Condensation heat transfer coefficients in an inclined smooth tube at low mass fluxes

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ABSTRACT

The purpose of this study was to present the heat transfer coefficients and flow patterns during the condensation of R134a inside an inclined smooth tube at low mass fluxes and different temperature differences (the temperature differences were between the saturation temperature and wall temperature). Condensation experiments were conducted at different inclination angles ranging from −90° (vertically downwards) to +90° (vertically upwards), at low mass fluxes of 50, 75, and 100 kg/m^{2}s, and temperature differences from 1 °C to 10 °C. Measurements were taken at different mean vapour qualities between 0.1 to 0.9 in a smooth tube test section with an internal diameter of 8.38 mm and length of 1.5 m. The average saturation temperature was kept constant at 40 °C. It was found that inclination significantly influenced the flow patterns and the heat transfer coefficients. Downwards flows accounted for an increase in heat transfer coefficient with the maximum heat transfer coefficient found at inclinations of −15° and −30° (downwards flow) at the corresponding minimum temperature difference was tested for in each case. The maximum inclination effect was approximately 60% and was obtained at the lowest mass flux of 50 kg/m^{2}s. In general, it was concluded that the heat transfer coefficients were more sensitive to the temperature difference for downwards flows than for upwards flows. Furthermore, there was no significant effect of temperature difference on the heat transfer coefficients for upwards flows. It was also found that the downwards and upwards vertical orientations were almost independent of the temperature difference. With respect to the inclination effect, it was found that in general, it decreased with an increase in temperature difference but decreased with an increase in mass flux and vapour quality.

Keywords: inclination, condensation, temperature difference, heat transfer coefficient, flow patterns

Highlights

- Measured heat transfer coefficients during condensation at low mass fluxes
- Inclination angle effects on heat transfer coefficients
- Temperature difference effects on heat transfer coefficients
- Optimum angle and temperature difference effects at low mass fluxes

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Nomenclature

EB	energy balance, %
G	mass flux, kg/m^2s
I	inclination effect
Q	heat transfer rate, W
T	temperature, °C
x	vapour quality

Greek symbols

α	heat transfer coefficient, W/m^2K
β	inclination angle (>0: upward, <0 downward) (rad)

Subscripts

m	mean
max	maximum
min	minimum
sat	saturation
w	water

1. Introduction

1.1 Background

Inclination is an option when designing condenser tubes that cannot be oriented horizontally or vertically owing to space constraints, operating conditions, performance optimization, or environmental conditions [1-16]. Examples in which condensation occurs in inclined tubes include steam condensers used for air-cooling, certain rooftop industrial air-cooled refrigeration systems, and in the condensers of motor vehicles and trucks driving up and down hills. However, little work [1, 12] has been published in the open literature which justifies the angles being used or which gives performance data at different inclination angles.

An example in which the environmental and space conditions are important factors is in dry regions of the world that lack large water resources for power-plant cooling. In such cases, large forced-convection, air-cooled power plant condensers are used. The condensers are normally constructed in an 'A' or 'V' frame configuration with the condensing steam in a downwards flow direction of approximately −60°. At least three countries (South Africa, Australia, and the U.S.) are currently increasingly using this technology. Some of the largest dry-cooling plants at present are found in South Africa, with an installed capacity of more than 10 GW. The typical water consumption of a dry-cooling plant is approximately 0.1 ℓ of water per kilowatt-hour of electricity produced. In comparison, a traditional wet-cooled plant requires nearly 2 l per kWh.

The cross-sectional geometry of the condensing channels of a dry-cooled power plant is finned on the outside. The channels are in many cases flat and rectangular and relatively large with dimensions of approximately 214 mm by 13 mm. The tube lengths are approximately 10 m in length with very low steam mass fluxes of lower than 10 kg/m^{2}s [17-19]. The reasons for these choices of inclination angle and low mass fluxes have not been addressed in literature. There is thus a gap in the literature that addresses condensation at different inclination angles as well as condensation at low mass fluxes

1.2 Inclination angles

Previous studies [1-4, 6-16, 20-38] on inclined tubes were at moderate to high mass fluxes. In those studies, it was found that varying the inclination angles altered the flow patterns with consequent effects on the heat transfer coefficients, pressure drops, and void fractions. It was also found that the effects of inclination became more pronounced as the mass flux decreased. For downwards inclinations, it was found that the effect of the gravity was dominant and caused a thinning of the liquid layer which led to a reduction in the thermal resistance within the tube surface, leading to higher heat transfer coefficients [1-5, 9]. For upwards inclinations, no concrete trend was established. However, there are two main challenges. The first is that there is no study that has systematically coupled the effect of inclination and temperature difference on the heat transfer coefficients and flow patterns at low mass fluxes (≤100 kg/m^{2}s). The other challenge is that there are contradictory reports [1, 4, 7, 8, 11] on the recommended inclination angle for optimum heat transfer performance. This is further evidenced by the fact that there is no unifying correlation that can properly predict the heat transfer coefficients in inclined smooth tubes. This may be attributed to the fact that the available models are either limited by tube size, working fluid, saturation temperature, mass flux, or tube orientation. A review of the most relevant works on inclination is presented below.

Tepe and Mueller [29] were arguably the first to publish their findings on the effect of inclination during condensation inside smooth tubes. They performed experiments during the condensation of benzene inside a smooth tube 18 mm in diameter at a single inclination angle of 15°. They observed that there existed an effect of inclination on their measured heat transfer coefficients. They also found that their measured heat transfer coefficients were approximately 50% higher than the predicted values when compared to the Nusselt [39] classical theory. Following closely were Hassan and Jakob [32], who performed numerical and empirical studies on the effect of inclination on the heat transfer coefficients during condensation outside horizontal tubes. They noticed an effect of inclination on the measured heat transfer coefficients. Furthermore, they applied the Nusselt [39] classical theory and compared the results of their experiments to that of their numerical analysis. They found that the heat transfer coefficients of their numerical study were between 28% and 100% lower than the results of their experiments. They attributed this to the rippling effect of the condensate film, which was not accounted for in their theoretical model. Later, Chato [31] also observed an inclination effect during condensation of R113, wherein he observed that slightly downwards inclinations led to an increase in heat transfer rates.

Chato [23] studied and developed analytical solutions for stratified laminar condensation in horizontal and inclined tubes. It was assumed that the condensate depth decreased along the tube length. Hence, he neglected the heat transfer in the liquid pool at the bottom of the tube and assumed that the void fraction did not change significantly with respect to vapour quality. These assumptions led to large errors at high mass fluxes and low vapour qualities, because convective heat transfer prevailed in those conditions. He further developed a Nusselt-type equation for the condensation of refrigerants at low vapour velocities inside horizontal and inclined tubes based on Chen’s [36] analysis of falling film condensation outside a horizontal cylinder.

Nitheanandan and Soliman [28, 40] obtained flow regime data during the condensation of steam inside a 13.4 mm diameter tube at upwards and downwards inclinations within ± 10°. In all their experiments, they achieved complete condensation inside the condenser. They found that the zones occupied by the wavy and slug regimes experienced significant shifts, whereas the effect on the annular flow boundary appeared to be insignificant at the present small inclination angles. They also compared their data with adiabatic gas–liquid flow regime maps developed analytically and experimentally for horizontal and inclined tubes.

Lips and Meyer [3-6] studied the heat transfer and pressure drops during the condensation of R134a inside a smooth inclined tube. They carried out experiments at different inclination angles for upwards and downwards flows. With the aid of a high-speed camera installed at the exit of their test section, they captured and studied the flow patterns by varying the mass fluxes, vapour qualities, and inclination angles. They found that at high mass fluxes and vapour qualities; the flow was independent of the angle of inclination and always remained annular. However, at high mass fluxes and low vapour qualities, the flow regime was largely intermittent and dependent on the inclination angle. They defined the impact of gravity on the heat transfer coefficients as the 'Inclination effect' (I_α) and presented an expression for it. They also found that the highest heat transfer coefficients were achieved at an inclination angle of between −15° and −30º (downwards flow). The gap in their work was that they did not investigate the combined effect of the temperature difference and inclination on the heat transfer coefficients.

Mohseni and Akhavan-Behabadi, Mohseni *et al*. [7, 10] conducted experiments for seven different tube inclinations between −90° and +90° and six (6) refrigerant mass fluxes between 53 and 212 kg/m^2s to measure the heat transfer coefficients and observe the flow patterns of R134a condensing inside smooth and microfin inclined tubes. They found that the tube inclination noticeably influenced the heat transfer coefficients. In terms of the flow regimes, they found an effect of inclination on the vapour and condensed liquid flow distribution leading to eight distinct flow regimes with respect to the different tube inclinations. They also found that the best heat transfer performance was achieved at an inclination angle of +30° (for all refrigerant mass fluxes). Their findings were in sharp contrast with the findings of Lips and Meyer [3-5, 9]. A holistic look at both studies showed that the difference between the experimental conditions was the length of the test sections, average saturation temperature, and the mass flux range, but would those variables be significant enough to sharply alter the inclination effect? They also found the effect of inclination angle on heat transfer coefficient to be more prominent at low vapour qualities and mass fluxes. Furthermore, they developed an empirical correlation that predicted the results of the heat transfer coefficient of their experiments. However, they did not investigate the combined effect the temperature difference and inclination on the heat transfer coefficients.

Meyer *et al*. [2] conducted condensation heat transfer experiments in an 8.38 mm diameter inclined smooth tube at saturation temperatures in the range 30–50 °C. Their study was an extension of the work of Lips and Meyer [4, 5, 9], who carried out similar experiments but at a single saturation temperature of 40 °C. They found out that in general, an increase in saturation temperature led to a decrease in the heat transfer coefficients. They also found out that the inclination effect on the heat transfer coefficients became more prominent as mass fluxes decreased. Similar to the result of Lips and Meyer [4] they found that the angle which gave the maximum heat transfer coefficients was between −15° and −30° (downwards flow). However, they did not investigate the influence of temperature difference on the heat transfer coefficients as they kept their heat transfer rate constant.

