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SPACE LOX VENT SYSTEM

DESIGN PHASE REPORT

CR-123826

GENERAL DYNAMICS
Convair Aerospace Division

This report was prepared by the Convair Aerospace Division of General Dynamics under Contract NAS8-26972, "Space LOX Vent System," for George C. Marshall Space Flight Center, Marshall Space Flight Center, Alabama 35812. The English system of units was used for the basic calculations presented in this report.

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1.0

INTRODUCTION AND SUMMARY

The basic objective of the present program is to design and build a prototype vent system capable of exhausting only vapor to space from an all liquid or two-phase mixture of oxygen, while operating under low or zero-gravity conditions. This report covers work performed under the detail design phase of the program. This "design phase" report replaces the quarterly report originally scheduled for submittal at the end of June 1972. Budget and work status information are not included since monthly reports containing this information are being submitted for all months of the contract following March 1972. A new overall program schedule is presented in Reference 1-1, reflecting new reporting requirements and the addition of evaluation testing.

Work completed prior to the detail design is presented in the first and second quarterly reports (References 1-2 and 1-3). Reference 1-2 reports results of a literature survey and screening analysis of various systems applicable to low-g LO_2 tank venting which resulted in the selection of the thermodynamic type vent system for further analysis and predesign. Reference 1-3 reports on studies and comparisons between two basic types of thermodynamic vent systems, one employing a forced convection compact heat exchanger with pump and the other utilizing a natural convection distributed heat exchanger. This work resulted in the selection of the compact heat exchanger vent system shown in Figure 1-1 as the best overall system for the requirements of the present program. Work reported in References 1-2 and 1-3 was accomplished under the 1971 Convair Aerospace Independent Research and Development (IRAD) program.

Work performed during the detail design phase of the program was concerned with the finalization of vent system performance, development of component specifications, solicitation of vendor bids, selection of components and overall system package design.

Initially, the compact system preliminary design defined for the comparisons presented in Reference 1-3 was reviewed in light of a desirability to demonstrate complete tank mixing at one-g. Also, performance of the system at low-g conditions with a full tank and maximum temperature stratification or maximum pressure rise between vent cycles was investigated. It was found that under these extreme conditions, not previously considered, that use of a larger pump mixer than defined in Reference 1-3 would be desirable. In addition, to simplify ground testing with only a small weight penalty, the exchanger vent pressure was increased from 5 psia to 22 psia nominal. This resulted in less than a 0.2% increase in system weight.

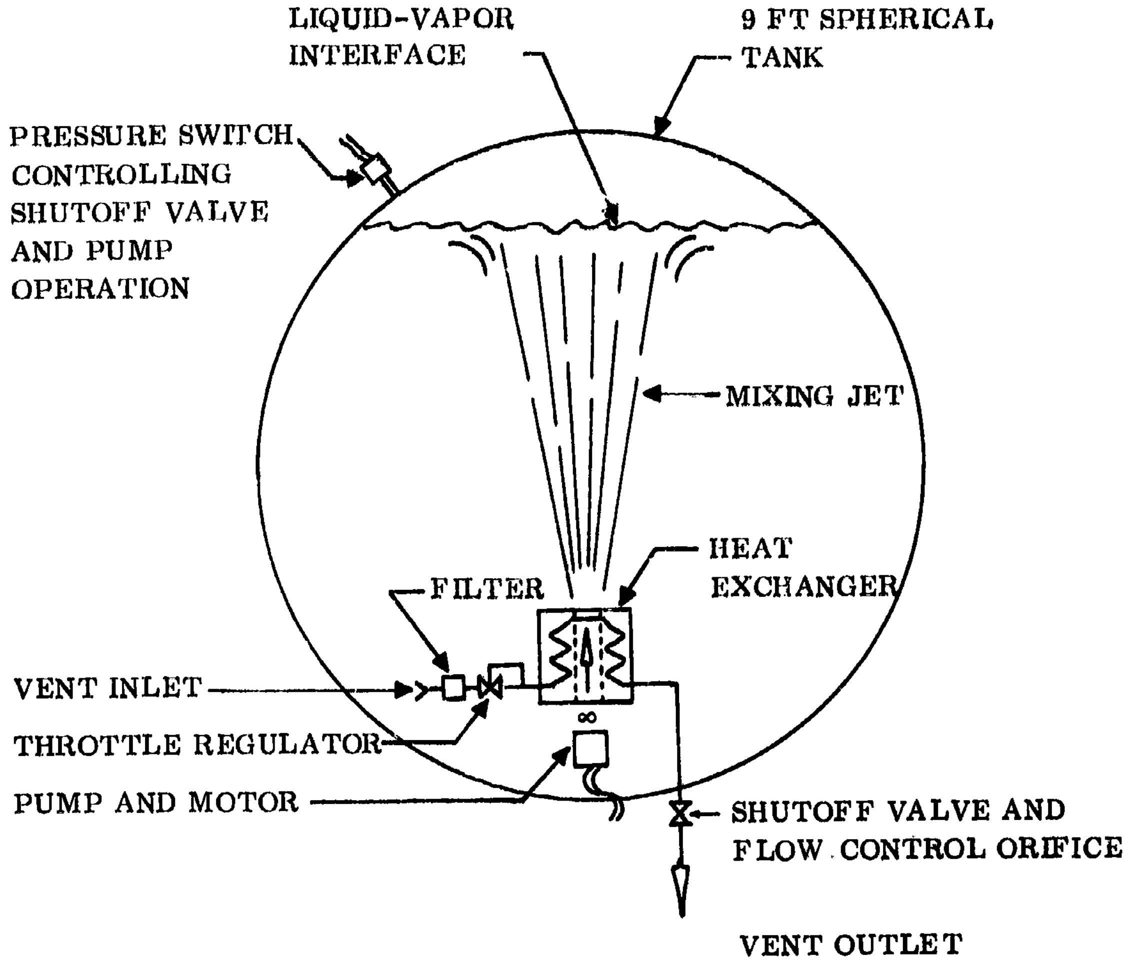


Figure 1-1. Compact Heat Exchanger Vent System Schematic

A summary of changes made to the Reference 1-3 system is presented below.

Pump Capacity:	Increased from .00067 m^3/sec @ .362 m heat (1.4 cfm @ 1.2 ft) to .00284 m^3/sec @ .91 m head (6.0 cfm @ 3.0 ft).
Pressure Switch Dead Band:	Increased from 3.45 to 10.3 kN/m^2 (0.5 to 1.5 psi) minimum.
Vent Flow Rate:	Decreased from .005 to .0047 kg/sec (40 to 37.5 lb/hr) nominal.
Throttling Pressure:	Increased from 34.5 to 152 kN/m^2 (5 to 22 psia) nominal.

It was determined that the above differences would not change the basic results of the previous system comparisons and screening studies reported in References 1-2 and 1-3.

A summary of procedures used to arrive at final system and component design criteria is presented in Section 2.0 with background data contained in Reference 1-4. The overall system package along with final system and component operating characteristics is presented in Section 3.0.

2.0

SYSTEM DEFINITION

This section summarizes the analytical procedures used to determine the specific operating requirements of the overall space LOX vent system and the individual system components. The basic vehicle requirements used in defining the vent system are presented in Table 2-1. System screening and analytical trade-offs described in References 1-2 and 1-3 resulted in the selection of the bulk heat exchanger concept illustrated in Figure 1-1. The basic analytical steps used in defining operating details of this system are presented below.

2.1 PUMP DEFINITION.

The first task was to define a family of pumps which could meet the basic low-g vent requirements, which in this case are to (1) pump saturated liquid and/or saturated or superheated gaseous oxygen through a heat exchanger and (2) to mix the fluid in a propellant tank at specified gravity levels. Basic design data are presented in Table 2-1.

