UNSTEADY CONDUCTION AND FREE CONVECTION FOR THE DESIGN OF A STEAM PIPING AND ITS CONDENSATE RECOVERY SYSTEM

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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 water-hammering conditions in the pipeline damaging all equipments installed. An unsteady state conduction model was developed to calculate the mass flow rate of the condensate formed, due to the heat stored in pipe wall and in the thermal insulation and due to the heat dissipation to surroundings. 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's criteria. The load of the condensate was analyzed in function of the steel and of the insulation Stefan numbers and their densities. Considering the coupling of the differential pressure, between upstream and downstream of the steam trap that the condensate piping demands of the steam trap and the differential pressure obtained from the characteristic curve of the steam trap, a governing global equation appears. The mass flow rate of the condensate that can flow 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 the condensate estimated in the unsteady conduction model. In the iterative process, the Lagrange or the spline 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 the heat* stored in the pipe wall and in the thermal insulation and due to the heat dissipation to surroundings.

Water hammer is caused by the accumulation of the 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 described by Telles (1982) and by McCauley (1995), has purpose to remove the 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 the condensate which then flows to the steam trap, as indicated in Fig. 1. The trap should discharge the condensate.

The *c*ondensate loads are relatively small and constant while in normal operation. Startup loads or during warming up, due to 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 Fig. 2.

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 in the insulation and a startup time varying between 5 to 10 minutes.

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

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

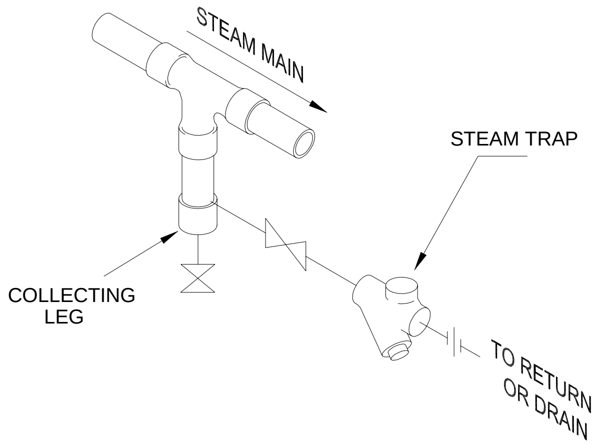

Figure 1. Collecting leg and steam trap on horizontal steam main

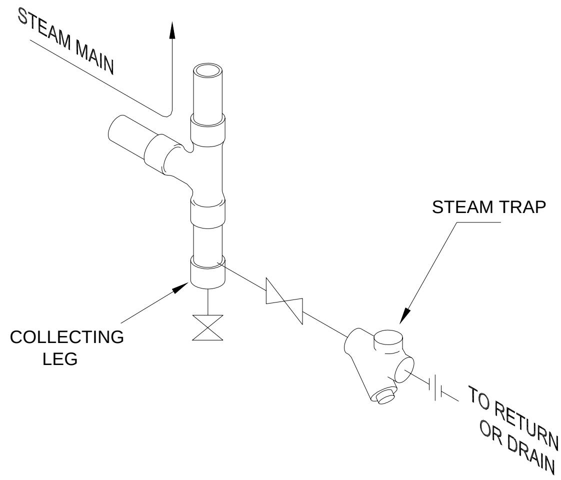

Figure 2. Collecting leg and steam trap on vertical steam main

2. UNSTEADY STATE CONDUCTION AND FREE CONVECTION MODEL

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

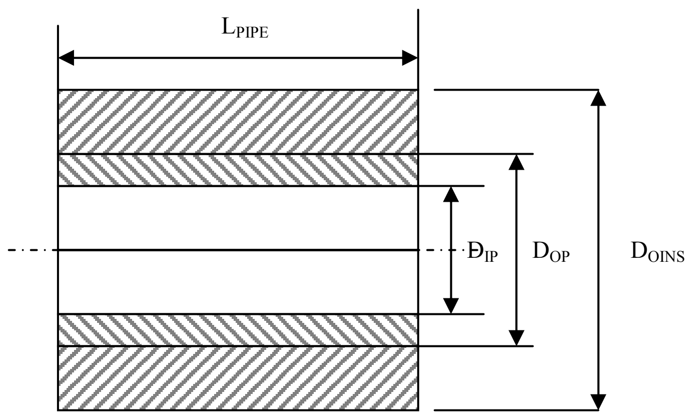

Figure 3 - Carbon steel pipe and its insulation

L is length and D is inner diameter
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 heat stored in pipe and insulation walls and the startup time, as:

