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"BY-PASS FLOWS AND TEMPERATURE DISTRIBUTION IN A HOT GAS DUCT INTERNALLY INSULATED BY CARBON STONE"

Aydin Konuk

ABSTRACT

A mathematical model has been developed to calculate by-pass flows and temperature distribution in a hot gas duct internally insulated by carbon stone rings. The equations of conservation of mass and momentum are solved for a piping system to obtain axial and radial by-pass velocities. The energy equation is solved next by a marching method to obtain the radial temperature distribution along the duct.

The results, although qualitative due to simplifications in the model, are useful to study the effects of duct geometry on its performance.

I – INTRODUCTION

Hot gas ducts are one of the most critical components of high temperature reactors (HTR), and in the last year many duct designs have been tested and mathematical models have been developed to predict their performance. At the "Instituto de Energia Atômica (IEA)", São Paulo, Brazil, tests are continuing on a vertical duct internally insulated by mineral fibers. At the "Institut Fur Reaktorbauelemente, Kernfornscungsanlage Julich GmbH", internal insulation for horizontal ducts have been tested, using metallic foils[3], mineral fibres[4] and lastly carbon stone[1] as the insulating material. The results are summarized by Brockerhoff[2]. Foil and fibre insulation have been found satisfactory for keeping the duct wall temperature and the heat losses at low values. However, they would be too costly and time consuming for long ducts. The carbon stone insulation offers the advantages of low cost and rapid construction, due to its simplicity. An other advantage is that, in constrast to the first two, it contains no metallic parts, in particular no hot gas liner is necessary. However, it was found to be a less effective insulating material than foils or fibres. One of the problems was the by-pass flows in the gaps between carbon stone rings and between duct wall and carbon stone.

At the "Instituto de Energia Atômica", some work has been done in model development. Computers programs have been developed to predict performances of fibrous insulation in horizontal and vertical cylinders[5], and of a ring – tubes ceramic duct[6]. In this report, a methematical model for the carbon stone insulation is presented.

The description below is taken from[2].

The carbon stone insulation tested at Julich[1,2] is shown in Figure 1. The insulation was supplied by the Sigri Company in Meitingen, Germany. The trade name of the carbon material is RUD-N.H consists of five rings, the first and the last acting as entrance and exit passages. The direction of gas flow is shown in Figure 1. Half tubes were welded to the outer surface of the duct wall to measure the heat losses. The average inner diameter of the duct (pressure – tube) is D_3 = 930.2 mm.

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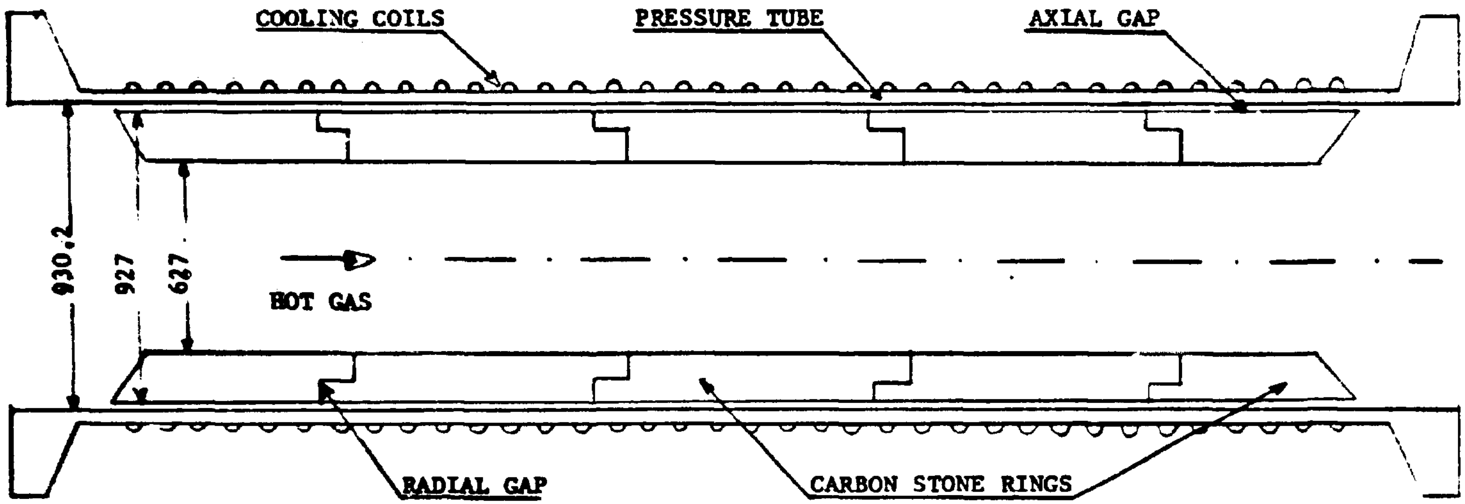

Figure 1 – Carbon Stone Insulation

The carbon rings were made with a smaller diameter than the pressure tube. The outside diameter was $D_2 = 927$ mm and the inner diameter was $D_1 = 627$ mm. Thus, when the rings are exactly positioned there is a gap of 1.6 mm between the pressure tube wall and the carbon ring. The rings were 800 mm long (except the first and the last, which were shorter). To avoid direct flow of hot gas through the gap between adjacent rings to the pressure tube, the rings on the cold side are tightly packed. On the inner part an expansion gap at 1.5 mm was left. Each carbon ring has a groove at 5 mm x 5 mm in which, if necessary, a seal may be placed to prevent the by-pass flows between the rings an the pressure tube.

In the following section, the mathematical model of the carbon stone insulation is given. Then, numerical results are presented, fallowed by conclusions and remarks on future work.

II – MATHEMATICAL MODEL

The hot gas duct has been modeled first as a piping network. The equations of conservation of mass and momentum, written for the resulting piping system, have been solved to obtain the axial radial leakage velocities in the duct. Then, the duct was divided into axial increments and the equation at conservation of energy has been solved to obtain the radial temperature distribution along the duct

1) Piping Network

The piping network corresponding to the duct shown in Figure 1 is given in Figure 2.

The horizontal branches correspond to the gap between the carbone stone and the pipe wall, and the vertical branches to the gap between carbon stone rings. The horizontal branches have velocity v, flow area A_v, hidraulic diameter DH_v, and lenght ℓ_v. The vertical branches have velocity u, flow area A_u, hidraulic diameter DH_u, and lenght ℓ_u. Velocity u represents the radial flow from inside to the outside of the rings. For the case where the gaps are not sealed, the branches are represented by the fallowing values:

$$A_v = \pi(D_3^2 - D_1^2)/4$$

$$A_u = \pi D_1 G$$

$$DH_v = D_3 - D_1$$

$$DH_u = 2G$$

G is the gap between rings. This gap has been taken as G = 1.5 mm.

The pressure of the main gas stream is denoted by Pg.

The pressure at the entrance of the pipe, P_g^1, must be supplied. The remaining gas pressures, P_g^2, $P_g^3, \ldots, P_G^6$ are calculated from:

$$P_g^{i+1} = \frac{l_v^i}{DH_v^i} f \rho \frac{V^2}{2g_c}$$

where f is the friction factor, ρ gas density and V gas velocity in the duct.

