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An Analysis on Direct-Contact Condensation in Horizontal Cocurrent Stratified Flow of Steam and Cold Water

Sukho Lee and Hho-Jung Kim
Korea Institute of Nuclear Safety
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동방향 성층이상유동에서의 직접접촉 응축현상에 대한 해석

이석호 · 김효정
한국원자력안전기술원
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Abstract

The physical benchmark problem on the direct-contact condensation under the horizontal cocurrent stratified flow was analyzed using the RELAP5/MOD2 and /MOD3 one-dimensional model. Analysis was performed for the Northwestern experiments, which involved condensing steam/water flow in a rectangular channel. The study showed that the RELAP5 interfacial heat transfer model, under the horizontal stratified flow regime, predicted the condensation rate well though the interfacial heat transfer area was underpredicted. However, some discrepancies in water layer thickness and local heat transfer coefficient with experimental results were found especially when there is a wavy interface, and those were satisfied only within the range.

요 약

동방향 성층이상유동에서의 직접적촉 응축현상을 일차원 모델인 RELAP5/MOD2와 /MOD3를 이용하여 해석하였으며, 해석결과를 Northwestern의 실험결과와 비교 · 검토하였다. 해석결과 RELAP5의 공유열전달 모델은 동방향 성층이상유동에서 응축율을 비교적 잘 예측하고 있다. 그러나 공유접촉면에 파형이 생기는 경우는 물경계두께 및 국부 열전달계수는 유사한 범위로 일치할 뿐 현상을 예측하는데 상당한 차이가 있다.

1. Introduction

The direct-contact condensation effects between liquid and vapor interface become one of the safety issues in nuclear reactor transients[1]. Especially, when the cold emergency core cooling water comes into contact with steam, the interfacial heat and mass transfer domains the transients. The condensation phenomena under various flow regimes can be characterized by the interfacial heat transfer coefficient and the interfacial heat transfer area. It is therefore important to predict them exactly, so there have been intensive studies in those fields through experiments as well as code assessments.

In this study, the condensation effect under the horizontal cocurrent stratified flow is calculated using RELAP5/MOD2 and /MOD3[2,3], and compared with experiment by Lim et al[4]. The junction-based interphase drag, which were incorporated into RELAP5/MOD3, uses the donor void fraction to evaluate the interphase drag rather than using the MOD2 method of averaging the interphase drag from the two volumes on either side of the junction. The void fraction calculations were significantly improved with this modification, although it was just done for specific junctions[5]. The cell-centered phasic velocity to compute the cell-centered phasic mass fluxes was used. The velocity calculational algorithm uses donored void fractions so that the value computed depends upon the void gradient between adjacent volumes, as well as on the void fraction in the volume in which the velocity is being computed.

It is shown in this study that the RELAP5 interfacial heat transfer model, under the horizontal stratified flow regime, predicted the condensation rate well though the interfacial heat transfer area was underpredicted.

2. Experimental Apparatus, Procedure

Measurement[4] of local steam condenstion rates of cocurrent stratified flow of steam and subcooled water was carried out at atmospheric pressure in a horizontal rectangular channel. Fig. 1 shows a schematic diagram of the system. The channel was constructed of 6.4 mm thick stainless steel with pyrex glass windows, and its dimensions are about 1.6 m long, 0.3 m wide, and 0.06 m high.

A data matrix for this experiment is shown in Fig. 2. The steam flow rate varied from 0.04 kg/sec to 0.16 kg/sec. The water flow rate varied

