Design of Test Apparatus to Analyze Two-Phase Refrigerant Flow Distribution for Furnace Coil Applications

by

Heather Gerallyn Smith

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Signature of Author ....
Department of Mechanical Engineering
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Certified by ....
Professor Peter Griffith
Thesis Supervisor

Accepted by ....
Professor Ain A. Sonin, Chairman
Departmental Committee on Graduate Students

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ABSTRACT

An apparatus designed and built to analyze the two phase fluid distribution characteristics of R-22 in a furnace coil application is outlined. A currently manufactured distributor assembly is utilized to test the apparatus as well as to analyze potential problems resulting in maldistribution.

In a typical vapor-compression refrigeration cycle saturated or slightly subcooled liquid from the condenser is passed through a throttle where the fluid pressure drops to that of the evaporator. Flashing of some liquid in this process results in a mixture of perhaps 20% quality. To maximize the heat transfer performance of the evaporator, thereby achieving the saturated or slightly superheated vapor required at inlet to the compressor, equivalent distribution of the two-phase mixture (flow rate and quality) through each circuit in the evaporator is required.

Development of an apparatus to simulate practical inlet and outlet conditions to the throttle and distributor assembly is presented. Consideration is given to various distributor characteristics including component geometry and component orientation as applied. Analysis of distribution is achieved through measurement of the liquid phase. Test results and analysis of maldistribution is presented for a currently manufactured distributor component.

Thesis Supervisor: Prof. Peter Griffith
Title: Professor, Mechanical Engineering

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1. Introduction

1.1 Discussion of Vapor-Compression Refrigeration Cycles

The standard vapor-compression cycle can be described from the thermodynamic behavior of the heat exchange medium as shown on the Temperature-Entropy plot of Figure 1.1-1.

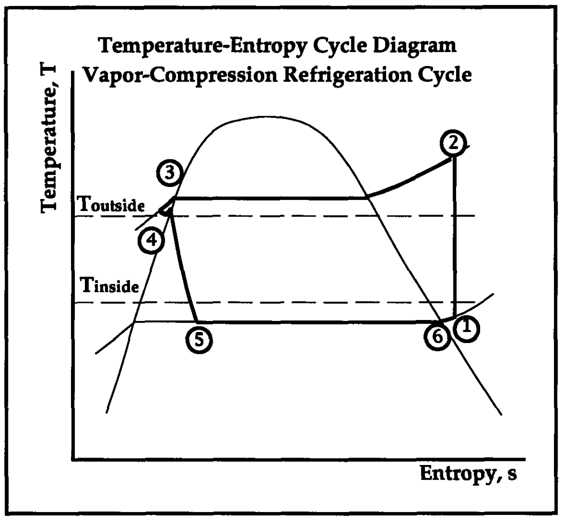

Figure 1.1-1 Vapor Compression Refrigeration Cycle T-s Diagram

For the cycle shown, saturated or slightly superheated vapor is taken from the evaporator coil exit (points 6,1) and compressed (the isentropic ideal is shown) to the higher pressure side of the cycle (point 2). Rejection of heat at constant pressure through the condenser desuperheats and condenses the refrigerant to a subcooled state (point 3). Flow through an expansion device (e.g. thermal expansion valve) in conjunction with a distributor/throttle assembly drops the flow pressure resulting in a mixture of perhaps 20% quality at inlet to the evaporator (point 5). Evaporation of the mixture at constant pressure gives the saturated or slightly superheated vapor required at inlet to the compressor. A simple schematic of the cycle is shown in Figure 1.1-2.

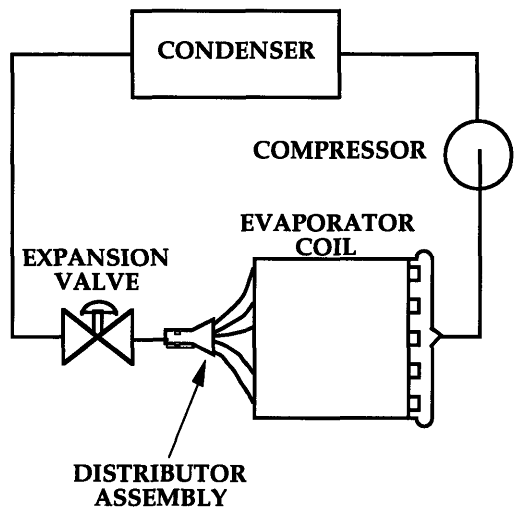

Figure 1.1-2 Vapor Compression Refrigeration Cycle Schematic

There are four primary components of the cycle each responsible for one "leg" of the cycle: 1) Compressor, 2) Condenser, 3) Throttle/Distributor, and 4) Evaporator. There are a multitude of additional components which can be included, e.g. devices for control under varying environmental conditions, systems to maintain purity of the refrigerant medium as it picks up dirt and moisture through the cycle, or measurement of flow behavior. Heat exchanger design, mechanical and electrical control, two-phase flow, boiling, condensation, and dry-out are fundamentals to be considered in the design and analysis of such a system. What is developed here will be utilized in the future to study the distributor assembly, an element complimentary to the throttle expansion valve, and the aspects of two-phase flow which control its performance.

1.2 Problem Statement

The behavior of the cycle as a whole depends upon the performance of its many elements. A key aspect in the measurement of design certainty involves reliable testing of individual cycle components. The apparatus presented herein was designed and built to test the behavior of the throttle, distributor and capillary assembly under specific environmental conditions. With Refrigerant-22 as the flow medium, two pressure vessels and a condenser are utilized to simulate the condenser and evaporator environment of a standard furnace coil unit. With extensive temperature and pressure control integrated into the system via cartridge heaters and chilled water and cold water/steam coils, additional environmental characteristics can be simulated.

With the apparatus build complete and tested for a vertical distributor orientation, experimental work can begin on testing different distributor assembly designs (focusing more on manufacturing characteristics, for example), the accuracy in applying the same design to different unit configurations as well as how the distributor behaves with different degrees of inlet subcool. Questions regarding the effectiveness of the distributor , its orientation, and the importance of head losses through the capillaries can be studied. Further, off-design conditions such as a loss in refrigerant can be approached and analyzed experimentally.

1.3 Current Distributor Assembly

This section has been included exclusively to introduce or perhaps refresh the reader on the geometry and function of the distributor assembly to be tested. (Note that the distributor 'assembly' includes throttle ring, distributor, and capillary tubes.) The primary function of the distributor (as the name implies) is to give a uniform distribution of refrigerant throughout the evaporator coil. A typical geometry is shown in Figure 1.3-1.

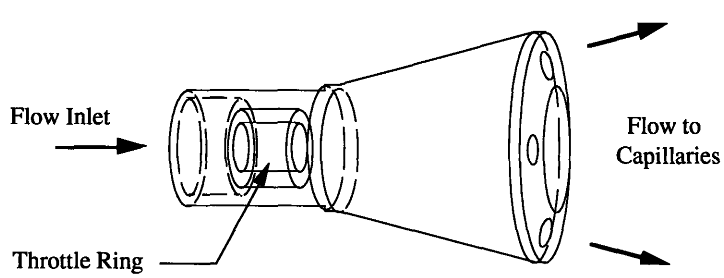

Figure 1.3-1: Distributor

The pressure drop associated with the throttle at inlet to the device serves to increase velocity and turbulence thereby improving the homogeneous characteristics of the two phase flow [5, 17]. The flash gas resulting from the drop in pressure further increases turbulence. The distributor assembly is typically used in conjunction with a separate expansion valve. The pressure drop from the high side to low side of the refrigeration cycle is therefore taken by both devices with a secondary amount resulting from frictional flow through the capillaries.

It has been noted in the text that operation of the distributor is not limited by orientation though a vertical orientation is "recommended" [5, 17]. Orientation is one parameter the apparatus, outlined in Chapter 3, will test in the future. Proper functioning of the distributor assembly definitely require, however, a precise equality in size, shape, and surface characteristics of the small capillary paths. Manufacturing considerations therefore play a role. These two points and others were considered in the apparatus design and have been briefly discussed in Chapter 2 along with two-phase flow relations.

2. Background

2.1 Aspects of Flow in Refrigeration Cycles

Flow through a vapor-compression cycle from a fundamental thermodynamic approach consists of superheated vapor, subcooled liquid, and mixtures at various thermodynamic qualities each at specific points in the cycle. From a black box analogy one can discern the function of cycle components and the corresponding condition of flow relative to a Temperature-Entropy or Pressure-Specific Volume diagram. Fully understanding the cycle is largely more complicated in that flow mixtures into and out of components are two-phase liquid-vapor systems where a homogenous analysis can not be applied with accuracy.

The primary cycle path of interest for a distributor analysis consists of flow from condenser outlet to evaporator coil inlet. As discussed in Chapter 1 an expansion valve or throttle of some type is required to traverse from the high to low pressure sides of the cycle (excluding line losses). The resulting variance in mixture quality as flash gas is produced promotes changing two-phase flow regime characteristics (Section 2.2.2) and the potential for such phenomena as choking (Section 2.2.4). The simulation of such flow in an apparatus designed to study flow distribution prior to the evaporator coil therefore requires an understanding of two-phase flow phenomena as well as vapor-compression refrigeration cycle basics.

2.2 Two-Phase Flow

The most difficult aspect in determining flow distribution results from its nature as a two-phase flow. The (homogeneous/) continuum relations for a single phase fluid and all that corresponds to a more fundamental study of fluid mechanics no longer applies as the effect of gravity on the distribution of the two phases and the variations in boundary conditions must be considered in the analysis. The difficulties associated with a compressible gas phase and a deformable gas-liquid interface make clear the above point. The production of flash gas and the resulting changes in mixture quality both prior and through the distributor assembly requires an understanding of gas-liquid two-phase flow. The fundamental parameters and relationships utilized in the distributor test apparatus design described herein will be presented first. For reference in the proceeding sections, an idealized model of two-phase flow through a pipe is presented in Figure 2.2-1 [4]:

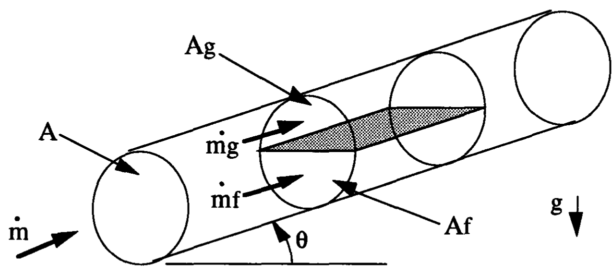

Figure 2.2-1: Model of Two-Phase Liquid-Vapor Flow in a Pipe

2.2.1 Relationships and Parameters of Interest

As a means of characterizing the refrigerant flow through the test section and thereby designing the measurement equipment required several fundamental parameters of two-phase flow theory were utilized. To determine the size necessary for the measurement glasses, for example, flow rate calculations for the two-phase flow out of the capillaries was required. The volumetric flow rate for each phase can be defined (2.1a,b):

$$Q_{gcap} = \frac{x \quad \dot{m}_T}{n \quad \rho_{ge}}$$
$$Q_{fcap} = \frac{(1-x) \quad \dot{m}_T}{n \quad \rho_{fe}} \qquad (2.1a,b)$$

where n refers to the number of circuits and x is the vapor mass quality or dryness fraction [4] (2.2):

$$x = \frac{\dot{m}_g}{\dot{m}_T} \tag{2.2}$$

An alternative definition for the mass quality is the thermodynamic quality if both phases are in thermodynamic equilibrium, i.e. each at the saturation temperature [6]. Note, however, that a further condition for this equivalence requires homogeneous flow. For the purposes of design, it is presumed here that though the flow is not homogeneous, the thermodynamic quality present at various points along the two-phase flow path through the distributor and capillaries is equal to the dryness fraction defined above.

With significant phase velocities presumed to exist, more severe of course for the vapor phase, some calculation was done to determine flow velocities at exit from the capillaries. The results would thereby discern if capturing the flow for measurement would be problematic. Phase velocities can be determined from (2.3a,b) [4]:

$$\begin{aligned} V_g &= \frac{G_{cap}}{\rho_g} \frac{x}{\alpha} \\ V_f &= \frac{G_{cap}}{\rho_f} \frac{(1-x)}{(1-\alpha)} \end{aligned} \tag{2.3a,b}$$

where G_{cap} is the mass flux through each capillary and can be defined for the flow from Equation 2.4.

$$G_{cap} = \frac{\dot{m}_T}{A} \tag{2.4}$$

The phase velocities can then be determined from a rearrangement of the above (2.5a,b):

$$V_g = \frac{Q_g}{A \quad \alpha}$$
$$V_f = \frac{Q_f}{A \quad (1-\alpha)} \tag{2.5a,b}$$

The void fraction, α, in the above relationships represents the ratio of vapor cross sectional flow area to total cross sectional flow area. Several analytical and empirical methods are available for calculation of the void fraction. The general relation for void fraction can be determined from continuity relations for each phase and is given by (2.6) [15, 21]:

$$\left(\frac{1-\alpha}{\alpha}\right) = \left(\frac{V_g}{V_f}\right)\left(\frac{\rho_g}{\rho_f}\right)\left(\frac{1-x}{x}\right) \tag{2.6}$$

where the velocity or "slip" ratio can be estimated from water-steam curves of specific volume versus slip ratio [15]. The result is then iterated upon until consistency in slip ratio is found. A general empirical form developed by Butterworth to represent several of the two-phase flow models was utilized as a comparison [4] (2.7).

$$\alpha = \left[1 + B_B\left(\frac{1-x}{x}\right)^{n_1}\left(\frac{\rho_g}{\rho_f}\right)^{n_2}\left(\frac{\mu_f}{\mu_g}\right)^{n_3}\right]^{-1} \tag{2.7}$$

For example, the prediction of Lockhart and Martinelli is applied with [4]:

$$B_B = 0.28 \quad n_1 = 0.64 \quad n_2 = 0.36 \quad n_3 = 0.07$$

2.2.2 Flow Regimes

The flow regime for a segment of two phase flow describes how the two phases are positioned relative to one another. This is important to the analysis of flow distribution if the regime just prior to distribution can be determined. For example, bubbly flow prior to distribution might be the optimum condition to assure equivalent flow (mass flow rate and quality) into each capillary whereas stratified flow could be quite detrimental to distribution (a condition potentially existing only if the flow is moving somewhat horizontally) with some circuits receiving all vapor and others all liquid. Further, finding the flow regime within the capillaries can be important to an analysis of the evaporator coil heat transfer characteristics. A short discussion, therefore, is presented here to gain an understanding of flow regimes in the areas of interest and what effects the regimes might have on distributor behavior. Note that though a thorough understanding of flow regimes is not so important for the test apparatus design, a brief introduction here will be helpful for further analytical studies of distribution.

As described in the recent work of Taitel [19] four primary flow regimes can be defined: stratified, intermittent, annular, and bubble. Each of these four regimes can be further divided into "subclasses" if necessary. Stratified flow is characterized by a distinct interface separating upper vapor and lower liquid flow layers. This flow regime exists only in horizontal or slightly inclined flow at relatively low phase velocities where gravity and buoyancy effects play a large role. Intermittent flow, on the other hand, is characterized by liquid slugs separated by large bubbles of vapor and can exist in various forms through all angles of flow inclination. The annular flow regime consists of a liquid film at the wall encasing a central gaseous core. Some liquid may or may not be entrained in this central region. And finally, the bubble regime is characterized by dispersed bubbles in a continuous liquid phase [19].

The intent of this section is to provide a means of predicting the two-phase flow regime at points of interest within the distributor assembly. A multitude of different models and maps exist each with different dependencies on phase characteristics. Until recently, these models emphasized the analysis of specifically horizontal or specifically vertical flow as these two cases were considered the most important. Currently a "unified" approach is favored which incorporates flow inclination into the analysis and is, therefore, applicable to any pipe inclination. Such an approach is presented by Taitel [19]. In his paper, he notes the eleven dimensional parameters affecting the flow pattern (and are therefore required for its prediction) under isothermal incompressible conditions:

(1) The liquid superficial velocity, j_f

(2) The gas superficial velocity, j_g

(3) Liquid density, ρ_f

(4) Gas density, ρ_g

(5) Liquid viscosity, μ_f

(6) Gas viscosity, μ_g

(7) Pipe diameter, d

(8) Acceleration of gravity, g (m/s2)

(9) Surface Tension, σ

(10) Pipe roughness, ε

(11) Pipe Inclination, θ

where the superficial velocities can be defined as follows [4] (2.8a,b):

$$j_g = \frac{G \quad x}{\rho_g}$$
$$j_f = \frac{G \quad x}{\rho_f} \qquad (2.8a,b)$$

An algorithm developed by Hodes [9] utilizes the dimensional parameters as input (SI units) with the output being the desired flow regime map.

2.2.3 Pressure Drop Analysis

For a single phase fluid the prediction of pressure drop over the length of a pipe (e.g.) is a straightforward application of physical laws. A similar calculation applied to a region of two-phase flow would ignore fluid-fluid interaction and thereby produce significant inaccuracies. The dependency of fluid-fluid interaction on flow regime indicates that uncertainty can be reduced in the approximation of pressure drop if predictive models developed specific to separate flow regimes are utilized. Applying these models with confidence, however, requires that the predicted flow regime is correct. To minimize error, so to speak, a conservative approach would utilize more general (i.e. not limited to any specific flow regime), though indeed also inaccurate, models for the approximation of pressure drop in two-phase flow. Two models are mentioned below with the latter being the method utilized for this analysis.

The homogeneous approach assumes that the characteristics of two-phase flow can be modeled as a single phase. The model assumes that vapor and liquid phase velocities are equal and "mean" or "effective" properties can be identified for the flow [4, 9, 21]. One can foresee the simplification in analysis such approximations would produce. Further, it would seem evident that such an analysis would be grossly inaccurate for all flows save, perhaps, bubbly flow with a fine, homogeneous dispersion of vapor bubbles in liquid flow. The more general separated flow model does not assume equivalence of phase velocities and therefore utilizes distinct characteristics for each flow [4, 9, 21]. The model defines pressure drop by developing mass and momentum conservation relations for each phase and adding them together (reference Figure 2.2-1).

Utilizing the separated flow model the following relation for pressure drop can be found [9, 21] (2.9):

$$-\frac{dp}{dz}=\frac{4}{A}\frac{\tau_W}{}+G^2\frac{d}{dz}\left[\frac{(1-x^2)}{(1-\alpha)\rho_f}+\frac{x^2}{\alpha\ \ \rho_g}\right]+\left[\alpha\rho_g+(1-\alpha)\rho_l\right]g\sin\theta \qquad (2.9)$$

The terms on the right hand side of the equation represent frictional, acceleration, and gravitational components of the pressure drop respectively. The frictional component is typically dominant and, ironically, the most difficult to determine.

As a result of the difficulties in predicting two-phase flow pressure drop directly many models instead predict a two-phase multiplier or a ratio of the two-phase pressure gradient to that of a single phase flow. From the discussion of Hodes (from Hewitt) and Whalley the model of Friedel is most accurate in predicting the frictional pressure gradient term when the viscosity ratio $\mu_f/\mu_g < 1000$ [9, 21]. The two-phase multiplier defined by Friedel is concisely presented in the appendices of Whalley [21]. The reader is directed to that reference or the original work of Friedel for further details.