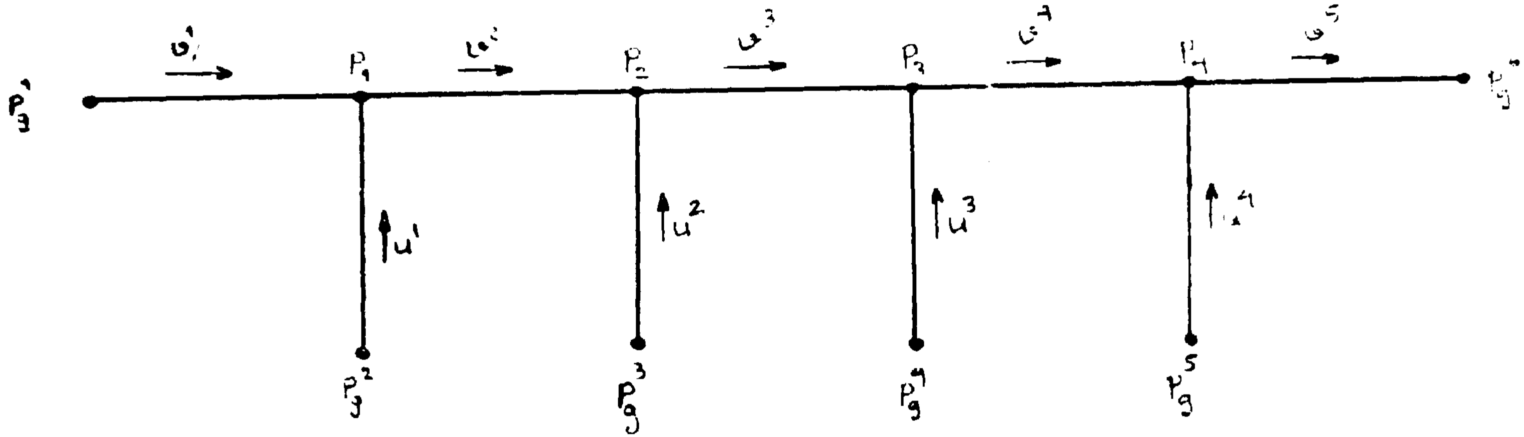

Figure 2 — Piping Network Model

The pressures P^1, P^2, P^3, P^4 in the horizontal branches and the velocities v^1, v^2, . . . , v^5, and u^1, u^2, u^3, u^4 are the unknows at the system. They are calculated by solving the fallowing equations:

Conservation of mass

$$A_v^{i+1} v^{i+1} - A_v^i v^i - A_u^i u_i = 0 \qquad i = 1,4 \tag{1}$$

Conservation of momentum in the horizontal branches:

$$p^i - p^{i-1} + \frac{l_v^i \rho f_v v^i |v^i|}{2 D_{Hv}^i g_c} = 0 \qquad i = 1,5 \tag{2}$$

with $p^0 = p_g^1$, $p^5 = p_g^6$

f_v is the friction factor in the horizontal branches

Conservation of momentum in the horizontal branches:

$$p^i - p_g^{i+1} + \frac{l_u^i \rho f_u u^i |u^i|}{2 DH_u^i g_c} = 0 \qquad i = 1,4 \tag{3}$$

The equations (2) and (3) are first linearized by assuming values of |v| and |u|, then the linear system of 13 unknowns (5v, 4u and 4P) is solved by Gauss elimination. The resulting velocities are used to linearize the system for the second iteration, and the procedure is continued until velocities converge within a certain precision.

The program has been run with and without sealing of the gaps. The results are shown in the next section.

2) Conservation of Energy

The calculated velocities are used in the solution at the energy equation in order to obtain temperature profiles in the duct. The duct was divided axially, and at each axial position, 5 unknown temperatures along the radius were considered (Figura 3). These are:

T_1 : Temperature at the inner surface of carbon stone

T_2 : Temperature at the outer surface of carbon stone

T_3 : Temperature of the axial by-pass stream (v)

T_4 : Temperature at the pressure tube wall

T_5 : Temperature of the cooling water

The equations of conservation of energy for an axial position are the following:

Conservation of energy at the inner surface of carbon stone: $(D = D_1)$

$$\pi D_1 \Delta x h_1 (T_g - T_1) - 2\pi \Delta x k_c \frac{T_2 - T_1}{\ln \frac{D_2}{D_1}} \tag{4}$$

Conservation of energy at the outer surface of carbon stone: $(D = D_2)$

$$2\pi \Delta x k_c \frac{T_2 - T_1}{\ln \frac{D_2}{D_1}} - \pi D_2 \Delta x h_2 (T_2 - T_3) = 0 \tag{5}$$

Conservation of energy for the axial by pass stream (Between D_2 and D_3)

$$\pi D_2 \Delta x h_2 (T_2 - T_3) - \pi D_3 \Delta x h_3 (T_3 - T_4) - \dot{m}_1 Cp_1 (T_3 - T_3^*) = 0 \tag{6}$$

Conservation of energy at the pressure tube wall $(D = D_4)$

$$\pi D_3 \Delta x h_3 (T_3 - T_4) - \pi D_4 \Delta x h_4 (T_4 - T_5) = 0 \tag{7}$$

Conservation of energy for the cooling water (Between D_4 and D_5)

$$\pi D_4 \Delta x h_4 (T_4 - T_5) - \dot{m}_2 cp_2 (T_5 - T_5^*) - \pi D_5 \Delta x h_5 (T - T_{out}) = 0 \tag{8}$$

The cooling water is assumed to flow in a coaxial pipe of diameter D_5. The definition of the variables is given below:

k_c = thermal conductivity of carbon stone, W/K [illegible]

h_1 = heat transfer coefficient between the hot gas and the inner wall of carbon stone, W/Km^2

h_2 = heat transfer coefficient between the outer surface of carbon stone and the axial by-pass stream, W/Km^2

h_3 = heat transfer coefficient between the axial by-pass stream and the inner surface of the pressure tube, W/Km^2

$\dot{m}_1$ = mass flow rate the axial by-pass stream, Kg/s. It is calculated from:

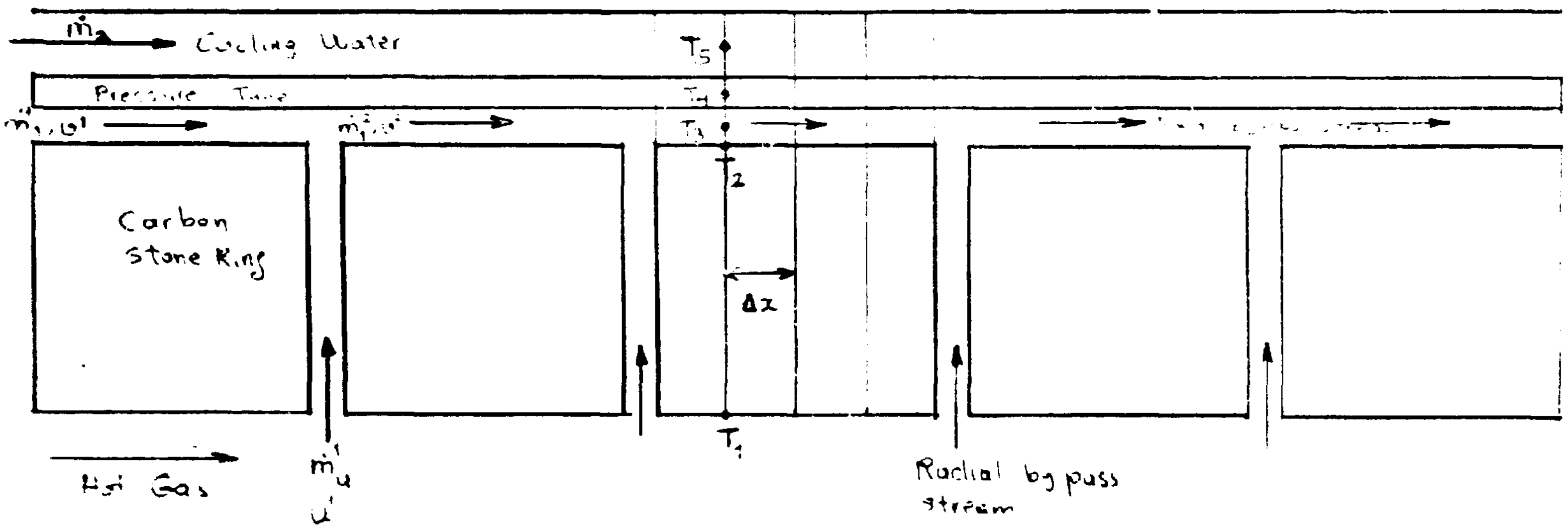

Figure 3 — Model for Energy Equation

Cp_1 = specific heat at constant pressure of axial by-pass stream, Ws/K kg

T_3 = temperature of the axial leakage stream one step before T_3, °C

h_4 = heat transfer coefficient between the outer surface at the pressure tube and the cooling water, W/Km²

h_5 = heat transfer coefficient between the cooling water an the ambient air, W/Km²

$\dot{m}_2$ = mass flow rate of the cooling water, Kg/s

Cp_2 = specific heat of cooling water, Ws/K kg

T_5 = temperature of the cooling water one step before T_5, °C.

