A GENERAL PREDICTION MODEL OF TWO-PHASE FLOW DISTRIBUTION IN A MULTIPASS EVAPORATOR

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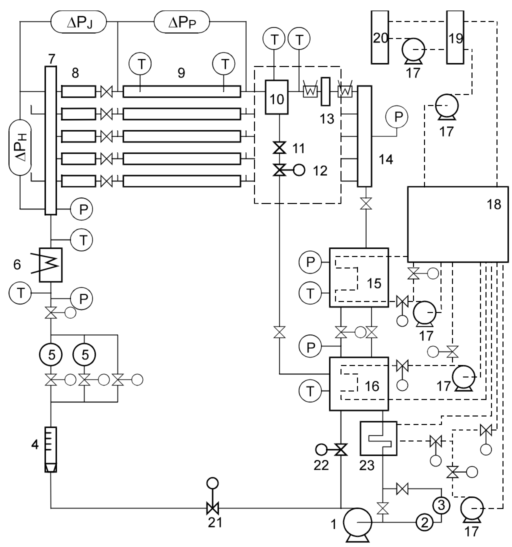
ABSTRACT

A simulation model that enables to predict two-phase flow distribution in a multipass evaporator is proposed. The model considers the multipass evaporator with a header as the combination of simple elements, i.e. straight tubes and T-junctions, and utilizes the correlations to predict the pressure drop at the elements. As a correlation for the phase separation characteristics at a T-junction, the empirical equations for liquid division ratio derived from our previous experimental data is used. By this model, gas phase flow distribution to each pass is determined as it makes the pressure at the outlet of each pass equal. Calculation results well predict the previous experimental data that were obtained under the condition of practical quality range at the inlet of an evaporator. The suitableness of this model suggests that the complexity of the two-phase flow distribution in multipass tube attributes to the phase separation phenomena in dividing two-phase flow.

INTRODUCTION

In order to enhance the heat transfer area and reduce the pressure drop simultaneously, so called multipass evaporator, that has two or more passes for evaporating refrigerant flow has come to be widely used. If a header were successfully used as a flow distribution device for multipass evaporator, it would bring higher energy efficiency and smaller size to the refrigeration equipments. So terrible maldistribution, however, generally occurs that the header is not used as flow distribution device except for some kind of automobile air-conditioners. In order to solve the maldistribution, we have to optimize the geometric conditions (i.e., length, diameter, orientation of each tube, shape of junction, etc.) and the operational conditions (i.e., circulating flow rate of refrigerant, quality at the inlet of evaporator, heating load on each pass, etc.), but there are so much parameters that the flow distribution characteristics becomes very complex.

Aiming to clarify the mechanism on which two-phase flow distribution is determined, authors have been experimentally studied the two-phase flow distribution in header-type multipass evaporator (e.g., Watanabe *et al*., 1995). On the basis of the results obtained in previous researches, the simulation model for predicting the two-phase flow distribution is proposed in this article.

