Int. J. Appl. Engng Ed. Vol. 7, No. 2, pp. 125–134, 1991
Printed in Great Britain.

0742–0269/91 $3.00+0.00

Microcomputer-Assisted Heat Transfer Measurement/Analysis in a Circular Tube*

A. J. GHAJAR†
Y. H. ZURIGAT‡
School of Mechanical and Aerospace Engineering, Oklahoma State University, Stillwater, OK 74078, *U.S.A.*

An interactive computer program has been developed to calculate the local inside wall temperatures and local peripheral heat transfer coefficients from local outside wall temperatures measured at different axial locations along an electrically heated horizontal circular tube. The program also calculates the pertinent fluid flow and heat transfer dimensionless parameters. The test fluids used were water and mixtures of ethylene glycol and water. The computer program can run on any microcomputer. It can be used to reduce the experimental data to a form suitable for development of forced and mixed convection flow heat transfer correlations in an electrically heated horizontal circular tube in the laminar, transitional and turbulent flow regimes.

INTRODUCTION

HEAT transfer measurements in pipe flow is essential for assessment of performance of heat exchanging equipment and development of design correlations. Usually, the experimental procedure for a constant wall heat flux boundary condition consists of measuring the tube outside wall surface temperatures at discrete locations and the inlet and outlet bulk temperatures in addition to other measurements like the flow rate, room temperature, voltage drop across the test section and current carried by the test section. The peripheral heat transfer coefficient (local average) and the Nusselt number (dimensionless heat transfer coefficient) thereafter are calculated based on the knowledge of the pipe inside-wall surface temperature. While measurement of inside-wall temperature is difficult, it can be calculated from the measurements of the outside wall temperature, the heat generation within the pipe wall and the thermophysical properties of the pipe material (electrical resistivity and thermal conductivity).

This paper discusses the data gathering, reduction and processing procedures and the finite-difference formulations used for the development of an interactive computer program. The program calculates local inside-wall temperature, local peripheral heat transfer coefficient and other pertinent information from local outside-wall temperature measurements at different axial locations along an electrically-heated horizontal circular tube.

EXPERIMENTS

A schematic diagram of the overall experimental setup is shown as Fig. 1. The experimental setup shown was also used for pressure drop and intermittency factor measurements. The heat transfer test section is a horizontal seamless 316 stainless steel circular tube with an inside diameter of 0.624 in (15.850 mm) and an outside diameter of 0.748 in (18.999 mm). The total length of the test section is 20 ft (6.10 m) providing a maximum length to diameter ratio (L/D) of 385. The end connections of the test section consisted of copper plates which were silver-arc soldered to the ends of the test section to secure a well defined electric circuit through the end plates. The test section was insulated from the environment using fiberglass pipe insulation and vapor-proof pipe tape. The total thickness of the insulation materials is about 1.25 in (3.18 cm). To ensure a uniform fluid bulk temperature at the exit of the test section, a mixing well which consisted of several baffles was utilized. A one-shell and two-tube pass heat exchanger was used to cool the fluid from the test section to an allowable and steady state inlet bulk temperature.

The calming and inlet sections in front of the test section were used to produce a uniform velocity distribution in the test fluid before entering the test section. The calming section consisted of a 7-in (17.8 cm) diameter acrylic plastic cylinder with three perforated acrylic plastic plates followed by tightly packed 4-in (10.2 cm) long soda straws sandwiched between galvanized steel mesh screens. The total length of the calming section is 24.25 in (61.6 cm). Flow leaving the calming section entered the inlet section and flowed undisturbed through 9.25 in (23.5 cm) of a 6.5-in (16.5 cm) diameter acrylic plastic tube before it

---

* Paper accepted 14 July 1990.

† Associate Professor.

‡ Research Associate. Currently at the ESM Department of Virginia Polytechnic Institute and State University, Blacksburg, VA 24061, U.S.A.

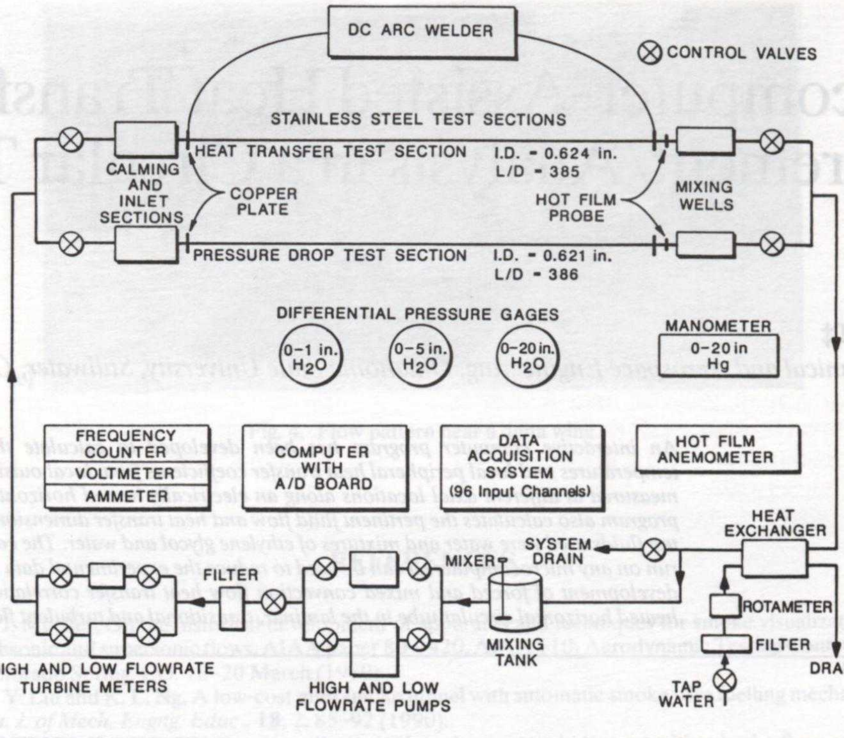


Fig. 1. Schematic of the experimental setup.

entered the test section. The combination of the inlet and test sections produced a square-edged entrance configuration for the test section.

