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ABSTRACT

An analysis is carried out for flow boiling in a vertical matrix aluminum heat exchanger. The prediction model, developed for thin film evaporation in a previous study, is extended to include heat transfer in the slug-flow regime that exists at low mass qualities. Appropriate criteria is used to switch from slug-flow to thin-film annular-flow analysis. The two-phase-flow convective heat transfer enhancement for the slug-flow is correlated with inclusion of fluid Reynolds and Prandtl numbers in addition to commonly used Martinelli parameter. This approach reflects transient nature of heat transfer in the slug-flow regime. The thin-film annular-flow analysis developed in the previous study is refined with inclusion of a reliable two-phase friction factor correlation. The experimentally measured pressure drop is used to validate the friction factor correlation. The resulting prediction method is used to predict exit mass qualities for ammonia and R-114. The experimental analysis includes flow boiling of ammonia and R-114 in a vertical brazed-aluminum matrix heat exchanger. The test unit has straight perforated fins on the fluid side and extruded rectangular channels on the single-phase (water) heating-media side. Only two parameters are adjusted to validate the analytical prediction method, the constant in the friction factor correlation, and the constant in the slug-flow heat transfer correlation. The results show that the combination of slug-flow and thin-film annular-flow model gives better prediction of the overall performance of the matrix heat exchanger than a single model applied for the whole range of mass qualities.

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INTRODUCTION

Matrix heat exchangers are being considered for various applications in single- and two-phase flow systems. The compactness, flexibility of selection for flow geometry, and relatively high thermal performance are reasons for increasing use of matrix heat exchangers. In recent years, significant progress has been made for understanding single- and two-phase flow heat transfer in such flow channels. However, review analysis of Robertson (1), and Westwater (2) indicated that no unified prediction methods are available for flow boiling in narrow passages of matrix heat exchangers. Scaling the geometry, fluid properties and test conditions from experimental apparatus to prototypical systems has a large uncertainty. The present analysis is an advancement of the previous studies, and it is focused on key observations made by the investigators regarding two-phase flow behavior in narrow flow passages.

The previous studies of phase-change heat transfer in matrix heat exchangers can be summarized as follows. Robertsons and coworkers (3,4,5,6,7,8,9,10) have reported experimental data with cryogenic fluids and refrigerants. The test section they have used represents prototypical flow channel, however, the constant heat flux electric system may not represent process conditions. Their data and limited analysis seem to provide key information necessary to develop prediction methods. Carey and coworkers (11,12,13,14,15) have conducted experiments with a short test section with large fin sizes (pitch and height). The test section may not represent the commercial unit, however, their data and visual observation of two-phase flow could help identifying the governing heat transfer mechanism. They correlated the experimental data using an empirical approach that is based on the Martinelli parameter conventionally used for two-phase flow in circular tubes. They did not differentiate flow regimes and developed separate correlation for slug-flow and annular-flow regimes. Westwater and coworkers (16,17,18,19,20) carried out experiments with a small multiflow channel test unit with condensing steam as heat source. However, temperature driving force was relatively high, and therefore, nucleate boiling was the principal mode of heat transfer. Their analysis concentrated on temperature distribution on the fin, and the effects of two-phase flow velocity on nucleate boiling. Argonne National Laboratory (21,22,23) has tested two-prototypical heat exchangers, one with serrated- and one with straight-perforated fins. The Argonne analysis is based on the assumption of annular flow with no entrainments, which may be justified for low mass flux of ammonia used for possible use of this heat exchanger for OTEC applications. Additional work can be grouped as early work (24,25,26) in which the matrix heat exchanger was considered for boiling and condensation applications. Recently adiabatic two-phase analysis for narrow channels have been carried out (27,28,29), which could be used for the development of prediction methods.

In the present analysis, the previous (21) study of flow boiling in matrix heat exchangers is extended to include slug-flow heat transfer for low mass qualities. Moreover, experimental data with two-fluids, ammonia and R-114 with widely different physical properties are used to validate the analysis. In addition, the analysis follows key observation made in the previous studies, as a result the prediction method can be considered as an evolving physical model rather than new empirical correlation developed for a specific geometry and test conditions. Owen and Hewitt (30) have argued that the prediction method based on so called "phenomenological models" should have wider range of application, and as more data and observation information become available, it can be continuously improved.

ANALYSIS OF PREDICTION METHODS

The thermal prediction method includes the single phase heat transfer, slug-flow analysis for low mass quality region, transition criteria from slug- to annular-flow, force balance on thin liquid film in annular-flow, and heat transfer in the film. Although the present analysis is for straight perforated fins, it can be extended to the serrated fin geometry with inclusion of appropriate single phase correlation for heat transfer and friction factor calculation. The following assumptions are made in the present analysis:

1. No entrainment occurs in the vapor core.
2. The flow of liquid is evenly distributed on the fin and plate.
3. Surface tension (Gregorig) effects are negligible.
4. Liquid-film is laminar for force balance, however, Hewitt's analysis (31) is included for calculation of film Nusselt number.
5. No nucleate boiling at any point i.e. subcooled and saturated boiling conditions.

Single-Phase Heat Transfer

In the previous analysis (21) the single-phase heat transfer was not considered, because for ammonia tests, the inlet subcooling was small. However, for R-114 tests average inlet subcooling is about 17°C, and heat transfer area required for the single-phase heat transfer cannot be ignored. The following correlation developed by Shah and London (32) for laminar flow in rectangular channel was used.

$$Nu = 7.541\ (1 - 2.61\ \alpha + 4.97\ \alpha^2 - 5.119\ \alpha^3 + 2.702\ \alpha^4 - 0.548\ \alpha^5) \tag{1}$$

This correlation was compared with the reported heat transfer coefficient by Robertson (3,4) for cyclohexane and liquid-nitrogen and a good agreement was observed. Moreover, the data showed that the transition occurred at Reynolds number of about 1500, and the turbulent-flow heat transfer behavior was seen for Reynolds numbers above 4000. In the present liquid Reynolds numbers was less than 2000 for most test runs , however, interpolation method was used for calculation of the heat transfer coefficient for Reynolds numbers between 1500 and 4000. A sensitivity study showed that the overall results were relatively insensitive to the exact transition Reynolds number within the range of uncertainty. For the friction factor calculation, the following correlation developed by Shah and London (32) was used.

$$f = 8.968 \left[\frac{(1+\alpha)}{\alpha^{0.5}}\right]^{0.75} / Re \qquad (2)$$

Slug-Flow Analysis

As reported in the previous analysis (21), the slug-flow region is expected to be less than 15% of total heat exchanger length for ammonia tests. Moreover, inclusion of the slug-flow has small effects on the prediction of the overall performance as shown by two previous analyses (21,22). However, for R-114 tests, the initial analysis showed that it was not possible to neglect the slug-flow. It should be borne in mind that understanding of the heat transfer mechanism for the slug-flow is far less than that for the annular-flow, therefore, semiempirical approach needs to be adopted. Most previous studies referred to in the introduction have correlated the data using the Martinelli parameter without differentiating flow regimes. However, Robertson (3) has indicated that the slug-flow should be treated as transient heat transfer to the liquid phase flowing between vapor bubbles. In that case a correlation of the following form should be used for heat transfer in the slug-flow regime.

