REPORT NO. CASD-NAS 75-021

(NASA-CR-120769) SPACE LOX VENT SYSTEM Final Report (General Dynamics/Convair) 133 p HC $5.75 CSCL 21H

N75-23685

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SPACE LOX VENT SYSTEM

FINAL REPORT

GENERAL DYNAMICS
Convair Division

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SPACE LOX VENT SYSTEM

FINAL REPORT

30 April 1975

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FOREWORD

This report was prepared by Convair Division of General Dynamics under Contract NAS8-26972, "Space LOX Vent System" for the George C. Marshall Space Flight Center of the National Aeronautics and Space Administration. The work was administered under the technical direction of the Astronautics Laboratory, George C. Marshall Space Flight Center with Mr. R. Stonemetz and Mr. G. Young acting as project managers. The current project manager is Mr. G. Young.

The project leaders at Convair have been Mr. J. A. Stark and Mr. R. C. Erickson. The current project leader is Mr. R. C. Erickson. In addition, the following Convair personnel contributed to the program: Messrs. M. D. Walter, J. R. Elliott and H. G. Brittain.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

LIST OF FIGURES

LIST OF FIGURES (Continued)

LIST OF FIGURES (Continued)

LIST OF TABLES

SUMMARY

The basic objective of this program performed under Contract NAS8-26972 has been to design, build and test a low gravity prototype vent system capable of exhausting only vapor to space from an all liquid or two-phase mixture of oxygen. This objective has been met with a compact heat exchanger thermodynamic vent system. This report documents the work completed to select and design the vent system and to demonstrate performance of the vent system to control pressure in a liquid oxygen tank. The primary design requirement is for the system to control pressure in a 2.74 m (9 foot) oxygen tank to 310 ± 13.8 kN/m^2 (45 ± 2 psia) when the external heating rate is 32.2 - 35.2 watts (110-120 Btu/hr).

During the preliminary analysis and concept selection task a literature survey was conducted to determine those system concepts capable of meeting the basic LOX vent requirements. The literature search was accomplished primarily for the period from 1965 up to the present. Work prior to 1966 was thoroughly covered under the first phase of the "Study of Zero-Gravity, Vapor/Liquid Separation," Contract NAS8-20146 performed for NASA/MSFC by Convair. This previous data was, however, reviewed for specific application to the present LOX vent program. The major difference between previous programs, which were oriented primarily to LH_2, and the present program is that the critical nature of the liquid oxygen environment necessitates a re-evaluation of hardware and system requirements for use with oxygen.

In the literature survey, the systems reviewed include devices using centrifugal, liquid surface and electric forces, and devices relying on heat transfer. In each of the concepts presented above, the use of local liquid vapor separation versus total fluid control was considered.

With regard to local separation, the most significant requirement is that the device be capable of operation with a 100% liquid inlet. The only concept found which can efficiently operate under such a condition was the thermodynamic heat exchanger vent system. With regard to total fluid control, vortexing of the entire tank fluid was discarded because of the potential interference with the overall vehicle control and personnel operations and the long time intervals potentially required for start-up and shutdown. In the case of electrical systems, there is still some question of LO_2 compatibility where electrical discharges may occur and development of the required high voltage feed-throughs and orientation electrodes has not been sufficient to demonstrate a high level of confidence.

Results of the literature survey and screening analysis was the selection of the thermodynamic vent system concept as the most viable for further study and design under the present program. A thermodynamic vent system ingests the fluid

either liquid or vapor, to be vented from a tank and throttles it at constant enthalpy to a lower pressure and temperature before passing it into a heat exchanger. In the heat exchanger, either a free convection distributed or a compact forced convection heat exchanger, energy is absorbed directly or indirectly from the bulk fluid in sufficient quantity to vaporize any liquid, if present, and superheat the vapor before it is discharged overboard.

Internally suspended, interior wall and exterior wall types of natural convection-distributed heat exchangers vent systems were considered before an exterior wall mounted type was selected. Analysis of the functional requirements of the forced convection compact heat exchanger system included sizing bulk propellant mixing, and motor/pump sizing. A comprehensive comparison between the compact heat exchanger vent system and the wall heat exchanger vent system was completed. The compact heat exchanger has a weight penalty of 28.1 kg (62 lb) for a 30-day mission. This weight can be predetermined and the concept is sufficiently developed to allow reasonable assurance that it can be further developed to meet the requirements of this program. On the other hand, the relative state of development of the wall heat exchanger concept is not sufficient to warrant its selection for this program. At this point in time reliable tools for predicting thermodynamic performance of wall heat exchangers operating on large tanks, over the full range of possible space conditions, are not available nor can any particular configuration be identified as the best design to pursue for development. Thus, the compact heat exchanger concept was selected as the prototype LOX vent system which best meets the requirements of this program.

The detail design phase of the study included with the finalization of vent system performance, development of component specifications, solicitation of vendor bids, selection of components and overall system package design. The following component requirements were defined.

Heat Exchanger:	Hot side	0.00284 m^3/sec (6.0 cfm) @ 0.3048 m (1 ft) LO_2 pressure drop
	Cold Side	0.0047 kg/sec (37.5 lb/hr) @ 3.45 kN/cm^2 (0.5 psia) pressure drop
Pump Capacity:	.00284 m^3/sec @ .91 m head (6.0 cfm @ 3.0 ft)	
Pressure Switch Dead Band:	to 10.3 kN/m^2 (1.5 psi) minimum	
Vent Flow Rate:	.0047 kg/sec (37.5 lb/hr) nominal	
Throttling Pressure:	152 kN/m^2 (22 psia) nominal	

The above requirements were given to hardware vendors. Component selection was based on bidder technical ability, minimum costs and the ability to deliver on schedule. The following items were procured and assembled into a complete test package.

a. A throttling regulator of aluminum construction with an evacuated bellows sensing downstream pressure (HTL Industries P/N 187250-4). W[illegible] the unit 0.22 kg (0.62 lb).

b. The filter is of stainless steel construction with a 10 micron-nominal rating (Western Filter Co. In. P/N 70-16510-10) Weight of the unit is .39 kg (0.87 lb)

c. The heat exchanger is of all aluminum construction and weights 2.95 kg (6.5 lb) (Geoscience P/N 02B2). The cold or vent side flow is through a single coil of .95 cm (3/8 inch) tubing, and the hot side flow is vortexed over the outside of this tubing. This design allows for highly efficient heat transfer of a boiling fluid and minimizes the possibility of liquid "carry-over."

d. The pump/motor is basically of stainless-steel construction and weighs 5kg (11 lbs) (Sunstr[illegible]d) P/N EP145603-100, SNX-1). The unit was modified from an existing pump/motor. The motor was modified from 60 Hz 240V unit to 60 Hz 75 v unit with an operating speed of approximately 1800 rpm with 60 Hz three phase and 75 v input. The speed and flow of the unit can be reduced to approximately one-sixth of design by proportionately reducing the frequency and voltage.

e. The pressure switch is of stainless steel construction and weighs 0.34 kg (0.75 lb) (Hydra-Electric P/N 83159). It is located external to the propellant tank in the vacuum chamber environment.

The entire test package, including instrumentation bosses and mounting bracketry weighs 11.36 kg (25 lb) (Figure 3-7). In this system, hea[illegible] wall s[illegible]ainless tubing and instrumentation bosses were used in order to be compatible with existing CRES temperature probe fixtures.

The system shown in Figure 3-7 was tested with oxygen in an 2.2 m by 1.88 m (87 inch) by 74 inch) oblate spheroid tank. This tank was superinsulated with 22 layers of goldized kapton Superfloc insulation and installed in the vacuum chamber at the Convair Sycamore Canyon Cryogenic Test Site. The test package was located approximately .5 m (20 inches) from the bottom of the test tank.

Prior to system evaluation testing, each component was individually tested to demonstrate component compatibility and operation in LO_2. Upon completion of the component testing, the system was assembled and its ability to operate in LO_2 was demonstrated.

The 240 hour test program to evaluate the LO_2 vent system was then completed. A total of 94 pressure cycles at 57 different test conditions were completed to define the performance envelope of the vent package. The test parameters included four liquid levels 0.18 m to 1.5m (7 inch to 60 inch), six total heat fluxes 67 watts to 439 watts (230 Btu/hr to 1500 Btu/hr), four pump speeds (230 RPM to 1800 RPM) four vent rates 1.1 gm/s to 9.95 gm/s (9 lb/hr to 75 lb/hr) and two pump discharge flow directions (nozzle vertical, toward the liquid/vapor interface and the nozzle horizontal, toward the tank wall).

The vent system was able to control test tank pressure for all conditions except at minimum pump speed, with the vent inlet and heat exchanger in liquid and the pump discharge nozzle in the horizontal position. The vent system vented vapor only for all test conditions except at minimum pump speed where it appeared that the system pulsed some liquid. The minimum pump speed represents a condition of 1/6 design pump speed, 280 RPM.

All work completed under this contract was accomplished using customary english units. As required by contract all units are reported with the International System of Units as the preferred primary system of units except for recorded test data which are reported in customary english units for which the instrumentation was calibrated.

SECTION 1

INTRODUCTION

This is the final report summarizing the work completed under contract NAS8-26972 "Space LOX Vent System." Work covered includes concept selection, design, fabricating and testing of a prototype compact heat exchanger thermodynamic vent system. The system is designed to operate in a 2.7m (9 foot) spherical liquid oxygen tank with a heating rate of 32.2 - 35.2 watts (110-120 Btu/hr) and to control pressure to 310 ± 13.8 kN/m^2 (45 ± 2.0 psia). The design mission is of 2,590 ks (30 days) duration on board a space shuttle orbiter.

Details of the system definition task to determine system requirements and specifications are contained in Section 2. Four primary separation concepts were considered:

a. Heat Exchange or Thermodynamic Vent - where the vent fluid is throttled to a low pressure and temperature and allowed to exchange heat with the tank fluid in order to vaporize any liquid initially present in the vent stream.

b. Mechanical Separation - employing a rotating element imparting centrifugal forces to the fluid to separate the gas from the liquid.

c. Dielectrophoresis - utilizing the forces caused by non-uniform electric fields acting upon dielectric fluids, such as hydrogen and oxygen. Both total liquid control and separator devices were considered.

d. Surface Tension - utilizing fluid surface forces to orient the liquid in a tank, employing baffles or screens, or to effect a separation in a vent separator device.

Detail design definition of the selected heat exchanger vent system is presented in Section 3. The various trade-offs which were made include:

a. Bulk heat exchanger versus wall type

b. Pump turbine drive versus electric motor

c. Determination of optimum vent flow rates and vent cycle

d. Determination of tank mixing requirements

Procurement of system components and system are discussed in Section 4. The overall design of the test facilities and system evaluation testing are presented in Section 5. Detail discussion of test results for the total vent system as well as individual components is given in Section 6.

Overall study conclusion and recommendations are presented in Section 7.

SECTION 2

VENT SYSTEM SCREENING AND SELECTION

The objective of this program was to design, build and test a low-gravity prototype vent system capable of exhausting only vapor to space from an all liquid or two-phase mixture of oxygen. Design criteria and ground rules are based on the vent system finding application in the LO_2 tank of the Space Shuttle Orbital Maneuvering System (OMS). A literature survey was completed to define possible vent systen concepts that would meet requirements and a comparative screening analysis was done to select the most promising system. The work in this section was performed under the Convair Aerospace 1971 Independent Research and Develcpment (IRAD) program, Reference 2-1, and is reported herein only for reference as it relates to the pertinent subject of liquid oxygen tank venting.

2.1 DESIGN REQUIREMENTS AND GROUND RULES

Space Shuttle operating data were received from the NASA/MSFC project manager (reference 2-8). These data were reviewed and compared with the data presented in References 2-1 through 2-7.

Based on these data, the LOX vent program was conducted under the following ground rules:

1. The basic vent system specifications are presented in Table 2-1. It is noted that data contained in the basic contract scope of work are also included for completeness. Tank material, shape and acceleration levels were taken from References 2-4, 2-6 and 2-7. The maximum acceleration of 10^{-4} g's comes from assuming that the shuttle is actually docked to the space station during a significant portion of the coast period.

2. The gross propellant use schedule, based primarily on data from Reference 2-6, is presented in Table 2-2. It is noted that the maximum time between the TPI maneuver and deoribt was increased from that given in Reference 2-6 to 2,578 Ks (716 hrs) to allow for the possiblity of a 30 day mission. Otherwise the data reflects overall minimum and maximum condition for the Orbit Maneuvering System (OMS), as presented in Reference 2-6. It is assumed that for the present vent application, long coast periods, such as in orbit without propellant usage, are the most critical and the only need for a usage schedule is to provide an approximate measure of these times and corresponding ullage volumes. For calculating LO_2 tank volumes, an initial ullage of 3% and a propellant mixture ratio of 5:1 are used. LO_2 density is taken to be 1,135 kg/m^3 (70.8 lb/ft^3) (Reference 2-6).

Table 2-1. Basic Vent System Specifications

Design Element	Specification
Propellant	Liquid Oxygen
Overall Tank Pressure Range	103.5 - 345 kN/m^2 a (15-50 psia)
Tank Pressure Control Range	310 ± 13.8 kN/m^2 a (45 ± 2 psia)
Operational Tank Fluid Temperature	89° to 130°K (160° to 185°R)
Overall Temperature Range	89° to 244°K (160° to 440°R)
Separator Inlet Quality	0 to 100%
Vapor Vent Flow Rate Range	.63 to 6.3 gm/s (5 to 50 lb/hr)
Total Operational Steady-State Heat Leak	32.2 to 35.2 watts (110 to 120 Btu/hr)
Suction Line to Tank Outlet Heat Leak	5.9 to 8.8 watts (20 to 30 Btu/hr)
Mission Duration	605 to 2,590 ks (7 - 30 days)
Minimum Life Time	100 Missions
Vibration Levels	20 g's nonoperational 5 g's operational
Component Failure Critiera	1 - operational 2 - safe
Tank Material	Aluminum
Tank Shape	Spherical
Coast Acceleration Levels	10^{-4} - 0 g's

Table 2-2. Typical OMS Propellant Use Schedule

Event	Mission Elapsed Time, ks (hr min. /max	Time From Last Burn, ks (hr) min. /max.	Total Propellant Used, kg (lb) min. /max
Phasing	2.88/2.88 (0.8/0.8)		988/3220 (2200/7100)
Height	5.76/79.92 (1.6/22.2)	2.88/77.04 (0.8/21/4)	2087/2560 (4600/5640)
Coelliptic	8.64/82.82 (2.4/23.0)	2.88/2.88 (0.8/0.8)	191/2130 (420/4690)
TPI	14.04/88.56 (3.9/24.6)	5.40/5.76 (1.5/1.6)	161/192.5 (355/424)
Deorbit	82.44/2,592 (22.9/720)	68.4/2,578 (19.0/716)	3620/4330 (7980/9550)
Contingency	-		1630/1975 (3590/4350)
Total	82.44/2,592 (22.9/720)		8700/14,1400 (19,145/31,754)

3. In order to evaluate the possible effects on vent system operation of having combined propellant storage for the OMS and Attitude Control Propulsion System (ACPS), it is desired to also define a typical ACPS operating cycle. Important factors are frequency and duration of operation, propellant used, and disturbing accelerations. Potential accelerations caused by the ACPS operation are important in that for systems utilizing accumulators it is feasible that venting may occur while these engines are firing. A survey of the data contained in References 2-6, 2-8 and 2-9 indicates a significant range of potential operating conditions. Based on a representative compilation of this data the following range of conditions was investigated for their effect on vent system performance.

   a. Firing duration of 6 to 500 sec.

   b. Time between firings of 10 to 7200 sec.

   c. Liquid oxygen used during a firing of 19 to 158.8 kg (42 to 350 lb).

   d. Disturbing acceleration during a firing of 0.141 g's.

   It is assumed that the above ACPS operations occur between the OMS firings presented in Table 2-2.

4. A completely autogeneous pressurization system was assumed for the basic analysis. Use of oxygen pressurant simplifies the vent system design in that heat transfer coefficients can be better predicted and an unknown loss of helium pressurant through the vent is eliminated. This reduces the need for a vent system which operates only with a liquid inlet.

2.2 LITERATURE SURVEY

This section describes the results of a literature survey to define all systems which may be applicable to the LOX vent requirements presented in Section 2.1. The literature search was accomplished primarily for the period from 1965 up to the present.

It is noted that work prior to 1966 was thoroughly covered under the first phase of the "Study of Zero-Gravity, Vapor/Liquid Separators," Contract NAS8-20146 performed for NASA/MSFC by Convair Aerospace (Reference 2-10).

Under that program various ways were studied of separating vapor from liquid in a low-acceleration field in order to permit venting of vapor. Four primary methods of separation were considered:

a. Heat Exchange or Thermodynamic Vent - where the vent fluid is throttled to a low pressure and temperature and allowed to exchange heat with the tank fluid in order to vaporize any liquid initially present in the vent stream.

b. Mechanical Separation - employing a rotating element imparting centrifugal forces to the fluid to separate the gas from the liquid.

c. Dielectrophoresis - utilizing the forces caused by non-uniform electric fields acting upon dielectric fluids, such as hydrogen and oxygen. Both total liquid control and separator devices were considered.

d. Surface Tension - utilizing fluid surface forces to orient the liquid in a tank, employing baffles or screens, or to effect a separation in a vent separator device.

Other separation methods including fluid rotation, a "hydrogen sublimator," and magnetic positioning were considered, but were not found to be of sufficient value to be included in the detailed predesign comparisons with the four methods listed above.

It was found that the dielectrophoretic and surface tension devices were consistently poorer from safety/reliability and efficiency considerations than either the mechanical or heat exchange separator systems. The latter two systems were competitive with each other on many of the criteria, but the heat exchange system was judged to be the most promising one for the three vehicle/mission cases considered in this study. A major consideration was the inability of systems other than the thermodynamic one to operate efficiently with a 100% liquid inlet. More detailed studies of the heat exchange type of system were then made paralleling a phase II program under the NAS8-20146 contract to define, design and test such a system for a hydrogen tank. Testing was satisfactorily completed on the LH_2 system and the data are presented in Reference 2-11. The major difference between the previous program and the LOX vent program is that the critical nature of the liquid oxygen environment necessitates a re-evaluation of hardware and system requirements for use with oxygen. As an example, the use of electric wiring in the oxygen should be eliminated and more emphasis needs to be placed on passive systems than was done for the hydrogen application.

The basic objective of the current literature search was to update the information contained in Reference 2-10 with specific application to the LOX vent program and to determine if any new system concepts or data had been generated which may change the previous conclusions, or make systems other than the thermodynamic systems more competitive for the LOX vent application . The main emphasis in the search was thus on basic vent concepts and their potential feasibility with respect to LOX operation, rather than detailed phenomenon such as surface tension and droplets separation theory. Such theory was only considered as it effects the determination of LOX vent feasibility.

The STAR, CSTAR, and IAA indexes were reviewed and a Defense Documentation Center search was conducted under the subject headings of "Liquid Vapor Separation" and "Cryogenic Propellant Venting." Where the document titles indicated application to the current LOX vent program, available abstracts were reviewed and pertinent documents obtained. New documents reviewed for specific application to the present program are listed in Appendix A along with a description of the general content or scope of the document, key technology advancements presented, and how the data contained may be applicable to the present LOX vent program. Where specifically cited in the following discussions, these documents are also listed in the Reference section by author and title.

Pertinent information was found for separation devices using centrifugal, liquid surface and electric forces and devices relying on heat transfer. In each case the use of local liquid vapor separation versus total fluid control was considered. Key points uncovered by the literature survey were:

1. The work done to date on dielectrophoretic systems has not been sufficient to demonstrate a high level of confidence as to their operating safety in oxygen. Development of reliable high voltage feed-throughs is still a problem and the potential arcing of electrodes is forever present. Therefore such systems were not considered further.

2. A variation of the basic thermodynamic heat exchanger vent system was noted in the literature which has not been considered in previous contracted studies. This system is described in Reference 2-12 and illustrated in Figure 2-1.

   It is similar to the thermodynamic vent system studied under Contract NAS8-20146 except that the mechanical pump is replaced by a high voltage DC field. The field is applied between the heat exchanger coils and the tank wall in order to cause tank fluid agitation. For a 500,000 volt field it was estimated that heat transfer coefficients at low-g (approximately 10^{-6} g's) car be increased to what they would be for natural convection at one-g. The Reference 2-12 data indicated that this system would even be applicable to fluorine tank venting and is better than either a wall exchanger or a bulk unit with a mechanical mixer. However, additional information obtained from MDAC (Reference 2-13) indicated that no further work had been done on this system since 1969. The major problem is with high voltage feed-throughs, in that the heat exchanger at high potential must be

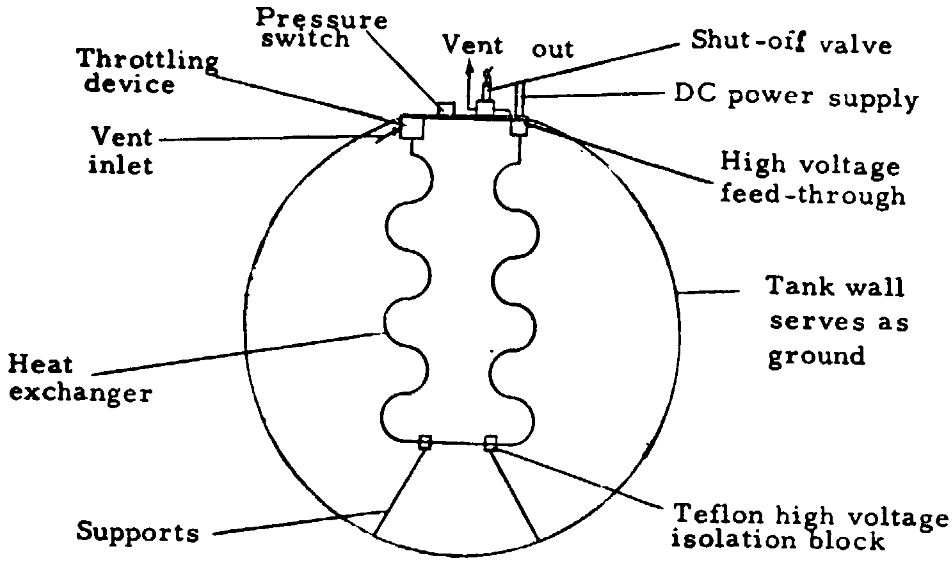


Figure 2-1. Electro Convection Thermodynamic Vent System

electrically insulated from the tank at low potential. The problems with this system are thus essentially the same as for the dielectrophoretic liquid separation systems, and, was thus not considered further.

3. The basic thermodynamic concept can be made to be compatible with the oxygen environment by isolating the pump motor windings from direct contact with the oxygen by use of a sealed can between the wiring and the rotor. Also, the possibility exists of using the vent gas to drive a turbine to power the pump, and the potential of using a wall heat exchanger without any pump exists. The heat exchanger vent concept is thus considered further under the final systems analysis and concept selection task described in Section 2-3.

4. With regard to local separation for systems other than the heat exchanger, the most significant operating problem is the requirement that the device vent vapor when the inlet is 100% liquid. It is noted that it may be feasible to operate such systems with a 100% liquid inlet if throttling were accomplished at the inlet to create some vapor for separation. However, in this case the separated liquid would need to be pumped back to the tank pressure and inlet qualities obtained in this manner would be very low. Based on data developed in Reference 2-10, it is concluded that local surface tension devices operating in this mode would not be feasible and were thus not considered further in the study. Use of fluid vortexing such as with a vortex tube or a mechanical separator has been proposed by some (Ref. 2-23) and is considered further in Section 2.3.

5. Total fluid separation with a surface tension screen system, such as described in Reference 2-9, does appear feasible and is analyzed and compared with other systems in Section 2.3.

6. Complete tank fluid rotation was discarded from further consideration due to potential interference with the overall vehicle control and personnel operations and the long time intervals potentially required for start up and shutdown.

2.3 SYSTEMS ANALYSIS AND CONCEPT SELECTION

This section describes and analyzes the candidate low-gravity LOX tank vent systems which showed sufficient merit to pass the preliminary screening reported in Section 2.2. Two basic concepts are considered. One is the use of a constant enthalpy throttling valve, which regardless of inlet fluid quality insures that the inlet of any downstream device will receive some vapor. The other is to control the entire bulk of liquid within the tank at all times such that only vapor is allowed to contact the vent system inlet. Systems which utilize an upstream throttling device, include vortex, cyclone or mechanically driven liquid-vapor separators and the heat exchanger which vaporizes inlet fluid due to temperature difference created by the throttling process. The only system found which utilizes the total liquid control concept employes surface tension devices.

2.3.1 THROTTLING CONCEPT. The constant enthalpy throttling process for typical vent system operating conditions is described with the aid of the temperature-enthropy diagram in Figure 2-2. The state points describing four possible throttling valve inlet conditions are shown as 1a through 1d. The final state points after expanding on constant enthalpy lines to a lower pressure are given as 2a through 2d. 1a to 2a describes the throttling process for superheated vapor, 1b to 2 b for a two phase inlet condition ending in a vapor quality of 50%, 1c to 2c is for a saturated liquid inlet ending in a mixture with 8% vapor quality, and 1d to 2d is for a subcooled liquid inlet (30 psia saturation pressure) ending in a mixture with 4% vapor quality. A device placed downstream of the throttling valve would then be required to operate with inlet conditions ranging from state points 2a through 2d.

2.3.2 MECHANICAL AND VORTEX SEPARATOR SYSTEMS. Figure 2-3 is a schematic of a typical system utilizing a mechanical or vortex tube type separator. It is assumed that in this system the separator is supplied with a fluid ranging in vapor quality from approximately 4% to 100%. The vapor is then separated from the liquid and vented overboard. The liquid must be pumped back up to a sufficient pressure head for return to the tank. Power to drive the pump and/or separator may be supplied from a turbine driven by vent gas, or from an electric motor.

The effectiveness of this system depends on the separating efficiency of the separator and the power required to drive the pump and separator.

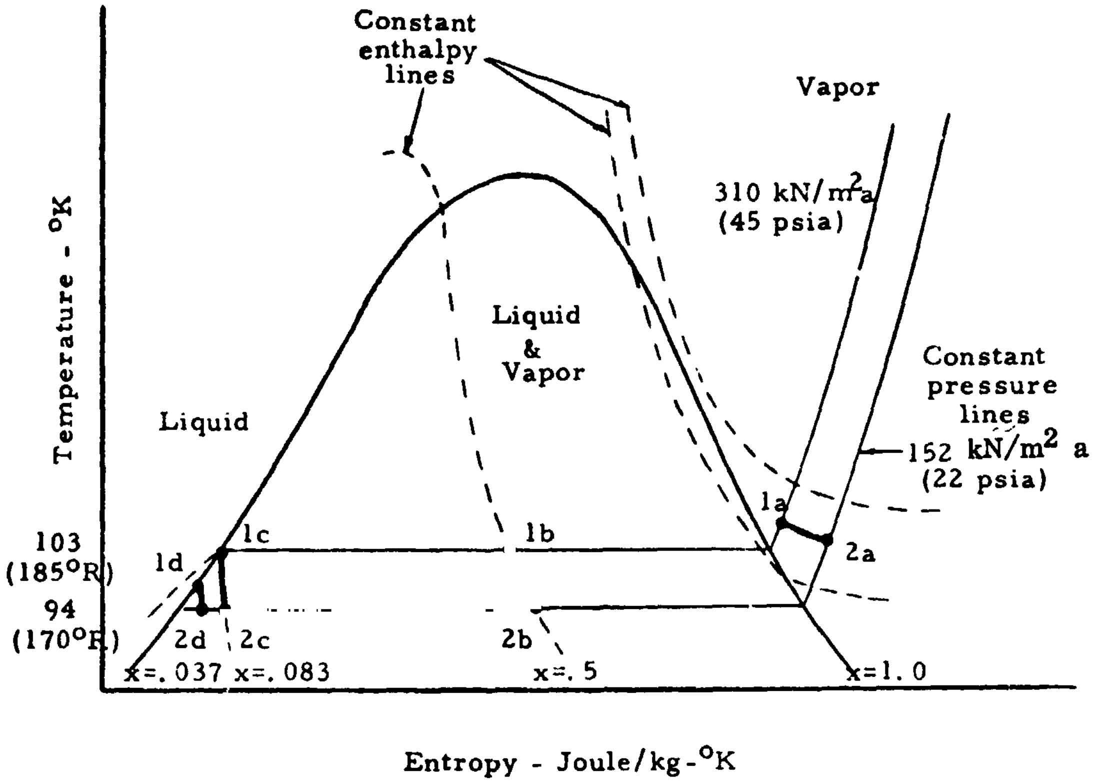


Figure 2-2. T-S Diagram for Oxygen

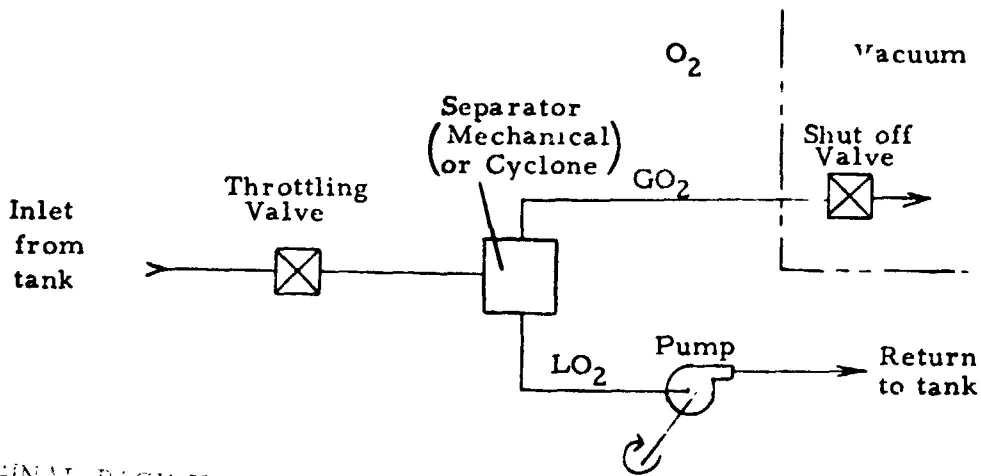


Figure 2-3. Separator Systems - General Schematic

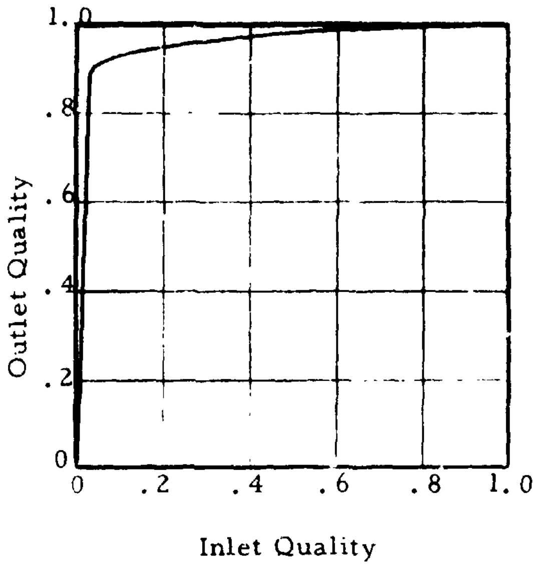


Figure 2-4. Mechanical Separator Performance

Figure 2-4, taken from Reference 2-14 as a typical performance curve for mechanically drive separators, shows that for the range of inlet conditions (i.e., vapor quality 1 to .04) 0 to 10% of the liquid flowing through the system can be vented overboard. For extended duration missions, such as anticipated for space shuttle, a considerable weight penalty in excess propellants would thus need to be carried along for venting. The mechanically driven separator is thus considered undesirable for the low-g LOX tank venting because inlet conditions cannot be controlled and if low vapor quality mixtures enter this system, significant quantities of liquid would be vented.

The vortex tube (using centrifical forces with separated flowstreams exiting parallel) and the cyclone type separator (a conical shaped devise using centrifical forces and secondary flow forces on the cone face to split the separated flow in opposite directions) were also considered for this basic system. Using an analysis developed in Reference 2-15 it was shown that the vortex tube would be acceptable purely from a pressure drop standpoint for use in the LOX vent system. Other considerations such as separation efficiency, however were not dealt with in this reference. Reference 2-16 on cyclonic two-fluid separation presented a more detailed analysis of the separation process itself. However, from the same reference, a test program using air and water spray indicates that even under favorable 1-g conditions, liquid carry-over in the gas will occur. The actual amount of carry-over and governing phenomena were not examined and thus no qualitative assessment of wasted propellant could be made for the present case.

The qualitative fact that liquid carry-over will occur, in a favorable 1-g environment under which these tests were performed, would indicate that this type of device would be unusable for the present LOX vent application. It is also unlikely that such a system would have better performance characteristics than those described by the Figure 2-4 data.

2.3.3 THERMODYNAMIC VENT SYSTEM. The thermodynamic vent concept is shown schematically in Figure 2-5 and operates in the following manner. The vent line between the tank and space contains a throttling regulator through which tank fluid is expanded to a lower pressure and temperature. The temperature difference between tank fluid and the throttled fluid is used to raise the quality of the throttled or vent fluid and produce superheated vapor at the outlet of a heat exchanger. This system can be designed to vent vapor regardless of inlet conditions.

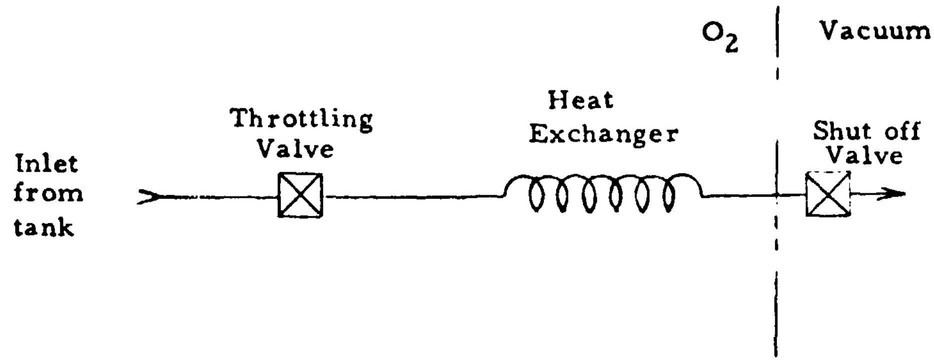


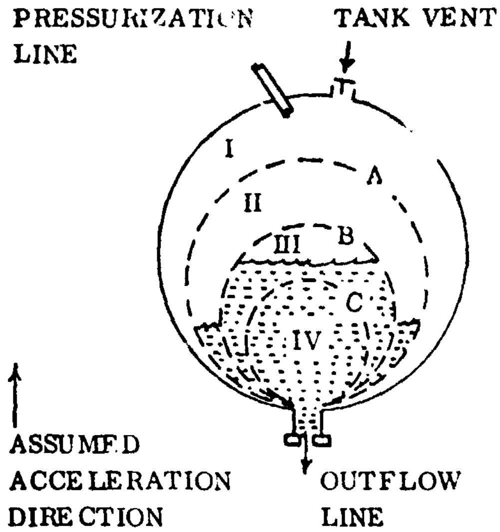
Figure 2-5. Thermodynamic System - General Schematic

The design is quite flexible and, other than knowledge of approximate boil off rate, is in no way vehicle or mission dependent. For example, in the case of the heat exchanger, a compact coiled tube or a wall-mounted tube configuration may be employed. The heat exchanger can also be used as the conditioning system for feedlines and acquisition devices. An electric or turbine driven mixer can be used to reduce heat exchanger length and promote temperature destratification. The vent operation can be continuous or intermittent. Intermittent venting can be actuated electrically or pneumatically.

Weight of a system designed to this concept for the space shuttle orbiter LO_2 OMS tank is estimated to be 7 to 18 kg (15 to 40 lbs) depending on which type of heat exchanger is used and whether or not a mixer is employed. This would include all hardware associated with the vent system plus incremental vehicle electrical fuel cell system weight in the case where an electric mixer is used. In addition to the relatively light hardware weight, a reduction in the mass of propellant vented is achieved due to the systems ability to vent superheated vapor.

2.3.4 <u>BULK CONTROL VENTING.</u> The only bulk control system considered feasible for the present application is a surface tension total management system designed by Martin Marietta Corporation, Reference 2-9. This device is schematically represented in Figure 2-6. It is designed to comply essentially with the ground rules presented in Section 2.1 and is fully described in Reference 2-9.

The intent of this system is to maintain the outer annulus (region I) as a liquid free region for gaseous venting and the inner sphere (Region IV) as a gas free volume for liquid expulsion. Screen "A" is sized to contain the initial liquid volume, therefore the volume of region I is equal to the anticipated initial ullage volume. Following boost and prepressurization most of the liquid in the tank will be positioned within screen "A" as follows: with screen "A" fully wetted, vaporization and/or injection of a pressurant will cause the pressure in region I to rise forcing the liquid there into the lower pressure inner regions. During engine operation liquid is expelled from



	DIA.
TANK	3.05m (10 FT)
SCREEN A	2.75m (8 FT)
SCREEN B	2.13m (7 FT)
SCREEN C	1.22m (4 FT)

Figure 2-6. Surface Tension Total Management System

region IV while pressurant, if required, is added to region I. The liquid will flow from region II through III into IV. During continued outflow the liquid level in region II will drop while the level in region III remains constant until the delta head exceeds the retention capabilities of screen "B". At this point the level in region III begins to drop with the delta head between regions II and III remaining constant until region II is drained. The same sequence will take place with regions III and IV until all remaining liquid is retained in region IV. The above operating sequence assumes a constant low-g acceleration direction. With varying acceleration directions the sequence will change but the intended function of keeping region IV free of gas and region I free of liquid would theoretically be accomplished.

This system has one obvious problem in venting only vapor, the possibility of screen "A" losing its liquid retention capability which could result in the venting of liquid. There are several conceivable conditions (some discussed in Reference 2-9) where liquid would be retained or spilled into region I; such as normal acceleration levels during engine firing (until tank is haft empty), an abnormal acceleration level, excess residuals before docking, and screen drying due to pressurant gas inflow or abnormal local heat flux. These are but five of the possibilities. In the case of the shuttle OMS tank, the probability of venting any liquid contained in region I appears high because of the tanks forward location. With the tank in such a position, liquid will be thrown to the forward or vent end by all vehicle pitch and yaw maneuvers.

2.3.5 <u>CONCEPT COMPARISONS AND RECOMMENDATIONS.</u> Based on the discussions of the previous paragraphs only two systems were considered for the space LOX system application. They were the surface tension bulk control system (Paragraph 2.3.4) and the thermodynamic vent system (Paragraph 2.3.3).

Table 2-3 gives a general comparison between the two systems It is noted that except for weight the data is qualitative and represents only a relative rating between the two systems.

Based on the data presented, the concept recommended for the detail analysis and predesign phases of the study was the thermodynamic vent system. The choice is based on the concept's versatility and inherent independence from other vehicle systems and the fact that it is the only system which can adequately guarantee all vapor venting. That is, it has the posssibility of using one basic design for any variety of tank configurations and vehicle mission profiles.