A constant wall heat flux boundary condition was maintained by a Lincoln DC-600 welder. Thermocouples were placed on the outer surface of the tube wall at close intervals near the entrance and at greater intervals further downstream (see Fig. 2 for details). Thirty-one thermocouple stations were designated with four thermocouples at each station. The thermocouples were placed 90 degrees apart around the periphery. Copper-Constantan T-type thermocouples were used. The thermocouples were attached to the outside of the tube wall with Omegabond 101, an epoxy adhesive with high thermal conductivity and electrical resistivity. The inlet and exit bulk temperatures were measured by means of thermocouple probes inserted in the calming section and the mixing well, respectively. The data acquisition system used for the temperature measurements consisted of a Cole-Parmer MAC-14 datalogger with 96 input channels interfaced with an IBM compatible AT personal computer. The flow rate was measured by a turbine meter located upstream from the test section. The voltage drop across the test section

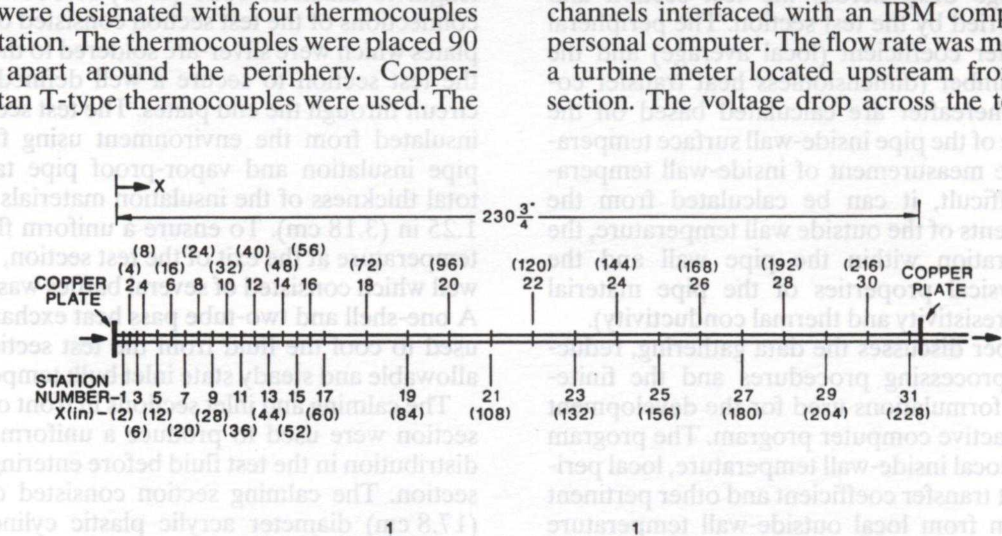


Fig. 2. Heat transfer test section thermocouple distribution.

and the current carried by the test section were measured by a voltmeter and an ammeter, respectively.

The test fluids used in the experiments were distilled water, ethylene glycol and different mixtures of distilled water and ethylene glycol.

COMPUTER PROGRAM

The computer program consists of four segments, the input data, the finite-difference formulations, the physical properties and the output.

1. *Input data*

A sample input data for the computer program is given as Fig. 3. Description of the information required for the first two lines of the input data file appear as follows:

Run number	1008
Total number of thermocouple stations used	26
Fluid index (1 = water, 2 = ethylene glycol)	1
Mass concentration of ethylene glycol	0.00
Flow rate (gal/min)	3.8610
Current carried by test section (A)	390.00
Voltage drop across test section (V)	21.55
Inlet bulk temperature (°F)	83.30
Exit bulk temperature (°F)	98.21
Room (ambient) temperature (°F)	79.43

All the subsequent lines (3 through 28) in Fig. 3 correspond to information about the outside tube wall temperatures measured at each station, see Fig. 2 for the layout of thermocouple stations. In lines 3 through 28 of Fig. 3, the first column corresponds to the thermocouple station number, the second column indicates the number of thermocouples at that particular station, the third column is the distance in inches between a given thermocouple station and the pipe entrance. The remaining columns (two or four) are the measured outside wall temperatures at each station.

```
1008 26
 1   0.00 3.8610 390.00   21.55   83.30   98.21  79.43
  1  4      2.00  94.61   95.12   95.09   94.90
  3  4      6.00  95.33   95.56   95.83   95.13
  4  4      8.00  95.50   95.66   95.67   95.24
  5  4     12.00  95.86   95.82   96.00   95.81
  6  4     16.00  96.44   96.16   96.14   96.18
  7  4     20.00  96.55   96.43   96.19   96.35
  8  4     24.00  96.93   96.69   96.50   96.53
  9  4     28.00  97.04   96.71   96.88   96.67
 10  4     32.00  97.03   97.20   96.77   96.98
 11  4     36.00  97.44   97.45   97.08   97.32
 12  4     40.00  97.71   97.62   97.17   97.31
 13  4     44.00  98.10   97.54   97.39   97.54
 14  4     48.00  98.04   97.89   97.80   97.71
 15  4     52.00  98.16   98.09   97.94   98.35
 16  4     56.00  98.41   98.42   98.23   97.66
 17  4     60.00  98.79   98.35   98.38   97.83
 18  4     72.00  99.19   99.07   99.24   98.92
 19  4     84.00  99.95  100.10   99.90   99.73
 20  4     96.00 100.47  100.52  100.77  100.19
 21  4    108.00 100.87  101.41  101.67  101.37
 22  4    120.00 101.80  102.44  102.35  101.74
 23  2    132.00 102.74  102.97
 25  2    156.00 104.35  104.24
 27  2    180.00 105.93  105.46
 29  2    204.00 107.60  107.20
 31  2    228.00 109.07  109.61
```