Xing *et al.* [11] performed experiments during the condensation of R245fa inside an inclined tube 14.81 mm in diameter and 1.2 m in length at an average saturation temperature of 55 °C. The mass fluxes considered to be in the range 191–705 kg/m^{2}s with the vapour quality ranging between 0.19 and 0.95. They further carried out a nondimensional study wherein they found influences of inertia and gravity on the condensation heat transfer coefficients. Their analysis showed that surface tension forces were insignificant during the two-phase process. With respect to the heat transfer coefficients, they found an influence of inclination angle on the heat transfer coefficients and posited that optimal inclination angles of 15° and 30° existed, at which the heat transfer coefficients reached a maximum. With respect to the flow patterns, they observed stratified-smooth flow, stratified-wavy flow, intermittent flow, churn flow, falling film, and annular flow. They developed a correlation which was also able to explain the influence of mass flux, vapour quality, and inclination angle on condensation heat transfer coefficient. This correlation also predicts that the measured heat transfer coefficient strongly depends on the Froude's number and vapour quality.

Shah [14, 41] developed new correlations for condensation inside smooth inclined micro- and macrochannels. Their correlation was an extension of the Shah [42, 43] models. However, their model has not been tested with new experimental data at low mass fluxes and varying temperature differences.

1.3 Low mass fluxes

Despite its wide range of utilization and huge potential for future applications, little information is available on the thermal performance of inclined condensers at low mass fluxes. A review of the open literature [1-6, 9-16, 20, 21, 24-26, 28, 29, 34, 41, 44-73] revealed that most studies on condensation inside smooth tubes focussed on horizontal and vertical configurations at mass fluxes typically greater than 200 kg/m^{2}s and normally reaching up to 1 000 kg/m^{2}s. Other studies [18, 19, 49, 63, 72, 74] showed that at low mass fluxes, the heat transfer coefficient was dependent on the temperature difference. Of these studies, none has quantitatively and systematically investigated the effects of both tube inclination and temperature difference on the heat transfer coefficients and flow patterns in smooth tubes at low mass fluxes. This underscores the need for more data collection by employing empirical studies to help in this regard. A review of the most relevant literature on low mass fluxes is presented below.

Davies *et al*. [17-19] conducted experimental and flow visualization studies during the condensation of steam in noncircular inclined steel tubes brazed with aluminium fins at steam mass fluxes lower than 10 kg/m^{2}s and a uniform air fin-face velocity of 2.2 m/s. Their test condenser was approximately 11 m in length and made of steel with brazed aluminium fins with a rectangular cross-section of dimensions 214 mm × 18 mm. Their test condenser was also cut in half lengthwise and covered with a polycarbonate viewing window to allow for simultaneous visualization and the heat transfer measurements. Their maximum inlet air temperature ranged from 35 °C and the average condensing temperature of the steam was maintained at 100 °C. With respect to inclination angle, they varied it from 0° (horizontal flow) to 75° (downward flow). Furthermore, they found that the depth of the condensate river at the bottom of the tube decreased with an increase in inclination angle. The average steam-side heat transfer coefficient was shown to increase with an increase in inclination angles. Overall, their results suggested that an improvement in steam-side heat transfer performance was achieved by varying the tube inclination angle. With respect to flow visualization, they found only the stratified flow regime for all test conditions at all locations along the condenser. They also observed both film-wise, drop-wise, and condensation on the tube wall. The steam-side heat transfer coefficient was found to be dependent on the wall-steam temperature difference, and not vapour quality or Reynolds number. As a result, the condensation heat transfer coefficient did not decrease along the condenser length, as is common for smaller condenser tubes with higher mass fluxes. Finally, they posited that the overall heat transfer coefficient of the condenser was found to increase linearly with increasing downwards inclination angle of the test condenser, at an approximate rate of 0.08% per degree of inclination beneath the horizontal reference. They attributed this increase to improved drainage and increased void fraction near the condenser outlet.

Lyulin *et al*. [13] studied the laminar convective condensation of pure ethanol vapour inside an inclined smooth circular tube of inner diameter 4.8 mm and of length 200 mm. The experiments were conducted at an average saturation temperature of 58 °C. The vapour mass flux was varied from 0.24 to 2.04 kg/m^{2}s. They investigated the dependence of the heat transfer coefficient on both the temperature difference between the saturated vapour and the wall and the condenser inclination. They found that the heat transfer coefficient reduced with an increase in the temperature difference. They also found the heat transfer coefficient was a maximum at an inclination angle in the range -35° to -15°. They attributed this to the complex gravity drainage mechanism of the condensate. They also posited that their results would be valuable in the development of compact cooling systems for ground and space applications.

Arslan and Eskin [63] measured the heat transfer coefficients during the condensation of R134a inside a vertical smooth tube. They covered only downwards flows for a mass flux range (20–175 kg/m^{2}s). Their condensation temperatures were varied from 20–30 °C. Their findings revealed that the heat transfer coefficients decreased with an increase in saturation pressure and that at low mass fluxes, the heat transfer coefficients were dependent on the temperature difference between the inner tube wall temperature and saturation temperature. They also found that the measured heat transfer coefficients increased as the mass flux increased. They posited that amongst the other correlations, that of Akers *et al.* [75] best predicted their results, with an average deviation of 23%. However, they did not represent their results as a function of vapour quality and only considered a vertical tube orientation, neglecting other possible inclinations.

Recently, Olivier *et al.* [1] investigated the effect of inclination on void fraction and heat transfer coefficient during the condensation of R134a inside a smooth inclined tube at mass fluxes between 100 and 400 kg/m^{2}s. They captured flow regimes and measured void fractions with a capacitive void fraction sensor mounted at both the inlet and outlet of their test section. They kept the heat transfer rate of their experiments at 200 W. They found that the inclination effect on heat transfer coefficients and measured void fractions became insignificant with increasing mass flux and vapour quality. The greatest effect of inclination on heat transfer coefficients was observed for combinations of low mass flux and low vapour quality. Their results at downwards inclinations were more sensitive to changes than for upwards inclinations. They also found that the void fraction and flow pattern map predictions were inadequate for inclined flow conditions. However, they did not investigate the influence of temperature difference on the heat transfer performance, keeping their heat transfer rate constant.

Most recently, Meyer and Ewim [74] investigated the effect of this temperature difference on heat transfer coefficients at low mass fluxes during the condensation of R134a in a smooth horizontal tube with an internal diameter of 8.38 mm. They carried out experiments at mass fluxes of 50, 75, 100, 150, and 200 kg/m^{2}s, at temperature differences from 3 °C to 10 °C, and at a condensing temperature of 40 °C. The flow patterns captured at the inlet and outlet of their test section and were found to be stratified and stratified wavy flows only. They found that the effect of temperature difference on the heat transfer coefficients began to manifest at a mass flux of 150 kg/m^{2}s, but only at a mean vapour quality of 0.1. They also found that the dependence of heat transfer coefficients on temperature difference increased at all vapour qualities when the mass fluxes were lower than 150 kg/m^{2}s. In general, they found that the maximum heat transfer coefficients were found at the lowest temperature differences and the minimum heat transfer coefficients at the maximum temperature differences. They also found that the dependence on the temperature difference became more pronounced as the mass flux reduced. To conclude, they suggested a revision of the Cavallini *et al*. [76] model. The results of this revision were that their 103 experimental data points could be estimated to within ±5%. The gap in their study was that it was limited to only horizontal tubes.

1.4 Problem statement and purpose of study

The review of previous works presented in Secs 1.2 and 1.3 shows that there is a gap in the literature on condensation at different inclination angles at low mass fluxes. Meyer and Ewim [74] showed that at low mass fluxes, the heat transfer coefficients are a function of not only mass flux and quality but also the temperature difference *(ΔT)* between the wall on which condensation occurs and the saturation temperature. It was, therefore, the purpose of this study to experimentally investigate the heat transfer coefficients and flow regimes at different inclination angles at low mass fluxes and different temperature differences, ΔT. This study is a continuation of the work of

Meyer and Ewim [74], which was limited to condensation in horizontal tubes. This study concentrates on the effect of different inclination angles.

2. Experimental apparatus and procedure

The test bench (Fig. 1) used for this investigation is a well-established test bench that has been used for previous condensation studies [1-5, 15, 16, 34, 67, 72, 77]. However, slight modifications were incorporated to accommodate the low-mass-flux requirements and these were discussed in detail by Meyer and Ewim [74] . The inclination angle (β) of the test section could be varied in this study from −90° (downwards flow) to 90° (upwards flow), with 0° (horizontal flow) as the reference. The inclination angles were measured with a digital inclinometer, which was calibrated to an accuracy of 0.01°.

The test section inner diameter and length were 8.38 mm and 1.5 m, respectively. The condensing fluid was R134a and was cooled with water flowing through an annulus in a counter-flow direction. The condensation experiments occurred at a condensation temperature of 40 °C and the inlet quality was controlled with a pre-condenser. The heat transfer rates in the test section were varied from 170 to 580 W.

A summary of the operating conditions and average energy balances of all experiments is given in Table 1. The energy balances of all the experiments varied between a minimum of 0.2% and a maximum of 5.2%. The average energy balance was 2.1% with a standard deviation of 1.2%.

