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FUNDAMENTAL STUDIES IN HYDROGEN BLOW-DOWN AND CRYOGENIC COOLING

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A hydrogen facility designed to conduct experiments relating to blow-down of supercritical hydrogen is described in detail. A theoretical study was conducted to determine the best way of supplying supercritical hydrogen at specified temperature and pressure; the different methods of meeting the required experimental criteria are discussed. The latter part of the study describes experimental arrangements designed and fabricated to study the fluid flow phenomena in free expansion of supercritical/liquid cryogens, and the heat transfer characteristics of cryogens under spray cooling and jet impingement.

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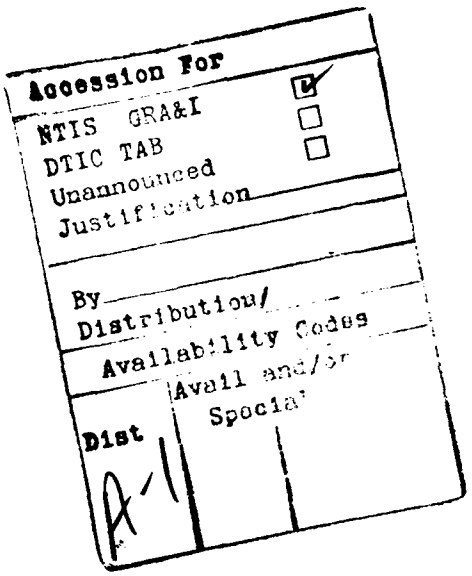


LIST OF ILLUSTRATIONS

NOMENCLATURE

C_p	Heat capacity at constant pressure
g	Acceleration due to gravity
h	Enthalpy
h_{fg}	Latent heat of vaporization
K_{sf}	Constant in Rosenhow correlation
m	Mass
Pr	Prandtl number
q"	Heat flux
T	Temperature
t	Time
V	Volume

Greek

δ	Infinitesimal difference
ρ	Density
μ	Viscosity
σ	Surface tension

Subscripts

e	Exit
g	Gas
i	Inlet
l	Liquid

1. INTRODUCTION

In some proposed Space Defense Initiative(SDI) concepts, high-power devices will be used to provide energy for advanced weapons concepts. In the best of microwave tubes, efficiencies from 70% (klystrons and klystrodes) to over 85% (crossfield devices) have been demonstrated. It appears certain that 10-30% of the input power to the tubes will have to be dissipated as heat. The high-speed computer systems under development also require a very high heat removal capability from the cooling system. A study [1] performed determined that the direct cooling with cryogenic hydrogen is the most promising for space applications. Liquid/supercritical hydrogen fuel already aboard the spacecraft can be used as the heat removal medium for spray/jet impingement/forced convection cooling or flow boiling.

In order to get a better insight into the thermal and fluid flow phenomena occurring during blow-down (i.e., the sudden introduction of supercritical fluid into the heated environment) and cryogenic cooling, experiments have to be designed and performed to obtain measurements which are critical to the performance of the system. From such results correlations will be sought and incorporated into numerical models to provide a prediction scheme for the respective processes.

This first part of the study is concerned with the design and building of a hydrogen test facility. This blow-down facility will be a scale-down version of a conceptual space weapon system. The scaling is done such that the gas dynamics, heat transfer and pressure drop can be realistically simulated. This is essential to the understanding of the thermal and flow instabilities that occur in a full-scale system. The primary objective of this part of the study is to provide a means of supplying supercritical hydrogen to heat generating devices which are initially at ambient conditions and have to be cooled to operating conditions with supercritical hydrogen within short stsrt-up times. As a part of this study, an experimental investigation will be carried out to study the transient fluid flow phenomena associated with free expansion of supercritical and liquid hydrogen.

The second part of the study is concerned with an experimental

investigation into the steady-state/transient heat transfer and structural dynamics aspects of cryogenic cooling. An experimental set-up has been constructed for studying these aspects with N_2 and H_2 as the cryogen (the set-up will initially be used with liquid/supercritical N_2 as the test fluid).

The experimental set-ups for both parts of the study and the hydrogen blow-down facility are described in detail in the Sections 4 and 5.

2. OBJECTIVES OF THE RESEARCH EFFORT

The objective of the research effort is to develop a fundamental understanding of the transient and steady-state phenomena occurring during cryogenic cooling. This will be attained through carrying out experimental studies for the free expansion, the heat transfer, fluid flow and structural dynamics during different forms of cryogenic cooling. Spray cooling and jet impingement cooling will be studied primarily as the heat removal methods. The experimental studies will provide the information needed for formulating predictive correlations/model.

3. BACKGROUND

In order to study the cryogenic cooling and blow-down using supercritical/liquid cryogens, it is essential to study the free expansion process and the cryogenic cooling characteristics.

3.1 FREE EXPANSION PROCESS

A supercritical fluid on being injected into a fully evacuated space undergoes free expansion with resultant temperature drop. The drop in temperature and pressure will cause the fluid to change to liquid or even solid phase depending on the temperature drop associated with this free expansion. This process is shown in the T-S diagram in Figure 3.1.1, the line AF represents the sudden drop in temperature associated with the free expansion process. No previous research exists on the study of this process

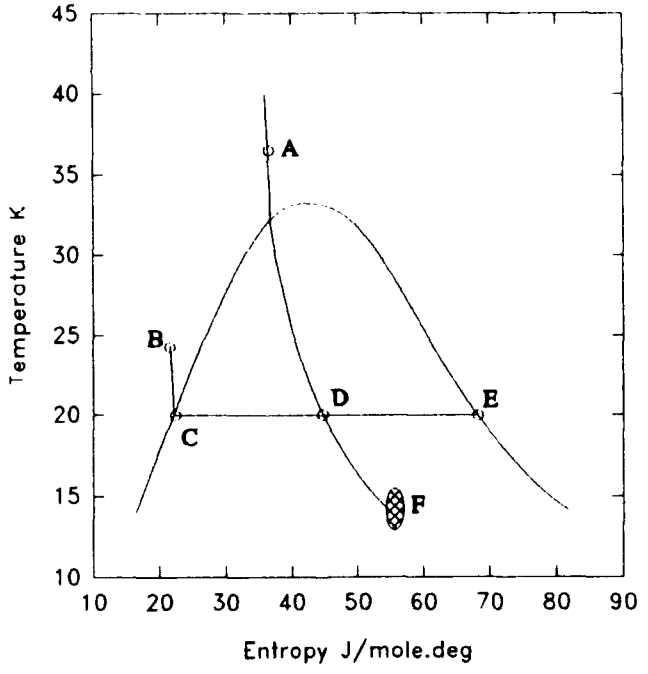


Figure 3.1.1 T-S Diagram

for cryogenic fluids. The phenomena associated with this process of free expansion of supercritical H_2 have to be studied in detail for a

basic understanding of the process and to provide design parameters for applications involving the free expansion process, e.g., start-up of spray cooling, jet impingement in low pressure environment. Referring to Figure 3.1.1 again, the trace BCE represents spray/jet impingement cooling using liquid hydrogen, the high pressure liquid on exiting the nozzle undergoes expansion (BC) and then undergoes phase change on the heat transfer surface (CE). Trace ADE in the Figure represents the spray/jet impingement cooling using supercritical hydrogen, as seen in the Figure the fluid undergoes expansion from A to D and then undergoes phase change in DE. It is important to study the process AD in order to formulate a model for heat transfer characteristics of supercritical cryogen. Hence, the first part of this research effort deals with the experimental study of free expansion of supercritical/liquid cryogens.

3.2 CRYOGENIC COOLING

Regarding heat transfer and fluid flow studies in cryogenic cooling, some studies have been performed dealing with liquid cryogen under pool boiling [2, 3] and flow boiling [4, 5] conditions. All of these studies indicate that correlations derived for normal fluids do not apply to cryogenic liquids. There have been no studies performed with supercritical cryogens. Very limited experimental data exist on the high heat flux removal methods of spray cooling and jet impingement cooling utilizing cryogens. Some studies have been performed for spray cooling with liquid nitrogen in the Leidenfrost region [6, 7], but these studies are not very relevant for the high heat flux region of spray cooling which occurs before the Leidenfrost temperature. All this indicates a need for experimental studies under non-film boiling conditions.

This research effort is directed towards studying the heat transfer characteristics under normal spray cooling/jet impingement conditions for liquid nitrogen and liquid hydrogen. Since these modes of heat transfer depend a lot on the flow conditions, a detailed study of the droplet distribution and flow situations under free expansion and normal conditions will be carried out. Another region of concern in cryogenic cooling is the enormous temperature gradients which occur in transient

conditions (e.g., start-up), these can cause substantial thermal stresses in the cooled device; an investigation into the structural dynamics of cryogenic cooling will, therefore, be carried out.

4. HYDROGEN BLOW-DOWN

As explained in the previous sections an experimental study into the blow-down of liquid/supercritical hydrogen is needed in order to formulate design criteria for the actual systems utilizing cryogenic cooling. A test facility was designed for the purpose of conducting a small scale blow-down test. Following is a description of the test facility and the supply arrangement for supercritical hydrogen needed for the tests.

4.1 TEST FACILITY

Taking into account National Fire Protection Agency (NFPA) regulations [8-10], the extremely high flammability of hydrogen demands that the blow-down test-bay be isolated from population centers. The hydrogen facility has been located on a farm 3 miles out of the city limits. The site was surveyed and bulldozed to build earth embankments which will act as explosion absorbers-deflectors. Figure 4.1.1 illustrates the elevation and plan views of the overall facility. The primary concern behind the design is fire and explosion safety.

4.1.1 Test-Bay

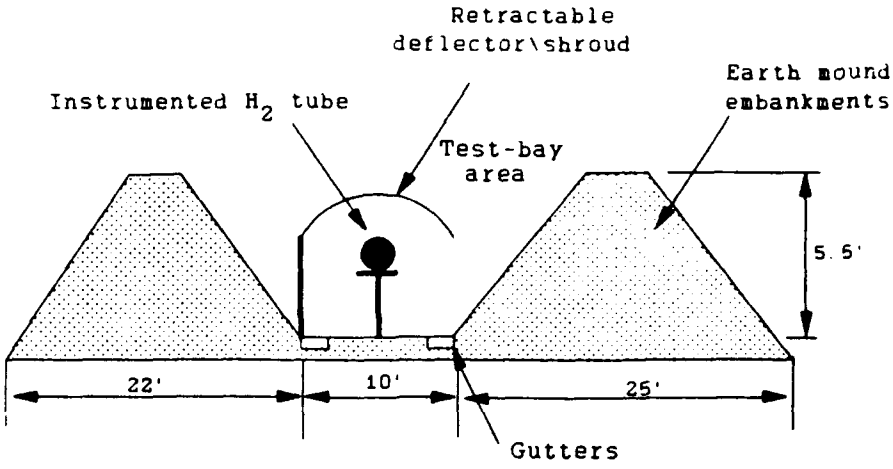
The test bay (see Figure 4.1.1), is located within raised embankments of soil (90 ft x 22 ft x 5.5 ft). The design of the embankments will:

1) Absorb into the soft soil any flying matter from an explosion.
2) Direct force of explosion into atmosphere.

The primary embankments are 70 feet in length with secondary 15 feet extensions set at an angle at either ends to provide 360° coverage at ground level.

The test bay can be visualized to be a trench 70 feet long. The blow-down tube(BT) will be supported on stands (Figure 4.1.2). Covering the tube will be retractable, semicircular steel shrouds to trap any flying fragments in the direction of the open sky.

Elevation view



Plan view

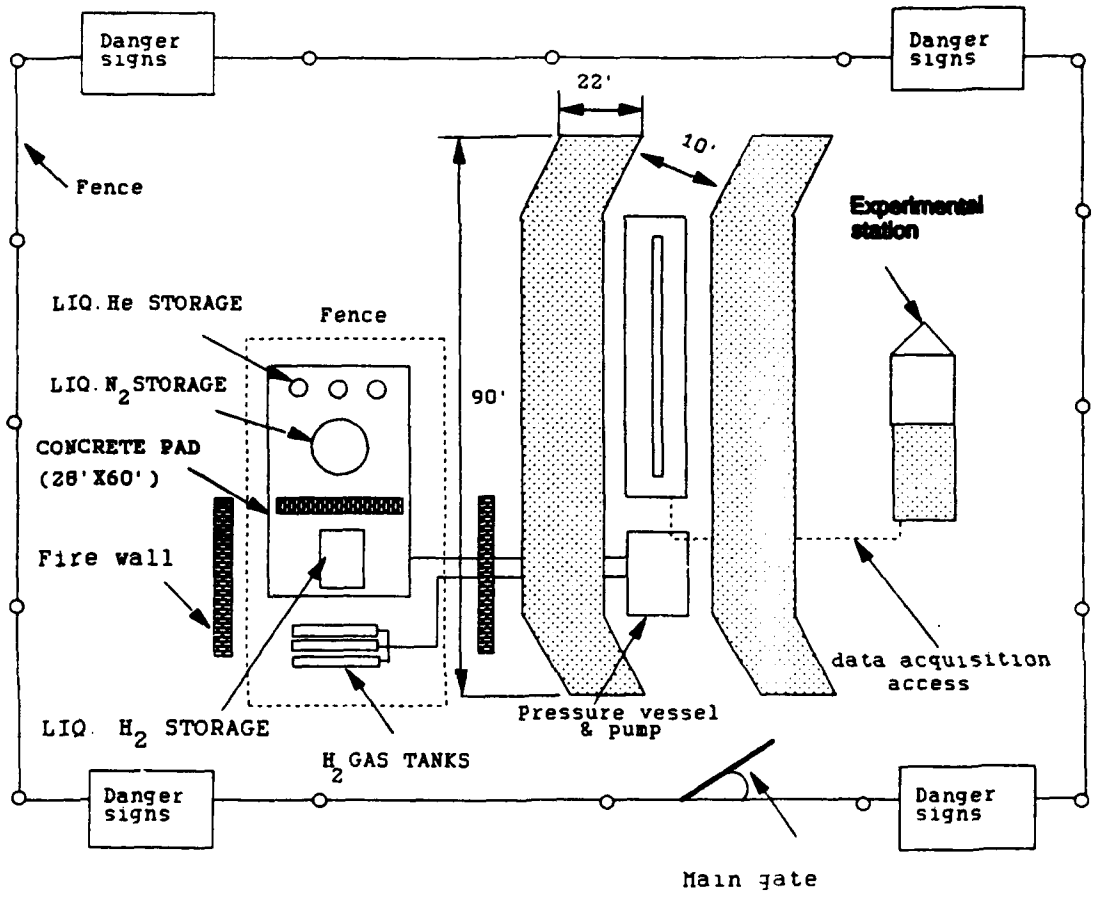


Figure 4.1.1 Test Facility

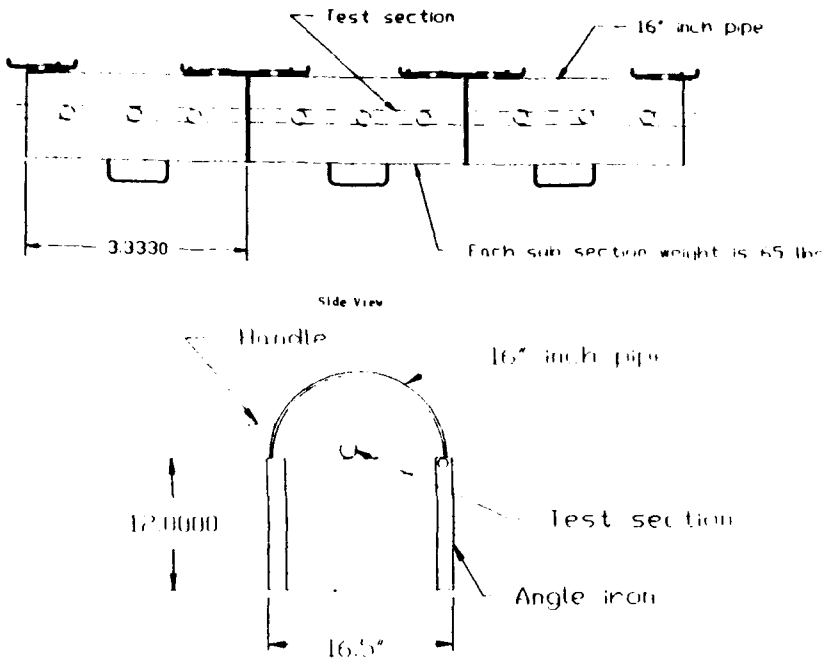


Figure 4.1.2 Blow-down Tube Supports

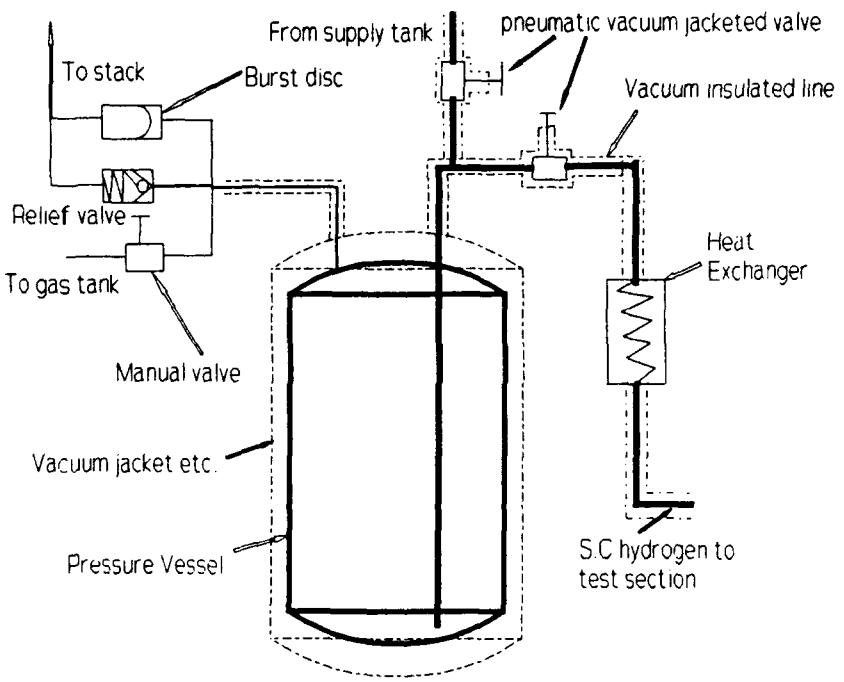


Figure 4.1.3 High Pressure H_2 Supply Set-up

For similar reasons the pressurized cylinder (see Figure 4.1.3) will be located within the trench with the necessary support and drainage facilities. Any fragments expelled due to fracture of the pressure vessel will be absorbed into the earth, and the rest directed into the sky. Storage tanks, (see Figure 4.1.1), and dewars located in a separately fenced area will be used to fill the pressure vessel via transfer lines. Cryogenic transfer lines will be used for transfer of the liquid hydrogen from the storage tanks to the pressure vessel.

