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The influence of tilt on natural convection in the elbow thermosyphon

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Abstract—The paper presents the results of an experimental study of natural convection in a water-filled, elbow thermosyphon. Data have been obtained with a small-scale rig in which the Rayleigh number was varied in the range $10^{5.6} < Ra < 10^{7.6}$ with 1.98 cm inner diameter tubes having heated and cooled lengths ranging from 20 to 80 cm. The main purpose of the study was to determine the effect of tilting an initially-vertical cooled section. The results show that forward, or positive, tilts produce very little effect on the heat transfer rate while backward, or negative, tilts produce a dramatic reduction within the first 20 degrees.

INTRODUCTION

THE TURBULENT thermosyphon has found extensive use in many spheres, e.g. ground freezing, waste heat recovery and nuclear reactor cooling. In the vast majority of these applications, the thermosyphon is linear; that is, it consists of a straight tube with one end exposed to a heat source while the other contacts a heat sink. There is, however, a growing range of circumstances for which this arrangement is unsuitable; in particular, it must sometimes be replaced by a right angled arrangement [1–4]. Typical among these is the configuration in which an upright, cooled section is connected to a horizontal, heated section.

To date, the most interest in this configuration has been shown in the evaporative form. Very little work has been done on the system in its most conservative form; namely, under single-phase conditions. Two recent papers have attempted to repair this deficiency through analysis and experiment. Lock and Park [5] have shown in a numerical study that laminar flow patterns in the vertical and horizontal sections are reconciled through a convective coupling mechanism in which a hot tongue of fluid moving over the upper surface of the heated section becomes the ascending core in the cooled section, while the descending annulus is transposed into the cold tongue entering the heated section below. Their results are limited to low Rayleigh numbers.

Lock and Ladoon [6], on the other hand, obtained experimental data for a similar system using water as the filling fluid. While this confirmed the coupling mechanism and provided useful information on the effect of tube geometry, as defined by the heated–cooled length ratio and the heated length–diameter ratio, it did not include the effect of tilting the tube explored in the companion numerical study. Accordingly, the present work was undertaken.

The numerical study revealed that positive tilt angles, as defined in Fig. 1, produced different behaviour from negative tilt angles. It was also noted that overall behaviour exhibited a symmetry about $+45°$ and $-135°$. These facts have been used to guide the present study which is consequently restricted to a relatively narrow range of tilts. As with the previous studies, the purpose here is to determine the heat transfer characteristics of the device: in particular, to explore regime behaviour; to measure the effect of tilt on heat transfer; and to probe the effect of tube geometry.

EXPERIMENTAL CONSIDERATIONS

The experiments were conducted on a small scale rig built from two copper tubes, both 1.98 cm inner diameter (2 mm wall thickness) and over 80 cm long. These were assembled in the configuration shown schematically in Fig. 2. As indicated, the heated section was wrapped with four equal lengths of electric heating tape connected in parallel. This provided four independent 20 cm heated lengths around which was wrapped a thick annulus of fibreglass thermal insulation to minimize heat lost to the atmosphere. Prior to the tests proper, each heated length was calibrated by measuring the heat supplied when the tube was

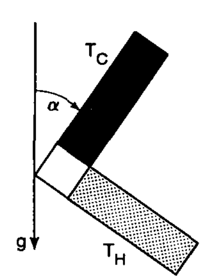

FIG. 1. Configuration and tilt angle.

NOMENCLATURE

D	diameter
g	gravitational acceleration
k	thermal conductivity
L	length
Nu	Nusselt number
$\dot{Q}$	heat flux
Ra	Rayleigh number
T	absolute temperature.

Greek symbols

α	tilt angle
β	thermal expansion coefficient
κ	thermal diffusivity
ν	momentum diffusivity.

Subscripts

C	cooled
D	based on diameter
H	heated.

filled with thermal insulation; this heat supply rate was then plotted against the temperature difference between the tube wall and the room air. The heat leakage thus determined was later found to be about 10% of the gross heat supplied during the experiments proper.

The initially-vertical section of the tube was cooled by four identical cooling jackets connected to the building water mains. No attempt was made to estimate the heat removed from the system since, under the steady conditions investigated, it is equal to the net heat supplied. At the right angle junction between the heated and cooled sections, a thick layer of thermal insulation was added to provide an adiabatic surface and thus minimize any diabatic effects near the junction.