$$Nu_{TP} = Nu_f \left[1 + F(1/X)^{0.36} (Re_f Pr_f)^{1/3}\right] \qquad (3)$$

where Martinelli's parameter, X, is defined as $[(-dp/dz)_g/(-dp/dz)_f]^{1/2}$. Here, the term expressed in terms of the Martinelli parameter represents the entrance length (length to equivalent diameter ratio). The term shown by Reynolds and Prandtl numbers raised to 1/3 gives the transient heat transfer to the liquid phase for time of contact of the given slug with the wall. In this approach it is assumed that the heat transfer resistance of vaporization at the interface is negligible, and slugs are completely mixed between consecutive wall contacts. It should be noted that the analysis of ammonia and R-114 data improved significantly by inclusion of Reynolds and Prandtl numbers in Eqn. 3. The exponent to Martinelli's parameter of 0.36 was taken from experimental results of Robertson (5). Other correlations of the form reported by Carey and coworkers (11,12,15) where Reynolds and Prandtl numbers are not included were tried, but a common constant could not correlate data for both fluids, ammonia and R-114.

The adiabatic two-phase data reported by Lowry (28) were used for calculating the multiplier in the friction factor equation. The data were correlated as follows:

$$\phi_f = 1.5\ (Jg^*)^{0.176} \quad \text{for} \quad Jg^* < 1.0 \tag{4}$$

$$= 1.5\ (Jg^*)^{0.311} \quad \text{for} \quad Jg^* > 1.0$$

The total pressure drop included frictional, gravity and momentum change losses. The frictional pressure drop in the slug-flow regime was relatively small, therefore, the uncertainty in the above correlation did not contribute significant error in the total pressure drop. The void fraction was estimated using the following equation which is based on the drift flux model. Later, it will be shown that a relatively smooth transition of void fraction occurs from the slug- to annular-flow where the void fraction is calculated using the calculated liquid film thickness.

$$\alpha_o = \frac{\mathring{x}}{\mathring{x} C_o(\rho_f-\rho_g)/\rho_f + (C_o+U_{gj}/U_{fj})\rho_g/\rho_f} \tag{5}$$

where

$$C_o = 1.13$$

$$U_{gj} = 1.41 \left[\frac{\sigma g(\rho_f-\rho_g)}{\rho_f^2}\right]^{0.5}$$

$$U_{fj} = m/\rho_f$$

Transition from Slug-Flow to Annular-Flow

Recent investigation of Damianides (27) for two-phase flow in horizontal serrated-fin flow channel shows that the flow regime maps developed for circular tubes are not applicable for flow in narrow finned channels. No such flow maps or transition criteria have been developed for matrix heat exchangers. However, it can be assumed that the annular-flow would prevail provided the interfacial shear stress produced by the fast moving vapor is able to support the liquid film without having the wall shear stress opposite to the interfacial shear stress. Similar argument was offered by Robertson (4) in his analysis of nitrogen data. However, they used dimensionless vapor velocity, Jg* equal to 1 for the transition from slug- to annular-flow regime. In the present a test was included in the analysis for balancing the force on the film. At the point where the wall shear stress becomes positive, the analysis is shifted to the annular-flow conditions. It should be noted that for most ammonia tests the transition occurred at Jg* of 0.8 to 1, while for R-114 tests Jg* values were in the range of 1.4 to 1.8. The following equation reported in the text book of Collier (33) was used for approximating the transition region.

$$Jg^* = 4(\rho_g/\rho_f)^{0.5}(J_f^* + R) \tag{6}$$

where

$$R = 0.23 + 0.13\ s/w$$

<u>Annular-Flow Analysis</u>

The annular-flow analysis followed the previous study (21) without any significant changes. The major change in the present analysis is for calculation of two-phase friction factor. However, for the sake of completeness the analysis is summarized here.

A force balance on the rising film (see Fig. 1) can be expressed as follows.

$$\mu_f \frac{du_f}{dy} = (-\ \rho_{fg} - dp/dz)(\delta - y) + \tau_i \tag{7}$$

The pressure gradient term, $-\frac{dp}{dz}$ is madeup of frictional, gravity and momentum change components:

$$-\frac{dp}{dz} = 4\frac{\tau_o}{d_e} + g\ [\alpha_o\rho_g + (1 - \alpha_o)\rho_f] \tag{8}$$

$$+ m^2 \frac{d}{dz} \quad \frac{\dot{x}^2}{\alpha_o\rho_g} + \frac{(1-\dot{x})^2}{(1-\alpha_o)\rho_f}$$

Integration of Eqn. 1 yields the following equation:

$$\Gamma = \frac{\rho_f(-\rho_f g - \frac{dp}{dz})\delta^3}{3\,\mu_f} + \frac{\tau_i \rho_f \delta^2}{2\,\mu_f} \tag{9}$$

The shear stress at the vapor-liquid interface is given by

$$\tau_i = \frac{f_{TP}}{2} \rho_g \, (\bar{U}_g - U_{fi})^2 \tag{10}$$

where f_{TP} is two-phase-flow friction factor, and the interfacial liquid velocity is 1.5 time average film velocity as determined by laminar film flow. The relationship between the interfacial and wall shear stresses can be expressed as follow by assuming linear shear stress distribution in the film.

$$\tau_o = \tau_i - \rho_f g \delta \tag{11}$$

Several investigations have been carried out to correlate the two-phase friction factor for the annular-flow. Most correlations have been based on the roughness factor principle, where the liquid film due to its thickness and the interfacial shear stress produces wavy surface. In the previous study (21), the correlation developed by Hanaratty (34) was used, and the experimentally measured pressure drop was used to determine a constant in the equation. However, this correlation was not successful for both ammonia and R-114 data, probably due to noniterative approach used in the analysis in which the film Reynolds number is used instead of film thickness. Therefore, correlations that included the liquid film thickness were considered in the present analysis. In correlation by Wallis as reported by Gilchrist (35) the friction factor is indirectly correlated by using the void fraction, which was used in the previous study (22) for flow boiling of ammonia in serrated-finned channels. The correlation by Whalley and Hewitt as reported by Gilchrist (35) is similar to Wallis correlation except that vapor to liquid density ratio is included. These two correlations did not give satisfactory agreement between predicted and experimentally measured pressure drops. It should be noted that in the present analysis a constant appearing as a multiplier in the correlation is adjusted to give minimum difference between experimental and calculated pressure drops. The correlation by Roberts and Hartley as reported by Gilchrist (35) was found to give the most satisfactory comparison among all correlations used in the present analysis. The correlation is expressed as follows:

$$f_{TP} = f_g + K \left[\frac{\delta}{d_e} - \frac{5}{Re_g} \left(\frac{2}{f_g} \right)^{1/2} \right] \tag{12}$$

Owen and Hewitt (30) derived an equation for friction factor in terms of roughness factor. They argued that the equivalent roughness of the interface should include the boundary sublayer in the gas core, which is represented by the second term of Eqn. 12, which is identical to that used by Owen and Hewitt (30). It should be noted that ammonia data have low gas-side Reynolds numbers in range of 2000 to 4000, as compared to that for R-114 data where Reynolds number was in the range of 4000 to 16000 for the annular-flow regime. Therefore, for a comparable film thickness as observed in the analysis, two-phase friction increase as compared to the single phase flow is less for ammonia than for R-114, which is reflected in the overall pressure drop measurements. Gilchrist (35) compared various correlations with their experimental data and it was observed that correlation of Roberts and Hartley predicted the experimental data better than all other correlations.