An experimental station shown in Figure 4.1.4 housing the necessary monitoring equipment, computers, data acquisitions and personnel will be located on the side and external to the test-bay.

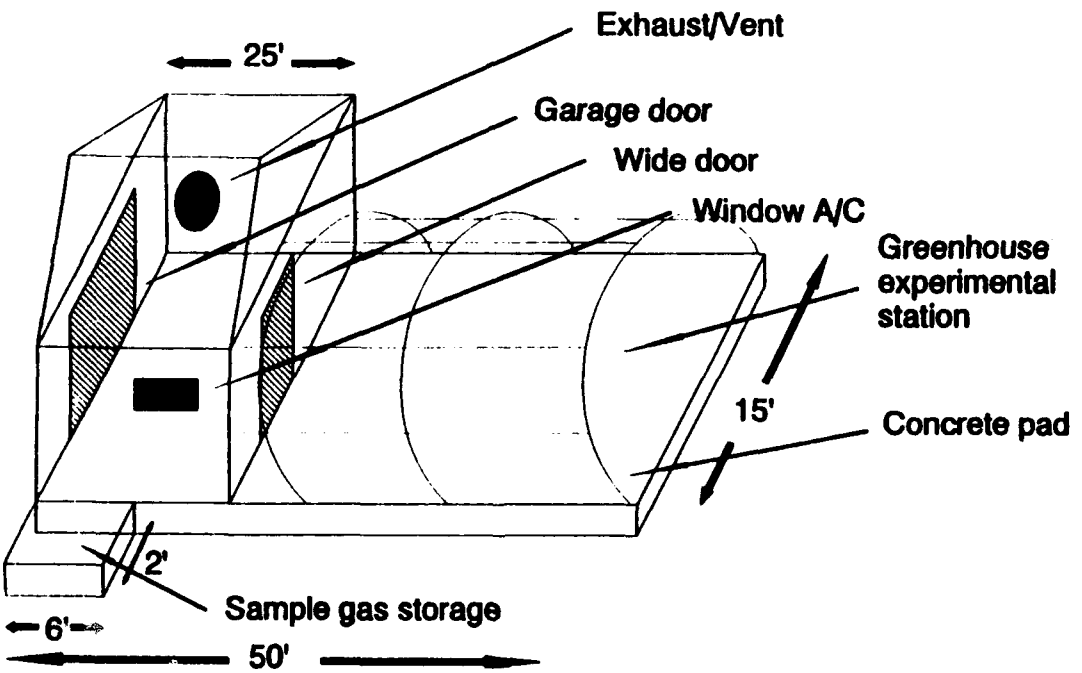


Figure 4.1.4 Experimental Station

The entire area will be cordoned off by a fence with the appropriate safety and warning signs posted to gain attention of any trespassers.

4.2 SUPPLY OF SUPERCRITICAL HYDROGEN

This project is directed towards the study of the thermophysical phenomena occurring when supercritical cryogenic hydrogen is introduced into a manifold (BT) which is initially at ambient temperature and vacuum pressure conditions. The supercritical hydrogen on flowing through the BT cools the walls of the tube and at the same time raises the pressure within the tube. After a certain period equilibrium conditions in temperature and pressure are reached. This transient period from start to equilibrium is the time period of interest. The time of the experiment is finite and is of the order of a few tens of seconds. The minimum quantity of supercritical hydrogen (at 35 K and 30 bars) required is estimated to be of the order of 275 liters.

The experiments involving supercritical hydrogen and helium can be performed using a flow through arrangement or a closed-loop arrangement. The following sections describe each method in detail.

4.2.1 Flow-through Arrangement

Supercritical hydrogen/helium is required at 30 bars and 35 K for the experiments. One way to achieve this is to compress liquid helium/hydrogen by some means to 30 bars and then increase its temperature to 35 K before it enters the test section. In the flow-through arrangement a pressure vessel is used to supply hydrogen/helium at 30 bars and a separate heating arrangement is used to raise its temperature to 35 K (Figure 4.1.3).

The pressure vessel shown in Figure 4.1.3 has to be charged with liquid helium/hydrogen at atmospheric pressure from the supply tank. Then the pressure in the vessel has to be raised to and maintained at 30 bars for the experiments. The pressure inside the vessel can be raised by three different methods; each will be described below in detail.

4.2.1.1 Heated Pressure Vessel

Here the pressure vessel is equipped with a heat exchanger (see Figure 4.2.1), which raises the temperature of the fluid inside, thereby increasing the pressure to 30 bars. If the vessel is initially full of liquid hydrogen at atmospheric pressure its temperature has to be raised

to about 23 K to raise its pressure to 30 bars. However, as the supercritical hydrogen starts exiting the vessel the density of the remaining fluid starts falling. This means that the temperature of the fluid has to be increased to maintain the pressure at 30 bars. After the temperature inside the vessel reaches 35 K, no more hydrogen can be pumped out at the required conditions. This implies that only a certain portion of fluid in the vessel can be used in the experiments;

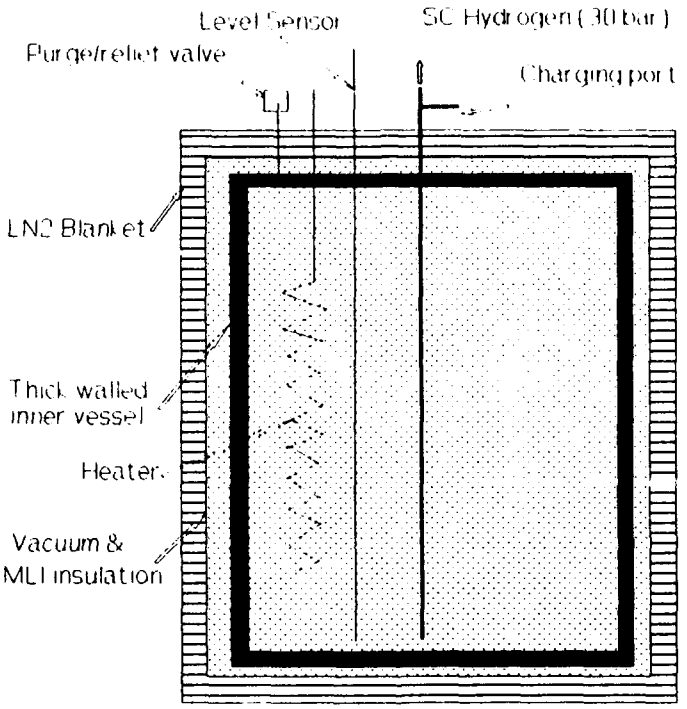


Figure 4.2.1 Heated Pressure Vessel

calculations showed that for a 0.2 kg/s hydrogen flow rate for 90 seconds the volume of the vessel needed will be about 900 liters.

The large size of the vessel needed and the lack of uniformity of exit temperature are the main disadvantages of this method.

4.2.1.2 Vessel with a Piston

Another way of maintaining the pressure in the vessel constant is by having a piston arrangement shown in Figure 4.2.2. Here the pressure is maintained constant at 30 bars by having compressed helium at 30 bars on one side of the piston. Thus the pressure of the outgoing fluid is always constant at 30 bars. The disadvantage of this system is that the

diameter of the vessel has to be kept small for ease of machining and, therefore, the length of the vessel increases with the result that three vessels of 2-m length and 0.3-m diameter have to be used to keep systems physically manageable. Also problems of adequate lubrication for the sliding piston are not easily satisfied.

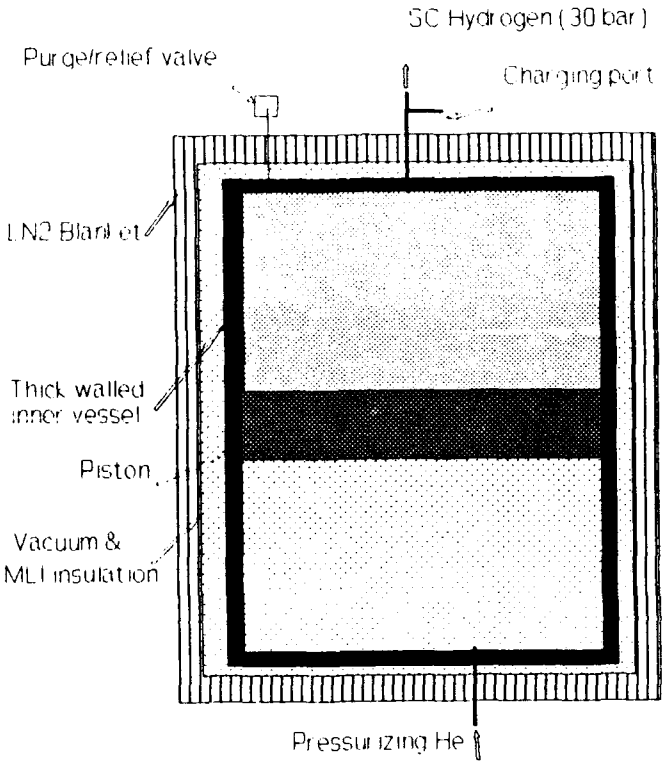


Figure 4.2.2 Pressure Vessel with Piston

4.2.1.3 Gaseous Piston Vessel

This method involves using compressed hydrogen/helium gas at room temperature to raise the pressure of the liquid hydrogen/helium in the pressure vessel. Figure 4.2.3 shows the schematic diagram of the arrangement. Since this method involves mixing of hydrogen gas at room temperature and liquid hydrogen at 20 K, a detailed thermodynamic analysis was necessary to determine the condition of the exiting fluid.

Figure 4.2.4 shows the control volume considered in the thermodynamic calculations pertaining to the problem of pumping hydrogen using compressed hydrogen gas. It is assumed that the vessel is perfectly insulated making the process adiabatic. Since no work is done by the gas inside the vessel and no heat flows in or out of the system,

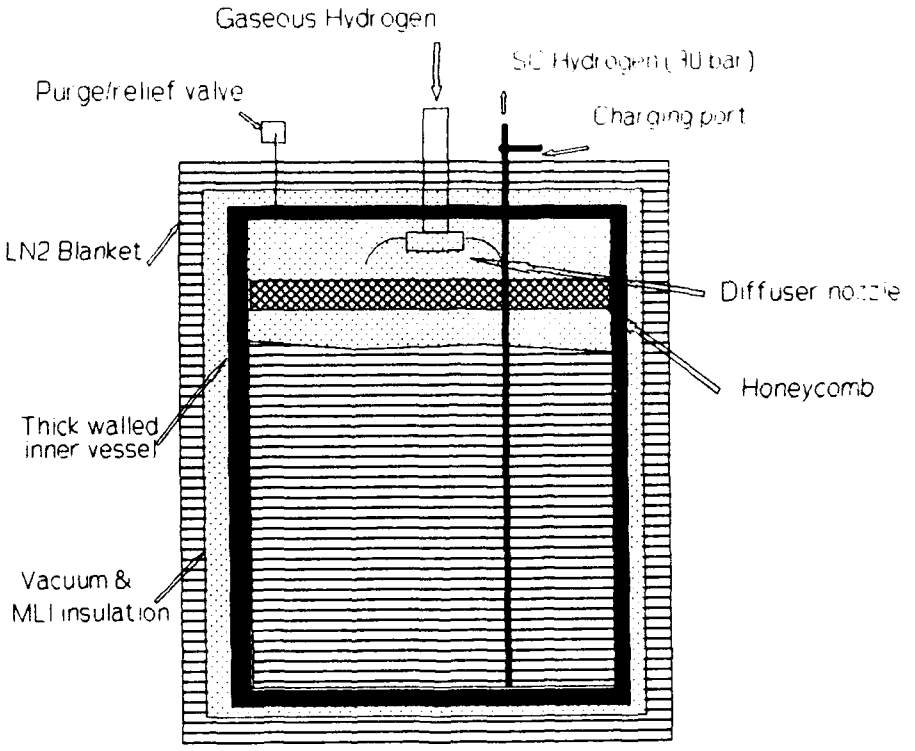


Figure 4.2.3 Gaseous Piston Vessel

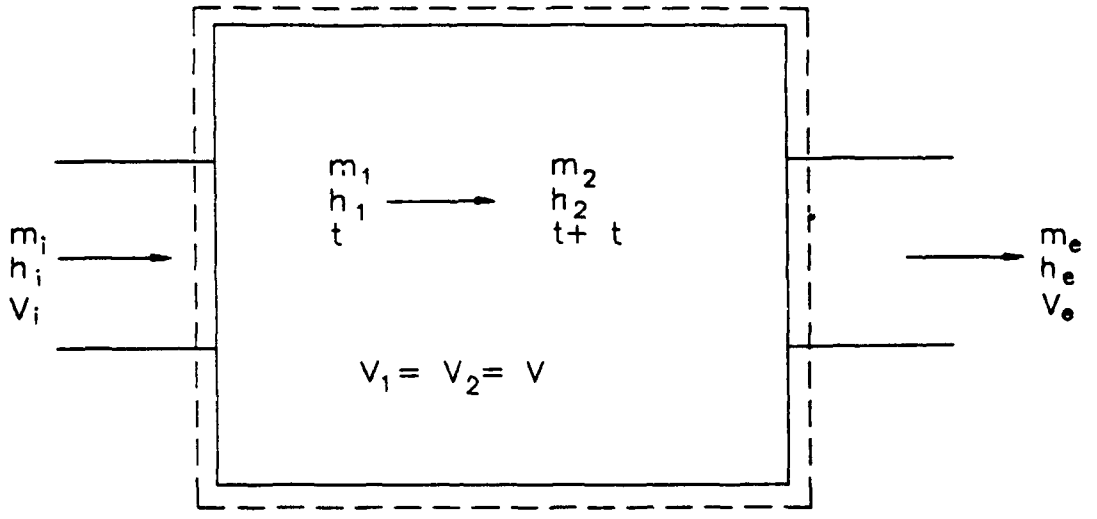


Figure 4.2.4 Control Volume

it essentially means that the enthalpy of the system is conserved. Assuming initial and final conditions are at steady state, we can write (for the total enthalpy of the control Volume):

Enthalpy in + Initial Enthalpy - Enthalpy out = Final Enthalpy

And from conservation of mass, we get:

Mass in + Initial Mass - Mass out = Final Mass

Considering the first equation we have three unknowns (initial enthalpy is the only known parameter). Since enthalpy of outgoing hydrogen will vary with time, the first equation can be written as:

$$\int_1^2 h_i \, dm_i + m_1 h_1 - \int_1^2 h_e \, dm_e = m_2 h_2 \qquad (4.2.1)$$

Writing this equation in terms of volume flow rate of incoming gas and mass flow rate of outgoing supercritical hydrogen, we obtain:

$$\int_1^2 h_i \rho_i \frac{dV_i}{dt} dt + \rho_1 V_1 h_1 - \int_1^2 h_e \frac{dm_e}{dt} dt = \rho_2 V_2 h_2 \qquad (4.2.2)$$

Since $V_1 = V_2 = V$, where V is the volume of the vessel, the conservation of mass equation in terms of the above parameters becomes:

$$\int_1^2 \rho_i \frac{dV_i}{dt} dt + \rho_1 V - \int_1^2 \frac{dm_e}{dt} dt = \rho_2 V \qquad (4.2.3)$$

To solve this problem we consider discrete time steps of duration δt. Considering the state of the hydrogen in the vessel at the time t as state 1 and at time $t + \delta t$ as state 2 and rearranging the terms in the conservation of mass equation, we get:

$$\rho_i \frac{dV_i}{dt} \delta t = (\rho_2 - \rho_1) V + \frac{dm_e}{dt} \delta t \qquad (4.2.4)$$

Now, assuming that the incoming gas mixes perfectly with the hydrogen already in the vessel during each time step, the state at the end of each time-step can be considered to be in a quasi-steady state. The conservation of enthalpy equation for each time step can then be written

as:

$$\rho_i h_i \frac{dV_i}{dt} \delta t + \rho_1 V h_1 - h_e \frac{dm_e}{dt} \delta t = \rho_2 V h_2 \qquad (4.2.5)$$

Substituting for the first term from the conservation of mass equation and rearranging we obtain:

$$\rho_2 (h_2 - h_i) V - \rho_1 (h_1 - h_i) V + (h_e - h_i) \frac{dm_e}{dt} \delta t = 0 \qquad (4.2.6)$$

From the tables it can be seen that the relation between density and enthalpy between density = 71 kg/m^3 to density = 30 kg/m^3 can be approximated by a linear relationship. This relationship can be written as:

$$h = -8.883\rho + 956.50 \qquad (4.2.7)$$

where ρ is in kg/m^3 and h is in kJ/kg.

