldown occurred only over the first few seconds of transfer and did not significantly effect the chilling and filling of the receiver tank. This was for a LH_2 inlet to a one-half inch diameter by 100 ft long Al Aly line and a 42.5 ft^3 Al Aly receiver tank. Transfer line and receiver tank chilldown were thus investigated further using the line and receiver tank programs separately. The analyses performed and the data generated are presented in the following paragraphs.

5.1 LINE CHILLDOWN ANALYSIS

In order to perform the line chilldown analysis, parametric line weights were defined for both Al Aly and CRES lines over a range of diameters from 0.5 to 3 inches and for operating pressures up to 2,000 psi. The total weight is taken to consist of the basic line plus mounting brackets and expansion loops or joints. Figures 5-1 and 5-2 present basic line weights for CRES and Al Aly ducting and Figure 5-3 gives corresponding weights for attachments. Both weights are per foot of line length. Line expansion and contraction are assumed to be taken care of by providing loops in the line or gimbal type flex joints. Expansion loops will be used for line diameters of 1.0 inches and less and gimbal joints for diameters greater than 1 inch. The weights for each loop or joint are presented in Figure 5-4 as a function of the basic line weight. The number of loops or joints required as a function of line length is presented in Figure 5-5. With respect to the number and weight of gimbal joints, a range of values was designated and the actual quantity needed would depend on the specific routing configuration under consideration. The curve drawn represents an average of the best available data.

An example of the determination of the total line weight for an 0.5 inch by 100 ft Al Aly line operating with 160 psia is presented below.

From Figure 5-2, $Wt_{line} = 100 \times 0.036 =$ 3.60 lb
From Figure 5-3, $Wt_{attach} = 100 \times 0.0027 =$ 0.27 lb
From Figure 5-5, No. Loops = 4
From Figure 5-4, $Wt_{exp.loops} = 4 \times 0.05 =$ 0.20 lb
TOTAL 4.07 lb