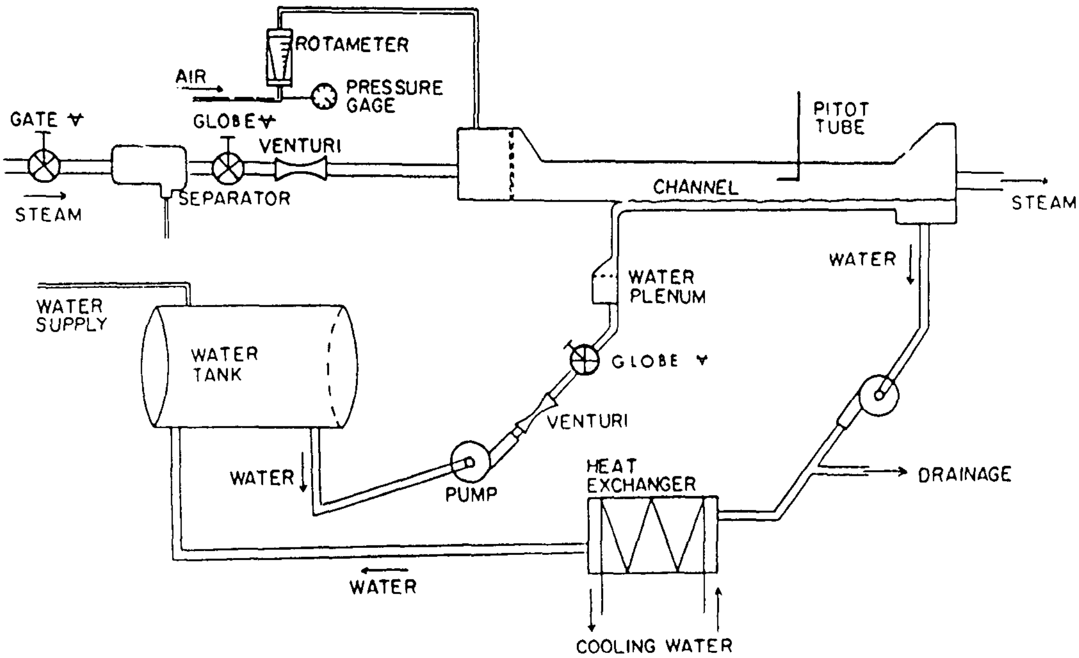

Fig. 1. Schematic Diagram of the System [4]

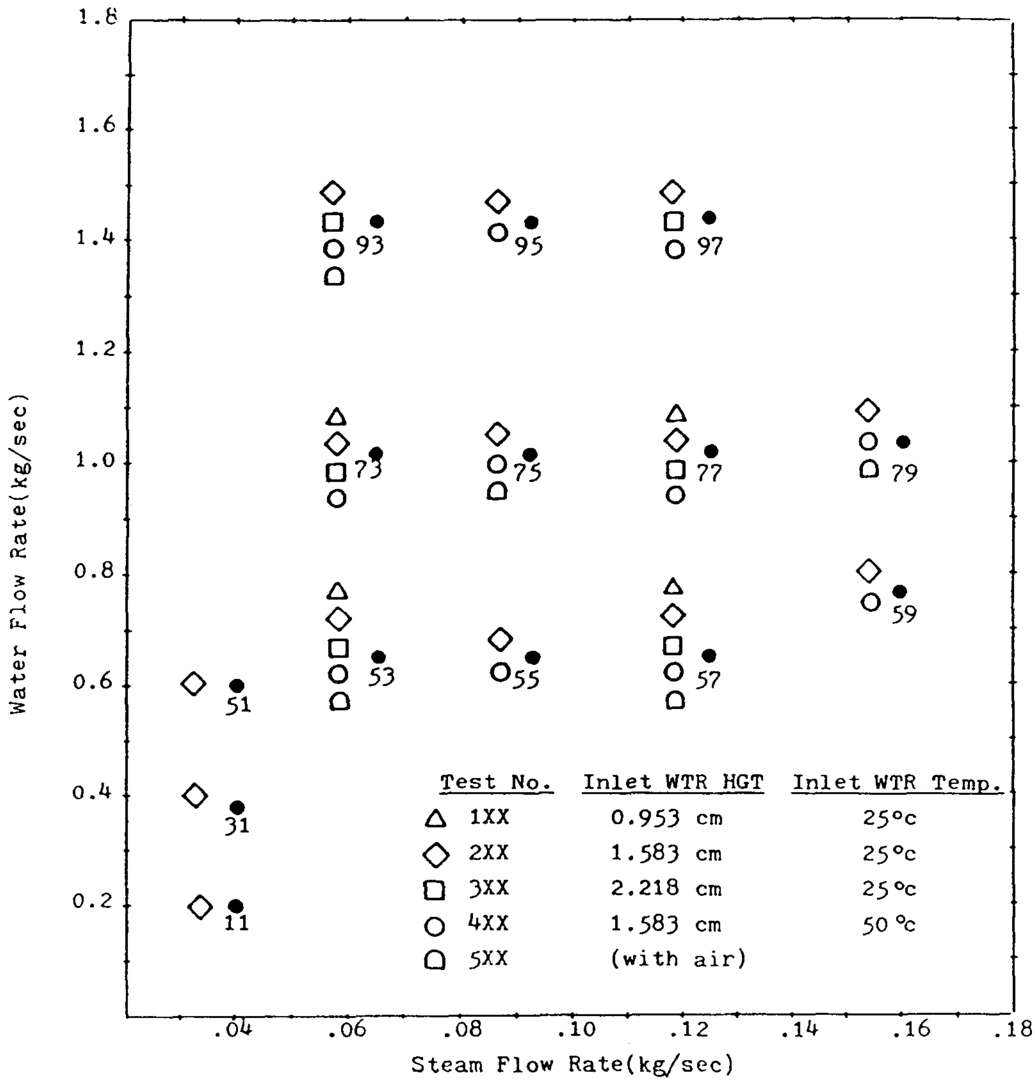

Fig. 2. Experimental Data Matrix [4]

from 0.2 kg/sec to 1.45 kg/sec. The maximum ranges of water and steam flow rates were restricted by either the initiation of bridging phenomena or the occurrence of a hydraulic jump near the entrance region.