3. Data reduction

The nomenclature used and data reduction have been discussed in detail by Meyer and Ewim [74] and are therefore not discussed in this paper. The only new term introduced in this study is the inclination effect, (I_α) defined by Lips and Meyer [4], as:

$$I_\alpha = \frac{\alpha_{max} - \alpha_{min}}{\alpha_{\beta=0}} \qquad (1)$$

In Eq. 1, α_{max} and α_{min} are the maximum and minimum heat transfer coefficients obtained for a specific mass flux and mean vapour quality for the various angles of inclination. Furthermore, $\alpha_{\beta=0}$ is the heat transfer coefficient obtained for the horizontal orientation, which is presented in Meyer and Ewim [74] . In addition, the '*temperature differences*' referred to in this paper, which were used to calculate the heat transfer coefficients, are given by Eq. 2:

$$\Delta T = T_{sat} - \overline{T}_{w,i} \tag{2}$$

This refers to the temperature difference between the average refrigerant saturation temperature, T_{sat}, and the inner wall temperature, $\overline{T}_{w,i}$, as explained in Meyer and Ewim [74] . The average refrigerant saturation temperature was taken as the average of the test section inlet and outlet temperatures. $\overline{T}_{w,i}$, was taken as the average of the 28 wall temperature measurements on the test section.

4. Uncertainty analysis and repeatability

An uncertainty analysis was conducted as prescribed by Dunn [78] and is presented in Table 2. The full details can also be found in Meyer and Ewim [74] . A selection of approximately 60% of the experiments was repeated three months later and the differences in results were compared. The maximum percentage differences of the heat transfer coefficients and qualities, when the experiments were repeated, was approximately 5%. This maximum difference was found at vapour qualities below 0.25 and inclination angles of +90° and −90°.

5. Validation

A validation study was conducted to establish the integrity and accuracy of our experimental test section and the data produced from it. The validation experiments were two-fold. First, condensation experiments were conducted at different mass fluxes in a horizontal configuration and these results were discussed in detail in Meyer and Ewim [74] .

Second, validation experiments were conducted at different inclination angles, as summarized in Table 3, which identifies the 45 different conditions that were used for experimental comparison purposes. The validation experiments were conducted at a saturation temperature of 40 °C, over a mass flux range of 200–400 kg/m^{2}s, at a mean vapour quality of 0.5, at inclination angles of $-90° \leq \beta \leq 90°$, and with heat transfer rates of approximately 200 W, as were done by Lips and Meyer [4] and Meyer *et al.* [2] The results of a part of the validation experiments are summarized in Fig. 2. In this figure, the results are given for a mass flux of 300 kg/m^{2}s and vapour quality of 0.5 for 15 different inclination angles from −90° to 90°, and are compared to the measurements of Lips and Meyer [4] and Meyer *et al.* [2]. In general, the measurements compared well and the differences in results were within the uncertainties of our measurements.

6. Results

The in-tube condensation heat transfer experiments were carried out at a condensation temperature of 40 °C, and mass fluxes of 50, 75, and 100 kg/m^{2}s, at different temperature differences, inclination angles, and mean vapour qualities. The inlet and outlet flow regimes were also captured and all the results are shown in Figs. 3–10. Where relevant, the heat transfer coefficients were the averaged over the test section length. The approximate mass fluxes, vapour qualities, and temperature differences are presented in an experimental matrix in Table 4, which shows that 900 experimental data points were produced. At a mass flux of 100 kg/m^{2}s, it was possible to take 375 measurements: 15 different inclination angles of −90°, −60°, −45, −30°, −15°, −10°, −5°, 0°, 5°, 10°, 15°, 30°, 45°, 60°, and 90° and at five different temperature differences of 1, 3, 5, 8, and 10 °C with mean vapour qualities varying between 0.25 and 0.9. At a mass flux of 75 kg/m^{2}s, it was possible to take 300 measurements at 15 different inclination angles between −90° and 90° and at four different temperature differences of 1, 3, 5, and 8 °C with mean vapour qualities varying between 0.10 and 0.9. At a mass flux of 50 kg/m^{2}s, it was possible to take 225 measurements at 15 different inclination angles between −90° and 90° and at temperature differences of 1, 3, and 5 °C with mean vapour qualities of 0.10–0.90. The measurement points that could not be produced were mainly at lower mass fluxes and vapour qualities. The challenges at these points were high differences between the temperatures of the condensing refrigerant at the inlet and outlet of the test section. This meant that we ended up having a sensible cooling process instead of a latent heat condensation (phase change) process. Other difficulties faced were measurement fluctuations in the desired test mass fluxes and vapour qualities, and the time taken to attain steady state conditions. These difficulties were more pronounced as temperature differences were increased, and as mass fluxes and vapour qualities were reduced.

6.1 Flow patterns

Fig. 3 summarizes the six flow patterns observed in this study. These flow patterns are stratified (S), stratified wavy (SW) (also observed in Meyer and Ewim [74]), annular (A), annular wavy (AW), intermittent (I), and churns flows (C). These flow patterns were adopted using the descriptions of flow regimes as prescribed by Thome [79, 80]. Bubbly flow was not observed on its own but was observed during intermittent flows. The flow pattern abbreviations S, SW, A, AW, I, and C are used to identify the flow patterns in Figs. 4 and 5. In these figures, the flow patterns are given for two different mass fluxes 100 kg/m^{2}s (Fig. 4) and 50 kg/m^{2}s (Fig. 5) as a function of temperature differences and inclination angles for mean qualities of 0.5 and 0.25, respectively. These were chosen to reflect most of the flow pattern descriptions observed during the experiments.

In Fig. 4, at a mass flux of 100 kg/m^{2}s, and mean vapour quality of 0.5, the flow patterns are given for temperature differences of 3, 5, and 10 °C, and inclination angles of −90°, −60°, −30°, 0°, 30°, 60°, and 90°. At an inclination angle of −90°, it was found that both the inlet and outlet flow regimes were either annular or annular wavy for all the temperature differences. At inclination angles of −60°, −30°, and 0°, it was found that both the inlet and outlet flow regimes were stratified wavy at all the temperature differences. The only exception was at a temperature difference of 10 °C and an inclination of 0°, for which stratified flow was observed at the exit of the test section. At inclination angles of 30° and 60°, it was found that both the inlet and outlet flow regimes were frequently changing from churn to stratified wavy for all the temperature differences. At an inclination angle of 90°, both the inlet and outlet flow regimes were churn flow.

In Fig. 5, at a mass flux of 50 kg/m^{2}s and mean vapour quality of 0.25, the flow patterns are given for temperature differences of 1, 3, and 5 °C, and inclination angles of −90°, −60°, −30°, 0°, 30°, 60°, and 90°. At an inclination angle of −90°, both the inlet and outlet flow regimes were intermittent and churn, respectively, for the three temperature differences. At inclination angles of −60°, −30° and 0°, both the inlet and outlet flow regimes were stratified wavy for all temperature differences. At inclination angles of 30° and 60°, both the inlet and outlet flow regimes were intermittent for the three temperature differences. At an inclination of 90°, all the flow patterns were churn flow at both the inlet and outlet.

6.2 Heat transfer coefficients

The heat transfer coefficients at mass fluxes of 100, 75, and 50 kg/m^{2}s are plotted as functions of different inclination angles with varying temperature differences at different mean vapour qualities of 0.25, 0.5, 0.62, and 0.75 in Figs. 6–8. Furthermore, the heat transfer coefficients at mass fluxes of 100 and 50 kg/m^{2}s are plotted as functions of temperature differences of 1, 3, 5, 8, and 10°C with varying inclination angles in Fig 9. In general, the results showed the same general trends of heat transfer coefficients as a function of mass flux, temperature differences, and vapour qualities that have been shown in previous work [74, 77]. Thus, the heat transfer coefficients increased with decreasing values of temperature differences and increased with increasing values of vapour quality and mass flux.

Fig. 6 shows that at a mean vapour quality of 0.25, the maximum heat transfer coefficients were found at the minimum temperature differences in each case; i.e., ΔT = 3° C (Fig. 6a), ΔT = 1° C (Fig. 6b), ΔT = 1° C (Fig. 6c) for mass fluxes of 100, 75, and 50 kg/m^{2}s, respectively. These occurred at inclination angles of −15°, −30°, and −15°. Furthermore, it was found that the minimum heat transfer coefficients for all the temperature differences were found at an inclination of −90° (downwards flow) and the corresponding maximum temperature differences were 10, 8, and 5 °C for mass fluxes of 100, 75, and 50 kg/m^{2}s, respectively.

It was also found that the inclination effect was more dominant for downwards flows than upwards flows. For downwards flows, the flow regimes were, in general, all stratified wavy (Figs. 4 and 5) with the gravity forces collecting the condensing liquid on the bottom part of the tube with a very thin condensing liquid layer around the circumference on the top part of the tube. As this layer is thin, the heat transfer resistance is small and therefore, the heat transfer coefficients are large. Furthermore, the condensing liquid did not only flow downwards to the bottom part of the tube but also in an axial direction to the tube outlet. The results show that in general, the optimal downwards angle is between −30° or −15°.

The trend of variations in heat transfer coefficient may be related to the prevailing flow regime. At $\beta = -90°$, the flow regime is churn which generally corresponds to low heat transfer coefficients. With an increase in the inclination angle to the optimum, the flow regime becomes stratified, which provides a direct contact between the vapour and tube wall and as a result, the heat transfer coefficient increases. With further increase in the inclination angle, the liquid film thickness seems to increase, which results in an increase in the heat transfer resistance and consequently, a decrease in the heat transfer coefficient. For the vertical upward flow directions, the flow regimes are almost churn and therefore, the heat transfer coefficients decrease, and the inclination effect is negligible.

For upwards flows, it was found that there seemed not to be any major effect of inclination on the heat transfer coefficients of different temperature differences. Furthermore, it was deduced that the effect of temperature difference was different for the vertically upwards (+90°) flow in comparison to the vertically downwards (−90°) flow. The temperature differences had a negligible effect on the heat transfer coefficients during both vertically downwards flows and vertically upwards flows.

