Experimental Results on Gravity Driven Fully Condensing Flows in Vertical Tubes, their Agreement with Theory, and their Differences with Shear Driven Flows' Boundary-Condition Sensitivities

Jorge Kurita, Michael Kivisalu, Soumya Mitra, Ranjeeth Naik, and Amitabh Narain

Department of Mechanical Engineering-Engineering Mechanics Michigan Technological University Houghton, MI 49931

ABSTRACT

This paper synthesizes experimental results reported here with computational results towards development of a reliable heat transfer correlation for a specific annular condensation flow regime inside a vertical tube. For fully condensing flows of pure vapor (FC-72) inside a vertical cylindrical tube of 6.6 mm diameter and 0.7 m length, the experimental measurements are shown to yield values of average heat transfer co-efficient, and approximate length of full condensation.

The experimental conditions cover: mass flux G over a range of 2.9 kg/m²-s $\leq G \leq 87.7$ kg/m²-s, temperature difference ΔT (saturation temperature at the inlet pressure minus the mean condensing surface temperature) of 5 °C to 45 °C, and cases for which the length of full condensation x_{FC} is in the range of $0 < x_{FC} < 0.7$ m.

The range of flow conditions over which there is good agreement (within 15%) with the theory and its modeling assumptions has been identified. Additionally, the ranges of flow conditions for which there are significant discrepancies (between 15 -30% and greater than 30%) with theory have also been identified.

The paper also refers to a brief set of key experimental results with regard to sensitivity of the flow to timevarying or quasi-steady (i.e. steady in the mean) impositions of pressure at both the inlet and the outlet. The experimental results support the updated theoretical/computational results that gravity dominated condensing flows do not allow such elliptic impositions, whereas, shear driven condensing flows (representative experimental results are also presented) are sensitive to such impositions.

Key Words: Internal Gravity Driven Condensing Flows, Condensation inside Vertical Tubes, Annular Condensing Flows

NOMENCLATURE

C_{p1}	Specific heat of the liquid condensate, J/(kg-K)
D	Inner diameter of the test-section, m
Fr	Froude number U ² /gD
$h_{\rm fg}(p_{\rm in})$	Heat of vaporization at pressure p _{in} , J/kg
ħ	Average heat transfer coefficient, \dot{Q}_{out} / (π ·D·L), W/(m^2 -K)
Ja	Condensate liquid Jakob number, $C_{p1} \Delta T / h_{fg}(p_{in})$
\mathbf{k}_1	Conductivity of condensate liquid, W/(m-K)
L	Length of the test-section, m
\dot{M}_{in}	Vapor flow rate at test-section inlet, g/s
$\dot{M}_{\rm L}$	Liquid flow rate at test-section exit, g/s
'n	Interfacial mass flux, kg/s/m ²
p_{B}	Evaporator (boiler) pressure, kPa
\mathbf{p}_{in}	Pressure at the test-section inlet, kPa
p _{exit}	Pressure at the test-section exit, kPa
Pr_1	Condensate liquid Prandtl number, $\mu_1 \cdot C_{p1} / k_1$
p_{xP-i}	Test-section pressures at locations $x_P = x_{P-i}$ (i = 1, 2,), kPa
$\overline{q''}$	Average convective heat flux, W/m ²
$\dot{Q}_{\mathfrak{b}}$	Net heat rate into the evaporator, W
\dot{Q}_{total}	Net heat rate out of the test-section, W
Re _{in}	Inlet vapor Reynolds number, $\rho_2 UL_c/\mu_2$
T_{B}	Evaporator fluid temperature, °C
Т	Non-dimensional time
$T_{sat}(p)$	Saturation temperature at pressure p, °C
$\overline{T}_{\rm w}$	Mean condensing surface temperature, °C
$T_w(x_{P-i})$	Condensing surface temperatures at different locations $x_P = x_{P-i}$ (i = 1, 2,), °C
$T_{V\text{-}in}$	Vapor temperature at test-section inlet, °C
$\boldsymbol{u}_{\mathrm{f}}$	Non-dimensional value of the x-component of interfacial speed
U	Average Inlet vapor speed, m/s
$\mathbf{x}_{\mathbf{P}}$	Physical distance along the test section, m
X(x)	Ratio of vapor mass flow rate to total mass flow rate at any location x along the test section
ĩ	Non-dimensional x (x/D)
\mathbf{x}_{FC}	Approximate length needed for full condensation (estimted by computations), m
ΔT	$T_{sat}(p) - \overline{T}_W, {}^{o}C$

 $\Delta p \qquad p_{in} - p_{exit}, \, kPa$

- Z(x) Another name for X(x)
- Δ Physical value of condensate thickness, m
- δ Non-dimensional value of condensate thickness
- ρ_2 Density of vapor, kg/m³
- ρ_1 Density of liquid, kg/m³
- μ_2 Viscosity of vapor, kg/(m-s)
- μ_1 Viscosity of liquid, kg/(m-s)
- $\pi(x)$ Non-dimensional pressure p

Subscripts

Exit	Test-section exit
in	Test-section inlet
Na	Natural steady case
Exp	Obtained from experiments
Sim	Obtained from computations/simulations

1. INTRODUCTION AND BACKGROUND

This paper presents a fundamental experimental investigation for gravity driven internal condensing flows inside a vertical tube. Only the pure vapor case (with FC-72 as a working fluid) is considered.

There are quite a few existing experimental papers with condensation in vertical tubes and passages ([1] - [4], etc.) which involve a number of pure fluids. Also there are several heat transfer correlations ([5] - [11], etc.) which have been developed to cover various different realms of internal condensing flow physics (including those involving horizontal and vertical tubes). The experiments as well as the correlations in the literature ([5]-[11]) cover a rather large set of internal condensing flow regimes and associated flow physics. For example, condensing flows in the literature involve: shear driven to gravity driven condensate motion (inside horizontal to vertical tubes and channels), laminar to turbulent nature of the flows in the vapor phase, and laminar (with or without waves) to turbulent (wavy) nature of condensate flows, and annular to various non-annular (plug/slug, bubbly, etc.) liquid/vapor interface-configuration patterns (also termed liquid/vapor morphologies). One of the goals of these types of investigations has been to synthesize analyses and experiments to provide reliable order of magnitude estimates for average heat transfer coefficients over the large set of flow physics conditions associated with condensing flows. In addition to the above, there are also modern condensing flow experiments for flows in μ m-mm scale ducts ([12] - [14]). These newer experiments typically involve shear driven and laminar condensate motion and exhibit even a greater variety of liquid-vapor morphologies (injection annular, plug/slug, etc.) for different steady/quasi-steady and oscillatory realizations of these flows.

In the above context, the goal of the reported experiments is to use a reliable synthesis of experiments with computational/theoretical results obtained for a specific flow physics category and to develop a reliable heat transfer correlation (s) for the chosen category. In this paper, gravity driven flows with laminar wavy and annular condensate flows are investigated. The theoretical/computational results [16] and its synthesis with experimental results reported here is, to begin with, for the flow physics category of laminar condensate and laminar vapor (in the near interface region) flows which are annular with small to moderate waviness. The theoretical/computational results in [16] do not employ any ad hoc models (as in [7], [8], [14]) for interfacial shear stress, pressure gradient, condensate turbulence, etc. Results from this approach, in conjunction with experimental result, is used to define flow physics boundaries. This is done by considering experimental data that agree with the theory in [16] as well as data that systematically deviates from the flow physics assumptions underlying the theory. Therefore this approach identifies nearby flow physics categories for which the agreement is not good and one needs to properly model the interactions between the vapor flow in the "near-interface" region and condensate waviness associated with laminar or turbulent condensate flows. Over the identified regime of disagreement with theory, the "near interface" vapor flow and condensate waviness interactions are strong enough to significantly enhance the heat transfer rates compared to the laminar/laminar (with or without waves) approach [16]. The eventual goal of this type of experiment/theory synthesis is that if one wants more accurate flow prediction tool and/or correlations, one should be able to develop (through appropriate synthesis of experiments and analysis) the theory and requisite correlations for other specific categories of condensing flow physics. The ability to develop this synthesis for laminar/laminar annular flows is reported here and, with the help of more and appropriate experimental data, the way to accomplish the same for more complex nearby flowphysics categories (such as cases involving significant interfacial waviness) is also outlined.

Another objective of the reported investigations is to develop an experimental flow control strategy that helps us understand the role of boundary-condition sensitivities associated with different impositions of quasisteady or time varying pressure-differences across the condenser. The reported experimental investigations have sufficient number of flow controls that allow one to independently vary the inlet mass flow rate \dot{M}_{in} , inlet or exit pressures, and vapor to condensing-surface temperature difference ΔT . The exit condition for the condenser is specified by exit pressure through appropriate flow control strategies. Due to an error in some of our earlier results ([17] - [20]) for gravity driven partial condensation flows inside a vertical tube, we wrongly stated that condensing flows are generally "elliptic" because, for a given set of inlet and wall conditions, one can always impose different *steady* exit conditions have been recently made ([21]), and we find that there exists sensitivity to exit conditions (termed "elliptic sensitivity" in [21]). But the sensitivity is limited only to presence of time-varying or quasi-steady pressure-difference impositions that are different than the *natural steady pressure-difference* ([16] and [21]) for a given quasi-steady mass flow rate and cooling conditions. Furthermore, this sensitivity is limited to shear driven or *mixed* driven (partially shear and partially gravity driven) internal condensing flows. The experimental and theoretical results confirming "elliptic-sensitivity" for shear driven flows (whether they occur in μ m-scale ducts or perfectly horizontal channels or zero-gravity duct flows) are reported separately in [21]. These results are of enormous significance in properly ensuring repeatable realizations of annular/stratified shear driven flows – which happen to be less repeatable and robust than the gravity driven flows reported in this paper. The shear driven flows' elliptic sensitivity results are also important in understanding the greater variety and complexity of shear driven condensing flows' liquid/vapor morphologies - such as those observed in large diameter horizontal tubes (see Carey [22]) and in μ m-scale ducts ([12], [14])). Improper or inadvertent imposition of exit-conditions may also be the cause for various flow transients ([23] - [25]) observed for "mixed" or purely shear driven internal condensing flows in horizontal tubes. Furthermore, as theoretically and experimentally shown in [21], availability of suitable time-periodic fluctuations in flow variables (mass flow rate, inlet/outlet pressures, etc.) is key to enabling multiple pressure-difference impositions that, independent of any impact on flow rates, affects condensate thickness and leads to different quasi-steady shear driven annular flows with large changes (> 20 – 30 %) in heat transfer rates.

It is shown here that gravity driven and gravity *dominated* internal condensing annular flows (see [16] for the definition of this flow) do not allow externally imposed changes in self-selected exit conditions if inlet pressure, inlet mass flow rate, and condensing-surface thermal conditions are specified (i.e. the problem is "parabolic" with no exit-condition sensitivity). Most of the vertical tube cases reported here fall in this gravity driven and gravity *dominated* annular internal condensing flow category.

Since this paper deals only with gravity dominated flows and [21] deals only with purely shear driven flows, we expect that our planned future experiments dealing with mixed driven flows will exhibit elliptic-sensitivity behavior that is intermediate between these two limiting cases.