Pressure gradient for the annular-flow regime is calculated using Eqn. 8, and the void fraction is calculated using the following equation, which assumes uniform film thickness in the channel.

$$\alpha_o = \left(1 - \frac{\delta}{a}\right)\left(1 - \frac{\delta}{b}\right) \tag{13}$$

The film heat transfer coefficient is calculated using the following equation:

$$h = \frac{k_f}{\delta} Nu_f \tag{14}$$

where Nu_f is film Nusselt number which accounts for turbulence and instability at the interface. Hewitt and Taylor (31) determined the Nusselt number as a function of film Reynolds number, and Robertson (4) found that the film Nusselt number could correlate the data for nitrogen in perforated finned channels. In the present analysis, the film Nusselt number was calculated by correlating tabulated values of Nu_f by Collier (33). For ammonia the film Reynolds numbers were less than 400, and the film Nusselt number was close to unity. However, for R-114 tests data the film Reynolds numbers were in the range of 1000 to 1800, and Nu_f values ranged from 2 to 3.

Computation Procedure

A computation procedure was employed for calculation of outlet conditions like exit mass quality, total pressure drop and rate of heat transfer. The fluid entered from side of the heat exchanger and made 90 degree turn before flowing in the straight section. A mean level was used as an inlet point. The effective length of heat exchanger was determined by equivalent straight section, however, inlet and outlet sections where fluid has to make 90 degree turn were relatively small compared to the total length. The heating media, water entered from a nozzle located at the end of heat exchanger giving parallel flow configuration. The heat exchanger was divided into n (generally 100) number of sections, and the integration was started from the bottom, where all inlet conditions were known. At each length the fluid side heat transfer coefficient was calculated after checking local conditions for single-phase, slug-flow or annular-flow. Corresponding calculation method was used for determination of the heat transfer coefficient. For the annular-flow regime an iterative method was used to calculate the film thickness. The iterative loop included Eqns. 9, 10, 12, and 13. The fin efficiency was calculated assuming uniform heat transfer coefficient in the channel. The water-side heat transfer was calculated using Sleicher and Rouse equation (36), which in the previous study (21) was found to be a better correlation for the present range of conditions.

In order to minimize the effects of local mean temperature difference caused by pressure distribution, an iterative method was used in which the inlet pressure was adjusted to match the outlet pressures at the top. The pressure gradient at the outlet was generally steep as compared to that at inlet, as a result, prediction of heat transfer coefficient was affected by calculation of local mean temperature difference. The total pressure drop is then compared with experimental values. Constant, in Eqn. 12 was adjusted to give minimum error for all test data for ammonia and R-114. Next, constant F in Eqn. 3 was adjusted to give smooth transition of the heat transfer coefficient at the point of transition from slug-flow to annular-flow. A common set of values for these constants was used for all test data of ammonia and R-114.

EXPERIMENTS

A complete description of the experimental facility and test unit is given elsewhere (23). Figure 2 shows an elemental section of the test unit, and appropriate dimension are reproduced in Table 1. All inlet and outlet parameters i.e. temperature pressure, and flow were measured with precision instruments to give accuracy of measurements within 2% for rate of heat transfer, exit quality and pressure drops. The rate of heat transfer, Q was determined from the water-side flow rate and change in temperature. The exit quality was then calculated using the inlet fluid flow rate and rate of evaporation calculated from the rate of heat transfer. The pressure drop was calculated using measured pressures at the inlet and outlet, where the pressure sensors were located at mean level of inlet and outlet nozzles.

Table 1 shows range of operating conditions for ammonia and R-114 tests. Comparable heat fluxes were used for both series of tests, therefore, mass flux for R-114 was significantly greater than that for ammonia. Moreover, inlet subcooling was higher for R-114 than that for ammonia. Except for the Wilson plot method, water flow rate was maintained constant at 2.2 ± 0.01 m/s. The outlet pressure was controlled to a predetermined value and maintained constant for a given series of tests, while the inlet water temperature floats over a narrow range depending upon heat flux and the overall heat transfer coefficient. A total of 37 test runs were taken for ammonia, of which 18 were for the Wilson plot analysis i.e. water flow rate was varied by keeping the fluid-side conditions constant. The remaining 19 test runs were taken for studying the performance with mass flux and exit mass quality as variables. For R-114, 22 test runs were taken, of which only 6 were for the Wilson plot analysis.

VALIDATION OF PREDICTION METHODS

The validation procedure centered around determination of constants K in Eqn. 12, and F in Eqn. 3. In the original correlation of Roberts and Hartley, value for K is 6, which was used to start the analysis. However, the predicted pressure drops for most test runs were small by more than 10%. The value is then increased to 9, which improved the comparison. Finally, the agreement was acceptable for K value of 12. Figure 3 shows the difference between experimental and predicted pressure drop as a function of mass flux. It shows that the pressure drops for most test run is within 10%, however, for some ammonia runs the value is relatively high. It can be argued that due to low mass flux for ammonia, the pressure drop was relatively small, therefore, small error could show up as large percentage error. It will be shown later that this degree of agreement in the pressure drop was adequate for heat transfer calculations, because for the film thickness calculation selection of K value was not as much sensitive as for the pressured drop determination. It can be argued that the original value of 6 for constant K was determined for circular tubes, and a higher value may be expected for a narrow rectangular channel with aspect ratio of about 7 in the present analysis.

The second constant, F in Eqn. 3 was determined by observing the film coefficient at the point of transition from slug- to annular-flow regime. The criteria was to have relatively smooth transition of the heat transfer coefficient for both fluids, ammonia and R-114. A value for F of 0.35 was determined iteratively along with that for K as discussed above. Figures 4 and 5 show axial profiles of the heat transfer coefficient for ammonia and R-114 respectively. The results show that for both fluids the heat transfer coefficient gradually changes from a slug-flow to annular-flow for both fluids. Similar trend was observed for all ammonia and R-114 test runs. However, for about 3 R-114 test runs, the slug-flow heat transfer coefficient was about 20% higher than the annular value at the point of transition. This could be attributed to the location of exact transition point, because in a very short distance from the transition point the annular-flow heat transfer coefficient shows continuity with that for the slug-flow. In the analysis the effects of moving the transition point within a small range was evaluated and no noticeable difference in the overall results was observed.

Figures 6 and 7 show void fraction distribution for ammonia and R-114 test runs. It should be noted that void fraction for slug-and-annular-flow regimes was calculated using Eqns. 5 and 13, respectively. Equation 5 is based on drift-flux model assumptions, while void fraction calculation for the annular-flow regime is based on the film thickness determined in the present analysis. Considering the uncertainty of the present analysis and the drift-flux model, the agreement shown in Figs. 6 and 7 for two fluids with widely different physical properties should be quite satisfactory. In all test runs, a slight discontinuity is observed at the point of transition, but the extended slug-flow curve merge smoothly with the annular-flow curve.