When comparing the heat transfer coefficients of the horizontal tube ($\beta = 0°$) orientation to that of downwards vertical ($\beta = -90°$) orientation, it was found that the heat transfer coefficients of the horizontal orientation were greater. This could be attributed to the stratification due to gravity, which enhanced the heat transfer by keeping the condensate thickness low in the upper region of the tube as compared to the vertically downwards flow. In this case, even though the heat transfer coefficient at the bottom was reduced, the heat transfer enhancement in the upper region prevailed and the mean cross-sectional heat transfer coefficient was increased as compared to the vertically downwards flow orientation. It can also be deduced that condensation heat transfer coefficients were more sensitive to changes in the inclination angles near the horizontal position. In these slightly inclined positions (either upwards or downwards) the flow patterns were mainly stratified smooth flow and stratified wavy.

When stratified smooth flow and stratified wavy flow occurred, the inclination angles had a heat transfer enhancement effect. As the inclination angles (Figs. 4 and 5) decreased from 0° to −30°, the liquid film thickness decreased because of gravity and consequently led to an increase in the convection effect. As a result, the thermal resistance decreased, and therefore, the heat transfer coefficient increased. Furthermore, the flow regimes were almost the same for this region (stratified wavy or stratified.) With a further decrease of the inclination angle from −30° to −60°, the flow regimes remained stratified wavy without any significant changes in the liquid film thicknesses. Therefore, the heat transfer coefficients remained approximately constant between these two inclination angles. This can be seen in Figs. 6–8. However, with the decrease in the inclination angle from −60° to −90°, there was a change in the flow regime from stratified wavy to either churn, intermittent or annular flows. When the flow regime changed to churn or intermittent flows, the liquid phase covered the tube surface sporadically, which caused an increase in thermal resistance and consequently a decrease in the heat transfer coefficients. However, when the flow regime changed to annular flow, the liquid film always covered the entire tube surface, which also caused a significant decrease in the heat transfer coefficients. This considerable decrease can be observed in Figs. 6–8 at an inclination angle is −90°. The same interpretation is valid for the upwards flow directions, but the difference is that in those regions, the flow regimes were always intermittent or churn, for which the inclination had no significant effect on the heat transfer coefficients. In summary, the variations of heat transfer coefficients with respect to the tube inclination angle can be attributed to the change of flow regime and liquid film thickness on the tube surface.

Comparing the heat transfer coefficients between the mass fluxes in Figs. 6 (a), (b), and (c), it was found that in general, there was an increase in heat transfer coefficient as the mass flux increased. This increase could be expressly attributed to an increase in shear forces because, for each comparison, the temperature differences and inclination angle were kept constant. This same trend was observed in our previous study on horizontal tubes, where in general, the heat transfer coefficients increased with mass flux. In that study, the effect of temperature difference on the heat transfer coefficients was found to be the main driving force of the heat transfer process. In the current study, the prevailing flow regime and inclination angles also played roles in the heat transfer process.

Fig. 7 shows the effect of inclination on the heat transfer coefficients at different temperature differences at a vapour quality of 50%. In general, the results are the same than in Fig. 6, for a vapour quality of 25%. The exception is that the maximum heat transfer coefficients were found to be at a slightly lower inclination angle of −15° or −30° (downwards flow). At this vapour quality of 50% which was complicated by the role of interfacial waves and inclination angles, there was an increased shear stress on the vapour–liquid interface causing a more unstable interface, thereby enhancing the condensation heat transfer. With respect to the minimum heat transfer coefficient, it was also found that the heat transfer coefficient was more sensitive to the combined effect of inclination angles and temperature difference for downwards flows than for upwards flows. In general, it can be deduced that because the heat transfer coefficient is closely related to the liquid film thickness on the tube wall, the higher heat transfer coefficient was found when there was the thinner liquid film thickness and the converse was true for lower heat transfer coefficients. Finally, it was found that the phenomenon of the vertically upwards flow, where higher heat transfer coefficients were found for the maximum temperature difference, was also experienced.

Fig. 8 shows the effect of inclination on the heat transfer coefficients at different temperature differences but at a higher vapour quality of 75%. These results do not differ significantly from the results in Figs. 6 and 7, which were for qualities of 25% and 50% respectively. However, for upwards inclination angles at this vapour quality, there is an increase in the vapour shear forces exerted on the liquid film interface, which slows down the downwards motion of the liquid film and subsequently causes the interfacial portion of the film to be carried upwards instead downwards to the drain at the bottom of the tube. This condition leads to the onset of flooding.

In Fig.8, it can also be deduced that the effect of mass flux on the condensation heat transfer coefficient gradually increases as the vapour quality increases. The converse is true for decreasing vapour qualities. This can be attributed to the changes of flow regimes with vapour quality and mass flux. At lower vapour qualities of 25% and 50%, and mass fluxes of 50 kg/m^{2}s and 75 kg/m^{2}s, the flow regimes are in general stratified, stratified-wavy or churn flow. Therefore, increasing the mass flux from 50 kg/m^{2}s to 75 kg/m^{2}s does not change the flow regime to annular flow. Furthermore, this increase in mass flux does not decrease the liquid film thickness on the tube surface. Therefore, the result will be only a slight manifestation of the effect of shear forces, and an increased interaction between liquid and vapour phases, which does not significantly increase the condensation heat transfer coefficients. However, when the vapour quality is 75% (Fig. 8), the increase in the mass flux changes the flow regime to annular flow where the effect of shear forces is significant and consequently, decreases the liquid film thickness on the tube surface. Therefore, the condensation heat transfer coefficient increases more significantly with the increase in the mass flux (Fig. 8) as compared to the lower vapour qualities (Figs 6 and 7). It should be noticed that the above explanation is valid for all inclination angles except for β = -90° and β = +90°, where the flow regimes are in general annular at all the operating conditions

Consistent with previous figures, it can also be deduced that there was practically no effect of temperature difference at both the upwards and downwards vertical orientations, even though there was a minimal increase in heat transfer coefficient as temperature difference was increased in the vertically downwards orientation. The converse was true for the vertically upwards orientation.

Fig. 9 more clearly shows the effect of the temperature differences and inclination angles at a mass flux of 100 kg/m^{2}s (Fig. 9a) and 50 kg/m^{2}s (Fig. 9b) for a mean vapour quality of 50%. The results show that the maximum heat transfer coefficients were at an inclination angle of −30° for all the temperature differences. Furthermore, they show that the minimum heat transfer coefficients occurred at an inclination angle of −90° for all the temperature differences. It was also found that the percentage difference of the maximum heat transfer coefficient and the minimum heat transfer coefficient for a mass flux of 100 kg/m^{2}s in Fig 9a was 73%, whereas at a mass flux of 50 kg/m^{2}s in Fig. 9b, it was 81%. This further lends credence to the fact that the heat transfer enhancement effect decreased with an increase in mass flux. In addition, the figure clearly shows that although the heat transfer slightly decreased for the vertically downwards flow as the temperature difference increased, this decrease was less than 2% (negligible). The converse was true for vertically upwards flow and the increase was also 2%. The figure also shows that the effect of temperature difference was more dominant for downwards flows.

Fig. 10 shows the effect of inclination on the temperature differences and mass fluxes. In the figure, the inclination effect at mass fluxes of 100, 75, and 50 kg/m^{2}s for mean vapour qualities of 0.25 and 0.5 are plotted against the different temperature differences. This figure shows that the inclination effect was inversely proportional to the mass fluxes. Hence, as the mass flux decreased, the inclination effect increased. It was also found that between the minimum and maximum temperature differences, there was a decrease in the inclination effect. For instance, at a mass flux of 75 kg/m^{2}s and quality of 0.5, the inclination effect of 51% was found at a temperature difference of 1 °C and an inclination angle of −30°. This was greater than the inclination effect of 49%, which was observed at a temperature difference of 3 °C and an inclination angle of −15° for a mass flux of 100 kg/m^{2}s. Furthermore, at a mass flux of 50 kg/m^{2}s and quality of 0.25, the average inclination effect across the temperature differences of 1, 3, and 5 °C was 56%, which was higher than at the average inclination effect of 75 kg/m^{2}s at the same quality. This lends credence to the fact that the inclination effect was more significant at lower vapour qualities. In general, the inclination effect increased with a decrease in mass flux and increased with a decrease in vapour quality.

7. Conclusions

Experiments were carried out during the condensation of R134a in a smooth inclined tube at mass fluxes of 50, 75, and 100 kg/m^{2}s. The mean vapour qualities ranged from 0.1 to 0.9 at temperature differences of 1, 3, 5, 8, and 10 °C. In total, 945 data points were obtained for the validation and actual experiments. The flow patterns were visualised using two high-speed cameras installed at the inlet and outlet of the test section. It was observed that an annular flow pattern is prevalent for vertically downwards inclination (−90°). In contrast, churn was common for the vertically upwards inclination (+90°), with wavy annular being observed at higher qualities and intermittent at lower qualities. In all cases, the maximum heat transfer coefficients were found at the minimum temperature difference tested per data point and at inclinations angles between −15° and −30°. However, the minimum heat transfer coefficients were consistently found at the maximum temperature difference tested per data point and at an inclination angle of −90° (vertically downwards flow). It was found that even though the heat transfer coefficients for vertically downwards flow decreased with an increase in temperature difference, the percentage differences were approximately 2% (negligible). The converse was true for vertically upwards flows. With respect to the inclination effect, it was found that it decreased with an increase in temperature difference. It was also found that at low qualities (below 0.35), that the inclination effect was more noticeable. On the contrary, at high vapour qualities (above 0.5), no additional significant effect of vapour quality was found on the inclination effect. In general, the maximum inclination effect was found at the lowest mass flux tested for and the converse was true for the maximum mass flux investigated for. For annular flows, it was found that the heat transfer coefficients were independent of the inclination angle. In such cases, the heat transfer coefficient increased with increasing mass flux and this could be attributed to thinning of the liquid film by the increasing effect of vapour shear forces. However, for flows found to be stratified and intermittent, the effect of inclination angles was significant. Beyond a certain downwards inclination angle, it was found that the heat transfer coefficient decreased, and this could be attributed to the decrease in the perimeter occupied by the thin film of condensation at the top of the tube, where most of the condensation occurs. In general, the heat transfer coefficient mainly depends on the perimeter occupied by the condensation film and its thickness, which were primarily a function of inclination and temperature difference. In conclusion, it is recommended that future inclined condensers be inclined at angles between −15° and −30°. Furthermore, it is required that more flow-pattern-dependent mechanistic models be developed to assist with the prediction of the heat transfer coefficients for condensing flows in inclined tubes.