The data of fourteen test conditions for liquid layer thickness, vapor flow rate, and the differential pressure were tabulated in Ref. 4 at the 5 locations along the flow path, together with flow rates and liquid temperatures at the inlet and outlet and inlet vapor temperatures.

3. The RELAP5 Model and Calculations

In RELAP5/MOD2 and MOD3, the interfacial mass transfer is modelled according to the thermodynamic process, interphase heat transfer regime and flow regime. After the thermodynamic process is decided, the flow regime map is used to determine the phasic interfacial area and to select the interphase heat transfer correlation [6]. The interfacial heat transfer between the gas and liquid phase phase actually involves both heat and mass transfer. Temperature gradient-driven interfacial heat transfer is computed between each phase and the interface. The temperature at the interface is assigned the saturation value for the local pressure. Heat transfer correlations for each side of the interface are provided in the code. The form used in defining the heat transfer correlations for superheated liquid, subcooled liquid, superheated gas, and subcooled gas is that for a volumetric heat transfer coefficient.

It has been reported in Ref. 3 that the RELAP5/MOD2 overpredicted the void fraction profile in the simulation of the ROSA-IV Two Phase Test Facility (TPTF) and overpredicted the hot leg void fraction in the simulation of ROSA-IV Large Scale Test Facility (LSTF). The TPTF void fraction calculations using RELAP5/MOD3 were significantly improved with this modification, although it was just done for specific junctions. However, it may be expected that this improvement is only for the case where the flow experiences a significant change (e.g., a change in direction, from horizontal to vertical) across the junction.

In RELAP5, the only apparent means of modelling a rectangular channel is by specifying a Pipe component which has one azimuthal segment. Thus it appears that RELAP5 can model the condensing flow parallel to a horizontal vapor-liquid interface only in Pipe component. The flow area of the test section is given equal to the actual flow area and the equivalent diameters of the test section is calculated using the relation between flow area and wetted perimeter.

Since heat transfer coefficients are given in the form of a dimensionless parameter (usually Nusselt number, Nu), the volumetric heat transfer coefficients are coded as follows[2] ;

$$H_i=(k/L)\ Nu\ a_{gf}$$

where

H_i=volumetric interfacial heat transfer coefficient

k=thermal conductivity

L=characteristic length

a_{gf}=interfacial area per unit volume

The volumetric interfacial area, a_{gf}, is based on simple geometric considerations.

For smooth interface, $a_{gf}=4\sin\theta/(\pi D)$, and for wavy interface, $a_{gf}=(4\sin\theta/\pi D))\ F_{27}$.

Where

θ=angle between the vertical and the stratified liquid level

D=pipe diameter

$F_{27}=1+\sqrt{|V_g/V_{crit}|}$ and a multiplicative parameter is applied to a_g^f in the code to attemp to account for an increase in a_{gf} due to a wavy surface. This parameter F_{27} appropriately increases as V_g increases. An evaluation of the validity of function F_{27} requires comparision with experiment.

The expression for Nusselt Number for horizontally stratified subcooled liquid is based on the Dittus-Boelter correlation, and is described as follows :

$$H_{if}=(k_f/D_{hf})(0.023\ R_{ef}^{0.8})a_{gf}$$

where D_{hf} is liquid phase hydraulic diameter, and is defined as

$$D_{hf}=\pi\ \alpha_f\ D/(\pi-\theta+\sin\ \theta)$$
$$Re_f=\rho_f\ D\ |V_f-V_g|\ /\ \mu_f$$

The Reynolds number used for the correlation does not employ the phasic hydraulic diameter, as is the widely accepted practice for this correlation. Also the Nusselt number upon which the expression for Hig for horizontally stratified superheated gas is based has two parts, the first of which is the Dittus-Boelter correlation. The other part upon which Nu is based is simply a large number. Details are described in Ref. 2.