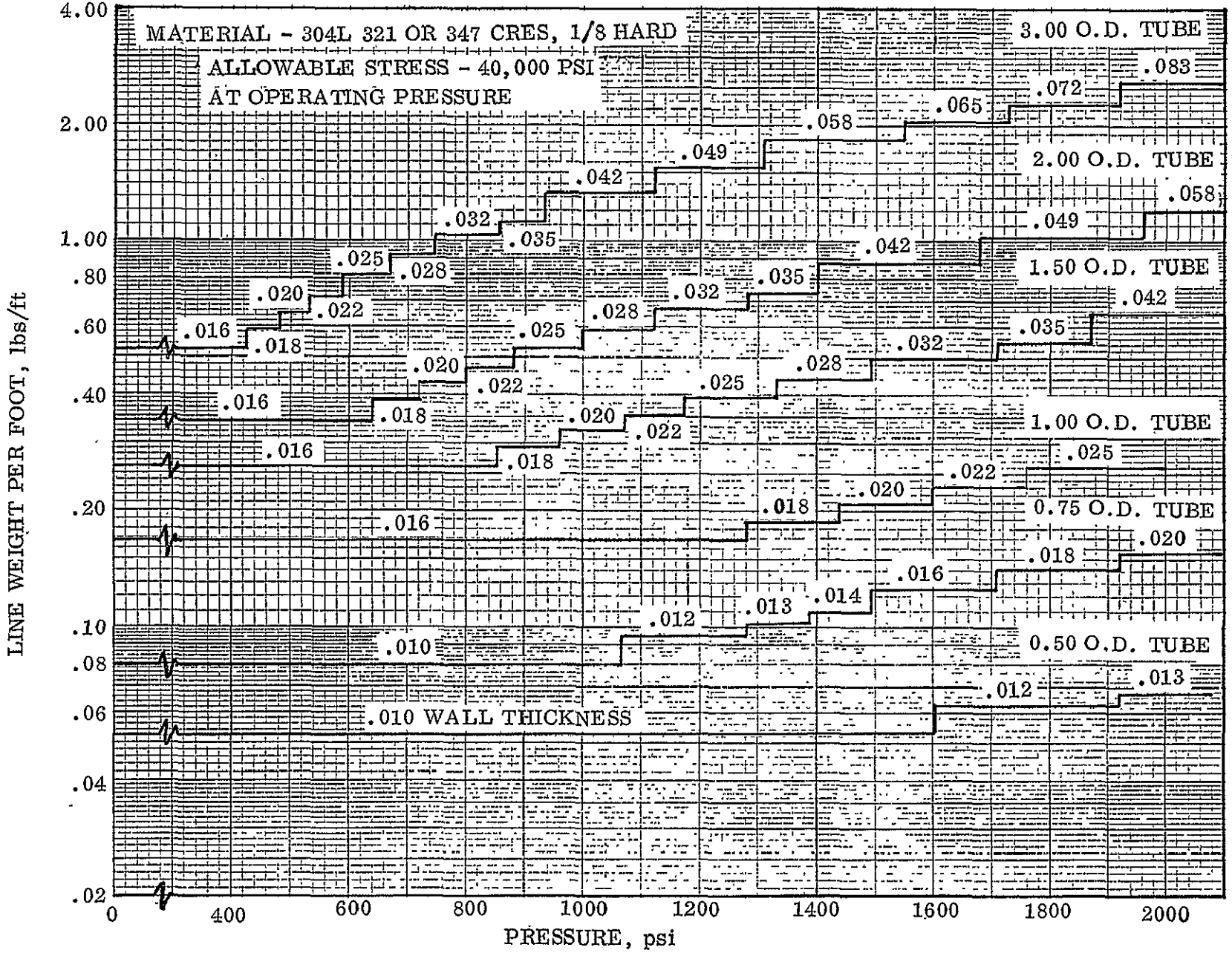


Figure 5-1. Line Weight Versus Pressure for CRES Tubing

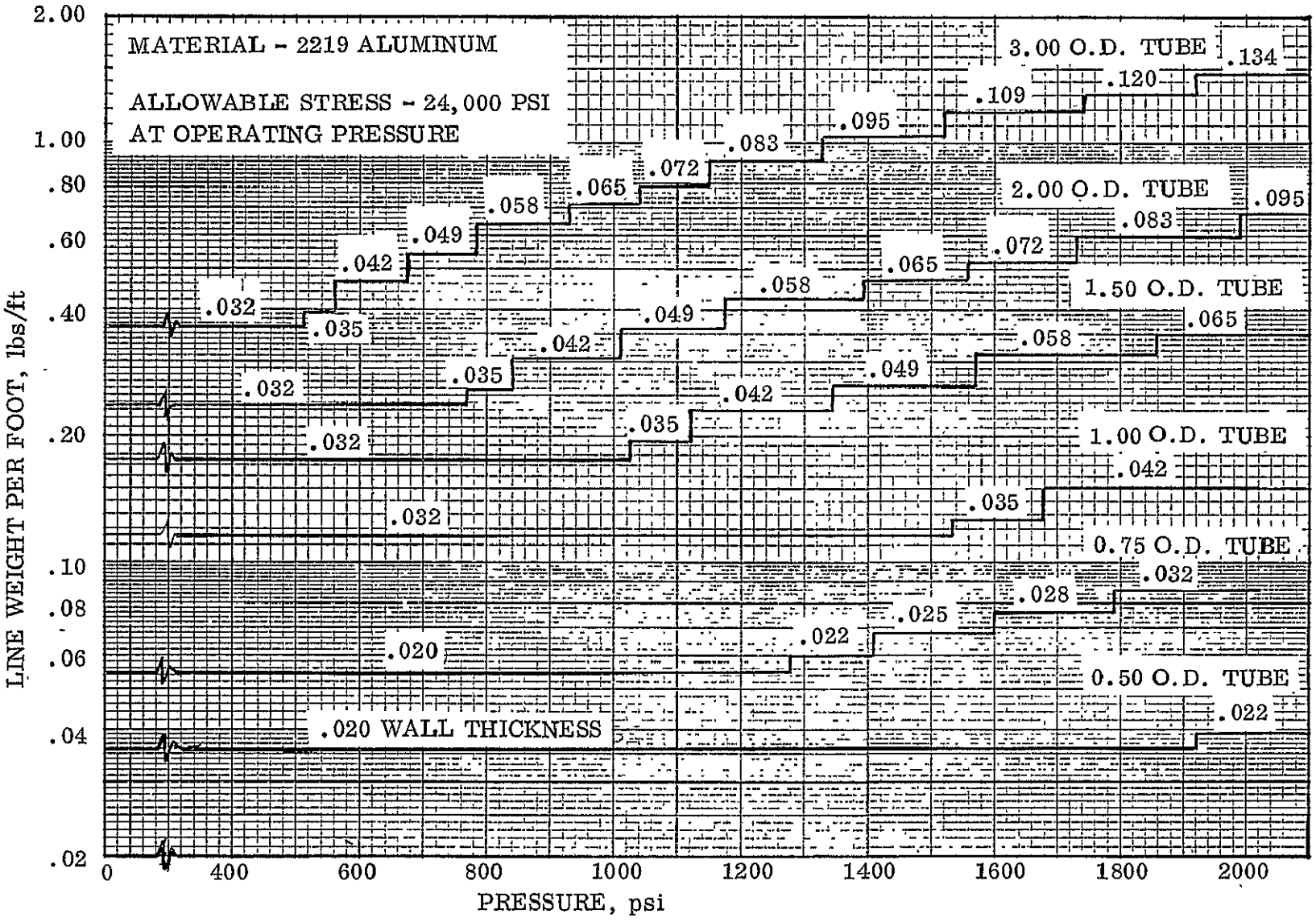


Figure 5-2.. Line Weight Versus Pressure for Aluminum Tubing

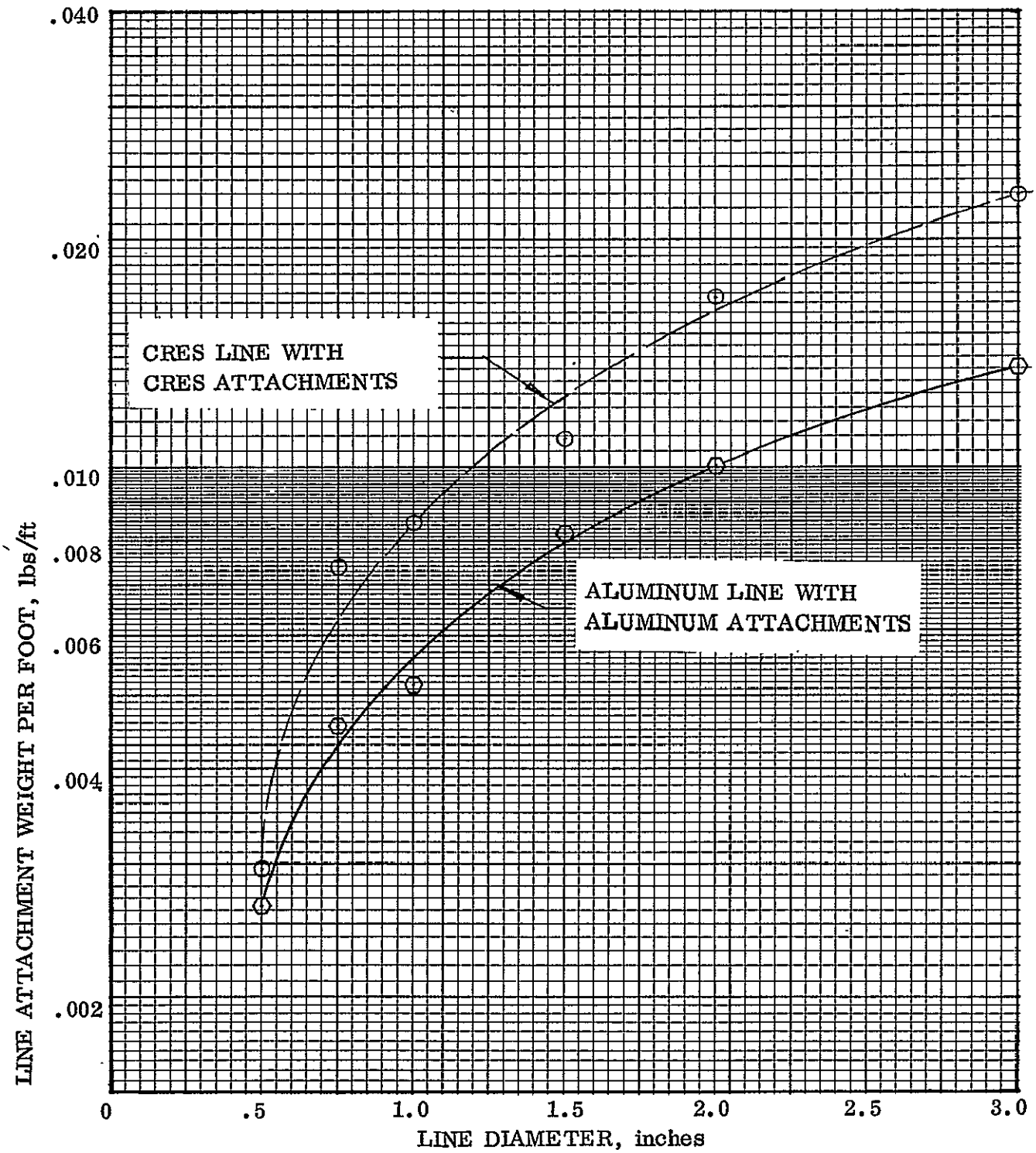


Figure 5-3. Attachment Weight Versus Line Diameter for CRES and Aluminum Tubing

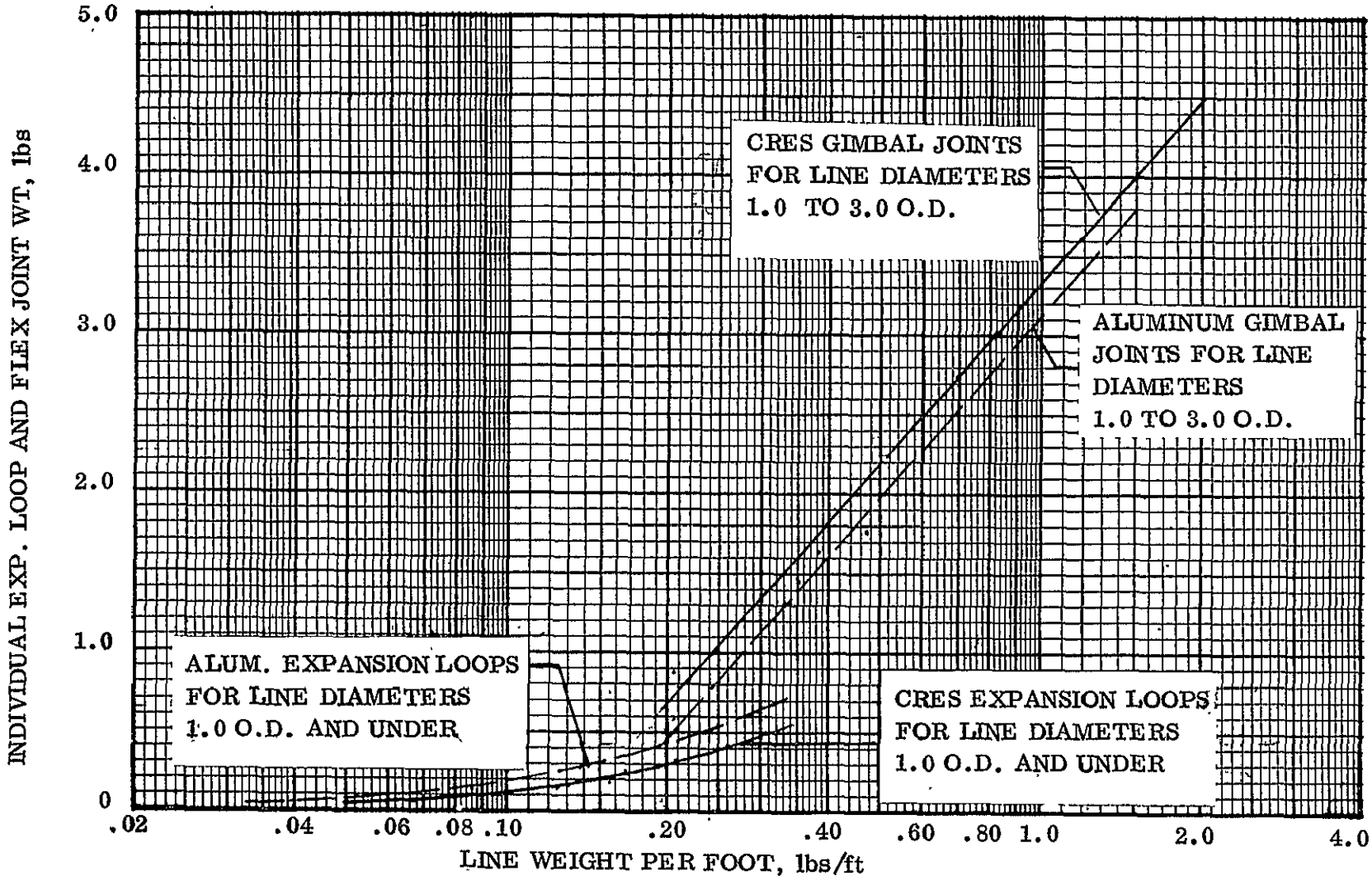


Figure 5-4. Line Weight Versus Flex Joint Weight

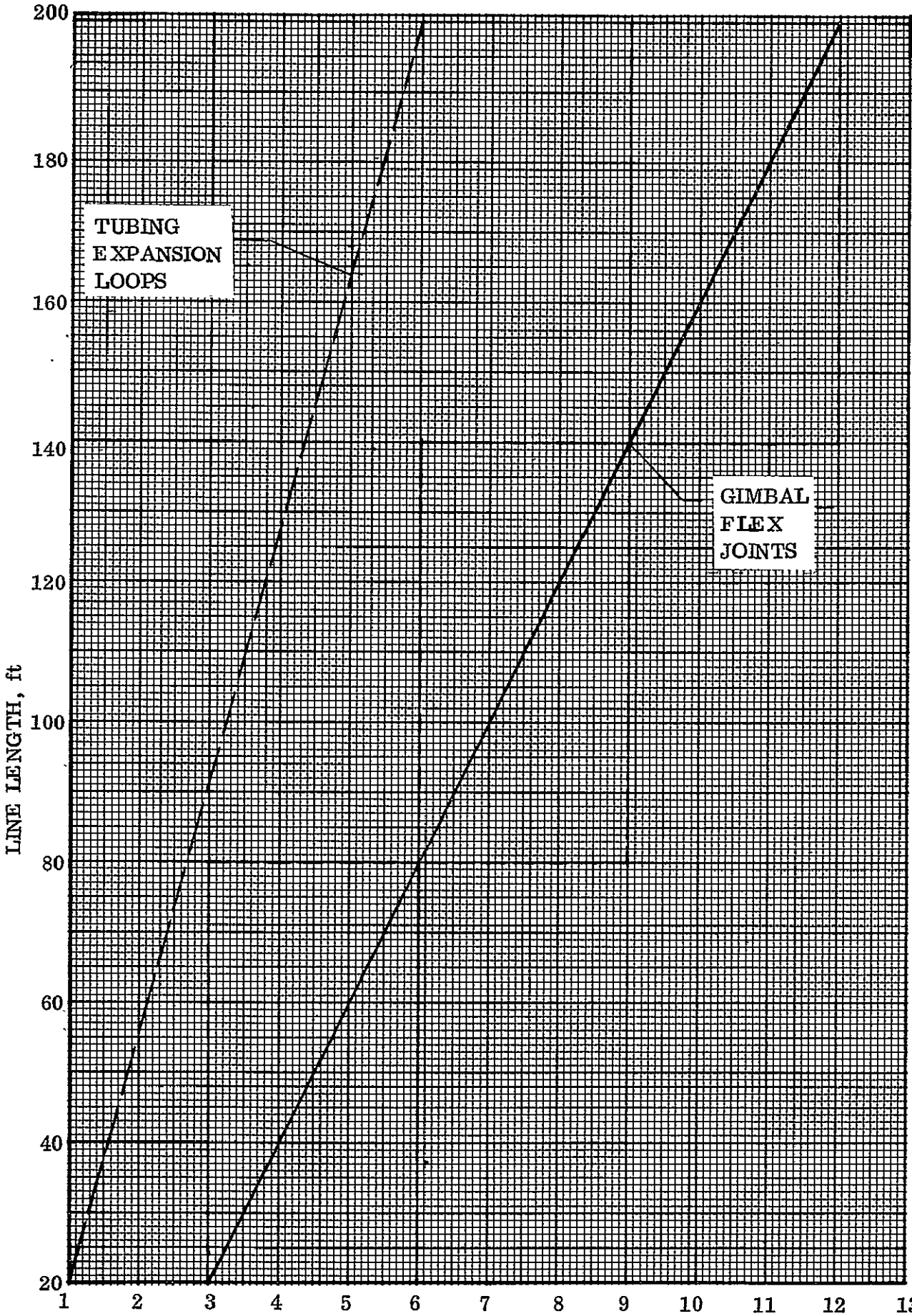


Figure 5-5. Flex Joint Quantity Vs. Line Length

It is noted that this weight represents just the line sections which are assumed to require chilling down and does not include any insulation or jacketing.

From the total weight, as determined from the above data, an equivalent wall thickness is then determined and used in the line chilldown computer program. Taking temperature at the end of the line as representative of chilldown a typical chilldown curve is presented in Figure 5-6. A chilldown time of 3.8 seconds is determined from this data. The mass flow rate following chilldown for this 0.5 inch by 100 ft line was found to be 0.67 lb/sec. At this rate it would take approximately 250 sec to provide 167 lb of LH_2 to a receiver tank. This fluid quantity is required to fill a 42.5 ft^3 receiver tank to 90% capacity. Therefore, in this case, the line chilldown time represents only 1.52% of the tank fill time.

Chilldown calculations were made over a range of line inlet pressures and for two different line lengths in order to illustrate the potential effect of the line on the overall transfer process. Chilldown times are presented in Figure 5-7 and final flow rates and the percent of the overall fill time required to chill the line are given in Figure 5-8. This percentage data gives an indication of the relative importance of the line in the overall tank fill process. It is seen that at low inlet