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ENERGY LOSSES IN HORIZONTAL STEAM LINES

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ABSTRACT

The present work shows a computarized model where joint equations from the fluid mechanics and heat transfer have been introduced applied to a process of polytropic expansion, in order to determine the energy losses that the steam suffers when it is conducted through long horizontal pipes of a big diameter which are thermically insulated. This model gave interesting step by step results of final conditions of the following parameters: pressure, fluid and insulator temperatures, enthalpy and steam dryness along any section pipe of a given diameter, thickness and type of insulation, starting from the initial conditions of pressure; mass rate of the fluid as much as with sorrounding air temperature and wind velocity.The model could be modified and utilized for non-geothermal purposes having as an objective improving the design of the piping system or for checking up the actual operating conditions.

NOMENCLATURE

A Pipe internal flow area, L^2
Cp Heat capacity at constant pressure, FL/MT
d Differential operator
D_0 Orifice plate diameter, L
D_1 Internal diameter of pipe, L
D_2 External diameter of pipe, L
D_3 Insulator diameter, L
f Friction factor, dimensionless
g Gravity acc., L/θ^2
gc Conversion factor, $ML/F\theta^2$
Gr Grashof number, dimensionless
h Heat transfer particular coef., $FL/L\theta T$
H Enthalpy, $FL/\theta M$
J Conversion factor from mech. to thermal energy
K Thermal conductivity, $Fl/LT\theta$
K_0 Stefan-Boltzmann coef., Fl/θ^2T^4
κ^1 Isentropic expansion coef, dimensionless
κ Polytropic expansion coef, dimensionless
L Lenght of traveled distance, L
N Pipe line number
Nu Nusselt number, dimensionless
P Total pressure, F/L^2
Pr Prandtl number, dimensionless
Q Heat flux per unit of lenght FL/θ
Re Reynolds number, dimensionless
R Gases universal constant, FL/MT
Sc Schmidt number, dimensionless
t Insulator thickness, L
T Temperature, T
U Overall heat transf. coef., $FL/L\theta T$
V Linear velocity, L/θ
v Specific volume, L^3/M
w Work, FL
W Mass rate, M/θ
X Dryness, dimensionless
Y Pipe position constant, dimensionless
Z Distance, L
β Thermal expansion coef, 1/T
γ Specific weight, F/L^3
Δ Difference
ε Pipe relative rugosity, dimensionless
ε_0 Emisivity
η Isentropic efficiency, dimensionless
η_0 Steam trap efficiency, dimensionless
λ Latent heat, FL/M
μ Absolute viscosity, $M/L\theta$
ν Kinematic viscosity, L^2/θ
ρ Density, M/L^3
θ Time, θ

Subindexing

1 Initial conditions
2 Final conditions
a Room or enviroment
i Internal
o External
f Water
s Steam
fs Water/steam mixture
p Wall

INTRODUCTION

Evaluation or energy loss in vapor piping in the field or the industrial processes currently existing or at the designing stage, constitutes a very important step for the setting up of the economical-technical parameters of the same operation, maintenance and development of the sistems and subsistems which take part in this processes.A big number of industries have requirements, of water vapor with certain degree of humidity -

temperature and pressure preestablished - for the process itself; however, during - the conduction of this fluid from its --- source up to the point of utilization, take place dynamics changes in its thermal- and transporting properties which may impact directly these terminal conditions. Knowledge of the increasing evolution of- these changes makes possible to anticipate or detect anomalies or else carry out- optimization in the group of variables so that corrective action can be taken in -- the design, operation or maintenance of - the vapor conducting system. When handling geothermal vapor as much as in other processes the distances to cover as much- as the diameters and fluxes came out to - be relatively big, furthermore the geothermal vapor presents the particularity- of transporting non-condensable gases, -- dragged water from the separation point - with its inherent salinity, as much as -- condensated water formed during the transportation, which implies a strict control in the operating conditions having in --- mind to prevent excesive condensation problems and the phenomena of scaling, corrosion and erosion in pipes and equipment,- which render the generation of electric - energy low.

EQUATIONAL PROCEDURE

As a marked difference to other methods that make a total estimation of variables, in this model the fluid mechanic and heat transfer equations have been put together in a single equation which may be integrated numerically between pipe's intervals, to the point or convergency of the loss - energy between a pressure range. The programme is based considering the polytropic expansion of a humid vapor with heat- and friction losses that occur in differential sections of an insulated pipe, -- asuming a homogeneous, unidirectional --- flow accompanied by condensation giving - rise to different types or flow patterns- such as anular, anular-disperse and/or -- estratified as it has been proved graphically using the Mandhane diagram {1} and- by experimental observations made by wairakei New Zealand {2}. The starting point is a general equation of the energy law - in its differential form {3}

$$\frac{dp}{\gamma} + \frac{VdV}{gc} + \frac{g}{gc}\,dz + d(\ell w) + dw = 0 \tag{1}$$

That in the case of a horizontal pipe - and in absence of work done by or, on the fluid it is reduced to.