The flow through the heat exchanger must be sufficient to provide heat transfer to vaporize any liquid oxygen, up to 100%, which may be present at the vent inlet. Pump flow rates and exchanger head losses for heat transfer purposes are defined by iterative calculations as a function of vent rate and exchanger sizing or flow geometry factors. The CHEAP computer program described in Reference 2-1 is used for this purpose. Typical calculations which were made for this study are presented in Reference 1-3.

Table 2-1. Basic Vehicle Requirements for Vent System Definition

Design Element	Specification
Fluid	Oxygen
Overall Tank Pressure Range	103.5 - 345 kN/m^2a (15-50 psia)
Tank Pressure Control Range	310 ± 13.8 kN/m^2a (45 ± 2 psia)
Operational Tank Fluid Temperature	89° to 103°K (160° to 185°R)
Overall Environmental Temperature Range	89° to 244°K (160° to 440°R)
Vent Inlet Quality	0 to 100%
Total Operational Steady-State Heat Leak	32.2 to 35.2 watts (110 to 120 Btu/hr)
Mission Duration	605 to 2,590 ks (7-30 days)
Minimum Life Time	100 Missions
Tank	Spherical (9 ft dia)
Coast Acceleration Levels	10^{-4} - 0 g's

The minimum energy required for mixing is determined on the basis of work (Ref. 2-2) performed by the Ft. Worth Operation of Convair Aerospace. Mixing is intended to be accomplished by a small high velocity jet issuing into the bulk fluid as shown in Figure 1-1. Minimum mixing velocities are based on requirements for penetrating the warm layer of liquid at the liquid/vapor interface.

The following equation from Reference 2-2 is used for determining the minimum energy required to penetrate the liquid/vapor interface:

$$(V_o D_o) = \frac{1}{2}\left[\frac{\beta \Delta T_{max} Z^3 aP}{\left[1-(V_{max}/V'_{max})^2\right](P+1)(P+3)}\right]^{1/2} \qquad (2\text{-}1)$$

where

$(V_o D_o)$ =	velocity-diameter product at mixer outlet required to penetrate warm liquid layer at vapor/liquid interface.
β =	coefficient of volumetric expansion for the liquid.
ΔT_{max} =	maximum temperature difference between bulk liquid and liquid/vapor interface (assumed to be 0.5K [1R]).
Z =	distance from mixer to liquid/vapor interface.
a =	local acceleration
P =	exponential constant (assumed to be .8 from Ft Worth water tests)
V_{max} =	maximum centerline velocity with a temperature gradient
V'_{max} =	maximum centerline velocity without a temperature gradient (V_{max}/V'_{max} assumed to be .9 from Ft Worth data)

Based on $a = 10^{-4}$ g's, Z = 2.58 meters (8.5 ft), and β = 0.00521/°K (0.0029/°R) the pump $V_o D_o$ required for mixing was determined from Equation 2-1 to be 0.00272 m^2/sec (0.0293 ft^2/sec).

In order to find the actual pump head and flow required for a given $V_o D_o$ mixing parameter the following equations were used.

$Q = A_o V_o$	Continuity Equation
$H = V_o^2/2g_c$	Total Free Stream Head Loss
$A_o = \pi D_o^2/4$	Geometry for Circular Discharge

were

Q = volume flow rate through pump

A_o = flow area at pump exit

V_o = flow velocity at pump exit

H = total head loss due to mixing

g_c = gravitational constant

D_o = diameter of flow at pump exit

By combining the above three equations, the following equation of head versus flow capacity in terms of V_oD_o was derived:

$$Q = \frac{(V_oD_o)^2 \pi}{4\sqrt{2g_cH}} \tag{2-2}$$

In the overall analysis three pumps with AC induction motors sealed from the oxygen environment were chosen for further analysis, as being capable of meeting the mixing energy requirements and still have adequate power remaining to accomplish hot side heat transfer in the exchanger. The pertinent characteristics of these pumps are presented below.

Pump	Flow m^3/sec (CFM)	Head m (Ft)	$(V_oD_o)_{max}$ m^2/sec (ft^2/sec)
No. 1	6.6×10^{-4} (1.4)	0.366 (1.2)	0.0474 (0.51)
No. 2	2.84×10^{-4} (6.0)	0.915 (3.0)	0.124 (1.33)
No. 3	2.74×10^{-3} (5.8)	1.203 (3.95)	0.13 (1.4)

It is noted that other pump devices such as a brushless D.C. motor and vent gas drive turbine were investigated and discarded in favor of the A.C. motor. Details of these tradeoffs are presented in Reference 1-3.

2.2 TOTAL SYSTEM WEIGHT

The next step was to estimate total vent system weight when using each of the pumps defined by the work described in Section 2.1.

The total system weight is taken to consist of vented propellant, electrical power supply and exchanger hardware.

The weight of vented propellant was calculated by the following formula from Reference 2-3:

$$Wt_{v_p} = \left[\frac{\dot{Q}_{in}}{\dot{m}_v\,[(e\,\lambda/1-e) + h_o - h_\ell] - \dot{P}_{in}}\right] t\,\dot{m}_v \qquad (2\text{-}3)$$

where

$\dot{Q}_{in}$ = total rate of heat transfer into tank

t = total mission time

$\dot{m}_v$ = vent rate while venting

e = ratio of saturated vapor to liquid density

λ = latent heat of vaporization at tank pressure

h_o = specific enthalpy at vent outlet

h_ℓ = specific enthalpy of bulk liquid

$\dot{P}_{in}$ = total power into tank via pump motor

The weight of the power supply necessary to drive the motor was assumed to be the same as for a d-c fuel cell operating on hydrogen and oxygen and represented by the following formula from Reference 2-4:

$$Wt_{ps},\ Kg = 42.5\ (\dot{P}_{in},\ KW) + 0.000365\ (\dot{P}_{in},\ KW)\ (t_o,\ sec)$$

or (2-4)

$$Wt_{ps},\ lb = 94\ (\dot{P}_{in},\ KW) + 2.9\ (\dot{P}_{in},\ KW)\ (t_o,\ sec)$$

where

t_o = time that pump is actually on.

This time (t_o) is determined from Equation 2-3 where

$$\text{Time On } (t_o) = \frac{\text{Total Propellant Vented}}{\text{Vent Rate While Venting}}$$

or

$$t_o = \left[\frac{\dot{Q}_{in}\ t}{\dot{m}_v\ [(e\lambda/1-e) + h_o - h_\ell] - \dot{P}_{in}} \right] \qquad (2\text{-}5)$$

Heat exchanger weights were determined for each pump system by iterative calculations, as described in Reference 1-5, using the CHEAP computer program (Reference 2-1). Exchanger weights were plotted as a function of vent flow rate and equations of the best fit curve derived. Resulting exchanger weight equations for pump no. 1 are presented below.

$$Wt_{ex},\ Kg = 12,172\ (\dot{m}_v,\ Kg/sec)^2 + 370\ (\dot{m}_v,\ Kg/sec) + 1.0$$

or (2-6)

$$Wt_{ex},\ lb = 0.000426\ (\dot{m}_v,\ lb/hr)^2 + 0.103\ (\dot{m}_v,\ lb/hr) + 2.245$$

For pumps no. 2 and no. 3

$$Wt_{ex},\ Kg = 10,614\ (\dot{m}_v,\ Kg/sec)^2 + 360\ (\dot{m}_v,\ Kg/sec) + 1.32$$

or (2-7)

$$Wt_{ex},\ lb = 0.000368\ (\dot{m}_v,\ lb/hr)^2 + 0.1\ (\dot{m}_v,\ lb/hr) + 3$$

The head loss allowed for flow through the exchanger, for each pump, is based on the condition where the exchanger head loss is the minimum necessary to prevent excessive heat exchanger weight, i.e. for a given hot side flow rate the pressure drop or head loss through the exchanger must be above a certain minimum in order to have efficient vortexing flow as required to provide forced convection heat transfer under all anticipated orientations and/or acceleration levels.

