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CONDUCTION INVESTIGATIONS INTO THE MAGNETIC PROPERTIES OF MATTER

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CONDUCTION INVESTIGATIONS INTO THE MAGNETIC PROPERTIES OF MATTER

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SUMMARY

Because of problems resulting from poor quality control of the magnetic samples required for the original study, the graduate research associate has been able to study a different problem. This report presents the thermal analysis of a heat exchanger for heating air to temperatures on the order of 3000°C for use in NASA's high temperature wind tunnel. It was ascertained that an externally finned shell-and-tube type of heat exchanger with counterflow could be considered for this application. The methods of estimating the convective heat transfer coefficients are outlined in this report. As a part of thermal design the required size of heat exchanger was predicted. As a result of extreme length of heat exchanger (≈ 1/2 km), it was determined that a conventional heat exchanger may not be the most suitable tool for this application.

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INTRODUCTION

A heat exchanger is any device which provides the transfer of thermal energy from one fluid to another. In the simplest type of heat exchangers the hot and cold fluids mix directly, as compared to more common types in which the fluids are separated by a wall. This type, called a recuperator, may vary in design from a simple plane wall between two flowing fluids to more complex configurations involving multiple passes, fins, or baffles (Ref. 1, 2). In this case conductive and convective and sometimes radiative modes of heat transfer must be considered.

Many factors including thermal analysis, size, weight, structural strength, pressure drop, and cost, contribute to the design of a heat exchanger. The heat transfer coefficient of each fluid is estimated by the geometry of the flow passages, fluid flow rates, temperatures and fluid properties. From these specified conditions an overall heat transfer coefficient is then calculated.

For our application it was determined to use an externally-finned shell and tube arrangement with counterflow to investigate the practicality of the task. Thermal analysis of this type of heat exchanger is presented in detail in the next section.

THERMAL ANALYSIS

The primary objective in thermal design of a heat exchanger is to determine the necessary surface area, A, required to transfer heat at a given rate for a given fluid temperature and flow rate.

One of the first tasks in the thermal analysis of a heat exchanger is to evaluate the total thermal resistance of the system. This is facilitated by employing the overall heat-transfer coefficient, U, defined as the reciprocal of the sum of the thermal resistances of the system. For a cylindrical wall it can be written as (Ref. 3)

$$U = \frac{1}{1/h_i + [r_i \ln(r_o/r_i)]/K + (\frac{r_i}{r_o})/h_o} \qquad (1)$$

here h is the film coefficient, K is the tube thermal conductivity, r is the radius of the tube, and the subscripts o and i represent the outside and the inside, respectively.

To determine the rate of heat transfer, Q, from the energy balance between the two fluids

$$Q = m\ C_p\ \Delta T)_{hot} = m\ C_p\ \Delta T)_{cold} \qquad (2)$$

where m is the mass flow rate of flow, C_p is the specific heat at constant pressure, and ΔT is the absolute value of the inlet and outlet temperature difference.

Heat exchangers are designed by the basic equation (Ref. 4)

$$Q = U\ A\ \overline{\Delta T}$$

For more complex type of heat exchangers, such as those involving multiple tubes, several shell passes or crossflow, the above equation is modified to include a correction factor (Ref. 4).

The remaining term, $\overline{\Delta T}$, in the above equation is the logarithmic mean temperature difference (LMTD). The temperatures of the fluids in a heat exchanger are generally not constant, but vary from point to point as heat flows. Even for a constant thermal resistance, the rate of heat flow varies along the path of the exchangers, since it depends on the temperature difference between the hot and cold fluid. Figure 1 illustrates the change in temperatures that may occur in a simple shell and tube counter flow heat exchanger.