2.4 HEAT EXCHANGE CONCEPT SELECTION

Functionally, a thermodynamic vent system ingests the [illegible] to be vented from a tank, either liquid or vapor, and throttles it at constant enth[illegible] a lower pressure and temperature before passing it into a heat exchanger. I[illegible] heat exchanger, energy

Table 2-3. Comparison of Low-g LOX Tank Vent Systems

Criterion	Surface Tension Bulk Control	Thermodynamics Heat Exchange
Hardware Weight	23 to 36 Kg* (51 to 78 lbs)	7 to 18 Kg■ (15 to 40 lbs)
Versatility	Low	ExcellentΔ
Complexity	Less	More
Basic Concept Successfully Tested	No	Yes
Possibility of Venting Liquid	High	Low
Vent Mass Reduction Capability	Moreø	Some
Confidence in Hardware Development	Smaller⊠	Higher
Safety	Same	Same+

* Excludes vent valves and/or other pressure control hardware.
■ Actual value depends on detail system, i.e., bulk or tank wall heat exchangers.
Δ Not dependent on tank shape or size, propellant load, acceleration environment, etc.
ø Reduction, however, gained at expense of increased hardware weight.
⊠ For large tanks only where fluid heads exceed retention capabilities of surface tension devices - for smaller tanks this would be rated "same."
\+ If electric motor is not immersed in O_2 environment.

is absorbed from the bulk fluid in sufficient quantity to vaporize any liquid, if present, and superheat the vapor before it is discharged overboard. Detailed analysis and tradeoffs were performed of options within the general concept of a thermodynamic vent system. The main objective was to make a selection between a vent system which utilized a free convection distributed heat exchanger and one which utilized a compact forced convection heat exchanger.

2.4.1 DISTRIBUTED HEAT EXCHANGER. Distributed heat exchanger configuration manufacturing and assembly and performance were determined to define the effectiveness of this design approach for the thermodynamic vent system.

2.4.1.1 Configuration Analysis. Three basic configurations for the distributed heat exchanger were examined. Each system included a filter, regulator and shut off valve. The three configurations considered are presented in Figure 2-7 as tubular heat exchangers a) internally suspended, b) mounted to the interior of the tank wall and c) tubes mounted to the exterior tank wall.

The internally suspended tubular heat exchanger has the advantage of distributing the heat transfer throughout the propellant bulk and thus reducing internal thermal gradients. However, in spite of this advantage the suspended tubular heat exchanger is not competitive with the wall mounted heat exchangers in either weight or performance. The suspended heat exchanger sizing and performance is extremely sensitive to the heat transfer coefficient between the tank fluid and heat exchanger surface, whereas the wall mounted exchangers are not. Reference 2-17 indicates that the film coefficient for O_2 will range from 0.06 to 170 watt/m^2•K (0.01 to 30 Btu/hr-ft^2-°R) for the various conditions expected inside the LO_2 tank. An order of magnitude decrease in film coefficient creates a requirement for an order of magnitude increase in tube surface area to vaporize or condition a fixed amount of vented fluid to the desired exit condition. Wall mounted exchangers use the tank wall as a heat transfer fin and, because they partially intercept the incident heat, are not as sensitive to internal film coefficient variation. Also by utilizing the tank wall as a fin, the effective heat exchanger area is significantly increased over that of a plain tube. Weight calculation for a 2.7m (9 foot) diameter spherical tank showed that the internally suspended exchanger would weight on the order of 34 Kg (75 lb) as compared to 2.7 Kg (6 lb) for a wall mounted unit. In this basis, the internally suspended heat exchanger was eliminated from further consideration.

When comparing internally mounted wall heat exchangers with externally mounted wall heat exchanger, the only criteria of consequence appears to be in manufacturing and inspection. Fabrication, assembly and inspection of internally mounted tank hardware is generally more complex, time consuming, and costly than for externally mounted hardware. Since no obvious performance advantage can be anticipated for the internally mounted wall heat exchanger, it was also eliminated from further consideration.

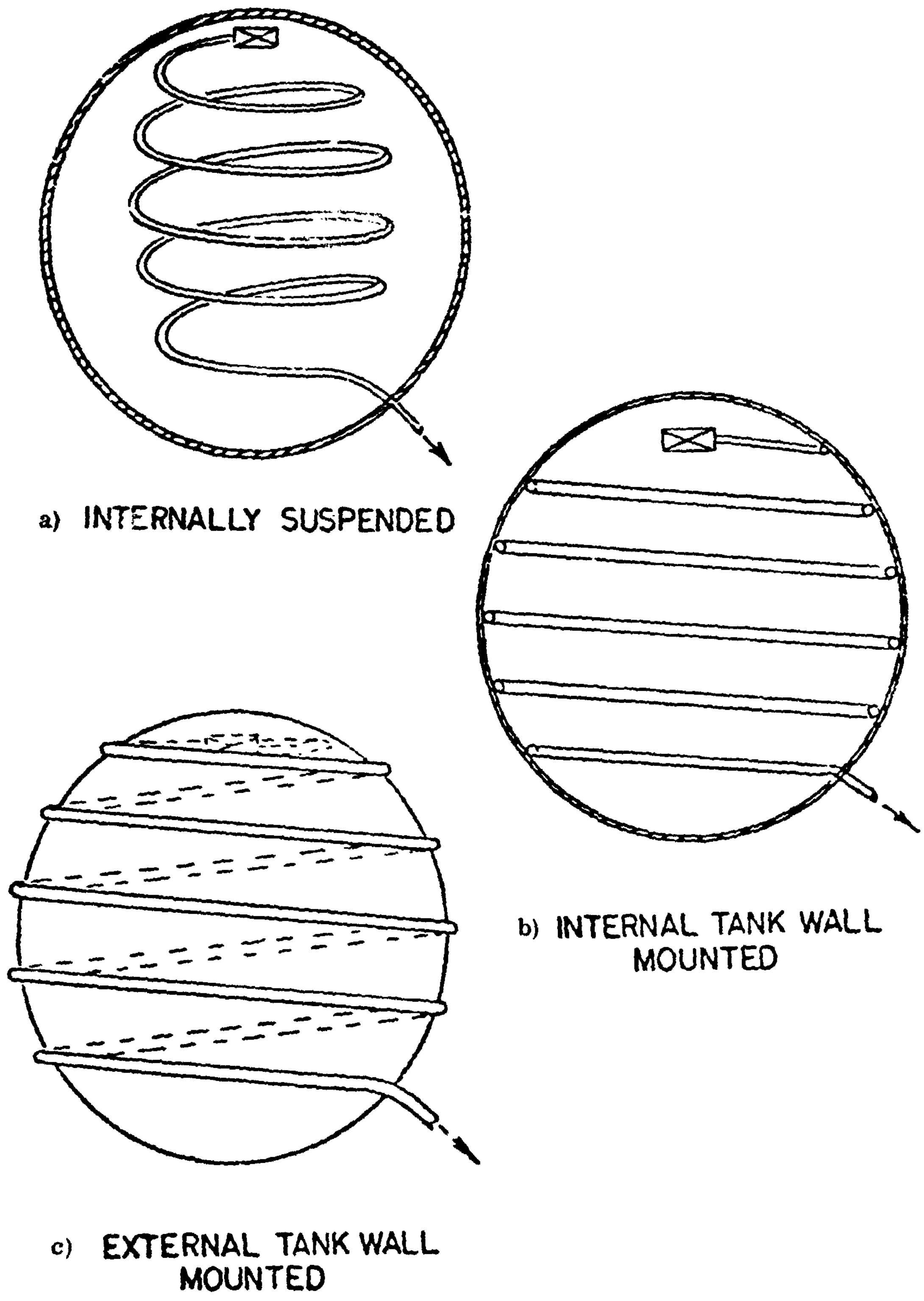


Figure 2-7. Distributed Heat Exchanger Configurations

Two types of externally mounted wall heat exchangers were considered; continuously attached tubular and point contact tubular. The continuously attached tube has the advantage of requiring less length to transfer a given quantity of heat. The point contact exchanger has the advantage of a more even distribution of heat flux and smaller tank wall temperature gradients because of the higher resistance to heat flux of the stud which in turn requires a longer tube.

2.4.1.2 Manufacture and Assembly. Several methods of continuously attaching an aluminum heat exchanger to an aluminum tank wall are discussed in Reference 2-14. The simplest method is brazing which may or may not require use of reinforced skin thickness, depending on the size of the tank. For this type of continuous attachment the assembled tank and heat exchanger will have to be re-heat treated to insure adequate strength of the parent tank metal.

A predesign analysis was conducted to determine the best method of point attaching the heat exchanger tube to the tank wall. On the basis of this analysis, the best overall method of point attaching an aluminum tube to an aluminum tank appears to be by brazing the tube to a number of machined studs and then resistance welding the studs to the tank wall with a commercially obtainable stud welding hand gun. The studs would be machined from round stock aluminum and would then be slipped over pre-bent tube coils and brazed at specified positions. This assembly would be brought to the tank and the studs resistance welded to the tank lands. The final step is to remove the stud shank and dress the stud. Local heating of the tank material is minimized and thus re-heat treating of the tank is not required. Also, the low profile of the finished heat exchanger installation will be compatible with multilayer insulation systems and this particular method of point contacting enables the designer to control heat flux by selecting separation distances between studs on the tube and/or contact area between the studs and tank wall.

2.4.1.3 Wall Mounted Heat Exchanger Performance. In order to determine the performance of the wall heat exchanger system for the LOX vent application a computer model was developed. The LOX vent computer program is designed to allow venting of either gas or liquid from the tank and to have either gas or liquid at the bottom or top of the tank. This is necessary in order to determine the effect on vent system performance of the various fluid orientations which may exist in space.

References 2-18 and 2-19 were the only literature found which attempted to analyze the effect of the contained fluid conditions on the performance of wall heat exchangers. The program was set up to develop an analytical model of a vent-free fluorine feed system and then fabricate a test article to generate data for verification of the model. Reference 2-20 reports test results with the Ref. 2-18 system and indicates that the analytical model was modified and improved and that the final model could predict test data with ±20% under all imposed conditions.

An overall description of the present model giving the basic equations, assumptions and execution technique is given in Reference 2-21. Figure 2-8 is a schematic of the model as set up for the analysis discussed below.

The model was used to determine the heat exchanger-to-tank wall contact area which gave the most reasonable performance when the tank contained 90% liquid and liquid was being vented. Most reasonable performance was defined as the least amount of mass vented for a fixed pressure change with an all gas exit condition. Pressure drop was not considered in the analysis. With the contact area defined and held constant the liquid quantity and liquid orientation were then changed to determine the effect on vent system performance.

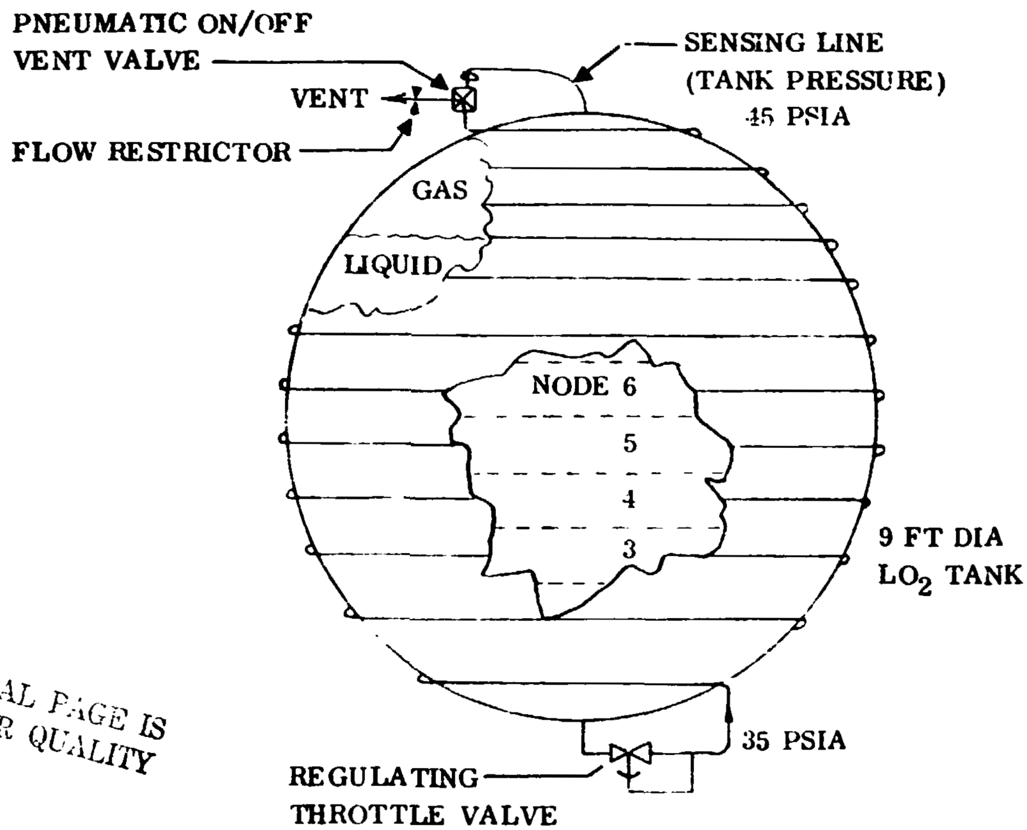


Figure 2-8. Schematic-Vent System Model Set-Up

Basic assumptions used were as follows: The tank was constructed of .15 cm (.060 in.) aluminum and was 2.7 m (9 ft) in diameter. The heat exchanger tubing was .635 × .056 cm (.25 × .022 in) aluminum with length and contact area as required. Initial ullage volume was set at 10%, system inlet was liquid, and the bulk liquid and heat exchanger inlet were at the same end of the tank. Initial fluid and tank wall temperatures were as given in Figure 2-9. These conditions represent a nearly full tank which has been allowed to stratifiy as might happen after a short engine burn. Vent valve settings were 312.6 kN/m^2 (45.5 psia) crack and 306.3 kN/m^2 (44.5 psia) reset. The flow restrictor was set at 3.25×10^{-6} kg/sec (2 lb/hr) and the heat exchanger inlet pressure was regulated at 241 kN/m^2 (35 psia).

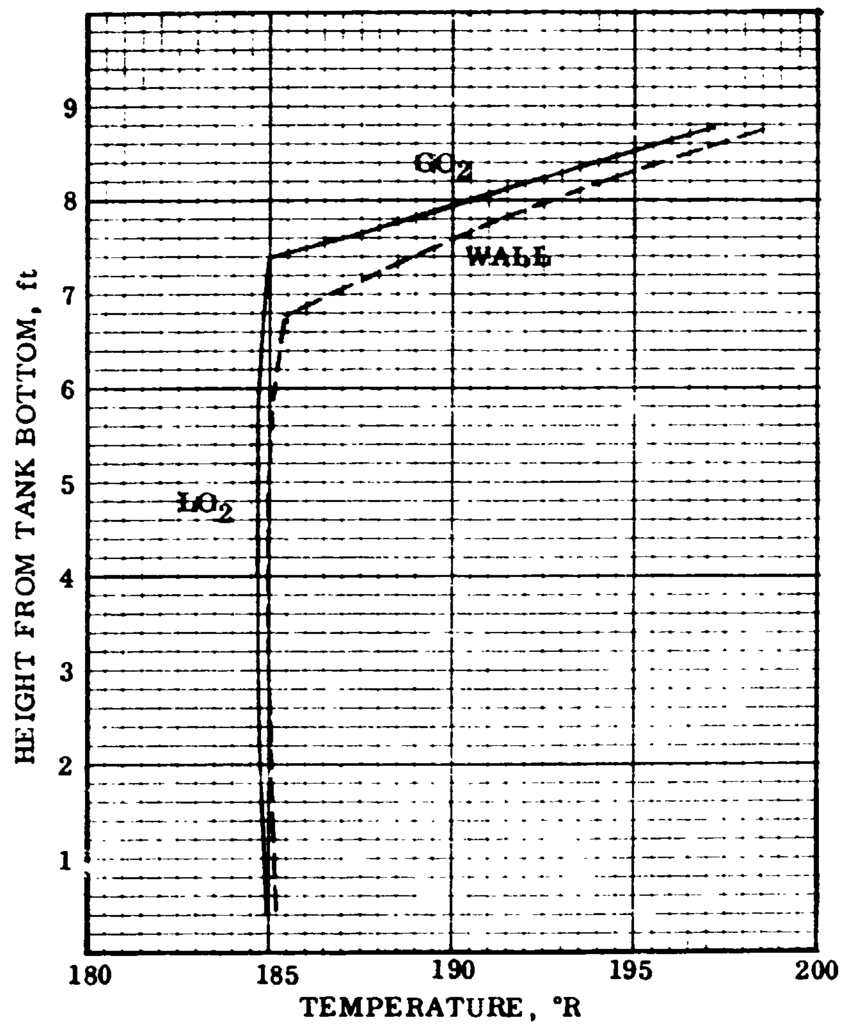


Figure 2-9. Initial Temperature Conditions for LOX Vent System Model

Figure 2-10 shows tank pressure decay and vapor quality of the vent fluid as functions of time for several heat exchanger configurations. Qualities greater than 1.0 indicate superheated vapor. Total mass vented to achieve the blowdown is also indicated on the figure. It is of interest to note the effect of under and over design (too little and too much contact area) on heat exchanger vapor quality and time to vent. When under designed the heat exchanger exit vapor quality is less than 1.0 (contains liquid). Decreasing the contact area lowers the exit vapor quality and increases the time to vent with the net result being wasted liquid to achieve a given pressure decay. Over-designing has the same effect, i.e. increased vent time and increased vented mass, however, the heat exchanger is superheated vapor. In this case it appears that increased contact area in the liquid region of the tank causes the liquid to subcool. This decreases the capacity of the heat exchanger fluid to cool the ullage and thus increases the time to achieve a given pressure decay. Unless the tank contents are on occasion thoroughly mixed by normal vehicle maneuvers, a system with too much contact area in the liquid region would be extremely costly in terms of excess vented oxygen.

Investigation of the effect of changing heat exchanger inlet pressure revealed that increasing the pressure and thus inlet temperature to a heat exchanger sized to operate at a lower pressure produced the same effect as operating with an undersized heat exchanger, i.e., exit vapor quality decreased and vent time increased. This could be an important consideration for systems which control pressure by varying mass withdrawl which in turn changes heat exchanger inlet pressure.

From thermodynamic considerations the ideal heat exchanger design would, for either gas or liquid at the vent system inlet, require the same amount of time and vented mass to achieve a given pressure decay. A design sufficient for comparative purposes is represented by the solid lines in Figure 2-9; .455 m^2 (4.20 ft^2) contact over the wall area containing liquid and .256 m^2 (2.77 ft^2) over the ullage. This design was selected since it vented a low mass and it did vent vapor only. Figure 2-11 was generated using this heat exchanger configuration. The solid line represents liquid venting with a 10% ullage volume. The dashed line represents liquid venting with an 80% ullage volume. The dotted line represents vapor venting with a 10% ullage volume. For this case the liquid volume is positioned at the opposite end of the tank from the vent system inlet. It is presumed that the difference in vent down time between the 10% ullage vapor and liquid venting cases is due to the heat exchanger being over designed for the liquid case which showed a slight subcooling of the liquid.

In all cases attempted with the computer model tank pressure was controlled with both inlet liquid and vapor venting. The major variation from case to case was the quantity of oxygen vented for pressure control. Tank pressure can be controlled within ± 29 kN/m^2 (2 psia) limits with a thermodynamic vent system utilizing a tank wall mounted heat exchanger. However, the efficiency of such a system is questionable. Once a heat exchanger is sized for a specific set of conditions it is easy to

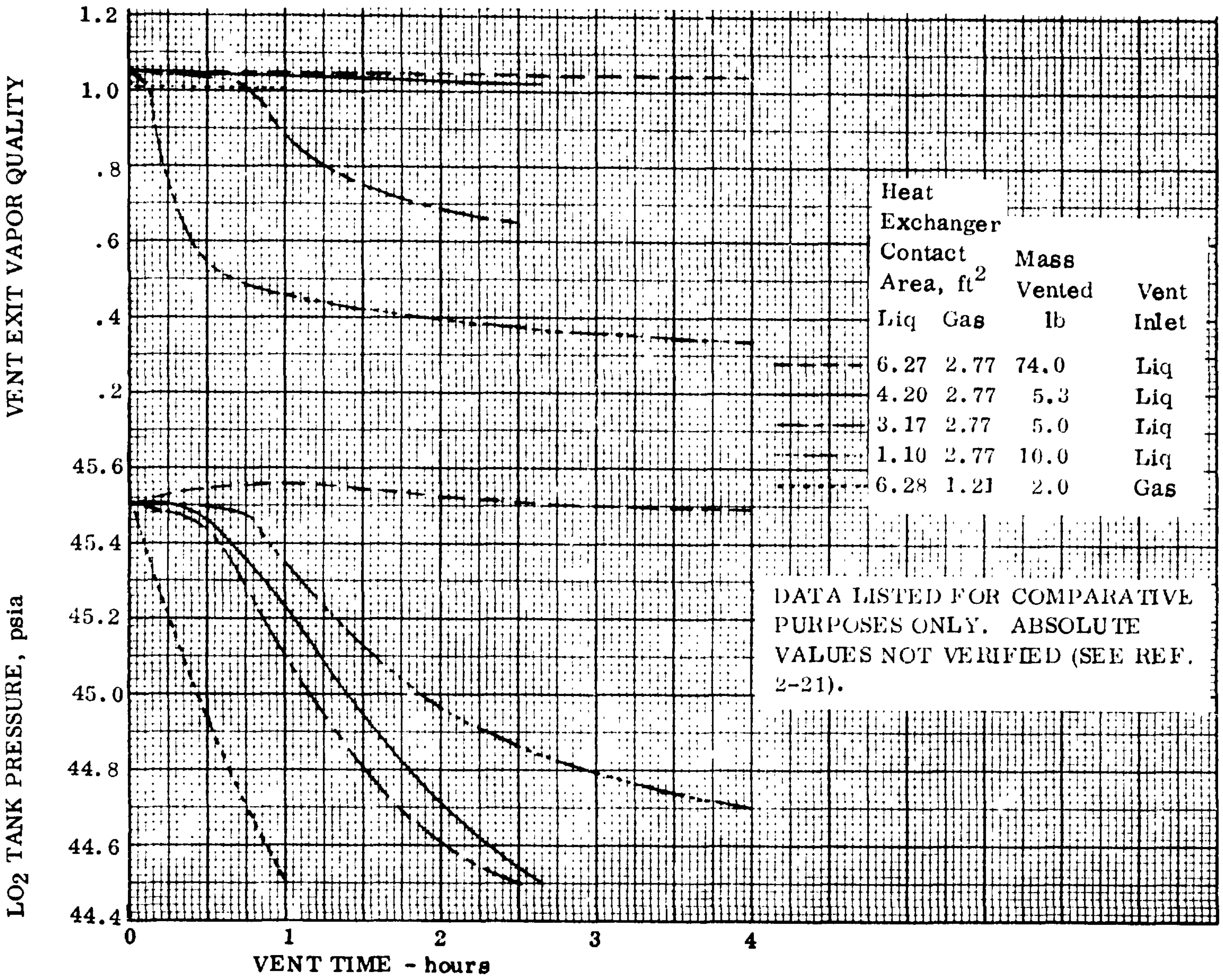


Figure 2-10. Effect of Wall Heat Exchanger Contact Area on Vent System Performance

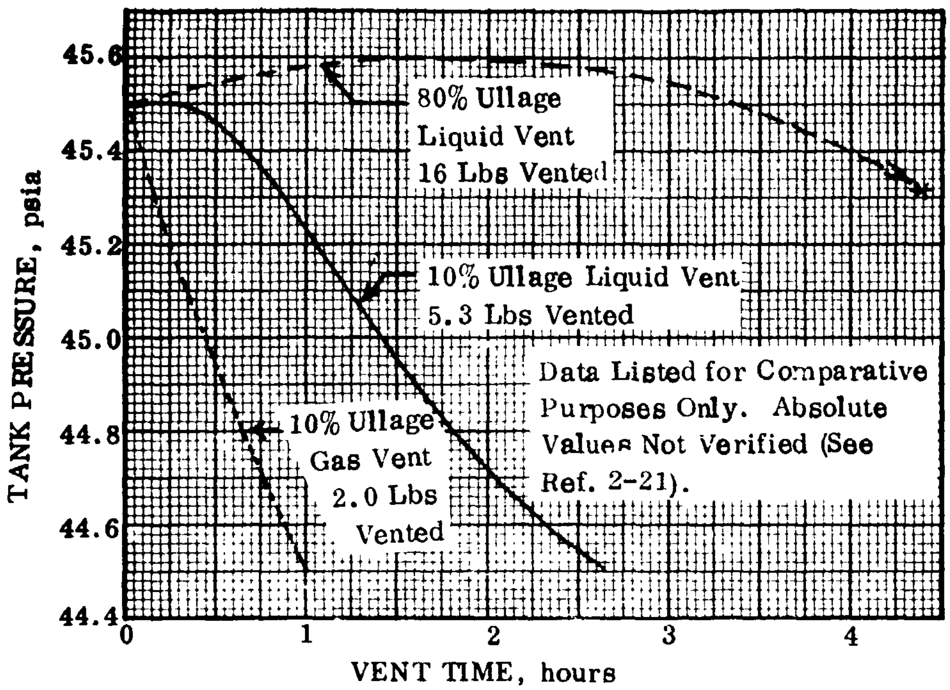


Figure 2-11. Effect of Variable Liquid Quantity and Orientation

conceive of a probable change in fluid orientation or quantity which would produce for example the effect of oversizing the heat exchanger, i.e. causing subcooling of the contained liquid. Furthermore, practical design practices would dictate that the heat exchanger be conservatively designed (i.e. oversized) to assure vapor venting during operation under all conditions. Thus it appears that liquid subcooling cannot be avoided with a thermodynamic vent system utilizing a wall heat exchanger; unless periodic mixing of the tank contents is achieved, the system could be very inefficient.

There are only two means of obtaining mixing for a wall mounted heat exchanger system: addition of a separate mechanical mixer or dependence on normal vehicle maneuvers. On extended duration missions where vehicles will be docked to space stations or otherwise be inactive for long periods of time, through propellant mixing cannot depend on vehicle maneuvering. Thus there is no obvious means of assuring efficient venting with the wall heat exchanger vent system, nor can the total amount of propellant vented during a long duration mission be reliably estimated.

2.4.2 BULK HEAT EXCHANGER. The concept of the bulk heat exchanger thermodynamic vent system has been demonstrated in Reference 2-22. Application of the concept for use in LO_2 possess no problems other than isolating the motor winding and wiring. Previous work (Ref. 2-11) has demonstrated that the concept is not mission dependent and that weight penalties can be defined for both system weights and vented propellant.

2.4.3 CONCEPT COMPARISON AND SELECTION. Data and discussion from Sections 2.4.1 and 2.4.2 was used to develop a concept comparison between a thermodynamic vent system which utilizes a free convection tank wall heat exchanger and one which uses a forced convection compact heat exchanger. Table 2-4 contains the generalized comparison. The wall heat exchanger concept hold several desirable advantages over the compact concept. For hardware operation the wall heat exchanger is inherently more reliable, and safer because the number of moving parts is minimized and electrically powered components are eliminated. In addition, the wall heat exchanger is easily integratable into capillary device and suction line cooling systems if it is set to operate on a continuous basis. This system also provides the lightest hardware weight design and would have the greatest potential for achieving maximum theoretical performance.

The compact heat exchanger system lists confidence in design, confidence in development, flexibility in application and ease of assembly as its comparative advantages over the wall heat exchanger system. Reasons for these advantages are given in the comment section of Table 2-4. The two items involving confidence are extremely important to making a final selection between the two concepts. There is little or no assurance from past experience or present analysis that the wall system could successfully perform the intended venting function. Also related to this lack of confidence in the all heat exchanger system is item 2 of Table 2-4, which indicates that at this time the performance of the wall heat exchanger cannot be adequately predicted Past experience does on the other hand indicate that the compact system is capable of performing as required and is predictable in its operation. The thermodynamic vent system utilizing a forced convection compact heat exchanger is therefore selected as the best prototype LOX vent system for this program.

It is noted that for missions significantly longer than 30 days, the compact exchanger system with electric pump may become less attractive from both a thermodynamic performance and hardware operational standpoint. As space residence time increases, electrical demand increases. As an example, the total amount of electrical power added to the tank contents increases as does the weight of the power supply. Increased power to the tank results in additional excess ventage. The increased time also means moving hardware must be designed for long life without maintenance.

If the wall heat exchanger concept (or for that matter the passive bulk propellant control concept evaluated in Reference 2-22) could be devloped, its thermodynamic performance and weight would theoretically be unaffected by increasing space residence times. These systems should therefore, be given full consideration in any future program which requires space residence times significantly lo[illegible] than 30 days.

2.4.4 COOLING OF SUCTION FEEDLINE. Integration of suction feedline cooling with the thermodynamic vent system is dependent upon the type of system chosen. As indicated in Table 2-4 the free convection wall heat exchanger concept can be easily integrated with a suction feedline cooling system since it is a continuous flowing

Table 2-4. Comparison of Heat Exchanger Concepts for Thermodynamic Vent System

Criterion	Free Convection Wall Heat Exchanger	Forced Convection Compact Heat Exchanger	Comments
Hardware weight	Comparable	Comparable	Compact includes weight of electrical power supply.
Total weight penalty (hardware wt. plus excess vented O_2)*	Unknown	Predictable	Discussion in Section 2.4 explains why excess ventage unknown.
Theoretical Performance Potential	Best	Least	Longer duration missions cause increase in electrical power supply weight and total power added to propellant by compact system. No effect on ideal wall system.
Hardware Operational reliability	Most reliable	Slightly less reliable	Compact system uses more moving parts
Assembly and installation	More difficult and costly	Easiest	Wall difficult because tank lands must be provided plus reheat treat of entire tank may be required.
Safety	Best	Slightly less	Compact slightly less because electrical power is required for operation.
Confidence in deriving successful design	Least	Most	Wall is "least" due to lack of sufficient analysis and test for large tasks with variable fluid quantities and fluid orientations.
Integration with other systems	Easily integrated with capillary device and suction line coding sys.	Requires separate systems for cooling of capillary devices and suction lines.	Compact is used in an on-off system and cannot provide continuous cooling.
Flexibility in application	Least	Greatest	Example - increased tank size: compact-larger pump or added mixer. Wall - less effective on increased liquid mass and more susceptable to subcooling.
Confidence in system development	Least	Greatest	Compact has been successfully demonstrated in large cryogenic tank. (Ref. 2-11).

* Excess in ventage caused by additional heat input or inefficient venting over that which would be vented at saturated vapor conditions with only environmental heating.

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system. Distributed wall heat exchangers could be designed to include the suction feedline. However, since the compact heat exchanger system is used in an intermittent manner, continuous cooling can not be provided. The forced convection compact heat exchanger system requires a separate system for cooling suction feedlines. Since the compact heat exchanger concept was selected for development as the most effective zero-g vent system, no work was done on a system to provide cooling to the suction feedline.

Design analysis of a system to provide cooling of capillary acquisition devices and feedline systems has been completed under a company sponsored IRAD program and is presented in Reference 2-24.

SECTION 3

DETAIL DESIGN DEFINITION

This section describes the analytical procedures used to determine the specific operating requirements of the overall space LOX vent systen and the individual components. Basic vehicle design criteria were defined in Table 2-1.

The bulk heat exchanger vent system consists of the following basic components; throttle valve, shutoff valve, heat exchanger, pump and motor. Figure 3-1 is a typical system with the heat exchanger hot side fluid used as a mixing jet. In order for the system to perform properly, mixing of the bulk fluid must be accomplished to minimize liquid temperature stratification and allow communication of the bulk fluid at the exchanger with that controlling tank pressure at the liquid/vapor interface. Hot side fluid circulated through the heat exchanger will be used to mix the bulk fluid in the tank.

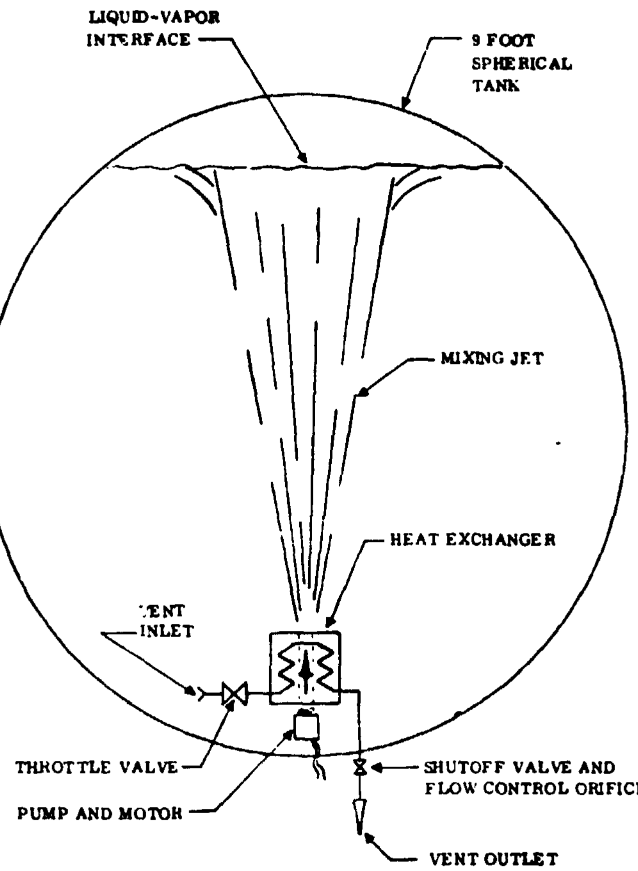

Figure 3-1. Schematic Bulk Heat Exchanger Vent System

The basic venting cycle is defined in Figure 3-2. 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.

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

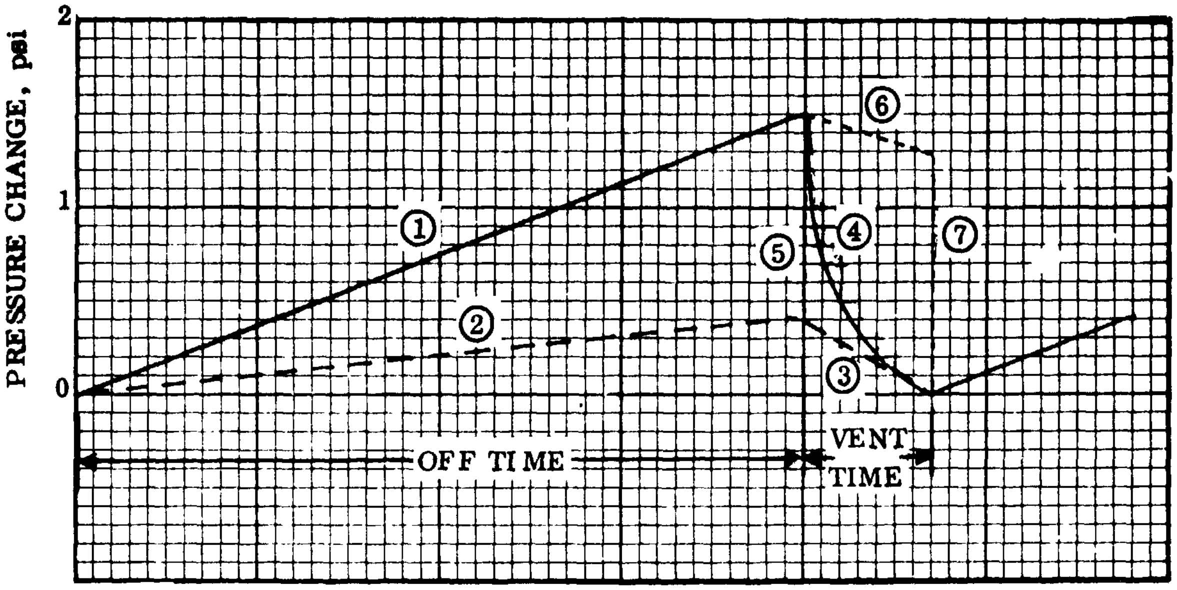

Figure 3-2. Typical Vent Cycle With Adequate Mixing

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 3-2.

Optimizations were performed to minimize the sum of exhcnager hardware, vented propellant, and power supply weights. The following step by step procedure was used to determine the minimum weight system.

1. Calculate the minimum pump energy required to mix a full tank (assumed to consist of a 3% ullage).
2. Calculate the pump flow and head required for mixing (with specific speed characteristics of a particular pump) as a function of mixing energy.
3. Calculate the time required to mix as a function of mixing energy.
4. Define time to vent as a function of vent flow rate and pump motor input power.
5. Determine the optimum heat exchanger configuration as a function of vent flow rate using the design point of pump assumed in step 2.
6. Define total system weight as a function of propellant vent rate for the motor assumed in step 2. Determine minimum total system weight and associated operating vent flow rate.
7. Determine that sufficient head is available from motor assumed in step 2 to allow system to operate at minimum total weight conditions. If not,

reconfigure system operating point as necessary to obtain minimum weight working system with this pump.

8. Repeat steps 2 through 7 with different size pump to evaluate effect of pump size on system weight.

3.1 MIXING ENERGY

The minimum energy required for mixing is based on work performed by the Fort Worth Division of General Dynamics. Mixing is intended to be accomplished by a small, high velocity jet issuing into the bulk fluid. Minimum mixing velocities are based on requirements for penetrating the warm layer of liquid at the liquid/vapor interface and mixing in a reasonable time.

The following equation from Reference 3-1 is used for determining the minimum energy required to penetrate the liquid/vapor interface:

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

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 .5K [1R])

Z = distance from mixer/vapor interface (assumed to be 2.58 meters [8.5 ft])

a = local acceleration

P = exponential constant (assumed to be .8 from Fort 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

$(V_o D_o)$ was calculated for different values of local acceleration, 10^{-4} g to represent the highest acceleration during coast periods and .141g to represent the disturbing acceleration during a typical OMS engine firing.

At an acceleration of 10^{-4} g, the minimum (V_oD_o) = .00272 m^2/sec (.0293 ft^2/sec) and at an acceleration of .141 g, the minimum (V_oD_o) = .10387 m^2/sec .118 ft^2/sec).

3.2 PUMP PERFORMANCE REQUIREMENTS FOR MIXING

In order to find the head and flow required for a given V_oD_o mixing parameter, the following equations were used:

$$\dot{m} = \rho AV, \quad Q = AV \text{ and } H = V^2/2g$$

where

$\dot{m}$ = mass flow rate

ρ = fluid density

A = flow area

V = velocity of fluid

Q = volume flow rate .

H = head of fluid

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

$$Q = \frac{(V_o D_o)^2 \pi}{4\sqrt{2gH}} \tag{3-2}$$

When this equation is combined with the pump specific speed equation, the head and capacity of the pump required for mixing are found. The pump specific speed equation is

$$q^{1/2} = \frac{N_s}{N} H^{3/4} \tag{3-3}$$

where

q = flow in gpm

N_s = pump specific speed in rpm (15,500 from Reference 3-2)

N = pump speed in rpm (5490, Reference 3-2)

H = head in feet

Solution of Equations 3-2 and 3-3 for H and Q in terms of V_oD_o gives the following minimum requirements to which a pump with the speed characteristics listed above must perform in order to provide mixing at .141g acceleration level:

$$V_oD_o = .1039 \text{ m}^2/\text{sec } (1.118 \text{ ft}^2/\text{sec})$$

$$Q = .00214 \text{ m}^3/\text{sec } (4.53 \text{ cfm})$$

$$H = .801 \text{ m } (2.625 \text{ ft})$$

These requirements are for mixing only. Additional requirements will be imposed when the heat exchanger is placed upstream of the mixing nozzle. The flow rate is the same for mixing as for the heat exchanger, but additional head is required for the heat exchanger pressure drop.

3.3 MIXING TIME

Time to mix is also important. It is necessary that the mixing be accomplished within the time allotted for venting to prevent the necessity of operating the mixing motor between vent cycles. It should be noted that if mixing is not complete, local subcooling will occur and the vent gas exit temperature will be lowered with a subsequent rise in weight of vented propellants. Thus, it is important that mixing be accomplished, preferably in as short a time as possible. If mixing is not accomplished within the venting time, the mixing pump must be operated between vent cycles requiring an additional input of power into the tank a more complicated control system.