Fig. 3. Sample input data file.

The computer program is capable of handling input data files with as many as eight thermocouples per station along the periphery of any given horizontal pipe. The user has the option of either using the format shown in Fig. 3 or a more generalized format. With the data gathered in the manner described above, the pipe wall inside surface temperature is calculated using a finite-difference method described next.

2. *Finite-difference formulations*

The numerical solution of the conduction equation with internal heat generation and variable thermal conductivity and electrical resistivity was originally developed by Farukhi [1]. The numerical solution is based on the following assumptions:

1. Steady state conditions exist.
2. Peripheral and radial wall conduction exist.
3. Axial conduction is negligible.
4. The electrical resistivity and thermal conductivity of the tube wall are functions of temperature.

Based on the above assumptions the expressions for calculation of local inside wall temperature and heat flux and local and average peripheral heat transfer coefficients will be presented next.

a. *Calculation of the local inside wall temperature and the local inside wall heat flux*

The heat balance on a segment (slice) of the tube wall at any particular station is given by (see Fig. 4):

$$Q_g = Q_1 + Q_2 + Q_3 + Q_4. \tag{1}$$

From Fourier's law of heat conduction in a given direction n we know that

$$Q = -KA\frac{dT}{dn} \tag{2}$$

Now substituting Fourier's law and applying the finite-difference formulation for the radial (i) and peripheral (j) directions in Eq. (1) we obtain:

$$Q_1 = \frac{(K_{i,j} + K_{i-1,j})}{2}\,\frac{2\pi\left(r_i + \frac{\Delta r}{2}\right)\Delta z}{N_{TH}}\,\frac{(T_{i,j} - T_{i-1,j})}{\Delta r} \tag{3}$$

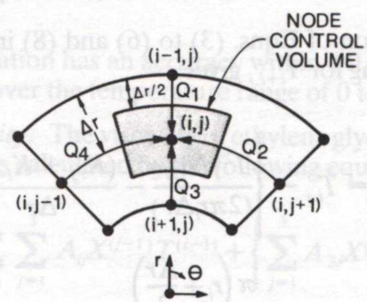


Fig. 4. Finite-difference grid arrangement.

$$Q_2 = \frac{(K_{i,j} + K_{i,j+1})}{2} (\Delta r \Delta z) \frac{(T_{i,j} - T_{i,j+1})}{\left(\frac{2\pi r_i}{N_{TH}}\right)} \quad (4)$$

$$Q_3 = \frac{(K_{i,j} + K_{i+1,j})}{2} \frac{2\pi\left(r_i - \frac{\Delta r}{2}\right)}{N_{TH}} \Delta z \frac{(T_{i,j} - T_{i+1,j})}{\Delta r} \quad (5)$$

$$Q_4 = \frac{(K_{i,j} + K_{i,j-1})}{2} (\Delta r \Delta z) \frac{(T_{i,j} - T_{i,j-1})}{\left(\frac{2\pi r_i}{N_{TH}}\right)} \quad (6)$$

where

K = thermal conductivity
r_i = tube inside radius
Q = rate of heat transfer
T = temperature
Δz = length of element
Δr = incremental radius
N_{TH} = number of finite-difference sections in the θ-direction (peripheral) which is equal to the number of thermocouples at each station.
i and j = the indices of the finite-difference grid points, i is the radial direction starting from the outside surface of the tube and j is the peripheral direction starting from top of the tube and increasing clockwise.

Heat generated at i, j element volume is given by:

$$Q_g = I^2 R \quad (7)$$

where

I = current
$R = \gamma l / A$ = resistance
γ = electric resistivity of the element
$l = \Delta z$ = length of the element
$A = (2\pi r_i / N_{TH})\Delta r$ = cross-sectional area of the element.

