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# HEAT EXCHANGER CHARACTERISTICS REQUIRED FOR A FLUID-COOLED VENTURI

A Thesis
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## ABSTRACT

Heat exchanger characteristics are defined and a heat exchanger was designed which will provide an adequate coolant inlet temperature to the cooling passage of a venturi (a convergent-divergent nozzle). The venturi is used to meter air flowing at high temperatures. The heat flux to the venturi must be controlled to prevent component failure at the maximum air flow temperature. A three-dimensional finite-element computer code is utilized to verify safe temperature distribution throughout the venturi and to provide the cooling requirements for the heat exchanger. Based on the cooling requirements, a five-pass counterflow heat exchanger was designed that incorporated a finned tube heat transfer surface.

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## NOMENCLATURE

| | |
|---|---|
| $A$ | Area, $ft^2$ |
| $A_c$ | Cross-sectional area, $ft^2$ |
| $B$ | Fin thickness, inches |
| Btu | British thermal unit |
| $C$ | Fluid heat capacity, Btu/sec-°F |
| $C_D$ | Venturi discharge coefficient |
| $C_p$ | Fluid specific heat, Btu/lbm-°F |
| $D$ | Diameter, ft |
| $D_h$ | Hydraulic diameter of the cooling passage, ft |
| $D_i$ | Inside diameter of an annulus, ft |
| $D_o$ | Outside diameter of an annulus, ft |
| $D_v$ | Local diameter of the venturi, ft |
| $d$ | Total derivative |
| $E$ | Energy flow across a boundary, Btu/sec |
| $E_{gen}$ | Energy generated in a control volume, Btu/sec |
| $E_{sto}$ | Energy stored in a control volume, Btu/sec |
| $E_{surr}$ | Energy in the surroundings, Btu |
| $E_{sys}$ | Energy in a system, Btu |
| $F$ | Friction factor |
| °F | Degree Fahrenheit |
| ft | Foot |
| $g_c$ | Gravitational constant equal to 32.2 ft-lbm/lbf-$sec^2$ |
| $H$ | Fin height, inches |
| $h$ | Convection coefficient, Btu/hr-$ft^2$-°F |

| | |
|---|---|
| $h_D$ | Convection coefficient based on diameter, Btu/hr-ft$^2$-°F |
| $h_{D,h}$ | Convection coefficient based on hydraulic diameter, Btu/hr-ft$^2$-°F |
| hr | Hour |
| K | Thermal conductivity, Btu/hr-ft-°F |
| °K | Degree Kelvin |
| $K_{eq}$ | Equivalent thermal conductivity, Btu/hr-ft-°F |
| L | Length, ft |
| lbm | pound mass |
| M | Mach number |
| m | Mass flow rate, lbm/sec |
| N | Number of fins |
| NTU | Number of transfer units |
| $n_{fin}$ | Fin efficiency, percent |
| $n_s$ | Percent of total area in thermal contact |
| P | Static pressure |
| Pr | Prandtl number |
| psi | Pounds force per square inch |
| psia | Pounds force per square inch absolute |
| q | Heat flow rate, Btu/sec |
| R | Gas constant, ft-lbf/lbm-°R |
| $R_{con}$ | Contact resistance, hr-ft$^2$-°F/Btu |
| $Re_D$ | Reynolds number based on diameter |
| $Re_{D,h}$ | Reynolds number based on hydraulic diameter |
| $R_g$ | Fouling factor, hr-ft$^2$-°F/Btu |
| r | Cylindrical coordinate |

| | |
|---|---|
| $r_m$ | Mean radius of contact spots, inches |
| sec | Second |
| T | Static temperature, °F |
| $T_{bulk}$ | Bulk temperature of the coolant, °F |
| $T_{wall}$ | Wall temperature, °F |
| t | Time, sec |
| $U_{tot}$ | Overall heat transfer coefficient, Btu/hr-ft$^2$-°F |
| V | Velocity, ft/sec |
| w | Width of elements representing contact resistance, ft |
| x | Rectangular coordinate |
| z | Cylindrical coordinate |
| $\alpha$ | Mean thickness of the fluid layer, inches |
| $\gamma$ | Ratio of specific heats for air |
| $\Delta$ | Designates change |
| $\delta$ | Boundary layer thickness at a plant wall, ft |
| $\varepsilon$ | Heat exchanger effectiveness |
| $\mu$ | Dynamic viscosity, centipoise |
| $\pi$ | 3.1714 |
| $\rho$ | Fluid density, lbm/ft$^3$ |
| $\phi$ | Cylindrical coordinate |
| $\partial$ | Partial derivative |

Subscripts

| | |
|---|---|
| a | Designates conditions of air |
| atm | Measurement of pressure in atmospheres |
| base | Designates properties at the base of the fin |
| bulk | Designates bulk temperature of the fluid |

| | |
|---|---|
| c | Designates properties of the cold fluid |
| cr | Designates critical conditions |
| e | Designates conditions of ethylene glycol |
| f | Designates fluid |
| fin | Designates properties of the fin |
| h | Designates properties of the hot fluid |
| i | Summation index |
| id | Designates properties at inner surface of the heat exchanger tube |
| in | Designates conditions at the system boundary inlet |
| j | Designates ratio |
| max | Designates maximum value |
| min | Designates minimum value |
| od | Designates properties at the outer surface of the heat exchanger tube |
| out | Designates conditions at the system boundary outlet |
| r | Designates radial coordinate |
| s | Designates conditions at the surface |
| t | Designates total conditions |
| v | Designates conditions for the venturi |
| z | Designates coordinate |
| $\infty$ | Designates freestream properties |

Superscripts

| | |
|---|---|
| * | Designates throat conditions |
| $\overline{\phantom{x}}$ | Designates average value across flow at given station |
| $'''$ | Designates per unit volume |

# CHAPTER I

## INTRODUCTION

Today, aerodynamic systems are built to operate in high temperature environments. Many examples of these systems exist in the ground test facilities required for flight simulation of high speed aircraft engines and rocket motors. The performance of these systems is often restricted by the temperature limitations of the components subjected to high heat fluxes. To ensure the satisfactory operation of the system over the complete test range of temperatures and pressures, each component must be able to withstand the operating conditions.

The critical component is often the one which is in contact with a hot gas. There are several methods to maintain these components below a maximum safe temperature: (1) shield the component by introducing a film of cool gas (or liquid, if applicable) between the part and the hot gas, (2) cover the surface with a material that absorbs heat flux by ablation, and (3) remove the heat from the component by passing a fluid through passages within the component.

A component which is common to ground test facilities (as well as many industrial systems) is a venturi. A venturi is an axisymmetric, convergent-divergent nozzle, required to accurately measure fluid flow past a fixed plane in a pipe or duct. Therefore, venturis are often in contact with hot fluid. Several advantages of the venturi as a flow measurement device are: (1) high accuracy of measurement, (2) a low pressure loss across the venturi, which is significant in pressure limited systems, and (3) reduced exit velocity from the venturi which makes it possible to re-establish uniform, low velocity flow conditions only a short distance downstream of the venturi [1][1].

A venturi flowmeter and its cooling system are the subject of this thesis. A venturi, built in two parts and joined at the throat, meters air flowing at a high total temperature and total pressure. The venturi is fluid-cooled at its highest heat flux location, the throat, with a rectangular cross-section coolant flow passage encircling the throat. The critical temperature of the venturi is the maximum operating temperature of the O-ring seals at the sealing face of the joined flanges of the venturi, which is about 600°F. The coolant leaving the venturi passage carries away the heat from the throat region. The specified coolant and its flow rate are designed to absorb a sufficient quantity of heat flow to allow the venturi to operate safely (i.e. protect the O-ring seals from excessive heat) at high airflow total temperature and total pressure.

A heat exchanger is incorporated in the cooling circuit to provide for energy exchange from the hot coolant stream (fluid leaving the venturi), to a secondary cold stream. The heat exchanger must be designed to cool the venturi coolant back to a suitably low temperature at the inlet of the venturi cooling passage. Unless the heat exchanger is adequately designed, the heat transfer from the venturi to the coolant will not be sufficient to protect the O-ring seals at high temperatures because the primary coolant stream temperature will continue to increase during operation.

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[1]Numbers in brackets refer to similarly numbered references in the Bibliography.

## Statement of the Problem

Considering specified airflow conditions, the maximum heat flux delivered to the given venturi will be calculated. A coolant, called the primary coolant, will be selected which will provide a sufficient heat capacity to transport heat from the venturi. Heat exchanger characteristics will be defined as required to maintain the primary coolant inlet temperature at an appropriate value for collecting heat from the venturi. The heat exchanger characteristics will be determined by its required heat transfer surface area, which in turn defines the type of heat exchanger needed. The success of the heat exchanger design characteristics will be demonstrated by verifying that the maximum temperature distribution in the venturi flange is below the critical temperature for the O-ring seals. The verification will be performed by applying a finite-element heat conduction program to predict isotherms in the venturi and around the O-ring seals during worst case operation.

# CHAPTER II

## A SYSTEM OVERVIEW

### System Details

The system is composed of three main components: (1) the venturi, (2) the heat exchanger, and (3) the primary coolant fluid pump (Figure 1)[2]. The venturi dimensions are fixed and are specified in Figures 2 and 3. The heat exchanger required will be justified on the basis of heat transfer surface area, which is interpreted to define the type of heat exchanger.

The venturi is constructed in two parts which are fastened together at the throat (Figure 3). The flanges are required for proper mounting of the venturi to a supporting wall. Airflow is forced through the venturi and only a small amount is considered to stagnate at the wall supporting the venturi. Two O-ring seals are located at the plane where fastening occurs (Figure 3). These seals, made of silicon rubber, provide an airtight joint for the venturi flanges.

Very close to the O-ring seals, located at the venturi throat, a rectangular cooling passage provides for thermal energy transfer from the venturi to the coolant (Figure 3). The cooling passage encircles the venturi throat and the cooling fluid enters and exits at the bottom of the venturi.

The venturi is made of Type 310 stainless steel throughout. The O-ring seals have a maximum operating temperature of 600°F [2].

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[2] Figures and tables are found in Appendix A.

The primary coolant system pump is assigned a pressure supply head of 25 psi. The primary coolant mass flow rate is assigned a value of 1 lbm/sec (therefore, the velocity is 4.3 ft/sec in the rectangular venturi cooling passage). By inspection, the system pressure losses are small due to the low flow rates.

## The Approach

The maximum heat load delivered to the coolant will be calculated based on the maximum operating conditions expected. A heat exchanger will be defined to provide an adequate steady state heat transfer rate from the hot stream (primary coolant flow from the venturi) to the cold stream (secondary coolant flow in the heat exchanger) to prevent O-ring failure. The maximum temperature at the O-ring seals will be verified by finite-element heat transfer analysis to be below 600°F for the highest heat load.

# CHAPTER III

## THE VENTURI OPERATING ENVIRONMENT

Operating conditions which the venturi is exposed to are defined by the following airflow characteristics: (1) total temperature, (2) total pressure, and (3) mass flow rate.

The venturi is exposed to airflow which ranges from total temperatures of -100 to 650°F and total pressures of 14 to 150 psia. The total pressure downstream of the venturi is low enough to support a Mach number equal to one of the venturi throat; thus ensuring choked operation.

The mass flow rate of air through the venturi can be calculated for steady flow conditions at the venturi throat using the continuity of mass equation [3]:

$$m_a = 2\pi \int_0^{r^*} \rho V \; r \; dr \Big|_{throat} = C_D \, \overline{\rho_a} \, A_{c,v} \, \overline{V_a} \tag{3-1}$$

where $m_a$ = mass flow rate

$\overline{\rho_a}$ = fluid density

$A_{c,v}$ = cross-sectional area

$\overline{V_a}$ = fluid velocity

$C_D$ = discharge coefficient for the venturi

Generally,

$$C_D = \text{function } (Re_{D*}, \text{ venturi geometry}).$$

This approach has been validated by Smith and Matz who have pioneered the use of venturi meters for accurate mass flow metering [1]. As defined by Obert and Young [4], a stream is in steady flow if all its variables are independent of time (i.e. $\rho$, A, and V are time independent). The continuity of mass equation can be put in terms of the so-called mass-averaged stagnation conditions of the flow upstream of the venturi inlet;

$T_t$ = total temperature

$P_t$ = total pressure

by assuming that air is an ideal gas undergoing adiabatic, isentropic flow expansion to the throat conditions. This assumption is substantiated by investigating the compressibility factor of air over the operating range previously defined. This procedure is illustrated in Appendix B.

Assuming a Mach number equal to unity across the venturi throat and using the adiabatic temperature and isentropic pressure relations, the continuity equation for an ideal gas is: (see Appendix C for a detailed derivation).

$$m_a = C_D \sqrt{\frac{\gamma\, g_c}{R_a}} \left(\frac{\gamma+1}{2}\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}} \frac{P_t\, A_{c,v}}{\sqrt{T_t}} \tag{3-2}$$

The maximum system operating conditions ($T_t$ = 650°F and $P_t$ = 150 psia) represent the worst-case design conditions for the heat exchanger (i.e. define the maximum heat load). If the heat exchanger characteristics provide a heat flux at the O-ring seals below a critical level at the maximum system operating conditions, then the system will function over the entire operating range.

Therefore, the system operating conditions considered for this analysis are:

1. $T_t$ = 650°F
2. $P_t$ = 150 psia
3. $m_a$ = 482 lbm/sec.

To prevent O-ring failure, the maximum allowable temperature is set 450°F; this provides a margin of safety over the maximum allowable temperature of 600°F. The margin of safety will allow for a variation in the actual and calculated maximum heat flows to the venturi. Although a conservative approach will be taken to calculate the maximum heat load on the venturi, the margin of safety, commonly used in good engineering practice, allows for any variations.

The heat exchanger will be used during cold airflow conditions to prevent primary coolant freeze up. During hot airflow conditions, the cooling circuit should be used subsequent to airflow shutdown to prevent the primary coolant from possible boiling as it stands in the venturi.

# CHAPTER IV

## PRIMARY COOLANT SPECIFICATIONS

The primary coolant for the system is defined using the following criteria:

1. The coolant must remain liquid at temperatures down to -30°F (for cold airflow conditions).
2. The coolant must not vaporize and therefore shall have a high boiling point.
3. The coolant shall be inexpensive and readily available.

A portion of the cooling system will be exposed to the ambient outside environment throughout the year. A coolant having a freezing point below -30°F should not freeze during winter at the site where this venturi is operated.

The maximum coolant temperature will be a function of the heat transferred from the air through the venturi to the coolant. This value will be calculated once the heat load is determined.

After reading the literature on coolants, and discussions with other engineers, an automotive type "antifreeze" consisting of 50 percent by weight aqueous solution of ethylene glycol was chosen to satisfy the specifications.

As shown in Figure 4, a 50 percent by weight aqueous solution of ethylene glycol has a freezing point of -32°F, and as shown in Figure 5, it has a boiling point of 265°F at one atmosphere pressure (or 315°F at the pump supply pressure of 25 psi) [5]. Other properties of ethylene glycol (specific heat, thermal conductivity, and dynamic viscosity) are shown in Appendix D.

Meeting the third criteria, ethylene glycol is readily available and comparatively inexpensive, and it is reasonably safe to handle, store, and discard.

CHAPTER V

# HEAT LOAD DETERMINATION

## Physical Origins of Heat Transfer

From the First Law of Thermodynamics,

$$\Delta E\}_{surr} + \Delta E\}_{sys} = 0. \tag{5-1}$$

Flowing internal energy is conserved in the absence of heat sources or sinks. The Second Law of Thermodynamics states that heat will travel from one body to another only when the bodies are at different temperatures and the heat will flow from higher to lower temperatures [6]. The First Law is the basis for most heat transfer analyses, although the Second Law often is used to evaluate the efficiency of such processes.

The Laws of Thermodynamics relate the states of a thermodynamic system, while the heat transfer theory allows the engineer to determine rates of thermal energy changes.

The word heat is often not used in a strict thermodynamic sense [4]. From the thermodynamic point of view, an energy flow is identified as heat only if (1) the energy transfer occurs across a system boundary due to a temperature difference, and (2) it is not transported by the mass flow. However, it is conventional in the subject of heat transfer to denote all energy flows that arise because of a temperature difference as "heat transfers".

Heat transfer occurs by three methods: (1) conduction, (2) convection, and (3) radiation. This analysis will consider only heat transfer by convection and conduction, and radiation will be neglected.

Conduction may be viewed as the transfer of thermal energy from the more energetic to the less energetic particles of a substance due to the collisional interactions between the particles of a substance [7]. Heat transfer processes can be quantified in terms of appropriate rate equations. For heat conduction, the appropriate rate equation is Fourier's Law:

$$q_x = -K(\partial T/\partial x) \tag{5-2}$$

where the minus sign is introduced so $q_x$ is positive in the direction of increasing x (and decreasing T) [8], (see Figure 6).

Convection heat transfer occurs in flowing fluids, and is comprised of two mechanisms:

1. Energy transferred from one particle to another particle by conduction, locally.
2. Energy transferred by the gas or liquid bulk motion.

For fluid in contact with a solid wall, the temperature gradient is confined to a narrow layer called the thermal boundary layer, close to the wall. The innermost part of the thermal boundary layer is called the laminar film regime, or film for short (Figure 7). In this thin film the largest temperature variations occur. Within the film, heat transfer takes place by conduction [9]. The rate equation for this mode of heat transfer is as follows:

$$q_x = K/\delta \; A_s \; (T_\infty - T_{wall}) \tag{5-3}$$

where

$$h = K/\delta \tag{5-4}$$

and h is called the convection (or film) coefficient. Unlike thermal conductivity, the convection coefficient, h, is not a property of the solid or fluid alone but is dependent on geometry, surface finish of the solid, velocity and transport properties of the fluid. Determination of the convection coefficient is a major portion of all convective heat transfer rate problems. Most heat transfer convection coefficients are determined experimentally, but have mathematical forms that result from simple boundary layer theory.

## Approach to Calculations

The heat load calculated is that heat flux delivered from the airflow through the venturi to the primary coolant. The temperature of the venturi flanges should be maintained below a value critical to the O-ring seals. The heat exchanger will transfer the heat flow from the hot stream (primary coolant) to a cold stream (secondary coolant) thus maintaining the venturi at a constant temperature below the critical operating limit for the O-ring seals.

The heat flux delivered to the primary coolant is based on the following preliminary assumptions:

1. A maximum service temperature of 450°F (as opposed to 600°F) for the O-ring seals will be used. This allows a 150°F margin of safety for the O-ring failure.
2. The maximum possible heat energy transferred to the venturi will be based on the maximum total temperature (650°F) and total pressure (150 psia) of the airflow through the venturi. These conditions will be worst-case

conditions for the O-ring seals and will provide the maximum heat transferred from the venturi (source) to the heat exchanger (sink). If the system can withstand this heat load, then it can withstand operating at lower temperatures and pressures.

3. Primary coolant temperature at the inlet of the venturi cooling passage cannot exceed 80°F.
4. The primary coolant is 50 percent by weight aqueous ethylene glycol.

<u>Control Volume Analysis</u>

The system under consideration is as shown in Figure 1. It consists of a venturi which acts as the heat source, a heat exchanger which acts as a heat sink, a primary coolant which collects and transports the heat from the source to the sink, a pump to drive the coolant through the circuit and a storage reservoir for the secondary coolant in the heat exchanger.

A differential control volume analysis is necessary to illustrate the importance of the various boundary conditions applied to the venturi as well as to form a foundation for determination of the heat load. Also, the differential control volume analysis will illustrate the credibility of simplifying assumptions. Because of the complicated venturi geometry, however, the differential control volume analysis will not readily lead to a direct solution of temperature distribution in the venturi.

The representative differential control volume is as shown in Figure 8. An energy balance on the control volume yields:

$$E_{in} - E_{out} + E_{gen} = E_{sto} \tag{5-5}$$

Because the venturi is axisymmetric about its centerline, the heat conduction equation for the control volume [7] is described by:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(Kr\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial}{\partial \phi}\left(K\frac{\partial T}{\partial \phi}\right) + \frac{\partial}{\partial z}\left(K\frac{\partial T}{\partial z}\right)$$

$$+ q''' = C_p\,\rho\,\frac{\partial T}{\partial t} \tag{5-6}$$

Solution to this equation will define the temperature distribution as a function of the coordinates ($r$, $\phi$, $z$, $t$). The heat transfer is found in the appropriate direction as follows:

$$q_r = -Krd\phi dz\,\frac{\partial T}{\partial r} \tag{5-7}$$

$$q_z = -Krd\phi dr\,\frac{\partial T}{\partial z} \tag{5-8}$$

$$q_\phi = -Kdrdz\,\frac{\partial T}{r\partial \phi} \tag{5-9}$$

At this point, some logical simplifying assumptions can be made which, for this design problem, will not induce significant error in the solution. The assumptions are as follows:

1. The heat transfer rate will be found at steady state. For this analysis, the maximum heat transfer rate is being sought. This occurs at the maximum total operating conditions which will be held steady for a sufficient time to induce a steady state temperature distribution in the venturi.

2. No internal heat generation. Internal heat generation is due to electrical resistance, chemical or nuclear reactions, etc.
3. The venturi material is isotropic and homogeneous.
4. The venturi material has a constant thermal conductivity. This is justified since this analysis is concerned with the heat transferred at the maximum temperature only.

The heat conduction equation can now be simplified to the following:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 T}{\partial \phi^2} + \frac{\partial^2 T}{\partial z^2} = 0 \qquad (5\text{-}10)$$

This equation can be solved for the temperature distribution in the r, $\phi$, and z directions.

## Boundary Conditions

The heat conduction equation is a second-order differential equation in r, $\phi$, and z, and therefore needs six boundary conditions to be solved (i.e. two each in the r, $\phi$, and z directions). Since the equation is not time dependent (i.e. steady state), no initial condition is necessary.

In general, the types of boundary conditions possible are, as described by Apraci [8]:

1. Prescribed temperature.
2. Prescribed heat flux.

3. Insulated surface (no heat flux).
4. Heat transfer to the ambient by radiation.
5. Heat transfer to the ambient by convection.
6. Prescribed heat flux acting at a distance.
7. Interface of two continua of different thermal conductivities.
8. Interface of two continua in relative motion.
9. Moving interface of two continua (change of phase).

Considering a finite control volume as shown in Figure 9, the applicable boundary conditions were found by analysis of the venturi installation and the flow configurations:

1. Heat transfer to the venturi by forced convection on the inner surface of the venturi exposed to airflow.
2. Heat transfer from the venturi by forced convection on the cooling passage surface exposed to coolant flow.
3. Negligible free convection heat transfer occurs on the remaining outer surfaces.
4. A small amount of heat transfer occurs by conduction from the bulkhead onto which the venturi is mounted. By calculation, this heat transfer is of the order of 10 percent of the total heat transferred to the venturi by forced convection. Since this fell within the accuracy of the overall analysis, it was neglected.

In the heat conduction path between the venturi inner surface and the cooling passage, there exists a temperature drop at the interface of the adjoining flanges where imperfect contact exists. Real surfaces are not smooth but consist of microscopic peaks and valleys. Heat transfer through such an interface takes place by combined mechanisms of conduction across true contact points, conduction across entrapped interstitial fluid, and radiation across interstitial gaps [10]. Resulting overall heat conductance at the joint is a function of the materials in contact (thermal conductivity, surface finish, and hardness), the contact pressure, the mean temperature across the joint and the presence of oxide films. In the present study, this boundary condition will be represented by contact resistance between the two joining flanges on the venturi.

## Quantified Boundary Conditions for Finite-Element Analysis

The boundary conditions are calculated from empirical correlations which have been proven accurate from previous studies. Quantifying boundary conditions will include the following:

1. Determination of contact resistance to heat transfer at the interface of the adjoining flanges.
2. Determination of convection coefficients at both the inner venturi surface exposed to air and the cooling passage surface exposed to the coolant.

Ozisik and Hughes [11] have developed a method to estimate the contact resistance. The resistance is given by:

$$R_{con} = \frac{1}{(1-n_s)K_f/\alpha + 2n_s K/(\pi r_m)} \qquad (5\text{-}11)$$

where $n_s$ = percent of total area in contact

$K_f$ = apparent fluid conductivity

$\alpha$ = mean thickness of the fluid layer

$K$ = thermal conductivity of solid

$r_m$ = mean radius of contact spots

Appendix E shows that the contact resistance is calculated to be equal to $0.662 \times 10^{-3}$ hr-ft$^2$-°F/Btu.

To determine an appropriate empirical correlation for the convection coefficient on the air side of the venturi, the Reynolds number must be calculated. Turbulent flow in pipes is represented by Reynolds numbers greater than 2300 [7]. The Reynolds number is defined as:

$$Re_D = \frac{\rho V D}{\mu} = \frac{4m}{\pi \mu D} \qquad (5\text{-}12)$$

Using the flow rate as defined previously on page 8, and the properties of air [12] as listed in Appendix D, the Reynolds number is calculated as a function of venturi diameter (Figure 10) and is shown in Figure 11. The flow is turbulent.