2.2.4 Choked (Critical) Flow

A flow is said to be choked when the critical flow velocity has been achieved (Mach = 1) and changes in downstream pressure cannot be communicated back to a point prior to the flow choke. For complete analysis of the distributor assembly, knowledge of the choke point can indicate how dependent distribution is on downstream pressure, e.g. if the capillary is not choked, the distributor is susceptible to discharge pressure as well as head losses in the capillaries or conversely, if choke occurs at the throttle, orientation of the distributor will not effect distribution. After the point of choke there is flexibility in pressure.

In the consideration of how two-phase mixtures behave under choked conditions, several points must be kept in mind. First, there is a critical pressure ratio associated with the occurrence of choke which can be helpful in defining where choke occurs. Further, the choke or critical velocity of the two phase mixture is less than that of either phase alone. Also, there is a dependency of choke velocity and the critical pressure ration upon flow regime.

2.3 Aspects of Maldistribution

The design and build of an apparatus to test flow characteristics of a distributor assembly requires a fundamental understanding of the phenomena to be tested. Looking at the potential for maldistribution of flow helps to identify critical junctures on the path from condenser to evaporator. The elements resulting in off-design behavior of the distributor assembly and the subsequent complications therefore become important. Two examples of the former and one example of the latter are discussed below.

2.3.1 Pressure Drop Dependence

The relationship between distribution and pressure drop is somewhat complicated in that off-design distribution can effect pressure drop and unforeseen pressure drops can effect distribution. The result in both cases is a loss in evaporator effectiveness. For example, if manufacturing defects give poor distribution, differing quality mixtures flowing into capillaries can result in varying pressure drops and inconsistent heat transfer characteristics across the evaporator coil. Looking, on the other hand, at the effect of pressure drop on distribution, choke at the TXV valve or at the throttle prior to distribution alleviates the concern of distributor orientation and the corresponding gravitational head losses. Or, for example, while high pressure drop with no choke prior to distribution gives annular flow and good distribution, low pressure drop with no choke and the potential for intermittent flow gives inconsistent distribution

2.3.2 Manufacturing Considerations

A discussion of manufacturing considerations is somewhat subjective in that no statistical studies were done for the analysis presented here. The only background information utilized resulted from observations made in machining fixtures to test the distributor assembly (distributor plus throttle). With flow area being of critical concern in controlling pressure drop through the assembly reliable tolerances become important. For example, a dent made into the inlet inner diameter of the soft brass distributor can reduce flow area thereby producing premature flash gas. The method utilized to divide the flow from one to several channels or "circuits" requires precision in machining, as each capillary inlet must be identical to the next with no burrs resulting from a tool being driven in too far. Burrs on flow walls can deflect flow and produce extra pressure drops leading to maldistribution. Any amount of slop in seating the capillaries into the distributor exit for soldering/brazing can result in reduced area and additional unforeseen pressure drops.

2.3.3 Effects on Heat Transfer

It would seem apparent that maldistribution of flow from the distributor would be detrimental to the heat transfer characteristics of the evaporator coil. If, as an extreme case, half of the circuits received completely vapor and the other half received only liquid varying amounts of superheated and saturated fluid would flow into the compressor. Such a flow not only affects compressor behavior, as the affinity of oil for refrigerant would result in a lubrication problems as liquid flows out, but also that of the thermal expansion valve which monitors suction gas superheat and controls flow into the evaporator. Further, with some coils of the evaporator receiving only vapor those sections are essentially inactive thereby reducing evaporator performance.

3. Experimental Apparatus

To physically model the refrigerant flow through the throttle and distributor assemblies requires the simulation of both high and low pressure side environments. The experimental apparatus discussed in this chapter was designed to achieve the condenser and evaporator conditions for a typical home/commercial air conditioning unit under practical field conditions (to be discussed). By designing the experiment around typical field conditions, direct correlation of the distributed flow can be made to units in current use. An example of typical environments is shown on the T-s diagram of Figure 3-1.

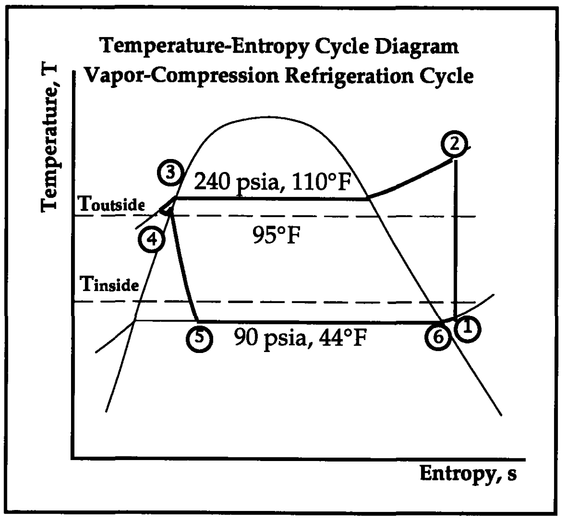

Figure 3-1: T-s Diagram Showing Typical Cycle Environments

Utilizing refrigerant, in this case Refrigerant 22, as the flow medium restricts the design of an experimental apparatus in three primary aspects. One, the flow loop must be closed to protect the environment. Two, the thermodynamic properties in the area of interest require use of pressure vessels rated to fairly high pressure. Finally, refrigerant is a "wonderful" solvent. Crazing of plastics and similar difficulties with rubbers limits the selection of materials available for seals and lighting systems (Section 3.3). The components to be presented in this chapter consist primarily of a refrigerant reservoir

(Section 3.2) designed to achieve condenser conditions (the inlet conditions to the expansion valve), a main pressure vessel (Section 3.3) sustained at evaporator conditions, the measurement equipment contained within this vessel (Section 3.3), and, finally, an ice-water-cooled heat exchanger (Section 3.4) included as a control to condense excess refrigerant vapor and assist in maintaining conditions in the main pressure vessel (Figure B-1). An introductory discussion is presented first on the distributed flow measurement technique.

3.1 Flow Measurement Technique

As observed previously, a two-phase mixture exits the distributor as some liquid flashes to vapor with successive pressure drops through the throttle and distributor assembly. The fluid into the evaporator therefore consists of both liquid and vapor refrigerant mixed with some lubricating oil. (Note, the effects of oil on distribution are considered negligible.) The liquid, with a dominant heat transfer coefficient, is the primary heat transfer mechanism through the evaporator coil and subsequently the phase of interest. Therefore, the measure of distribution as achieved in this experimental apparatus is the volume of liquid over a specified time from each of the capillaries exiting the distributor. The fixture designed and built to make these measurements is discussed in section 3.3.

The observation is made here that different mixture qualities resulting from different mass quantities of liquid into each capillary will result in varying flow regimes and pressure drops throughout the evaporator coil. Analyses can hence be made on individual flow regimes and pressure drops resulting from the differing mixture qualities if maldistribution has occurred. The components required to capture the distribution are now presented in flow order.

3.2 The Reservoir

Testing of the distributor assembly required simulation of the inlet and exterior environments. The standard unit chosen to be analyzed did not contain a thermal expansion valve in the circuit connecting the high and low pressure sides of the vapor-compression cycle. Without that additional pressure loss, inlet conditions to the distributor can be taken as the outlet condenser environment less any frictional or head pressure drops in the connecting line from condenser to distributor. Assuming these losses to be negligible the distributor inlet conditions were taken at a 15°F subcool from standard saturation conditions i.e. the outlet condenser condition. For the experiment, some type of reservoir maintained at this state and containing sufficient volume to run the experiment was required. Note that the design described below is not restricted to the subcooled conditions chosen.

3.2.1 Volume Requirements

A minimal volume of refrigerant in the reservoir must be maintained to assure unimpeded control of environments throughout the apparatus as well as a test of sufficient duration to achieve reliable results. The volume requirements were determined assuming the volume is achieved initially from charging. Note that measuring refrigerant mass required is a viable option but volume calculation is necessary to size the reservoir. (Knowledge of mass required will help in assuring proper charge, however.) Note further that because each of the required volume sources were at different thermodynamic conditions, the mass requirements were determined from local conditions and the equivalent volume from properties at ambient conditions. Three primary volumes of refrigerant fluid were considered. The first required volume is that necessary to run one test for the chosen duration of time. Typical mass flow rates for the refrigerant mixture through a standard coil were supplied by the sponsor for this project. Using average mass flow rate, the test duration, and the approximate ambient liquid specific volume, the first volume requirement was calculated from (3.1).

$$V_{f_{TEST}} = \dot{m}_f \quad \Delta t \quad \upsilon_f\left(@T_{ambient}\right) \tag{3.1}$$

As briefly discussed at the beginning of this chapter, the main pressure vessel was designed to simulate the evaporator conditions. As will be discussed below, the evaporator coil saturation pressure is achieved by evaporating sufficient refrigerant vapor prior to the actual blow-down. The second volume, therefore, is the R22 liquid required to achieve these conditions. A simple calculation of vessel volume and condensing heat exchanger volume, specific volume of vapor at the evaporation temperature, and the approximate ambient liquid specific volume was utilized (3.2).

$$V_{f_{PV}} = \left(\frac{V_{PV} + V_{Cond}}{\upsilon_g @T_{evap.}}\right) \quad \upsilon_f @T_{Subcool} \tag{3.2}$$

The third volume is related to the cartridge/immersion heaters utilized to evaporate the required vapor as discussed above. To prevent burnout of the heaters at any time during set up of the test environment or during the test, the heaters must be covered in fluid at all times. The required volume is somewhat of an approximation as the duration of heater use can vary with the surrounding environmental conditions (heat transfer into the vessel relative to ambient conditions in the lab). The additional volume can therefore be taken as that geometrically required to cover the heaters plus an approximate amount to maintain the vessel environment once the initial conditions are achieved. Again, a simple calculation of sump volume, saturated liquid specific volume at the evaporator temperature, and the approximate subcooled liquid specific volume was required (3.3).

$$V_{f_h} = \left(\frac{V_{Sump}}{\upsilon_f @T_{evap.}}\right) \quad \upsilon_f @T_{Subcool} \tag{3.3}$$

Because the liquid volume in the reservoir is a significant parameter in both the actual test and in maintaining the environments, a sight glass was fitted to the vessel for continuous observation of the liquid level.

3.2.2 Pressure Vessel Design

With the reservoir environment (temperature and pressure) and approximate size known, a pressure vessel design including pipe size and wall thickness (schedule), flange class, and weld types could be finalized. As the volume of this vessel falls below the bounds prescribed by the ASME Pressure Vessel Code, only stress considerations needed to be made, i.e. the reservoir design did not require ASME certification. The following discussion refers primarily to both the reservoir and condensing heat exchanger vessel designs with minor consideration given to the Code. Certain ASME Code requirements will be included in the main vessel design of Sec 3.3 for reasons specified there.

Various handbooks are available to assist the pressure vessel designer in choosing the appropriate pipe, flanges, and bolts for a given application. In the development of the reservoir and condensing heat exchanger vessels the equations and tables of Marks', Moss, Bednar, and Harvey were utilized to assist in making preliminary material decisions [2, 3, 7, 14]. Complimentary stress calculations were made to check the resulting factors of safety. The designation is made at this point between "operating" pressure as the pressure at the internal top of the vessel under operating conditions and the "design" pressure which is the pressure for which pressure relief valves are gauged. Note that the operating pressure must be less than the design pressure.

Using the required reservoir volume, a pipe diameter and length were determined (Figure B-1). Considering moderate temperatures and approximately 250 psia pressure pipe type and material was chosen for the vessel [2, 14]. Typically pipe specification is given with reference to a nominal pipe size (NPS) and Schedule number. The schedule number is a measure of wall thickness and can be approximated from (3.4) [2]:

$$Schd.No. = 1000\frac{P}{S\ E} \tag{3.4}$$

where P is the operating pressure (psig), S is the allowable stress for the pipe material chosen, and E is a quality factor or measure of joint efficiency (e.g. relative to joint welds) which can be determined from Tables 8.7.16 and 8.7.17 of Marks) [1, 2]. The greater the schedule number, the larger the wall thickness, therefore 3.4 represents a minimum. Using the allowable stress values corresponding to the operating temperatures of the reservoir the minimum required wall thickness can also be calculated from (3.5) [1, 2]:

$$t_m = \frac{PD}{2(SE + Py)} + A \tag{3.5}$$

The above relationship is an adaptation of that given in the ASME Code for Pressure Piping (B31.1). A simple stress analysis for a cylindrical shell gives the following result.

The reservoir was analyzed as a cylinder under internal pressure. The pipe chosen from volume requirements and analysis of the code was such that thin-walled theory could not be applied. The following relationships of Shigley and Mischke for stresses in cylinders under static equilibrium were therefore utilized in the analysis (3.6), (3.7), (3.8) [13]:

$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2 - r_i^2 r_o^2 (p_o - p_i) / r^2}{r_o^2 - r_i^2} \tag{3.6}$$

$$\sigma_t = \frac{p_i r_i^2 - p_o r_o^2 + r_i^2 r_o^2 (p_o - p_i) / r^2}{r_o^2 - r_i^2} \tag{3.7}$$

$$\sigma_l = \frac{p_i r_i^2}{r_o^2 - r_i} \qquad (3.8)$$

where σ_r, σ_t, and σ_l represent the radial, tangential (hoop), and longitudinal stresses respectively. The above were applied assuming the representative element was far from stress concentrations and free of defects. The maximum principle stress, the hoop stress, was then compared to the yield strength of the material to determine if the subsequent factor of safety was satisfactory as deemed by the Code. Note that the ASME Boiler and Pressure Vessel Code prescribe a factor of safety equal to 4 [7].

For access to the heating coils described below, it was decided that one end of the reservoir have a flange seal and the other a flat head welded to the shell. The flange and bolt assembly was chosen based on pressure and temperature requirements for the reservoir and the mating nominal pipe size (Reference Tables 8.7.36 and 8.7.38 of Marks'). The type of welds required for both the flange assembly and flat head were determined from discussions and similar applications found in the literature [1, 3, 14] (Figure B-10).

Gasket material and type were determined based on the enclosed medium as well as flange configuration.

3.2.3 Control of Environment

With the size and material requirements of the reservoir determined, the control system can now be developed. As discussed in the introduction to this section (3.2), the reservoir must be maintained at the condenser outlet conditions i.e. subcooled to a temperature lower than greater than condenser saturation. It was presumed the environment could be achieved and maintained through a combination of pressurized nitrogen vapor introduced into the vessel and heating via a cool water/hot steam mixture running tube-side through the vessel (Figure B-10). The use of cool water is beneficial for two reasons beyond it availability. One, in combination with hot steam it allows for adjustable control of heat exchange inside the reservoir (e.g. to produce varying degrees of subcool), and two, cool water is utilized to assist in the final function of the reservoir at the end of a test.

In designing a control for the reservoir environment both charging of the system and actually creating and maintaining condenser conditions were considered. The refrigerant comes as a saturated liquid-vapor mix under pressure in 30 lbm drums. Charging the reservoir consists of creating and maintaining a pressure differential between the supply tank (high) and reservoir (low) until the desired mass of vapor (which will condense) has been transferred. Using a mixture of cool water and hot steam through a coil in the reservoir can give a steady rise in temperature toward that of the condenser exit subcooled environment. Application of regulated nitrogen vapor would then raise the pressure above the corresponding saturation point toward the required subcooling. Having created the appropriate conditions in the main vessel, blow-down of the actual experiment can be expedited.

Given the readily available heat exchange media, water/steam, the specifications for a heat exchanger were developed. Though utilized for stability of R22 stored in the system, the primary function of the heat exchanger involves achieving the subcooling temperature desired.

The heat exchanger with warm water flowing tube-side in a coil configuration was modeled as a horizontal cylinder immersed in an R-22 pool. The analysis was simplified further by considering a segment of the coil as a cylinder of constant surface temperature. There was a degree of freedom in the design in that the required heat exchange could always be achieved provided the lengthy transient periods were not an issue. As a result, there are some approximations in the analysis. To determine the length of coil required to achieve the distributor inlet conditions, the amount of heat necessary to raise the temperature of subcooled R22 from room temperature to condenser saturation conditions was determined from a First Law relation (3.9).

$$W - Q = m \quad C_p \quad (T_2 - T_1)$$
$$\dot{Q} = \left[\frac{\rho \quad V \quad C_p \quad (T_2 - T_1)}{\Delta t} \right] \tag{3.9}$$

In the relation above both ρ, the saturated liquid density, and C_p, the specific heat were found for R22 at the bulk temperature, T_b. The volume V is, of course, the volume of liquid in the reservoir and t is a logical period of time for heat transfer based on the careful preparation of subcooled liquid as mentioned earlier. T_1 and T_2 are the initial and final temperatures of the liquid respectively. The length of coil can now be determined from the convective heat transfer relation 3.10):

$$\dot{Q} = \bar{h}_o \quad A_{s,o} \quad (T_{s,o} - T_{\infty,o}) \tag{3.10}$$

where Q is the total heat transfer rate for the coil, equation (3.2.9), h_o is the average convection coefficient for the liquid over the entire exterior surface of the coil, $A_{s,o}$, $T_{s,o}$ is the outer surface temperature of the coil and $T_{\infty,o}$ can be taken as equal to the bulk temperature. The equation can be simplified further to identify the length of coil, L:

$$\dot{Q} = \bar{h}_o \quad 2\pi r_o L \quad (T_{s,o} - T_{\infty,o}) \tag{3.11}$$

The convection coefficient was determined from empirical relations for a long horizontal circular cylinder geometry and isothermal surface. Incropera and Dewitt give two such relations [10] (3.12):

$$Nu_D = h_o \frac{D}{k} = C \quad Ra_D{}^n \tag{3.12}$$

as suggested by Morgan [10] where C and n are tabulated, and (3.13):

$$Nu_D = \left\{ 0.6 + \frac{0.387 Ra_D{}^{1/6}}{\left[1 + (0.559 / \Pr)^{9/16}\right]^{8/27}} \right\}^2 \tag{3.13}$$

as suggested by Churchill and Chu [10]. Both relationships above determine the average Nusselt number over the circumference of the circle as Nu_D is influenced by boundary layer growth. The Rayleigh number, Ra, is found from the Grashoff and Prandtl numbers by (3.14) [10]:

$$Ra_L = Gr_L Pr = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} Pr \tag{3.14}$$

where L is the characteristic length of the geometry equal in this case to the outer diameter of the copper tubing used in the coil. All properties in the above relations are calculated for the liquid at the film temperature (3.15).

$$T_f = \frac{T_{s,o} + T_b}{2} \tag{3.15}$$

The relation of Morgan given above (3.12) gave the most conservative approximation to the length of coil required and therefore was utilized in the design. To solve for the convection coefficient, and therefore the length of coil required to achieve the desired temperature of R22, the outer coil surface temperature $T_{s,o}$ was approximated. For this heat exchanger the above analysis was assumed satisfactory due to the freedom of operating parameters. However, this analysis will be completed in Sec. 3.2.3 for the condensing heat exchanger design with modifications made for condensation.

A final note is made before leaving this section. If the coil requirements had exceeded the preliminary geometry of the vessel, the additional length of nominal 8 in. pipe required would not have effected the pressure vessel specs.