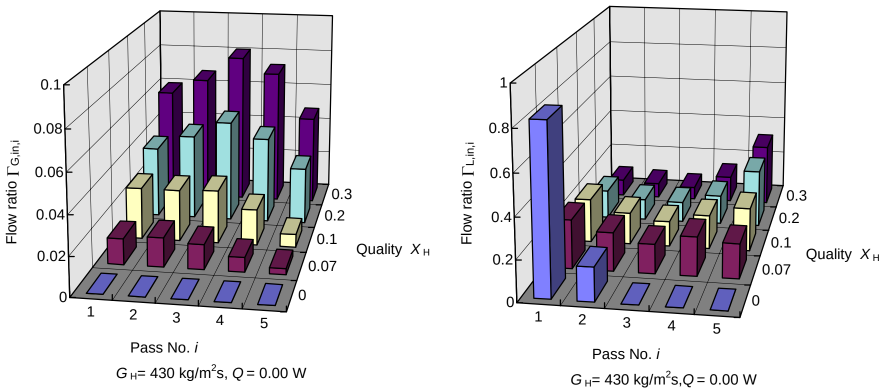


1. Refrigerant pump
2. Filter
3. Dryer
4. Flow meter(rotameter)
5. Flow meter(gear type)
6. Pre-evaporator
7. Distribution header
8. Glass tube
9. Evaporator
10. Separator
11. Ball cock
12. Needle valve
13. Orifice
14. Manifold
15. Condenser
16. Reservoir for R123
17. Water pump
18. Reservoir for water
19. Chilling unit
20. Cooling tower
21. Main valve
22. Bypass valve
23. Subcooling water jacket

Fig. 1 Experimental apparatus

1 SUMMARY OF THE EXPERIMENT

Since this article focuses on a simulation model, a brief review about the experiment on which the model is based (Wantanabe *et al.*, 1996ab) should be made here. Fig. 1 shows a schematic diagram of the experimental apparatus. Test fluid HCFC-123 was circulated by a pump (1). Its flow rate was controlled by main valve (21) and bypass valve (22), and was measured by gear-type flow meters (5). At a pre- evaporator (6), refrigerant flow was electrically heated to be controlled its quality before entering the test section. The test section was consisted of a vertical main pipe and five horizontal passes; all of them were 6 mm in i.d. Five passes were identified by pass No. 1 ~ 5 in order from the inlet side of main pipe. Electric heaters were applied to each pass to simulate various patterns of heating load profiles that would be occurred in the actual multipass evaporators.



(a) Gas flow ratio (b) Liquid flow ratio

Fig. 2 Examples of previous experimental data

$$\Phi_{L,i} = 2.79\frac{We_{l,i}^{0.203}}{Re_{G,l,i}^{0.288}}\cdot\left(1+\frac{q_{i+1}-q_i}{q_{i+1}+q_i+0.0125\cdot G_{L,l,i}\cdot u_{L,l,i}^2}\right)^{1.807}\cdot B \tag{1}$$

$$\text{Here,}\quad B=\begin{cases}1 & (\text{for } i=1,2,3)\\ 1.64+7.09\times10^{-4}\cdot\left(\dfrac{\rho_{L,l,5}\cdot g\cdot L_{Pitch}}{0.5\cdot\rho_{L,l,4}\cdot u_{L,l,4}^2}-18.3\right)^2 & (\text{for } i=4)\end{cases}$$

At the outlet of each pass, gas-liquid separator (10) was installed to measure the flow rate of each phase independently. By

Table 1 Experimental conditions

Header inlet mass flux G_H kg/(m^{2}s)	430
Header inlet quality X_H	(0) 0.07 0.1 0.2 0.3
Total heat input Q W	0 300 450 600 900
Step heating pattern	F D1 D2 U1 U2

Table 2 Heat flux on each pass for each condition

		Test code	Heat flux kW/m^2					Total heat input W	Hypothetic air flow rate m^3/hour
			Pass 1	Pass 2	Pass 3	Pass 4	Pass 5		
Flat load		0F	0					0	
		300F	3.98					300	
		450F	5.97					450	
		600F	7.96					600	
		900F	11.94					900	
Step load	Type-D	300D1	1.59	2.39	3.45	5.04	7.43	300	300
		450D1	2.39	3.51	5.17	7.63	11.14	450	300
		600D1	3.25	4.71	6.90	10.15	14.85	600	300
		300D2	0.93	1.66	2.92	5.17	9.22	300	200
		450D2	1.39	2.52	4.38	7.76	13.86	450	200
	Type-U	300U1	7.43	5.04	3.45	2.39	1.59	300	300
		300U2	9.22	5.17	2.92	1.66	0.93	300	200

closing the valve mounted under the separator, liquid flew into the separator was accumulated to be measured its flow rate by timed efflux. Separated gas flow, on the other hand, was taken to an orifice (13). After the flow measuring section, gas phase flow was gathered at a manifold (14) and returned to a reservoir (16) after getting liquidized at a condenser (15) while liquid phase flow gathered was directly returned to the reservoir (16). Notice that the group of devices that are bordered by broken lines is installed to every pass. Test conditions are shown in Table 1. “Step load” means the condition under which uneven heating load are applied to each pass. Amounts of heat input applied to each pass in various experimental conditions are shown in Table 2.