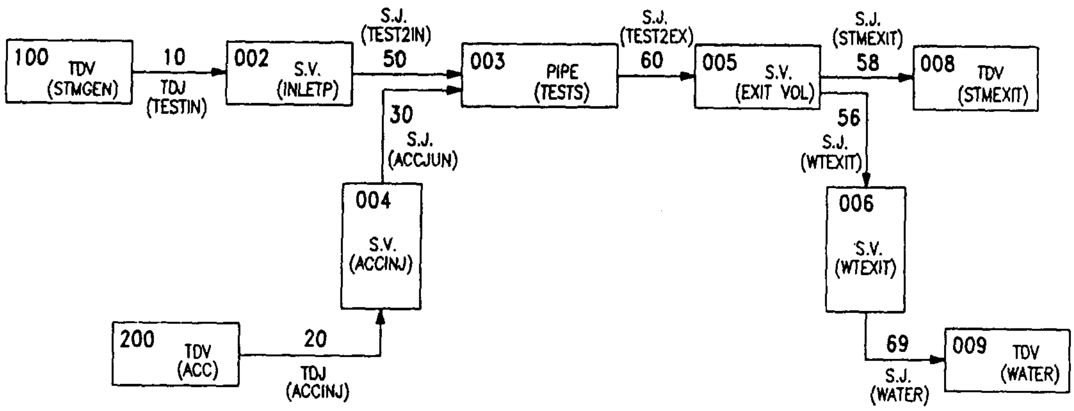

Fig. 3. Nodalization(A) of Cocurrent Condensation Test

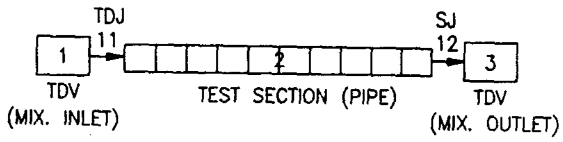

Fig. 4. Nodalization(B) of Cocurrent Condensation Test

It was found to be difficult to obtain steady state conditions, and to prescribe proper outlet conditions even though the calculation reached steady state condition when using a nodalization of test section in Fig. 3. Therefore, a simple nodalization was introduced to describe the experimental outlet conditions as shown in Fig. 4. The main pipe, representing the main test section, consists of 10 cells.

3.1 Calculation without a fixed interfacial area

Under stratified flow condition, the interfacial heat transfer area is calculated from the hydraulic diameter and the void fraction. In this experiment, the cross section of the test section is 0.30 m$\times$ 0.063 m, hence the hydraulic diameter is about 0.105 m. That is, the maximum width of the flow is not 0.30 m but 0.195 m in the calculation. The interfacial heat transfer area calculated by RELAP5 is thus about 3 times smaller than actual interfacial area, which mean that the heat transfer coefficient correlation of RELAP5 overestimates the condensation rate.

3.2 Calculation with a fixed interfacial area

RELAP5 would not give highly accurate quantitative results compared to the experimental data, principally because a pipe has a circular cross section. The geometrical restriction has the obvous consequence that the liquid level and interfacial area are coupled in the calculations. Calculated interphase transfer of mass, momentum, and energy at a point in the flow path are thus more strongly affected by upstream condensation than would be the case with a constant interface area. Thus the interfacial area in the PHAINT Subroutine of the RELAP5/MOD2 was fixed to be the same as the area of the test section instead of being calculated by the RELAP5/MOD2.

4. Calculated Results and Discussions

Four calculations among the experimental data(test no. 253, 259, 279, and 293) were done using RELAP5/MOD2 with and without a fixed interfacial area, and RELAP5/MOD3 without a fixed interfacial area.

The calculated and measured steam flow rate and change in static pressure as a function of axial position for the test no. 253 are shown in Fig. 5

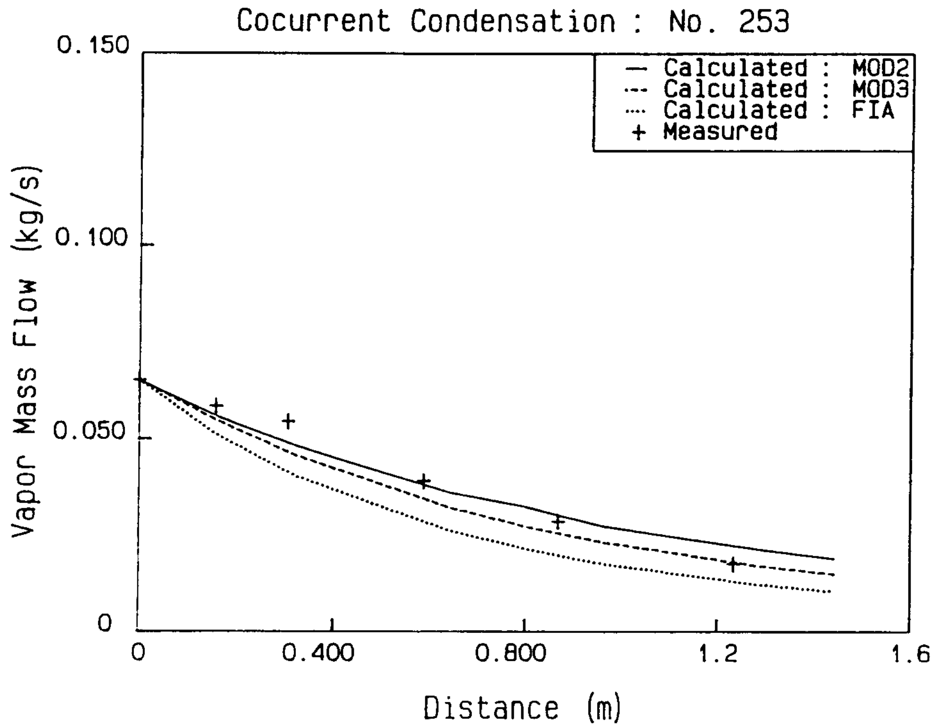

Fig. 5. Vapor Mass Flow (No. 253)