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

$$\theta_m = 3 \times \frac{118\sqrt{H}\, D_t}{(V_oD_o)^{2/3}\, g_c^{1/6}} \left[\frac{V_o D_o \rho}{\mu}\right]^{-1/6} \qquad (3\text{-}4)$$

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

The above equation is obtained from Reference 3-3 as a reasonable estimate of mixing time based on past destratification testing performed at Convair with LH_2. The solution to Equation 3-4 is presented in Figure 3-3 as a function of liquid height above the mixing nozzle (H) and the product of mixing nozzle diameter and jet exit velocity (V_oD_o).

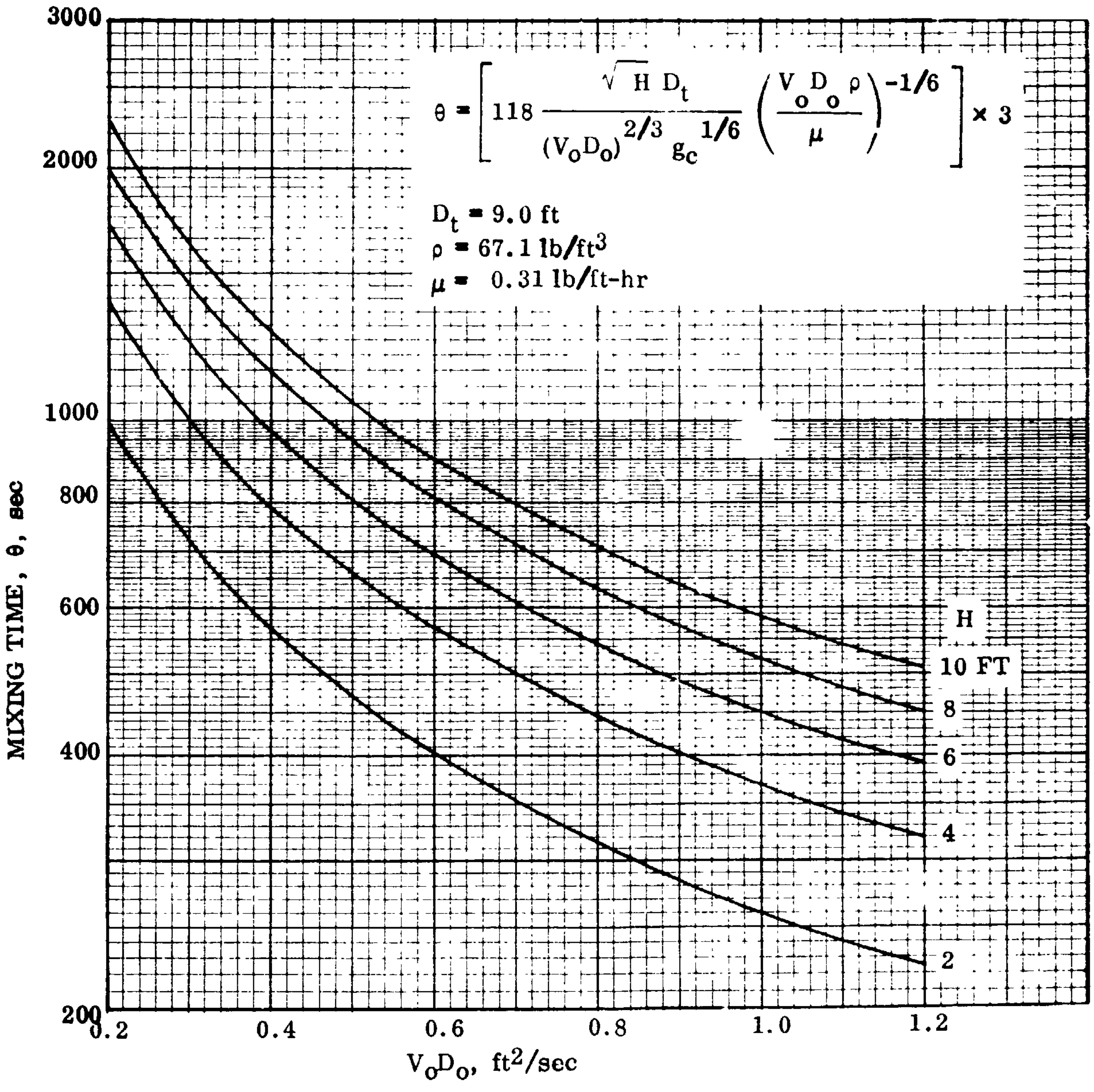

Figure 3-3. Solution of Mixing Time Equation as Function of V_oD_o and Liquid Height

3.4 VENT TIME DEFINITION

The time required to vent in a cycle 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.

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

1. Total pressure change (pressure switch deadband)
2. Non-homogeneous pressure rise rate (slope of line 1)
3. Homogeneous pressure rise rate (slope of line 2)
4. Homogeneous venting pressure decay rate (slope of line 3)

The pressure switch deadband was earlier redefined as 1.5 psia. The non-homogeneous pressure rise rate is taken from Reference 3-4 as

$$\frac{\Delta p}{\Delta t} = 1450\,(\dot{Q}/MS)^{1.14} \qquad (3\text{-}5)$$

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 rate for a homogeneous system is taken from a development presented in Reference 3-5 as

$$\frac{\Delta p}{\Delta t} = \left(\frac{\partial p}{\partial T}\right)_s \left(\frac{\dot{Q}}{M_\ell C_{p_\ell} + M_v C_{p_v}}\right) \qquad (3\text{-}6)$$

where

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

$(\partial p/\partial T)_s$ = partial of pressure with respect to temperature at saturation conditions, psi/°R

Q = tank heating rate, Btu/hr

M_ℓ = liquid mass in tank, lbm

M_v = vapor mass in tank, lbm

C_{p_ℓ} = specific heat of saturated liquid at constant pressure, Btu/lb-°R

C_{p_v} = specific heat of saturated vapor at constant pressure, Btu/lb-°R

The pressure decay rate during homogeneous venting is taken with slight modification (to include the heat capacity of the vapor) from Equation 3-5 of Reference 3-6 as

$$\frac{\Delta p}{\Delta t} = \left[\frac{\partial p}{\partial T}\right]_s \left[\frac{\left[(e\lambda/1-e) + h_o - h_\ell\right]\dot{m}_v - \dot{Q}_{in} - \dot{P}_{in}}{C_{p_\ell} M_\ell + C_{p_v} M_v}\right] \tag{3-7}$$

where

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

$(\partial p/\partial T)_s$ = partial of pressure with respect to temperature at saturation conditions, psi/°R

C_{p_ℓ} = specific heat of saturated liquid at constant pressure, Btu/lb_m-°R

C_{p_v} = specific heat of saturated vapor at constant pressure, Btu/lb_m-°R

λ = latent heat of vaporization, Btu/lb_m

h_o = specific enthalpy at vent outlet, Btu/lb_m

h_ℓ = specific enthalpy of bulk liquid, Btu/lb_m

$\dot{m}_v$ = vent weight flow rate, lb_m/hr

$\dot{Q}_{in}$ = total rate of heat transfer into the tank, Btu/hr

$\dot{P}_{in}$ = total power into the tank fluid via pump motor, Btu/hr

M_ℓ = liquid mass in tank, lb_m

M_v = vapor mass in tank, lb_m

e = ratio of vapor density to liquid density

The solutions to Equations 3-5, 3-6 and 3-7 are presented in Figure 3-4 as functions of percent ullage volume. Data from this figure can then be used to compute the vent time for any ullage condition or volume. Referring to Figure 3-2, the vent "off" time is first defined by dividing the total pressure change (switch deadband) by the non-homogeneous or quiescent pressure rise rate obtained from Figure 3-4. With "off" time defined, the total pressure rise which would occur in this time under homogeneous conditions is then determined. This is done by multiplying the homogeneous pressure rise rate from Figure 3-4 by the "off" time. Allowable vent time can be determined by dividing this homogeneous pressure rise (ΔP) by the homogeneous venting pressure decay rate.

3.5 TOTAL SYSTEM WEIGHT

The total system weight consists of vented propellant, electrical power supply and exchanger hardware.

The weight of vented propellant was calculated by the following formula from Reference 3-5:

$$Wt_{v_p} = \left[\frac{\dot{Q}_{in}}{\dot{m}_v \left[(e\lambda/1-e) + h_o - h_\ell \right] - \dot{P}_{in}} \right] t\, \dot{m}_v \qquad (3\text{-}8)$$

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

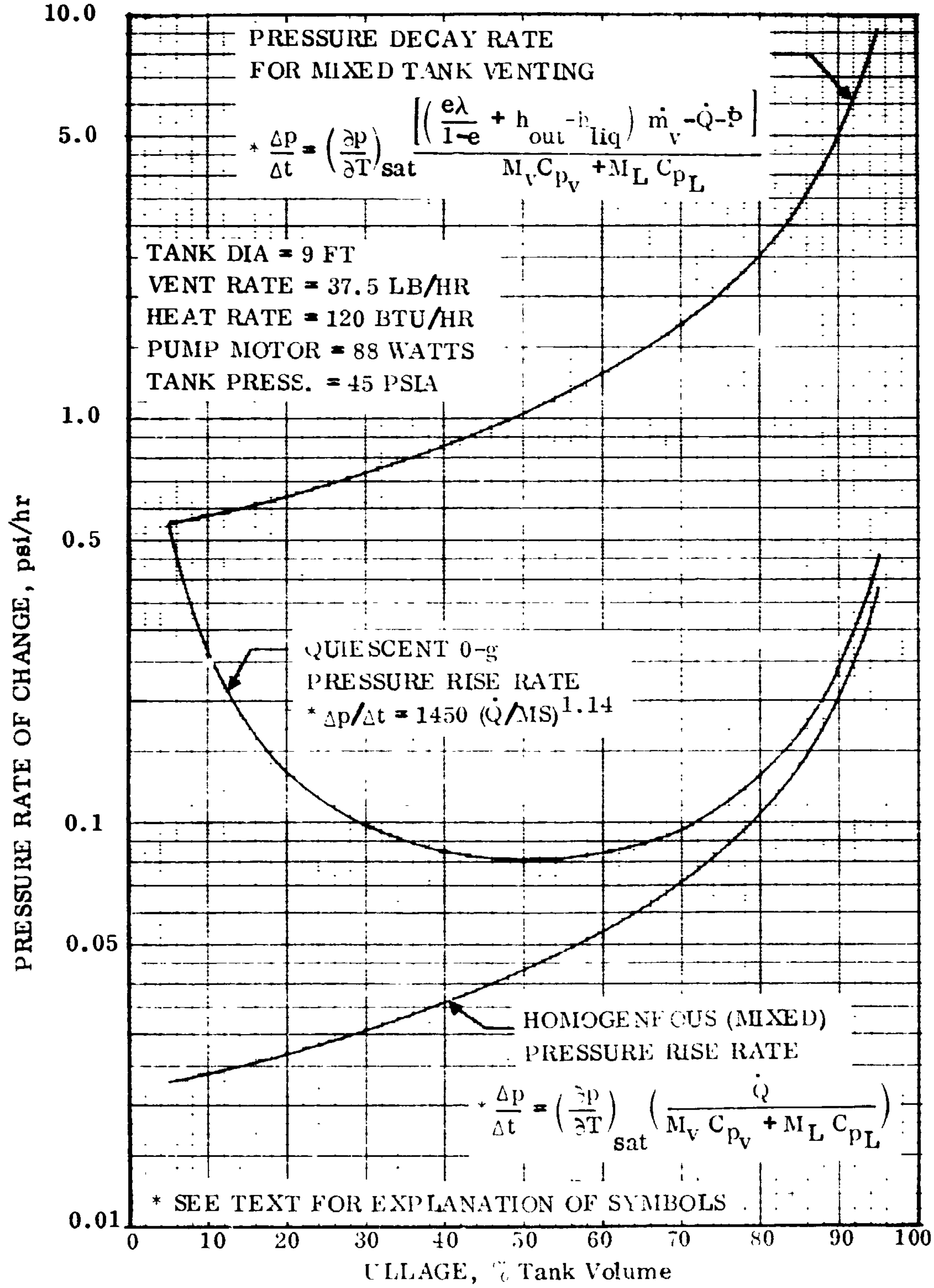

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Figure 3-4. LO_2 Vent System Pressure Rate Data vs Percent Ullage

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 3-7

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

or (3-9)

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

where

t_o = time that pump is actually on.

This time (t_o) is determined from Equation 3-8 where

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

or

$$t_o = \left[\frac{\dot{Q}_{in}\, t}{\dot{m}_v\, [(e\lambda/1-e) + h_o - h_l] - \dot{P}_{in}} \right] \tag{3-10}$$

Heat transfer calculations and heat exchanger sizing are based on the methods described in References 3-6 and 2-15. The exchanger is divided into three heat transfer sections based on the vent or cold side fluid condition.

I Boiling up to 90-percent quality

II Constant temperature vapor, 90-percent to 100-percent quality

III Variable temperature, superheated gas

Cold side heat transfer coefficients in Section I are based on the Kutateladze equation:

$$\frac{\dot{Q}}{A_{SC}[.5555\,(\Delta T_{WC})]^{2.5}} = 1.547 \times 10^{-7}\left[\frac{112.3\, C_{P\ell C}}{(h_{SV}-H_{S\ell})\,\rho_{VC}}\right]^{1.5} \times \left[\frac{0.0173\, k_{\ell C}\,(.01603\,\rho_{\ell C})^{1.282}\,(6.894\times 10^{4}\,P_{Ci})^{1.75}}{(\sigma_{\ell C})^{.906}\,(14.88\,\mu_{\ell C})^{.626}}\right] \tag{3-11}$$

In the foregoing equation the following units apply.

$\dot{Q}$, Btu/hr	$k_{\ell C}$, Btu/hr-ft-°R	$(h_{SV}-h_{S\ell})$, Btu/lb
A_{SC}, ft^2	$\rho_{\ell C}$, lb/ft^3	$\sigma_{\ell C}$, dynes/cm
ΔT_{WC}, °R	P_{Ci}, lb/in^2	$\mu_{\ell C}$, lb/ft-sec
$C_{P\ell C}$, Btu/lb-°R		

In Sections II ard III, cold side coefficients are obtained from the Dittus-Boelter Equation;

$$\frac{h_f D}{k} = 0.023\,(Re)^{0.8}\,(Pr)^{0.4} \tag{3-12}$$

Hot side heat transfer coefficients are based on the following equation from McAdams (Reference 3-10) for flow over tubes.

$$\frac{h_f D}{k} = \left[0.35 + 0.56\,(Re)^{0.52}\right]\,(Pr)^{0.3} \tag{3-13}$$

Heat transfer sizing for each section is based on a heat balance between hot and cold sides and the enthalpy change required in the vent fluid, as follows:

$$\dot{Q} = h_{f_H}\,A_H\,(T_H - T_w) = h_{fc}\,A_c\,(T_w - T_c) \tag{3-14}$$

where

$$(T_H - T_w) + (T_w - T_c) = (T_H - T_c) \tag{3-15}$$

and

$$\dot{Q} = \dot{m}_v \, h, \text{ enthalpy change in the exchanger section} \tag{3-16}$$

Iterations are made to determine the minimum unit size (tubing length and diameter) meeting both heat transfer and pressure drop requirements. The allowable hot side pressure drop is based on the pump head available. Weight is then determined assuming the use of aluminum tubing and shrouding.

In order to determine the heat exchanger weight as a function of vent flow rate, the following step-by-step procedure was used.

1. A specific motor-pump combination was assumed to provide the flow for the hot side of the heat exchanger and for mixing the bulk fluid. The design point for pump flow and head are used as an arbitrary starting point.

2. An exit temperature for the vent flow through the heat exchanger was assumed. Initial calculations indicated that this temperature should be within one degree of the bulk liquid saturation temperature to gain the maximum change in enthalpy between the liquid and the vented gas without overly increasing the size and weight of the heat exchanger.

3. A vent flow rate was assumed.

4. The head required for mixing was subtracted from head available and the remainder used in determining the heat exchanger hot side geometry.

5. The heat exchanger geometry was determined by use of the CHEAP computer program (Reference 3-8) by varying (a) the cold side coil inside diameter, (b) the cold side coil tube size, (c) the hot side free flow area, (d) the hot side free flow length, (e) the heat exchanger outside diameter and (f) the heat exchanger inlet pressure. The computer program calculated the pressure drop through both the cold side and the hot side for each configuration. The weight of each heat exchanger configuration was calculated and heat exchanger weight versus pressure drop for each configuration was then plotted to find the effects of the different variables. Typical data is presented in Figures 3-5 and 3-6. It was found, in general, that heat exchanger weight decreased with decreasing coil diameters, tube diameter and hot side free flow area.

6. Various configurations were checked with both liquid and gas at the vent inlet. The vent flow rate is decreased, venting will be longer with a subsequent increase in the time the pump motor is on. This will cause an increase in the weight of vented propellants. If the vent flow rate is increased, venting will be completed before mixing can occur. Local subcooling will then take place with a lowering of the vent gas enthalpy and a subsequent increase in weight of the vented propellants (liquid may actually be expelled from the vent if sufficient subcooling is allowed).

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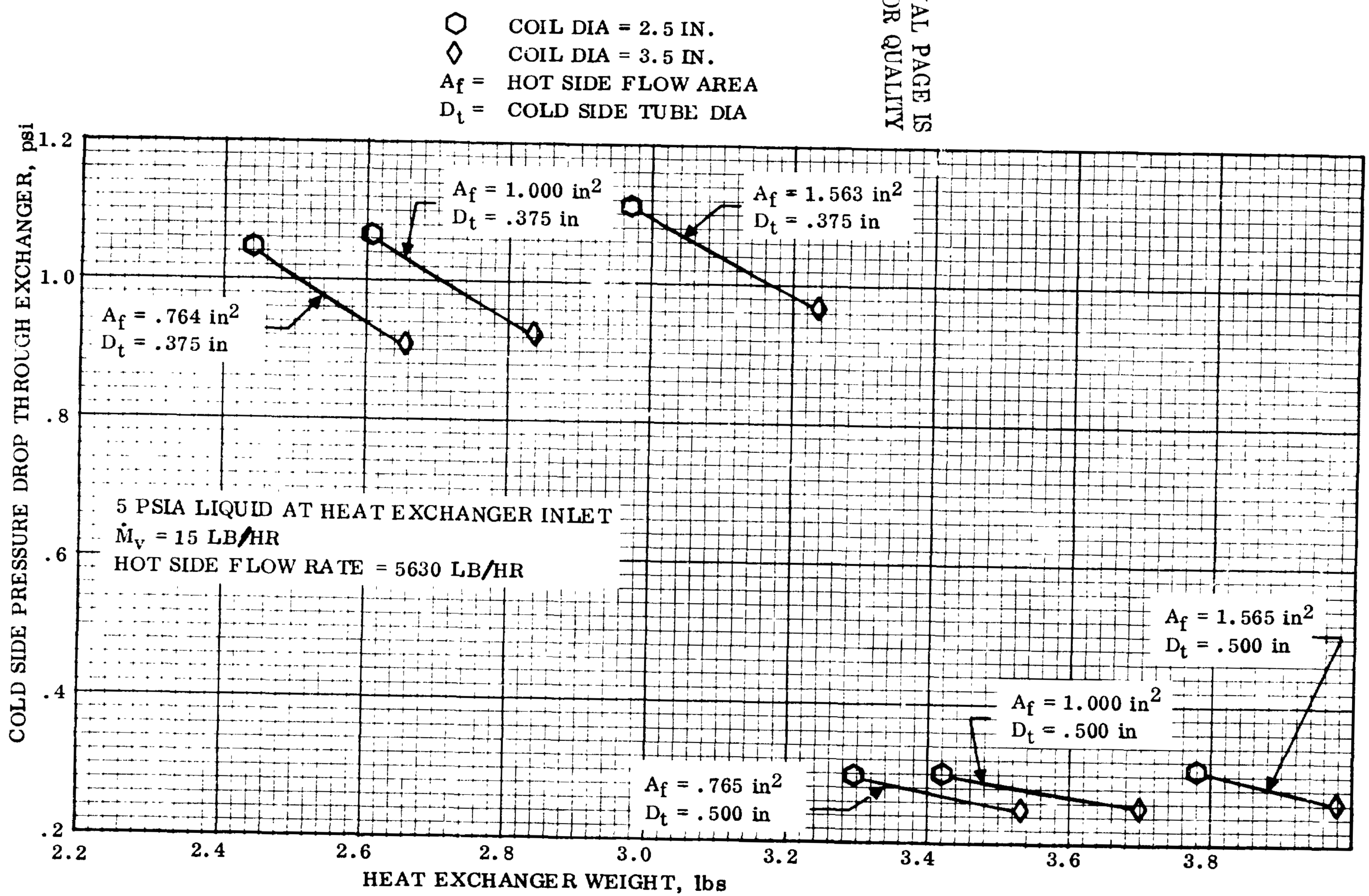

Figure 3-5. Effect of Heat Exchanger Configuration on Cold Side Pressure Drop and Heat Exchanger Weight

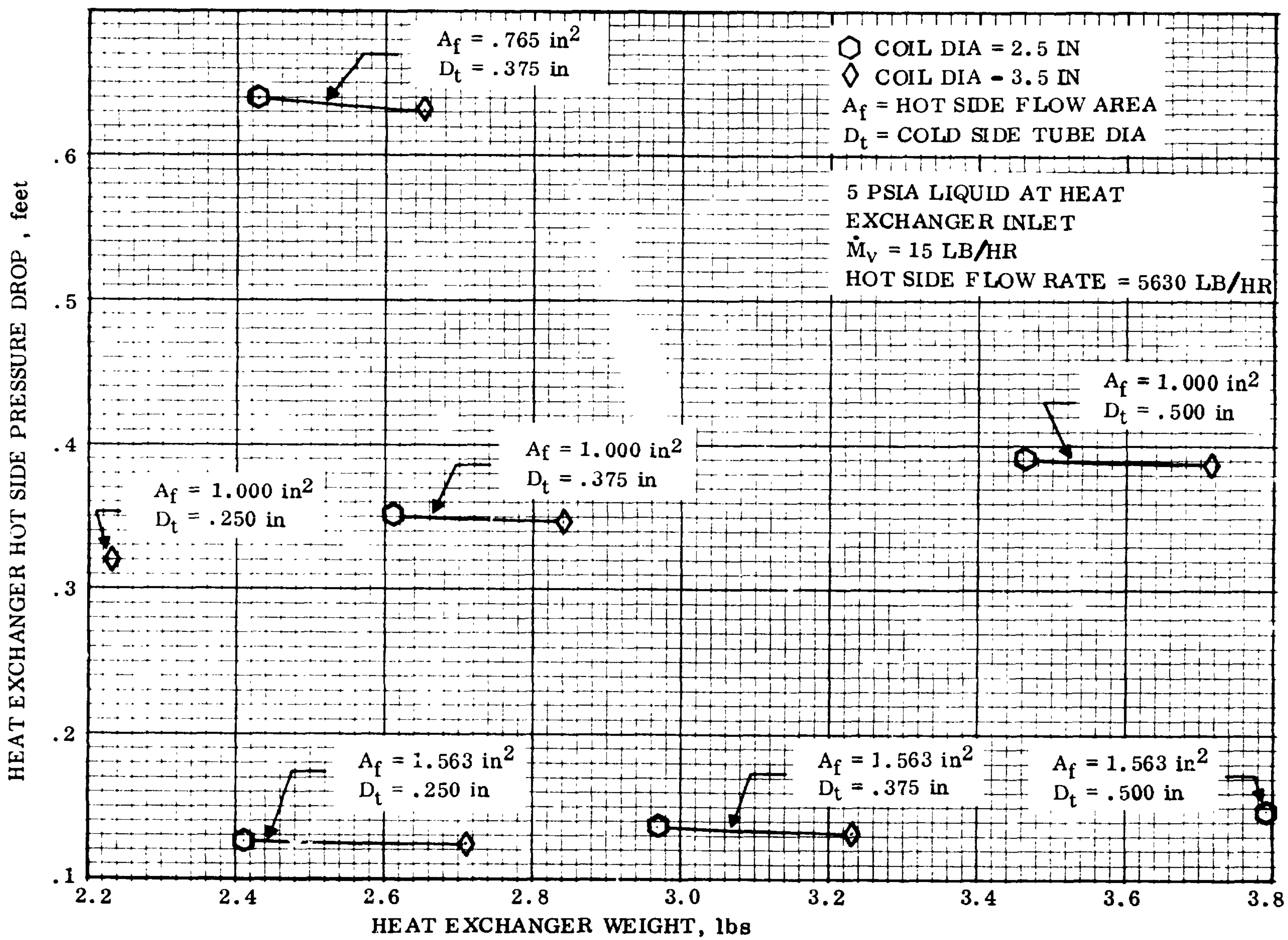

Figure 3-6. Effect of Heat Exchanger Configuration on Hot Side Pressure Drop and Heat Exchanger Weight

7. The above processes were repeated for increasing vent flow rates to find the optimum configuration and heat exchanger weight for each vent flow rate.

The heat exchanger weights, as calculated for each configuration, were plotted as a function of vent flow rate and an equation of the best curve was derived so that these weights could be added to the equation for the weight of vented propellants. The equation for the heat exchanger weight is:

$$Wt_{HX} = 12,171.6\, \dot{m}_v^2 + 370.4\, \dot{m}_v + 1.0$$

where Wt and $\dot{m}_v$ units are kg and kg/sec respectively. Or

$$(Wt_{HX} = .000426\, \dot{m}_v^2 + .10288\, \dot{m}_v + 2.245) \qquad (3\text{-}17)$$

for Wt and $\dot{m}_v$ units of lb and lb/hr respectively.

In the overall analysis, three pumps with AC induction motors sealed from the oxygen environment were chosen for further analysis, each 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.

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.

Pump system characteristics used are summarized in Table 3-1.

Table 3-1. 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 3-8, 3-9, 3-10, and 3-17, the total weight for pump system No. 1 is

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

$$+ .0426\ \dot{P}_{in} + 12171.6\ \dot{m}_v^2 + 370.4\ \dot{m}_v + 1.0 \qquad 3\text{-}18$$

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

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

$$+ .094\ \dot{P}_{in} + .000426\ \dot{m}_v^2 + .103\ \dot{m}_v + 2.245 \qquad 3\text{-}19$$

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

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

$$+ .0426\ \dot{P}_{in} + 10.514\ \dot{m}_v^2 + 360\ \dot{m}_v + 1.32 \qquad (3\text{-}20)$$

where units are the same as for Equation 3-18 or

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

$$+ .094\ \dot{P}_{in} + .000368\ \dot{m}_v^2 + .100\ \dot{m}_v + 3.0 \qquad (3\text{-}21)$$

with units as for Equation 3-19

Weight data obtained from Equations 3-19 and 3-21 are presented in Figure 3-7 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 3-9) for the 6 cfm pump No. 2 indicated a maximum power of 88 watts. Subsequent vendor data (Reference 3-11) showed a potential range of 60 to 80 watts. Power estimates for pump No. 3 are presented in Reference 3-6. Curves for all cases are included in Figure 3-7.

An examination of Figure 3-7 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 vent flow rates above approximately 2.9 lb/hr (to offset 120 Btu/hr heat input and 42 watts pump power) intermittent venting must be accomplished and the time required to mix becomes an important parameter. The time to mix requirement fixes the maximum allowable vent rate for each pump system.

Referring to Figure 3-7, 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 3-1) over that of pump No. 2 to warrant the increased system weight.

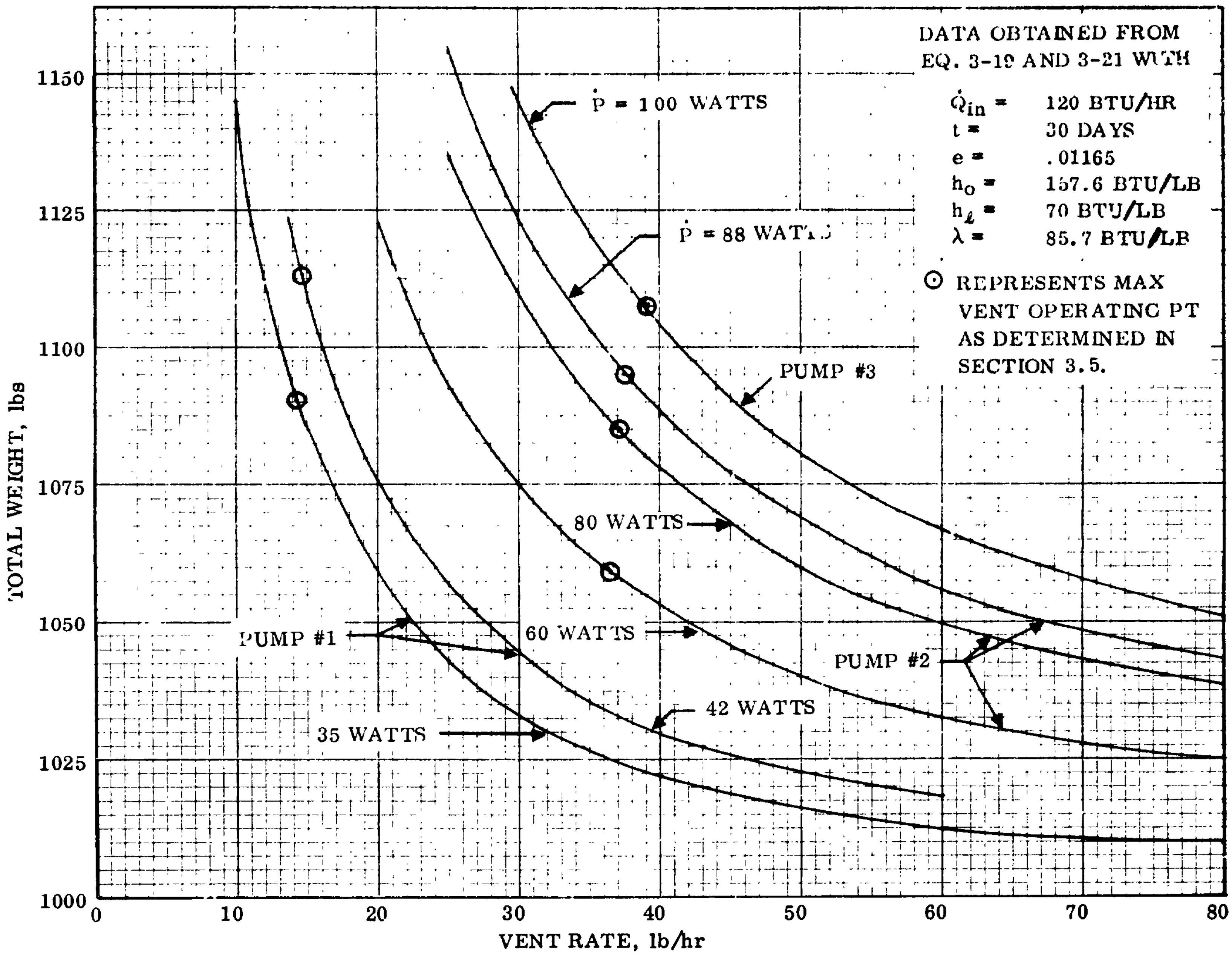

Figure 3-7. System Weight Versus Vent Rate

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SECTION 4

FINAL DESIGN PACKAGE

This section presents the overall systems package resulting from the analyses discussed in Section 3 and the selection of specific vendor hardware. Final overall system and component operating characteristics are included.

An assembly drawing of the in-tank vent system hardware, including provisions for pressure and temperature instrumentation, is presented in Figure 4-1. The pressure switch and shutoff valve are located separately outside the LO_2 tank and are thus not shown in Figure 4-1.

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

4.1 OVERALL SYSTEM

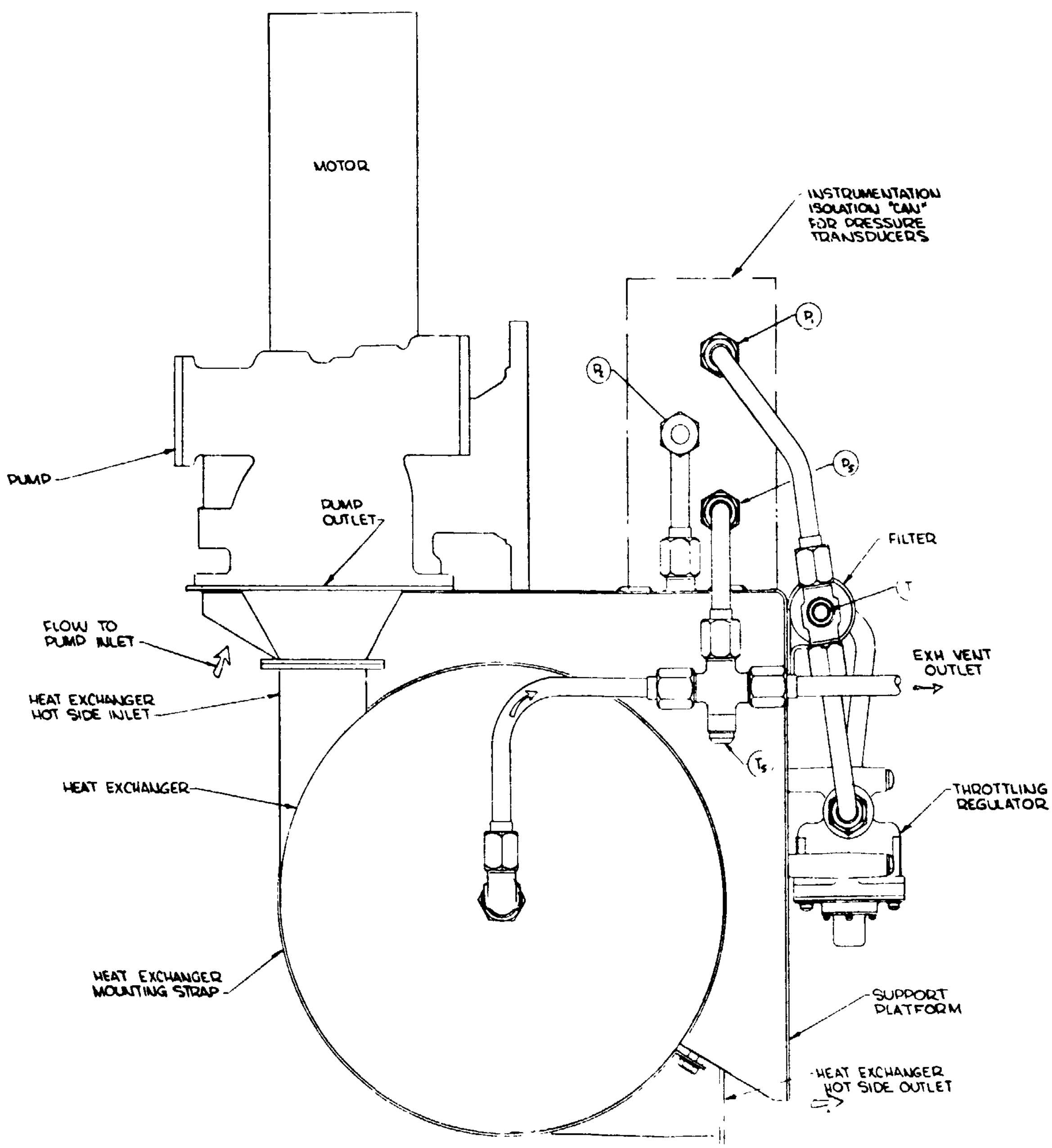
The system schematic is presented in Figure 3-1. The overall function is to control oxygen tank pressure to 310 ±13.8 kN/m^2 (45 ±2 psia) while allowing only superheated vapor to be exhausted to space. Operation is intermittent and the vent flow is nominally 4.7 gm/s (37.5 lb/hr) while venting. External heating of the tank is nominally 32.3 to 35.2 watts (110 to 120 Btu/hr).

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

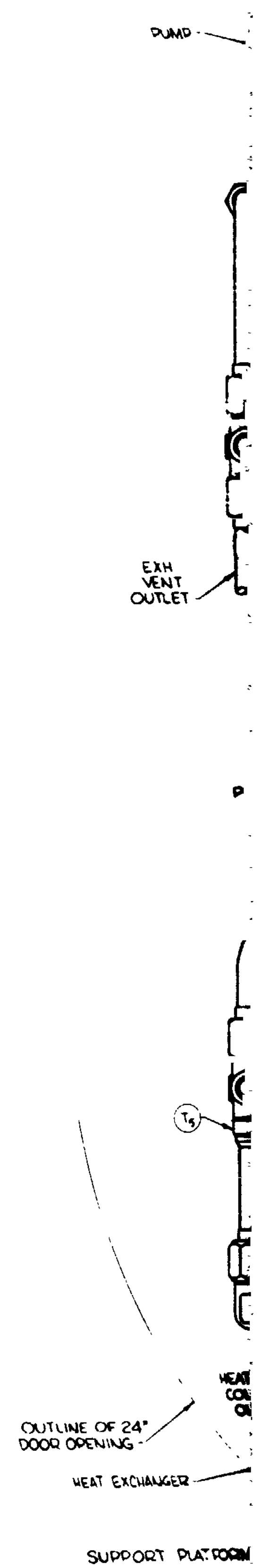
Service Life:	3000 hours	Operating
	69000 hours	Non-operating
	72000 hours	Total
Run Duration:	Maximum continuous run time, 4.0 hours. Maximum continuous run time, 15 seconds.	
Cycles:	Minimum of 30,000 start-run-stop cycles.	
Temperature Shock:	That experienced during the normal loading of a LO_2 storage tank.	
Acceleration Levels:	0 to 1g continuously applied in any direction.	
Cleanliness:	To Convair LOX clean Specification 0-75192-2 or equivalent.	

