CONDENSATION COEFFICIENT AND WETNESS THERMAL LOSSES OF WET STEAM FLOW AT LOW PRESSURES

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ABSTRACT

Condensation coefficient of wet steam flow at low pressures has been evaluated theoretically. It was found from the theoretical results that wet steam condensation coefficient depends upon: the pressure at Wilson point, vapour supercooling, droplet size, evaporation coefficient and index of isentropic expansion.

Available data on the condensation coefficient of wet steam flow through nozzles are found to be dependent on the pressure values of both initial flow conditions and Wilson point. A comparison has been conducted between the present values of the condensation coefficient and that reported and required to minimize the difference between the predicted and measured pressure distributions in wet steam flow through nozzles. The agreement in the comparison is good. In order to verify the model, another comparison between a measured pressure distribution and a predicted one through a nozzle based on selected values for the condensation and evaporation coefficients is also carried out. The quality of agreement in this comparison is good.

Moreover, entropy increase due to the irreversibility of heat and mass transfer processes in wet steam has been estimated numerically. The numerical results show that increasing the condensation coefficient, droplet size and expansion rate as well as decreasing the evaporation coefficient tends to increase the wetness thermal losses during wet steam flow.
1. INTRODUCTION

In the last few decades, problems of wetness loss in the last stages of steam turbines of both traditional and nuclear power stations have been studied with concern to the condensation characteristics of flowing steam [1-3]. During nucleation and condensation processes, two different coefficients are of great significance and they are referred to as condensation and evaporation coefficients. Condensation coefficient defining the fraction of the impinging vapour molecules which adhere to the droplet surface. Similarly, the evaporation coefficient denotes that fraction of the impinging molecules which rebound from the droplet surface.

In view of pioneering measurements of these coefficients, Sherwood and Johannes [4] obtained different values for the condensation coefficient of condensing liquids and they concluded that not all liquids have condensation coefficient equal to unity. Recently, Young [5] has suggested that these coefficients have similar values in steam flow under equilibrium conditions when the condensation rate is zero and they have different values when a net condensation effect is occurring and nonequilibrium conditions prevail. Nevertheless, it is common to use the value of the condensation coefficient for water as unity and the evaporation coefficient to be zero in most of the numerical models which treat nucleation and condensation processes [5]. Gajewski and others [6] have presented a theoretical model that permits to vary the values of both the condensation and evaporation coefficients for a nonequilibrium condensation process in water vapour.

It may be observed from the previous literature, which declare the full gasdynamic behavior of condensing streams, that there are pronounced differences among the predicted pressure distributions beside the droplets radii and their measured values. The good agreement between the computed parameters of condensing flows using condensation theories and the measured ones can facilitate the difficulties in describing and solving many of the condensation problems such as flow discontinuities and losses. The unique solution to bring the predicted pressure distribution of a condensing flow closely to the measured one was obtained by Hill [7] and Saltanov et al. [8]. This solution is based on using different values for the condensation and evaporation coefficients as shown in Fig. (1). It is interesting to note that they used random values for these coefficients. At present, the only attempt to conjoin the condensation coefficient with the pressure value at Wilson point in free molecular, intermediate and continuum flow regimes has been reported by Young [5]. He used most of the available data to estimate the required values of the condensation coefficient in order to match the computed pressure distributions of steam flow through nozzles with that of experiments in literature.
It is well known that occurrence of interphase temperature difference and spontaneous condensation in an expanding steam flow within nozzles or blades causes an irreversible heat and mass transfer resulting in an overall entropy increase on the steam flow behind the nozzle or blade exit. This entropy increase is known as the thermodynamic wetness loss and may be defined as the thermodynamic wetness loss coefficient \([2,9]\). Intuitively that rate of condensation or moisture accumulation in wet steam and, consequently, the condensation coefficient must control the thermodynamic wetness loss. Therefore, using inaccurate values of steam condensation coefficient alters the wetness losses of such flows. Lately, several authors have derived theoretical \([2,9]\) or empirical \([10]\) formulations for the thermodynamic wetness loss coefficient. None of these formulations correlate the dependence of the condensation or evaporation coefficients on the thermodynamic wetness loss.

The work in this paper is divided into two sections. In the first, a theoretical formulation for the condensation coefficient evaluation as a function of wet steam main characteristics has been carried out. The results of this theoretical formulation are compared with those required to combine both the predicted and measured pressure distributions in condensing wet steam. In the second section, both the condensation and evaporation coefficients have been introduced in a theoretical model which quantifies the thermodynamic wetness loss of steam in order to study the effect of these coefficients variation on the irreversibility of condensing steam.