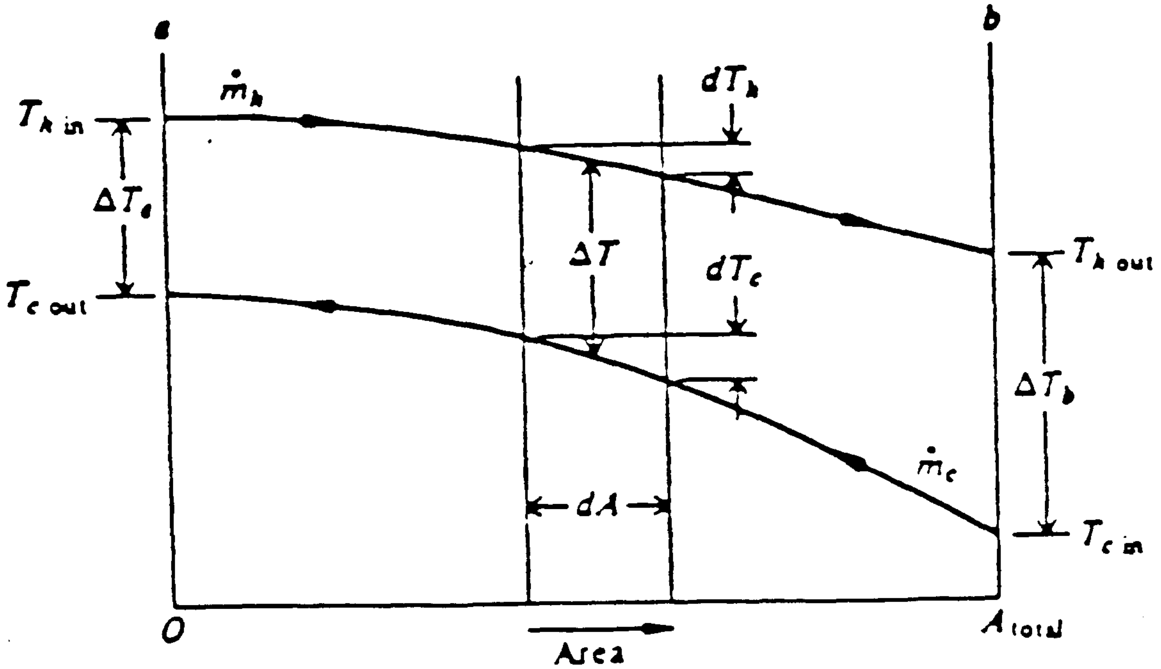

Figure 1. Temperature distribution in single-pass counterflow heat exchanger.

For a counter flow heat exchanger the LMTD is defined as (Ref. 3)

$$\overline{\Delta T} = \frac{\Delta T_a - \Delta T_b}{\ln \Delta T_a/\Delta T_b} \tag{4}$$

based on the following assumptions:

1. Rate of flow of each fluid is constant.
2. Specific heat of each fluid is constant.
3. Overall heat transfer coefficient through the exchanger is constant.

4. The system is adiabatic, heat exchange takes place only between the two fluids.

5. Fluid flow is countercurrent.

For the case of the finned heat exchanger, a convenient concept that can be used to provide a value for the heat transfer rate from a fin is the fin efficiency. The efficiency of the bar fin is given by (Ref. 3)

$$\phi = \frac{\tanh\ (pL)}{pL} \tag{5}$$

Here $p = h_c/ky$, L is the length of the fin and y is half the fin thickness.

Use of Reynolds Analogy to Estimate the Convection Coefficients

Empirical correlations for the heat transfer coefficients needed in this study were not available, but could be estimated using Reynolds analogy. By utilizing the boundary layer concept of no slip in the vicinity of a surface, the fluid is essentially stationary. As a result the transfer of heat takes place primarily by conduction. Reynolds analogy (Ref. 5) results from making the approximation that temperature and velocity change at proportional rates through the boundary layer as

$$-\frac{1}{C_p}\frac{q''_s}{\tau_s}\int_0^u du = \int_{T_s}^{T} dT \tag{6}$$

Here q'' and τ represent heat transfer and shear stress, respectively. The subscript s denotes the surface values.

For the case of turbulent flow in a tube, integration of the above equation with upper limit of $u = V$ and $T = T_b$, where V is mean velocity and T_b is the bulk temperature, gives

$$-\frac{1}{C_p}\frac{q''_s}{\tau_s}V = T_b - T_s \tag{7}$$

or

$$\frac{\overline{h}_x}{\rho C_p V} = \frac{\tau_s}{\rho V^2} \tag{8}$$

Shear stress at the surface, τ_s, can be evaluated from the force balance on a cylindrical control volume of length L and diameter D, as