The radial temperature profile in the carbon stone is taken as:

$$T = \frac{T_2 - T_1}{\ln \frac{D_2}{D_1}} \ln \frac{2r}{D_1} + T_1$$

which corresponds to the radial heat conduction with constant thermal conductivity. Thus, angular and axial heat conduction are not considered. This assumption simplifies the problem by avoiding radial and angular divisions in the model. Then, the heat flow across a ring of lenght Δx is given by

$$Q = -2\pi\Delta x \frac{T_2 - T_1}{\ln \frac{D_2}{D_1}}$$

which was used to derive equations (4) and (5).

In the derivation of equation (7), axial conduction in the pressure tube is neglected. Also, the temperature drop across the tube wall is taken as zero.

The convective heat flow into the axial by-pass stream v from the radial by pass stream u is accounted for by an energy balance at the intersection of these two streams:

$$(\dot{m}_u + \dot{m}_1)\, Cp_1\, T_3^* = \dot{m}_u C_{pu}\, T_{gas} + \dot{m}_1 C_{p1}\, T_3 \qquad \text{for } u > 0 \qquad (9)$$

$$T_3^* = T_3 \qquad (9^1)$$

where Cp_u and $\dot{m}_u$ are respectively the specific heat and mass flow of the radial by-pass stream u. T_3^* at the beginning of the rings 2 to 5 is calculated from Eqs (9) and (9^1). Tgas is temperature of the hot gas flowing in the duct. Thus, it is assumed that the radial stream u reaches axial stream v without any temperature drop. In reality, stream u becomes colder as it gives off heat to the carbon stone. This effect however could not be considered within the framework of this simplified model.

Solution of the system given by Eqs. (4), (5), (6), (7), and (8) starts by assuming T_3^* = Tgas and

$T_5^* = T_{in}$, where Tin is the inlet temperature at the cooling water. The system of the 5 equations and 5 unknowns is solved again by Gauss elimination to obtain the 5 unknown temperatures (T_1, T_2, T_3, T_4 and T_5) at the first axial position (at a distance Δx from the entrance) of the first ring. The subsequent steps are solved similarly T_3^* and T_5^* being now T_3 and T_5 at the previous axial step until second ring is reached. For the first axial position of the second ring (again at a distance Δx from its beginning), T_3^* is calculated from Eqs. (9) or (9[1]).

The calculation is carried out similary for the remaining rings.

The results of the temperature calculations are shown in the next section.

III – RESULTS

The cumputer program was first run to study the effect of sealing the gaps in the hot gas duct. The hot gas in the duct was choosen as helium, at 40 bars and 300° C, flowing at a velocity of 70 m/s. The friction factors for the helium in the duct and all by-pass stream have been taken as .03. The pressure drop in the duct per ring (ℓ_v = .8 m) has been calculated as .003 bar.

Figure 4 shows the velocity distribution in the duct when none of the gaps are sealed. In this case, an interesting result is obtained: there is no radial leakage from the hot gas to the pressure tube. There is only the axial by-pass stream v, with a constant velocity of 5 m/s.

Figure 5 shows the case where the axial gap between the first ring and the pressure tube is closed. The hot gas enters through the radial gaps 1, 2 and 3 with respective velocities at 6.5 m/s, 1.3 m/s and .09 m/s; flows in the axial gaps 2, 3, 4 and leaves through the last axial gap.

In Figure 6, the first axial and the first radial gaps are both closed, and the velocities are the same as in the previous, only shifted to the right by one ring.

It can be observed that sealing the first rings does not help reducing the by-pass streams in the remaining rings. In Figure 7, the first and last axial rings are sealed, and the results is similar to that of Figure 5, except that the axial by-pass stream leaves through the last radial gap instead of the last axial gap.

In Figure 8, the first and last axial and radial gaps are closed. The hot gas enters tnrough the second radial gap, flows in the third axial gap, and leaves through the third gap. Therefore, to eliminate leakage flows from the hot gas to the pressure tube, almost all the gaps must be sealed.

For the solution of the energy equation, the heat transfer coefficient h_1 between the hot gas and the inner surface of the rings has been taken as 1000 W/Km². The same value was used for h_4, the heat transfer coefficient between the cooling water and pressure tube. The water pipe was considered as completely insulated, giving h_5 = 0. The heat transfer coefficients h_2 and h_3 for the axial by-pass stream have been calculated from $Nu_2 = Nu_3 = 4$ (laminar flow), which gave $h_2 = h_3$ = 50 W/Km². Since this is a low value, program was run once with $h_2 = h_3$ = 500 W/Km². The cooling water inlet temperature and mass flow rate were taken as 20°C and 0.1 Kg/s respectively for all runs.

Figure 9 shows the velocity and temperature dsitributions for V = 70 m/s with none at the gaps sealed, and $h_2 = h_3$ = 50 W/Km². The temperature at the inner surface of carbon stone, T_1, remains a few degrees below the gas temperature (Tgas = 300°C); its outer surface temperature, T_2, drops to about 230°C at the end of duct. The axial by-pass stream, entering at T_3 = 300°C, is cooled down to about 190°C. The pressure tube is heated from about 40°C to 145°C, and the cooling water from 20°C to about 140°C.

In Figure 10, the conditions are the same as in the previous case, but the helium velocity has

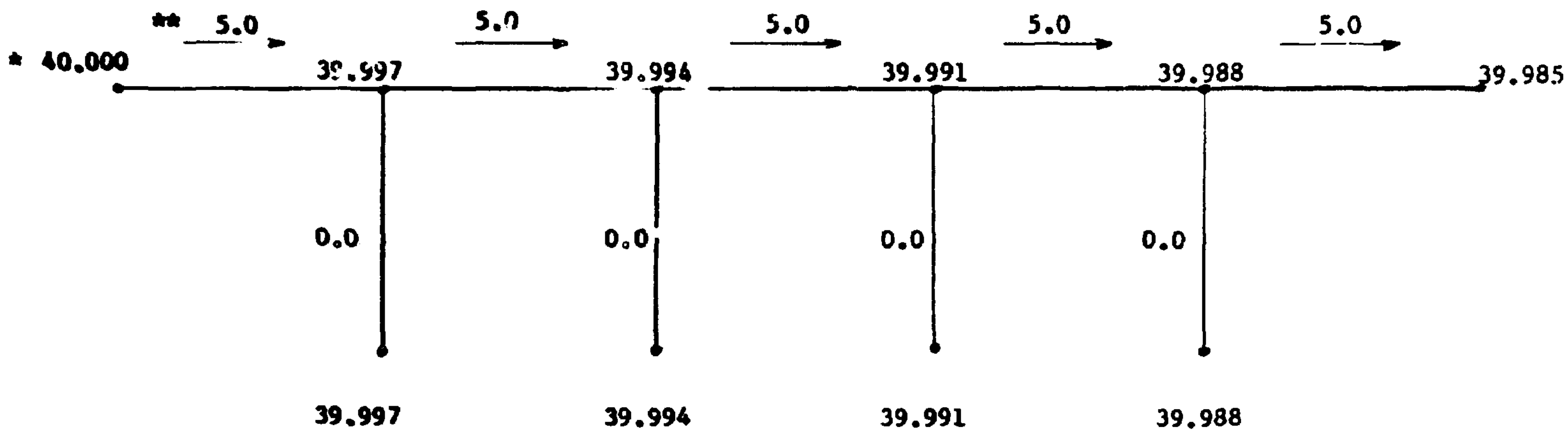

Figure 4 — Velocity Distribution, no Gaps Sealed, V = 70 M/S
** Velocities: m/s
* Pressure: bar