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Fig. 7. Condensation heat transfer coefficients, α, as a function of inclination angle, β, at different wall and refrigerant temperature differences, ΔT, at a mean quality of 0.50 during condensation: (a) Mass flux of 100 kg.m^2/s, (b) mass flux of 75 kg.m^2/s, and (c) mass flux of 50 kg.m^2/s.

Fig. 8. Condensation heat transfer coefficients, α, as a function of inclination angle, β, at different wall and refrigerant temperature differences, ΔT, at a mean quality of 0.75 during condensation: (a) Mass flux of 100 kg.m^2/s, (b) mass flux of 75 kg.m^2/s, and (c) mass flux of 50 kg.m^2/s.

Fig. 9. Condensing heat transfer coefficients as function of temperature differences, ΔT, and different inclination angles, β, at a mean quality of 0.50: (a) Mass flux of 100 kg.m^2/s and (b) mass flux of 50 kg.m^2/s.

Fig. 10. Inclination effect as function of temperature differences, ΔT, at different mass fluxes during condensation: (a) Vapour quality of 0.25 and (b) vapour quality of 0.50.

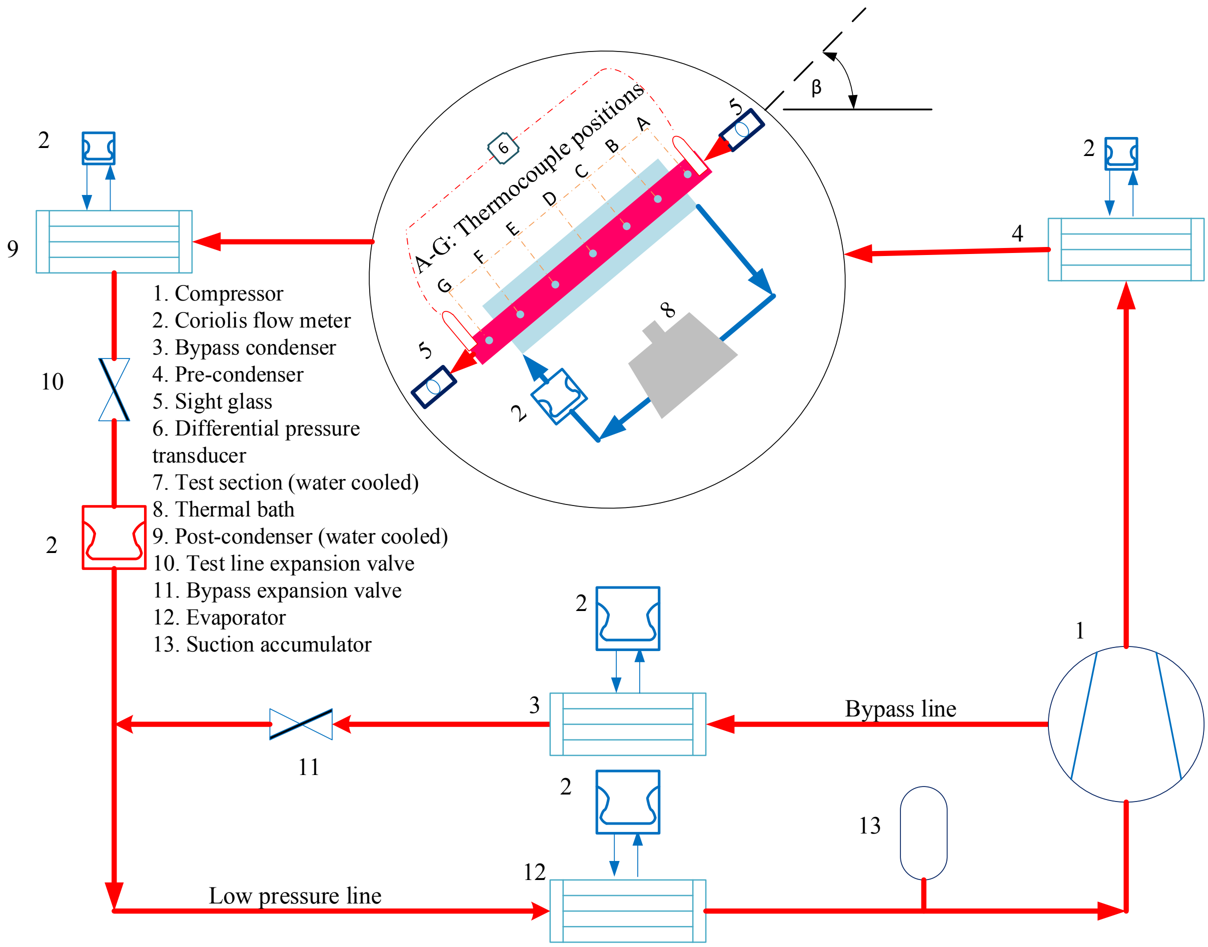
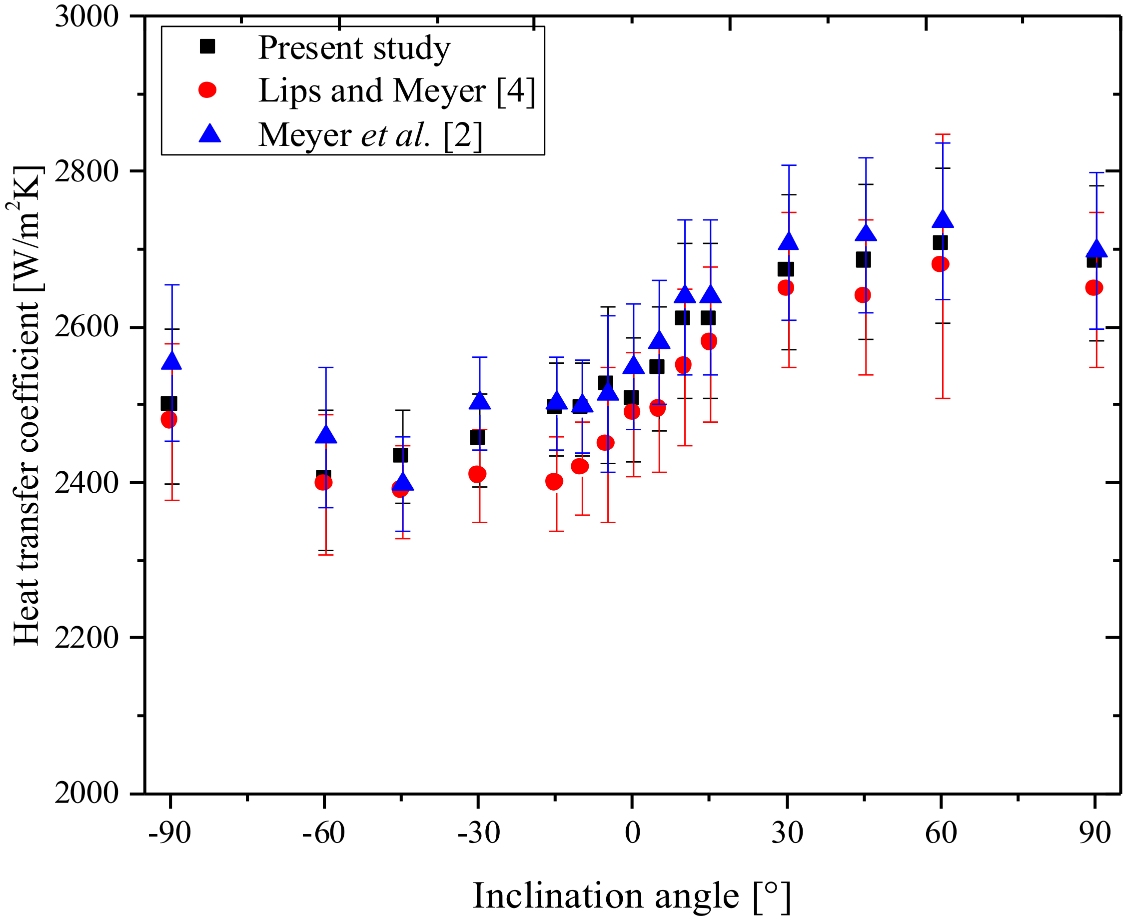

A-G: Thermocouple positions
A
B
C
D
E
F
G
β
1. Compressor
2. Coriolis flow meter
3. Bypass condenser
4. Pre-condenser
5. Sight glass
6. Differential pressure transducer
7. Test section (water cooled)
8. Thermal bath
9. Post-condenser (water cooled)
10. Test line expansion valve
11. Bypass expansion valve
12. Evaporator
13. Suction accumulator
Bypass line
Low pressure line

Figure 1



Present study
Lips and Meyer [4]
Meyer et al. [2]
Heat transfer coefficient [W/m²K]
3000
2800
2600
2400
2200
2000
-90
-60
-30
0
30
60
90
Inclination angle [°]