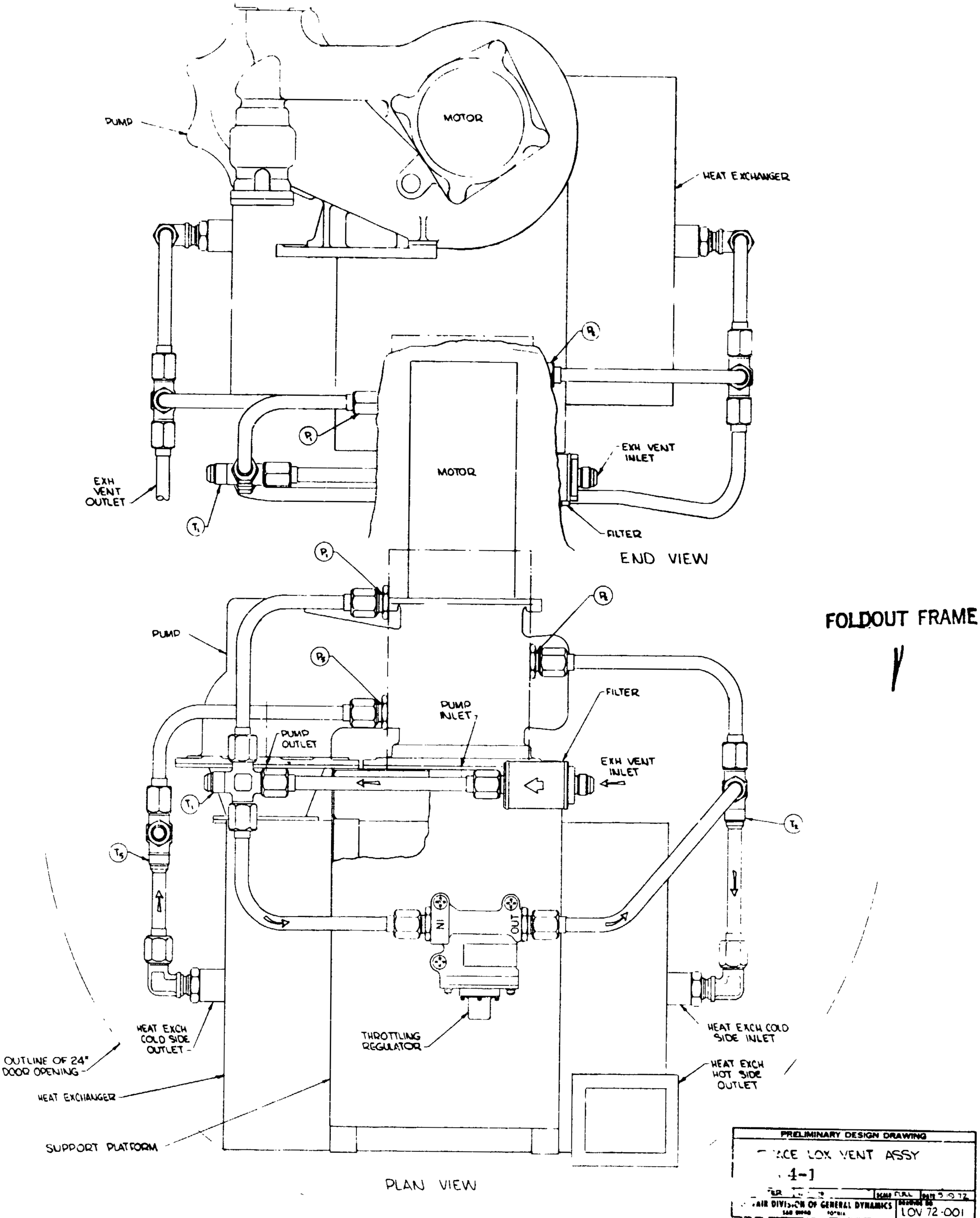
NOTES:
1- (P)&(T) CALLOUTS REFER TO POINTS PROVIDED FOR PRESSURE & TEMPERATURE TAPS
2- SCALE
10 INCHES

FOLDOUT FRAME



SIDE VIEW





FOLDOUT FRAME

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)
Pressure:	296 to 324 kN/m^2 (43 to 47 psia), operating to exact performance requirements.
	103 to 344 kN/m^2 (15 to 50 psia), off design operation, non-operating, and functional checkout.
Temperature:	88.9 to 111.1°K (160 to 200°R) operating to exact performance requirements.
	88.9 to 244°K (160-440°R), off design operation and non-operating.
	88.9 to 311.1°K (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 to space vacuum
Pressure:	0 to 103 kN/m^2 (0 to 15 psia)
Temperature:	294 ±28°K (70 ±50°F)

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

4.2 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:	2.84×10^{-3} m^3/s (6.0 cfm) at 0.915 m (3.0 ft) minimum static head rise with LO_2 at 1073 kg/m^3 (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 1073 kg/m^3 (67 lb/ft^3).

Motor Design:	Motor stator and lead wires fully enclosed ("canned") in stainless steel. Illustrative schematic presented in Figure 4-2.
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.

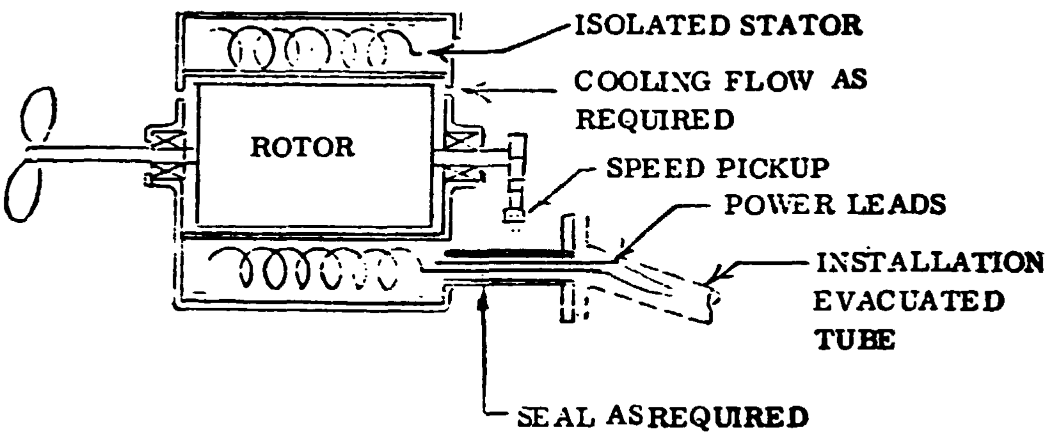


Figure 4-2. Pump Motor Schematic

4.3 HEAT EXCHANGER

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

Performance:

<u>Hot Side</u>

Inlet Media:	Saturated LO_2 and GO_2, separate or mixed, or superheated GO_2.
Flow:	2.84×10^{-3} m^3/s (6.0 cfm) of LO_2 at 1073 kg/m^3 (67 lb/ft^3).
Pressure Loss:	0.305 m (1.0 ft) (maximum) of LO_2 at 1073 kg/m^3 (67 lb/ft^3)
Inlet Pressure:	296 to 324 kN/m^2 (43 to 47 psia)
Inlet Temperature:	101.8 to 102.9°K (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 — 4.7 gm/s (37.5 lb/hr).
Pressure Loss:	3.44 kN/m^2 (0.5 psi) with GO_2 at 102.7°K (185°R).
Inlet Pressure:	Design Point — 151 ±6.9 kN/m^2 (22 ±1 psia).
Inlet Temperature:	94.6°K (170.3°R) max for design.
Outlet Temperature:	Design Point — 100.6°K (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 13.76 kN/m^2 (2 psi) greater than ambient. Hot side pressure 344.2 kN/m^2 (50 psi) greater than cold side. Ambient pressure 344.2 kN/m^2 (50 psi) greater than cold side.
Checkout Differential Pressure:	Hot side 34.4 kN/m^2 (5 psi) greater than ambient or cold side. Cold side 34.4 kN/m^2 (5 psi) greater than ambient or hot side.
Weight:	4.09 kg (9 lb) (max).

4.4 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:	5.72 ±0.31 gm/s (37.5 ±2.5 lb/hr) saturated LO_2 or GO_2.

Outlet Pressure:	151.4 ±6.9 kN/m^2 (22 ±1.0 psia) design operating.
Internal Leakage:	25×10^{-3} gm/s (0.02 lb/hr) allowable with 324 kN/m^2 (47 psia) LO_2 at the inlet and 206.5 kN/m^2 (30 psia) at the outlet.
Differential Pressures:	3.4 kN/m^2 (0.5 psi) crush load on upstream body, operating.
	0 kN/m^2 (0 psi), non-operating.
	206.5 kN/m^2 (30 psi), crush load on downstream body, operating.
	13.8 kN/m^2 (2 psi), burst load on downstream body, non-operating.
	344. kN/m^2 (50 psi), maximum design load on evacuated bellows.

4.5 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 5.04 gm/s (40 lb/hr) of saturated GO_2 at 296 kN/m^2 (43 psia).
Maintenance:	Filter element can be easily replaced for any required periodic maintenance.

4.6 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:	324 kN/m^2 (47.0 psia) (maximum).
Deactuation Pressure:	296 kN/m^2 (43.0 psia) (minimum).
Deadband:	10.3 kN/m^2 (1.5 psi) (minimum).
Internal Temperature:	294 ±28°K (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:	103 to 344 kN/m^2 (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.

4.7 SHUTOFF VALVE

In the final configuration shown in Figure 4-1 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 100°K to 311°K (180 to 560°R). Maximum pressure drop is 6.87 kN/m^2 (1 psi) of 311°K (560°R) GO_2 at 5.04 gm/s (40 lb/hr) flow.

SECTION 5

COMPONENT TESTING

Following system definition and preparation of component specifications, as presented in Section 3.0 requirements were provided to hardware vendors, bids were received, and selections were made on the basis of technical ability and economy.

The components were individually tested in LN_2 and LO_2 to demonstrate their compatibility with and capability to operate in the oxygen environment to be used in the evaluation testing.

5.1 DELIVERED HARDWARE

The requirements from section 4 were given to hardware vendors, bids received and selections made on the basis of technical ability, minimum costs and the ability to deliver on schedule. The following items were procured and assembled into a complete test package:

a. A throttling regulator of aluminum construction with an evacuated bellows sensing downstream pressure (HTL Industries P/N 187250-4). Weight on the unit is 0.28 kg (0.62 lb).

b. The filter is of stainless steel construction with a 10 micron-nominal rating (Western Filler Co. In. P/N 70-1-16510-10) Weight of the unit is 0.39 kg (0.87 lb).

c. The heat exchanger is of all aluminum construction and weighs 2.95 kg (6.5 lb) (Geoscience P/N 02B2). The cold or vent side flow is through a single coil of .95 cm (3/8 inch) tubing, and the hot side flow is vortexed over the outside of this tubing. This design allows for high efficient heat transfer of a boiling fluid and minimizes the possibility of liquid "carry-over."

d. The pump/motor is basically of stainless steel construction and weights 5 kg (11 lb.) (Sunstrand P/N EP145603-100, SNX-1). The unit was modified from an existing pump/motor. The motor was modified from a 60 cps 240 v unit to 60 cps 75 v unit with an operating speed of approximately 1800 rpm at 60 cycles, three phase, 75 v input current.

e. The pressure switch is of stainless steel construction and weighs 0.34 kg (0.75 lb). (Hydro-Electric 93159). It is located external to the propellant tank in the vacuum chamber environment.

5.2 COMPONENT TESTING

The objective of this portion of the test program was to demonstrate system components compatiblity with and satisfactory operation in LO_2. Tests were performed both before and after assembly into the vent system package in accordance with the requirements defined in Reference 5-1.

The regulator was initially checked out in LN_2 and found to be out of tolerance at flow rates from 3.4 to 23.6 kg (7.5 to 52 lb/hr). At 17 kg (37.5 lb/hr) the outlet pressure was too high and the regulator was returned to the vendor. After rework, the regulator was retested at flow rates from 3.4 to 31.7 kg (7.5 to 70 lb/hr.). Performance as shown in Figure 5-1 in LO_2 was in tolerance for all LO_2 flow rates. However, in cold GO_2 the maximum obtainable flow with the required outlet pressure was less than 13. kg (30 lb/hr). At this time the vendor indicated a special fitting in the regulator outlet would bring performance within tolerance. A modified fitting, supplied by the vendor, was installed and the test rerun, there was no significant improvement. Since the regulator was in tolerance flowing LO_2 which represents the major portion of the testing and there was sufficient GO_2 flow to control pressure, the operation of the pressure regulator was accepted.

The pump was initially checked out in LN_2 and GN_2 followed by the LO_2 and GO_2 tests. Performance was satisfactory in all respects. The minimum stable operating speed in this test set up was determined to be 690 rpm. At slower speeds pump operation was very erratic. The controlled variables during pump testing were input frequency and voltage. Test points were:

Hz	at	Vrms
60		87
50		73
40		58
30		44
25		36

The pump was operated for 7 hours at the 60 Hz design point and for 12 hours total at all speeds in both LO_2 and GO_2. The DC resistance of the pump motor field windings were measured at various temperatures from 294 to 94°K (+70 to -290°F) and found to vary from a minimum of 10.1 ohms cold to a maximum of 45.6 ohms at ambient temperatures. One further series of test points was taken to determine the ideal (minimum electrical power input) combination of frequency and voltage for a fixed pump speed. This was done by varying the input frequency to maintain a constant pump speed of 1680 rpm while varying the voltage from 87 to 55 Vrms. With the pump in LO_2, minimum input power was approximately 48 watts total at 60 Vrms

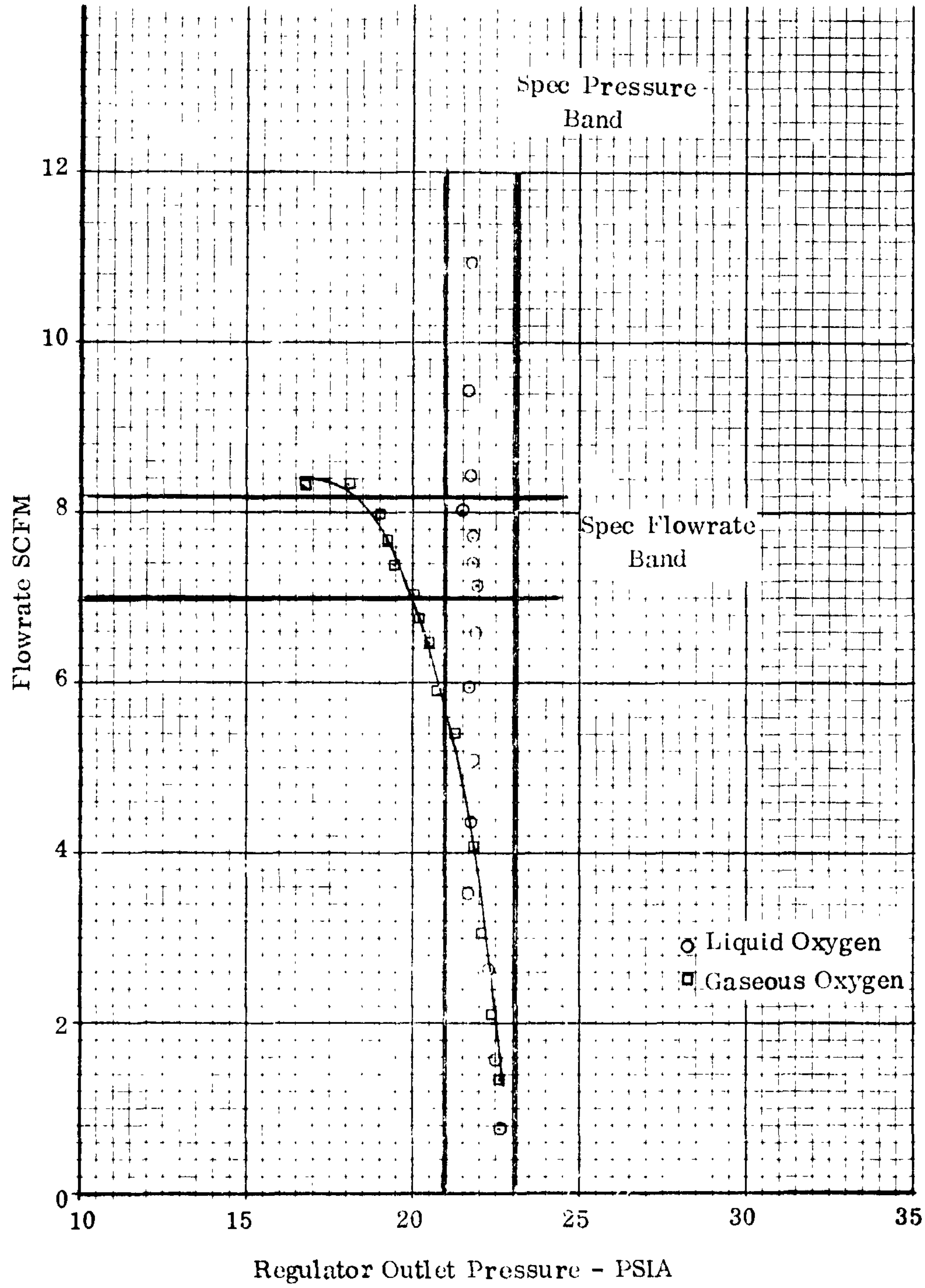


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Figure 5-1. Pressure Regulator Performance Test Results and 71 Hz. In GO_2 the pump speed was very nearly synchronous down to less than 10 Vrms.

The actuation of the pressure switch was checked and found to be in tolerance (45.4 psia make and 43.4 psia break). The switch was then exposed to the operating environment (installed on the test tank) during all the regulator and pump tests.

The components were then assembled into the complete system package and operation of the system was verified in both LN_2 and LO_2. Regulator and pump performance were virtually unchanged from that noted during individual testing.

SECTION 6

EVALUATION TESTING

The primary objective of the test program was to demonstrate the feasibility of the compact heat exchanger vent system to vent superheated gas and control tank pressure to 310 ±13.8 kN/m^2 (45 ±2 psia) when operating in either gaseous or liquid oxygen. Automatic pressure control characteristics of the system were measured when the system cycled between 297.6 kN/m^2 (43.2 psia) and 312.76 kN/m^2 (45.4 psia). The performance of each component was monitored to ensure that each portion of the system was operating correctly. The test system, testing performed, and test results are described in the following paragraphs.

6.1 VENT SYSTEM

The prototype vent system is pictured in Figures 6- , 6-2 and 6-3. The overall function is to control oxygen tank pressure to 310 ± 13.8 kN/m^2 (45 ±2 psi) while allowing only superheated vapor to be exhausted to space. Operation is intermittent and the vent flow is nominally 37.5 lb/hr during venting. External heating of the tank is nominally 32.2 to 35.2 watts (110 to 120 Btu/hr).

The system is assembled from the components identified in Sections 2 and 4. Added to the vent system package is a pneumatic cylinder and rotating joint at the base of the heat exchanger outlet nozzle. This will allow the direction of the heat exchanger hot side flow to be changed from the axial to the radial direction remotely. Instrumentation noted in Figures 6-1, 6-2 and 6-3 is identified as follows:

Temperature		Pressure	
T_1	Vent Inlet Temperature	P_1	Vent Inlet Pressure
T_2	HX Inlet Temperature — Cold Side	P_2	Inlet Pressure
T_3	HX Inlet Temperature — Hot Side	P_3	Outlet Pressure
T_4	HX Outlet Temperature — Hot Side		
T_5	HX Outlet Temperature — Cold Side		

The testing was to demonstrate vent system functioning capability in both gaseous and liquid oxygen, tank pressure control by venting only superheated vapor, and verification of predicted thermodynamic performance. System performance was evaluated under the following basic conditions.

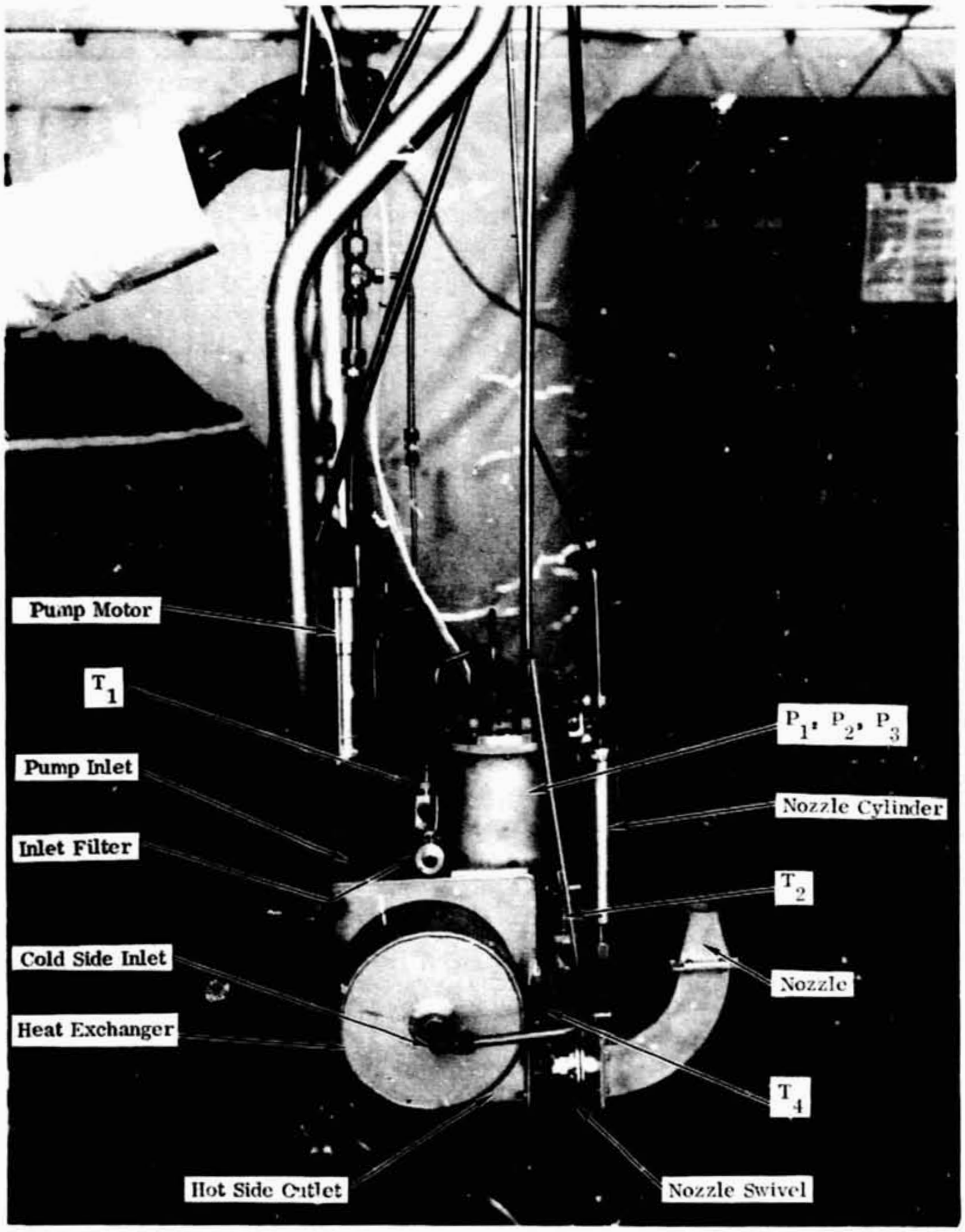

Figure 6-1. [illegible]ype Compact Heat Exchanger Thermodynamic Vent System, Side View — Cold Side Inlet

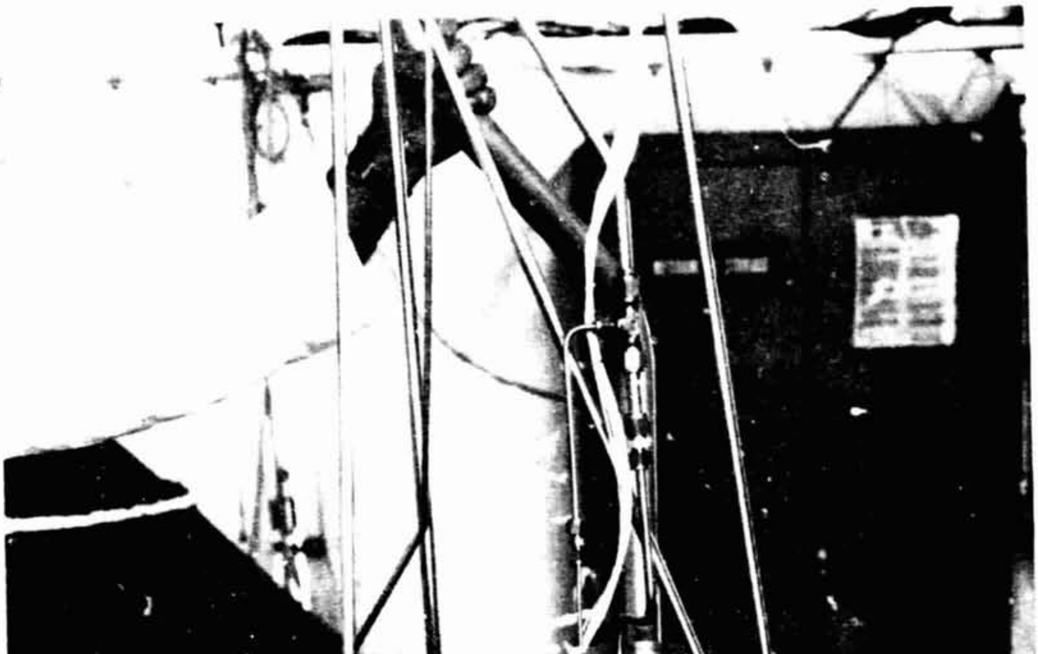

Figure 6-2. Vent System, Side View — Cold Side Outlet

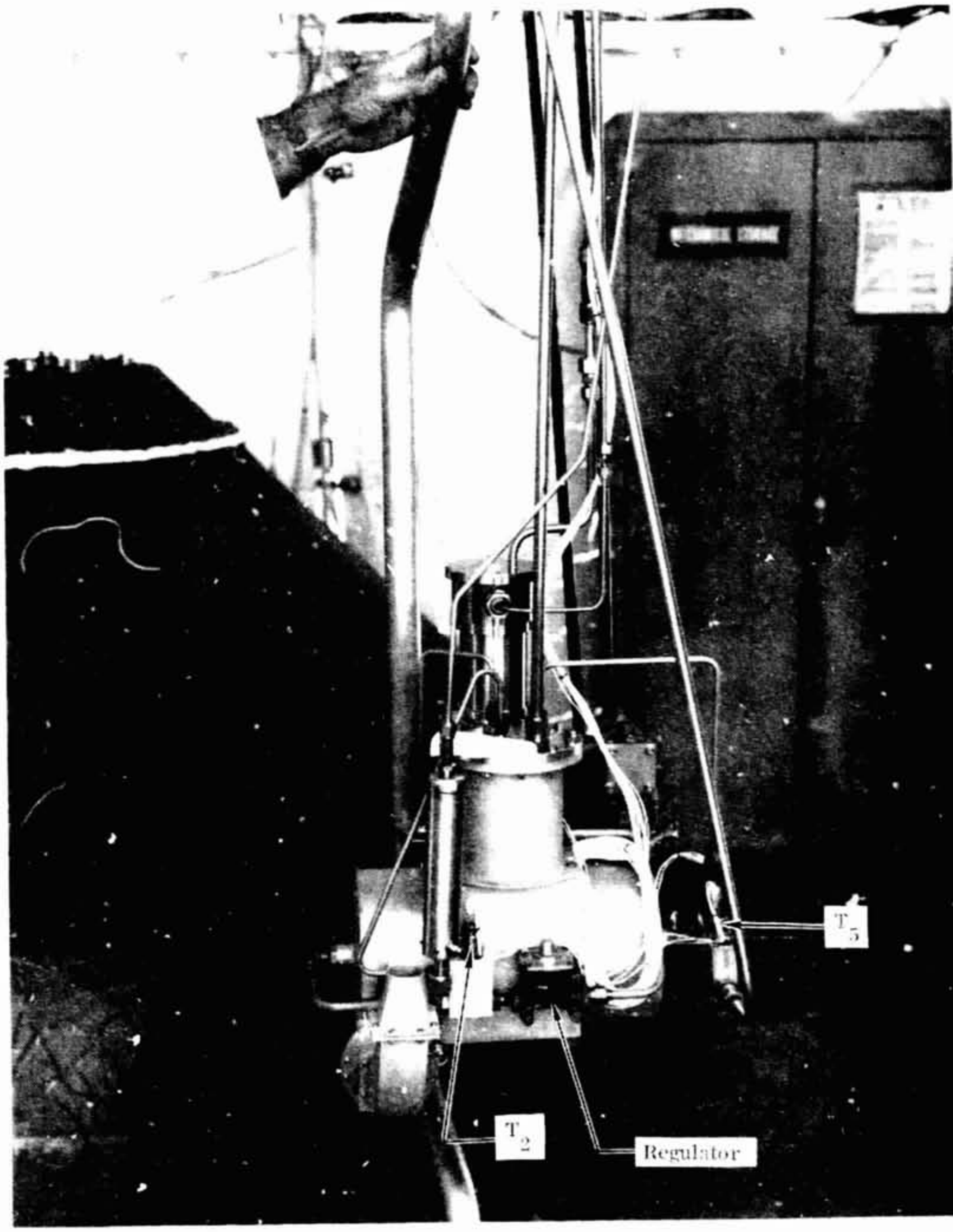

Figure 6-3. Vent System Test Package — Front View

1. Vent system actuation and deactuation with the test package or vent system located in gas and in liquid.
2. Test at two different hot side fluid discharge orientations to determine the effects on fluid mixing and tank pressure control.
3. Variation of pump speed, heat exchanger hot side flow and vent flow rates to determine limits of efficient system operation.

In all the above cases system operation efficiency was based on controlling tank pressure with the vapor vented at temperature above saturation. These objectives were achieved by controlling the following parameters.

1. Pump Speed, 280-1540 rpm.
2. Vent flow rate, 4.5 to 34 kg/hr (10 to 75 lb/hr).
3. Tank mixing requirements.
   a. Four liquid levels, .18, .33, .94, and 1.52 m (7, 13, 37, and 60 inches).
   b. Four wall heat flux rates, 67, 146, 293, and 439 watts (230, 500, 1000, and 1500 Btu/hr).
   c. Flow direction of heat exchange "hot side" discharge, axial or radial.

6.2 TEST TANK

The test tank is a 2.21/1.89 m (87.6/74.5 in.) diameter oblate spheroid, as shown in Figure 6-4, fabricated from 2219-T62 aluminum alloy and is equipped with a 0.61 m (24 in.) diameter access door containing ten 37-pin electrical passthrough fittings. The tank assembly also includes a co-axial vent/fill and drain tube assembly which penetrates the access door. An instrumentation tree mounted internal to the tank is equipped with temperature sensors. Tank total surface area and volume are 14.2 sq m (152 ft^2) and 4.95 cu m (175 ft^3), respectively.

The tank assembly has a 22-layer goldized Kapton Superfloc multilayer insulation blanket section. This blanket is supported from the fairings with pins and interconnected at the seams with rigid "twin pin" fasteners. Individual MLI layers are applied over the vent line and the six tank support struts. The upper flat area of the fairing assembly includes provisions for the vent, purge, and electrical penetrations.

In the original test plan, there were to have been two 22-layer goldized Kapton Superfloc multilayer insulation blankets. As a requirement of the "Design and Development of Pressurization and Repressurization Purge System for Reusable Space Shuttle Multilayer Insulation Systems", contract NAS 8-27419, one of the two layers was removed for inspection following the completion of that testing program.

The one remaining blanket had not been designed to be used alone and did not have sufficient "twin pin" fasteners to securely close the gap at the seams. As a result, the minimum heat leak that could be obtained by the system was 67 watts (230 Btu/hr). Although this is twice the nominal value for which the system was designed it is well within the capability of the system to control pressure. At a vent rate of 17 Kg/hr (37.5 lb/hr), pressure control could be maintained with heat leak to the system up to 2050 watts (7000 Btu/hr) which would require continuous venting.

Figure 6-4. LOX Vent Test Tank with Superfloc Insulation

6.3 TEST FACILITIES

The Site "B" thermal vacuum facility shown in Figures 6-5 and 6-6 was used to perform the vent system evaluation tests in the 2.21 m (87 in.) diameter tank. All fluid supply, vent, and control systems required for the subject tests were in existence and thus available for use during system testing. Figure 6-7 presents a schematic of the test configuration fluid control systems. Guard heat exchangers are used on the instrumentation lines to reduce system heat leaks.

The test chamber is a 3.66 m (12 ft) diameter by 4.89 m (16 ft) vacuum chamber and is serviced by a 81.3 cm (32 in.) oil diffusion pump and an LN_2 cold trap backed by two 14.2 m^3/min (500 ft^3/min) Kinney mechanical vacuum pumps. Controls for these pumps, along with all fluid system control and the data acquisition equipment, are located in a blockhouse approximately 23 m (75 ft) from the test pad.

Ambient temperature GO_2, GHe and GN_2 are available at the test site for use as purge gas and to control tank heat flux.

Figure 6-5. Liquid Hydrogen Test Site

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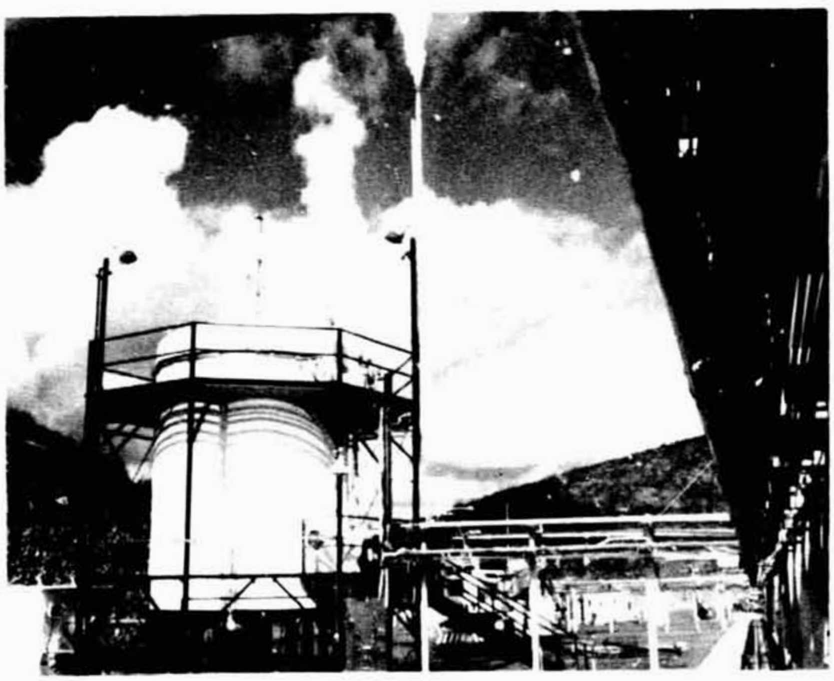
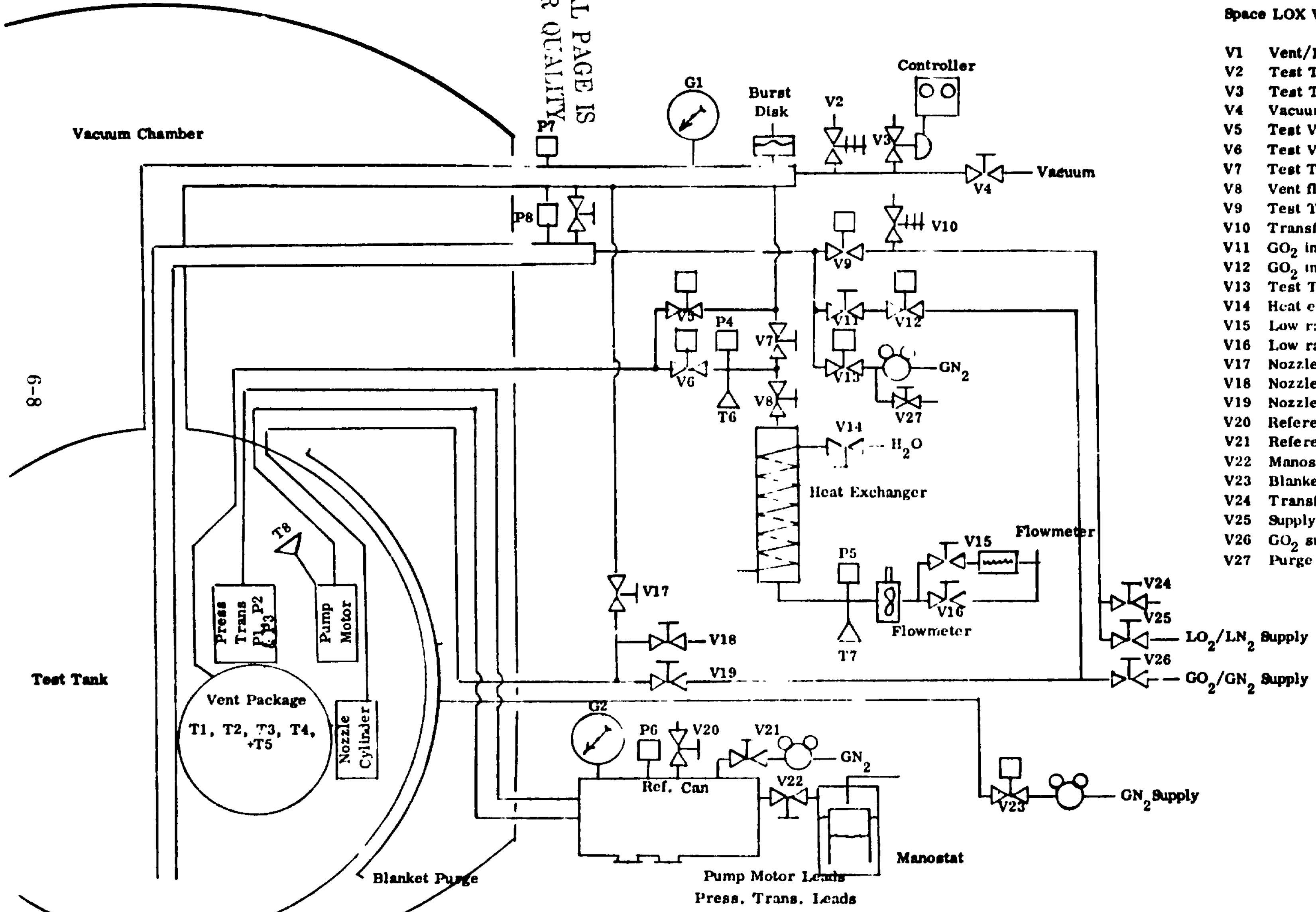


Figure 6-6. Twelve-Foot Diameter Vacuum Chamber



Space LOX Vent Test Setup

Space LOX Vent Setup - Valve Identification

V1	Vent/Fill lines equalization valve
V2	Test Tank relief valve
V3	Test Tank facility vent valve
V4	Vacuum purge valve
V5	Test Vent bypass valve
V6	Test Vent Shutoff valve
V7	Test Tank metered vent valve
V8	Vent flow control valve
V9	Test Tank fill valve
V10	Transfer line relief valve
V11	GO_2 injection flow control valve
V12	GO_2 injection valve
V13	Test Tank GN_2 purge valve
V14	Heat exchanger water supply valve
V15	Low range flowmeter shutoff valve
V16	Low range flowmeter bypass valve
V17	Nozzle cylinder equilization valve
V18	Nozzle cylinder vent valve
V19	Nozzle cylinder pressurization valve
V20	Reference can vent valve
V21	Reference can flow control valve
V22	Manostat shutoff valve
V23	Blanket GN_2 purge valve
V24	Transfer vent valve
V25	Supply tank drain valve
V26	GO_2 supply valve
V27	Purge regulator vent

Figure 6-7. Schematic of Space LOX Vent Test Facilities

6.4 TEST TANK REWORK AND INSTRUMENTATION

The basic test tank was initially designed to be compatible with both LO_2 and LH_2, however, subsequent testing has been oriented specifically to LH_2 that likely resulted in the introduction of non-LO_2 compatible contaminants. The instrumentation tree had to be modified to remove non-LO_2 compatible elements and the tank thoroughly LO_2-cleaned.

The existing instrumentation tree was modified by replacing the Lexan and fiberglass support arms with stainless steel and Teflon and removing the carbon resistors used previously for liquid level sensing. Thirty-two of the existing platinum temperature sensors will remain on the tree as temperature sensors and also to serve as liquid level sensors.

All thermocouples were calibrated at cryogenic temperatures before installation and the data used in the formulation of the data reduction computer program. LO_2 vent flowrates were measured using a turbine flowmeter. Liquid levels were monitored using the platinum temperature sensors and differential pressure in the tank.

Thermocouples and ion gages were used to measure and record test chamber pressures. A Granville-Phillips auto ranging vacuum meter was used for readout and coupling to the recorder.

The 84 channels of data listed in Table 6-1 were recorded digitally as dc voltages with a Dymec 2010 recorder. The dc volts were then converted to engineering units and printed using a Data Systems 620 and a Stromberg-Carlson-4020 computer. The five channels listed below were also recorded on a Sanborn Model 7700 strip chart analog recorder. The analog data was required to obtain the vent system start and stop transients.