Back, Massier, and Gier [13] have previously investigated the heat transfer in convergent-divergent nozzles. The investigation concluded that a pipe flow equation of the same form as that developed by Sieder and Tate [7] was within 5 percent of actual data for airflow at high total pressures (greater than 75 psia). The one-dimensional pipe flow equation provided results within the required accuracy desired for this study. Back, et al., defined the Reynolds number as a function of nozzle diameter, which is also used in the present study. Bartz [14] also developed simple Seider-Tate type formulas that were weighted by the ratio $\rho\mu/(\rho\mu)in$.

The convection coefficient on the air side is defined by the pipe flow equation developed by Sieder and Tate as:

$$h_D = 0.027\ (Re_D^{0.8})(Pr_a^{1/3})K_a/D_v\ (\mu_a/\mu_{s,a})^{0.14} \qquad (5\text{-}13)$$

where $Nu_D$ = local Nusselt number as a function of diameter

$Re_D$ = local Reynolds number as a function of diameter

$Pr_a$ = Prandtl number for air

$K_a$ = thermal conductivity of air

$D_v$ = local venturi diameter

$\mu_a$ = dynamic viscosity of free stream

$\mu_{s,a}$ = dynamic viscosity of fluid at the surface

This equation is valid for both constant surface temperature and constant heat flux, and for a large temperature range. Also it is assumed that the flow in the venturi is both hydrodynamically and thermally fully developed.

Using the pipe flow equation, the Nusselt number and the convection coefficient can be calculated as a function of diameter. The results are as shown in Figures 12 and 13.

The distribution of the convection coefficient along the venturi surface enables the heat flux to be calculated at each axial station, and thus completes the specification of the boundary condition for this inner surface.

The cooling passage is rectangular with a cross-sectional area equal to 0.5156 inches$^2$, an aspect ratio of 3.7, and a wetted perimeter of 3.5 inches (the entire perimeter is wetted).

To calculate the liquid-side convection coefficient, the average coolant bulk temperature (defined by $\{T_{in} + T_{out}\}/2$) should be used to determine fluid properties. However, the average temperature cannot be determined since the outlet temperature of the primary coolant is not known. Therefore, it is assumed that the bulk coolant temperature is 80°F throughout the cooling passage. The significance of this assumption will be evaluated once the average temperature can be calculated. The final answer can thus be determined by an iterative procedure.

The Reynolds number must be calculated to determine if the flow is laminar ($Re_{D,h} \leq 2300$) or turbulent ($Re_{D,H} > 2300$). For a rectangular tube, the Reynolds number is based on the hydraulic diameter as follows:

$$Re_{D,h} = m_e D_h/(A_c \mu_e) \qquad (5\text{-}14)$$

where $A_c$ = cross-sectional area of cooling passage

$D_h$ = hydraulic diameter of cooling passage

$m_e$ = assigned mass flow rate of coolant (1 lbm/sec)

Using the properties of 50 percent by weight aqueous ethylene glycol cooling fluid (as defined in Appendix D) and the specified flow rate (1.0 lbm/sec), the Reynolds number is equal to 6515. Therefore, the flow is turbulent [7]. The flow is assumed to be hydrodynamically fully developed at the entrance to the cooling passage since the rectangular shape of the cooling passage extends upstream in the tubing several feet from the venturi entrance. As shown by Kays [15], the thermal entry length is insignificant for fluids with Prandtl numbers above ten. For this study, the Prandtl number for the 50 percent by weight aqueous ethylene glycol solution is approximately twenty (varying slightly with temperature). Therefore, it is reasonable to assume a fully developed thermal flow throughout the venturi cooling passage.

Because the flow is turbulent, a circular tube prediction equation can be used with the hydraulic diameter as the characteristic dimension [16]. According to Incropera and Dewitt [7], for the Reynolds number of 6515, the following correlation will apply to turbulent flow in a rectangular tube:

$$h_{D,h} = 0.027\ (Re_{D,h}{}^{0.8})(Pr_e{}^{1/3})K_e/D_h(\mu_e/\mu_{s,e})^{0.14} \qquad (5\text{-}15)$$

The average coefficient is equal to 446 Btu/hr-ft$^2$-°F. This heat transfer coefficient represents that for the entire cooling passage surface. The coolant viscosity at the surface is assumed equal to the freestream viscosity. The value of surface viscosity, a function of temperature, can be estimated once the surface temperature is known. The significance of this assumption relative to the heat transfer coefficient will be evaluated once the temperature is known.

The entire outer surface of the venturi is assumed to be insulated (i.e. no heat transfer). This assumption is reasonable since the heat transfer due to free convection is insignificant compared to the heat transferred due to forced convection on the inner surface of the venturi.

The boundary conditions have been established in terms of convection coefficients. Now, the heat conduction equation can be solved, and the boundary conditions applied to the venturi to determine the temperature distribution in the venturi and its flanges.

## Solution to Heat Conduction Equation

An analytical solution to the heat conduction equation would be very complicated, time consuming, and probably inaccurate. For this analysis, an axisymmetric numerical solution will provide very adequate results. The method chosen for this analysis is a finite-element technique using a computer program developed by Swanson Analysis Systems, Inc. [17]. In general, the finite-element method is a discretization technique which approximates a system of differential equations by a large number of algebraic equations to be solved by a digital computer [18].

The program input data include:

1. Venturi geometry.
2. Loadings: prescribed temperatures, prescribed heat fluxes, convection coefficients with freestream bulk temperatures, or insulated surfaces.
3. Material properties: thermal conductivity, density, and specific heat of the material being analyzed (Type 310 steel).

The output is temperature distribution in the r and z directions.

Model Development

The venturi geometry will be interpreted by the computer in terms of nodes and elements. The element used is an axisymmetric ring element with a three-dimensional thermal conduction capability. The element has four nodal points with a single degree of freedom, temperature, at each node (Figure 14). The more elements that the model contains, the better the resolution of the answers. The elements cannot have an aspect ratio (length divided by width) greater than five, else the accuracy of the results will suffer [17]. The venturi is broken down into logical elements and the nodes numbered accordingly. Each node is defined in (x,r) coordinates with an origin at the intersection of the venturi centerline and the plane of the venturi entrance. The coordinates of each node is defined in inches. The model as represented by the computer is shown in Figures 15, 16, and 17.

The contact resistance is represented in the model by a thin layer of elements with a thermal conductivity equivalent to the resistance per foot of material. (See Appendix F for the calculations.)

The boundary conditions applied to the model are:

1. Convection coefficients for the inner venturi surface in contact with airflow.
2. Convection coefficients for the cooling passage surface in contact with the cooling fluid.
3. All remaining surfaces were assumed to be insulated.

The convection coefficients are applied to the appropriately numbered element faces (as defined by Figure 14) and the bulk fluid temperature for both the air (650°F) and the ethylene glycol (80°F) are specified for the analysis.

Additional inputs include the venturi material conductivity, density, and specific heat [19] as defined in Appendix C. These properties are assumed to be constant throughout the venturi.

## Model Results

The calculated temperature distribution in the venturi flanges is represented by the contours of Figure 18. The maximum temperature experienced by the O-ring seal is 360°F, below the design limit of 450°F. Therefore, at the highest total airflow conditions, a bulk ethylene glycol temperature of 80°F provides a 240°F margin of safety on O-ring failure. Local boiling is not a problem since the boiling point for 50 percent aqueous ethylene glycol is 315°F.

Because the temperature of the cooling passage is known, the surface viscosity of the coolant can be determined and the effects on the overall convection coefficient calculated. The average surface temperature of the venturi is 160°F. The ratio of coolant viscosity of the freestream to the viscosity at the surface increases the convection coefficient by 10 percent. This is insignificant compared to the overall accuracy of heat transfer.

## Quantifying Head Load

The calculated temperature of each node located around the perimeter (Figure 19) of the cooling passage is listed in Table I. The heat transfer to the cooling fluid is calculated from the following equation [7]:

$$q_i = h_s A_{s,i} \, (T_s - T_{bulk}) \tag{5-16}$$

where $i$ = represents the heat flux contributions from each individual surface

$h_s$ = the heat transfer coefficient and is equal to the previously calculated value of 446 Btu/hr-ft$^2$-°F

$A_{s,i}$ = the heat transfer area of the individual element

$T_s$ = the wall temperature as defined by the model results

$T_{bulk}$ = the bulk fluid temperature and is equal to 80°F (assumed to be constant).

The total heat transferred to the cooling fluid is equal to the sum of the individual contributions of each elemental surface area in contact with the cooling fluid. The temperature of the surface is the arithmetic mean of the temperatures of the connecting nodes. Figure 20 represents the calculated temperature for each node surrounding the cooling passage and Figure 21 shows the surface temperature for the $i^{th}$ surface. Table II lists the heat transfer contributions from each surface element.

The total heat energy transfer to the fluid, based on a bulk temperature of 80°F throughout, is 56,700 Btu/hr.

## CHAPTER VI

## VENTURI COOLING PASSAGE OUTLET TEMPERATURE

### The Calculation

The outlet temperature of the aqueous ethylene glycol solution is found from the following equation [7]:

$$T_{out} = q_t/(m_e C_{p,e}) + T_{in} \tag{6-1}$$

where $q_t$ = total heat transferred to the fluid

$m_e$ = mass flow rate of aqueous ethylene glycol

$C_{p,e}$ = specific heat of aqueous ethylene glycol

$T_{in}$ = fluid inlet temperature

The outlet temperature is calculated to be 100°F.

### Evaluation of Assumptions

The basic assumption made for convection coefficient calculations for the cooling passage surface, temperature distribution calculations in the model, and heat load delivered to the cooling fluid, is a constant fluid bulk temperature of 80°F.

The resulting average bulk temperature of the aqueous ethylene glycol solution is 90°F. The variation of fluid properties based on the average temperature and not the initial temperature is insignificant and will induce no extraneous errors.

As the fluid temperature rises, the maximum calculated temperature at the O-ring seal may increase. However, as shown in Appendix G, there is no increase in the maximum exposure temperature with a bulk coolant temperature of 110°F.

The maximum heat load delivered to the coolant occurs at a bulk temperature of 80°F (as compared to a higher temperature). With a bulk coolant temperature of 110°F (10°F higher than the actual coolant outlet temperature), the cooling passage surface temperature increases slightly (Table III). As shown in Table IV, the heat flux transferred to the coolant at 110°F is approximately equal to the heat flux transferred to the coolant at 80°F. As the coolant temperature increases, the viscosity decreases thus increasing the Reynolds number and the convection coefficient; however, the total heat transfer increases only slightly because the temperature difference between the wall and the coolant decreases.

Therefore, the assumption that the bulk coolant temperature of 80°F is constant throughout the cooling passage does not significantly affect the calculated heat load transferred to the coolant.

# CHAPTER VII

# THE REQUIRED HEAT EXCHANGER CHARACTERISTICS

## Approach

As described by Shah [20], a heat exchanger is a device which is used for transfer of internal energy between two or more fluids at differing temperatures. Common examples of heat exchangers are shell and tube exchangers, automobile radiators, condensers, evaporators, air preheaters, and dry cooling towers.

A heat exchanger consists of active heat exchanging elements such as a core of a matrix containing the heat transfer surface, and passive fluid distribution elements such as headers, manifolds, tanks, inlet and outlet nozzles or pipes. Usually there are no moving parts in a heat exchanger, except for recuperators. The heat transfer surface is a surface of the exchanger core which is in direct contact with fluids and through which heat is transferred by conduction.

Heat exchangers are classified according to the heat transfer processes, number of fluids, degree of surface compactness, construction type, flow arrangements, and heat transfer mechanism. For this study, a counterflow heat exchanger (Figure 22) was chosen. As shown in Figure 23, the counterflow heat exchanger requires the least surface area throughout the temperature operating range [21].

To arrive at a design of a new exchanger to meet the specified performance is referred to as a sizing problem. For the counterflow heat exchanger, the specified performance is that the aqueous ethylene glycol cooling fluid, flowing at 1.0 lbm/sec, is cooled from 100°F (this is the calculated primary coolant temperature at the exit of the venturi cooling passage, see page 27) to 80°F prior to re-entering the venturi.

Because of the abundance of heat exchanger literature, a complete heat exchanger design is not provided here. Only the heat exchanger characteristics required to deliver an adequate coolant temperature at the inlet of the venturi cooling passage are defined. The required characteristics are as follows:

1. Heat transfer material type.
2. Secondary coolant type.
3. Secondary coolant flow rate.
4. Secondary coolant inlet temperature.
5. Heat transfer area.

## Results

The material chosen for the heat transfer surface area (the tube in the counterflow arrangement) is copper. Copper has a high thermal conductivity (see Appendix D) [7] and will offer little resistance to heat transfer. Copper is commonly used for such an application when high system pressures are not experienced.

The secondary coolant is raw water stored in a reservoir exposed to the environment. The raw water is inexpensive and readily available. The water is pumped from the bottom of the reservoir, and therefore, should maintain a temperature range of 50 to 70°F year round. A temperature of 70°F will be used for this analysis and will provide conservative results.

The heat transfer surface area is based on the specified performance and empirical correlations of the convection coefficient. A tube radius will be defined. The required area will be described for the given tube radius. The heat transfer surface area will be optimized in terms of secondary coolant flow rate, tube size, and shell size.

The required heat transfer area is calculated by the NTU method [20] where an NTU is defined as:

$$NTU = \frac{U_{tot}\ A_s}{C_{min}} \tag{7-1}$$

where NTU = number of transfer units

$U_{tot}$ = overall heat transfer coefficient

$A_s$ = heat transfer surface area

$C_{min}$ = minimum stream heat capacity rate

The counterflow heat exchanger NTU is related to heat exchanger effectiveness, $C_{min}$ and $C_{max}$, as shown in Figure 24 [7].

The effectiveness of a heat exchanger is

$$\varepsilon = \frac{q_t}{q_{max}} \tag{7-2}$$

where $q_t$ = actual heat transferred

$q_{max}$ = the maximum heat that could be transferred between the two streams.

The methodology of calculating the area is as shown in Appendix H. Because repeated calculations are necessary, a computer program was written and used to optimize the heat transfer surface area, (see Appendix I).

Three tube sizes (listed as outside diameter) were investigated: (1) 1 inch, (2) 2 inches, and (3) 4 inches. In general, the results as shown in Figures 25 through 30 are:

1. The required heat transfer surface area for the counterflow heat exchanger does not decrease significantly for all three cases for secondary coolant mass flow rates greater than 4 lbm/sec.
2. The required heat transfer surface area increases for all three cases as shell diameter increases.

Therefore, the optimum counterflow single pass shell and tube heat exchanger contains a 1-inch tube (outside diameter is 1.12 inches), a 2-inch shell (outside diameter is 2.5 inches), a secondary coolant mass flow rate equal to 4 lbm/sec, and a total surface area of 28 $ft^2$. As illustrated in Figure 25, increasing the flow rate above 4 lbm/sec does not significantly lower the required heat transfer surface area. A low flow rate of 4 lbm/sec allows the system to operate at low pressures; thus permitting the use of copper tubing (Appendix J).

To attain the required heat transfer surface area (28 $ft^2$), the secondary coolant tube must be 107 feet long. This is unreasonably long for a counterflow heat exchanger. Another type of heat exchanger should be used to meet requirements which would be space conservative.

A high surface area per unit volume is an alternative approach to attaining the required surface area. This type of heat exchanger is referred to as a compact heat exchanger. One method for increasing surface area density is to build up a heat exchanger by stabilizing parallel plates, with fins or extended surfaces. Devices that simply increase the level of turbulence in a tube wall increase the heat transfer coefficient, but are inefficient when the added friction power expenditure is considered.

A finned tube will provide increased surface area, and therefore decrease the required tube length of the proposed counterflow heat exchanger substantially. Compared to a compact heat exchanger, the finned tube will not induce large pressure drops. Therefore, the finned tube will be used in the counterflow heat exchanger to provide for adequate heat transfer.

The tube must be finned on both sides to assure that the heat transferred from the hot primary coolant to the tube is equivalent to the heat transferred from the tube to the cold secondary coolant.

The required length of the finned tube can be calculated by specifying the number of fins along the inner tube wall and the fin height and thickness. Because the tube diameter is 1 inch, the number of fins is limited inside the tube. A fin with a rectangular cross section was chosen because it has a good efficiency [7].

As shown in Appendix K, ten fins, each with a height of 0.25 inches, will provide 28 $ft^2$ of surface area along the inner tube wall if the tube is 60 feet long. Increasing the fin height and thickness does not decrease the number of fins required and will increase the pressure drop through the finned tube. If the tube length is decreased slightly, the number of fins required increases rapidly, and this is not desirable. Ten fins along the inner tube surface will allow adequate heat transfer and not induce significant pressure drop.

Identical fins 0.5 inches in height are required on the outer tube surface to provide adequate heat transfer, (see Appendix K).

Therefore, the addition of fins has decreased the required tube length from 107 ft to 60 ft. A multipass counterflow arrangement utilizing the finned tube will allow for a more compact heat exchanger system (Figure 31).

## CHAPTER VIII

## CONCLUSIONS

Considering specified airflow conditions, the maximum heat flux delivered to the given venturi was determined using a finite-element computer program to solve for the temperature distribution in the critical portion of the venturi, the flanges. A 50 percent by weight aqueous ethylene glycol solution was selected as the primary coolant because of its low freezing point and high boiling point. The primary coolant flow rate was assigned 1 lbm/sec at a pump pressure head of 25 psi. The pressure drop in the primary coolant system was insignificant because of the low flow rate. A primary coolant temperature at the inlet of the venturi cooling passage of 80°F was determined to be sufficient to allow the O-ring seals to operate at the maximum expected critical airflow conditions ($T_t$ = 650°F and $P_t$ = 150 psia). The maximum calculated O-ring temperature was 360°F, a 240°F margin of safety.

Heat exchanger characteristics were defined in terms of the required heat transfer surface area which would provide a venturi cooling passage inlet temperature of 80°F given a cooling passage outlet temperature of 100°F. For a counterflow exchanger, the heat transfer surface area was optimized in terms of the tube flow rate, tube size, and shell size. The minimum required area was 28 ft$^2$. This is an equivalent tube length of 107 ft. This is unreasonably large for a counterflow heat exchanger. A compact heat exchanger was considered to provide the necessary area in a reasonably small container. However, the compact heat exchanger would create system pressures greater than desired (thus damaging the copper tubing). Compact heat exchangers most often are used in gas-to-gas or gas-to-liquid heat exchangers where large pressure losses are not significant to system performance.

Therefore, a finned tube design was selected to provide the required surface area in a reasonable sized counterflow design. This type of counterflow heat exchanger will provide the necessary surface area and will not induce significantly large pressure losses. An analysis of the design was done to specify fin geometry and fin area. The analysis determined the fin effectiveness for worst-case operation of "no fin tip heat transfer." Based on the analysis, an optimum finned tube with both inner and outer fins reduced the required tube length to 60 feet. To accommodate the finned tube in a reasonable area, a multipass counterflow arrangement was designed (Figure 31). Five counterflow passes, each containing 12 feet of the designed finned tubing, will provide the required heat transfer surface area.

In conclusion, therefore, a heat exchanger has been designed that satisfies the requirements for cooling a fluid-cooled venturi metering high temperature air.

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# APPENDIXES

APPENDIX A

TABLES AND FIGURES

Table I. Node Temperatures at the Cooling Passage Surface with a Bulk Coolant Temperature of 80°F

| Node Number | $r^3$ (inches) | $X^4$ (inches) | Node Temperature (°F) |
|---|---|---|---|
| 318 | 8.75 | 25.69 | 264 |
| 319 | 8.75 | 25.88 | 257 |
| 320 | 8.75 | 26.06 | 274 |
| 330 | 8.92 | 26.06 | 213 |
| 332 | 9.00 | 26.06 | 196 |
| 392 | 9.13 | 26.06 | 179 |
| 397 | 9.25 | 26.06 | 168 |
| 407 | 9.38 | 26.06 | 159 |
| 412 | 9.50 | 26.06 | 154 |
| 422 | 9.63 | 26.06 | 149 |
| 427 | 9.75 | 26.06 | 147 |
| 437 | 9.88 | 26.06 | 146 |
| 442 | 10.00 | 26.06 | 147 |
| 455 | 10.13 | 26.06 | 153 |
| 454 | 10.13 | 25.97 | 137 |
| 453 | 10.13 | 25.87 | 133 |
| 452 | 10.13 | 25.69 | 143 |
| 441 | 10.00 | 25.69 | 130 |
| 436 | 9.88 | 25.69 | 126 |
| 426 | 9.73 | 25.69 | 129 |
| 411 | 9.50 | 25.69 | 139 |
| 396 | 9.18 | 25.69 | 144 |
| 391 | 9.08 | 25.69 | 158 |
| 329 | 8.92 | 25.69 | 190 |

[3] Radial distance from venturi centerline

[4] Axial distance from venturi entrance

Table II. The Heat Transferred from Each Individual Surface as Identified in the Model for a Coolant Bulk Temperature Equal to 80°F

| Surface | Node Numbers | Surface Length (inches) | Surface Area ($in.^2$) | Surface Temp. (°F) | Heat Transferred (Btu/hr) |
|---|---|---|---|---|---|
| 1 | 318-319 | 0.19 | 10.44580 | 260.4 | 5836.3 |
| 2 | 319-320 | 0.18 | 9.89602 | 265.7 | 5692.0 |
| 3 | 320-330 | 0.17 | 9.43703 | 243.5 | 4779.6 |
| 4 | 330-332 | 0.08 | 4.50379 | 204.3 | 1733.9 |
| 5 | 332-392 | 0.13 | 7.40442 | 187.5 | 2465.5 |
| 6 | 392-397 | 0.12 | 6.92910 | 173.6 | 2007.7 |
| 7 | 397-407 | 0.13 | 7.60862 | 163.8 | 1974.1 |
| 8 | 407-412 | 0.12 | 7.11759 | 156.6 | 1689.1 |
| 9 | 412-422 | 0.13 | 7.81283 | 151.5 | 1729.9 |
| 10 | 422-427 | 0.12 | 7.30609 | 148.0 | 1538.3 |
| 11 | 427-437 | 0.13 | 8.01703 | 146.2 | 1643.2 |
| 12 | 437-442 | 0.12 | 7.49458 | 146.4 | 1540.5 |
| 13 | 442-455 | 0.13 | 8.22123 | 150.2 | 1786.4 |
| 14 | 455-454 | 0.09 | 5.72838 | 145.2 | 1157.1 |
| 15 | 454-453 | 0.1 | 6.36487 | 134.9 | 1082.2 |
| 16 | 453-452 | 0.18 | 11.45676 | 137.8 | 2051.0 |
| 17 | 452-441 | 0.13 | 8.22123 | 136.4 | 1436.7 |
| 18 | 441-436 | 0.12 | 7.49458 | 127.8 | 1109.2 |
| 19 | 436-426 | 0.15 | 9.24099 | 127.2 | 1350.5 |
| 20 | 426-411 | 0.23 | 13.89495 | 133.5 | 2301.8 |
| 21 | 411-396 | 0.32 | 18.77918 | 141.3 | 3565.7 |
| 22 | 396-391 | 0.1 | 5.73655 | 151.0 | 1262.1 |
| 23 | 391-329 | 0.16 | 9.04779 | 174.1 | 2637.5 |
| 24 | 329-318 | 0.17 | 9.43703 | 227.1 | 4299.5 |
| Total | | 3.5000 | 207.60 | | 56669.80 |

Table III. Node Temperatures at the Cooling Passage Surface with a Bulk Coolant Temperature of 110°F

| Node Number | $r^5$ (inches) | $X^6$ (inches) | Node Temperature (°F) |
|---|---|---|---|
| 318 | 8.75 | 25.69 | 266 |
| 319 | 8.75 | 25.88 | 257 |
| 320 | 8.75 | 26.06 | 275 |
| 330 | 8.92 | 26.06 | 219 |
| 332 | 9.00 | 26.06 | 203 |
| 392 | 9.13 | 26.06 | 189 |
| 397 | 9.25 | 26.06 | 180 |
| 407 | 9.38 | 26.06 | 173 |
| 412 | 9.50 | 26.06 | 168 |
| 422 | 9.63 | 26.06 | 164 |
| 427 | 9.75 | 26.06 | 162 |
| 437 | 9.88 | 26.06 | 162 |
| 442 | 10.00 | 26.06 | 163 |
| 455 | 10.13 | 26.06 | 169 |
| 454 | 10.13 | 25.97 | 155 |
| 453 | 10.13 | 25.87 | 150 |
| 452 | 10.13 | 25.69 | 161 |
| 441 | 10.00 | 25.69 | 149 |
| 436 | 9.88 | 25.69 | 145 |
| 426 | 9.73 | 25.69 | 148 |
| 411 | 9.50 | 25.69 | 156 |
| 396 | 9.18 | 25.69 | 160 |
| 391 | 9.08 | 25.69 | 171 |
| 329 | 8.92 | 25.69 | 199 |