2- FORMULATION OF THE PROBLEM

Assumptions were made in the theoretical formulation leading to the evaluation of the condensation and/or evaporation coefficients and the thermodynamic wetness loss as follows:

a- Steam is wet and consequently consists of a homogenous mixture of vapour and spherical water droplets.

b- The two phases are in thermal disequilibrium characterized by the temperature difference \((T_e - T_g)\).

c- Velocity slip between the two phases is neglected. This is because the droplets formed in nozzles and blades during fog condensation are very small and with diameters in the range of 0.04-2.0 μm. This assumption was considered in most of the previous researches (e.g. Young \([1,2 & 3]\), Hill \([7]\) and Saltanov et al.\([8]\)).

d- Different values for the effective expansion index of the vapour phase are considered to indicate the effect of the degree of steam relaxation.

e- The thermodynamic properties of the steam two phases are obtained using the empirical equations deduced by \([5,11]\) to simulate the accurate values in steam tables by Keenan, et al. \([12]\).
2.1 Formulation of the Condensation Coefficient

The condensation coefficient is defined, using the kinetic theory of gases, as the ratio of the vapour molecules sticking upon the droplet surface to those impinging on it [13]. Following this definition, the condensation coefficient is introduced as

\[ q_c = \frac{N_e}{N_i} \]  

(1)

where, \( N_i \) is the flux of incident vapour molecules and \( N_e \) is the condensed flux.

By analogy with Eqn. (1), the vaporation coefficient of the droplet in wet steam is defined as

\[ q_v = \frac{N_e}{N_{em}} \]  

(2)

where, \( N_{em} \) is the total flux of molecules emitted from the droplet surface under equilibrium vapour conditions while \( N_e \) is the emitted flux of molecules under nonequilibrium conditions in wet steam.

Condensation of steam in low pressure situations within steam turbines is considered as a free molecular process [5,14]. Therefore, simultaneous molecular heat and mass transfer is considered here. Upon this approach, a droplet at a temperature \( T_r \) and radius \( r \) is bombarded by vapour molecules from a stationary Maxwellian vapour of pressure \( P \) and temperature \( T_v \). Under this nonequilibrium state, condensation effect occurs and the rate of droplet growth is expressed as

\[ m_c = m_a - m_e = 4\pi r^2 \rho_r \frac{dr}{dt} \]  

(3)

where, \( m_a \) is the mass of vapour molecules adhering to the droplet surface in unit time and \( m_e \) is the mass of vapour molecules rebound (evaporating) from the droplet surface in unit time also. These rates were derived by Gyarmathy in [13] as

\[ m_a = q_c \frac{4\pi r^2 P}{\sqrt{2\pi R_g T_v}} \]  

(4)

and,

\[ m_e = q_e \frac{4\pi^2 P(T_r,r)}{\sqrt{2\pi R_g T_v}} \]  

(5)

Introducing equations (4) and (5) into Eqn. (3) gives

\[ 4\pi r^2 \rho_r \frac{dr}{dt} = q_c \frac{4\pi r^2 P}{\sqrt{2\pi R_g T_v}} - q_e \frac{4\pi^2 P(T_r,r)}{\sqrt{2\pi R_g T_v}} \]  

(6)

The heat balance of growing droplet by condensation in wet steam is given by

\[ \dot{Q} = \dot{Q} \]  

(7)
where, $\dot{Q}$ is the rate of heat transfer from the droplet surface and $\dot{Q}_r$ is the rate of latent heat liberation due to vapour condensation upon the droplet surface. These heat transfer rates are defined hereinafter according to the molecular laws of heat transfer as

$$\dot{Q} = 4\pi r^2 q_c \frac{k+1}{k-1} \frac{PR}{\sqrt{8\pi R_g T_g}} (T_r - T_g)$$  \hfill (8)

$$\dot{Q}_r = h_{fg} A \pi r^2 \rho_c \frac{dr}{dt}$$  \hfill (9)

By combining equations (8) and (9) with Eqn. (7), it is possible to obtain an expression for the rate of droplet growth $dr/dt$. Substituting this expression into Eqn. (6), yields the droplet heat balance as

$$\frac{4\pi r^2 q_c}{h_{fg}} \frac{k+1}{k-1} \frac{PR}{\sqrt{8\pi R_g T_g}} (T_r - T_g) = q_c \frac{4\pi r^2}{\sqrt{2\pi R_g T_g}} - q_c \frac{4\pi r^2}{\sqrt{2\pi R_g T_r}}$$  \hfill (10)

