UNSTEADY CONDUCTION AND FREE CONVECTION FOR THE DESIGN OF A STEAM PIPING

Carlos Alberto de Melo
Universidade Federal de Uberlândia
carlosam@mecanica.ufu.br

Abstract: *As steam flows through the distribution piping additional steam condensation is continuously taking place. This condensate has to be removed immediately to do not cause severe waterhamering conditions in the pipeline damaging all equipments installed. An unsteady state conduction model was developed to calculate the mass flow rate of condensate formed, due to heat dissipation to surroundings and due to heat stored in pipe wall and in thermal insulation. This load of condensate is used for the design of recovery condensate system, which contains the steam trap. The transient outer heat transfer coefficient between the insulation outer surface and surrounds was evaluated, iteratively at each instant, by free convection. Unsteady conduction equation, with boundaries and initial conditions was solved using the control volume difference finite formulation, presented by Spalding and by Patankar. The full implicit method was used to formulate the time term. The discretization equations satisfy the convergence Scarborough criteria. The load of condensate was analyzed in function of steel and insulation Stefan numbers. Considering the coupling between the pressure loss of the condensate piping system (system curve) and the pressure loss obtained from the characteristic curve of the steam trap, a governing global equation appears. The mass flow rate of condensate, that can flows in the recovery condensate system, is obtained by the solution of the global equation. This solution, that is called the effective condensate, must be greater than or equal to the load of condensate estimated in the unsteady conduction model. In the iterative process, the Lagrange or the spine cubical interpolating function is used to obtain the intermediate points of the characteristic curve and the secant method is used to calculate the root of the global equation.*

Key Words: *steam piping, unsteady conduction, free convection and condensate recovery system.*

1. INTRODUCTION

As steam flows through the distribution piping additional condensation is continuously taking place, due *to heat* dissipation to surroundings and due to heat storage in the pipe wall and in the thermal insulation.

Water hammer is caused by the accumulation of condensate (water) trapped in a portion of horizontal steam piping. The velocity of the steam flowing over the condensate causes ripples in the water. Turbulence builds up until the water forms a solid mass or slug, filling the pipe. This slug of condensate can travel at the speed of the steam and strike the first elbow in its path with a force comparable to a hammer blow.

Steam trap, as indicated by McCauley (1995), has purpose to remove condensate from piping to prevent damage to the piping and control valves, while assuring that steam users receive dry steam.

Adequately sized drip pockets or collecting leg at the bottom of piping or upstream of heat exchanges, collect condensate which then flows to the steam trap, as indicated in Figure 1. The trap should discharge the condensate.

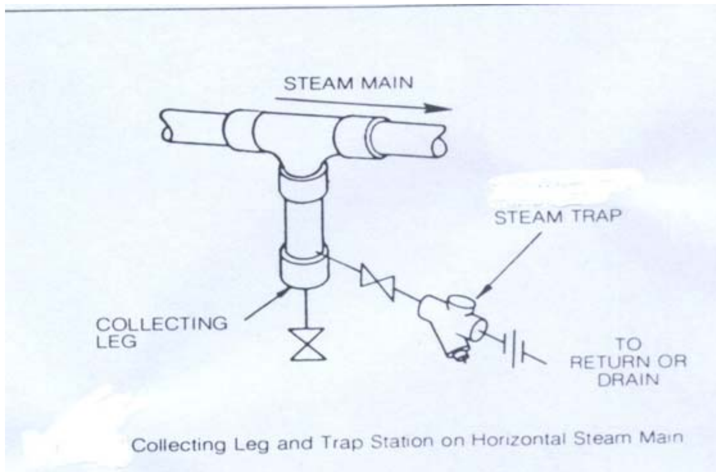

Figure 1. Collecting leg and steam trap on horizontal steam main

The c*ondensate loads* are relatively small and constant while in normal operation. Startup loads or during warming up, due the heat stored in metal and insulation walls, can be heavier. Boiler carry-over produces slugs of condensate, which are unpredictable in magnitude and frequency. The d*rainage of condensate to trap* is usually by gravity with the steam trap installed below the steam line. Occasionally piping in trenches or underground have steam traps installed above the pipe, but the condensate collecting point is below the pipe, as indicated in figure 2.