2. EXPERIMENTAL APPROACH

2.1 Experimental Facility and Setup

The vertical tube condenser test-section, which is shown in Fig. 1, is a part of a closed flow loop shown in Fig. 2. The flow loop is designed for investigation of partial and fully condensing flows although fully condensing flows alone are of interest to this paper. It has three independent feedback control strategies that can fix and steady the values of: inlet mass flow rate \dot{M}_{in} through active feedback control of the heat input to the evaporator/boiler, condensing surface temperature $T_W(x)$ (uniform or non-uniform) through feedback control of one of the controllable displacement pumps in the set up. The flow in Fig. 2 is also especially designed to allow development of procedures under which condensing flows can be allowed to reach steady state while seeking their own self-selected exit conditions (exit quality or pressure). These flows with self-selected exit conditions

are termed 'natural' flows. This closed flow loop also allows development of procedures (see section 6) which are able to tell us, in somewhat different ways than some other open loop experiments (e.g. [12] - [13]), as to whether or not gravity driven (and gravity dominated) condensing flows will accommodate different externally imposed quasi-steady pressure-difference values that are different from the self-selected natural value.

A 0-540 W electric heater wrapped around the evaporator/boiler in Fig. 2 causes the working fluid (FC-72) to evaporate and/or experience nucleate boiling near the heater surface. The vapor mass flow rate out of the evaporator \dot{M}_{in} is fed into the test section. This mass flow rate is measured by a Coriolis flow meter F_1 and, this value is controlled, as needed, by a feedback controlled heating of the evaporator (for this, electric heating is controlled by a solid state relay that undergoes a PID control through Labview that is also sensing the mass flow rate from F_1). Under steady conditions though, the value of \dot{M}_{in} gets approximately fixed by a constant steady electrical heating rate in to the evaporator. This is due to the restriction imposed by the evaporator energy balance, viz., $\dot{M}_{in} \approx \dot{Q}_b/h_{fg}(p_B)$. Here \dot{Q}_b is the net heat rate into the evaporator, p_B is the steady evaporator pressure, T_B is the steady evaporator temperature (which, depending on the nature of the phase-change process in the boiler, may or may not nearly equal the saturation temperature T_{sat} (p_B) of the fluid at the evaporator pressure p_B), and h_{fg} is the heat of vaporization at the mean pressure p_B in the evaporator.

The test-section in Fig. 1 is a 0.8 m long (however the condensing-surface itself is 0.7 m long), vertical, stainless steel (316 SS) tube of 6.6 mm inner diameter, D, and 7.94 mm outer diameter. At the entrance of the test-section, the inlet vapor temperature is denoted as T_{V-in} , the inlet pressure is denoted as p_{in} , and the inlet vapor is kept slightly superheated (i.e. T_{V-in} is 2-10°C above $T_{sat}(p_{in})$). A suitable thermocouple and an absolute pressure transducer respectively measure the temperature, T_{V-in} , and pressure, p_{in} , of the vapor at the inlet. The dynamic view from an axial boroscope, mounted at the top of the test-section shown in Fig. 1, is used to visualize and ascertain the nature of the flow in the test-section. The boroscope videos verified that the quasi-steady flows reported here were indeed wavy annular. The use of this boroscope also allowed us to ascertain and ensure the dryness of the incoming vapor at the entrance to the test-section.

This test-section (see Fig. 1) is suitably instrumented with various sensors (thermocouples, pressure transducers, etc.). The technique used for mounting all the sensors in the test-section is described in detail in Siemonko [26]. An improved method that allows sensors to be replaced or removed more easily is used in the new test section of Fig. 1 ([27]).

The test-section in Fig. 1b (not shown to size relative to the outer tube in Fig. 1a) is centrally aligned in the hollow space of a larger diameter stainless steel (314 SS) tube. This outer tube has an inner diameter of 35 mm and an outer diameter of 42 mm. The test-section tube is cooled by the counter flow of cold water in the annulus formed by the outer surface of the test-section tube and the inner surface of the outer tube. As shown in Fig. 2, this primary flow of coolant water is made possible by a closed loop that contains a heat-exchanger and a pump.

A 3/4 hp centrifugal pump drives water through the loop at a high flow rate (> 26.5 liters/min) to minimize temperature variations along the wall of the test-section. A separate closed loop (not shown in the figure but described in Kurita [27]) has water, as secondary coolant, flowing through the tube-side of the shell-in-tube heat exchanger in the primary loop. This separate loop assures secondary coolant (water) flow at a steady constant temperature which is adjusted with the help of a chiller and a semi-automatically adjusted value of the water flow rate (0 – 17 liters/min) through the tube-side of the heat exchanger in Fig. 2. This arrangement provides for a good control of the steady value of condensing-surface temperatures $T_w(x)$ and temperature T_{C-in} at the coolant inlet location in Fig. 2.

Resistance temperature device (RTDs) and type-T thermocouples measure temperatures at different locations of the test-section (see Fig. 1) and at other flow loop locations marked by points B', B, T', T'', C₁, C₂, D₁, D₂, D₃, P₁', and P' in Fig. 2. Absolute pressure transducers measure pressures at: test-section locations 1, 3, 6, and 9 in Fig. 1 and at locations B, P₁' and D₂ in Fig. 2. A sensitive, variable reluctance differential pressure transducer measures pressure differences in the test-section (in Fig. 1, this is between locations 1 and 6). Two electronically controllable displacement pumps P₁ and P₂ (see Fig. 2) can pump liquid FC-72 at a steady or unsteady specification of volume or mass flow rates. Flow meters at locations marked F₁ (Coriolis meter that directly measures mass flow rate), F₂ (a volume flow rate measuring rotameter), P₁ (volume flow rate meter imbedded in pump P₁), and P₂ (volume flow rate meter imbedded in pump P₂) are corrected as necessary to yield mass flow rates through those locations. Most of the details of the employed data acquisition system are available in Siemionko [26] and Kurita [27]. The purging process ([26] - [27]) ensures that the vapor flowing in the test section is pure over the duration of the experimental run and that non-condensable air in the flow loop has an insignificant presence.

For convenience in discussing the behavior of this flow loop, the system in Fig. 2 is broken into the following sub-systems. (i) Sub-system A is the portion of the flow loop between points P' and T' (this portion contains the flow into the evaporator, the evaporator, the flow meter F_1 , valve V_1 , and the tubing leading the flow into the test-section). (ii) Sub-system B in the test-section portion of the flow loop between points T' and T'' in Fig. 2. (iii) Sub-system C is the portion of the flow loop between points T'' and T'' in Fig. 2. (iii) Sub-system C is the portion of the flow loop between points T'' and P' (this portion consists of the L/V separator used for partially condensing flows, the two branches of the flow in the liquid line and the auxiliary-condenser line, and pumps). (iv) Sub-system D consists of the primary coolant loop shown in Fig. 2 and a portion of the secondary coolant loop (not shown here but described in Kurita [27]). These sub-section names are used, as needed, for later discussions of the experimental results.

The facility reported above can be and has been used for partially condensing flows. However, since a good comparison for gravity driven partially condensing flows experimental data of Lu and Suryanarayana [28] with the theory [16] has already been achieved, they are not considered in this paper. Though, in future, the use of the above experimental facility for investigations of boundary condition sensitivities of partially condensing flows is planned.

2.2 Operating Procedure

Procedure for "Natural/Unspecified" Exit Condition Cases for Fully Condensing Gravity Driven Flows

To realize fully condensing flows in the flow loop schematic of Fig. 2, the auxiliary condenser flowsection is removed (valves V3 and V4 are closed). The liquid completely fills the line from some "point of full condensation" in the test-section all the way to the boiler/evaporator (this means that the liquid fills all the locations marked as T", L/V separator, and the electronically controllable displacement pump P₁). "Natural/selfselected" fully condensing flow cases is achieved in this set-up by two different procedures.

In procedure-1, the inlet mass flow rate \dot{M}_{in} is held fixed by the evaporator heater control, the condensing surface temperature $T_w(x)$ by coolant temperature and flow rate control of the coolant in the secondary loop, and, inlet vapor temperature T_{V-in} is steadied (at a desired value through an electric heating tape on the vapor line feeding into the test-section). The inlet pressure is allowed to seek a natural value of $p_{in} = p_{in}|_{Na}$ as well as the exit pressure is allowed to seek a certain natural value of $p_{exit} = p_{exit}|_{Na}$ by making the exit liquid mass flow rate \dot{M}_{L-e} (through the displacement pump P₁) track the inlet mass flow rate \dot{M}_{in} (measured through Coriolis meter F₁) through the tracking equation $\dot{M}_{L-e|P_1} = \dot{M}_{in}$. The steady values of the inlet and exit pressures that are eventually achieved lead to a well defined self-selected "natural" pressure difference $\Delta p|_{Na} = p_{in}|_{Na} - p_{exit}|_{Na}$. The dryness of the vapor at the inlet is ensured through boroscope visualization and a check that the superheat condition $T_{V-in} > T_{sat}$ (p_{in}) is satisfied.

The same value of $\Delta p|_{Na}$ is also achieved by a second (and preferred) procedure, viz. procedure-2. In this procedure-2, different long term steady values of inlet vapor mass flow rate \dot{M}_{in} are achieved by active evaporator heat control while the displacement pump P₁ holds fixed the inlet pressure $p_{in} = p_{in}^*$ at different set values. This is possible because the pump P₁ is controlled through a Proportional-Integral-Derivative (PID) controller which, through ups and downs in the flow rate passing through P₁ is able to attain a steady $p_{in} = p_{in}^*$ and a steady in the mean flow rate through P₁. These mass flow rate shifts through the pump P₁ eventually become minuscule around the steady continuous mass flow rate value of \dot{M}_{in} through the Coriolis meter F₁. During this operation, inlet temperature T_{V-in} (manipulated through the control of the tape heater on the line between F₁ and the test-section inlet), and the mean wall temperature \overline{T}_W (manipulated through the control of the secondary coolant's flow rate and its chiller controlled temperature) are steadied and held fixed at approximate desirable values. When the flow steadies to some "self selected/natural" steady exit pressure p_{exit}^* , a steady value of $\Delta p = p_{in}^* - p_{exit}^* = \Delta p|_{Na}$ is obtained.

It is found, for the same \dot{M}_{in} and ΔT , Δp values obtained from procedure- 2 is the same as the one obtained through procedure-1. The advantage of procedure-2 is that it allows one to change temperature difference $\Delta T \equiv T_{Sat}(p_{in}) - \overline{T}_W$ by changing pressure p_{in} without any need for changing T_w (x) to achieve different mean temperatures \overline{T}_w .

2.3 Error Analysis, Repeatability, and Flow Morphology for the Experimental Data

The accuracies of all the instruments (except the Coriolis meter F_1) used for reported values of directly measured variables were established after their in-house calibrations with the help of suitable and reliable reference instruments of known resolution and appropriate reference physical conditions (temperature, flow rate, pressure, etc.).

The accuracy of the Coriolis meter was established by the vendor support staff at the time of its installation. The error estimates for the calculated variables reported in Table 1 were obtained by well-known standard procedures (see [29]) described in the Appendix. When experimental results are given in a tabular form (such as Table 1), the accuracies of both measured and calculated variables within a column is taken into account. For variables whose errors are either due to resolution in the measurement or are significantly lower order of magnitude than the variables involved, the maximum value of these errors are reported in the headers of appropriate columns in Table 1. For variables whose errors correlate with the mean values of the variables itself, percentage relative errors (both mean and maximum values) are reported in the relevant column headers.

