A SEMI EMPIRICAL METHOD FOR DESIGN OF A STEAM PIPING AND ITS CONDENSATE RECOVERY SYSTEM

Carlos Alberto de Melo
Universidade Federal de Uberlândia – Faculdade de Engenharia Mecânica
carlosam@mecanica.ufu.br

Abstract. *As steam flows through the distribution piping additional condensation is continuously taking place. This condensate has to be removed immediately as steam is carrying it forward at high velocities. This could cause severe waterhamering conditions and erosion in the pipeline damaging all equipments installed. The carbon steel piping and its thermal insulation are designed to have standard dimensions, withstand mechanical stress and cause minimal resistance to the steam flow. A computational algorithmic was developed to calculate the global heat transfer coefficient and the mass flow rate of condensate formed due to heat dissipation. Considering the heat stored in the pipe wall and in the thermal insulation, a model to calculates the mass flow rate of condansate formed during heating of the line was developed. The total load of condensate is called the estimated condensate. The design of condensate piping that contains the steam trap, is done considering the coupling between the pressure loss of the condensate piping system and the pressure loss at the steam trap characteristic curve. The mass flow rate of condensate, that can flows in the piping is obtained by the solution of the global equation. This condensate load,which is called the effective condensate, must be greater than or equal to the estimated condensate. 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. Several characteristic curves and inner diameters of the condensate piping are tested until the load estimated condensate is reached.*

Keywords*: steam piping, condensate recovery, energy, design and optimization*

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.

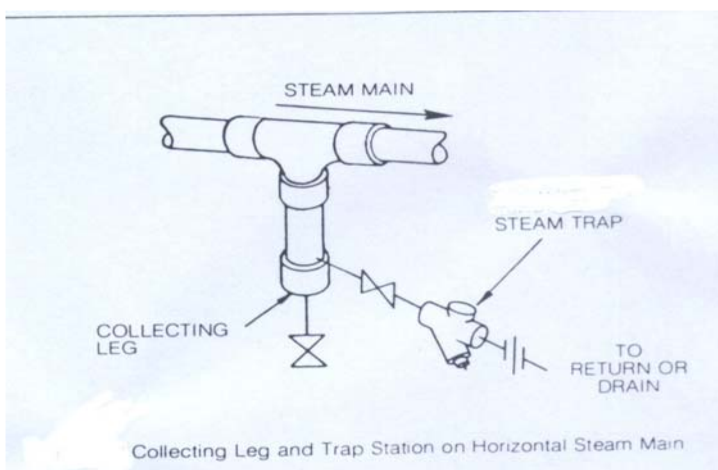

Figure 1. Collecting leg and steam trap on horizontal steam main

As reported by Telles (1982) and by Crooker and King (1987), the models for calculate the load of estimated condensate are simplified and oversized. The purpose of this paper is present a simplified method, that use models, equipments curves and experimental correlations, for design of a steam piping and its condensate recovery system, with good precision.

2. Formulation of the problem

Figure 2 shows the carbon steel pipe and its thermal insulation:

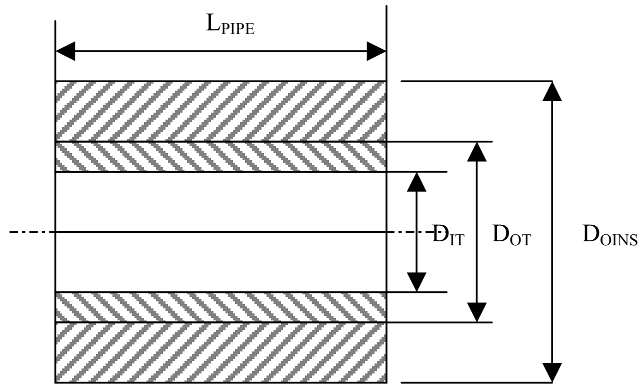

Figure 3 - Carbon steel pipe and its insulation

D_{IT} - Inner diameter of the tube
D_{OT} - Outer diameter of the tube
D_{OINS} - Outer diameter of the insulation

Due to heat dissipation to surroundings and due to heat stored in pipe wall and in thermal insulation, steam condensation is continuously taking place.