Considering the enthalpy of outgoing hydrogen during the time step as equal to the enthalpy of hydrogen in the vessel at the beginning of the time step (h_1), we can write:

$$\rho_2 (h_2 - h_i) - \rho_1 (h_1 - h_i) + \frac{(h_1 - h_i)}{V} \frac{dm_e}{dt} \delta t = 0 \qquad (4.2.8)$$

Substituting for enthalpy of incoming gas h_i from tables and h_1, h_2 from equation 4.2.7 and considering a time step of 1 sec we get:

$$\rho_2 (-8.883\rho_2 + 956.5 - 4235) - \left(\rho_1 - \frac{1}{V} \frac{dm_e}{dt}\right)(-8.883\rho_1 + 956.5 - 4235) = 0 \qquad (4.9)$$

This is a quadratic equation in terms of density of state 2 and it can be easily solved for each time step.

A computer program was written to compute the density of hydrogen in the vessel and the hydrogen gas input during each time step using the above equations. The results presented in Figure 4.2.5 show the variation in density of outgoing hydrogen with time for different vessel

sizes at a mass flow rate of 0.2 kg/s. The total volume of gaseous hydrogen needed to compress and eject the supercritical hydrogen ranged from 0.8 to 1.15 m^3 (at 30 bars) for varying vessel sizes. However, all these results are obtained with the assumption of uniform mixing. In actuality there is no reason to expect total mixing but the formation of a stratified medium. In that case, the hydrogen near the top would be expected to mix with incoming hydrogen, whereas, that below would remain practically unaffected; consequently the variation in density of the outgoing hydrogen would be significantly smaller.

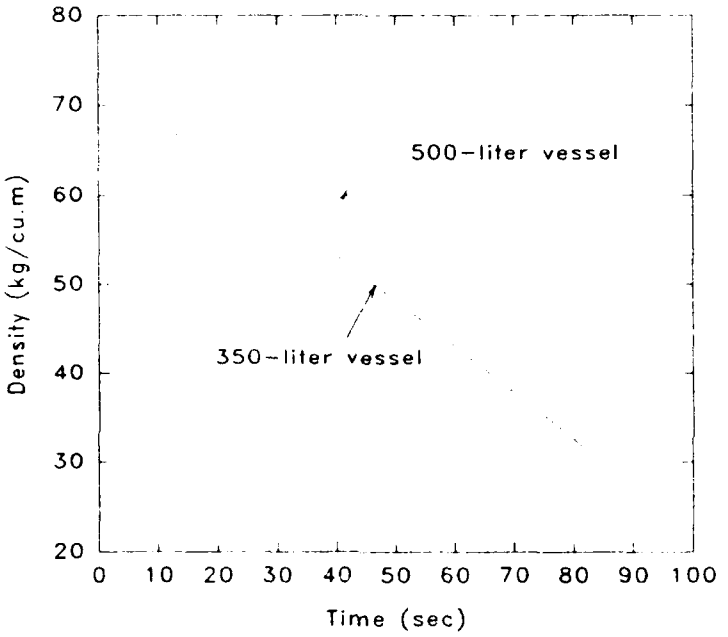


Figure 4.2.5 Variation in Exit Density

The pressure vessel used in the flow-through arrangement will be designed to minimize the mixing of incoming gaseous hydrogen and supercritical hydrogen in the vessel (see Figure 4.2.3). Based on the calculations, it was decided that a 500-liter vessel is needed for the blow-down experiments. The pressure vessel will be designed to ASME code with vacuum and multilayered insulation (MLI) blankets and the necessary feedthrough accoutrements. The pressure vessel is initially successively evacuated and flushed several times with high purity nitrogen. The vessel is then chilled using liquid nitrogen. Finally, a vacuum is again drawn and then liquid hydrogen at 1 bar and 20 K (ρ = 71 kg/m^3) from storage tanks is loaded into the pressure vessel via

transfer lines.

4.2.2 Closed-Loop Arrangement

For the liquid helium tests the flow-through system will have to be charged with liquid helium and this requires the supply vessel to be cooled down to 4 K; considering the extremely low heat of vaporization of helium a large quantity of it is required for this purpose. Now, the helium required for the tests has to be at 35 K; this means that we have to waste a lot of helium just to charge the vessel with liquid helium at 4 K when we don't really need it at such a low temperature. Thus, it would be preferable to employ a method that eliminates this cooling and

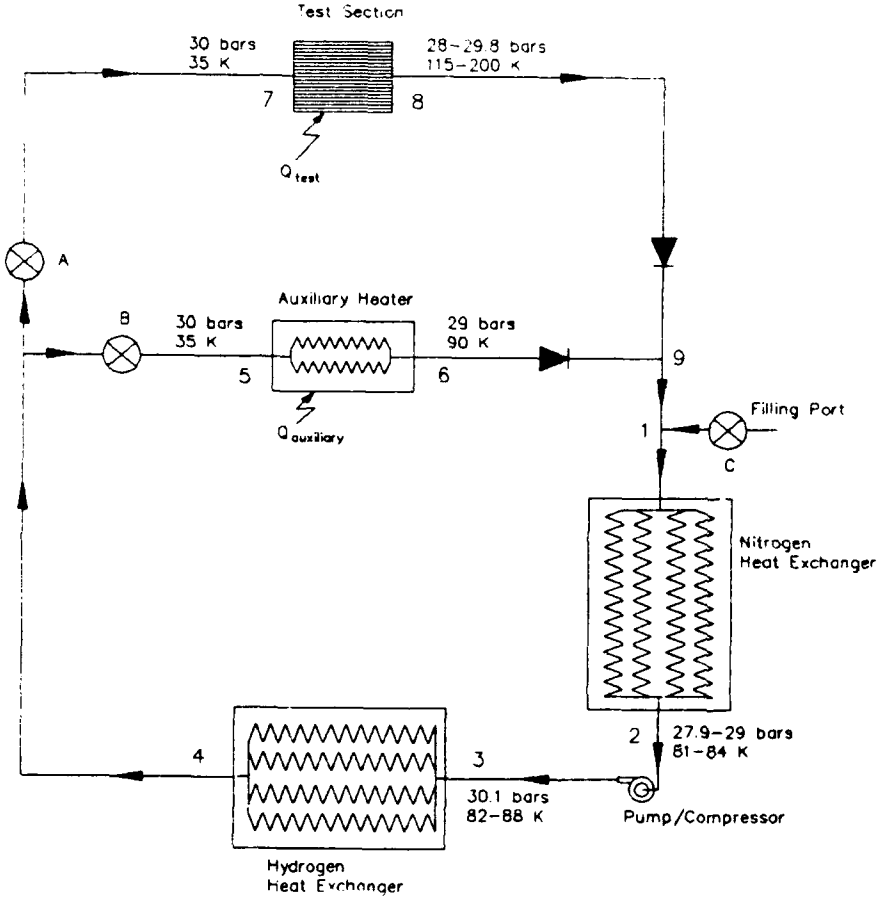


Figure 4.2.6 Closed-Loop Arrangement

reheating. A method which does precisely this is a closed-loop system in which gaseous hydrogen/helium is cooled down to 35 K thus eliminating the costly liquid helium requirement. Another motivation for using a closed-loop arrangement is the controllability of such a system compared to the open loop system.

The schematic diagram of the closed loop design for the tests is shown in Figure 4.2.6. The gaseous hydrogen/helium at room temperature and 30 bars is charged into the loop from valve C. Upon entering the system at 1, the gas passes through a shell and tube heat exchanger with liquid nitrogen on the shell side and gaseous hydrogen/helium on the tube side. The hot gas flowing inside the heat exchanger tubes loses heat to the liquid nitrogen which boils off on the outside. The gas exiting the heat exchanger at 2 is compressed to the required pressure by the pump and it enters a second heat exchanger at 3. This heat exchanger is identical to the first one, except the shell side contains liquid hydrogen at atmospheric pressure and 20 K. The gas loses heat to the liquid hydrogen and comes out of the heat exchanger at 4 whereupon it comes to a junction from where it can either go to 5 or 7 depending on the status of valves A and B.

During the charging process valve A is kept closed and B is kept open, so the gaseous/supercritical hydrogen/helium goes to 5 during the charging process. At 5 it enters a heater which will be used to heat the fluid. The fluid on coming out of the heater section at 6 passes through a check valve and mixes with charging gas at 1; the heat input to the heater section is controlled to maintain the temperature of this mixed fluid above liquid nitrogen temperature to avoid freezing of nitrogen in the first heat exchanger.

The fluid is kept circulating in this loop (1-2-3-4-5-6-9-1) till the hydrogen/helium exiting the second heat exchanger at 4 is at the temperature and pressure needed for the tests. On reaching the desired condition at 4, valve A is opened and valve B closed enabling the fluid to go through the test section instead of the auxiliary heater.

During the tests the test section is heated to simulate actual load conditions; the fluid passing through the test section gets heated and goes to the liquid nitrogen heat exchanger via 8, 9 and 1. The liquid nitrogen heat exchanger is designed to cool the fluid entering it to 84K

K or lower at 2; the pump is then used to increase the pressure of the fluid to over 30 bars to account for pressure drop in the lines. This increase in pressure would increase the fluid temperature to 88 K or lower at 3, the hydrogen heat exchanger then cools the fluid to below 35 K so that the fluid on reaching 7 is at 35 K. Thus the conditions at 7 are maintained constant by controlling the pump and the hydrogen heat exchanger. The following sections describe the design of the two heat exchangers and the pump requirements.

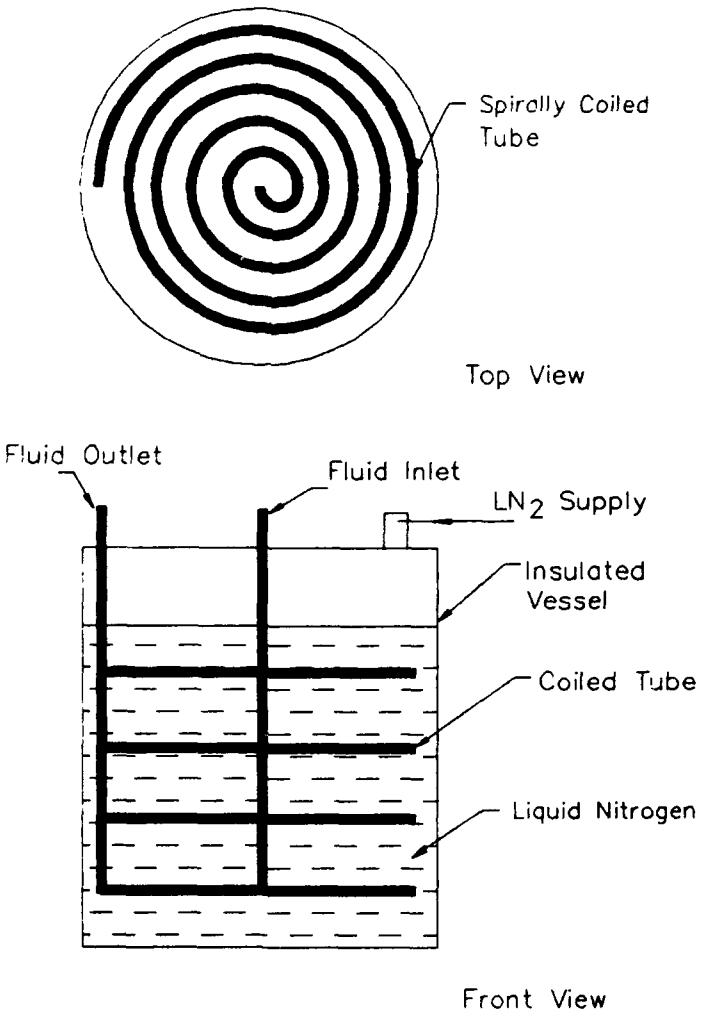


Figure 4.2.7 Heat Exchanger Design

4.2.2.1 Liquid Nitrogen Heat Exchanger

The liquid nitrogen heat exchanger was designed to cool the fluid exiting the test section at various conditions to at least 84 K. This is a shell and tube type heat exchanger with the test fluid passing through the tubes and the liquid nitrogen at atmospheric pressure on the shell side. The schematic diagram of this heat exchanger in Figure 4.2.7 shows the tube configuration. The tube bundle in the heat exchanger is made up of a number of tubes, each coiled in a spiral in the horizontal plane. These spiral sections are connected in parallel to avoid excessive fluid velocity and pressure drop.

The fluid entering the tubes of the heat exchanger is always above the temperature corresponding to the critical heat flux (CHF) for liquid nitrogen (90 K). Thus, in almost all cases we will have film boiling on the outside of the tubes for the initial portion of the tubes. The film boiling heat flux is nearly $1/8^{th}$ the maximum nucleate boiling heat flux (CHF) and hence it is preferable to minimize the length over which it occurs by somehow reducing the tube outside temperature. The temperature on the external surface of the tubes depends on the bulk temperature of the fluid in the tube T_b, the heat transfer coefficient on the inside h_i, the conductivity of the tube material k_t, and the heat transfer coefficient on the outside h_e. Once the tube material is chosen the only way to keep the tube outside temperature low is by reducing the heat transfer coefficient h_i. A good way to reduce h_i is to decrease the Reynold's number on the inside by reducing the mass flow rate in the tube. This is done by splitting the flow among parallel sections entering the heat exchanger. However, h_i should not be reduced so low that it goes to the other extreme of not allowing the liquid to boil off the tube.

A computer program was written to calculate the length of the tube in each spiral section given the number of these parallel sections. The program calculates the temperature and pressure profiles in each tube by dividing it into small elements over which the conditions are assumed uniform. The bulk temperature of the fluid entering the first element is known from the exit conditions of the test section. The mass flow rate in the tube is determined by the number of parallel sections; given the mass flow rate and the bulk temperature, the heat transfer

coefficient h_i is calculated. The heat flux on the outside is calculated from Rosenhow's correlation [11]:

$$\frac{C_{pl}(T_w - T_{sat})}{h_{fg}} = K_{sf}\left\{\frac{q''}{\mu_l h_{fg}}\left[\frac{\sigma}{g(\rho_l - \rho_g)}\right]^{\frac{1}{2}}\right\}^{\frac{1}{3}} Pr_l^{1.7} \qquad (4.2.10)$$

here, K_{sf} is a constant depending on the surface-liquid combination; from previously published pool boiling data for liquid nitrogen it was found to be equal to 0.01 [11]. Since q" depends on the external surface temperature we have to iterate till the heat flux on the inside surface matches the heat flux from the outside surface of the tube. Thus, the heat transferred in each element can be determined and a temperature and pressure profile of the fluid inside the tube be obtained.