In addition to error estimates, repeatability of the fully condensing flows for independent runs were established for a few randomly selected flow cases. For brevity, this The repeatability is only shown for a set of fully condensing flows in Figs. 3a - b. Identical results were obtained over three independent time segments shown in Figs. 3a - b. These repeatable condensing flow results also indicate the repeatable performance and control of the pumps, cooling systems, temperature controls, etc. that form the various subsystems shown in Fig. 2. Furthermore, procedure - 1 for fully condensing flows (described in section 2.2) was used for achieving the steady flow in the first time segment of Fig. 3b and procedure - 2 was used for achieving steady flows shown in the second and third time segments of Fig. 3b. Since Δp values in Fig. 3b are the same for all the three steady states (which have approximately the same \dot{M}_{in} and ΔT values), the result experimentally demonstrates the fact that these gravity driven flows are sufficiently stable [20] and are able to self-seek their own exit conditions (like "parabolic" flows) in multiple experimental realizations.

The boroscope video mounted on the top of the test-section indicated that all the gravity driven flows reported and investigated here (see Fig. 4) exhibited an ability to seek and retain robust wavy annular flows.

3. EXPERIMENTAL RESULTS FOR VERTICAL TUBE AND COMPARISONS WITH THEORY

Notations Used for Reporting Experimental Results

Let \dot{M}_{in} be the inlet mass flow rate of the vapor, and ΔT be a fixed representative value of a certain vapor to wall temperature difference profile. Let $\dot{M}_{L}(x)$ and $\dot{M}_{V}(x)$ be the condensate and the vapor mass flow rates at any given location x ($0 \le x \le L$) along the test section length. Let the inlet Reynolds number be given as Rein $\rho_2 UD/\mu_2$ where U is the average speed of vapor at the inlet Ξ (i.e.

 $\dot{M}_{in} \equiv \pi/4 D^2 * \rho_2 U$). Here ρ_2 and μ_2 are, respectively, representative constant values of vapor density and viscosity. Note that all vapor properties are assumed to be approximately uniform and their representative constant values are evaluated at the pressure of p_{in} (the test-section inlet pressure) and a temperature of 1°C superheat over the saturation temperature associated with p_{in} . Thus the Reynolds number represents a specific non-dimensional form of \dot{M}_{in} . The non dimensional temperature difference ΔT is expressed as the ratio of condensate Jacob Number ($Ja \equiv Cp_1 \cdot \Delta T/h_{fg}$) to the condensate Prandtl Number ($Pr_1 = \mu_1 \cdot Cp_1/k_1$) i.e. $Ja/Pr_1 = \Delta T \cdot \mu_1/h_{fg} \cdot k_1$. Here ρ_1 , μ_1 , k_1 , and C_{p1} are, respectively, representative constant values of liquid density, viscosity, thermal conductivity, and specific heat. All liquid properties are evaluated at the pressure of p_{in} and a temperature of " $\overline{T}_W + 0.5 \cdot \Delta T$." Also $h_{fg} \approx h_{fg}(p_{in})$ is the heat of vaporization at a representative pressure that is close to the mean condenser pressure for the reported experimental runs. The gravity vector is expressed in terms of the components g_x - along the test section and g_y - in the transverse direction, where their non dimensional values are $Fr_x^{-1} = g_x \cdot D/U^2$ and $Fr_y^{-1} = g_y \cdot D/U^2$. Since $g_y = 0$ for a vertical tube, $Fr_y^{-1} = 0$. For convenience, we have replaced Fr_x^{-1} by a new non dimensional gravity parameter $G_P = Fr_x^{-1} \cdot Re_{in}^2 = \rho_2^2 \cdot g_x D^3/\mu_2^2$ which is independent of the inlet flow rate. Therefore the non-dimensional parameter set that impacts and characterizes the vertical in-tube condenser flow is: {x/D, Re_in, G_P, Ja/Pr_1, $\rho_2/\rho_1, \mu_2/\mu_1$ }.

Because, for most reported data, the counter flow of cooling water surrounding the test-section is at a fixed flow rate (8 gallons/min) and at nearly fixed temperatures (20-30°C – sufficient to ensure that water properties do not change significantly), the condensing-surface temperature variations $T_W(x)$ is adequately modeled by the assumption of a uniform surface temperature \overline{T}_W . This is because for any two representative experimentally measured wall temperature variations $T_{W1}(x)$ and $T_{W2}(x)$ shown in Fig. 5a, it is found that the variation of non-dimensional condensing-surface temperature $\theta_W(\tilde{x}) (\equiv (T_W(\tilde{x}) - \overline{T}_W)/\Delta T)$ with non-dimensional distance $\tilde{x} \equiv x/D$ is nearly the same function (within experimental errors) for all cases. This means that, in the non-dimensional formulation of the problem ([16]), the thermal condition for the condensing-surface remains the same. Furthermore we verified that the simulation results (based on the methods described in [16]) were nearly the same, with regard to the overall quantities of interest to this paper, for a non-uniform temperature $T_W(x)$ and for the associated uniform temperature \overline{T}_W . The non-monotonous trend of $\theta_W(\tilde{x})$ in Fig. 5a is understandable and its \tilde{x} variations associated with varying flow physics of increasing thickness of a nearly smooth condensate, onset of significant waviness, and eventual single phase nature of the flow.

Experimental Range and Results

The experimental results given in Table 1 for the fully condensing flows show key details of different and representative flow runs. These results were obtained by the flow loop arrangement in Fig. 2 and the procedures 1 and 2 described in section 2.2. The experimental runs reported in Table 1 cover only a portion of the non-dimensional parameter space { \tilde{x} , Re_{in}, G_p $\equiv ((\rho_2^2 g_x D_h^3) / \mu_2^2)$, Ja/Pr₁, ρ_2 / ρ_1 , μ_2 / μ_1 } that governs this intube problem. With distance x from the inlet satisfying the constraints $0 \le x \le x_{FC}$ for full condensation cases

considered here, the remaining non-dimensional parameters cover the zones shown in Figs. 5b - c. The ranges of the non-dimensional numbers involved in Figs. 5b - c are:

$$\begin{array}{l} 0.0025 < \rho_2/\rho_1 < 0.11 \\ 0.015 < \mu_2/\mu_1 < 0.025 \\ 471000 < G_p < 9510000 \\ 0.004 < Ja/Pr_1 < 0.069 \end{array} \tag{1}$$

It should be noted that our ability to cover a limited range of ρ_2/ρ_1 and μ_2/μ_1 values in Fig. 5c results from the fact that $\Delta T \equiv T_{Sat}(p_{in}) - \overline{T}_W$ is systematically changed by changing the inlet pressure p_{in} and the representative vapor properties change when $T_{Sat}(p_{in})$ changes.

It was observed that, for the chosen and reported values of \dot{M}_{in} and $\Delta T \equiv T_{Sat}(p_{in}) - \overline{T}_W$, the flow selfselected an effective point of full condensation whose distance x_{FC} from the inlet was such that $0 < x_{FC} < L$ and the test-section was filled with liquid downstream of the point/zone of full condensation (in the video, unlike the still picture in Fig. 4b, the upper level of liquid that filled the tube was above the green light and was quite visible). Most gravity driven vertical tube flows visually appeared to be annular wavy up to some point x_A close to the point $x_{FC} (> x_A)$ of full condensation. The length $(x_{FC} - x_A)$ of the non-annular zone could not be quantified but appeared small and somewhat dependent on flow conditions. This length x_{FC} is typically estimated (by extrapolation of the computationally obtained [16] values of the 'quality' curve with x – beyond x_A to x_{FC}) to cover points where at least 90 to 100% of the incoming vapor mass flow rate has already condensed. The experimental results given in Table 1 for these "self-selected/natural" fully condensing flows show a hydrostatic component in absolute pressure readings at tranducer loactions below the point of full condensation. For most cases considered here, $x_{FC} < x_9 < L$, and therefore there is always a hydrostatic component of pressure at location - x_9 in Fig. 1. This implies:

$$p_{in} - p_{x9} = (p_{in} - p_{X_{FC}}) + (p_{X_{FC}} - p_{x9}) \cong (p_{in} - p_{X_{FC}}) + \rho g(x_{FC} - x_9) .$$
(2)

Because both the reported experiments (Fig. 6a) and associated theory ([16]) confirm that, for most cases, the pressure differences in the two-phase (mostly) annular region is negligible compared to the hydrostatic pressure (i.e. $|p_{in} - p_{x_{FC}}| \ll |\rho g(x_{FC} - x_9)|$), Eq. (2) simplifies to:

$$\mathbf{p}_{\rm in} - \mathbf{p}_{\rm x9} \cong \rho \mathbf{g} (\mathbf{x}_{\rm FC} - \mathbf{x}_{\rm 9}) \tag{3}$$

Because the left side of Eq. (3) is available from the experimental results, Eq. (3) is used to obtain experimental estimates for x_{FC} , the distance between the inlet and the effective point of full condensation. Furthermore the computational solution approach described in the companion paper [16] gives - under the assumption of nearly smooth interface, steady laminar vapor flow (in the near interface zone) and laminar condensate flow - direct theoretical estimates for x_{FC} and $(p_{in} - p_{x-FC})$ values. Cases for which the prediction tool in [16] does not allow (or is not valid) one to go right up to 100% condensation, the theoretical estimate for x_{FC} is obtained by extrapolating from the theoretically obtained distances for 80 – 90% condensation of the incoming vapor. Although predictions of $|(p_{in} - p_{x-FC})|$ is off, its order of magnitude is correct and, therefore it is found that the right side of Eq. (3) correctly approximates the right side of Eq. (2).

For fully condensing flows, the experimental value of the average heat-transfer coefficient \overline{h} (or \overline{h}_{Exp}) is obtained from Eq. (4) below. For this, the experimental estimates of \dot{M}_{in} , x_{FC} , and $\Delta T \equiv T_{Sat}(p_{in}) - \overline{T}_W$ are used in Eq. (4).

$$\dot{Q}_{out} \approx \dot{M}_{in} h_{fg} = (\pi D \cdot x_{FC} \cdot \bar{h}) \Delta T$$
 (4)

The theoretical value of the average heat-transfer coefficient \bar{h} (or \bar{h}_{Sim}) is obtained by employing the computationally obtained values for the distance x_{FC} in Eq. (4). The resulting value of \bar{h}_{Sim} is found to be equivalent ([16]), or nearly the same, as the one obtained from the relationship:

$$\bar{h} \cong \frac{k_1}{x_{FC}} \cdot \int_0^{x_{FC}} \frac{1}{\Delta(x)} \cdot dx$$
(5)

where the film thickness $\Delta(x) \equiv D \cdot \delta(x)$ and the expression on the right of Eq. (5) is evaluated by the computational solution approach described in [16].

As shown in Figs. 6b-c, whenever the laminar vapor/laminar condensate assumption is adequate, there is an excellent agreement (within 3 - 4%) between theoretical and experimental estimates of x_{FC} . Equivalently, because of Eq. (3) we also have an agreement between experimental and theoretical estimates of $\Delta p \equiv (p_{x9} - p_{in})$. In Figs. 6b-c, the reported plots show the functional dependence structure of $x_{FC} = x_{FC}$ (\dot{M}_{in} , ΔT) as one of the two variables (\dot{M}_{in} or ΔT) is varied while the other is held fixed. Figures 6b-6c show the cases for which the experimentally calculated x_{FC} starts becoming significantly smaller than the theoretically calculated x_{FC} at larger values of \dot{M}_{in} and ΔT . Therefore, Figs. 6b-6c, provide us with a good opportunity to experimentally obtain and develop a quantitative criteria as to when the near interface region of the vapor flow starts interacting strongly with laminar waviness or turbulence of the condensate in the near interface region. A reasonable procedure for quantifications of these effects, that appear to be the most probable cause (given the videos show the annularity of these flows) for theory and experiment to deviate from one another, is described in section 4.

