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Modeling of Saturated Two-Phase Flow Expansion across an Abrupt Area Change

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Abstract- In this study, the pressure recovery of two-phase saturated steam-water mixture is determined across sudden expansion fittings with the vapor phase being compressible. The pressure recovery is determined by employing the conservative equations of mass, momentum and energy with entropy generation based on two-phase separated flow model. The pressure and steam quality at the inlet of the abrupt flow area are in a range of $0.15-8.5$ *MPa* and quality of $0.5-1$. respectively. Also, the ratio of the inlet to outlet flow area is in the range of $0.1-1$. During an adiabatic expansion process, the profile of the entropy does not increase monotonically, but it reaches a maximum value at the inlet quality of $x_1 = 0.3$, before it goes to zero and the process becomes an isentropic process at the specified inlet pressure of $p_1 = 0.2$ *MPa*. At this steam quality the conversion rate of the dynamic pressure head into static change becomes a minimum. The trend of the curves for pressure recovery against two phase mass velocity is proportional to the inverse of the inlet pressure so that the higher the inlet pressure the lower increase in pressure recovery is obtained. The results also show a good agreement with the experimental data in the literature.

Keywords- *Two-Phase Floe,* *Sudden Expan, Heat Transfer, Numerical Method*

I. INTRODUCTION

Two-phase pressure drops caused by abrupt flow area changes have many applications in various industries, such as power generation, refrigeration, distillation, pneumatic system and food processing. The inevitable use of constructional fittings leads to the increase in minor pressure losses. Roui and Dash [1] numerically investigated two-phase flow pressure drops through 2 thin and thick orifices with air–water flows in horizontal pipes. Two-phase computational fluid dynamics calculations, using the Eulerian–Eulerian model have been employed to calculate the pressure drop through orifices. The operating conditions cover the gas and liquid superficial velocity ranges 0..3 ‒ 4 m/s and 0.6 ‒ 2 m/s, respectively. The local pressure drops have been obtained by means of extrapolation from the computed upstream and downstream linearized pressure profiles to the orifice section. The expansion and contraction loss coefficients are found to be different for single-phase flow of air and water. Ozmen-Cagatay and Kocaman [2] studied the dam-break flow over dry channel with an abrupt contracting part in certain downstream section. A new experiment was carried out in a smooth-prismatic channel with rectangular cross section and horizontal bed. A digital imaging technique was adopted for flow measurement and thus flood wave propagation was sensitively obtained. Chakrabarti et al. [3] numerically simulated the performance of a sudden expansion with fence viewed as a diffuser using SIMPLE algorithm. The Reynolds number was in the range of 20–100 and fence subtended angle between 10 deg and 30 deg with an aspect ratio of 2. The results revealed that for higher Reynolds number, the use of a fence always increases the effectiveness of the diffusion process when compared with a simple sudden expansion configuration. In low Reynolds number regime, depending on the positioning of the fence and the fence subtended angle, the fence may increase or decrease the diffuser effectiveness in comparison with sudden expansion without fence. Ohtake et al. [4] experimentally and analytically obtained the frictional pressure drops of gas-liquid two-phase flow in mini-micro pipes and at vena contracta with water and argon at room temperature. The diameter of the test mini-pipe was 1.0 and 0.5 mm, respectively. Each test tube was connected at both ends to small tanks. The diameter of the small tank was 15 mm for 1.0 mm diameter of test tube and 5 mm for 0.5 mm diameter of test tube, respectively. Yan, et al. [5] simulated Nitrogen gas flow in two-dimensional micro channels with a sudden contraction and expansion by direct simulation Monte Carlo method in both slip and transition regimes. It was found that the resistance coefficient for micro channels differs from that for conventional-scale channels in two aspects. One the resistance coefficient for micro channels is much smaller and the divergence between micro channels and conventional-scale channels increases with the increase of Knudsen number. On the other hand, the resistance coefficient for micro channels is more sensitive to the expansion ratio of channels, especially when the expansion ratio is less than 3. Chen et al. [6] conducted some newly measured pressure drops for an abrupt expansion. They found that the existing correlations all fail to provide a reasonably predictive capability against the newly collected data. Furthermore, a unique flow pattern called "liquid jet-like flow pattern" occurs at a very low quality region of total mass flux of 100 kg/m^2.s, and it raises a setback phenomenon of pressure drop. By contrast, an appreciable increase of pressure difference is seen when the liquid jet-like flow pattern is completely gone. In addition, a similar conclusion is drawn for the data of contractions. For the correlations/predictive models, the homogeneous model gives satisfactory prediction for conventional macro-channels but fails to do so when the channels become smaller. Gundogdu et al. [7] analytically studied the static pressure recovery and the minor loss coefficient through an axis-symmetric, circular cross-section, sudden expansion fitting of a horizontal pneumatic conveying line with air–solid particle flow. They proposed a new original analytical slip flow model, which takes into account the slip velocity between gas and solid phases evaluated by coupling the well-known separated flow model with the empirical slip ratio predictions in the literature. Yang and Zhang [8] developed a non –equilibrium two-fluid model for refrigerant two-phase critical flow inside the short tube orifice. Comparisons were made between the results of the two-fluid model and the experimental data of refrigerants R134a, R12, R22, R410A and R407C flowing through short tubes. The predictions by the two-fluid model and by the homogeneous equilibrium model show that the two-fluid model gives acceptable predictions with the deviations of 20%, while the homogeneous equilibrium model underestimates the flow rate by 20% or so. Chen et al. [9] examined the two-phase flow pattern change and pressure drop at the sudden contraction from small rectangular channels (3-9 mm^2 and 3- 6 mm^2) into a 3 mm diameter tube. A unique deflection of contraction pressure drop vs. vapor quality is observed at a very low quality regime with an unusual "liquid like vena contracta". They compared the measured pressure drops with existing correlations' models, but none of them could accurately predict the available database. By proposing a correction factor accounting for the influence of surface tension (Bond number and contraction ratio) to the original homogeneous model, considerable improvement of the predictive ability of homogeneous model was arrived. Schmidt and Friedel [10] studied experimentally two-phase pressure drop across sudden contractions using mixtures of air and liquids. They developed a model to calculate the two-phase pressure drop considering the annular-dispersed flow pattern based on momentum and mass balances as well as on their experimental results. They indicated that the vena-contracta phenomenon did not occur in their system at all. On contrary, many of the published studies have assumed the occurrence of the vena-contracta phenomenon in analogy with single-phase flow and have assumed that dissipation occurs downstream of the vena-contracta point. Kondo et al. [11] experimentally investigated the multidimensional behavior of upward gas-liquid two-phase flow in a vertical pipe with an axisymmetric sudden expansion. The void fraction distributions were measured and then the cross-sectional averaged void fractions were calculated for various locations in the flow direction at the below and above of the sudden expansion for various flow conditions. The prediction of averaged void fraction in the flow direction was carried out using one-dimensional two-fluid model. However, they also revealed that the two-phase flow behavior even in the sudden expansion might be predicted to a certain extent using the one-dimensional two-fluid model.