Using the computer program the number of parallel sections was increased till it was felt that the addition of another section would not help much. Based on the results it was decided that the heat exchanger should have four parallel stainless steel tubes of 1/2-in ID and 5/8-in OD with a tube length of 5.8 m. The temperature distributions inside the heat exchanger tubes for this configuration at various test section exit conditions are shown in Figure 4.2.8.

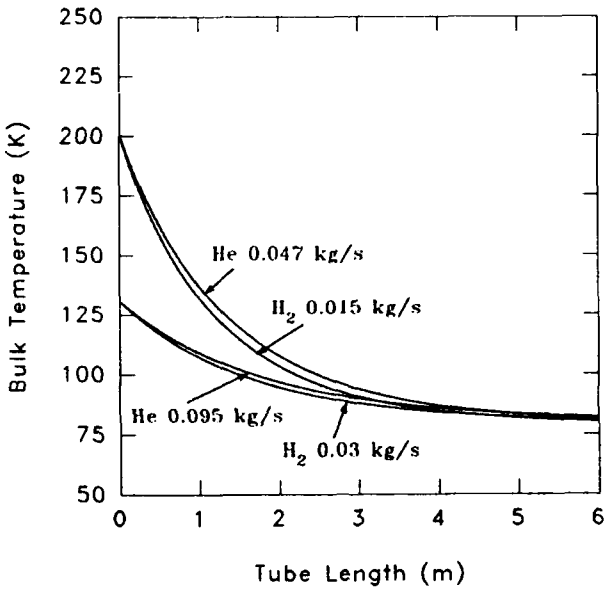


Figure 4.2.8 Temperature Distribution in N_2 Heat Exchanger

4.2.2.2 Liquid Hydrogen Heat Exchanger

The liquid hydrogen heat exchanger is designed with the same criterion as the liquid nitrogen heat exchanger. The temperature and pressure of the fluid entering this heat exchanger do not vary much compared to the previous. The calculations showed that four tubes similar to the other heat exchanger with lengths of 5.5 m each would be sufficient for this heat exchanger. Figure 4.2.9 shows the temperature distribution in the tubes under the expected inlet conditions.

4.2.2.3 Pump Specifications

The pump will be used to raise the pressure of the fluid to compensate for the pressure drop through the test section, the lines and the heat exchanger tubes. The total pressure drop through the loop is nearly 2.75 bars and the maximum volume flow rate through the pump is about 3.75 liters/sec.

If a suitable pump is not found, the other alternative would be pursued.

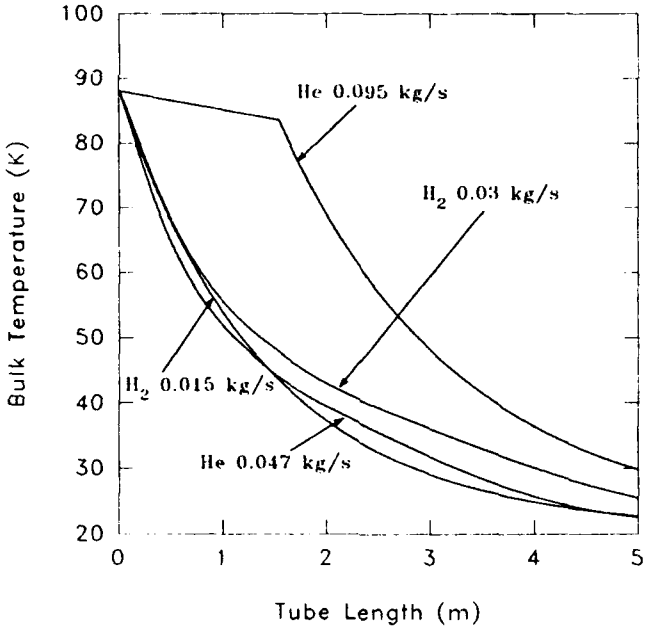


Figure 4.2.9 Temperature Distribution in H_2 Heat Exchanger

4.3 FLOW RATE MEASUREMENT

The flow rate of hydrogen is determined by performing a mass

balance with gas and liquid flow measurements from the liquid recuperation and gas disposal system described in "Hydrogen Disposal" and Figure 4.3.1.

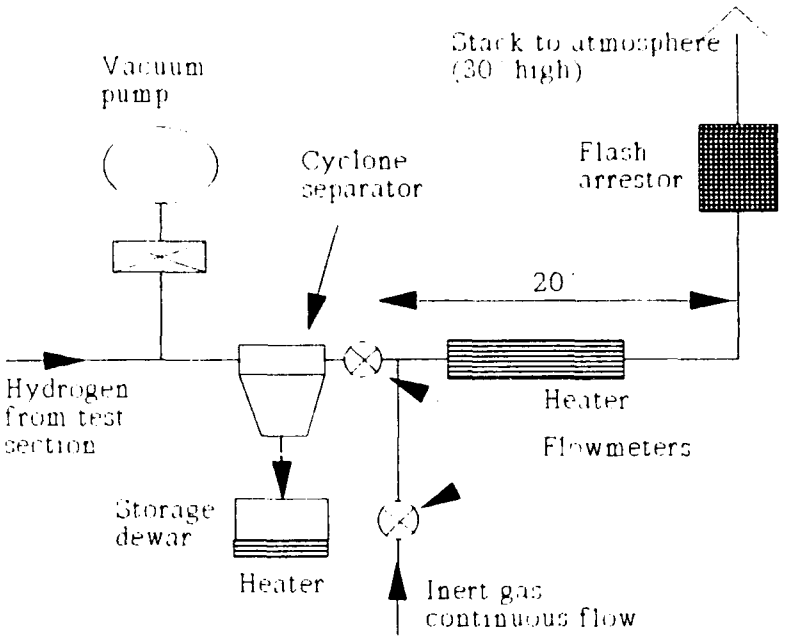


Figure 4.3.1 Hydrogen Disposal System

4.4 HYDROGEN DISPOSAL

Referring to Figure 4.3.1, the supercritical hydrogen on passing through the BT takes up heat, some of it changing phase to liquid and the rest to gas. The unspent liquid is recovered by separation in a centrifuge cyclone type separator and collected in a dewar. The gas is allowed to escape into the stack located around 20 feet from the test bay. The hydrogen gas being much lighter than air, will rise into the atmosphere. A flash arrestor prevents any flashback into the exhaust lines. The lines leading from the end of the BT to the stack are continuously flushed with nitrogen. Flow meters monitor the individual flow rates of the gaseous nitrogen and hydrogen. After the experiment the unspent liquid hydrogen collected in the separator dewar is heated and the ensuing gas flow measured. From the above flow rates a mass balance of the hydrogen leaving the test section as liquid and gas can be obtained. The above parameters are strongly a function of the diameter d of the tube and the mass flowrate of hydrogen . This implies

that if the diameter is reduced by a factor of 5 from d = 5 in to 1 in, then the mass flowrate can be reduced by a factor of 25 from 5 kg/s to 0.2 kg/s.

4.5 EXPERIMENTAL STATION

The experimental station, shown in Figure 4.1.4, will consist of a well insulated permanent building 25 ft x 15 ft with a drive-in door at one end and a wide access door at the other. This building will house the personnel, monitoring instrumentation, and the electrical mains. The building will sit on a 6 in thick concrete extended an extra 25 ft. On the extended 25 ft will be located a top ventilated plastic greenhouse. Ultra-small scale experiments with liquid hydrogen will be performed in this external space. The plastic envelope provides sufficient shelter yet least resistance to any possible explosions.

4.6 LIQUID HYDROGEN AND HELIUM SUPPLY

Three manufacturers of liquid hydrogen were contacted, namely:

1) Airco Products
2) Union Carbide
3) Air Products

The liquid hydrogen would be supplied in a 1500-gallon permanent storage vessel which can be filled periodically. Rental charges on the vessel are $1200/month. The price of the liquid hydrogen in these quantities is $2.08/gal. The liquid helium would be supplied in 600-liter portable dewars at a price of $3.75/liter. The liquid hydrogen storage vessel would be mounted on a 1ft thick concrete pad adjoining the blow-down facility (see Figure 4.1.1). Fire walls will separate the vessel from any other dewars. The pad will be separately fenced with provisions for supply truck access.

5. FUNDAMENTAL STUDIES IN FREE EXPANSION AND CRYOGENIC COOLING

In order to get a better insight into the thermal and fluid flow phenomena occurring during cryogenic cooling, experiments have to be designed and performed so as to obtain measurements which are critical to the performance of the system. From such results correlations will be sought and incorporated into numerical models to provide a prediction scheme for the various modes of heat removal.The first part of this study deals with experimental investigation into the transient fluid flow phenomena associated with the free expansion of high pressure liquid or supercritical hydrogen. The second part deals with experimental study of the heat transfer, fluid flow and structural dynamics characteristics of cryogenic cooling. The experimental set-ups designed and constructed for the purpose are described in the following sections.

5.1 FREE EXPANSION STUDY

In order to study the phenomena associated with free expansion process we need to be able to inject a limited quantity of supercritical H_2 into a space evacuated to the highest degree possible. The critical point for H_2 is at 13.16 bar and 33.23 K, therefore, the system has to be capable of supplying hydrogen in a sufficient range around these conditions. It was determined that the system should be capable of injecting H_2 at up to 30 bar and at temperatures as low as 20 K in order to obtain enough data for the analysis. The other criterion for system design was the ability to observe the phenomena occurring in the free space with a high speed digital camera in order to study the process in detail. Another point of interest is the pressure fluctuation inside the free space upon the injection of supercritical H_2. The experimental system described below was designed considering all these requirements.

5.1.1 Experimental System

The experimental system was designed with a glass (pyrex) vacuum chamber into which the high pressure liquid or supercritical H_2 is injected via a nozzle. The high pressure fluid is injected into a

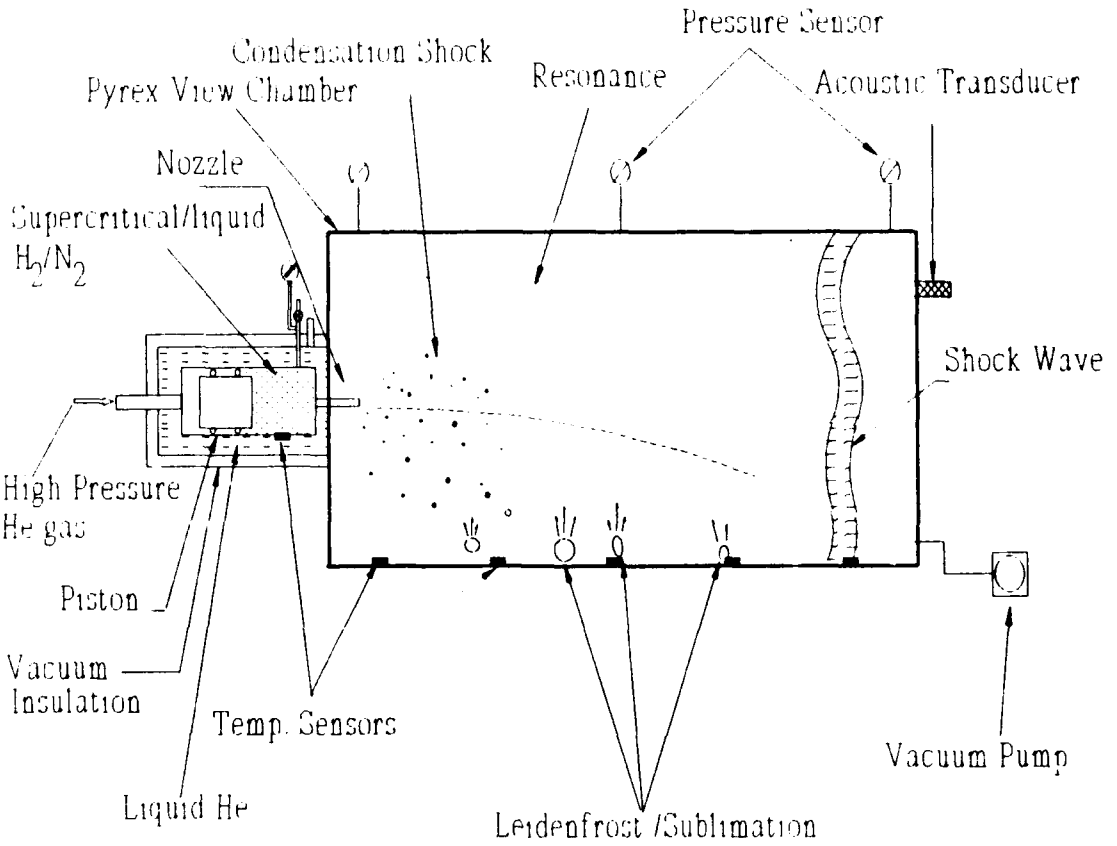


Figure 5.1.1 Experimental Schematic for Free Expansion Study

vacuum using a very small piston pump with gaseous He or N_2 as the driving force for the piston. The schematic of this experimental set-up is shown in Figure 5.1.1. The difficulty in obtaining liquid hydrogen in small quantities necessitated the use of a hydrogen liquefier employing liquid helium as the coolant. The detailed assembly drawing of the apparatus in Figure 5.1.2 shows the arrangement used for liquefying the hydrogen and pumping it at a high pressure into the vacuum chamber. Gaseous hydrogen is fed into the innermost chamber containing the piston (the pump) and liquefied there by means of liquid helium in the surrounding shell. The liquefied hydrogen is then pressurized to the required pressure by pumping high pressure He or N_2 gas on the other side of the piston. When the required temperature and pressure conditions inside the pump are reached, a valve connecting the pump to the injector nozzle inside the vacuum chamber is opened and high