Final pump system characteristics used are summarized in Table 2-2.

Table 2-2. Pump System Operating Parameters Used in Final Design Analysis

	Pump No. 1	Pump No. 2	Pump No. 3
Flow, m^3/sec (CFM)	6.6×10^{-4} (1.4)	2.84×10^{-3} (6.0)	2.74×10^{-3} (5.8)
Total Head, m (ft)	0.366 (1.2)	0.915 (3.0)	1.203 (3.95)
$(V_oD_o)_{max}$, m^2/sec (ft^2/sec)	0.0474 (0.51)	0.124 (1.33)	0.13 (1.4)
Exchanger Head Loss, m (ft)	0.244 (0.8)	0.305 (1.0)	0.305 (1.0)
$(V_oD_o)_{mixing}$, m^2/sec (f^2/sec)	0.0361 (0.389)	0.1115 (1.2)	0.121 (1.302)

Combining equations 2-3, 2-4, 2-5 and 2-6, the total weight for pump system No. 1 is

$$W_T, \text{Kg} = \left[\frac{\dot{Q}_{in}\, t}{\dot{m}_v\,(e\lambda/1-e) + h_o - h_\ell) - \dot{P}_{in}}\right](\dot{m}_v + 3.65 \times 10^{-7}\,\dot{P}_{in})$$

$$+ .0426\,\dot{P}_{in} + 12171.6\,\dot{m}_v^2 + 370.4\,\dot{m}_v + 1.0 \qquad (2\text{-}8)$$

where

$\dot{Q}_{in}$ joules/sec, t sec, $\dot{m}_v$ Kg/sec, e dimensionless, λ joules/Kg, h_o and h_ℓ joules/Kg, $\dot{P}_{in}$ watts

or

$$W_T, \text{lb} = \left[\frac{\dot{Q}_{in}\, t}{\dot{m}_v\,[(e\lambda/1-e) + h_o - h_\ell] - 3.419\,\dot{P}_{in}}\right](\dot{m}_v + .0029\,\dot{P}_{in})$$

$$+ .094\,\dot{P}_{in} + .000426\,\dot{m}_v^2 + .103\,\dot{m}_v + 2.245 \qquad (2\text{-}9)$$

where

$\dot{Q}_{in}$ Btu/hr, t hr, $\dot{m}_v$ lb/hr, e dimensionless, λ Btu/lb, h_o and h_ℓ Btu/lb, $\dot{P}_{in}$ watts

For pump systems no. 2 and no. 3

$$W_T, \text{Kg} = \left[\frac{\dot{Q}_{in}\, t}{\dot{m}_v\,[(e\lambda/1-e) + h_o - h_\ell) - \dot{P}_{in}}\right](\dot{m}_v + 3.654 \times 10^{-7}\,\dot{P}_{in})$$

$$+ .0426\,\dot{P}_{in} + 10.514\,\dot{m}_v^2 + 360\,\dot{m}_v + 1.32 \qquad (2\text{-}10)$$

where units are the same as for Equation 2-8, or

$$W_T, \text{lb} = \left[\frac{\dot{Q}_{in}\, t}{\dot{m}_v\,[(e\lambda/1-e) + h_o - h_\ell) - 3.419\,\dot{P}_{in}}\right](\dot{m}_v + .0029\,\dot{P}_{in})$$

$$+ .094\,\dot{P}_{in} + .000368\,\dot{m}_v^2 + .100\,\dot{m}_v + 3.0 \qquad (2\text{-}11)$$

with units as for Equation 2-9.

Equations 2-8 through 2-11 are the same as Equations on pages 3-13 and 3-20 of Reference 1-3 except that when using English units a factor of 3.419 times $\dot{P}_{in}$ was found to be missing in the denominator of the Reference 1-3 Equations.

Weight data obtained from Equations 2-9 and 2-11 are presented in Figure 2-1 for the three pumps. It is noted that prior to actual fabrication and test there is some uncertainty as to the actual input power required for the "canned" (sealed from O_2) motors. For the small pump, no. 1, predicted power inputs were between 35 and 42 watts. Initial estimates (Reference 1-4) for the 6 cfm pump no. 2 indicated a maximum power of 88 watts. Subsequent vendor data (Reference 2-5) showed a potential range of 60 to 80 watts. Power estimates for pump no. 3 are presented in Reference 1-3. Curves for all cases are included in Figure 2-1.

An examination of Figure 2-1 shows that the small pump operating at vent flows on the order of 80 lb/hr has the lowest total system weight. However, with the system operating at flow rates above approximately 1.4 lb/hr intermittent venting must be accomplished and the time required to mix becomes an important parameter. In fact this requirement fixes the maximum allowable vent rate for each pump system, as described in the following section.

2.3 VENT OPERATING POINT

It is assumed that the mode of operation for this vent system is intermittent and propellant mixing will occur only during venting and the propellant may be quiescent between vent periods, resulting in a non-homogeneous pressure rise. It is further assumed that for efficient system operation, the propellant must be completely mixed during each vent period. Thus the time to vent would need to be at least as great or greater than the time required to mix.

The time required to mix is obtained from the following equation.

$$\theta_m = 3 \times \frac{118 \sqrt{H}\, D_t}{(V_o D_o)^{2/3}\, g_c^{1/6}} \left[\frac{V_o D_o \rho}{\mu} \right]^{-1/6} \qquad (2\text{-}12)$$

where

θ_m = mixing time

H = height of liquid/vapor interface above mixing nozzle

D_t = tank diameter

V_o = mixing nozzle outlet velocity

D_o = mixing nozzle diameter

g_c = gravitational const.