Attou et al. [12] developed a semi-analytical model for two-phase pressure drop in sudden enlargements, based on the solution of one-dimensional conservation equations downstream of the enlargement. They compared the predictions of three models, such as homogeneous flow, frozen flow, and bubbly flow models, with experimental data and found that bubbly flow model provides the best agreement with data. Abdelali et al. [13] experimentally investigated pressure drops across sudden expansion and contraction in small circular channels, using air and water at room temperature and near-atmospheric pressure as the working fluids. The measured total two phase pressure changes indicated the occurrence of significant velocity slip. The assumption of a velocity slip ratio in accordance with minimum entropy production in annular flow regime led to a reasonable agreement between the data and a simple one-dimensional flow theory. Salcudean et al. [14] studied the effect of various flow obstructions on pressure drops in horizontal air– water flow and derived pressure loss coefficients and two-phase multipliers. Delhaye [15] studied a detailed review of possible procedures for pressure recovery calculations. He started with single-phase flows and then expanded his derivations to two-phase flow. The major differences among the previous models are based on three sources, the definition of the densities according to the two-phase model used; the simplifications introduced in the models such as incompressibility of the fluid and constant void fraction in the control volume in the case of the heterogeneous flow models; the calculation of the void fraction from primary flow parameters. In view of these drawbacks, a new model is derived based on the two-fluid model with the conservative equations of mass, momentum, energy and entropy generation in terms of the compressibility of the vapor phase.