3.3 The Main Pressure Vessel

The main pressure vessel is signified as such due to the central role of this component in the apparatus. The environment maintained inside the vessel is that of the distributor and capillary assembly exit (evaporator inlet) as determined from literature provided by the sponsor. The vessel contains all of the measurement equipment as well as means to control the environment (Figures B-1 through 9). Prior to actual blow-down the desired environment is achieved by producing vapor from an initial volume of refrigerant in the base of the vessel (as described below, Section 3.3.3). Heat is added for this task from the surrounding ambient conditions and from heaters installed in the sump blind flange. During the test, additional vapor produced by flashing liquid through the throttle, distributor, and capillary assembly is controlled by communication with a condensing heat exchanger designed for that purpose (Section 3.4).

3.3.1 Measurement Equipment

As discussed in Section 3.1, the liquid phase is the primary phase of interest and the apparatus was subsequently designed to determine the liquid distribution into the evaporator coil. Several complimentary components were designed and built to enable liquid measurements of flow exiting the distributor and capillary tube assembly (Figures B-5 through 9). The methodology utilized included measuring the liquid volume from each capillary with individual sealed glass cylinders and valved drains. Viewing for measurement was achieved with a sight glass installed in one blind flange of the vessel. The vessel size was determined after the measurement equipment needs were known.

It is noted here that the design intent included simplicity and ease of use. Reliability was placed on mechanical motion to eliminate the unnecessary complication of electrical components. It was mentioned previously that refrigerant creates problems for plastics and rubber. For that reason, seals throughout the apparatus, including gaskets for the flanges, were achieved with either neoprene, the compatibility is noted by Parker Company (Parker ® O-ring), or Teflon. The small amount of wiring utilized for the lamp was protected by neoprene tubing where exposed to refrigerant.

3.3.1.1 Volume Measurements

To measure the liquid phase, multiple considerations were required. First, the maximum liquid volume attainable for a test of the chosen duration was determined to properly size the measurement glasses. Second, the flow rate of the two-phase mixture was studied in a separate water-air apparatus (Section 3.3.1.2) to determine how flow direction into the glass cylinders could effect the amount of liquid retained while allowing the vapor to escape. Third, a fixture was designed to direct the flow as prescribed by those experiments. Fourth, a drainage system from each cylinder was required to retain the closed cycle/system nature of the apparatus while further allowing a series of successive tests. Note, however, that given sufficient time in a warm ambient (laboratory) environment, evaporation would achieve the same goal. Fifth, based on the glass cylinder design a mechanical control system was necessary to bring each cup within sight of the window for measurement (Figure B-7). Sixth, based on the results of the above design, a fitting was required to allow controlled rotation of the distributor as measurements were taken (Figure B-9). Finally, and perhaps most pivotal, a method for illumination inside the vessel was necessary. This interesting problem had few applicable solutions that were discovered during the course of the apparatus design.

Once the basics of how liquid measurement was to be achieved were determined, the above ideas were checked for feasibility mechanically and geometrically. Only the first consideration required flow calculation. Using the two-phase relations of Chapter Two and the data given on mixture mass flow rate through the distributor, the volume requirements for an assembly of five circuits during a specified blow-down period were determined. Sufficient measurement volume was included for eight circuits or less where connections between a full and empty cylinder could be achieved if volumes became excessive.

A few comments regarding illumination inside the vessel are in order. It would seem that ordinary household light bulbs would not sustain pressures in excess of 240 psia. A hunt was therefore undertaken to acquire some type of light source and housing assembly that could be utilized inside the main pressure vessel. No such assembly was found to commercially exist after consulting several sources. Discussion, however, with a member of the oceanography community [12] resulted in several ideas as well as the current solution. It was learned that small utility and appliance light bulbs could indeed sustain such pressures. Other suggestions included fluorescing tape and an electro-luminescent apparatus. Both were unacceptable in a refrigerant environment. LED's and fiber optics were also considered. The final design includes a small light bulb with a porcelain fixture. Power is provided via electrical hookup through a neoprene plug. Consideration was given to cost, sealing, and the amount of light necessary.

3.3.1.2 Velocity Apparatus

As discussed in Chapter 2 the flow velocity through the distributor is such that containment of the liquid flow out of the capillaries as well as retainment of the capillaries was a concern in the design. As a means of preempting any difficulties regarding the above, a small air-water apparatus was designed and built to analyze the potential flow patterns existing at exit from the capillaries as well as how much the flow momentum could disrupt the placement of the capillary (Figure 3.3-1).

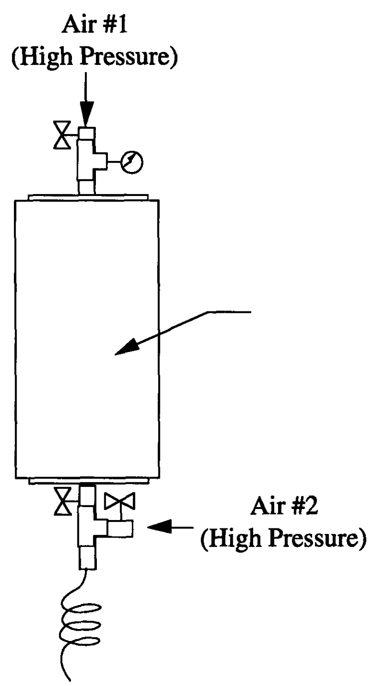

Figure 3.3-1: Velocity Test Apparatus

Using a two-circuit (two paths from the distributor) pattern as the most conservative analysis (highest flow velocities) approach, the liquid velocity was determined from Equation (3.16):

$$V_{f_2} = \frac{Q_{f_2}}{\frac{\pi}{4} d_i^2 (1-\alpha)} \tag{3.16}$$

where the number 2 corresponds to the number of circuits and Q is the corresponding liquid volumetric flow rate. Applying results from above to Bernoulli for steady, frictionless flow out of a tank, the pressure required to achieve that velocity with water was determined. With the aid of a pressure regulator to specify air pressure, several tests were run with water flowing from a capillary. Both the inclination of flow into the receiver and the momentum effects of the liquid were observed. Volumetric flow rate was also measured to expose errors and potential leaks in the apparatus.

After achieving a satisfactory arrangement for liquid alone, the flow spiraled into the glass smoothly and with no disruption, a second air inlet (#2 in Figure 3.3-1) was added to the experiment. Though the air pressure at point of injection was very approximate to that required for distributor vapor flow, the tests results were useful is showing how the higher speed vapor can disrupt the flow. The brief experiments above helped determine how varying degrees of inlet inclination affected not only the way in which flow was received but how easily the vapor could move out of solution.

3.3.1.3 Back Pressure Control

One consideration was included in the apparatus design, not as a requirement for the preliminary experiments included here, but in preparation for further tests of the distributor assembly. As discussed in Chapter 2, the varying pressure gradients for flow across the distributor and capillaries can result in changing flow regime, e.g. "off-design" conditions, and the potential for maldistribution. As a means of controlling the pressure gradient across the assembly modification of the receiving pressure for separate capillaries was made possible. It was important that in controlling the back pressure, the integrity of the fluid characteristics is not jeopardized. Included in the design, therefore, was the placement of cups connected to the receiving glasses via a sealed line. The static head of these secondary cups would then produce the receiving pressure desired. Note that this method has not been fully investigated. The probability of choked flow for the pressure gradient existing between the condenser and evaporator coils may eliminate the usefulness of such a design (i.e. the back pressure would have to be significantly higher than the evaporator conditions).

3.3.2 Pressure Vessel Design

In the reservoir discussion (Section 3.2) a bound was mentioned regarding the ASME requirements on pressure vessel builds. Because the main vessel lies outside the region of exemption, i.e. the volume exceeds one cubic foot, its design and build were subject to ASME certification. Study of the code resulted in a preliminary design which subsequently went to an outside source with license to design and build such a vessel. It is noted that all characteristics of the basic preliminary design save weld specifications were approved by this ASME certified vendor (Appendix C).

Similarly to the reservoir design, pipe material and type were determined for the body of the vessel based on design pressure and temperature requirements. The design was complicated, however, by the desired 'tee' shape (Figure B-2) and resulting stress concentrations. Using an inner diameter and vessel length prescribed by the measurement equipment, the Code was consulted to determined size limitations and reinforcement requirements for openings ([1], Part UG).

The required wall thickness for a pipe used as a pressure vessel was determined in accordance with Part UG-27 of the Code for cylindrical shells [1, 14] (3.17):

$$t = \frac{PR}{SE - 0.6P} \qquad (3.17)$$

The above calculation was done for both the main shell and attachment. Note that the shell thickness was determined first and then a required reinforcement for the attachment was determined. An iteration would have been possible had the required reinforcement been excessive. The resulting circumferential stress was found from (3.18) [1, 14]:

$$\sigma_t = \frac{P(r_i + 0.6t)}{2Et} \qquad (3.18)$$

and compared to the yield strength of the material for a factor of safety calculation. Note that S, the maximum allowable stress was found from Section III of the Code for the operating temperature of the vessel and E, the joint efficiency (circumferential in this case) was found from Table UW-12, Section VIII, Div. 1 of the Code [1].

3.3.3 Control of Environment

The environment to be sustained inside the main vessel corresponds to that of the evaporator inlet as discussed earlier. For analysis of flow distribution, the drop in pressure between condenser (reservoir) and evaporator (main vessel) is the primary variable. Sufficient heat exchange is achieved from ambient conditions (greater than evaporator saturated temperature) to maintain R-22 vapor at the evaporator saturation pressure inside the main vessel. The condensing heat exchanger described in Section 3.4 below in combination with a bleed, maintains the desired condition. Note that cartridge heaters were added to the design as a supplementary heat source for evaporation. Typical laboratory conditions, however, produce sufficient warmth for evaporation to occur.

A thermocouple mounted to the main vessel base in combination with the pressure gauge on the condenser gives a clear indication of conditions within these environments. The system pressure release valve is affixed to the main vessel as required for ASME certification.

3.4 The Condensing Heat Exchanger

The final component of the apparatus to be discussed is the condensing heat exchanger included to control the main vessel environment. To maintain the saturated evaporator conditions of the main vessel during blow-down, the increase in pressure resulting from flashed vapor must be controlled. Control is achieved by condensing the extra vapor via a heat exchanger with continuously circulating ice water coils as a heat sink and a nitrogen vapor feed and bleed combination maintaining pressure (Figure B-1).

Prior to blow-down, with the heaters or surrounding ambient environment preparing vapor, the condensing heat exchanger is open to the vessel environment to both scrub the condenser of air and to prepare the condenser for a distinct interface with the nitrogen vapor. During a test, the nitrogen-R22 vapor interface moves up as excess R22 vapor flows into the heat exchanger causing the pressure to rise toward the required evaporator condition. As the R22 vapor comes in contact with the coils condensation occurs. The corresponding decrease in pressure induces a potential feed of nitrogen into the chamber and the interface moves back down. The evaporator pressure is therefore maintained. Note that the bleed is hand controlled via a ball valve whereas the feed is controlled via a pressure regulator from the nitrogen tank.

3.4.1 Control of Environment

Unlike the reservoir which is primarily a pressure vessel the condenser is primarily a heat exchanger. It's size was therefore determined based on the heat transfer characteristics required. Because the analysis is iterative as approximations are made and checked, the following was incorporated into a short Matlab code (mathematics software) to ease computation. The basis is presented here.

The primary design parameter was the length of tube required to condense R22 vapor at the saturated evaporator conditions existing in the main vessel. The following analysis was undertaken with warm R22 vapor flowing over tubes carrying cold water (Figure 3.4-1).

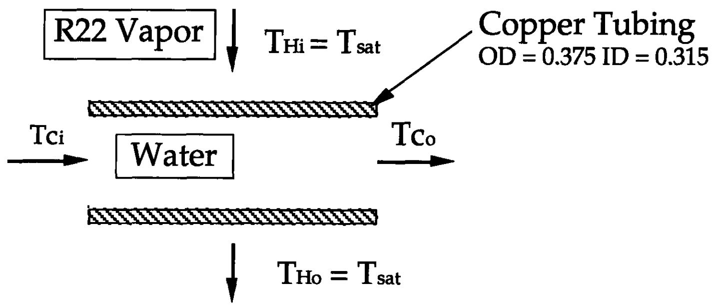

Figure 3.4-1: Condenser Heat Exchange Flow Model

The coil heat exchanger was designed based on the analogy of laminar film condensation on a bank of tubes of constant surface temperature. Two primary assumptions must be stated. First, the R22 vapor was assumed to have negligible velocity over the tubes so that the laminar film condensation relation discussed below can be utilized. Second, there is no significant change in either potential or kinetic energy of the system and heat exchange occurs only between the vapor and coil [10].

Using the model from above, the following heat transfer rate relationship was utilized (3.19):

$$\dot{q} = U_i \quad A_{Si} \quad \Delta T_{lm} \tag{3.19}$$

where U_i is the overall heat transfer coefficient based on the interior surface of the coils, A_{si} is the interior surface area, and ΔT_{lm} is the log mean temperature difference. Note that the log mean temperature difference is independent of heat exchanger configuration (e.g. parallel flow vs. counterflow) as one side is changing phase (i.e. the temperature is constant). Written in terms of the desired parameter the relation becomes (3.20):

$$\dot{q} = U_i \quad \pi d_i L \quad \Delta T_{lm} \tag{3.20}$$

There is typically a substantial number of degrees of freedoms in the design of any heat exchanger, enough to confuse the designer. Two choices were made, therefore, at the onset of this analysis. First, the tubing is chosen as commercially pure copper. Second, the mass flow rate of cooling water was chosen (though this value can be limited to the size of pump required to achieve it). Several parameters must be calculated to determine the length, L, of coil required. The overall heat transfer coefficient based on the inner surface area of the coil can be found from (3.21) (Note that the addition of fouling resistance was assumed negligible):

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{d_i}{2}\frac{ln(d_o/d_i)}{k} + \frac{1}{h_o}\frac{d_i}{d_o}} \tag{3.21}$$

Calculation of the convection coefficient, h_o, for the R22 vapor required an initial approximation for the exterior surface temperature, T_{so}, of the coil. The analysis by Nusselt of film condensation on vertical plates surrounded by stagnant, saturated vapor has been extended to laminar film condensation on the outer surface of horizontal tubes [10]. The resulting relationship for the average convection coefficient, h_D [10] is (3.22):

$$ho = \bar{h}_D = C\left\{\frac{\rho f(\rho f - \rho g)k_f{}^3 h'_{fg}}{\mu_f(T_{sat} - T_{s,o})d_o}\right\}^{0.25} \tag{3.22}$$

where C=0.729 for horizontal tubes. The relationship was modified for a vertical tier of N tubes [10] (3.23):

$$ho = \bar{h}_D = C\left\{\frac{\rho f(\rho f - \rho g)k_f{}^3 h'_{fg}}{N\mu_f(T_{sat} - T_{s,o})d_o}\right\}^{0.25} \tag{3.23}$$

In the relationship above, the density, conduction coefficient and viscosity for saturated R22 liquid were found at the film temperature (3.15). The density for saturated R22 vapor was found at the saturation temperature, T_{sat}. Not knowing the final configuration of tubes, the number of tubes, N, was approximated and then checked after the required length of coil was determined. The h_{fg}' term in (3.22, 23) is the "corrected" heat of vaporization which can be found from (3.24) [18]:

$$h_{fg}' = h_{fg} + 0.68 C_p \left(T_{sat} - T_{s,o} \right) \tag{3.24}$$

where the heat of vaporization, h_{fg} was found at the saturation temperature, T_{sat} and the specific heat, C_p, at the film temperature, T_f. If the heat of vaporization had not been corrected, Equation 3.22 would give an average convection coefficient assuming that all heat transfer to the wall acted in the condensation of saturated vapor to saturated liquid. A small portion of the heat transfer, however, acts to cool the condensate to some temperature between $T_{s,o}$ and T_{sat}. Modifying h_{fg} accounts for the resulting slight decrease in condensation rate.

To determine the interior convection coefficient, hi, a convection correlation for internal flow in a circular tube was required. The assumption was made that the water flow is fully developed through the coil. The single phase flow regime (laminar vs. turbulent) was determined from the Reynolds number calculation (3.25):

$$Re_i = \frac{\rho_w \; u \; d_i}{\mu_w} = \frac{\dot{m}_w \; d_i}{A_i \; \mu_w} = \frac{4 \; \dot{m}}{\pi \; d_i \; \mu_w} \tag{3.25}$$

For fully developed turbulent flow in a smooth pipe the empirical Dittus-Boelter equation was utilized to determine the local Nusselt number [10] (3.26):

$$Nu_D = 0.023 \quad Re_D^{4/5} \quad Pr^n \qquad \begin{bmatrix} 0.7 \le \Pr \le 160 \\ \mathrm{Re}_D \ge 10{,}000 \\ \frac{L}{d} \ge 10 \end{bmatrix} \tag{3.26}$$

where n=0.4 for heating of the internal flow and the restriction of moderate temperature difference between the bulk water temperature and the tube interior surface temperature must apply [10]. As noted previously, the small difference between the temperature of the ice water and the saturation temperature of the evaporator results in a large heat exchanger. With a long tube, the average Nusselt number over the length was assumed approximately equal to the local value found from (3.26). The fluid properties in the internal flow relations above were found at the bulk flow temperature of the water. To determine the bulk temperature, the mean inlet and exit conditions must be know. The inlet temperature was taken as 32°F. The exit temperature was determined from the First Law relation (3.27):

$$\dot{q} = \dot{m}_C \quad C_C \quad (T_{Co} - T_{Ci}) = \dot{m}_H \quad h_{fg}' \tag{3.27}$$

The value of T_{co} was first estimated to find the specific heat for the water at its bulk temperature. The resulting value of T_{co} was then compared and iterated until two successive values approximated each other. The final value of T_{co} was utilized to find the bulk temperature and, hence, the properties for the internal flow relations above. The mass flow rate of vapor, m_H was taken as the flow rate of vapor from the distributor.

A comparison was made at this juncture between the bulk temperature of the water, T_{bw}, and the coil exterior surface temperature, T_{so}. If $T_{bw}>T_{so}$ it is necessary to go back and choose a larger value of T_{so}. If $T_{bw}<T_{so}$ a sufficient number of puzzle pieces are now present to go back and check the T_{so} approximation directly (3.28):

$$\pi d_o \quad h_o \quad (T_{sat} - T_{s,o}) = \frac{T_{sat} - T_{bw}}{\dfrac{1}{\pi d_i h_i} + \dfrac{1}{2\pi k}\ln\left(d_o/d_i\right) + \dfrac{1}{\pi d_o h_o}} \tag{3.28}$$

The above interior flow analysis from was then iterated until two successive values approximated each other.

With the convection coefficients known, the overall heat transfer coefficient can be calculated from (3.21). With the inlet and outlet cooling water temperatures known, the log mean temperature difference can be calculated [10] (3.29):

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{log(\Delta T_2/\Delta T_1)} \tag{3.29}$$

where $\Delta T_2 = (T_{sat} - T_{co})$ and $\Delta T_1 = (T_{sat} - T_{ci})$. With the heat transfer rate from (3.27) the length of coil required was determined from (3.20).

The height of coil, and subsequent size of the heat exchanger was determined assuming the coils were spaced at outer diameter intervals (Figure B-12). If the resulting heat exchanger size logically seem unreasonable, a increased value of cooling water mass flow rate could have been utilized.