$$\frac{dp}{\gamma} + \frac{VdV}{gc} + d(\ell w) = 0 \tag{2}$$

Where ℓw stands for loss work by fluid- irreversibilities and may include: friction or heat losses through the wall pipe, sliding, frictional effects between phases viscosity effects, superficial tension etc., in this study are considered - exclusively the effects of wall friction- (Ef) and the heat loss (Q) so that:

$$\frac{dp}{\gamma} + \frac{VdV}{gc} + dEf + dQ = 0 \tag{3}$$

Considering a polytropic expansion where an initial pressure P_1 is known, and - where temperature T_1 and specific weight- may be known, the following relations can be study {4}

$$\frac{P}{\gamma^{\kappa}} = \frac{P_1}{\gamma_1^{\kappa}} \tag{4}$$

and

$$TP^{\frac{1-\kappa}{\kappa}} = T_1\,P_1^{\frac{1-\kappa}{\kappa}} \tag{5}$$

Substituting equations (4) and (5) and- utilizing the continuity equation, the -- Darcy equation and the heat transfer general equation. considering U constant for- small pipe's sections the following is -- obtained:

$$\text{First term}\quad dP/\gamma = (P_1^{1/\kappa}/\gamma_1)(dP/P^{1/\kappa}) \tag{6}$$

$$\text{Second term}\quad VdV/gc = -(W^2P_1^{2/\kappa}/A^2\gamma_1^2\kappa gc)\,(dP/P^{\frac{2+\kappa}{\kappa}}) \tag{7}$$

$$\text{Third term}\quad dEf = (fW^2P_1^{2/\kappa}/2gD\gamma_1^2A^2)\,(dL/P^{2/\kappa}) \tag{8}$$

$$\text{Fourth term}\quad dQ = (JU/W)\{(P_1^{\frac{1-\kappa}{\kappa}}T_1/P^{\frac{1-\kappa}{\kappa}}) - Ta\}\,dL \tag{9}$$

Substituting equations (6), (7), (8) -- and (9) in (3) and multiplying for $\gamma_1 P^{2/\kappa}$ putting constant terms if:

$$p^{1/\kappa}P_1^{1/\kappa}dP - (W^2P_1^{2/\kappa}/A^2\gamma_1\kappa g\,c)dP/P +$$

$$(fW^2P_1^{2/\kappa}/2gD\gamma_1A^2)dL + (JU\gamma_1/W)$$

$$(P^{\frac{1+\kappa}{\kappa}}\;P_1^{\frac{1-\kappa}{\kappa}}\;T_1 - Ta\;P^{2/\kappa})dL = 0 \tag{10}$$

Putting constant terms if:

$$C_1 = P_1^{1/\kappa} \tag{11}$$

$$C_2 = W^2P_1^{2/\kappa}/A^2\gamma_1\kappa g\,c \tag{12}$$

$$C_3 = fW^2P_1^{2/\kappa}/2gD\gamma_1A^2 \tag{13}$$

$$C_4 = JU\gamma_1P_1^{\frac{1-\kappa}{\kappa}}/W \tag{14}$$

$$C_5 = JU\gamma_1Ta/W \tag{15}$$

Equation (10) may be expressed as:

$$C_1P^{1/\kappa}dP - C_2P^{-1}dP + C_3dL + C_4P^{\frac{1+\kappa}{\kappa}}dL -$$

$$C_5P^{2/\kappa}dL = 0 \tag{16}$$

Separating variables and integrating:

$$\int_{P_1}^{P_2} \frac{C_1P^{1/\kappa} - C_2P^{-1}}{C_3 + C_4P^{\frac{1+\kappa}{\kappa}} - C_5P^{2/\kappa}}\,dP + \int_0^L dL = 0 \tag{17}$$

Equation (17) should be solved by numerical methods until the value of the first integral be aproximated to the length of the proposed pipe's section: the shorter the pipe's section and the more subdivisional intervals, the aproximation will be more precise. In these study has been utilized the Simpson's rule for its simplicity and without sacrificing precision, in the search for abreviating computing time at equal o smaller times than 5 minutes per pipe's section.

SUPPLEMENTARY EQUATIONS

For the development of the this model is was necessary to adjust the following variables: temperature specific volumes, enthalpies and polytropic's constants as a function of pressure and humidity, utilizing the vapor tables {5} in the normal operational range of the Cerro Prieto I geothermal field (Mexicali, México). Another variables such as viscosity and specific heat of liquid and vapor were obtained from ref {6}, always considering a correlation coefficient bigger or equal to 99°/°.

Horlock equation {7} was utilized in the calculation of factor (κ) in the humid vapor region, which comes out to be very useful when applying it, particular to bare pipe and finding the isentropic efficiency for future developments.

$$\kappa = \frac{\kappa'}{1-\kappa' R\;(\eta-1)/Cp} \quad \text{(Horlock)} \tag{18}$$

In order to calculate the heat transfer coefficient several equations were taken into account, as those which appeared in Holman's book {8}, in Metais and Eckert {9} with a computarized diagram for free, forced and combined convection; as much as from the work by Chesney {10} Schroder {11} Koenig {12} and Ballesca {13}. The friction factor was estimate using the correlation presented by Churchill {15}, {16} and is valid for all flow regimes.