[5] Radial distance from venturi centerline

[6] Axial distance from venturi entrance

Table IV. The Heat Transferred from Each Individual Surface as Identified in the Model for a Coolant Bulk Temperature Equal to 110°F

| Surface | Node Numbers | Surface Length (inches) | Surface Area ($in.^2$) | Surface Temp (°F) | Heat Transferred[7] (Btu/hr) |
|---|---|---|---|---|---|
| 1 | 318-319 | 0.19 | 10.44580 | 261.5 | 6153.5 |
| 2 | 319-320 | 0.18 | 9.89602 | 266.4 | 6019.4 |
| 3 | 320-330 | 0.17 | 9.43703 | 247.0 | 5028.9 |
| 4 | 330-332 | 0.08 | 4.50379 | 211.3 | 1773.4 |
| 5 | 332-392 | 0.13 | 7.40442 | 196.6 | 2493.6 |
| 6 | 392-397 | 0.12 | 6.92910 | 184.7 | 2013.4 |
| 7 | 307-407 | 0.13 | 7.60862 | 176.5 | 1968.3 |
| 8 | 407-412 | 0.12 | 7.11759 | 170.6 | 1676.8 |
| 9 | 412-422 | 0.13 | 7.81283 | 166.3 | 1711.9 |
| 10 | 422-427 | 0.12 | 7.30609 | 163.5 | 1519.4 |
| 11 | 427-437 | 0.13 | 8.01703 | 162.1 | 1622.9 |
| 12 | 437-442 | 0.12 | 7.49458 | 162.4 | 1527.1 |
| 13 | 442-455 | 0.13 | 8.22123 | 166.2 | 1795.2 |
| 14 | 455-454 | 0.09 | 5.72838 | 161.9 | 1156.7 |
| 15 | 454-453 | 0.1 | 6.36487 | 152.7 | 1056.9 |
| 16 | 453-452 | 0.18 | 11.45676 | 155.6 | 2031.9 |
| 17 | 452-441 | 0.13 | 8.22123 | 154.6 | 1426.9 |
| 18 | 441-436 | 0.12 | 7.49458 | 147.0 | 1078.0 |
| 19 | 436-426 | 0.15 | 9.24099 | 146.5 | 1310.8 |
| 20 | 426-411 | 0.23 | 13.89495 | 151.9 | 2261.7 |
| 21 | 411-396 | 0.32 | 18.77918 | 158.0 | 3505.8 |
| 22 | 396-391 | 0.1 | 5.73655 | 165.7 | 1243.3 |
| 23 | 391-329 | 0.16 | 9.04779 | 185.4 | 2653.9 |
| 24 | 329-318 | 0.17 | 9.43703 | 232.5 | 4495.1 |
| Total | | 3.5000 | 207.60 | | 57524.85 |

[7]The convection coefficient is 560 Btu/hr-ft-°F

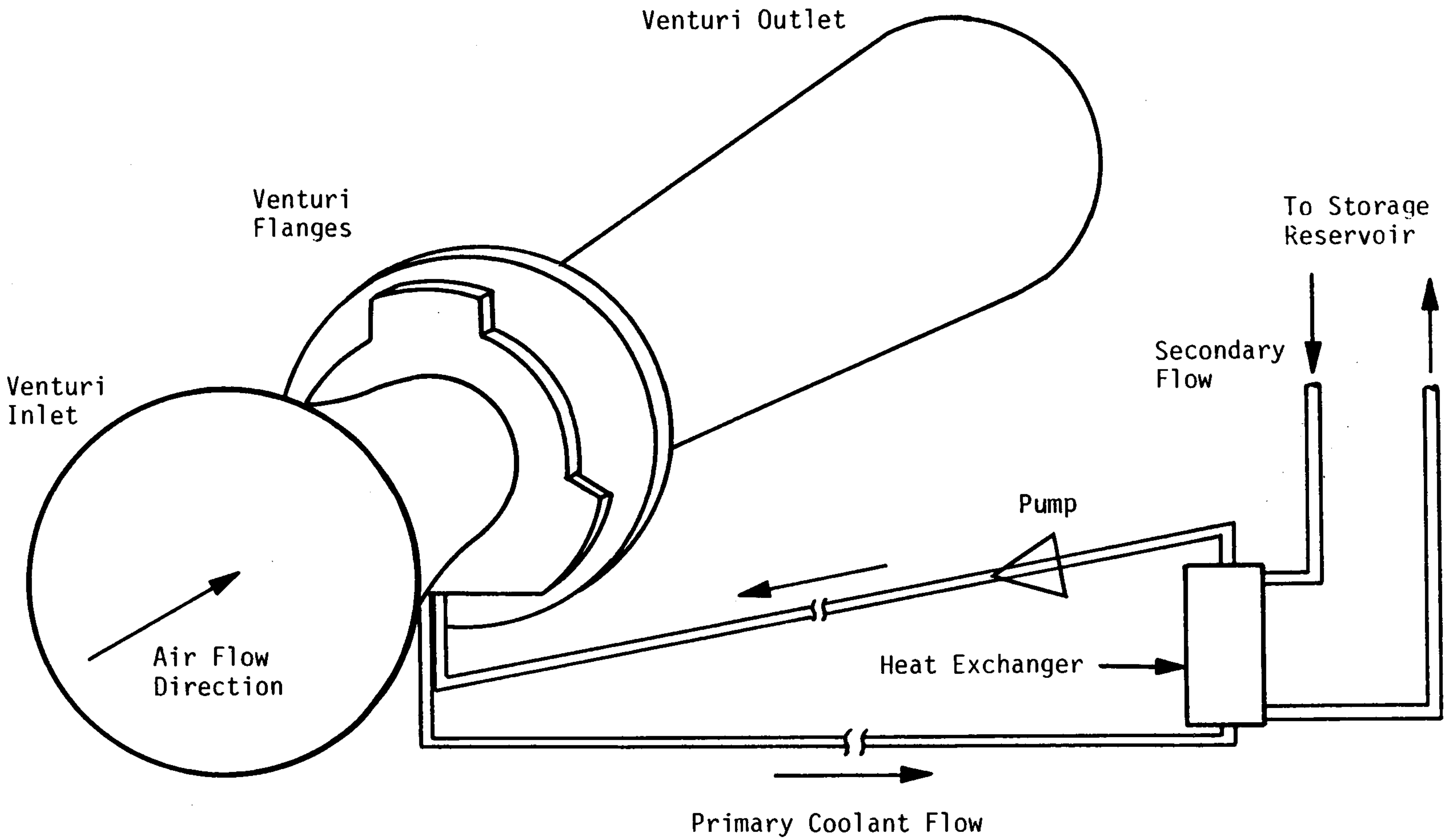



Figure 1. The System; The Venturi and Heat Exchanger

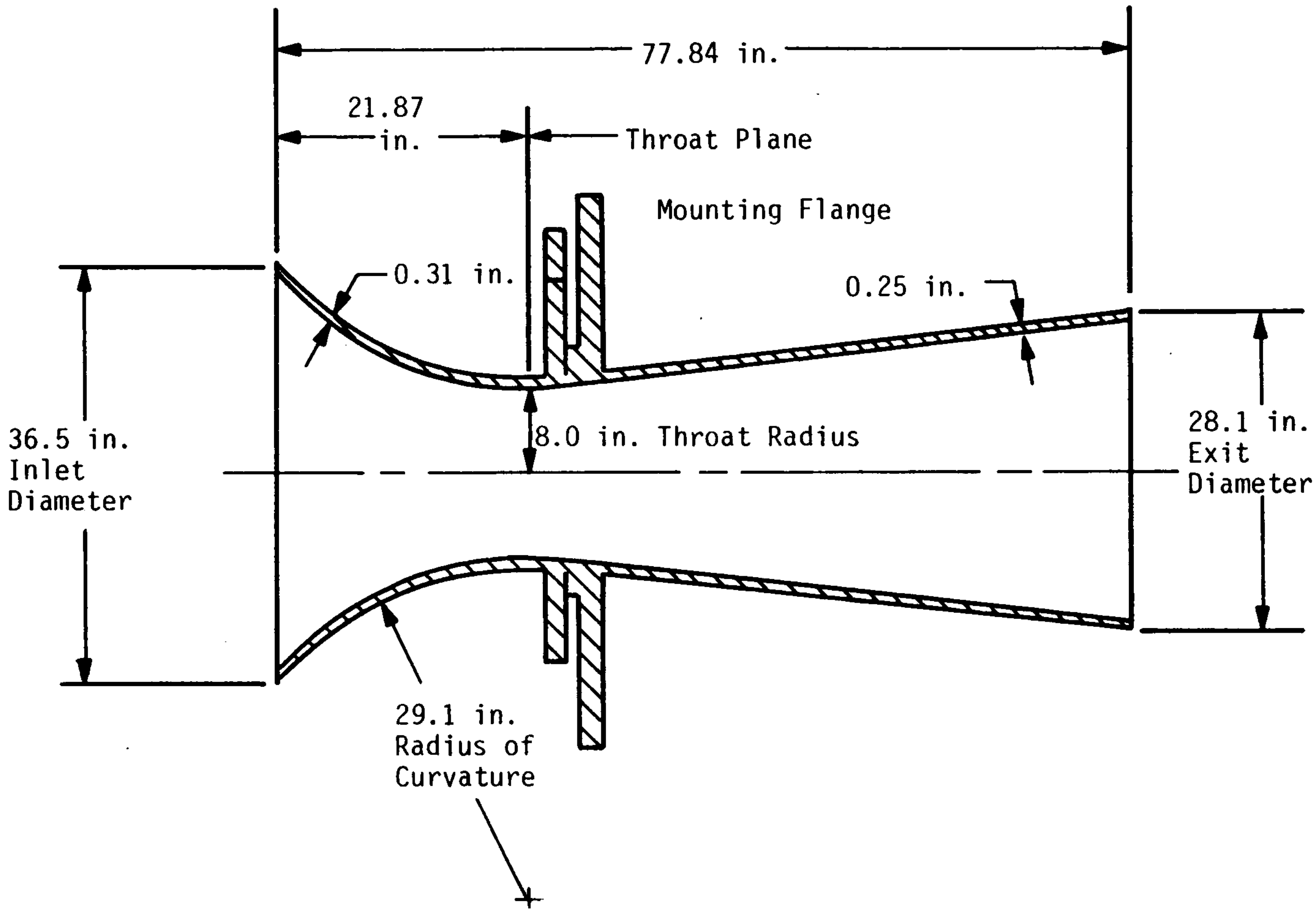



Figure 2. The Venturi Dimensions

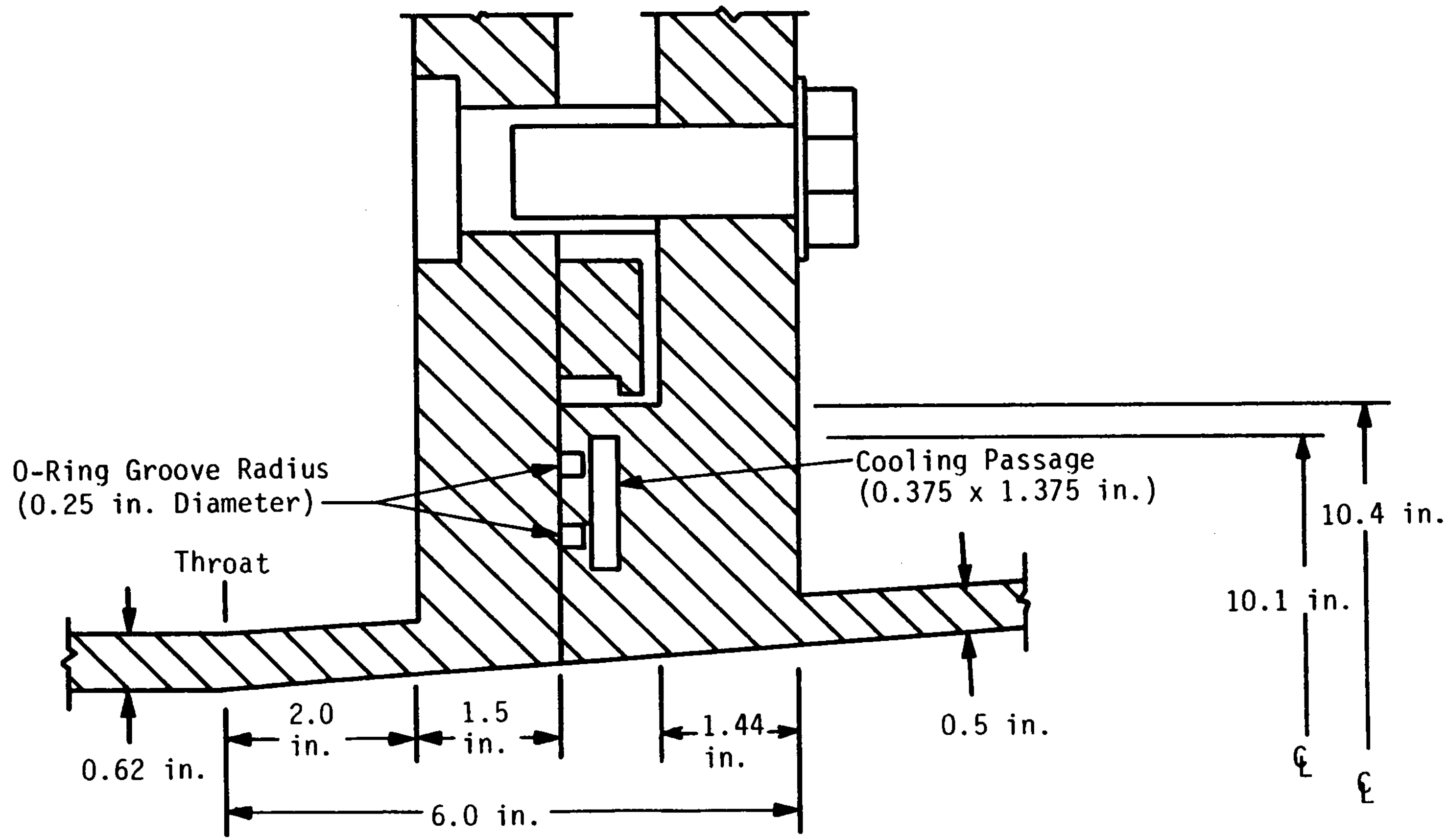



Figure 3. An Enlarged Cross-Sectional View of the Venturi Flanges

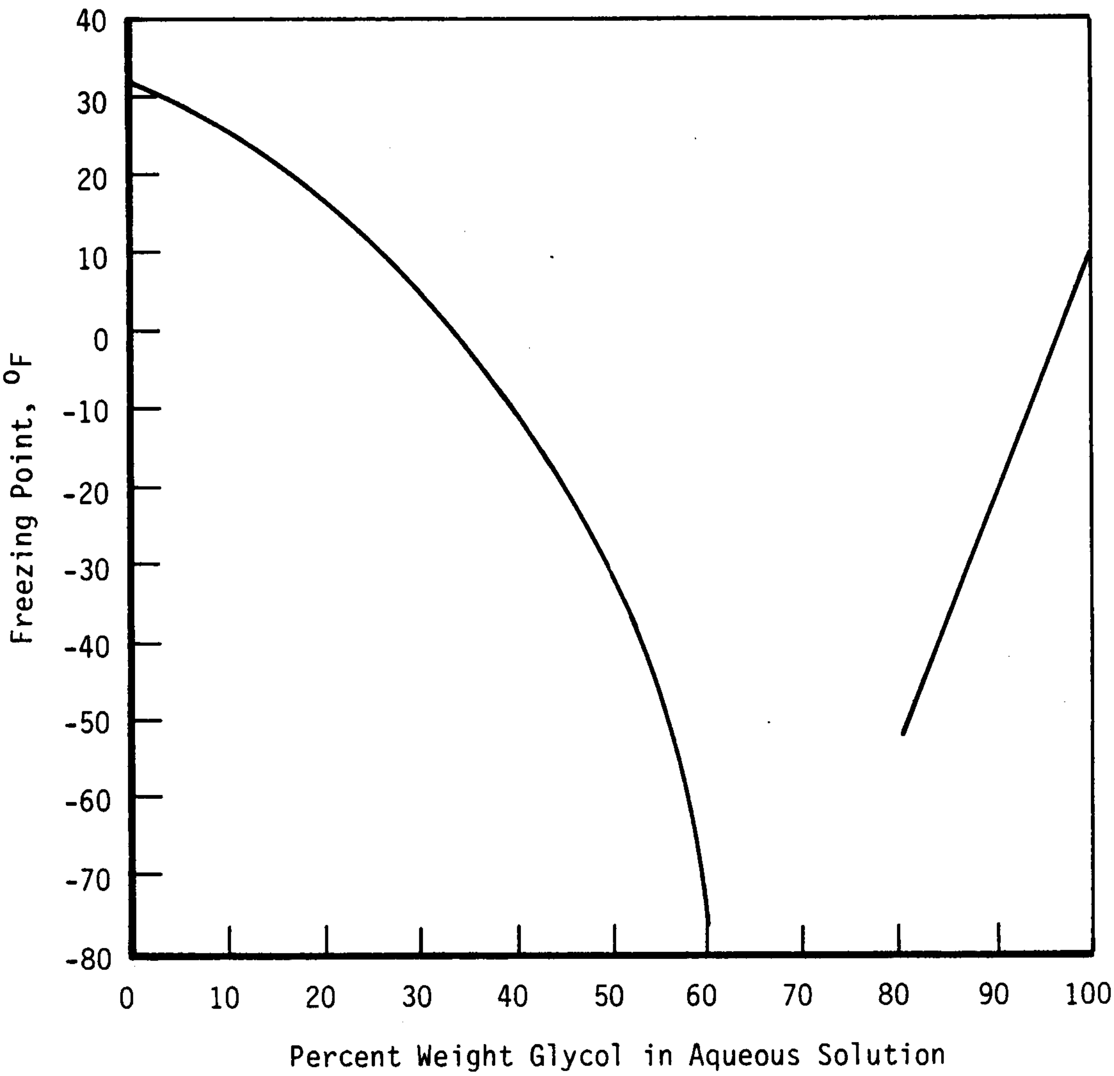



Figure 4. Freezing Point of Ethylene Glycol as a Function of Percent by Weight Aqueous Solution

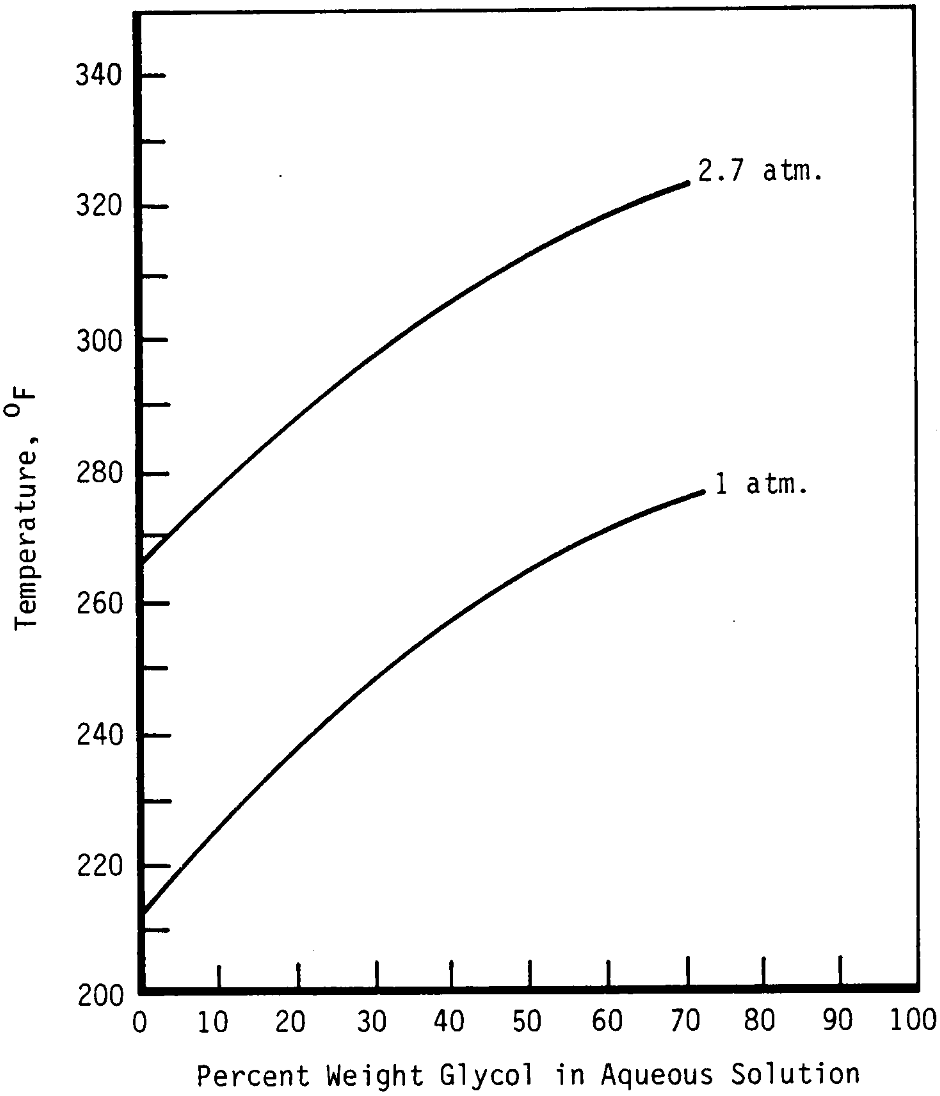



Figure 5. Boiling Point of Ethylene Glycol as a Function of Percent by Weight Aqueous Solution

