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 proceses 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. Condesnation coefficient defining the fraction of the impinging vapour molecules which adhere to the droplet surface. Similarly, the evaporation coeficient denotes that fraction of the impinging molecules which rebound from the droplet surface.

In view of poineering 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 differnet values when a net condensation effect is occuring 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 nulceation and condensation processes [5]. Gajewski and others [6] have presented a theoretical model that permitis 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 behaviour 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 intersting 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 authours have derived theoretical [2,9] or empirical [10] formulations for the thermodynamic wetness loss coefficient. None of theses 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 theroetical 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 vairation 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 consequantly consists of a homogenous mixture of vapour and spherical water droplets.

b- The two phases are in thermal disequilibrium characterized by the temperture difference (T_s-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 Formualtion of the Condensation Coefficient

The condensation coeficient 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

$$q_c = \frac{N_c}{N_i}$$ (1)

where, N_i is the flux of incident vapour molecules and N_c is the codnensed flux.

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

$$q_e = \frac{N_e}{N_{em}}$$ (2)

where, N_{em} is the total flux of molecules emitted from the dropelt 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 sationary Maxwellian vapour of pressure P and temperature T_g. Under this nenequilibrium state, condensation effect occurs and the rate of droplet growth is experssed as

$$\dot{m}_c = \dot{m}_a - \dot{m}_e = 4\pi r^2 \rho_\ell \cdot \frac{dr}{dt}$$ (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

$$\dot{m}_a = q_c \cdot \frac{4\pi r^2 P}{\sqrt{2\pi R_g T_g}}$$ (4)

and, $$\dot{m}_e = q_e \cdot \frac{4\pi r^2 \cdot P_s(T_r, r)}{\sqrt{2\pi R_g T_r}}$$ (5)

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

$$4\pi r^2 \rho_\ell \cdot \frac{dr}{dt} = q_c \cdot \frac{4\pi r^2 P}{\sqrt{2\pi R_g T_g}} - q_e \cdot \frac{4\pi r^2 \cdot P_s(T_r, r)}{\sqrt{2\pi R_g T_r}}$$ (6)

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

$$\dot{Q} = \dot{Q}_r$$ (7)

where, $\dot{Q}$ is the rate of heat transfer from the droplet surface and $\dot{Q}_\ell$ 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 lows of heat transfer as

$$\dot{Q} = 4\pi r^2 . q_e . \frac{k+1}{k-1} . \frac{PR_g}{\sqrt{8\pi R_g T_g}} . (T_r - T_g) \tag{8}$$

$$, \dot{Q}_\ell = h_{fg} . 4\pi r^2 . \rho_\ell . \frac{dr}{dt} \tag{9}$$

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

$$\frac{4\pi r^2 . q_e}{h_{fg}} . \frac{k+1}{k-1} . \frac{PR_g}{\sqrt{8\pi R_g T_g}} . (T_r - T_g) = q_c . \frac{4\pi r^2 . P}{\sqrt{2\pi R_g T_g}} - q_e . \frac{4\pi r^2 . P_s(T_r, r)}{\sqrt{2\pi R_g T_r}} \tag{10}$$

Rearranging the above equation gives

$$\frac{R_g}{2h_{fg}} . \frac{k+1}{k-1} . (T_r - T_g) = \frac{q_c}{q_e} - \frac{P_s(T_r, r)}{P} \sqrt{\frac{T_g}{T_r}} \tag{11}$$

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

$$q_c = q_e . \left\{ \left[\frac{R_g}{2h_{fg}} . \frac{k+1}{k-1} . (T_r - T_g) \right] + \left[\frac{P_s(T_r, r)}{P} . \sqrt{\frac{T_g}{T_r}} \right] \right\} \tag{12}$$

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

$$T_r - T_g = \Delta T - \Delta T_c \tag{13}$$

where ΔT_c is the capillary supercooling,

$$\Delta T_c = \frac{2\sigma T_s}{\rho_\ell h_{fg} r} \tag{14}$$

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

$$q_c = q_e . \left\{ \left[\frac{R_g}{2h_{fg}} . \frac{k+1}{k-1} . (\Delta T - \Delta T_c) \right] + \left[\frac{P_s(T_r, r)}{P} . \sqrt{\frac{T_g}{T_r}} \right] \right\} \tag{15}$$