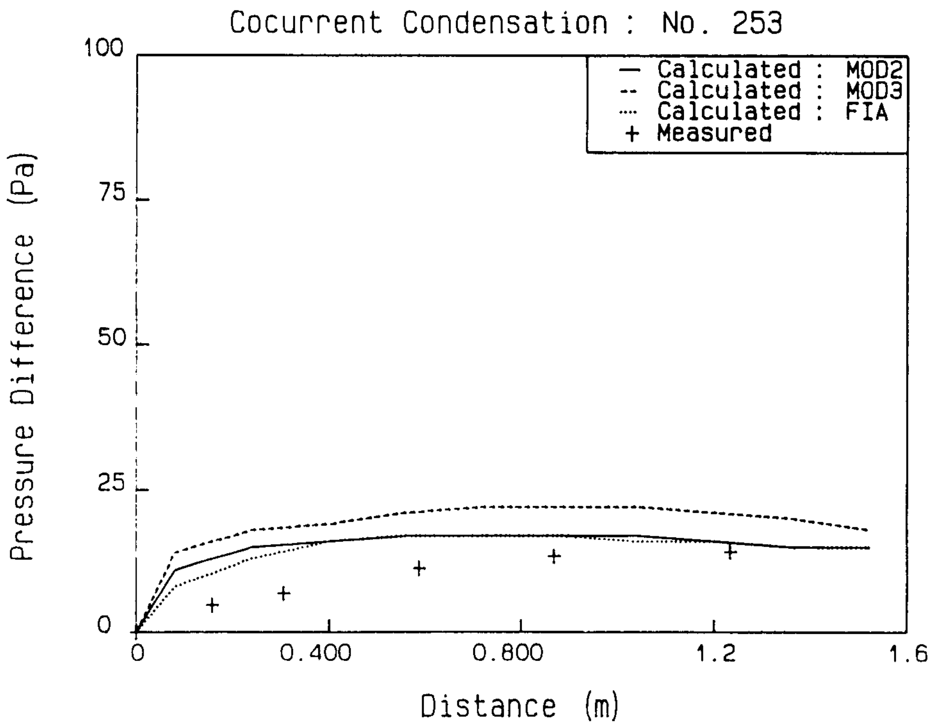

Fig. 6. Pressure Difference (No. 253)

and Fig. 6, respectively. In the figures "FIA" represents the calculation the fixed interfacial area. The decrease in steam flow rate is a measure of the mass exchanged through condensation. Due to large steam condensation, the pressure difference increases with axial distance. Fig. 7 shows the effect of varying the inlet steam flow rate, and by increasing the steam flow rate, the condensation rates increase and heat transfer coefficients increase as shown in the experiment. The experiment indicatied that the thermal resistance in the gas side is negligible compared to the thermal resistance in the liquid side. The condensation rate therefore depends on the ability of the liquid motion to transport thermal energy away from the interface into the liquid main stream. In this case, it is the interfacial wave agitation which enhances the convection. This explains the increase of condensation rates with higher steam flow rates.

The effect of momentum exchange is to retard the steam and accelerate the water. Most of the exchanges occur between the inlet and the first calculating stations. The steam flow rate drops monotonically from their initial value at the inlet ;· the magnitude of the drop increase with the inlet steam flow rate. The calculated and experimental flow rates are in good agreement. And the RELAP5/MOD3 calculations show slighter improvement than the RELAP5/MOD2 comparing with experiment. The principal effects of increasing the inlet steam flow rate are an increase in the level of turbulence creation in the water and an increase in momentum exchange through shear, resulting in a rapid decrease in the water layer thickness.