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

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

$$R=\frac{D_{IT}}{D_{OINS}} \ ; \ \theta=1 \quad (1.1)$$

The boundary condition (1.1) is realistic and it was considered because the inner heat transfer coefficient between the steam condensate and inner pipe surface is very big, as described by Kern (1950).

$$R=\frac{D_{OP}}{D_{OINS}} \ ; \ \alpha=\alpha_{AV} \ \text{(The mean value } \alpha_{AV} \text{ will be defined in the numerical solution process)} \quad (1.2)$$

$$R=1 \ ; \ \frac{\partial \theta}{\partial R}=-Biot\theta \ ; \ Biot=\frac{h_O D_{OINS}}{K_{INS}} \quad (1.3)$$

The heat transfer coefficient h_o and Biot number will be determined, iteratively, at each instant.

$$\tau=0 \ : \ \theta=0 \quad (1.4)$$

The dimensionless variables were defined as follows:

$$\theta=\frac{T-T_{SUR}}{T_{SAT}-T_{SUR}} \quad (2)$$

$$R=\frac{r}{2D_{OINS}} \quad (3)$$

$$\tau=\frac{4\alpha_{INS}t}{D_{OINS}^2} \quad (4)$$

Air outer layer (AOL) dimensionless temperature is defined as:

$$\theta_{AOL}=\frac{\theta_{OINS}}{2} \quad (5)$$

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

ρ_{AOL} = 1,83714 - 2,3148E-03 $(T_{SAT} - T_{SUR})\theta_{AOL}$ (6)
μ_{AOL} = 8,676E-06 + 3,582E-08 $(T_{SAT} - T_{SUR})\theta_{AOL}$ (7)
α_{AOL} = -2,41766E-05 + 1,544E-07 $(T_{SAT} - T_{SUR})\theta_{AOL}$ (8)
K_{AOL} = 4,03833E-03 + 7,41E-05 $(T_{SAT} - T_{SUR})\theta_{AOL}$ (9)

$\nu_{AOL} = \mu_{AOL} / \rho_{AOL}$ (10)

$Pr = \nu_{AOL} / \alpha_{AOL}$ (11)

The subscripts are:

SAT - saturation
SUR - surrounds

The Grashof number, described by Holman (1983), in air outer layer temperature is:

$$Gr = \frac{\rho_{AOL}^2 g \beta_{AOL} (T_{SAT} - T_{SUR}) \theta_{AOL} D_{OINS}^3}{\mu_{AOL}^2} \quad (12)$$

The Rayleigh number is:

$Ray = Gr\ Pr$ (13)

The experimental correlation to determine the Nusselt number, described by Holman (1983), is:

$Nu = 0{,}53(Ray)^{0{,}25}$ (14)

The outer heat transfer coefficient, by free convection between the insulation outer surface and surrounds, is:

$h_o = K_{AOL}\ Nu / D_{OINS}$ (15)

The Biot number is:

$$Biot = \frac{h_0 D_{OINS}}{K_{INS}} \quad (16)$$

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 tri-diagonal matrix algorithm (TDMA).

Figure 4 shows the discretized control volumes:

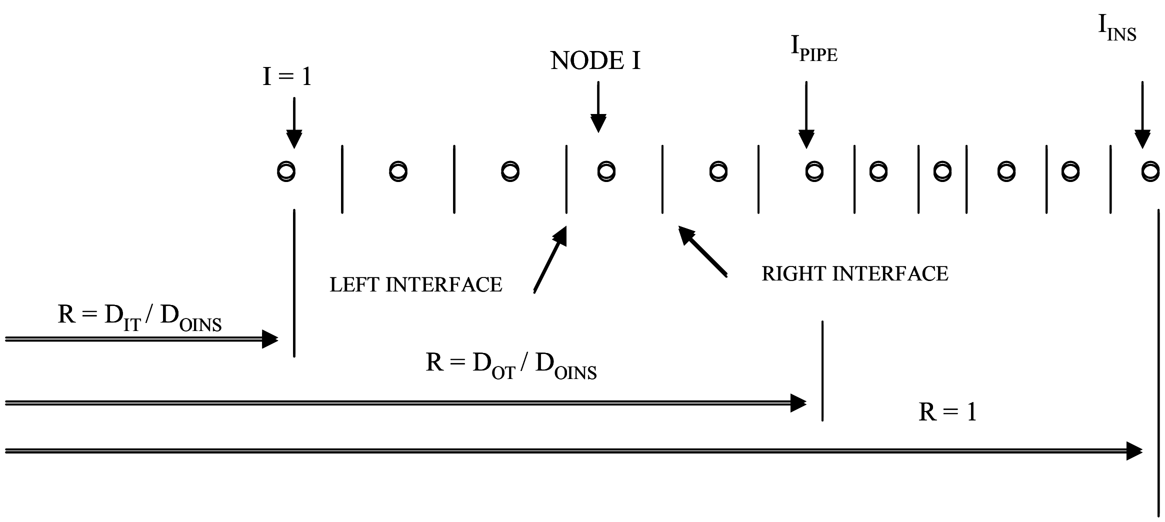

Figure 4 – Discretized control volumes

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

$$\Delta R_1 = \frac{\frac{1}{D_{OINS}}\left(D_{OINS} - D_{IP}\right)}{I_{PIPE} - 1} \quad ; \quad \Delta R_2 = \frac{\frac{1}{D_{OINS}}\left(D_{OINS} - D_{OP}\right)}{I_{INS} - I_{PIPE}} \tag{17}$$

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

$$\alpha_{AV} = \frac{\Delta R_1 \alpha_{ST} + \Delta R_2 \alpha_{INS}}{\Delta R_1 + \Delta R_2} \tag{18}$$

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

$$\dot{m}_{stor} = \frac{\alpha_{INS} L}{\Delta \tau}\left[\rho_{ST} Ste_{ST} \sum_{I=1}^{I_{PIPE}} V_I \theta_I + \rho_{INS} Ste_{INS} \sum_{I=I_{PIPE}+1}^{I_{INS}} V_I \theta_I\right] \tag{19}$$

$\Delta\tau$ is the dimensionless time step to reach the steady state or startup dimensionless time
L is the length of the pipe

The heat dissipation, in the stead state, is:

$$Q_{DISS} = h_o \pi D_{OINS} L\left(T_{SAT} - T_{SUR}\right)\left(\theta_{I_{INS}} - \theta_{SUR}\right) \tag{20}$$

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

$$\dot{m}_{DISS} = \frac{Q_{DISS}}{h_{lv}} \tag{21}$$

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

$$Ste_{ST} = \frac{C_{ST}\left(T_{SAT} - T_{SUR}\right)}{h_{lv}} \quad ; \quad Ste_{INS} = \frac{C_{INS}\left(T_{SAT} - T_{SUR}\right)}{h_{lv}} \tag{22}$$

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

$$\dot{m}_C = F\left(\dot{m}_{DISS} + \dot{m}_{STOR}\right) \tag{23}$$

F is the safety factor, used by steam trap manufactures handbooks, varying between 2 and 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.

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

$$\dot{m}_{STOR} = \frac{\pi L\left(T_{SAT} - T_{SUR}\right)}{4\Delta t h_{lv}}\left[\left(D_{OP}^2 - D_{IP}^2\right)\rho_{ST} C_{ST} + \left(D_{OINS}^2 - D_{OP}^2\right)\rho_{INS} C_{INS}\right] \tag{24}$$

Δt is the startup time varying between 5 and 10 minutes. This estimated time step does not obeys any criteria.

The insulation critical radius concept consider constant the heat transfer coefficient between insulation outer susrface and surrounds. When the insulation radius is equal the critical radius the heat transfer between the insulation and surrounds is equal to zero. This consideration is false because this heat is a function of the insulation outer susrface temperature, which is a function of the insulation thickness.

The insulation critical radius is:

$$r_{critical} = \frac{k_{INS}}{h_o} \quad (25)$$

When the insulation radius is equal the critical radius, has:

$$\frac{r_{critical}}{r_{INS}} = \frac{k_{INS}}{h_o r_{INS}} = 1 \quad (26)$$

Equation (26) indicates that the when the inverse of Biot number is equal to 1, in the steady state, the heat transfer to surrounds is equal to zero.