pressures and long line lengths, with resulting low flow rates, line chilldown begins to become a significant factor. These data serve to illustrate under what conditions it is important to couple the line chilldown and tank fill analyses. Preliminary data for CRES lines indicates less of a contribution due to their lower heat capacity.

During the next reporting period, further analysis will be performed to determine the final effect of various line configurations on overall chilldown and fill operations.

5.2 RECEIVER TANK CHILLDOWN

Preliminary analysis indicates that the maximum pressure which can occur in a locked-up (no venting) receiver tank during chilldown and fill is a strong function of the rate of heat transfer between the tank wall and fluid and also the inflow rate. This is illustrated by the data presented in Figure 5-9, where tank pressure histories during chilldown are given for several different wall to fluid heat transfer coefficients and inflow rates. It is seen that the minimum pressure rise occurs for the high inflow, low heat transfer case. The low heat transfer rate would be representative of a condition where cold liquid is not being injected on the hot tank wall and most of the wall cooling is by gaseous convection. The maximum pressure for this case was only 66 psia versus 189 psia for the highest rate heat transfer case.

The various key points in the transfer process are illustrated on the curves shown in Figure 5-9. In all the cases presented here the first thing to occur is the start of liquid accumulation in the tank. Then, for the high heat transfer cases, the wall becomes essentially chilled and the filling proceeds from there on at a fairly constant wall temperature. The pressure, however, decreases due to the continued addition of low energy liquid. For the low heat transfer high flow case the tank becomes filled before