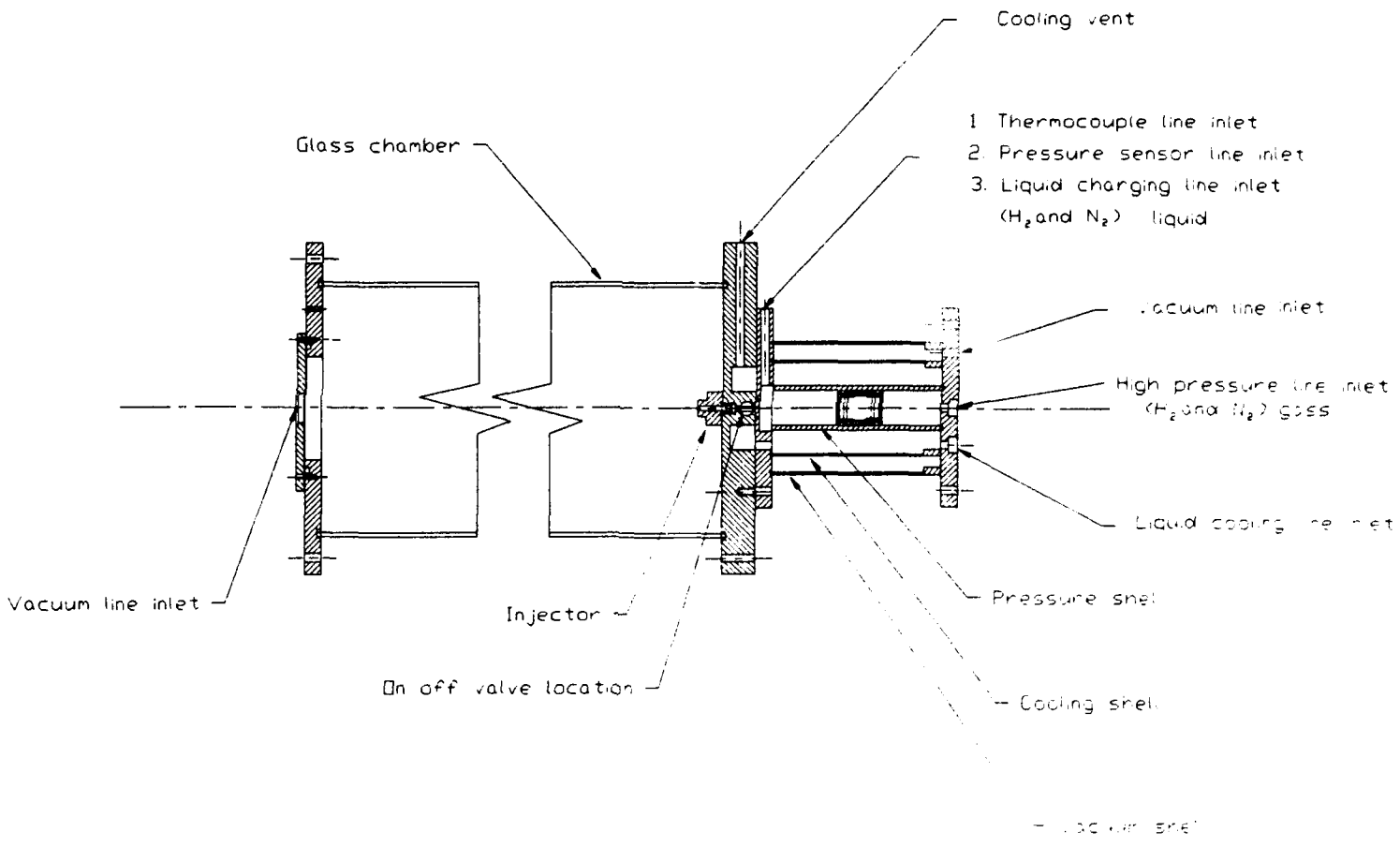


Figure 5.1.2 Assembly Drawing of Experimental Set-up

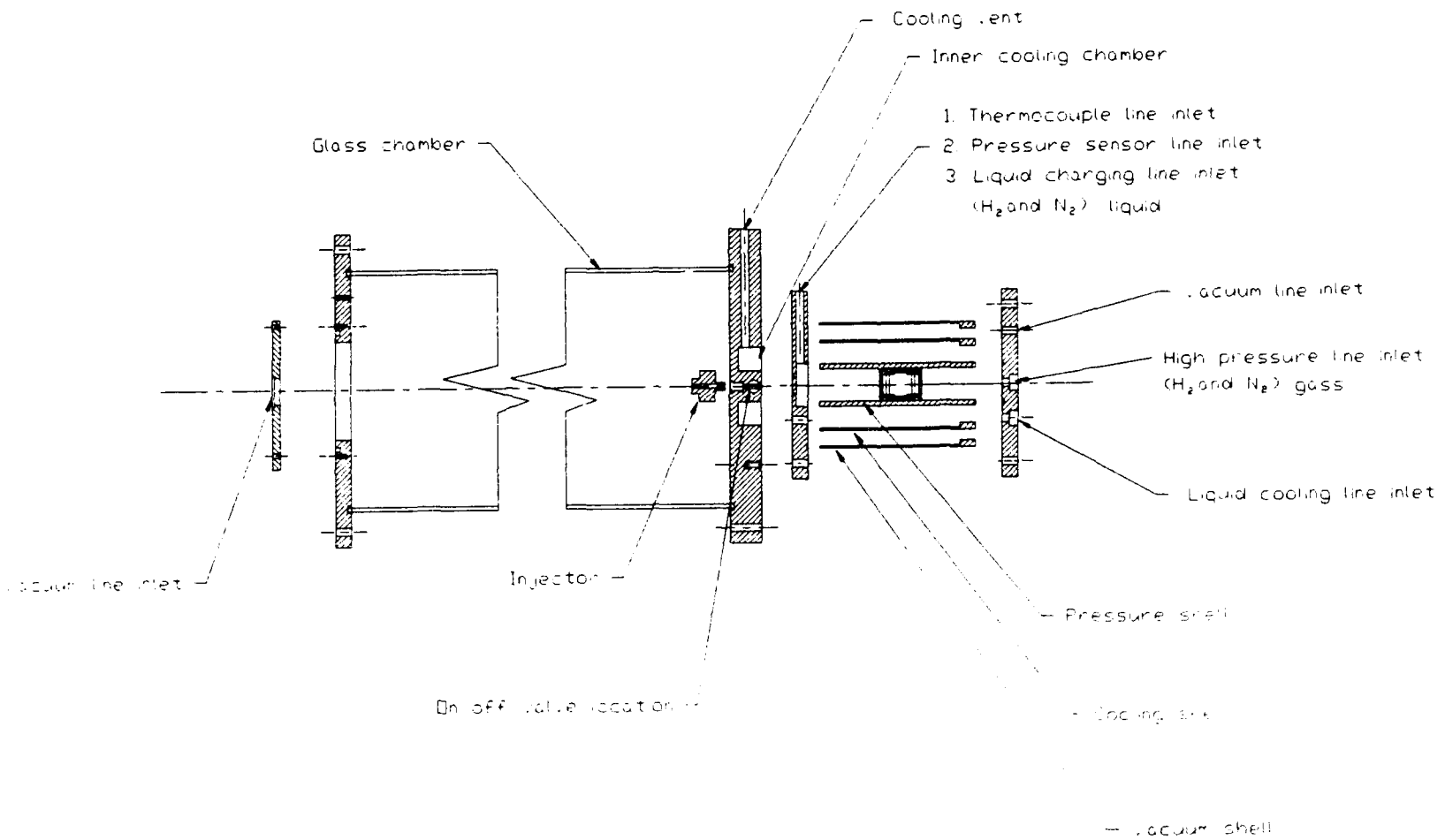


Figure 5.1.3 Exploded View of Experimental Set-up

pressure supercritical H_2 is injected into the evacuated space.

As seen in Figure 5.1.2 and in the exploded view in Figure 5.1.3, the pump and liquefier assembly are made of three concentric cylinders. The innermost cylinder is machined to a high finish to enable the piston to slide inside. The piston uses two spring energized TFE seals (EnerSeal made by Advanced Products Co.,Inc.) as piston rings; these seals are circular in shape and have a C crosssection with a helical metal spring installed inside the C. The metal spring provides the elasticity needed for the seal at the low temperatures; the sealing surface of the rings is made of virgin TFE which is self-lubricating and, hence, excellent for cryogenic service. Surrounding the innermost cylinder is the shell for liquid helium with a shell for vacuum insulation around it. The ports for gas and liquid supply are in the two end caps/flanges. One of the flanges connects to the glass vacuum chamber flange; the vacuum chamber flange has the injector nozzle and the valve for the nozzle. A cooling chamber is cut into the glass chamber flange in order to cool the injector nozzle to the fluid temperature. The nozzle injecting the fluid into the chamber is removable and thus various nozzles can be tested in this set-up.

5.1.2 Instrumentation

The temperature at various positions inside the vacuum chamber and pump will be measured using silicon diode sensors. The pressure variations inside the vacuum chamber will be measured using and Bayard-Alpert (from 10^{-2} to 10^{-7} mm Hg) and thermocouple type (up to 10^{-2} mm Hg) vacuum gauges. The phenomena occurring inside the glass chamber will be recorded and analyzed using a high-speed digital camera and a high-speed data acquisition system.

5.2 HEAT TRANSFER STUDY

The objective of the heat transfer study is to obtain a predictive correlation/model for cryogenic cooling using liquid/supercritical N_2 ad H_2 under the following heat transfer modes:

a) Spray cooling

b) Jet Impingement cooling

In order to accomplish this, a large amount of heat transfer data is needed under different flow and temperature conditions.

5.2.1 Experimental System Requirements

In order to meet the objectives the experimental apparatus has to satisfy the following criteria:

a) Supply of N_2 and H_2 in liquid and supercritical phases at flow rates up to 30 l/hr. The pressure and temperature ranges for the two fluid are:

Pressure range: For N_2 : 1 to 50 bar
For H_2 : 1 to 30 bar

Temperature range: For N_2: 78 K and above
For H_2: 21 K and above

b) Heater arrangement capable of providing up to 300 W/cm^2 and ability to simulate different surface conditions (roughness, coatings etc.). The heater arrangement and the associated instrumentation should give an accurate estimation of the surface temperature and heat flux.

c) The experimental set-up should provide for measurement of the spray characteristics using Phase-Doppler-Particle-Analyzer (PDPA) and high-speed digital camera.

d) Since the spray cone angle changes with flow rate, there should be provision for movement of the nozzle along the normal direction with respect to the heater surface in order to allow all the fluid to impinge the surface. Also since it is difficult to center the nozzle exactly over the heater surface in a fixed arrangement, there should be some provision to move the nozzle in the plane parallel to the heater surface to accomplish this. Thus the nozzle is required to have maneuverability along all the three axes.

e) Ability to measure flow rate of the fluid being used.

f) Finally, the whole arrangement should be leak proof to the highest degree considering the hazardous nature of H_2.

5.2.2 Experimental Set-up

An experimental set-up has been designed and constructed based on

the preceding requirements. The experimental arrangement consists of four major parts:

1. Test chamber
2. Fluid supply arrangement
3. Fluid transfer line
4. Instrumentation

A schematic diagram of the whole system shown in Figure 5.2.1 indicates the general arrangement. The following is a detailed description of the individual components.

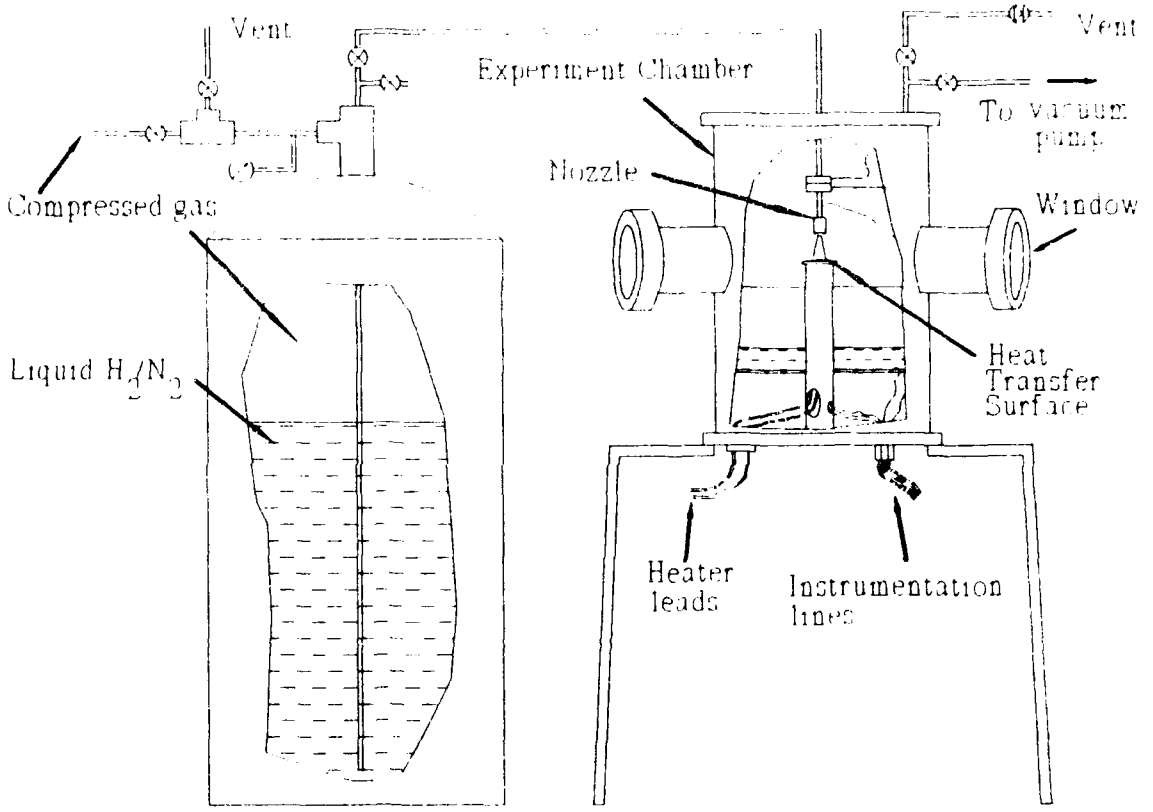


Figure 5.2.1 Schematic of the Cryogenic Cooling Set-up

5.2.2.1 Test Chamber

The test chamber assembly shown in Figure 5.2.2 consists of the heater arrangement, nozzle feedthrough, windows for PDPA and digital camera measurements and feedthroughs for instrumentation and power lines. The entire chamber is made out of stainless steel 304; all parts are assembled using metal-to-metal seals. The chamber consists of three individual assemblies: 1) the main chamber assembly, 2) the top cap