Fig. 2 shows experimental results obtained under the condition of no heating load on every pass. Gas flow ratio $\Gamma_{G,in,i}$ was defined as gas mass flow rate at the inlet of each pass divided by circulating mass flow rate; liquid flow ratio $\Gamma_{L,in,i}$ was defined by a same manner. $\Gamma_{G,in,i}$ and $\Gamma_{Lin,i}$ are shown in Fig. 2 (a) and 2 (b), respectively, as the function of pass No. i and of header inlet quality X_H. From the Fig.2, it is clear that the profiles of flow distribution are quite different for gas phase and for liquid phase. After the detailed discussion, we finally obtained the following two conclusions: ① The gas phase flow distribution was closely connected with the pressure drop along each pass. ② Contrary to this, liquid flow was affected by the local flow state in the vicinity of each junction. On the basis of latter conclusion, liquid division ratio $\Phi_{L,i}$ (defined as the fraction of liquid mass flow rate taken off to the pass) was introduced and correlated with each phase flow rate at the main pipe just before the junction. Empirically obtained correlation is shown as Eq. (1), which will be substituted as an equation for presenting phase separation characteristics.

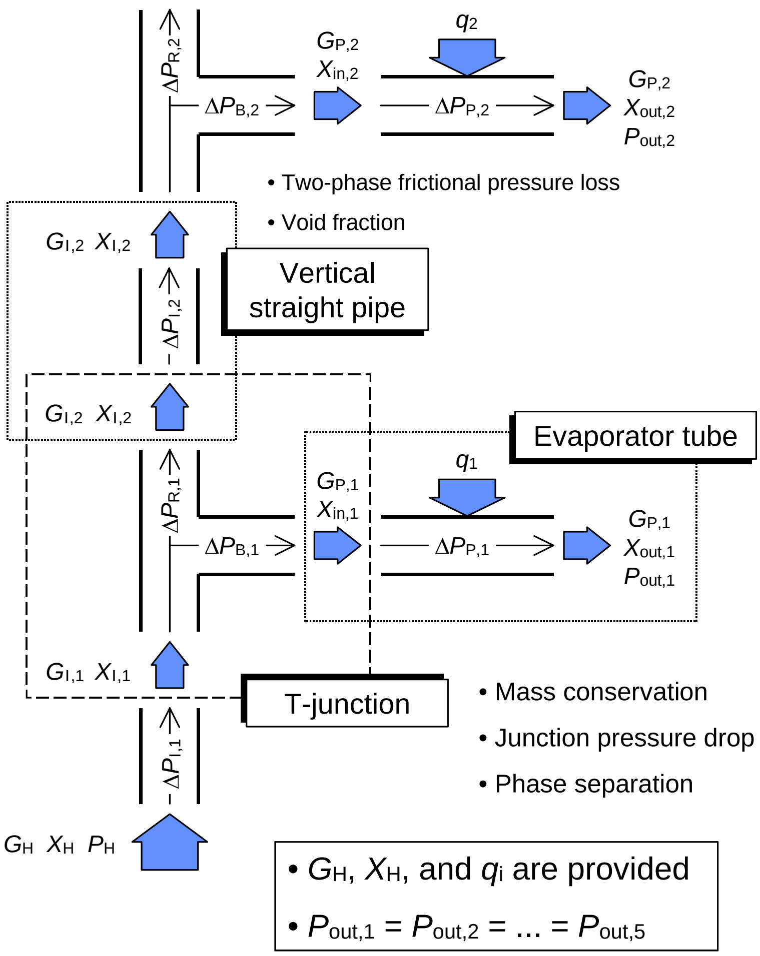


Fig.3 Present simulation model

2 DETAILS OF SIMULATION MODEL

When two-phase flow enters the header, so terrible maldistribution usually occurs that the header is merely used as a flow distribution device. For single-phase flow, e.g. water, air, steam and so on, however, headers are being universally used as a flow distribution device. It would be of great use for investigating the two-phase flow distribution to have a discussion about the single-phase flow distribution in a header. Water flow distribution at the header had been studied by Kubo and Ueda (1968) in both theoretical and experimental way. They considered the main pipe where the flow velocity get reduced due to the flow split as a diffuser, and expressed the pressure recovery by the diffusion effect as a function of flow division ratio Φ. Combined with another equation for the pressure drop along the pass, also presented as a function of the flow division ratio, they concluded they could predict the flow distribution as it made the total pressure change along every pass equal. Utilizing their basic concept, we considered a multipass evaporator as a combination of three elements; upward straight tube, T-junction, horizontal evaporator tube, as shown in Fig. 3. In following sub-sections, the three elements are discussed in detail to formulate the relationship between the pressure changes and the flow rates of gas and liquid.

