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Two-phase flow model of a horizontal symmetrical impacting T-junction

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Abstract. A symmetrical distribution of the two-phase refrigerant mixture before the evaporator is crucial to achieve optimal performance of heat pumps. In a previous study experiments on two-phase flow in a horizontal symmetric impacting T-junction were conducted in our lab. The measurements performed include both pressure drop and phase distribution data of refrigerants. This data is unique as almost all previous experiments in literature investigate air-water flows. With this data a mechanistic model was constructed which is capable of predicting the phase distributions and pressure drop depending on the flow regime and the fluid properties. The model is capable of predicting 90% of the data with a maximum mean deviation of 5%.

1. Introduction
Two-phase flows are a common occurrence in a wide variety of applications, ranging from boiling water reactors to domestic heat pumps. In many of these applications the flow is split up over multiple tubes. An example of this is the distributor head before the evaporator in a heat pump. However, it is not known how the two phases split up over these channels. Currently no methods exists to determine this for the complex structure of a real distributor head. Therefore, this geometry is often simplified in literature to a T-junction, which is the simplest example of a distributor head, only having 2 outlet tubes. This geometry has been researched before and multiple experiments on horizontal impacting T-junctions were made [1]. For horizontal impacting T-junctions the models and modelling techniques that exist are described next.

The first type of modeling technique is the dividing streamline concept, which constructs fictional barriers within the flow. The fluid to one side of this fictional barrier will flow to the outlet branch on the same side. Both Azzopardi et al. [2] and Hwang et al. [3] applied this methodology to construct a model. Another method, applied by Ottens et al. [4], is the double stream model, the basis of which is a macroscopic steady-state mechanical energy balance for each phase and for each outlet separately. Pure empirical models have also been constructed [5]. One of these empirical models stands out, namely the one developed by Chen et al. [6], as it incorporates fluid properties by comparing different mixtures of water and nitrogen. Lastly the model of El-Shaboury et al. [7] is very comprehensive, it incorporates the mass, momentum and energy balances for the phases.

The models mentioned a have as main drawback that they were constructed with air-water flows. They generally don’t incorporate fluid properties and often only describe one inlet flow regime. The presented model aims to fill this gap by using a dataset [8] consisting of different refrigerants and flow regimes.
2. Modelling
The model technique applied here is based on the previous model of El-Shaboury et al. [7], but with a different pressure drop model and an extra droplet entrainment term. The fitting parameters are applied in a wider range of inlet flow regimes and fitted with refrigerant data. The control volume on which the conservational laws will be applied is represented in Figure 1.

![Figure 1](image)

**Figure 1.** Control volume of the phase distribution model.

The control volume is extended into each branch, allowing to assume fully developed flow at the control volume boundaries. In the following analysis, steady-state operation is assumed. Data was averaged over a long enough time frame, during experiments, even though some flow regimes are momentarily unsteady such as slug flow.

2.1. Conservation of mass
The conservation of mass, split in phases, is expressed in equation (1).

\[-\dot{m}_{l,1} - \dot{m}_{g,1} + \dot{m}_{l,2} + \dot{m}_{g,2} + \dot{m}_{l,3} + \dot{m}_{g,3} = 0\]  

(1)

In which the subscripts 1, 2 and 3 stand for the inlet, right outlet and left outlet respectively as shown in Figure 1. The subscripts l and g stand for the liquid and vapour (gas) phase respectively.

2.2. Conservation of momentum
The forces acting on the control volume are the pressure force, the gravitational force, the friction force and the surface tension force. The surface tension forces are negligible compared to the other forces as the experiments were carried out with macro-scale tubes [9]. Furthermore, the control volume size is small, which leads to negligibly small friction forces as well. Equation (2) then represents the simplified conservation of momentum in the x-direction as defined in Figure 1.

\[\dot{m}_{l,2}v_{l,2} + \dot{m}_{g,2}v_{g,2} - \dot{m}_{l,3}v_{l,3} - \dot{m}_{g,3}v_{g,3} = (P_3 - P_2)A_c\]  

(2)

In which v is the velocity of the corresponding phase and branch, P is the pressure and $A_c$ the cross-sectional area of the tube.