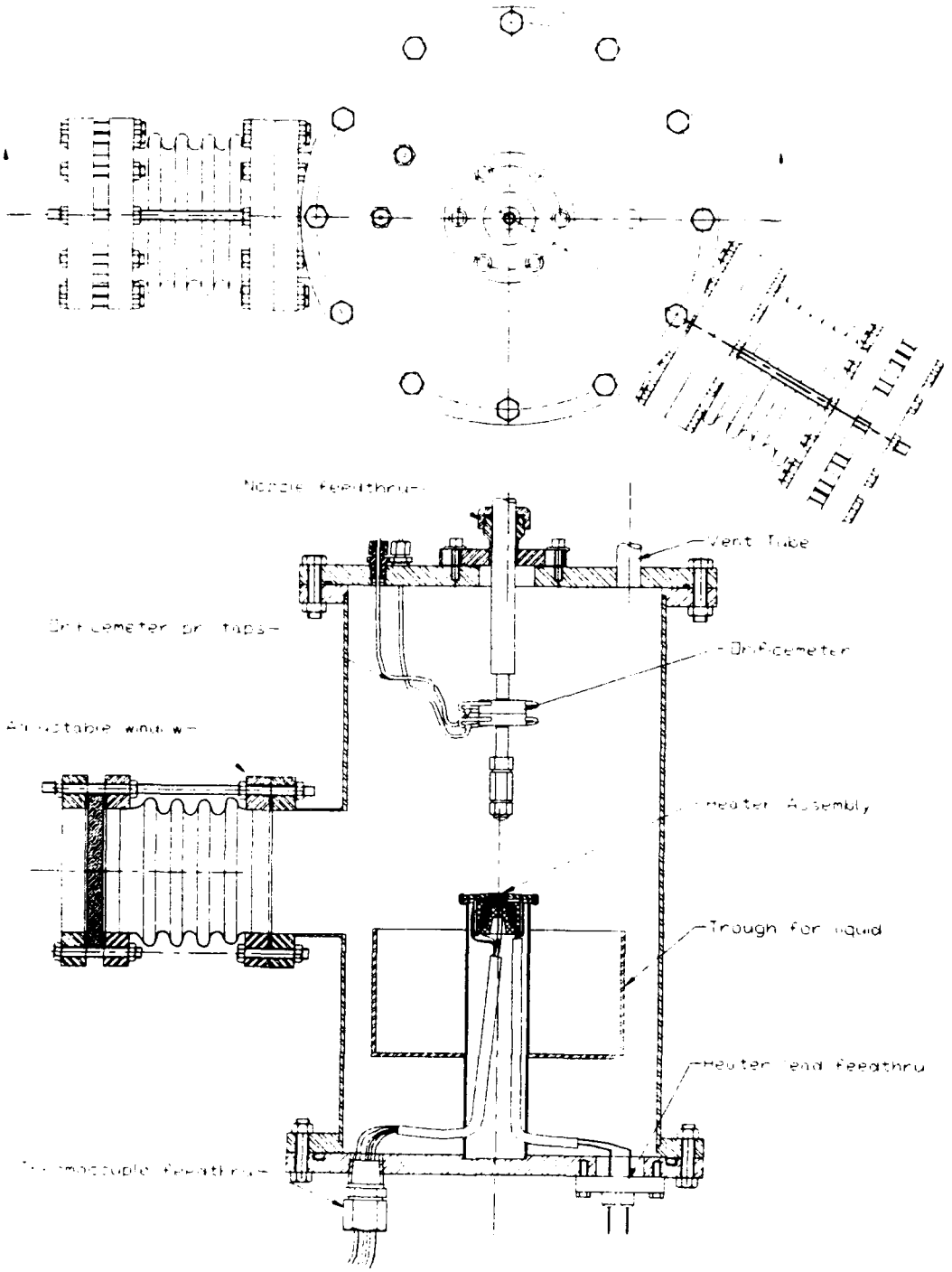


Figure 5.2.2 Test Chamber Assembly

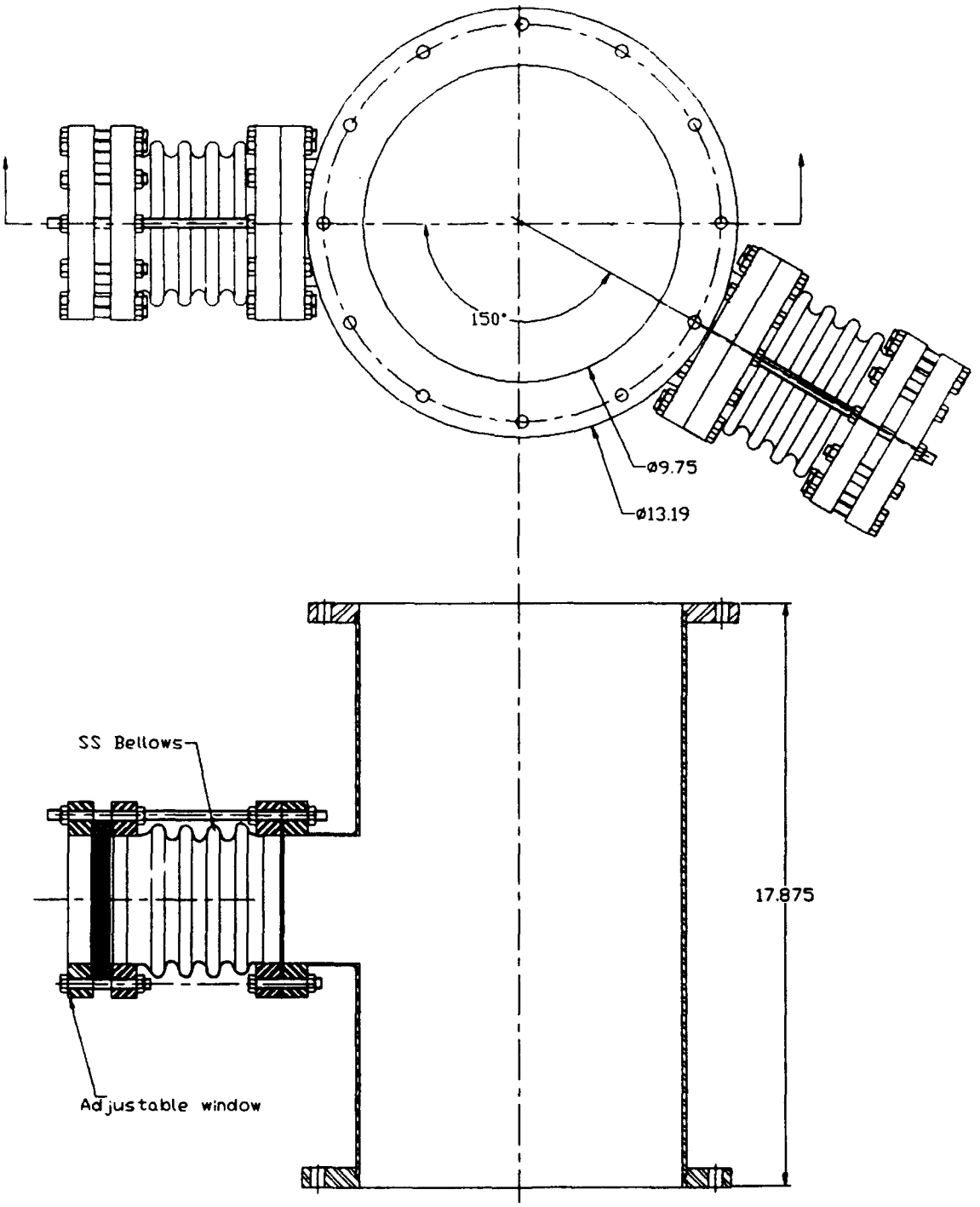


Figure 5.2.3 Main Chamber Assembly

assembly, and 3) the bottom cap assembly assembled together to form the test chamber.

The main chamber assembly shown in Figure 5.2.3 is essentially a cylindrical shell with end flanges for assembly to the top and bottom cap assemblies. The shell has two welded windows placed 150° apart at the same horizontal level. Each window arrangement consists of a formed bellows assembly interjected between the shell and the glass view port. These bellows allow the view port glass to be moved for proper alignment with the PDPA system (the PDPA systems requires the two windows normal to 150° angle). Leak prevention in the viewing glass assembly to the

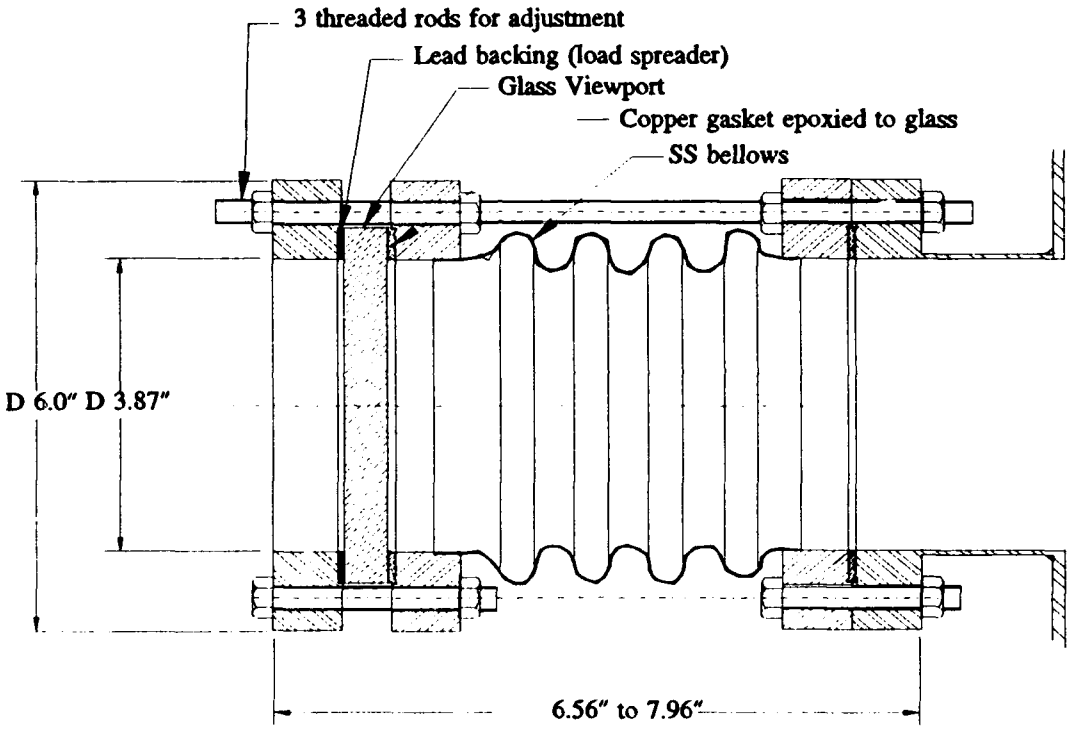


Figure 5.2.4 Window Details

stainless steel flange of the bellows presents a special problem. The glass generally contains non-flat surface which has to be mated to a flat stainless steel flange; this is difficult to accomplish using metal O-rings or C-rings. This problem was overcome by the arrangement shown in Figure 5.2.4, the mating glass surface is bonded to a soft metal gasket (copper) using a low temperature resistant epoxy cement; the gasket provides the seal against the stainless steel flange employing a

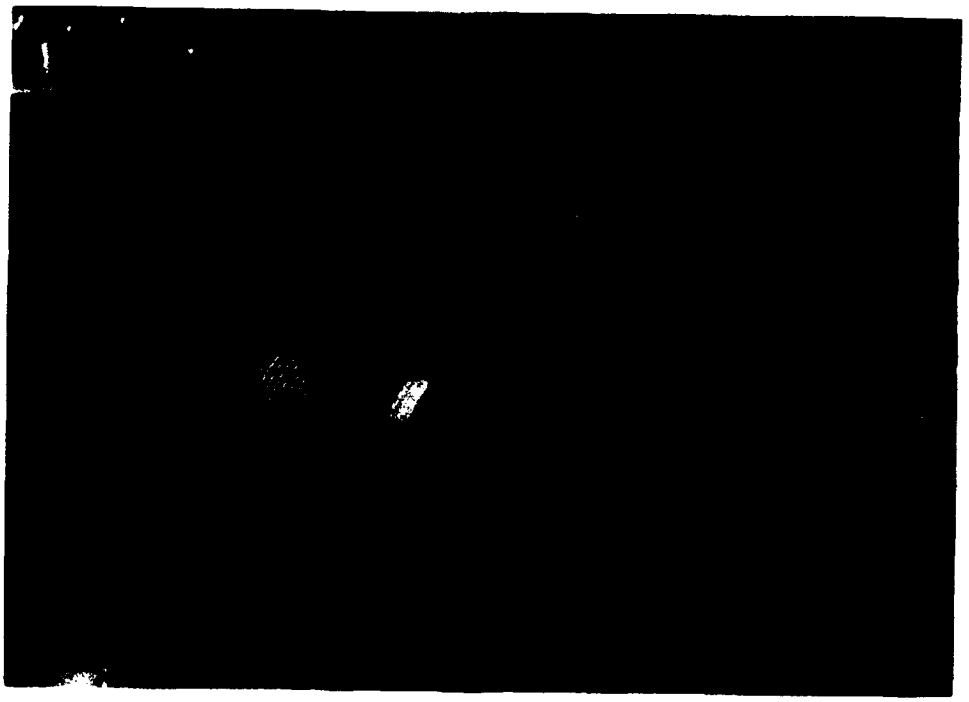


Figure 5.2.5 Window Assembly

knife edge seal. The uniform loading on the glass is provided by a lead gasket between the backing flange and the glass. Figure 5.2.5 shows a picture of the window assembly, as seen in the photograph, three threaded rods enable the alignment of the glass to the required angle.

The top cap assembly contains the feedthrough for the nozzle, the orifice meter pressure tap feedthroughs and gas vent connection. The assembly is shown in Figure 5.2.6, the fluid flow is measured using the orifice meter preceding the nozzle, the feedthrough is vacuum jacketed for thermal insulation of the fluid line. The orifice meter details are shown in Figure 5.2.7, the upstream and down stream flanges have three radial pressure taps for average pressure measurement; the lead lines from the pressure taps on each flange are interconnected and the two resulting lead lines go through the compression fittings on the top cap and are connected to the differential pressure transducer. As can be seen from Figure 5.2.6, the whole nozzle feedthrough assembly allows the nozzle to be moved in all three axes. A quick calculation for the vent line shows that at a flow rate of 50 l/hr of LH_2 the maximum possible gas exhaust flow rate is 12 l/sec at steady state. Hence, a vent line of about 7mm inside diameter will be sufficient to prevent choking. The

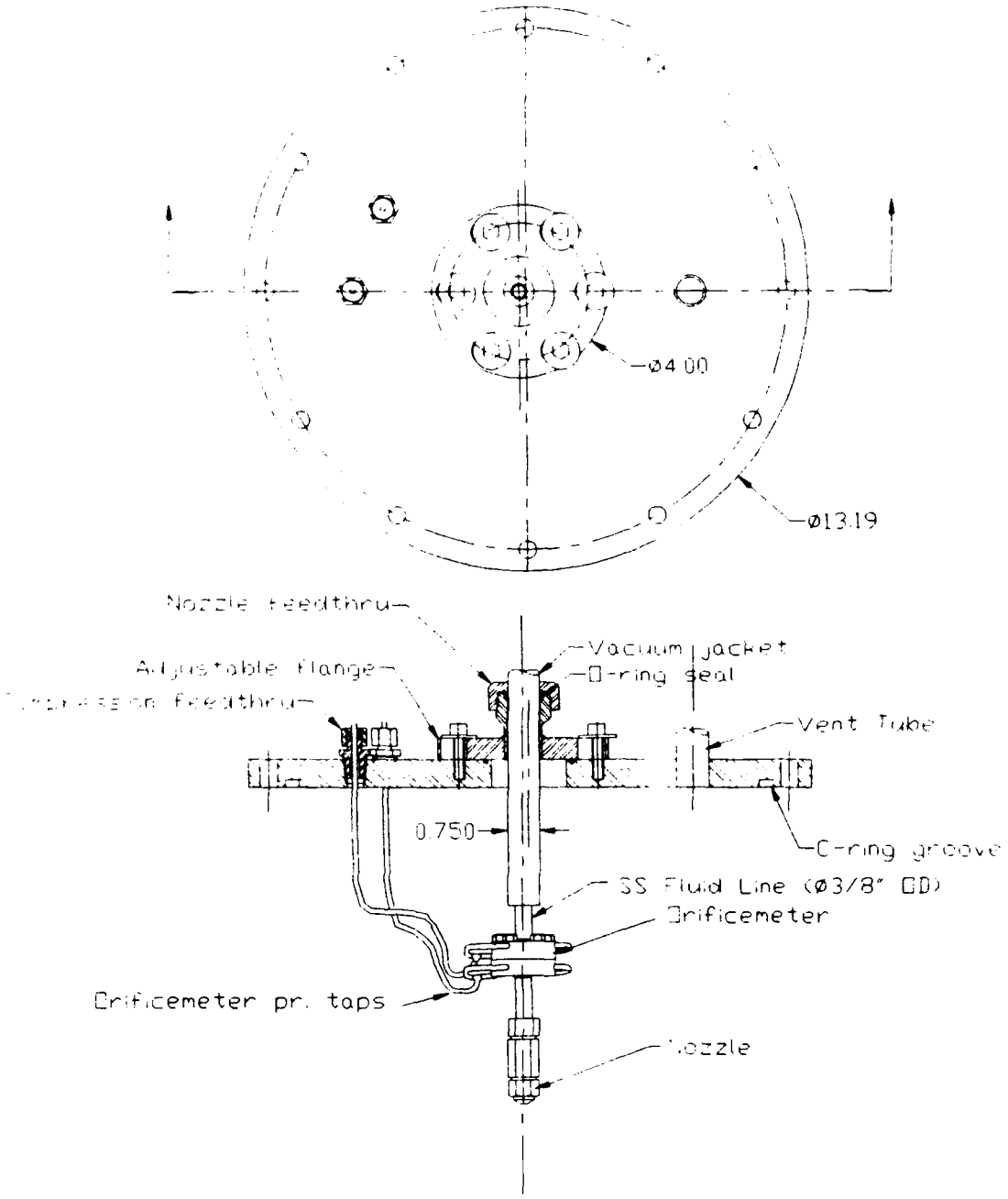
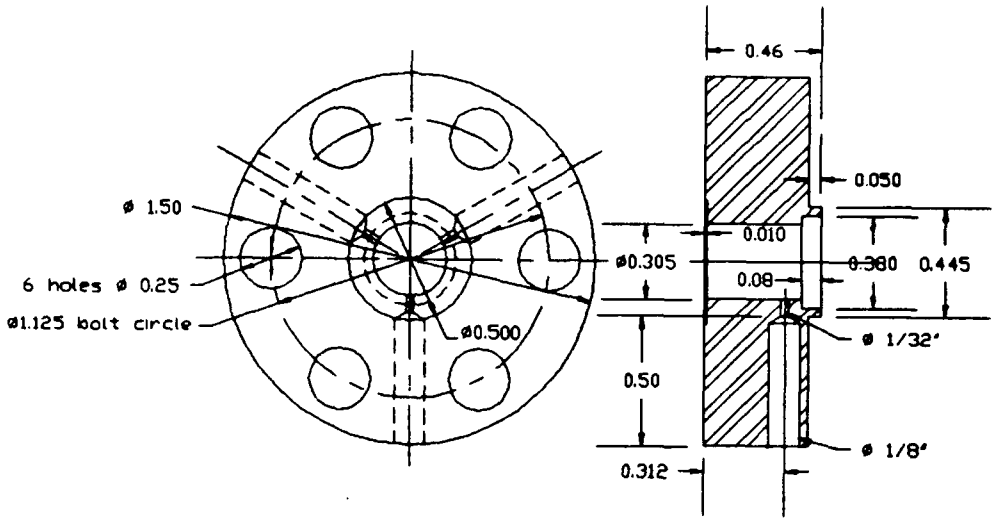
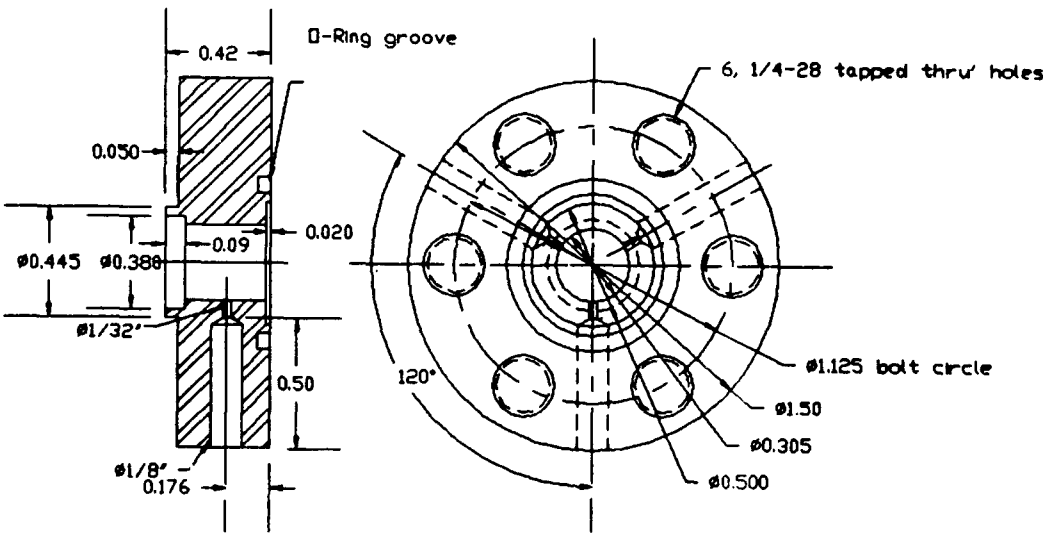