2.1 Straight tube

Because there is no heat exchange at the upward main pipe, general correlations for frictional pressure loss concerning the adiabatic two-phase flow can be used. In this research, however, the tube diameter is so small as 6 mm in i.d. that whether existing expressions that had been developed for the tube of larger diameter are successfully used is not sure. In order to examine the applicability of those expressions, some preliminary tests for 6 mm i.d.-tube were made to compare the experimental data and the correlated data. As a result, a homogeneous model that adapted an equation of two-phase viscosity proposed by Dukler *et al*. (1964) was found to be successfully used. In addition, an expression for void fraction proposed by Zivi (1964) is used because prediction of the pressure drop along the vertical tube needs to estimate static pressure difference.

For the horizontal evaporator tube, the homogeneous model of Dukler *et al.* (1964) is used with some modification. The original homogeneous model can not be applied to evaporating two-phase flow where the quality is getting changed along the tube, so that the Sympson method is used to numerically integrate the two-phase friction factor which is expressed as a function of quality. One more additional correlation for an acceleration term due to phase change is also employed.

Here, one thing has to be remembered about the experiment that the intensity of heating load on each pass was restricted as it did not make quality at the outlet of each pass excess unity. This was because electric heater was installed to each pass as a heating source to simulate the actual heating load on the evaporator. In this experiment, consequently, pressure drop along each pass could be estimated by relatively easy operations: (i) Calculating the quality at the outlet of pass using heat balance equation. (ii) Integrating two-phase friction factor with respect to quality through the pass-inlet quality to the pass-outlet quality.

If the simulation is applied to practical evaporators, more complicated calculation that divided an evaporator tube into a lot of small cells has to be applied to determine the point where evaporation completes.