Changes in system geometry were effected by the use of styrofoam pistons inserted through the closed end of each section, as indicated in Fig. 2. The presence of each piston ensured that the length over which it extended was shut down, thereby giving control over the active lengths in both heated and cooled sections. A few experiments were undertaken with the right angled junction piece replaced by a smooth bend.

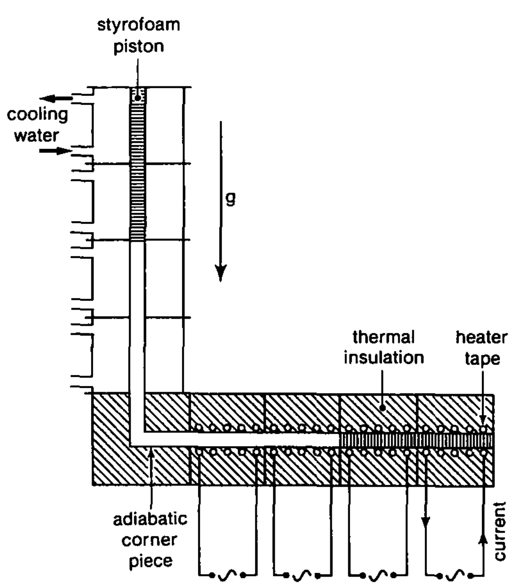

Fig. 2. Schematic of elbow thermosyphon rig.

Temperatures were measured throughout using copper–constantan thermocouples connected via a set of switching boxes to a digital thermometer. The thermocouple junctions were laid in axial grooves, 1 mm deep and about 0.5 cm long, cut in the outer surface of the copper tubes. The wires for each section were led out axially, to minimize end conduction errors, and bunched together before exiting either through a specially-sealed boss, in the cooled sections, or between layers of insulation, for the heated sections. Wall thermocouples were located every 2 cm on each of four rows running longitudinally over the entire 80 cm length of each section; the rows were equidistant around the tube circumference. The gross heat supply rate to each heater was measured in the usual way with an ammeter and voltmeter.

Following assembly and the calibration runs, the experimental procedure was as follows. With chosen styrofoam pistons in place, the tube was carefully filled with bubble free water and the cooling water for the active sections turned on. Power to the corresponding active heated sections was then set at a low level and the system left to reach steady state; this typically required a time period of about 3 h during which power adjustments were made, as necessary, to establish a reasonably uniform wall temperature. Once the system had stabilized, readings were taken of temperature (at the walls and in the atmosphere) and of the gross power supplied. The average wall temperatures T_H and T_C were calculated from 20 representative individual thermocouples: they were then used to determine the heat leakage and finally the Nusselt and Rayleigh numbers defined, respectively, by

$$Nu_D = \frac{\dot{Q}}{\pi k L_H (T_H - T_C)}$$

$$Ra_D = \frac{\beta g (T_H - T_C) D^3}{\nu \kappa}.$$

Fluid properties were estimated at the mean temperature $(T_H + T_C)/2$.

Table 1. Test schedule

Test series	L_H (cm)	L_H (cm)	α (deg)
1	40	40	0
2	40	40	21.5
3	40	40	33.0
4	40	40	36.0
5	40	40	42.0
6	40	40	-5.5
7	40	40	-10.5
8	40	40	-19.5
9	20	20	36.0
10	80	80	36.0

$D = 1.98$ cm, $3.1 < Pr < 7.1$, $-1.0 < \log_{10} Nu_D < 0.8$, $5.6 < \log_{10} Ra_D < 7.6$.

This procedure was then repeated, first by increasing the power level in steps towards the safe maximum, and second by decreasing the power levels. In this way, the possibility of hysteresis could be investigated. None was observed. The range of non-dimensional variables covered in the study is given in Table 1. A formal error analysis was also conducted. The uncertainty bands thus calculated are shown on the figures discussed below.