The full condensation runs/cases associated Figs. 5b-c also allows a comparison of theoretically and experimentally obtained values of the average heat-transfer coefficient \bar{h} . These comparisons are shown in Fig. 6d.

4. EFFECTS OF INTERACTIONS BETWEEN NEAR-INTERFACE VAPOR FLOW AND CONDENSATE FLOWS' INTERFACIAL WAVINESS

4.1 Observed Deviations Between Experimental and Theoretical Results

As seen in Figs. 7a-7b, there are significant numbers of experimental data points for fully condensing flows where the experimentally measured values of the heat transfer coefficients are significantly more than the idealized theoretical estimates. For convenience of analysis, we have sorted the data points based on their deviation from the idealized theory and defined the measure as: Deviation $\equiv \frac{\overline{h}_{Exp} - \overline{h}_{Th}}{\overline{h}_{Exp}} \times 100$. The sorting in Fig. 7a is done in "Ja/Pr₁ - Re_{in}" plane and in Fig. 7b in "Ja/Pr₁ - Re_{$\delta|x=x_{FC}$} ($\equiv \frac{4\dot{M}_L(x_{FC})}{\pi Du_1}$)" plane. A pictorial depiction of this sorting, as shown in Figs. 7a-7b, is in the following three groups: (i) experimental values of \overline{h} that are within 15% (represented by filled diamond points), (ii) experimental values of \overline{h} that are between 15% -30% above the correlated or theoretical values (represented by hollow diamond points), and (iii) experimental values of h that are more than 30% above the correlated or theoretical values (represented by the filled circular points). These experimentally obtained data are also the same for which, in Figs. 6a-6b, the indirectly measured experimental values of x_{FC} are smaller than the predicted values of x_{FC} . The fact that the experimental heat transfer rate is higher and the point of full condensation is smaller is understandable. Near the inlet, despite turbulence of the vapor in the core regions, the condensate is thin, laminar, and nearly smooth (except for vibration induced waves). The large value of interfacial mass-flux and associated large bending of vapor stream lines at small distances from the inlet (see, e.g., stream line patterns in [20]) makes the "near interface" vapor flow zone laminar despite turbulence in the core regions. However, for these gravity driven condensate flows, the condensate accelerates and by a certain downstream distance, the laminar condensate motion develops significant instability induced (different from wall noise induced) interfacial waves and, further downstream, the condensate flow becomes turbulent - as is the also the case for the gravity driven Nusselt problem ([5] and [30]). At such downstream distances, despite laminarization of the slowing vapor in the core regions, the near interface vapor flow regions can become turbulent due to the accelerating condensate flow becoming turbulent at x locations where $\text{Re}_{\delta}(x) (\equiv \frac{4\dot{M}_{L}(x)}{\pi D\mu_{1}})$ is sufficiently large. Note that [30] suggests that the order of magnitude of Re_{δ} values where gravity driven laminar condensate becomes wavy is 30 to 100 and where it becomes turbulent is 500 to 1000.

The curves C_1 (and C_2) in Fig. 7a suggests that the critical inlet Reynolds number - which determines whether or not near-interface laminarity or turbulence of the vapor is possible over the length of interest ($0 \le x \le x_{FC}$) - also depends on the temperature-difference parameter Ja/Pr₁ which is known to control the interfacial mass transfer rates (see Eq. (6) in [16]). Similarly curves C_1' (and C_2') in Fig. 7b suggests that the critical condensate Reynolds number $\text{Re}_{\delta|x=x_{FC}} (\equiv \frac{4\dot{M}_L(x_{FC})}{\pi D\mu_1} = \frac{4\dot{M}_{in}}{\pi D\mu_1})$ - which determines at what distance x relative to x_{FC} the laminar condensate becomes significantly wavy and/or significantly turbulent – also depends on the average interfacial mass transfer rate modeled by the parameter Ja/Pr_1 . This dependence of critical values of Re_{in} or $Re_{\delta|x=x_{FC}}$ on interface mass transfer rate (as modeled by Ja/Pr_1) is expected because vapor lines bend and pierce the interface (see, e.g., [19]) with significant reduction in the x- component of the vapor speed in the near interface region. The extent of reduction in the x-component of vapor speed by the time vapor reaches the interface also depends on the value of $Re_{\delta}(x)$ - a measure of the increasing inertia of the condensate.

By reprocessing the results in Figs. 7a-7b in terms of the parameters used for the x-axes in Figs. 8a-8b, we arrive at the following results for FC-72 flows' annular condensation inside vertical tubes:

I. The effects of near-interface vapor turbulence and near-interface condensate waviness are small (in the sense that the laminar/laminar smooth-interface model [16] for these flows are adequate) provided

$$Re_{in} < 60000 \left(\frac{Ja}{Pr_{1}}\right)^{0.2718}$$

and
$$Re_{\delta|x=x_{FC}} < 700 \left(\frac{Ja}{Pr_{1}}\right)^{0.668}$$
(6)

II. The effects of near-interface vapor turbulence and near-interface condensate waviness enhance the average heat transfer coefficient \overline{h} by a factor between 1.15 and 1.3 provided

$$60000 \left(\frac{Ja}{Pr_{1}}\right)^{0.2718} \le Re_{in} \le 88000 \left(\frac{Ja}{Pr_{1}}\right)^{0.2718}$$

and $700 \left(\frac{Ja}{Pr_{1}}\right)^{0.1145} \le Re_{\delta|x=x_{FC}} \le 1250 \left(\frac{Ja}{Pr_{1}}\right)^{0.1145}$ (7)

III. The effects of near-interface vapor turbulence and near-interface condensate waviness enhance the average heat transfer coefficient \overline{h} by a factor greater than 1.3 provided

$$Re_{in} > 88000 \left(\frac{Ja}{Pr_{1}}\right)^{0.2718}$$

and
$$Re_{\delta|x=x_{FC}} > 1250 \left(\frac{Ja}{Pr_{1}}\right)^{0.1145}$$
(8)

4.2 Reasons for Deviations and Suggested Modeling for Wavy and Turbulent Annular Flows

Based on the above results, it is likely that, as and when one has access to experimentally measured "local" values of heat transfer coefficient h_x , or heat flux $q''_w(x)$, then one could use – in conjunction with the existing steady [16] and a full unsteady simulation capability being developed by our group (as in [21]) - a local interfacial vapor Reynolds number $\text{Re}_{VL}(x)$ to characterize local transition from smooth interface steady laminar/laminar conditions to significant interfacial waviness without turbulence as well as significant waviness with turbulence. This number is defined here as $\text{Re}_{VL}(x) \equiv \rho_2 \{\text{Uu}_f(x)\}^2 / \tau^i$ where, as in [16], $\{\text{Uu}_f(x)\}$ and τ^i are, respectively, time-averaged mean of the *physical* values of interfacial speed and shear stress. Because the condensate continues to speed up under gravity, the same number $\text{Re}_{VL}(x)$ can also be used to characterize

subsequent transition to near-interface vapor/liquid turbulence. This parameter $\text{Re}_{VL}(x)$ is also the reciprocal of the much sought after interfacial friction factor $f(x) \equiv \tau^i / \rho_2 \{ Uu_f(x) \}^2$ whose non-dimensional definitions as well as models vary in the literature (see [15]).

From the one-dimensional smooth-interface laminar/laminar theory presented in the companion paper [16], we know that the flow is predictable in terms of three variables viz., non-dimensional interfacial speed $u_f(x)$, non-dimensional film thickness $\delta(x)$, and non-dimensional pressure gradient $d\pi(x)/dx$. This is done in [16] with the help of an appropriate approximate model for friction factor f and, concurrently, a model for the vapor momentum flux (with the interfacial shear from the vapor profile preferably matching the value from the interfacial shear model) through their known explicit functional dependence on the non-dimensional variables: $u_f(x)$, $\delta(x)$, and $d\pi(x)/dx$. The approximate explicit analytical model for f(x) in the 1-D approach [16] is easily obtained from using the thin film, laminar/laminar flow, and negligible inertia assumptions leading to condensate profile given by Eq. (7) of [16]. Utilizing this profile and setting $\tau^i \equiv \rho_2 U^2 \cdot u_f^2(x) \cdot f(x)$ equal to its value in terms of interfacial velocity gradient $\tau^i \equiv \mu_1 U/D \cdot \partial u_1 / \partial y|^i$, one obtains an explicit model for $f(x) \equiv 1/Re_V(x)$. The resulting explicit model/relationship for the friction factor f(x)is: $\mathbf{u}_{f}^{2}(\mathbf{x})\cdot\mathbf{f}(\mathbf{x}) = \left[\mathbf{u}_{f}(\mathbf{x}) - \left\{\left(-d\pi(\mathbf{x})/d\mathbf{x} + \rho_{1}/\rho_{2}\cdot\mathrm{Fr_{x}}^{-1}\right)\cdot\mathrm{Re_{in}}\right\}\delta^{2}(\mathbf{x})\cdot\left(1/2\right)\cdot\left(\mu_{2}/\mu_{1}\right)\right]\cdot\left[\frac{2}{\delta(\mathbf{x})}\cdot\frac{\mu_{1}}{\mu_{2}}\right].$ This functional

dependence for f(x) exhibits the general dependence on the three x-dependent non-dimensional flow variables of $u_f(x)$, $\delta(x)$, and $d\pi/dx$ (x) along with dependence on the non-dimensional constant parameters: ρ_2/ρ_1 , μ_2/μ_1 , Re_{in} , and Fr^{-1}_x . This formulation as well as the model for $f(x) \equiv 1/Re_{VL}(x)$ can also be equivalently obtained in terms of three other suitable variables, such as: non-dimensional interfacial speed $u_f(x)$, non-dimensional film thickness $\delta(x)$, and non-dimensional local condensate Reynolds number $Re_{\delta}(x) \equiv 4\dot{M}_L(x)/\pi D\mu_1$ (where $\dot{M}_L(x)$ is the physical value of the local condensate mass flow rate).