Figure 5.2.6 Top Cap Assembly



Upstream Flange



Downstream Flange

Figure 5.2.7 Orifice Meter Details

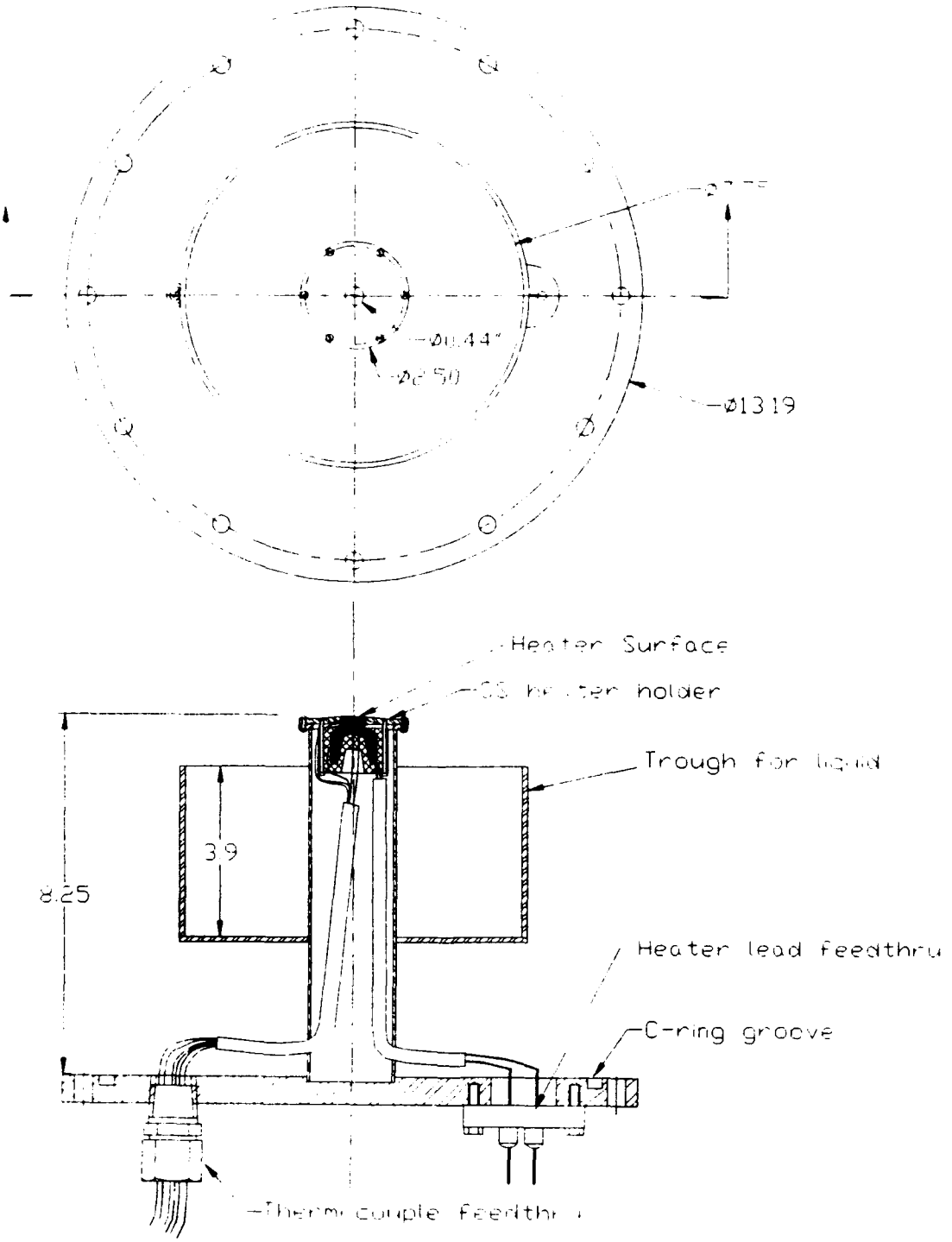


Figure 5.2.8 Bottom Cap Assembly

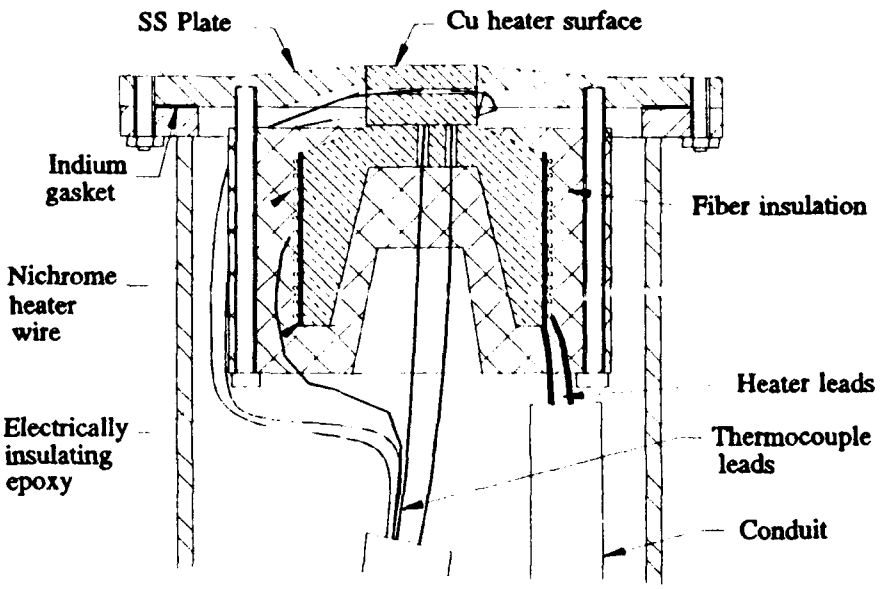


Figure 5.2.9 Heater Details

The vent line was made to be a 3/4-in copper tube soldered to the top cap, the tube is oversized to prevent any chance of pressure buildup in the chamber.

The bottom cap assembly contains the heater arrangement with the associated instrumentation. The two feedthroughs allow for thermocouple leads and power leads into the chamber. The whole assembly is shown in Figure 5.2.8. The details of the heater arrangement are shown in Figure 5.2.9; it consists of the copper heater block soldered to a stainless steel cap which seals the liquid from the rest of the assembly. Four thermocouples attached to the heater block allow for the estimation of the heat transfer surface temperature. All parts of the copper heater block except the top heat transfer surface are insulated from the liquid by a combination of fiber insulation, radiation shields and vapor barrier. The nichrome heating wire wound around the bottom part of the heater block is electrically insulated from the block by an electrically insulating and thermally conductive cement. A finite element analysis was carried out while designing the heater to estimate the heat loss through the stainless steel cap and the resulting error in surface temperature estimate. It was found that the errors were within 8% for

all the cases considered. Details of the finite element analysis of the heater section are presented in the Appendix.

The heater leads and thermocouple wires pass through separate metal conduits to prevent noise in thermocouple readings. The thermocouple leads pass through a feedthrough fitting in the bottom cap which allows six sheathed thermocouple wire pairs to pass through. The heater lead wires pass through a different feedthrough as shown in Figure 5.2.8. The trough around the heater assembly is for collecting any liquid flowing off the heater surface and thus preventing any liquid contact with the bottom cap.

All three of these assemblies are assembled together using metal C-rings for sealing; the whole test chamber is capable of operating at up to 4 bars internal pressure; this limitation is mainly because of the bellows used in the windows.

5.2.2.2 Fluid Supply Arrangement

As discussed in detail in Section 4, the two main options for fluid supply at the required conditions are the gas pressurized vessel and the pump. Both of these methods will be used in this study; the pressure vessel shown in Figure 5.2.10 is simply a tank with a Tee fitting which has a tube fed to the bottom of the tank from the top opening and the side opening is used as a vent and pressurizing port. The tank is insulated using polyurethane foam and is surrounded by a protective steel shell. This arrangement will be used for experiments with N_2 supply above 10 bar and up to 50 bars. This pressure vessel has a volume of about 30 liters and, hence, it can supply enough N_2 to run the experiments at the maximum expected flow rate for about an hour for each fill.

A gear pump will be used at lower pressures for better controllability and lower cooldown requirements. The gear pump is manufactured by Tuthill Corp. and is capable of providing a pressure head of 8 bar and can handle a fluid pressure of 34 bar. The pump speed can be controlled in order to vary the flow rate of the fluid. The pump head assembly and parts are shown in Figure 5.2.11. This pump has a magnetic coupling and, hence, all the moving parts are hermetically sealed inside the pump head; there are no moving seals in the pump. The

Figure 5.2.10 Pressure Vessel

bearing materials were tested at low temperatures and found adequate. Although this pump is not manufactured for cryogenic service, it was felt that the only concern regarding its use with LN_2 and LH_2 are the static seals used in the pump head. The silicone O-rings used in the pump will be replaced with metal C-rings for use at the cryogenic temperatures.

5.2.2.3 Fluid Transfer Line

The fluid transfer line connecting the fluid supply to the test chamber nozzle line should have a minimum of heat leak and be capable of handling thermally induced stresses due to temperature cycling. Figure 5.2.12 shows the details of the vacuum jacketed transfer line designed for the purpose; the fluid line is a 3/8-in stainless steel tube

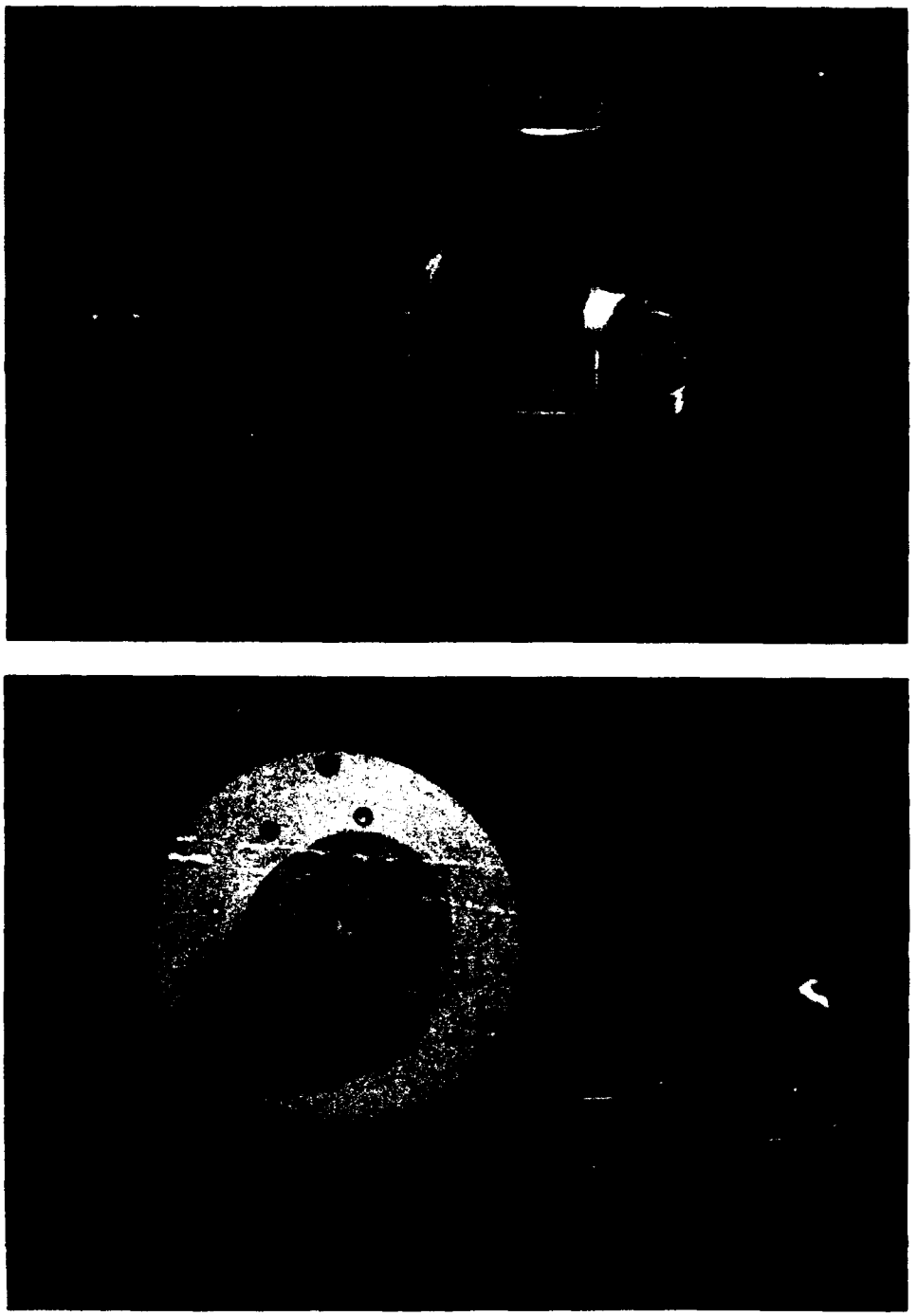


Figure 5.2.11 Pump Head Assembly and Parts

and the vacuum jacketing is made from 1-1/8-in copper tubing. As shown in the Figure, the inner fluid line is covered with three layers of aluminum sheet for radiation shielding. The whole transfer line shown in the photographs in Figure 5.2.13 is about 10 ft long and has a U bend

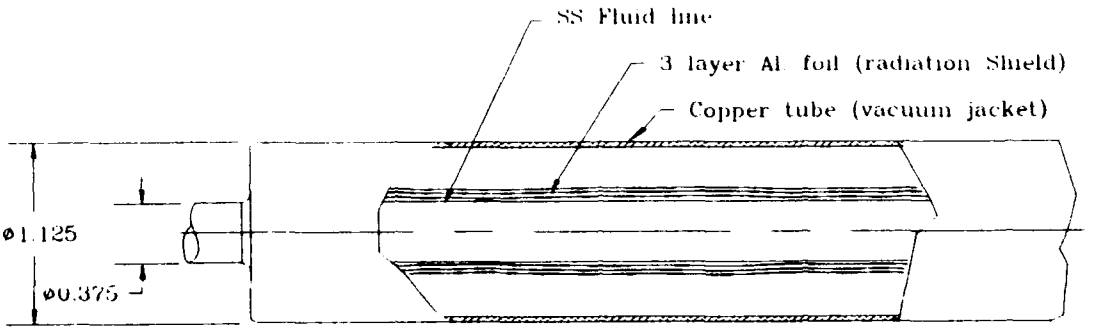


Figure 5.2.12 Transfer Line Details

for thermal contraction compensation; the bends are vacuum jacketed with bellows which are connected to the copper jacketing.

The whole experimental arrangement with the pressure vessel, transfer line, test chamber, LN_2 dewar, the high pressure gas tank and the associated valves and gauges is shown in the photographs in Figure 5.2.13. The pressure vessel is filled at low pressure from the liquid N_2 dewar shown in the photograph and then pressurized by high pressure gas from the N_2 gas cylinder. The high pressure liquid/supercritical N_2 passes trough the transfer line and is sprayed or injected onto the heat transfer surface in the test chamber. The flow rate of the fluid is measured by the orifice meter preceding the nozzle. The heat flux input and temperatures in the heater block are monitored by the data acquisition.