Channel No.	Measurement
42	Heat Exch. cold side inlet temp T_2
43	Heat Exch. hot side inlet temp T_3
45	Heat Exch. cold side outlet temp T_5
92	Regulator outlet pressure P_2
97	Test Tank pressure P_7

6.5 LOX VENT SYSTEM EVALUATION TESTING

The basic requirement of the system is to control tank pressure to 310 ±13.8 kN/m^2 (45 ±2 psia) when the external heating rate to the tank is 32.2 to 35.2 watts (110-120 Btu/hr). Control must be maintained with the system operating with either gas or liquid at the pump and/or at the vent inlet. The test program to evaluate the LO_2 vent

Table 6-1. LOX Vent Test Instrumentation

Channel No.	Transducer
11 through 40 (Fluid Temp & Liquid Level)	Platinum resistor, Rosemont Model 118L. Excitation was 0.5 madc when used as temperature probes and 20 madc when used for liquid level
41 through 45 (Fluid Temp)	Platinum resistor, Rosemont Model 150R16
51 through 88 (Fluid & Surface Temp.)	Chromel/Constantan Thermocouple with LH_2 reference junction
91 through 97 (Pressure)	Full bridge strain gage pressure transducer, Statham Model 350
101 (rpm)	6 lobe armature with induction coil
102 (Flow)	Turbine meter, Flow Technology Model FT-12M-GL with induction coil
103 (Flow)	Hot Film constant temp. anemometer, Thermo Systems, Inc., Model 1053/1352-2G
104 and 105 (Vacuum)	Ionization gage tube, Westinghouse Model WL7903 with Ion gage controller. Granville Phillips Model 224020

system was completed during 240-hour period of testing. A total of 94 pressure cycles at 57 different test conditions were completed to define the performance envelope of the vent package. The test parameters included four liquid levels, four total heat fluxes, four pump speeds, four vent rates and two pump discharge flow directions (nozzle vertical, toward the liquid/vapor interface and the nozzle horizontal, toward the tank wall). Table 6-2 presents the test schedule that was accomplished during evaluation testing.

The vent system was able to control test tank pressure for all conditions except the condition of minimum pump speed, vent inlet and heat exchanger in liquid and the pump discharge nozzle in the horizontal position. The vent system vented vapor only for all test conditions except for the condition of minimum pump speed. For these conditions it appeared that the system pulsed some liquid. The minimum pump speed represents a condition of operating at 1/6 design pump speed (280 RPM).

Table 6-2. LO_2 Vent Evaluation Test Schedule

Test	Axial Discharge Position					
Cond. No.	Liquid[(1)] Level	HX Environment	Vent Inlet	Heat Input Btu/hr	Vent Flow lb/hr	Pump Speed rpm
1	1	Liquid	Liquid	500	37.5	1540
2	1	"	"	500	75.0	1540
3	1	"	"	500	5.0	1540
4	1	"	"	500	20.0	1540
5	1	"	"	500	37.5	1080
6	1	"	"	500	37.5	550
7	1	"	"	500	37.5	280
8	1	"	"	250	37.5	1540
9	1	"	"	250	37.5	550
10	1	"	"	250	37.5	280
11	1	"	"	1000	37.5	1540
12	1	"	"	1000	37.5	1080
13	1	"	"	1000	37.5	550
14	1	"	"	1000	37.5	280
15	2	"	"	500	37.5	1540
16	2	"	"	500	37.5	280
17	3	Vapor	"	500	37.5	1800
18	3	"	"	500	75.0	1800
19	3	"	"	500	20.0	1800
20	3	"	"	500	10.0	1800
21	3	"	"	500	37.5	1200
22	3	"	"	500	37.5	600
23	3	"	"	500	37.5	300
24	4	"	Vapor	500	20.0	1800
25	4	"	"	500	20.0	300
26	4	"	"	500	27.5	1800
27	4	"	"	500	10.0	1800
Sp 1	1	Liquid	Liquid	500	20.0	280
Sp 2	1	"	"	1000	75.0	1540
Sp 3	1	"	"	1000	20.0	1540
Sp 4	2	"	"	500	75.0	1540
Sp 5	2	"	"	500	20.0	1540
Sp 6	3	Vapor	"	500	50.0	1800
Sp 7	3	"	"	500	45.0	1800
Sp 8	3	"	"	1000	20.0	1800
Sp 9	3	"	"	1000	20.0	300
Sp 10	3	"	"	1000	27.5	1800
Sp 11	3	"	"	1000	10.0	1800

(1) Liquid Level 1. Above 60, Below 61 3. Above 13, Below 14
2. Above 37.5, Below 38 4. Above 2, Below 7

Table 6-2. Continued

Test Cond. No.	Radial Discharge Position					
	Liquid Level	HX Environment	Vent Inlet	Heat Input Btu/hr	Vent Flow lb/hr	Pump Speed rpm
1	1	Liquid	Liquid	1000	37.5	1540
2	1	"	"	1000	37.5	1080
3	1	"	"	1000	37.[illegible]	550
4	1	"	"	1000	37.5	280
5	1	"	"	500	37.5	1540
6	1	"	"	500	37.5	1080
7	1	"	"	500	37.5	550
8	1	"	"	500	37.5	280
9	2	"	"	500	37.5	1540
10	2	"	"	500	37.5	280
Sp 1	2	"	"	500	75.0	550
Sp 2	2	"	"	500	20.0	550
Sp 3	3	Vapor	"	500	37.5	1800
Sp 4	3	"	"	500	37.5	300
Sp 5	3	"	"	500	37.5	OFF
Sp 6	3	"	"	500	75.0	1800
Sp 7	4	"	"	500	20.0	1800
Sp 8	4	"	Vapor	500	27.5	1800
Sp 9	4	"	"	500	27.5	OFF
Sp 10	4	"	"	500	27.5	600
Sp 11	4	"	"	500	10.0	OFF

Tests were run at four liquid levels as follows:

1. 1.52 m (60 inches) — Vent inlet in liquid, pump and heat exchanger well submerged. in liquid.
2. 0.94 m (37 1/2 inches) — Vent inlet in liquid, pump and heat exchanger barely submerged in liquid.
3. 0.33 m (13 inches) — Vent inlet in liquid, pump and heat exchanger in gas.
4. 0.18 m (7 inches) — Vent inlet in gas, pump and heat exchanger in gas.

Liquid level was determined by using platinum resistance temperature probes as LO_2/GO_2 detectors by increasing the excitation voltage by a factor of three.

Tests were run at various vent flow rates from 4.5 to 34 kg/hr (10 to 75 lb/hr). Minimum flow was limited by the heat input to the test tank at the lowest chamber pressure and maximum flow was limited by the vent system regulator capacity. Flow rate was measured as ambient gas with a turbine meter and/or a hot film anemometer.

Tests were made at various heat leak rates from 67 to 439 watts (230 to 1500 Btu/hr). Since the heat input to the test tank was by conduction through the MLI, it was controlled by regulating the chamber pressure. The heat input was measured using the liquid oxygen in the test tank as a slug calorimeter. At any specific chamber pressure the heat input was determined by locking up the test tank with the pump running to prohibit stratification and timing the tank pressure rise rate which is proportional to the change in liquid temperature.

The minimum heat leak to the tank was 67 watts (230 Btu/hr). Values of 146, 239 and 439 watts (500, 1000 and 1500 Btu/hr) were then selected to provide a significant range of input heat to determine its effect on vent system performance. Although these selected input heat conditions are as much as three times higher than those in the original test plan, they are well within the capability of the vent system which could maintain pressure with input heat as high as 7000 Btu/hr.

Four nominal pump speeds, full, 2/3, 1/3 and 1/6, of the 1800 rpm design speed were used. The frequency and voltage settings for the three phase power supply and the actual pump speeds were as follows:

Power Supply		Actual Pump Speed	
Frequency (hz)	Volts (rms)	In Gas (rpm)	In Liquid (rpm)
60	75	1800	1540
40	50	1200	1080
20	25	600	550
10	15	300	280

The actual pump speed was measured using the speed coil built into the pump. Due to the extremely low signal level from the speed coil, the rpm as measured by the frequency converter and recorder was ±100 rpm and very unreliable below 1000 rpm. The rpms listed above wer measured using an oscilloscope and oscillator to read the speed coil and are ±10 rpm.

6.6 TEST RESULTS

6.6.1 SYSTEM PERFORMANCE. The series of tests defined in Table 6-2 verified the ability of the prototype LO_2 vent system to control tank pressure and vent only superheated vapor over a wide range of operating conditions. The test results which define the ability of the system to control pressure and vent only vapor are summarized in Table 6-3. Data presented are nominal values for pressure switch actuation and deactuation, vent flow rate and heat exchanger vent side outlet temperature and pressure. Values for cycle time off (pressure rise), cycle time on (pressure decay-pump on), liquid level and input heat are also presented.

Test cycles are identified by Run No., test condition and cycle. Run No. identifies the grouping of digital data that was reduced in one computer run. The test condition is that defined in Table 6-2, and the cycle number is the number of cycles repeated at a given test condition. For all cases except Run No. 13, test condition 8, the vent system was able to control pressure. For all cases except 11 cycles throughout the test, the vent system was clearly able to vent vapor only. The 11 exceptions were where the nominal heat exchanger vent side outlet temperature was less than 100°K (180°R). These conditions will be examined in detail in following sections.

The boundaries of the performance envelope for the vent system are partially defined by the data presented in Figures 6-8 and 6-9. Figure 6-8 depicts the vent system performance for the design condition of 17 kg/hr (37.5 lb/hr LO_2) and 1540 rpm pump speed at liquid level 1, and heat input of 146 watts (500 Btu/hr). The initial rapid pressure decay is due to mixing of the tank fluid followed by the removal of accumulated input heat from the LO_2 bulk. The vent fluid is throttled into the heat exchanger with a resultant exit pressure of 150.4 kN/m^2 (21.8 psia). The temperature at heat exchanger vent side outlet shows 102°K (183.6°R), well above the oxygen saturation temperature at 150.4 kN/m^2 (21.8 psia).

Figure 6-9 depicts the vent system performance for the test condition that the vent system did not control pressure (Run 13, condition 8). After the vent system actuated, the pressure continued to rise with no mixing at the pump speed of 280 rpm. The vent flow was throttled down to 152.5 kN/m^2 (22.1 psia) but the pump did not establish sufficient flow to the heat exchanger hot side to fully vaporize the vent flow. At 17 minutes into the run, the vent flow was reduced to 9.1 kg/hr (20 lb/hr). At this reduced vent flow rate there was sufficient heat exchange to vent only vapor. The pressure continued to rise even though energy was being removed at a higher rate than it was being input. At 35 minutes into this cycle the vent flow was increased to 12 kg/hr (26.5 lb/hr) and the system still vented only vapor, but without mixing to break up the liquid/vapor interface, the pressure continued to rise. At 47 minutes into the run, the pump speed was increased to 1540 rpm and the vent flow rate increased to 34 kg/hr (75 lb/hr). At this condition the tank quickly mixed and pressure rapidly decayed to the deactuation pressure since the input heat had already been removed from the bulk of the LO_2.

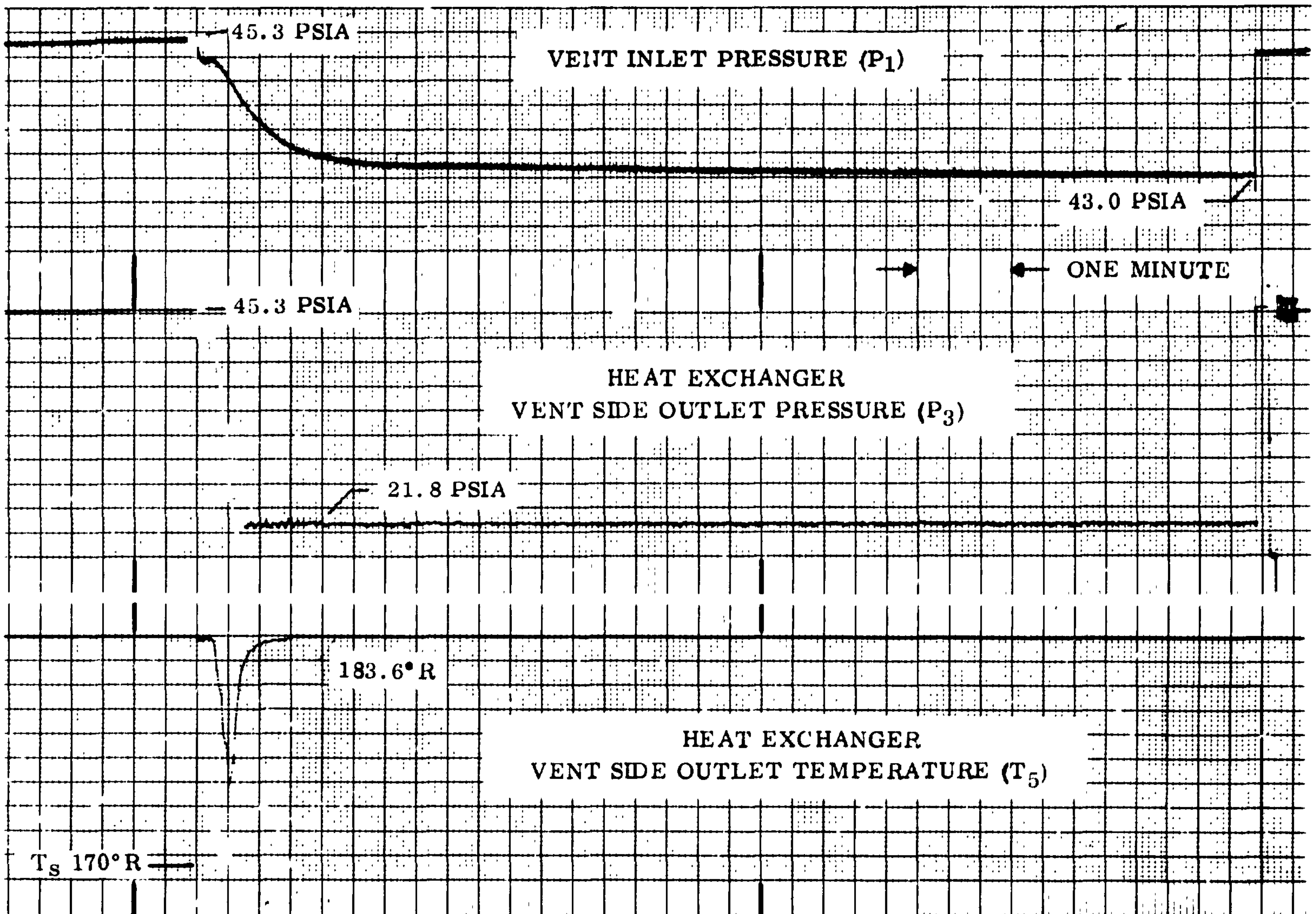
These two test runs demonstrate the effectiveness of the thermodynamic liquid/vapor separator as a system for venting vapor only from an oxygen tank subjected to

Table 6-3. LO_2 Vent System Evaluation Test Results (Axial Discharge Position)

Run No.	Test Condition	Cycle No.	Actuation Press. psia	Deactu. Press. psia	Cycle Time Off Min.	Cycle Time On Min.	Pump Speed RPM	Vent Flow lb/hr	Liquid Level In. Above	Liquid Level In. Below	Hx Outlet Temp °F	Hx Outlet Press psia	Input Heat Btu/hr
9	8	1	45.34	42.93	42	1	NR	34.5	60	61	-282	21.8	250
		2	45.51	43.23	58	5	NR	34.5	60	61	-276	21.9	250
		3	45.50	43.28	77	7	NR	35.5	60	61	-276	21.8	250
	9	1	45.48	43.25	110	6	NR	35.5	60	61	-279	21.9	250
	10	1	45.48	43.43	144	19	NR	36.0	60	61	-288	22.0	250
	8	1	45.33	43.21	81	1	1540	56.5	60	61	-276	NR	250
10	17	1	45.41	43.53	34	9	NR	37.5	13	14	-258	21.3	500
		2	45.06	42.31	36	[illegible]	NR	41.0	13	14	-259	21.0	500
		3	45.22	42.36	35	10	NR	40.0	13	14	-259	20.8	500
		4	44.33	42.41	NR	NR	NR	41.0	13	14	-263	21.0	500
	18	1	45.17	42.45	27	5	1805	77.0	13	14	-267	19.4	500
	19	1	45.21	42.29	36	25	1815	22.5	13	14	-257	22.0	500
	23	1	45.3[illegible]	42.69	35	9	NR	38.0	13	14	-267	21.1	500
	22	1	45.33	42.51	40	11	500	37.5	13	14	-265	21.2	500
	21	1	45.41	42.54	39	11	1207	37.5	13	14	-265	21.3	500
	Spec-6	1	45.34	42.36	40	6	1811	52.5	13	14	-269	20.0	500
		2	45.37	42.32	40	6	1811	51.5	13	14	-272	20.4	500
	Spec-7	1	45.39	42.59	40	9	1811	45.5	13	14	-271	20.8	500
	17	1	45.32	42.62	42	12	1811	37.5	13	14	-270	21.3	500
11	Spec-8	1	45.18	42.97	23	23	1811	21.0	2	7	-276	19.2	1000
		2	45.14	42.99	25	27	1811	21.0	2	7	-267	18.9	1000
		3	45.05	42.87	25	27	1808	20.0	2	7	-263	1[illegible].0	1000
	Spec-9	1	45.01	43.20	24	24	NR	20.0	2	7	-267	19.3	1000
	Spec-10	1	44.68	42.80	25	11	1809	29.5	2	7	-150	15.6	1000
	Spec-11	1	45.35	45.02	25	-	1810	10.0	2	7	-142	21.0	1000
	24	1	45.35	42.95	-	17	1810	20.0	2	7	-144	19.0	500
	25	1	45.25	43.03	37	18	NR	20.5	2	7	-157	19.0	500
	2	1	45.24	43.31	40	10	1812	29.5	2	7	-151	15.7	500
	27	1	45.29	43.31	39	107	1812	9.5	2	7	-132	20.8	500
	24	1	45.36	43.13	42	16	1810	11.5	2	7	-146	19.0	500
12	1	1	45.49	42.96	174	41	1540	38.5	60	61	-275	21.7	1500
	Spec-1	1	45.41	43.12	44	90	650/1075	37.5	60	61	-289/-276	21.7	1500
	2	1	45.35	43.30	43	54	1075	39.0	60	61	-276	27.0	1500
	1	1	45.12	43.42	41	56	1540	37.5	60	61	-276	22.0	1500
		2	45.16	43.26	46	41	1540	37.5	60	60	-276	22.0	1500

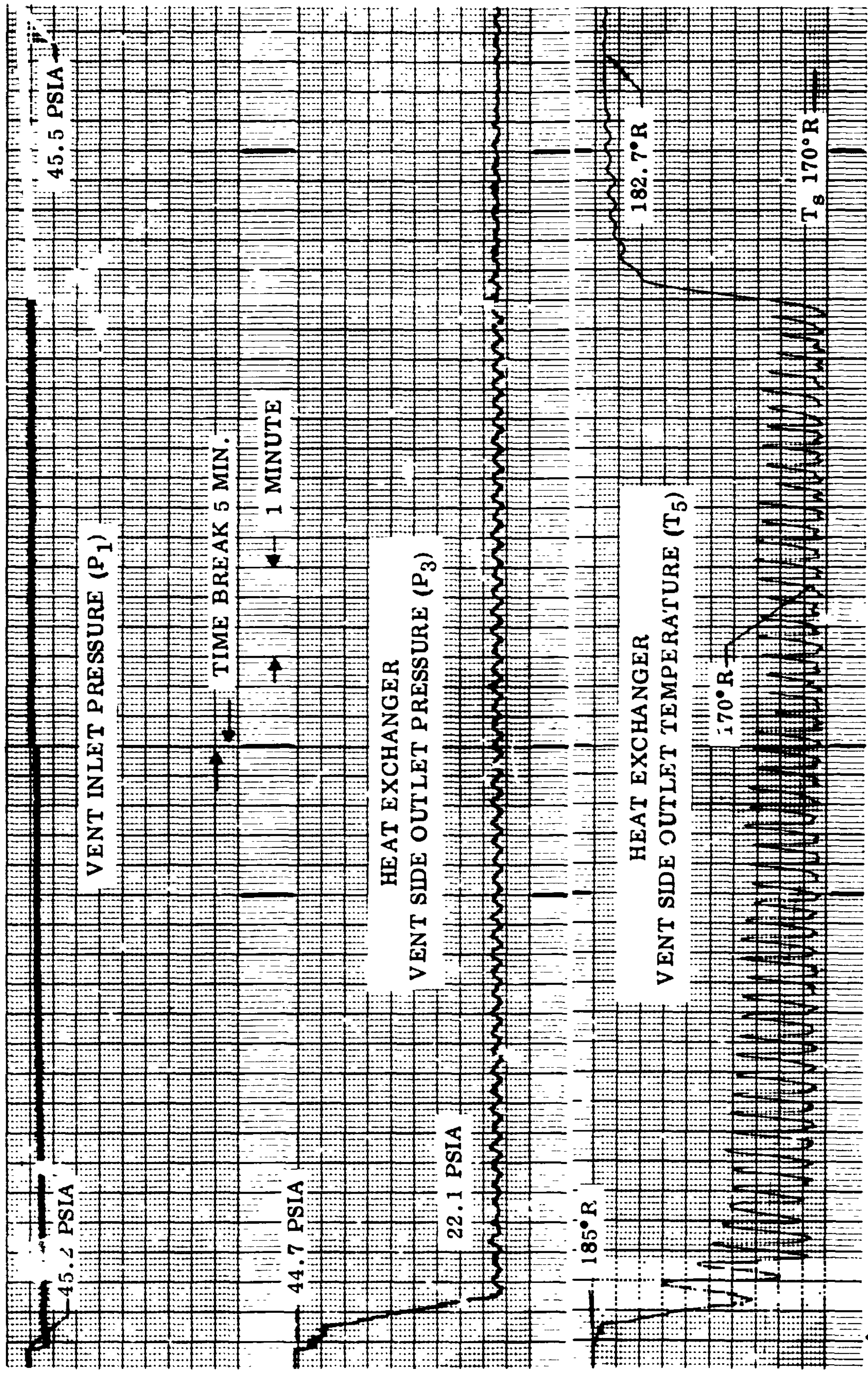
Table 6-3. LO_2 Vent System Evaluation Test Results (Radial Discharge Position)

Run No.	Test Cond.	Cycle No.	Actuation Press. psia	Deact. Press. psia	Cycle Off Min	Time On Min	Pump Speed RPM	Vent Flow lb/hr	Liquid Level, In Above	Liquid Level, In Below	Hx Outlet Temp. °F	Hx Outlet Press. psia	Input Heat Btu/hr
13	5	1	45.50	43.20	56	28	1535	39.0	60	61	-276	21.5	500
		2	45.31	43.03	64	33	1537	33.5	60	61	-276	21.8	500
	6	1	45.26	43.10	67	33	1072	35.0	60	61	-276	21.6	500
	7	1	45.26	42.97	68	36	400	33.5	60	61	-279	21.6	500
	8	1	45.17	46.18	60	51	NR	35/25	60	61	-289	22.0	500
	5	1	45.29	43.21	78	11	1540	37.5	60	61	-276	21.6	1000
	1	1	45.32	43.21	54	48	1537	37.5	30	61	-276	21.6	1000
	3	1	45.26	NR	63	63	190	36.5	60	61	-278	22.0	1000
14	9	1	NR	NR	83	24	NR	37.5	37.5	38	-276	NR	500
		2	NR	NR	99	26	NR	37.5	37.5	38	-274	NR	500
		3	45.37	43.11	99	29	NR	37.5	37.5	38	-275	21.5	500
	10	1	45.34	43.06	101	37	1536	36.5	37.5	38	-289/-276	22.0	500
	7	1	45.29	43.24	119	23	380	37.5	37.5	38	-279	22.0	500
	Spec-1	1	45.42	43.20	110	10	200	79.5	37.5	38	-291	20.4	500
	Spec-2	1	45.32	OFF	86	-	NR	25.0	37.5	38	-275	22.2	500
15	Spec-3	1	45.13	43.57	41	15	1800	33.5	12	13	-256	21.5	500
		2	45.11	42.67	36	14	1812	32.5	12	13	-259	21.5	500
		3	45.01	43.11	36	11	1810	37.5	12	13	-259	21.5	500
		4	45.09	43.23	34	12	1811	36.0	12	13	-259	21.4	500
	Spec-4	1	45.27	42.81	34	11	NR	36.0	12	13	-267	21.5	500
	Spec-5	1	45.51	42.60	37	12	OFF	36.5	12	13	-268	21.2	500
	Spec-3	1	45.41	42.73	37	12	1811	36.0	12	13	-257	21.4	500
	Spec-6	1	45.48	43.00	33	5	1811	76.0	12	13	-267	19.5	500
	Spec-7	1	45.21	43.13	[illegible]	26	1812	20.5	2	7	-222	19.9	500
		2	45.15	43.31		20	1811	21.0	2	7	-214	20.0	500
	Spec-8	1	45.26	43.26	40	10	1811	33.0	2	7	-211	16.3	500
	Spec-9	1	45.37	43.02	39	10	OFF	33.0	2	7	-217	16.3	500
	Spec-10	1	45.41	43.07	44	10	500	33.0	2	7	-217	16.2	500
	Spec-11	1	45.39	43.41	44	77	OFF	10.0	2	7	-205	21.6	500
	Spec-7	1	45.44	43.35	49	17	1812	21.5	2	7	-195	19.8	500



VENT ON 17:53 — TEST CONDITIONS: PUMP SPEED 1540 RPM, LIQUID LEVEL 1, VENT FLOW RATE 37.5 LB/HR, INPUT HEAT 500 BTU/HR — 18:04

Figure 6-8. Vent System Performance Run No. 13, Condition No. 5



15:49 TEST

16:06 VENT FLOW 20 LB/HR

CONDITIONS: PUMP SPEED 280 RPM, VENT FLOW RATE 37.5 LB/HR, LIQUID LEVEL 1, INPUT HEAT 500 BTU/hr

Figure 6-9. Vent System Performance Run No. 13, Condition No. 8

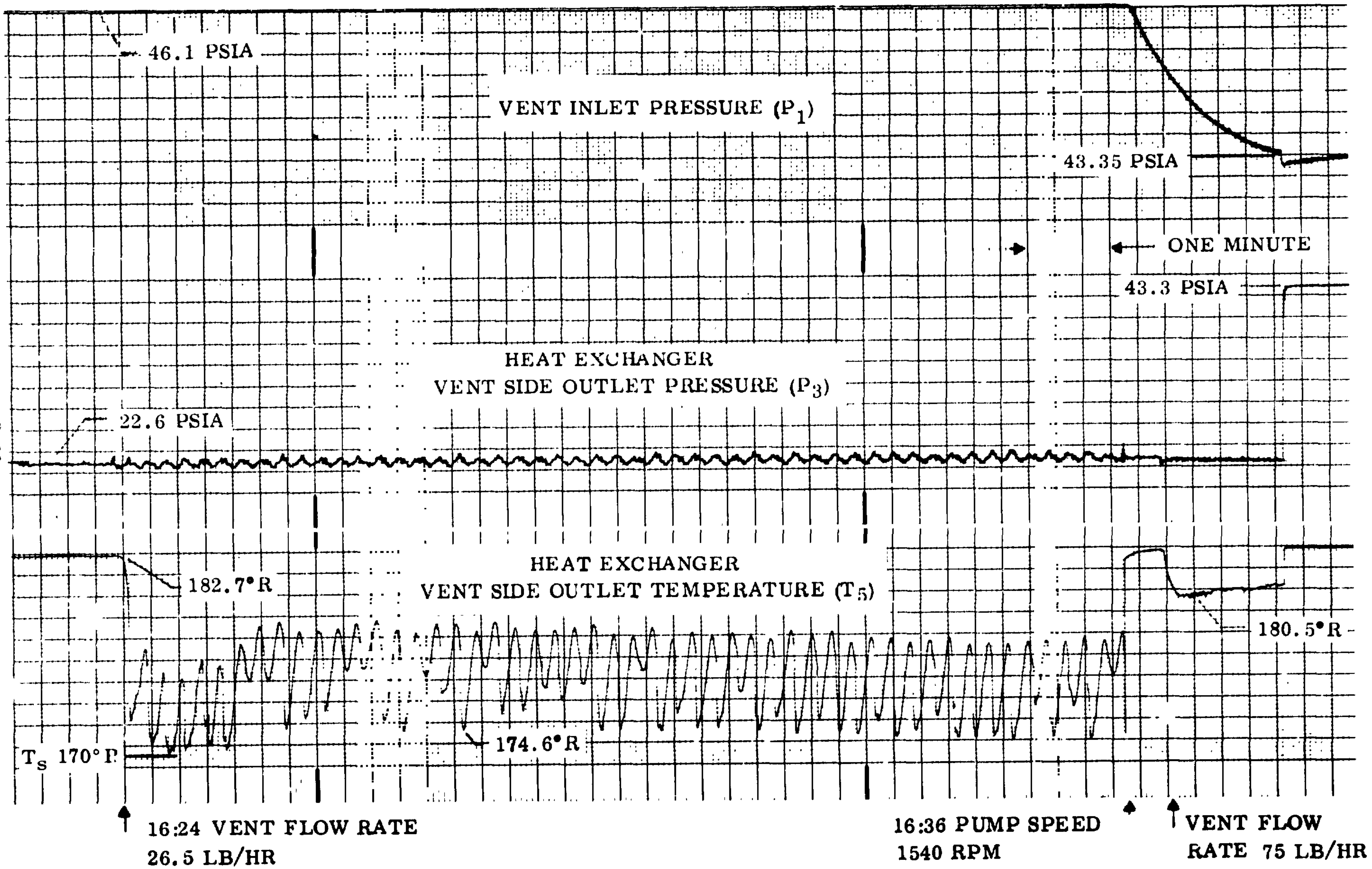


Figure 6-9. (Contd) Vent System Performance Run No. 13, Condition No. 8

orbital heating rates. It also showed the importance of tank fluid mixing and liquid-ullage coupling in controlling tank pressure.

6.6.2 SYSTEM PERFORMANCE BOUNDARIES. The liquid level, vent flow rate, heat exchange hot side flow rate and mixing jet discharge direction were varied in accordance with Table 6-2 to determine the effect on system performance.

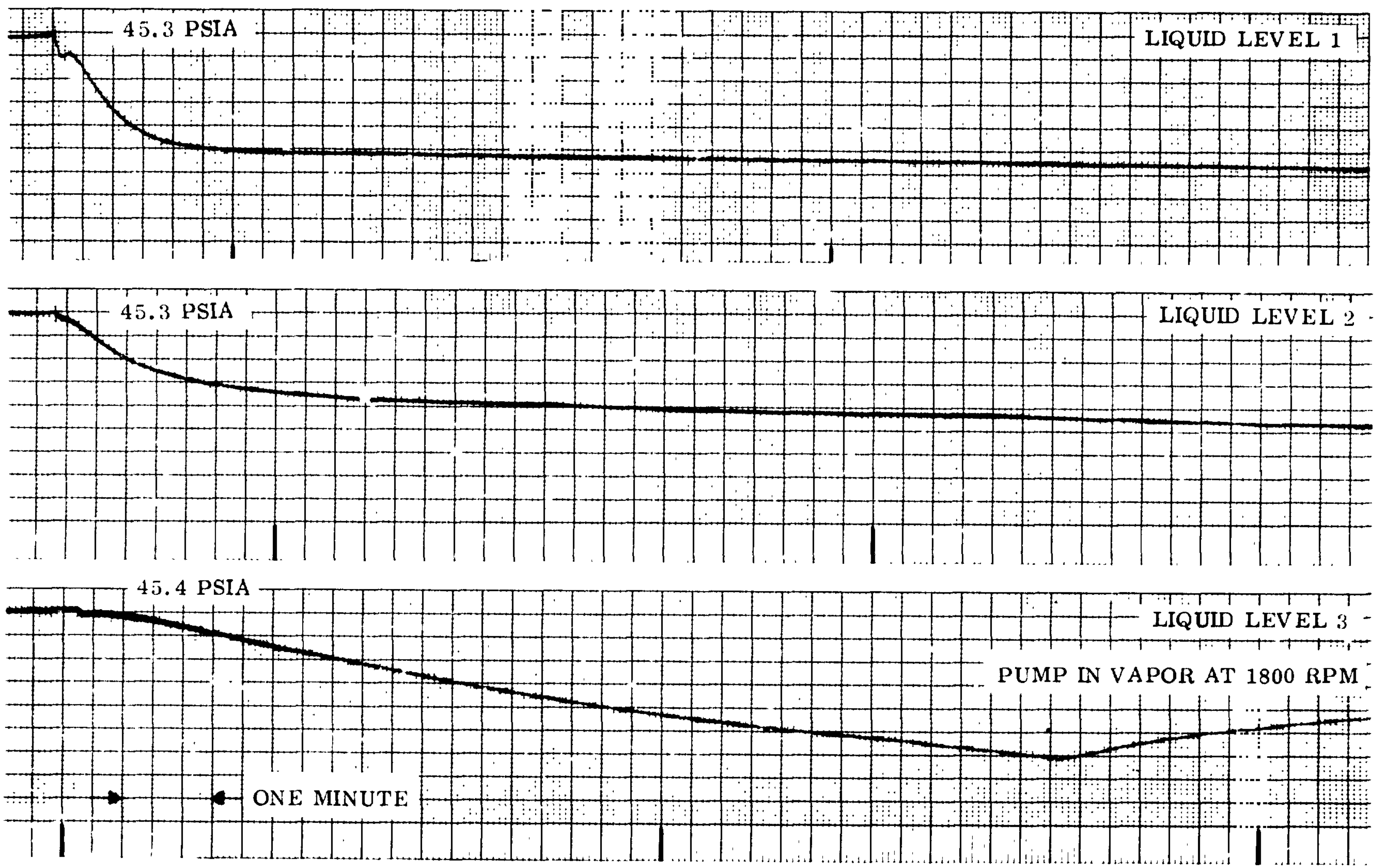
Four liquid levels were used during evaluation testing to determine the effect of liquid level on system performance and to operate the vent package both in liquid and gas. Two liquid levels were above the heat exchanger discharge nozzle and two were below the heat exchanger hot side inlet. The heat exchanger hot side discharge is at the level of .61 m (24 inches) above the tank bottom. The heat exchange hot side inlet is at .61 m (24 inches) and the vent inlet is at .28 m (11 inches).

The change in pressure decay as a function of liquid level is shown in Figures 6-10 and 6-11. Liquid level 1 is at 1.52 m (60 inches) to .91 m (36 inches) above the mixing jet outlet, level 2 is at .95 m (37.5 inches) to .34 m (13.5 inches) above the jet outlet, liquid level 3 is at .33 m (13 inches) below the heat exchanger inlet but above the vent inlet and liquid level 4 is at .18 m (7 inches) below both the heat exchanger and vent inlet. With decreasing liquid level and increasing ullage volume, the initial pressure drop due to fluid mixing is damped until the heat exchanger is in vapor at liquid levels 3 and 4, the pressure decay is linear. Time to vent is as follows: level 1, 28 min.; level 2, 26 min.; level 3, 11 min.; level 4, 17 min.

With the heat exchanger in liquid, liquid levels 1 and 2, the fluid is mixed by the discharge of the heat exchanger hot side fluid. At liquid level 2, the fluid is mixed more rapidly. But, since the pressure decay rate is dominated by ullage effects the initial decay in pressure is not as rapid since the ullage volume is larger. However, the steady stage pressure decay rate is greater at liquid level 2 and since there is less bulk fluid heat to be removed, the time to vent is less.

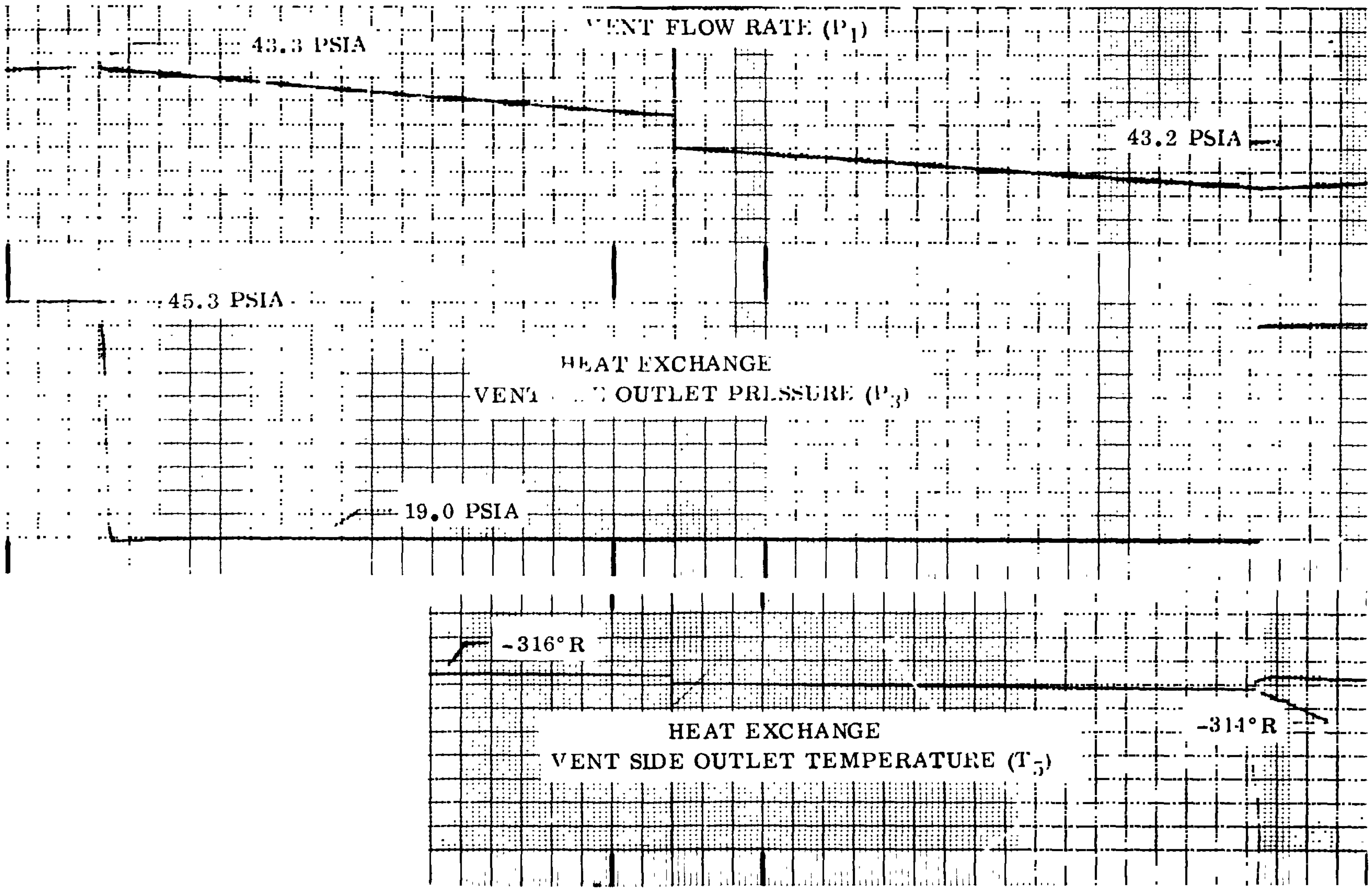
At liquid levels 3 and 4 the heat exchanger is in vapor and there is no fluid mixing. The thermodynamic cooling ability of the vent system is available at liquid level 3 since the vent inlet continues in liquid. This has the effect of cooling the ullage and producing the shortest time to vent. There is sufficient heat transfer from the ullage to continue to vent superheated vapor, as can be seen in Figure 6-11.

The performance of the system in providing mixing at various pump speeds is depicted in Figure 6-12. Mixing was accomplished at the noted pump speeds with the heat exchanger hot side flow discharge directed both towards the liquid/vapor interface (axially) and in the radial direction (rotated). With decreasing pump speed, the fluid mixing time increased. Also, it can be seen that the mixing time is longer in the radial position and that change of pump speed has a stronger effect on mixing time when the nozzle is in the radial discharge position.