ρ = liquid density

μ = liquid viscosity

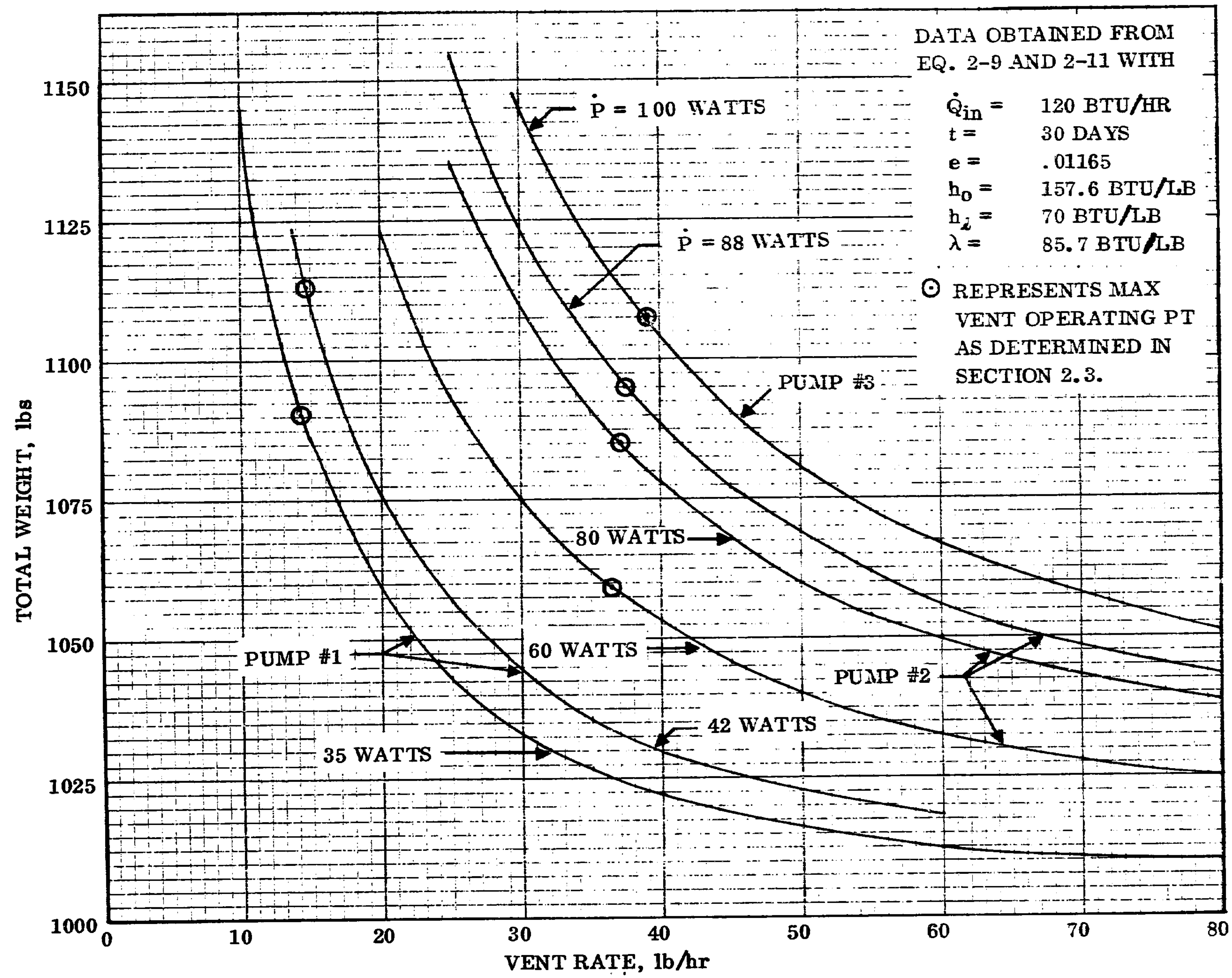


Figure 2-1. System Weight Versus Vent Rate

The above equation is obtained from Reference 2-1 as a reasonable estimate of mixing time based on past destratification testing performed at Convair Aerospace with LH_2. The solution to Equation 2-12 is presented in Figure 2-2 as a function of liquid height above the mixing nozzle (H) and the product of mixing nozzle diameter and jet exit velocity ($V_o D_o$). It is noted that this figure is the same as Figure 2-3 of Reference 1-4 except that a typographical error was found in the Reference 1-4 figure with respect to labeling of the abscissa.

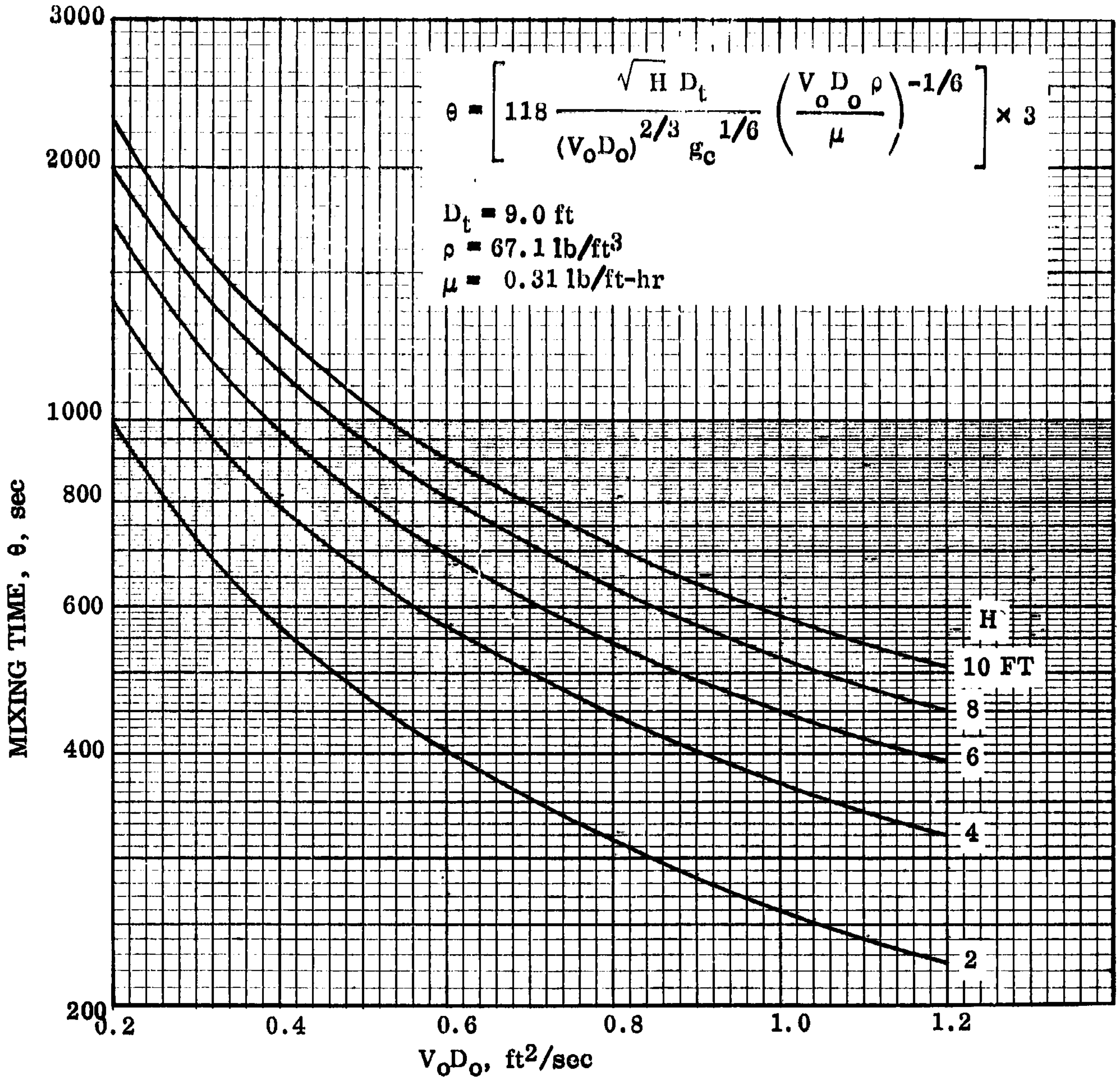


Figure 2-2. Solution of Mixing Time Equation as Function of $V_o D_o$ and Liquid Height

Determination of the time available for mixing during the vent cycle is outlined below. This time is dependent on the total pressure decrease to be achieved and the rate of change of pressure during the vent. The pressure switch dead band defines the total pressure change. The rate of pressure decay is dependent on a number of variables, including tank conditions during the pressure rise prior to venting, and can be related to the rate of change of pressure of a mixed (homogeneous) tank. To illustrate, a complete pressure cycle is graphically depicted in Figure 2-3.

