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MODELING OF COMPONENTS IN ABSORPTION REFRIGERATION SYSTEMS DURING TRANSIENT OPERATION

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Abstract

In this study, two models have been developed to simulate the dynamic behavior of an orifice in an absorption chiller. They are developed based on homogeneous as well as separated flow models. Several two-phase flow models have been deemed to anticipate the performance of the orifice in the absorption chiller. The dynamic behavior of the orifice in the transient stage rigorously depends on some flow variables such as vapor quality, void fraction, pressure loss and two-phase mass flow rate. Results show that the homogeneous model predicts more pressure drops than that the separated model during transient throttling process of working fluid.

KEY WORDS: absorption chiller, expansion device; two-phase flow; refrigerant; pressure drop

INTRODUCTION

Because of time-consuming of the absorption chiller during start up operation, the study of the dynamic simulation will be important. The overall performance of the absorption chillers are related to the dynamic behavior of their expansion devices during start up operation. Short and long-tube orifices are commonly used because of easy manufacturing, reliability and cost effectiveness. Despite of simplicity in shape, orifice shows some complicated behavior during the expansions process.

Kim and O'Neal [1] derived a semi-empirical model to find out the mass flow rate through a short-tube orifice as a function of tube geometry, inlet conditions and downstream pressure. Zhang and Yang [2] developed a non-equilibrium two-fluid model (TFM) for refrigerant two-phase critical flow inside a short-tube orifice. They took into account both inter-phase velocity and temperature differences in their model. They ultimately demonstrated the applicability of TFM to refrigerant two-phase flow in a short tube. This verifies the existence of thermodynamic and hydraulic non-equilibrium characteristics of two-phase flow in a short tube as well as non-equilibrium effects on mass flow rate.

Kim et al. [3] by assuming a critical flow through an orifice developed a two-phase flow model to simulate dynamic behavior of a small-scale absorption chiller. Some previous works have focused on dynamic simulation of absorption chillers without using any special model [4, 5].

This paper highlights the importance of the throttling simulation of a short-tube orifice between the condenser and evaporator sections during the transient process by using both homogeneous as well as separated flow models. The results of the short-tube orifice throttling by using different flow models are also compared.

MATHEMATICAL MODELING

After boiling off LiBr-H_2O solution in the generator of an absorption chiller, the water vapor is condensed by a water-cooled heat exchanger in the condenser. Subsequently, the mixture of liquid-vapor as a refrigerant substance flows through the orifice and being throttled between condenser and evaporator sections from high to low pressure respectively.

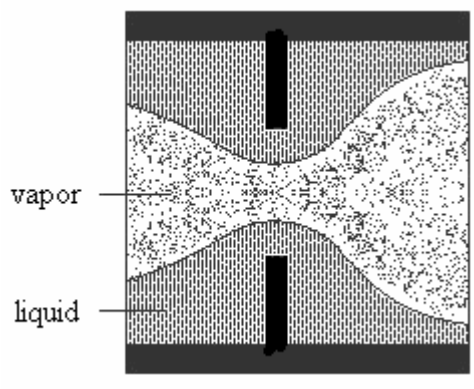

Fig.1. Evaporation of refrigerant in an annular flow from the left to right direction during expansion through an orifice

In an ideal situation, the pressures of the generator and condenser sections are assumed to be identical and are determined by their corresponding saturation temperature. Temperature and concentration of the LiBr-H_2O solution are determined by numerical solution of mass and energy equations. NTU approach is also used for heat exchanger and mass flow rate in the condenser.

Vapor Quality

The vapor quality in the evaporator as a refrigerant substance is first obtained by the adiabatic process when the two-phase flow expands between the condenser and evaporator as:

$$x_e = \frac{1}{(h_{fg})_e}(h_c - h_{fc} - \frac{u_e^2}{2}) \tag{1}$$

where the velocity is readily obtained by mass continuity as:

$$u_e = G\,(\upsilon_{fe} + x_e \upsilon_{fg}) \tag{2}$$

x_e is first guessed by the isenthalpic process as the two-phase flow expands between the same pressures as:

$$x_e = \frac{h_{fc} - h_{fe}}{(f_{fg})_e} \tag{3}$$

There are traditionally two models to determine pressure drop in a two-phase flow: (i) The homogeneous flow model, (ii) the separated flow model. These models are presented below.