Following the determination of constants K and F, the exit mass quality was calculated for all test runs and compared with the experimental data as shown in Fig. 8. On average exit mass quality for ammonia agreed with the predictions within about 3 %. For some high test runs where exit mass qualities were greater than 80 %, the predicted values were about 10% higher, which could be contributed to local dryout which is included in the present analysis. For R-114 test runs, the predicted mass qualities were generally lower by about 5 to 8% as compared to experimental values for test runs with exit mass qualities greater than 70%. For other test run, the agreement was within 5%.

DISCUSSION OF RESULTS

The degree of agreement shown in Fig. 8 for experimental and predicted exit mass qualities and the previous figures for local profiles, indicates that the combination of slug- and annular-flow models have potential to predict the thermal performance. The use of pressure drop data became the focal point of the analysis. It is however important to develop necessary two-phase data required in an analysis of this nature. The key feature of the analysis was to examine local heat transfer behavior in addition to prediction of the overall performance. The proposed slug-flow correlation reflects the transient nature of heat transfer in this flow regime. Due to widely different physical properties and operating conditions of ammonia and R-114, it was possible to identify the need to include fluid Reynolds and Prandtl numbers in the slug-flow heat transfer correlation.

It is important to scale the experimental data to prototypical heat exchanger flow geometries for flow boiling of single- and multicomponent-fluid systems. Unless a satisfactory prediction method is developed for single-components, it would be quite difficult to extrapolate the experimental data for multicomponent-fluid systems, where additional resistances due to mass transfer have to be included in the analysis. The present analysis can be easily extended to flow boiling of multicomponent systems. Moreover, the present analysis identifies specific two-phase flow information required for the development of prediction methods for the thermal performance.

The major objective of the present analysis was to develop a prediction method that can be used for two fluids with widely different physical properties tested with the same heat exchanger. It followed the similar logic used by Robertson (8), where a film model was proposed to satisfy nitrogen and R-11 test data. Therefore, the present analysis extends the previous work and uses the established results to develop a prediction method. It should be borne in mind, that most experimental analysis have been carried out with fixed flow geometries. Therefore, a major uncertainty prevails for extrapolation of the literature data to other commercial flow geometries, e.g., fin-density, -height, and -thickness and straight vs. offset fins. Validated analyses are the only way to extrapolate the data to other geometries, unless a major experimental program is carried out. Flexibility of design parameters for matrix heat exchangers are significantly greater than those for conventional shell-and-tube heat exchangers. Therefore, such analyses become important for utilization of the design flexibility available for matrix heat exchangers.

CONCLUSIONS

The theoretical analysis developed in the previous study is extended to include slug-flow heat transfer at low mass qualities. It is proposed that fluid Reynolds and Prandtl numbers should be included in the slug-flow heat transfer correlation. The annular-flow analysis is refined with incorporation of the film Nusselt number and use of a better two-phase friction factor correlation. A criteria of supporting the liquid film by the interfacial shear stress is used for locating transition point from slug- to annular-flow.

The analysis is validated with ammonia and R-114 test data. The pressure drop data are used to determine a constant in the two-phase friction factor correlation. Similarly, a constant in the slug-flow heat transfer correlation is determined by matching the film coefficient at the transition point. A single set of constants is used to validate ammonia and R-114 test data. The resulting analysis predicted the exit mass qualities for both fluids within the overall uncertainties.

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NOTATIONS

a	half channel width
b	half channel height
d_e	equivalent diameter
f	friction factor
F	undetermined constant in Eqn. 3
g	gravitation constant
h	heat transfer coefficient
Jg^*	dimensionless vapor velocity
Jf^*	dimensionless fluid velocity
k_f	fluid thermal conductivity
K	undetermined constant in Eqn. 12
m	mass flux
Nu	Nusselt number, $(h\ k_f/d_e)$
p	pressure
Pr	Prandtl number
U_{gj}	superficial velocity for vapor
U_{fj}	superficial velocity for fluid
S	channel height
W	channel width
X	Martinelli parameter
$\dot{X}$	mass quality
y	defined in Fig. 1
z	axial distance

Greek letters

α	channel aspect ratio, a/b
α_0	void fraction
δ	liquid film thickness
Γ	liquid mass flow per unit perimeter
ϕ	friction factor multiplier defined in Eqn. 4
ρ	density
μ	viscosity
τ_i	interfacial shear stress
τ_0	wall shear stress
σ	coefficient of surface tension

Subscipt

f	fluid
g	vapor

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Table 1. Heat exchanger geometry and test conditions

Parameters		Ammonia	R-114
<u>Operating Conditions</u>			
Working fluid			
Outlet pressure	(10^5 N/m^2)	9.4	3.5
Mass flux	(kg/s m^2)	20 to 43	85 to 250
Inlet subcooling	(°C)	3 to 7	13 to 19
Exit mass quality	m^2	0.34 to 0.85	0.53 to 0.82
Water			
Velocity	(m/s)	1 to 3.4	1 to 3.4
Inlet temperature	(°C)	25 to 30	44 to 49
<u>Heat Exchanger Geometry</u>			
Material		Aluminum Al-3003	
Effective length	(m)	4.71	
Flat-plate heat transfer area	(m^2)	49	
Water flow area	(m^2)	0.093	
Fluid flow area	(m^2)	0.036	
Working fluid			
Fin thickness	(mm)	0.4	
Fin spacing	(fins/mm)	0.72	
Fin height	(mm)	7.1	
Perforated area	(%)	3.3	
Water			
Channel size	(mm)	19.7x29.0	
Fin thickness	(mm)	3.2	
Wall thickness	(mm)	2.9	

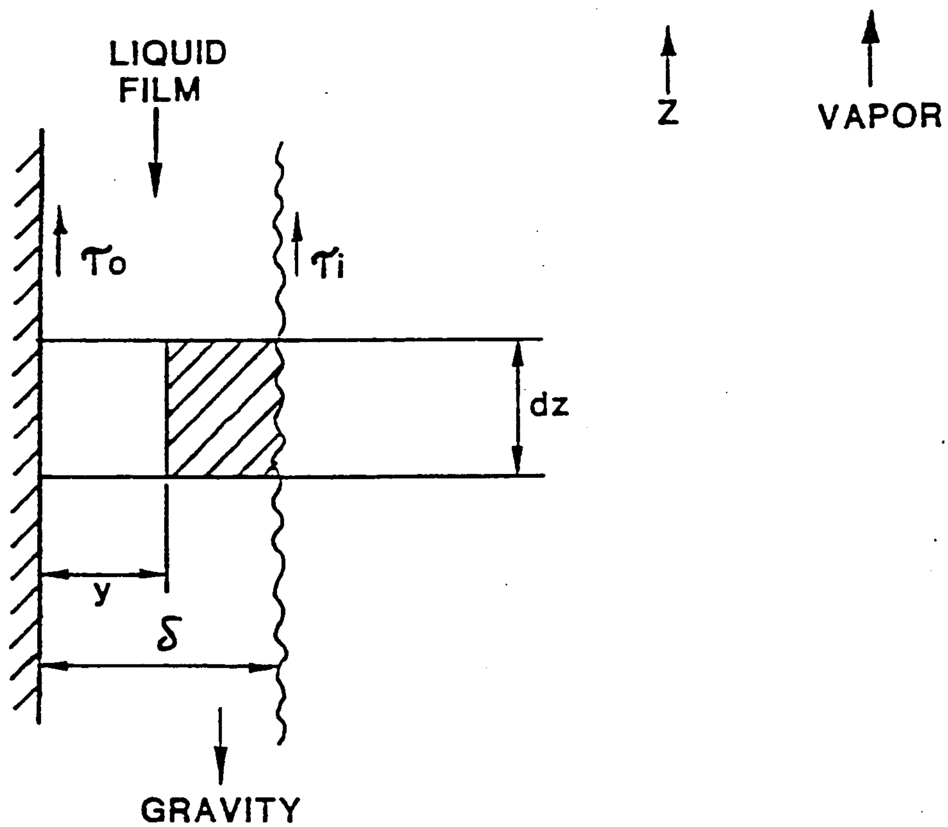

Fig. 1. Force balance on liquid film.