Results.

As a manner of an example the results which could be obtained from this programme are shown and the they are compared against lowering of pressure results calculated with some of the conventional equations that appeared in ref {14}

Entering data

Initial pressure (psia)	120
Mass rate (lbs/hr)	110 000
Humidity (Fraction)	0.9997

Steam trap efficiency	is not considered in this case
Orifice plate diameter	is not considered in this case
Insulator thermal conductivity (Btu-ft)/(hr-ft²°F)	0.03
Aluminum's insulator covering emissivity	0.09
Insulator thickness (inches)	3"
Room temperature (°F)	32
Wind velocity (miles/hr)	5
Pipe type and diameter	carbon steel 12" ST40S
Pipe relative rugosity	0.00015
Pipe total length -- (Ft)	1000

Step by step results are showed in the - table 1, the computing of this model makes easier for changes in the input data in -- such a way that it is possible to utilize- it to check up field test or else in vapor line design.

Following are given results of presure - drop obtained, using this model versus results obtained using other equations quo-- ted in other works {14}

ΔP (psi)	Equations
11.21	This model
8.65	Ideal isothermal flow
9.42	Darcy, modified for compre-- ssible flow
9.03	Babcock for steam (empirical)

In this results may be observed the pressure difference when the heat losses are- included with implicit variations associated thermodynamical properties. This model is going to be tested with operational -- data from Cerro Prieto (México) geother-- mal field and the results will appear in- a second report in June 1986.

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TABLE 1

APRIL 24 1986
WELL NUMBER XXX
PRESSURE DROP IN ORIFICE PLATE = 0 LBS/FT^2
STARTING ENTALPHY= 1190.15 BTU/LBM

LENGHT FTS.	INITIAL P. LBS/IN^2.	INITIAL T. FAR. DEG.	DRYNESS %	INITIAL H. BTU/LBM	FINAL P. LBS/IN^2.	FINAL T. FAR. DEG.	SUP. TEMP. FAR.DEG	FINAL H. BTU/LBM.
50.0000	120.0000	341.3130	0.9993790	1190.1500	119.4790	341.0680	47.6011	1189.7900
100.0000	119.4790	340.9860	0.9990560	1189.7900	118.9580	340.7400	47.5789	1189.4300
150.0000	118.9580	340.6580	0.9987360	1189.4300	118.4380	340.4110	47.5677	1189.0700
200.0000	118.4380	340.3290	0.9984040	1189.0700	117.9170	340.0810	47.5453	1188.7100
250.0000	117.9170	339.9980	0.9980750	1188.7100	117.3960	339.7500	47.5341	1188.3400
300.0000	117.3960	339.6660	0.9977440	1188.3400	116.8400	339.4170	47.5115	1187.9700
350.0000	116.8400	339.3110	0.9974120	1187.9700	116.2850	339.0600	47.4990	1187.6000
400.0000	116.2850	338.9540	0.9970770	1187.5900	115.7290	338.7030	47.4752	1187.2200
450.0000	115.7290	338.5960	0.9967410	1187.2100	115.1740	338.3430	47.4626	1186.8400
500.0000	115.1740	338.2360	0.9964020	1186.8300	114.6180	337.9830	47.4385	1186.4500
550.0000	114.6180	337.8760	0.9960610	1186.4500	114.0630	337.6210	47.4259	1186.0700
600.0000	114.0630	337.5130	0.9957180	1186.0600	113.5070	337.2570	47.4016	1185.6800
650.0000	113.5070	337.1490	0.9953730	1185.6800	112.9170	336.8920	47.3888	1185.2900
700.0000	112.9170	336.7600	0.9950260	1185.2800	112.3260	336.5030	47.3632	1184.8900
750.0000	112.3260	336.3700	0.9946760	1184.8800	111.7360	336.1110	47.3491	1184.4900
800.0000	111.7360	335.9790	0.9943240	1184.4800	111.1460	335.7180	47.3233	1184.0800
850.0000	111.1460	335.5850	0.9939700	1184.0700	110.5560	335.3240	47.3091	1183.6700
900.0000	110.5560	335.1900	0.9936120	1183.6600	109.9650	334.9270	47.2830	1183.2600
950.0000	109.9650	334.7930	0.9932530	1183.2500	109.3750	334.5290	47.2688	1182.8500

STEAM TRAP PURGED WATER = 0 LBM/HR

TOTAL MASS FLOW IN STEAM LINE = 110000 LBM/HR

FINAL ENTHALPY IN THIS TOTAL SECTION = 1182.43 BTU/LBM

TOTAL ENTALPHY DROP = 7.71448 BTU/LBM

TOTAL HANDLED LENGHT = 1000 FTS

PRESSURE IN THIS TERMINAL POINT = 108.785 LBS/IN^2

DRYNESS IN THIS POINT IS = .992891

ANOTHER RESULTS OF INTEREST
############################

FRICTION FACTOR (LAST) .0133609
OVERALL HEAT TRANSFER COEF. .441416 BTU/HR.-FT-FAR DEG.
CONDENSATE FLOW (LAST 50 FT.) 7.54246 LBS/HR