II. THEORETICAL BACKGROUND

A. Pressure Recovery in Single Phase Flow

The pressure change of the single-phase flow across an abrupt flow area (Fig. 1a) from a simplified momentum and mechanical energy balance equations are respectively as:

$$P_2 - P_2 = G_1^2 \vartheta \sigma (1-\sigma) \tag{1}$$

$$P_2 - P_2 = \frac{1}{2} G_1^2 \vartheta (1-\sigma^2) \tag{2}$$

ϑ is the specific volume, σ is the ratio of the inlet to outlet flow area and G_1 is mass velocity related to the inlet area.

A typical change of static pressure along the axis for the single-phase flow across the abrupt flow area is illustrated in fig. 1(b). The static pressure of the first part decreases due to the wall shear forces then increases at the abrupt flow area due to the deceleration of the flow in the transitional region. After reaching the maximum, the pressure gradient merges with the downstream pressure gradient line in the second part of the developed region.

B. Pressure Recovery in Two-Phase Flow

At low Much number when the two-phase flow mixture expands and decelerates through an abrupt flow area, a part of the dynamic pressure head is recovered and a rise in static pressure is observed. The other part is dissipated, i.e. it converts from mechanical into thermal energy. For the first time, Romie [15] derived the pressure recovery for two-phase flow through an abrupt flow area. He used a simplified momentum balance based on heterogeneous model by neglecting wall shear and gravitational forces without any heat and mass transfer between two phases.

$$P_2 - P_2 = G_1^2 \sigma \left\{ \begin{aligned} &\left[\frac{x^2}{\alpha_1} \vartheta_g + \frac{(1-x)^2}{1-\alpha_1} \vartheta_f \right] - \\ &\sigma \left[\frac{x^2}{\alpha_2} \vartheta_g + \frac{(1-x)^2}{1-\alpha_2} \vartheta_f \right] \end{aligned} \right\} \tag{3}$$

Subscripts 1 and 2 are conditions at plane 1 and 2. x and α are the vapor quality and void fraction respectively. Also, subscripts g and f are vapor and liquid phases respectively. This model shows more information about the flow pattern and an additional equation is needed for the prediction of the mean void fraction from primary flow parameters. In this regard, the following void fraction correlation proposed by Rouhani [16] is applied.

$$\alpha = \frac{x\vartheta_b}{1 + 0.2(1-x)\ (\vartheta_f + x\vartheta_{fg}) + \dfrac{\overline{u_{gj}}}{G}} \tag{4}$$

where $\bar{u}_{gj}$ is weighted mean drift velocity and is caused by local relative motion between the phases recommended by Wallis [17] as:

$$\bar{u}_{gj} = 1.18 \left[\frac{\sigma\, g(\rho_f - \rho_g)}{\rho_f^2} \right]^{0.25} \tag{5}$$

σ is surface tension, g is gravity acceleration, ρ_f and ρ_g are the liquid and gas densities respectively. Another momentum balance equation with further simplification is attributed to Lottes [18]. He assumed that all loss of dynamic pressure head takes place in the liquid phase as:

$$P_2 - P_2 = \frac{\sigma(1-\sigma^2)\vartheta_f G_1^2}{(1-\alpha)^2} \tag{6}$$

The next formula suggested by Chisholm and Sutherland [19] is based on the Lockhart-Martinelli [20] parameter, X as:

$$P_2 - P_2 = G_1^2 \vartheta_f \sigma (1-\sigma)(1-x)^2 \left[1 + \frac{1}{X} + \frac{1}{X^2} \right] \tag{7}$$

$$X = \frac{1-x}{x} \sqrt{\frac{\vartheta_g}{\vartheta_f}},\ C = \left[1 + 0.5 \sqrt{\frac{\vartheta_{fg}}{\vartheta_g}} \right] \left[\sqrt{\frac{\vartheta_g}{\vartheta_f}} + \sqrt{\frac{\vartheta_f}{\vartheta_g}} \right] \tag{8}$$

Besides of the above momentum based models, Richardson [21] derived a correlation based on the mechanical energy balance equation by considering only the liquid velocity as:

$$P_2 - P_2 = \frac{\sigma(1-\sigma^2)(1-x)^2}{2(1-\alpha)} \vartheta_f G_1^2 \tag{9}$$