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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

$$\xi_T = \frac{T_s.\Delta s}{C_e^2/2} \tag{16}$$

where, C_e is the flow exit velocity. The total entropy increase Δs 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

$$s = (1-Y).s_g + Y.s_f \tag{17}$$

This equation can be expressed in differential form by

$$ds = (1-Y).ds_g - s_g.dY + Y.ds_f + s_f.dY \tag{18}$$

It is seen from the basic differential equations of thermodynamics that

$$T.ds = dh - v.dP \tag{19.a}$$

or

$$T_g.ds_g = dh_g - \frac{dP}{\rho_g} \tag{19.b}$$

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

$$T_f.ds_f = dh_f - \frac{dP}{\rho_f} \tag{20}$$

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

$$(1-Y).dh_g - dY.(h_g - h_f) + Y.dh_f + C.dC = 0 \tag{21}$$

$$C.dC + \frac{1-Y}{\rho_g}.dP = 0 \tag{22}$$

gives after arranging that

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$$ds = \left[\frac{h_g - h_t}{T_g} - (s_g - s_t)\right].dY - Y.\left(\frac{1}{T_g} - \frac{1}{T_t}\right).dh_t \qquad (23)$$

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

$$ds = \left\{h_{fg} - C_{pg}.\left[T_s(P) - T_g\right]\right\}.\left[\frac{1}{T_g} - \frac{1}{T_s(P)}\right].dY \qquad (24)$$

Neglecting the term $\{C_{pg}.[T_s(p) - T_g]\}$ in Eqn. (24) and assuming for low pressure situations that $\Delta T_c \to 0$ yields

$$ds = h_{fg}.\left(\frac{1}{T_g} - \frac{1}{T_s}\right).dY \qquad (25)$$

Equation (13) can be simplified to

$$\Delta T = T_s - T_g$$
$$\text{or} \qquad\qquad\qquad \left.\right\} \qquad\qquad\qquad (26)$$
$$T_g = T_s - \Delta T$$

Substituting T_g from Eqn. (26) into Eqn. (25) gives

$$ds = h_{fg}.\frac{\Delta T}{T_s(T_s - \Delta T)}.dY \qquad (27)$$

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

$$ds = h_{fg}.\frac{\Delta T}{T_s^2}.dY$$
$$\text{or} \qquad\qquad\qquad \left.\right\} \qquad\qquad\qquad (28)$$
$$\frac{ds}{dt} = h_{fg}.\frac{\Delta T}{T_s^2}.\frac{dY}{dt}$$

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

$$\frac{dY}{dt} = \frac{C_{pg}.(1 - Y)}{\tau_T.h_{fg}}.(\Delta T - \Delta T_c) \qquad (29)$$

where τ_T is the thermal relaxation time defined by

$$\tau_T = \frac{(1-Y).C_{pg}.r^2.\rho_\ell.\left[1+3.78(1-\upsilon).\dfrac{Kn}{Pr_g}\right]}{3\lambda_g Y} \qquad (30)$$

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

$$\Delta T - \Delta T_c = \frac{\dfrac{q_c}{q_e} - \left[\dfrac{P_s(T_r,r)}{P}.\sqrt{\dfrac{T_g}{T_r}}\right]}{\dfrac{R_g}{2h_{fg}}.\dfrac{k+1}{k-1}} \qquad (31)$$

Combining equations (31) and (29) gives

$$\frac{dY}{dt} = \frac{C_{pg}.(1-Y)}{\tau_T}.\frac{2\left\{\dfrac{q_c}{q_e} - \left[\dfrac{P_s(T_r,r)}{P}\sqrt{\dfrac{T_g}{T_r}}\right]\right\}}{R_g.\dfrac{k+1}{k-1}} \qquad (32)$$

In wet steam circumstances, the socalled "expansion rate" {i.e., P = (1/P).(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

$$\frac{ds}{dt} = \frac{h_{fg}.\Delta T}{T_s^2}.\frac{dY}{dt} \cong \frac{(1-Y).C_{pg}}{T_s^.}.\frac{\Delta T^2}{\tau_T} \qquad (33)$$

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

$$\Delta s = \frac{(1-Y).C_{pg}}{T_s^2}.(F.P.\tau_T)^2.\left[\frac{t}{\tau_T} - (1-e^{-\frac{t}{\tau_T}})\right] \qquad (34)$$

where,

$$F = \frac{P}{(1-Y).C_{pg}.\rho_s}.\left[\frac{C_p.T_s}{h_{fg}} - (1-Y).(\alpha.T_g).\frac{\rho_s}{\rho_g}\right] \qquad (35)$$

3. NUMERICAL CALCULATIONS

Now, the theroetical 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 evalute 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_{fg} and $P_s(T)$ 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

$$k = 0.603 + a_o\left[1 - \left(\frac{P}{P_*}\right)^{3/2}\right]^{a_1} \tag{36}$$

where,

$$a_o = 0.522 - 0.1418\left[Y/(1-Y)\right]$$
$$,a_1 = 1.34565 - 0.76825(1-Y)$$

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 intervales 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: $Y_i = 0.01$, $d_i = 2.0$ μm , $\Delta T_i = 1.0$ 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] predictes the variations in wet steam properities 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 0.1μm.