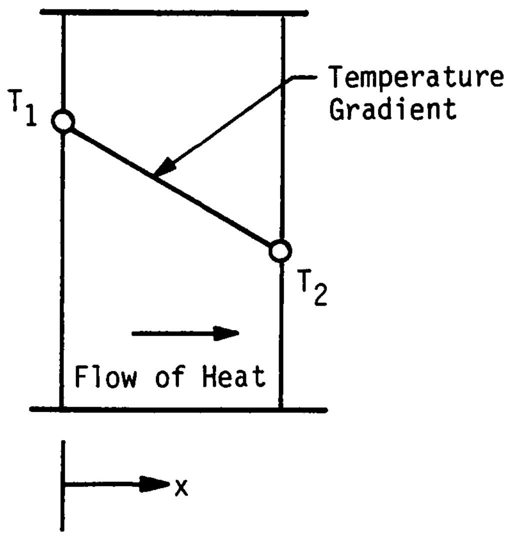



Figure 6. Conduction of Heat Through a Plane Wall

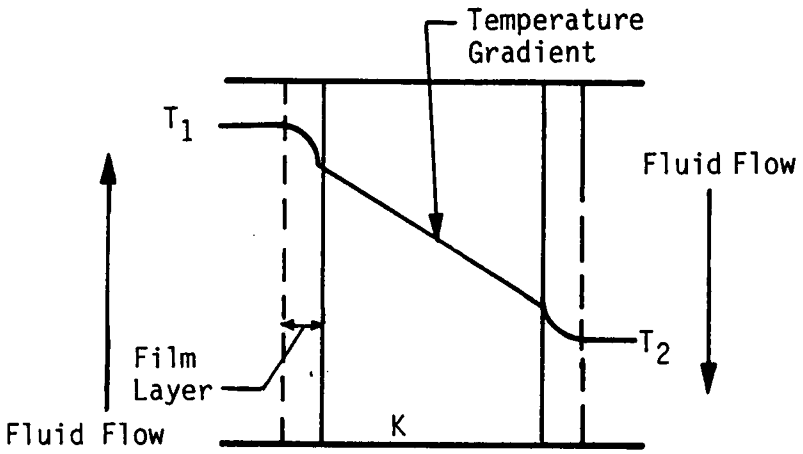



Figure 7. Temperature Variation in a Thin Film of Fluid at a Plane Wall

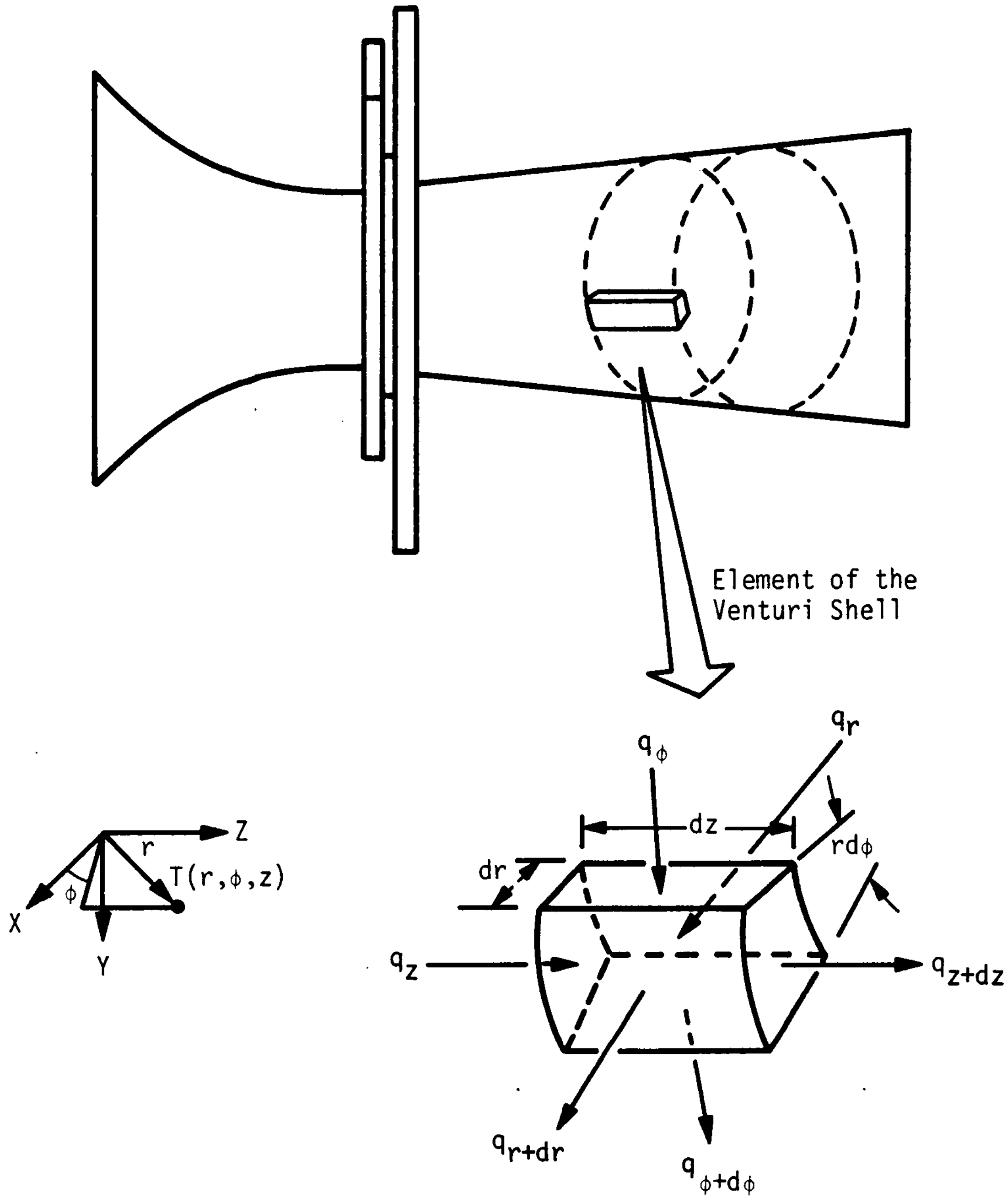
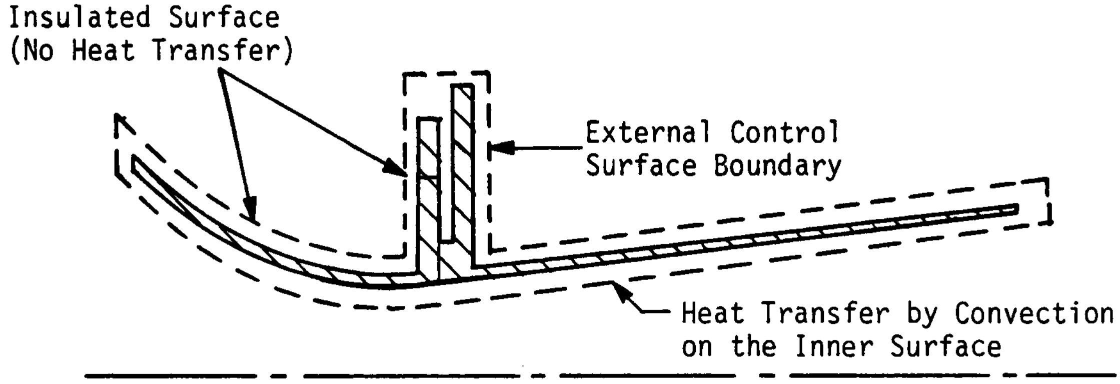



Figure 8. A Cylindrical Differential Control Volume Analysis of the Venturi



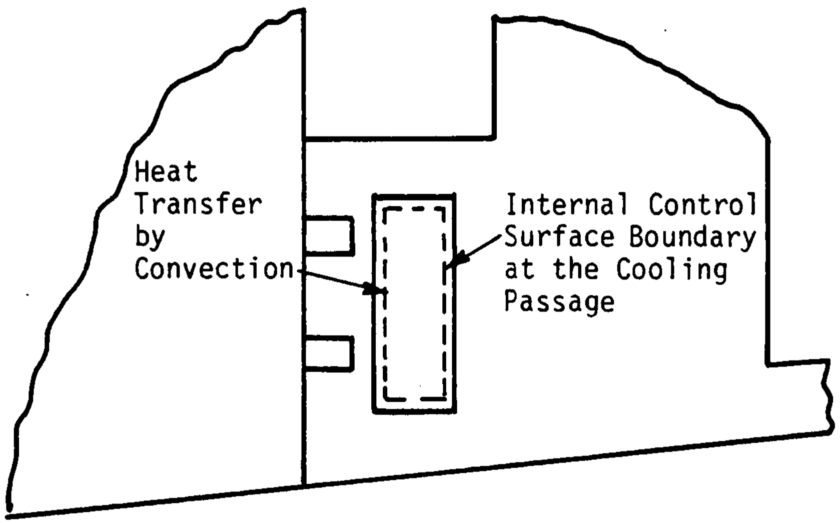




Figure 9. A Finite Control Volume Analysis of the Venturi with the Boundary Conditions Applied

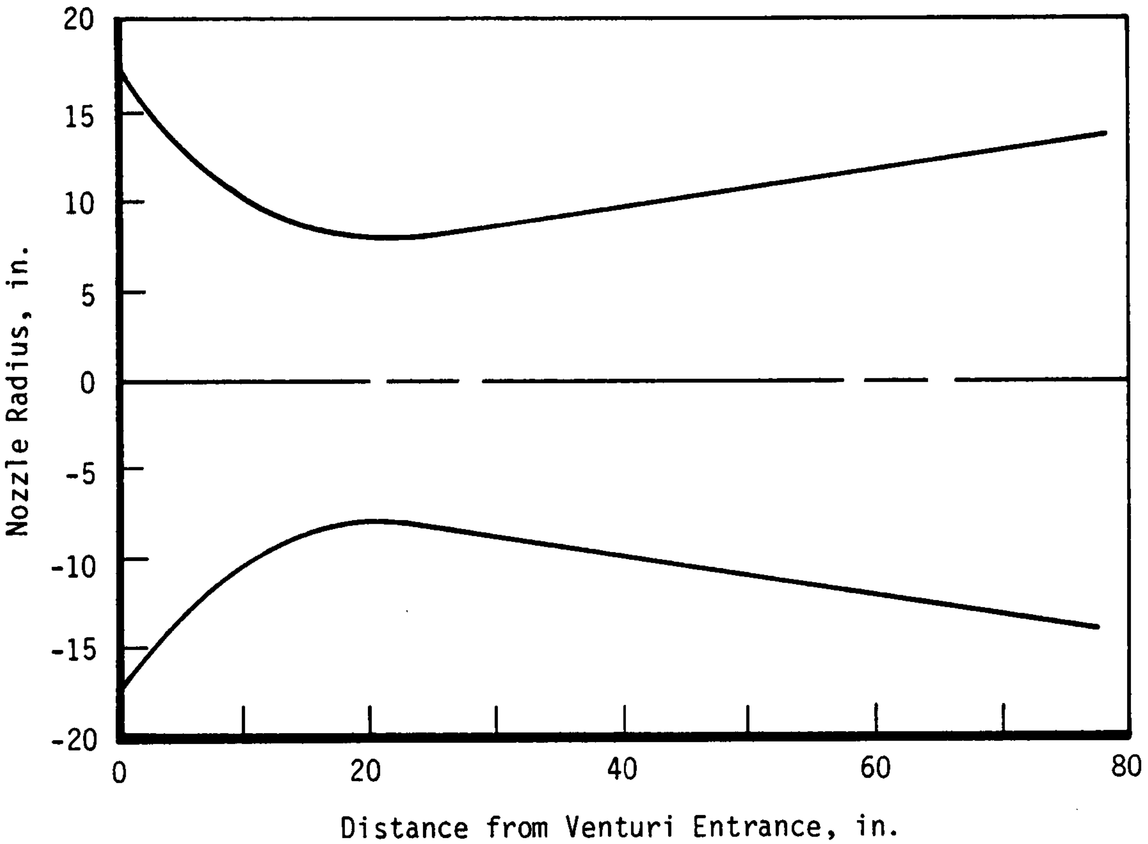



Figure 10. The Venturi Profile as a Function of Diameter

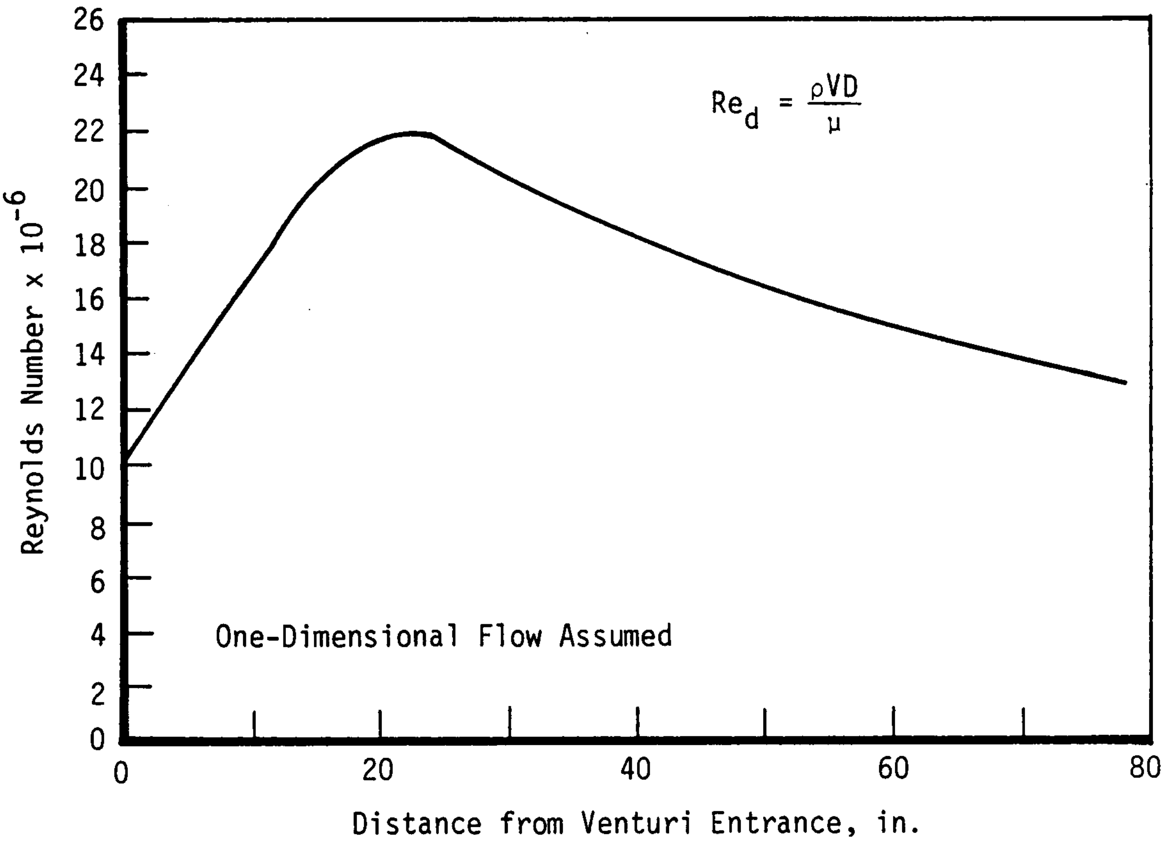



Figure 11. Reynolds Number on the Inner Face of the Venturi as a Function of Diameter

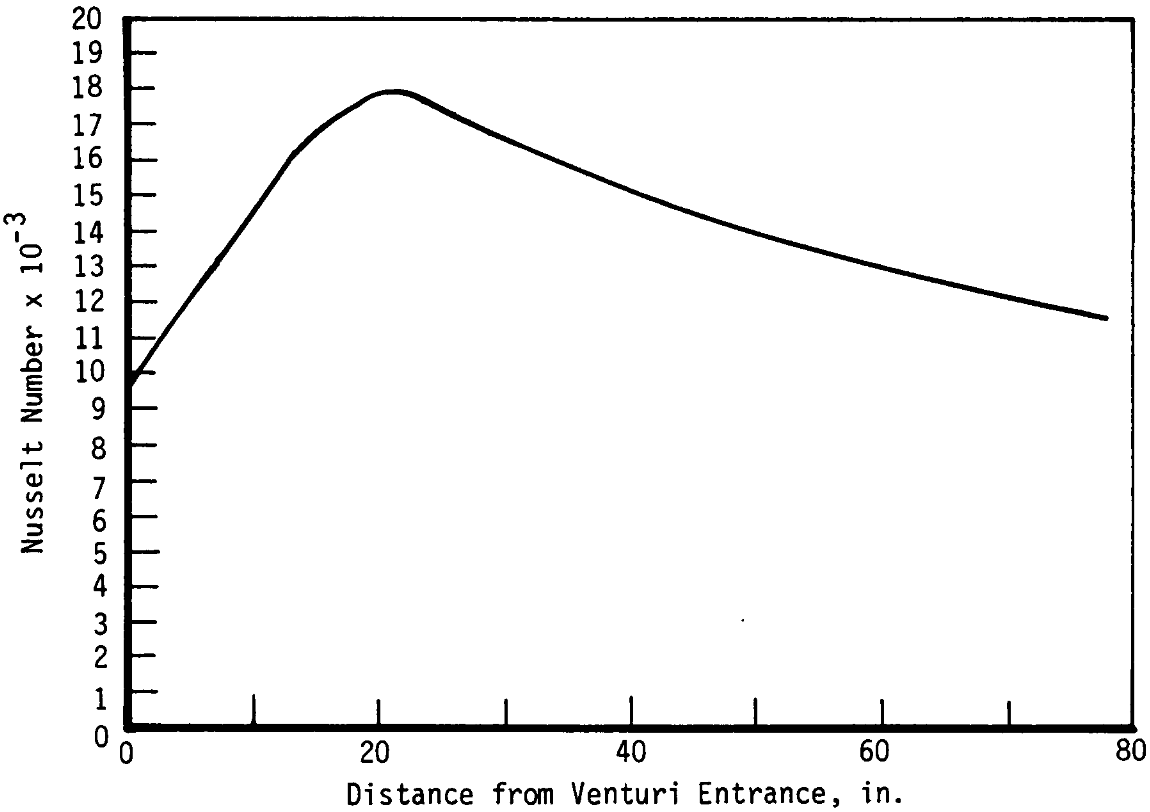



Figure 12. Nusselt Number on the Inner Face of the Venturi as a Function of Diameter

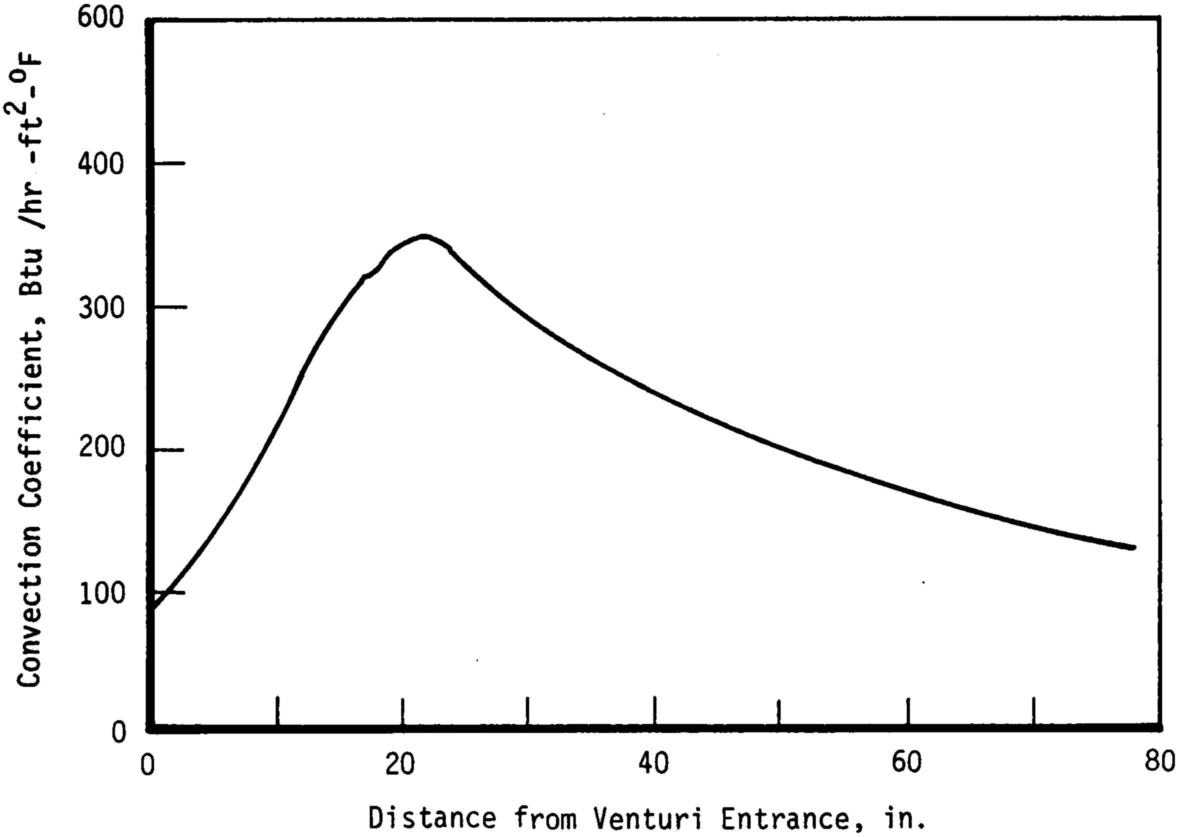



Figure 13. Convection Coefficient on the Inner Face of the Venturi as a Function of Diameter

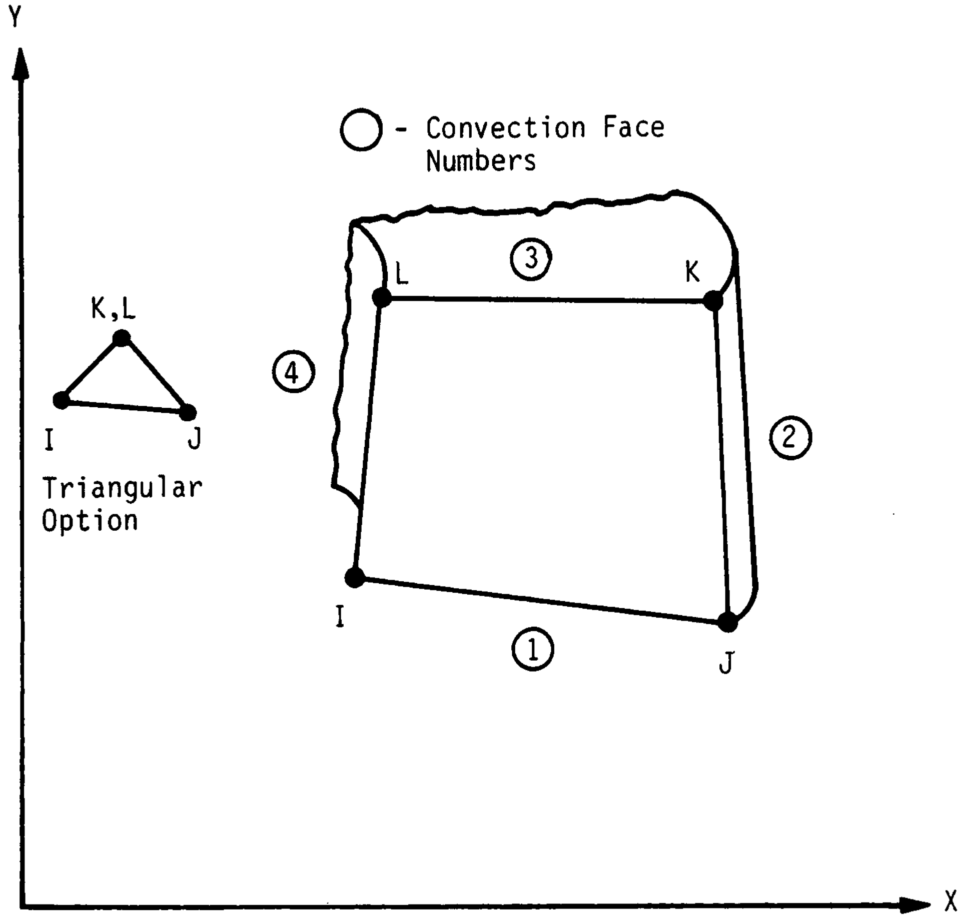



Figure 14. The Harmonically Loaded Axisymmetric Thermal Solid Element as Defined by ANSYS

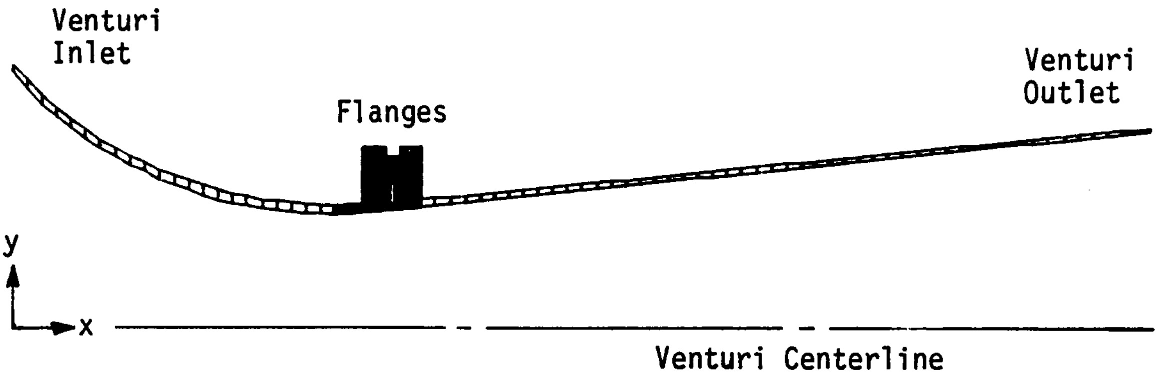



Figure 15. The Venturi as Represented by the Finite-Element Computer Model

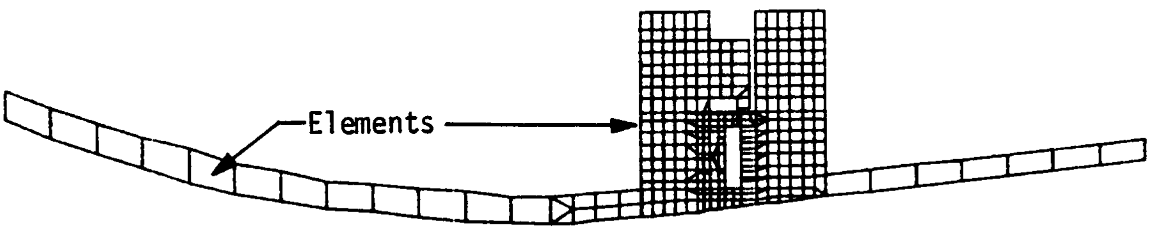



Figure 16. Axisymmetric View of a Portion of the Venturi as Represented by the Finite-Element Computer Model