Figure 2

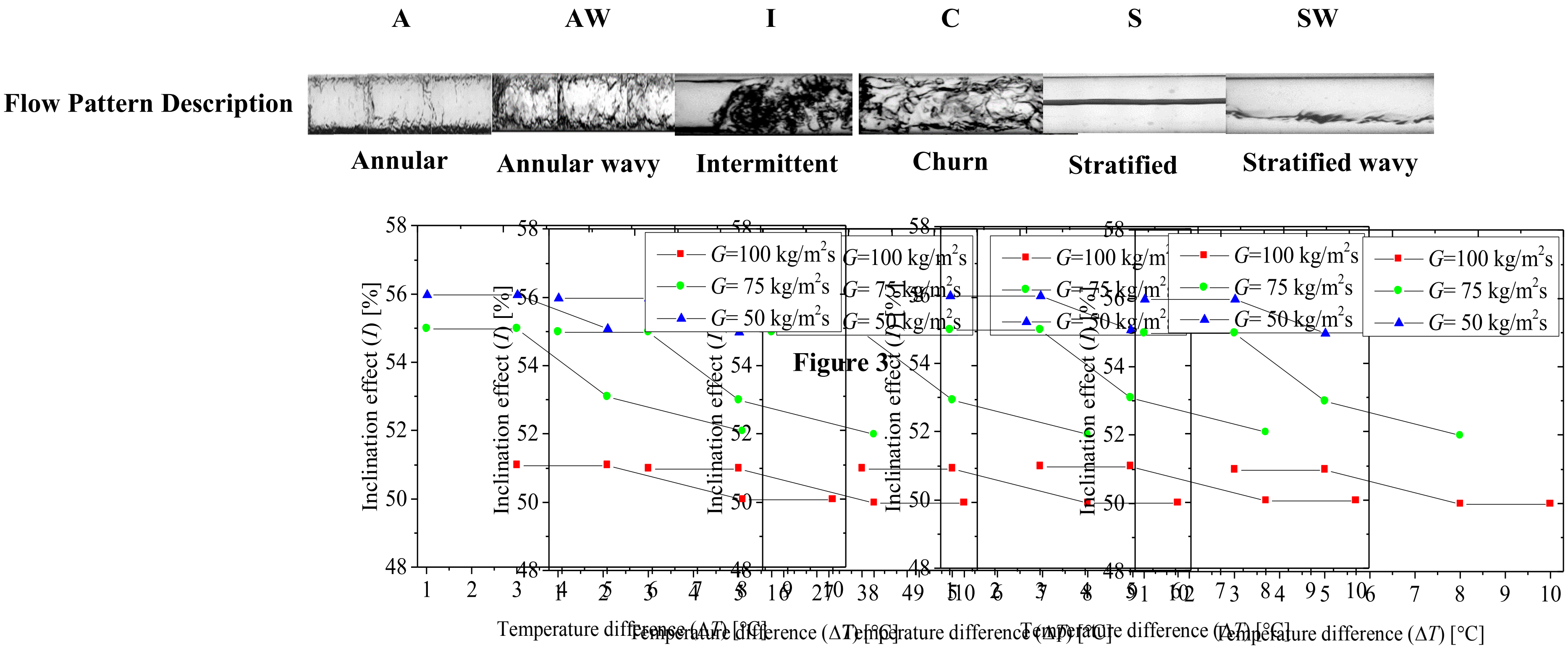

A
AW
I
C
S
SW
Flow Pattern Description
Annular
Annular wavy
Intermittent
Churn
Stratified
Stratified wavy
Inclination effect (I) [%]
Temperature difference (ΔT) [°C]
G=100 kg/m²s
G= 75 kg/m²s
G= 50 kg/m²s

Figure 3

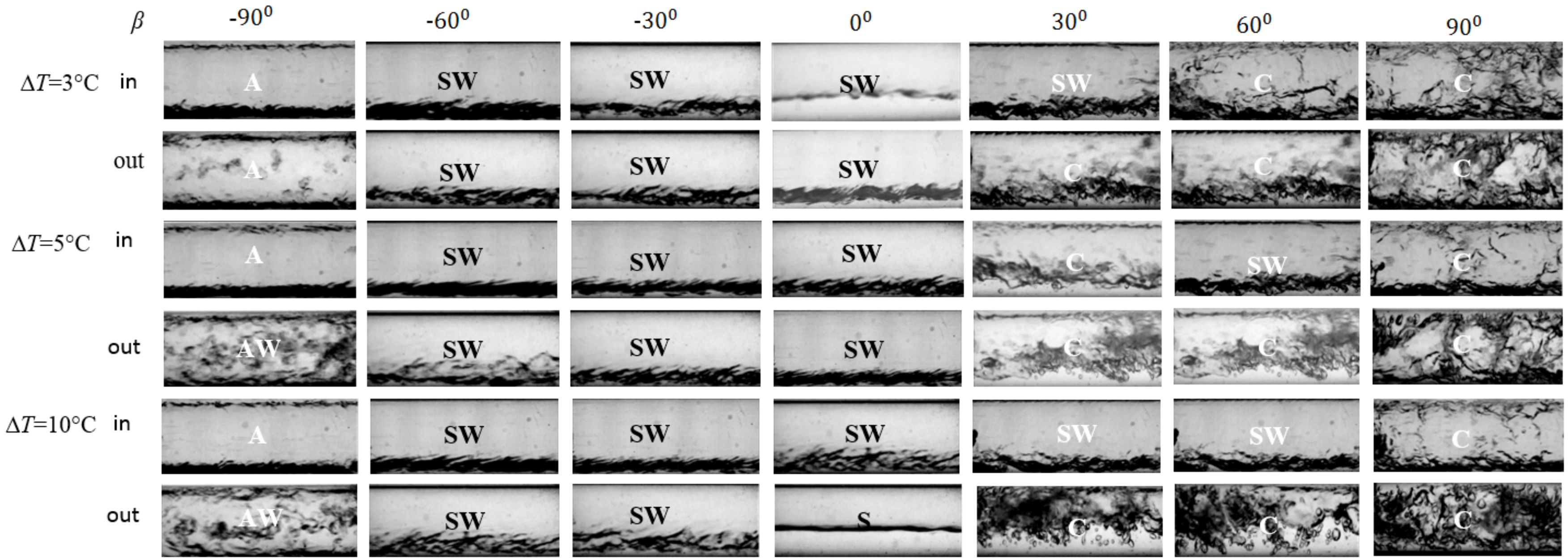

β
-90⁰
-60⁰
-30⁰
0⁰
30⁰
60⁰
90⁰
ΔT=3°C in
out
ΔT=5°C in
out
ΔT=10°C in
out
A SW SW SW SW C C
A SW SW SW C C C
A SW SW SW C SW C
AW SW SW SW C C C
A SW SW SW SW SW C
AW SW SW S C C C