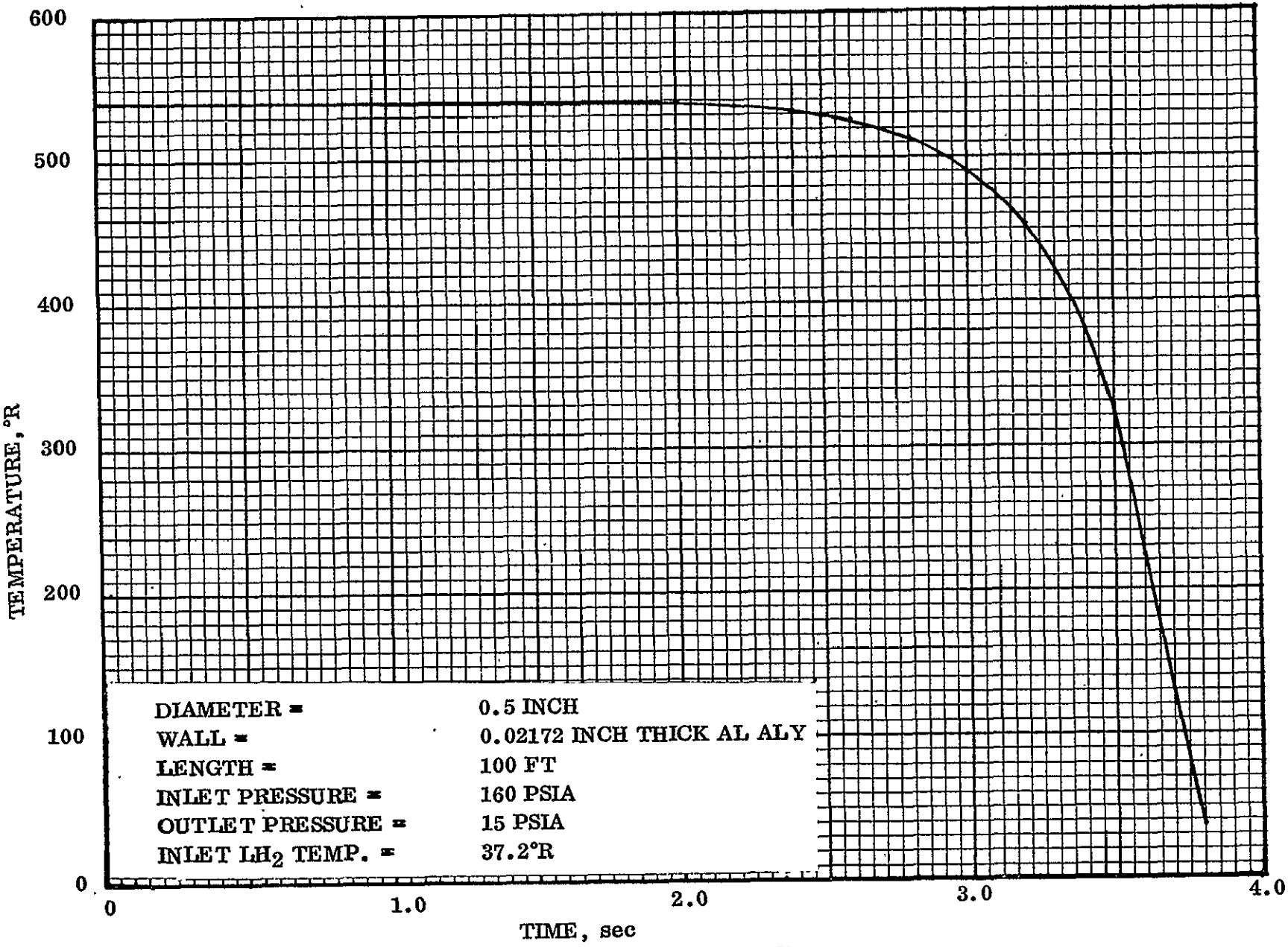


Figure 5-6. Line Exit Temperature Versus Time

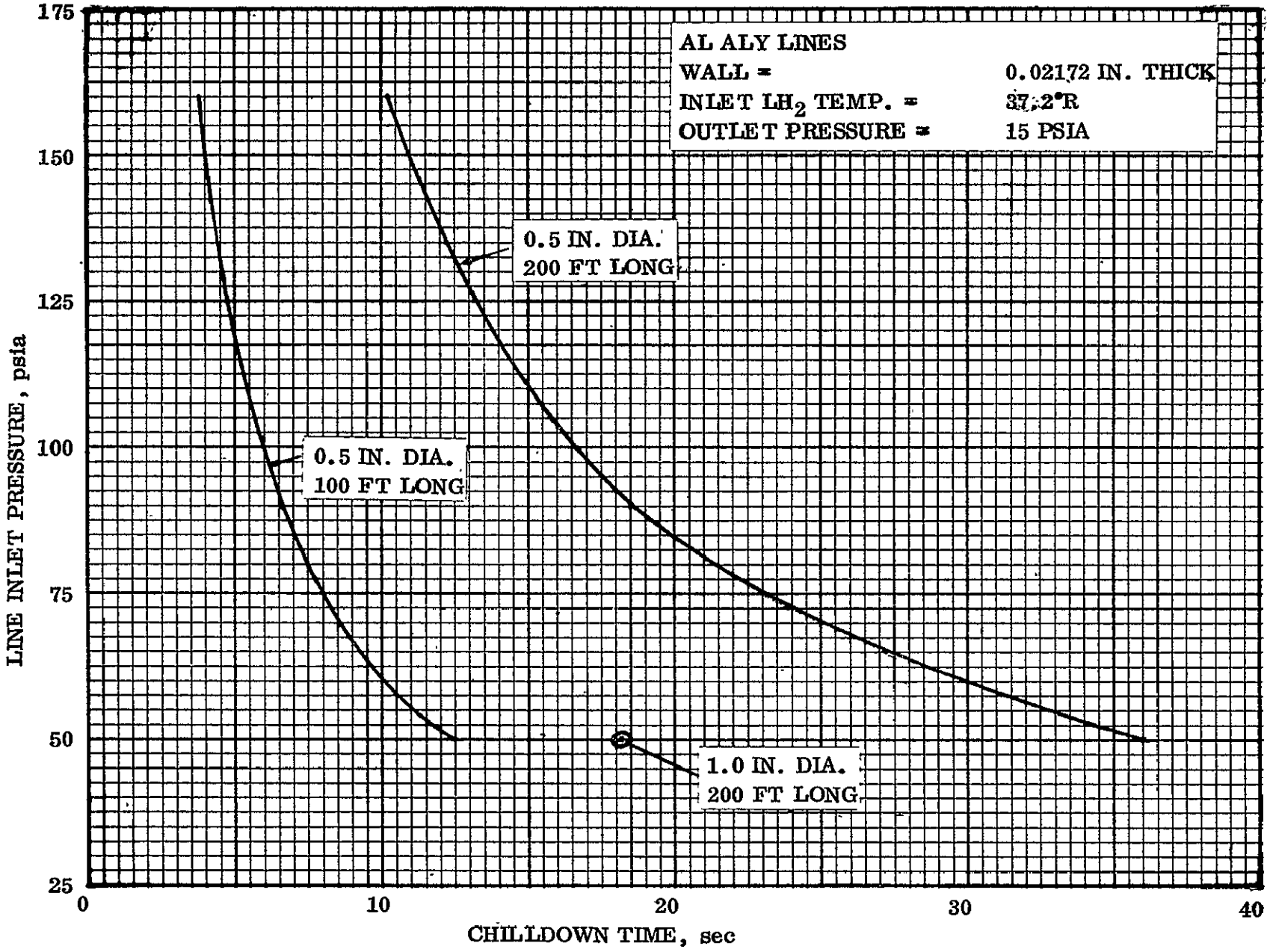


Figure 5-7. Chilldown Time Vs Line Inlet Pressure

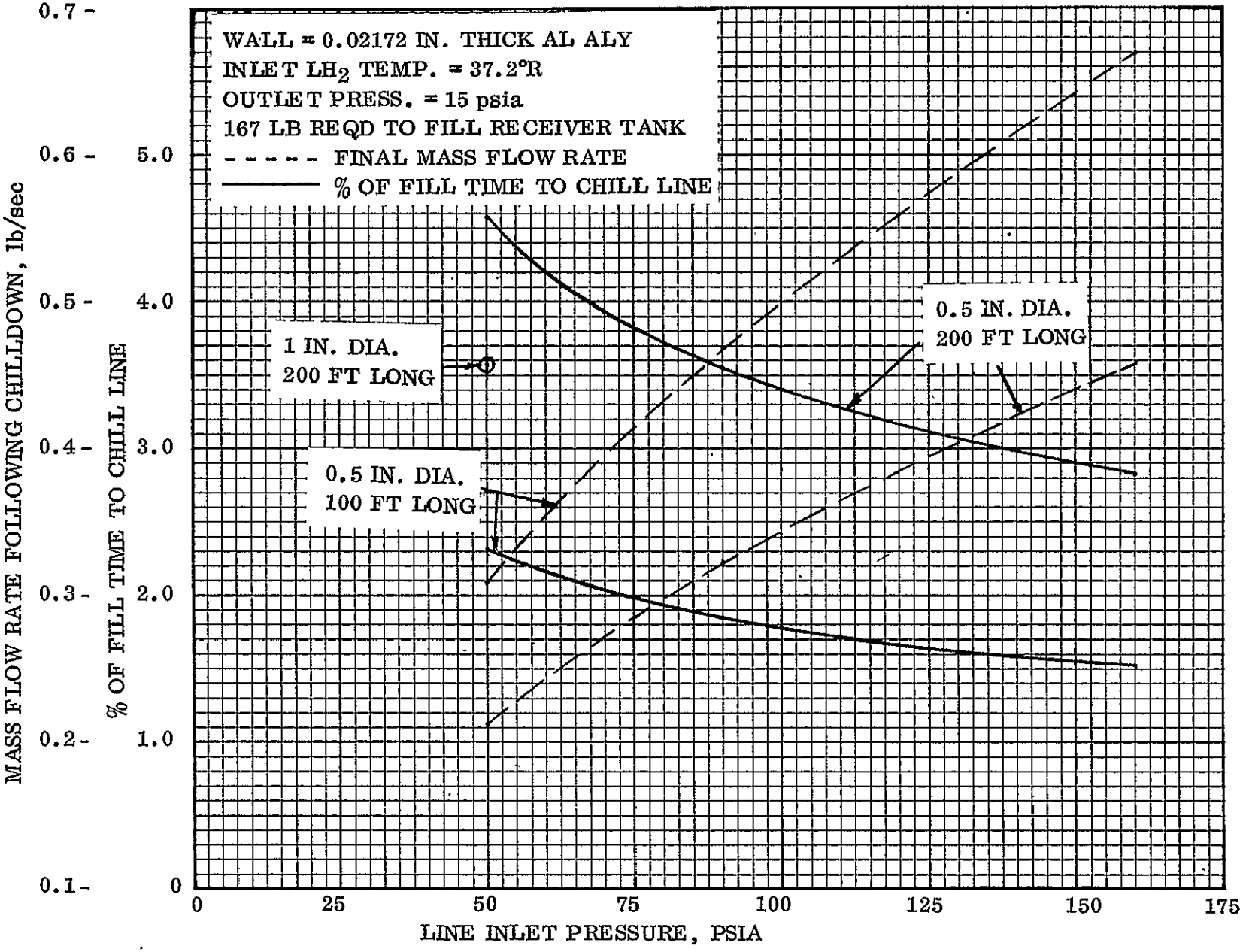


Figure 5- 8. Effect of Line Length and Flow on Relative Chilldown Time

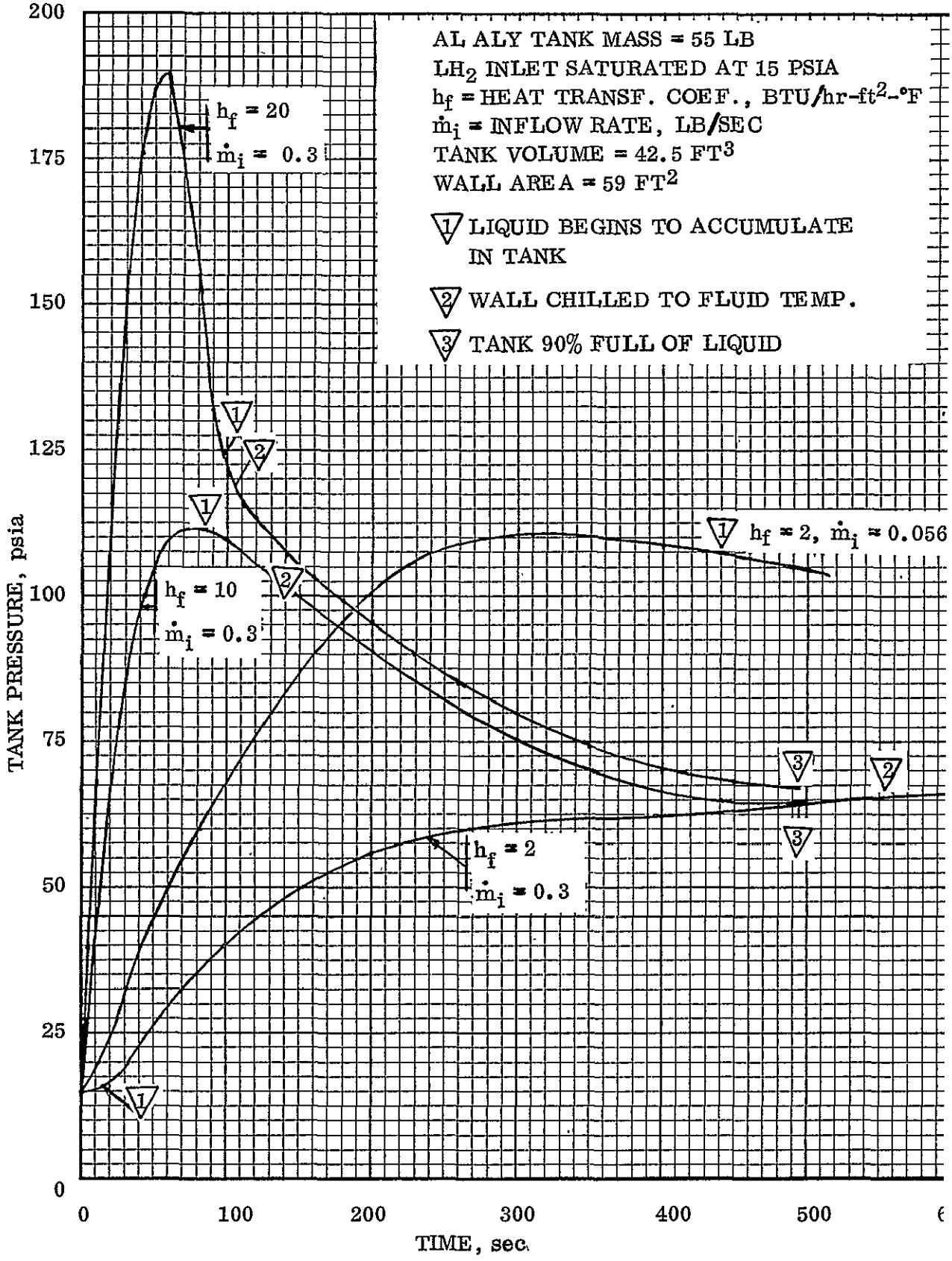


Figure 5-9. Receiver Tank Pressure During Chilldown and Fill

the wall has had a chance to completely chill down. The tank fluid thus absorbs additional heat following fill and the tank pressure rises by a small amount. It is noted that for the low inflow rate case the problem was terminated prior to tank filling or completion of wall chilldown. However, it appears that the peak pressure has been reached. For the transfer processes which were carried to completion, the final pressures are seen to be approximately equal. This is expected since essentially the same overall mass and energy have been transferred in each case.

The above data serves to show that significant variations can occur in peak tank pressure, depending upon the conditions imposed. Further analyses will be accomplished during the next reporting period for different tank sizes and masses and for various inflow conditions and heat transfer processes in order to fully assess the importance of chilldown on the overall transfer operation. The weights and hardware effects of venting the receiver tank will also be assessed.

SECTION 6