For the gravity *dominated* flow runs (as defined in [16]) that are experimentally considered in Figs. 6 – 7 above, the condensate motion is approximately characterized by the Nusselt solution [5] for a range of inlet flow rates and one can either use the 1-D theory in [16] or use the single degenerate ordinary differential equation in $\delta(x)$ (given in [5]) to obtain the Nusselt solution that yields $\delta(x)$ as well as $u_f(x)$ and $Re_{\delta}(x)$. In this degenerate gravity *dominated* limit, vapor momentum balance is completely independent of the condensate motion (and need not be solved if the interest is in the condensate motion alone) and vapor mass balance yields the variations in the average value of the vapor speed. This degeneracy for gravity dominated flows arise from the fact that interfacial shear τ^i or $u_f^2(x) \cdot f(x)$ is nearly zero relative to wall shear. Yet one can use the full 1-D solution scheme [16] to obtain the still negligible values of $u_f^2(x) \cdot f(x)$ up to a certain $x = x^*$ and this is done and resulting values of a gradually increasing $Re_{VL}(x) = 1/f(x)$ is schematically (i.e. not to scale) shown in Fig. 9. However, downstream of a certain $x = x^*$, finite amplitude instability induced waves develop ([16], [30]). In the zone $x > x^*$, interfacial shear τ^i or $u_f^2(x) \cdot f(x)$ is significantly non-zero and should not be ignored. The value of x^* itself can be obtained by more detailed experiments or by the unsteady computational approaches described in [19] - [21]. Once x^* is obtained, one can still use the full 1-D solution scheme [16] to obtain the still negligible values or by the unsteady computational approaches described in [19] - [21]. Once x^* is obtained, one can still use the full 1-D solution scheme [16] to obtain the still negligible

values of $u_f^2(x^*) \cdot f(x^*)$. If one has to match the negligible shear behavior for $x \approx x^*$ to the wave-induced nonnegligible shear behavior for $x > x^*$, one must discard the Nusselt model for the condensate velocity profile and use the one given by Eq. (7) in [16]. For $x > x^*$, one can set $u_f^2(x) \cdot f(x) = w(x) \cdot u_f^2(x^*) \cdot f(x^*)$ where w(x) > 1 is an empirically/computationally knowable function of x that can be determined to model the wave-enhanced interfacial shear so as to make the predicted mean values of the flow variables (such as mean film thickness and wall heat-flux) agree with their experimentally measured values for $x > x^*$. Finding "w(x)" or "f(x)" this way will be far superior to ad hoc modeling of interfacial shear ([15]).

The knowledge of $u_f^2(x) \cdot f(x)$ from 1-D theory [16] for $x < x^*$ along with its modification $u_f^2(x) \cdot f(x) = w(x) \cdot u_f^2(x^*) \cdot f(x^*)$ for $x > x^*$, can be used to express the pressure gradient $d\pi(x)/dx$ as a function of $u_f(x)$ and $\delta(x)$ (or $\text{Re}_{\delta}(x)$) for $x > x^*$. This suggested modification of the 1-D scheme for $x > x^*$ would only allow two unknown independent variables (say $u_f(x)$ and $\delta(x)$ or $u_f(x)$ and $\text{Re}_{\delta}(x)$) in the modeling and solution for $x > x^*$. This is a natural outcome of using an empirically obtained function w(x) to replace one of the unknowns (say $d\pi(x)/dx$) for $x > x^*$. The suggested formulation and solution (not done here) of the problem for $x > x^*$ in terms of the two chosen variables (say $u_f(x)$ and $\text{Re}_{\delta}(x)$) can be used to show that, as x approaches x^* in Fig. 9, the value of $\text{Re}_{VL}(x)$ approaches a certain critical constant value of $\text{Re}_{VL}|_{Cr-1}$.

The trends of $\text{Re}_{VL}(x)$, $\text{Re}_{\delta}(x)$, $\text{Re}_{in}(x)$, in Fig. 9 for $x < x^*$ is for a specific gravity dominated flow situation and is representative (though not to scale) of the trends obtained by the solution scheme described in [16]. The trends in Fig. 9 for the laminar/laminar wavy zone over $x^* \le x \le x^{**}$ and the turbulent/turbulent wavy zone for $x \ge x^{**}$ are entirely schematic and need to be obtained with the help of sufficient experimental input and the modeling approach suggested in the previous paragraph. In Fig. 9, the suggested trend of the parameter $\text{Re}_{VL}(x)$ for $x < x^*$ denotes nearly smooth interface steady laminar/laminar conditions, $\text{Re}_{VL}(x) > \text{Re}_{VL}|_{\text{Cr-1}}$ for $x > x^*$ denotes significant laminar/laminar waviness in the near interface zone, and $\text{Re}_{VL}(x) > \text{Re}_{VL}|_{\text{Cr-2}}$ for $x > x^{**}$ denotes significant turbulent/turbulent waviness in the near interface zone. The above suggested developments for gravity dominated flows will also yield proper models for the interfacial shear τ^i (or f) and would make the existing models and one dimensional analyses much more reliable outside the currently available [16] laminar/laminar smooth interface zone.

With the above understanding of $\text{Re}_{VL}(x)$, we see that the parameter zone characterized by Eq. (6) above is one for which the ratio x^*/x_{FC} dominates the ratio $(x_{FC} - x^*)/x_{FC}$, the parameter zone characterized by Eq. (7) above is one for which the ratios $(x_{FC} - x^*)/x_{FC}$ and x^*/x_{FC} are comparable in magnitude, and the parameter zone characterized by Eq. (8) above is one for which the ratio $(x_{FC} - x^*)/x_{FC}$ are comparable in magnitude, and the parameter zone characterized by Eq. (8) above is one for which the ratio $(x_{FC} - x^*)/x_{FC}$ starts dominating the ratio x^*/x_{FC} .

It is reiterated that the above described synthesis of theory with experiments require experimental data on distances where transition to waviness and turbulence occur, mean local values of heat-flux in the wavy and turbulent condensate zone, etc. for different working fluids. As and when such a synthesis is achieved, one should be able to replace the numerical multipliers and exponents appearing in Eqs. (6) – (8) by multipliers and exponents that are functions of G_P , ρ_2/ρ_1 , and μ_2/μ_1 . Such a generalization of Eqs. (6) – (8), along with heat transfer co-efficient values obtained from a one dimensional theory (see [16]), will yield more reliable estimates of vertical in-tube condensation heat-transfer coefficients (for different working fluids) in the wavy and turbulent regimes.

5. REMARKS ON COMPARISONS WITH WELL KNOWN CORRELATIONS AND RECOMMENDATIONS

As seen from the theory ([16], [21]) and notations defined in section - 3, the non-dimensional numbers that affect these condensing flows can be represented by the set { $\tilde{x} = x/D$, Re_{in}, G_P, Fr_y⁻¹, Ja/Pr₁, ρ_2/ρ_1 , μ_2/μ_1 }. Well known experimental correlations ([5] - [10], etc.), including those listed in Table 2, often replace the parameter set { $\tilde{x} = x/D$, Re_{in}, Ja/Pr₁} by {Re_δ(x), Re_{in}, X(x)} where the condensate's film Reynolds number is Re_δ(x) = 4 $\dot{M}_L(x)/\pi D\mu_1$ and the local vapor quality is X(x) = Z(x) = $\dot{M}_V(x)/\dot{M}_{in}$. As a result of the above choices, one finds that most of the existing correlations ([4] - [11], etc.), directly or indirectly use the nondimensional parameter set: {X(x), Re_δ(x), Re_{in}, G_P, ρ_2/ρ_1 , μ_2/μ_1 }. One significant drawback of this choice of parameters is that even when a heat transfer correlation provides a reasonable order of magnitude estimate, the correlation itself is not useful for estimating the length of a condenser for a given heat load unless the correlation is supplemented by experimentally measured or analytically/computationally obtained (as is the case here) axial variation in quantities such as X(x), Re_δ(x), etc. This drawback is not present for the explicit correlations (involving x) presented in this and the companion paper [16] for annular wavy gravity driven flows.

In actual practice, for a given fluid, the fluid parameter ρ_2/ρ_1 and μ_2/μ_1 do not affect the flow as much as the other parameters. Of these remaining parameters, the set {Re_{in}, G_P, Re_{\delta}} have a profound impact on the flow physics in the sense that together they determine: (i) certain well defined low to high ranges of Re_{in} for which the core vapor flow is turbulent over a significant length of the flow and it affects the pressure-difference across the two-phase region, (ii) certain well defined low, high, and intermediate ranges of values for G_P (see [16]) decides whether the flow is shear driven, gravity driven, or driven by a combination of shear and gravitational forces, and (iii) certain well defined low to high ranges of Re_{\delta} (also discussed in Eqs. (6) - (8)) determine whether or not the near-interface region of the condensate flow exhibits sufficient waviness (with or without turbulence).

The above discussion suggests that the parameters {Re_{in}, G_P, Re_{δ}} in the header of Table 2 determine, loosely speaking, at least eight distinct pairs of permutations of these three parameters (e.g. {high, low, low}) that define different flow physics groups - each of which require separate experimental and modeling attention. Of these, for purely shear driven flows, one typically does not need to consider high Re_{δ} cases involving condensate turbulence as the condensate flows' Re_{δ} is seldom as high as those for gravity driven flows (which may, see [30], high Re_{δ} in the range of 500-1000).

As seen in Table 2, the popular correlations in the literature ([4] - [11], etc.) are developed over multiple flow physics groups because they attempt an ambitious synthesis of experimental data obtained for flows in vertical tubes, horizontal tubes, and horizontal to inclined channels. Even when an attempt is made (as in [11]) to

classify and propose different correlations for different flow regimes (annular, stratified, plug/slug, etc.), the correlations for annular flow rely heavily on horizontal tube data. Now horizontal tube annular flows require shear forces to dominate the effects of azimuthal component of gravity vector which requires relatively large Re_{in} values (see $J_G > 2.5$ criteria in [11]) in an altogether different {Re_{in}, G_P} space as compared to vertical intube annular wavy flows considered here. Unlike high Re_{in} and low G_p three dimensional flows in the correlations given in [11], the flows considered here involve low Re_{in} and high G_p values that allow two-dimensional annular flows. As a result of the aforementioned reasons, even where the agreement between the experimentally obtained heat transfer coefficients in this paper and those obtained from related theoretical correlations [16] are found to be good, the values obtained from other correlations of Shah [6], Cavallini [7] and [11], Dobson-Chato [8], etc. are not as good (see Fig. 10). The existing correlations can, at best, only provide an order of magnitude estimate of the average heat transfer coefficient. If higher accuracy correlations need to be developed, one must seek correlations for the specific physics based subgroups.

As an example, the experiments and analysis [16] are synthesized here to propose correlations that limit themselves (see the first row of Table 2) to a clear and sharp categorization of the boundaries among shear, gravity and mixed flow regimes within laminar vapor (low Re_{in}) and laminar condensate (low Re_{δ}) flow regimes. Furthermore, as discussed in section-4, this paper also makes some contribution, through our experimental results, to define the boundary of this annular wavy flow regime where an assumption of smallness of wave effects and near-interface "laminarity" of the vapor and the condensate flows can be considered to be adequate.

In Figs. 7a-7b, most of the experimental data, that agree well (i.e. within 15%) with the theoretical model in [16], are found to correspond to gravity dominated flows for which either the Nusselt correlations [5] for high G_P or "near" Nusselt correlations for moderate to high G_P are adequate. As a result, for the data in the range specified by Eq. (1), the following correlations can be used:

$$q_{w}^{''}(x) = h_{x} \Delta T \tag{9}$$

$$Nu_{x} = h_{x}D/k_{1} \cong 1/\delta \tag{10}$$

where, if G_P is large and in the gravity *dominated* zone defined in [16], we have

$$Nu_{x} = \left[4\left(\frac{Ja}{Pr_{l}}\right)\left(\frac{\tilde{x}}{Gp}\right)\right]^{-1/4},$$
(11)

and, if G_P is in the moderate to large G_P zone defined in [16], we have

$$Nu_{x} = \left[\frac{0.748\hat{x}^{0.35}(Ja/Pr_{1})^{0.361}(\rho_{2}/\rho_{1})^{0.238}}{Re_{in}^{0.353}(\mu_{2}/\mu_{1})^{0.595}}\right]^{-1}$$
(12)

For reasons discussed above, unlike the Shah [6], Cavallini [7], Cavallini [11], Dobson-Chato [8], etc correlations in Fig. 10, the specialized correlations given in Eqs. (9) - (12) - and restricted by Eqs. (1) and (6) - are superior for this specific gravity driven annular wavy flow regime (also see Table 2).