Figure 4

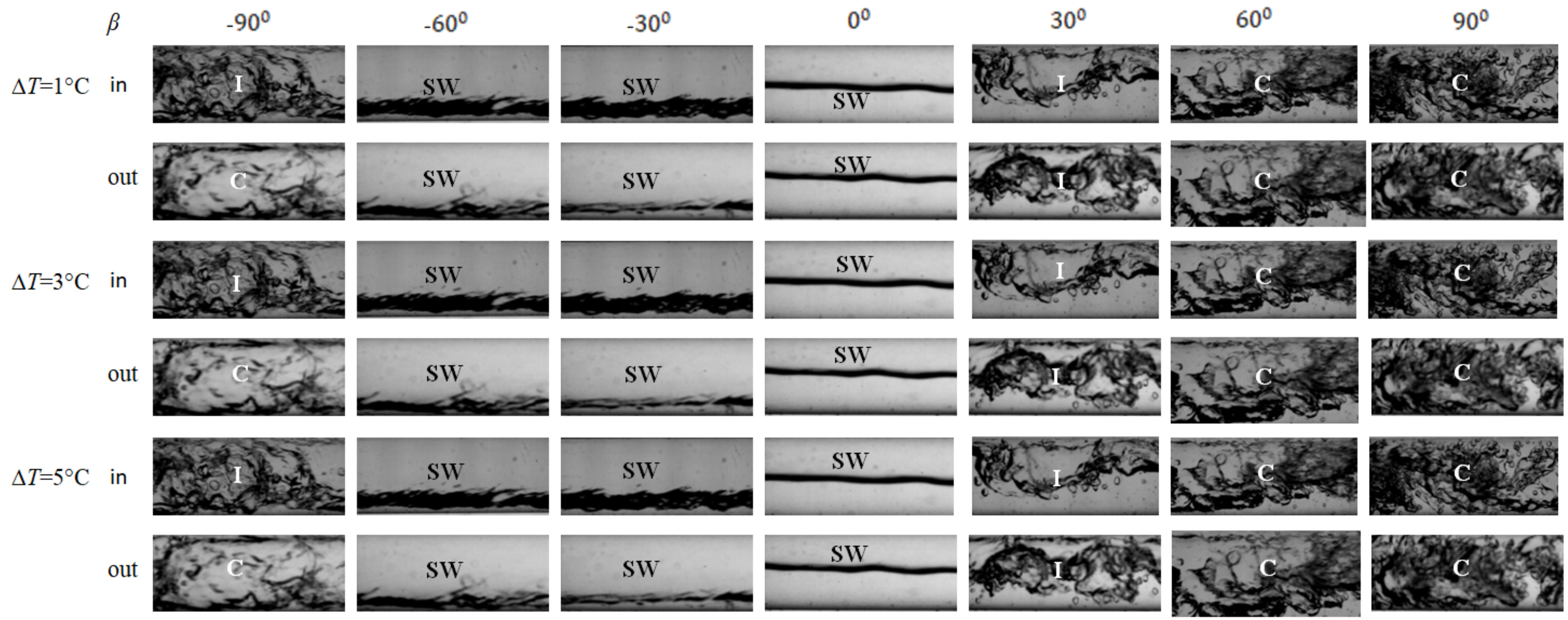
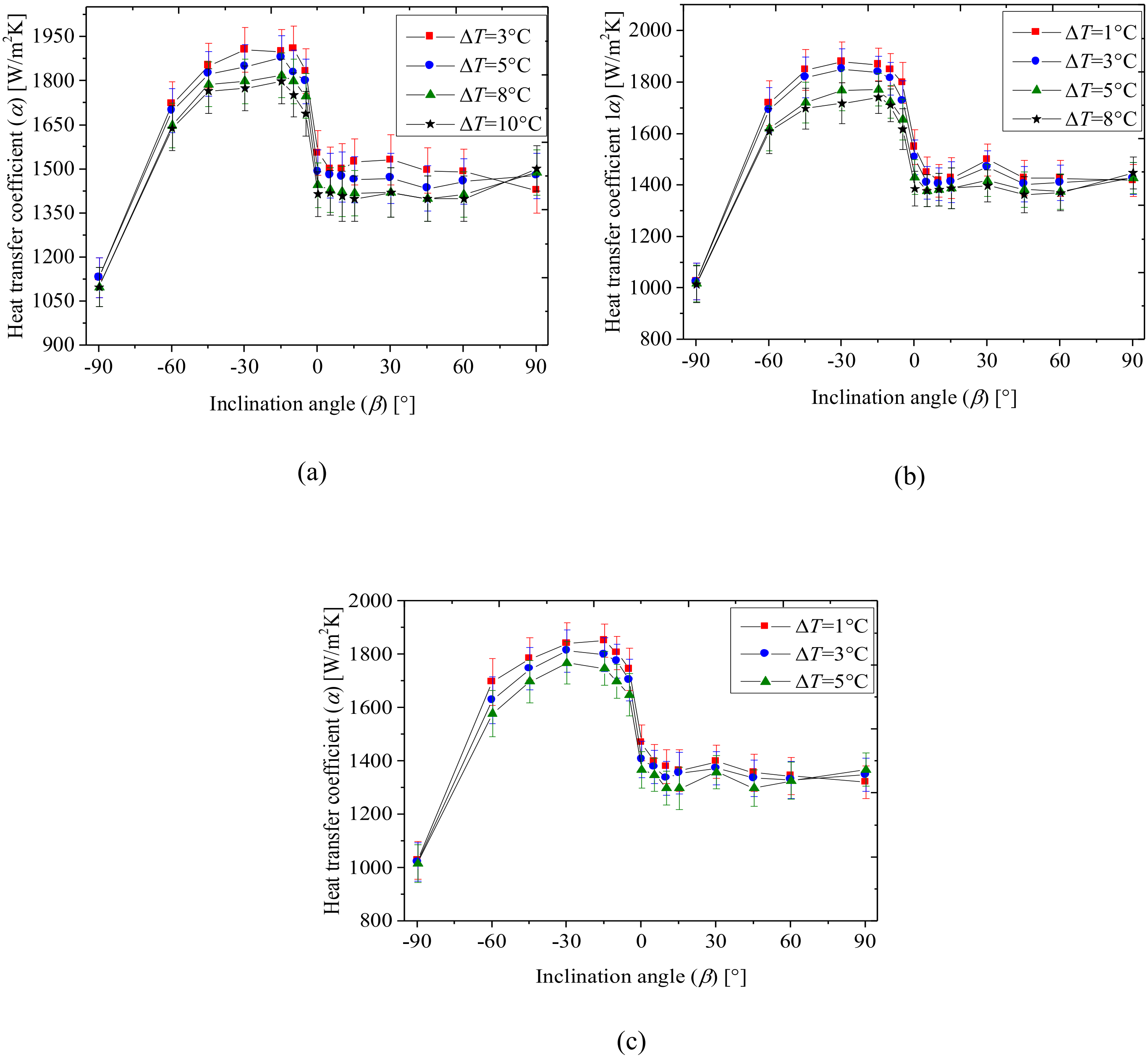
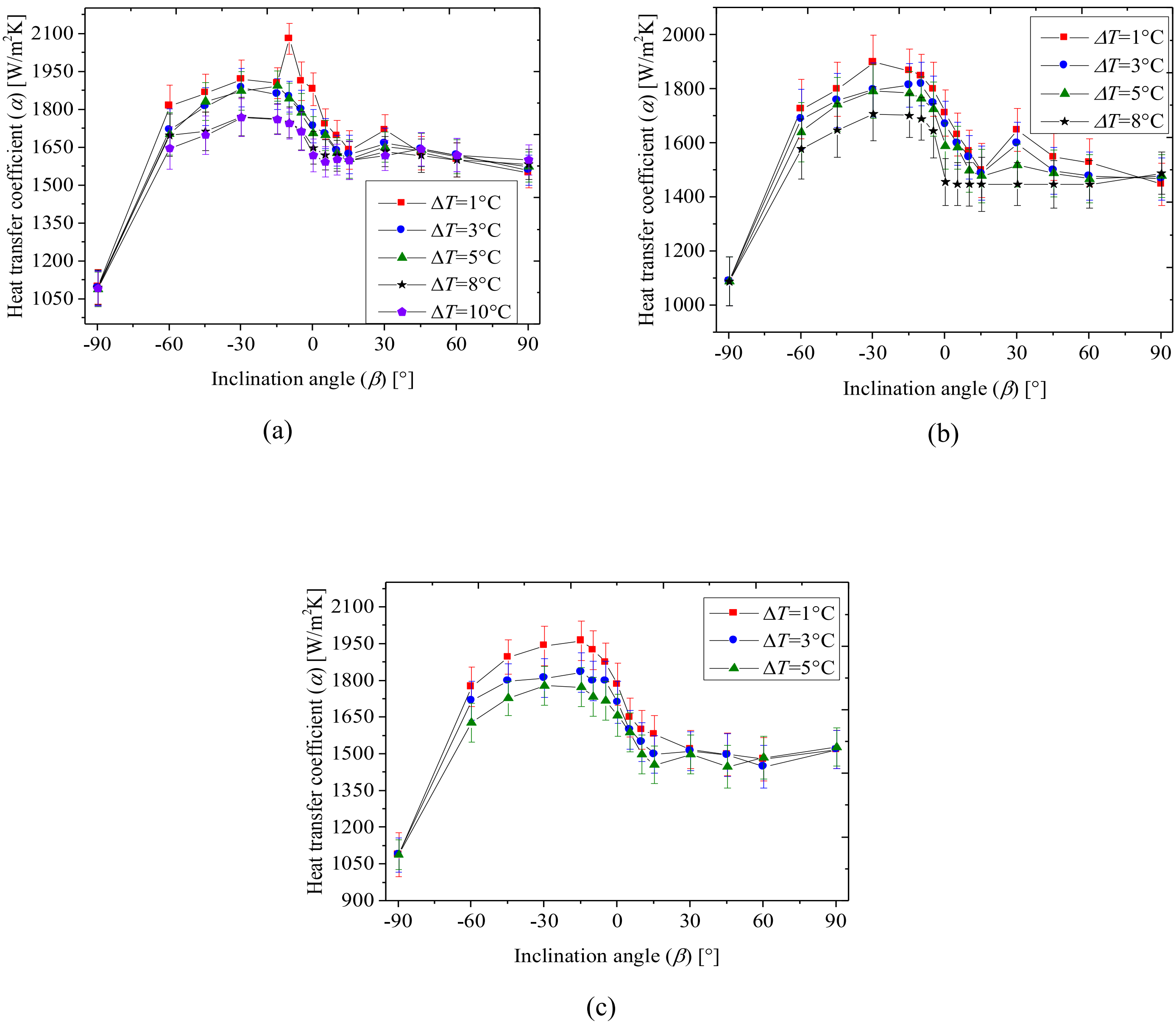

β
-90⁰
-60⁰
-30⁰
0⁰
30⁰
60⁰
90⁰
ΔT=1°C in
out
ΔT=3°C in
out
ΔT=5°C in
out
I
SW
C

Figure 5



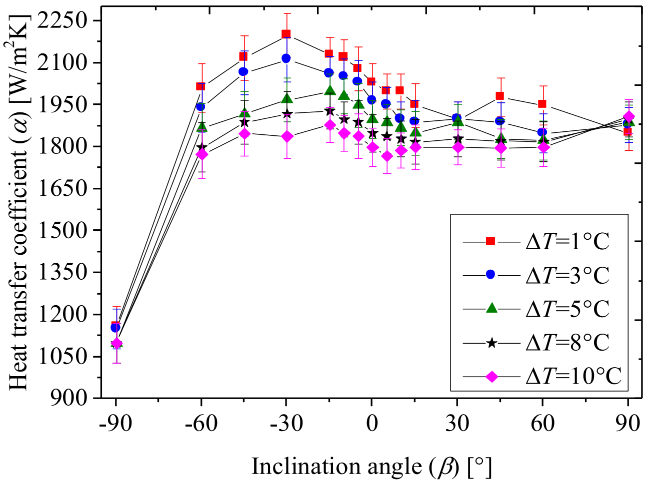
(a)
Heat transfer coefficient (α) [W/m²K]
Inclination angle (β) [°]
ΔT=3°C
ΔT=5°C
ΔT=8°C
ΔT=10°C
(b)
Heat transfer coefficient 1(α) [W/m²K]
Inclination angle (β) [°]
ΔT=1°C
ΔT=3°C
ΔT=5°C
ΔT=8°C
(c)
Heat transfer coefficient (α) [W/m²K]
Inclination angle (β) [°]
ΔT=1°C
ΔT=3°C
ΔT=5°C

Figure 6

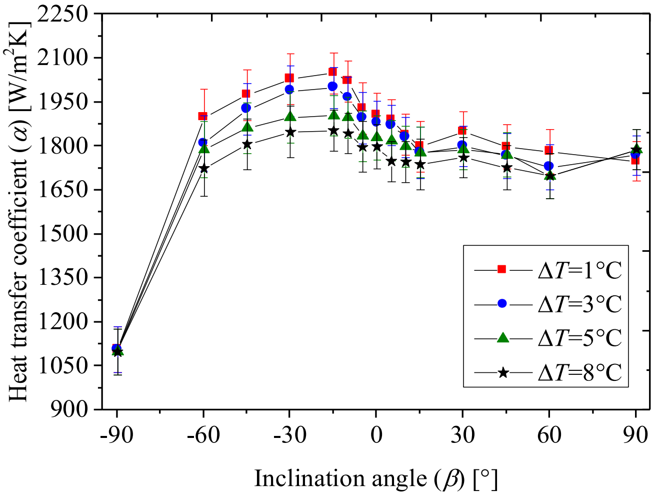


Heat transfer coefficient (α) [W/m²K]
Inclination angle (β) [°]
ΔT=1°C
ΔT=3°C
ΔT=5°C
ΔT=8°C
ΔT=10°C
(a)
(b)
(c)