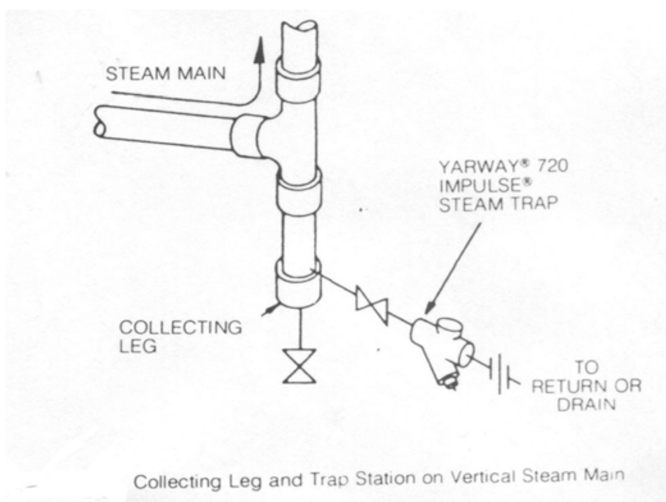

Figure 2. Collecting leg and steam trap on vertical steam main

The models for calculate the load of estimated condensate are simplified and oversized, as proposed by Telles (1982), by Crooker and King (1987) and by McCauley (1995). They consider saturation uniform temperature condition, in pipe wall and insulation and a startup time varying between 5 to 10 minutes.

Melo (2005) developed a semi empirical method for design of a steam piping and its condensate recovery system, nevertheless was not considered the unsteady state condition for outer heat transfer coefficient and for Biot number.

The purpose of this paper is develop a unsteady conduction and free convection model, where the Biot number is determined iteratively, at each instant. The temperature field in pipe wall and insulation allows calculate, with good precision, the load of condensate for the design of steam piping and its condensate recovery system.

2. UNSTEADY STATE CONDUCTION AND FREE CONVECTION MODEL

Figure 3 shows the carbon steel pipe and its thermal insulation:

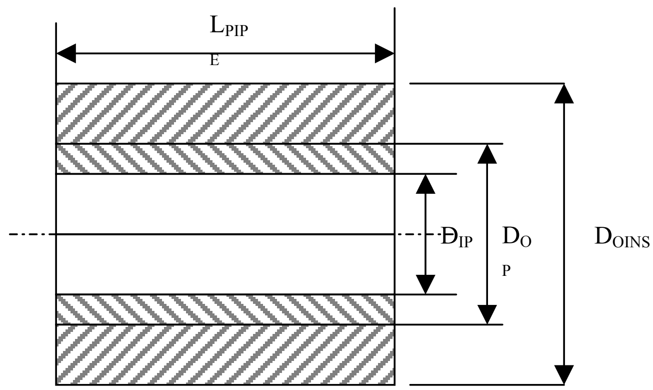

Figure 3 - Carbon steel pipe and its insulation

The subscripts are:

P - pipe
INS - insulation
IP - inner pipe
OP - outer pipe
OINS – outer insulation

The following unsteady heat conduction equation, in axis-symmetric coordinate, can be used to model the stored heat in pipe wall and insulation and the startup time, as:

$$\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \theta}{\partial R}\right) = \frac{\alpha_{INS}}{\alpha}\frac{\partial \theta}{\partial \tau} \tag{1}$$

$$\alpha = \alpha_P \ : \ \frac{D_{IT}}{D_{OINS}} \le R \le \frac{D_{OT}}{D_{OINS}} \ \text{(pipe region)} \ ; \ \alpha = \alpha_{INS} \ : \ \frac{D_{OT}}{D_{INS}} \le R \le 1 \ \text{(insulation region)}$$

$$R = \frac{D_{IT}}{D_{OINS}} \ ; \ \theta = 1 \tag{1.1}$$

$$R = \frac{D_{OT}}{D_{OINS}} \ ; \ \alpha = \alpha_{AV} \ ; \quad \alpha_{AV} \ \text{(will be defined in numerical process)} \tag{1.2}$$

$$R = 1 \quad ; \quad \frac{\partial \theta}{\partial R} = -Bi\ \theta \quad ; \quad Bi = \frac{h_O D_{OINS}}{K_{INS}} \tag{1.3}$$

h_o and Bi is determined iteratively at each instant.