$$\tau_s = \frac{(\Delta P)D}{\Delta L} \tag{9}$$

but from Darcy's equation (Ref. 2),

$$\frac{\Delta P}{L} = \frac{f}{D}\frac{\rho V^2}{2} \tag{10}$$

where f is Darcy's friction factor. Therefore,

$$\tau_s = f\ V^2/8 \tag{11}$$

finally

$$St_x = \frac{h}{\rho C_p V} = f/8 \tag{12}$$

where St_x is the local stanton number. The same form is also valid for averaged values

$$\overline{St} = \frac{\bar{h}}{\rho C_p V} = f/8 \qquad (13)$$

In order to estimate the internal convection coefficient $\bar{h}$ an approximation of the friction factor f is needed. From Schilichting (Ref. 6) the friction factor, in terms of Reynolds number for smooth pipe, is given as

$$\frac{1}{f} = 2.0 \log (Re \cdot \quad f \quad) - 0.8 \qquad (14)$$

for our $Re_D \approx 10^7$ a friction factor of $f \approx 0.008$ was obtained. For calculating the convection coefficient parallel to the long pipe, tne flat plate approximation can be used. Similarly, the flow outside the pipe is turbulent, and from Reynolds analogy integration of equation (5)

$$\frac{h_x}{\rho C_p V} = \frac{C_f}{2} \qquad (15)$$

where C_f is the average skin friction. For a flat plate, C_f can be approximated as (Ref. 7)

$$C_f = 0.072(Re_L)^{1/5} \qquad (16)$$

with $Re_L \approx 5 \times 10^6$ an average skin friction of $C_f \approx 0.0035$ was resulted.

Calculation of Thermodynamic and Transport Properties of Flow

Hot fluid is the result of the complete combustion of methane in the combustion chamber with excess oxygen in the process. The product is a mixture of the following gases.

Gas	Mole Fraction
CO_2	13.3%
H_2O	26.5
Air	49.7
O_2	10.5

At high pressure (≈ 100 atm) and temperature (mean average temperature of approximately 1300°C) the properties of pure gases are obtained from references 8 and 9.

The coefficient of viscosity of a gas mixture of n components may be calculated by Wilke's semi-imperical correlation (Ref. 10) as

$$\mu_{mix} = \sum_{i=1}^{n} \frac{\mu_i}{\sum_{j=1}^{n} \phi_{ij} \frac{x_j}{x_i}} \tag{17}$$

where μ_i are the viscosities of the component gases. x_i and x_j are mole fractions. The coefficients ϕ_{ij} are given as functions of the viscosities and molecular weights of species i and j as

$$\phi_{ij} = \frac{[1 + (\mu_i/\mu_j)^{1/2} (M_j/M_i)^{1/4}]^2}{[8(1 + M_i/M_j)]^{1/2}} \tag{18}$$

Similarly, the thermal conductivity of a mixture and polyatomic gases may be divided into two portions as (Ref. 10)

$$k_{mix} = k'_{mix} + k''_{mix} \tag{19}$$

Here, k'_{mix} represents the monatomic thermal conductivity of the mixture, whereas k''_{mix} accounts for the diffusional transport of internal energy. Since the combustion product has no monatomic component the term k'_{mix} is zero, and k''_{mix} is also calculated from equations (16) and (17) as μ_i is replaced by the thermal conductivity of the gas components, K_i (Ref. 11). A computer program was written in BASIC to calculate the viscosity and thermal conductivity of the mixture. A listing of the program appears on the next page.

The assumption of ideal gas made it possible to calculate the density of the mixture gas with the molecular weight of 28.25. Also, Prandtl number and specific heat at constant pressure were obtained based on the mole fraction of the mixture.

PRESENTATION OF RESULTS

Calculations are performed for heat exchanger tubes of various diameters of 5, 10, and 20 inches (0.125, 0.250, and 0.500 meters, respectively). The condition of staying within the compressibility limit of Mach $<$ 0.3 was maintained throughout the analysis. With sonic velocity of approximately 660 m/sec, a maximum speed of 200 m/sec is allowed. For the present analysis, it was assumed that tube thicknesses were both 1/16" (0.016 meters). For all cases the fin length and thickness were taken as 0.016 meters and 0.014 meters, respectively. They were assumed to be one fin thickness apart from each other. As a result a fin efficiency of 40 percent was calculated.