6. EXPERIMENTAL RESULTS ON DIFFERENCES BETWEEN GRAVITY AND SHEAR DRIVEN FLOWS' BOUNDARY CONDITION SENSITIVITIES

6.1 Background for Boundary-Condition Sensitivity Results

New and careful theoretical and computational investigations ([16], [21]) show that purely shear driven flows exhibit a certain *elliptic-sensitivity* to unsteady impositions. *Elliptic-sensitivity* of condensing flows means that different independent impositions of pressures with suitable time variations (whether they are steady in the mean or not) is possible at the inlet and the exit while all other conditions (mass flow rate, condensing-surface thermal boundary condition associated with the cooling approach, etc.) remain quasi-steady at the same mean values. For purely shear driven quasi-steady flows, elliptic-sensitivity has been demonstrated (see [21]) for the case of unsteady or steady-in-the-mean pressure-difference impositions (through concurrent control of inlet and exit pressures). As a result of this sensitivity, the flow realizations (including heat transfer rates and condensingsurface temperatures) significantly change in response to different quasi-steady pressure-difference impositions. This paper reports and summarizes our experimental finding that similar elliptic sensitivity does not exist for gravity dominated flows. That is, 'non-natural' elliptic pressure-difference impositions (different from selfselected, or "natural" pressure-difference) for the same inlet mass flow rate and coolant flow conditions could not be imposed for experimental realizations of gravity dominated flows in closed loop systems considered here. Whenever such non-natural pressure-difference is experimentally imposed on a vertical tube condenser, unlike the shear driven case in [21], the flow variables within the test-section do not make direct adjustments but they do so only because the other mechanical boundary condition (namely the quasi-steady mass flow rate) also changes.

For the gravity *dominated* in-tube vertical annular flows considered here (see definition of gravity dominated in [16]), the condensate motion and the associated film thickness (interface location) is completely determined by gravity as they must follow the Nusselt solution [5]. This makes the coupling between liquid motion and vapor motion uni-directional in the sense that the liquid motion dictates and achieves the kind of vapor motion that must be there in order to achieve the Nusselt [5] type behavior for the condensate motion. Under these conditions, computational simulations (of the type reported in [21]) also fail to show that imposition of different quasi-steady exit pressure conditions is possible if the inlet conditions (i.e. mass flow rate \dot{M}_{in} and pressure p_{in} at the inlet of the test section in Fig. 11) as well as the condensing-surface cooling approach are kept steady/quasi-steady.

In the next two sub-sections, we describe the experimental procedure and results for boundary condition sensitivity of gravity dominated flows.

6.2 Procedure for Imposition of "Non-Natural" Pressure-difference for Vertical In-tube Flows

For describing the experimental approach needed to assess the elliptic-sensitivity effect for fully condensing gravity driven flows, the flow loop in Fig. 2 has been slightly modified, and this version is schematically shown in Fig. 11. This set up is similar to the one used for shear driven flows [21], except that, for the reported experimental results, the evaporator does not have the water bath within which the evaporator in Fig. 11 is immersed. The test-section in Fig. 11 is the vertical tube test-section in Fig. 1. In addition, there is an option to use a controllable compressor between the evaporator and the inlet of the test-section. Since the compressor is in a bypass loop, if the mass flow rate \dot{M}_{in} is fixed, the degree to which the compressor contributes energy and adjusts the mean value and fluctuation content of the inlet pressure p_{in} of the vapor as it enters the test-section is adjusted by the control of the compressor's rotational speed (rpm) and through the level of opening/closing of the valve V_{BP} in Fig. 11.

First the self-sought "natural" pressure-difference for fully condensing flows is achieved using a variation of the procedure described in section 2.2. That is, for these fully condensing cases, the valve V in Fig. 11 is closed and the pump P_2 in Fig. 11 is removed and eventually attained steady operating values of \dot{M}_{in} , p_{in} , and $\Delta T \equiv T_{sat}(p_{in}) - \overline{T}_W$ are such that the point of full condensation is within the test-section and the "Collection" Chamber" in Fig. 11 is filled with liquid. The first few steps for the procedure involve: (i) holding fixed the Coriolis mass flow meter F_C reading of the mass flow rate \dot{M}_{in} by manually adjusting and fixing the electrical power supplied to the evaporator heating element, (ii) fixing the inlet pressure $p_{in} = p_{in}^*$ with the help of the PID control of the rpm of the compressor, and (iii) arriving at one well defined steady condensing surface temperature $T_W(x) = T_W(x)|_{Na}$ through control on the primary coolant (water that flows along the outside surface of the internal refrigerant tube in the test section) flow rate and its temperature. (iv) The "natural" exit pressure $p_{exit} = p_{exit}|_{Na}$ is achieved (and sensed) by using the controllable displacement pump P_1 to merely "track" the liquid pumping rate through it and set it equal to Coriolis meter mass flow rate M_{in}. (v) Then a new PID control is implemented on the pumping rate of pump P_1 , using feedback from the pressure sensor which measures p_{exit} , to maintain the same exit pressure ($p_{exit}|_{C-1} = p_{exit}|_{Na}$). (vi) Afterwards, attempts are made to change the set point in the pump P₁'s PID control, to achieve various different "non-natural" ($\neq p_{exit}|_{Na}$) values of exit pressure p_{exit} $(p_{exit}|_{C-2}, p_{exit}|_{C-3}, etc.)$ while holding the inlet pressure at $p_{in} = p_{in}^*$. It is expected that, if quasi-steady non-natural pressure-difference impositions associated with exit pressure specifications p_{exit}|_{C-2}, p_{exit}|_{C-3}, etc. are possible, eventually different quasi-steady states should be reached for the fixed heater adjusted steady value of vapor mass flow rate \dot{M}_{in} (at the Coriolis-meter F_1).

6.3 The Inability to Impose Quasi-steady "Non-natural" Pressure-Differences at a Fixed Quasi-steady Mass Flow Rate for Gravity Dominated Flows.

In the time duration t_1 to t_1 + 20 min (see Figs. 12-13), the test section boundary conditions were constrained as indicated above such that the total test section pressure drop for case C-1 is in the vicinity of a "natural" case for the indicated mass flow rate \dot{M}_{in} , inlet pressure p_{in} , and wall temperature condition \overline{T}_W . Although the Na-1 case itself is not shown in Figs. 12 and 13, many such natural cases have been obtained (including all of the experimental cases for the vertical tube condenser used in Figs. 5 - 6) where it was observed, for representative cases, that long term quasi-steady flow in the test-section is achieved along with a long term quasi-steady flow in the rest of the system. Therefore, although the time duration for case C-1 ($t_1 \le t \le t_1+20$ min) shown in Figs. 12 - 13 is not long enough to assure long term system steadiness outside of the test-section, based on our previous experience, it is expected that test-section ($p_{in}|_{C-1}$, $\overline{T}_W|_{C-1}$, $\dot{M}_{in}|_{C-1}$, etc.) and other system flow variables would remain quasi-steady at their depicted values.

Between time t_1+20 min and time t_2 , the exit pressure (p_{exit}) boundary condition was shifted upward from $p_{exit}|_{C-1}$ to $p_{exit}|_{C-2}$ (see Fig. 12) by changing the set point on the pump P₁'s PID control, which resulted in a temporary slowing down of the liquid pumping rate to allow the level of the liquid in the test section to rise, thereby reducing the length of full condensation (not shown). Due to such an imposition, the inlet pressure (p_{in}) would normally rise to compensate for the adjustment and regain a natural Δp_T ($\equiv \Delta p$); however, the compressor control on p_{in} prevented this from happening (see Figs. 12-13). Instead, between time t_1+20 min and time t_2 , the compressor PID control reduced the speed of the compressor, $\dot{\omega}_C$, relieving pressure at the inlet to keep it at the quasi-steady value of p_{in} shown in Fig. 12.

Case C-2, from time t₂ to time t₂* appears to have achieved a non-natural quasi-steady value of $\Delta p_T|_{C-2} \neq \Delta p_T|_{Na-1}$ flow with other test section variables showing little to negligible shift in their mean values (note small shift down in wall temperature). However, unlike the case C-1 and Na-1, the flow outside of the test-section did not achieve any steady-state and this is shown (in Fig. 13) by the unsteady compressor speed $\dot{\omega}_{C|C-2}$ and the evaporator pressure $p_{evap}|_{C-2}$ over this time duration. An outcome of this unsteadiness is that, at time t_2^* , $p_{evap}|_{C-2}$ has shifted upward (see Fig. 12) enough that the evaporation thermodynamics inside the evaporator has changed to prevent the evaporator from continuing to supply the mass flow rate of $\dot{M}_{in}|_{C-2} \approx \dot{M}_{in}|_{C-1}$; therefore, $\dot{M}_{in}|_{C-2^*}$ reduces between time t_2^* and time t_2^{**} . During this time period, the values of p_{in} and p_{exit} are held fixed in the mean at p_{in} and $p_{exit}|_{C-2}$, respectively, by the compressor and the liquid pump P₁. Note from Fig. 13 that $\dot{\omega}_{C}|_{C-2^*}$ is still going down between times t_2^* and t_2^{**} , but not as quickly as $\dot{\omega}_{C}|_{C-2}$ between times t_2 and t_2^* . Also note that $p_{evap}|_{C-2^*}$ is still unsteady from time t_2^* to time t_2^{**} , although in the mean it is increasing more slowly than $p_{evap}|_{C-2}$ between time t_2 and time t_2^* . In addition, $\overline{T}_W|_{C-2^*}$ changes to achieve a new steady value (nearly equal to $\overline{T}_W|_{C-1}$ at time t_2^{**}) consistent with a changed new value of $\dot{M}_{in}|_{C-2^{**}}$. Some time at or beyond time t_2^{**} , the condensing flow in the test section would achieve a new natural case, case N-2 (not shown in Figs. 12 - 13) for which both the test-section and the system flow variables will be steady.

The key observation and difference from the shear driven case's elliptic-sensitivity [21] is that it was not possible to have $\dot{M}_{in}|_{C-2^{**}} \approx \dot{M}_{in}|_{C-1}$ (at time t_2^{**}) despite the evaporator heater's attempt to do so. In other words, the condensing flow did not readjust to accommodate a different Δp for the same \dot{M}_{in} .

From the example reported above and additional experimentation (not reported here for brevity), the flow loop's response as to which mechanical boundary condition variables are shifted (mass flow rate \dot{M}_{in} , inlet pressure p_{in} , etc.) and which could be held fixed depended on the procedure used and relative strengths of different portions of the flow loop external to the test-section.

An intermediate response for "mixed" driven flows (see definition of "mixed" driven flow in [16]) is expected and will be demonstrated by us when we redo the experiments in [21] for slightly inclined channels.

The results presented here and in [16] and [21] correct some of our misunderstandings from [17]-[18] in which a non-existent control for fixing inlet pressure - combined with a non-functional sensing of the inlet pressure - caused us to erroneously confirm the incorrect conclusion (based on an earlier CFD code result which has since been corrected in [16] and [21]) that steady gravity dominated flows *also* exhibited elliptic-sensitivity.