Pressure Drop

The two-phase flow pressure drop through a vertical orifice up to down can be obtained by a homogeneous model, [6]. In this model, the two-phase mixture is simulated as a single phase fluid flowing with the mean fluid properties and thermodynamic equilibrium between two phases.

$$\Delta P = \frac{2 f_{TP} L G^2 \upsilon_f}{D}(1 + \frac{x}{2}\frac{\upsilon_{fg}}{\upsilon_f}) + G^2 \upsilon_{fg} x + \frac{gL \sin\theta}{x\,\upsilon_{fg}}(1 + x\,\frac{\upsilon_{fg}}{\upsilon_f}) \tag{4}$$

where the two-phase Fanning friction factor is estimated as below.

$$f_{TP} = \frac{0.079}{(\frac{GD}{\mu_{TP}})^{0.25}} \tag{5a}$$

Where the two-phase viscosity is determined by:

$$\mu_{TP} = \left[\frac{1-x}{\mu_f} + \frac{x}{\mu_V}\right]^{-1} \tag{5b}$$

In this equation the physical property of the fluid is calculated based on the mean pressure between downstream and upstream sections.

In the transient period, it is also practice to assume that the two phases flow annularly through the orifice. A similar model has been investigated by Kim et al. [1]. Another model for the two-phase pressure drop is suggested by Chisholm [7] based on the liquid flow which is only flowing alone inside the orifice as:

$$\Delta P_{TP} = (1 + \frac{C}{X} + \frac{1}{X^2})\Delta P_f \tag{6}$$

or based on the vapor flow which is only flowing alone as:

$$\Delta P_{TP} = (1 + CX + X^2)\Delta P_V \tag{7}$$

X is named the Lockart-Martinelli parameter and defined as:

$$X^2 = \frac{(-\frac{dP}{dz}F)_f}{(-\frac{dP}{dz}F)_V} = (\frac{1-x}{x})^{1.75}(\frac{\mu_f}{\mu_V})^{0.25}(\frac{\upsilon_f}{\upsilon_V}) \tag{8}$$

and C is a constant and yields.

$$C = k(\frac{\upsilon_f}{\upsilon_V})^{0.5} + \frac{1}{k}(\frac{\upsilon_V}{\upsilon_f})^{0.5} \tag{9}$$

where $k = (\upsilon_f / \upsilon_V)^{0.5}$ if $X \geq 1$ and $k = (\upsilon_V / \upsilon_f)^{0.5}$ if $X \leq 1$.

The pressure drop of the liquid and vapor are respectively determined by:

$$\Delta P_f = \frac{f_f L}{2D} G^2 (1-x)^2 \upsilon_f \tag{10a}$$

$$\Delta P_V = \frac{f_V L}{2D} G^2 x^2 \upsilon_V \tag{10b}$$

Assuming no shear force between phases, Murdock [8] has suggested the following correlation, based on the separated model, for the pressure drop as:

$$\Delta P_{TP} = (1 + 1.26X)^2 \Delta P_V \tag{11}$$

Another correlation extracted from the continuity equation for one-phase flow has been suggested for two-phase flow as below, [8]:

$$\sqrt{\Delta P_{TP}} = 1.26\sqrt{\Delta P_f} + \sqrt{\Delta P_V} \tag{12}$$

In all above equations, the quality of refrigerant is to be determined in upstream.
The required void fraction is suggested by Zivi [9] as:

$$\alpha = \left[1 + \frac{1-x}{x} \times \left(\frac{\upsilon_f}{\upsilon_V}\right)^{\frac{2}{3}}\right]^{-1} \tag{13}$$

And for an annular flow, it can be obtained by the following correlation.

$$\phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2} = \frac{1}{(1-\alpha)^2} \tag{14}$$

SOLUTION METHOD

In the procedure developed for the two-phase throttling model, the condenser pressure as well as the flow rate is known. On the other hand, the pressure drop and the vapor quality before and after the throttling are unknown. After determining the quality before expansion, downstream pressure is assumed and from which vapor quality is evaluated at this location. Then, pressure drop across the orifice will be determined using one of the aforementioned models. The procedure is repeated until the guessed and calculated pressure drop are the same or the difference between them is within acceptable range. In this procedure, the pressure drop accuracy is 0.1 Kpa.