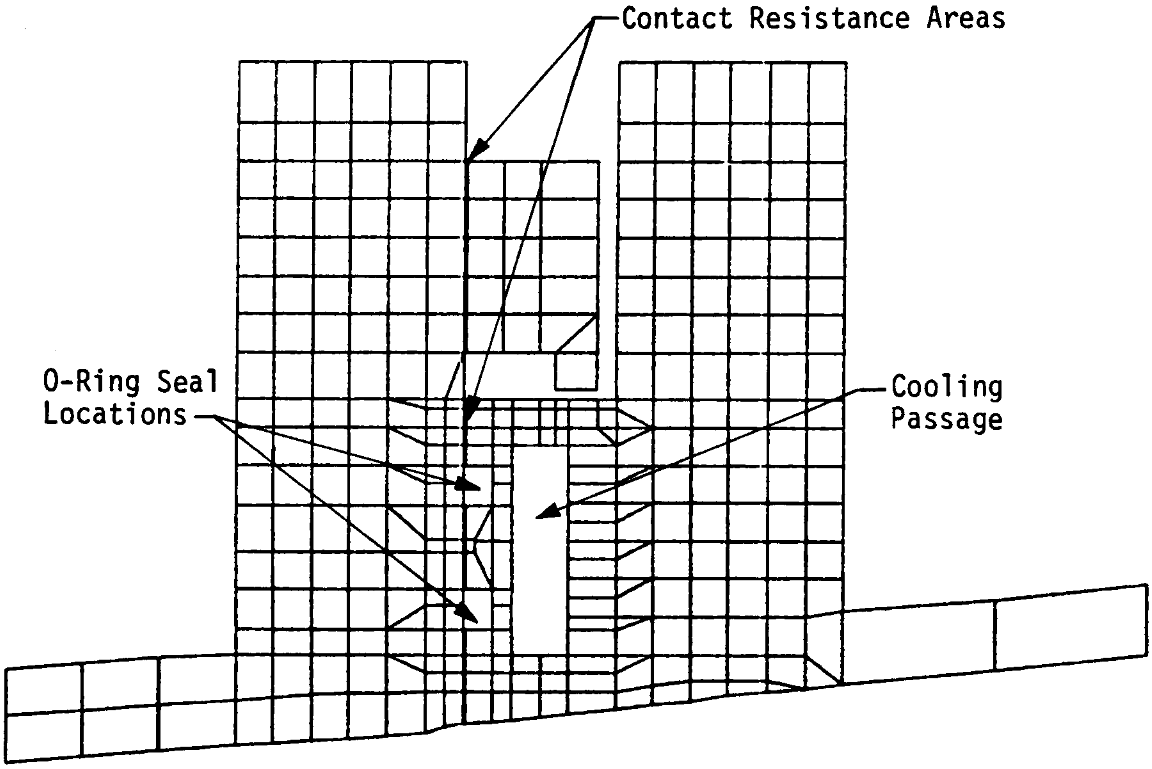



Figure 17. Axisymmetric View of the Venturi Flanges as Represented by the Finite-Element Computer Model

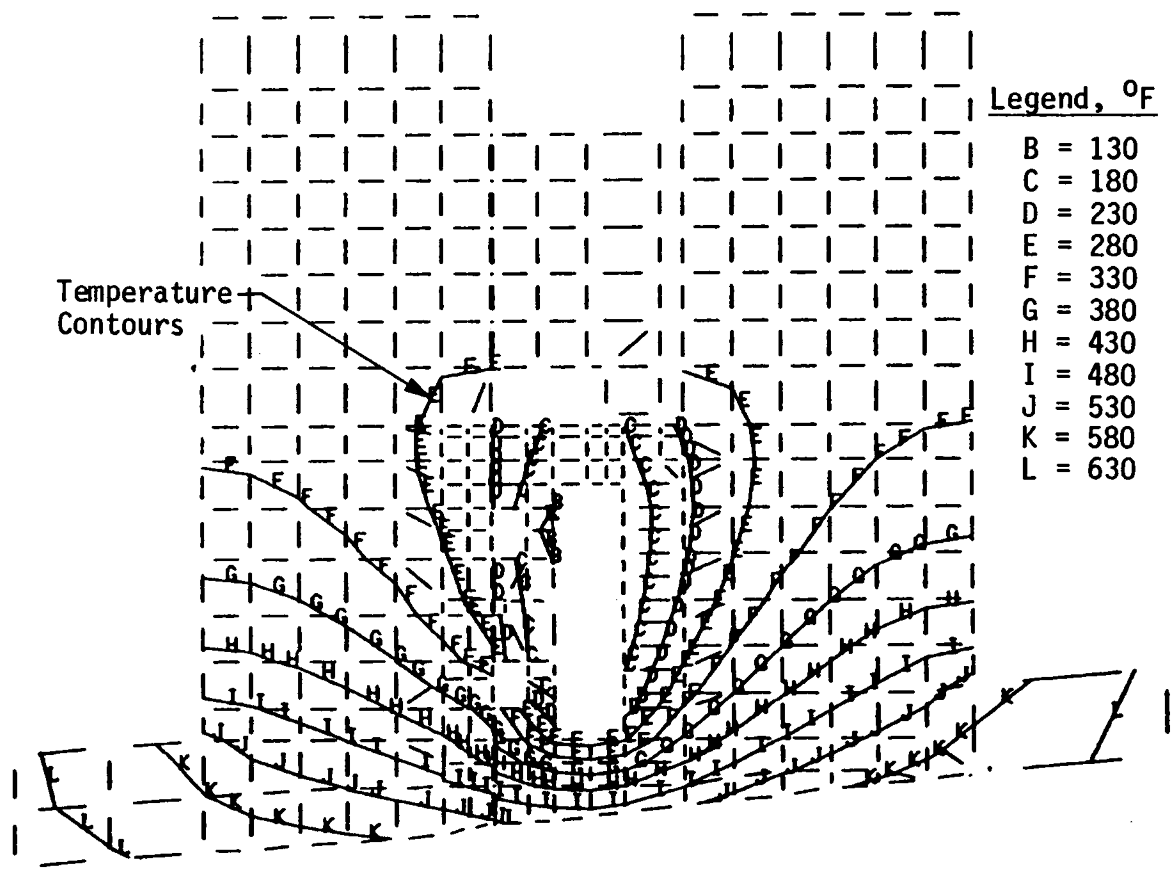



Figure 18. The Temperature Distribution in the Venturi Flanges (Surrounding the O-ring Seals and Cooling Passage) for a Bulk Coolant Temperature of 80°F and Maximum Air-Side Surface Temperature

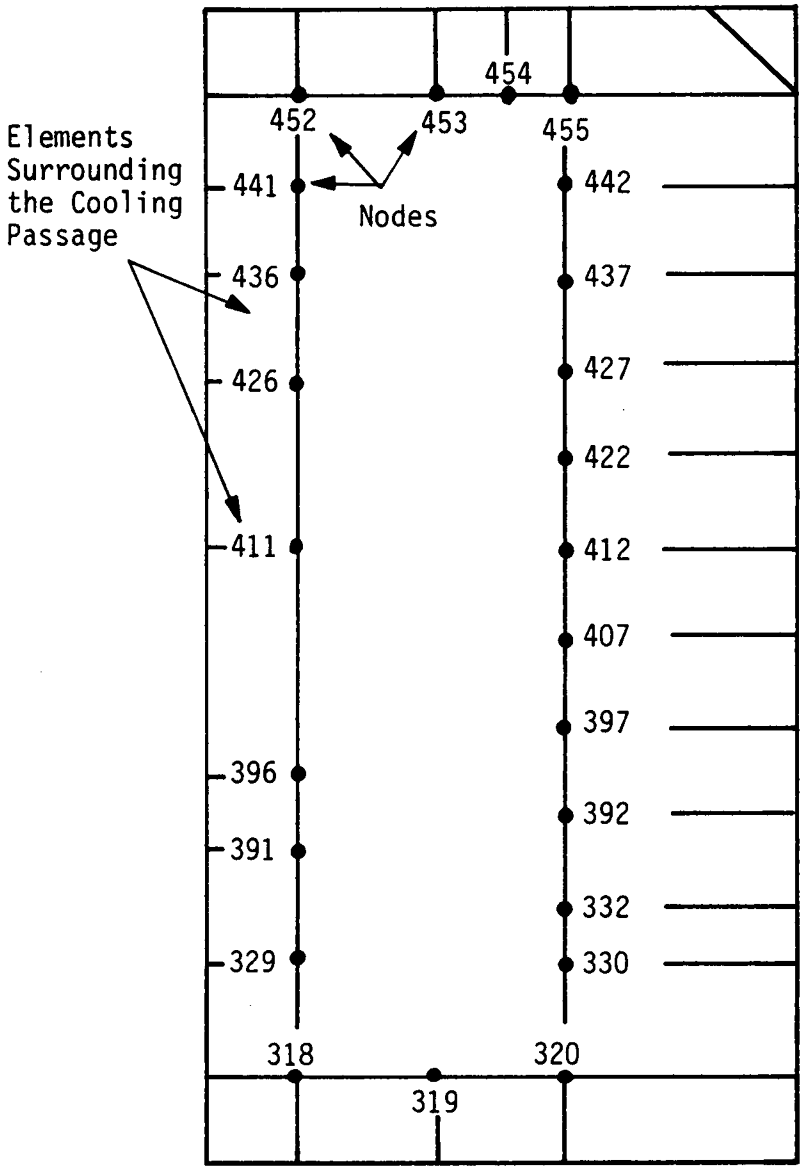



Figure 19. The Node Numbers Identifying Points on the Cooling Passage Surface

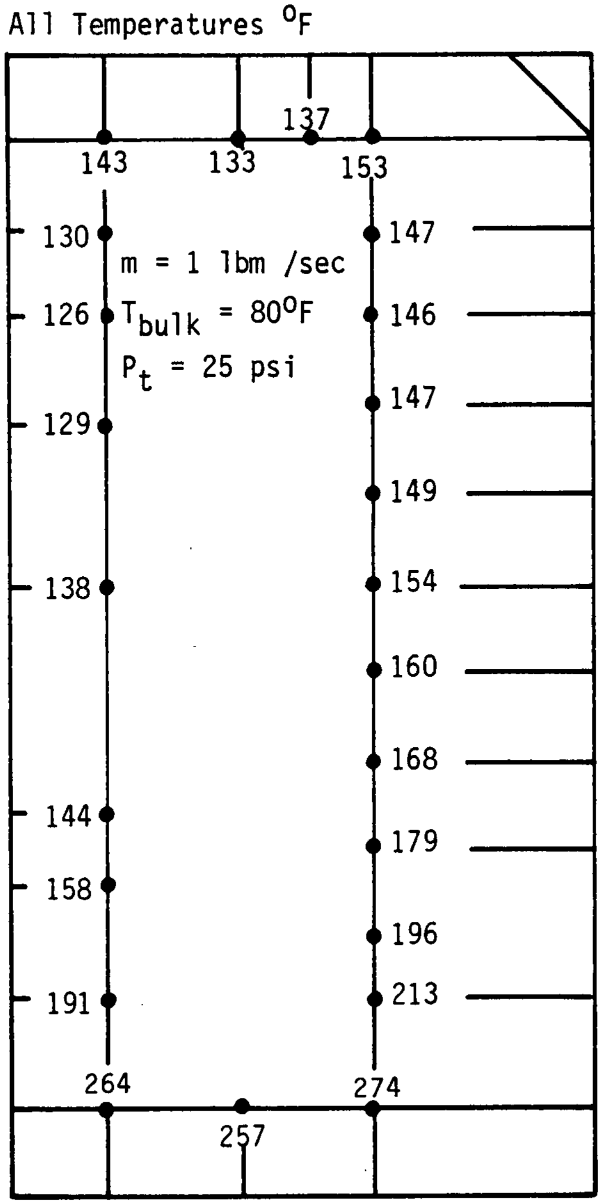



Figure 20. The Local Temperatures of the Nodes on the Cooling Passage Surface

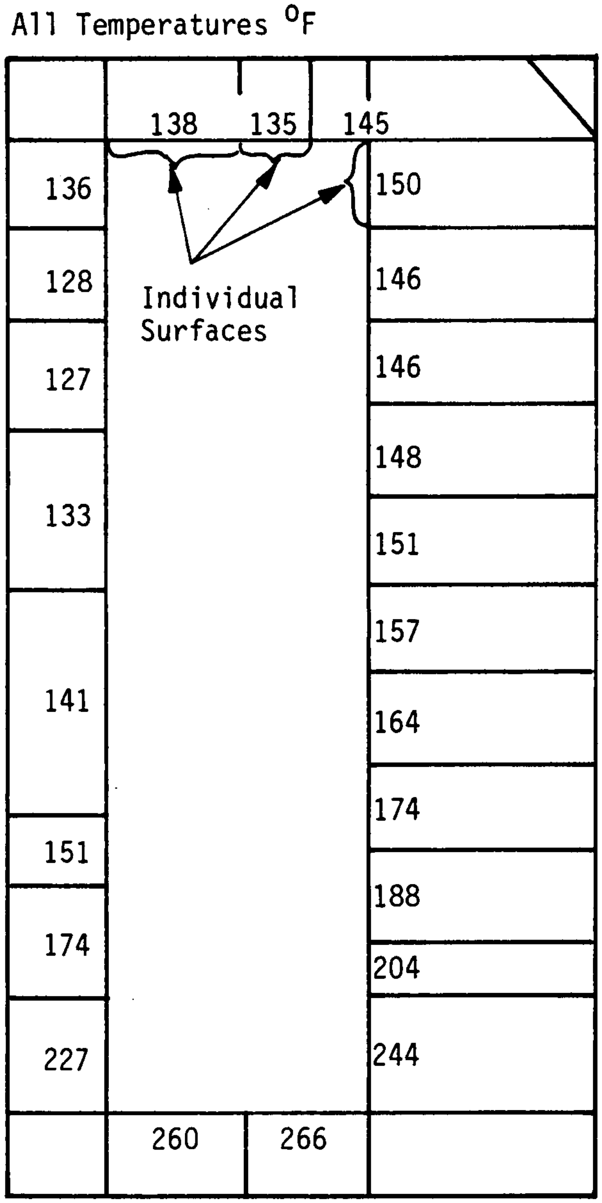



Figure 21. The Average Temperatures of the Individual Surfaces of the Cooling Passage

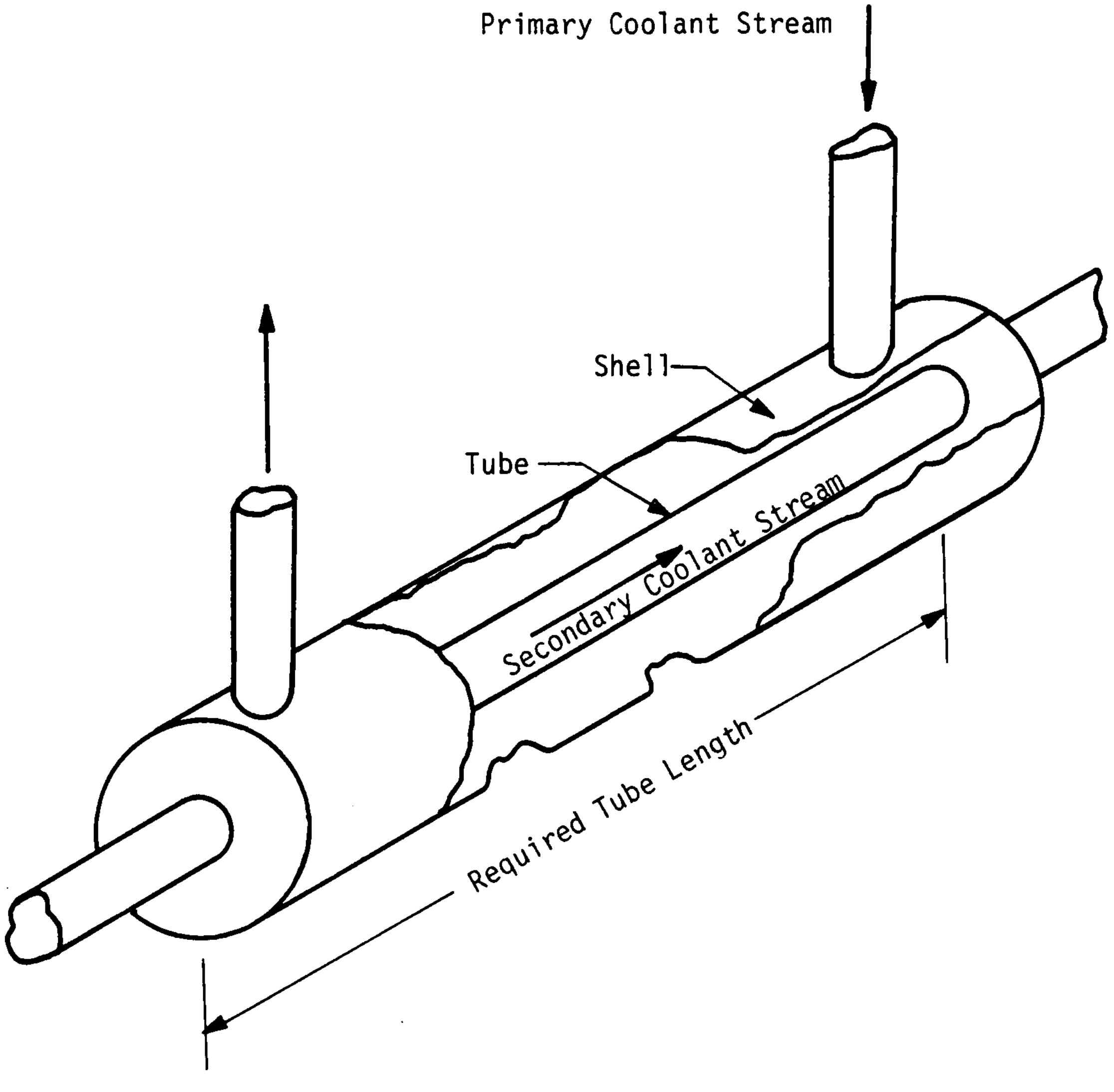



Figure 22. Typical Single Pass Shell and Tube Counterflow Heat Exchanger

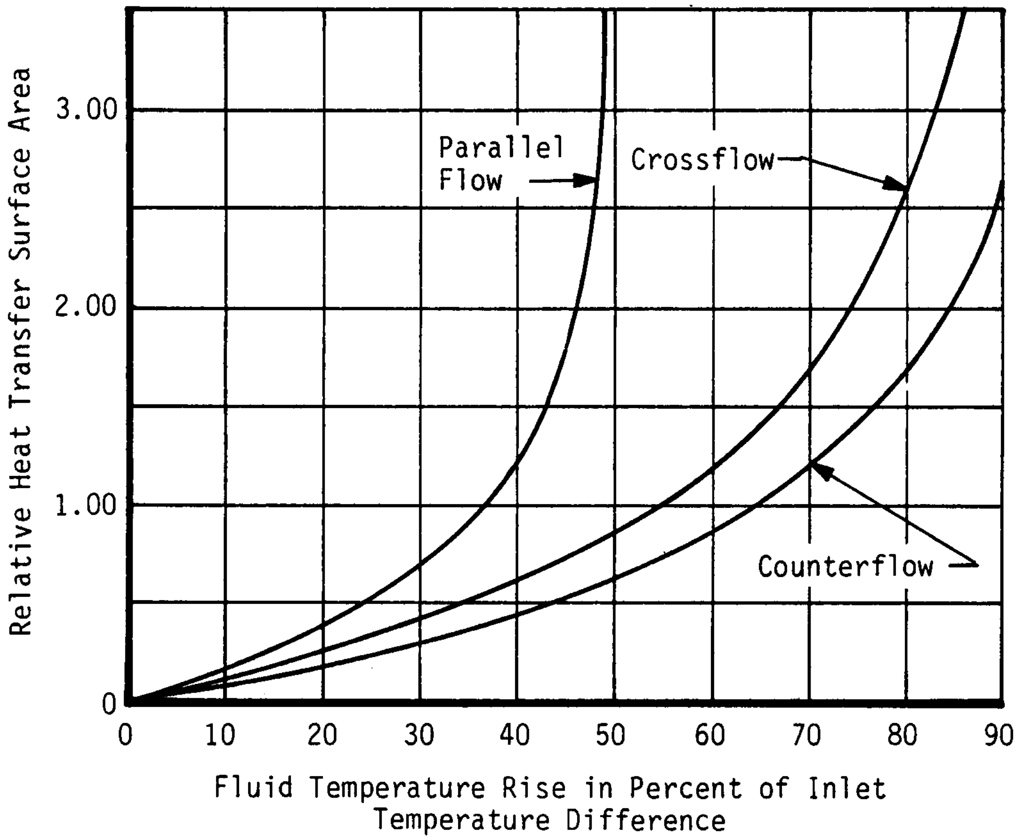



Figure 23. The Relative Heat Transfer Surface Area Required as a Function of the Ratio of the Temperature Rise in the Fluid Stream Having the Greater Change in Temperature to the Difference in Temperature Between the Inlet Streams

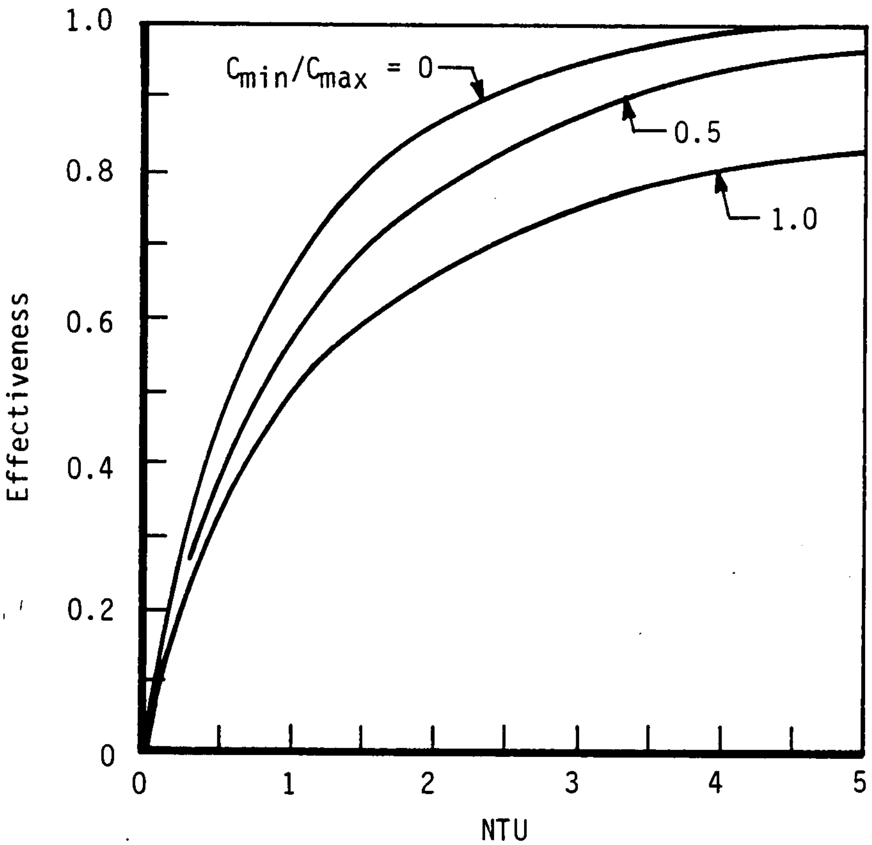



Figure 24. The Effectiveness of a Counterflow Heat Exchanger as a Function of $C_{min}$, $C_{max}$, and NTU

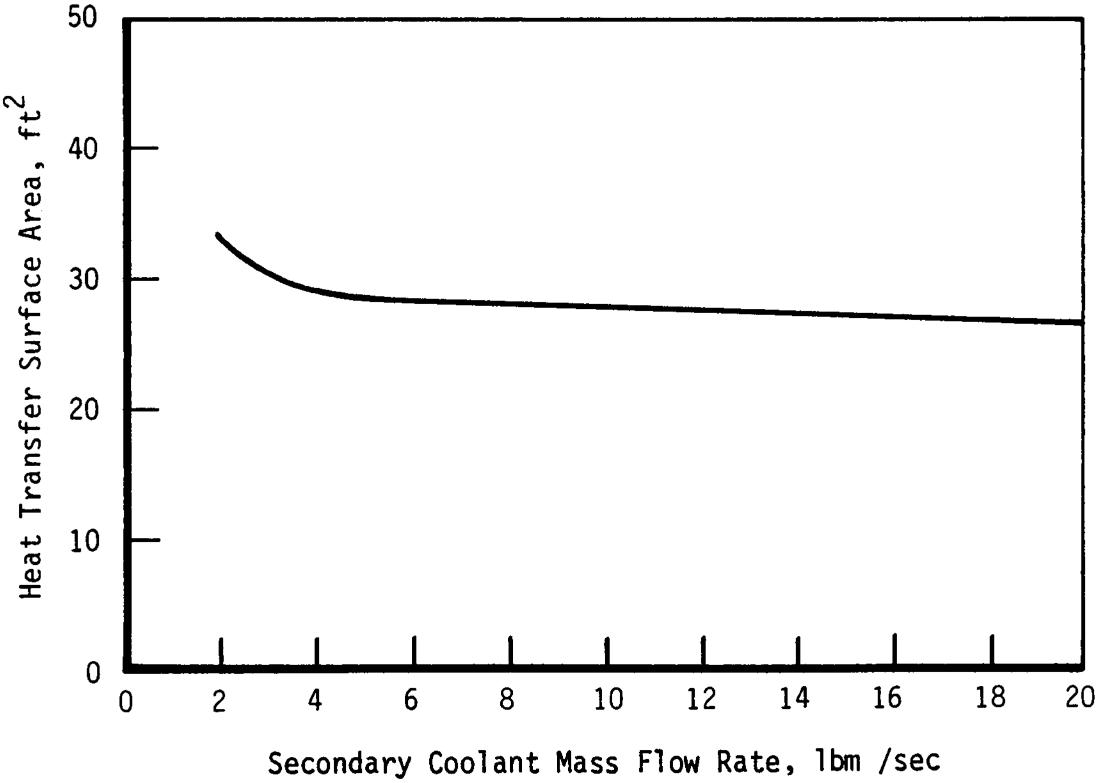



Figure 25. Heat Transfer Surface Area Required as a Function of Secondary Coolant Mass Flow Rate for a 1-Inch Tube with a Shell Inside Diameter Equal to 2 Inches