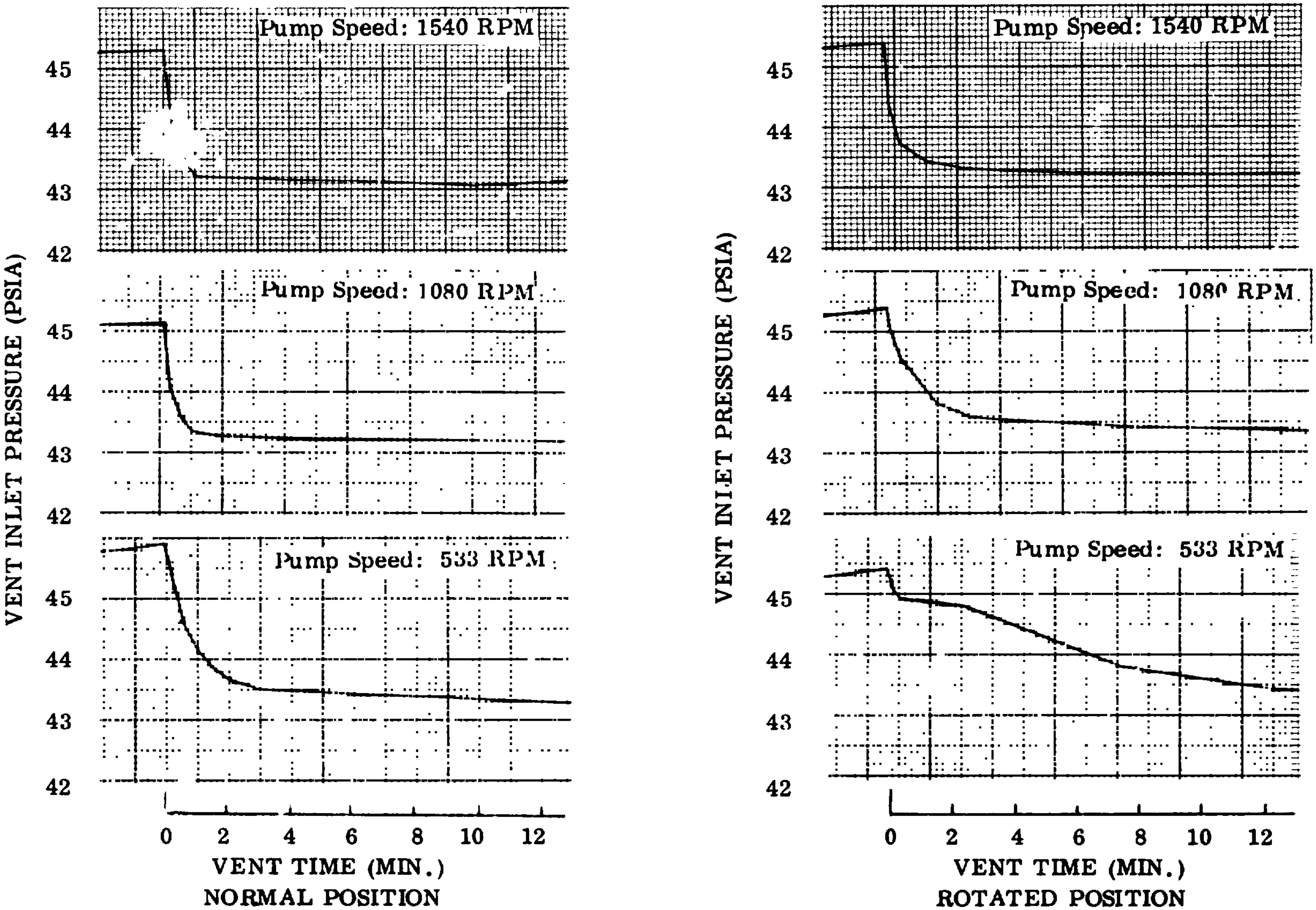
TEST CONDITIONS: PUMP SPEED, 1540 RPM
VENT FLOW RATE, 3.75 LB/HR
LIQUID LEVEL NOTED
INPUT HEAT, 1000 BTU/HR

Figure 6-10. Effect of Change of Liquid Level on Vent Inlet Pressure Decay



TEST CONDITIONS: PUMP SPEED 1800 RPM LIQUID LEVEL 4
VENT FLOW RATE 20 LB/HR INPUT HEAT 500 BTU/HR

Figure 6-11. Vent System Performance Run No. 11 Condition 24



TEST CONDITIONS: PUMP SPEED - NOTED, LIQUID LEVEL 1, VENT FLOW RATE 7.5 SCFM

Figure 6-12. Effect of Change in Pump Speed on Vent Inlet Pressure Decay

The performance of the system in venting superheated vapor at various pump speeds is presented in Figures 6-13 thru 6-16. At 1540 and 1080 rpm, Figures 6-13 and 6-14, the heat exchanger superheats the vent flow (22 psia) to the bulk fluid temperature. At the pump speed of 550 rpm, Figure 6-15, there is a reduction in the effectiveness of the heat exchanger with the vent flow approximately five degrees below the bulk temperature but still with nine degrees of superheat. As can be seen in Figure 6-16 and as was pointed out in Figure 6-9, at a pump speed of 280 rpm the system vents saturated oxygen which may contain some quantity of liquid. This performance is dependent only on pump speed, operating pressure and vent flow rate. It is independent of heating rate and liquid level as long as the heat exchanger and vent inlet are both in liquid.

For the case at liquid level 3 (heat exchanger inlet in vapor-vent inlet in liquid), there was no effect on ability to vent superheated vapor with pump speed even with the pump off. The heat exchanger primarily transferred heat from the shroud to the ullage at liquid level 3.

The performance of the system in venting superheated vapor at various vent flow rates is presented in Figures 6-17 through 6-19. These figures show the performance of the system at liquid level 2, input heat of 146 watts (500 Btu/hr) and pump speed of 550 rpm. At pump speeds of 1080 and 1540 rpm the system vented superheated vapor over the full range of vent flow rates tested. At 550 rpm, with the vent flow rate of 17 kg (37.5 lb/hr), Figure 6-17, the system vented superheated vapor at a temperature approaching the bulk fluid temperature. When the flowrate was reduced to 9.1 kg/hr (20 lb/hr) the vent fluid temperature came up to the bulk liquid temperature, approximately 8.3°K (15°R) superheated. At a vent flow rate of 34 kg/hr (75 lb/hr), the vent system was venting saturated fluid as shown in Figure 6-19. At the cold side flow rate of 34 kg/hr (75 lb/hr), there was insufficient hot side flow to produce the heat exchanger effectiveness necessary to vent superheated vapor. The system pressure decay time was approximately proportional to the changes in vent flow rate so apparently or no liquid oxygen was being vented.

The effect of change in vent flow rate at liquid level 3 is presented in Figures 6-20 and 6-21. In all cases the system was able vent superheated vapor. The only effects of doubling the vent flow rate from 17 kg/hr (37.5 lb/hr) to 34 kg/hr (75 lb/hr) was to reduce the vent exit temperature and to decrease vent time.

6.6.3 <u>HEAT EXCHANGER PERFORMANCE</u>. In order to obtain a quantitative indication of heat exchanger thermal performance as a function of flow rate, heat exchanger effectiveness, $T_{co} - T_{ci}/T_{Hi} - T_{ci}$ is used. Heat exchanger effectiveness is plotted against vent side flow rate in Figure 6-22 for liquid oxygen on the hot side and in Figure 6-23 for gaseous oxygen on the hot side. In both cases, the vent side inlet was liquid oxygen. As shown in Figure 6-22, the actual effectiveness exceeded the effectiveness calculated for the design performance.

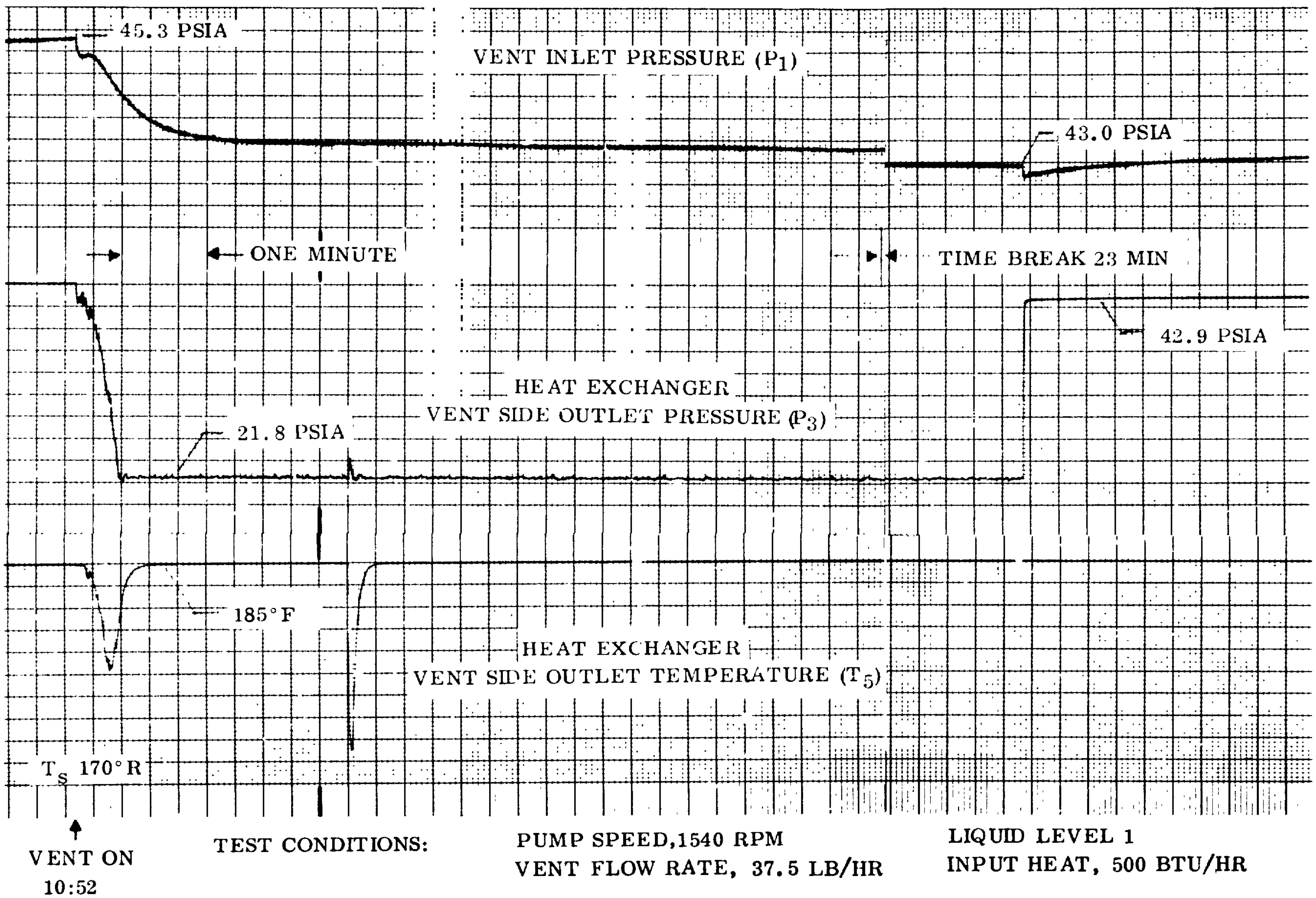
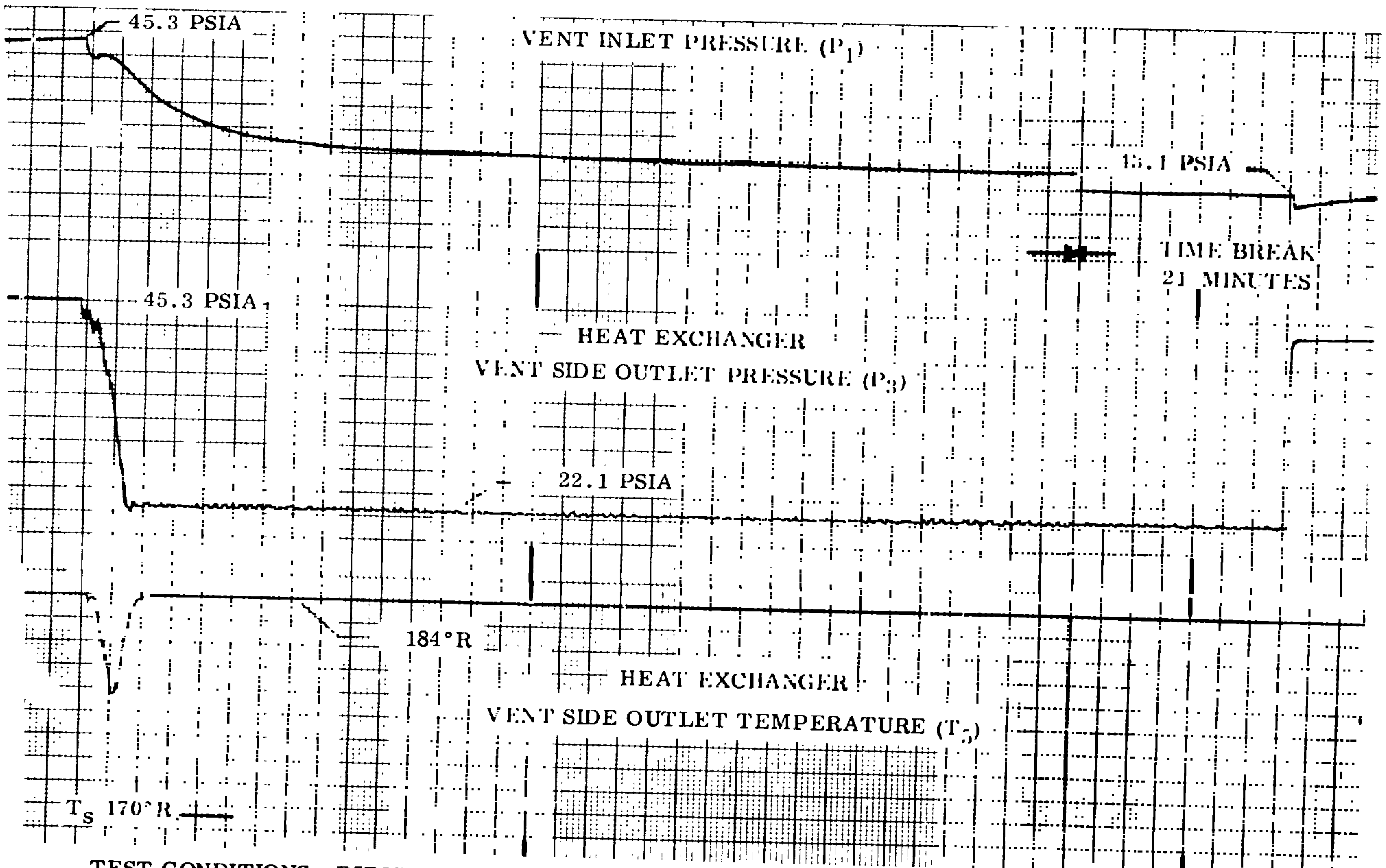
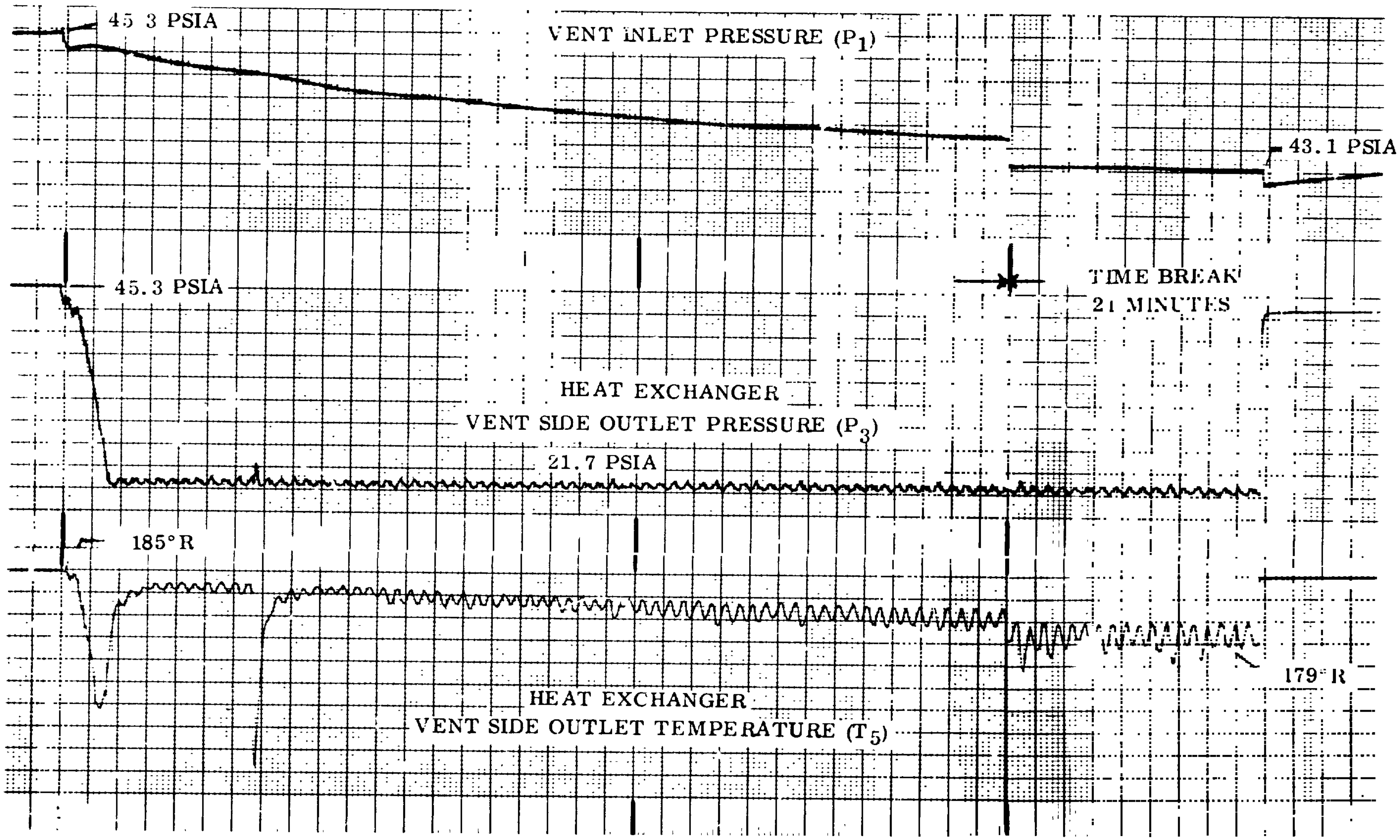


Figure 6-13. Vent System Performance Run No. 13, Condition No. 5, Cycle 2



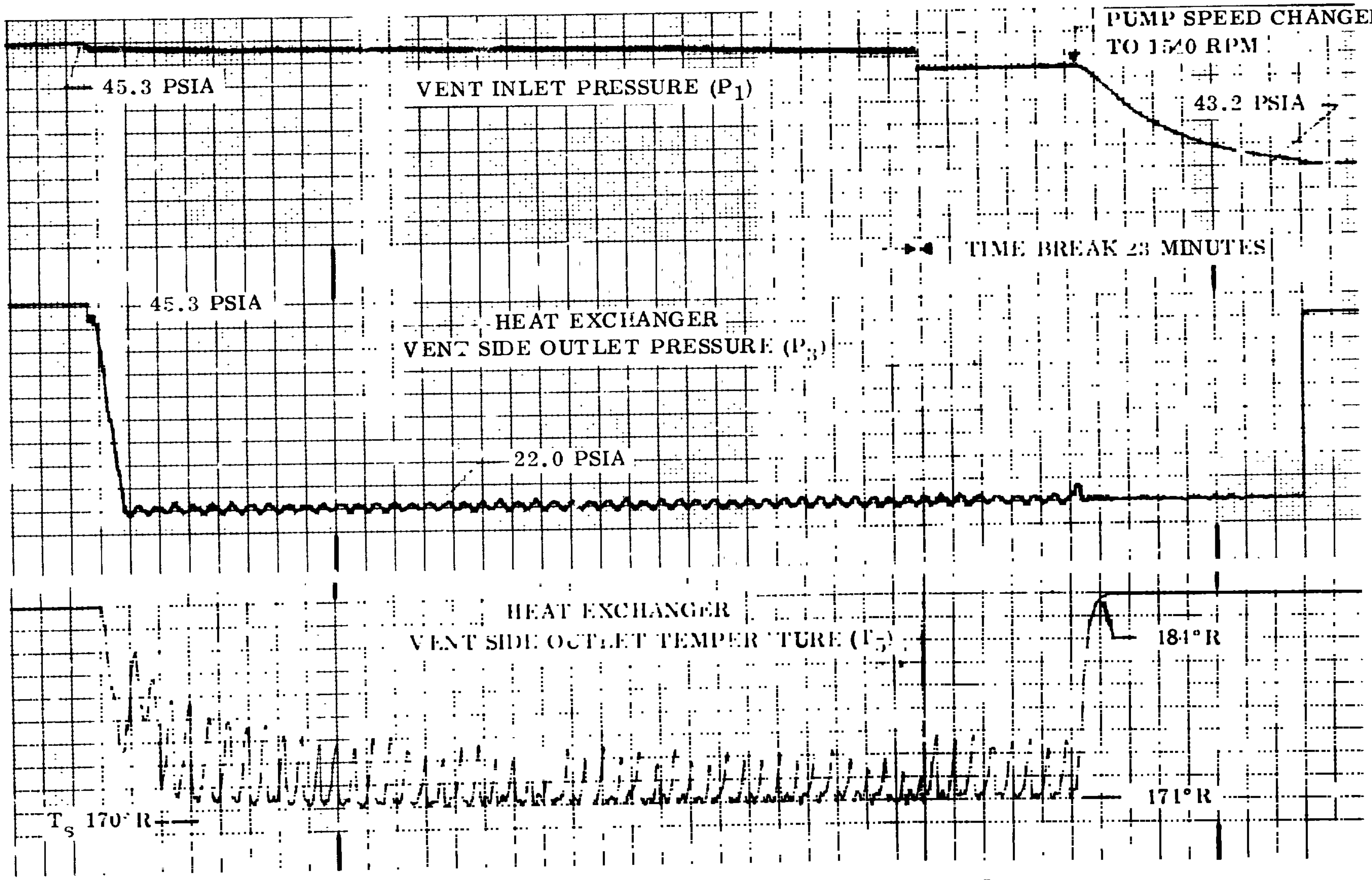
TEST CONDITIONS: PUMP SPEED, 1080 RPM LIQUID LEVEL 1
VENT FLOW RATE, 37.5 LB/HR INPUT HEAT, 500 BTU/HR

Figure 6-14. Vent System Performance Run No. 13, Condition No. 6



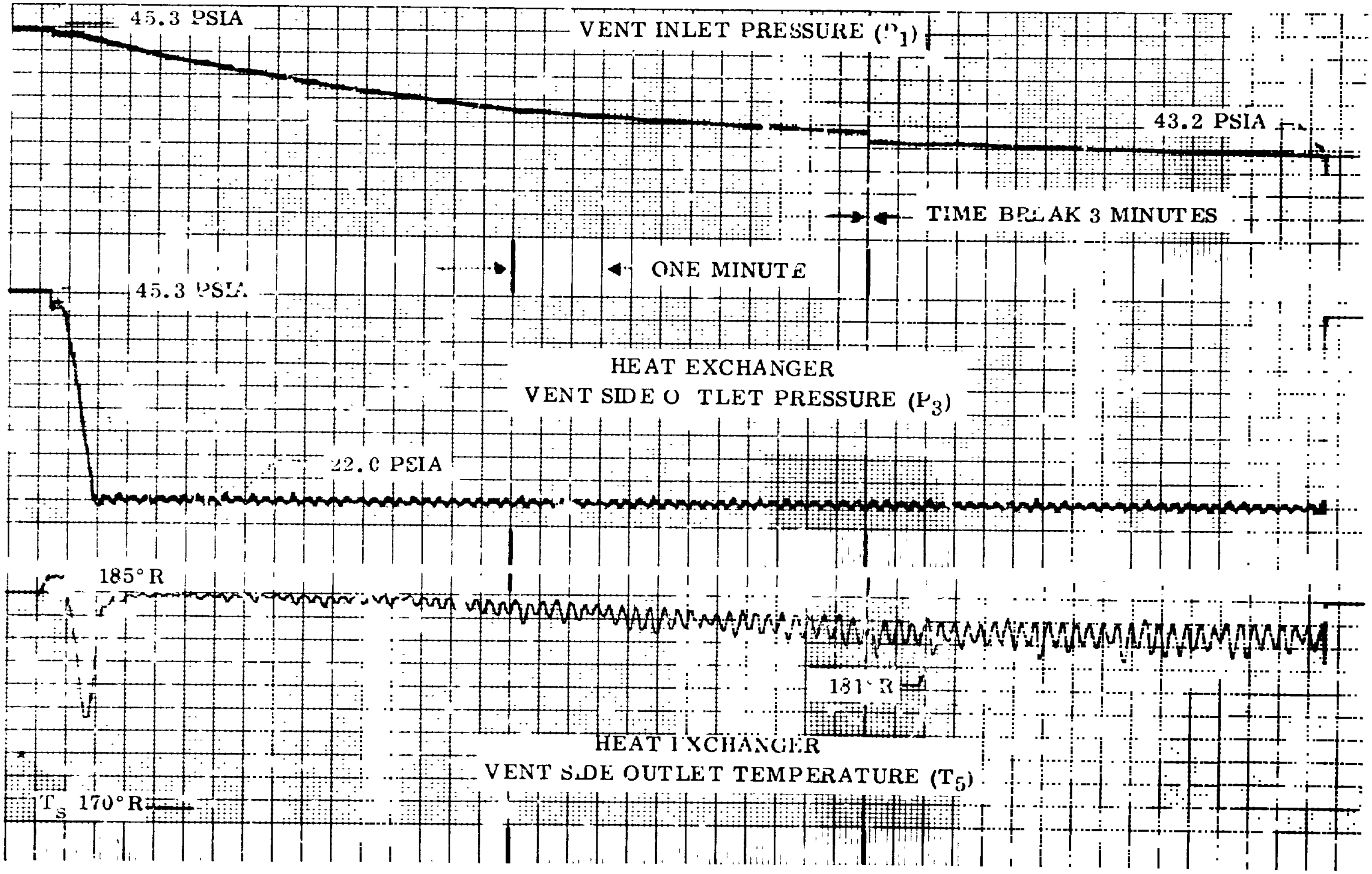
TEST CONDITIONS: [illegible] P SPEED, 550 RPM LIQUID LEVEL 1
VENT FLOW RATE, 37.5 LB/HR INPUT HEAT, 500 BTU/HR

Figure 6-15. Vent System Performance Run No. 13, Condition No. 7



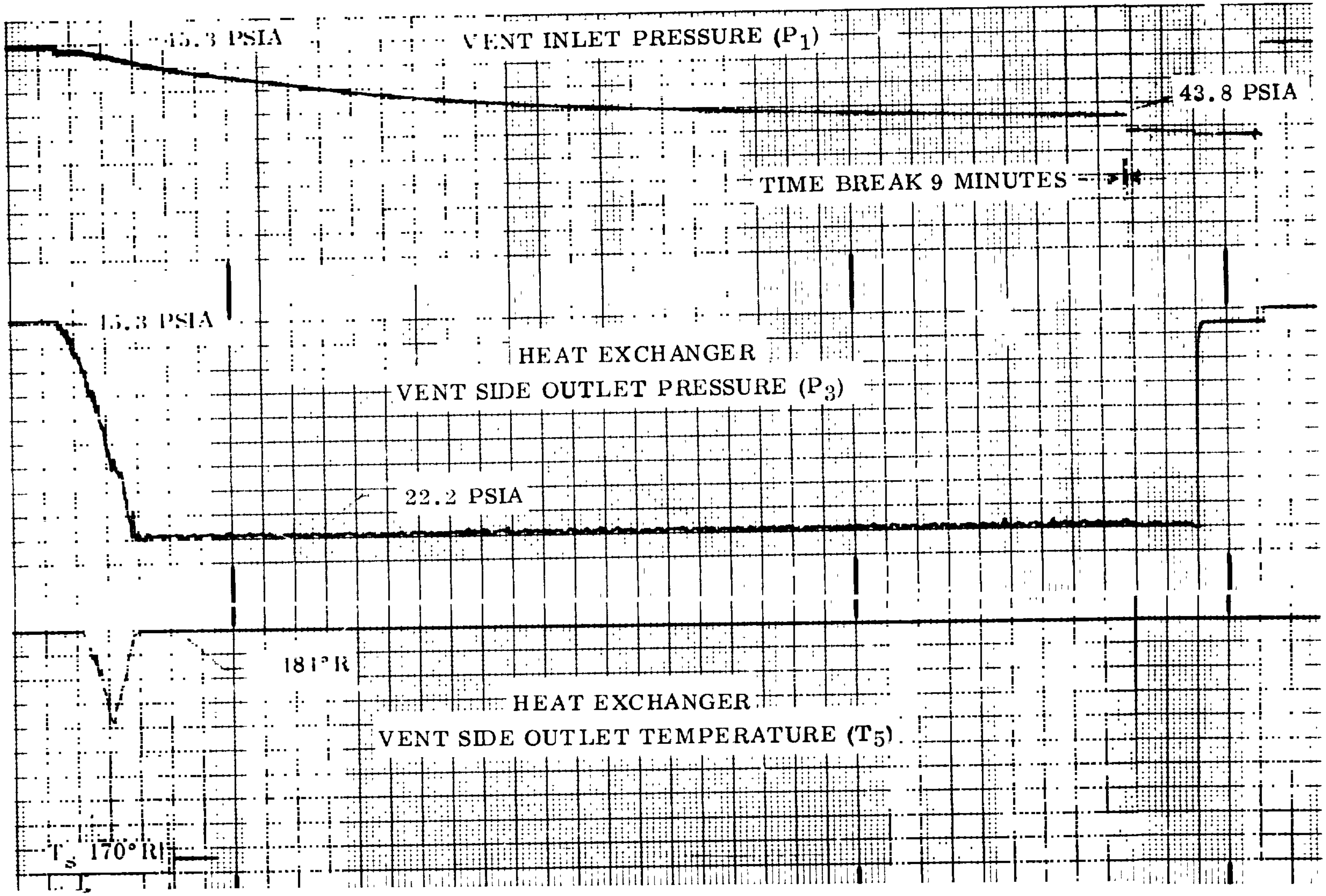
TEST CONDITIONS: PUMP SPEED, 280 RPM
VENT FLOW RATE, 7.5 LB/HR
LIQUID LEVEL 2
INPUT HEAT, 500 BTU/HR

Figure 6-16. Vent System Performance Run No. 14, Condition No. 10



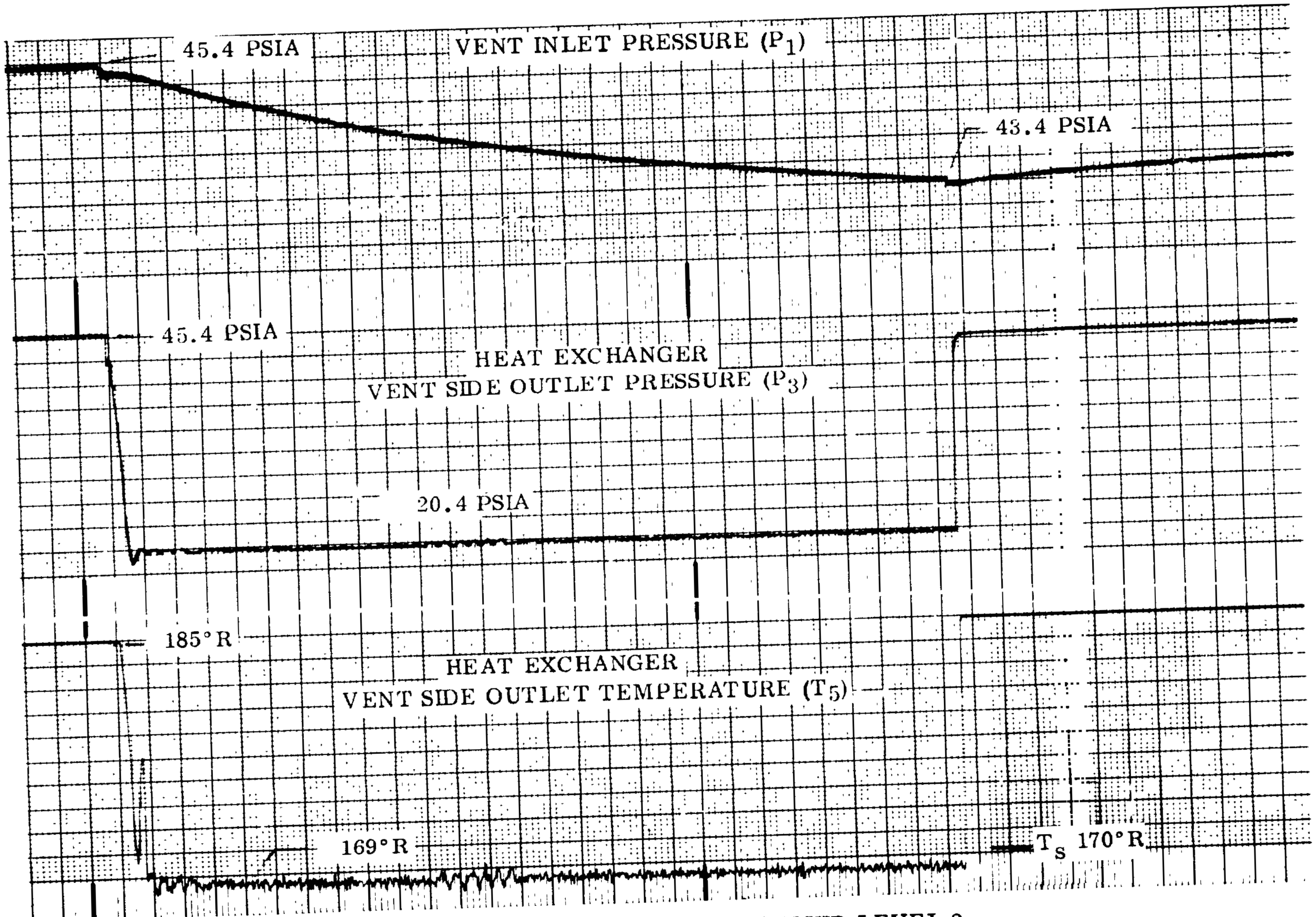
TEST CONDITIONS: PUMP SPEED, 550 RPM LIQUID LEVEL 2
VENT FLOW RATE, 37.5 LB/HR INPUT HEAT, 500 BTU/HR

Figure 6-17. Vent System Performance Run No. 14, Condition No. 7



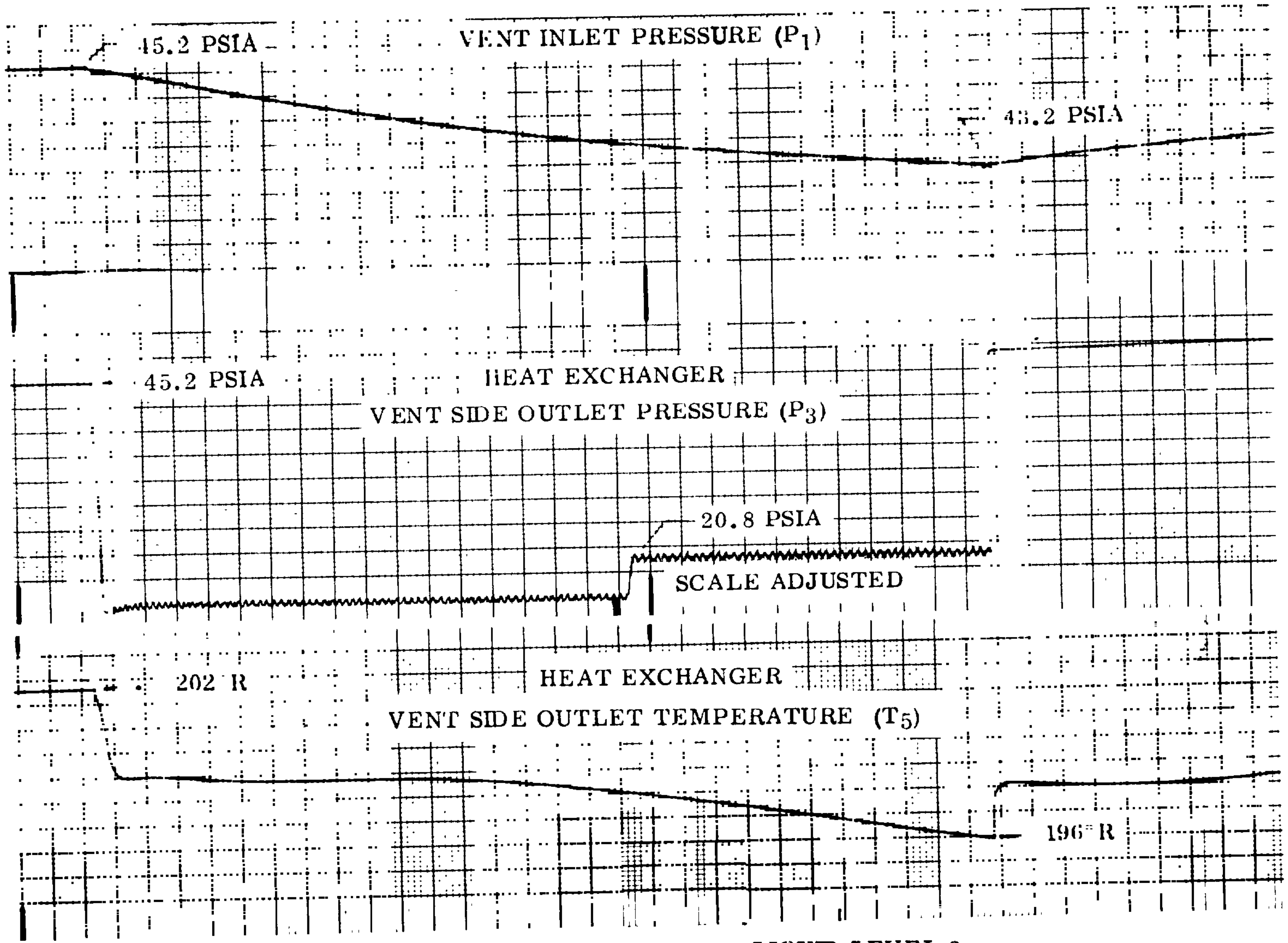
TEST CONDITIONS: PUMP SPEED, 550 RPM — LIQUID LEVEL 2
VENT FLOW RATE, 20 LB/HR — INPUT HEAT, 500 BTU/HR