Rearranging the above equation gives

$$\frac{R_g}{2h_{fg}} \frac{k+1}{k-1} (T_r - T_g) = q_c \frac{P}{q_c} \frac{T_g}{P} \sqrt{T_r}$$  \hfill (11)

Now, the condensation coefficient is deduced from the final form of the droplet heat balance; Eqn. (11); to be

$$q_c = q_c \left\{ \frac{R_g}{2h_{fg}} \frac{k+1}{k-1} (T_r - T_g) \right\} + \left\{ \frac{P(T_r, r)}{P} \sqrt{T_r} \right\}$$  \hfill (12)

The difference between droplet and vapour temperatures, $(T_r - T_g)$, was related by Gyarmathy [14] to the vapour supercooling as

$$T_r - T_g = \Delta T - \Delta T_c$$  \hfill (13)

where $\Delta T_c$ is the capillary supercooling,

$$\Delta T_c = \frac{2\sigma T_r}{\rho h_{fg} r^2}$$  \hfill (14)

Combination of equations (13) and (14) into Eqn. (12) gives the condensation coefficient finally as,

$$q_c = q_c \left\{ \frac{R_g}{2h_{fg}} \frac{k+1}{k-1} (\Delta T - \Delta T_c) \right\} + \left\{ \frac{P(T_r, r)}{P} \sqrt{T_r} \right\}$$  \hfill (15)
Equation (15) reveals that the condensation coefficient is directly proportional to the evaporation coefficient, steam gas constant, vapour supercooling, capillary supercooling, ratio of droplet pressure to vapour pressure and ratio of vapour temperature to droplet temperature and is inversely proportional to the heat of vaporization.

2.2 Formulation of Condensation Thermal Losses

It has been shown by a number of investigators that the energy loss coefficient is the most convenient parameter to evaluate the thermodynamic wetness loss of wet steam expanding in a nozzle, blade or any similar devices. This coefficient is defined in [2,9] by

\[ \xi_Y = \frac{T_y \Delta s}{C_e^2/2} \] (16)

where, \( C_e \) is the flow exit velocity. The total entropy increase \( \Delta s \) of wet steam is derived here due to Young [5]. It is seen from the basics of two phase flow thermodynamics that the total specific entropy of nonnucleating wet steam is given by

\[ s = (1 - Y)_s + Y.s_t \] (17)

This equation can be expressed in differential form by

\[ ds = (1 - Y)ds_g - s_g.dY + Y.ds_g + s_t.dY \] (18)

It is seen from the basic differential equations of thermodynamics that

\[ T_ds = dh - v.dP \] (19.a) or

\[ T_g ds_g = dh_g - \frac{dP}{\rho_g} \] (19.b)

Similarly, the change in water phase entropy can be expressed by

\[ T_ds_t = dh_t - \frac{dP}{\rho_t} \] (20)

Combining equations (18), (19) and (20) with the following differential forms for the energy and momentum equations

\[ (1 - Y)_s dh_g - dY.(h_g - h_t) + Y.dh_t + C.dC = 0 \] (21)

\[ C.dC + \frac{1 - Y}{\rho_g}.dP = 0 \] (22)

gives after arranging that
Introducing suitable thermodynamic relations for the enthalpy and entropy differences between the vapour and liquid phases into Eqn. (23) and after some manipulation it becomes

\[
\begin{align*}
\frac{ds}{dT} &= \left[\frac{h_f - h_L}{T_f} - (s_f - s_L)\right]dY - Y\left(\frac{1}{T_f} - \frac{1}{T_L}\right)dh_t \\
&= \left\{h_f - C_p\left[T_s(P) - T_f\right]\right\}\left[\frac{1}{T_f} - \frac{1}{T_L(P)}\right]dY
\end{align*}
\]  

(24)

Neglecting the term \(\{C_p\left[T_s(P) - T_f\right]\}\) in Eqn. (24) and assuming for low pressure situations that \(\Delta T_c \to 0\) yields

\[
\begin{align*}
\frac{ds}{dT} &= h_f\left(\frac{1}{T_f} - \frac{1}{T_L}\right)\frac{1}{T_L(P)}dY \\
&= h_f\frac{\Delta T}{T_s(T_f - \Delta T)}dY
\end{align*}
\]  