An increase in the water flow rate leads to an

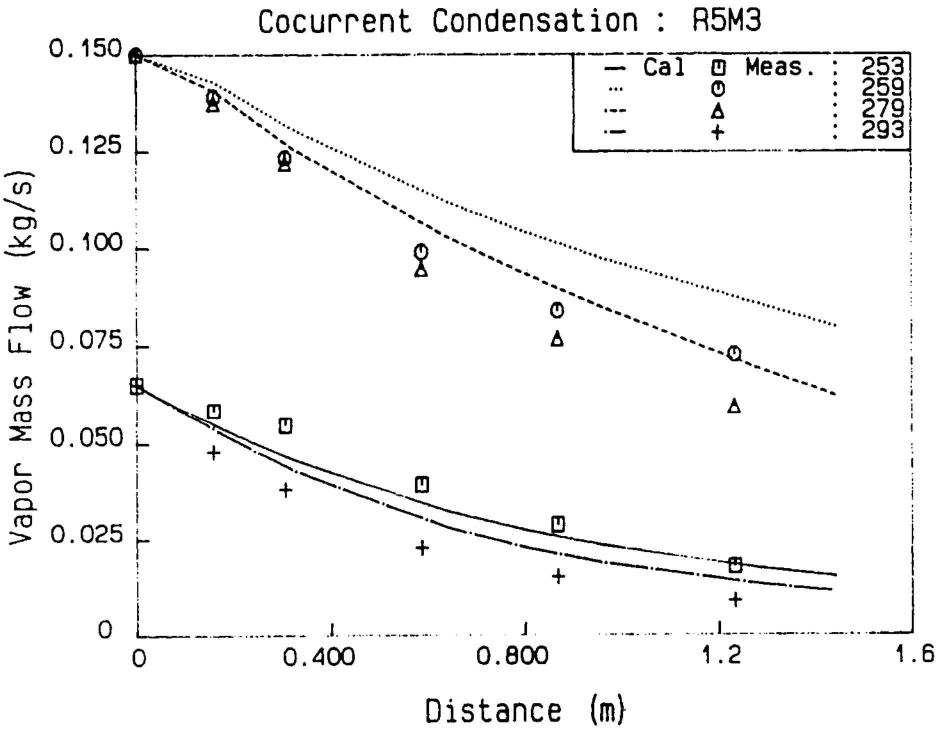

Fig. 7. Vapor Mass Flow (RELAP5/MOD3)