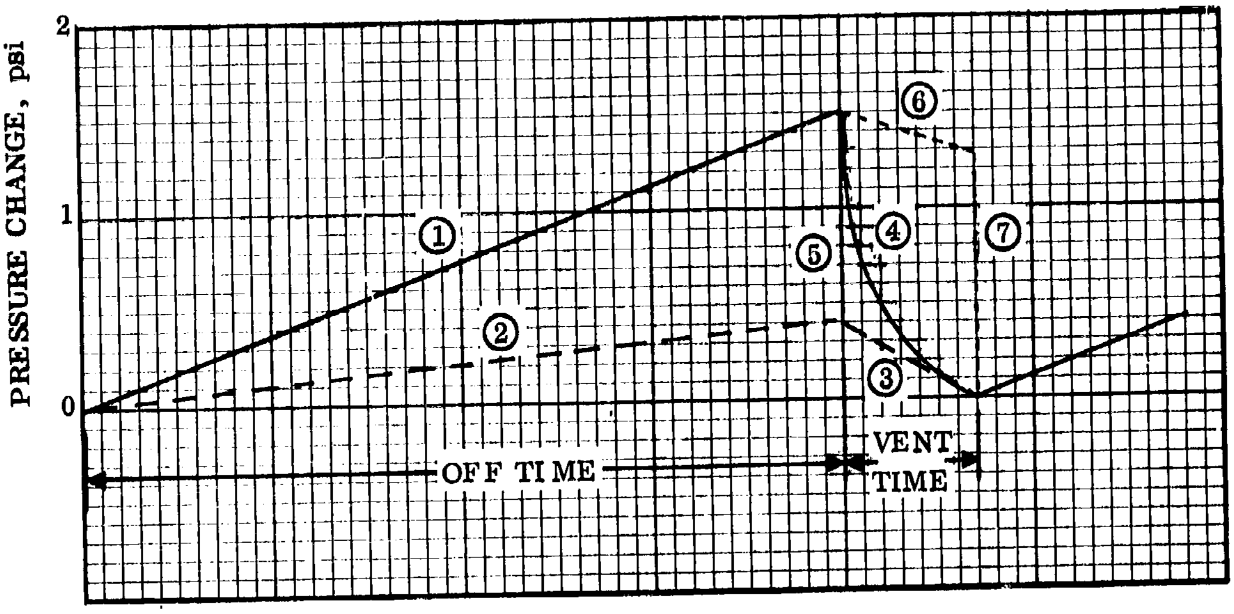


Figure 2-3. Typical Vent Cycle With Adequate Mixing

The heavy line, consisting of segments 1 and 4, represents what is expected to be a typical pressure history during a cycle in which non-homogeneous conditions prevail during the non-vent portion of the cycle. The two dashed lines (2 and 3) represent the pressure cycle for mixed tank conditions with the pressure rise (vent system off) time restricted to that for the non-mixed pressure rise case.

The pressure profile of lines 5 and 3 represent an idealized case in which instantaneous mixing of the bulk propellant occurs at the start of venting. This is illustrated to show that in this case the major portion of the overall pressure change is due to mixing. Line 4 represents the combination effect of simultaneous mixing and venting. The profile of lines 6 and 7 represent a limiting case in which mixing is delayed until the end of the vent time. In reality the vent profile (line 4) may fall anywhere within the envelope defined by lines 3, 5, 6 and 7 for efficient vent performance as long as complete mixing occurs within the time defined as "vent time" in Figure 2-3.

For the pressure cycle (lines 1 and 4) defined by Figure 2-3 and per the above discussion, the vent time can be computed from the following four variables.

a. Total pressure change (pressure switch deadband)

b. Non-homogeneous pressure rise rate (slope of line 1)

c. Homogeneous pressure rise rate (slope of line 2)

d. Homogeneous venting pressure decay rate (slope of line 3)

The non-homogeneous pressure rise rate is taken from Reference 2-6 as

$$\frac{\Delta p}{\Delta t} = 1450 \, (\dot{Q}/MS)^{1.14} \qquad (2\text{-}13)$$

where

$\Delta p/\Delta t$ = pressure rise rate, psi/hr

$\dot{Q}$ = tank heating rate, BTU/hr

M = total mass of O_2 in tank, lb_m

S = Ullage volume as % of tank volume

The pressure rise and decay rates for a homogeneous system are determined from the EQPR computer program described in Reference 2-1.

Data presented in Reference 1-4 shows that available vent times and allowable mixing times are dependent on ullage volume and that the most critical condition (maximum allowable vent rate) occurs at the minimum ullage.

It was decided (Reference 1-4) that designing for operation at lower ullages than 5% was not necessary to meet the requirements of the present application. Following are the steps accomplished to determine the maximum allowable vent rate for each pump system.

a. Calculate the maximum expected pressure rise rate under stratified conditions from Equation 2-13. For the present case with 5% ullage $\Delta P/\Delta t$ was found to be 0.99 N/m^2-sec (0.516 psi/hr).

b. Determine the minimum time for the tank pressure to rise from system deactuation to actuation. For the present case with a minimum deadband of 1.5 psi and pressure rise rate from (a) this is 0.000805 sec (2.9 hrs).

c. Determine the pressure rise of the mixed case over the same time as determined in (b) above. The mixed rise rate is determined using the EQPR computer program described in Reference 2-1. For the present case $(\Delta P/\Delta t)_{mixed}$ = 0.0441 N/m^2-sec (0.0231 psi/hr) and ΔP_{mixed} = 461 N/m^2 (0.067 psi).

d. Determine the maximum allowable mixed pressure decay rate for mixing to occur within the available vent down time. Mixing time is determined from Figure 2-2 with H = 8 ft (5% ullage) for each pump using the V_oD_o mixing values presented in Table 2-2. The allowable mixed pressure change is obtained from (c) and for the present case the allowable decay rate = 0.067 psi/mixing time.

e. Calculate values of the mixed pressure decay rate as a function of vent rate for each of the pump input powers of interest. The EQPR program is used and data obtained for the present case is plotted in Figure 2-4, for 35, 42, 60, 80, 88 and 100 watt power inputs.

f. The allowable maximum vent rate is then determined from matching the allowable pressure decay rate determined in (d) with actual decay rates obtained in (e) as a function of vent rate. Maximum values of vent rate obtained in this manner are marked on the curves of Figure 2-1 showing the minimum total system weight obtainable with each pump system.

2.4 SELECTION OF PUMP SYSTEM

Referring to Figure 2-1 it is seen that when operating at maximum allowable flow rates the total weights for the three pump systems are comparable; with pump no. 2 having somewhat the lowest potential weight. In comparison with pump no. 1 it is seen that the total weight for the pump no. 2 system is less sensitive to slight changes in vent rate which can be caused by inaccuracies in the flow control hardware. Also, the additional mixing power available with the larger pump, no. 2, will facilitate demonstration testing at 1-g and will allow flexibility in testing since its speed can be reduced and tank mixing investigations made at reduced flow and power. The 100 watt pump, no. 3, does not provide enough additional mixing power (Table 2-2) over that of pump no. 2 to warrant the increased system weight.

Based on the above discussion, pump no. 2, having a nominal flow of 2.84×10^{-3} m^3/sec (6.0 cfm) and head of 0.915 m (3.0 ft), was chosen to form the nucleus of the present LOX vent system.