HIGH PRESSURE SYSTEM ANALYSIS

The work presented in this section was performed under the Convair 1971 Independent Research and Development (IRAD) program and is reported herein only for reference as it relates to the pertinent subject of cryogenic propellant transfer in space.

Work to date has been directed toward the conception, definition (sizing) and feasibility of supercritical (high pressure) cryogen transfer systems under low-g conditions. Systems or methods of high pressure cryogen transfer presented in this section include: (a) simple blowdown, (b) expulsion by heating, (c) pumping, (d) cooling receiver-heating supply, (e) cooling receiver-heating supply plus regeneration, and (f) vortex tube assisted transfer.

The definition analyses for these systems was performed with the Plumber Code as described in Section 4 of Reference 3-4. All work to date has been based on hydrogen. In all cases except systems (d) and (e) initial receiver pressure was assumed to be 100 ± 20 psia in accordance with the Reference 2-1 baseline. It is further assumed that the state of the fluid in the supply system must be supercritical or gaseous at all times to insure expulsion of a homogeneous fluid.

Initial fluid densities for a depleted receiver were assumed to vary between 0.1 and 0.5 lbs/ft^3. Required mass transfer to replenish a depleted receiver is assumed to be 137 lbs of hydrogen. A receiving station is assumed to be comprised of 8 such receivers.

