Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp265-270)

Initiation of Water Hammer in a Steam/Water Pipe with a Non-Condensable Gas

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Abstract: The effect of a non-condensable gas on the initiation of water hammer in a condensing water-steam system has been investigated by extending Bjorge's work. Based on computational results, it is concluded that the pipe slope is the dominant parameter which affects the initiation of water hammer. The effective temperature difference in driving energy across the liquid interface was found to be $(T_i - T_l)^b$ instead of previously accepted expression of $(T_{sat} - T_l)^a$. As a result, the effect of the subcooling of inlet liquid in inducing water hammer is not always a favorable factor. The effects of the parameters including liquid flow rate, pipe length, and diameter are similar to Bjorge's conclusion. Nusselt number is used and correlated directly by mass transfer theory. More experiments are needed in order to find a more accurate correlation of the occurrence of water hammer.

Key Words: Water hammer, Condensation, Non-condensable gas, Daitel-Dukler parameter.

1 Introduction

Bjorge [1] developed a one- dimensional flow model to describe the initiation mechanism associated with water hammer induced by collapsing of steam bubbles in a pipe containing steam and subcooled water. He defined an "absolute stability limit" which is a conservative prediction of water hammer formation. This limit is postulated as the minimum liquid flow rate needed to satisfy the so-called Taitel-Dukler [2] criterion, assuming that the subcooled liquid entering a pipe is quickly heated up to saturation temperature by steam flow.

A mathematic model for predicting the occurrence of water hammer in a large circular piping system has been established. The parameters such as liquid depth, liquid temperature and steam mass flow rate, all along the pipe, are investigated. Taitel-Dukler's [2] criterion is employed to predict whether the occurrence of water hammer will initiate or not. A computer code was developed to solve the mathematicam model. In case the computer code does not yield a convergent solution for some cases, the Wallis' [3] full limit analysis can also be introduced to assess if water hammer occurs.

Jackobek and Griffith [4] followed the methodology developed by Bjorge and calculated the impact velocity and peak pressure rise during the course of the occurrence of water hammer by assuming: (a) zero pressure exists inside the steam bubbles upon collapse; (b) liquid plug velocities diminish upon impact; and (c) condensation is the dominant process in which the pressure is reduced. A threshold of the steam mass flow rate is needed to achieve a thermal equilibrium. It means that the energy transferred from latent heat into sensible heat is treated as an upper limit for energy mixing.

There are some researches related to the transition from stratified flow to slug flow which is the initiation state of water hammer. Kordyban et al [5] concluded that, for given channel geometries and fluid properties, the primary factors which would influence this transition are gas velocity, liquid level, wavelength, and wave height. Taitel-Dukler assumed that the pressure difference on the wavy surface is due to Bernoulli effect. They postulated that slug formation would occur when the pressure difference was large enough to overcome the gravity force acting on the wave. Mishima and Ishii [6] postulated that a stability criterion is obtained by introducing the wave deformation limit and the "most dangerous wave" concept in stability analysis.

Minkowycz and Sparrow [7,8] found that the presence of non-condensable gas can reduce the condensation rate based on their boundary layer experiment. This may be a practical way to prevent the initiation of water hammer. The non-condensable gas provides a void space, equivalent to a buffer, to reduce the impact velocity and simultaneously reduce the peak pressure.

2 Condensation Phenomena with the Presence of Non-condensable Gas for flat plate

The current study investigates the effect of the non-condensable gas in a stratified-flow condensation system by extending Bjorge's work. Base on previous researches, it is generally accepted, in a steam piping system, water hammer is initiated mainly by a rapid condensation rate. In light of this, it is considered feasible to inhibit the initiation of water hammer by adding a small fraction of non-condensable gas into the concerned system.

Now, consider stationary fluids covering an infinite flat plate with a constant temperature while vapor is condensing on the liquid/vapor interface.

Based on conservation of energy, heat transfer per unit width of the plate can be expressed as:

$$k_f \frac{\Delta T}{\delta} dx = d\dot{m}_v h_{fg} = \rho_l \bar{u} dx h_{fg}$$

In which, $\Delta T = T_{sat} - T_w$; $\bar{u} = d\delta/dt$.