2.3. Conservation of energy
The general expression for the conservation of energy can be reduced to equation (3).

\[\dot{m}_{l,1}\left(h_{l,1} + \frac{v_{l,1}^2}{2}\right) - \dot{m}_{l,2}\left(h_{l,2} + \frac{v_{l,2}^2}{2}\right) - \dot{m}_{l,3}\left(h_{l,3} + \frac{v_{l,3}^2}{2}\right) + \]  

\[\dot{m}_{g,1}\left(h_{g,1} + \frac{v_{g,1}^2}{2}\right) - \dot{m}_{g,2}\left(h_{g,2} + \frac{v_{g,2}^2}{2}\right) - \dot{m}_{g,3}\left(h_{g,3} + \frac{v_{g,3}^2}{2}\right) = 0\]  

(3)
As the model is for horizontal impacting T-junctions the height of the inlet and outlets is the same, which cancel out from the general equation due to the conservation of mass. As the friction force is negligibly small, the viscous work is also neglected. The heat transferred to the environment is also omitted because the T-junction is assumed to be perfectly insulated. Lastly there is also no shaft work within the control volume as only a simple T-junction is considered.

2.4. Pressure difference correlation
Equation (2) contains the pressure difference between both outlet branches, which needs to be modelled. This is achieved by considering two streamlines, from the inlet to each outlet. Applying the Bernoulli principle on these streamlines results in equation (4) which represents the pressure drop from outlet $i$ with respect to the inlet [10].

$$\Delta P_{i1} = \left( \frac{\dot{\rho}_i \frac{\dot{v}_1^2}{2} - \frac{\dot{\rho}_1 \dot{v}_1^2}{2}}{2} \right) + K_{i1} \dot{\rho}_1 \frac{\dot{v}_1^2}{2}$$

$$K_{i1} = a + b \frac{\dot{m}_i}{\dot{m}_1}$$

Wherein the subscript $i$ represents either outlet 2 or 3, $\dot{\rho}$ is the homogeneous density, $\dot{v}$ is the homogeneous velocity and $K$ is a correction factor which is a function of the mass fraction that flows through outlet $i$. The last term in equation (4) represents the irreversible term. The constants $a$ and $b$ in equation (5) are fitted to the experimental data depending on the inlet flow regime.

The pressure difference between both outlets can be written as the difference of the pressure drop over both streamlines. As an effect of droplet entrainment was noticed, a correction term was added to the final equation. This is represented in equation (6) by the last term which includes the Weber number. The constants were fitted using the experimental data and the resulting values are listed in table 1.

$$P_3 - P_2 = \left( \frac{\dot{\rho}_2 \dot{v}_2^2}{2} - \frac{\dot{\rho}_3 \dot{v}_3^2}{2} \right) + a_1 \left( \frac{\dot{m}_2 - \dot{m}_3}{\dot{m}_1} \right) \frac{\dot{v}_1^2}{2} + a_2 (v_{g,2} - v_{g,3}) \dot{m}_{l,1} W_{e1}$$

| Inlet flow regime         | Annular | Intermittent | Slug flow | Stratified-wavy | Slug+Stratified-wavy |
|---------------------------|---------|--------------|-----------|-----------------|----------------------|
| $a_1$                     | 2.70    | 2.62         | 2.72      | 2.84            | 2.66                 |
| $a_2$                     | -0.003  | -0.009       | 0.03      | 0.10            | 0.05                 |

3. Model operation
The model predicts both outlet conditions (i.e flow regime, vapour quality, …) by applying an iterative loop. The required information are the inlet mass flow rate, one outlet mass flow rate, inlet vapour quality, inlet saturation pressure, tube dimensions and the inlet flow regime. By taking a guess for the outlet vapour qualities, all other variables can be calculated. The secondary outlet mass flow rate follows from equation (1). Based on the calculated outflow flow regimes the void fractions in each tube can be calculated using the drift-flux model of Rouhani-Axelsson [11]. The saturation pressures in the outlet tubes follow from equation (4), enabling the calculations of the enthalpies. In a final calculation, it is checked whether the conservation laws close. If this is not the case, the outlet vapour quality guesses are updated and the process is repeated. Figure 2 compares the numerical model to the experimental campaign. The left Figure is the pressure difference between the exit branches according to equation (6) and the right Figure is the liquid mass fraction that leaves the junction through the right branch.
4. Conclusion

Not many models in literature are able to calculate the phase distribution over junctions with varying refrigerants. These models are also only valid for a specific inlet flow regime. The presented model was developed using refrigerant experimental data, incorporating both the inlet flow regime and the fluid properties. The model predicts 90% of the experimental data with a maximum mean deviation of 5%. For further work, this model can be adapted to Y-junctions or more complex structures with more outlet tubes. The capability of describing phase maldistribution results in more optimal design considerations in parts or machines downstream of the junction to in turn increase overall design efficiency.

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