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_c = 0.4, q_e = 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 supercoling, 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_c will be decreased. The reason for q_c constancy at the lower values of P_w is due to the small values of vapour supercooling and consequantly the small values of vapour temperature. This will keep the predicted values of q_c to be constant. Whilst the reason of q_c 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

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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 ΔT tendes to decrease T_g and then decreases q_c. While the effect of droplet size on q_c could be drown 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 μm approximatly coincide with those predicted for d = 4.0 μm. Effect of q_e on q_c as illustrated in Fig.(3.c) can easly 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 vairation 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_c 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:

$$P_o = -0.106723 + 4.31596 P_w - 1.99157 P_w^{\,2} \qquad (37)$$

This relation was used to conjugate the two -x axises in Fig. (5) to show q_c as a function in both P_o and P_w. The symboled 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 symboled 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_c and q_e which are requried to combine theory and experiment.

Author	P_o, bar	t_o,°C	P_w,bar	t_w,°C	$\Delta T = t.(P_w)\text{-}t_w$	q_c	q_e
Barschdorff & Ludwig [15]	1.45	134.2	0.539	46.6	36.61	0.114	0.149
	1.16	119.8	0.465	41.4	38.12	0.20	0.265
	0.93		0.275	28.0	39.1	0.73	0.85
	0.78	123.0	0.236	21.4	42.3	0.865	0.925
	0.78	108.0	0.318	32.9	37.57	0.60	0.825
Gyarmathy & Lesch[17]	1.03	125.0	0.38	38.6	36.06	0.28	0.378
Gyarmathy&Meyer[16]	2.05	175.0	0.515	47.5	34.56	0.13	0.1702
Moore et al. [25]	0.5		0.10			1.0	1.0
Moses & Stein [18]	1.0	85.0	0.80	67.1	26.4	0.07	0.087
	0.7	103.7	0.25	24.6	40.4	0.80	0.8875
	0.68	112.0	0.33	43.3	28.0	0.49	0.666
	0.58		0.14			1.0	1.0

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 obtaind under initial conditions of; $P_o = 2.3$ bar, $Y_i = 0.006$, $\Delta T_i = 3.0$°C and $C_i=37.50$ m/s. This comparison is based upon selecting suitable values of 0.06 and 0.07 for q_c and q_e respectively from Fig. (5). Furthermore, other values of q_c of 1.0 and 0.01 are taken with a constant value of q_e 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_c and q_e from Fig. (5), while the predictions of the other values of q_c and q_e differ clearly from the experimental one.

Figures (7-8) indicates the variation of the coefficinet 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_c and q_e on ξ 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 $20.0^{\circ}K$ respectively. It will be observed in Figs. (7.a &b) that the coefficient of wetness loss (ξ) increases with increasing q_c or decreasing q_e. Explanation of this tendency can be found in Eqn. (32). This equation shows that increasing q_c 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 ξ increasing through two regions of nozzle flow namely $(0.4 < M_g < 0.5)$ and $(1.4 < M_g < 1.8)$ is followed by a notable reduction. This is attributred to the similar trend of supercooling variation which was obtained in the results of nozzle flow parameters. Finally, variation of ξ 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 speeking, results of Fig. (8) show that ξ 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 coeffcieint and wetness loss variations in the transient and continum 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_{fg}	specific enthalpy of evaporation.
Kn	Knudsen number, $= (1.5\, \mu_g/d\rho_g)\, [1/(R_g T_g)]^{0.5}$
k	index of isentropic expansion.
L	nozzle length.
M_g	Mach number of vapour phase.
$\dot{m}$	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_{em}	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, $= C_{pg}\, \mu_g / \lambda_g$
$\dot{Q}$	rate of heat transfer.
q_c	condensation coefficeint.
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_ℓ	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.
Δs	total entropy increase of wet steam.
ΔT	vapour supercooling.
ΔT_{cap}	capillary supercooling.
λ_g	thermal conductivity of vapour phase.
μ_g	dynamic viscosity of vapour phase.
ν	coefficient presented in Eqn. (30) by Young. [5].
ζ_T	wetness thermal loss coefficient.
ρ	density.
σ	surface tension.
τ_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.

fg   phase transition.
g   of the vapour phase.
i   of the initial conditions.
l   of the water phase.
o   of the initial stagnation conditions.
r   of the droplet surface.
s   saturation value.
t   at throat.
w   at Wilson point.
*   at critical conditions.