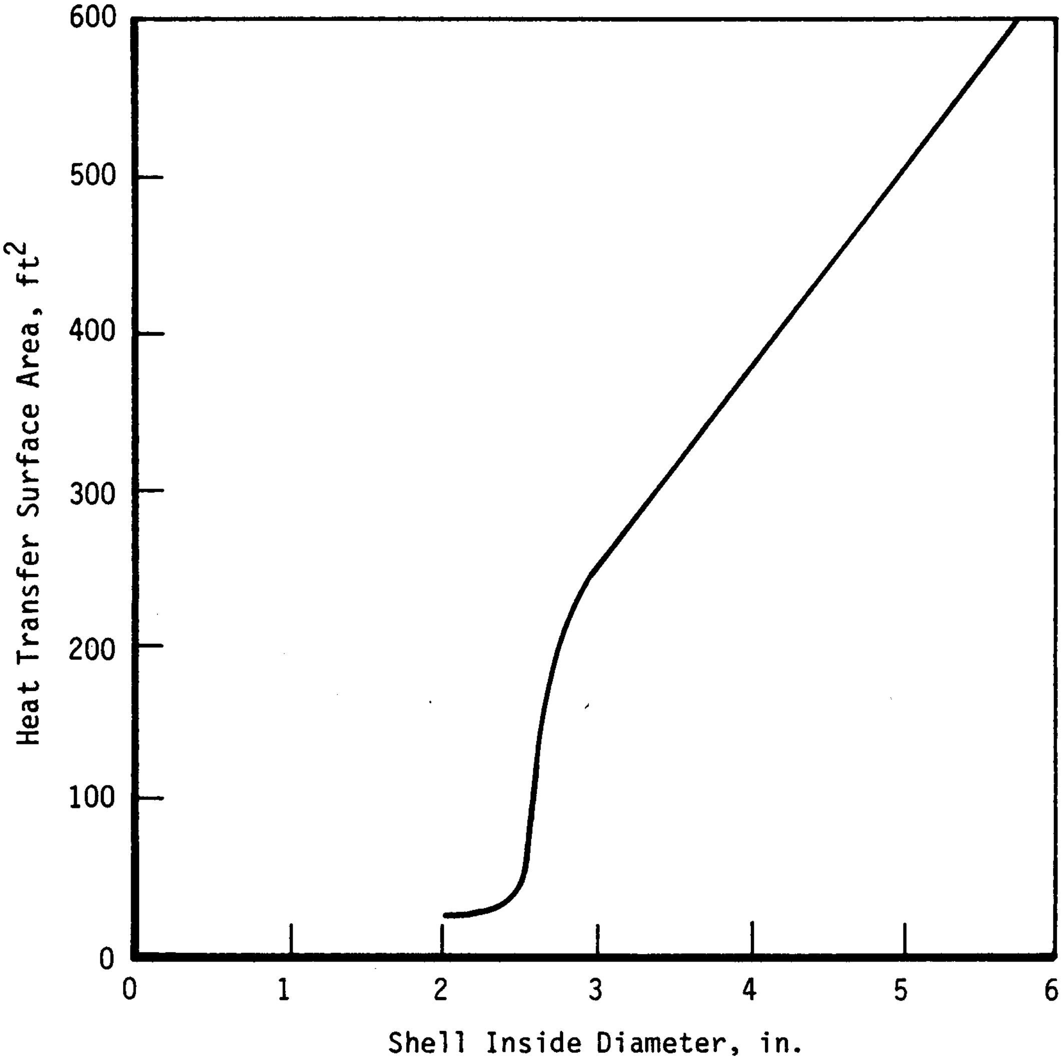



Figure 26. Heat Transfer Surface Area Required as a Function of Shell Inside Diameter for a 1-Inch Tube with a Tube Mass Flow Rate Equal to 16 lbm/sec

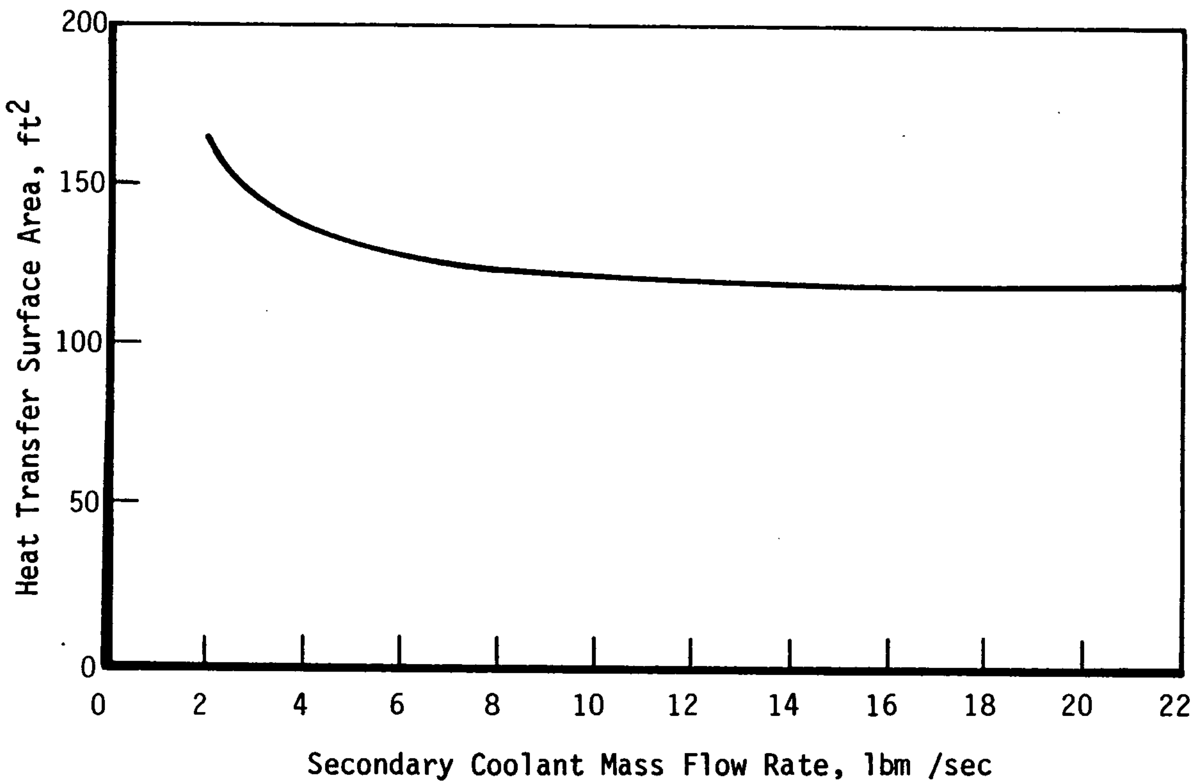



Figure 27. Heat Transfer Surface Area Required as a Function of Secondary Coolant Mass Flow Rate for a 2-Inch Tube and a Shell Inside Diameter Equal to 3 Inches

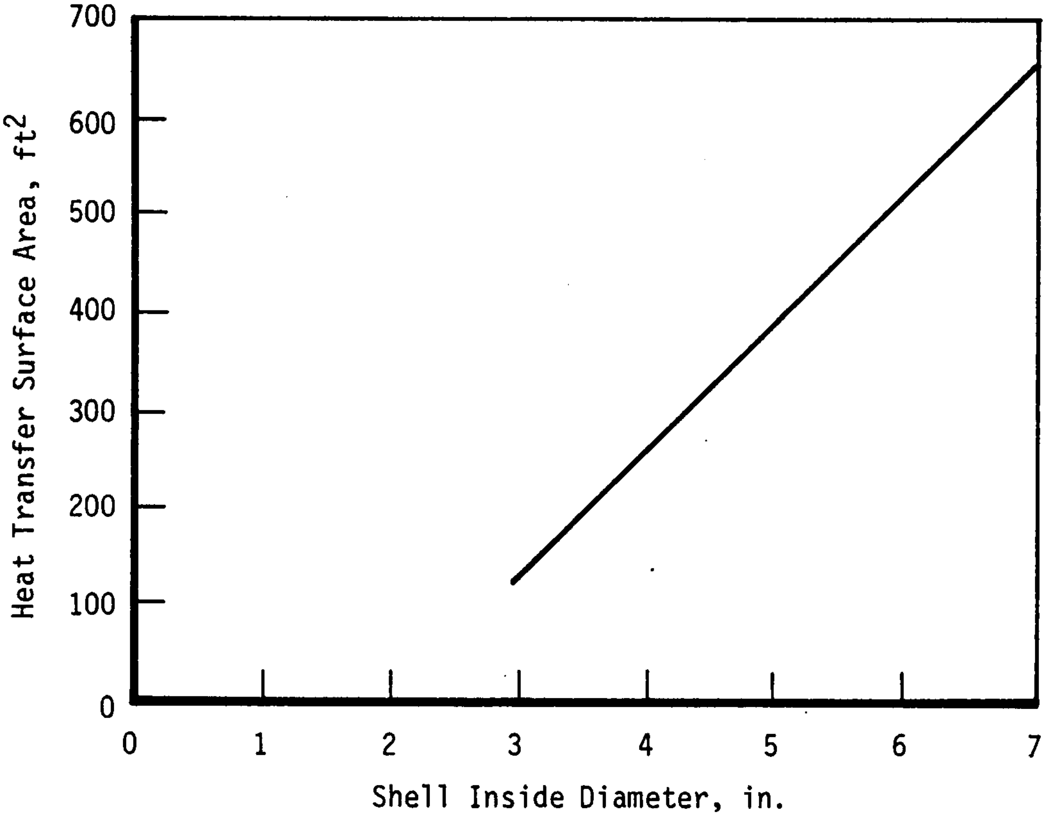



Figure 28. Heat Transfer Surface Area Required as a Function of Shell Inside Diameter for a 2-Inch Tube with a Tube Mass Flow Rate Equal to 16 lbm/sec

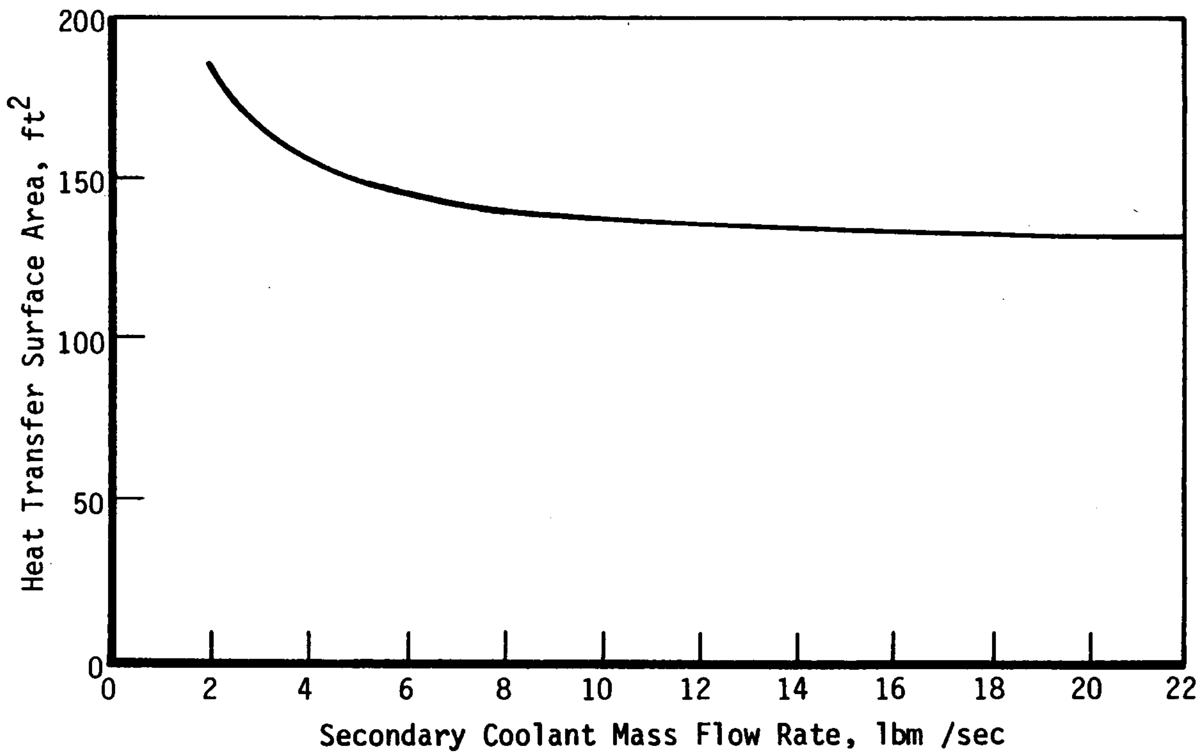



Figure 29. Heat Transfer Surface Area Required as a Function of Secondary Coolant Mass Flow Rate for a 4-Inch Tube and a Shell Inside Diameter Equal to 5 Inches

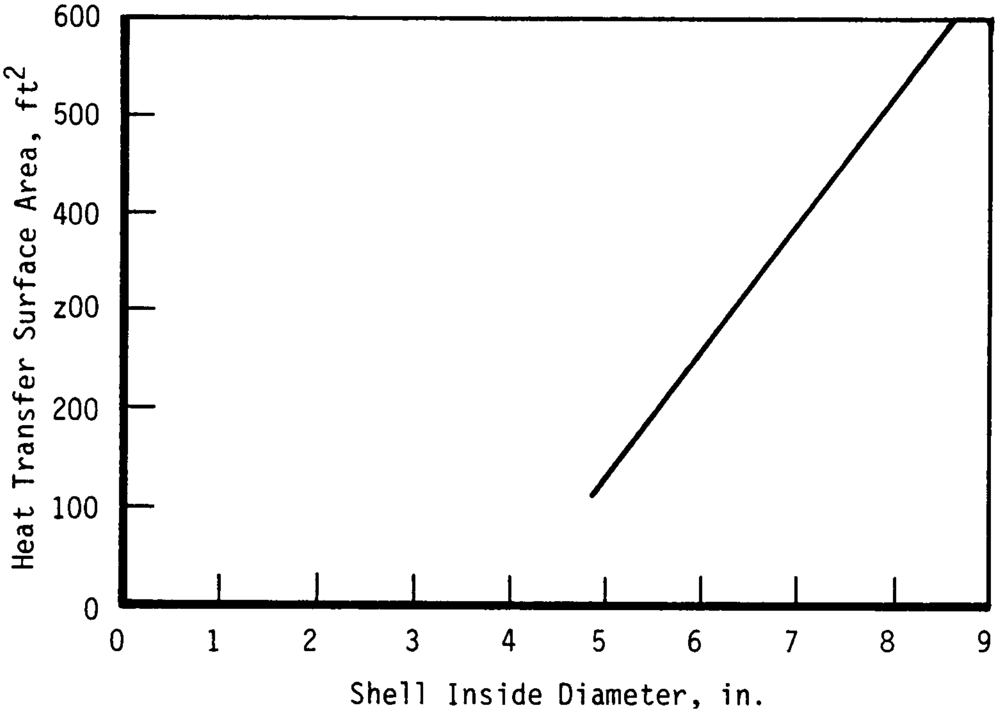



Figure 30. Heat Transfer Surface Area Required as a Function of Shell Inside Diameter for a 4-Inch Tube and a Tube Mass Flow Rate Equal to 16 lbm/sec

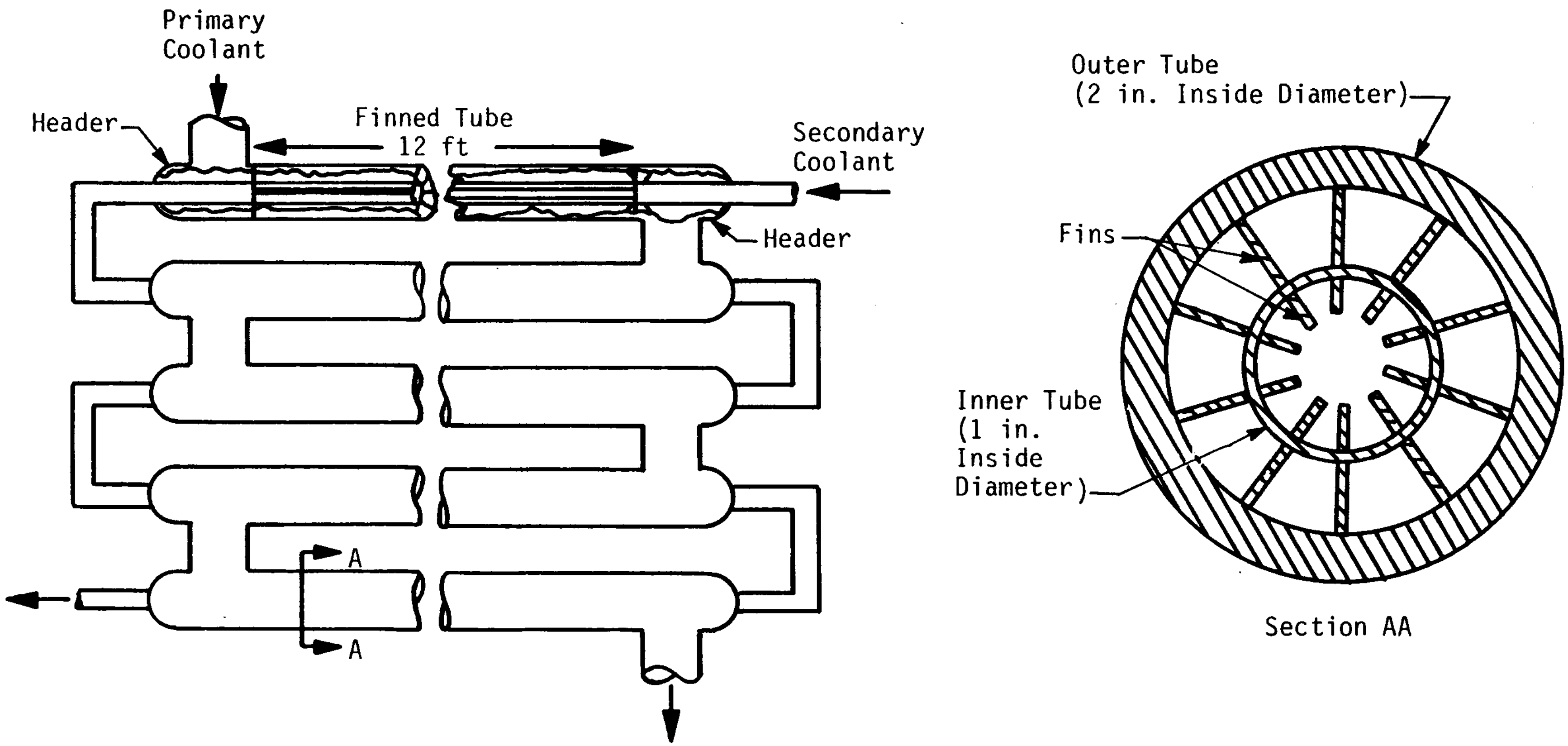



Figure 31. The Multipass Counterflow Heat Exchanger with a Finned Tube

# APPENDIX B

# TREATMENT OF AIR AS AN IDEAL GAS FOR HEAT TRANSFER ANALYSIS

Air can be approximated as an ideal gas if the compressibility factor is within a few percent of one. Figure B1 shows the compressibility factor as a function of the temperature ratio, $T_j$, and the pressure ratio, $P_j$, where:

$$T_j = T/T_{cr} \tag{B-1}$$

$$P_j = P/P_{cr} \tag{B-2}$$

The critical temperature, $T_{cr}$, and critical pressure, $P_{cr}$, are defined for air as:

$$T_{cr} = 132.41°K \tag{B-3}$$

$$P_{cr} = 37.25 \text{ atm.} \tag{B-4}$$

Considering the complete operating range, the ratio of static pressure to critical pressure decreases, as Mach number increases from zero, faster than the ratio of static temperature to critical temperature decreases. Therefore, the affect of temperature on the compressibility factor vanishes. The compressibility factor is nearly equal to one for the range of static pressures from venturi inlet to throat plane.