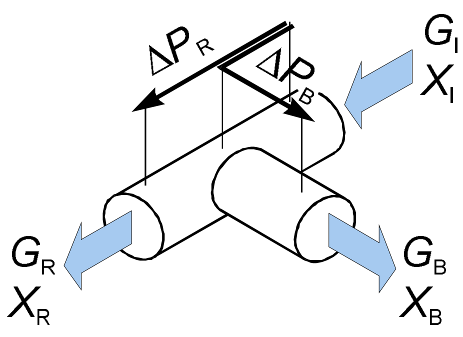


Fig. 3 Definition of variables for dividing two-phase flow at a T-junction

2.2 T-junction

Dividing flow model proposed by Saba and Lahey (1984) has been the only one that took the effect of pressure gradients at the junction into account for predicting the two-phase flow division. They considered eight independent variables concerning to dividing two-phase flow; i.e. gas flow rate and liquid flow rate at inlet main pipe (referred to as INLET in followings), outlet main pipe (RUN, as the same) and branch pipe (BRANCH) and two kinds of junction pressure gradient. Since three of them had to be provided by operational condition, five equations were needed to solve the problem. Saba and Lahey employed the mass conservation equations for gas and liquid flows, correlations for pressure drop along the INLET-RUN stream and the INLET-BRANCH stream. And one more equation namely "the fifth equation" was needed and they proposed to apply the linear momentum equation integrated along free streamline from INLET to BRANCH.

$$\Delta P_{\mathrm{R,i}} = \frac{1}{2}\left(\frac{{G_{\mathrm{P,i}}}^2}{\rho_{\mathrm{h,in,i}}} - \frac{{G_{\mathrm{I,i}}}^2}{\rho_{\mathrm{h,I,i}}}\right) + K_{\mathrm{R}}\left(\frac{{G_{\mathrm{I,i}}}^2}{2\rho_{\mathrm{h,I,i}}}\right) \tag{2}$$

$$\Delta P_{\mathrm{B,i}} = \frac{\rho_{\mathrm{h,in,i}}}{\rho_{\mathrm{h,I,i}}} K_{\mathrm{B}} \frac{{G_{\mathrm{I,i}}}^2}{2\rho_{\mathrm{h,I,i}}} + \frac{\rho_{\mathrm{h,in,i}}}{2}\left[\left(\frac{G_{\mathrm{P,i}}}{\rho_{\mathrm{h,in,i}}}\right)^2 - \left(\frac{G_{\mathrm{I,i}}}{\rho_{\mathrm{h,I,i}}}\right)^2\right] \tag{3}$$

$$K_{\mathrm{R}} = 1.06\Phi_{\mathrm{T}} + 0.395 \tag{4}$$

$$K_{\mathrm{B}} = -1.55{\Phi_{\mathrm{T}}}^2 + 1.74\Phi_{\mathrm{T}} + 1.00 \tag{5}$$

If the same discussion is made on dividing single-phase flow, there are five independent variables; i.e. flow rate at the INLET, RUN, BRANCH and two pressure drops. Since two of them are provided by operational condition, the problem can be easily solved by following three expressions; mass conservation equation and correlations of two kinds of pressure drop. In other words, analyzing the single-phase flow division does not need "the fifth equation". Accordingly, it is indicated that the essential aim of "the fifth equation" is to evaluate the characteristics of phase separation, which is the phenomenon peculiar to dividing two-phase flow. Here, phase separation means being unequal of Φ_G and Φ_L. From the above discussion, it has come to be clear that the expressions for two kinds of junction pressure drop and phase separation are required for analyzing the characteristics of two-phase flow divided at a T-junction.

For junction pressure drop, some semi-empirical correlations have been proposed but they were based on the data from experiments under the condition of large diameter T-junction and they were not sufficiently studied weather they can be successfully applied to T-junction of various configurations. So that authors made experiments to measure the junction pressure drop in a vertical T-junction of 6 mm in i.d. in which air-water adiabatic two-phase flow was used as test fluids to find out the correlation which can successfully predict the junction pressure drop concerning to small diameter T-junction. Detailed results are not discussed here but they suggested that ΔP_R is relatively well predicted by a homogeneous model formulated as Eq.(2) and ΔP_B showed better agreement with the correlation derived by Reimann and Seeger (1986), Eq.(3). In these equations, ρ_h presented the homogeneous density and functions that present K_R and K_B were determined as Eqs.(4), (5) by fitting of our experimental data.

As the equation for phase separation, the empirical correlation for the liquid division ratio derived from the data of two-phase flow distribution in header-type multipass evaporator, already introduced as Eq.(1), was applied. Since Φ_L was presented as a function of $G_{G,I,i}$ and $G_{L,I,i}$ as shown in Eq. (1), this equation did not provide a phase separation characteristics, whose restrict meaning is the relationship between Φ_G and Φ_L , but it can be sufficiently used for the aim of closing the simultaneous equations.