The final development of environment control for the condensing heat exchanger was to determine pumping power requirements for the chosen mass flow rate of recirculating cooling water. A reasonable mass flow rate of water had been chosen based on experience in the heat transfer field [15] and it was therefore expected that the pump requirements would be reasonable. Using the system schematic (Figure B-1) and the approximated length of flow line outside the heat exchanger the following head relationship was utilized throughout the circuit to determine the pressure rise required across the pump (3.30):

$$\left(\frac{p_1}{\rho_1} + \frac{1}{2}u_1^2 + gz_1\right) = \left(\frac{p_2}{\rho_2} + \frac{1}{2}u_2^2 + gz_2\right) + W_S + g\Delta h_{TotalLosses} \tag{3.30}$$

The head loss term can be determined from (3.31) [22]:

$$\Delta h_{TOT} = \frac{u^2}{2g}\left(\frac{fL}{d_i} + \sum K\right) \tag{3.31}$$

where the velocity u can be written in terms of the mass flow rate and density at the bulk temperature, f is the Moody friction factor, and K is a loss coefficient found for flow through elbows, sudden contractions, etc. [22]. By summing the losses throughout the cooling water circuit the total head loss was determined from (3.31). Calculation of the pressure drop of the entire circuit allowed for proper pump sizing. Two bypass loops were included with the pump installation to help achieve fine tuning of the coolant mass flow rate.

3.4.2 Pressure Vessel Design

The design of the condensing heat exchanger as a pressure vessel follows precisely the analysis utilized for the reservoir. Knowing the length of coil required to achieve the desired heat exchange, the nominal pipe size and subsequent flange size were determined. The pipe thickness and weld requirements were determined as described in Section 3.2.2.

3.4.3 Flooding Considerations

A subtle, but important aspect of the condenser design involved the potential for flooding to occur in the pipe connecting the heat exchanger to the main pressure vessel. Flooding is a phenomena which develops in a vertical or severely inclined pipe if the flow of a falling liquid film is impeded by an up flow of vapor. The most predominant explanation for the phenomena involves the growth of waves at the liquid-vapor interface for certain combinations of liquid and vapor flow rate [21, 22]. The visualization experiments of McQuillan [21] showed that the onset of flooding is marked by the slowing down of wave motion at the interface and the appearance of one stationary wave.

As other waves collide with the stationary wave, a wall of liquid grows until the vapor sweeps it upward.

During blow-down through the distributor the increase in pressure resulting from flashed vapor is relieved by the condensation of that vapor in the condensing heat exchanger. The pipe connecting the heat exchanger and main pressure vessel is a region susceptible to a flooding instability. Sufficient vapor flow up the incline will prevent the gravitational downward flow of the condensate. The pipe was chosen with sufficient diameter to prevent such an occurrence.

A brief analysis was done by Whalley to determine the vapor flow rate required to hold a liquid wave stationary at the liquid-vapor interface. The resulting correlation [21] (3.32):

$$j_g^* = j_g \quad \rho_g^{1/2} \quad \left[gd_i\left(\rho_f - \rho_g\right)\right]^{-1/2} \tag{3.32}$$

$$j_g^* \approx 0.89$$

looks very much like the experimental results found in Wallis [20] (3.33):

$$j_g^* \approx 0.9 \tag{3.33}$$

The symbol jg* refers to a dimensionless group developed by Wallis as noted above [20] (3.34):

$$jg^* = jg \quad \rho_g^{1/2} \quad \left[gd_i\left(\rho_f - \rho_g\right)\right]^{-1/2} \tag{3.34}$$

where j_g refers to the superficial velocity of the vapor defined as in Chapter 2. The inner diameter of the pipe, d_i, was chosen to achieve below the j_g* empirical limit. Note that a conservative approximation to the vapor flow rate limit is made in the analysis above as the flow rates at the onset of the flooding instability can be higher for inclined flow relative to vertical flow [20].

One further avenue for the flooding instability must be addressed and that is the onset of flooding corresponding to or resulting from an hydraulic jump. Hydraulic jumps can be characterized as a supercritical fluid changing quickly back to a subcritical fluid through a short turbulent transition region. The supercritical and subcritical descriptions of flow refer here to the speed of the flow relative to an infinitesimal surface wave. Therefore, the critical parameter effecting hydraulic jump performance is the up stream Froude number [22] (3.35):

$$Fr = \frac{C}{\sqrt{gh}} \qquad \begin{array}{ll} Fr < 1.0 & \text{Subcritical Flow} \\ Fr = 1.0 & \text{Critical Flow} \\ Fr > 1.0 & \text{Supercritical Flow} \end{array} \tag{3.35}$$

where h is the height of flow to the free surface. Downstream parameters, a change in pressure for example, can typically be communicated up stream and downstream allowing for smooth transitions in the flow, for the example a change in velocity. If, however, flow upstream is supercritical and at a point downstream subcritical flow is required, the need for transition can not be communicated and an abrupt instability results. The instability is an hydraulic jump.

As the condensate flows down from the condensing heat exchanger it must negotiate a turn and continue its descent on an incline a few degrees from horizontal. The flow just past the turn might experience an hydraulic jump. The sudden increase in condensate flow area would represent the wave growth discussed previously. With the appropriate relative velocities of liquid and vapor, a flooding instability can result. To minimize the fluid area transition as it moves through the turn, the condensate inlet was designed to limit flow in the tube to the outside edge.

4. Results

4.1 Apparatus Specifications

The results of a study to develop an apparatus consist of the apparatus specifications and some type of proof that the concept works. The apparatus presented was designed and built to test the behavior of the distributor, throttle and capillary assembly of a standard furnace coil unit under specific environmental conditions. To achieve those conditions two pressure vessels and a condenser were sized and organized into a system with both temperature and pressure control. The scaled down system schematic is shown in Figure 4.1-1. A complete set of component drawings and some scanned photographs of the rig can be found in Appendix B.

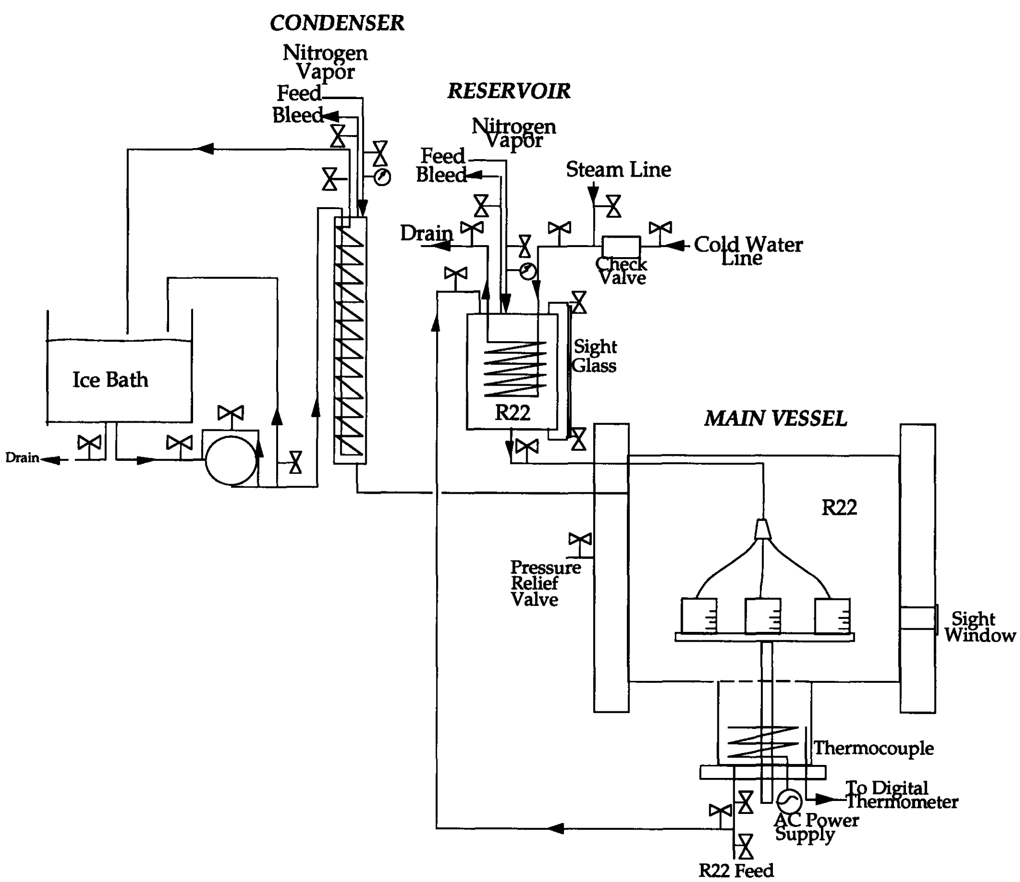

Figure 4.1-1: Rig Schematic

Table 4.1-1: Details of Apparatus Specifications:

CONDENSER	***RESERVOIR***	***MAIN VESSEL***
Body:	Body:	Body:
5in. Nom. Carbon Steel Pipe	8in. Nom. Carbon Steel Pipe	16in. Nom. Carbon Steel Pipe
Schedule 40	Schedule 40	Schedule 40, SA53B
Gaskets and Seals:	Gaskets and Seals:	Class 300 Slip-On Flange Asms
Neoprene	Neoprene	Gaskets and Seals:
Connections:	Connections:	Neoprene
3/4 in. OD Commercially Pure	3/8 in. OD Commercially Pure	Connections:
Copper Refrig. Tubing	Copper Refrig. Tubing	3/4, 3/8 in. OD Commercially Pure
Cooling Coils:	Water/Steam Coils:	Copper Refrig. Tubing
3/8 in OD Commercially Pure	3/8 in OD Commercially Pure	Cartridge Heaters:
Copper Refrig. Tubing	Copper Refrig. Tubing	(4) 505W, 120V
32°F Ice Water Bath	41°F Water, 212°F Steam	Pressure Release Valve:
Circulation:	1/2 in Black Iron Pipe	Gauged to 300psia
1 hp Bell and Gosset	Nitrogen:	Operating Conditions:
Centrifugal Hydraulic Pump	Pressure regulated supply tank	Saturated R22 Liquid and Vapor
3/8 in PVC piping	Operating Conditions:	90 psia
Nitrogen:	Nitrogen vapor	
Pressure regulated supply tank	Subcooled R22 Liquid	
Operating Conditions:	240 psia, 95∞F	
Nitrogen and R22 vapor		
R22 condensate		
90 psia		

4.2 Test of Current Distributor Design

A typical distributor assembly (supplied by the sponsor) was installed into the apparatus with a unique fitting (reference Appendix B) to test not only the apparatus design but to look at potential flow distribution. The preparation of proper environments took roughly ten hours with perhaps excessive time being allowed for settling of transients. A large portion of that time can also be attributed to familiarization of the experimenter with the thermodynamic behavior of Refrigerant 22 under varying conditions. The environmental conditions just prior to blow-down and experimental results are presented below (Table 4.2-1):

Component	Medium	State
Condenser	Nitrogen and R22 Vapor	90 psia
Main Vessel	R22 Vapor	65 deg. F
Reservoir	R22 Subcooled Liquid	240 psia, 95 deg. F

Table 4.2-1: Test Environmental Conditions

The resulting distribution of the assembly is shown as liquid heights in measurement glasses and percentage from the average in Table 4.2-2. The duration of the test was 30.54 seconds with approximately 3 in. of subcooled liquid being drained from the reservoir.

Capillary	Liquid Height (in.)	% from Average
1	4.1875	1.75
2	4.1250	0.227
3	3.7813	8.12
4	4.2656	3.64
5	4.2188	2.51

Table 4.2-2: Distribution Data - Liquid Height by Capillary

There was no visible leakage from any element within the main vessel. Flow into the glasses appeared as a uniform spray (liquid and vapor) with the liquid spiraling smoothly and being retained. Taking data immediately assured that evaporation did not severely effect results.

5. Conclusions

The apparatus satisfies the requirements for accurate and modifiable testing of a distributor assembly for a standard furnace coil unit. The components were included and subsequently designed to allow for a variety of potential thermodynamic conditions. The rig is therefore capable of simulating differing air conditioning units (condenser outlet and evaporator inlet states) subject to varying environmental conditions. With a basic understanding of the thermodynamic behavior of R22, the control of states is elementary provided time is allowed for transient behavior, in particular the homogenizing of thermal layers. All aspects of the test section and measurement equipment inside the main vessel were designed as non-destructive, i.e. each element can be removed and replaced without effecting other components. The primarily mechanical aspects of the design further assure modularity as well as space for additional measurement equipment and experimental apparatus.

6. Recommendations for Future Work

6.1 Apparatus Modifications

Of primary concern in achieving accurate representation of flow distribution is the control of pressure drops from the condenser outlet conditions to the evaporator inlet conditions. Included in the current apparatus on this flow line (from reservoir to main vessel) are two compression fittings, an elbow, and one ball valve. The valve is a necessity and can not be replaced and any associated pressure drop must therefore be accounted for in the flow analysis. The compression fittings utilize ferrules to produce a seal. Ferrules deform the tubing and subsequently create a reduction in flow area. It is suggested that these ferrules be replaced with O-rings and the fittings be supported to prevent blowing out of the tube. Similar action can be taken with the elbow. Replacing the elbow with a careful bend in the copper tubing is a possibility but the bending process typically produces a gradual reduction in flow area as well. A potential goal to achieve is the reduction of these 'excess' pressure drops to less than 10% of the total.

Another region of concern regarding flow area occurs on the 3/4 inch refrigeration line connecting the main vessel to the condenser. The size of the line was determined from flooding considerations and therefore reductions in area may inhibit proper counterflow of vapor and condensate. The compression fittings sealing this line also utilize ferrules resulting in a reduction of flow area and improved chances for flooding.

On each of the three vessels there is some type of measurement device to determine thermodynamic state (e.g. pressure gauge, thermocouple). If the fluid monitored is saturated the state is completely described, however, if subcooling exists it would be beneficial to have additional information beyond intuition and clear knowledge of one physical property. It is recommended, therefore, to improve the collection techniques for state data by, for example, including a separate pressure gauge on the main vessel (in case communication is lost between the vessel and the condenser where a gauge is already located), as well as looking into more accurate means of measurement (e.g. using voltage readout for thermocouples).

At several junctures in both the preparation of states for test and the post test collection of refrigerant, nitrogen vapor was bled from the system. It would be advantageous and environmentally correct to have a halide detector or some other means of measuring refrigerant concentrations on hand to assure that being bled is indeed nitrogen or air. The hand held detectors are expensive so it is suggested the sponsor be consulted to arrange a loan.

6.2 Analytical

The discussion presented for two-phase flow analysis and flow regime determination is by no means comprehensive enough to achieve a full understanding of the flow both prior and after distribution. An in-depth use of the separated flow model as well as the 1990 Taitel unified flow regime model is necessary to gain a full understanding of how pressure drops can affect flow distribution.

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Appendix A: Nomenclature

A	Crossectional Flow Area
A	Coefficient [1]
A_s	Surface Area for Heat Transfer
d_i	Inner Diameter
d_o	Outer Diameter
D	Outer Pipe Diameter
E	Quality Factor, Measure of Joint Efficiency [1]
G	Mass Flux
$\bar{h}_D$	Average Convection Coefficient - Outer Flow
h_{fg}	Heat of Vaporization
h_{fg}'	Corrected Heat of Vaporization
h_i	Convection Coefficient - Inner Flow
h_o	Convection Coefficient - Outer Flow
j_g	Vapor Superficial Velocity
j_f	Liquid Superficial Velocity
k	Conduction Coefficient
$\dot{m}_f$	Liquid Phase Mass Flow Rate
$\dot{m}_g$	Vapor Phase Mass Flow Rate
$\dot{m}_T$	Mixture (Total) Mass Flow Rate
p_o	External Applied Pressure
p_i	Internal Applied Pressure
P	Maximum Operating Pressure (for Vessel)
$\dot{q}$	Rate of Heat Transfer
$\dot{Q}$	Rate of Heat Transfer
Q	Volumetric Flow Rate

r_i	Inner Radius
r_o	Outer Radius
R	Outer Pipe Radius
S	Maximum Allowable Stress (for Vessel material)
t	Material Thickness (for Vessel)
Δt	Time
u	Velocity (Condenser Cooling Water)
U_i	Overall Heat Transfer Coefficient - Based on Inner Surface Area
V_f	Liquid Phase Velocity
V_f	Refrigerant Volume
V_g	Vapor Phase Velocity
y	Coefficient [1]
α	Void Fraction
β	Volumetric Thermal Expansion Coefficient (Tabulated)
ρ_g	Vapor Density
ρ_f	Liquid Density
υ_f	Liquid Specific Volume
υ_g	Vapor Specific Volume
θ	Flow Inclination