RESULTS AND DISCUSSION

In a commercial absorption chiller with 150 ton refrigeration, the pipe nominal diameter carrying refrigerant from the condenser to the evaporator is assumed to be $D = 2.5$ inch. The condenser pressure is varied in the range of $1.5 < P_C < 7.5$ Kpa, [10].

Fig. 2 shows the pressure drop of the refrigerant across the orifice against the inlet vapor quality for different two-phase mass flow rates by Chisholm model. It can be inferred that by elapsing time during the start-up, the pressure drop will be increased due to more refrigerant being condensed in the condenser. Therefore, more liquid and fewer vapors pass through the expansion device. In all figures the inlet vapor quality is usually high, because the condensation rate of the vapor coming from the generator is low and more vapor remains in the condenser at the early stage of the operation. For $\dot{m} > 0.04\,kg/s$ there is a critical quality below that the pressure drop increases with quality and above that the opposite action occurs. Moreover, the slope of the curve is very steep and limited to a very small range of quality.

Fig. 3 again depicts the two-phase mixture pressure drop across the orifice versus inlet vapor quality for different mixture mass flow rate based on the homogeneous model. Similar to the previous figure, at each mass flow rate, ΔP drops against the inlet vapor quality but with moderate slope. In addition, the figure exhibits less sensitivity relative to the inlet vapor quality at low mass rate.

Fig. 4 compares the pressure drop against inlet vapor quality for different models when $\dot{m} = .04\ kg/s$. It is seen that the homogeneous model predicts more pressure drop in comparison with the other ones but they have nearly a similar slope. It is clear that the first part of the curve has a higher slope than the second part. This enhancement of the slope may be attributed to the two-phase flow effect.

Fig. 5 illustrates the two-phase flow pressure drop against the two-phase mixture mass flow rate. Each P_C curve approaches toward the one corresponding to $P_C = 7$ Kpa as the mass flow rate of the

mixture becomes small. The results show that the pressure drop across the orifice is suddenly rises after a certain value of the mixture mass flow rate.

Fig. 6 presents the pressure drop against mass flow for different condenser pressure in homogeneous model. The results show that all the curves have a similar trend and the pressure loss is nearly constant and equal to about $\Delta P = 1.5\ \text{Kpa}$ when $.015 < m < .055\ \text{kg/s}$ and all P_c curves have the same asymptotes.

Fig. 7 compares the pressure drop against the mass flow rate of the different models at $P_c = 7\,\text{Kpa}$. . The discrepancy between the homogeneous model results and the other ones becomes larger as the mass flow rate increases. In other words, this difference is related to the physical properties effects of the mixture which have different effects in the homogeneous as well as the separated models.

CONCLUSION

The dynamic behavior of an expansion device between the condenser and evaporator has been studied by a homogeneous as well as a separated model. Several two-phase pressure drop models have been developed to simulate the dynamic behavior of the refrigerant flow through the orifice in the transient process. They predict the pressure and quality of the refrigerant after the expansion device. The three separated two-phase models have almost the same results whereas the homogeneous results are different and with higher values relative to the three separated models. The results show that at any given condenser pressure, there exists a mass flow of the two-phase mixture that the slope of the pressure drop rises sharply.

NOMENCLATURE

C	constant
D	diameter
f	friction factor
G	mass velocity
h	enthalpy
K	constant
L	length
P	pressure
u	velocity
x	quality
X	Lockart-Martinelli parameter

Subscripts

c	condenser
e	evaporator
f	liquid
fc	condenser liquid saturation
fe	Evaporator liquid saturation
fg	latent heat
f	friction
in	inlet
TP	Two-phase
v	vapor

Greek Letters

α	void fraction
Δ	difference
ϕ^2	two-phase pressure ratio
θ	vertical angle
υ	specific volume