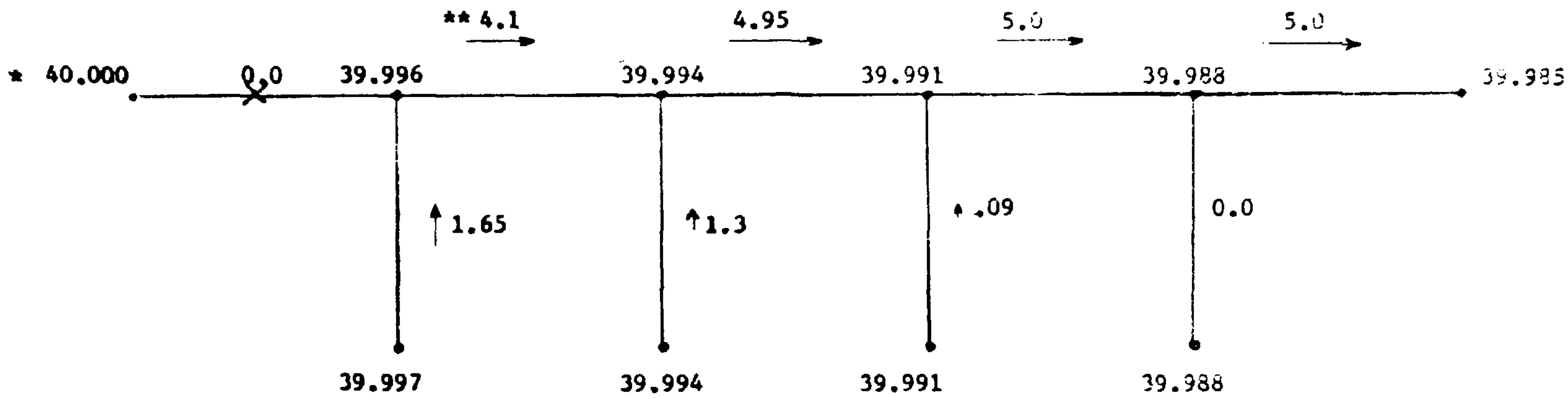

Figure 5 — Velocity Distribution, First Axial Gap Sealed, V = 70 M/S
** Velocities: m/s
* Pressures: bar

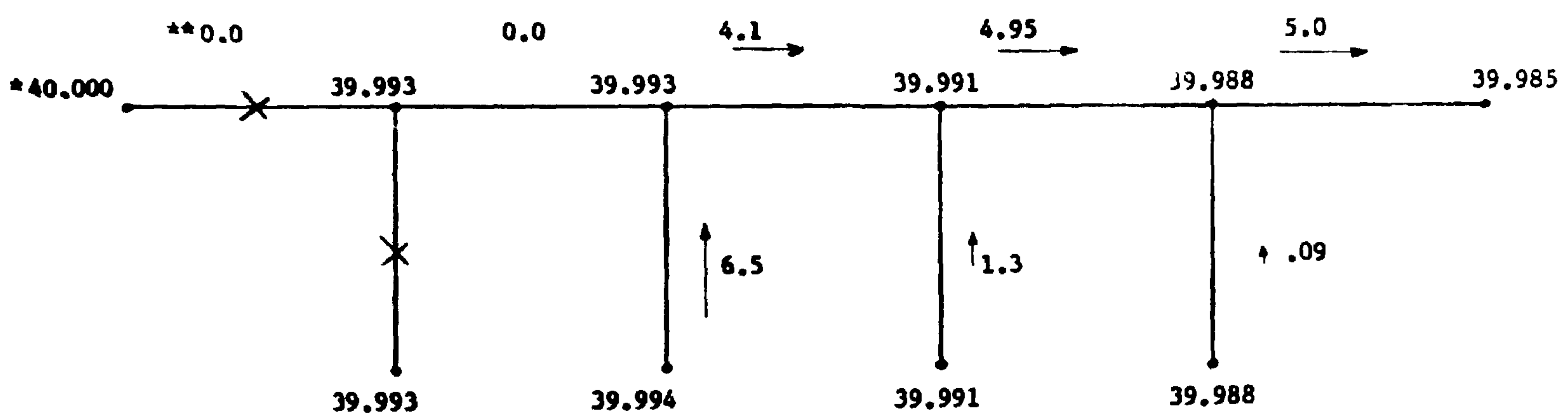

Figure 6 — Velocity Distribution, First Axial and Radial Gaps Sealed V = 70 M/S
** Velocities: m/s
* Pressures: bar

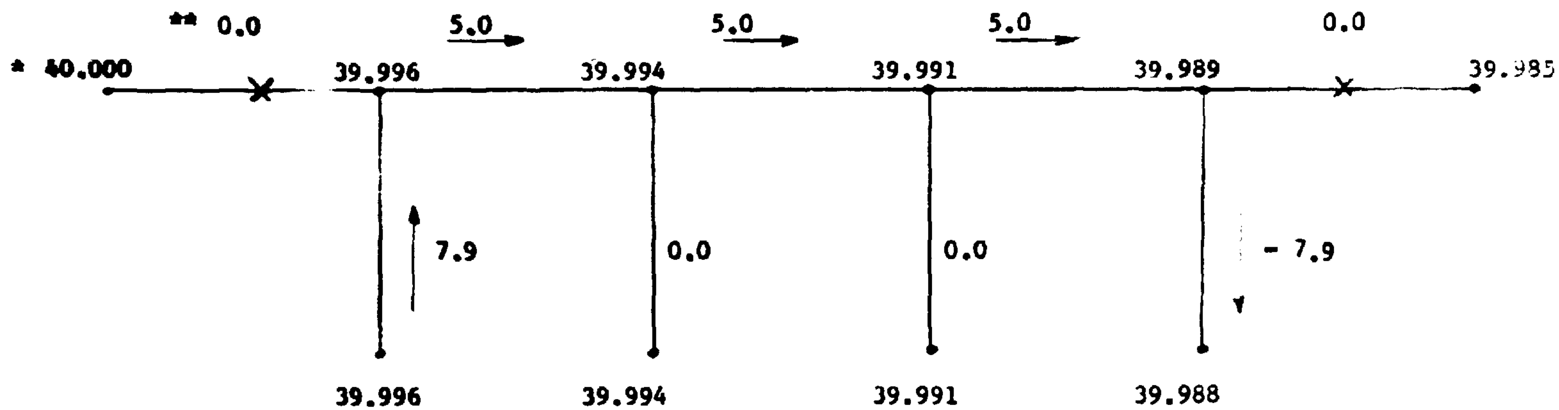

Figure 7 — Velocity Distribution, First and Last Axial Gaps Sealed, V = 70 M/S
** Velocities: m/s
* Pressures: bar

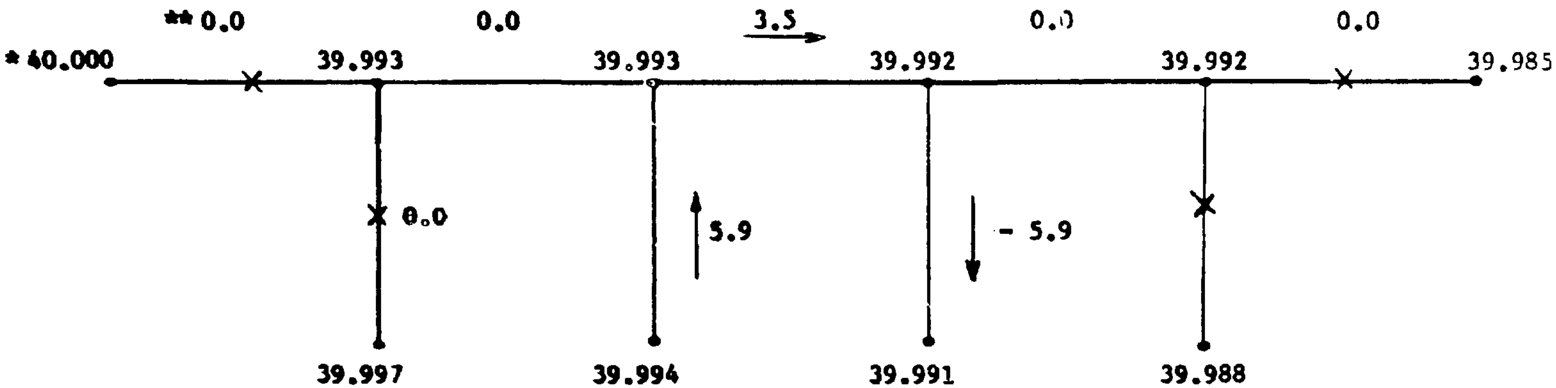

Figure 8 – Velocity Distribution, First and Last Axial and Radial Gaps Sealed, V = 70 M/S
** Velocities: m/s
* Pressures: bar