3 PROCEDURE OF THE SIMULATION

A flow chart of the simulation is shown in Fig. 4. First, operational condition must be determined to provide mass flux, quality and pressure at the inlet of header, represented as G_H , X_H and P_H. The pressure at the outlet of passes $P_{out,ini}$ is also postulated here. Since the flow condition at the inlet main pipe of junction 1 is same as that at the inlet of header, the pressure drop at the inlet main pipe of junction 1 can be calculated to determine the pressure at the junction 1. Then, postulated gas division ratio $\Phi_{G,1}$ and the liquid division ratio $\Phi_{L,1}$ which is calculated by using Eq. (1) provide the flow rates of gas and liquid at the inlet of pass 1. Subsequently, junction pressure drop along the dividing stream at the junction 1 $\Delta P_{B,1}$ is calculated by Eq. (3), then by using a correlation of pressure drop along the evaporator tube, the pressure at the outlet of pass 1 $P_{out,1}$ can be computed. If the computed $P_{out,1}$ differs from initially postulated $P_{out,ini}$, this operation is repeated with changing $\Phi_{G,1}$ until the difference between computed $P_{out,1}$ and $P_{out,ini}$ comes to be less than adequate allowance. Once the $\Phi_{G,1}$ is obtained, the amount of gas flow which is not diverted into the pass 1 is obtained. It means that the flow condition at the inlet main pipe of junction 2 becomes clear, so that the junction pressure drop along the penetrating stream $\Delta P_{R,1}$ and the pressure drop at the inlet main pipe of junction 2 can be calculated to determine the pressure at the junction 2. These series of calculation are represented as LOOP1 as shown in Fig. 4 and the same series are applied to pass 2, 3, 4, shown as LOOP2, LOOP3, LOOP4 in Fig. 4. Consequently, the flow conditions at the inlet main pipe of junction 5 are clarified, and because the junction 5 is actually not a junction but an elbow, the flow rate of each phase at the inlet of pass 5 is automatically obtained and the pressure at the outlet of pass 5 $P_{out,5}$ can be computed without any assumption. If the computed $P_{out,5}$ is differ from $P_{out,ini}$, the whole simulation are repeated with changing initially postulated $P_{out,ini}$.

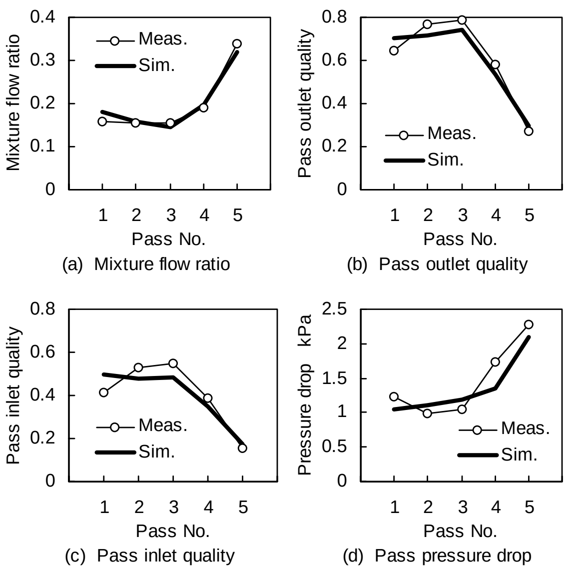


Fig. 5 Examples of simulation result
(Condition: Flat heating, Q=450 W, X_H=0.3)