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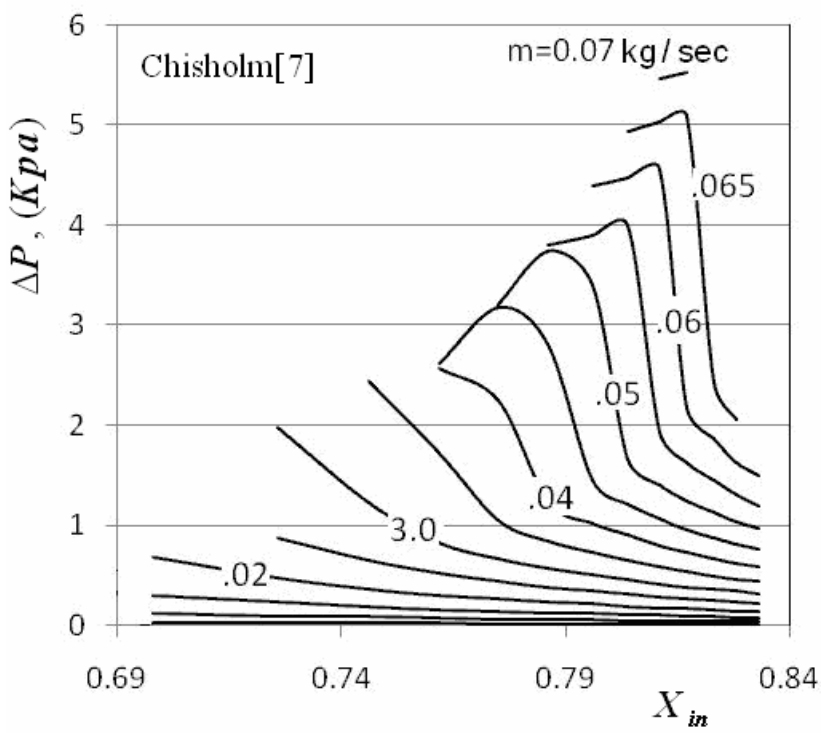

Fig. 2 Two-phase flow pressure drop against inlet vapor quality based on Chisholm model for different two-phase mass flow rates

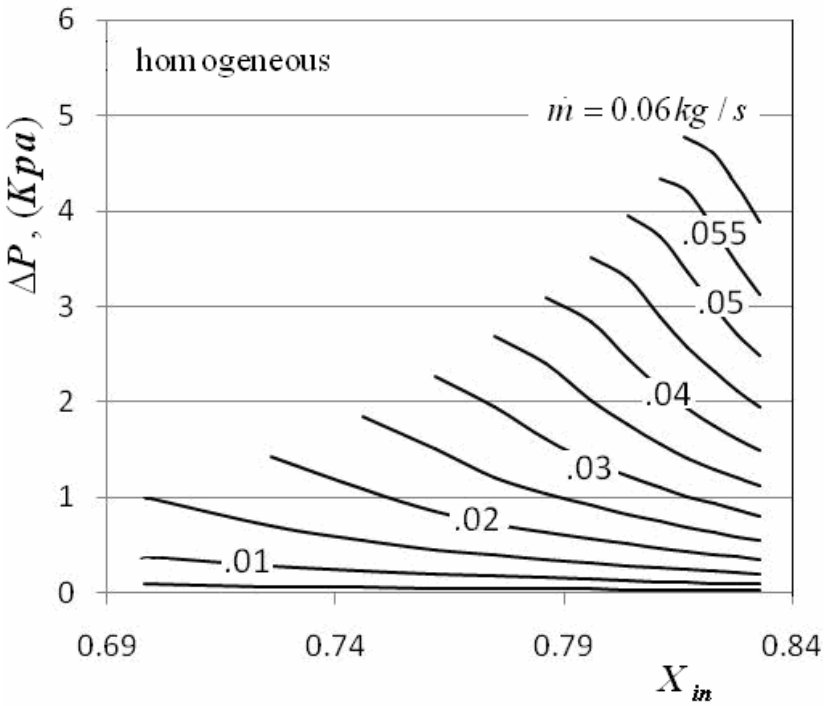

Fig. 3 Two-phase flow pressure drop against inlet vapor quality based on homogeneous model for different two-phase mass flow rates