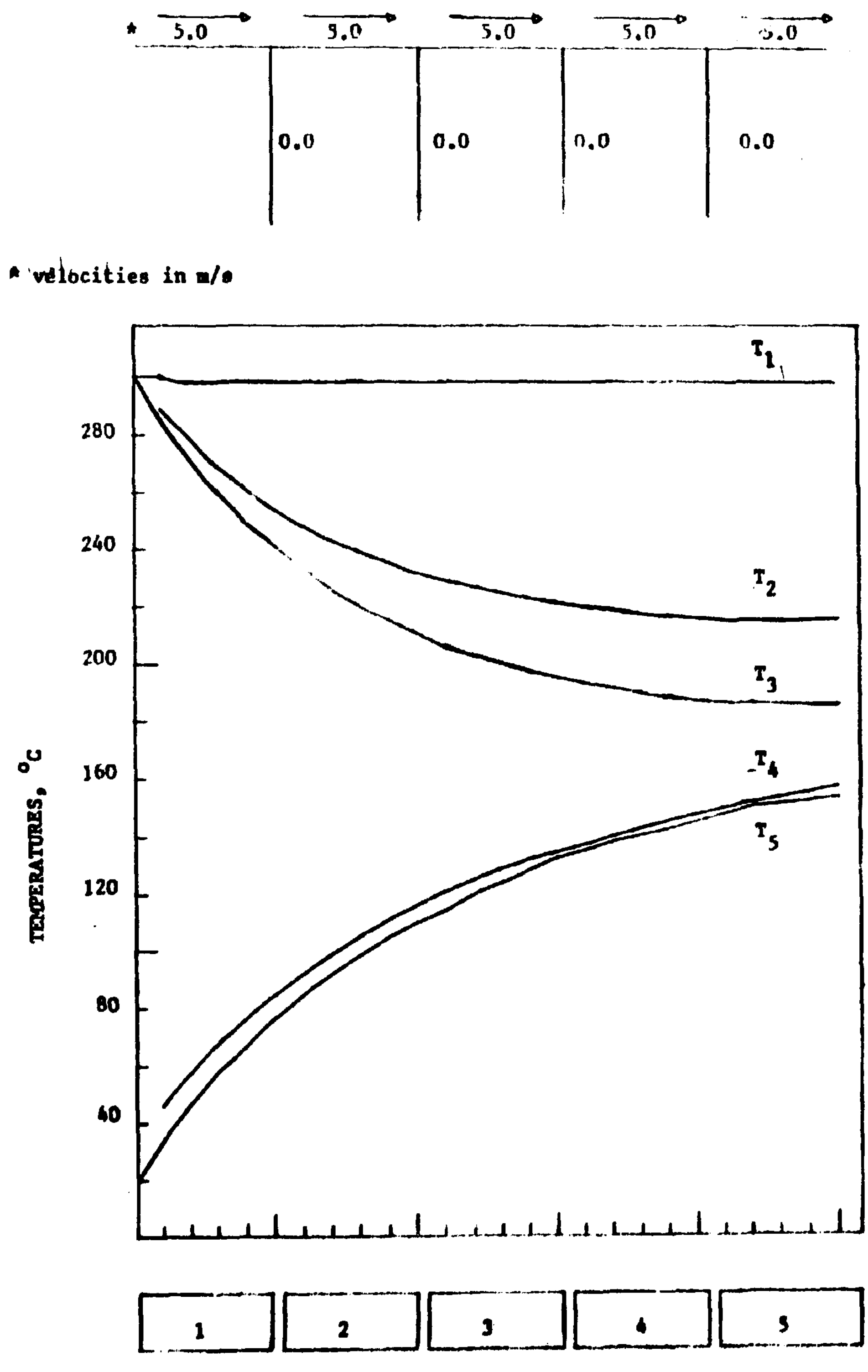

Figure 9 – Velocity and Temperature Distribution, no Gaps Sealed, V = 70 M/S, $h_2 = h_3 = 50$ W/Km^2

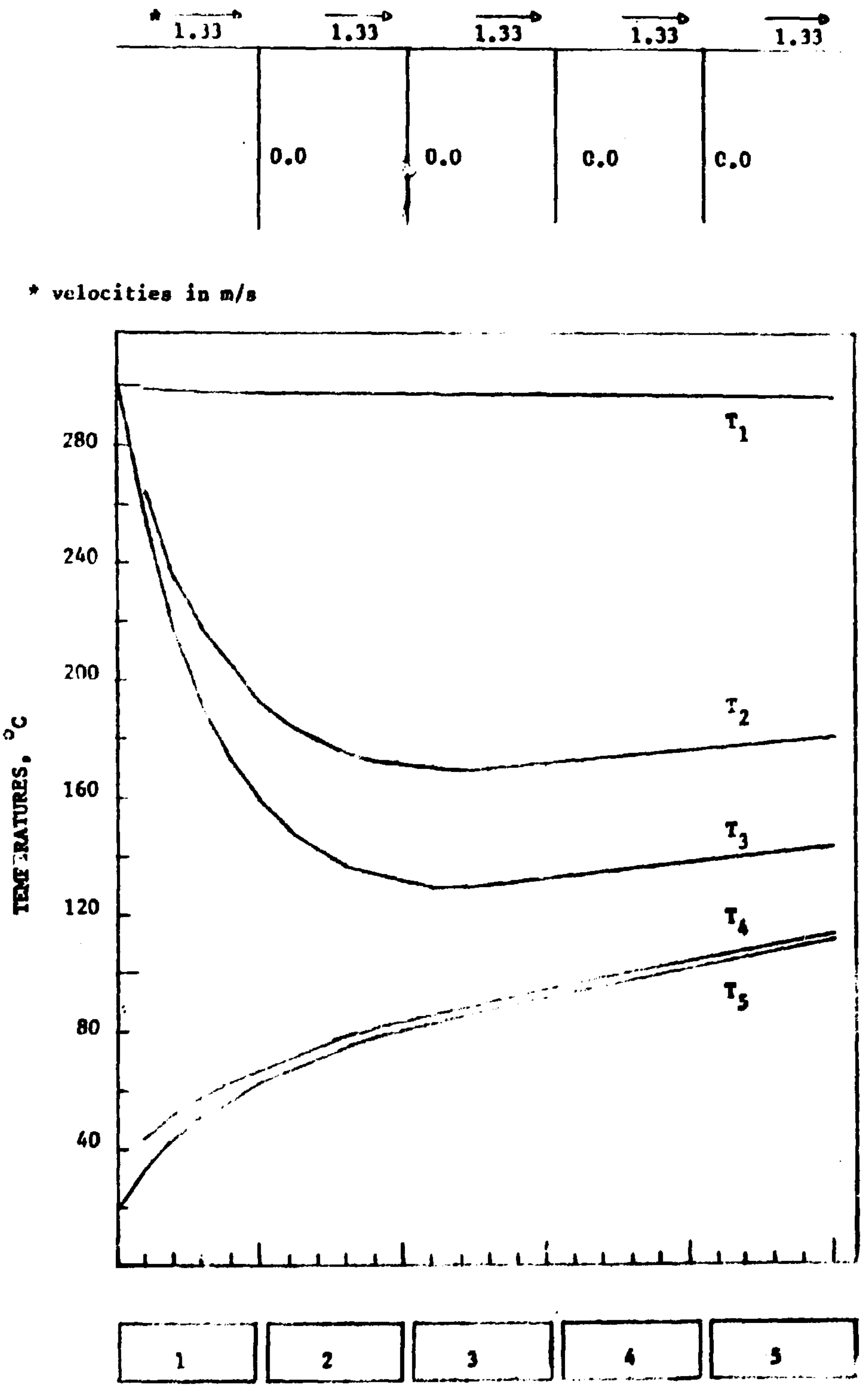

Figure 10 – Velocity and Temperature Distribution, no Gaps Sealed, V = 18 m/s, h_3 = h_3 = 60 W/Km²

been decreased from 70 m/s to 18 m/s. Thus, the pressure gradient for the main flow is smaller, leading to smaller axial by-pass velocities (1.33 m/s instead at 5 m/s). The temperature profiles are similar to those in Figure 9, but the cooling water, and consequently the pressure tube temperatures, are lower by about 40°C. On the other hand, the temperature of the outer surface of the carbon stone is kept at lower temperatures, causing higher temperature differences across the rings. Decreasing the gap G between the rings and the pressure tube would have the same effect as decreasing the velocity V of the hot gas, ie, lower mass flow rates for the axial by-pass stream, thus the temperature profiles with $V = 70$ m/s, but a smaller G would be similar to those in Figura 10.