Substituting the above definitions into Equation (7) gives:

$$Q_g = I^2 \frac{\gamma \Delta z}{\left(\frac{2\pi r_i}{N_{TH}}\right)\Delta r} \quad (8)$$

Substitution of Eqns. (3) to (6) and (8) in Eq. (1) and solving for $T_{i+1,j}$ gives:

$$T_{i+1,j} = T_{i,j} - \left\{ \frac{I^2 \gamma N_{TH}}{(2\pi r_i \Delta r)} - \frac{(K_{i,j} + K_{i-1,j})}{\Delta r} \frac{\pi\left(r_i + \frac{\Delta r}{2}\right)}{N_{TH}} (T_{i,j} - T_{i-1,j}) - (K_{i,j} + K_{i,j+1}) \frac{\Delta r N_{TH}}{4\pi r_i} (T_{i,j} - T_{i,j+1}) - (K_{i,j} + K_{i,j-1}) \frac{\Delta r N_{TH}}{4\pi r_i} (T_{i,j} - T_{i,j-1}) \right\} \Bigg/ \left\{ (K_{i,j} + K_{i+1,j}) \frac{\pi\left(r_i - \frac{\Delta r}{2}\right)}{\Delta r N_{TH}} \right\} \quad (9)$$

Equation (9) is used to calculate the temperature of the interior nodes. In this equation the thermal conductivity and electrical resistivity of each node control volume is determined as a function of temperature from the following equations for 316 stainless steel:

$$K = 7.27 + 0.0038T \quad (10)$$

$$\gamma = 27.67 + 0.0213T \quad (11)$$

where T in °F, K in Btu/hr-ft-°F and γ in micro-ohm-in.

Once the local inside wall temperatures are calculated from Eq. (9), the local peripheral inside wall heat flux can be calculated from the heat balance equation, see Eq. (1).

b. *Calculation of the local peripheral and local average heat transfer coefficients*

From the local inside wall temperature, the local peripheral inside wall heat flux and the local bulk fluid temperature, the local peripheral heat transfer coefficient can be calculated as follows:

$$h_i = \dot{q}_i''/(T_{wi} - T_b) \quad (12)$$

where

h_i = local peripheral heat transfer coefficient
$\dot{q}_i''$ = local peripheral inside wall heat flux
T_{wi} = local inside wall temperature
T_b = bulk fluid temperature at the thermocouple station.

Note that in these analyses it is assumed that the bulk fluid temperature increases linearly from the inlet to the outlet according to the following equation:

$$T_b = T_{in} + (T_{out} - T_{in})X/L \quad (13)$$

where

T_{in} = bulk inlet temperature
T_{out} = bulk outlet temperature
X = distance from the pipe inlet to the thermocouple station
L = total length of the test section.

The local average heat transfer coefficient at each station can be calculated by the following equation:

$$\bar{h}_i = \bar{\dot{q}}_i''/(\bar{T}_{wi} - T_b) \quad (14)$$

where

$\bar{h}_i$ = local average heat transfer coefficient

$\dot{q}_i''$ = average peripheral inside wall heat flux at a station

$\bar{T}_{wi}$ = average inside wall temperature at a station.

3. *Physical properties*

The test fluids used in the experiments were distilled water, ethylene glycol and different mixtures of distilled water and ethylene glycol. The equations for physical properties of these fluids are presented next.

a. *Physical properties of water*

The correlation equations used for water were those used by [2] except density [3] and coefficient of thermal expansion [4].

Density. The density of water is calculated by the following equation:

$$\rho = 999.86 + 6.1464 \times 10^{-2} T - 8.4648 \times 10^{-3} T^2 + 6.8794 \times 10^{-5} T^3 - 4.4214 \times 10^{7} T^4 + 1.2505 \times 10^{-9} T^5 \quad (15)$$

where

ρ = density, kg/m^3 and T = temperature, °C

This equation is valid for the temperature range from 0 to 100°C and is within the accuracy of the steam tables [5] which is ±0.05 kg/m^3.

Viscosity. The viscosity of water is calculated by the following equation:

$$\log_{10}[\mu_T/\mu_{20}] = \{1.327(20 - T) - 1.053 \times 10^{-3}(20 - T)^2\}/(T + 105) \quad (16)$$

where

μ_{20} = viscosity of water at 20°C = $1.0 \times 10^{-3} N - \text{s/m}^2$

μ_T = viscosity of water at T°C, $N - \text{s/m}$

T = temperature, °C

This equation is valid within the temperature range from 10 to 100°C. It has an accuracy within ±1%.

Specific heat. The specific heat of water is calculated by the following equation:

$$C_p = 4.216 - 2.2 \times 10^{-3} T + 3.66 \times 10^{-5} T^2 - 1.475 \times 10^{-7} T^3 \quad (17)$$

where

C_p = specific heat, kJ/(kg − K), and T = temperature, °C

This equation has an accuracy within ±1% for the range from 0 to 100°C.

Thermal conductivity. The thermal conductivity of water is calculated by the following equation:

$$K = 5.6276 \times 10^{-1} + 1.874 \times 10^{-3} T - 6.80 \times 10^{-6} T^2 \quad (18)$$

where

K = thermal conductivity of water, $W/(m - K)$ and T = temperature, °C

This equation is applied in the temperature range of 0 to 100°C. It has an accuracy within ±1%.

Coefficient of thermal expansion. The thermal expansion coefficient of water is calculated by the following equation:

$$\beta = \{0.0615 - 0.01694 T - 2.06 \times 10^{-4} T^2 - 1.77 \times 10^{-6} T^3 + 6.3 \times 10^{-9} T^4\}/\{999.86 + 0.06146 T - 0.00847 T^2 + 6.879 \times 10^{-5} T^3 - 4.42 \times 10^{-7} T^4 + 1.25 \times 10^{-9} T^5\} \quad (19)$$

where

β = coefficient of thermal expansion, $1/K$

ρ = density at T, kg/m^3

T = temperature, °C.