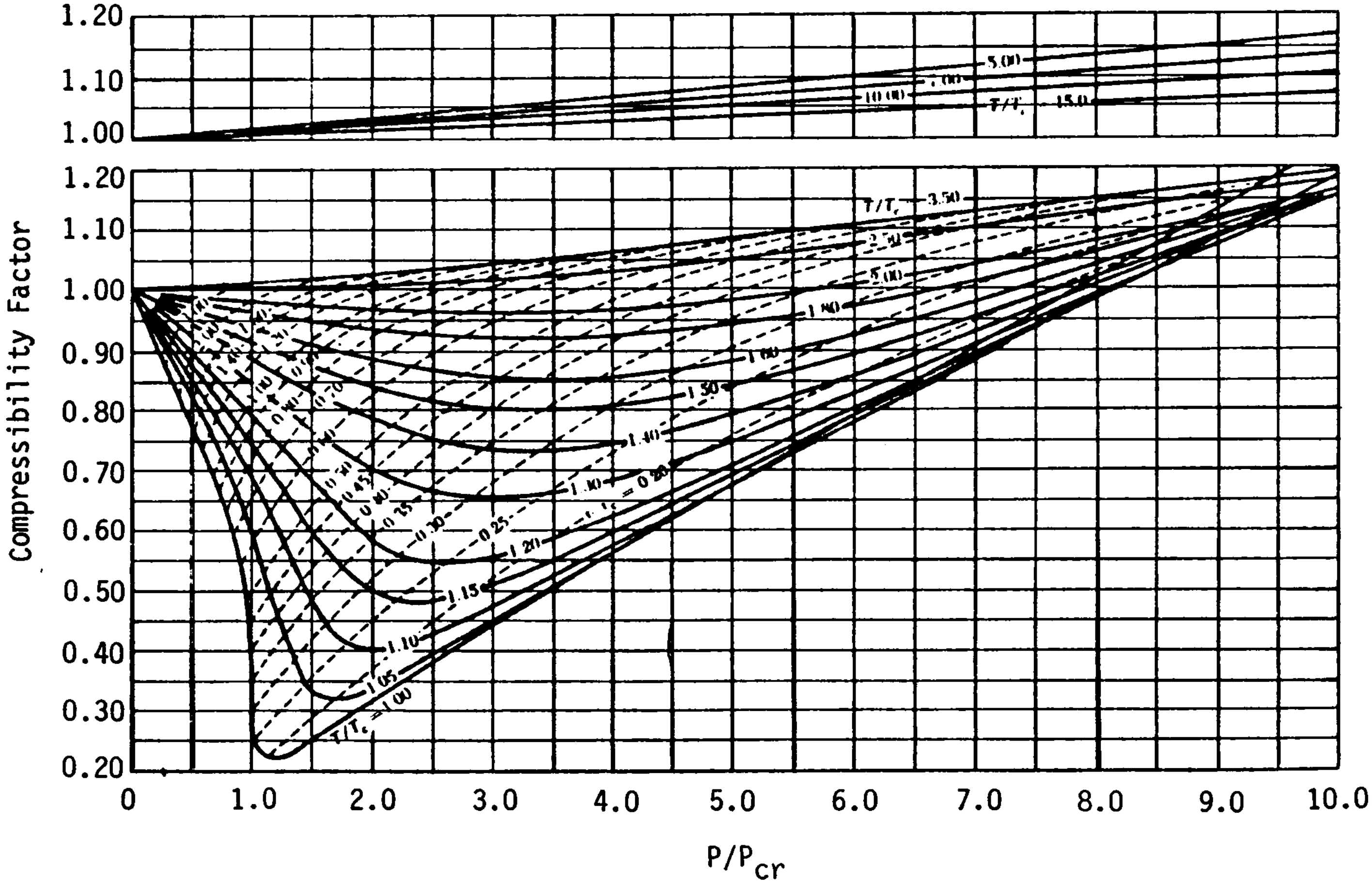



Figure B1. Compressibility Chart

# APPENDIX C

## PROCEDURE FOR CALCULATING MASS FLOW RATE FOR AN IDEAL GAS

The flow rate of air through the venturi can be expressed in terms of total temperature and total pressure by using the continuity equation in combination with the ideal gas law. Given the continuity of mass equation:

$$m_a = C_D \rho_a A_c V_a \tag{C-1}$$

density is substituted as follows:

$$\rho_a = \frac{P}{R_a T} \tag{C-2}$$

to give:

$$m_a = \frac{C_D P A_{c,v} V_a}{R_a T} \tag{C-3}$$

Rewriting the velocity in terms of the Mach number:

$$V_a = M\sqrt{\gamma g_c R_a T} \tag{C-4}$$

and the continuity of mass is:

$$M_a = C_D P A_{c,v} M \sqrt{\frac{\gamma g_c}{R_a T}} \tag{C-5}$$

The static temperature and static pressure can be replaced by total conditions with the following relationships:

$$T_t = T \left( 1 + \frac{\gamma - 1}{2} M^2 \right) \tag{C-6}$$

$$P_t = P \left( 1 + \frac{\gamma - 1}{2} M^2 \right)^{\frac{\gamma}{\gamma - 1}} \tag{C-7}$$

The continuity of mass is now:

$$m_a = \frac{C_D \; P_t \; A_{c,v} \; M}{\left(1 + \frac{\gamma-1}{2} M^2\right)^{\gamma/(\gamma-1)}} \sqrt{\frac{\gamma \; g_c \left(1 + \frac{\gamma-1}{2} M^2\right)}{R_a \; T_t}} \tag{C-8}$$

At the venturi throat, it is assumed that the Mach number is equal to one; the continuity equation reduces to

$$m_a = C_D \sqrt{\frac{\gamma \; g_c}{R_a}} \left(\frac{\gamma+1}{2}\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}} \frac{P_t \; A_{c,v}}{\sqrt{T_t}} \; . \tag{C-9}$$

APPENDIX D

PROPERTIES OF AQUEOUS ETHYLENE GLYCOL, WATER,

AIR, TYPE 310 STAINLESS STEEL, AND COPPER

Table D1. Properties of Aqueous Ethylene Glycol, Water, Air, Type 310 Stainless Steel, and Copper

| Medium | Thermal Conductivity (Btu/hr-ft-°F) | Specific Heat (Btu/lbm-°F) | Viscosity (centipoise) | Density ($lbm/ft^3$) |
|---|---|---|---|---|
| Ethylene Glycol | 0.245 | Figure D1 | Figure D2 | 66.1 |
| Water | Figure D3 | 1.0 | Figure D4 | 62.4 |
| Air | 0.0259 | 0.255 | 0.0305 | ---- |
| Type 310 Steel | 9.0 | 0.11 | ---- | 488 |
| Copper | 231 | ---- | ---- | ---- |

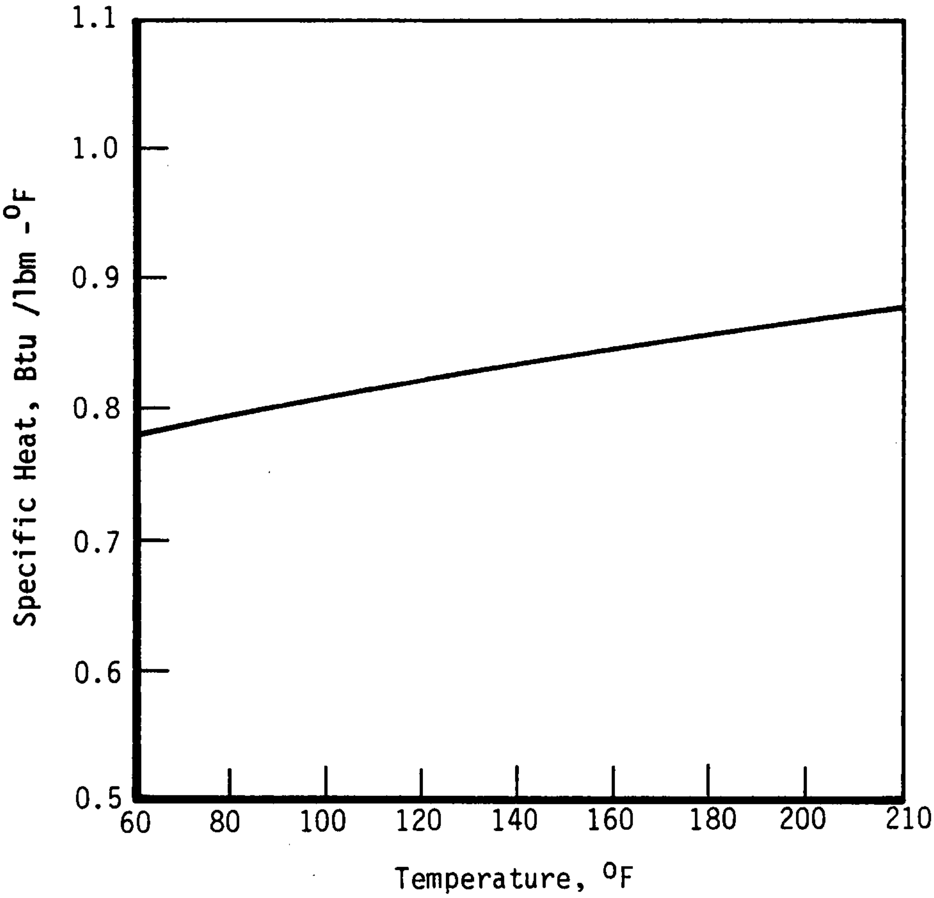



Figure D1. Specific Heat of 50 Percent by Weight Aqueous Ethylene Glycol

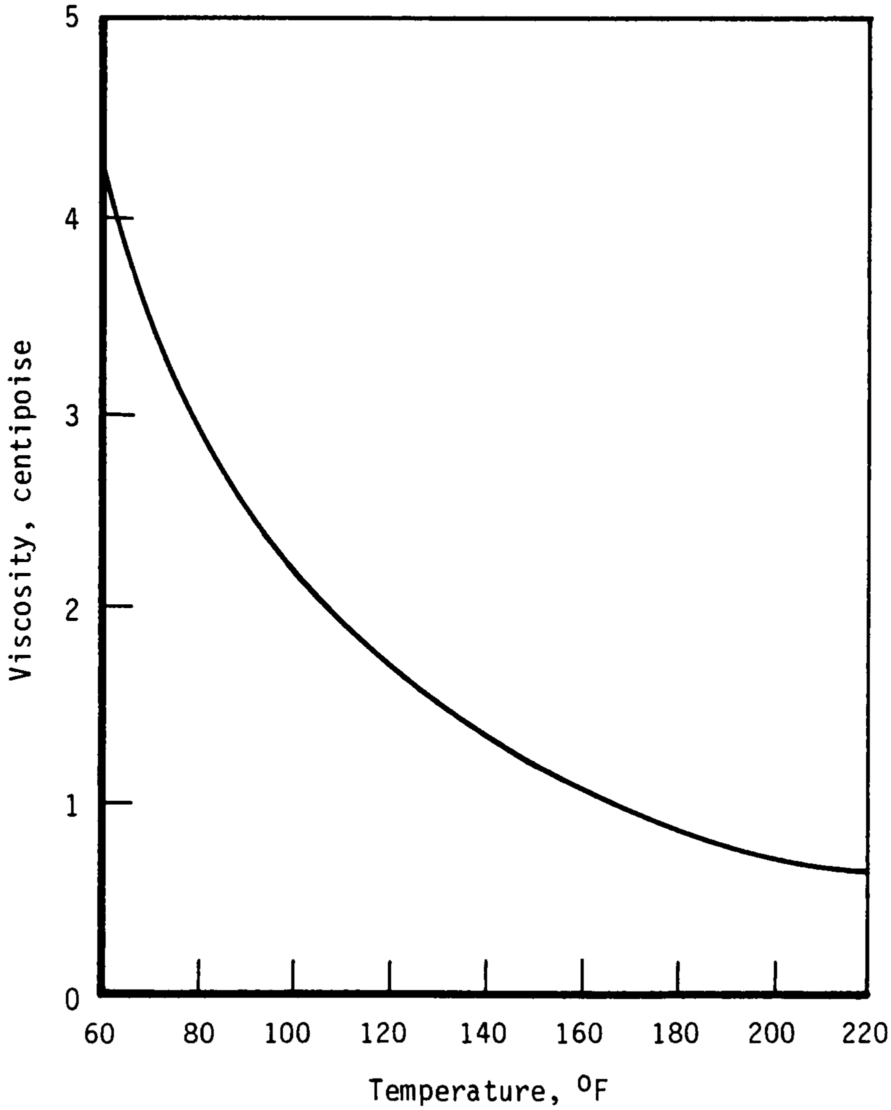



Figure D2. Viscosity of 50 Percent by Weight Aqueous Ethylene Glycol

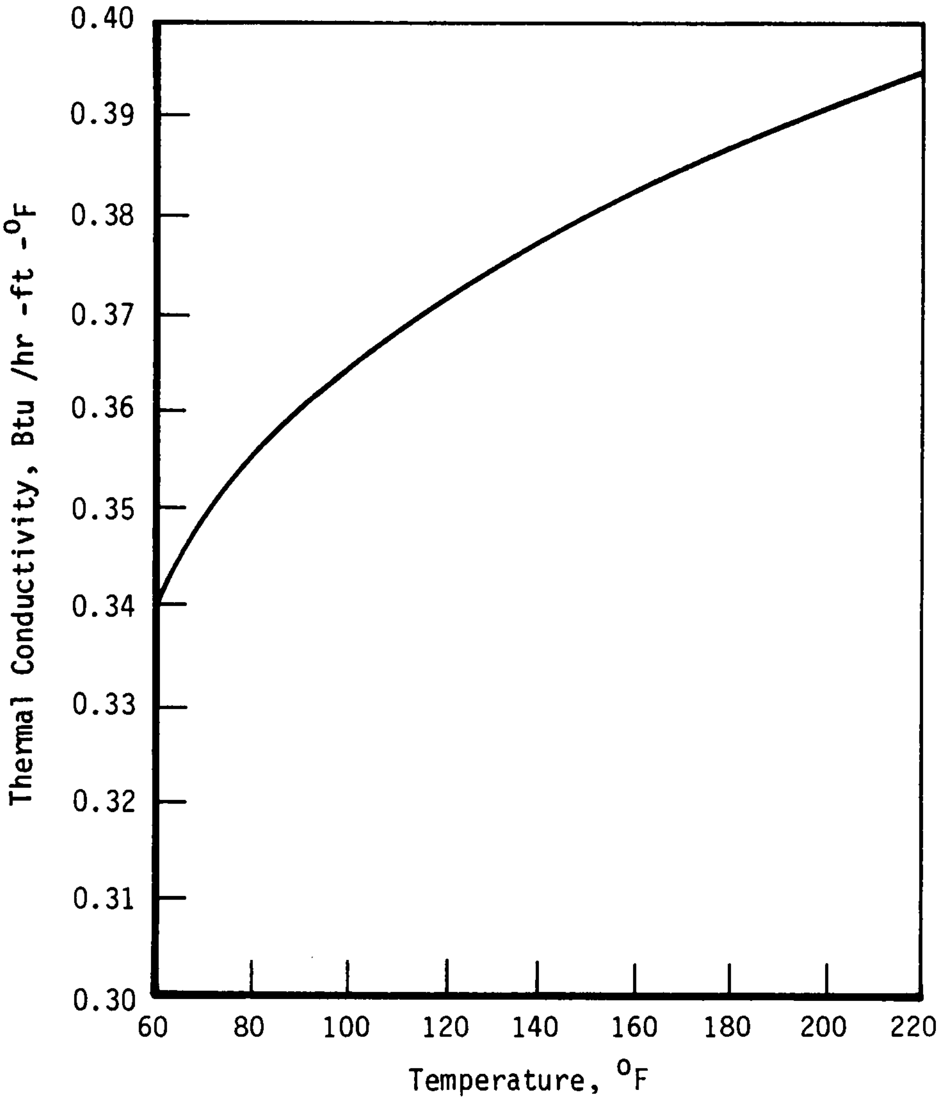



Figure D3. Thermal Conductivity of Water as a Function of Temperature

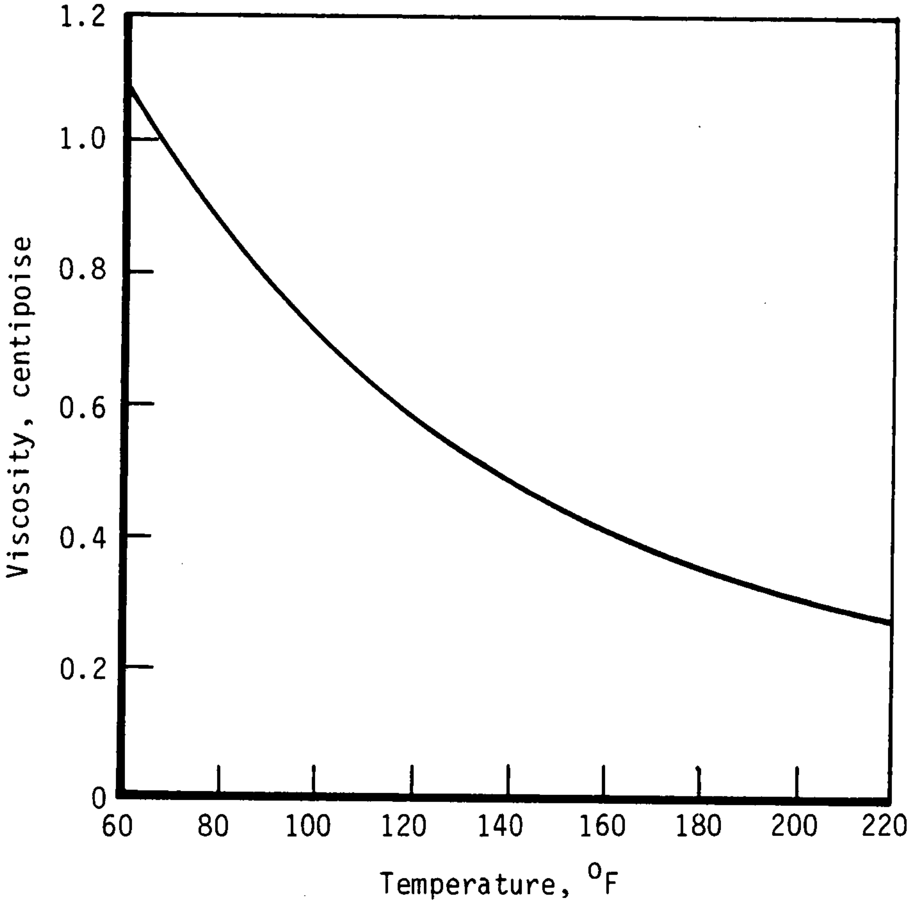



Figure D4. The Viscosity of Water as a Function of Temperature

APPENDIX E

CALCULATION OF CONTACT RESISTANCE

As defined by Ozisik and Hughes, the contact resistance is:

$$R_{con} = \frac{1}{(1-n_s)\,\frac{K_f}{\alpha} + 2\,n_s\,K/(\pi r_m)} \qquad \text{(E-1)}$$

where $n_s$ = (actual contact area of solid)/(total area of the joint)

$K_f$ = apparent fluid conductivity

$\alpha$ = mean thickness of the fluid layer

$K$ = thermal conductivity of solids

$r_m$ = mean radius of contact spots.

The fraction of solid surfaces in actual contact depends on the type of material, type of surface finish, and the applied load. As specified for the venturi, the interfacing surfaces have a ground surface finish (for Type 310 steel, roughness is 50 x $10^{-6}$ inches) and are tightened to a pressure greater than 1000 psi (as required by specifications). From Figure E1, $n_s$ can be estimated to be equal to 0.0135.

The apparent thermal conductivity of the fluid filling the voids, which is air, is taken as the thermal conductivity of the fluid at the mean temperature of the interface. Assume that the temperature at the interface is 450°F, equivalent to the maximum operating temperature of the O-ring seals. At this temperature, the thermal conductivity (i.e. apparent fluid thermal conductivity) is equal to 0.02355 Btu/hr-ft-°F.

Ozisik and Hughes suggest that the mean thickness of the fluid layer, $\alpha$, is the root mean square value of the roughness of the two surfaces in contact. This method generally underestimates the mean distance due to high spots. Therefore, a constant of multiplication is used to correct this. Ozisik and Hughes recommend a value equal to seven as an appropriate correction. The mean thickness of the fluid layer is defined as:

$$\alpha = \sqrt{(50 \times 10^{-6})^2 + (50 \times 10^{-6})^2} \qquad \text{(E-2)}$$

The thermal conductivity of each solid has been previously defined as equal to 9 Btu/hr-ft-°F.

The mean radius, $r_m$, is a function of contact pressure as shown in Figure E2. At an interface pressure equal to 1000 psi, the radius is equal to $98 \times 10^{-6}$ inches.

The contact resistance is calculated to be equal to $0.662 \times 10^{-3}$ hr-ft$^2$-°F/Btu.

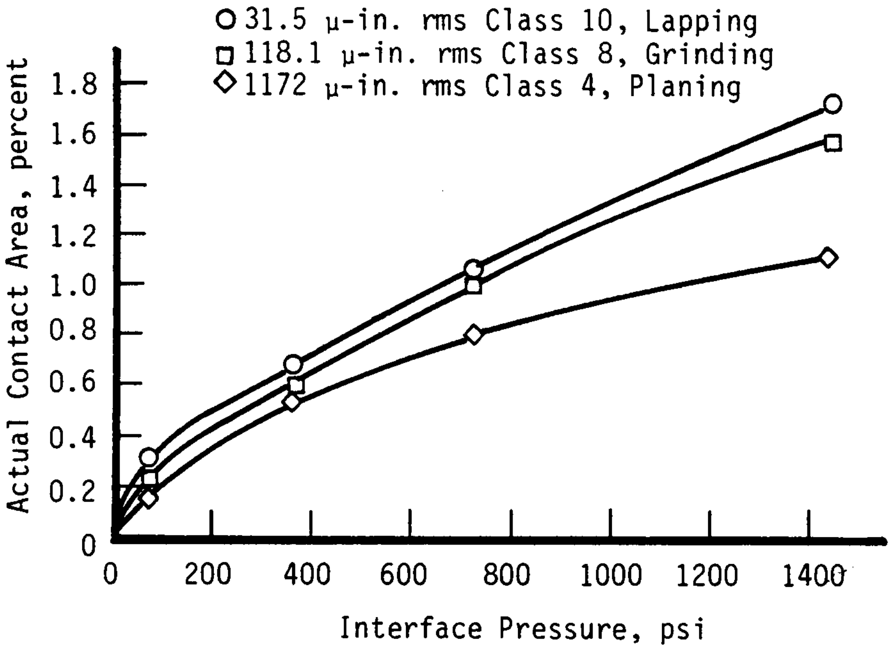



Figure E1. Actual Contact Area as a Function of Interface Pressure

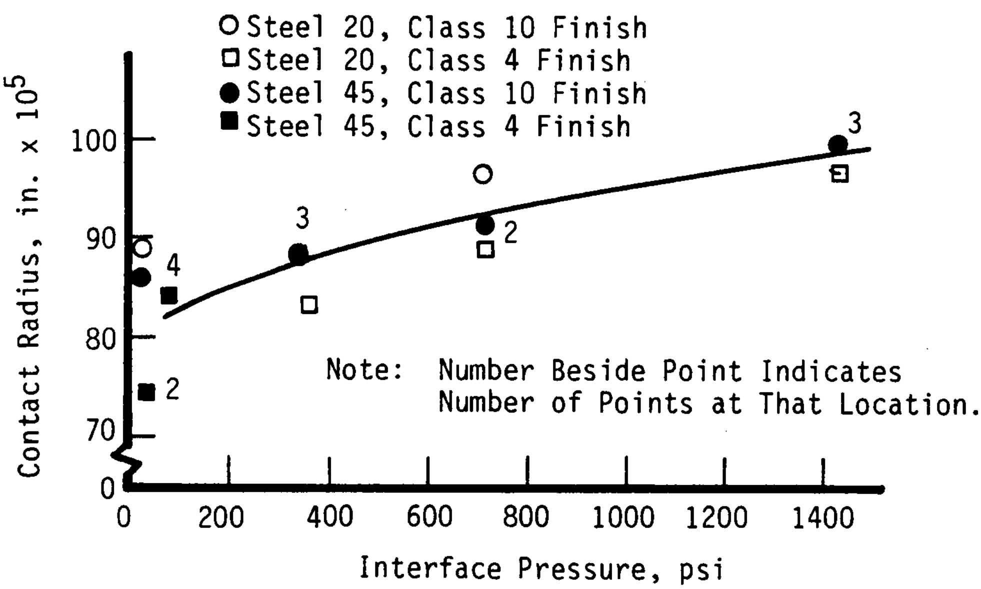



Figure E2. Contact Radius Estimated Interface Pressure

# APPENDIX F

# REPRESENTATION OF CONTACT RESISTANCE IN THE MODEL

The contact resistance is equal to:

$$R_{con} = 0.662 \times 10^{-3} \text{ hr-ft}^2\text{-°F/Btu} \qquad \text{(F-1)}$$

as defined in Appendix E. This is represented in the model by a thin strip of elements, 0.01 inches thick, with a thermal conductivity equivalent to the resistance of heat flow over the width of the element. The resistance per foot is the reciprocal of conductivity. The thermal conductivity is then equal to:

$$K_{eq} = \frac{w}{R_{con}} \qquad \text{(F-2)}$$

and is equal to a value of 1.34 Btu/hr-ft-°F.