$$\tau = 0 \quad : \quad \theta = 0 \tag{1.4}$$

The dimensionless variables are defined as follows:

$$\theta = \frac{T - T_{SUR}}{T_{SAT} - T_{SUR}} \quad ; \quad R = \frac{r}{2D_{OINS}} \quad ; \quad \tau = \frac{4\alpha_{INS} t}{D_{OINS}^2} \text{ (Fourier number)} \tag{2}$$

The Grashof number and Nusselt correlation, related by Holman (1983), in air outer layer temperature (AOL), that can be used to determine the outer heat transfer coefficient by free convection between the insulation outer surface and surrounds, is:

$$Gr = \frac{\rho_{AOL}^2 g \beta_{AOL} (T_{SAT} - T_{SUR})(\theta_{GOINS} - \theta_{SUR}) D_{OINS}^3}{\mu_{AOL}^2} \tag{3}$$

$$\text{Ray} = \text{Gr Pr} \tag{4}$$

$$\text{Nu} = 0{,}53(\text{Ray})^{0{,}25} \tag{5}$$

$$h_o = K_{AOL}\ \text{Nu} / D_{OINS} \tag{6}$$

$$Biot = \frac{h_0 D_{OINS}}{K_{INS}} \tag{7}$$

Air outer layer dimensionless temperature is defined as:

$$\theta_{AOL} = \frac{\theta_{GOINS} + \theta_{SUR}}{2} \tag{8}$$

Air outer layer properties ρ, α, K, ν, Pr and β in SI system were adjusted by least square method, considering table A-5 from Holman (1983):

$$\rho_{AOL} = 1{,}83714 - 2{,}3148E\text{-}03\ (T_{SAT} - T_{SUR})\,\theta_{AOL} \tag{9}$$

$$\mu_{AOL} = 8{,}676E\text{-}06 + 3{,}582E\text{-}08\ (T_{SAT} - T_{SUR})\,\theta_{AOL} \tag{10}$$

$$\alpha_{AOL} = -2{,}41766E\text{-}05 + 1{,}544E\text{-}07\ (T_{SAT} - T_{SUR})\,\theta_{AOL} \tag{11}$$

$$K_{AOL} = 4{,}03833E\text{-}03 + 7{,}41E\text{-}05\ (T_{SAT} - T_{SUR})\,\theta_{AOL} \tag{12}$$

$$\nu_{AOL} = \mu_{AOL} / \rho_{AOL} \tag{13}$$

The subscripts are:

SUR - surrounds
SAT - saturation
AOL - air outer layer
GOINS - guessed outer insulation
OINS - outer insulation

Equation (1), with boundaries conditions (1.1) to (1.3) and initial condition (1.4) is solved using the control volume difference finite formulation, presented by Spalding (1972) and by Patankar (1980). The full implicit method is used to formulate the time term.

The discretization equations satisfy the convergence criteria, as indicated by Scarborough (1958).

The procedure to solving algebraic equations is the tridiagonal matrix algorithm (TDMA).

Figure 4 shows the discretized control volumes:

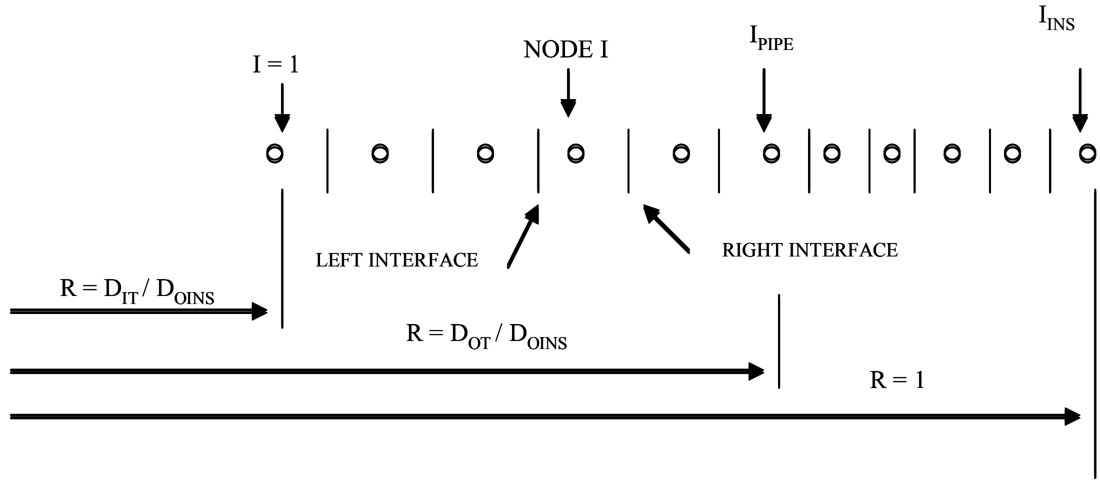

Figure 4 – Discretized control volumes

I_{PIPE} and I_{INS} are, respectively, the nodes of pipe and insulation outer surfaces.

$$\Delta R_1 = \frac{\frac{1}{D_{0ins}}\left(D_{OINS} - D_{IT}\right)}{I_{PIPE} - 1} \quad ; \quad \Delta R_2 = \frac{\frac{1}{D_{0ins}}\left(D_{OINS} - D_{OT}\right)}{I_{INS} - I_{PIPE}} \tag{14}$$

ΔR_1 and ΔR_2 are, respectively, the radial increments in pipe and insulation.

The heat dissipation is:

$$Q_{DISS} = h_o \pi D_{OINS} L\left(T_{SAT} - T_{SUR}\right)\left(\theta_{I_{INS}} - \theta_{SUR}\right) \tag{15}$$

The mass flow rate of condensate formed, due to heat dissipation, is calculated as:

$$\dot{m}_{DISS} = \frac{Q_{DISS}}{h_{lv}} \tag{16}$$

The mass flow rate of condensate formed, due to heat stored in pipe wall and insulation is calculated in the present model as:

$$\dot{m}_{stor} = \frac{\alpha_{INS} L}{\tau}\left[\rho_{ST} Ste_{ST} \sum_{I=1}^{I_{PIPE}} V_I \theta_I + \rho_{INS} Ste_{INS} \sum_{I_{PIPE}}^{I_{INS}} V_I \theta_I\right] \tag{17}$$

V_I is the dimensionless discretized control volume
Ste_{ST} and Ste_{INS} represent, respectively, steel and insulation Stefan numbers, defined as:

$$Ste_{ST} = \frac{C_{ST}\left(T_{SAT} - T_{SUR}\right)}{h_{lv}} \quad ; \quad Ste_{INS} = \frac{C_{INS}\left(T_{SAT} - T_{SUR}\right)}{h_{lv}} \tag{18}$$

The total load of condensate, which is called the estimated condensate, is:

$$m_C = F\left(\dot{m}_{DISS} + \dot{m}_{STOR}\right) \tag{19}$$

F is the safety factor, used by steam trap manufactures, varying from 2 to 5. This method uses the factor 3 for small nominal diameter. For big nominal diameter of steam pipe, the factor 1,5 is used. As indicated, this method is never oversized.

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3. DESIGN OF CONDENSATE PIPING

The design of condensate piping, that contains the steam trap, is done considering the coupling between the pressure loss of the condensate piping system (system curve), and the pressure loss of the characteristic curve of the steam trap.

Figure 5 shows the Layout of steam and condensate piping and Figure 6 shows system and steam trap characteristic curves.

The differential pressure that the condensate piping demand of the steam trap (system) can be written as:

$$\Delta p_{syst} = p_{cl} - p_{rc} + \rho_C g\left(z_{cl} - z_{bst} + z_{ast} - z_{rc}\right) - \frac{8 f\left(L_{bst} + L_{ast}\right)\dot{m}_C^2 v_c}{\pi^2 D_{ict}^5} \tag{20}$$

L_{bst} Sum of straight and component equivalent lengths before steam trap, L_{ast} sum of straight and component equivalent lengths after steam trap and D_{ict} is inner diameter of condensate tube

f is the friction factor, obtained by the Colebrook and White formula.