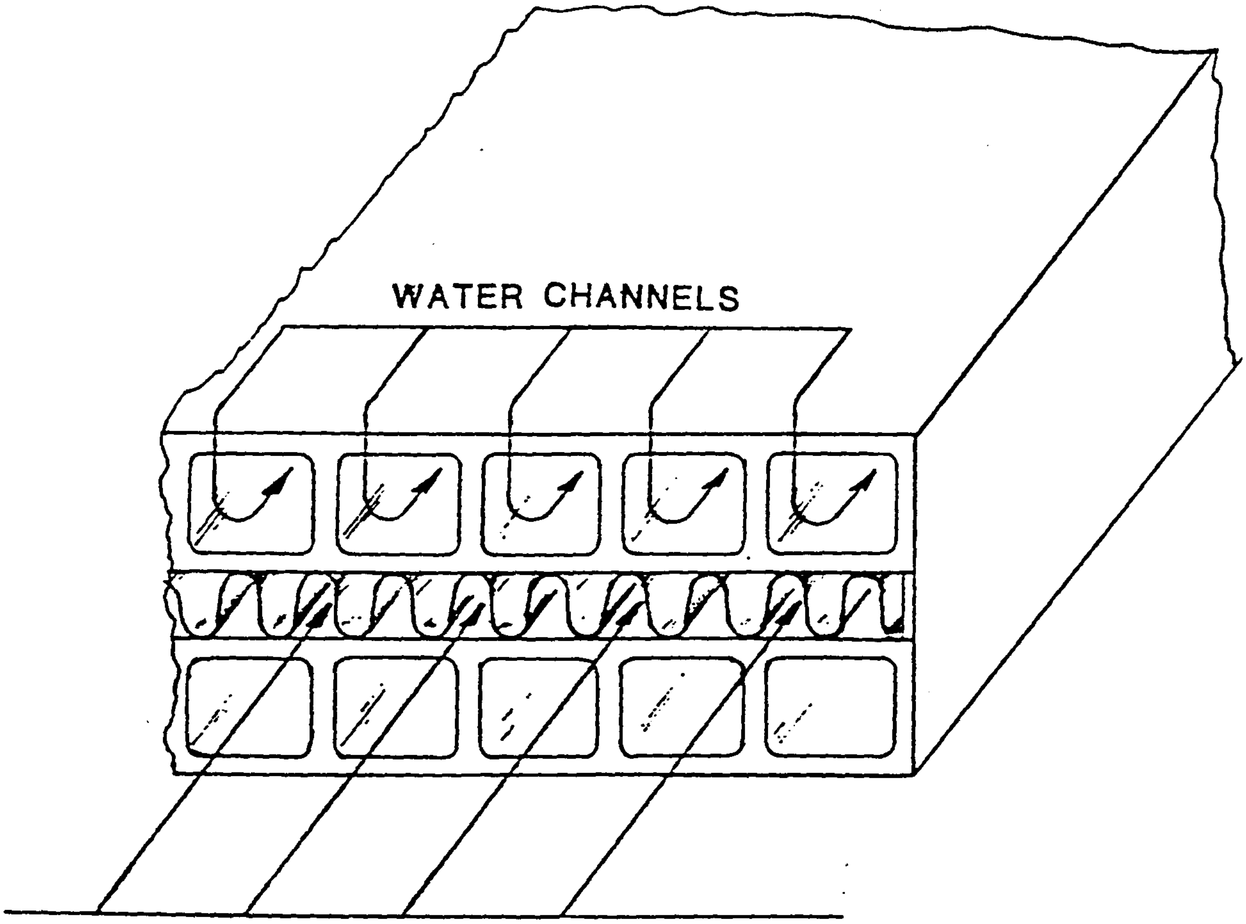

Fig. 2. Elemental section of matrix heat exchanger having straight, perforated fins on working-fluid side and extruded water channels.