Therefore, it yields

$$\delta d\delta = \frac{k_f \Delta T}{\rho_l h_{fg}} dt$$

Then, heat flux can be obtained by introducing depth δ_o with respect to time t_o, and it yields

$$q = \frac{k_f (T_{sat} - T_w)}{\delta} = \frac{k_f (T_{sat} - T_w)}{\left[\left(\frac{2k_f (T_{sat} - T_w)(t - t_o)}{\rho_l h_{fg}}\right) - \delta_o^2\right]^{1/2}} \tag{1}$$

By including mass transfer with the presence of a non-condensable gas, it can be obtained from [10],

$$\frac{q_m Lc}{\bar{\rho} h_{fg} D_m (m_{v\infty} - m_{vi})} = 0.425 \left[GrSc\left(\frac{1 + 0.4 Ja_l}{Ja_l}\right)\right]^{1/4} \tag{2}$$

where $Lc = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$

Substituting T_{sat} in Eq.(4) by T_i, it yields

$$q = \frac{k_f (T_i - T_w)}{\left[\left(\frac{2k_f (T_i - T_w)(t - t_o)}{\rho_l h_{fg}}\right) - \delta_o^2\right]^{1/2}} \tag{3}$$

Equation (3) can be substituted back to Eq. (2) and T_i can be solved by iteration. Finally, by taking the ratio of heat fluxes expressed in Eq. (3) and Eq. (2), the effect of the presence of a non-condensable gas on heat transfer can be evaluated.

Now, consider a convective flow above an infinite flat plate with a constant temperature while vapor is condensing on the liquid/vapor interface. Also by assuming a Prandtl number very close to unity, the momentum boundary layer is then identical to the thermal boundary layer. The boundary conditions of the boundary layer are:

$y = 0$, $T = T_w$ and $y = \delta$, $T = T_{sat}$.

It can be obtained that

$$T_{sat} - T = \left(T_{sat} - T_w\right)\left(1 - {y}/{\delta}\right) \qquad (4)$$

and the overall energy equation is

$$q = k_f \left.\frac{\partial T}{\partial y}\right|_{y=0} = \frac{d\dot{m}}{dx}\left[h_{fg} + \frac{1}{\dot{m}}\int_0^{\delta} \rho_f u Cp_f \left(T_{sat} - T\right) dy\right] \qquad (5)$$

where $\dot{m} = \int_0^{\delta} \rho_f u dy = \frac{2}{3}\rho_f u_{\max}\delta = \frac{\rho_f}{3\mu_f}\left(-\frac{\partial P}{\partial x}\right)\delta^3$

By assuming a parabolic velocity distribution and substituting Eq. (4) into Eq. (5), it yields

$$q = \frac{k_f\left(T_{sat} - T_w\right)}{\delta} = \frac{d\dot{m}}{dx} h_{fg}\left(1 + \frac{3}{8} Ja\right) \qquad (6)$$

where $$\frac{d\dot{m}}{dx} = \frac{\rho_f}{\mu_f}\left(-\frac{\partial P}{\partial x}\right)\delta^2 \frac{d\delta}{dx} \qquad (7)$$

Substituting Eq. (7) into Eq. (6), then it yields

$$\frac{k_f\left(T_{sat} - T_w\right)}{\delta} = \frac{\rho_f}{\mu_f}\left(-\frac{\partial P}{\partial x}\right)\delta^2 \frac{d\delta}{dx} h_{fg}\left(1 + \frac{3}{8} Ja\right) \qquad (8)$$

Equation (8) can be integrate to obtain δ with the initial condition $\delta = \delta_o$ at $x = 0$, and then heat flux will be

$$q = \frac{k_f\left(T_{sat} - T_w\right)}{\left[\int_0^x \frac{4k_f \mu_f \left(T_{sat} - T_w\right)}{\rho_f\left(-\frac{\partial P}{\partial x}\right) h_{fg}\left(1 + \frac{3}{8} Ja\right)} dx + \delta_o^4\right]^{1/4}} \qquad (9)$$

The Nusselt number on a flat plate with a parallel laminar flow is [10]:

$$Nu_x = 0.332 \operatorname{Re}_x^{1/2} \operatorname{Pr}^{0.33}, \quad \operatorname{Pr} \geq 0.5$$

Therefore, the mass transfer with the existence of a non-condensable gas becomes:

$$\frac{q_m x}{\bar{\rho} h_{fg} D_m \left(m_{v\infty} - m_{vi}\right)} = 0.332 \operatorname{Re}_x^{0.5} Sc^{0.33} \qquad (10)$$