(25)

Equation (13) can be simplified to

\[
\Delta T = T_s - T_f
\]

or

\[
T_f = T_s - \Delta T
\]  

(26)

Substituting \(T_f\) from Eqn. (26) into Eqn. (25) gives

\[
\frac{ds}{dT} = h_f\frac{\Delta T}{T_s(T_f - \Delta T)}dY
\]  

(27)

Considering \(\Delta T.T_f \approx 0\) comparing to \(T_s^2\) in Eqn. (27) for the microscopic-sized droplets of \(d << 1.0 \mu m\) which were generated in wet steam within the situations of low pressure reduces the entropy increase to the final form of

\[
\frac{ds}{dT} = h_f\frac{\Delta T}{T_s^2}dY
\]

or

\[
\frac{ds}{dT} = h_f\frac{\Delta T}{T_s^2} = \frac{dy}{dt}
\]  

(28)

Solving equations (28) and (16) in order to obtain the wetness loss coefficient requires a knowledge about the rate of steam wetness increase \(dY/dt\). This rate was expressed previously by Gyarmathy [14] and Young [9] as

\[
\frac{dY}{dt} = C_p\frac{(1 - Y)}{\tau_f.h_f}\left(\Delta T - \Delta T_c\right)
\]  

(29)
where \( \tau_T \) is the thermal relaxation time defined by

\[
\tau_T = \frac{(1 - Y).C_{pes}.r^2.\rho_t}{3\lambda_y Y} \left[ 1 + 3.78(1 - v).\frac{K_h}{Pr_s} \right]
\]

In order to introduce the effect of changing the values of the condensation and evaporation coefficients on the obtained numerical values of the wetness losses, Eqn. (15) and be rewritten to be

\[
\Delta T - \Delta T_e = \frac{q_e - \left[ \frac{P_s(T_e,r)}{P} \right]^{\frac{T_e}{T_r}}}{R_x \frac{k + 1}{k - 1}} \tag{31}
\]

Combining equations (31) and (29) gives

\[
\frac{dY}{dt} = \frac{C_{pes}.(1 - Y)}{\tau_T} \frac{2}{\left[ q_e - \frac{P_s(T_e,r)}{P} \right]^{\frac{T_e}{T_r}} \frac{k + 1}{k - 1}} \tag{32}
\]

In wet steam circumstances, the so-called "expansion rate" \( \{i.e., P = (1/P). \frac{dP}{dt}\} \) plays a significant role in affecting the condensation characteristics. Therefore, the effect of expansion rate variation on the calculated values of wetness losses is inserted also here. Insertion of the expansion rate effect into the computed values of wetness losses requires the approximation of Eqn.(28) to a suitable form. This approximation was carried by a number of authors [2,3] as follows

\[
\frac{ds}{dt} = \frac{h_{es} \cdot \Delta T}{T_i^2} \frac{dY}{dt} \approx \frac{(1 - Y).C_{pes} \Delta T^2}{\tau_T} \tag{33}\]

Considering the semi-analytical technique which was presented by Young[2] to integrate Eqn.(33), the required entropy increase which includes the expansion rate is given by

\[
\Delta s = \frac{(1 - Y).C_{pes}.(F.P.\tau_T)^2}{T_i^2} \left[ \frac{t}{\tau_T} - (1 - e^{-\frac{t}{\tau_T}}) \right] \tag{34}
\]

where,

\[
F = \frac{P}{(1 - Y).C_{pes} \rho_s} \left[ \frac{C_p \cdot T_s}{h_{es}} - (1 - Y).(\alpha \cdot T_s) \frac{P_s}{\rho_s} \right] \tag{35}
\]

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3. NUMERICAL CALCULATIONS

Now, the theoretical formulation that has been developed in the previous section for both the condensation/or evaporation and wetness loss coefficients are used to produce a computer program. This program is executed using a personal computer in order to evaluate the expected variations of these coefficients.