On the other hand, Wadle [22] proposed a new formula for the recovery pressure in a pipe where the contour of the expansion pipe follows a steep hyperbolic tangent. He carried out a theoretical and experimental study on the flow of two-phase steam–water and air–water mixtures. This correlation is not based on the momentum or the energy balance equation, but it is derived in terms of superficial velocities of the two phases. The author explains that two-phase flow phenomena caused by internal effects may be described in terms of the superficial velocity as Wallis [17] used in the flooding correlation or Mandhane et al. [22] used in the flow patterns.

$$P_2 - P_2 = \frac{k}{2}(1-\sigma^2)\left[x^2 \vartheta_g + (1-x)^2 \vartheta_f \right] G_1^2 \tag{10}$$

Wadle states that the factor k is to be experimentally adjusted and suggested $k = 2/3$.

III. PROBLEM STATEMENT AND METODOLOGY

To describe the behavior of saturated vapor-liquid two-phase flow during expansion in an abrupt flow area, we investigate an expansion process without any heat or mass transfer. In other words, the vapor quality remains constant during the process. Ignoring the dissipation and potential energies, the energy equation for two–phase vapor and liquid mixture between sections 0 and 2 is as follows, (Fig. 1).

$$d(h + 0.5u^2) = d\left[xh_g + (1-x)h_f + 0.5u^2 \right] = 0 \tag{11}$$

where h is enthalpy and u is average velocity of the mixture.

Differentiating the above equation as for $x =$ constant will be.

$$xdh_g + (1-x)dh_f + 0.5du^2 = 0 \tag{12}$$

The thermodynamics relation for a pure substance involving enthalpy [24] is.

$$dh = CdT + \left[\vartheta - T(\frac{\partial \theta}{\partial T})_P \right] dP \tag{13}$$

where, C_P , ϑ, T and P are the specific heat at constant pressure, specific volume, temperature and pressure respectively. Applying the above equation for both the vapor and liquid phases separately under saturation condition and then introducing their corresponding enthalpy into Eq. (12), we have

$$xC_{pg} + (1-x)C_{pf} \; dt_{sat} + 0.5du^2 + \left\{ \begin{array}{l} x\left[\vartheta_g - T_{sat}(\frac{\partial \vartheta_g}{\partial T_{sat}})_P\right]dP + \\ (1-x)\left[\vartheta_f - T_{sat}(\frac{\partial \vartheta_f}{\partial T_{sat}})_P\right] \end{array} \right\} dP = 0 \quad (14)$$

During changes from saturated liquid to saturated vapor, the slope of the vapor pressure as a function of temperature is expressed by the following the Clapeyron equation.

$$\left.\frac{\partial \theta}{\partial T}\right|_{sat} = \frac{h_{fg}}{\vartheta_{fg} T_{sat}} \quad (15)$$

Based on the mass flow rate, the average density of the two-phase mixture is obtained by:

$$u = \frac{\dot{m}}{\rho A} = \bar{\vartheta} G \quad (16)$$

$\bar{\vartheta} = x\vartheta_g + (1-x)\vartheta_f$. We will now derive a relation for the change of kinetic energy of the vapor-liquid mixture.

$$\frac{1}{2}du^2 = \frac{1}{2}\bar{\vartheta}^2 dG^2 + x\bar{\vartheta}G^2 \frac{\partial \vartheta_g}{\partial P} dP \quad (17)$$

The quantities dt_{sat} and du^2 are now replaced by Eqs. (15) and (17) into Eq. (14) and common terms are collected. The result will be.

$$\frac{dP}{dG} = \frac{\vartheta^2 G}{\bar{\vartheta}(1 + xG^2 \frac{\partial \vartheta_g}{\partial P} + \frac{\bar{C}_P T_{sat} \vartheta_{fg}}{h_{fg}} - \frac{h_{fg}}{\vartheta_{fg}} \frac{\partial \bar{\vartheta}}{\partial P}} \quad (18)$$

where, $\bar{C}_P = x(C_P)_g + (1-x)(C_P)_f$.

In addition, entropy of the vapor-liquid two-phase mixture is:

$$dS = d\bar{S} \quad (19)$$

where, $\bar{S} = xS_g + (1-x)S_f$.