DISCUSSION OF RESULTS

Regime behaviour

It is well established that there are two principal laminar flow regimes in the linear, vertical thermosyphon [7]. For high Rayleigh numbers, substantial velocity and temperature gradients in the radial direction are confined to a narrow region close to the tube wall. In this boundary layer regime, an annulus of fluid flows from each extremity towards the junction region, while central cores flow in the opposite direction, replenishing the annuli. As the Rayleigh number is reduced, the annuli thicken thereby experiencing greater resistance from the opposing cores. In this impeded regime, the 'boundary layer' fills the tube, and the slope of the $Nu \sim Ra$ curve is much steeper leading, eventually, to a conduction regime where, for all practical purposes, circulation ceases.

Similar behaviour occurs in the linear, horizontal thermosyphon, except that it is complicated by a strong secondary flow [8]. The primary flow then consists of a pair of filaments moving in opposite directions. As noted earlier, the elbow thermosyphon combines the vertical and horizontal flow structures through a convective coupling mechanism; otherwise, the primary flows remain unaltered [5, 6]. It is therefore natural to expect that these laminar flow regimes might be found in the elbow thermosyphon.

Figure 3 shows the data obtained in this study with a representative heated length–diameter ratio $L_H/D = 20:1$. As a general observation, it is evident that the positive tilt data obtained with $Ra_D > 10^{6.5}$ display a noticeably lower slope than for $Ra_D \leqslant 10^{6.0}$, although the influence of tilt and the greater error in the lower range does not permit an unequivocal

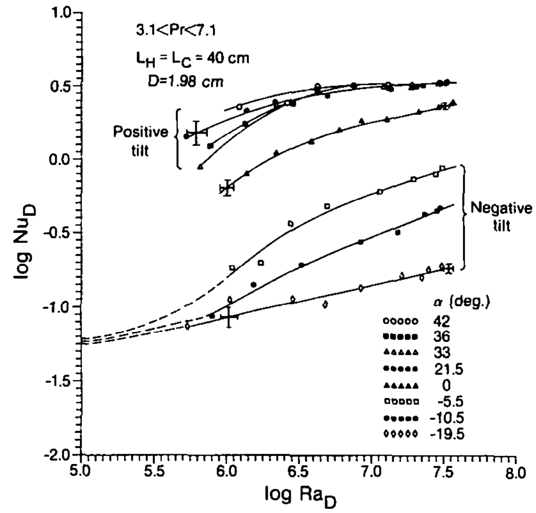

FIG. 3. The effect of Rayleigh number on heat transfer in an elbow thermosyphon.

assertion. The observation is, however, consistent with the data presented later in Fig. 5 and with the earlier findings of Lock and Ladoon [6] who suggested that the high range data were not in fact obtained under laminar conditions. It thus appears that $Ra_D \simeq O(10^{6.5})$ separates a laminar impeded regime from a fully-mixed turbulent regime. Martin [9], detected a similar transition in a water-filled, linear open thermosyphon. Under fully-mixed conditions, turbulent diffusion extends across the entire tube width in the absence of a mean circulation; thermal energy is transmitted along the tube primarily as the result of low intensity, large scale eddy motion. There is no evidence in the linear thermosyphon literature of a subsequent transition from a fully-mixed turbulent regime to a turbulent boundary layer regime at even higher Rayleigh numbers. However, this transition may occur in nonlinear tubes [10].

For negative tilts, the situation changes substantially, as Fig. 3 reveals. For Rayleigh numbers less than about 10^6, the heat transfer rate (and hence the circulation rate) is very low; this presumably represents the conduction regime. For higher Rayleigh numbers, the heat transfer rate is still significantly lower than for positive tilts, thus suggesting a laminar boundary layer regime separated from the conduction regime by a brief impeded regime. Slopes along the curves for $\alpha = -5.5°$ and $-10.5°$, for example, are significantly greater than the value of 0.25 characteristic of laminar boundary layer flow; but this is the slope of the higher data points when $\alpha = -19.5°$. This latter behaviour tends to confirm the appearance of the single, bifilamental circulation loop predicted numerically for tilt angles less than $-15°$, though for lower Rayleigh numbers [5].

Selected data were also obtained with the right angled junction piece replaced by a smooth bend of radius 7.6 cm. Within experimental error, no difference in heat transfer was observed.