Appendix B: Component Machine Drawings

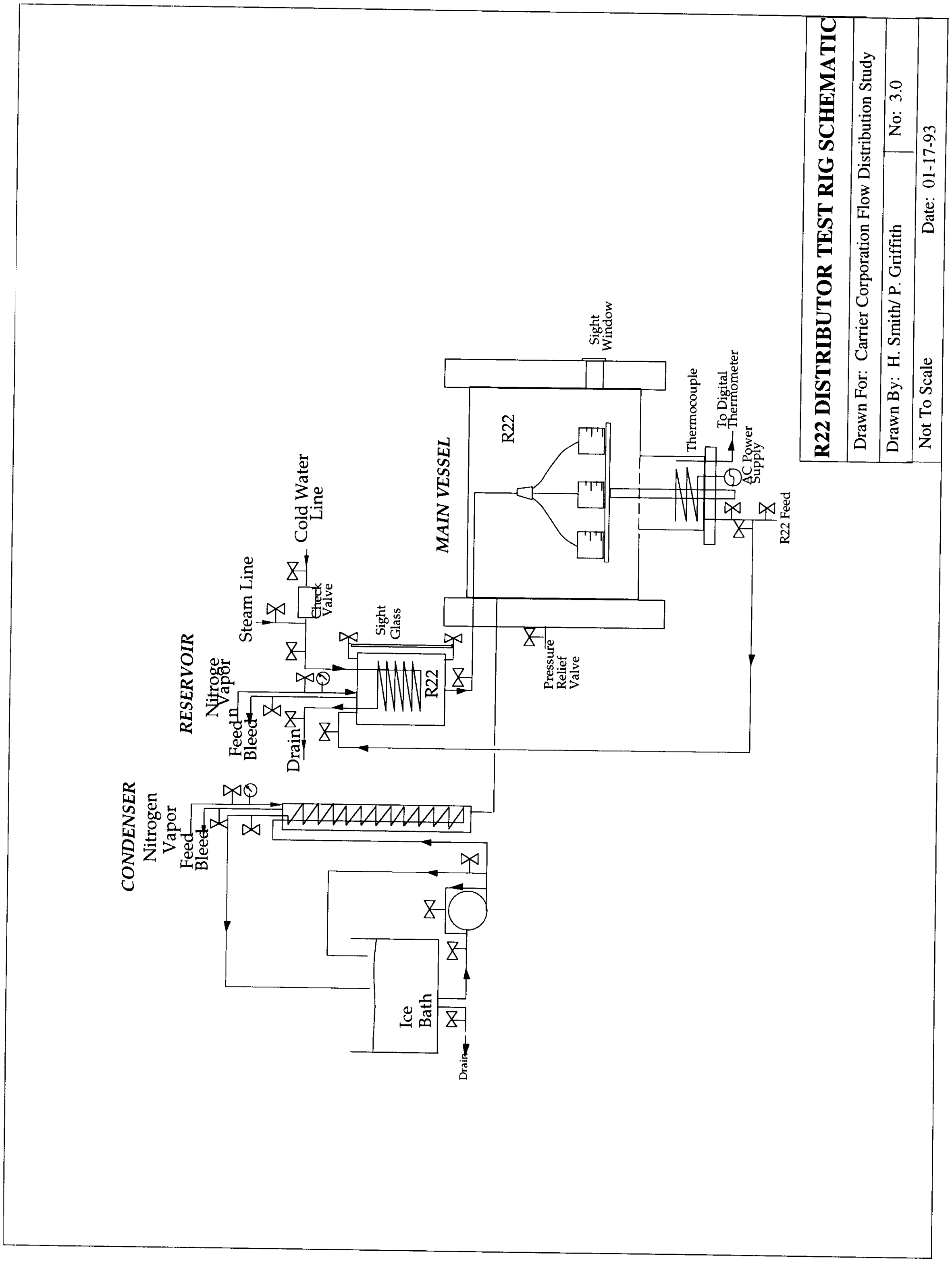


Figure B-1: R22 Distributor Test Rig Schematic

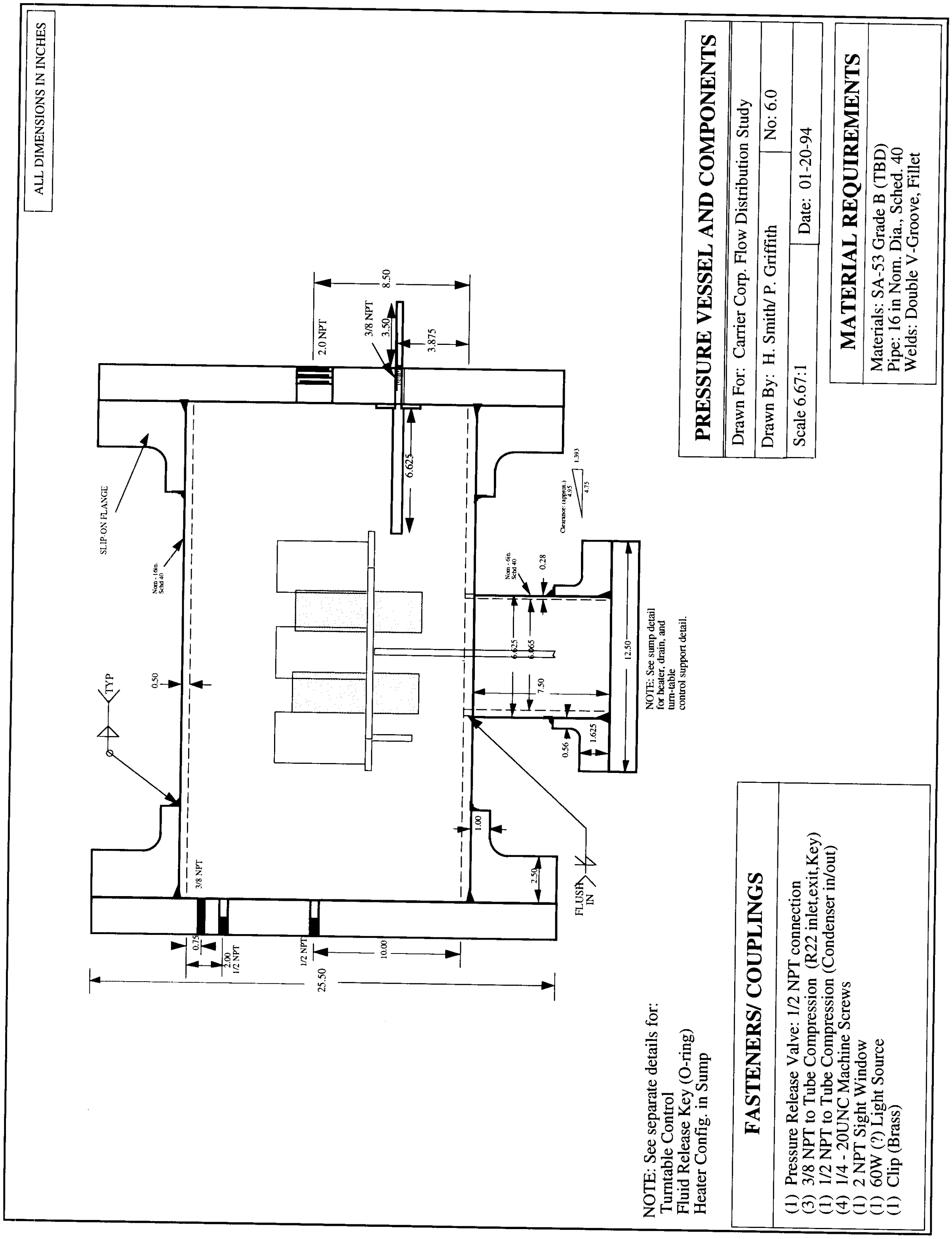


Figure B-2: Pressure Vessel and Components

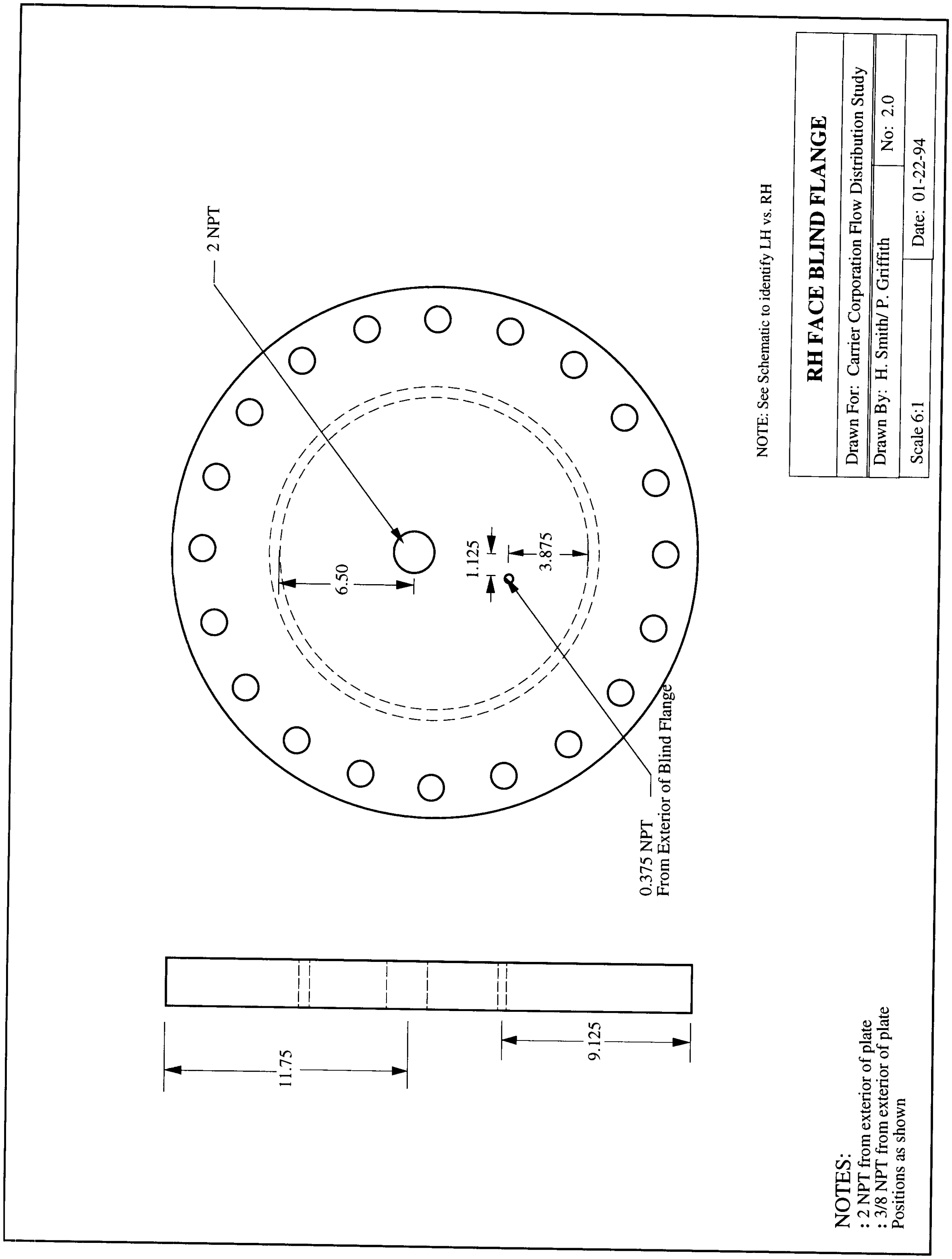


Figure B-3: RH Face Flange Assembly

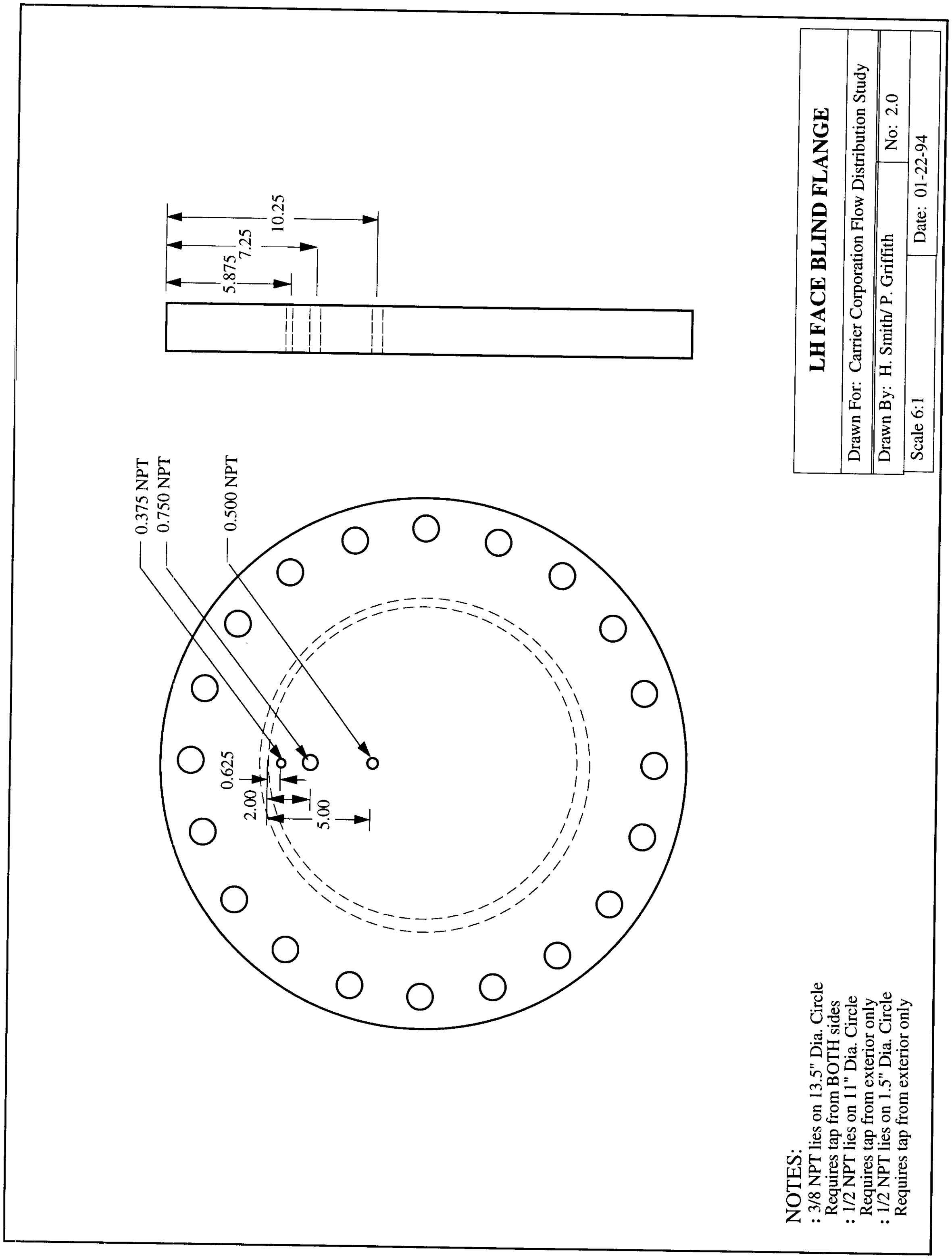


Figure B-4: LH Face Flange Assembly

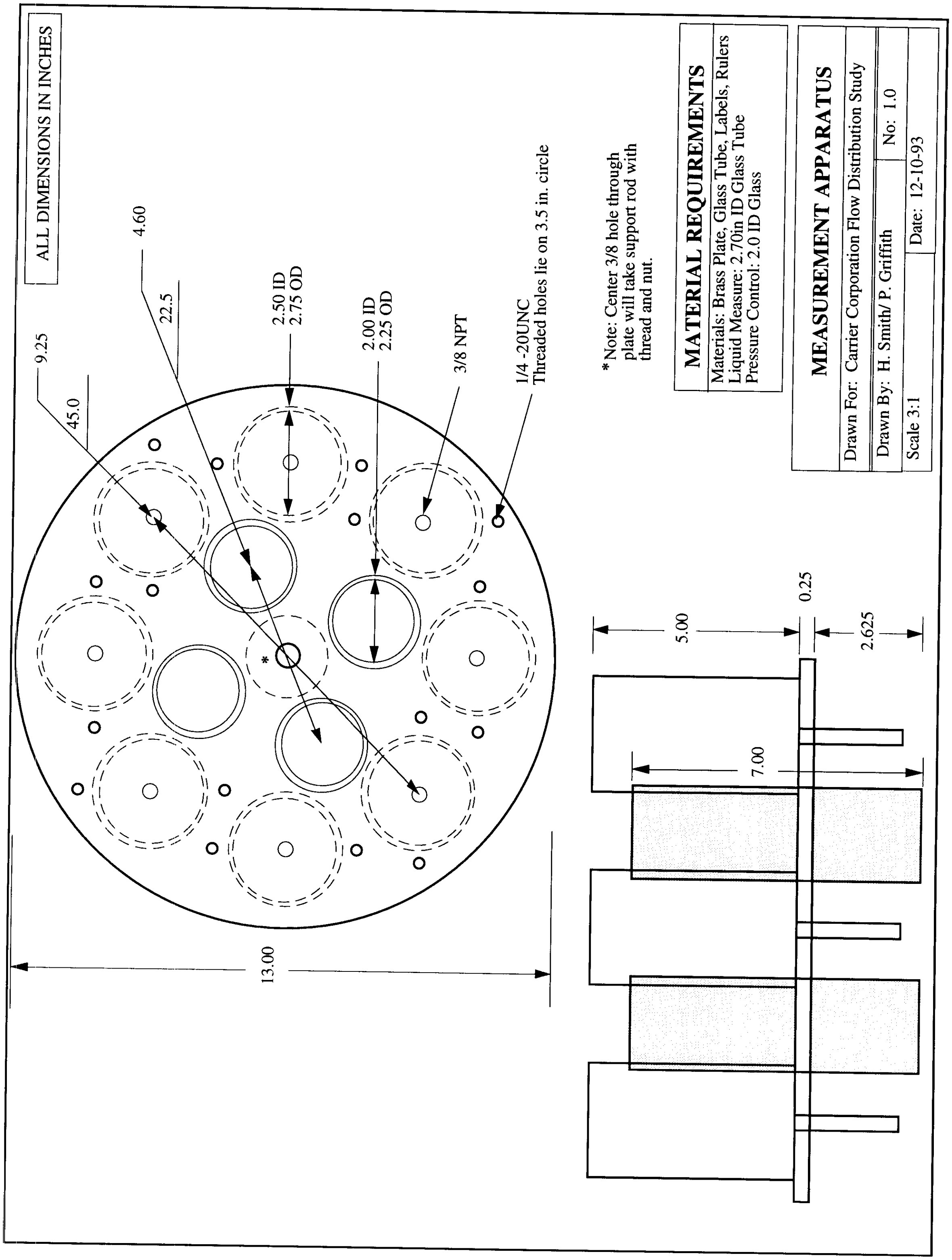


Figure B-5: Measurement Apparatus

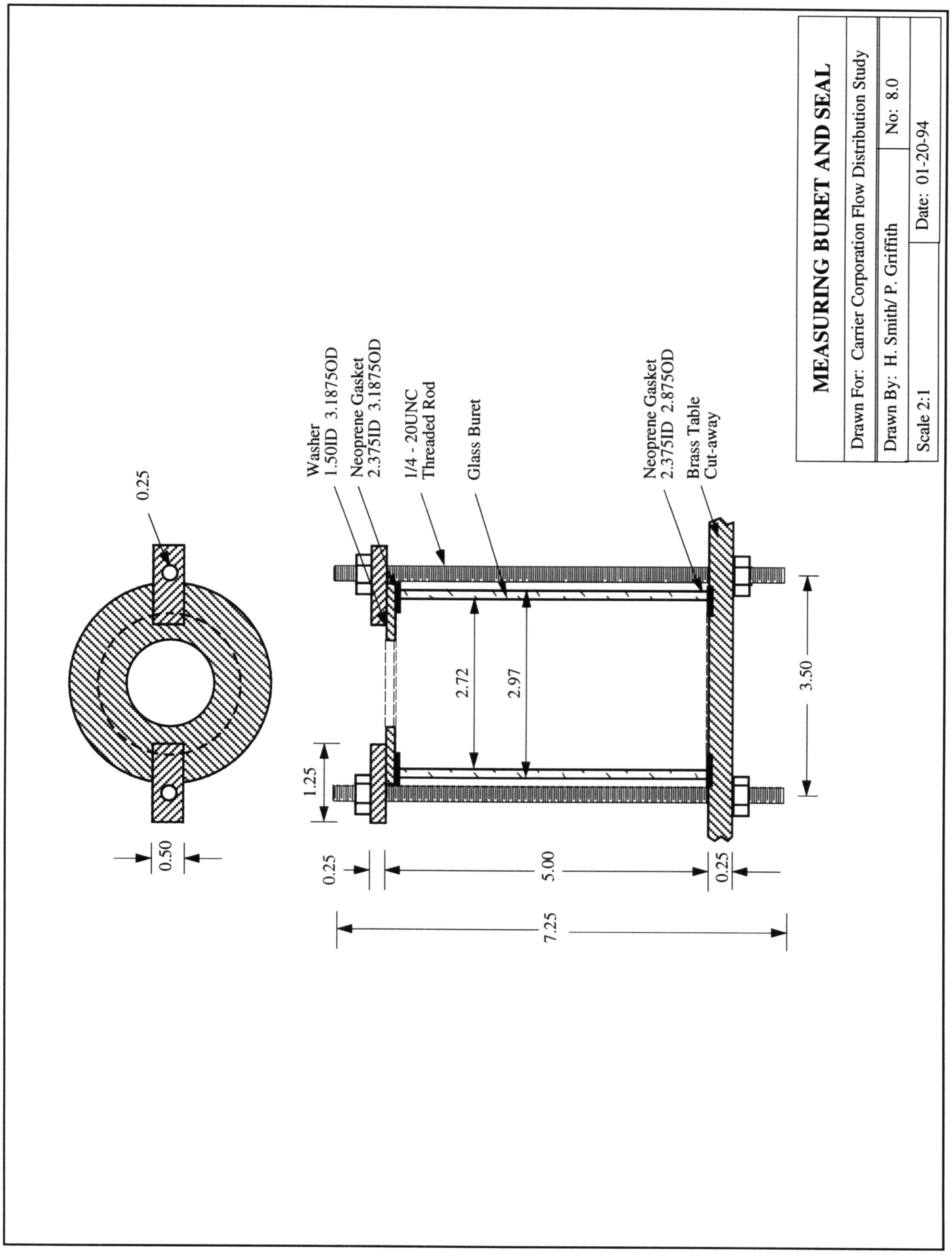


Figure B-6: Measurement Glass Assembly

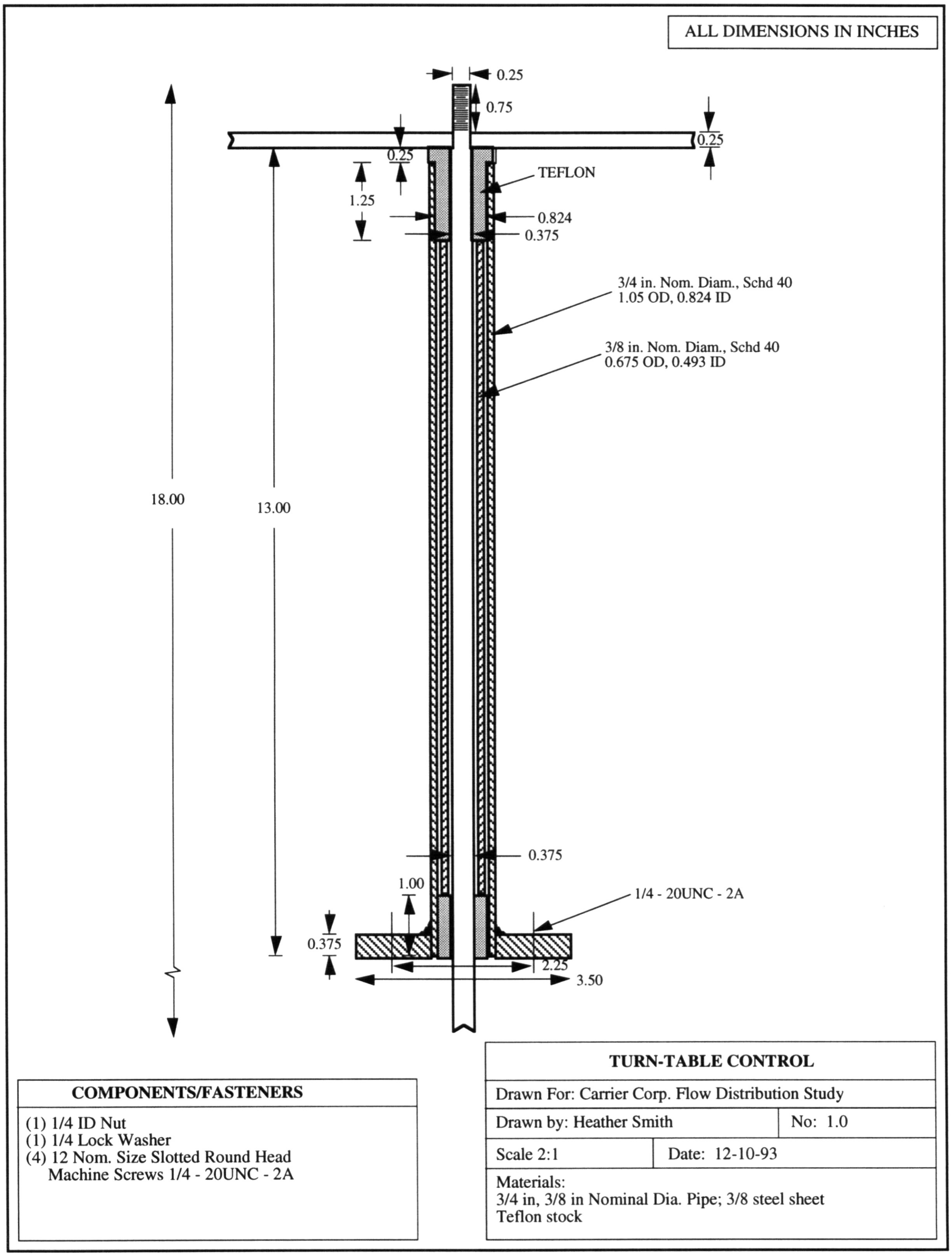


Figure B-7: Turn-Table Control

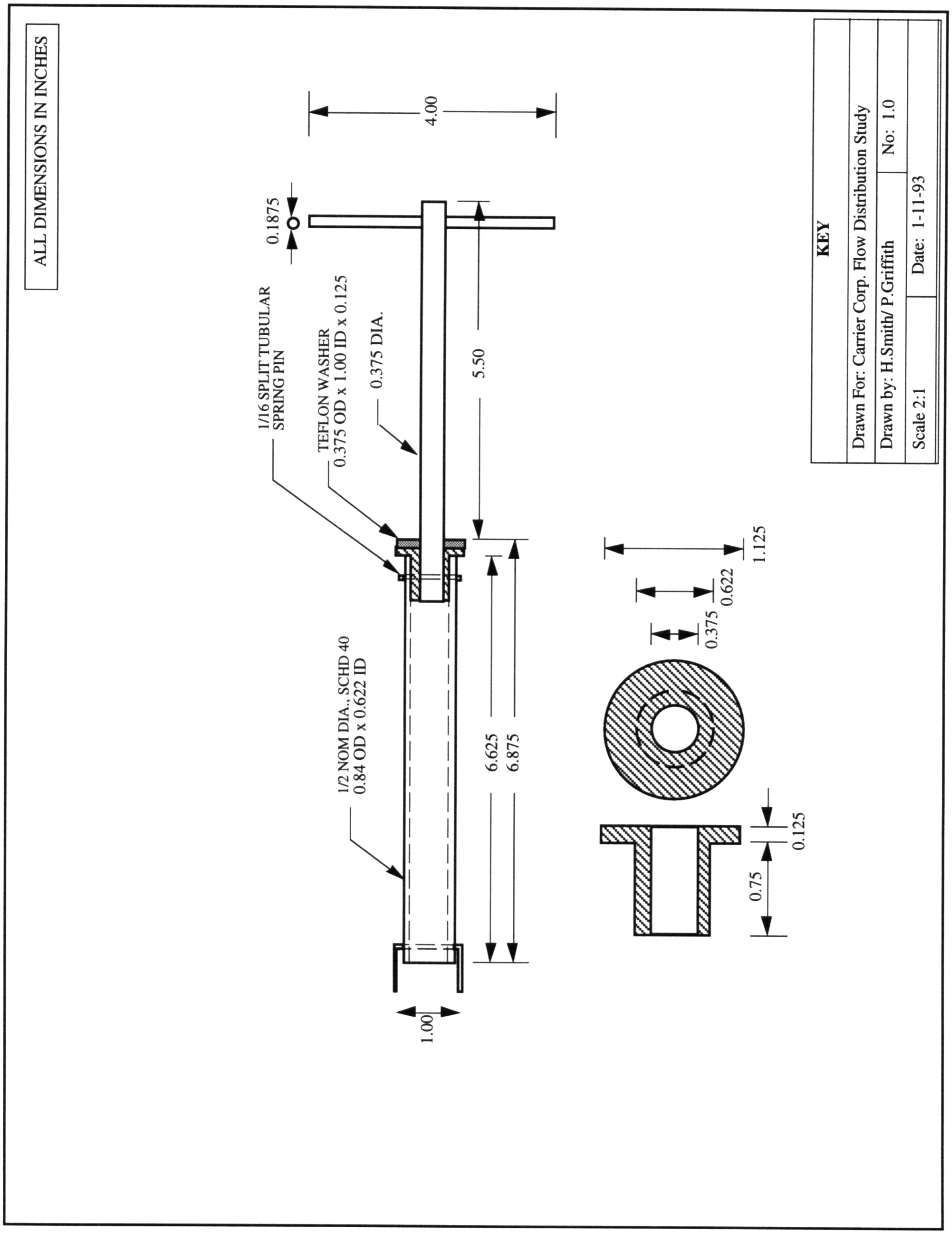