7. CONCLUSIONS

The results presented in this paper achieved the following:

- Steady gravity driven condensing flows of pure FC-72 considered here were experimentally realized for gravity dominated situations and were found to lead to robust annular/stratified flows.
- The paper establishes a range of conditions for which the flow is adequately modeled (within 15%) by the laminar vapor/laminar condensate smooth-interface theory and associated correlations obtained in the companion paper [16].
- The paper described an approach that allowed demarcation of the flow zones where near-interface vapor turbulence and near-interface condensate waviness interact to significantly enhance the heat transfer rates and reduce the length of full condensation as compared to the theory in the companion paper [16].
- The paper recommends that higher accuracy heat transfer correlations, if desired, must be obtained by analysis and experiments that are specialized for a particular flow physics category of internal condensing flows (eight broad categories have been identified).
- The experimental results on boundary-condition sensitivity show that pure shear driven flows are parabolic with "elliptic-sensitivity" whereas gravity dominated flows are strictly "parabolic".

ACKNOWLEDGEMENT:

Parts of this research were supported by NASA grants NNC04GB52G and NNX10AJ59G and parts through NSF grant NSF-CBET-1033591. The experimental results for vertical tube are taken from the Ph.D thesis of J.Kurita.

REFERENCES

- J.H. Goodykoontz, R.G. Dorsch, Local heat transfer coefficients for condensation of steam vertical down flow within a 5/8-inch diameter tube, NASA TN D-3326, 1966.
- [2] J.H. Goodykoontz, R.G. Dorsch, Local heat transfer coefficients and static pressures for condensation of high-velocity steam within a tube, NASA TN D-3953, 1967.
- [3] F.G. Carpenter, Heat transfer and pressure drop for condensing pure vapors inside vertical tubes at high vapor velocities, Ph. D Thesis, University of Delaware, 1948.
- [4] A. Cavallini, R. Zechchin, High velocity condensation of R-11 vapors inside vertical tubes, Heat Transfer in Refrigeration (1971) pp. 385-396.
- [5] W. Nusselt, Die Oberflächenkondesation des Wasserdampfes, Zeitschrift des Vereines Deutscher Ing. 60 (27) (1916) pp. 541-546.
- [6] M.M. Shah, A General Correlation for Heat Transfer during Film Condensation inside Pipes, Int. J. of Heat and Mass Transfer 22 (1979) pp. 547-556.
- [7] A. Cavallini, J.R. Smith, R. Zechchin, A Dimensionless Correlation for Heat Transfer in Forced Convection Condensation, 6th Int. Heat Transfer Conference, Tokyo, Japan, 1974.
- [8] M.K. Dobson, J.C. Chato, Condensation in Smooth Horizontal Tubes, ASME J. of Heat Transfer (1998) pp. 193-213.
- [9] M. Soliman, J.R. Schuster, P.J. Berenson, A General Heat Transfer Correlation for Annular Flow Condensation, ASME J. of Heat Transfer (1968) pp. 267-276.
- [10] N.Z. Azer, L.V. Abis, T.B. Swearinger, Local Heat Transfer Coefficients during Forced Convection Condensation inside Horizontal Tubes, Trans. ASHRAE 77 (1977) pp. 182-201.
- [11] A. Cavallini, G. Gensi, D. Del Col, L. Doretti, G. A. Longo, L. Rossetto, C. Zilio, Condensation Inside and Outside Smooth and Enhanced Tubes – A Review of Recent Research, International Journal of Refrigeration 26 (2003) pp. 373-392.
- [12] H.Y. Wu, P. Cheng, Condensation Flow Patterns in Microchannels, Int. J. Heat and Mass Transfer 48 (2005) pp. 286-297.
- [13]X.J. Quan, P. Cheng, H.Y. Wu, Transition from Annular to Plug/Slug Flow in Condensation in a Microchannel, Int. J. Heat and Mass Transfer 51 (2008) pp. 707-716.
- [14] J.W. Coleman, S. Garimella, Two-phase Flow Regimes in Round, Square, and Rectangular Tubes during Condensation of Refrigerant R134a, Int. J. Refrig. 26 (2003) pp. 117-128.
- [15] A. Narain, G. Yu, Q. Liu, Interfacial Shear Models and Their Required Asymptotic Form for Annular Film Condensation Flows in Inclined Channels and Vertical Pipes, Int. J. of Heat and Mass Transfer 40 (1997) pp. 3559 – 3575.
- [16] S. Mitra, A. Narain, R. Naik, S.D. Kulkarni, Quasi One-dimensional Theory for Shear and Gravity Driven Condensing Flows and their Agreement with Two-dimensional Theory and Selected Experiments, submitted for publication in the Int. J. of Heat and Mass Transfer, 2011.

- [17] A. Narain, S. D. Kulkarni, S. Mitra, J. H. Kurita, M. Kivisalu, Internal Condensing Flows in Terrestrial and Micro-Gravity Environments - Computational and Ground Based Experimental Investigations of the Effects of Specified and Unspecified (Free) Conditions at Exit, Interdisciplinary Transport Phenomenon, Annals of New York Academy of Sciences 1161 (2009) pp. 321-360.
- [18] A. Narain, J. H. Kurita, M. Kivisalu, S. D. Kulkarni, A. Siemionko, T. W. Ng, N. Kim, L. Phan, Internal Condensing Flows Inside a Vertical Pipe – Experimental/Computational Investigations of the Effects of Specified and Unspecified (Free) Conditions at Exit, ASME Journal of Heat Transfer (2007) pp. 1352-1372.
- [19] A. Narain, Q. Liang, G. Yu, X. Wang, Direct Computational Simulations for Internal Condensing Flows and Results on Attainability/Stability of Steady Solutions, Their Intrinsic Waviness, and Their Noise-Sensitivity, Journal of Applied Mechanics 71 (2004) pp. 69-88.
- [20] L. Phan, X. Wang, A. Narain, Exit Condition, Gravity, and Surface-Tension Effects on Stability and Noise-sensitivity Issues for Steady Condensing Flows inside Tubes and Channels, International Journal of Heat and Mass Transfer 49 Issues 13-14 (2006) pp. 2058-2076.
- [21] S.D. Kulkarni, A. Narain, S. Mitra, M. Kivisalu, M. M. Hasan, Exit-Condition Sensitivity for Shear and Gravity Driven Annular/Stratified Internal Condensing Flows. Accepted for publication in the *International Journal of Transport Processes*, 2010.
- [22] V.P. Carey, Liquid-Vapor Phase-Change Phenomena, Series in Chemical and Mechanical Engineering, Hemisphere Publishing Corporation, 1992.
- [23]G.L. Wedekind, B.L. Bhatt, An Experimental and Theoretical Investigation in to Thermally Governed Transient Flow Surges in Two-Phase Condensing Flow, ASME Journal of Heat Transfer 99 (4) (1977) pp.561-567.
- [24] B.L. Bhatt, G. L.Wedekind, Transient and Frequency Characteristics of Two-Phase Condensing Flows: With and Without Compressibility, ASME Journal of Heat Transfer 102 (3) (1980) pp. 495-500.
- [25] B.L. Bhatt, G.L.Wedekind, A Self Sustained Oscillatory Flow Phenomenon in Two-Phase Condensing Flow Systems, ASME Journal of Heat Transfer 102 (4) (1980b) pp. 695-700.
- [26] A. Siemonko, Design, Fabrication, and Operation of a System to Control FC-72 Condensation Inside a Vertical Tube, Ph.D Thesis, Michigan Technological University, 2006.
- [27] J.H. Kurita, Experimental investigation of fully condensing downward vapor flows in a vertical tube unspecified (free) exit condition cases, Masters Thesis, Michigan Technological University, 2007.
- [28]Q. Lu, N. V. Suryanarayana, Condensation of a vapor flowing inside a horizontal rectangular duct, Journal of Heat Transfer 117 (1995), pp. 418-424.
- [29] P. Holman, Experimental Methods for Engineers, sixth edition, McGraw Hill, 1994.
- [30] Incropera, F. P. and D. P. Dewitt, *Fundamentals of Heat and Mass Transfer*, Sixth Edition, Wiley, 2006.

APPENDIX

The reported error calculations involve three kinds of variables, namely: (i) measured variables x_{Mi} (i = 1 to N0), (ii) handbook variables x_{Hi} (i = 1 to N1) that report values of various fluid properties, and (iii) calculated variables x_{Ci} (i = 1 to N2) that depend on other (measured, handbook, or calculated) variables. For the directly measured variables in Tables 1 to 2, we have N0 = 6 with $\{x_{M1} \equiv \dot{M}_{In}, x_{M2} \equiv p_{in}, x_{M3} \equiv \bar{T}_w, x_{M4} \equiv p_{exit} = p_{x9}, x_{M5} \equiv D$, $x_{M6} \equiv L\}$. The errors Δx_{Mi} have been calculated by in house calibration experiments and/or values reconfirmed by the vendor of the instrument involved. These error values are: $\{\Delta x_{M1} = \pm 0.05 \text{ g/s}, \Delta x_{M2} = \pm 0.15 \text{ kPa}, \Delta x_{M3} = \pm 0.9^{\circ}\text{C}, \Delta x_{M4} = \pm 0.27 \text{ kPa}, \Delta x_{M5} \equiv \pm 0.5 \text{ mm}, \Delta x_{M6} \equiv \pm 0.001 \text{ m}\}$. For the handbook variables in Tables 1 to 2, we have N1 = 7 with $\{x_{H1} \equiv \rho_1, x_{H2} \equiv \rho_2, x_{H3} \equiv \mu_1, x_{H4} \equiv C_{p1}, x_{H5} \equiv k_1, x_{H6} \equiv h_{fg} (p_{ref}), x_{H7} \equiv T_{sat} (p_{ref})\}$. For these variables, a nominal experimental error of 2% is assumed. That is the error $\Delta x_{Hi} = \pm 0.02^* x_{Hi}$ (i = 1 to M1). For the calculated variables (some of them are reported in Table 1), we have N2 = 13 with $\{x_{C1} \equiv T_{sat} (p_{in}|_{measured}), x_{C2} \equiv h_{fg} (p_{in}), x_{C3} \equiv \Delta T, x_{C4} \equiv \Delta p, x_{C5} \equiv \rho_2/\rho_1, x_{C6} \equiv \mu_2/\mu_1, x_{C7} \equiv G, x_{C8} \equiv Re, x_{C9} \equiv Ja/Pr_1, x_{C10} \equiv \dot{Q}_{out}, x_{C11} \equiv \bar{h}|_{Expt}, x_{C12} \equiv \bar{q}'', x_{C13} \equiv x_{FC}|_{Expt}\}$. For all these calculated variables x_{Ci} , the defining equations are such that they can be reassembled to express their dependence only on the measured and handbook variables. As a result, one can express the dependences and errors for x_{Ci} (assuming a random Gaussian variation in errors) as per the well known (Holman [29]) expressions:

$$\mathbf{x}_{Ci} = f_i \left(\mathbf{x}_{M1}, \mathbf{x}_{M2}, \dots, \mathbf{x}_{MN0}, \mathbf{x}_{H1}, \mathbf{x}_{H2}, \dots, \mathbf{x}_{HN1} \right)$$
(A.1)

and

$$\Delta \mathbf{x}_{\mathrm{Ci}} = \sqrt{\sum_{i=1}^{\mathrm{N0}} \left\{ \left(\frac{\partial \mathbf{f}_i}{\partial \mathbf{x}_{\mathrm{Mi}}} \right) \cdot \Delta \mathbf{x}_{\mathrm{Mi}} \right\}^2 + \sum_{i=1}^{\mathrm{N1}} \left\{ \left(\frac{\partial \mathbf{f}_i}{\partial \mathbf{x}_{\mathrm{Hi}}} \right) \cdot \Delta \mathbf{x}_{\mathrm{Hi}} \right\}^2} , \qquad (A.2)$$

where, i = 1 to N2, and each function f_i is known as a single explicit function or a function given by a set of nested explicit functions. Because of these explicit forms of f_i , (unlike situations where f_i itself is approximately known with some statistical uncertainty), their derivatives and their values in Eq. (A.2) are easily calculated for each x_{Ci} for any given run in Table 1.