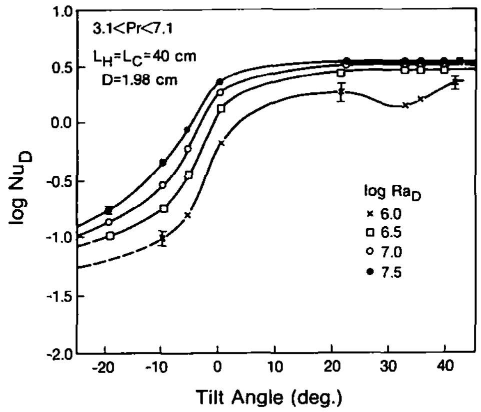

FIG. 4. The effect of tilt on heat transfer in an elbow thermosyphon.

The effect of tilt and geometry

The effect of tilt on heat transfer is shown in Fig. 4 which is a cross plot of Fig. 3 after the data had been smoothed. Given the anticipated symmetry about $\alpha = 45°$, the effect of tilt may be divided into two ranges. With $\alpha > 0$, tilting the tube produced only a modest change in heat transfer for any given Rayleigh number. Although the highest measured value occurred at $\alpha = 36°$, it is possible that, within the experimental error indicated, a tilt of 45° represents an optimum, provided $Ra_D > 10^{6.5}$ and keeping in mind that temperature dependent fluid properties may alter the situation. However, this theoretical optimum (or any other) is very shallow, thereby offering greater flexibility to the designer.

It thus appears that for $0° < \alpha < 90°$, any alteration in the performance of the cooled section is compensated by a corresponding alteration in the performance of the heated section. In this tilt and Rayleigh number range, the coupling mechanism appears to be unchanged. Vigorous turbulent mixing caused by separation at the junction is consistent with this suggestion; it effectively converts the heated and cooled sections into two open thermosyphons fed from the same pool of mixed, and virtually isothermal, fluid. Only when the Rayleigh number is lowered substantially would there be reason to suggest alternative (laminar) behaviour. Some evidence of this is seen in Fig. 4 near $\alpha = 30°$ but the attendant error renders it inconclusive. More definite evidence is found in the numerical study of Lock and Park [5] who noted bifurcations, and a reduced heat transfer rate, in the vicinity of $\alpha = 45°$ under impeded conditions, e.g. $Ra_D < 10^5$.

Figure 4 also reveals that the behaviour of the system changes dramatically if the tilt angle is negative. This feature was heralded in Fig. 3 which shows the negative tilt data clearly separated from the positive tilt data. Under normal circumstances, a thermosyphon designer would avoid this region where the heat transfer rates are especially low. From symmetry, the heat transfer rate reaches a minimum at $\alpha = -135°$ when, with the circulation virtually absent, heat is transferred downwards by conduction. For all practical purposes, this minimum will also be obtained at any tilt angle when $Ra_D \to 0$. A simplified conduction analysis reveals that $Nu_D = O(10^{-3})$ under these conditions, but the use of a copper tube (with a thermal conductivity several hundred times greater than that of water) prevented this value from being reached in these experiments. A more realistic asymptote for the apparatus used appears to be in the range $10^{-1.5} \leqslant Nu_D \leqslant 10^{-1.2}$.

With this rough estimate in mind, the negative tilt data (see Fig. 3) have the following interpretation. As the negative tilt is introduced, it has the initial effect of reducing the circulation rate and thereby tends to convert a fully-mixed (separated) turbulent flow into a laminar (unseparated) flow. Further tilting thus creates a laminar impeded flow with a steeper slope in the $Nu_D \sim Ra_D$ curve near $Ra_D = 10^6$, but increasing proximity to the conductive regime evidently produces an overriding effect and a lower slope, particularly for $Ra_D > 10^7$ when a single circulation loop may occur. It is clear from Fig. 4 that this dramatic reduction in heat transfer occurs largely in the first 20° of negative tilt.

It is worth noting that the symmetry about $+45°$ (or $-135°$) is characterized by only two extrema. For lower Rayleigh numbers, and laminar flow, Lock and Park [5] found that $\alpha = +45°$ marked a local minimum, while the maximum was shifted to $\alpha = -10°$. This is a significant difference under positive tilt conditions and may be attributed to the presence of turbulence in the experiments reported here. For $Ra_D \geqslant 10^{6.5}$, the laminar convective coupling is evidently destroyed and replaced by simple mixing if the tilt is positive.