With the similar processes as mentioned before, the heat transfer rate with the consideration of the mass transfer included becomes:

$$q_m = \frac{k_f\left(T_i - T_w\right)}{\left[\int_0^x \frac{4k_f \mu_f \left(T_i - T_w\right)}{\rho_f\left(-\frac{\partial P}{\partial x}\right) h_{fg}\left(1 + \frac{3}{8} Ja\right)} dx + \delta_o^4\right]^{1/4}} \qquad (11)$$

Equation (11) can be substituted into Eq. (10), and T_i can be obtained by iteration on Eq. (11). Note that $\left(-\partial P/\partial x\right)$ and Re_x are not determined yet and these terms will be discussed in the following section.

3 Condensation Phenomena with the Presence of Non-condensable Gas for Circular Pipe

In generally, thermal layer available for mass transfer at the vapor/liquid interface is very thin in comparison with the pipe diameter [7]. The heat flux in a circular pipe can be approximated by Eq. (11) as long as the ratio of the liquid depth to the pipe diameter is not too large. Also, δ_o in Eq. (11) can be set to zero since because heat transfer only occurs at the liquid-vapor interface. By replacing T_w to T_l, Eq. (11) is simplified to be:

$$q_m = \frac{k_f\left(T_i - T_l\right)}{\left[\int_0^x \frac{4k_f \mu_f \left(T_i - T_l\right)}{\rho_f\left(-\frac{\partial P}{\partial x}\right) h_{fg}} dx\right]^{1/4}} \qquad (12)$$

where thermodynamic properties are evaluated with respect to the liquid temperature. For mass transfer, the Nusselt number is

$$Nu_m = \frac{jD}{\bar{\rho} D_m \left(m_{v\infty} - m_{vi}\right)} = \frac{q_m D}{\bar{\rho} h_{fg} D_m \left(m_{v\infty} - m_{vi}\right)} \qquad (13)$$

where $\bar{\rho} = (\rho_i + \rho_\infty)/2$. Several studies regarding interfacial condensation heat transfer under a condition of stratified flow of steam and subcooled water have been listed in [1]. Here, Bankoff's correlations is adopted:

$$Nu = 0.236 C_1 \, \mathrm{Re}_v^{0.027} \, \mathrm{Re}_l^{0.49} \, \mathrm{Pr}_l^{0.42} \qquad \text{(14-a)}$$

for smooth interface, and

$$Nu = 1.17 \times 10^{-10} C_1 \, \mathrm{Re}_v^{2.1} \, \mathrm{Re}_l^{0.56} \, \mathrm{Pr}_l^{1.16} \qquad \text{(14-b)}$$

for rough interface. C_1 is 2.5 for circular pipe.

When considering mass transfer with the existence of a non-condensable gas, the Prandtl number Pr_l and Reynolds number of the vapor phase Re_v should be replaced by Schmidt number Sc and Reynolds number of the vapor-gas phase, respectively. Here, $Sc \cong \nu_v/D_m$ and $Re_{v\&g} = (\rho u D_h/\mu)_{v\&g}$ where $\rho_{v\&g} = \rho_v m_v + \rho_g m_g$,

$$\mu_{v\&g} = \frac{\lambda_v \mu_v}{\lambda_v \eta_{vv} + \lambda_g \eta_{vg}} + \frac{\lambda_g \mu_g}{\lambda_v \eta_{gv} + \lambda_g \eta_{gg}}$$

and

$$\eta_{jk} = \frac{\left[1 + \left(\frac{\mu_j}{\mu_k}\right)^{1/2} \left(\frac{M_k}{M_j}\right)^{1/4}\right]^2}{\sqrt{8}\left[1 + \left(\frac{M_j}{M_k}\right)\right]^{1/2}} \qquad (15)$$

The modified Bankoff's correlations for condensation with the presence of a non-condensable gas can therefore be expressed as a function of the form

$$Nu_m = f\left(\mathrm{Re}_l, \mathrm{Re}_{v\&g}, Sc\right) = 0.59 \, \mathrm{Re}_l^{0.49} \, \mathrm{Re}_{v\&g}^{0.027} \, Sc^{0.42} \qquad \text{(16-a)}$$

for a smooth interface, and

$$= 2.925 \times 10^{-10} \, \mathrm{Re}_l^{0.56} \, \mathrm{Re}_{v\&g}^{2.1} \, Sc^{1.16} \qquad \text{(16-b)}$$

for a rough interface. Interfacial temperature can be obtained by combining Eqs. (12), (13) and (16) and undergoing iterative processes.