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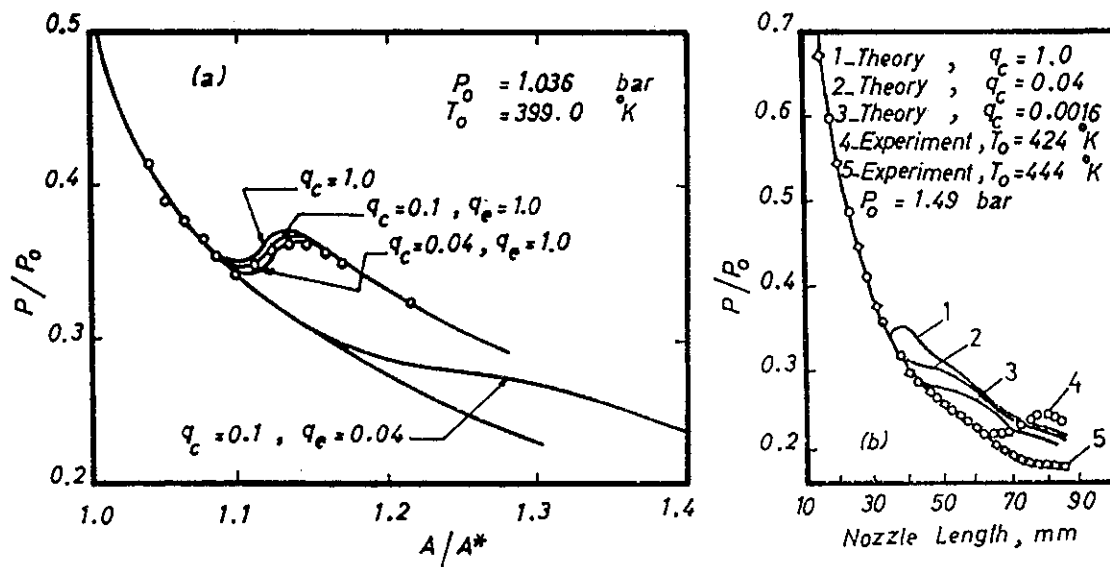
Figure 1: Effect of variations in the condensation coefficient and evaporation coefficient on the shape of the pressure distribution in condensing steam flow through a nozzle:
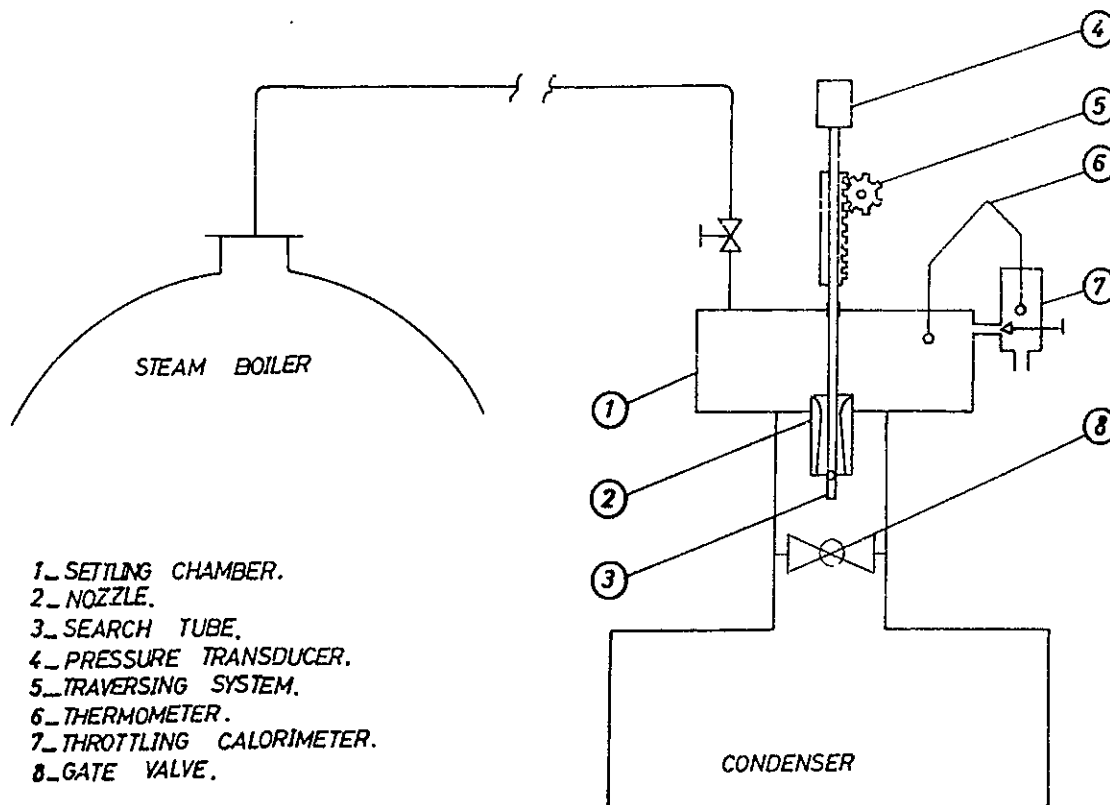a- data of Hill [7], and b- data of Saltanov et al. [8].