Calculations of condensation coefficient variations due to different conditions of steam within the investigated range of pressure at Wilson point (0.1-1.0 bar) are carried out by solving Eqn. (15) with the aid of Eqns. (13-14) and empirical equations for $h_0$ and $P_e(T)$ as in [5,11]. These equations are accurate to within 1% for the pressure range investigated. Conditions of steam used in the calculations are as follows:

i- Vapour supercooling of 1.0, 5.0, 10.0, 20.0 and 40.0 K.
ii- Droplet size (diameter) of 1.0, 2.0, 4.0 and 6.0 μm.
iii- Evaporation coefficient of 1.0, 0.8, 0.4, and 0.2.
iv- Isentropic index of expansion as 1.1, 1.2, 1.3 and $k = f(Y, P)$. In order to perform the relation $k = f(Y, P)$ in the calculations, a recent formula for $k$ in wet steam that has been discussed in [1] is used in the present study as

$$k = 0.603 + a_i \left[ 1 - \left( \frac{P}{P_f} \right)^{\gamma_i} \right]$$  \hspace{1cm} (36)

where,

$$a_i = 0.522 - 0.1418 \left[ \frac{Y}{(1-Y)} \right]$$

and

$$a_i = 1.34565 - 0.76825(1-Y)$$

These values of $k$ were selected because the index $k$ of wet steam was found to vary within these values in the whole range of condensation phenomena in water vapour. Numerical computations executed in this section have been taken over equal finite pressure intervals of 0.01 bar. It can be noticed here that there are some initial conditions are kept constant unless those which are required to be changed such as: $Y_i = 0.01$, $d_i = 2.0 \, \mu m$, $\Delta T_i = 1.0 \, K$, $q_e = 1.0$ and $k = 1.2$.

Solution of Eqn.(16) with the aid of equations (28-35) is performed using the results of steam nozzle flow obtained from a computer program [11] predicts the variations in wet steam properties during expansions through nozzles. Now, solution of Eqn.(16) which presents the coefficient of condensation thermal loss can be obtained against the flow pressure distribution (i.e., nozzle pressure ratio) or the corresponding vapour Mach number. To illustrate the variation of the wetness loss coefficient with changing both the condensation and evaporation coefficients, computations are released for the following:
i- Condensation coefficient of 1.0, 0.8, 0.4, 0.2 and 0.1.
ii- Evaporation coefficient of 1.0, 0.8, 0.4 and 0.1.
iii- Droplet size of 1.0, 0.5, 0.2 and 0.1 µm.
iv- Expansion rate of -15000.0, -10000.0, -5000.0 and -1000.0 s⁻¹.

All these numerical calculations of the wetness loss coefficient are taken for constant initial conditions of \( P_0 = 2.0 \) bar, \( q_e = 0.4 \), \( q_c = 1.0 \), \( k = 1.2 \) and \( d_i = 2.0 \) µm.

4. EXPERIMENTAL APPARATUS AND MEASUREMENTS

The general arrangement of the experimental apparatus is shown in Fig. (2). This apparatus consists mainly of a fire-tube boiler, nozzle (2) and a surface condenser. The boiler supplies the apparatus with wet steam of wetness \( Y = 0.5\% \) at a rate of one ton per hour and at a pressure of 6.0 bar. The nozzle was made of brass and is fitted (screwed), vertically in downward direction for its exit section, at the bottom of the settling chamber (1).

The axial pressure distribution in nozzle flow, steam wetness fraction, static vapour temperature and steam flow rate were measured in the experimental program. The axial pressure distribution along the nozzle axis was obtained using Stodola search tube (3) equipped with a pressure transducer (4). The steam dryness fraction ahead of nozzle entrance besides static vapour temperature were measured using throttling calorimeter (7) and thermometer (6) respectively. Flowing steam in the nozzle was rated using a metering tank after the steam condensation in the condenser. The steam flow rate is used to obtain the vapour velocity within the settling chamber.