The maximum differential pressure is:

$$\left(\Delta p\right)_{\max} = p_{cl} - p_{ch} + \rho_C g\left(z_{cl} - z_{bst} + z_{ast} - z_{ch}\right) \tag{21}$$

p is pressure and z is the point height

The subscripts are:

c - condensate

cl - condensate collecting leg

ch - condensate header

bst - before steam trap

ast - after steam trap

The Lagrange function is used to obtain the interpolated points of the steam trap characteristic curve, as:

$$\Delta p_{inn} = \sum_{I=1}^{NP} \Delta p_I \prod_{J=1}^{NP} \frac{\left(\dot{m}_C - m_J\right)}{\left(m_I - m_J\right)} \qquad J \neq I \tag{22}$$

m_I represents the mass flow rate vector with NP positions (points of steam trap characteristic curve) and Δp_I represents the differential pressure vector with NP positions and correspond the mass flow rate m_I

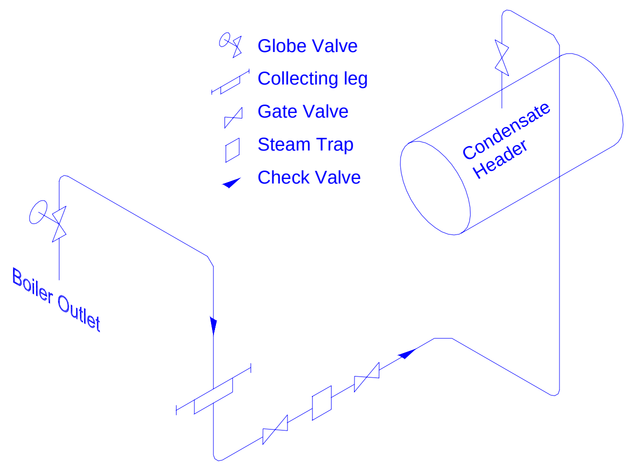

Figure 5. Layout of steam and condensate piping

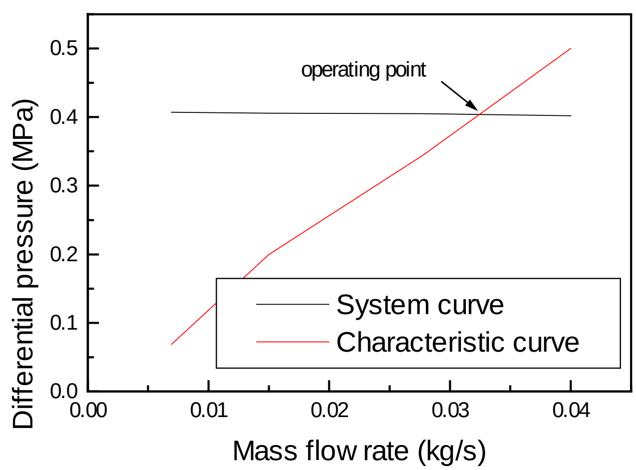

Figure 6. System and steam trap characteristic curves

As shown in Figure 6, we can write:

$$\Delta p_{syst} = \Delta p_{inn} \tag{23}$$

$$\Delta p_{syst} - \Delta p_{inn} = s\left(\dot{m}_C\right) = 0 \tag{24}$$

Equation (24) is the global governing equation. Its solution is the mass flow rate of condensate that will flow in the recovery condensate system.

The mass flow rate of condensate that will flow in the condensate recovery system is determined by secant method, as follows:

MA = m_1 ; MB = m_{NP}
Repeat (Iterative process)

SMA= s(MA)
SMB =s(MB)

$$MC = \frac{SMA * MB - SMB * MA}{SMB - SMA}$$

SMC = s(MC)
MA = MB
MB = MC
Until ABS(SMC) < Tolerance

The Lagrange routine was used to generate Δp_{inn} and the Colebrook routine was used to generate Δp_{syst}. The convergence value MC is the solution of equation (24) and represents the operating condensate mass flow rate.