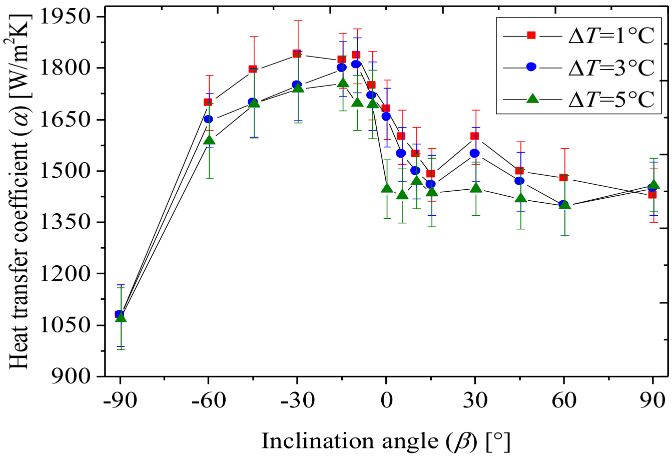
Figure 7


Heat transfer coefficient (α) [W/m²K]
Inclination angle (β) [°]
ΔT=1°C
ΔT=3°C
ΔT=5°C
ΔT=8°C
ΔT=10°C
900 1050 1200 1350 1500 1650 1800 1950 2100 2250
-90 -60 -30 0 30 60 90

(a)

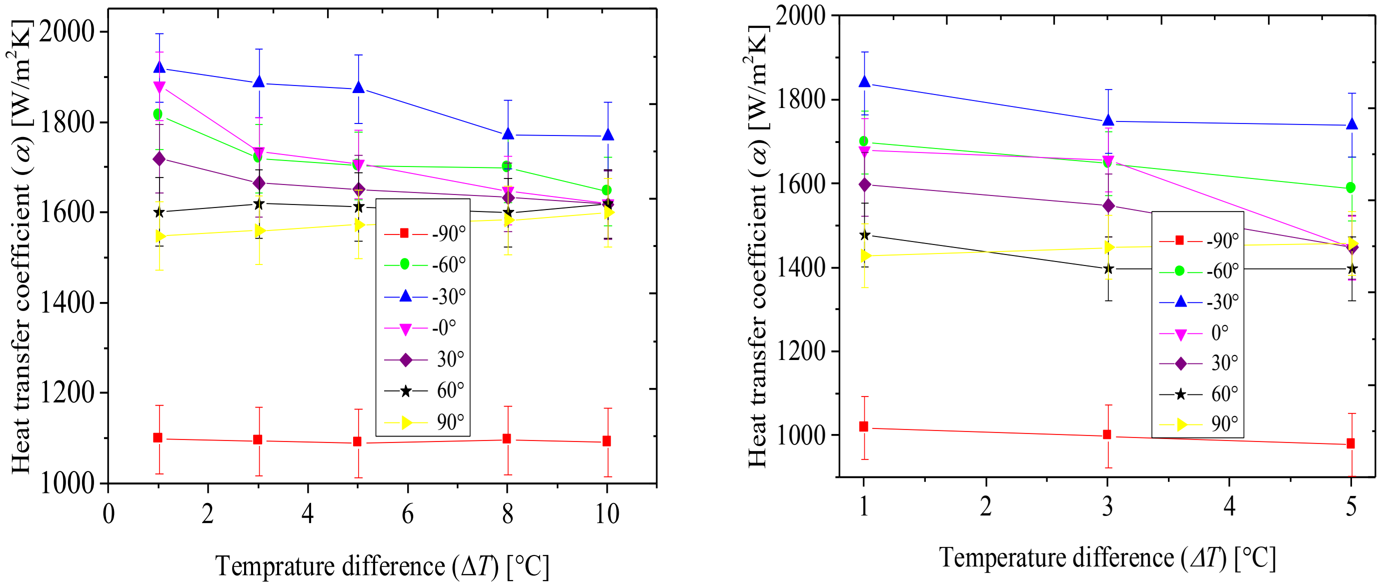

Heat transfer coefficient (α) [W/m²K]
Inclination angle (β) [°]
ΔT=1°C
ΔT=3°C
ΔT=5°C
ΔT=8°C
900 1050 1200 1350 1500 1650 1800 1950 2100 2250
-90 -60 -30 0 30 60 90

(b)


Heat transfer coefficient (α) [W/m²K]
Inclination angle (β) [°]
ΔT=1°C
ΔT=3°C
ΔT=5°C
900 1050 1200 1350 1500 1650 1800 1950
-90 -60 -30 0 30 60 90

(c)

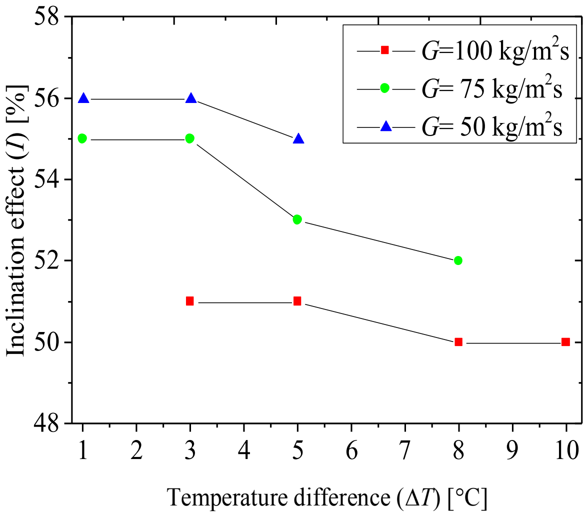
Figure 8


Heat transfer coefficient (α) [W/m²K]
Temprature difference (ΔT) [°C]
-90°
-60°
-30°
-0°
30°
60°
90°
Temperature difference (ΔT) [°C]
0°

(a)

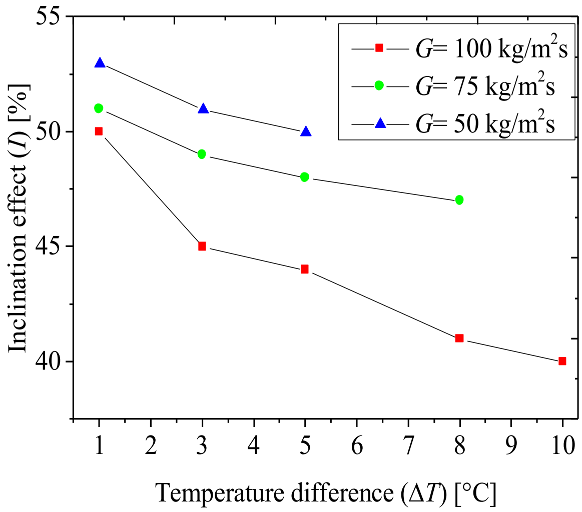
(b)

Figure 9



58
56
54
52
50
48
Inclination effect (I) [%]
1 2 3 4 5 6 7 8 9 10
Temperature difference (ΔT) [°C]
G=100 kg/m2s
G= 75 kg/m2s
G= 50 kg/m2s

(a)



55
50
45
40
Inclination effect (I) [%]
1 2 3 4 5 6 7 8 9 10
Temperature difference (ΔT) [°C]
G= 100 kg/m2s
G= 75 kg/m2s
G= 50 kg/m2s

(b)

Figure 10

List of Tables

Table 1

Parameter	**Average**	**Minimum**	**Maximum**	**Standard deviation**
Condensation temperature	40.0 °C	39.6 °C	40.5 °C	0.28 °C
Saturation pressure	1 052 kPa	1 031 kPa	1 074 kPa	9.8 kPa
Energy balance (*EB*)	2.1%	0.2%	5.2%	1.2%

Table 2

Parameter	Range	Uncertainties
T_{sat}	40 °C	±0.1 °C
G	50–100 kg/m^{2}s	±1%
x_m	0.1–0.9	±5%
α	1 300–2 800 W/m^{2}K	±11%
Q_w	180–600 W	±1%

Table 3

G [kg/m²s]	x_m [-]	***β*** [°]	**Points**
200	0.5	−90, −60, −45, −30, −15, −10, −5, 0, 5, 10, 15, 30, 45, 60, 90	15
300	0.5	−90, −60, −45, −30, −15, −10, −5, 0, 5, 10, 15, 30, 45, 60, 90	15
400	0.5	-90, −60, −45, −30, −15, −10, −5, 0, 5, 10, 15, 30, 45, 60, 90	15
			Total = 45 points

Table 4

$\boldsymbol{G}$	$\Delta \boldsymbol{T}$	$\boldsymbol{x}_{\mathbf{m}}$	$\boldsymbol{\beta}$	**Points**
[kg/m^{2}s]	**[°C]**	**[-]**	**[°]**	
50	1,3,5	0.10, 0.25, 0.5,	−90, −60, −45, −30, −15, −10, −5,	
		0.62, 0.75, 0.9	0, 5, 10, 15, 30, 45, 60, 90	225
75	1,3,5,8	0.10, 0.25, 0.5,	−90, −60, −45, −30, −15, −10, −5,	
		0.62, 0.75, 0.9	0, 5, 10, 15, 30, 45, 60, 90	300
100	1,3,5,8,10	0.10, 0.25, 0.5,	−90, −60, −45, −30, −15, −10, −5,	
		0.62, 0.75, 0.9	0, 5, 10, 15, 30, 45, 60, 90	375

Total = 900 points