This equation was derived from the relation $\beta = (1/\rho)(\partial\rho/\partial T)_p$, with the use of the density correlation listed previously. It is applied in the temperature range of 0 to 100°C and is accurate within the accuracy of the steam tables [5].

b. *Physical properties of ethylene glycol-water mixtures*

The correlation equations used for ethylene glycol-water mixtures were those reported in [6], except thermal expansion coefficient [4].

Density. The density of ethylene glycol-water mixture is calculated by the following equation:

$$\rho = \sum_{i=1}^{3} \sum_{j=1}^{3} A_{ij} X^{(i-1)} T^{(i-1)} \quad (20)$$

where

Values of A_{ij}

	$j = 1$	$j = 2$	$j = 3$
$i = 1$	1.0004	0.17659	−0.049214
$i = 2$	-1.2379×10^{-4}	-9.9189×10^{-4}	4.1024×10^{-4}
$i = 3$	-2.9837×10^{-6}	2.4164×10^{-6}	-9.5278×10^{-8}

ρ = density, g/cm^3

X = mass fraction of ethylene-glycol

T = temperature, °C.

This equation has an accuracy within ±0.25%. It is applied over the temperature range of 0 to 150°C.

Viscosity. The viscosity of ethylene glycol-water mixture is calculated by the following equation:

$$ln\,\mu = \sum_{i=1}^{2} \sum_{j=1}^{3} A_{ij} X^{(j-1)} T^{(i-1)} + \left\{ \sum_{j=1}^{3} A_{3j} X^{(j-1)} \right\}^{\frac{1}{4}} T^2 \quad (21)$$

where

Values of A_{ij}

	$j=1$	$j=2$	$j=3$
$i=1$	0.55164	2.6492	0.82935
$i=2$	-2.7633×10^{-2}	-3.1496×10^{-2}	4.8136×10^{-3}
$i=3$	-6.0629×10^{-17}	2.2389×10^{-15}	-5.8790×10^{-16}

μ = viscosity, mPa-s
X = mass fraction of ethylene-glycol
T = temperature, °C

This equation has an accuracy within ±5%. It is applied over the temperature range of −10 to 100°C.

Prandtl number. The Prandtl number of ethylene glycol-water mixture is calculated by the following equation:

$$ln\ \mathrm{Pr} = \sum_{i=1}^{2} \sum_{j=1}^{3} A_{ij} X^{(j-1)} T^{(i-1)} + \left\{ \sum_{j=1}^{3} A_{3j} X^{(j-1)} \right\}^{\frac{1}{4}} T^2 \tag{22}$$

where

Values of A_{ij}

	$j=1$	$j=2$	$j=3$
$i=1$	2.5735	3.0411	0.60237
$i=2$	-3.1169×10^{2}	-2.5424×10^{-2}	3.7454×10^{-3}
$i=3$	1.1605×10^{-16}	2.5283×10^{-15}	2.3777×10^{-17}

Pr = Prandtl number, $\mu C_p / K$
X = mass fraction of ethylene-glycol
T = temperature, °C.

This equation has an accuracy within ±5%. It is applied over the temperature range of 0 to 150°C.

Thermal conductivity. The thermal conductivity of ethylene glycol-water mixture is calculated from the following equation:

$$K_W = 0.56276 + 1.874 \times 10^{-3} T - 6.8 \times 10^{-6} T^2$$
$$K_{EG} = 0.24511 + 1.755 \times 10^{-4} T - 8.52 \times 10^{-7} T^2$$
$$K_F = 0.6635 - 0.3698 X - 8.85 \times 10^{-4} T$$
$$K_M = (1 - X) K_W + X K_{EG} - K_F (K_W - K_{EG})(1 - X) X \tag{23}$$

where

K_W = thermal conductivity of pure water, $W/m-K$
K_{EG} = thermal conductivity of pure ethylene glycol, $W/m-K$
K_F = Filippov's constant
K_M = thermal conductivity of ethylene glycol-water mixture, $W/m-K$
X = mass fraction of ethylene glycol
T = temperature, °C

The equation has an accuracy within ±1%. It is applied over the temperature range of 0 to 150°C.

Coefficient of Thermal Expansion. The thermal expansion coefficient of ethylene glycol-water mixture is calculated by the following equation:

$$\beta = -(1/\rho)\{-1.2379 \times 10^{-4} - 9.9189 \times 10^{-4} X + 4.1024 \times 10^{-4} X^2 + 2(-2.9837 \times 10^{-6} \times T + 2.4614 \times 10^{-6} XT - 9.5278 \times 10^{-8} X^2 T)\} \tag{24}$$

where

β = coefficient of thermal expansion, 1/°C
ρ = density, g/m^3
X = mass fraction of ethylene glycol
T = temperature, °C.

This equation was derived from the relation $\beta = (1/\rho)(\partial\rho/\partial T)_p$, with the use of the density correlation listed previously. It is applied over the temperature range of 0 to 150°C.

4. *Output*

A sample output data file from the computer program is given as Figs 5a to 5d. The user has the option of specifying SI or English units for the output data. The output file starts with a summary of some of the important information about the experimental run for quick reference and then lists the measured outside wall temperatures, see Fig. 5a. In Fig. 5b the calculated inside wall temperatures and the Reynolds number of the flow based on the inside wall temperature are displayed. Next, the inside surface peripheral heat fluxes and the peripheral heat transfer coefficients for each thermocouple station along the pipe are shown as Fig. 5c. Finally, Fig. 5d gives a tabular summary of the most important results of the experiments. This table gives for each station its location from the tube entrance, the bulk fluid temperature, the bulk Reynolds, Prandtl, Nusselt and Grashof numbers, the ratio of bulk to wall absolute viscosities, the ratio of top to bottom peripheral heat transfer coefficients and finally the local average heat transfer coefficient.