Telles (1982) and steam trap manufacturers handbooks oversize the following mass flow rate of condensate formed, due to the heat stored in pipe wall and insulation,, as:

$$\dot{m}_{STOR} = \frac{\pi L(T_{SAT} - T_{SUR})}{4\Delta t_{STOR} h_{lv}} \left[\left(D_{OP}^2 - D_{IP}^2\right)\rho_{ST} C_{ST} + \left(D_{OINS}^2 - D_{OP}^2\right)\rho_{INS} C_{INS} \right] \tag{25}$$

4. RESULTS AND DISCUSSIONS

The results were obtained considering saturated steam, at pressure of 1 MPa (T_{SAT} = 453 K), flowing through the distribution piping of nominal diameter 150 mm (6"), schedule 40 (D_{IP} = 153,8 mm , D_{OP} = 168,2 mm), with straight length of 10,5 m, and a mass flow rate of 2,7777 kg/s (10000 kg/h) . In this case, the tube wall thickness withstands the internal steam pressure and the pressure loss was less than 3 percent of boiler outlet steam pressure.

The insulation tested, in above algorithm, was pre-molded calcium silicate or rock silicate. The outer wall insulation temperature of 321 K was less than 333 K (safety condition), for insulation thickness of 50 mm.

The straight length of condensate piping is 18 m ; $z_{ch} - z_{cl}$ = 7 m and $z_{ast} = z_{bst}$; Dict = 12,5 mm

Table 1 shows the parameters that were used in numerical algorithm:

Table 1. Numerical algorithm parameters

D_{IP} / D_{OINS}	D_{OP} / D_{OINS}	I_{PIPE}	I_{INS}	$\Delta\tau$	Δt
0,5735	0,6271	10	50	$1.x10^{-06}$	0,03934 s

Table 2 shows the parameters, obtained by unsteady conduction model..

Table 2. Results of unsteady state conduction model

T_{OWINS} (K)	h_o (W/m^2K)	Q_{DISS} (W)	Biot Steady state	τ Steady state	m_{DISS} (kg/s)	Q_{STOR} (W)	m_{STOR} (kg/s)	Condensate estimated (kg/s)
321	4,044	960,14	7,532	5,62E-02	4,755E-04	13515,4	6,6928E-03	2,1505E-02

Table 3 shows load of condensate parcels (m_{PIPE}) and (m_{INS}), pipe and insulation Stefan numbers and densities.

The parcels m_{PIPE} and m_{INS} represent, respectively, the condensate formed due energy stored in pipe and insulation.

Stefan number is the fraction of steam latent heat that stores in pipe or insulation.

Table 3. Load of condensate parcels

m_{STOR} (kg/s)	m_{PIPE} (kg/s)	m_{INS} (kg/s)	Ste_{PIPE}	Ste_{INS}	ρ_{PIPE} kg/m^3)	ρ_{INS} (kg/m^3)
6,6928E-03	5,7448E-03	9,4799E-04	0,036282	0,05694	7840	700

Figure 7 shows temperature profiles in tube and insulation for each instant, for insulation thickness of 50 mm.

The dimensionless start time was 5.62×10^{-02} , that corresponds a time of 2210 s. This time indicates that the steady state condition is reached.

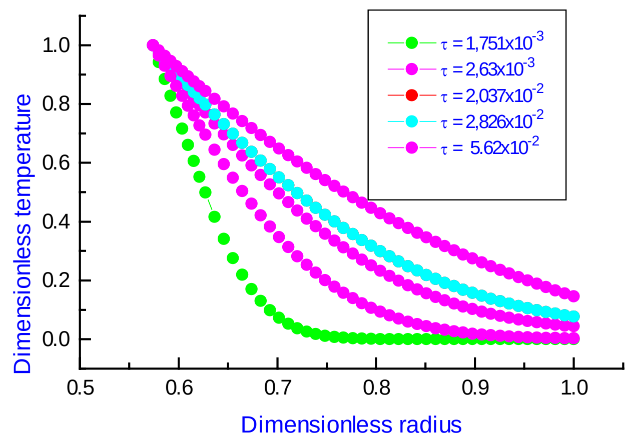

Figure 7. Temperature profiles in tube and insulation

Figure 8 shows the steady state temperature profile in a pipe insulated for each inverse Biot number.