3. DESIGN OF CONDENSATE PIPING

The differential pressure, between upstream and downstream of the steam trap that the condensate piping (system) demands of the steam trap is called system curve.

The differential pressure obtained from steam trap manufacturer is called characteristic curve (inn).

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

Figure 5 shows the layout of steam and condensate piping.

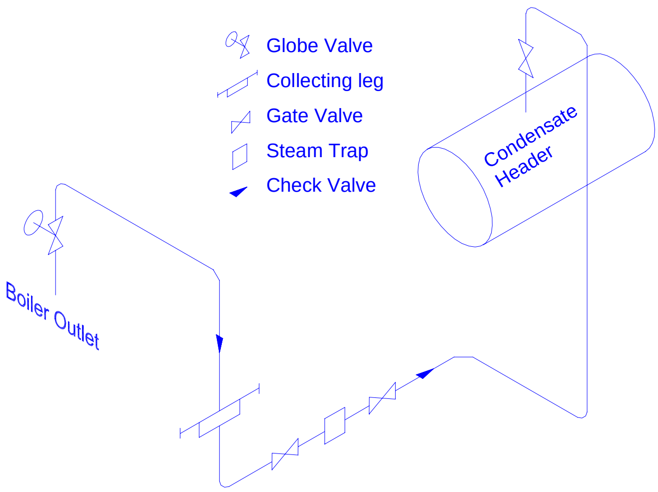

Figure 5. Layout of steam and condensate piping

Based on Bernoulli and Darcy equations, the differential pressure that the condensate piping demands of the steam trap, is:

$$\Delta p_{syst} = p_{cl} - p_{ch} + \rho_C g(z_{cl} - z_{bst} + z_{ast} - z_{ch}) - \frac{8 f (L_{bst} + L_{ast}) \dot{m}_c^2 v_c}{\pi^2 D_{icp}^5} \quad (27)$$

L_{bst} is sum of straight and component equivalent lengths before steam trap, L_{ast} is sum of straight and component equivalent lengths after steam trap, D_{icp} is inner diameter of condensate pipe, f is the friction factor, 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

icp – inner condensate pipe
.

$$(\Delta p)_{\max} = p_{cl} - p_{ch} + \rho_C g(z_{cl} - z_{bst} + z_{ast} - z_{ch}) \quad (28)$$

Figure 6 shows the differential pressure obtained from the system curve and the differential pressure obtained from the characteristic curve.

As shown in Fig. 6, in the operation point, we can write:

$$\Delta p_{syst} = \Delta p_{inn} \quad (29)$$

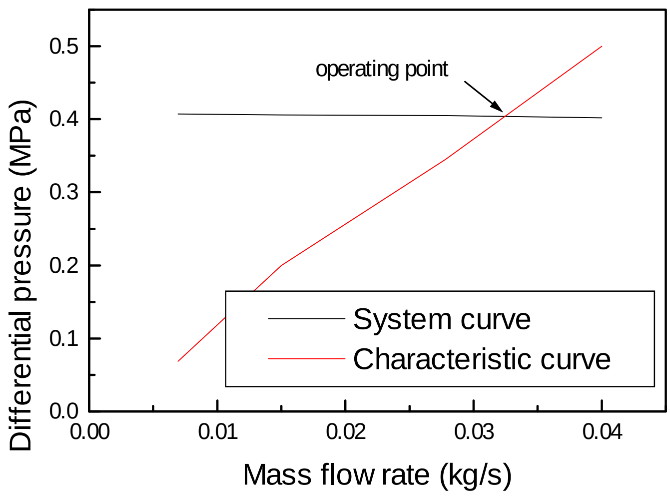

Figure 6. System and steam trap characteristic curves

$$\Delta p_{syst} - \Delta p_{inn} = s\left(\dot{m}_C\right) = 0 \quad (30)$$

The subscript syst indicates system and the subscript inn indicates inner or characteristic curve.
f is the friction factor that can be obtained by the Colebrook and White's formula as:

$$x = \frac{1}{\sqrt{f}}$$

$$g(x) = x - 1{,}74 + \ln\left(\frac{2e}{D_{ict}} + \frac{18{,}7x}{\mathrm{Re}_{D_{ict}}}\right) = 0 \quad (31)$$

$$\mathrm{Re}_{D_{ICP}} = \frac{4m_C}{\pi D_{icp}\mu_C} \quad (32)$$

$$g'(x) = 1 - \frac{\dfrac{18{,}7x}{\mathrm{Re}_{D_{icp}}}}{\dfrac{2e}{D_{icp}} + \dfrac{18{,}7x}{\mathrm{Re}_{D_{icp}}}} \quad (33)$$