In addition to the complete output presented in Figs. 5a to 5d, the user has the option of creating a short version of the output file (similar to Fig. 5d). This version of the output file is created in a headingless format and can be used for further analysis of the experimental results. For example it can be used as an input data file for a graphics software for developing plots or a curve fitting software for development of heat transfer correlations.

The output presented in Figs 5a to 5d contain all the necessary information for an in depth analysis of the experiments. For example, if the ratio of bulk to wall absolute viscosities is greater than 1.0 this

```
*-----------------------------------------------*
              RUN NUMBER 1008
       TEST FLUID IS DISTILLED WATER
*-----------------------------------------------*


VOLUMETRIC FLOW RATE =      3.86    GPM
MASS FLOW RATE       =    1925.4    LBM/HR
MASS FLUX            =    906637    LBM/(SQ.FT-HR)
FLUID VELOCITY       =      4.03    FT/S
ROOM TEMPERATURE     =     79.43    F
INLET TEMPERATURE    =     83.30    F
OUTLET TEMPERATURE   =     98.21    F
AVERAGE RE NUMBER    =     25829
AVERAGE PR NUMBER    =      5.11
CURRENT TO TUBE      =     390.0    AMPS
VOLTAGE DROP IN TUBE =     21.55    VOLTS
AVERAGE HEAT FLUX    =      9129    BTU/(SQ.FT-HR)
Q=AMP*VOLT           =     28677    BTU/HR
Q=M*C*(T2-T1)        =     28648    BTU/HR
HEAT BALANCE ERROR   =       .10    %


   OUTSIDE SURFACE TEMPERATURES - DEGREES F


      1        3        4        5        6        7        8        9       10

1   94.61    95.33    95.50    95.86    96.44    96.55    96.93    97.04    97.03
2   95.12    95.56    95.66    95.82    96.16    96.43    96.69    96.71    97.20
3   95.09    95.83    95.67    96.00    96.14    96.19    96.50    96.88    96.77
4   94.90    95.13    95.24    95.81    96.18    96.35    96.53    96.67    96.98


     11       12       13       14       15       16       17       18       19

1   97.44    97.71    98.10    98.04    98.16    98.41    98.79    99.19    99.95
2   97.45    97.62    97.54    97.89    98.09    98.42    98.35    99.07   100.10
3   97.08    97.17    97.39    97.80    97.94    98.23    98.38    99.24    99.90
4   97.32    97.31    97.54    97.71    98.35    97.66    97.83    98.92    99.73


     20       21       22       23       25       27       29       31

1  100.47   100.87   101.80   102.74   104.35   105.93   107.60   109.07
2  100.52   101.41   102.44
3  100.77   101.67   102.35   102.97   104.24   105.46   107.20   109.61
4  100.19   101.37   101.74
```

Fig. 5a. Sample output data file (page 1).

indicates large variations of that property and constant property analysis should not be used. Another interesting result is the ratio of top to bottom peripheral heat transfer coefficients. If this ratio is close to 1.0, forced convection dominates and heat transfer is primarily dependent on the Reynolds and Prandtl numbers. However, if the ratio is less than 1.0, natural convection exists and heat transfer is by mixed convection. In this case secondary flow caused by the difference between the fluid density at the wall and at the pipe center also affects the temperature profile. For mixed convection the heat transfer in addition to Reynolds and Prandtl numbers is dependent on the Grashof number (which accounts for the variation in density of the test fluid). In extreme cases, where the ratio is much less than 1.0, free convection dominates and heat transfer primarily depends on the Prandtl and Grashof numbers.

SUMMARY

The development of an interactive computer program for heat transfer measurement/analysis has been described. The program uses finite-difference formulation to calculate local inside wall temperatures and local peripheral heat transfer coefficients from measured local outside wall temperatures and other pertinent experimental data. In the analysis both radial and peripheral heat conduction exists. However, axial heat conduction has been neglected. The program also calculates important dimensionless parameters such as Nusselt, Prandtl, Reynolds and Grashof numbers.