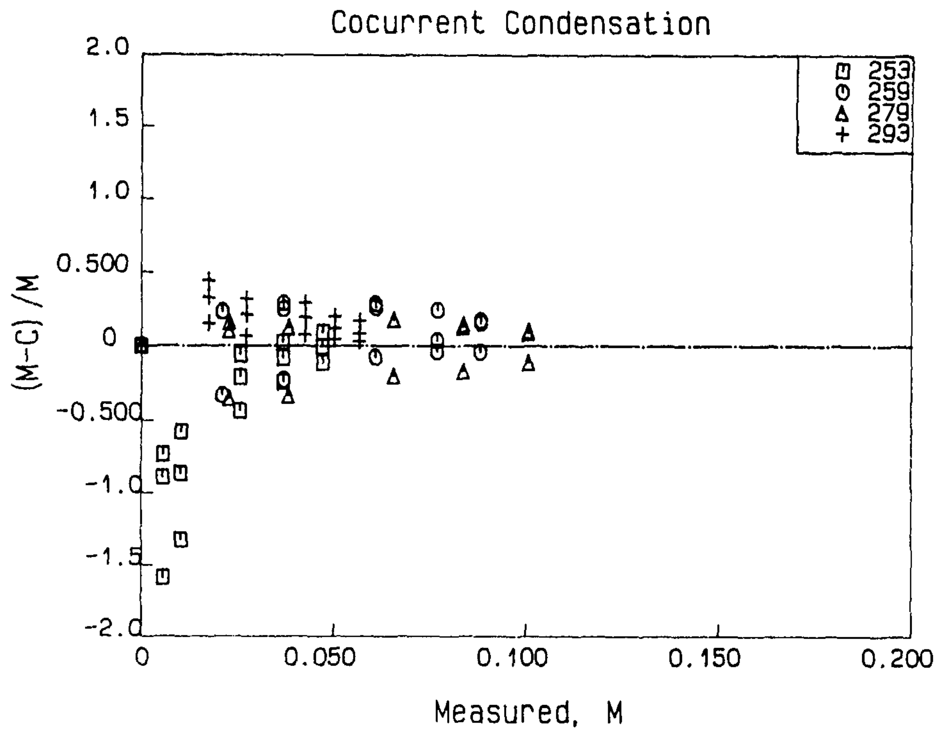

Fig. 8. Prediction of Local Condensation Rate

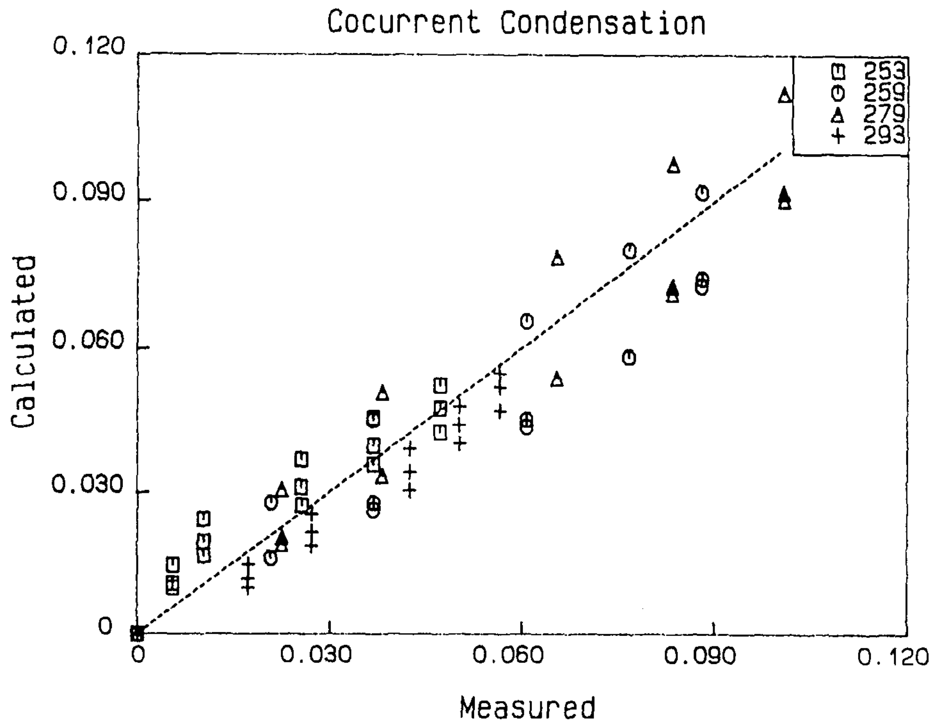

Fig. 9. Comparison of Local Condensation Rate

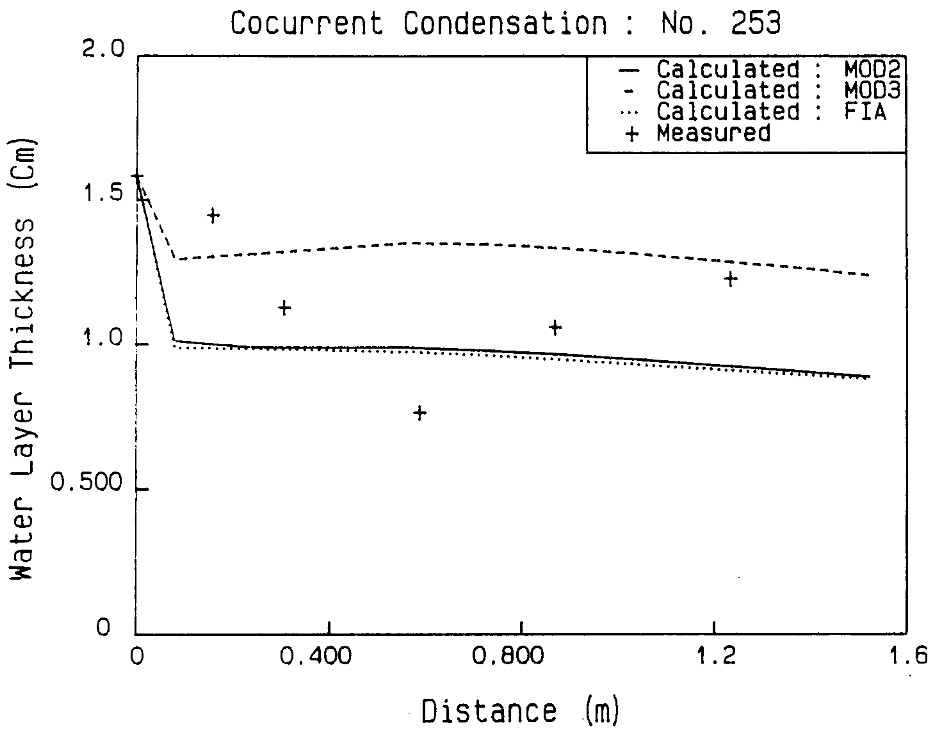

Fig. 10. Water Layer Thickness (No. 253)