Figure B-8: Key

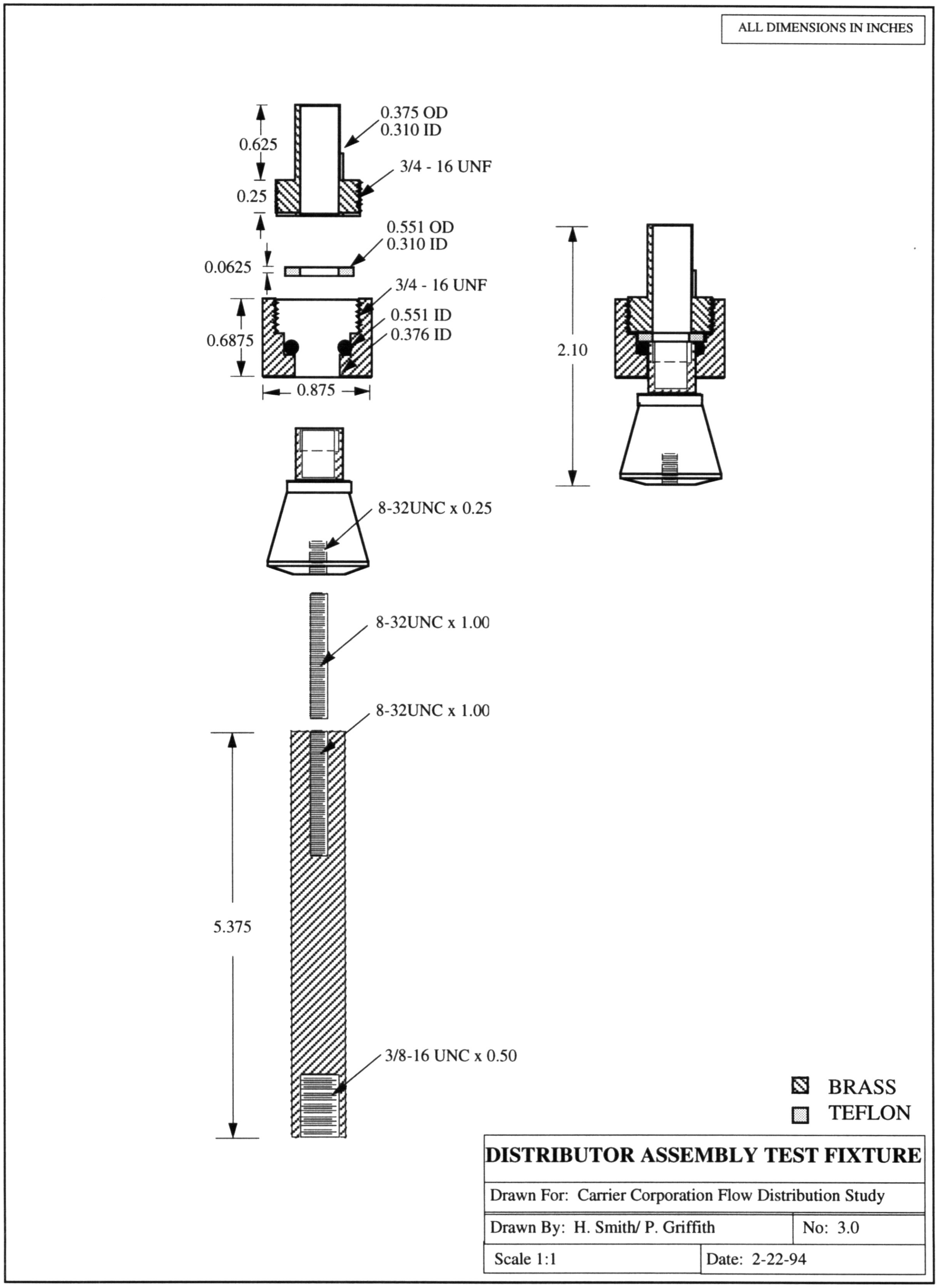
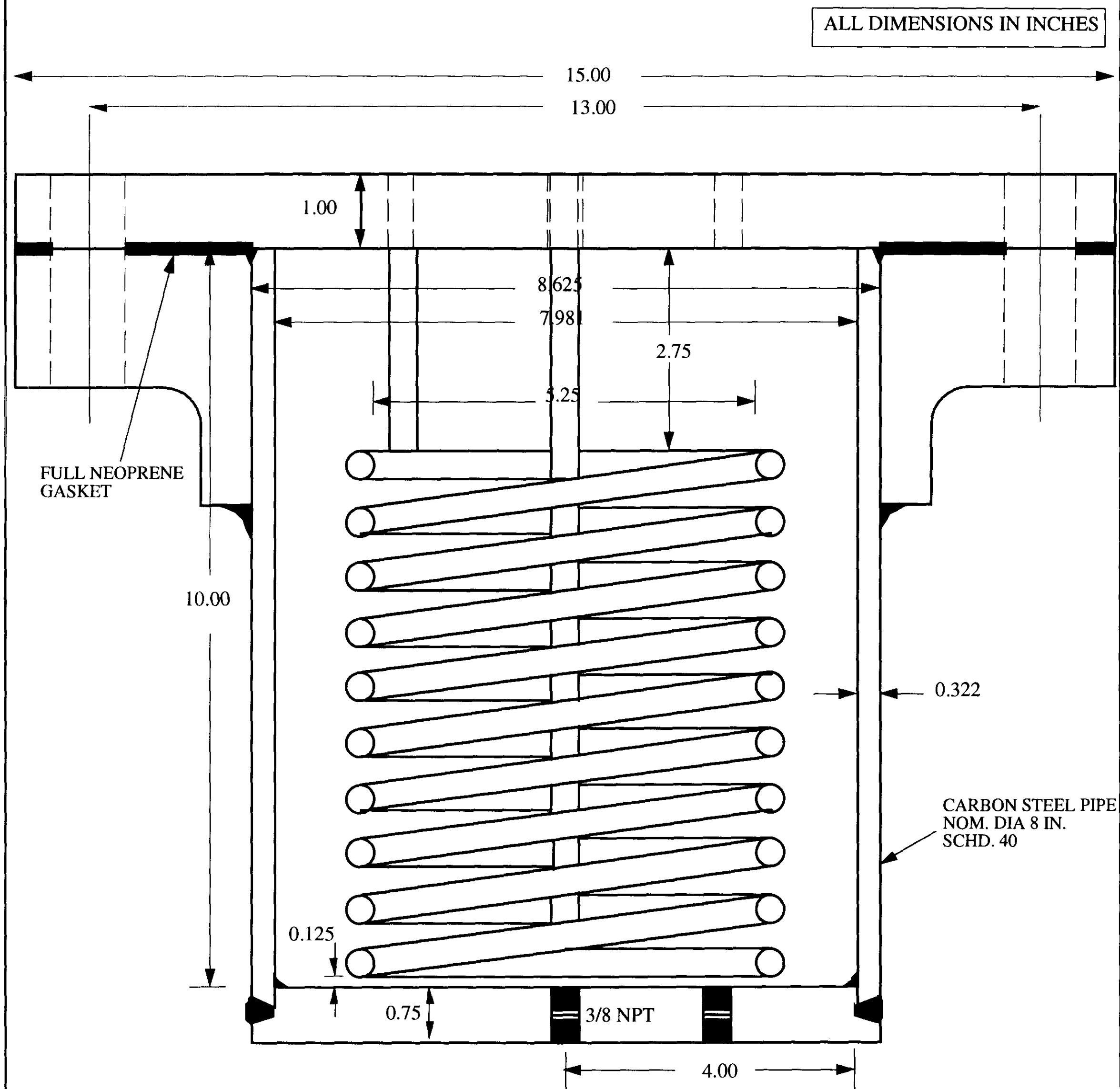


Figure B-9: Distributor Assembly Test Fixture



COMPONENTS/FASTENERS

(4) 3/8 in. through Couplings
(2) 1/2 in. through Couplings
(1) 1/2 in. 2-Way Ball Valve
(3) 3/8 in. 2-Way Ball Valve

REFRIGERANT RESERVOIR	
Drawn For: Carrier Corp. Flow Distribution Study	
Drawn by: H.Smith/P.Griffith	No: 4.0
Scale 2:1	Date: 11-04-93

Materials:
Body- 8.0 in Nominal Diam. SA-53 Grade B
Coil- 3/8 OD Comm. Pure Cu
Exterior- Insulation Tape
Flange- 8.0 in Nom. Pipe Size
Welds - Fillet, Dble V-groove
Gasket- Neoprene
Bolts - (12) 1.0 Diam.