1_ SETTLING CHAMBER.
2_ NOZZLE.
3_ SEARCH TUBE.
4_ PRESSURE TRANSDUCER.
5_ TRAVERSING SYSTEM.
6_ THERMOMETER.
7_ THROTTLING CALORIMETER.
8_ GATE VALVE.

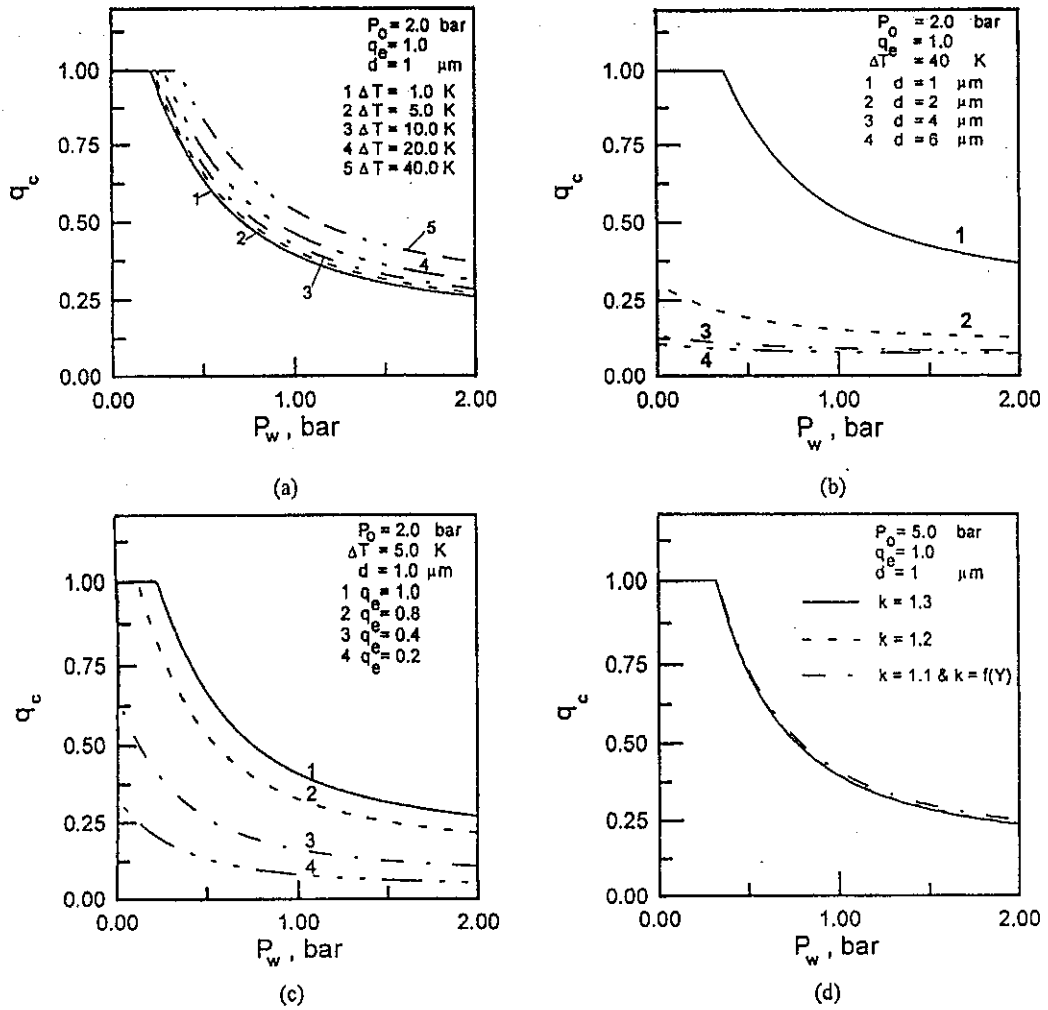*Figure 2 : Schematic of the experimental apparatus.*

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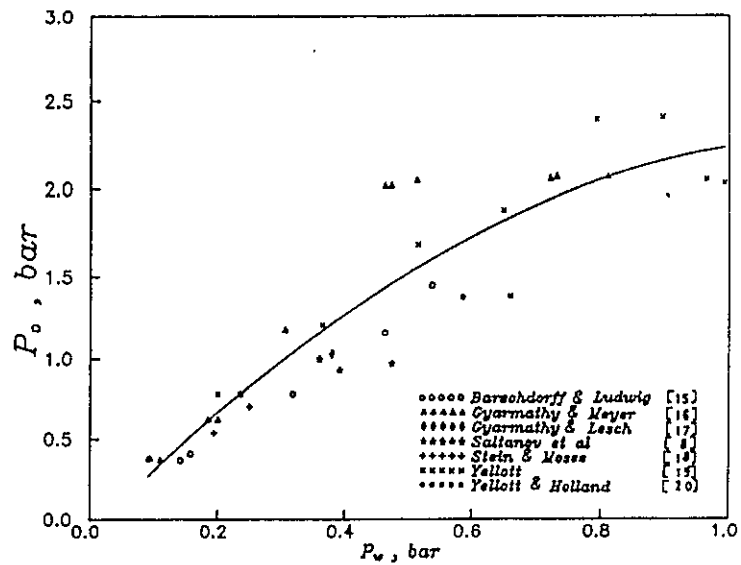c- evaporation coefficient and d- index of isentorpic expansion.