Figure 6-18. Vent System Performance Run No. 14, Condition Spec 2



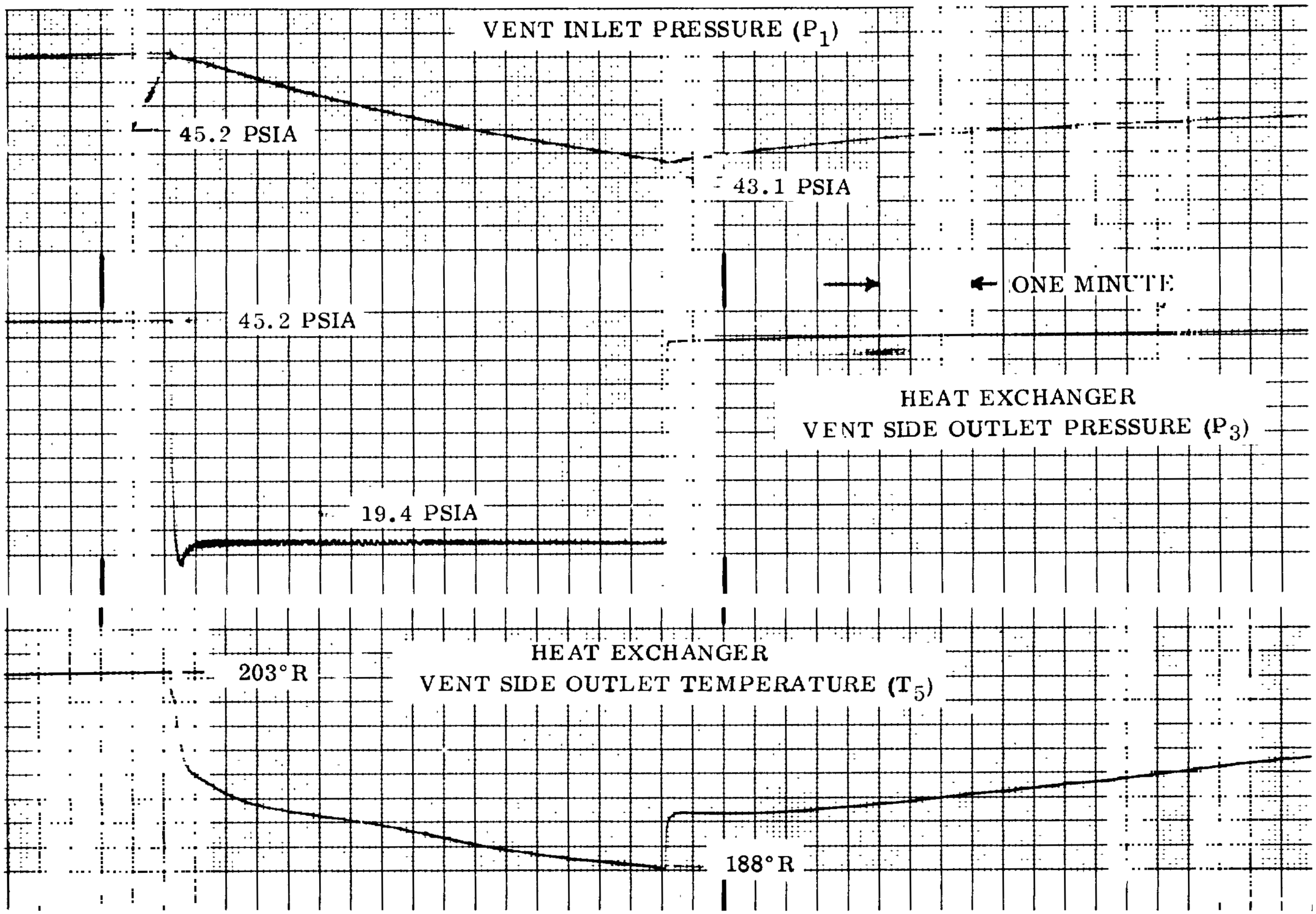
TEST CONDITIONS: PUMP SPEED, 550 RPM — LIQUID LEVEL 2
VENT FLOW RATE, 75 LB/HR — INPUT HEAT, 500 BTU/HR

Figure 6-19. Vent System Performance Run No. 14, Condition Spec 1



TEST CONDITIONS: PUMP SPEED 1800 RPM LIQUID LEVEL 3
VENT FLOW RATE 37.5 LB/HR INPUT HEAT 500 BTU/HR

Figure 6-20. Vent System Performance Run No. 10, Condition 17



TEST CONDITIONS: PUMP SPEED, 1800 RPM — LIQUID LEVEL 3
VENT FLOW RATE, 75 LB/HR — INPUT HEAT, 500 BTU/HR

Figure 6-21. Vent System Performance Run No. 10, Condition 18

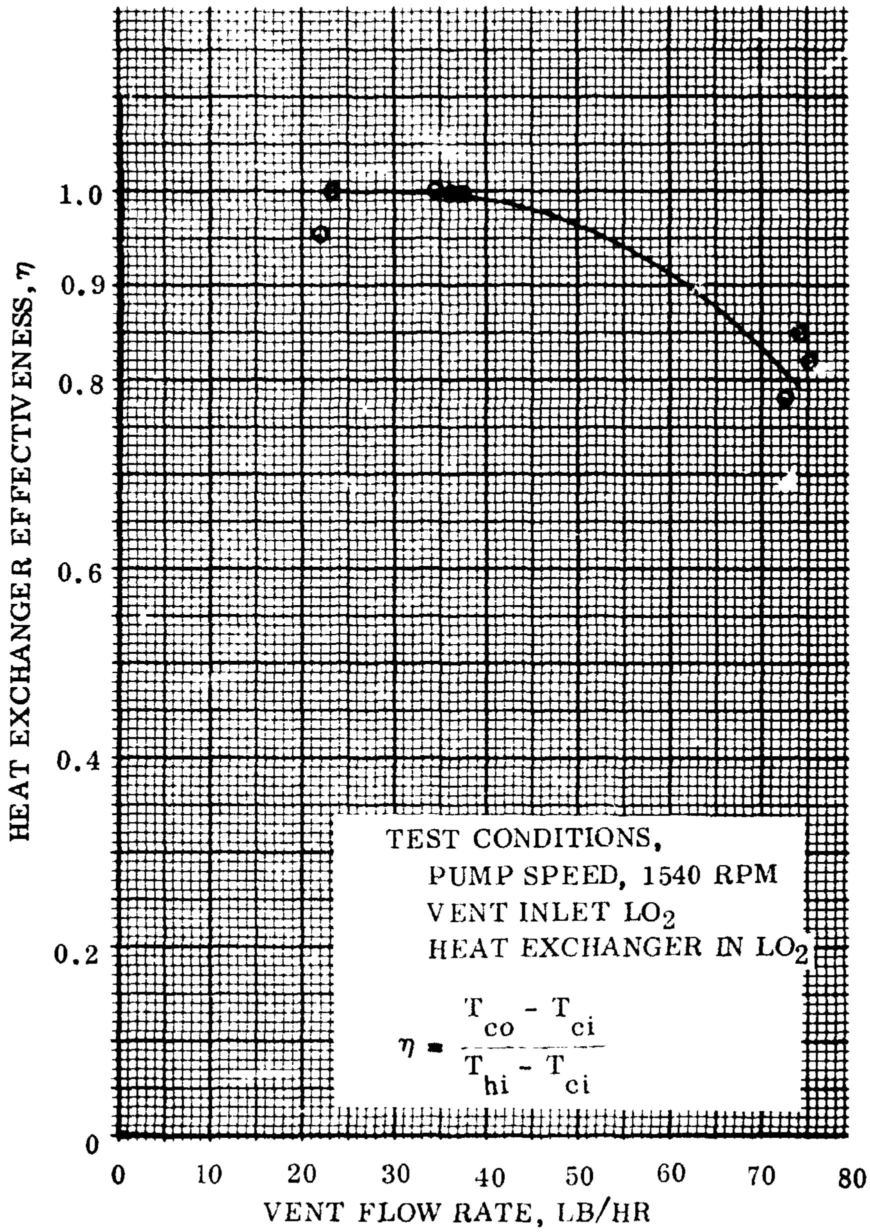


Figure 6-22. Heat Exchanger Effectiveness in Liquid Oxygen

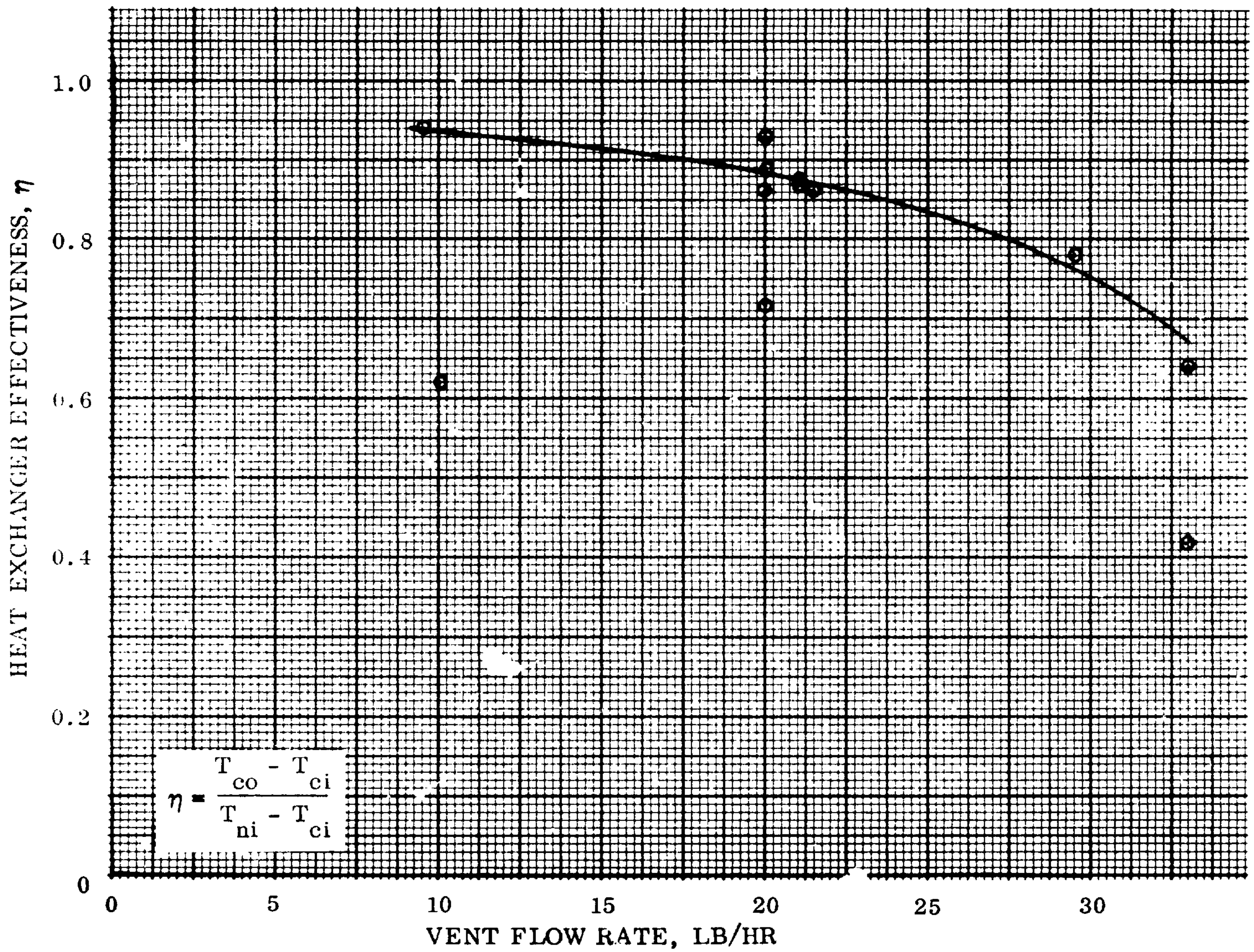


Figure 6-23. Heat Exchanger Effectiveness in Gaseous Oxygen

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The exchanger effectiveness values when operating with gaseous oxygen on the hot side, as shown by Figure 6-23, were lower than those presented in Figure 6-22 for liquid oxygen on the hot side. This was due to the lower hot side heat transfer coefficients when operating with a superheated gas. It was expected that, with saturated gas on the hot side, condensation would occur and heat transfer coefficients would be as high as the liquid. The high hot side temperatures when the unit was in gas (liquid level at 13 inches) were due to a significant amount of temperature stratification existing in the ullage. These hot side temperatures were such that the actual vent side exchanger outlet temperature was at least as high when the system was in gas as when it was immersed in liquid, even though the exchanger effectiveness was lower. The definition of effectiveness as used here is not an absolute measure of the exchanger performance when operating with a boiling fluid. For a given exchanger vent side outlet pressure, vent side outlet temperature is the primary measure of efficient system operation. This effectiveness is, however, a good measure of the performance of the exchanger in the superheat region, giving an indication of the system's ability to vent pure vapor. A zero effectiveness value would mean that saturated vapor was being vented, with a good chance that liquid could also be present in the vent stream.

Heat exchanger performance was further determined as a function of hot side flow rate by varying the pump speed. Both the vent and heat exchanger hot side inlets were in liquid, and the vent flow rate was maintained constant at approximately 17 kg/hr (37.5 lb/hr). The results of these tests are presented in Figures 6-24 and 6-25. Figure 6-24 shows exchanger effectiveness versus pump speed which is proportional to hot side flow rate. It is seen that the effectiveness is zero (saturated outlet fluid) at pump speeds below 80 percent of the pump design speed (1540 rpm). This is also illustrated in Figure 6-25 where heat exchanger outlet temperature is shown to decrease as pump speed decreases. With the outlet temperature at saturated conditions, some liquid was assumed to be at the system outlet. This occurred at the test condition with a pump speed at 280 rpm, corresponding to flow rate of 2210 lb/hr. Plotted readings of exchanger outlet temperature, showed that saturated outlet conditions would continue to occur at hot side flow rates slightly above this value.

From the data recorded in the LH_2 thermodynamic vent test, reference 2-11, it was shown that liquid was not present at the exchanger outlet until completely saturated conditions were reached. Therefore, any superheating of the vent gas with the present exchanger design results in a pure gas vent.

Heat exchanger vent side pressure drop was also measured during testing. These data are plotted in Figure 6-26 as a function of vent flow rate, for both gas and liquid inlets.

6.6.4 PUMP PERFORMANCE. Data obtained during testing of the pump and the heat exchanger are presented in Figures 6-27 and 6-28. Pump performance curves from Pesco (Reference 3-11) are presented in Figure 6-27. Pressure drop data to obtain the Convair curve were taken from results of testing done on the heat exchanger at

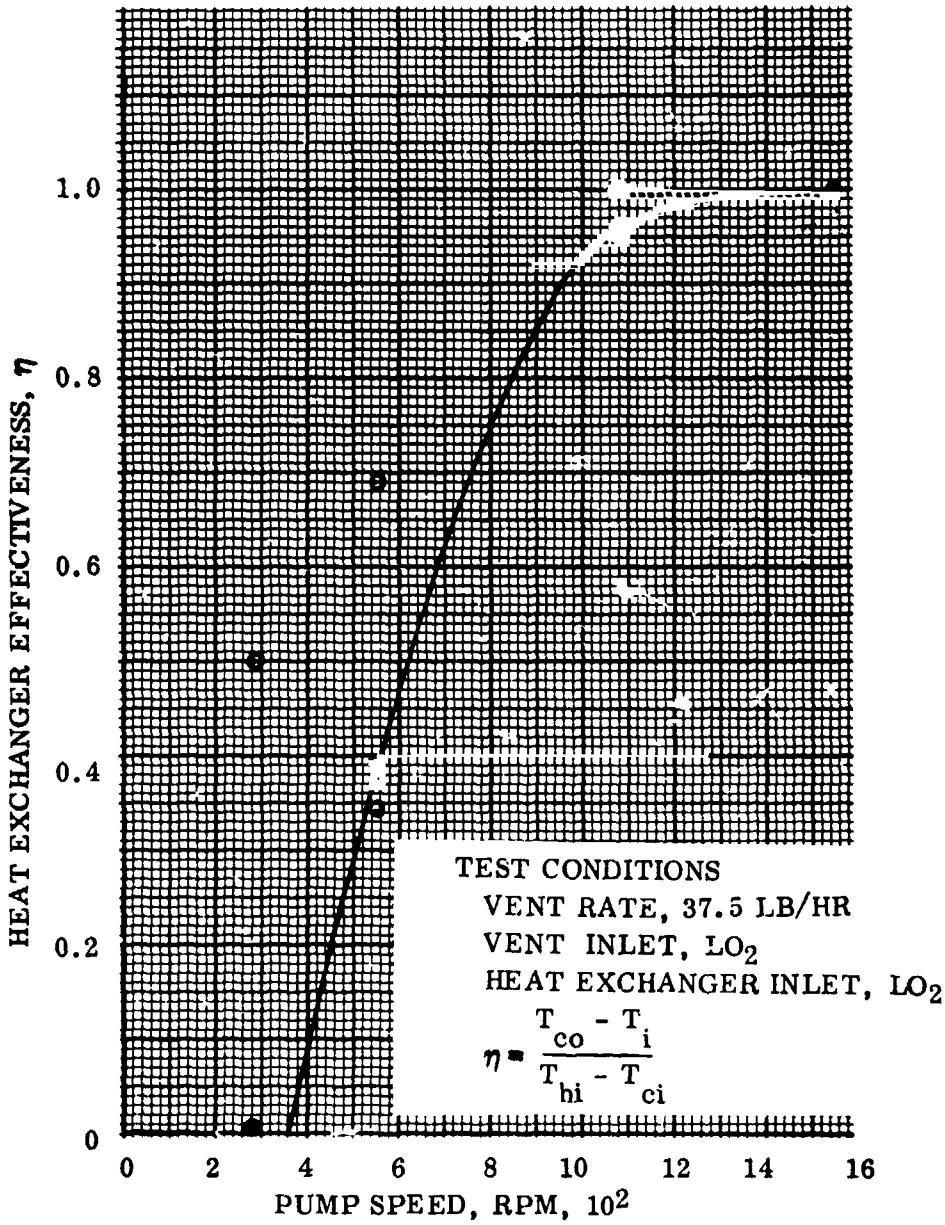

Figure 6-24. Heat Exchanger Effectiveness vs Pump Speed

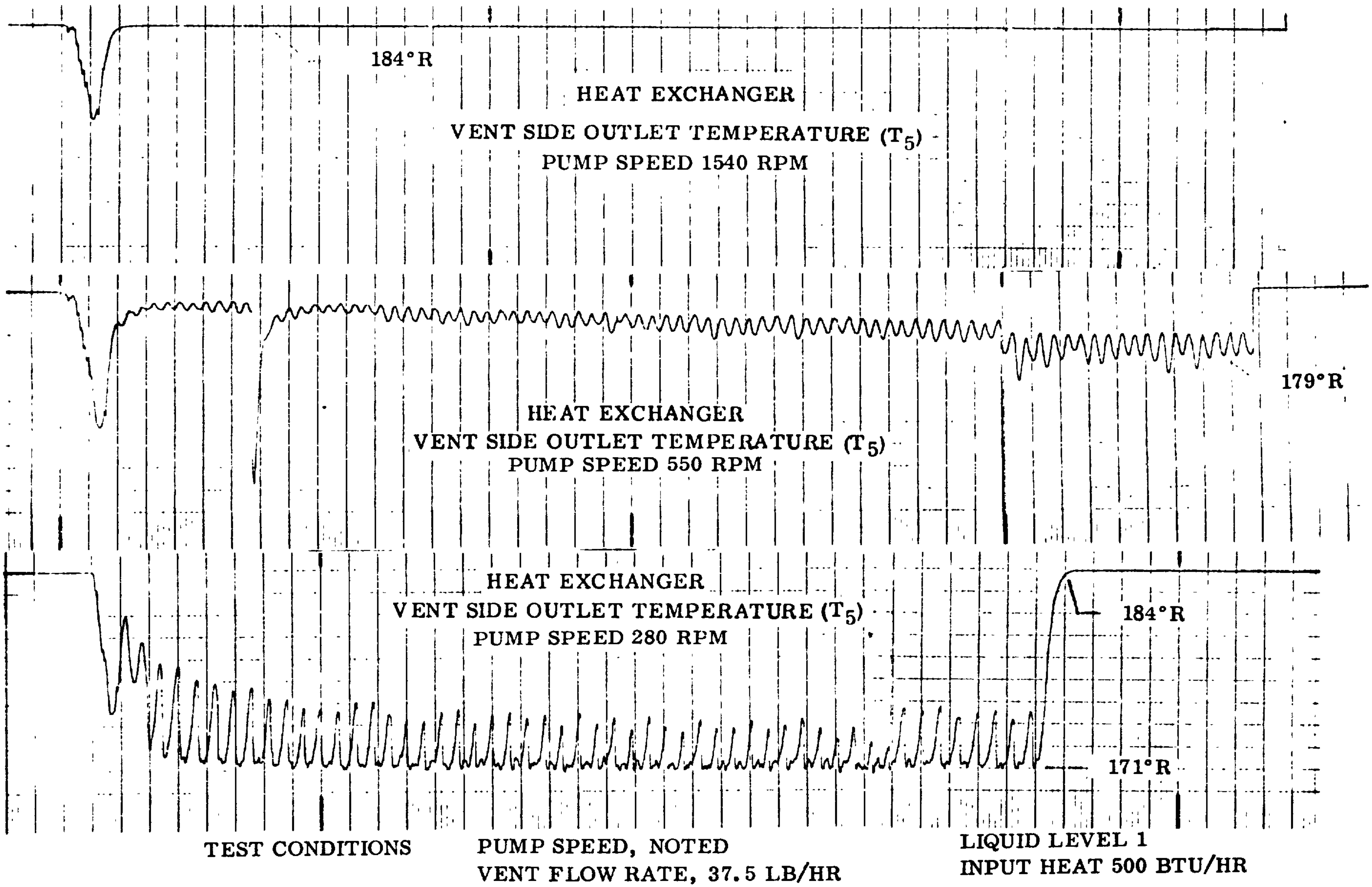

Figure 6-25. Heat Exchanger Vent Side Outlet Temperature at Various Pump Speeds

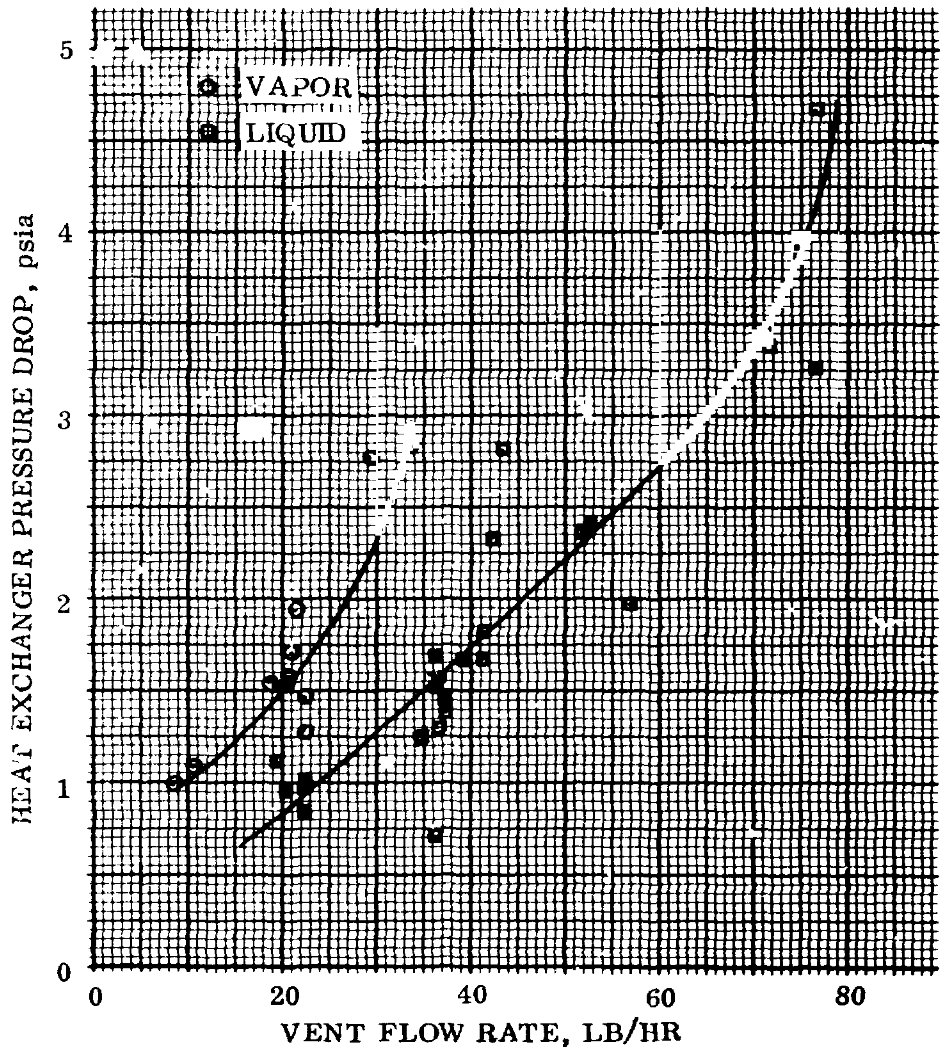

Figure 6-26. Heat Exchanger Pressure Drop Vs Vent Flow Rate

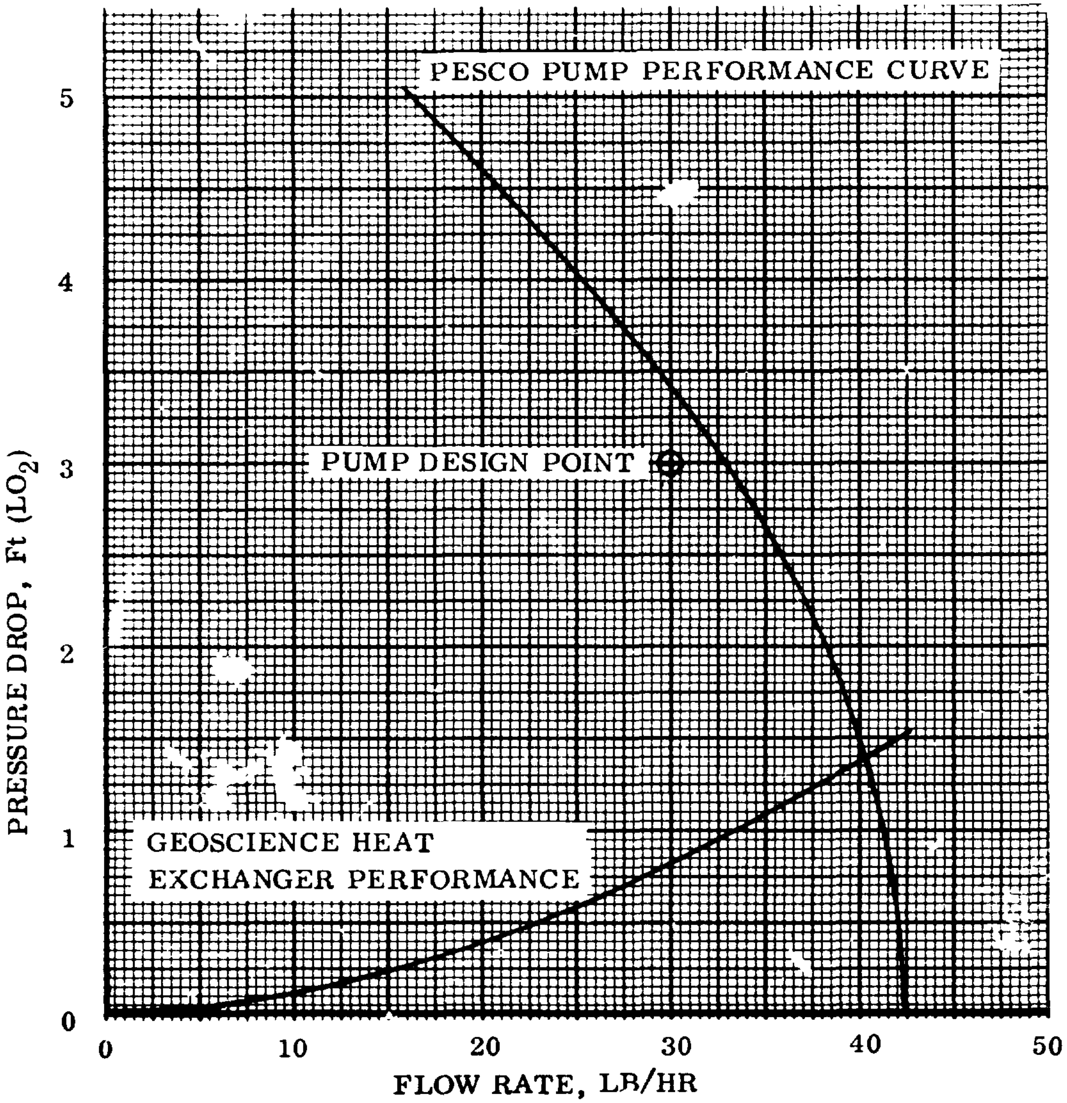

Figure 6-27. Pump and Heat Exchanger Performance Data

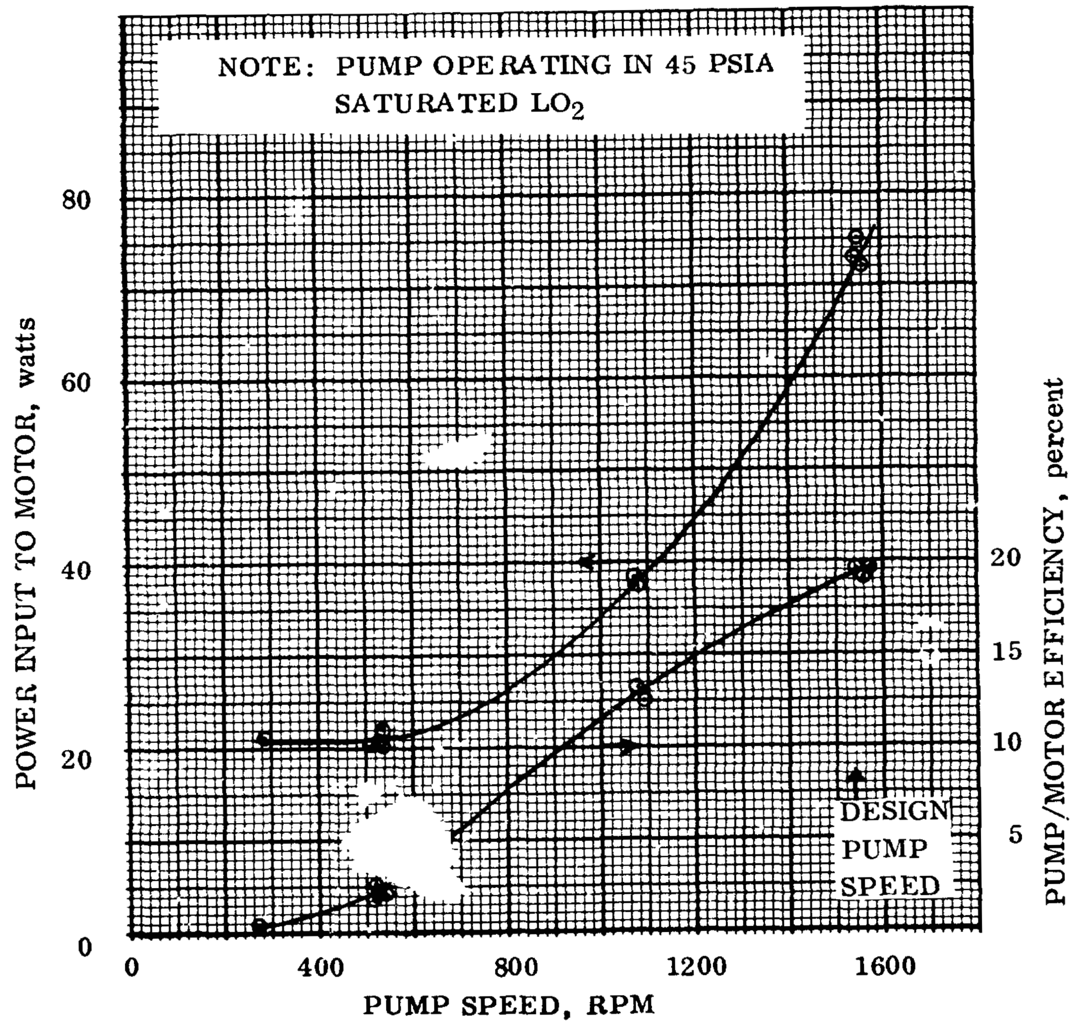

Figure 6-28. Pump Performance Vs Pump Speed

Geoscience (Reference 3-12). The hot side flow coefficient (C_p) was determined to be 2.0 based on a flow area of 1.63 by 1.94 inches.

In order to determine flow rates, heat rise, and pump fluid power, the pump affinity laws are used. In the present case, where the flow resistance is assumed constant;

$$Q \approx N$$

$$H \approx N^2$$

$$P_h \approx N^3$$

Where P_h is the hydraulic output power of the pump ($P_h = Q\,H$).

From the above data, the pump flow rate in LO_2 as a function of rpm was determined to be:

$$Q,\ \text{cfm} = 6.0 \left(\frac{\text{rpm}}{1540}\right) = 1.5 \times 10^{-3}\ (N,\ \text{rpm})$$

or for a LO_2 density of 67.5 lb/ft^3

$$\dot{m},\ \text{lb/hr} = 0.393\ (N,\ \text{rpm})$$

The static head rise is similarly determined to be

$$H,\ \text{ft} = 0.0738\ (Q,\ \text{cfm})^2$$

6.6.5 PRESSURE SWITCH PERFORMANCE. The pressure switch is required to sense tank pressure and actuate the pump and shut-off valve at a maximum pressure of 324 kN/m^2 (47 psia). Deactuation should occur at a minimum pressure of 269 kN/m^2 (43 psia). The minimum deadband of the unit should be 10.3 kN/m^2 (1.5 psi). A summary of the data obtained during oxygen testing is presented in Table 6-3.

During the Convair testing, the actuation point was sometimes slightly out of tolerance on the low side. The total band was, however, in tolerance and discussions with the vendor indicate that with a slight adjustment to the switch setting, control of the pressure within 296 to 324 kN/m^2 (43 to 47 psia) could be easily accomplished.

6.6.6 REGULATOR PERFORMANCE. The regulator performed satisfactorily throughout testing except for the problem noted during the compatibility test series when the unit regulated low and would not pass sufficient gaseous oxygen.

Regulator outlet pressure during normal operation is presented in Figure 6-29 as a function of vent-flow rate for both liquid and gaseous oxygen. The regulated pressure was lower with the gas inlet than with the liquid inlet, as would be expected. Also,

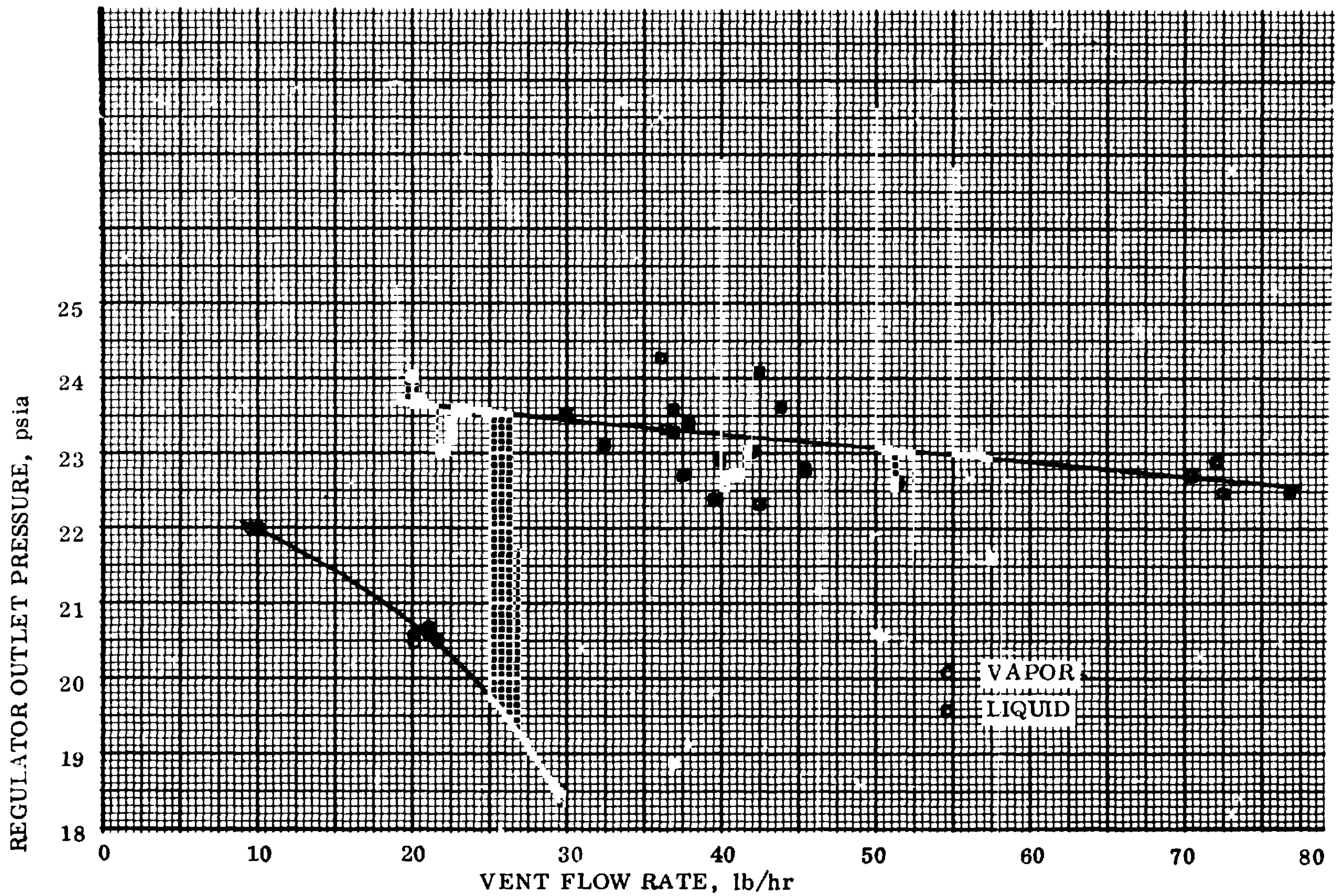

Figure 6-29. Regulator Outlet Pressure Flowing Liquid or Gaseous Oxygen

regulation was approximately 20.7 kN/m^2 (3.0 psi) lower than the original requirements of 151.8 ±3.4 kN/m^2 (22 ±0.5 psia) when operating with gas. Control of the pressure with a liquid oxygen inlet was, however, the most critical for heat transfer purposes and was generally within the requirements. In any case, the pressure regulation obtained resulted in satisfactory overall system performance.

6.6.7 ANALYSIS OF TANK PRESSURE DECAY RATES. The theoretical tank pressure decay rate while venting can be calculated when the vent fluid properties are known and the tank pressure decay model is assumed. The pressure change in a tank under equilibrium can be expressed as,

$$\frac{\Delta p}{\Delta t}, \frac{\text{psia}}{\text{hr}} = \left(\frac{\partial p}{\partial T}\right)_s \left(\frac{\dot{Q}, \text{Btu/hr}}{m_T, \text{lb}_m}\right) \tag{6-1}$$

Where Q is the heat removed from the tank and m is the total mass of fluid within the tank.

The net heat removed from the tank is the sum of the heat removed in the vent fluid and the heat added to the tank by normal heat leak and by input pump power. The heat removed from .he tank is given by

$$\dot{Q}_{out} = \left(\frac{e\lambda}{1-e}\right) + h_v - h_L \quad \dot{m}_v \tag{6-2}$$

where h_L is the enthalpy of the saturated liquid, h_V is the enthalpy of the fluid leaving the tank, e is the vapor to liquid density ratio, is the heat of vaporization at saturated conditions, m_V is the vent flow rate, and $\left(\frac{\partial p}{\partial T}\right)_s$ at 100°K (180°R) is 2.0.

Thus,

$$\frac{\Delta p}{\Delta t} = \frac{2.0\left[\left(\frac{e\lambda}{1-e} + h_v - h_L\right)\dot{m}_v - \dot{Q}_{in} - P_{in}\right]}{m_T\, C_{pL}} \tag{6-3}$$

The fluid vented is essentially at uniform conditions at or near the design flow rate. For the case when the nominal heat leak into the tank is 146 watts (500 Btu/hr). The pump input power at the design speed is nominally 73 watts (250 Btu/hr).