In Figure 11, the conditions are the same as in Figure 9, bu the first and last both axial and radial gaps are closed. The hot gas enters through the second radial gap, raising the T_2 and T_3 to almost 290°C. The pressure tube and cooling water temperature are raised past 100°C at the end of the third ring, and they reach about 120°C at the end of the duct. These values are lower those in Figure 9, but the sharp increases in the carbon stone and pressure tube temperatures show that partial sealing can damage the duct; higher but smoother temperature profiles are obtained if no sealing is used.

Figure 12 shows the effect of heat transfer coefficients for the axial leakage stream, h_2 and h_3. This case was run with $V = 13$ m/s, $h_2 = h_3 = 500$ W/km^2. Comparison with Figure 10 ($V = 18$ m/s, $h_2 = h_3 = 50$ W/Km^2) indicates that the pressure tube wall temperature depends little of h_2 and h_3, but the temperature of the outer surface of the rings decreases when h_2 and h_3 increase. However, 500 W/Km^2 is too high a number, and was used to see the effect of h_2 and h_3 on the results.

Considerations on natural convection

As it was seen, natural convection is not calculed in the model. The velocities are calculed by evaluating the densities of various gas streams at the temperature of the hot gas and based on the rasulting velocities, temperatures are calculated. However, since the temperature of the main gas stream is higher than that of the axial and radial by-pass streams, some natural convection may occur in the system. Specially, on the upper half of a horizontal duct, main gas stream with a lower density is located below the axial by-pass stream with a higher density. This is not a stable condition and at high pressures (high densities), natural convection could be possible. The equations of conservation of momentum can be modified as fallows to account for density differences and body forces:

Conservation of momentum for the axial by-pass stream:

$$p^j - p^{j-1} + \frac{l_v^j \rho_v^j f_v \, v^j |v^j|}{2 DH_v^j g_c} = 0 \tag{10}$$

Conservation of momentum for the radial by-pass stream:

$$p^j - p_g^{j+1} + \frac{l_u^j \rho_u f_u \, u^j |u^j|}{2 DH_u^j g_c} = \frac{l_u^j \rho_u^j \bar{g}}{g_c} \tag{11}$$

Since the gas densities ρ_v^j and ρ_u^j are functions of temperature, the equations of conservation of mass, momentum and energy are coupled now, so they must be solved simultaneously to consider free convection. An itirative method was used by Konuk[5] to solve a natural convection problem. The equations of conservation of mass and momentum are solved first with initial guesses of temperatures,

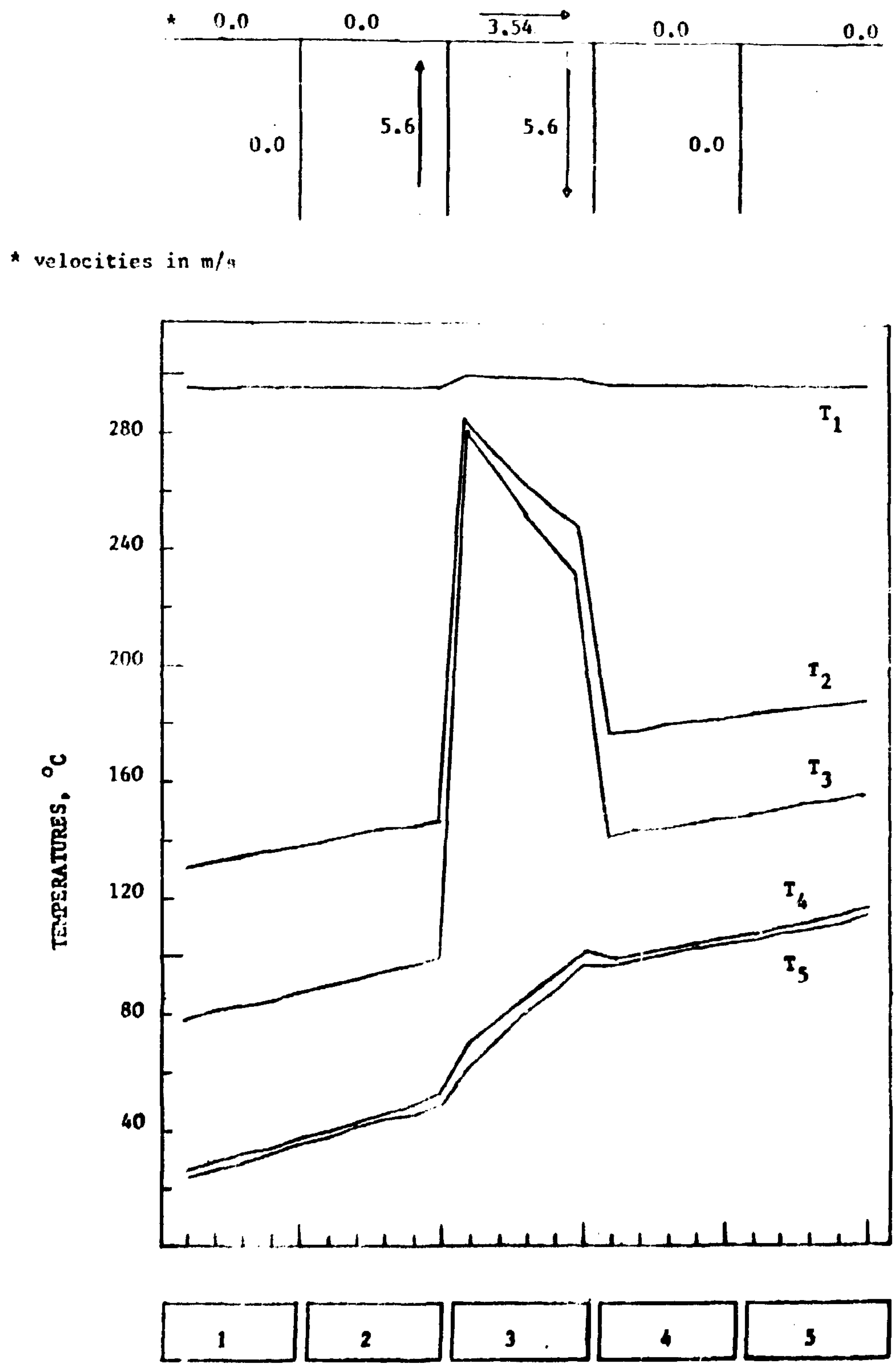

Figure 11 -- Velocity and Temperature Distribution, First and Last Axial and Radial Gaps Sealed. V = 70 m/s, $h_3 = h_5$ = 50 W/km²