Holman (1983) presents a heat transfer correlation, between the condensate laminar layer and pipe inner wall, proposed by Nusselt, as follows:

$$h_i = 0{,}725\left(\frac{\rho_l(\rho_l-\rho_v)h_{lv}K^3_{CIL}}{\mu_{CIL}D_{IT}(T_{SAT}-T_{GPIW})}\right)^{\frac{1}{4}} \tag{1}$$

The above symbols are:

ρ - density
h_{lv} - steam latent heat
K - conductivity
μ - dynamic viscosity
T - temperature

The subscripts are:

l - saturated liquid
v - saturated steam
CIL - condensate inner layer
SAT - saturation
GPIW - guessed pipe inner wall
PIW - pipe inner wall
SUR - surrounds
AOL - air outer layer
GOINS - guessed outer insulation
OINS - outer insulation

The Grashof number, that must be used to determine the outer heat transfer coefficient by free convection between the insulation outer surface and surrounds, is:

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

Properties K and μ ,.in average condensate inner layer Kelvin temperature, are written as:

μ_{CIL} = 3,66689E-04 + 1,16498E-06 (T_{CIL} - 273) (3)

K_{CIL} = 0,714493 - 2,26329E-04 (T_{CIL} - 273) (4)

Properties ρ, β.and μ,.in average air outer layer temperature in Kelvin, are written as:

ρ_{AOL} = 1,83714 - 2,3148E-03 T_{AOL} (5)

μ_{AOL} = 8,676E-06 + 3,582E-08 T_{AOL} (6)

α_{AOL} = -2,41766E-05 + 1,544E-07 T_A (7)

K_{AOL} = 4,03833E-03 + 7,41E-05 T_{AOL} (8)

ν_{AOL} = . μ_{AOL} / ρ_{AOL} (9)

Pr_{AOL} = ν_{AOBL} / α_{AOL} (10)

β_{AOL} = 1 / T_{OAL} (11)

2.1. Heat dissipation computational algorithmic

The following model is used to calculate the heat dissipation and others thermal parameters:

Consider an estimated value for T_{GPIW} less than T_{SAT} and another estimated value for T_{GOINS} greater than T_{SUR}

Repeat (Iterative process)

$T_{CIL} = (T_{SAT} + T_{GPIW}) / 2$; $T_{AOL} = (T_{GOINS} + T_{SUR}) / 2$

h_i is calculated by Eq. (1), using Eq. (3) and Eq. (4) for transport properties.

Grashof number Gr is calculated by Eq. (2), using Eq. (5) to (11) for transport properties.

Ray = Gr Pr

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

$h_o = K_{AOL}$ Nu / D_{OINS}

$$U = \frac{1}{\dfrac{D_{OINS}}{D_{IT}\,\bar{h}_i} + \dfrac{D_{OINS}\ln\left(\dfrac{D_{OT}}{D_{IT}}\right)}{K_{ST}} + \dfrac{D_{OINS}\ln\left(\dfrac{D_{OINS}}{D_{OT}}\right)}{K_{INS}} + \dfrac{1}{\bar{h}_o}}$$

$$Q_{DISS} = U\pi D_{OINS} L\left(T_{SAT} - T_{SUR}\right)$$

$$T_{PIW} = T_{SAT} - \frac{Q_{DISS}}{h_o \pi \bar{D}_{OINS} L}$$

$$T_{OINS} = T_{SUR} + \frac{Q_{DISS}}{h_o \pi D_{OINS} L}$$

$$CRIT1 = \left|\frac{T_{PIW} - T_{GPIW}}{T_{PIW}}\right| \quad \text{AND} \quad CRIT2 = \left|\frac{T_{OINS} - T_{GOINS}}{T_{OINS}}\right|$$