6.1 TRANSFER BY SIMPLE BLOWDOWN

6.1.1 DESCRIPTION AND OPERATION. The system is shown schematically in Figure 6-1.

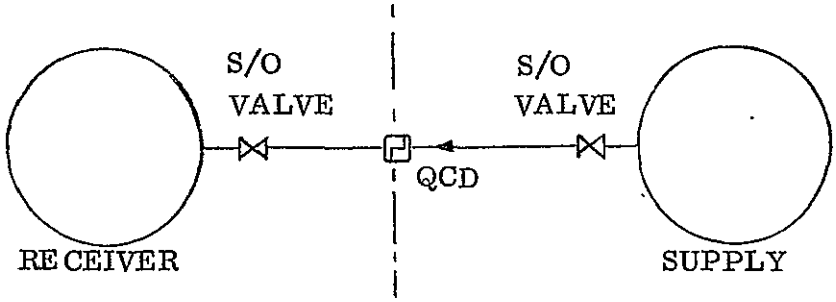


Figure 6-1. Schematic - Simple Blowdown System

Fluid is transferred from supply to receiver by pressure differential between the tw bottles. Transfer is terminated when pressure equilibrium is attained between the bottles. Control or crew involvement beyond connecting the bottles and operating th shutoff valves is not required.

6.1.2 DISCUSSION AND RESULTS. The sizing and feasibility analysis for this met of transfer was performed under the following conditions.

Equivalent Orifice Diameter of Transfer Line	= 0.5 inch	= constant
Heat Through Transfer Line	= 0	= constant
Heat to Supply Fluid	= 0	= constant
Heat to Receiver Fluid	= 0	= constant
Initial Fluid Mass in Supply	= 168 ± 2 lbs	= constant
Initial Receiver Pressure	= 100 psia	= constant
Initial Receiver Density	= 0.05 to 0.5 lb/ft^3	= variable
Receiver Volume	= 42.5 to 600 ft^3	= variable
Initial Supply Pressure	= 300 to 5000 psia	= variable
Supply Volume	= 42.5 to 200 ft^3	= variable

Figure 6-2 shows the effect of receiver volume on total mass transferred for the conditions listed. This figure indicates that the mass transferred for the listed conditions is far short of the desired quantity. In examining the entire range of data

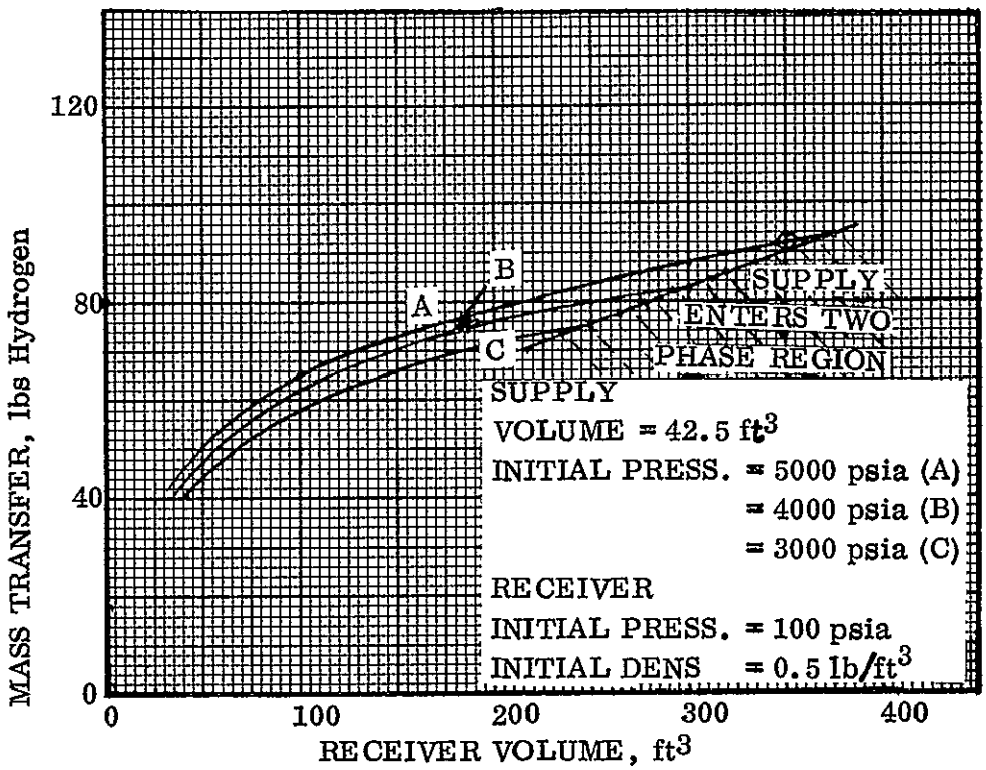


Figure 6-2. Supercritical Fluid Transfer by Simple Blowdown

generated from this study no combination of supply and receiver sizes that will transfer 137 lbs of hydrogen has been found. The trend is obvious however, that if a combination is found, the weights of the bottles will be prohibitively high.

A modification to the transfer procedure was made in an attempt to increase mass transfer. In this procedure the depleted receiver bottle was blowndown to 5 psia or less before initiating the transfer of fluid from the supply. This doubled fluid transfer under some conditions, but in all cases where the fluid state in the supply bottle remained supercritical or gaseous, the increase in transferred mass did not offset mass discarded during the initial receiver blowdown. When used in conjunction with simple blowdown this pre-blowdown scheme is not considered realistically feasible.

The following is a weight breakdown for the simple blowdown system transferring the most fluid, as determined from Figure 6-2. These weights should be used only for comparison with other systems discussed herein.

Receiver Bottle (Bare), 350 ft^3 @ 200 psia (peak) =	748 lbs
Supply Bottle (Bare), 425 ft^3 @ 5000 psia =	2467 lbs
Weight of System per 90 lbs Transferred Excluding Insulation, Instrumentation, etc.	3215 lbs

Based on the above discussion and weight analysis this system is considered unfeasible and no further consideration will be given to it.

6.2 TRANSFER BY HEATING SUPPLY FLUID

6.2.1 DESCRIPTION AND OPERATION. The system is shown schematically in Figure 6-3.

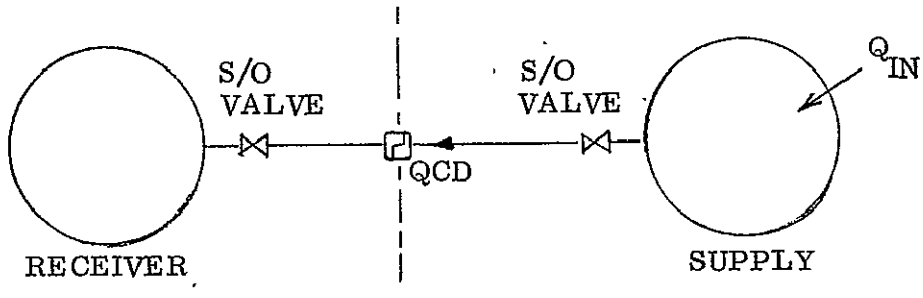


Figure 6-3. Schematic - Supply Heating System

Fluid is transferred from supply to receiver by pressure differential. The supply bottle pressure is maintained at a constant level by the addition of heat energy. Receiver bottle pressure is allowed to increase until it reaches the level of the supply pressure. Fluid transfer is terminated at this point.

6.2.2 DISCUSSION AND RESULTS. The feasibility analysis for this method of transfe was conducted under the following conditions.

Supply Bottle Volume	= 42.5 ft^3	= constant
Initial Supply Density	= 3.9 lb/ft^3	= constant
Receiver Bottle Volume	= 42.5 to 300 ft^3	= variable
Initial Receiver Density	= 0.1 to 0.5 lb/ft^3	= variable
Equivalent Orifice Dia of Transfer Line	= 0.06 to 0.12 in.	= variable
Heat Through Transfer Line	= 0	= constant
Heat Through Receiver	= 0	= constant
Heat to Supply Fluid	= as required to maintain supply bottle pressure constant	= variable
Supply Bottle Pressure	= 225 psia	= constant

Figure 6-4 shows the effect of receiver volume on total fluid mass transferred for the conditions stated. The required rate of heat energy addition to the supply fluid