TABLE CAPTIONS

 Table 1: Representative experimentally measured data and some key calculated and computed variables

 for natural fully condensing steady flows.

 Table 2: Representative coverage of the parameter space under different well known empirical or semiempirical correlations.

TABLES

Table 1:

RınNo	М _{іп}	$\overline{T}w$	ΔT	p _{in}	p _{xP-3}	p _{xP-6}	p _{xP-9}	X _{FC-Exp}	X _{FC-Sim}	ρ_2/ρ_1	μ_2/μ_1	(G	Re		Q out		व		\overline{h}_{Exp}		\overline{h}_{Sim}				
	(g⁄s)	3	(°C)	(kPa)	(kPa)	(kPa)	(kPa)	(m)	(m)			(kg	ím ² s)			(J/s)		(Wm^2)		(₩m ² K)		(W/m^2K)				
								%Error				%Error		%Enor		%Error		%Error		%Error						
Error	±0.05	±0.9	±1.7	±0.15	±0.15	±0.15	±0.27	mean max		±0.0010	±0.0009	mean	max	mean	max	mean	max	mean	max	mean	max					
								±11% ±27%				±34%	±36%	±21%	±25%	±13%	<u>+20%</u>	<u>+22</u> %	±37%	±25%	±34%					
1	0.25	296	12.8	47.67	49.52	52.24	57.41	0.107	0.111	0.0089	0.0164	7.	7.31 4426		26	22.71		22.71		2.71 10278		80	4	775		
2	0.25	296	9.0	40.77	42.01	44.72	49.83	0.149	0.159	0.0033	0.0157	7.	.31	44	4479		22.97		7423		Ø	769				
3	0.25	295	5.6	33.69	34.23	36.25	41.28	0.238	0.259	0.0028	0.0149	7.	.31	4540		23.26		4706		83	6	768				
4	0.25	295	5.0	32.29	32.82	34.25	39.26	0.276	0.293	0.0027	0.0147	7.	.31	4557		23.34		4085		81	9	771				
5	0.25	295	4.5	31.65	32.20	33.47	38.46	0.285	0.326	0.0026	0.0147	7.30 4559		59	23.34 3944		86	7	760							
6	0.25	295	4.3	31.31	31.86	32.85	37.84	0.302	0.339	0.0026	0.0146	7.	7.30 4563		З	23.36		23.36 3725		86	2	770				
7	0.25	295	4.1	30.92	31.48	32.37	37.36	0.308	0.356	0.0026	0.0146	7.	.31	4571		23.40		23.40		23.40		3661		90	в	783
8	1.00	296	10.3	43.09	43.78	43.66	43.14	0.694	0.895	0.0085	0.0160	29	9.22	17836		91.50		91.50		91.50		63	59	61	.7	478
9	1.00	296	11.2	44.82	45.51	45.39	45.06	0.683	0.816	0.0037	0.0161	29	9.23	17798		91.30		91.30		91.30		64	49	57	6	482
10	1.00	296	12.0	46.54	47.23	47.09	47.42	0.644	0.764	0.0038	0.0163	29	29.23 17740		91.03 6816		56	Ð	480							
11	1.00	296	12.8	48.26	48.94	48.80	49.67	0.612	0.711	0.0089	0.0165	29.22 1768		84	90.75		90.75		7150		56	Ø	482			
12	1.00	297.2	16.9	58.61	59.34	59.27	62.38	0.469	0.528	0.0048	0.0174	29	9.25	174	41	89.	50	92	0B	54	3	483				
13	1.00	306.1	17.5	82.74	83.40	59.14	86.62	0.459	0.508	0.0066	0.0204	29	9.26	169	71	86.	90	91	25	52	1	480				
14	1.00	303	42.4	165.48	166.07	168.67	174.33	0.159	0.186	0.0138	0.0246	29	9.25	159	67	80.	53	24	397	57	5	492				

Table	2:
-------	----

Correlations	Lov	v G _P	High	Low	High	Low	High
	Very Low	Moderately Low	G _P	${\sf Re}_{\delta}$	${\sf Re}_{\delta}$	Re _{in}	Re _{in}
Proposals based on this paper							
Cavallini [1974]							
Shah [1979]							
Dobson & Chato[1998]							
Azer et.al [971]							
Travis et.al [1973]							
Soliman et.al [1968]							
Cavallini [2003]							

FIGURE CAPTIONS

Fig. 1: (a) The photograph of condenser test-section. (b) The test-section schematic (diameters in (a) and (b) are not to the same scale). The condensing surface covers the zone $x_0 \le x \le x_{10}$.

Fig. 2: The schematic of the flow loop for achieving unspecified exit condition flows for partial or full condensation cases. For full condensation cases, valve V_3 is closed and the auxiliary condenser is not in use.

Fig. 3a: These time histories of inlet mass flow rate, inlet saturation temperature, and average wall temperature are nearly the same for three separate steady realizations over three separate time intervals. They are termed unspecified ("natural") exit condition cases - namely Natural-1, Natural-2, and Natural-3.

Fig. 3b: Time histories of p_{in} , p_{exit} , and Δp for the three unspecified pressure-difference runs shown in Fig. 3a. The first time-segment's steady flow realization was by procedure-1 and the steady realizations in the second and the third time segments were by procedure-2. Though p_{in} in the first segment is different, the

natural pressure-difference Δp values are seen to be equal for all the three runs because \dot{M}_{in} and $\Delta T \equiv T_{Sat}(p_{in}) - \overline{T}_W$ values are approximately the same for all the three cases.

Fig. 4: The light reflected from the interface depicts typical wavy annular flows realized in vertical tube experiments. The borescope views are taken from the top of the test-section and the blue/green lights are vertically down at a location below the test-section. The blue light picture on the left is for a representative partial condensation case and the green light picture on the right is for a representative full condensation case.

Fig. 5a: The figure shows two different condensing-surface temperature variations $T_{W1}(x)$ and $T_{W2}(x)$ corresponding to runs 12 and 13 in Table 1. It also shows corresponding non-dimensional temperatures $\theta_W(\tilde{x})$ with repect to non-dimensional $\tilde{x} = x/D$.

Fig. 5b: The investigated full and partial condensation flow cases cover the parameter space {Re_{in}, G_p, Ja/Pr₁} above.

Fig. 5c: The investigated full and partial condensation flow cases cover the parameter space $\{\rho_2/\rho_1, \mu_2/\mu_1\}$ space above.

Fig. 6a: For a representative case (run 14 in Table 1), the plot shows test-sectional pressure variation with distance x. The measurements are at locations x_0 , x_3 , x_6 , and x_9 marked in Fig. 1. The value of x_{FC} is marked to indicate its estimated value.

Fig. 6b: The comparison of theoretically and computationally obtained values of x_{FC} as a function of \dot{M}_{in} . Here the the dashed lines indicates the simulation results and the solid lines indicate the experimental results.

Fig. 6c: The comparison of theoretically and computationally obtained values of x_{FC} as a function of ΔT . Here the dashed lines indicates the simulation results and the solid lines indicate the experimental results.

Fig. 6d: Comparison of experimentally and computationally obtained values of average heat transfer coefficients for fully condensing flows.

Fig. 7a: The plot shows deviation levels (between theory and experiments) with respect to the variables Ja/Pr_1 (representing ΔT) and the inlet Reynolds number Re_{in} (representing \dot{M}_{in}). The solid black diamond points show less than 15% deviation from the idealized theory. The hollow diamond points show experimental heat transfer coefficient values higher by 15% - 30% and hollow circular points show that the experimental heat transfer values were greater than 30% with respect to the idealized theoretical estimates.

Fig. 7b: The plot shows deviation levels (between theory and experiments) with respect to the variables Ja/Pr₁ (representing Δ T) against the condensate film's Reynolds number Re_{δ} at x = x_{FC}. The solid black diamond points show less than 15% deviation from the idealized theory. The hollow diamond points show experimental heat transfer coefficient values higher by 15% - 30% and hollow circular points show that the experimental heat transfer values were greater than 30% with respect to the idealized theoretical estimates.

Fig. 8a: Plot of deviation against inlet parameter $\operatorname{Re}_{in}/(\operatorname{Ja/Pr}_1)^{0.2718}$. The solid black diamond points show less than 15% deviation from the idealized theory. The hollow diamond points show experimental heat transfer coefficient values higher by 15% - 30% and the hollow circular points show that the experimental heat transfer values were greater than 30% with respect to the idealized theoretical estimates.

Fig. 8b: Plot of discrepancy against inlet parameter $(\text{Re}_{\delta|x=x_{FC}})/(\text{Ja}/\text{Pr}_1)^{0.1145}$ at $x=x_{FC}$. The solid black diamond points show less than 15% deviation from the idealized theory. The hollow diamond points show experimental heat transfer coefficient values higher by 15% - 30% and the hollow circular points show that the experimental heat transfer values were greater than 30% with respect to the idealized theoretical estimates.

Fig. 9: For modeling heat transfer enhancement greater than 15%, the figure shows the schematic of variations, with distance x, for the near interface vapor turbulence parameter $\text{Re}_{VL}(x)$, vapor flow Reynolds number $\text{Re}_{V}(x)$, and the condensate Reynolds number $\text{Re}_{\delta}(x)$.

Fig. 10: The figure shows a comparison of experimentally obtained heat transfer coefficient with heat transfer coefficients obtained from evaluating different correlations with the help of our simulation tool's predictions [16]. The figure limits itself to assessing the efficacy of various correlations for the gravity dominated flows considered here.

Fig. 11. Flow loop arrangement for investigation of various boundary-condition sensitivities of internal partially or fully condensing flows (gravity or shear driven).

Fig. 12. Representative experimental time histories for test-section and flow-loop pressures demonstrating inability of gravity driven internal condensing flows to achieve steady state (at a fixed inlet mass flow rate) under imposition of "non-natural" pressure-difference values.

Fig. 13. Representative experimental time histories for test-section and flow-loop temperatures, inlet flow rate, and compressor speed demonstrating inability of gravity driven internal condensing flows to achieve steady state (at a fixed inlet mass flow rate) under imposition of "non-natural" pressure-difference values. This data corresponds to the same periods of time over which the data in Fig. 12 were obtained.

FIGURES



📷 : Nylon entrance and exit sections 📻 : Refrigerant tube wall 🔯 : Developed liquid film V: Flowing vapor

Fig. 1



Fig. 2



Fig. 3a



Fig. 3b



Fig. 4



Fig. 5a



Fig. 5b



Fig. 5c



Fig. 6a



Fig. 6b







Fig. 6d







Fig. 7b



Fig. 8a



Fig. 8b



Fig. 9



Fig. 10



Fig. 11



Fig. 12



Fig. 13