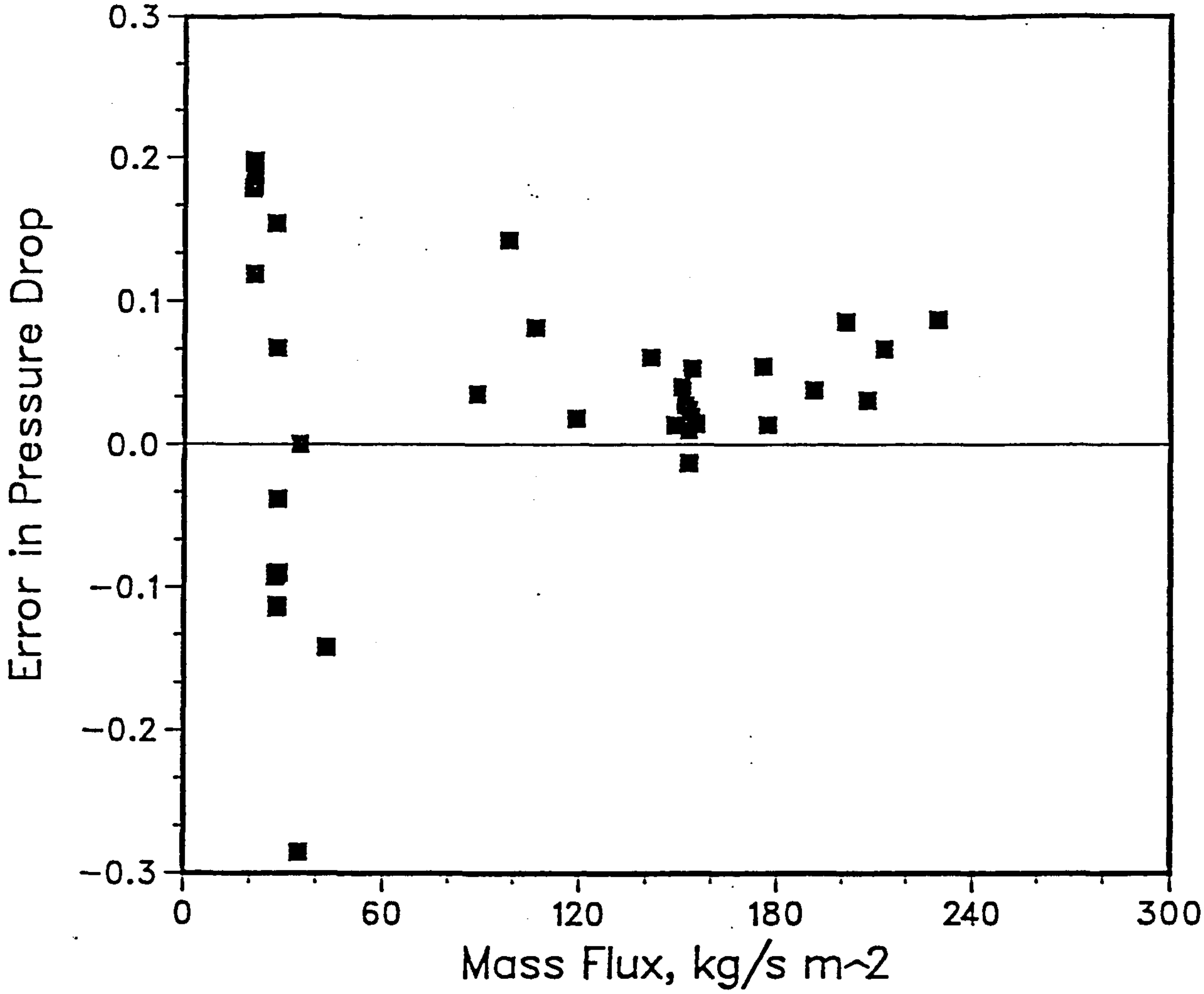

Fig. 3. Fractional error (experimental-prediction/experimental) in total pressure drop as a function of mass flux of ammonia and R-114.

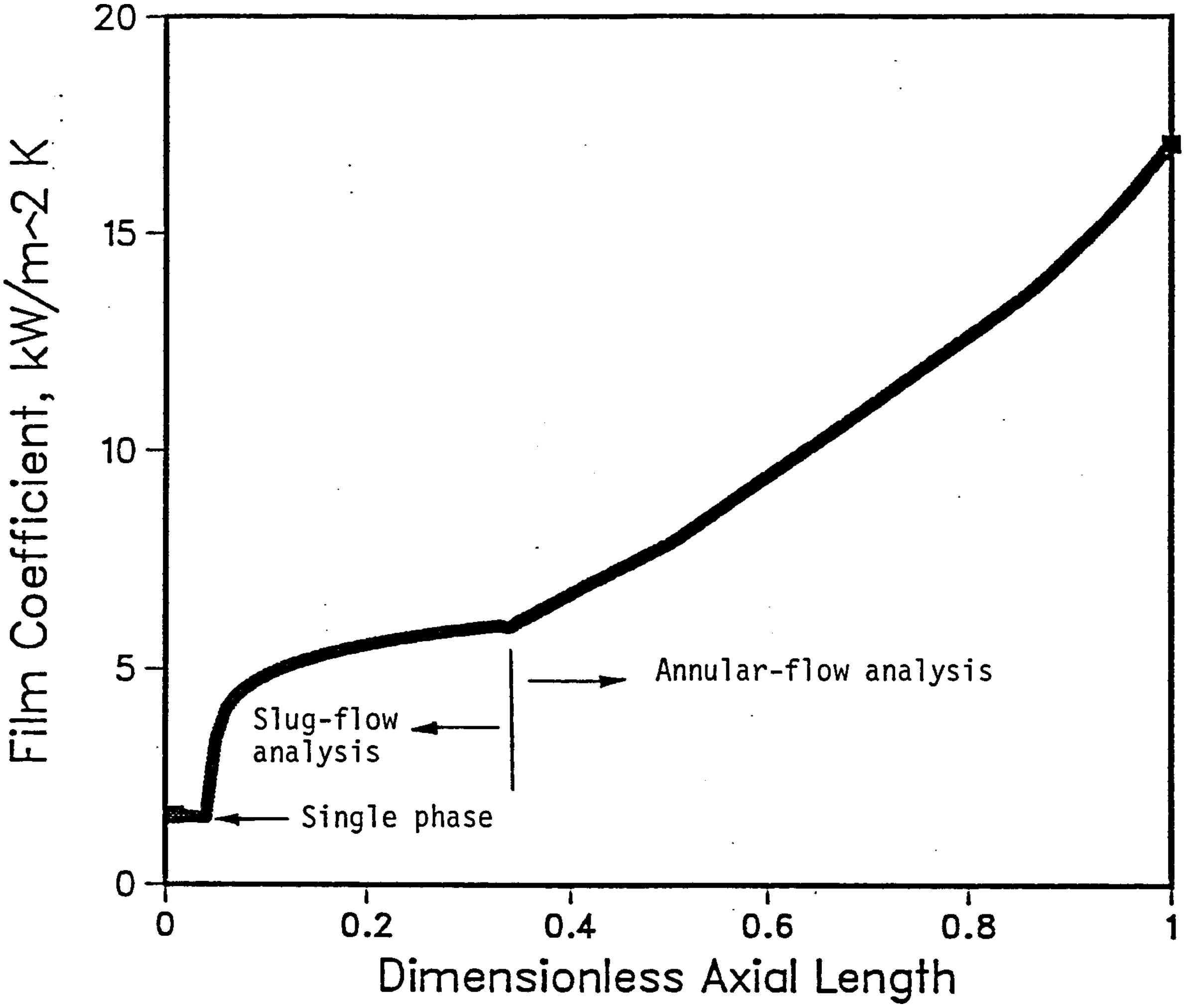

Fig. 4. Ammonia side heat-transfer coefficient as a function of evaporator length for test conditions of $\dot{m} = 27.5$ kg/s m^2 and $\dot{x} = 0.79$.