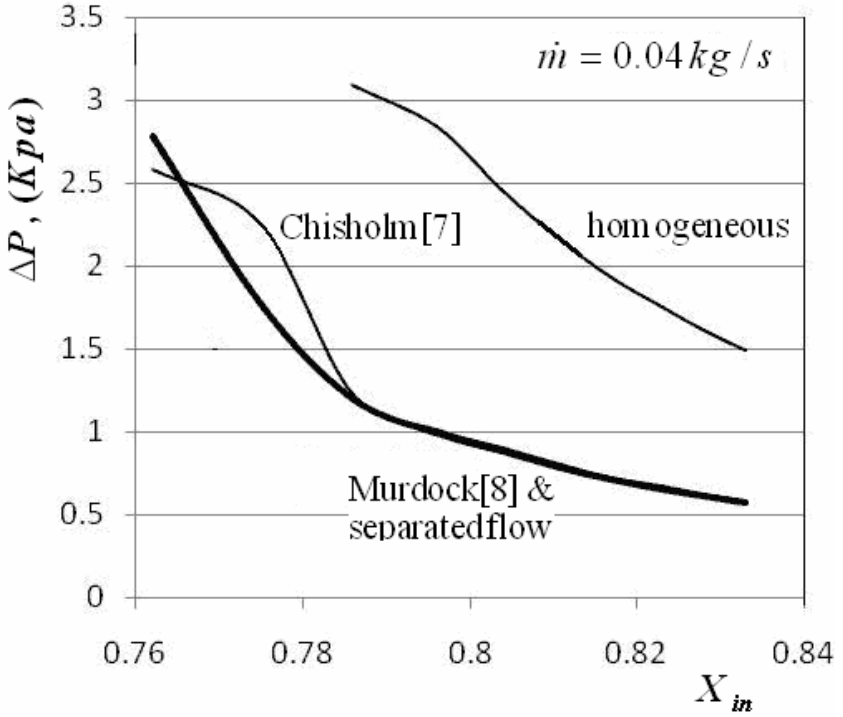

Fig. 4 Pressure drop versus inlet quality for different models at $\dot{m} = 0.04 kg/s$

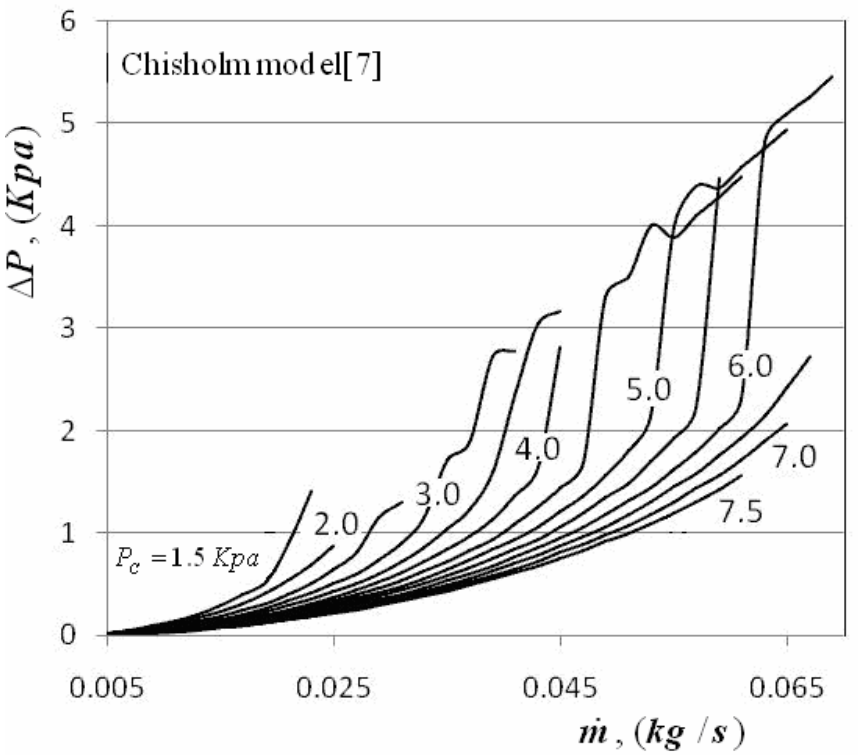

Fig. 5 Two-phase flow pressure drop against to-phase mass flow rate based on Chisholm model for different condenser pressures