$T_{GPIW} = T_{PIW}$ AND $T_{GOINS} = T_{OINS}$

Until (CRIT1) AND (CRIT2) $\leq$ tolerance

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

$$\dot{m}_{DISS} = \frac{Q_{DISS}}{h_{lv}} \tag{12}$$

Handbooks of steam trap manufacturers consider an estimated value of global heat transfer coefficient U and uniform temperature condition, in pipe wall and insulation. They consider a start up time varying between 5 to 10 minutes, as reported by Telles (1982) and by Crooker and King (1987). This time step is undersized for steam distribution piping, as shown by Melo (1992), which developed a generalized model for the design of thermal storage unit with phase change material. The results obtained by Melo were in good agreement with those obtained experimentally by Kalhori and Ramadhyani (1985). Melo's model will result in the startup time model.

The following mass flow rate of condensate formed, due to the heat stored in pipe wall and insulation, is oversized:

$$\dot{m}_{STOR} = \frac{\pi L\left(T_{SAT} - T_{SUR}\right)}{4\Delta t_{STOR} h_{lv}}\left[\left(D_{OT}^2 - D_{IT}^2\right)\rho_{ST} C_{ST} + \left(D_{OINS}^2 - D_{OT}^2\right)\rho_{INS} C_{INS}\right] \quad (13)$$

Where C is thermal capacity and Δt_{STOR} is the startup time.
The subscript ST and INS represent, respectively, steel and insulation.

2.2. Startup time model

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, with a good precision:

$$\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \theta}{\partial R}\right) = \frac{\alpha_{INS}}{\alpha}\frac{\partial \theta}{\partial \tau} \quad (14)$$

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

$$R = \frac{D_{IT}}{D_{OINS}} \quad ; \quad \frac{\partial \theta}{\partial R} = -Bi_1\left(1-\theta\right) \quad ; \quad Bi_1 = \frac{h_i D_{INS}}{K_T} \quad (14.1)$$

$$R = \frac{D_{OT}}{D_{OINS}} \quad ; \quad K_T\frac{\partial \theta}{\partial R} = K_{INS}\frac{\partial \theta}{\partial R} \quad (14.2)$$

$$R = 1 \quad ; \quad \frac{\partial \theta}{\partial R} = -Bi_2\theta \quad ; \quad Bi_2 = \frac{h_O D_{OINS}}{K_{INS}} \quad (14.3)$$

$$\tau = 0 \quad : \quad \theta = 0 \quad (14.4)$$

The dimensionless variables, above, were defined as follows:

$$\theta = \frac{T - T_{SUR}}{T_{SAT} - T_{SUR}} \quad ; \quad R = \frac{r}{2D_{OINS}} \quad ; \quad \tau = \frac{4\alpha_{INS} t}{D_{OINS}^2} \quad (15)$$

Equation (14), with boundaries conditions (14.1) to (14.3) and initial condition (14.4) 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 mass flow rate of condensate formed, due to heat stored in pipe wall and insulation, is calculated in the present model as:

$$\dot{m}_{stor} = \frac{\alpha_{INS} L}{\Delta \tau_{STOR}}\left[\rho_{ST} Ste_{ST}\sum_{I=1}^{N_{ST}} V_I\theta_I + \rho_{INS} Ste_{INS}\sum_{N_{ST}}^{N_{INS}} V_I\theta_I\right] \quad (16)$$

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

$$Ste_{ST} = \frac{C_{ST}\left(T_{SAT} - T_{SUR}\right)}{h_{lv}} \quad ; \quad Ste_{INS} = \frac{C_{INS}\left(T_{SAT} - T_{SUR}\right)}{h_{lv}} \quad (17)$$

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

$$m_C = F\left(\dot{m}_{DISS} + \dot{m}_{STOR}\right) \tag{18}$$

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, the factor 1 is used. As indicated, this method is never oversized.

2.3. 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 3 shows the Layout of steam and condensate piping and Figure 4 shows system and steam trap characteristic curves.