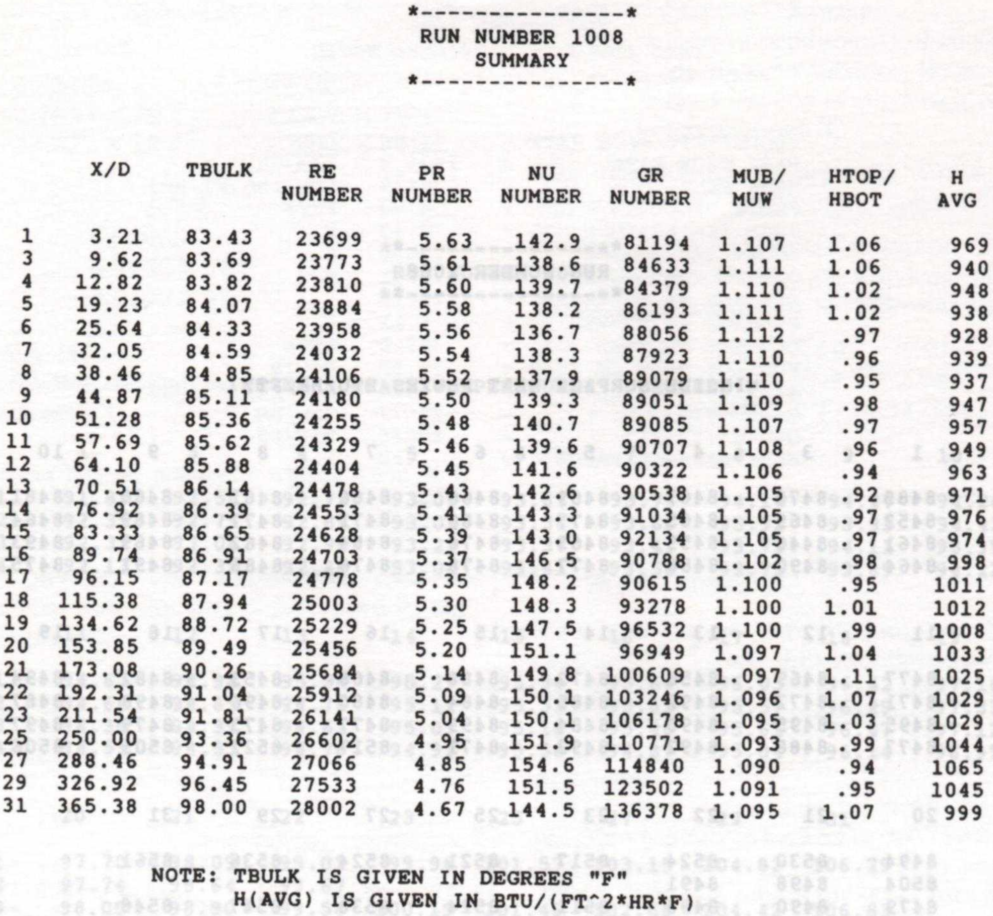
The type of computer program described in this study can be utilized as an effective design or instructional tool. As a design tool it can be used to perform parametric studies on the influence of certain parameters on the heat transfer characteristics of the flow or to develop heat transfer correlations in terms of pertinent dimensionless

```
                       *-----------------*
                        RUN NUMBER 1008
                       *-----------------*


           INSIDE SURFACE TEMPERATURES - DEGREES F

      1       3       4       5       6       7       8       9      10

 1  91.83   92.56   92.73   93.09   93.67   93.78   94.16   94.27   94.26
 2  92.35   92.79   92.89   93.05   93.38   93.66   93.92   93.93   94.43
 3  92.32   93.06   92.90   93.23   93.37   93.41   93.72   94.11   93.99
 4  92.13   92.35   92.46   93.04   93.41   93.58   93.75   93.89   94.21

     11      12      13      14      15      16      17      18      19

 1  94.67   94.94   95.34   95.27   95.38   95.64   96.03   96.42   97.18
 2  94.68   94.85   94.76   95.12   95.32   95.65   95.57   96.29   97.33
 3  94.30   94.39   94.61   95.03   95.16   95.46   95.61   96.47   97.12
 4  94.55   94.53   94.76   94.93   95.58   94.87   95.04   96.14   96.95


     20      21      22      23      25      27      29      31

 1  97.70   98.09   99.02   99.96  101.57  103.15  104.82  106.29
 2  97.74   98.64   99.67
 3  98.00   98.90   99.58  100.19  101.46  102.68  104.42  106.83
 4  97.41   98.60   98.96



             REYNOLDS NUMBER AT THE INSIDE TUBE WALL

      1       3       4       5       6       7       8       9      10

 1  26148   26363   26414   26521   26696   26728   26843   26876   26871
 2  26302   26432   26462   26509   26610   26692   26769   26774   26923
 3  26293   26515   26466   26564   26605   26619   26712   26827   26792
 4  26236   26302   26335   26506   26616   26667   26720   26762   26856

     11      12      13      14      15      16      17      18      19

 1  26994   27076   27195   27175   27210   27288   27404   27522   27752
 2  26998   27049   27023   27129   27189   27289   27267   27484   27798
 3  26884   26911   26978   27102   27143   27232   27278   27537   27736
 4  26958   26954   27023   27074   27269   27057   27107   27438   27684

     20      21      22      23      25      27      29      31

 1  27910   28028   28313   28602   29098   29587   30107   30566
 2  27924   28196   28512
 3  28002   28276   28484   28673   29063   29440   29981   30738
 4  27822   28184   28294
```