5. RESULTS AND DISCUSSION

Results of numerical calculations for condensation coefficient variation due to different values of vapour supercooling, droplet size, evaporation coefficient and index of isentropic expansion in wet steam are plotted in Fig. (3). In this figure, it will be firstly seen that increasing the pressure at Wilson point through a certain interval kept the condensation coefficient to be constant and with further \( P_w \) increasing \( q_e \) will be decreased. The reason for \( q_e \) constancy at the lower values of \( P_w \) is due to the small values of vapour supercooling and consequently the small values of vapour temperature. This will keep the predicted values of \( q_e \) to be constant. Whilst the reason of \( q_e \) decreasing with \( P_w \) increasing for a certain amount of vapour supercooling is attributed to the inverse proportionality between \( P_w \) or \( P_0 \) and the latent heat. This causes to increase the capillary supercooling and then it is expected that the first term in the right side of Eqn. (15) becomes with negative sign and
therefore lowering the calculated values of \( q_c \). Besides, this plot of results reveals that \( q_c \) decreases for a constant value of \( P_w \) as the vapour supercooling decreases (Fig. 3.a), the droplet size increases (Fig.3.b), the evaporation coefficient decreases (Fig. 3.c) and the index of isentropic expansion increases from 1.1 to 1.3 (Fig. 3.d). Effect of vapour supercooling on \( q_c \) is explained with respect to Eqn. (13). One observes that increasing \( \Delta T \) tends to decrease \( T_g \) and then decreases \( q_c \). While the effect of droplet size on \( q_c \) could be drawn from the fact that as the droplet size decreases, the capillary supercooling from Eqn.(14) increases also and therefore the first term in Eqn.(15) will decrease and then lowering the predicted values of \( q_c \). It can be also seen in Fig.(3.b) that the predicted values of \( q_c \) for constant droplet diameter of 6.0 \( \mu \)m approximately coincide with those predicted for \( d = 4.0 \) \( \mu \)m. Effect of \( q_c \) on \( q_e \) as illustrated in Fig.(3.c) can easily be explained as follows. It can be noticed that decreasing \( q_e \) in the left side of Eqn. (15); while all the equation parameters are kept constant; results in decreasing the obtained values of \( q_c \). A similar result for the effect of changing \( q_e \) on the predicted values of \( q_c \) was performed previously by Hill [7]. Finally, results in Fig. (3.d) are concerned with the effect corresponding to the variation of wet steam thermodynamic behaviour with changing the index of isentropic expansion. Plots in Fig.(3.d) indicate that increasing the index \( k \) leads to decrease \( q_c \). This is because increasing \( k \) tends to increase the vapour phase enthalpy and then attenuates the incidence of vapour molecules upon the droplet surface.

In order to apply the model presented here in combining both the predicted and measured pressure distributions of wet steam flow, a preparative step is required before starting this combination. In this step, correlations between the measured values of inlet pressure (\( P_o \)) and Wilson point pressure (\( P_w \)) and also between \( P_w \) (or \( P_o \)) and values of \( q_e \) required to combine theory and experiment are performed and presented in Figs. (4) and (5) respectively. Figure (4) indicates a plot of \( P_o \) against \( P_w \) within the investigated pressure range. This plot yields the following correlation between \( P_o \) and \( P_w \):

\[
P_o = -0.106723 + 4.31596P_w - 1.99157P_w^2
\]  

This relation was used to conjugate the two -x axes in Fig. (5) to show \( q_c \) as a function in both \( P_o \) and \( P_w \). The symbolized values of \( q_c \) in Fig. (5) are required to match the predicted pressure distribution in wet steam flow through nozzles using theory of spontaneous condensation and that obtained previously in the experimental investigations. These symbolized values according to Ref. [5]. The computed values of \( q_c \) and \( q_e \) using present procedure, Eqn. (15), are presented also in this figure. Table (1) summarizes the boundary conditions at which the calculations in Fig. (5) were carried out. It will be observed in plots of Fig.(5) that the computed values of \( q_c \) combine completely with that required to verify theory with
experiments. Furthermore, the trend of $q_c$ variation with $P_w$ in Fig. (5) coincides completely with those of Fig. (3). This coincidence confirms the present model.

Table (1): Boundary conditions of numerical computations of both $q_e$ and $q_c$ which are required to combine theory and experiment.

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<th>Author</th>
<th>$P_o$, bar</th>
<th>$t_o$, $^\circ$C</th>
<th>$P_w$, bar</th>
<th>$t_w$, $^\circ$C</th>
<th>$\Delta T = (t(P_o)-t_w)$</th>
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<td>0.58</td>
<td>112.0</td>
<td>0.14</td>
<td>43.3</td>
<td>28.0</td>
<td>1.0</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Figure (6) shows a comparison between present prediction and measurement of a pressure distribution for wet steam flow through a converging-diverging nozzle. Prediction of pressure variation during wet steam flow through a nozzle with such geometry is carried out using the computer program in [11]. Nozzle geometry used in computations and experiment is showed also in Fig.(6). Numerical and experimental results in Fig. (6) are obtained under initial conditions of; $P_o = 2.3$ bar, $Y_1 = 0.006$, $\Delta T_1 = 3.0^\circ$C and $C=37.50$ m/s. This comparison is based upon selecting suitable values of 0.06 and 0.07 for $q_e$ and $q_c$ respectively from Fig. (5). Furthermore, other values of $q_e$ of 1.0 and 0.01 are taken with a constant value of $q_c$ equals unity in order to declare the discrepancy in the degree of agreement between theory and experiment of the pressure distribution. This figure reveals quite agreement between the measured pressure distribution and the predicted one which was obtained using the suitable values of $q_e$ and $q_c$ from Fig. (5), while the predictions of the other values of $q_e$ and $q_c$ differ clearly from the experimental one.