EQPR COMPUTER PROGRAM (REFERENCE 2-1) USED WITH 380 FT^3 TANK VOLUME, 5% ULLAGE, 120 BTU/HR TOTAL HEAT INPUT, 45 PSIA TANK PRESSURE, 87.6 BTU/LB DIFFERENCE IN ENTHALPY BETWEEN SATURATED TANK LIQUID AND VENT GAS

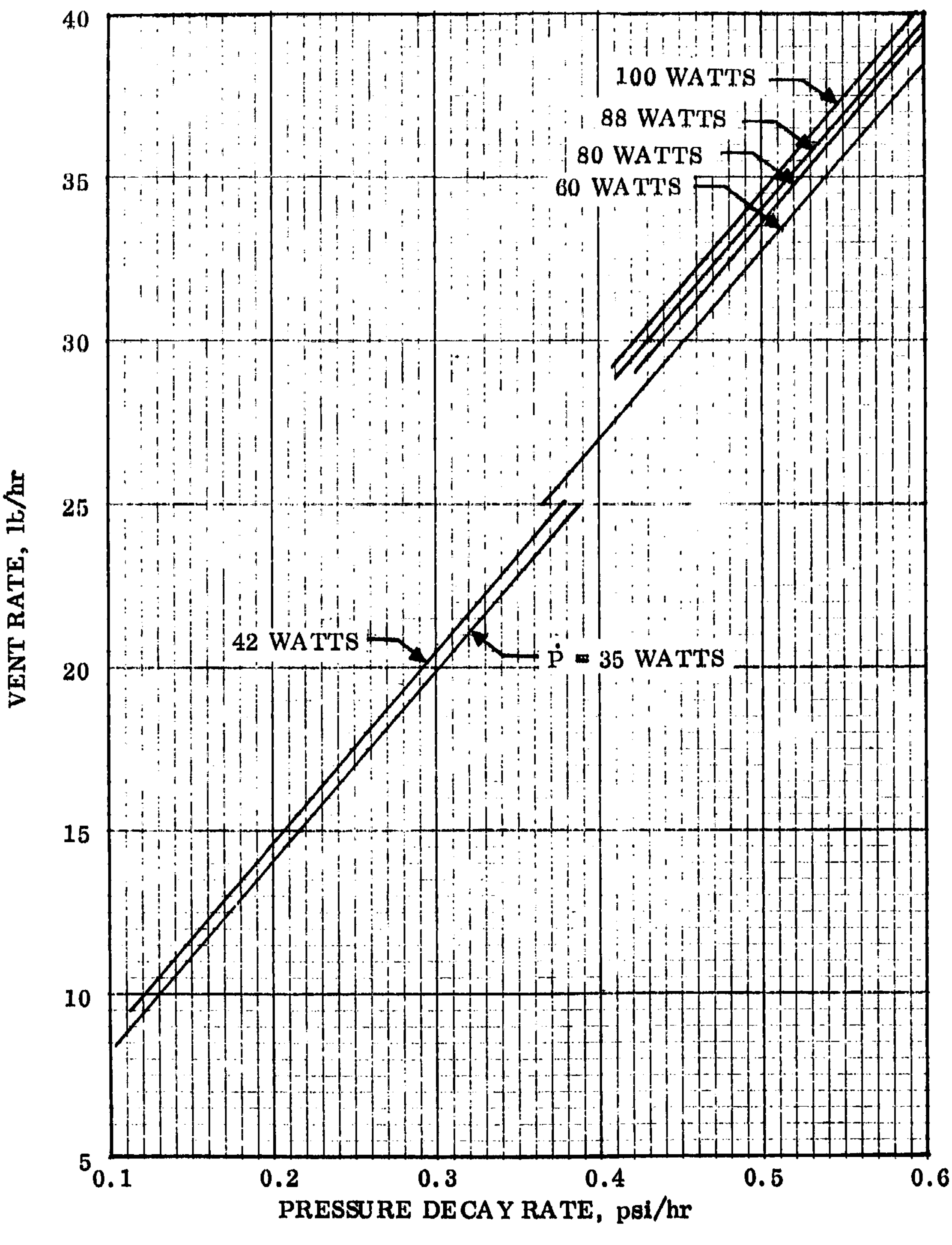


Figure 2-4. Mixed Fluid Pressure Decay Rate for a Venting O_2 Tank

3.0

FINAL DESIGN PACKAGE

This section presents the overall systems package resulting from the analyses discussed in Section 2.0 and the selection of specific vendor hardware. Final overall system and component operating characteristics are included.

It is noted that the data presented in Section 2.0 was based on an exchanger vent pressure of 5 psia. However, in order to simplify ground testing with only a small weight penalty, the exchanger vent pressure was increased from 5 psia to 22 psia nominal. The effect on weight of this change in pressure is illustrated by the data in Reference 1-3, and for the present case resulted in a total weight increase of less than 2 lb or 0.2%.

An assembly drawing of the in-tank vent system hardware, including provisions for pressure and temperature instrumentation, is presented in Figure 3-1. The pressure switch and shutoff valve are located separately outside the LO_2 tank and are thus not shown in Figure 3-1.

Overall system and component operating characteristics are outlined in the following sections.

3.1 OVERALL SYSTEM

The system schematic is presented in Figure 1-1. The overall function is to control oxygen tank pressure to 45 ± 2 psia while allowing only superheated vapor to be exhausted to space. Operation is intermittent and the vent flow is nominally 37.5 lb/hr while venting. External heating of the tank is nominally 110 to 120 Btu/hr.

The following general performance characteristics apply to each of the components as well as the overall system.

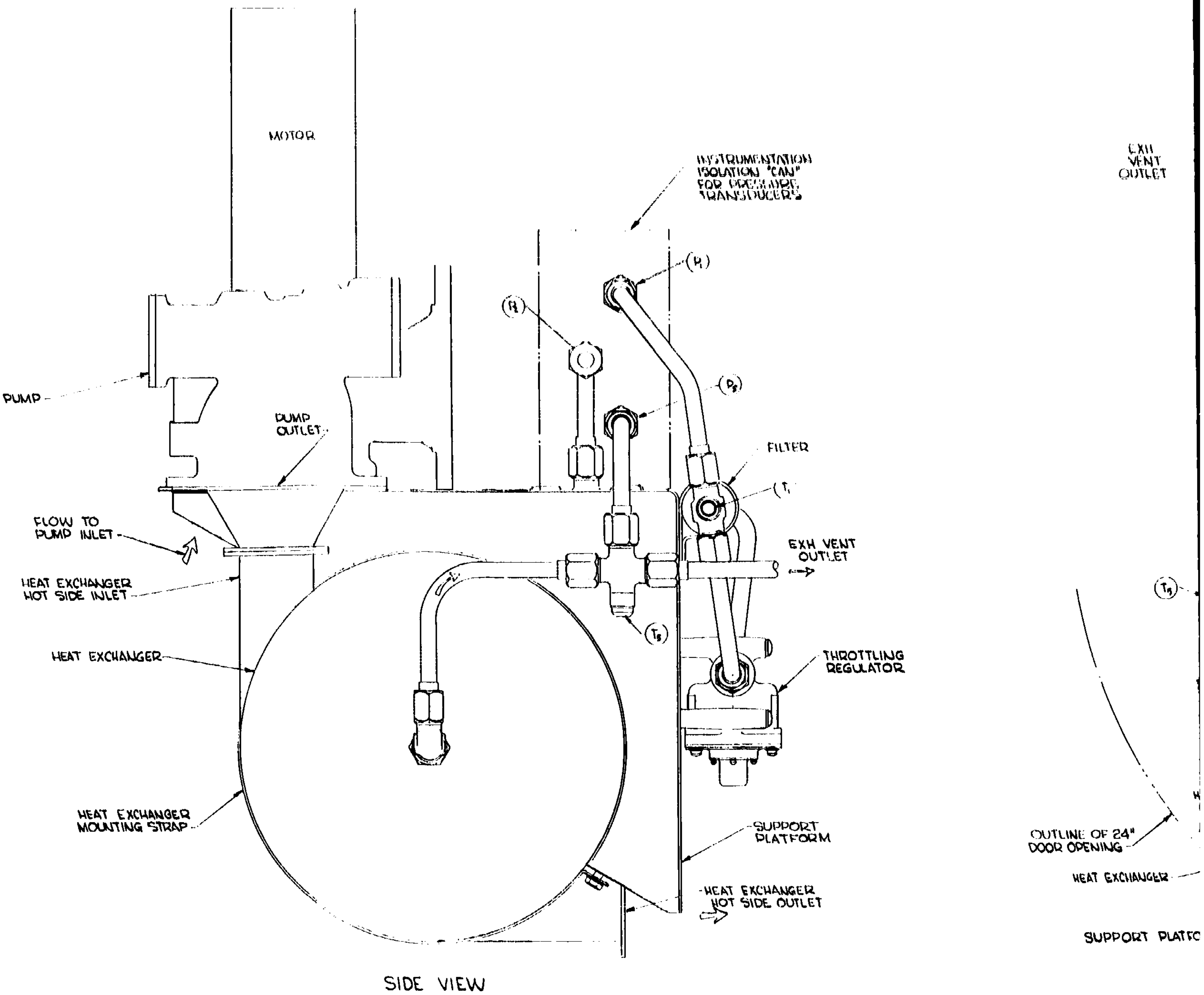
Service Life:	3000 hours	Operating
	69000 hours	Non-operating
	72000 hours	Total
Run Duration:	Maximum continuous run time, 4.0 hours. Minimum continuous run time, 15 seconds.	
Cycles:	Minimum of 30,000 start-run-stop cycles.	
Temperature Shock:	That experienced during the normal loading of a LO_2 storage tank.	