ORIGINAL PAGE IS
OF POOR QUALITY

```
1   !                                   -----
2   !                              --------------
5   !                          -----------------------
10  ! ..THIS PROGRAM CALCULATES VISCOSITY & THERMAL CONDUCTIVITY OF MIXTURES.>>
11  ! ------------------------                          -------------------------
12  !
13  !
21  REAL U(4),K(4),M(4),X(4),Fe(4,4),Fi(4,4)
22  !
23  !  ----------------------------------------
25  ! N IS NUMBER OF COMPONENTS IN THE MIXTURE
26  ! ----SYMBOLS LISTING ----
27  ! ----                ----
28  N=4!--# OF THE COMPONENTS --
29  ! ------------------------
30  !    1=O2
31  !    2=CO2
32  !    3=AIR
33  !    4=H2O
34  !
35  !
39  T=1175    !<<< T : TEMPERATURE IN DEGREES KELVIN.>>>
40  ! -- VISCOSITY & THERMAL CONDUCTIVITY AS A FUNCTION OF TEMPERATURE --
41  ! ---                                                              --
50  U(1)=237.9195+.2562*T
51  U(2)=140.4831+.2572*T
52  U(3)=153.1153+.2657*T
53  U(4)=-13.9480+.4050*T
60  K(1)=-121.3161+.1789*T
61  K(2)=82.2
62  K(3)=-51.5259+.113*T
63  K(4)=-6.4821+.1108*T
64  !   ------------------------------------
65  !       M(I) : MOLECULER Wt. OF COMPONENTS
66  !       X(I) : MOLE FRACTION OF COMPONENTS
70  M(1)=32
71  M(2)=44
72  M(3)=28
73  M(4)=18
74  X(1)=.105
75  X(2)=.133
76  X(3)=.497
77  X(4)=.265
80  Umix=0
90  Kmix=0
100 FOR I=1 TO N
110 Sumu=0
111 Sumk=0
115 FOR J=1 TO N
120 Fe(I,J)=(1+(U(I)/U(J))^.5*(M(J)/M(I))^.25)^2/(8*(1+M(I)/M(J)))^.5
130 Fi(I,J)=(1+(K(I)/K(J))^.5*(M(J)/M(I))^.25)^2/(8*(1+M(I)/M(J)))^.5
140 Sumu=Sumu+Fe(I,J)*(X(J)/X(I))
150 Sumk=Sumk+Fi(I,J)*(X(J)/X(I))
160 NEXT J
170 Umix=Umix+U(I)/Sumu
180 Kmix=Kmix+U(I)/Sumk
190 NEXT I
200 PRINT Umix,Kmix ! <<<RESULTS>>>
500 STOP
510 END
```

For the inner tube the inlet and outlet velocities are calculated based on changes in the fluid density as a function of temperature. Methods of obtaining the convective coefficients have been already outlined in this report. As a main purpose of this study the required length and pressure drop across a smooth pipe as well as finned pipe heat exchanger were calculated.

Results for different tube diameters are presented in Table 1.

Table I

Inner tube diam (m)	$\dot{m}_{max}$ (kg/sec)	No. of tube required	Inner tube fluid velos.		Convective coeff. ($kw/m^2 \cdot K^o$)		U ($kw/m^2 \cdot K^o$)	Smooth pipe		Finned pipe	
			V_{out}	V_{in}	h_i	h_o		length (m)	Δp (atm.)	length (m)	Δp (atm.)
0.125	27.3	14	190.4	30.6			1.974	2699	197.0	901	65.8
0.250	108.7	4	166.9	26.8	4.72	13.36	1.925	1384	39.7	455	13.1
0.500	434.6	1	167.0	26.9			1.898	702	9.9	229	3.2

NOTE: Outer tube diameters are twice the inner tube diameters.

CONCLUSION AND RECOMMENDATION

Calculation of heat exchanger area for smooth as well as finned tubes indicates that a conventional heat exchanger is not the most feasible approach to this problem. Even if the heat exchanger pipes are assembled in a compact form of a loop, the very high pressure drop across the heat exchanger cannot be tolerated. For the shortest length of finned tube in the case of 20-inch pipe, a head loss of at least 3.2 atmospheres was calculated.

As an alternative, a system which utilizes the energy released from fusion process is recommended. This process can be adopted by applying electrical current to a bank of sealed metal containers to melt the metal. The energy released to the surrounding air can provide the high temperature air needed for use in the wind tunnel.

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