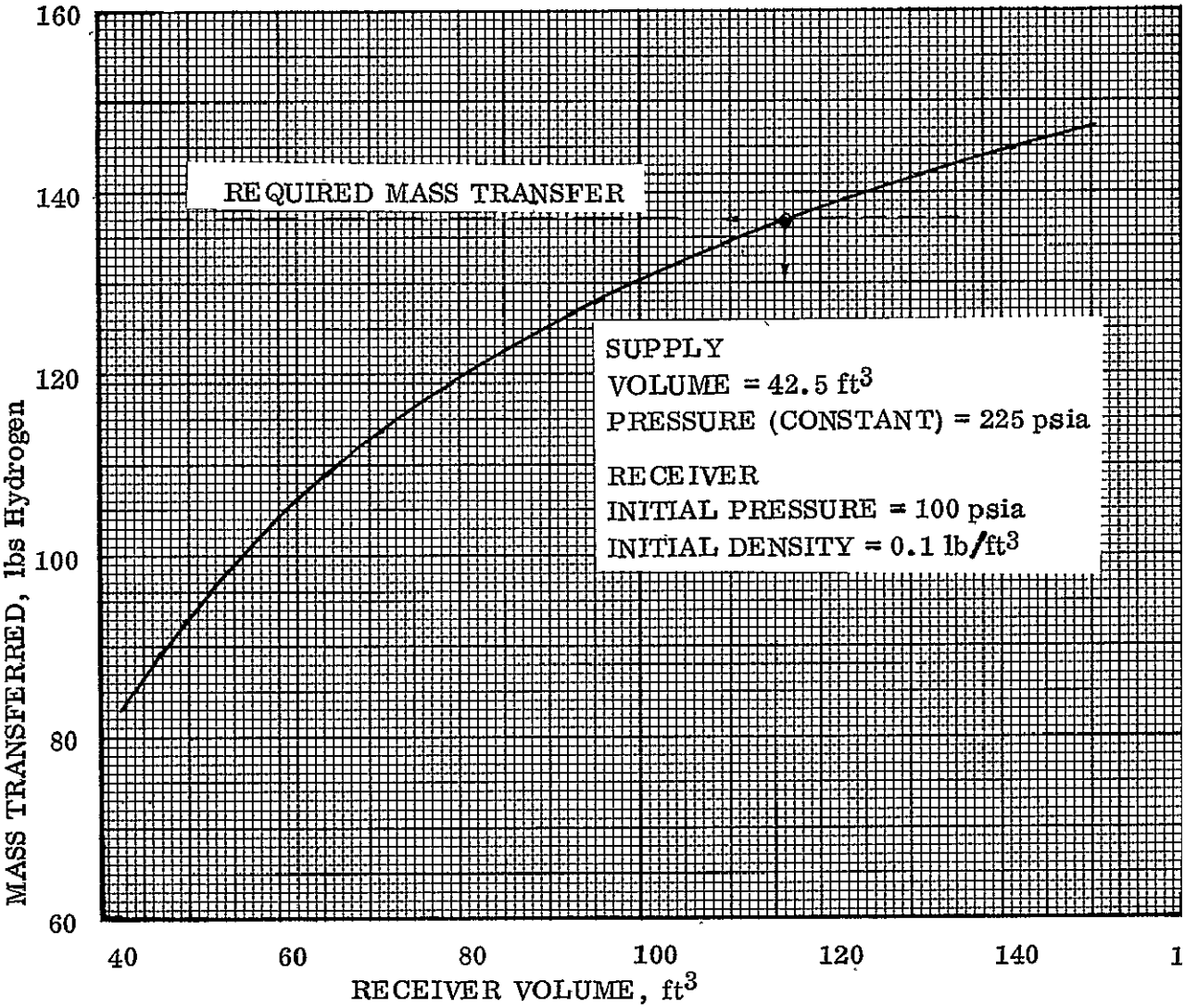


Figure 6-4. Supercritical Fluid Transfer by Heating Supply Bottle

is dependent on the transfer line diameter. Diameter increases require heat rate increases. The total energy required to transfer 137 lbs of hydrogen under the conditions stated in Figure 6-4 is approximately 18,000 Btu. The peak rate of energy addition was approximately 4 Btu/sec for a 0.06 inch transfer orifice diameter. At this rate the elapsed time to transfer was approximately 3.5 hours.

Weight analysis for this system must consider the source of heat energy. If the required energy is obtainable from the receiver stations existing power sources or waste processes, weight of the energy source would not be included as part of the fluid transfer system. The following weight summary, however, assumes an additional fuel cell type power supply is required. Furthermore its weight is shared with eight other supply bottles. Fuel cell weight is assumed to be 94 lbs/KW + 2.9 lbs/KW-hr.

The systems weight summary is as follows:

Supply Bottle (Bare), 42.5 ft^3, 225 psia	= 180 lbs
Receiver Bottle (Bare), 116 ft^3, 225 psia	= 350 lbs
Power Supply (see computation below)	= 65 lbs
Weight per 137 lbs Hydrogen Transferred (Excluding Insulation, Instrumentation, etc.)	= 595 lbs

Power supply weight was determined as follows:

4 Btu/sec = 4,210 KW @ 94 lb/KW = 396 lbs × 1/8*	= 49.5 lbs
18,000 Btu = 5.27 KW-hr @ 2.9 lb/KW-hr =	15.3 lbs
	64.8 lbs

*This assumes that one receiver is filled at a time.

Based on the above discussion and weight summary, this system is considered feasible and will be given further consideration.

6.3 TRANSFER BY PUMPING

6.3.1 DESCRIPTION AND OPERATION. The system is shown schematically by Figure 6-5.

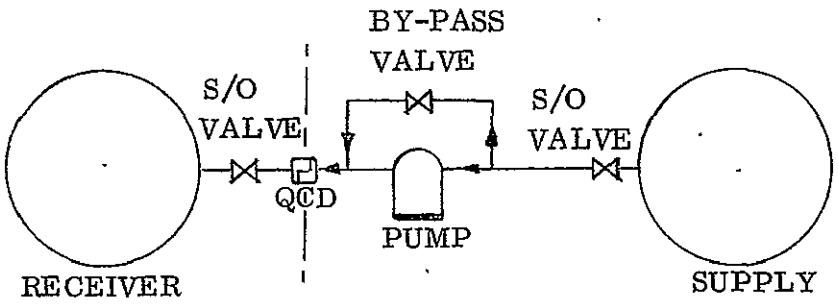


Figure 6-5. Schematic - Pumping System

In this system fluid is first transferred from the supply to the receiver by simple blow-down. When pressures are equalized the by-pass valve is closed and the compressor started. Transfer is terminated when the state of the supply fluid becomes two phase or when the receiver is fully loaded.

6.3.2 DISCUSSION AND RESULTS. The feasibility analysis for this method of transfer was accomplished under the following conditions.

Supply Bottle Volume	= 100 ft^3	= constant
Initial Supply Density	= 1.7 lb/ft^3	= constant
Equivalent Orifice Dia of Transfer Line	= 0.25 inch	= constant
Heat Through Transfer Line	= 0	= constant
Heat to Fluid	= 0	= constant
Initial Supply Pressure	= 1500 psia	= constant
Receiver Volume	= 42.5 to 100 ft^3	= variable
Initial Receiver Density	= 0.2 lb/ft^3	= constant

The Plumber Code was modified by adding subroutine MACHINE which computes work output required to compress the fluid from the supply bottle state to the receiver state.

Initial conditions of the supply bottle (volume, pressure, and density) were selected as optimum for an isentropic discharge of 140 lbs of hydrogen with the fluid state remaining gaseous. This would be a prime requirement for an actual system. The compressor was idealized as an electrically driven constant displacement device (constant speed piston type) operating at a rate of 1 ft^3/sec with a thermal efficiency of 85%.

The best system analyzed requires the compressor to put out approximately 600 Btu at a peak rate of 10.5 Btu/sec. At this rate the elapsed time to transfer was 5 minutes. Peak receiver bottle pressure was 1550 psia.

To reduce the size of the required electrical power supply, a smaller compressor and longer transfer time would be used in actual practice. In the weight summary presented below the electrical power supply is assumed to be the same as that used in Paragraph 6.2.2.

Weight summary:

Receiver bottle (bare), 100 ft^3 @ 1500 psia =	1700 lbs
Supply bottle (bare), 100 ft^3 @ 1500 psia =	1700 lbs
Compressor (est. by Ref. 3-6 × 1/8) =	30 lbs
Power Supply (est. as in Paragraph 6.2.2) =	50 lbs
Weight of System per 137 lbs Hydrogen Transferred = (Excluding Insulation, Instrumentation, etc.)	3480 lbs

Based on a comparison of the above weight with the equivalent weight from Paragraph 6.2, this system is considered unfeasible. No further consideration will be given to it.