The following algorithmic is used to determine the friction factor f:
Guess a value to x_0 for x (any value between 6 and 10) and other for tolerance
Repeat (iterative process)
Calculate $g(x_0)$
Calculate $g'(x_0)$

$x_1 = x_0 - \frac{g(x_0)}{g'(x_0)}$

$CRIT = \left| \frac{x_1 - x_0}{x_1} \right|$

$x_0 = x_1$

Until CRIT < Tolerance

$f = \frac{1}{x_1^2}$

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

$$\Delta p_{inn} = \sum_{I=1}^{NP} \Delta p_I \prod_{J=1}^{NP} \frac{\left(\dot{m}_C - m_J\right)}{\left(m_I - m_J\right)} \qquad J \neq I \tag{34}$$

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

Equation (30) is the global governing equation. Its solution is the mass flow rate of condensate that can flow in the recovery condensate system.

The mass flow rate of condensate that can flow in the condensate recovery system, which is called effective condensate load, 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 function was used to generate the intermediate points of Δp_{inn} and the Colebrook's routine was used to generate the friction factor of Δp_{syst}. The MC value is the solution of equation (30) and represents the condensate load that can flows in the condensate recovery system (effective condensate). This amount must be greater than or equal to the load of estimated condensate given by equation (23).

The tested characteristic curve must represent a steam trap that operates with differential pressure below of that given by Eq. (28). The steam trap must operates with load of condensate below and above of estimated condensate.

4. **RESULTS AND DISCUSSIONS**

The results were obtained considering the 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} = 154 mm , D_{OP} = 168,2 mm), with straight length of 12,5 m, and a mass flow rate of 2,7777 kg/s (10000 kg/h) . In this case, the pipe 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 331 K was less than 333 K (safety condition), for insulation thickness of 38 mm.

The straight length of condensate piping is 18 m ; $z_{ch} - z_{cl}$ = 7 m ; Dicp = 12,5 mm ; p_{ch} = 0,5 MPa . The characteristic curve of the Figure 6 was the tested steam trap characteristic curve in the algorithm. This steam trap is the smallest available.

Table 1 shows the parameters that were used in unsteady conduction model:

Table 1. Numerical algorithm parameters

D_{IP} / D_{OINS}	D_{OP} / D_{OINS}	I_{PIPE}	I_{INS}	$\Delta\tau$	Δt
0,6306	0,6888	10	61	$1.x10^{-07}$	0,003261 s

Table 2 shows the parameters, obtained by unsteady conduction model and the oversized condensate estimated by Telles.

Table 2. Results of unsteady state conduction model

T_{OWINS} (K)	h_o (W/m^{2}K)	Biot Steady state	τ Steady state	t (s) Steady state	m_{DISS} (kg/s)	m_{STOR} (kg/s)	F Safety factor	Condensate estimated (kg/s)	Oversized by Telles (kg/s)
331	4,5443	7,7064	0,09346	3047	7,1630E-04	4,6256E-03	3	1,6026E-02	0,165

Figure 7 shows temperature profiles in the tube and in the insulation for each instant, for insulation thickness of 38 mm.

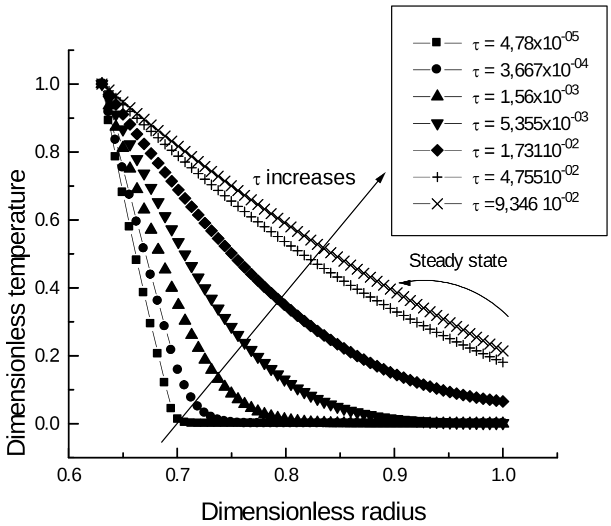

Figure 7. Temperature profiles in tube and insulation for each instant

Figure 8 shows the Biot number in function of the time.