Figures (7-8) indicates the variation of the coefficient of condensation thermal loss for wet steam flows through a nozzle with changing some parameters such as the condensation and evaporation coefficients, droplet size and expansion rate. Effect of changing both $q_e$ and $q_c$ on $\xi$ for a nozzle flow of wet steam is given in Fig. (7). Nozzle flow is represented here as a predicted variation in the Mach number of vapour phase in Fig.(7) and as a pressure ratio in Fig.(8). Parameters of nozzle flow are obtained numerically for inlet stagnation pressure, wetness
fraction and vapour supercooling ahead of nozzle entrance as 2.0 bar, 0.02 and 20.0°K respectively. It will be observed in Figs. (7.a &b) that the coefficient of wetness loss ($\xi$) increases with increasing $q_e$ or decreasing $q_e$. Explanation of this tendency can be found in Eqn. (32). This equation shows that increasing $q_e$ and lowering $q_e$ yield to increase the wetness accumulation and then rise the flow entropy as shown clearly by Eqn. (28). It is evident also from Fig. (7) that $\xi$ increasing through two regions of nozzle flow namely ($0.4 < M_e < 0.5$) and ($1.4 < M_e < 1.8$) is followed by a notable reduction. This is attributed to the similar trend of supercooling variation which was obtained in the results of nozzle flow parameters. Finally, variation of $\xi$ with both the droplet size and expansion rate for the investigated nozzle flow is shown in Fig.(8). Results of Fig.(8) have approximate behaviour like those of Fig.(7). Generally speaking, results of Fig. (8) show that $\xi$ of wet steam flow increases with increasing both the droplet size and flow expansion rate. These results were confirmed previously by Kostyuk and Frolov [10] and Young [9].

6. CONCLUSIONS

A numerical procedure has been developed to estimate the coefficients of condensation and wetness thermal loss in wet steam at low pressures. In the theoretical results, the condensation coefficient has been shown to depend significantly upon the pressure at Wilson point, vapour supercooling, droplet size, evaporation coefficient and index of isentropic expansion. The theoretical results declare also that wetness thermal losses in steam flow through nozzles depend greatly upon the condensation and evaporation coefficients, droplet size and expansion rate. The predicted values of the condensation coefficient from the present model are in good agreement with those required to match the pressure distributions in wet steam flow using condensation theories and measured ones. Furthermore, a predicted pressure variation based on selected values of the condensation and evaporation coefficients from present model was in good agreement with a measured one in wet steam flow through a converging-diverging nozzle. These encouraging results make the model to be extended to investigate the behaviour of condensation coefficient and wetness loss variations in the transient and continuum flow regimes.

NOMENCLATURE

A       nozzle cross-sectional area.
C       absolute velocity of vapour phase.
$C_p$   isobaric specific heat capacity of the mixture.
\(C_{pg}\) isobaric specific heat capacity of vapour phase.

\(d\) droplet diameter.

\(h\) specific enthalpy.

\(h_{fe}\) specific enthalpy of evaporation.

\(Kn\) Knudsen number, \(= (1.5 \mu_d / dR_g) [(1/(R_g T_g))]^{0.5}\)

\(k\) index of isentropic expansion.

\(L\) nozzle length.

\(M_e\) Mach number of vapour phase.

\(m\) mass flow rate-rate of mass variation.

\(N_c\) flux of condensed vapour molecules on droplet surface.

\(N_e\) flux of emitted vapour molecules from droplet surface.

\(N_{env}\) total flux of molecules emitted from droplet surface under equilibrium vapour conditions.

\(N_i\) flux of incident vapour molecules upon droplet surface.

\(P\) vapour pressure.

\(Pr_g\) Prandtl number of vapour, \(= C_{pg} \mu_g / \rho_g\)

\(Q\) rate of heat transfer.

\(q_c\) condensation coefficient.

\(q_e\) evaporation coefficient.

\(R\) radius of nozzle passage.