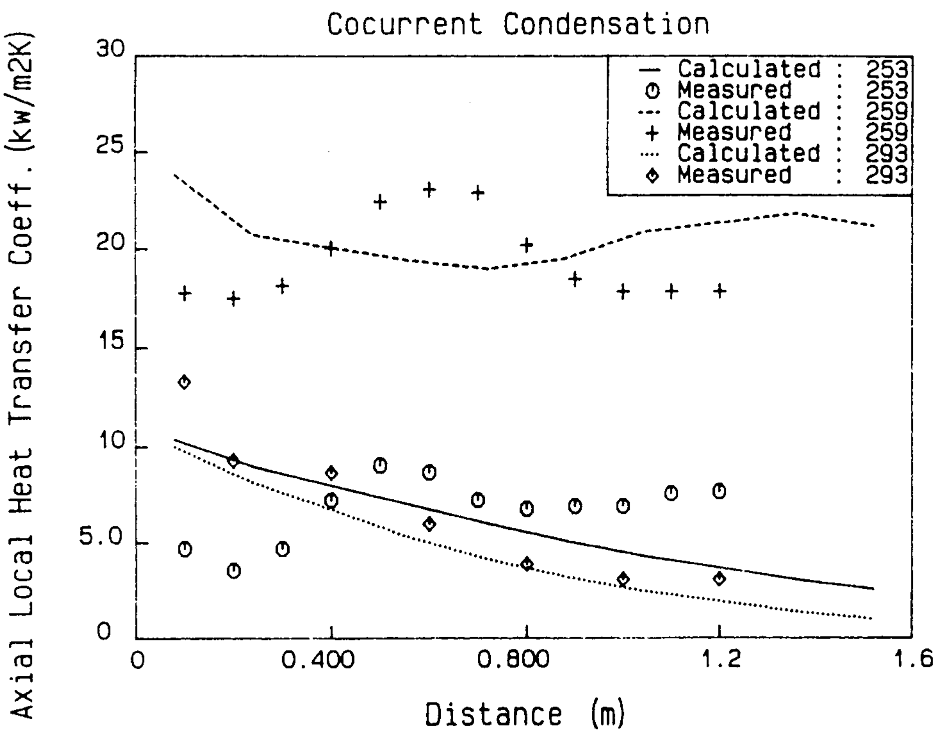

Fig. 11. Axial Local Heat Transfer Coefficient (RELAP5/MOD3)

increase in the condensation rate and therefore to a more rapid decrease in the steam flow rate downstream from the inlet. At the higher water flow rates, water temperature slowly increases due to the larger thermal capacity of the liquid, and therefore the temperature difference remains relatively larger. Because the steam velocity is much larger than water velocity, an increase in water velocity does not increase the interfacial wave agitation rather than the steam velocity. However, an increase in water flow rate may increase the liquid side turbulence intensity in the water and therefore increase the heat transfer coefficient. The condensation rate is higher at high flow rate because the residence time of the water in the channel is less in that case. Thus less heat is transferred to a given volume of water as it flows through the channel, so the subcooling remains high and a large mass exchange rate persists downstream from the inlet.

Comparisons of the calculated condensation rates with the experiment are shown in Fig. 8 and Fig. 9. A relatively large deviatin from the experiment was occurred at very lower condensation rate ; however, the order of the magnitude was quite small and this deviation became smaller for higher condensation rates. It was also observed that the absolute difference between these results were nearly constant along the channel.

The water layer thickness as a function of axial position is shown in Fig. 10. Since it is not possible to predict that with one–dimensional model, there are some discrepancies with experimental results, however, those were not altogether unreasonable.

Similar discrepancies were found in the local heat transfer coefficient as shown in Fig. 11. Visual Observations of the flow pattern in the test [4] showed that the interface between steam and water is characterized by the presence of wavelike disturbances. Different interfacial conditons were observed depending on the inlet steam and water flow velocity and the distance from the entrance region. At very low steam and water flow rates the interface remains smooth along the whole channel, resulting in low heat transfer coefficient with manomatric decrease. At higher flow rates, the smooth interface changes into the disturbed interface along the channel due to instabilities of two–phase flow, and therefore heat transfer is enhanced. The RELAP5 calculations show that the heat transfer coefficient is not predicted well especially when there there exists a wavy interface and is satisfied with experimental results only within the range.

The RERAP5 may not give highly accurate quantitative results in comparison with the experimental data, mainly because of the difference in the pipe geometry used in the code and the experiments. The geometrical restriction has the obvious consequence that the liquid level and interfacial area are coupled in the calculations. In order to examine the effect of the interfacial area, another calculation with a fixed interfacial area, which is the same as the experiment, was performed by implementing that into the PHAINT subroutine in the RELAP5/MOD2. However, the results did not give any improvement on reflecting the effect of the wavy interface accurately.

5. Conclusion

Local condensation rate in horizontal cocurrent stratified flow was predicted using RELAP5/MOD2 and /MOD3. It appears that RELAP5 can model condensing flow parallel to a horizontal vapor–liquid interface only in Pipe component. Therefore the only apparent means of modelling a rectangular channel is by specifying a Pipe component which has one azimuthal segment.

The RELAP5 prediction of the condensation rates is in good agreement with the experiments. However, some discrepancies in water layer thickness and local heat transfer coefficient with experimental results were found especially when there is a wavy interface, and those were satisfied only within the range. Therefore an effort to develope models including the effect of the wavy interface, which was shown in the experiment, might be taken into consideration to enhance the capability of the RELAP5 by considering the interfacial shear as well as the interfacial heat transfer.

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