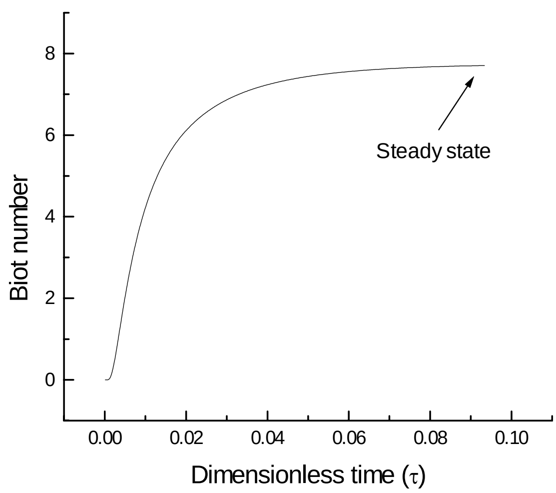

Figure 8.Biot number in function of the time.

Figure 9 shows steady state temperature profiles in the pipe and in the insulation in function of the inverse Biot number (Biot inverse).

As shown in the figure 9, the critical radius concept is not applied. The Biot inverse decreases and never tends to 1. When the insulation thickness increases the Biot inverse and temperature gradient in the insulation outer surface decreases and tend to zero.

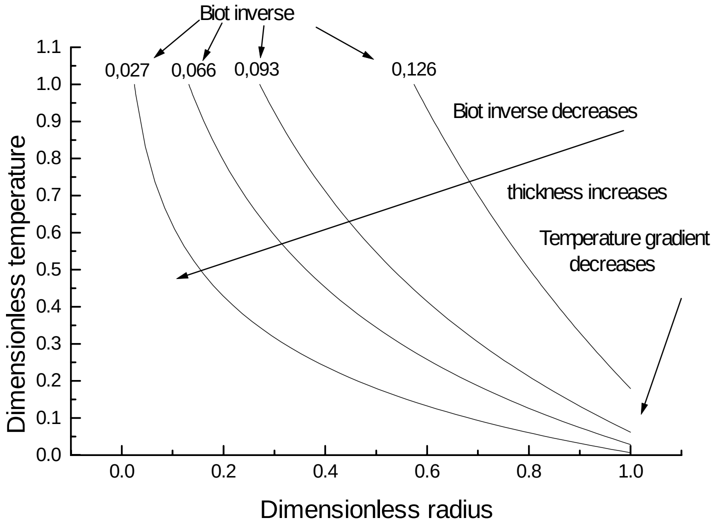

Figure 9. Steady state temperature profiles

Table 3 shows the maximum differential pressure of the steam trap (Δp_{max}), the operating mass flow rate of condensate or effective condensate (m_C) and the operating diferential pressure (Δp).

Table 3. Maximum and effective differential pressure

Δp_{max} (MPa)	m_C (kg/s)	Δp (Mpa)
0,4	0,0348	0,039

5. CONCLUSIONS

The models presented can be used to optimize the steam piping, its insulation and the heat dissipation. They can be used for the design of the condensate recovery system, with a good precision. As indicated in Fig. 7, the model based on unsteady state Biot number leads to continuous temperature profiles in pipe and insulation. As indicated in the Fig. 9, the insulation critical radius concept is not realistic for industrial piping and does not apply here. The insulation condition is reached when the Biot inverse tends to zero. As indicated in Tab. 2, Telles's model is about ten times oversized. The effective condensate of 0,0348 kg/s is greater than the estimated condensate of 0,01603 kg/s.

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.

Kern, D. Q., 1950, "Process Heat Transfer", McGraw-Hill, New York, 671 p.

McCauley, J. F., 1995, "The Steam Trap Handbook", The Fairmount Press, Inc., Liburn, Georgia.

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

Patankar, S. V., 1980, "Numerical Heat Transfer and Fluid Flow" , Hemisphere Publishing Corporation, New York, 1977, 197 p.

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.