Figure 4: Variation of pressure at Wilson point with the inlet
pressure.

P$_o$, bar

0.50   0.80   1.10   1.40   1.70   2.00   2.30

O	Barschdorff et al	[15]
☐	Binnie et al	[21], [22]
△	Delch et al	[23]
☆	Gyarmathy & Lesch	[17]
◇	Krol	[24]
▲	Moore et al	[25]
X	Moses & Stein	[18]
——	q$_c$, present model	
- - - -	q$_e$, present model	

q$_c$ & q$_e$

1.00

0.75

0.50

0.25

0.00

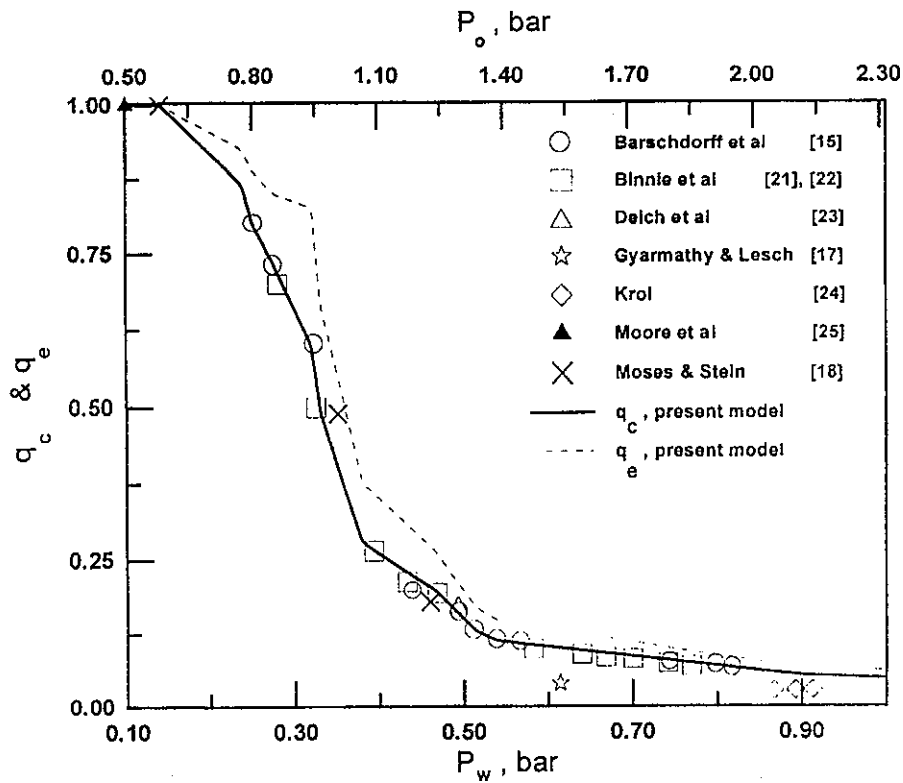0.10   0.30   0.50   0.70   0.90

P$_w$, bar

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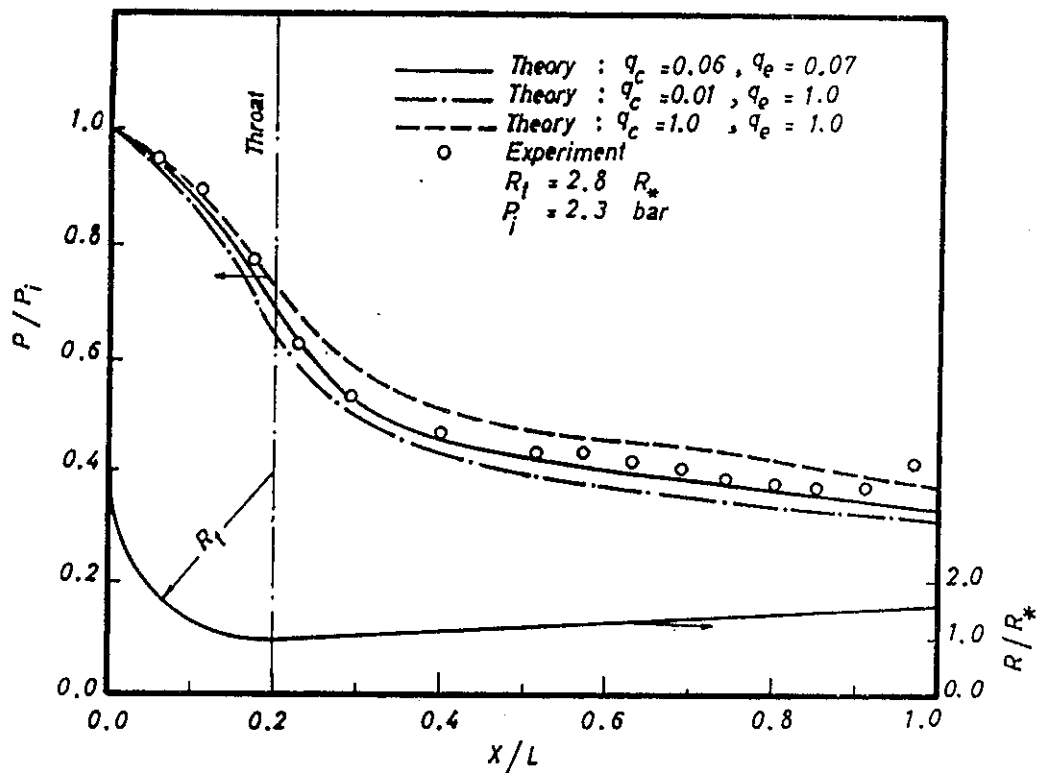(Symbols denote results were reported by Young [5]).