The pressure-gradient term in Eq. (12) was derived by Bjorge [1] based on the momentum balance of the liquid phase and by the assumption of a negligible interfacial velocity.

$$-\frac{dP}{dx} = \frac{1}{A_l}\left(\tau_l s_l + \tau_i s_i\right) + \rho_l g \theta + \rho_l g \frac{d\delta_l}{dx} + \frac{2\dot{m}_l}{\rho_l A_l}\frac{d\dot{m}_l}{dx} + \frac{\dot{m}_l s_i}{\rho_l A_l}\frac{d\delta_l}{dx} \qquad (17)$$

where $\frac{d\dot{m}_l}{dx} = \frac{q_m}{h_{fg}}$;

$$\frac{d\delta_l}{dx} = \frac{-\tau_l^* - \tau_i^* - \tau_{v\&g}^* - \theta + 2Fr^2 q^* (\psi - 1)}{1 - Fr^2 (1 + \Phi)},$$

$$\psi = \frac{1-\alpha}{\alpha}\frac{u_{v\&g}}{u_l}, \quad \Phi = \frac{1-\alpha}{\alpha}\frac{\left(\rho u^2\right)_{v\&g}}{\rho_l u_l^2},$$

$q^* = \frac{q_m}{\dot{m}_l h_{fg}}$ and $Fr^2 = \frac{\dot{m}_l^2 s_i}{\rho_l^2 g A_l^3}$.

The liquid temperature varies along the pipe distance by solving the following equation,

$$\frac{dT_l}{dx} = \frac{1}{\dot{m}_l C p_l}\frac{dq_m}{dx} \qquad (18)$$

4 Results and Discussion

As was predicted, the presence of a small quantity of a non-condensable gas in the condensing vapor significantly increases thermal resistance at the liquid-vapor interface. As shown in Fig. 1, Nusselt number is in general reduced by a factor of 10^5 when argon gas with a small amount of just 0.5% in pressure fraction is introduced. This is due to a dramatically increased mass transfer barrier imposed on condensation. Figure 2 shows the corresponding curves for Taitel-Dukler parameter in the prediction of the initiation of water hammer. It is apparent that the occurrence of water hammer in significantly inhibited when a non-condensable gas is present.

The effect of liquid subcooling on the initiation of water hammer is shown in Fig. 3, and it shows that a higher liquid subcooling would yield lower values of Taitel-Dukler parameter. This result is contrary to the case where a non-condensable gas is absent. This is because that heat transfer is proportional to $\left(T_i - T_l\right)^b$ instead of $\left(T_{sat} - T_l\right)^a$ when a non-condensable gas is present. The interfacial temperature T_i is always a little higher than liquid temperature. Thus, the effect of subcooling is negligible compared to the effect of the variation in thermophysical properties as determined by the system conditions. A liquid with a higher temperature will have a lower viscosity and hence a higher Reynolds number, and, according to Eqs. (12), (13), and (16), a higher Nusselt number and a higher Taitel-Dukler parameter.

The effect of pipe slope on Taitel-Dukler parameter is shown in Fig. 4, and it shows that even a small angle of pipe slope will remarkably increase the possibility of the initiation of water hammer. Similar to Bjorge's conclusion, increasing liquid flow rate and pipe length will increase the possibility of the initiation of water hammer. However, the effects are not very obvious.

5 Conclusion and Suggestions

The presence of a non-condensable gas in a condensing water-steam system with the occurrence of stratified flow has been investigated by extending Bjorge's work. A mathematic model for this condensation phenomena occurred in a large circular piping system has been established and its solution scheme has been programmed.

Based on computational results, it is concluded that the pipe slope is the dominant parameter which affects the initiation of water hammer. The effective temperature difference in driving energy across the liquid interface was found to be $\left(T_i - T_l\right)^b$ instead of $\left(T_{sat} - T_l\right)^a$. As a result, the effect of the subcooling of the inlet liquid in inducing water hammer is not always a favorable factor. The effects of liquid flow rate, pipe length, and diameter are similar to Bjorge's conclusion. Nusselt number is used and correlated directly by mass transfer theory. More experiments are needed in order to find a more accurate correlation of the occurrence of water hammer.