4 RESULTS AND DISCUSSION

Some examples of simulation results are shown in Fig. 5 compared with the previous experimental data (Watanabe et al., 1995a). These data were obtained under the condition of X_H=0.3 and test code 450F, defined in Table 2. Fig. (a) ~ (d) presents the mixture flow ratio $\Gamma_{T,in,i}$, quality at the outlet of pass $X_{out,i}$, quality at the inlet of pass $X_{in,i}$, pressure drop along the pass $\Delta P_{P,i}$, respectively. From the figures, it is clear that the simulation results well predict the measured data. For the other conditions, the agreement of simulated and measured data is approximately well as long as X_H remains high. Under the condition of lower X_H, unfortunately, the simulation is not successfully solved for some worst cases; or simulation, even solved, can no longer predict the measured data for some other cases. Although above mentioned problems are exist, it can be concluded that proposed simulation model predict the experimental data approximately well at least under the condition of high X_H that appears in the practical condition of refrigeration equipments.

According to the process for establishing the simulation model, it has come to clear that all the expression needed for calculation, even with some difficulties, could be found from existing researches, except "the fifth equation" that present the phase separation characteristics of dividing two-phase flow. This fact suggests us that the complexity for predicting two-phase flow distribution in multipass evaporator is essentially due to the complicated characteristics of phase separation phenomena in dividing two-phase flow. It means that proposed simulation model could be a universal method, which can be applied to multipass evaporator of arbitrary geometric configuration, if the appropriate expression is employed as "the fifth equation". Since there are a number of researches about the phase separation in two-phase flow divided at a T-junction (e.g., Azzopardi, 1986), it is expected that the good expression would be found. Authors have been making that kind of attempts (Watanabe *et al.*, 1998), and obtained good results by utilizing general prediction model for phase separation in dividing annular flow.

Consequently, it is concluded that the suitableness of proposed simulation model indicated the possibility of a universal method for predicting two-phase flow distribution in multipass evaporator.

5 CONCLUSIONS

The simulation model for predicting two-phase flow distribution in header-type multipass evaporator was proposed. Simulation results showed approximately well agreement with measured data from authors' previous experiment. The suitableness of the proposed method indicated that the complexity of the two-phase flow distribution in multipass evaporator was essentially due to the phase separation characteristics in dividing two-phase flow. It could be possible to establish a universal model that predicts the two-phase flow distribution in multipass evaporator of arbitrary geometric condition.

NOMENCLATURE

G	Mass flux	kg/m^2s	P	Pressure	kPa	q	Heat flux	kW/m^2
Q	Total heat input	W	X	Quality	—	Φ	Flow division ratio	—