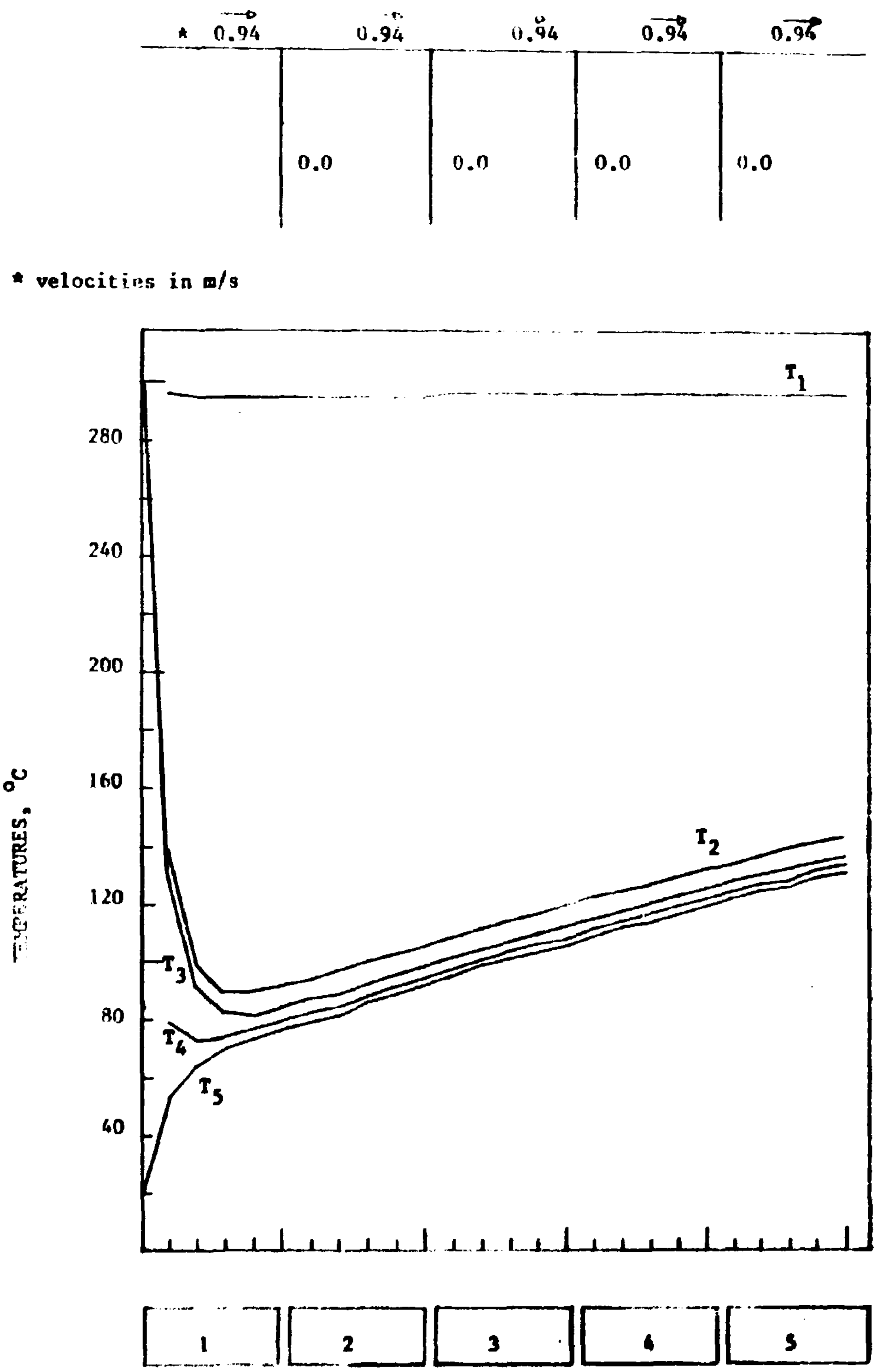

Figure 12 – Velocity and Temperature Distribution, no Gaps Sealed V = 13 m/s, $h_g = h_s = 500$ W/km^2

then the energy equation is solved using the calculated velocities. In the second itiration equations of conservation of mass and momentum are solved by evaluating the gas densities at the calculated temperatures, and then the energy equation is solved using the resulting velocities. The procedure is continued until temperatures converge. A similar method can be used for the present problem.

IV – CONCLUSIONS

The model developed allows the calculation of by-pass streams and temperatures in a hot gas duct internally insulated by carbon stone. The results presented, although qualitative can be helpful to find out the effects of the duct geometry on its performance.

To obtain quantitative results, the model should be improved by including natural convection and possibly making 4 angular divisions, 90° each. Also, it would be necessary to use realistic values for the radial gaps between the rings, and consider variation of the axial gap along the circoference due to non-centering of the rings in the pressure tube. Heat transfer coefficients h and friction factors f can be calculated more accuratly by introducing known experimental correlations for f and Nu into the computer program.

After the model has been improved, comparisons can be made with experimental results given in (3) and (4).