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

$$\Delta p_{syst} = p_{cl} - p_{rc} + \rho_C g\left(z_{cl} - z_{bst} + z_{ast} - z_{rc}\right) - \frac{8 f\left(L_{bst} + L_{ast}\right)\dot{m}_C^2 v_C}{\pi^2 D_{ict}^5} \tag{19}$$

L_{bst} Sum of straight and component equivalent lengths before steam trap
L_{ast} Sum of straight and component equivalent lengths after steam trap
f is the friction factor, obtained by the Colebrook and White formula as follows:

$$g(x) = x - 1{,}74 + \ln\left(\frac{2e}{D_{ict}} + \frac{18{,}7x}{\mathrm{Re}_{D_{ict}}}\right) = 0 \tag{20}$$

$$\mathrm{Re}_{D_{ict}} = \frac{4m_C}{\pi D_{itc}\mu_C} \tag{21}$$

$$g^{'}(x) = 1 - \frac{\dfrac{18{,}7x}{\mathrm{Re}_{D_{ict}}}}{\dfrac{2e}{D_{itc}} + \dfrac{18{,}7x}{\mathrm{Re}_{D_{ict}}}} \tag{22}$$

The following algorithmic is used to determine the friction factor f :
Estimate a value x_0 for x (any value between 6 to 10)
Repeat (Iterative process)
Calculate $g(x_0)$, by Eq. (20)
Calculate $g^{'}(x_0)$, by Eq. (22)

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

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

$$x_0 = x_1$$

Until CRIT < Tolerance

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

The maximum differential pressure is:

$$(\Delta p)_{\max} = p_{cl} - p_{ch} + \rho_C g\left(z_{cl} - z_{bst} + z_{ast} - z_{ch}\right) \tag{23}$$

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:

$$\Delta p_{inn} = \sum_{I=1}^{NP} \Delta p_I \prod_{J=1}^{NP} \frac{\left(\dot{m}_C - m_J\right)}{\left(m_I - m_J\right)} \qquad J \neq I \tag{24}$$

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

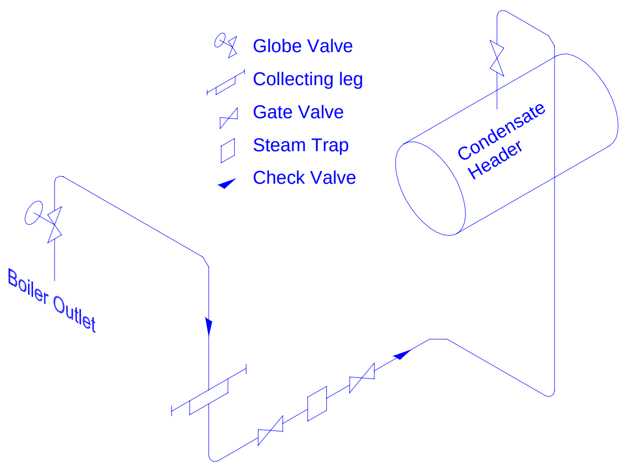

Figure 3. Layout of steam and condensate piping

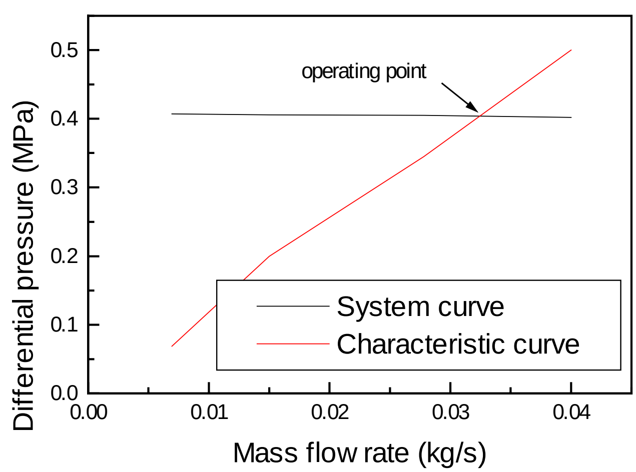

Figure 4. System and steam trap characteristic curves

As shown in Figure 4, we can write:

$$\Delta p_{syst} = \Delta p_{inn} \tag{25}$$

$$\Delta p_{syst} - \Delta p_{inn} = s\left(\dot{m}_C\right) = 0 \tag{26}$$

Equation (26) is the global governing equation. The mass flow rate of condensate that will flow in the piping is the solution of the global equation, which 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 (26) and represents the operating condensate mass flow rate.

3. Results obtained

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_{IT} = 153,8 mm , D_{OT} = 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 inlet steam pressure.

The insulation tested, in above algorithmic, was pre-molded calcium silicate or rock silicate. The outer wall insulation temperature 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 thermal parameters, obtained by algorithmic and Table 2 shows the results of start up time model, where (Man) represents steam trap Manufacturer.

Table 1. Results of Thermal algorithmic

T_{IWT} (K)	T_{OWINS} (K)	h_i (W/m^{2}K)	h_o (W/m^{2}K)	U (W/m^{2}K)	Q_{DISS} (W)	m_{DISS} (kg/s)	Bi_1	Bi_2
452,9	330	39100	4,403	0,9121	1250	0,619E-03	96,39	8,2

Table 2. Results of Startup time model

$\Delta t(s)$	$\Delta\tau$	m_{DISS} (kg/s)	m_{STOR} (kg/s)	m_C F=1 (kg/s)	m_C F=3 (kg/s)	m_C (Man) F=3 (kg/s)
1357	3,45E-02	0,619E-03	0,5311E-02	0,593E-02	0,1779E-01	0,1528

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 4 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 nominal diameter of 3/8 " (D_{ict} = 12,5 mm and D_{oct} = 17,1 mm)

Table 3 shows the operating condensate mass flow rate (effective condensate), the operating differential pressure and $(\Delta p)_{max}$

Table 3. 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

Figure 5 shows temperature profiles in tube and insulation, for each instant and the star up time of 1357 s.

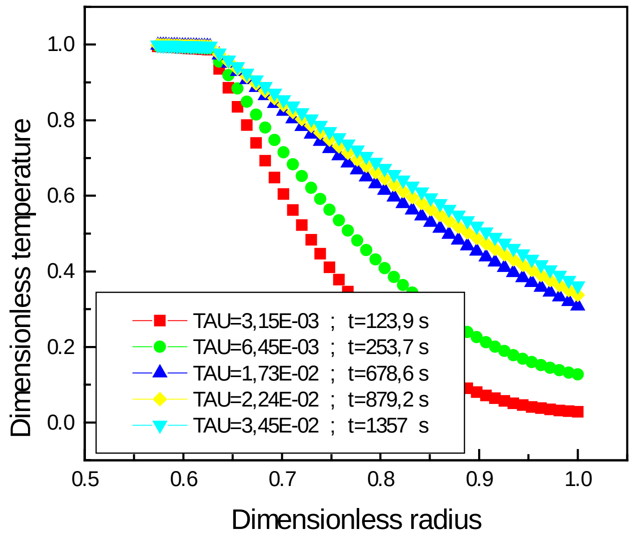

Figure 5. Temperature profiles in tube and insulation, for each instant

4. Conclusions

The models presented in this paper can be used to optimize the standard insulation thickness and dissipation heat transfer rate. They can design the load of condensate formed due to heat stored in pipe wall and insulation and the condensate recovery system, with a good precision. Table 3 shows that the load of condensate estimated by the steam trap Manufacturers Handbooks was about nine times oversized (0,1528/0,01779). As indicated, all condensate recovery system (piping and steam trap) are oversized by steam trap manufacturers handbooks. The effective condensate (0,03032 kg/s) is greater than or equal to the estimated condensate (0,01779 kg/s).

6. Acknowledgements

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