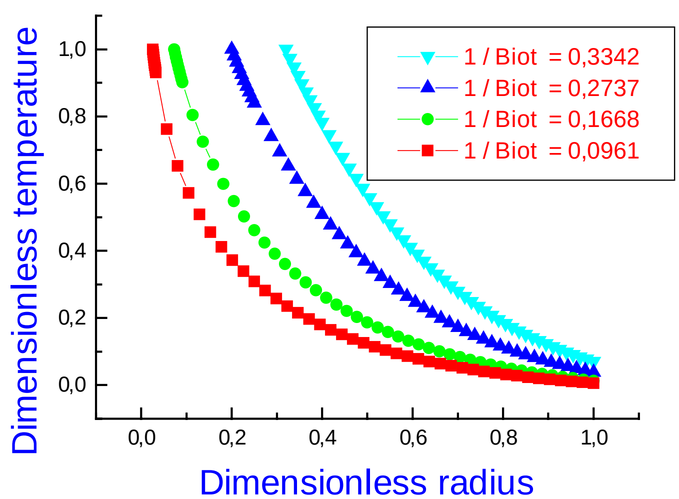

Figure 8. Outer wall insulation temperature gradient

The external radius of the insulation was the characteristic length for the Biot number. The insulation critical radius concept considers h_0 or Biot constant. It is not realistic for industrial piping, because when insulation thickness increases, the h_0 decreases and the Biot increases. When 1 / Biot tends to 1, the temperature gradient in the insulation outer surface is not equal to zero. When this concept is applied the insulation critical radius is less than pipe inner radius (no realistic).

Table 4 shows the operating condensate mass flow rate (effective condensate), the operating differential pressure and $(\Delta p)_{max}$

Table 4. Variables of condensate recovery system

$(m_C)_{operating}$ (kg/s)	$(\Delta p)_{operating}$ (MPa)	$(\Delta p)_{max}$ (MPa)
0,3032E-01	0,4037	0,41

The differential pressure of the steam trap characteristic curve, in the operating band, must be less than $(\Delta p)_{max}$ which is 0,41 MPa . This curve appears in Figure 6 as characteristic curve. It represents the 3/8" thermodynamic steam trap curve. This steam trap is the smallest available. The condensate piping has the least standard nominal diameter of 3/8 " (D_{ict} = 12,5 mm and D_{oct} = 17,1 mm).

The m_{STOR} obtained by steam trap manufacturers was 0,051 kg/s or about eight times oversized (0,051/0,006928).

5. CONCLUSIONS

The models presented in this paper can be used to optimize the standard insulation thickness and dissipation heat transfer rate in steam piping. They can be used for the design of condensate recovery system, with a good precision. As indicated in Figure 7, the model based on unsteady state Biot number leads to continuous temperature profiles in pipe and insulation. The insulation critical radius concept is not realistic for industrial piping and does not apply here. When it is applied the insulation radius is smaller than the pipe inner radius. As indicated, the models can be used for the design of condensate recovery system with a good precision. The effective condensate 0,03032 kg/s is greater than or equal to the estimated condensate 0,021505 kg/s.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

Crooker, S. and King, R., 1987, "Piping Handbook", 7ª ed., McGraw-Hill.
Holman, J. P., 1983, "Heat Transfer"', McGraw-Hill, São Paulo, Brazil, 629 p.
McCauley, J. F., 1995, "The Steam Trap Handbook" , The Fairmount Press, Inc., Liburn, Georgea.
Patankar, S. V., 1980, "Numerical Heat Transfer and Fluid Flow" , Hemisphere Publishing Corporation, New York, 1977, 197 p.
Melo, C. A., 2005, "A semi empirical method for design of a steam piping and its condensate recovery system" , 18th International Congress of Mechanical Engineering, Ouro Preto, M.G , Brasil
Scarborough, J. B., 1958, "Numerical Mathematical Analysis", 4th ed., Johns Hopkins Press, Baltimore.
Spalding, D. B., 1977, "GEMIX – A General Computer Program for two-dimensional Parabolic Phenomena", Pergamon, New York.
Telles, P. C. S., 1982, "Tubulações Industriais", 6ª edição, Livros Técnicos e Científicos editora S. A, Rio de Janeiro, Brasil, 367 p.