APPENDIX

COMPUTER PROGRAM

```
      DIMENSION X(20),V(10),C(10),PN(10),VN(10),CN(10),AV(10),AC(10)
      DIMENSION DL(10),DPV(10),DPL(10),PO(10),PL(10),PV(10) ,ALI(10)
      DIMENSION NP(10)
      COMMON A(20,8),IC(20,8),IN2(20),B(20),XN(20)
      DATA V,C,VN,CN/40*.1/
      DATA PN,X/30*.1/
      DATA AC,DC/10*.6,10*.001/
      DATA PV,PL/20*.03/
      DATA NP/10*5/
      DATA D1,D2,D3,D4,D5/.627,.527,.5302,.5702,1./
      RC=.22
      RL=3.2
      KOUNT=0
      CC=10.**5
      W=.5
      TOL=10.**(-5)
      TOL=10.**(-7)
      KMAX=20
      EPS=.01
      KMAX=10
      II=5
      III=II-1
      N=2*III+1
      DIO=.5302
      CSOD=.527
      CSID=.627
      DP=-.003
      PO(1)=40.
      DL(III)=10.**(-6)
      DL(II)=10.**(-6)
      DO 1 I=1,II
      DPV(I)=DIO-CSOD
      DPV(I)=10.**(-6)
      DPV(III)=10.**(-6)
      DPL(I)=2.*DL(I)
      AV(I)=3.1416*DIO*DPV(I)/2.
    1 AL(I)=3.1416*DL(I)*CSID
      DO 3 I=1,II
    3 PO(I+1)=PO(I)+DP
      WRITE(6,100)
   66 L=0
      DO 2 I=1,III
      L=L+1
      A(L,1)=-AV(I)
      IC(L,1)=L
      A(L,2)=-AL(I)
      IC(L,2)=L+2
      A(L,3)=AV(I+1)
      IC(L,3)=L+3
      B(L)=.0
      IN2(L)=3
      L=L+1
      A(L,2)=1.
      IC(L,2)=L
      A(L,1)=+RL*PV(I)*ABS(V(I))*AL(I)/(DPV(I)*2.*GC)
      IC(L,1)=L-1
```

```
      IF(I.EQ.1)GOTO 10
      A(L,3)=-1.
      IC(L,3)=L-3
      INZ(L)=3
      B(L)=.0
      GOTO 20
   10 B(L)=FQ(1)
      INZ(L)=2
   20 L=L+1
      A(L,1)=1.
      IC(L,1)=L-1
      A(L,2)=+WC*FL(I)*ABS(Q(I))*(CSCE-CSTL)/(4.*CC*DFQ(I))
      IC(L,2)=L
      B(L)=FQ(I+1)
    2 INZ(L)=2
      L=L+1
      A(L,1)=-1.
      IC(L,1)=L-2
      A(L,2)=+WC*FV(II)*ABS(V(II))*AL(II)/(DFV(II)*2.*GC)
      IC(L,2)=L
      B(L)=-FQ(II+1)
      INZ(L)=2
      CALL SPARAT(N,TOL)
      KOUNT=KOUNT+1
      DO 25 I=1,II
      VN(I)=XN(3*I-2)
      FN(I)=XN(3*I-1)
      QN(I)=XN(3*I)
   25 WRITE(6,1010)I,VN(I),QN(I),FN(I)
      DO 35 I=1,N
      IF(KOUNT.EQ.MAX)GOTO 50
   35 IF(ABS((X(I)-XN(I))/XN(I)).GE.EPS)GOTO 55
      GOTO 50
   55 DO 45 I=1,II
      V(I)=W*V(I)+(1.-W)*VN(I)
   45 Q(I)=W*Q(I)+(1.-W)*QN(I)
      GOTO 60
   50 CONTINUE
      AM2=.1
      TL=300.
      TOL1=20.
      AF1=1000.
      AF2=50.
      AF3=50.
      AF4=1000.
      AF5=.0
      AKC=2.9
      CF1=5000.
      CF2=4000.
      XN(3)=TG
      XN(5)=20.
      WRITE(6,1100)
      DO 70 I=1,II
      APL=WC*AL(I)*QN(I)
      APV=WC*AV(I)*VN(I)
      DX=AL(I)/AN(I)
      DXF=DX*3.1416
      C1=AKC/ALOG(D2/D1)
```

```
      WRITE(6,111)J
      NH1=NH(1)
      DO 71 J=1,NH1
      L=1
      A(L,1)=-DXF*(D1*AH1+2.*[illegible])
      IC(L,1)=L
      A(L,2)=2.*DXF*D1
      IC(L,2)=L+1
      B(L)=-DXF*D1*AH1*T0
      INZ(L)=2
      L=2
      A(L,1)=2.*DXF*D1
      IC(L,1)=L-1
      A(L,2)=-DXF*(2.*D1+D2*AH2)
      IC(L,2)=L
      A(L,3)=DXF*D2*AH2
      IC(L,3)=L+1
      B(L)=.0
      INZ(L)=3
      L=3
      A(L,1)=DXF*D2*AH2
      IC(L,1)=L-1
      A(L,2)=-DXF*(D2*AH2+D3*AH3)-APV*CP1
      IC(L,2)=L
      A(L,3)=DXF*D3*AH3
      IC(L,3)=L+1
      B(L)=-APV*CP1*XN(3)
      INZ(L)=3
      L=4
      A(L,1)=DXF*D3*AH3
      IC(L,1)=L-1
      A(L,2)=-DXF*(D3*AH3+D4*AH4)
      IC(L,2)=L
      A(L,3)=DXF*D4*AH4
      IC(L,3)=L+1
      B(L)=0
      INZ(L)=3
      L=5
      A(L,1)=DXF*AH4*D4
      IC(L,1)=L-1
      A(L,2)=-DXF*(D4*AH4+D5*AH5)-AP2*CP2
      IC(L,2)=L
      B(L)=-DXF*D5*AH5*TC(L) -AP2*CP2*XN(5)
      INZ(L)=2
      CALL SPARAT(5,ICL)
      WRITE(6,112)XN(1),XN(2),XN(3),XN(4),XN(5),J
   71 CONTINUE
      IF(APL.GT.0) XN(3)=(APV*XN(3)+APL*TG)/(APV+APL)
   70 CONTINUE
  110 FORMAT('1',3X,'T1',10X,'T2',10X,'T3',10X,'T4',10X,'T5',/)
  111 FORMAT('0',5X,'RING NUMBER',I2,/)
  112 FORMAT('0',3X,5F10.1,3X,'J=',I2)
  100 FORMAT('1','1',10X,'VN',10X,'UN',10X,'FN',/)
  101 FORMAT('0',I2,3E13.5)
      STOP
      END
```

```
C      SUBROUTINE SPAMAT
C      SOLVES A SYSTEM OF LINEAR EQUATIONS BY GAUSS ELIMINATION METHOD
C      USING SPARSE MATRIX TECHNIQUES
       SUBROUTINE SPAMAT(N,TOL)
       COMMON A(20,8),IC(20,8),INZ(20),B(20),X(2
       NZM=1
       DO 1 I=1,N
       X(I)=FLOAT(I)
     1 CONTINUE
       N1=N-1
       DO 10 I=1,N1
       NZ=INZ(I)
       AM=A(I,1)
       K2=1
       DO 15 K1=1,NZ
       IF(ABS(A(I,K1))-ABS(AM))15,15,16
    16 AM=A(I,K1)
       K2=K1
    15 CONTINUE
       IF(IC(I,K2).EQ.I)GO TO 21
       CALL MODCOL(I,N,K2)
    21 A1=A(I,K2)
       IF(ABS(A1)-TOL)101,101,22
   101 WRITE(6,102)I,A1
   102 FORMAT(/,10X,'NA LINHA ',I3,5X,'VALOR MAX.=',E8.4)
    22 CALL TRANS(K2,I)
       B(I)=B(I)/A1
       DO 23 K=1,NZ
    23 A(I,K)=A(I,K)/A1
       IP1=I+1
       DO 40 I1=IP1,N
       IF(INZ(I1).GT.NZM)NZM=INZ(I1)
       NZ=INZ(I1)
       DO 41 K=1,NZ
       IF(IC(I1,K).EQ.I)GO TO 42
    41 CONTINUE
       GO TO 40
    42 A2=A(I1,K)
       CALL TRANS(K,I1)
       NZ=INZ(I)
       NZ1=INZ(I1)
       DO 50 K1=1,NZ1
       DO 60 K=1,NZ
       IF(IC(I1,K1).EQ.IC(I,K))GO TO 51
    60 CONTINUE
       GO TO 50
    51 A(I1,K1)=A(I1,K1)-A2*A(I,K)
    50 CONTINUE
       NZ1=INZ(I1)
       DO 70 K=1,NZ
       DO 80 K1=1,NZ1
       IF(IC(I,K).EQ.0)GO TO 70
       IF(IC(I1,K1).EQ.IC(I,K))GO TO 70
    80 CONTINUE
       DO 71 K1=1,NZ1
       IF(IC(I1,K1).EQ.0)GO TO 72
```

```
11    CONTINUE
      INZ(II)=INZ(II)+1
      K1=NZ1+1
      NZ1=NZ1+1
12    A(II,K1)=-AZ*A(I,K)
      IC(II,K1)=IC(I,K)
13    CONTINUE
      B(II)=B(II)-AZ*B(I)
40    CONTINUE
10    CONTINUE
      B(N)=B(N)/A(N,1)
      NMS1=N-1
      DO 90 J=1,NMS1
      IB=N-J
      IA=N
      NZ=INZ(IB)
      DO 90 M=1,J
      DO 91 K=1,NZ
      IF(IC(IB,K).EQ.IA)GO TO 92
91    CONTINUE
      GO TO 93
92    B(IB)=B(IB)-A(IB,K)*B(IA)
93    IA=IA-1
90    CONTINUE
      DO 94 I=1,N
      L=IFIX(X(I))
      X(I)=B(L)
94    CONTINUE
      WRITE(6,103)NZM
103   FORMAT(/,1X,'NUM MAX OF ELEM.=',I4)
      RETURN
      END
```

```
      SUBROUTINE REDUCE(I,N,K2)
      COMMON A(20,8),IC(20,8),IN2(20),B(20),X(20)
      K=K2
      IC1=IC(I,K)
      N2=IN2(I)
      DO 30 IK=1,N2
      IF(IC(I,IK).EQ.I)GO TO 31
30    CONTINUE
      GO TO 32
31    IC(I,IK)=IC1
32    IC(I,K)=I
      DO 11 KX=1,N
      IF(X(KX).EQ.FLOAT(IC1))GO TO 12
      GO TO 13
12    X(KX)=I
      GO TO 11
13    IF(X(KX).EQ.FLOAT(I))X(KX)=IC1
11    CONTINUE
      DO 10 II=1,N
      IF(II.EQ.I)GO TO 10
      N2=IN2(II)
      DO 20 K=1,N2
      IF(IC(II,K).EQ.I)GO TO 21
      IF(IC(II,K).EQ.IC1)IC(II,K)=I
      GO TO 20
21    IC(II,K)=IC1
20    CONTINUE
10    CONTINUE
      RETURN
      END
```

```
      SUBROUTINE TRANS(K,I)
      COMMON A(20,8),IC(20,8),IN2(20),B(20),X(20)
      N2=IN2(I)
      K1=K+1
      IF(K1.GT.N2)GO TO 20
      DO 10 K2=K1,N2
      IC(I,K)=IC(I,K2)
      A(I,K)=A(I,K2)
      K=K+1
10    CONTINUE
      IC(I,K2)=0
      A(I,K2)=0
      GO TO 30
20    IC(I,K)=0
      A(I,K)=0
30    IN2(I)=IN2(I)-1
      IF(IN2(I).EQ.0)IN2(I)=1
      RETURN
      END
```

RESUMO

Foi desenvolvido um modelo matemático para calcular a distribuição de temperatura e fluxo secundário num canal de gás quente isolado internamente com anéis de carvão de pedra. As equações de conservação de massa e momento são resolvidas para um tubo a fim de obter as velocidades radial e axial. A seguir, é solucionada a equação da energia passo a passo, a fim de obter a distribuição de temperatura ao longo do tubo.

Os resultados, embora qualitativos devido às simplificações do modelo, são úteis para estudar os efeitos de geometria do tubo no desempenho do isolamento térmico.

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