Fig. 5b. Sample output data file (page 2).

```
                        *-----------------*
                         RUN NUMBER 1008
                        *-----------------*


              INSIDE SURFACE HEAT FLUXES BTU/HR/FT2

      1      3      4      5      6      7      8      9      10

 1  8485   8470   8467   8469   8460   8466   8461   8460   8481
 2  8453   8469   8465   8477   8480   8471   8477   8489   8464
 3  8461   8446   8459   8463   8476   8484   8482   8469   8493
 4  8464   8490   8486   8477   8478   8476   8485   8491   8475

     11     12     13     14     15     16     17     18     19

 1  8477   8469   8455   8472   8488   8468   8453   8482   8494
 2  8471   8472   8492   8485   8484   8480   8496   8498   8487
 3  8495   8495   8490   8484   8499   8477   8473   8479   8497
 4  8477   8488   8492   8493   8471   8517   8522   8505   8506

     20     21     22     23     25     27     29     31

 1  8494   8530   8524   8517   8521   8524   8536   8561
 2  8504   8498   8491
 3  8479   8490   8496   8511   8524   8538   8547   8546
 4  8520   8500   8526


        PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-F)

      1      3      4      5      6      7      8      9      10

 1  1009    955    950    939    905    921    908    923    953
 2   947    930    933    944    936    934    934    961    933
 3   951    900    931    924    938    961    955    940    984
 4   973    980    981    945    934    943    952    966    958

     11     12     13     14     15     16     17     18     19

 1   937    934    919    954    972    969    954   1000   1004
 2   935    944    984    972    979    970   1011   1017    985
 3   978    998   1001    983    998    991   1003    994   1010
 4   949    980    984    994    948   1069   1082   1037   1032

     20     21     22     23     25     27     29     31

 1  1035   1090   1068   1044   1037   1033   1019   1033
 2  1030   1015    983
 3   996    983    994   1015   1052   1098   1073    967
 4  1076   1020   1076
```

Fig. 5c. Sample output data file (page 3).



```
                        *----------------*
                         RUN NUMBER 1008
                             SUMMARY
                        *----------------*


       X/D     TBULK     RE       PR       NU       GR      MUB/    HTOP/      H
                       NUMBER   NUMBER   NUMBER   NUMBER    MUW     HBOT      AVG

  1     3.21   83.43   23699     5.63    142.9    81194   1.107    1.06      969
  3     9.62   83.69   23773     5.61    138.6    84633   1.111    1.06      940
  4    12.82   83.82   23810     5.60    139.7    84379   1.110    1.02      948
  5    19.23   84.07   23884     5.58    138.2    86193   1.111    1.02      938
  6    25.64   84.33   23958     5.56    136.7    88056   1.112     .97      928
  7    32.05   84.59   24032     5.54    138.3    87923   1.110     .96      939
  8    38.46   84.85   24106     5.52    137.9    89079   1.110     .95      937
  9    44.87   85.11   24180     5.50    139.3    89051   1.109     .98      947
 10    51.28   85.36   24255     5.48    140.7    89085   1.107     .97      957
 11    57.69   85.62   24329     5.46    139.6    90707   1.108     .96      949
 12    64.10   85.88   24404     5.45    141.6    90322   1.106     .94      963
 13    70.51   86.14   24478     5.43    142.6    90538   1.105     .92      971
 14    76.92   86.39   24553     5.41    143.3    91034   1.105     .97      976
 15    83.33   86.65   24628     5.39    143.0    92134   1.105     .97      974
 16    89.74   86.91   24703     5.37    146.5    90788   1.102     .98      998
 17    96.15   87.17   24778     5.35    148.2    90615   1.100     .95     1011
 18   115.38   87.94   25003     5.30    148.3    93278   1.100    1.01     1012
 19   134.62   88.72   25229     5.25    147.5    96532   1.100     .99     1008
 20   153.85   89.49   25456     5.20    151.1    96949   1.097    1.04     1033
 21   173.08   90.26   25684     5.14    149.8   100609   1.097    1.11     1025
 22   192.31   91.04   25912     5.09    150.1   103246   1.096    1.07     1029
 23   211.54   91.81   26141     5.04    150.1   106178   1.095    1.03     1029
 25   250.00   93.36   26602     4.95    151.9   110771   1.093     .99     1044
 27   288.46   94.91   27066     4.85    154.6   114840   1.090     .94     1065
 29   326.92   96.45   27533     4.76    151.5   123502   1.091     .95     1045
 31   365.38   98.00   28002     4.67    144.5   136378   1.095    1.07      999


                NOTE: TBULK IS GIVEN IN DEGREES "F"
                      H(AVG) IS GIVEN IN BTU/(FT^2*HR*F)
```

Fig. 5d. Sample output data file (page 4).

parameters in a given flow regime. In a classroom setting, the program can be employed to demonstrate the basic principles of heat conduction and convection.

Acknowledgements—Support for this research provided by the National Science Foundation under grant no. CBT-88103342. The assistance of Mr D. R. Maiello with the development of the computer program is gratefully acknowledged.

REFERENCES

1. M. N. Farukhi, An Experimental Investigation of Forced Convective Boiling at High Qualities Inside Tubes Preceded by 180 Degree Bend, Ph.D. thesis, Oklahoma State University, Stillwater, OK (1973).
2. A. Abdelmessih, Laminar Flow Heat Transfer Downstream from U-Bends, Ph.D. thesis, Oklahoma State University, Stillwater, OK (1986).
3. J. H. Chen, Heat Transfer in High Laminar, Transition and Lower Turbulent Flow Regimes for Square-Edged Contraction Entrance in a Circular Tube, Ph.D. thesis, Oklahoma State University, Stillwater, OK (1988).
4. J. R. Augustine, Pressure Drop Measurements in the Transition Region for a Circular Tube with a Square-Edged Entrance, M.S. thesis, Oklahoma State University, Stillwater, OK (1990).
5. L. Haar, J. S. Gallagher and G. S. Kell, *Steam Tables*, Hemisphere, New York (1983).
6. D. Bohn, S. Fischer and E. Obermeir, Thermal Conductivity, Density, Viscosity and Prandtl Numbers of Ethylene Glycol-Water Mixtures, *Ber. Bunsenges. Phys. Chem*., **88**, 739–742 (1984).