\(R_g\) gas constant (=461.51 J/Kg.\(°\)K).

\(r\) droplet radius.

\(s\) specific entropy of wet steam.

\(T\) temperature.

\(T_g, T_e\) vapour, water temperature.

\(T_r\) droplet surface temperature.

\(T_s\) saturation temperature.

\(t\) time.

\(x\) distance in flow direction.

\(v\) specific volume.

\(Y\) wetness fraction.

\(\Delta s\) total entropy increase of wet steam.

\(\Delta T\) vapour supercooling.

\(\Delta T_{cap}\) capillary supercooling.

\(\lambda_g\) thermal conductivity of vapour phase.

\(\mu_g\) dynamic viscosity of vapour phase.

\(\nu\) coefficient presented in Eqn. (30) by Young. [5].

\(\zeta_T\) wetness thermal loss coefficient.

\(\rho\) density.

\(\sigma\) surface tension.

\(\tau_T\) time of thermal relaxation.

**Subscripts**

- \(a\) related to adhering molecules.
- \(c\) related to condensing molecules - condensation.
- \(e\) related to evaporating molecules - evaporation - exit conditions.
phase transition.
of the vapour phase.
of the initial conditions.
of the water phase.
of the initial stagnation conditions.
of the droplet surface.
saturation value.
at throat.
at Wilson point.
at critical conditions.

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Figure 1: Effect of variations in the condensation coefficient and evaporation coefficient on the shape of the pressure distribution in condensing steam flow through a nozzle:
- data of Hill [7], and b- data of Saltanov et al. [8].

Figure 2: Schematic of the experimental apparatus.
Figure 3: Variation of the condensation coefficient with pressure at Wilson point for different values of: a- vapor supercooling, b- droplet size, c- evaporation coefficient and d- index of isentropic expansion.

Figure 4: Variation of pressure at Wilson point with the inlet pressure.
Figure 5: Values of the condensation coefficient required to combine theory and experiment against the pressure at Wilson point. (Symbols denote results were reported by Young [5]).

Figure 6: A comparison between the predicted pressure distribution using $q_c$ & $q_e$ from the present model and the measured one through a nozzle.
Figure 7: Variation of the wetness loss coefficient with a nozzle Mach number for various values of: a – condensation coefficient, and b- evaporation coefficient.

Figure 8: Variation of the wetness loss coefficient with a nozzle pressure ratio for different values of: a- droplet size, and b- expansion rate.
معامل التكثف والمفاقيم الحرارية للرطوبة لانسياب البخار الرطب عند ضغوط منخفضه

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شبين الكوم - مصر

ملخص البحث:

تم في هذا البحث تقييم معامل التكثف لانسياب البخار البخار الرطب عند الضغوط المنخفضة نظرياً، وقد أظهرت النتائج النظرية أن معامل تكثف البخار الرطب يعتمد على مقدار الضغط عند نقطة ويسون - مقدار فرط تبريد البخار - حجم قطرات الماء - معامل التبخر وسن التمدد الأيزتروبي.

أظهرت البيانات المتاحة عن معامل التكثف لانسياب البخار البخار الرطب أن هذا المعامل يتوقف على قيم الضغط لكل من حالة الإنسيباب الإبتدائية عند نقطة ويسون، تم عقد مقارنة بين القيم الحالية لمعامل التكثف والأخرى المنشورة والمطلوبة لتقليد الفارق بين التنبؤات النظرية والقياسات العملية لتوزيع الضغط في بخار رطب بنسب خلا الأبواق، وعند التوافق في هذه المقارنة جيداً، وتأكيّد النموذج النظري لهذه البحث 00 فإن مقارنة أخرى قد أجريت بين قياس عملي لتوزيع الضغط في بخار رطب بنسب خلا بوق وبين التنبؤ نظري له مؤسس على قيم مختارة لمعامل التكثف والتبخر، ولهن جا التوافق في هذا المقارنة جيداً أيضاً.

آكثر من ذلك 00 فقد اهتم البحث بإستنباط زيادة الإيزتروبي الناجمة عن لا إرجاعية إجراءات إنقلال الحرارة والكتلة عديماً، وأظهرت النتائج النظرية أن زيادة كل من معامل التكثف حجم القطرات ومعدل التمدد بالإضافة إلى خفض معامل التبخر يفضى إلى زيادة المفاقيم الحرارية للرطوبة أثناء إنسيبان البخار الرطب.