5.2.2.4 Instrumentation

The control circuit for the heater input is shown in Figure 5.2.14. The heater is supplied with ac voltage (0-100 V) from a variac; the power supplied to the heater is measured by a watt transducer which gives a dc signal proportional to the power input. This dc signal is read by the data acquisition system and converted to show the power input. A temperature controller reads the temperature of the heater wires and shuts of the power if the heater temperature goes above a

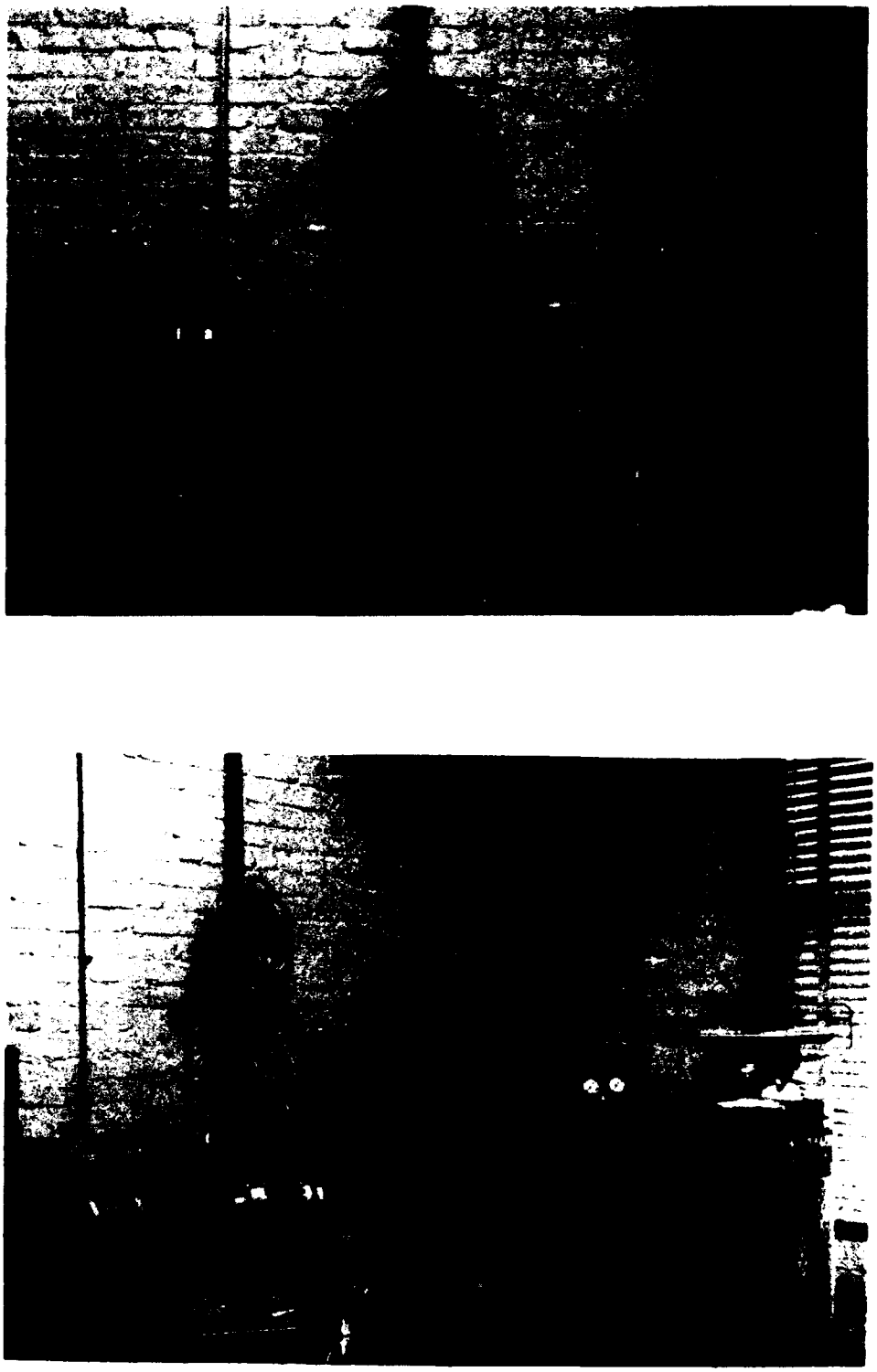


Figure 5.2.13 Experimental Set-up

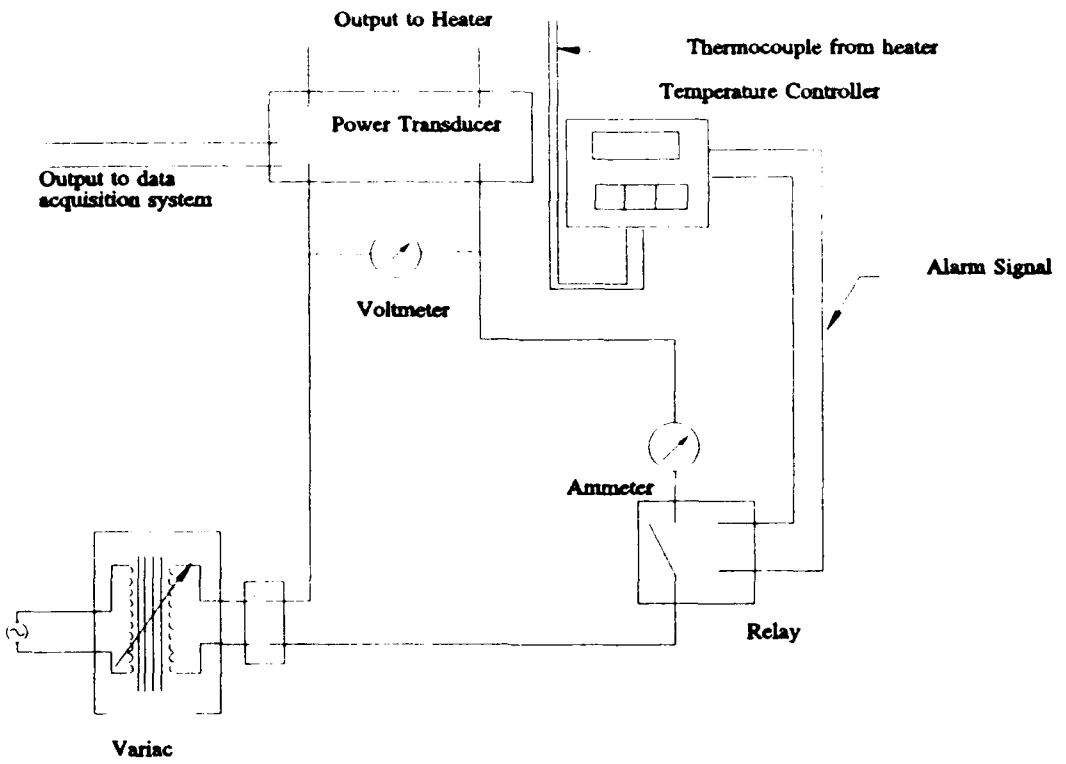


Figure 5.2.14 Heater Control Circuit

preset limit. A voltmeter and ammeter provide additional measurements for the purpose of control and verification. As mentioned before four E-type thermocouples are connected at various points on the heater block to provide surface temperature estimation. E-type thermocouples were chosen for their higher emf output and low thermal conductivity. Another thermocouple connected to the nozzle measures the temperature of the fluid being sprayed onto the surface. Other thermocouples are connected to the pressure vessel to monitor the cool-down process.

All the thermocouples and other dc signals are read by the data acquisition (Fluke 2287A). The differential pressure for the orifice meter is measured by a digital manometer connected to the differential pressure transducer. The gauge pressures at various points are measured by bourdon-type test gauges.

6. CONCLUSIONS AND FUTURE PLANS

A hydrogen blow-down facility capable of providing supercritical hydrogen at 30 bar and 35 K at up to 0.2 kg/s was designed. The site for the facility was prepared; however, the construction of the full facility was halted. The blow-down facility will be modified for the small-scale cryogenic cooling experiments with hydrogen.

The experimental set-up for the free expansion study is in the final stages of assembly. Detailed flow visualization and gas dynamics tests will be carried out using hydrogen helium and nitrogen in supercritical and liquid conditions.

The experimental set-up for the cryogenic cooling experiments has been completed and the preliminary experiments are currently underway. Initially, experiments are being performed with liquid nitrogen; depending on the results, appropriate modifications will be made to the experiments and the tests will be conducted with hydrogen at the hydrogen facility.

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APPENDIX FINITE ELEMENT ANALYSIS OF HEATER BLOCK

While designing the heater block it is essential to make certain the temperature uniformity of the heat transfer surface under the conditions encountered in the experiments. For this purpose a Finite Element Analysis (FEA) was carried out for the heater design under different boundary conditions.

A.1 FINITE ELEMENT ANALYSIS

A finite element analysis was carried out using NISA II/DISPLAY III program to optimize the heater design. The heater block being axisymmetric can be modeled using axisymmetric elements. The model shown in Figure A.1.1 is the axisymmetric model for the copper heater and the stainless steel cap; the axis of symmetry being the vertical axis. The nichrome heating wire used to supply the heat is simulated as a heat flux boundary condition represented by the arrows in the Figure; in this case a total heat input of 200 watts was considered. The heat transfer surface on the top is assumed to have convection boundary condition (to simulate spray or jet), since almost all the spray or jet will impact only the copper heat transfer surface; a heat transfer coefficient of 1.25×10^5 $W/m^2.K$ was used as the boundary condition. The stainless steel cap holding the copper heater was assumed to have a heat transfer coefficient equal to the pool boiling coefficient for nitrogen; hence, the boundary condition for the stainless steel cap was $h = 1.0 \times 10^4$ $W/m^2.K$. The convection boundary conditions are shown by the dark triangles inside the circles. A fluid temperature of 78 K was used, all the other surfaces were assumed to be perfectly insulated.

The resulting contour graph is shown in Figure A.1.2. As expected the temperature profiles are fairly flat in the upper portion of the copper block. The heat transferred through the stainless steel cap was calculated to be 13 watts out of the total of 200 watts supplied by the heater wires. This means that an error of 6.5% is introduced in the heat flux estimate because of the cap. The heat transfer temperature varied from 92.48 K in the center to 91.80 K at the edge; thus, no significant error is introduced in the temperature uniformity of the

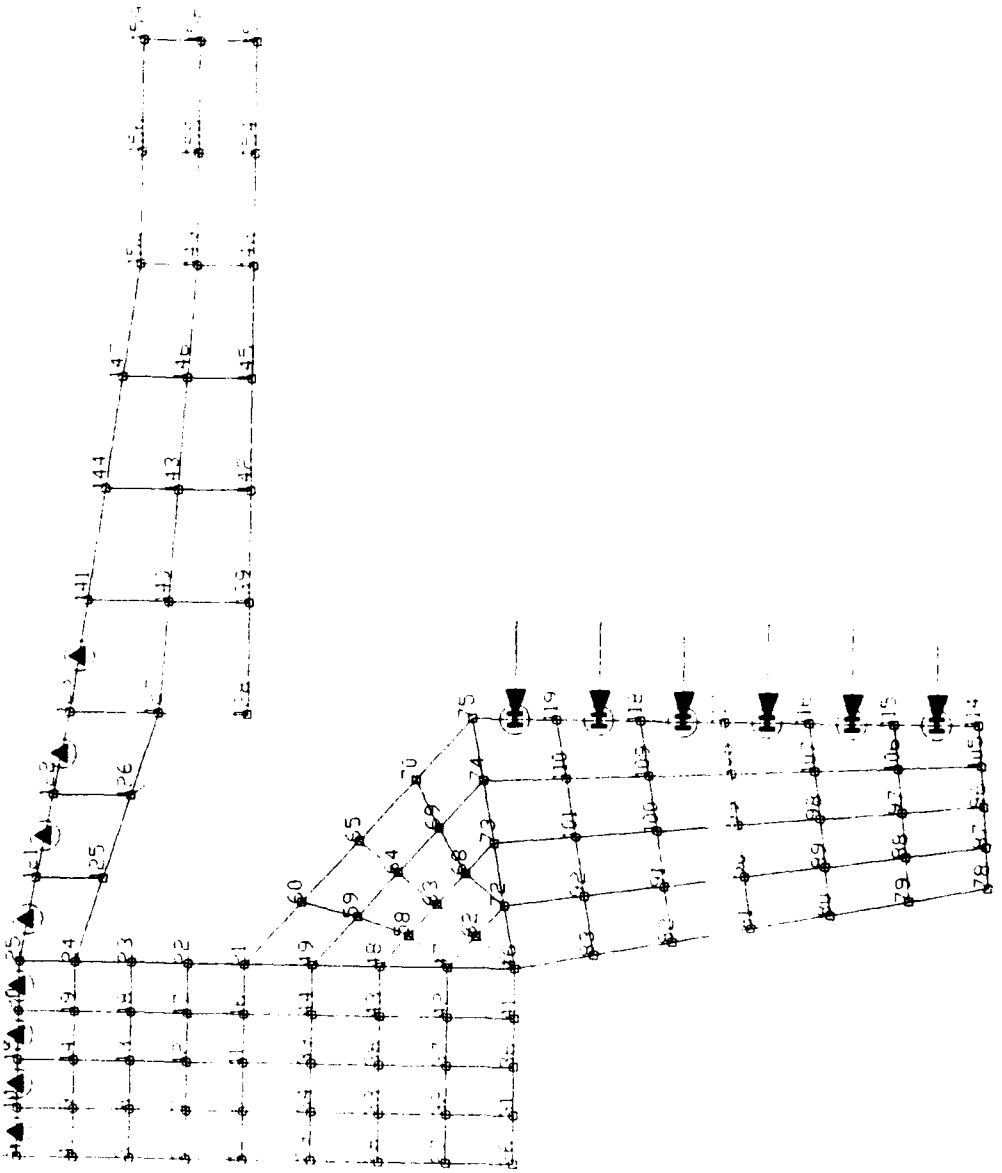

Figure A.1.1 Axisymmetric Heater Model

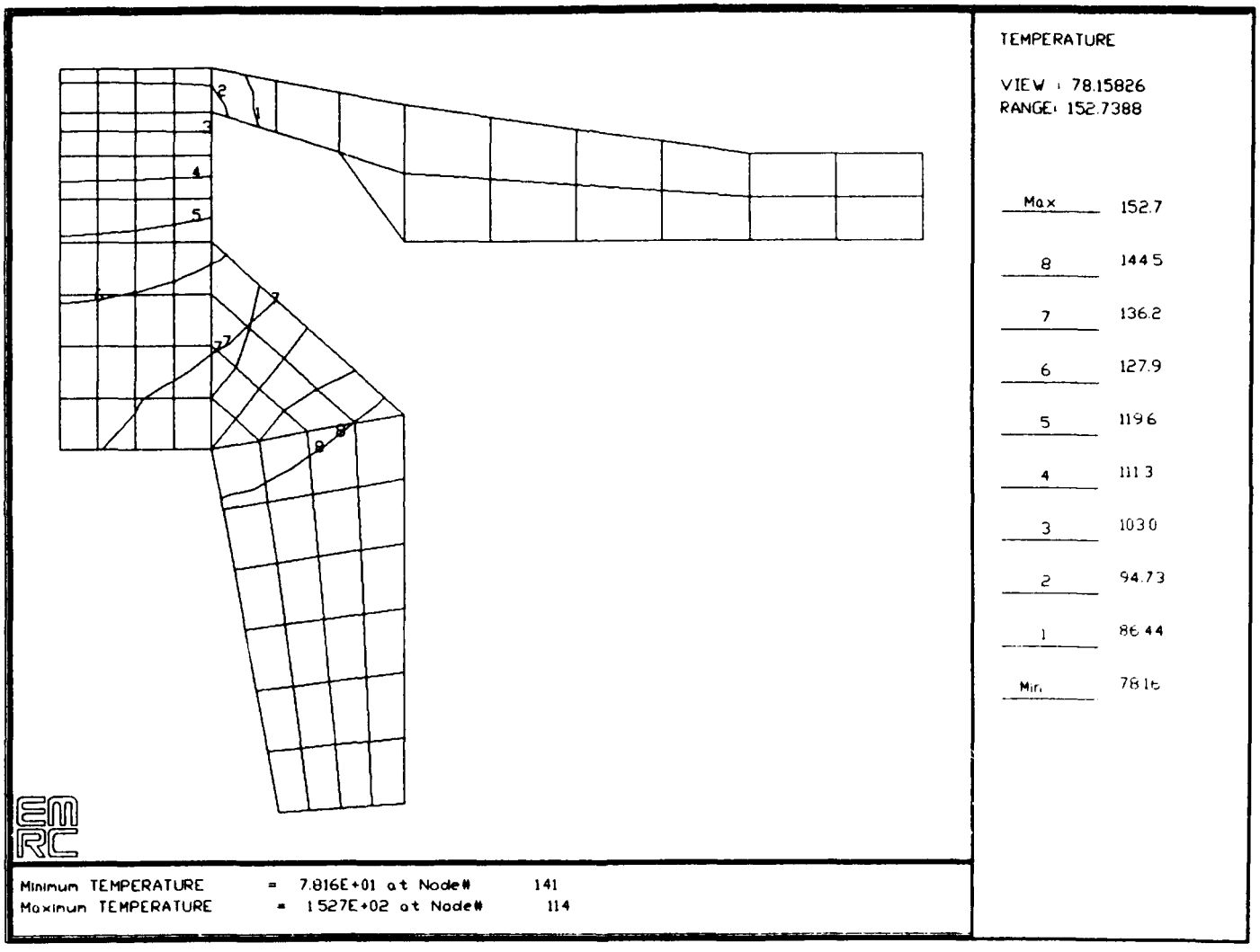

Figure A.1.2 Temperature Contours

surface. The surface temperature estimated from the extrapolation of thermocouples installed near nodes 1 and 3 is 93.6 K, i.e., an error of about 1.2%; the surface temperature estimated from thermocouples at nodes 11 and 13 is 93 K, an error of less than 1%. Hence it can be safely assumed that the temperature extrapolation will be very accurate and the heat flux estimate will be accurate to within 10% (taking other losses into account). The temperature difference cannot be used to measure heat flux since thermal conductivity of copper varies a lot by composition for the temperature range under consideration; also the conductivity may vary within the block itself because of the temperature variation.

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