However, the entropy relation for a pure substance as given by Sonntag et al. [24] is

$$dS = C_P \frac{\partial T}{T} - (\frac{\partial \vartheta}{\partial T})_P dP \quad (20)$$

Applying the above equation for both the vapor and liquid phases separately under saturation condition, then introducing their corresponding entropies into Eq. (19).

$$dS = \bar{C}_P \frac{dT_{sat}}{T_{sat}} - \left[x(\frac{\partial \vartheta_g}{\partial T_{sat}})_P + (1-x)(\frac{\partial \vartheta_f}{\partial T_{sat}})_P\right]dP \quad (21)$$

Inserting dT_{sat} from Eq. (15) into the above equation for constant x, we found

$$\frac{dS}{dP} = \frac{\bar{C}_P \vartheta_{fg}}{h_{fg}} - \frac{h_{fg}}{T_{sat}\vartheta_{fg}} \frac{\partial \bar{\vartheta}_P}{\partial P} \quad (22)$$

Combing Es. (18) and (22) yields

$$\frac{dS}{dG} = -\frac{\bar{\vartheta}^2 G \left[\frac{\bar{C}_P \vartheta_{fg}}{h_{fg}} - \frac{h_{fg}}{T_{sat}\vartheta_{fg}} \frac{\partial \bar{\vartheta}_P}{\partial P}\right]}{\bar{\vartheta}(1 + xG^2 \frac{\partial \vartheta_g}{\partial P}) + \frac{\bar{C}_P T_{sat} \vartheta_{fg}}{h_{fg}} - \frac{h_{fg}}{\vartheta_{fg}} \frac{\partial \bar{\vartheta}_P}{\partial P}} \quad (23)$$

ϑ_{fg} and h_{fg} represent the increase in specific volume and the increase in specific enthalpy when the state changes from saturated liquid to saturated vapor respectively. One of the major advantages of the Eqs. (18) and (23) is that they only depend on primary flow parameters as well as fluid properties at saturation pressure which are available from tables of thermodynamic properties of pure substances in a computer program or data form.

IV. SOLUTIONMETOD

The problem includes two unknown variables p and S which are functions of physical properties of the two-phase fluid. In the saturated region, pressure and temperature are dependent properties and, therefore, for each pressure all fluid properties are only functions of pressure given in thermodynamic tables. In addition, the effects of both phases are considered. Hence, the models should cover the whole void fraction region and should not be restricted to low-void flows, where neglecting the influence of the vapor phase may not be justified. To solve numerically Eqs. (18) and (23), we use the fourth order Runge-Kutta method [25] for p and S unknown variables under initial conditions p (x_1, G_1)= p_1 and S (x_1, G_1)= S_1 . For simplicity solution, it is possible the existing derivatives of $d\phi / dP$ for $\phi = \vartheta_f$, ϑ_g and $d\phi / dP$ in Eqs. (18) and (23) can be obtained by fitting the fluid properties data versus pressure from the thermodynamic table of the substance.

V. RESULTS AND DISCUSSION

To predict the expansion of vapor-liquid two-phase flow through an abrupt flow area, the saturated steam-water has been examined by the present model as a working fluid. The expansion process is assumed to be adiabatic without any heat and mass transfer at the interface between two phases. The range of pressure and that of the vapor quality at the inlet of the channel are $0.15\,MPa \le P_1 \le 8.5\,MPa$ and $0.05 \le x_1 \le 1$ respectively. Also, the ratio between the inlet to outlet surface area ranges $0.1 \le \sigma \le 1$. The problem analyzed is illustrated in Fig. 1.