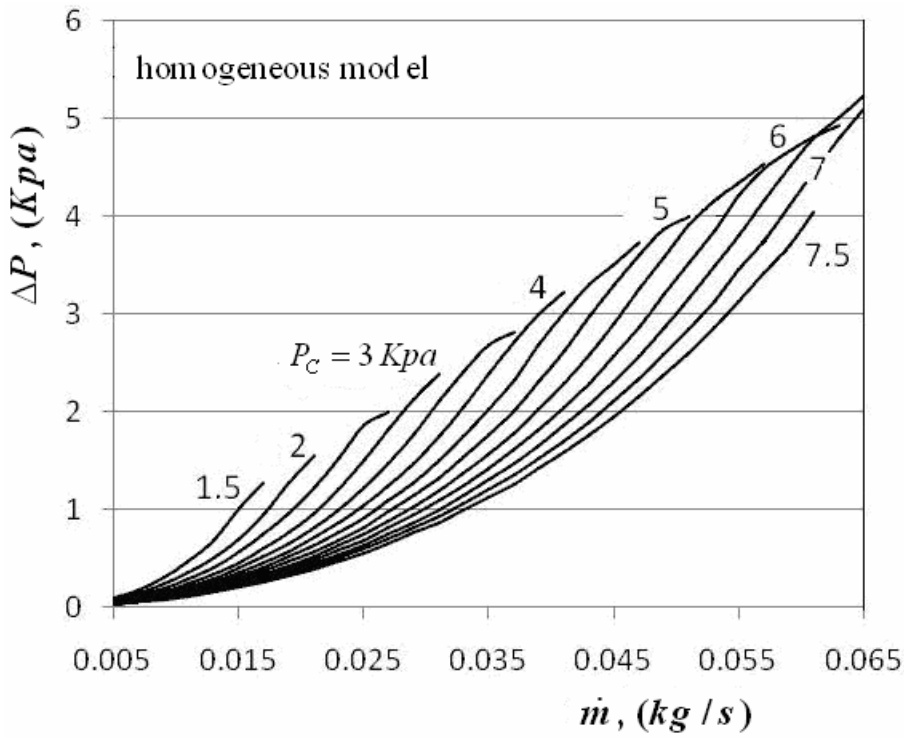

Fig. 6 Two-phase pressure drop against two-phase mass flow rate based on homogeneous model for different condenser pressures

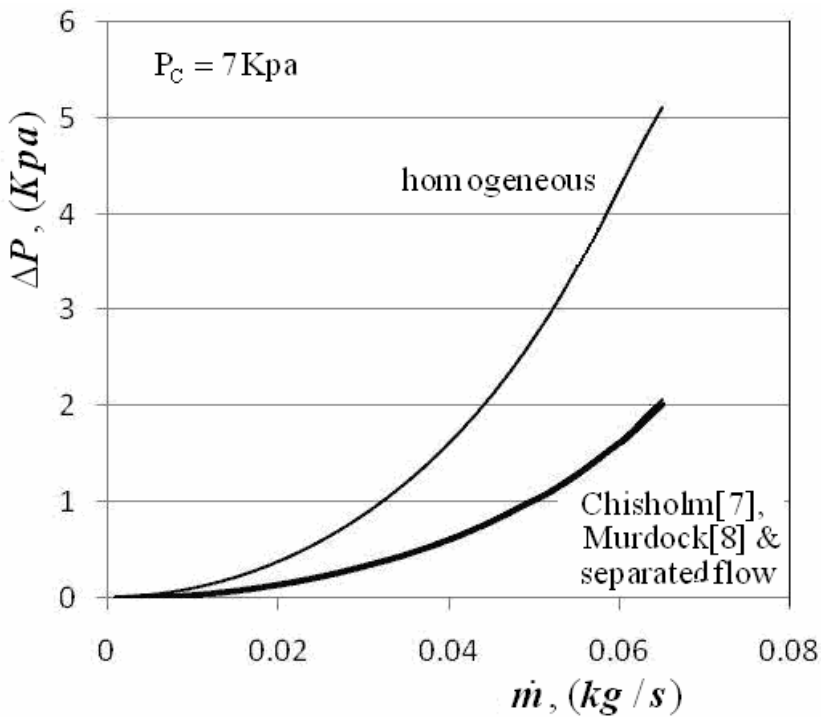

Fig. **7** Pressure drop versus mixture mass flow rate for different Models at condenser pressure $P_C = 7$ Kpa