FOLDOUT FRAME 1

NOTES:

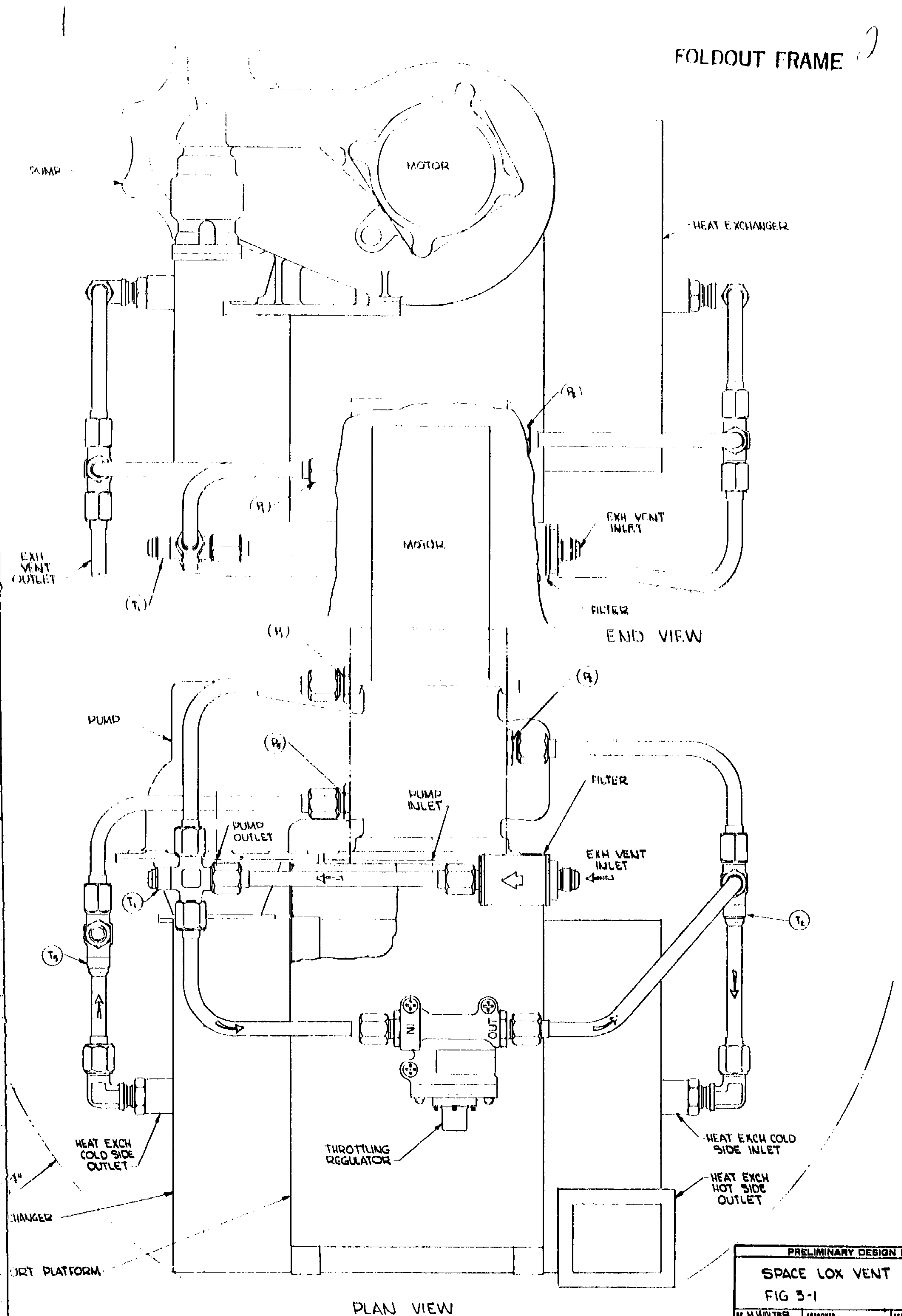
1. (P)(T) CALLOUTS REFER TO POINTS PROVIDED FOR PRESSURE & TEMPERATURE TAPS

2. SCALE
10 INCHES



SIDE VIEW

FOLDOUT FRAME



PRELIMINARY DESIGN DRAWING

SPACE LOX VENT ASSY

FIG 3-1

BY M. WALTER	APPROVED	SCALE FULL	DATE 3-10-72
CONVAIR DIVISION OF GENERAL DYNAMICS SAN DIEGO, CALIFORNIA		DRAWING NO LOV 72-001	

Acceleration Levels:	0 to 1 g continuously applied in any direction.
Cleanliness:	To Convair LOX clean Specification 0-75192-2 or equivalent.

Environment of System Package and Associated Hardware Inside Tank

Media:	Saturated LO_2 and GO_2, separate or mixed, or superheated GO_2. (Operating and Non-Operating). GN2 and GO_2 functional checkout
Pressure:	43 to 47 psia, operating to exact performance requirements.
	15 to 50 psia, off design operation, non-operating, and functional checkout
Temperature:	160 to 200°R operating to exact performance requirements.
	160 - 440°R, off design operation and non-operating.
	160 - 560°R, for checkout long enough to determine that electrical and mechanical operation is satisfactory.

The above environmental conditions are also considered to exist at the inlet to the pump, filter and throttling regulator.

Environment of Components Outside Tank

Media:	Air and space vacuum
Pressure:	0 to 15 psia
Temperature:	70 ± 50°F

Operating characteristics peculiar to the individual components are presented in the following sections.

3.2 PUMP

The basic operation of the pump is to provide hot side heat transfer in the exchanger and to mix the tank fluid to destroy temperature stratification within the normal vent down time.

Rating:	6.0 cfm at 3.0 ft minimum static head rise with LO_2 at 67 lb/ft^3.