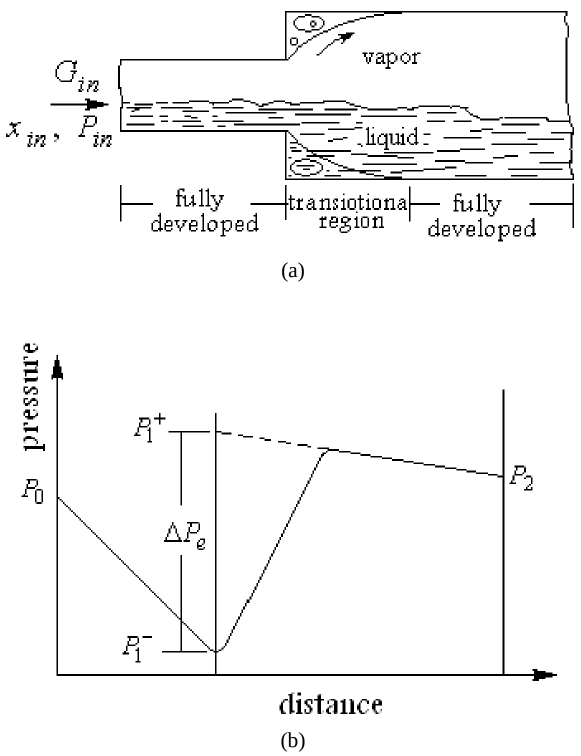

Figure 1. Two-phase mixture across an abrupt flow area

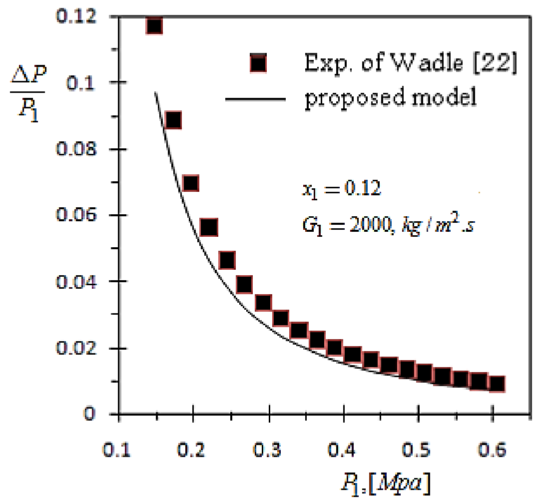

Figure 2. Comparison between the results of the proposed model and the experimental data of Wadle [22]

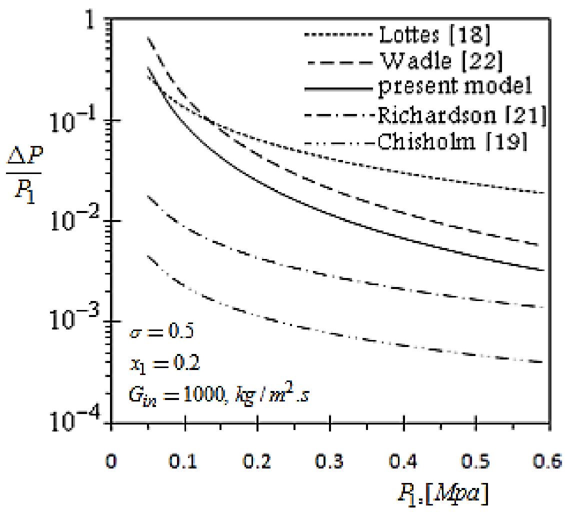

Figure 3. Comparison between present model and different theoretical models

The vapor phase may be either be the vapor of the liquid being decelerated, in which case the flow is termed "one-component" like the steam-water mixture in this study, or a different chemical species from liquid, in which case the flow is termed "two-component". To compare the results of this study with the experimental data of Wadle [22], a set of pressure recovery data versus inlet two-phase pressure was chosen for the mixture mass velocity of $G_1 = 2000\,kg/m^2.s$ and inlet steam quality $x_1 = 0.12$. The results, which were found to be in good agreement with the existing results, are presented in Fig. 2. The ordinate axis has been dimensionless by the inlet pressure.

Fig. 3 compares the results of the present model with the results of the different theoretical models. These results are calculated based on theoretical Eqs. (6), (7), (9) and (10) for the saturated steam-water mixture. The inlet vapor quality is $x_1 = 0.2$, the inlet mass velocity is $G_1 = 1000\,kg/m^2.s$ and the ratio of the inlet to outlet flow area is $\sigma = 0.5$. The results of Lottes [18] and Wadle [22] represent the upper bound whilst the results of Richardson [21] and Chisholm [19] represent the lower bound relative to the present model. These discrepancies among the theoretical results is probable due to the lack of information about the flow patterns and/or the void fraction model which usually holds only for fully developed straight pipe flows and no procedure is available for calculating the void in the abrupt flow area.