Figure B-10: Refrigerant Reservoir

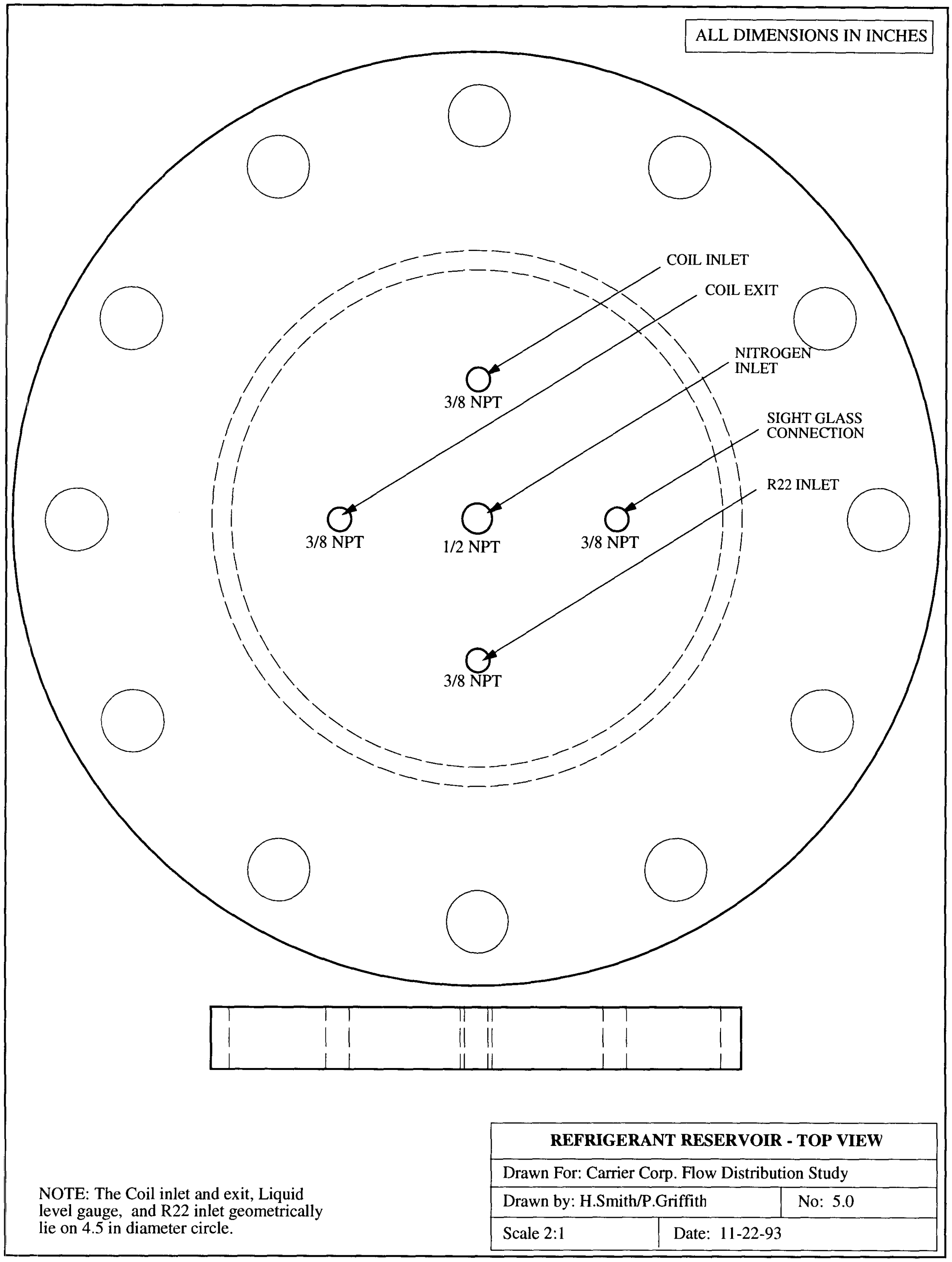
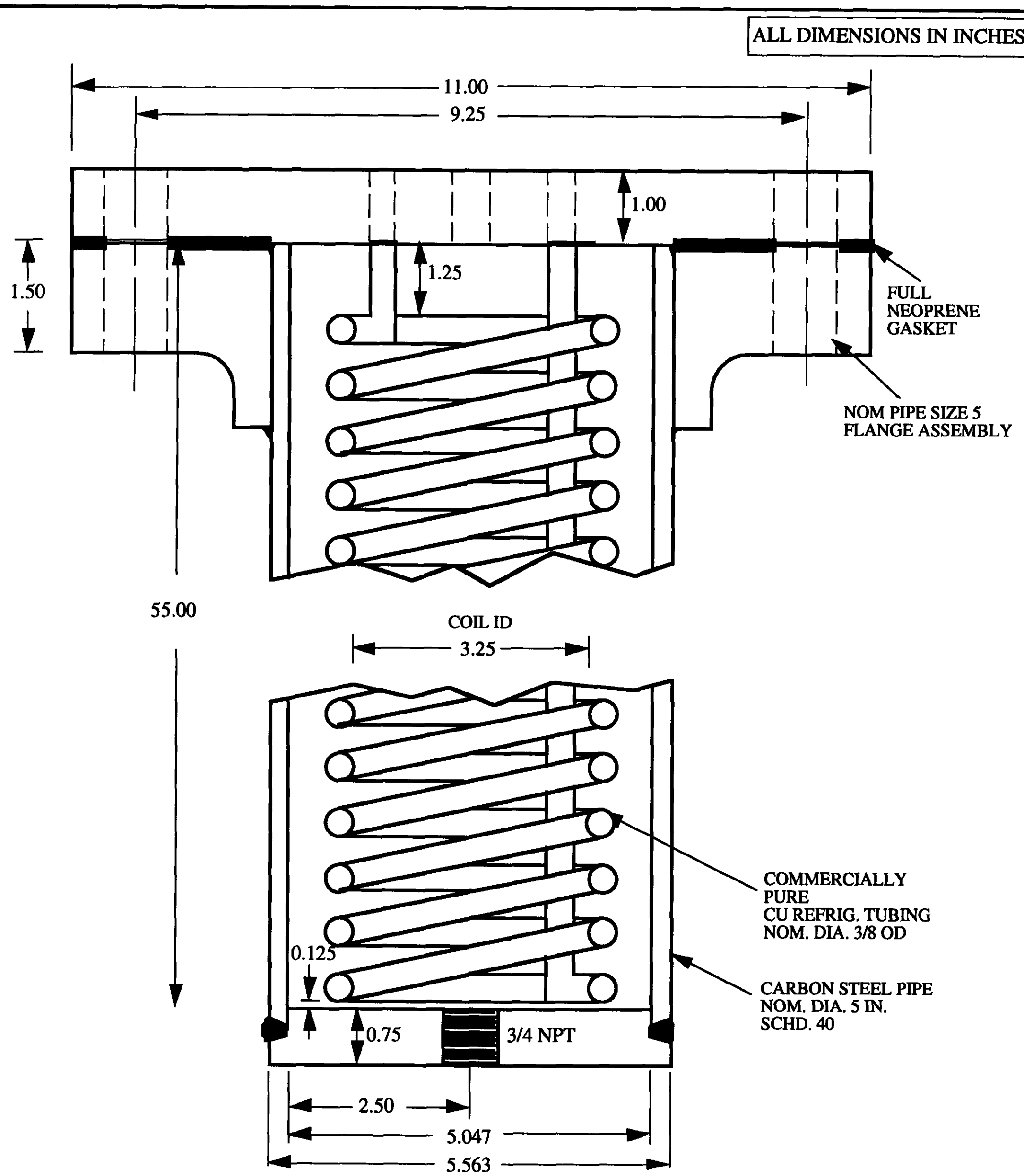


Figure B-11: Refrigerant Reservoir - Top View



COMPONENTS/FASTENERS
(2) Pressure Release Valve: 1/2 NPT conn (NUPRO B-8CPA2-NE-TS-150) Nitrogen Feed and Bleed (4) 3/8 in. through Couplings (2) 3/4 in. through Couplings (2) 1 in. 2-Way Ball Valves (1) 3/4 in. 2-Way Ball Valve (2) 3/8 in. 2-Way Ball Valve

CONDENSER	
Drawn For: Carrier Corp. Flow Distribution Study	
Drawn by: Heather Smith	No: 1.0
Scale 2:1	Date: 10-23-93
Materials: Body- 5.0 in Nominal Diam. Coil- 3/8 OD Comm. Pure Cu Exterior- Insulation Tape	Flange- 5.0 in Nom. Pipe Size Gasket- Neoprene Bolts - (8) 7/8 Diam.

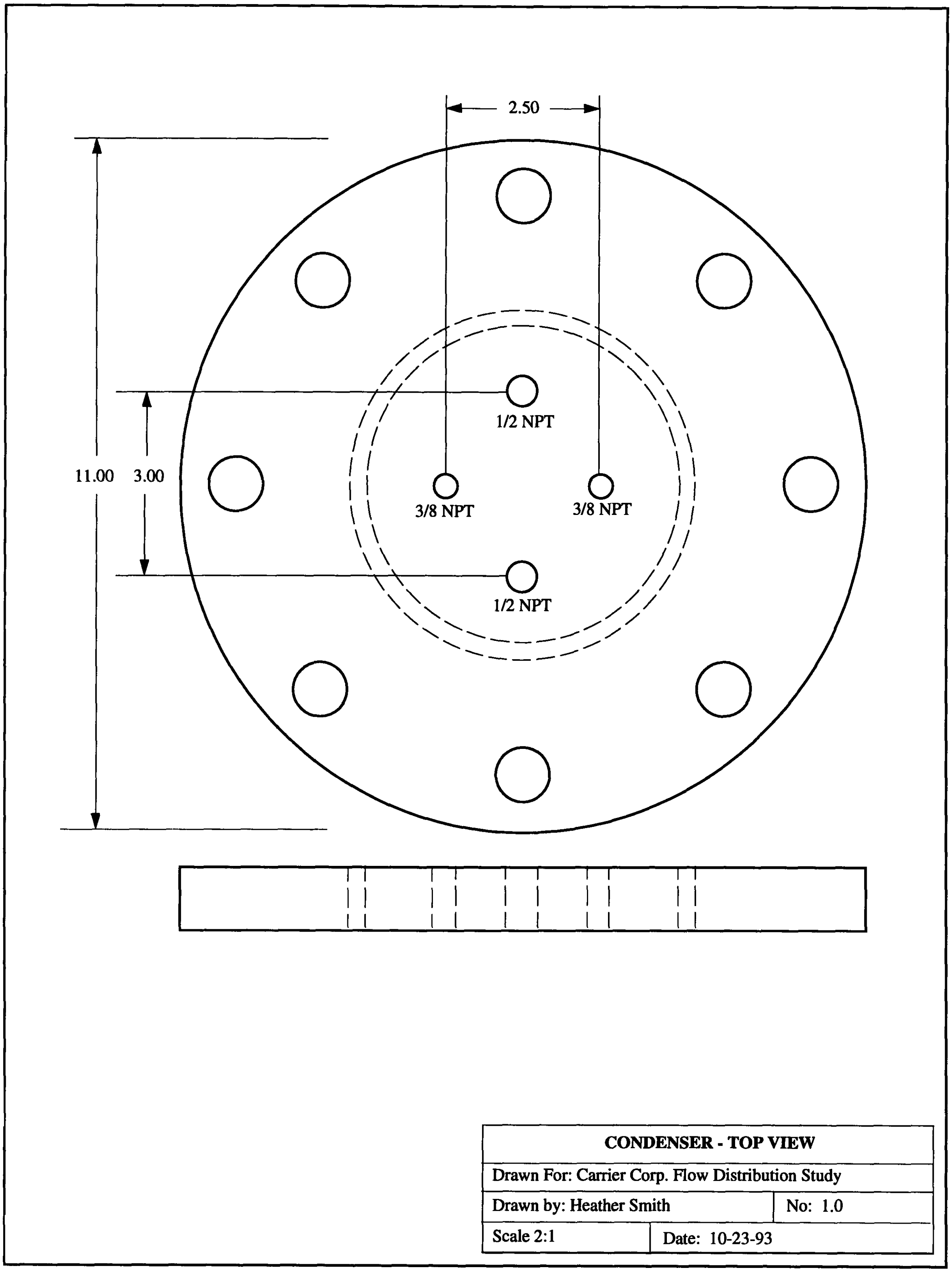


Figure B-13: Condenser - Top View

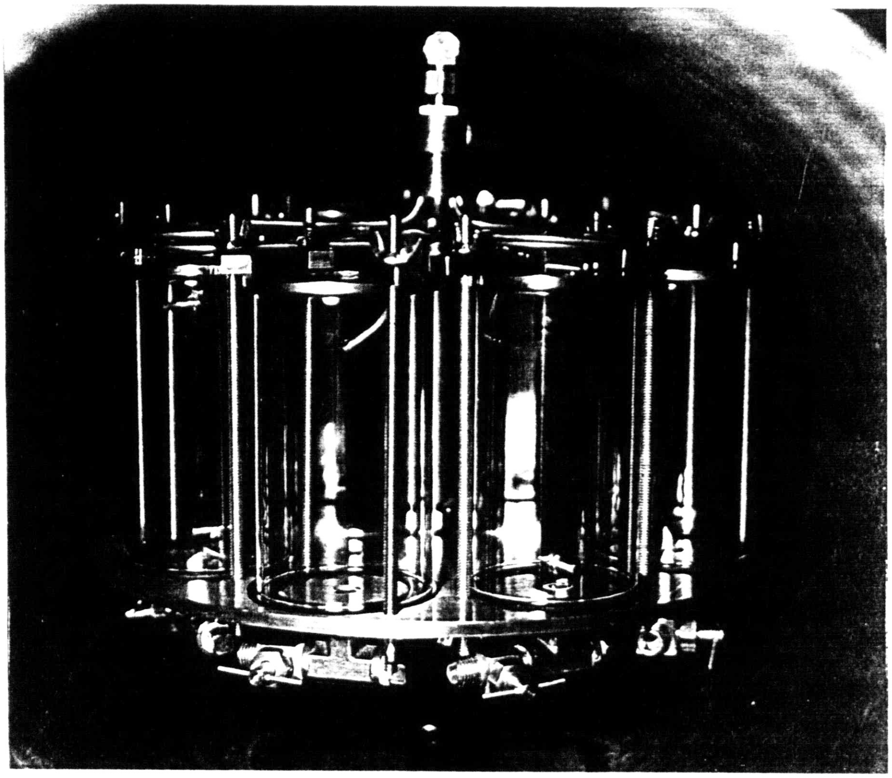


Figure B-14: Measurement Glass Assembly

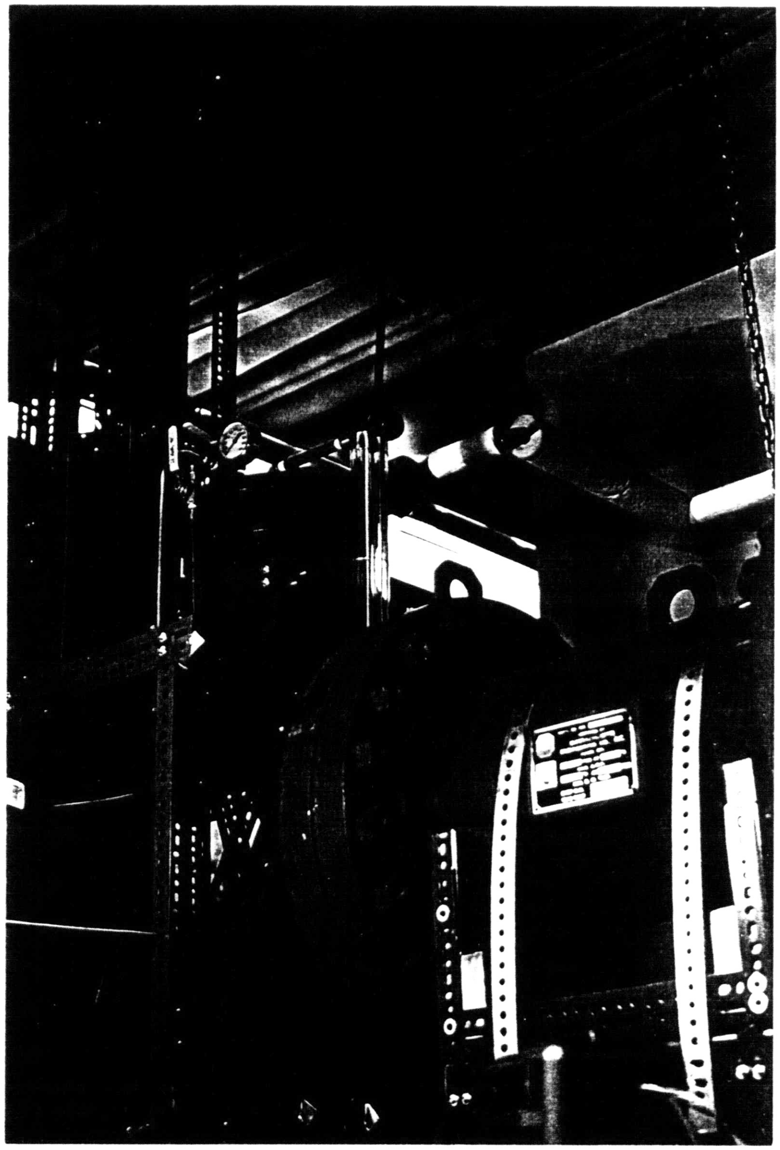


Figure B-15: Test Apparatus - (Left to Right) Condenser, Reservoir, Main Vessel

Appendix C: Main Vessel Certification

FORM U-1A MANUFACTURER'S DATA REPORT FOR PRESSURE VESSELS — PAGE 1 OF 1

(Alternative Form for Single Chamber, Completely Shop-Fabricated Vessels Only)

As Required by the Provisions of the ASME Code Rules, Section VIII, Division 1

1. Manufactured and certified by MASSACHUSETTS ENGINEERING CO., INC., 40 MURPHY DRIVE, AVON INDUSTRIAL PAR AVON, MA 02322 (Name and address of manufacturer)

2. Manufactured for MASSACHUSETTS INSTITUTE OF TECHNOLOGY, CAMBRIDGE, MA 02139-4307 (Name and address of purchaser)

3. Location of installation MASSACHUSETTS INSTITUTE OF TECH., BLDG. 3, 60 VASSER ST., CAMBRIDGE, MA 0213 (Name and address)

4. Type HORIZ. (Horiz. or vert., tank) C 1985 (Mfg's serial No.) — (CRN) A931123 REV. 3 (Drawing No.) 5854 (Nat'l. Bd. No.) 1994 (Year built)

5. The chemical and physical properties of all parts meet the requirements of material specifications of the ASME BOILER AND PRESSURE VESSEL CODE. The design, construction, and workmanship conform to ASME Rules, Section VIII, Division 1 1992 (Year) to A92 (Addenda (Date)) — (Code Case Nos.) — (Special service per UG-120(d))

6. Shell: SA106 - B (Matl. (Spec. No., Grade)) .500 (Nom. Thk. (in.)) — (Corr. Allow. (in.)) 1' - 4" OD (Diam. I.D. (ft. & in.)) 2' - 3" (Length (overall) (ft. & in.))

7. Seams: SEAMLESS (Long. (Welded, Dbl., Sngl., Lap, Butt)) — (R.T. (Spot or Full)) 85 (Eff. (%)) — (H.T. Temp. (°F)) — (Time (hr)) DBL FILLET LAP (Girth (Welded, Dbl., Sngl., Lap, Butt)) — (R.T. (Spot, Partial, or Full)) 1 (No. of Courses)

8. Heads: (a) Matl. SA105 (Spec. No., Grade) (b) Matl. SA516-70 (Spec. No., Grade)

	Location (Top, Bottom, Ends)	Minimum Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a)	END	2.25	—	—	—	—	—	—	1'-5 5/16	—
(b)	END	2.25	—	—	—	—	—	—	1'-5 5/16	—

If removable, bolts used (describe other fastenings) SA325 BOLTS SA563 NUTS (40) - 1.25" (Matl., Spec. No., Gr., Size, No.)

9. MAWP 315 psi at max. temp. 450 °F

* Min. design metal temp. 0 °F at 315 psi. Hydro., pneu., or comb. test pressure 473 psi.

10. Nozzles, inspection and safety valve openings: ELSEWHERE IN SYSTEM**

Purpose (Inlet, Outlet, Drain)	No.	Diam. or Size	Type	Matl.	Nom. Thk.	Reinforcement Matl.	How Attached	Location
—	1	6"	FLG	SA106-B/SA105	.280/300#	—	WELDED	SHELL

11. Supports: Skirt NO (Yes or no) Lugs 0 (No.) Legs 0 (No.) Other — (Describe) Attached — (Where and how)

12. Remarks: Manufacturer's Partial Data Reports properly identified and signed by Commissioned Inspectors have been furnished for the following items of the report: — (Name of part, item number, Mfg's name and identifying stamp)

*IMPACT TESTING NOT REQ'D/UG 20(F) 20 GALLON HORIZONTAL VESSEL OAL - 2' - 8 1/2"

NO OF LIFTING LUGS ON TANK - 3 - TANK HYDROSTATICALLY TESTED IN HORIZONTAL POSITION.

**USER MUST PROVIDE OVERPRESSURE PROTECTION BEFORE TESTING OR USING VESSEL

CERTIFICATE OF SHOP COMPLIANCE

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Pressure Vessels, Section VIII, Division 1. "U" Certificate of Authorization No. 162 expires 28 AUG, 1996.