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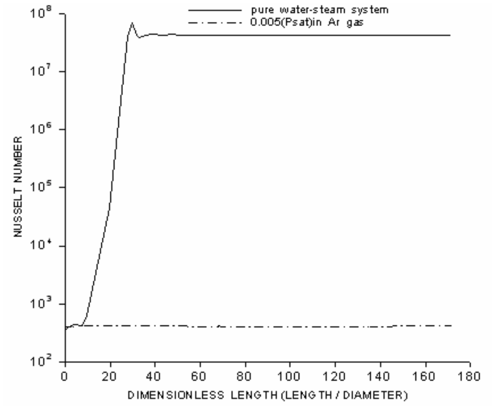

Fig. 1. Nusselt number distribution along the pipe with and without the effect of a non-condensable gas. L=60m, D=0.35m, θ=0°, (Tsat)in=570K, (Psat)in=8.2 Mpa, (Tl)in=373K, (Wl)in=70.6 kg/s

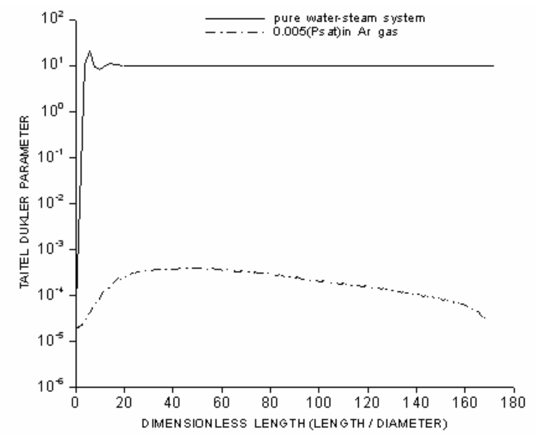

Fig. 2. Initiation of water hammer with and without the effect of a non-condensable gas. L=60m, D=0.35m, θ=0°, (Tsat)in=570K , (Psat)in=8.2 MPa (Tl)in=373K, (Wl)in=70.6 kg/s.

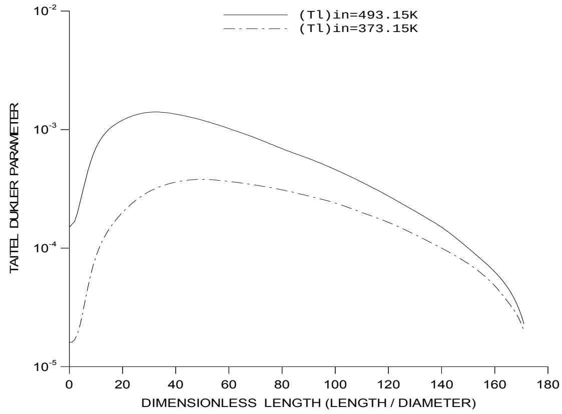

Fig. 3. Effect of inlet liquid subcooling on the nitiation of water hammer. L=60m, D=0.35m, θ =0°, (Tsat)in= 570K, (Psat)in= 8.2 Mpa, (Wl)in= 70.6 kg/s , He gas fraction =0.005(Psat)in.

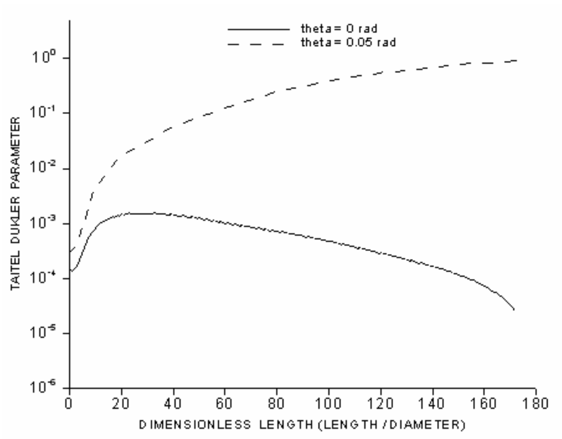

Fig. 4. Effect of pipe slope on the initiation of water hammer. L=60m, D=0.35m, (Tsat)in =570K, (Psat)in =8.2 Mpa, (Tl)in =493K, (Wl)in =70.6 kg/s.