Fig. 4 depicts the entropy change of the steam-water two-phase mixture as a function of the steam quality for three different mass velocities. Each curve has been divided into two parts by a horizontal line corresponding to $\Delta S = 0$. We can see that during the expansion process, the profiles of the entropy do not increase monotonically, but they reach a maximum at the inlet quality of $x \approx 0.3$. before they go to zero and the process becomes an isentropic process. In other words, at *this quality* the entropy change becomes a maximum and the conversion rate of the dynamic pressure head into static change is strongly influenced by bulk dissipation, i.e., the transfer of the mechanical energy into thermal energy is a minimum. On the other hand, when the inlet quality is equal to $x \approx 0.7$ the two-phase mixture expands in an isentropic process because of $\Delta S = 0$. At this quality there is no any dissipation energy and the conversion rate of the dynamic pressure head into the static change becomes a maximum. When the inlet quality is more than $x > 0.7$, the entropy change becomes negative and the two-phase mixture is impossible to expand under this specified inlet conditions and the authentic parts of the curves come to an end at $x_1 \approx 0.7$.

Fig. 5 represents the pressure recovery as a function of the two-phase mass velocity of the mixture for three different inlet pressures. The inlet steam quality is $x = 0.25$. The trend of the curves is proportional to the inverse of the inlet pressure so that the higher the inlet pressure the lower increase in the pressure recovery is obtained. The reason is that, at low fluid working pressure, the specific volume of the steam is large and its contribution in the momentum change is more significant than that of the steam at higher working pressure.

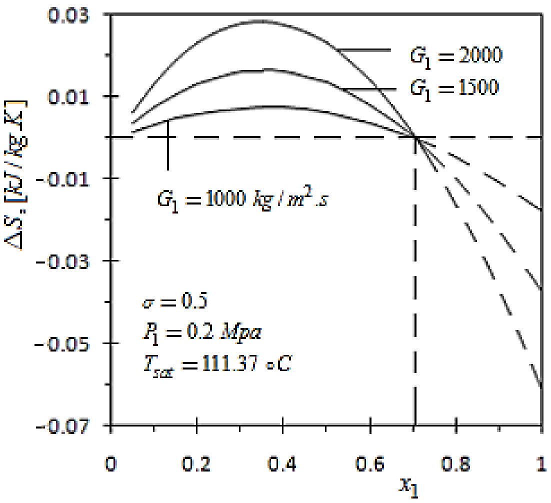

Figure 4. Entropy change between outlet and inlet vs. steam quality

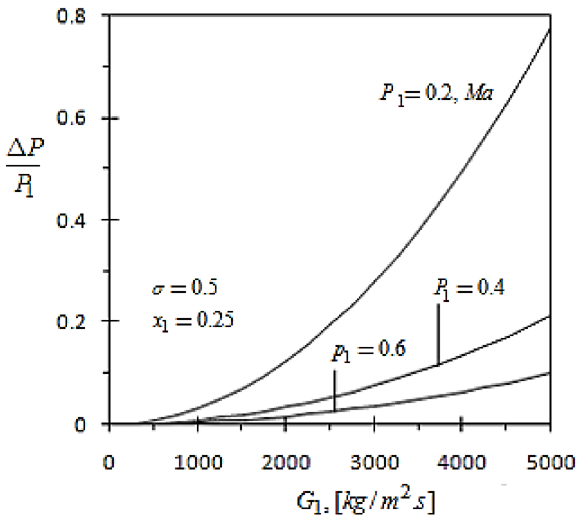

Figure 5. Pressure recovery vs. mass velocity of two-phase mixture

Fig. 6 indicates the pressure recovery against the inlet pressure for different mass velocities. The results show that the increasing percentage of the pressure recovery for a lower inlet pressure is higher than that of higher inlet pressure. Since P_1 is the denominator of $\Delta P / P_1$ and when P_1 increases the increasing percentage of $\Delta P / P_1$ decreases. In addition, this behavior is more significant at the beginning of the process especially for bigger mass velocities.