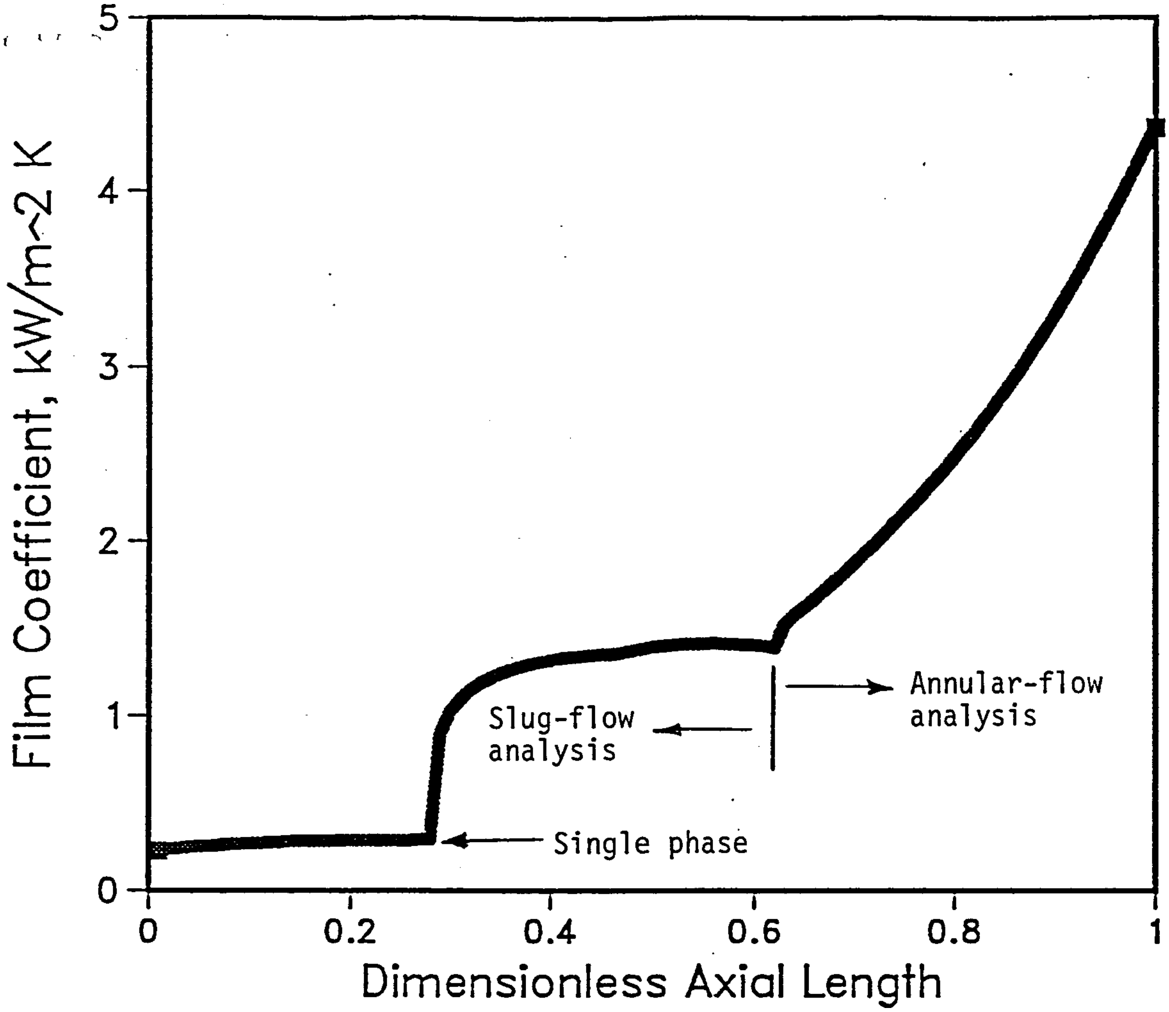

Fig. 5. R-114 side heat transfer coefficient as a function of evaporator length for test conditions of $\dot{m}$ = 153.4 kg/s m^2 and $\dot{x}$ = 0.77.

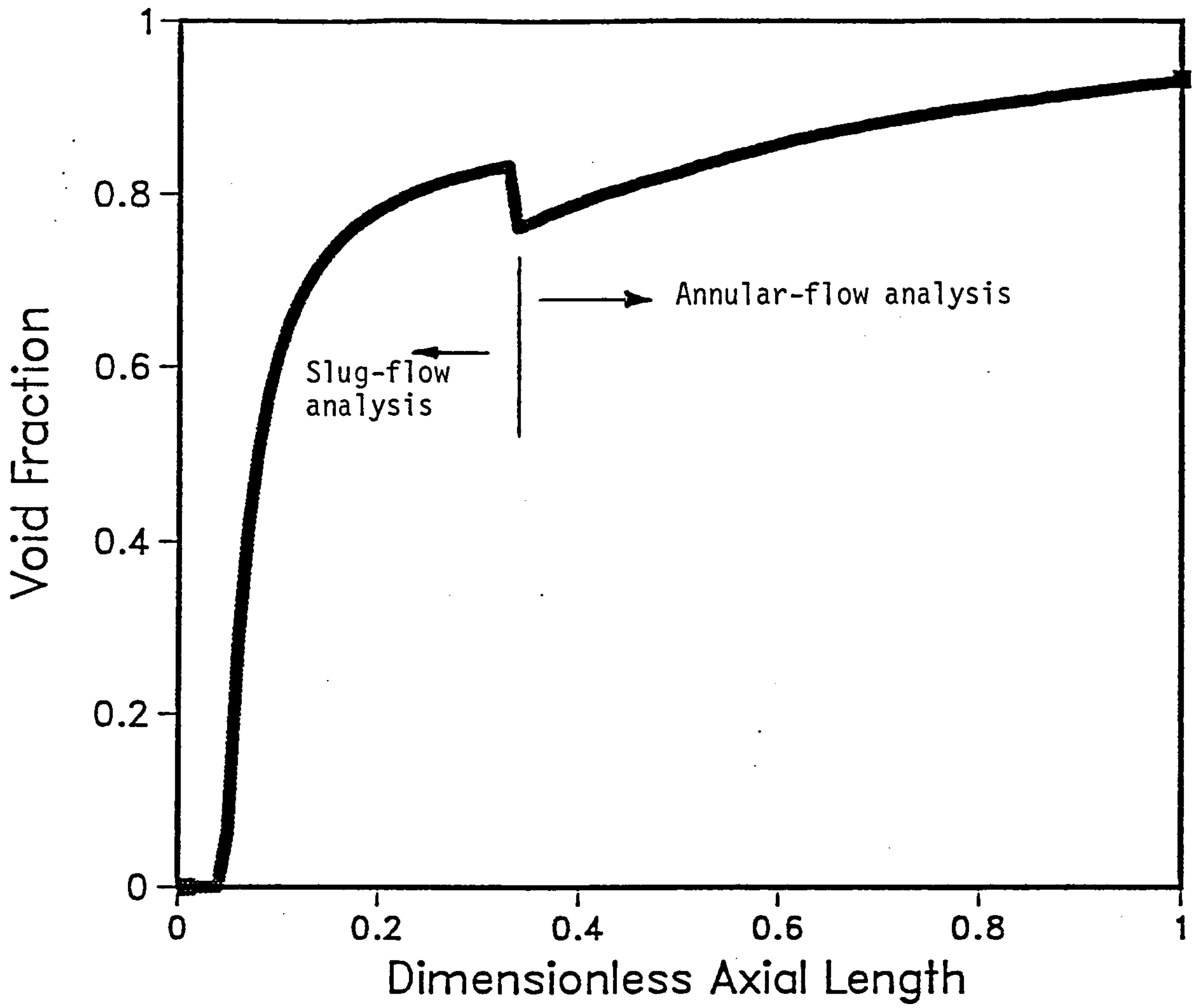

Fig. 6. Ammonia void fraction as a function of evaporator length for test conditions of $m = 27.5$ kg/s m^2 and $\dot{x} = 0.79$.