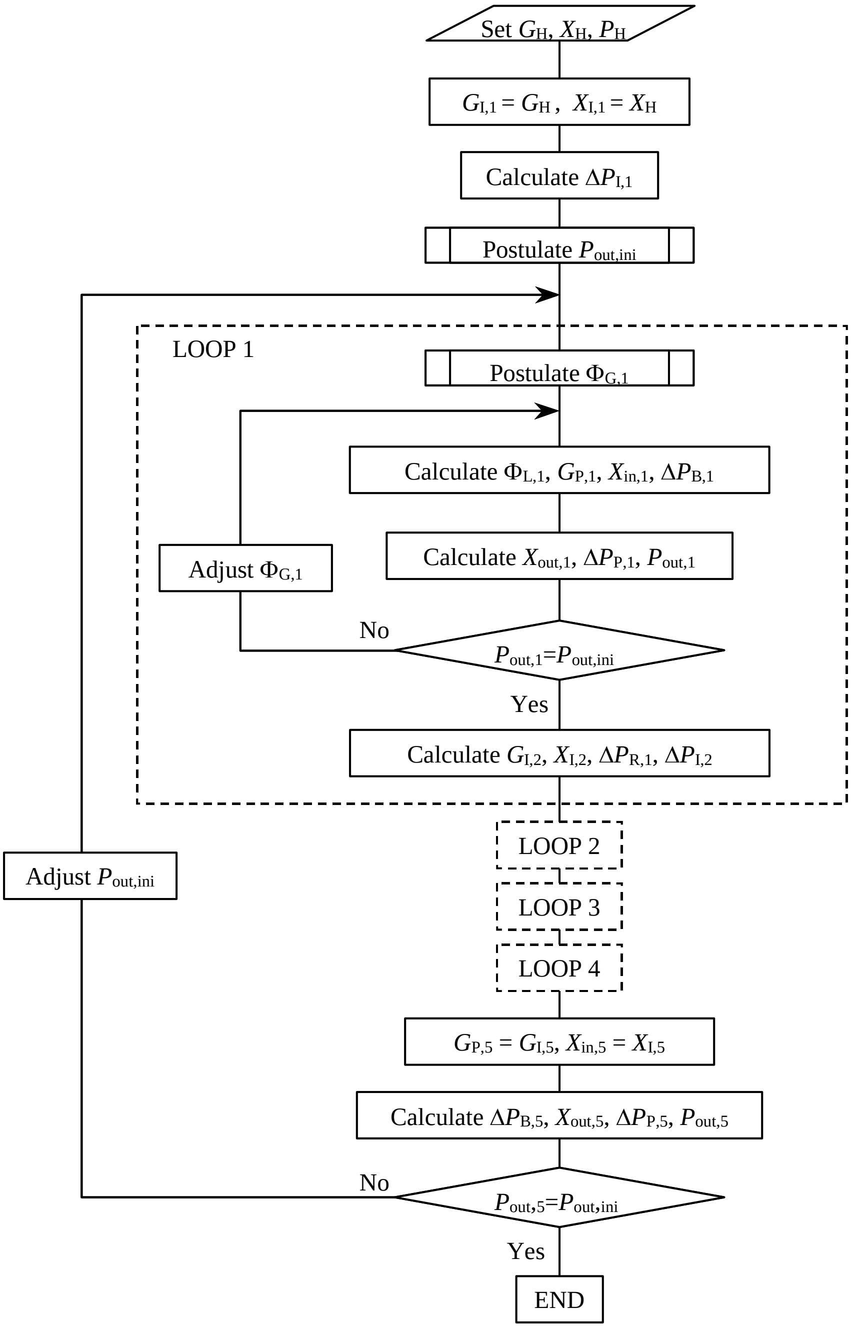


Fig. 4 Flow chart of proposed simulation

Subscripts

B : BRANCH of T-junction	G : Gas phase	H : inlet of header	I : INLET of T-junction
i : number of pass	in : inlet of pass	ini : initial value	L : liquid phase
out : outlet of pass	P : pass	R : RUN of T-junction	T : two-phase mixture

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Le modèle général de prédiction pour la distribution d'écoulement diphasique dans un évaporateur à parcours multiples

RESUME : Un modèle de computation est proposé, par qui on peut prévoir la distribution d'écoulement diphasique dans un évaporateur à parcours multiples. Dans ce modèle, il est considéré que l'évaporateur à parcours multiples avec une bordure soit une combinaison des éléments simples – c'est-à-dire, tuyaux droits et les jonctions à T - et les corrélations pour prévoir la diminution de pression aux éléments sont utilisées. Comme une corrélation pour les caractéristiques de la séparation de phase à la jonction à T, les équations empiricales pour le rapport de la division liquide sont utilisées, qui sont dérivées de nos données expérimentales précédentes. Par ce modèle, la distribution d'écoulement de phase gazeuse pour chaque parcours est déterminée parce que la pression à l'orifice de sortie de chaque parcours est égalisée. Les résultats de computation prévoient bien les données expérimentales précédentes qui étaient obtenues dans le domaine de la qualité pratique à l'orifice de sortie d'un évaporateur. La pertinence de ce modèle suggère que la complexité de la distribution d'écoulement diphasique dans le tuyau à parcours multiples attribue aux phénomènes de la séparation de phase en divisant l'écoulement diphasique.