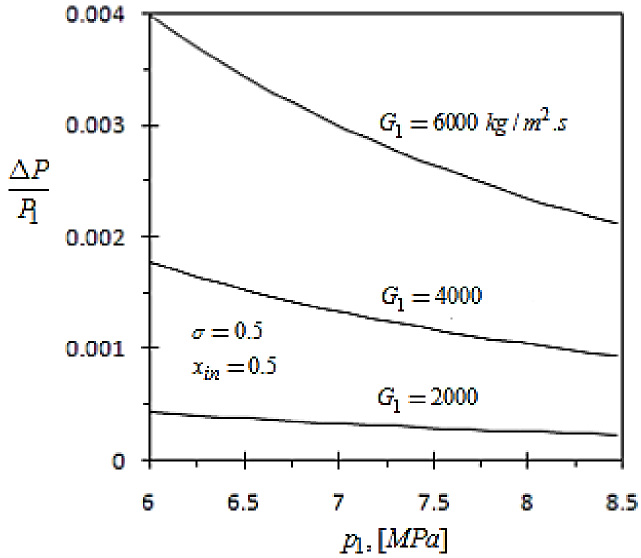

Figure 6. Pressure recovery vs. inlet pressure for different mass velocities of two-phase mixture

Fig. 7 depicts the pressure recovery against the ratio of the inlet to outlet flow area for three different two-phase mass velocities. The slope of $G_1 = 3000\,kg/m^2.s$ is very steeper in comparison with the others. The pressure recovery goes to zero as the ratio of the inlet to outlet cross section area approaches one. Because it was assumed that the steam quality remains constant during the expansion process as well as the wall friction was neglected (for a very short tube length). Thus it is expected that the pressure recovery is proportional to the momentum change between the inlet and outlet and in this case the pressure recovery would be zero.

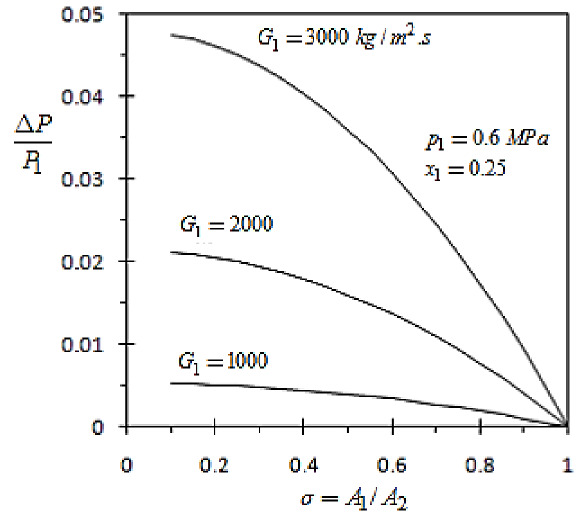

Figure 7. Pressure recovery vs. ratio of inlet to outlet cross section area

VI. CONCLUSIONS

To predict the expansion of vapor-liquid two-phase flow through an abrupt flow area, the saturated steam-water has been examined as a working fluid by the two-fluid model. The expansion process is assumed to be adiabatic. During the expansion process, the profile of the entropy does not increase monotonically, but it reaches a maximum at the inlet quality of $x = 0.3$ before it goes to zero and the process becomes an isentropic process. At the steam quality equal to $x = 0.3$, the entropy change is a maximum and the conversion rate of the dynamic pressure head into static change is strongly influenced by the bulk dissipation, i.e., the transfer of the mechanical energy into thermal energy is a minimum. The trend of the curves for pressure recovery against two-phase mass velocity is proportional to the inverse of the inlet pressure so that the higher the inlet pressure the lower increase in pressure recovery is obtained. This is because of the specific volume of the steam which is large and its contribution in the momentum change would be more significant than that of the steam at higher working pressure.

NOMENCLATURE

A	surface ares (m^2)
g	gravity acceleration (m/s^2)
G	mass velocity (kg/m^2.s)
h	enthalpy (kH/kg)
P	pressure (MPa)
S	entropy (kJ/kg.K)
T	temperature (K)
u	velocity (m/s)
u_{gj}	drift velocity (m/s)
x	vapor quality
X	Lockhart-Martinelli parameter

Greek Letters

α	void fraction
Δ	difference
ϑ	specific volume (m^3/kg)
ρ	density (kg/m^3)
σ	surface tension (N/m), inlet to outlet surface area

Subscript

1	inlet
2	outlet
f	fluid
fg	saturated vapor liquid difference
g	vapor
sat	saturation

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