# APPENDIX G

## TEMPERATURE CONTOURS WITH A COOLANT BULK TEMPERATURE OF 110°F

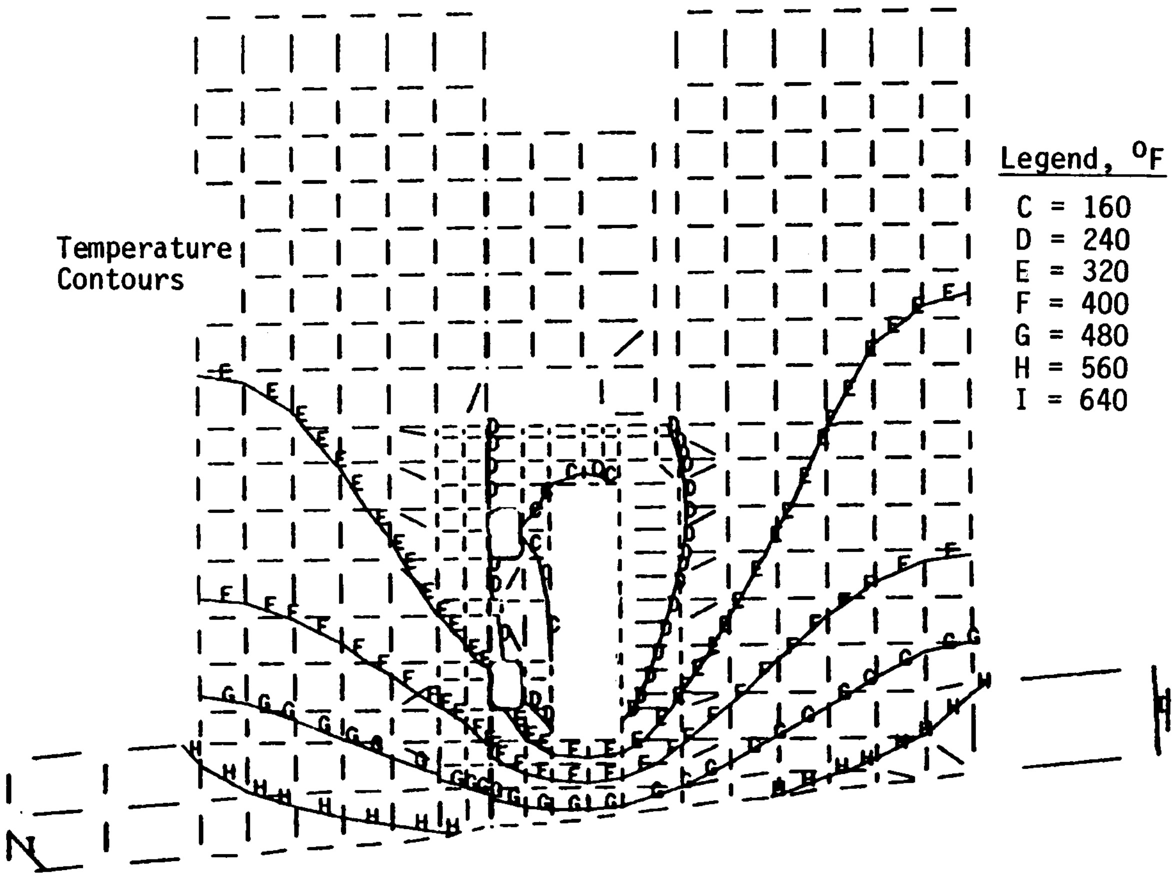



Figure G1. Temperature Contours of the Venturi Flanges with a Coolant Bulk Temperature of 110°F

APPENDIX H

METHODOLOGY FOR CALCULATING THE REQUIRED HEAT TRANSFER AREA

The methodology for calculating the required heat transfer area given:

1. inlet temperature of hot fluid
2. outlet temperature of hot fluid
3. mass flow rate of hot fluid
4. heat exchanger geometry
5. inlet temperature of cold fluid
6. mass flow rate of cold fluid

is as follows:

1. calculate the total heat transferred from the hot fluid:

$$q_t = m_h C_{p,h}(T_{h,in} - T_{h,out}) \tag{H-1}$$

2. calculate the maximum heat transfer possible:

$$q_{max} = C_{min} \, (T_{h,in} - T_{c,in}) \tag{H-2}$$

3. calculate the effectiveness:

$$\varepsilon = \frac{q_t}{q_{max}} \tag{H-3}$$

4. calculate the number of transfer units:

$$NTU = f(\varepsilon, \, C_{min}, \text{ and } C_{max}) \tag{H-4}$$

5. calculate the overall heat transfer coefficient:

$$U_{tot} = \frac{1}{\frac{1}{h_{id}} + R_{g,id} + \frac{r_{id}}{K} \ln \frac{r_{od}}{r_{id}} + \frac{r_{id}}{r_{od}} R_{g,od} + \frac{r_{id}}{r_{od}} \frac{1}{h_{od}}} \tag{H-5}$$

6. calculate the area:

$$A = \frac{NTU\ C_{min}}{U_{tot}} \tag{H-6}$$

The heat transfer coefficient for a circular tube is:

$$h = 0.027\ Re_D^{0.8}\ Pr^{1/3}\ K/D\ (\mu/\mu_e)^{0.14} \tag{H-7}$$

assuming turbulent flow. The heat transfer for the concentric tube annulus is defined by the following:

$$h = 0.023\ Re_{D,h}^{0.8}\ Pr^{1/3}\ (K)/D_h \tag{H-8}$$

where $D_h$ is defined as the hydraulic diameter of the annulus

$$D_h = D_o - D_i \tag{H-9}$$

All properties are evaluated at the mean fluid temperature.

APPENDIX I

HEAT EXCHANGER OPTIMIZATION PROGRAM LISTING

```
C     THIS PROGRAM IS DESIGNED TO CALCULATE THE REQUIRED SURFACE AREA
C     OF A SHELL AND TUBE HEAT EXCHANGER KNOWING THE FOLLOWING:
C
C     MTUB=TUBE FLUID MASS FLOWRATE (lbm/sec)
C     TTUBIN=TUBE FLUID INLET TEMP (deg F)
C     TTUBOT=TUBE FLUID OUTLET TEMP (deg F)
C     RITUB=INSIDE RADIUS OF TUBE (inches)
C     MSHEL=SHELL FLUID MASS FLOWRATE (lbm/sec)
C     TSHELI=SHELL FLUID INLET TEMPERATURE (deg F)
C     ROTUB=OUTSIDE RADIUS OF TUBE (inches)
C     RISHEL=INSIDE RADIUS OF SHELL (inches)
C     KMATL=THERMAL CONDUCTIVITY OF MATERIAL (BTU/hr/ft/deg F)
C
C     OTHER INPUTS ARE :
C     RFOULS=THE FOULING FACTOR FOR THE OUTTER SURFACE OF THE TUBE
C     RFOULT=THE FOULING FACTOR FOR THE INNER SURFACE OF THE TUBE
C
C
C
C     ***************************************************************
C
      REAL L1,MTUB,MSHEL,KTUB,KSHEL,HTUB,HSHEL,NTU,NUTUB,NUSHEL,KMATL
      REAL LS,NT,HIDEAL,LC,NSS,NC,NP,NCW,NB,LSISTR,LSOSTR,LSI,LSO,NX
      REAL HSHEL2,L2,LMTD,JI,JC,JL,JB,JRSTAR,JR,JS
C
C
      OPEN(4,FILE='INPUT.DAT',STATUS='OLD')
      READ(4,5)RUN
    5 FORMAT(F4.1)
      READ(4,10)MSHEL,TSHELI,TSHELO,RISHEL,ROSHEL
   10 FORMAT(5F10.5)
      READ(4,15)MTUB,TTUBIN,ROTUB,RITUB,RFOULS,RFOULT,KMATL
   15 FORMAT(7F10.5)
C
C     **OPEN AN OUTPUT FILE**
      OPEN(8,FILE='OUTPUT.DAT',STATUS='NEW')
C
      DO 500 I=1,8
      mtub=20.0
      DO 501 J=1,5
C
C     **UNIT CONVERSIONS (inches to feet)**
      RITUB=RITUB/12.0
      ROTUB=ROTUB/12.0
      RISHEL=RISHEL/12.0
      DITUB=2.0*RITUB
      DOTUB=ROTUB*2.0
      DISHEL=RISHEL*2.0
      ROSHEL=ROSHEL/12.0
      DOSHEL=ROSHEL*2.0
C
```

```
C
C     **CONSTANTS**
      PI=3.14159265
C
C     *****************************************************************
C     *                                                               *
C     **CALCULATIONS FOR THE MULTI-PASS HEAT EXCHANGER ****************
C     *                                                               *
C     *****************************************************************
C     **FIND THE AVERAGE TEMPERATURE OF THE SHELL FLUID****************
      TSHELA=(TSHELI+TSHELO)/2.0
C
C     **PROPERTIES OF THE SHELL FLUID (50% by weight ethylene glycol
C       aqueous solution) VALID FROM 80 TO 110 deg F *****************
C
C     **VISCOSITY (lbm/ft/hr)
      USHEL=10.0**(((-0.00674484)*(TSHELA-50.0))+0.69897)*2.4191
C
C     **THERMAL CONDUCTIVITY (BTU/hr/ft/deg F)
      KSHEL=0.245
C
C     **SPECIFIC HEAT (BTU/lbm/deg F)
      CPSHEL=0.082/70.0*(TSHELA-80.0)+0.7975
C
C     **PRANDTL NUMBER**
      PRSHEL=USHEL*CPSHEL/KSHEL
C
C
C     **CALCULATE THE TOTAL HEAT TRANSFERRED (BTU/hr)**
      QTOT=MSHEL*CPSHEL*(TSHELI-TSHELO)*3600.0
C
C     **SPECIFIC HEAT OF TUBE FLUID (WATER) (BTU/lbm/deg F)**
      CPTUB=1.0
C
C     **FIND THE OUTLET TEMPERATURE OF THE TUBE FLUID**
      TTUBOT=QTOT/(MTUB*CPTUB*3600.0)+TTUBIN
C
C     **FIND THE AVERAGE TEMPERATURE OF TUBE FLUIDS**
      TTUBA=(TTUBIN+TTUBOT)/2.0
C
C     **PROPERTIES OF TUBE FLUID (WATER) VALID FROM 50 TO 110 deg F**
C
C     **VISCOSITY (lbm/ft/hr)
      IF(TTUBA.LE.70.0)THEN
      UTUB=(-0.02)*(TTUBA-50.0)+3.15
      ELSE
      UTUB=(-0.725)/30.0*(TTUBA-70.0)+2.375
      ENDIF
C
C     **THERMAL CONDUCTIVITY (BTU/hr/ft/deg F)**
```

```
      KTUB=0.00052*(TTUBA-50.0)+0.338
C
C     **PRANDTL NUMBER**
      PRTUB=UTUB*CPTUB/KTUB
C
C
C     **FIND CMIN AND CMAX (BTU/hr/deg F)**
      CPTUB2=MTUB*CPTUB*3600.0
      CPSHL2=MSHEL*CPSHEL*3600.0
      IF(CPTUB2.LE.CPSHL2)THEN
      CMIN=CPTUB2
      ELSE
      CMIN=CPSHL2
      ENDIF
      IF(CPTUB2.GE.CPSHL2)THEN
      CMAX=CPTUB2
      ELSE
      CMAX=CPSHL2
      ENDIF
C
C     **CALCULATE THE MAXIMUM HEAT TRANSFERRED (BTU/HR)**
      QMAX=CMIN*(TSHELI-TTUBIN)
C
C     **CALCULATE THE EFFECTIVENESS**
      EFF=QTOT/QMAX
C
C     **FIND CSTAR**
      CSTAR=CMIN/CMAX
C
C     **CALCULATE THE NUMBER OF TRANSFER UNITS**
      NTU=-(ALOG((EFF-1.0)/(CSTAR*EFF-1.0)))/(1.0-CSTAR)
C
C     **CALCULATE THE REYNOLDS NUMBER OF THE TUBE FLUID**
      RETUB=(4.0*MTUB)/(PI*DITUB*UTUB)*3600.0
C
C     **CHECK LAMINAR OR TURBULENT**
      IF(RETUB.LE.2300)THEN
      NUTUB=4.36
      ELSE
      NUTUB=0.027*RETUB**0.8*PRTUB**(1.0/3.0)
      ENDIF
C
C     **CALCULATE THE CONVECTION COEFFICIENT**
      HTUB=(NUTUB*KTUB)/(DITUB)
C
C     **CALCULATE THE AREA OF THE ANNULUS**
      DRATIO=DOTUB/DISHEL
C     **HYDRAULIC DIAMETER**
      DSHEL=DISHEL-DOTUB
      ASHEL=PI/4.0*(DISHEL**2-DOTUB**2)
```

```
C
C     **CALCULATE THE REYNOLDS NUMBER OF THE SHELL FLUID**
      RESHEL=(MSHEL*DSHEL)/(ASHEL*USHEL)*3600.0
C
C     **CHECK LAMINAR OR TURBULENT**
      IF(RESHEL.GT.2300.0)THEN
      NUSHEL=0.023*RESHEL**0.8*PRSHEL**0.4
      ELSE
      NUSHEL=4.36266*EXP(((ALOG(DRATIO)+17.59718482)**2)/
     &467.7259851)
      ENDIF
C
C     **CALCULATE THE CONVECTION COEFFICIENT FOR THE SHELL FLUID**
      HSHEL=(NUSHEL*KSHEL)/DSHEL
C
C     **CALCULATE THE OVERALL COEFFICIENT FOR THE SHELL FLUID**
      UTOT=1.0/(1.0/HTUB+RFOULT+RITUB/KMATL*ALOG(ROTUB/RITUB)+
     &(RITUB/ROTUB)*RFOULS+(RITUB/ROTUB)*(1.0/HSHEL))
C
C     **CALCULATE THE TOTAL SURFACE AREA REQUIRED**
      AREA=(NTU*CMIN)/UTOT1
C
C     **CALCULATE THE TOTAL LENGTH REQUIRED**
      Li=AREA/DITUB/PI
C
C     **UNIT CONVERSIONS (feet to inches)**
      RITUB=RITUB*12.0
      DITUB=RITUB*2.0
      ROTUB=ROTUB*12.0
      DOTUB=ROTUB*2.0
      RISHEL=RISHEL*12.0
      DISHEL=RISHEL*2.0
      ROSHEL=ROSHEL*12.0
      DOSHEL=ROSHEL*2.0
      AREA=AREA*144.0
      L1=L1*12.0
C
C
C     ***************OUTPUT FOR SINGLE-TUBE SINGLE PASS***************
      WRITE(8,100)RUN
  100 FORMAT(25S,'SOLUTION FOR RUN # ',F4.1////)
      WRITE(8,110)
  110 FORMAT(5X,'REQUIREMENTS:'//)
      WRITE(8,120)MTUB
  120 FORMAT(16X,'TUBE:      1. FLUID MASS FLOWRATE= ',F5.2,' lbm/sec'/)
      WRITE(8,130)TTUBIN
  130 FORMAT(25X,'2. FLUID INLET TEMPERATURE= ',F5.1,' deg F'/)
      WRITE(8,132)DITUB
  132 FORMAT(25X,'3.  INSIDE DIAMETER= ',F5.2,' in'/)
      WRITE(8,134)DOTUB
```

```
 134 FORMAT(25X,'4. OUTSIDE DIAMETER= ',F5.2,' in'/)
     WRITE(8,136)KMATL
 136 FORMAT(25X,'5. THERMAL CONDUCTIVITY= ',F6.2,' BTU/hr/ft/deg F'//)
     WRITE(8,140)MSHEL
 140 FORMAT(16X,'SHELL:  1. FLUID MASS FLOWRATE= ',F5.2,' lbm/sec'/)
     WRITE(8,150)TSHELI
 150 FORMAT(25X,'2. FLUID INLET TEMPERATURE= ',F6.2,' deg F'/)
     WRITE(8,160)TSHELO
 160 FORMAT(25X,'3. FLUID OUTLET TEMPERATURE=',F6.2,' deg F'/)
     WRITE(8,162)DISHEL
 162 FORMAT(25X,'4. INSIDE DIAMETER= ',F6.2,' in'/)
     WRITE(*,164)DOSHEL
 164 FORMAT(25X,'5. OUTSIDE DIAMETER= ',F6.2,' in'///)
     WRITE(8,170)
 170 FORMAT(5X,'FLUID PROPERTIES:'//)
     WRITE(8,180)USHEL
 180 FORMAT(10X,'SHELL FLUID:   1.  VISCOSITY= ',F6.4,' lbm/ft/hr '/)
     WRITE(8,190)CPSHEL
 190 FORMAT(25X,'2. SPECIFIC HEAT= ', F8.4,' BTU/lbm/deg F'/)
     WRITE(8,200)KSHEL
 200 FORMAT(25X,'3. THERMAL CONDUCTIVITY= ',F6.2,' BTU/hr/ft/deg F'/)
     WRITE(8,210)PRSHEL
 210 FORMAT(25X,'4. PRANDTL NUMBER= ',F6.2/)
     WRITE(8,220)UTUB
 220 FORMAT(11X,'TUBE FLUID:  1.  VISCOSITY= ',F8.4,' lbm/ft/hr'/)
     WRITE(8,230)CPTUB
 230 FORMAT(25X,'2. SPECIFIC HEAT= ',F4.1,' BTU/lbm/deg F'/)
     WRITE(8,240)KTUB
 240 FORMAT(25X,'3.  THERMAL CONDUCTIVITY= ',F8.4,' BTU/hr/ft/deg F'/)
     WRITE(8,250)PRTUB
 250 FORMAT(25X,'4. PRANDTL NUMBER= ',F5.2////)
     WRITE(8,260)
 260 FORMAT(5X,'HEAT EXCHANGER:'//)
     WRITE(8,270)QTOT
 270 FORMAT(25X,'TOTAL HEAT TRANSFERRED= ',F10.2,' BTU/hr'/)
     WRITE(8,272)TTUBOT
 272 FORMAT(25X,'TUBE FLUID OUTLET TEMPERATURE= ',F6.2,' deg F'/)
     WRITE(8,280)QMAX
 280 FORMAT(25X,'MAXIMUM HEAT TRANSFERRED= ',F10.2,' BTU/hr'/)
     WRITE(8,290)EFF
 290 FORMAT(25X,'EFFECTIVENESS= ',F6.4/)
     WRITE(8,300)AREA
 300 FORMAT(25X,'TOTAL SURFACE AREA REQUIRED= ',F10.2,' in**2'/)
     WRITE(8,310)L1
 310 FORMAT(25X,'TOTAL LENGTH REQUIRED= ',F10.2,' in'///////)
C    ****************************************************************
C
     RUN=RUN+1
     MTUB=MTUB+4.0
 501 CONTINUE
```

```
      RISHEL=RISHEL+0.25
      ROSHEL=RISHEL+0.10
  500 CONTINUE
      STOP
      END
```

APPENDIX J

PRESSURE DROP IN PIPE

The pressure drop in a given pipe section is:

$$\Delta p = FL\rho V^2/(2Dg_c) \qquad \text{(J-1)}$$

where $F$ = friction factor

$L$ = pipe length, 107 ft

$\rho$ = density of water, 62.4 lbm/sec

$V$ = fluid velocity

$D$ = pipe diameter, 1 inch

The fluid velocity is:

$$V = m/(\rho A) \qquad \text{(J-2)}$$

where $m$ = 4 lbm/sec

$\rho$ = 62.4 lbm/ft$^3$

$A$ = 0.0055 ft$^2$

and the velocity is equal to 11.5 ft/sec.

The friction factor, a function of the Reynolds number and the roughness of the copper tubing, is equal to 0.013 as interpolated from the Moody diagram. Given this, the pressure loss in the tubing is equal to 15 psia.

APPENDIX K

THE REQUIRED FINS FOR THE TUBE

The total heat transfer to and from the tube surface is defined by:

$$q_t = q_{fin} + q_{base} \qquad \text{(K-1)}$$

where

$$q_t = hA_t\Delta T \qquad \text{(K-2)}$$

$$q_{fin} = n_{fin}hA_{fin}\Delta TN \qquad \text{(K-3)}$$

$$q_{base} = hA_{base}\Delta T \qquad \text{(K-4)}$$

Note that the heat transfer from the fin is a function of the fin efficiency, $h_{fin}$, and the number of fins, N. For a rectangular fin, the efficiency is defined in Figure K1.

Simplifying, Equation (K-1) reduces to:

$$A_t = n_{fin}NA_{fin} + A_{base} \qquad \text{(K-5)}$$

For the given counterflow heat exchanger, the required total inner tube surface area is 28 $ft^2$. Therefore,

$$28\ ft^2 = n_{fin}N(2HL) + (D_{id} - NB)L \qquad \text{(K-6)}$$

For a given fin size (height, H, length, L, and thickness, B), the number of fins required to produce 28 $ft^2$ can be calculated (Figure K2).

The surface area required on the outer tube surface is 32 $ft^2$ as calculated using the program listed in Appendix I. As performed above, the fin height required to provide the surface area is 0.5 inches.

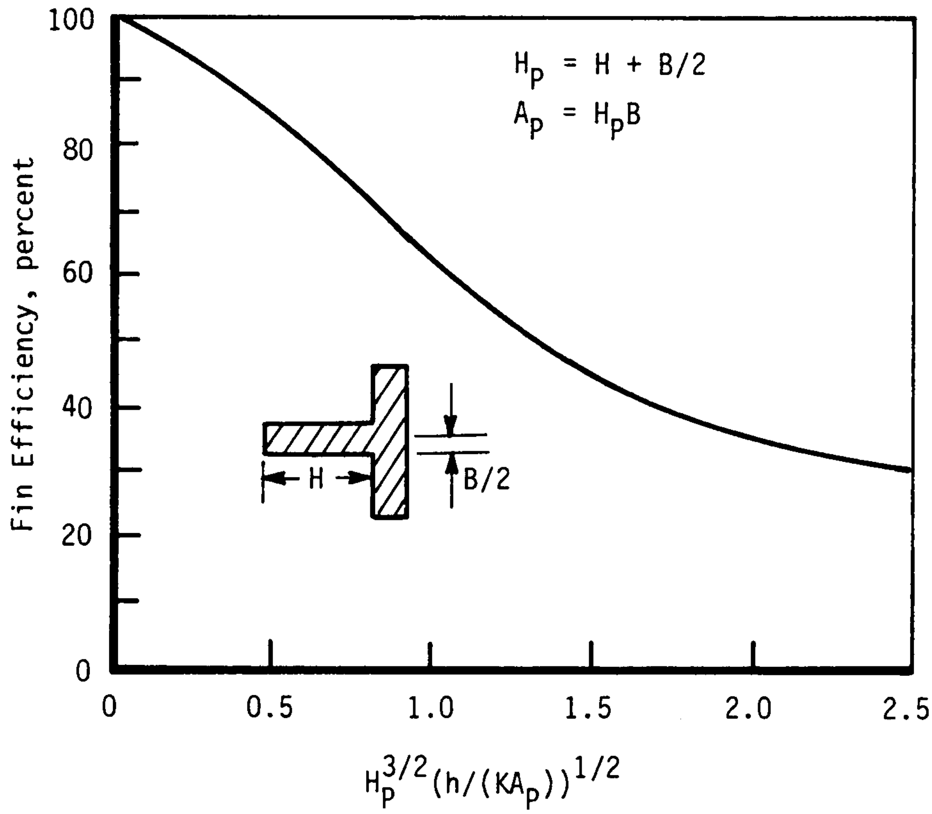



Figure K1. The Efficiency of a Rectangular Fin

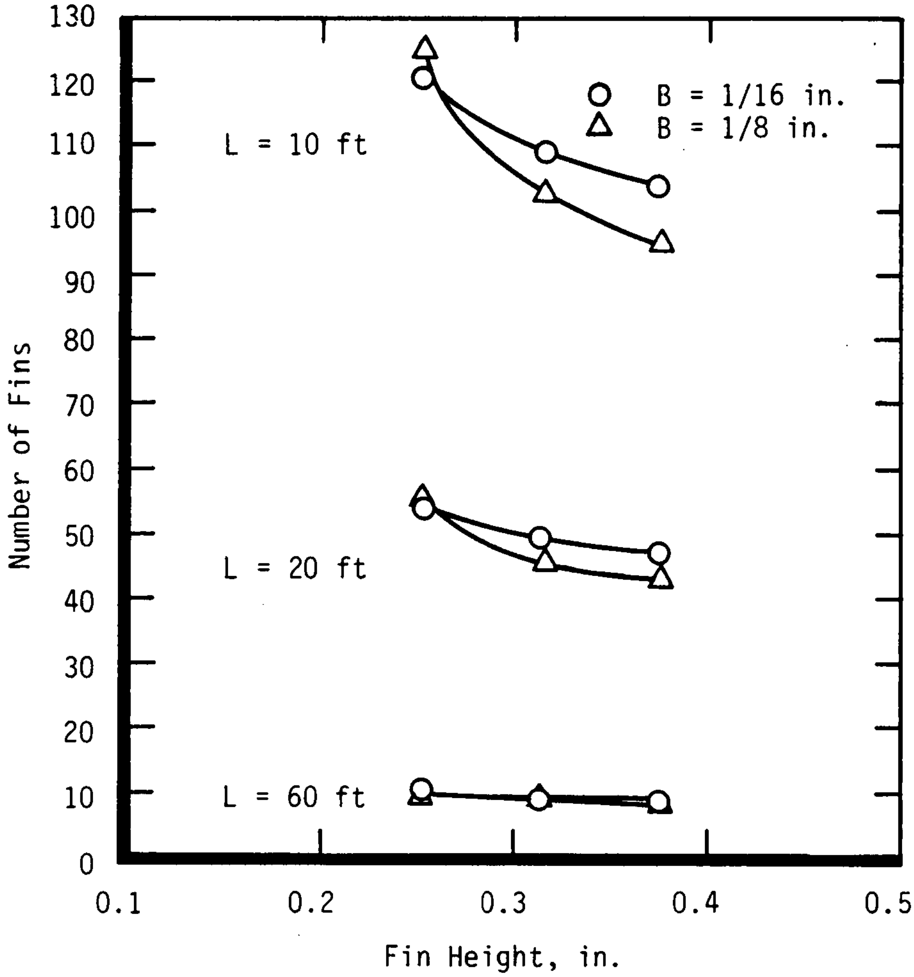



Figure K2. The Number of Fins Required (Along the Inner Tube Wall) as a Function of Fin Height to Provide 28 $ft^2$ of Heat Transfer Surface Area

## VITA

Robert Wood McAmis was born in Knoxville, Tennessee on February 24, 1962 to Charles and Betty Jo. He has one brother, Steve, and three sisters, Priscilla, Cathy and Julie. He attended Chilhowee Elementary School for eight years. After starting High School at Holston, he moved to Dandridge, Tennessee, as a freshman and graduated from Jefferson County High School in June 1980. In the fall of 1980, he entered Tennessee Technological University. Four years later, in June 1984, he graduated and accepted a job with Sverdrup Technology, Inc. at the Arnold Engineering Development Center. He was married to Mary K. Lee on September 8, 1984.

From the fall of 1984 until present, he has been a graduate student at the University of Tennesee Space Institute supported by the company.