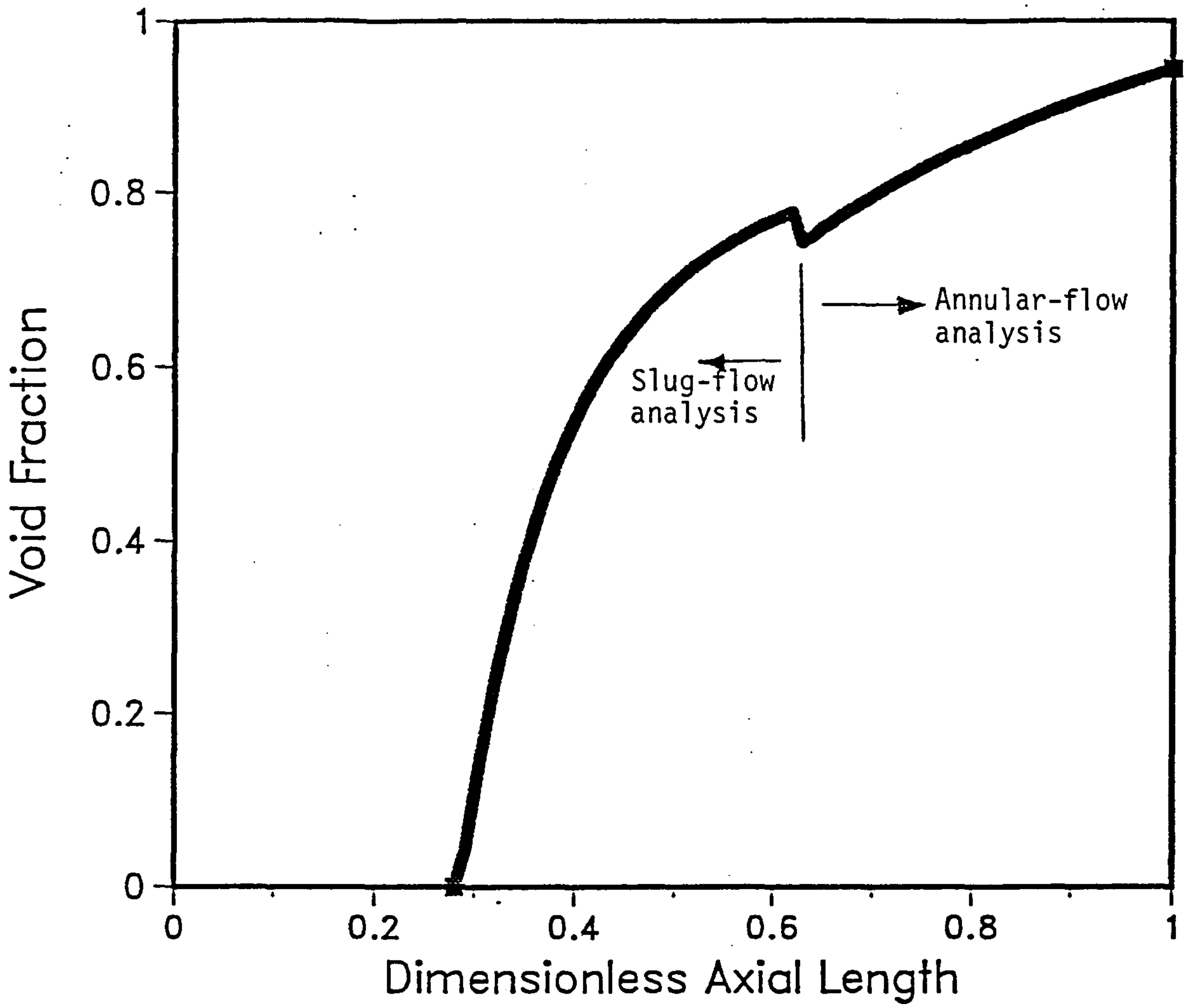

Fig. 7. R-114 void fraction as a function of evaporator length for test conditions of $\dot{m} = 153.4$ kg/s m^2 and $\dot{x} = 0.77$.

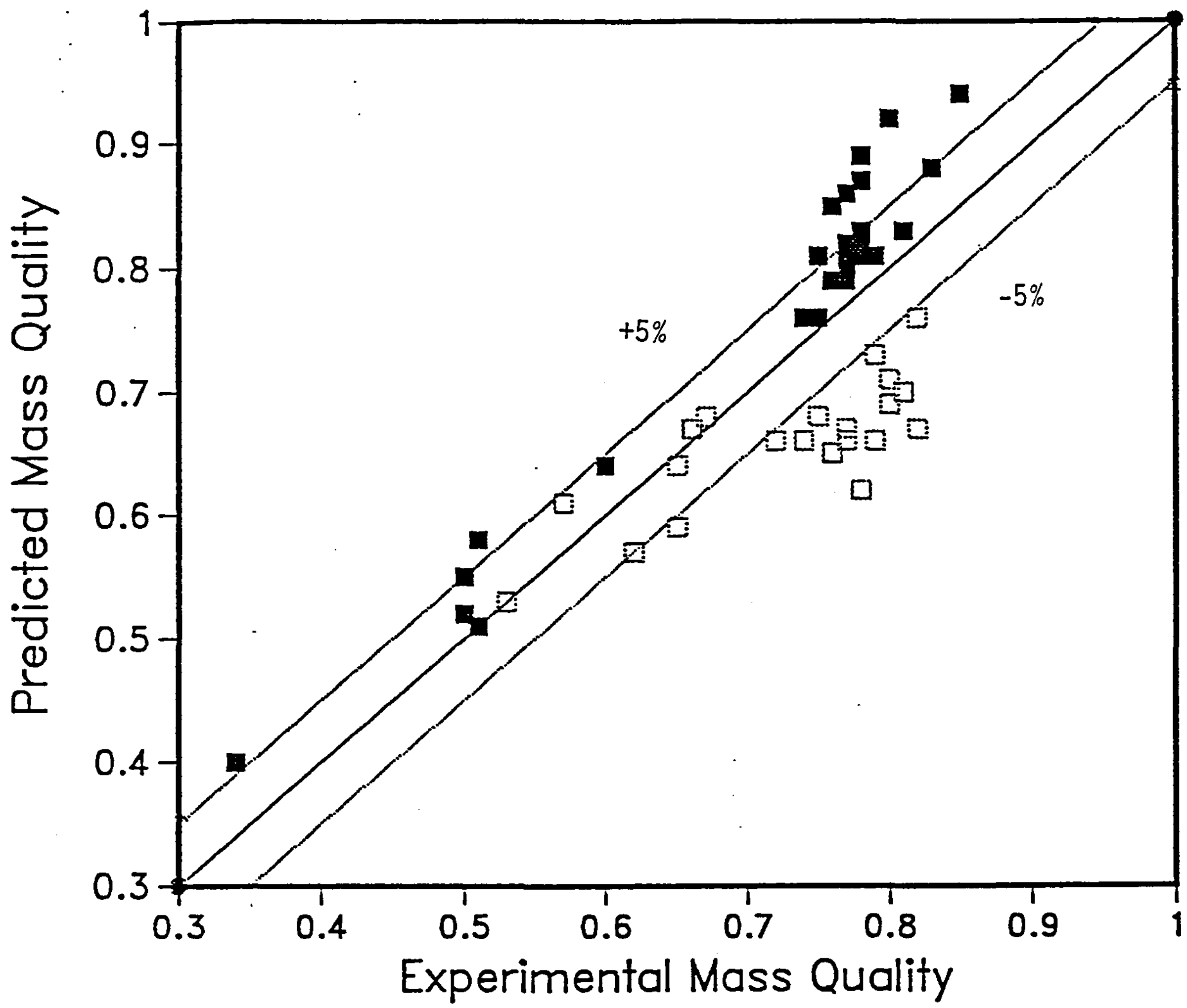

Fig. 8. Comparison of predicted mass quality with experimental results for ammonia (■) and R-114 (□).