—————— Theory : q$_c$ =0.06 , q$_e$ = 0.07
—·—·— Theory : q$_c$ =0.01 , q$_e$ = 1.0
— — — Theory : q$_c$ =1.0 , q$_e$ = 1.0
   o   Experiment
        R$_I$ = 2.8 R$_*$
        P$_i$ = 2.3 bar

Throat

P / P$_i$

1.0

0.8

0.6

0.4

0.2

0.0

R$_1$

2.0

1.0

0.0

R / R$_*$

0.0   0.2   0.4   0.6   0.8   1.0

X / L

Figure 6: A comparison between the predicted pressure dis-
tribution using q$_c$ & q$_e$ from the present model
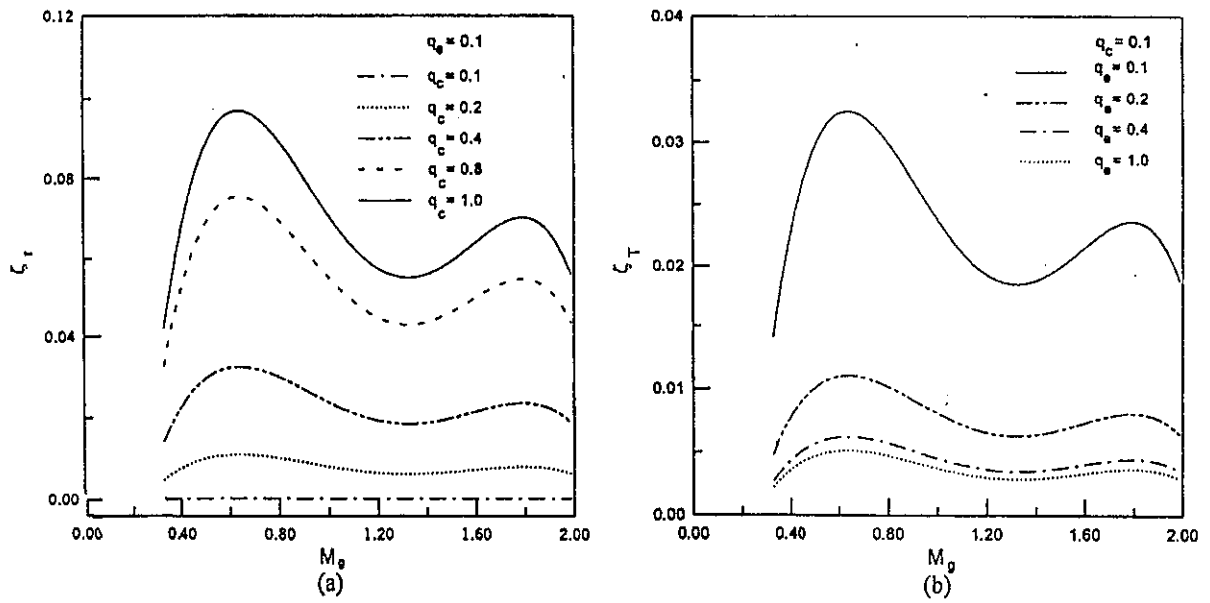and the measured one through a nozzle.