Operating RPM:	1600 to 1700 (max. no load rpm = 1800 at synchronous speed)
Power Input:	60 Hz, 3 phase, 240 volts, 60 to 80 watts with LO_2 at 67 lb/ft^3.
Motor Design:	Motor stator and lead wires fully enclosed ("canned") in stainless steel. Illustrative schematic presented in Figure 3-1.
Fail Safe Electrical Design:	Instantaneous surge on starting estimated at 4.0 (max.) times running current. Electrical fusing will be provided for currents above this to deactuate the unit in case of failure.
Instrumentation:	The unit design will include a rotor speed sensor.

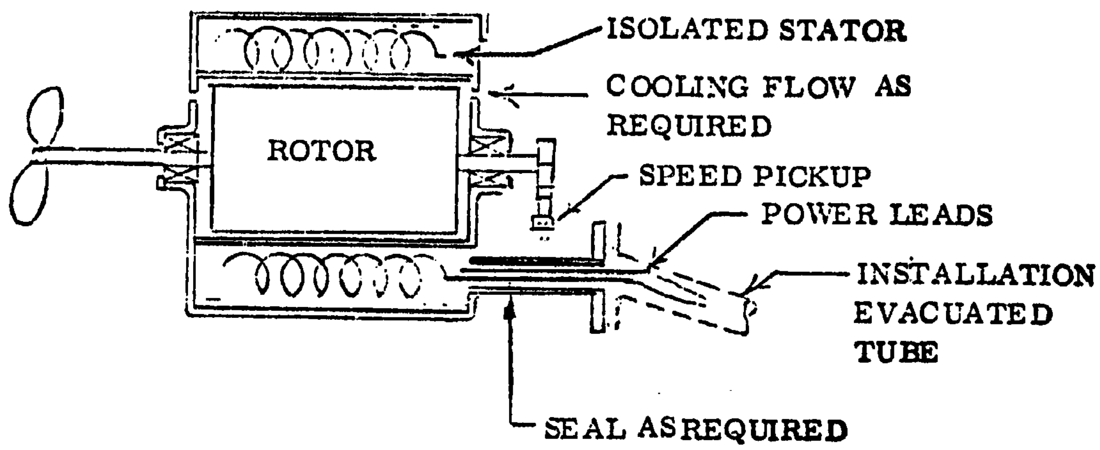


Figure 3-1. Pump Motor Schematic

3.3 HEAT EXCHANGER

The LOX vent exchanger is designed to vaporize and superheat any LO_2 which may be present at the vent inlet.

Performance:

Hot Side

Inlet Media:	Saturated LO_2 and GO_2, separate or mixed, or superheated GO_2.
Flow:	6.0 cfm of LO_2 at 67 lb/ft^3.
Pressure Loss:	1.0 ft (maximum) of LO_2 at 67 lb/ft^3.

Inlet Pressure:	44 to 48 psia
Inlet Temperature:	183.3 to 185.3°R for saturated O_2, higher for superheated O_2

Cold Side

Inlet Media:	Saturated LO_2 or GO_2 or both, or superheated GO_2. Design Point - Saturated LO_2
Flow:	Design Point - 37.5 lb/hr
Pressure Loss:	0.5 psi with GO_2 at 185°R
Inlet Pressure:	Design Point - 22 ± 1 psia
Inlet Temperature:	170.3°R Max for design
Outlet Temperature:	Design Point - 181°R (minimum)
Outlet Media:	Design Point - GO_2 (superheated)
Checkout	The unit will be capable of flowing GN_2 or GO_2 at 560°R through either side for checkout purposes

Structural:

Max. Operating Differential Pressure:	Hot side pressure 2 psi greater than ambient. Hot side pressure 50 psi greater than cold side. Ambient pressure 50 psi greater than cold side.
Checkout Differential Pressure:	Hot side 5 psi greater than ambient or cold side. Cold side 5 psi greater than ambient or hot side.
Weight:	9 lb (max)

3.4 THROTTLING REGULATOR

This unit provides an isenthalpic expansion of LO_2 and/or GO_2 between a variable inlet pressure and a downstream pressure controlled by the unit. This pressure expansion provides a temperature difference allowing the heat exchanger to vaporize any liquid which may be present in the vent.

Inlet:	Saturated LO_2 and GO_2, separate or mixed, or superheated GO_2, filtered to 10 micron particle size.

Flow: Maximum 40 lb/hr saturated LO_2 or GO_2.
Minimum 35 lb/hr saturated LO_2 or GO_2.

Outlet Pressure: 22 ± 1.0 psia design operating.

Internal Leakage: 0.02 lb/hr allowable with 47 psia LO_2 at the inlet and 30 psia at the outlet.

Differential Pressures: 0.5 psi crush load on upstream body, operating.
0 psi, non-operating.
30 psi, crush load on downstream body, operating.
2 psi, burst load on downstream body, non-operating.
50 psi, maximum design load on evacuated bellows.

3.5 FILTER

This unit is employed to prevent contamination of the throttling regulator and downstream flow hardware.

Rating: 10 micron-nominal.

Pressure Drop: 0.5 psi maximum while flowing 40 lbs/hr of saturated GO_2 at 43 psia.

Maintenance: Filter element can be easily replaced for any required periodic maintenance.

3.6 PRESSURE SWITCH

This unit senses the pressure of an LO_2 tank and causes electrical actuation of a pump and opening of a shutoff valve at an upper pressure limit and causes pump deactuation and shutoff valve closure at a lower pressure limit. Mounting is external to the LO_2 tank.

Actuating Media: GO_2 (operational), GN_2 (checkout)

Actuation Pressure: 47.0 psia (maximum)

Deactuation Pressure: 43.0 psia (minimum)

Deadband: 1.5 psi (minimum)

Internal Temperature: 70 ± 50°F

Electrical:

Circuit 1:	Triple Pole Single Throw - Operates up to 100 watt electric pump for durations of 15 seconds to 4 hours. Pump operates on 240 volt line to ground, 60 Hertz, 3 phase power. Contacts close at actuation pressure.
Circuit 2:	Single Pole Double Throw - Operates up to 60 watt solenoid in either position for durations of 5 seconds. The solenoid can operate on either 28 VDC or 120 VAC at 60 or 400 Hertz.
Isolation:	All electricity carrying components of the unit are isolated from the actuating media.

Structural:

Internal Pressure:	15 to 50 psia
Connection:	Pressure sensing port per MS 33656-4.
Leakage:	No external leakage even after the switch has undergone a single internal failure.
Failure Criteria:	First failure causes the switch to deactuate.

3.7 SHUTOFF VALVE

In the final configuration shown in Figure 1-1 the vent system shutoff valve is located external to the propellant tank and downstream of the heat exchanger and has no design requirements which are uniquely required to demonstrate satisfactory performance of the basic LOX vent system. Therefore, a facility type shutoff valve will be used during testing and procurement of a special valve was not required at this time.

The external environment and basic flow rate requirements are per Section 3.1. Internal fluids are GO_2 and GN_2 at temperature from 180 to 560°R. Maximum pressure drop is 1 psi of 560°R GO_2 at 40 lb/hr flow. Electrical operation will conform to the limitations of circuit #2 of the pressure switch (Section 3.6).

4.0

NEW TECHNOLOGY

In compliance with the New Technology clause of this contract, personnel assigned to work on the program have been advised, and periodically reminded, of their responsibilities in the prompt reporting of items of New Technology. In addition, response is made to all inquiries by the company-appointed New Technology Representative and copies of reports generated as a result of the contract work are submitted to him for review as a further means of identifying items to be reported. When deemed appropriate, conferences are held with the New Technology Representative to discuss new developments arising out of current work that may lead to New Technology items. The New Technology Representative will be responsible for transmitting New Technology to the Technology Utilization Officer. Contract plans to continue New Technology monitoring and surveillance as described above in the ensuing period to assure all items of New Technology are reported as they develop.

5.0

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