6.4 TRANSFER BY HEATING SUPPLY AND COOLING RECEIVER

6.4.1 DESCRIPTION AND OPERATION. The system is shown schematically in Figure 6-6.

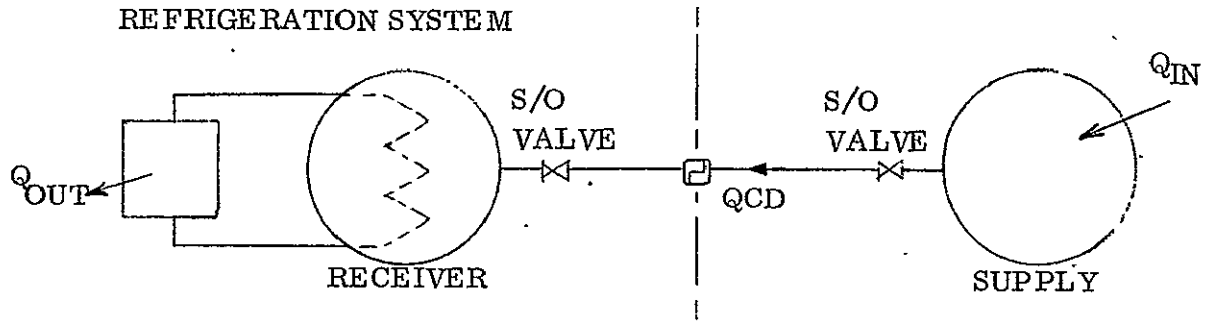


Figure 6-6. Schematic - Supply Heating/Receiver Cooling System

In this system fluid is transferred from the supply to the receiver by pressure differential. The absolute pressure levels in each bottle are maintained constant by adding heat energy to the supply and removing it from the receiver. Fluid transfer is terminated when the required mass transfer is achieved.

6.4.2 DISCUSSION AND RESULTS. The feasibility analysis for this system was conducted under the following conditions:

Supply bottle volume =	42.5 ft^3 =	constant
Receiver bottle volume =	42.5 ft^3 =	constant
Supply bottle pressure =	250 to 400 psia =	variable
Receiver bottle pressure =	200 to 300 psia =	variable

Analysis indicates that a minimum of approximately 26,000 Btu addition to the supply bottle and 36,000 Btu removal from the receiver is required to operate this system. In order to accomplish the transfer within the allotted time (3 hours/bottle) the average rates of heat transfer would be 2.4 Btu/sec added and 3.4 Btu/sec removal. Peak rates are estimated to be approximately twice the average or 4.8 Btu/sec and 6.8 Btu/sec respectively for addition and removal. According to the data presented in Reference 3-6, the weight of a spaceborne closed cycle refrigeration system with a capacity to remove 6.8 Btu/sec at a low temperature of 30°R is prohibitively high. As an example, 1000 lbs are required for 0.019 Btu/sec). This system is therefore considered unfeasible, and no further consideration will be given to it.

6.5 TRANSFER BY HEATING SUPPLY AND COOLING RECEIVER PLUS REGENERATI

6.5.1 DESCRIPTION AND OPERATION. The system is shown schematically in Figure 6-7.

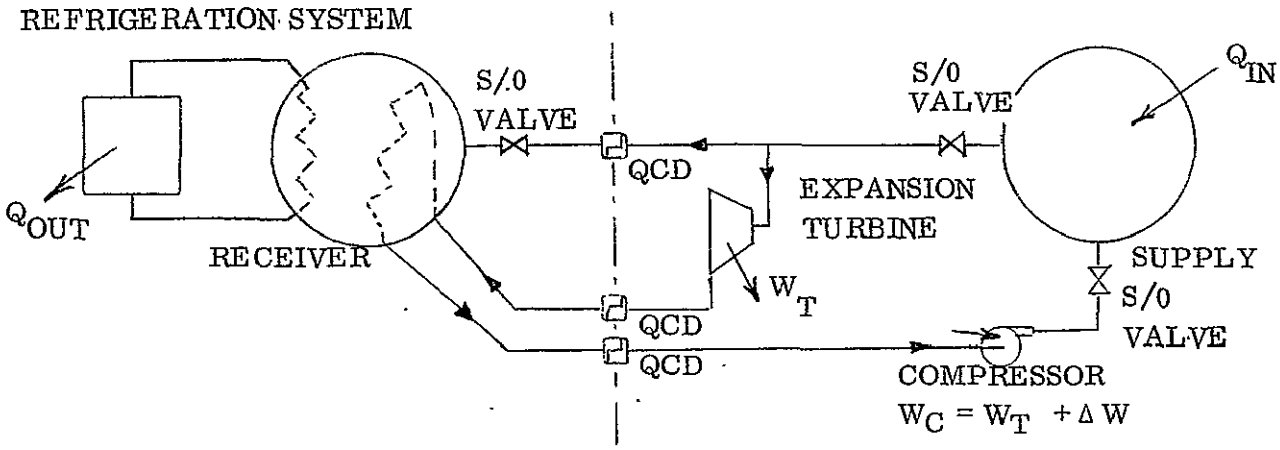


Figure 6-7. Schematic - Supply Heating/Receiver Cooling Plus Regeneration

In this system fluid is transferred from supply to receiver by a pressure differential. Pressure levels in each bottle are maintained constant by the addition and removal of heat energy as required. Fluid transfer is terminated when the required mass transfer is achieved.

6.5.2 DISCUSSION AND RESULTS. This system is basically the same as the one discussed in Paragraph 6.4 with the addition of an expansion turbine, return compressor, and extra valves and disconnects. In the analysis conducted, energy requirements for driving the system were reduced approximately 10 percent by the regenerative setup. This makes no appreciable difference in the refrigeration requirements and therefore the refrigeration package weights remain prohibitively high.

Based on the above discussion this system is also considered unfeasible and no further consideration will be given to it.

6.6 TRANSFER BY HEATING SUPPLY WITH VORTEX TUBE ASSIST

6.6.1 DESCRIPTION AND OPERATION. The system is shown schematically in Figure 6-8.

In this system fluid is transferred from supply to receiver by a pressure differential. The supply pressure is maintained at a constant level by adding heat energy from an external source and by circulating the hot fluid discharge of the vortex tube. A mixture of cold discharge fluid and cooled hot discharge fluid is transferred to the receiver. Transfer is terminated when the receiver pressure reaches the level of the supply pressure.

6.6.2 DISCUSSION AND RESULTS. A formal analysis of this system has not yet been conducted. In a preliminary examination this system was compared to the pure supply-heating type transfer system discussed in paragraph 6.2. By qualitative reasoning it

is anticipated that a reduction in receiver bottle size can be achieved because of the colder fluid being transferred. It is not expected however, that weight savings realized from such a reduction will offset added weight and complexity of the vortex tube and circulation pump. However, this system will be given further study to determine the validity of the above arguments.

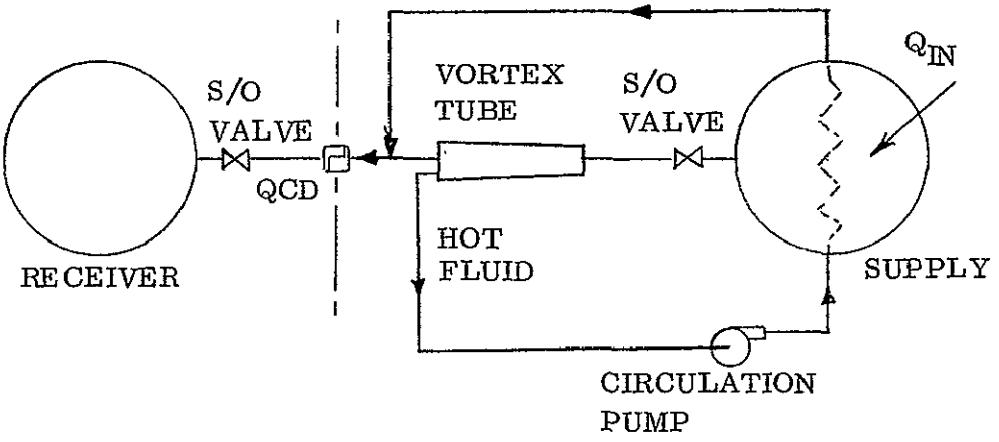


Figure 6-8. Schematic - Supply Heating Plus Vortex Tube Assist

SECTION 7

PROGRAM COST DATA

The following cost data estimates are presented as required by the NAS8-26236 contract document. All costs are approximate and are without fee.

1. Expenditures to date: $10,000

2. Estimated funds to completion: $22,966

3. Problem areas: None

SECTION 8

WORK TO BE PERFORMED

During the next month, analysis to develop a complete set of parametric data for the high pressure system will be continued. Work will be completed on the basic receiver tank and line chilldown analysis over the full range of line lengths and sizes and receiver tank sizes and design pressures, as presented in Reference 2-1.

SECTION 9

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