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Figure 7: Variation of the wetness loss coefficient with a nozzle Mach number for various values of : a – condensation coefficient, and b- evaporation coefficient.
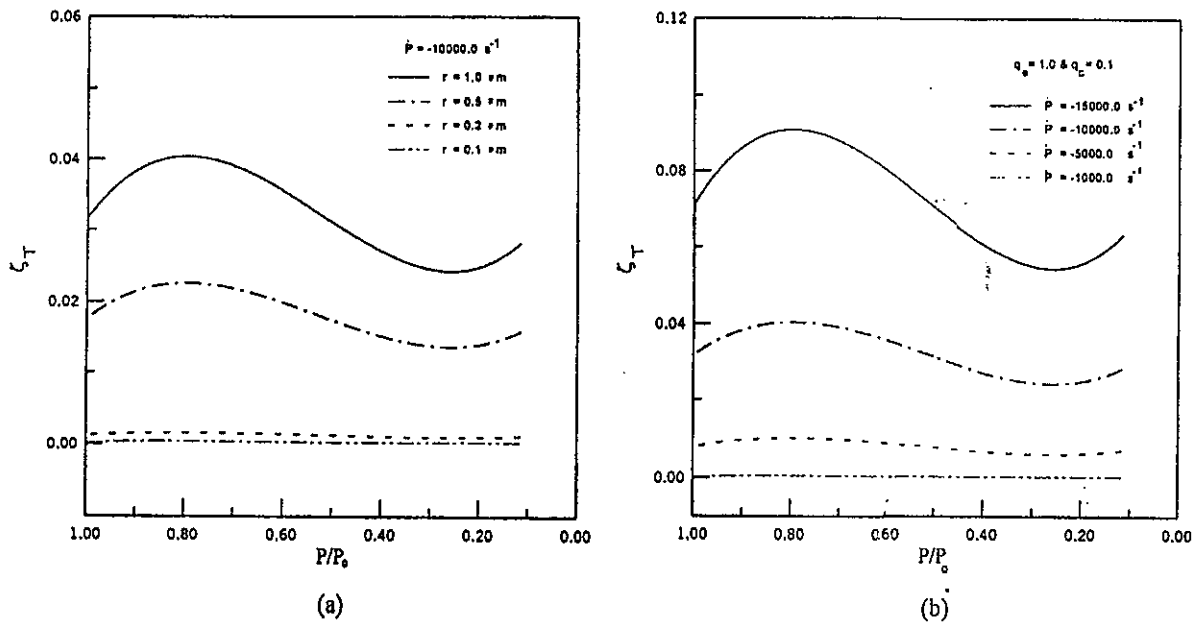

Figure 8: Variation of the wetness loss coefficient with a nozzle pressure ratio for different values of : a- droplet size, and b- expansion rate.

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معامل التكثف والمفاقيد الحرارية للرطوبة لانسياب البخار الرطب عند ضغوط منخفضه

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ملخص البحث:

تم فـى هذا البحث تقييم معامل التكثف لانسياب البخـار الرطب عند الضغوط المنخفضة نظريا ، ولقد أظهرت النتـائج النظريـة أن معـامل تكثف البخـار الرطب يعتمد على مقدار الضغط عند نقطـة ويلسون – مقدار فرط تبريد البخـار – حجم قطرات الماء – معامل التبخر وأس التمدد الايزنتروبى .

أظهرت البيانـات المتاحـة عن معامل التكثف لإنسياب البخـار الرطب أن هـذا المعامل يتوقف على قيم الضغط لكل مـن حالـة الإنسياب الإبتدائية وعند نقطـة ويلسون ، تم عقد مقارنـة بين القيم الحالية لمعامل التكثف والأخرى المنشورة والمطلوبه لتقليل الفارق بين التنبؤات النظرية والقياسات المعملية لتوزيعات الضغط فى بخار رطب ينساب خلال الأبواق ، ويعد التوافق فى هذه المقارنة جيدا ، ولتـأكيد النموذج النظرى لهذا البحث ٠٠ فإن مقارنة أخرى قـد أجريت بين قياس معملى لتوزيع الضغط فى بخار رطب ينساب خلال بوق وبين تنبؤ نظرى له مؤسس على قيم مختارة لمعاملى التكثف والتبخر ، ولقد جاء التوافـق فـى هذا المقارنـة جيدا أيضا .

أكثر من ذلك ٠٠٠ فقد إهتم البحث بإستنباط زيادة الإنتروبى الناجمـة عـن لا إرجاعية إجراءات إنتقال الحرارة والكتلة عدديا ، وأظهرت النتائج النظرية أن زيادة كل من معامل التكثف – حجم القطرات ومعدل التمـدد بالإضافة الى خفض معامل التبخر يفضى الى زيادة المفاقيد الحرارية للرطوبة أثناء إنسياب البخار الرطب .