Thus:

$$\frac{\Delta p}{\Delta t} = \frac{(86.7\dot{m}_v - 750)\ 2.0}{m_T\ C_{pL}} \tag{6-4}$$

This equation is plotted in Figure 6-30 for liquid levels of .33 m and 1.52 m (13 and 60 inches) over a range of flow rates. The data shown in Figure 6-30, for the two levels agrees fairly well with the calculated results. The .33 m (13-inch) liquid level data fall below the theoretical line. This difference was due to incomplete mixing, since the heat exchanger was in the vapor.

6.6.8 ANALYSIS OF TANK PRESSURE RISE RATES. The theoretical pressure rise rate for a non-homogeneous fluid in a tank subjected to uniform heat can be calculated using equation 3-5. This equation is plotted in Figure 6-31 along with a presentation of the pressure rise rates experienced during testing. Pressure rise rates generally fall below those predicted by equation 3-5. This is due at least in part, to the stratification of the ullage as depicted in Figure 6-32. Equation 3-5 assumes uniform heating and uniform ullage properties which clearly cannot be the case with the degree of ullage stratification present.

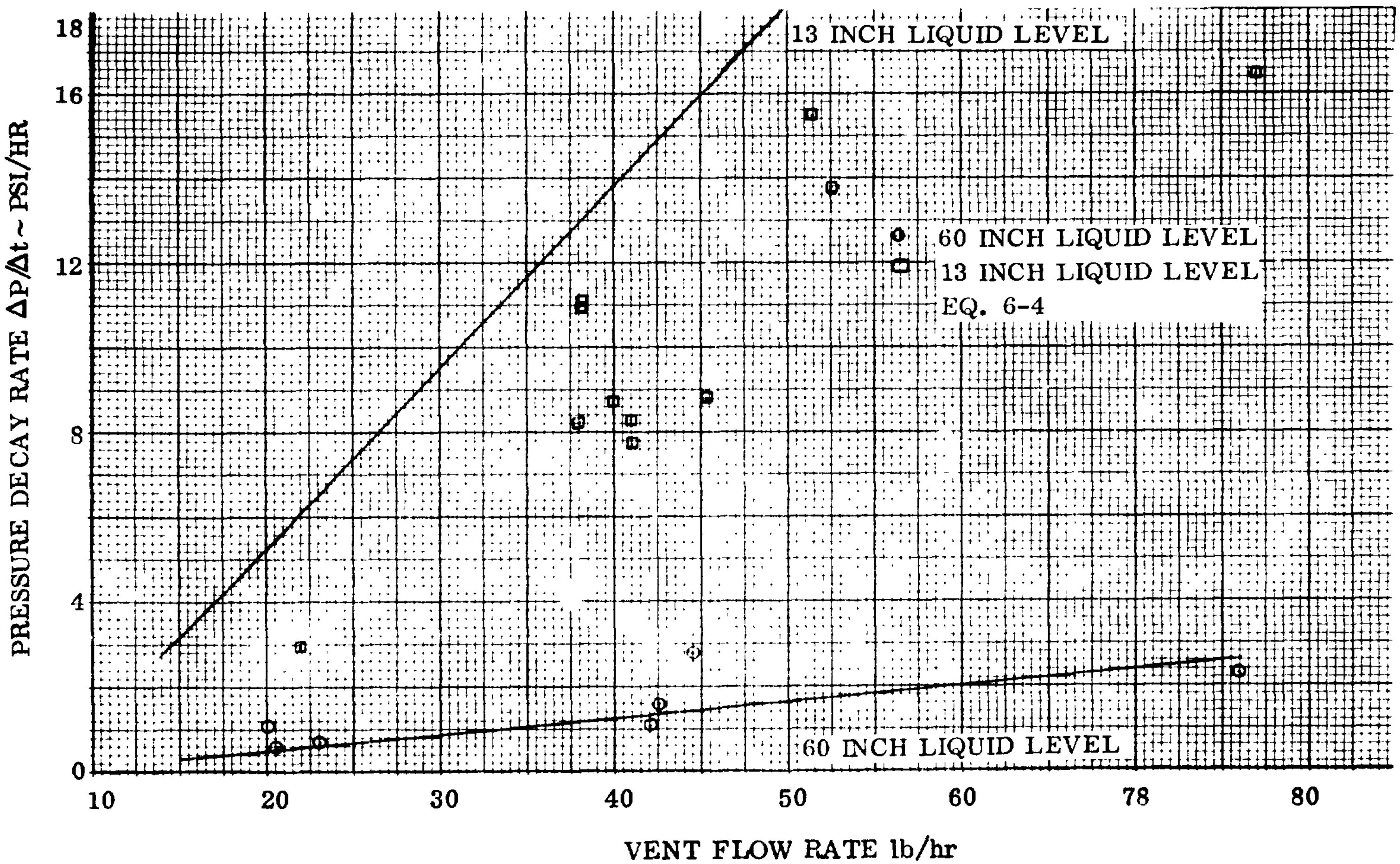

Figure 6-30. Pressure Decay Rates Versus Vent Flow Rate

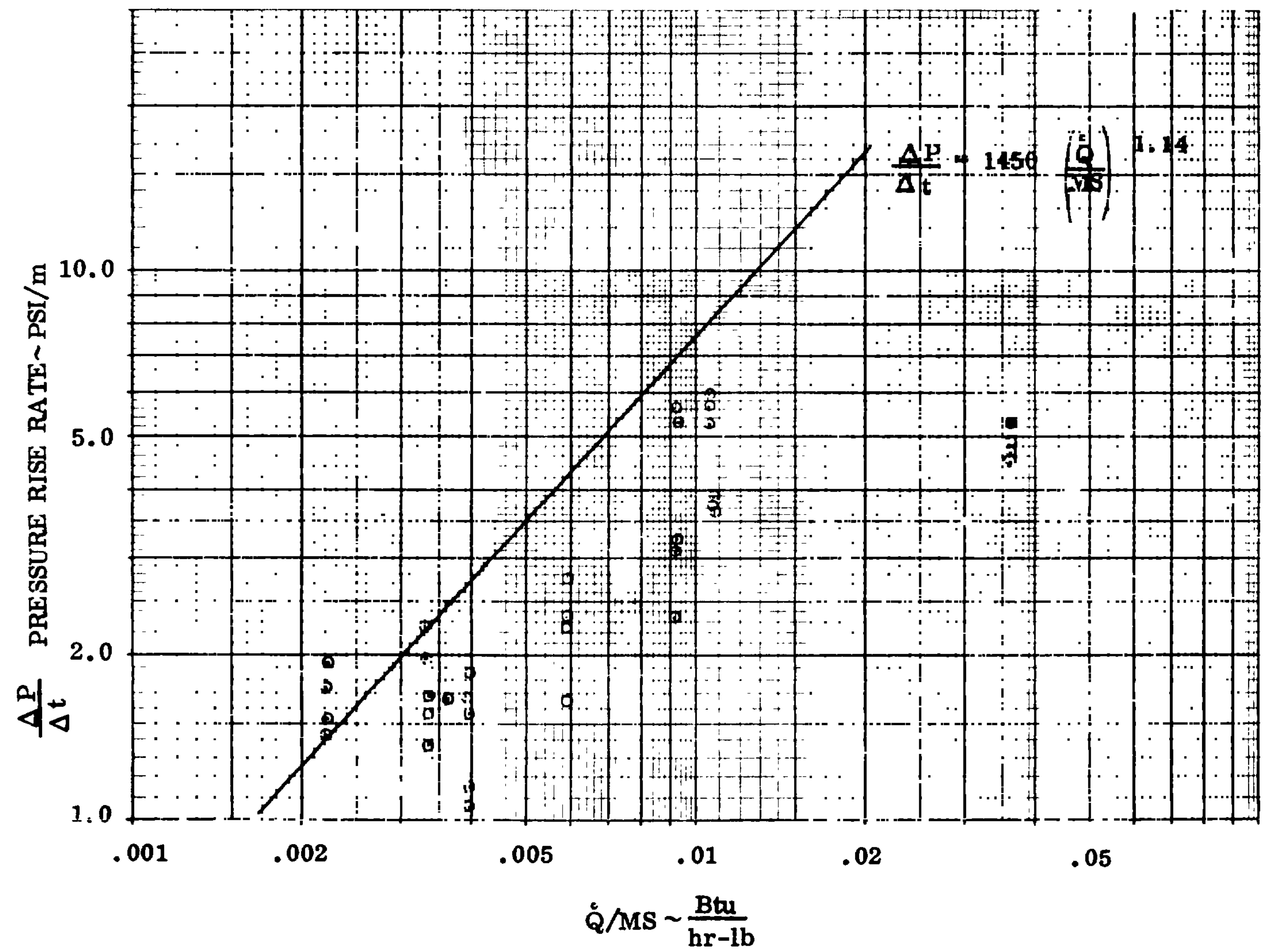

Figure 6-31. Correlation for LO_2 1-g Tank Pressure Rise Rate

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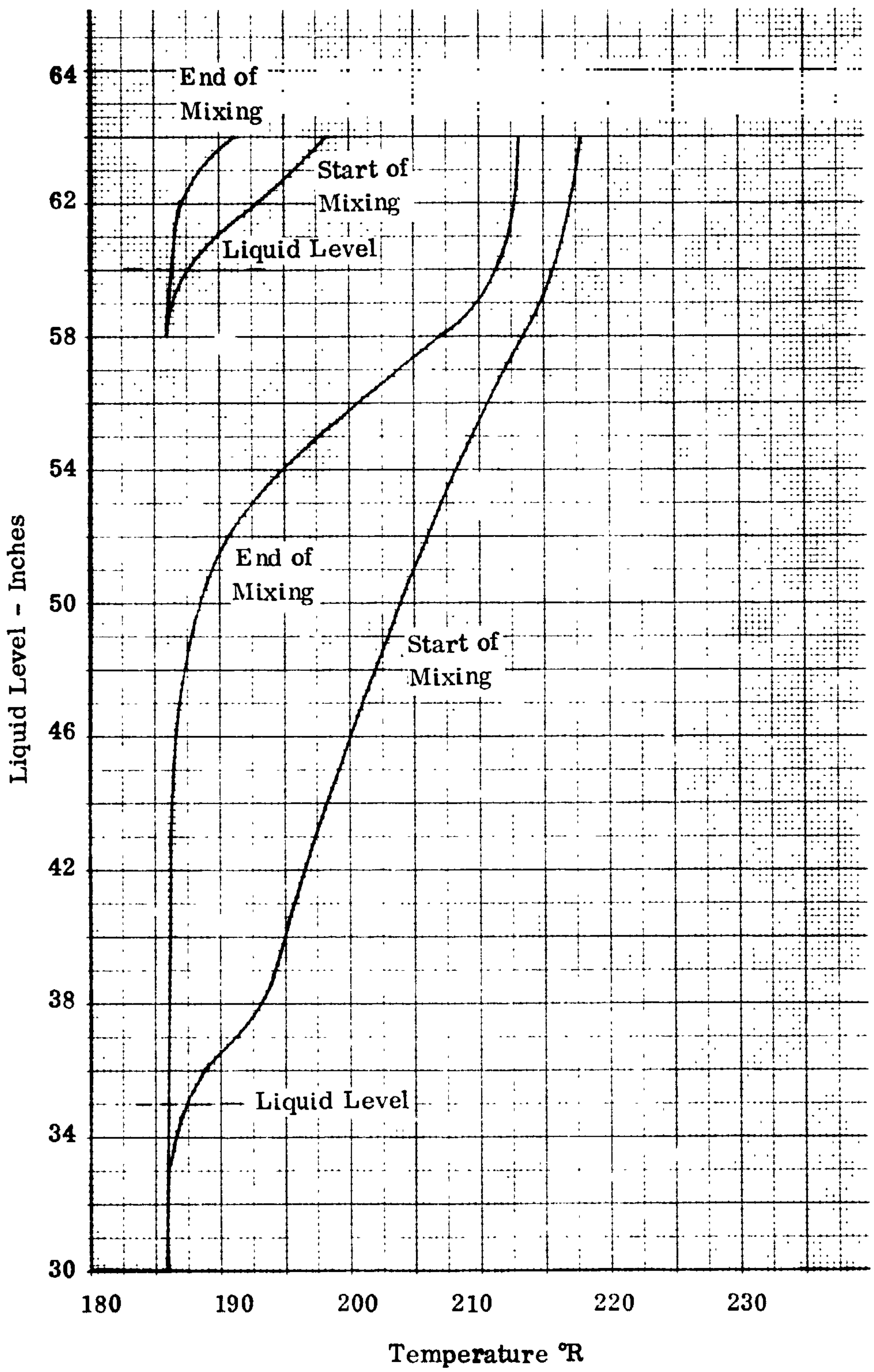

Figure 6-32. Effect of Mixing on Ullage Stratification at 35 Inch and 60 Inch Liquid Level

SECTION 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 CONCLUSIONS

The following conclusions are made as a result of this program.

1. The feasibility and efficiency of the system to control tank pressure while venting only vapor when operating in an environment at least as severe as that of the orbital experiment has been demonstrated.
2. Tank fluid mixing and liquid/ullage coupling are extremely important for efficient pressure control. Tank pressure decay with the vent inlet in liquid and with the heat exchanger outlet directed radially was slower than when directed axially. This is attributed to the fact that liquid mixing and subsequent liquid/ullage coupling was less effective in reducing the tank pressure.
3. Pressure control was accomplished with both gas and liquid at the vent and heat exchanger hot side inlets.

7.2 RECOMMENDATIONS

The following recommendations are made as a result of this program.

1. It is recommended that in preparing for operational use of the system a complete qualification testing of the system be performed.
2. Since it was verified that tank mixing is an essential criteria for efficient operation of this system and is integral with it, it is recommended that further analyses and testing be accomplished with this system to determine its mixing characteristics in zero gravity at various liquid levels, pump speeds, and power levels.
3. It is recommended that for orbital testing the system be located near one end of the tank with the heat exchanger outlet flow directed toward the other end.

SECTION 8
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APPENDIX A

DOCUMENT REVIEW DATA

The data contained in this section was obtained under the Convair Aerospace 1971 IRAD program and consists of the literature survey discussed in Section 2.0. General content, significant advancements, and applicability to the present LOX vent program are presented for each of the following documents.

A-1 Burge, G. W., "System Effects On Propellant Storability and Vehicle Performance," by Douglas for AFRPL, Contract No. AF 04(611)-10750, October 1966, AFRPL-TR-66-258.

The objective of this program was to investigate the tradeoff between propellant storability and vehicle performance for the propellant storage and feed subsystem of a hypothetical LF_2/LH_2 space propulsion system.

Parametric analysis of venting requirements and provisions was included. Design recommendations were derived and a full-scale test article was fabricated and tested to simulate the complete storage and feed subsystems. The test system utilizes LH_2 and LN_2. In the area of venting, potential propellant losses from uncontrolled venting at low-g were investigated and surface tension, mechanical separator and thermodynamic systems were analyzed for application to hydrogen tank venting. It was concluded that the thermodynamic system was the most promising for the H_2 and that there would not be a need to vent the F_2 tank for most missions considered.

Surface tension screens were discarded for the following reasons: (1) unknown behavior of moving fluids when encountering a dry screen, (2) problems associated with keeping the screen wet under all fluid and potential heating conditions, and (3) cleaning and compatibility problems with fine mesh screens. The following operating characteristics of mechanical separators are presented; (1) with turbine drive - 90% efficiency for inlet qualities down to 25% and (2) with electric motor drive - 80% efficiency for inlet qualities down to 10%.

It is noted that tests associated with low-g venting problems were not included. Basic exchanger design data and comparative data presented for the various low-g vent concepts are of a general interest to the LOX vent program.

A-2 Burge, G. W., "System Effects on Propellant Storability and Vehicle Performance," Contract No. AF-04(611)-10750, AFRPL-TR-68-227, December 1968.

This report presents the results of testing performed at AFRPL on the system described in the previous document in the area of system pressurization and high performance insulation. Data presented are not specifically applicable to the current LOX vent program.

A-3 Parmley, R. T., "Phase I Final Report, In-Space Propellant Orientation and Venting Experiments," by LMSC for Air Force Rocket Propulsion Laboratory, Contract AF 04(611)-11403, 31 October 1966. Dynatech performed dielectrophoretic component designs under subcontract.

This report describes the work done under phase I of a two phase program to provide two surface tension and two dielectrophoretic hardware units for orbital flight testing.

The primary experiment objectives are demonstration of low-g liquid orientation and ullage vapor venting. The first phase was designed to provide experiment definition, analysis, predesign and test plans. Storable propellant combinations are used for the surface tension analysis and cryogenic fluids for the dielectrophoretic analysis. The surface tension experiment was to be conducted in a 0.305 m (12 inch) diameter transparent tank using allyl alcohol as the test fluid, while the dielectrophoretic experiment was to use LN_2 in a 0.406 m (16-inch) diameter transparent tank. A significant amount of effort was performed on development of optimum materials and designs for the transparent test tanks.

The data presented serve to illustrate several possible dielectrophoretic and surface tension system designs, but is otherwise not directly applicable to the current LOX vent program.

A-4 Parmley, R. T., "Phase II Final Report, In-Space Propellant Orientation and Venting Experiments," by LMSC for AFRPL, Contract AF 04(611)-11403, April 1968.

This report summarizes work performed on the final design, fabrication, and ground testing of the surface tension experiment apparatus described in the previous document. As a result of the work reported two surface tension experiment hardware units were delivered to AFRPL. The companion dielectrophoretic experiment effort was terminated during the early fabrication phase. No specific reasons were given in this

document for termination, however, for future reference, all design and fabrication work performed up to the termination date of 2 March 1967 is summarized.

The only value to the present LOX vent program is in the illustration of potential surface tension vent concepts.

A-5 Stark, J. A. and Blatt, M. H., "Analysis of Zero-Gravity Receiver Tank Vent Systems," by Convair for MSFC, Contract NAS8-20146, GDC-DDB69-001, July 1969.

The information contained in this report is the result of a Phase III study under Contract NAS8-20146 to define and design an optimum vent system for use in an orbital propellant transfer receiver tank at low-g.

In this application, vent flow require[illegible]ts due to initial cooldown transients are high in relation to no[illegible] coast vent requirements and it was assumed that the unit did not ha[illegible] operate with 100% vapor. Data showed that the mechanical sep[illegible] was best for this application. Some work was also done with wall exchangers to improve the overall vent efficiency.

Some general comparison data and mechanical separator performance information may be applicable to the present LOX vent program.

A-6 Stark, J. A., et. al., "Cryogenic Propellant Control and Transfer," GDC-ERR-1538, December 1970.

The work described in this report is divided into three main categories, (1) propellant conditioning, (2) propellant orientation and control, and (3) orbital propellant transfer. Data pertinent to the present LOX vent program are found under both the propellant conditioning and orbital propellant transfer tasks. In the area of propellant conditioning, the reduction and analyzing of data obtained during cryogenic testing of a low-gravity heat exchanger vent system and associated temperature stratification and destratification was accomplished.

Also, and most pertinent to the present LOX vent study was the development and documentation of a computer program capable of analyzing the heat transfer and flow characteristics of the exchanger vent concept when operating with either H_2 or O_2. Pertinent work under the orbital propellant transfer task was an analysis of the safety aspects of a dielectrophoretic LO_2 orientation device. This analysis showed significant problems still existed with respect to the reliability of high voltage feed-throughs and the potential arcing of the electrodes and that the safety of such systems with O_2 had not yet been fully demonstrated.

A-7 Kavanagh, H. M. and Rice, P. L., "Development of Subcritial Oxygen and Hydrogen Storage and Supply Systems, "NAS9-1065, by AiResearch for MSC, 22 September 1964.

Oxygen and hydrogen subcritical storage and supply systems were designed, fabricated, and ground tested. The primary objective was to advance the state-of-the-art with respect to the required analytical tools, design procedures, and fabrication techniques, and, in particular, to resolve the phase separation problems associated with subcritical storage under zero-g conditions. The systems are designed to supply gas for fuel cells and life support systems.

Operation is similar to a thermodynamic vent system in that tank fluid is throttled to a lower temperature and pressure through a pressure regulator and then passed through a heat exchanger to vaporize any liquid which may be present. In this case the exchanger is located at the outer wall of the vessel which is vacuum jacketed. Also, the use rate is such that some of the existing fluid may be heated and passed back through a heat exchanger within the bulk fluid to maintain storage pressure above a certain level. The primary purpose of this system is to deliver gas rather than provide vent pressure control. The O_2 tank diameter was 0.635 m (25 in.).

The main value of the data to the present LOX vent program is in a general assessment of hardware and system configurations for a wall exchanger vent application. It is noted that actual testing was done with LN_2 rather than O_2.

A-8 Allgeier, R. K., "Subcritical Cryogenic Storage Development and Flight Test," NASA/MSC, February 1968, NASA-TN-D-4293.

This report describes both phases of a two phase program to develop subcritical cryogenic storage systems capable of supplying warm vapor regardless of the liquid orientation. Under the phase I program operational prototypes of LH_2 and LO_2 systems were designed, fabricated and ground tested. For the oxygen system N_2 was actually used as the test fluid. The phase I program data are also presented in the previous document (A-7).

The phase II program was to evaluate the performance of follow-on hardware in earth orbit using LN_2 as the test fluid. The tank was a dewar on the order of 0.635 m (25 in.) inside dia. The flight test system was aboard Apollo-Saturn flight 203 on 5 July 1966. This flight completed development of this particular subcritical system. Satisfactory system operation was demonstrated. The system exhibited good pressure control and stability and the heat-exchanger outlet temperature was significantly above saturation, indicating only vapor removal. Vapor was delivered at a constant pressure and the capability of the regulator throttling valve and wall heat exchanger to operate within the design limits was verified.

The main value of the data presented with respect to the current LOX vent program is in the assessment of hardware and system configurations for a wall exchanger vent application and in the state-of-the-art knowledge that such a system has been successfully operated in earth orbit.

A-9 Hill, D. E. and Salvinski, R. J., "A Thermodynamic System for Zero-g Venting, Storage, and Transfer of Cryogenic Propellants," Journal of Spacecraft and Rockets, Vol. 4, No. 7, TRW, July 1967.

Analysis and one-g LH_2 testing of a thermodynamic vent system with an exchanger mounted on the tank wall is described. The test system consisted of a 0.61m (24-in.)dia. aluminum tank insulated with 80 layers of crinkled aluminized nylar. The exchanger consisted of .0031 m (.125 in.) dia. aluminum tubing mounted with a spacing of 0.228 m (9 in.) to the outside surface of the tank.

Basic equations and weight and hardware configuration data may be applicable to evaluating wall exchanger systems for the LOX vent program.

A-10 Zara, E. A., "Flight Test of A Subcritical Cryogenic Storage Vessel," Air Force Flight Dynamics Laboratory, Wright-Patterson Air Force Base, June 1968, AFFDL-TR-68-46.

The results of an 1800 second, zero gravity flight test of a nitrogen subcritical cryogenic storage vessel are presented. The flight objective was to evaluate the performance of a throttling valve and heat exchanger, designed to deliver a controlled flow of gas from a stored two-phase mixture.

Some basic hardware and wall exchanger performance data may be pertinent to the current LOX vent program, however, the basic system is designed as a gas supplier, with external heating available for gas vaporization, rather than a vent system. The test vessel was a CRES dewar with an inner shell diameter of 0.178 m (7.0-in.).

A-11 Dowty, E. L. and Murphy D. W., "Heat Exchanger Design for Cryogenic Propellant Tanks," Paper Presented at the 1967 Cryogenic Engineering Conference.

This paper describes the design and test of a tank wall mounted heat exchanger with application to the thermodynamic vapor separation system. Analysis was accomplished on the basis of considering the tank wall to be a thermal fin with tubing attached to the outside to provide a sufficient fin root to absorb an amount of heat equal to that entering the tank fr... external sources.

The test tank was a 0.304 m (1 ft) dia. copper sphere and liquid argon was used to simulate the propellant and helium gas the coolant.

Some basic analytical data may be applicable to the current LOX vent wall exchanger concept, however, it is noted that energy balances and testing were associated with an exchanger attached only in the ullage region of a tank operating at one-g and would not have the same characteristics as one required to operate at low-g with any liquid orientation.

A-12 Murphy, D. W., et. at., "Vent-Free Fluorine Feed System," by Martin for AFRPL, Contract F04611-67-C-0044, March 1968.

A two phase program was conducted to investigate thermodynamic liquid vapor separator heat exchange systems that will efficiently maintain a vent-free fluorine condition by using vented hydrogen to cool the fluorine. This report presents the work accomplished under phase I of the program.

A parametric study was conducted to define an optimum system for a typical 90-day space mission. In this regard a computer program was developed to analyze the vehicle and define the amount of insulation, vented hydrogen and heat exchanger required for maximum vehicle ΔV. The fluorine is cooled by attaching aluminum tubing over the outside of the fluorine tank.

The data and analytical tools developed are applicable to the LOX vent study in the analysis of the wall exchanger thermodynamic vent concept.

A-13 Murphy, D. W. and Rose, L. J., "Vent-Free Fluorine Feed System," by Martin Marietta for AFRPL, Contract F04611-67-C-0044, June 1968, AFRPL-TR-67-323.

The objective of the program was to investigate thermodynamic liquid vapor separator heat exchange systems that will efficiently maintain a vent-free fluorine condition by using vented hydrogen to cool the fluorine. This report contains the results of phase II of this program in which a test article was designed and fabricated for use in the AFRPL space simulation facility to generate data for verifying the vent-free analytical model developed during phase I (reference previous document).

The test article consisted of a 1.83 m (6 ft) dia. by 3.35 m (11 ft) high structure housing a 0.915 m (3 ft) dia. and a 1.22 m (4 ft) dia. spherical propellant tank. The test fluids were to be LN_2 and LH_2, and the tanks were made of stainless steel. The nitrogen heat exchanger consisted of 15.25 m (50 ft) of copper tubing attached to the outside of the tank. The tanks were insulated with 1-inch of aluminized mylar multilayer insulation.

The system was helium leak tested and functionally tested with liquid nitrogen and hydrogen before shipment.

Some basic hardware and system fabrication techniques may be useful in evaluation of the LOX vent wall exchanger concept.

A-14 Page, R. G. and Tegart, J. R., "Vent-Free Fluorine Feed System Analysis - Final Technical Report, " Contract FO4611-69-C0033, September 1969.

The objective of this program was to reduce the experimental data obtained at the AFRPL on the system described in the previous document, correlate it with the analytical model described in document A-12 and modify the model and computer program if required.

The most significant test run was a 34-hour run to demonstrate the feasibility of the vent heat exchanger system. Based on results of the test the analytical model was modified and improved. In general the final model could predict the test data within ±20% under all imposed conditions.

The data and analysis are somewhat applicable to the LOX vent wall exchanger concept. It is noted, however, that the data will only be of a basic nature since the actual exchanger fluid was hydrogen rather than the O_2 or N_2.

A-15 Sterbentz, W. H., "Liquid Propellant Thermal Conditioning System," for LeRC by Lockheed, Interim Report, NAS3-7942, April 1967.

The overall program was designed to investigate the effectiveness of the thermal conditioning concept for cryogenics tank pressure control.

During the study various concepts were studied for use as fluid removal units, expansion units, heat exchanger units, and propellant mixers. Parametric performance data were developed for these components and are presented in the report. Also, some criteria for establishing propellant mixing requirements are included. It is noted that the fluid removal unit was originally required to provide a liquid-only inlet to the vent in order to conserve helium. However, in the analysis presented it was decided that none of the various concepts considered (wicking device, capillary standpipe, dielectrophoresis, dynamic separators) were reliable, or practical from a weight standpoint. Liquid removal was, therefore, not included in the final system design. The parametric data and mixing criteria were applied to three reference missions and a thermal conditioning system defined for the liquid hydrogen tank on each.

The data contained in the report is generally for hydrogen, however, some of the basic equations and parametric component performance data may be useful to the present LOX vent program.

A-16 Sterbentz, W. H., "Liquid Propellant Thermal Conditioning System," Final Report, NAS3-7942, by Lockheed for NASA/LeRC, 15 August 1968, LMSC K-07-68-2.

This document reports the design, fabrication, and test of a hydrogen thermal conditioning system. Also the experimental data are compared with the analytical data presented in the previous document.

The information contained in this report is only important to the LOX vent program as it adds to the general feasibility demonstration of thermal conditioning type vent systems.

A-17 Cady, E. C., "Thermodynamic Vapor-Liquid Separator," from Proceedings of Low-G Seminar at Douglas in May 1969. Edited by J. B. Blackmon, N 71-13113.

The objective of this program was to define an optimum low-gravity vent system for use with upper stage vehicles using cryogenics, including LF_2. Various thermodynamic vent concepts were considered.

The report discusses previous work done on wall and bulk heat exchanger systems by Martin (Document A-12), Lockheed (A-15) and Convair (Ref. 3-2), and presents a new concept of their (MDAC) own. This new concept is called an electro convection system, and is essentially the same as a bulk exchanger system, except that the mechanical pump is replaced by a high voltage electric field to provide increased heat transfer coefficients for exchanger operation. The report isn't clear as to whether or not fluid mixing, if any, will be sufficient to prevent the unit from subcooling a local liquid volume. The system electrical requirements appear to be similar to that for dielectrophoretic liquid orientation. However, based on their comparisons this system seems to have some advantages over the other systems and was considered further in the initial screening.

A-18 Leonhard, K. E., et. al., "Cryogenic Tank Test Program," GDC-ERR-1419, December 1969.

As part of this report the test system and test tank used for hydrogen heat exchanger vent testing is described. The test tank was a 2.23 m (87.6 in.)/1.89 m (74.5 in) oblate spheroid. Corresponding data reduction and information most pertinent to the present LOX vent study are presented in document A-6.

A-19 Fahimian, E. J. and Hurwitz, M., "Research and Design of a Practical and Economical Dielectrophoretic System for the Control of Liquid Fuels Under Low Gravity Environmental Conditions," Final Report, by Dynatech for NASA/MSFC, Contract NAS8-20553, May 1967, Dynatech report No. 723.

Design and weig. t data are presented for Saturn and uprated Saturn orbital tankers and for both three-stage and Saturn IV lunar mission vehicles. Data are based on the use of practical state-of-the-art components. Mechanical design and weight analysis were performed under subcontract by LMSC. Design and weight data generated for complete liquid orientation in the various lunar mission vehicles would be most applicable to the present LOX vent program.

A-20 Blutt, J. R., "Operating Safety of Dielectrophoretic Propellant Management Systems," Final Report, NAS8-20553, by Dynatech for MSFC, 31 March 1968, Dynatech Teport No. 768.

An experimental program was conducted to demonstrate the safety of a typical dielectrophoretic propellant orientation system with oxygen and hydrogen propellants and helium pressurization.

The program was conducted in three steps: (1) determination of compatibility of materials with oxygen and hydrogen, (2) generation of basic data on electrical breakdown in oxygen/helium and hydrogen/helium mixtures using model electrode elements, and (3) investigation of the extrapolation of the small-scale electrical breakdown data by performing "full scale" tests using 0.1525 m (6 in.) wide electrodes with 0.0254 m (1 in) and 0.076 m (3 in) spacing in a 1.052 m (41.5 in) diameter spherical tank. The study conclusion was that small-scale module electrical breakdown tests can be used to predict the performance of full scale designs.

It is noted, however, that the O_2 testing is considered the most critical to the present LOX vent application and, due to the failure of the high voltage feedthrough, the test series was cut short such that O_2 system safety was not conclusively proven.

An analysis of the safety demonstrations reported herein was made by Convair under Contract NAS8-26236 and is reported in document A-6.

A-21 Blutt, J. R. and Hurwitz, M., "A Dielectrophoretic Liquid Oxygen Converter for Operation in Weightless Environments," Dynatech Corp. for Wright Patterson Air Force Base, AMRL-TR-68-21, July 1966.

Analyses, experiments, and designs were accomplished to establish the performance characteristics, optimum weight, and safety of a dielectrophoretic liquid oxygen converter.

Application is to provide complete orientation of LO_2 in a 25-liter (0.88 ft^3) tank in order that liquid can be withdrawn and vapor vented. One-g testing was accomplished with both Freon and LN_2.

The main value of this work was in the development of realistic electrode configurations and weight data. In addition to the 25-liter (.88 ft^3) bottle operating at 10^{-3} g's weights are presented for systems up to 1.017 m (40 in) diameter (19 ft^3) and operating between 10^{-3} and 10^{-5} g's.

A-22 Bovenkerk, P. E., "Zero-Gravity Test of an Advanced Surface-Tension Propellant Orientation and Ullage Vapor Venting Device," AFRPL-TR-69-1[illegible] June 1969.

This report, summarizes the results of a low and zero-gravity test program to determine the functional capabilities of a representative liquid propellant surface-tension orientation and vapor venting system that resulted from Contract AF 04(611)-990 with Bell Aerosystems. This Bell work was covered under the literature survey presented in Reference 3-1.

This device utilized screens and capillary traps to accomplish the orientation and venting functions. Testing was accomplished in a KC-135 aircraft. The results of the test program successfully demonstrated the use of surface-tension screens for propellant orientation. The vapor venting tests that were performed were "successful," however, several unstable situations are possible which could not be investigated in the test facility utilized. In the venting system described in this report, there is no positive means of insuring that liquid will not be vented. There is the possibility of forcing some liquid through the vent baffle on every vent cycle.

It is concluded in the report, that orienting all the propellant is probably the soundest approach to solving the venting problem and the specific system described here would not be applicable to the LOX vent application.

A-23 Warren, Richard P., et. at., "Passive Retention/Expulsion Methods for Subcritical Storage of Cryogens," Martin Marietta Corp./Denver Report MCR-71-58, Contract NAS 9-10480, July 1971.

This report gives a detailed description and analysis of a surface tension device which is designed to control all of the propellant, thus enabling control of both a liquid free vent gas volume and a gas free liquid volume for expulsion. A heavily modified design of the same concept is presented as a combination acquisition and vent control system for the space shuttle orbiter OMS tanks.

A discussion of this system's design, operation and deficiencies is presented along with a comparison to a separate acquisition system and a thermodynamic type vent system. No testing has been performed to date to prove the systems performance capabilities under shuttle orbiter operating conditions.

This system is applicable to the LOX vent requirements and appears to be theoretically feasible. Thus, further consideration will be given to this system in the initial systems analysis and concept selection to be performed under the current LOX vent program.

A-24 Nice, M. and Ostrach, S., "Liquid-Gas Separation and Containment," Research sponsored by theAir Force Office of Scientific Research, August 1970.

The objective of this research was to determine some of the containment characteristics of a closed liquid-gas cyclone separator and how the containment characteristics are affected by shaping the separator's walls.

Performance curves are presented as a function of inlet Reynolds number, separator geometries, and collection times. Special attention is given to the effect of geometry on secondary flows. This report gives a good introduction to the general field of vortex separation along with a number of references. However, the data contained would only be useful to the LOX vent program if such a vortex system were pursued to detailed analysis and design.

A-25 Macklin, M., "Water Handling in the Absence of Gravity," Aerospace Medicine, Vol. 37, October 1966.

The use of centrifugal force field and static impingement methods of separating or collecting water in the absence of gravity are discussed. The report presents a good general discussion of the various separation processes, but does not contribute any specific data for solving the present LOX vent problem.

A-26 Lawler, M. T. and Ostrach, S., "A Study of Cyclonic Two-Fluid Separation," by Case Institute of Technology for AFOSR, Contract AF-AFDSR-194-65, June 1965.

This study presents a consideration of some of the problems of two-phase flow in order to analyze the separation of a two-phase fluid mixture in a cyclone separator. An experimental separator was designed and built. The test fluid was water and air. Separation efficiency and pressure drop data are presented for a wide range of operating conditions and geometric configurations.

Again this data would only be directly useful for the LOX vent program if a detailed analysis and design of a vortex type system were to be initiated.

A-27 Ward, W. D., "Vapor-Liquid Separator," NASA, Patent No. 3,397,512, 28 December 1966.

A device for separating vapor from a heterogeneous vapor-liquid mixture ranging from 100% liquid to 100% vapor at zero-g is described. The system utilizes a static vortex tube separator to provide vapor to drive a turbine which drives a pump to force the separated liquid back to the tank.

It is assumed that expansion into the vortex tube will vaporize enough gas to allow separation to be accomplished when the inlet is 100% liquid. At first glance it would seem that turbine and pump control over the full range of potential inlet qualities would be a significant problem. Also, the efficient operation of the vortex tube over the full range of conditions would be questionable. This type system is, however, further considered in the analysis and comparison task.

A-28 Hamm, J. R., et. al., "Technical Proposal, to Convair Aerospace Division General Dynamics Corporation for Fabrication and Acceptance Testing of A Liquid Oxygen Motor-Pump," Pesco Engineering Report No. 5551, 14 January 1971.

A pump designed for use in LO_2 is described. The basic pump was one originally designed to be immersed completely in the oxygen environment, both GO_2 and LO_2. The unit was satisfactorily tested in this environment under contract to MSC. The proposed unit is to be the same as that tested except that additional protection is incorporated by isolating the electrical wiring from actual contact with the oxygen. This is accomplished by utilizing a sealed can to separate the rotor and static elements.

A-29 Bradshaw, R. D., et. al., "Thermodynamic Studies of Cryogenic Propellant Management," GDC-ERR-AN-1144, December 1967.

This report describes a variety of work performed during 1967 under the Convair IRAD program. Two areas are pertinent to low gravity venting (1) analysis of a vortex tube type separator for receiver tank low-g venting and (2) feasibility analysis of destratification capabilities of a tank wall heat exchanger.

In the case of the vortex tube basic equations were developed for determining pressure drops required to separate vapor and liquid. Parametric data are presented for hydrogen, however, the basic equations may have application to the present LOX vent program. With respect to the tank wall exchanger it was determined that temperature stratification could be reduced by fairly small exchanger coil spacings. This particular analysis was for hydrogen and constant cooling temperatures were assumed with no account of how one actually distributes the vent cooling throughout such an exchanger. This information would only be useful to the present LOX vent program to the extent that the basic equations developed can be applied.

A-30 Warren, R. P. and Anderson, J. W., "A System for Venting a Propellant Tank in the Absence of Gravity," Advances in Cryogenic Engineering, Vol. 12, June 1966.

This document describes the analysis and 1-g testing of a thermodynamic vent system with an exchanger attached to the outside tank wall. The basic test objective was to demonstrate over a range of heat flux that good pressure control could be achieved with liquid at the inlet to the vent while maintaining the same boiloff rate as with a normal gas vent.

The test tank was aluminum of 0.66 m (26 in) dia. with hemispherical heads separated by a 0.254 m (10 in) cylindrical section. Two heat exchangers were employed with one attached in the liquid area and the other in the ullage region. These exchangers could be operated in parallel or separately. The exchanger consisted of 0.00635m (0.25 in) dia. aluminum tubes spiraled around the tank and welded to the outer surface at approximately every 0.203 m (3 in) via connecting aluminum tabs. LN_2 is the test fluid and is extracted from the bottom of the tank for vent through the exchanger(s).

The testing appeared to be successful and the basic concept and exchanger design and control criteria may be applicable to a detailed LOX vent wall exchanger system design.

A-31 Paynter, H. L., "Experimental Investigation of Capillary Propellant Control Devices for Low Gravity Environments," Martin for MSFC, June 1970, NAS8-21259, MCR-69-585, II Volumes.

Results are presented of drop tower tests to investigate; (1) the liquid/gas interfacial stability provided by perforated plate and square-weave screen under accelerations normal and parallel to the surfaces, (2) various passive schemes for preventing the passage of settled propellants, and (3) liquid filling of capillary annuli and the removal of undesired vapor pockets during filling.

Basic hydrodynamic scaling parameters were verified and the data presented would be useful in the detailed design of a surface tension device for the LOX vent application. The basic design of such a system which is used in the initial system comparisons was presented in document A-23.