Date 12 JAN 94 Co. name MASSACHUSETTS ENGR. CO. INC. (Manufacturer) Signed [signature] (Representative)

CERTIFICATE OF SHOP INSPECTION

Vessel constructed by MASSACHUSETTS ENGR. CO. INC. at AVON, MA 02322.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State or Province of MASSACHUSETTS and employed by ARKWRIGHT MUTUAL INS. CO., NORWOOD, MA *** have inspected the component described in this Manufacturer's Data Report on 1/12, 1994, and state that, to the best of my knowledge and belief, the Manufacturer has constructed this pressure vessel in accordance with ASME Code, Section VIII, Division 1. By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this Manufacturer's Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection. *** FACTORY MUTUAL ENGINEERING ASSOC.

Date 2/2/94 Signed [signature] (Authorized Inspector) Commissions NB 5321 MA 1480 "A" (Nat'l. Board (incl. endorsements), State, Prov. and No.)

(12/87) This form (E00117) may be obtained from the ASME Order Dept., 22 Law Drive, Box 2300, Fairfield, NJ 07007-2300.

REPRINT 8/90 (2300)

Appendix D: Experimental Procedure

The following encompasses the preparation and procedural requirements to charge the system with refrigerant and run a test. Note that charging of the reservoir with refrigerant is required only once at the onset of several tests. The system is closed-loop to ensure environmental safety as well as consecutive use of the same refrigerant volume. Note further that disassembly of the cups is required (and therefore assembly) only if a seal fails or fouling of the glass occurs.

The apparatus design included efforts to maintain the environment within the main vessel for realistic simulation of flow thermodynamic properties at evaporator inlet. Prior to actual blow-down the environment is achieved by bleeding vapor from the reservoir at ambient conditions. During the test, additional vapor is produced by flashing liquid through the throttle, distributor, and capillary assembly. The condensing heat exchanger was included in the design to condense excess vapor which would otherwise raise the pressure in the vessel. The main vessel though maintained at the proper pressure will have initial ambient temperature conditions (roughly the exterior temperature), i.e. greater than saturation. As liquid is injected for test the temperature will move down towards saturation but there will still be an opportunity for evaporation to occur (there is a transient in maintaining proper pressure). It is important, therefore, to expedite measurements immediately after the test is completed.

The procedure described below assumes that the total R22 volume required to prepare environments and run one test is charged into the reservoir at onset. The vapor required to prepare 'evaporator' environment in the main vessel is then bled from the reservoir.

As in any experiment it is important here to read the procedure and understand the interaction of apparatus components prior to running any tests. Because the active fluid is a freon mixture (and under pressure) safety precautions must be rigorously maintained. The use of safety glasses and gloves is recommended.

D.1 Preliminary Set-up

Charging and tests should proceed only if the system has been air tested to 100 psig over at least two hours (overnight is preferred). Reference the plot of Figure D.1-1 for example results. A drop in pressure of roughly 10 psia over two hours may be attributed to adjustment of interior relative to exterior environment. Excessive pressure drop signifies leakage and must be investigated. Soap and water (e.g. Snoop®) should be used liberally on all seals and fittings to test for leaks. Calibration curves for vessel pressure gauges are shown in Figure D.1-2.

After blind flanges are in place and a sealed system has been assured, observe glass cups to assure operation of light fixture and "turn-table" control. Nut on key fitting must be tight to assure seal.

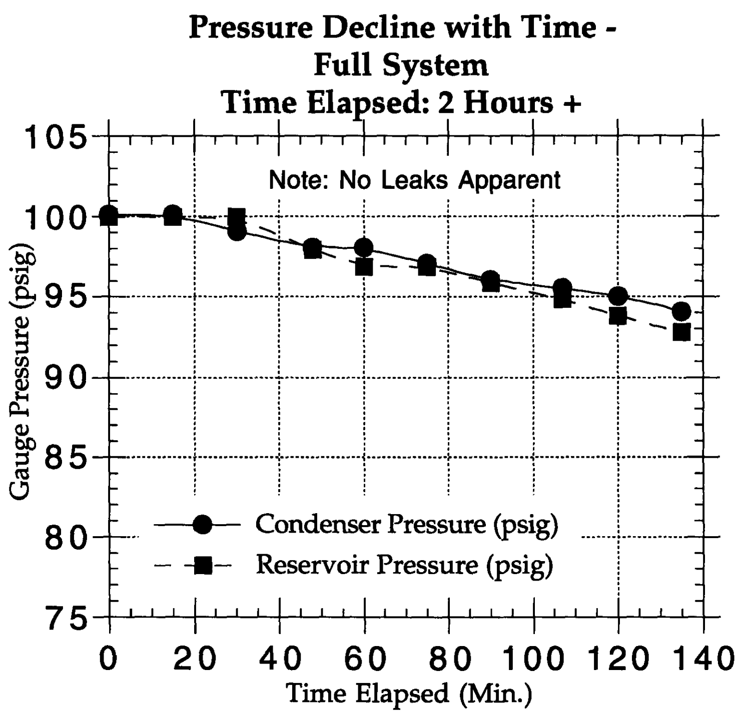


Figure D.1-1: Pre-Test Air Pressurization Results

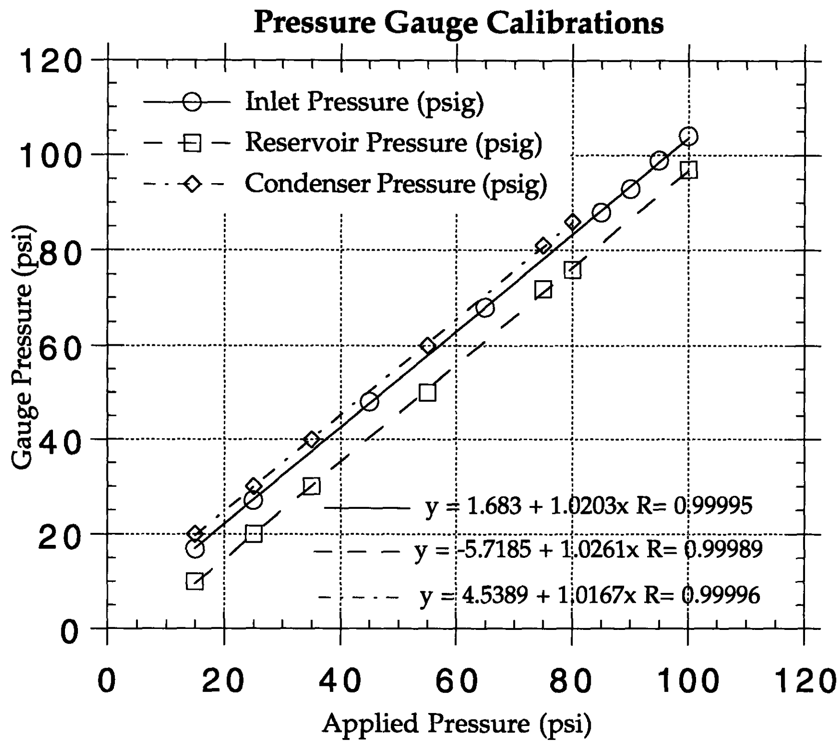


Figure D.1-2: Pressure Gauge Calibration Curves

D.2 Charging the System

Throughout this process make certain the valve connecting reservoir base to main vessel (the blow-down line) is CLOSED. Initially, all other valves should also be closed.

1. Obtain a scale with greater than 50 lbf capacity and place it beneath the R22 tank. (Note that the tank should be valve side UP. R22 vapor will then be the charging medium. Condensation will occur in the reservoir as noted below.) Monitor the mass of the tank continuously through the proceeding steps. Note when changes take place and why.

2. If a two-valve brass manifold is available, connect the center hose to the R22 tank flared connection. Make certain both manifold valves are closed then connect the larger scale gauge hose to the apparatus inlet at the base of the main vessel. (If a

manifold is not available secure a refrigerant compatible hose with 1/4 in. female flared connections. A 200 psi gauge is also necessary.)

3. Turn on cooling water for reservoir coil and achieve 42°F (may take as long as thirty minutes). Cooling water temperature can be taken with a thermometer at the drain opening.

4. Open valve on supply tank and observe pressure. It should be the saturation pressure at ambient conditions (i.e. the tank contains both liquid and vapor).

5. The system needs to be purged of air. Open valves at base of main vessel connecting reservoir and main vessel to apparatus inlet ("inlet valve"). Open valve connecting supply tank to apparatus approximately 20°. Blow in sufficient R22 vapor to fill system and then close the inlet valve. After a brief transient period, release air from condenser vent.

6. Close valves at base of main vessel connecting reservoir and main vessel to inlet.

7. Record current mass of supply tank.

With the system purged of air (won't be completely purged of course) and the cooling line cycling cold water through the reservoir, the system is ready to be charged. A pressure gradient must be achieved between the supply tank (high) and the reservoir (low) to feed sufficient R-22 vapor into the apparatus.

8. Record pressure of supply tank. Open inlet valve. Slowly open valve connecting reservoir to inlet. Vapor from the supply tank will move with the pressure gradient until equilibrium is achieved. Conditions in the supply tank will drop to meet that of the reservoir within roughly 5 minutes. Approximately 2 lbf will have been transferred. Bubbling liquid will be visible in reservoir sight glass.

To transfer the remaining refrigerant it is necessary to provide an additional temperature control. Submerging the supply tank in a warm water bath prevents the temperature from dropping to that of the reservoir and thereby maintains a pressure difference between the supply tank and pressure vessel.

9. Prepare a water bath of sufficient size to partially submerge the supply tank. Monitor the tank pressure periodically throughout this process. Use either water boiled separately (an electric kettle or hot plate will suffice) or a cartridge heater

partially submerged in the bath (the former is more expedient) to achieve a gradient of 40 psi or more. (e.g. Reservoir: 80 psig, Tank: 120 psig.) Monitor the reservoir r refrigerant level continuously. After sufficient amount is transferred, close the inlet valve.

10. Remove supply tank from the bath. Wearing hand protection, relieve pressure in lines connecting tank to apparatus. Close valve on tank.

D.3 Running Tests

D.3.1. Pre-Test Preparation

11. Fill condenser cooling supply bucket with water to approx. 16 in from base. There should be 8 - 10 inches of water below the ice layer at all times to assure smooth pump suction. Note that pump performance is improved with increased inlet head.

12. Close the bleed line valve on condenser cooling line. Assured that both the inlet and the bypass valves are completely open, turn on the 1 hp pump. Open bleed valve 50%. Monitor mass flow rate until 0.2+ lbm/sec flow is achieved. Monitor ice supply and restock as required.

13. Monitor both reservoir and condenser cooling lines until desired temperatures are a achieved (e.g. Reservoir: 42°F, Condenser: 32°F).

D.3.2 Setting Up Environment

With the condenser cooling activated, any refrigerant vapor bled into the vessel/condenser system will condense and the remaining air will rise to the highest point where it can be bled off. Further, excess vapor bled into the system will begin condensed as the 90 psia environment is attained. Note that if there is too much air in the system, the refrigerant can't make it into the condenser and the condensing process will cease.

Sufficient vapor must be bled from the reservoir into the base of the vessel to achieve the desired evaporator conditions. A pressure gradient is required to induce flow.

14. With a steady temperature in the reservoir cooling line, open the steam valve approximately 1/6 of a turn. After the condensate has been pushed out of the steam line, a knocking sound will ensue as the steam and cold water meet in the mixing tee.

15. Reaching the desired 90°F (the drain water/steam mix temperature) temperature must be done slowly allowing each temperature change to thoroughly mix into the R22. Note that different degrees of subcooling in the reservoir environment will require different water/steam mix temperatures. Continue opening the steam line valve 1/6-1/4 turn every 15 minutes. Monitor the water/steam drain temperature throughout.

16. After 90°F has been attained (reservoir pressure should be approximately 175 psig) open valve at base of vessel connecting reservoir to main vessel valve. Slowly open (to approximately 10 degrees) the main vessel valve while monitoring reservoir liquid level. (Doing it slowly protects measurement equipment inside the vessel from sudden pressure changes and large vapor flow rates.)

17. Bleed one inch of vapor (liquid level will go down) or less in increments until sufficient liquid remains in reservoir for test plus a safety factor (i.e. to protect main vessel environment from the high reservoir pressure allow for 0.5 in. or more to remain in the reservoir after blow-down).

18. Monitor temperature/pressure of all vessels. Bleed off of air will be required from ball valve on condenser shell as system pressure goes up.

19. Be prepared to supplement condenser/vessel pressure with condenser N_2 tank if the need arises. (It should not be necessary.)

20. At this juncture, the desired (evaporator) pressure has been attained in the main vessel and the fluid remaining in the reservoir is well mixed at the desired subcooled temperature. Just prior to blow-down, supplement reservoir pressure with N2 gas until the 'condenser' pressure is attained (240 psia).

D.3.3 Test

21. Assure conditions of both reservoir and vessel are as desired (e.g. 240 psia, 95°F and 90 psig - respectively).

22. Blow down for 1 minute and close valve. Monitor level in reservoir as calculations may be incorrect. Do not allow reservoir to run "dry".

23. Turn on light source and, using table control, take depth readings of fluid in cups.

24. Use the key to drain measurement glasses. Note that the O-ring seal for the key will have to be broken in order to maneuver the key. Caution must be taken to assure minimal loss of refrigerant vapor. (Note that evaporation of all liquid in the cups can be achieved within approximately 3 hours as the main vessel returns to ambient conditions.) With the glasses empty, tighten the seal.

25. Close the valve connecting N_2 tank to reservoir. Turn off steam line to reservoir coil and allow temperature/pressure in reservoir to drop below that of evaporator conditions (reservoir cooling line is still 'on').

26. Carefully open drain valves (reservoir valve then vessel valve) at base of main vessel and allow R22 liquid/vapor to flow back to reservoir for storage. The reservoir may be maintained at ambient conditions after excess N_2 has been bled.

Appendix E: Proposed Design - Flashing Turbine Distributor

In a typical vapor-compression refrigeration cycle saturated or slightly sub-cooled liquid from the condenser is passed through a throttle where the fluid pressure drops to that of the evaporator. Flashing of some liquid in this expansion process results in a mixture of perhaps 20% quality. To maximize the heat transfer performance of the evaporator, thereby achieving the saturated or slightly superheated vapor required at inlet to the compressor, a uniform distribution of the two-phase mixture (flow rate and quality) through each circuit in the evaporator is required. A turbine and distributor assembly was proposed to satisfy the expansion and distribution requirements for refrigerant into the evaporator as well as to utilize the work available from the expansion process. The patent paperwork was initiated (M.I.T. Technology Disclosure) in December 1993.

The proposed design utilizes a nozzle and turbine rotor combination to achieve the expansion and distribution of two-phase flow into the evaporator (Figure E-1). Expansion of the flow for a typical system of 3-ton capacity results in an enthalpy reduction which, via the turbine rotor blades, could provide approximately 150W of power. One possible application of this concept would be to use this turbine to power a fan for the purpose of circulating air through an evaporator (Figure E-2).

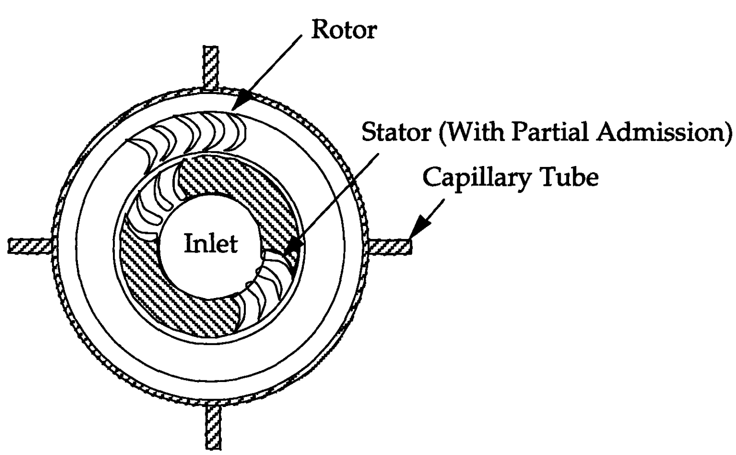


Figure E-1: Single-Stage Turbine Example (Radial Flow Impulse Turbine)

It is proposed that distribution of the refrigerant flow can be achieved effectively if the turbine stage is assembled inside a circular, horizontal plenum from which flow tubes to the evaporator originate at equal elevation (Figure E-3). The flow is thus divided from a single inlet path to the required number of exit paths for distribution throughout the evaporator. The turbine blading must be designed such that flow separation does not occur and entrainment of the flashing vapor in the liquid flow is maintained.

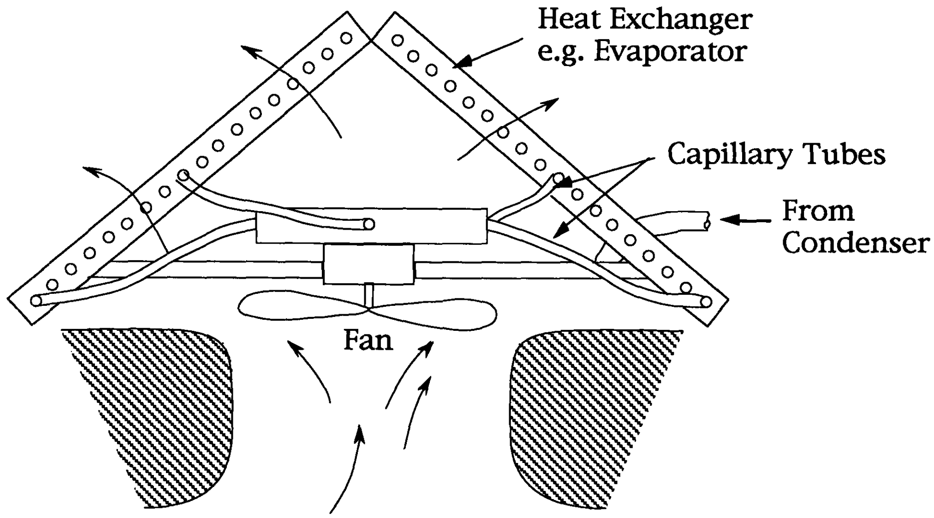


Figure E-2: Turbine Powered Fan Assembly

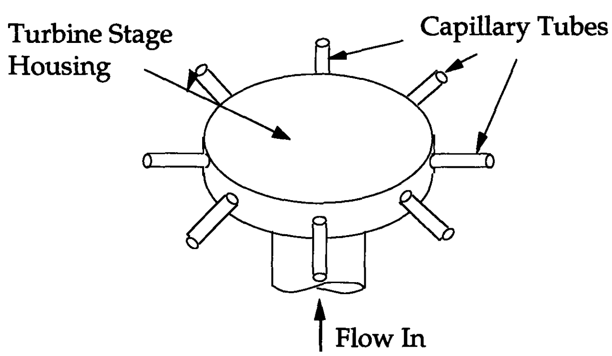


Figure E-3: Horizontal Plenum

The proposed design outlined briefly here utilizes the work produced from expansion of refrigerant fluid while distributing flow. The circular arrangement of tubes has the additional benefit of distributing flow independently of the flow rate or quality. Utilizing the expansion work to power a fan for a standard evaporator results in a unique coupling between power demand for the overall unit and fan speed (the power to the fan and the flow of refrigerant being expanded are coupled). Further, a partial recovery of the unit efficiency lost from waste of expansion work can be achieved through the proposed system.