The effect of geometry on the 'optimum' heat transfer rate is shown in Fig. 5. These data were obtained with $\alpha = +36°$. This tilt was chosen simply because it

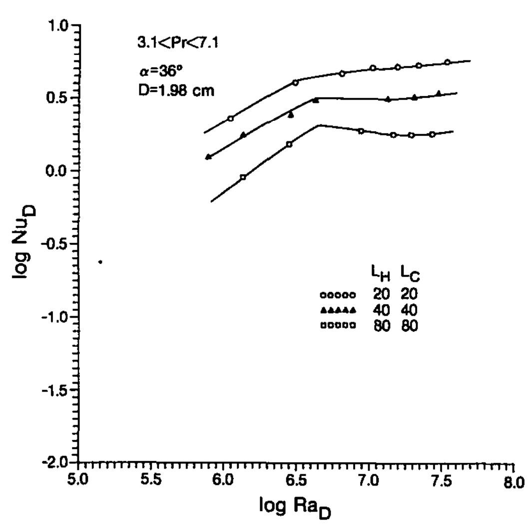

FIG. 5. The effect of geometry on heat transfer with positive tilt.

gave the highest measured heat transfer rate but Figs. 3 and 4 suggest that geometrical effects would not themselves change much in the range $10^\circ < \alpha < 80^\circ$. The results displayed in Fig. 5 reveal a monotonic decrease in heat transfer rate as the heated length–diameter ratio increases when the heated–cooled length ratio is fixed at 1.0. This is in agreement with the findings of previous work [6] which also revealed that the effect of heated–cooled length ratio was negligible when $\alpha = 0^\circ$; it is therefore reasonable to assume that similar behaviour would occur with the tubes tilted, especially in the range $0^\circ < \alpha < 90^\circ$.

Figure 5 also supports the earlier suggestion that a Rayleigh number of around $10^{6.5}$ marks a division between laminar impeded flow (for lower values) and turbulent mixed flow (for higher values) when the tilt is positive. The transition is evidently brought into sharper focus as the length–diameter ratio is increased. Lock and Ladoon [6] noted that fully-mixed turbulent data in the absence of tilt could be correlated in the form

$$Nu_D = A\left[Ra_D\left(\frac{D}{L_H}\right)^n\right]^m$$

and found that $n = 4$. A similar correlation for a linear tube was found by Lock and Simpson [11]. Using the high Rayleigh number data in Fig. 5, again reveals a correlation of this form with $n = 4$. As the figure illustrates, m is not strictly a constant for these data. However, it is seen to be a small number of order 0.1, thus indicating a weak dependence on Rayleigh number. To a first approximation, a designer could ignore the effect of Rayleigh number entirely for $Ra_D > 10^7$.

CONCLUSIONS

The paper describes the results of an experimental investigation into the thermal behaviour of a water-filled, elbow thermosyphon under single-phase conditions. The principal purpose of the investigation was to determine the heat transfer characteristics of the device when tilted. In keeping with previous work, the data have been used to explore regime behaviour and measure the effect of tube geometry on heat transfer. An attempt has been made to interpret behaviour in terms of a flow model previously developed from the behaviour of linear tilted thermosyphons and right angled, untilted thermosyphons.

The data discussed here were, by and large, consistent with observations on related thermosyphons. In particular, a steeper slope in the $Nu_D \sim Ra_D$ curves for positive tilt angles was taken as evidence of a laminar impeded flow regime when $Ra_D \lesssim 10^{6.5}$. For higher values of Rayleigh number, the data contained evidence of turbulent flow, and hence it is suggested that $Ra_D \simeq O(10^{6.5})$ marks the division between a laminar impeded regime and a fully-mixed turbulent regime. Laminar flow apparently prevailed for negative tilts which provided evidence of the conduction, impeded and boundary layer regimes.

The effect of tilt on heat transfer was found to divide into two ranges. For positive tilts, the heat transfer rate varied only slightly with $0^\circ < \alpha < 45^\circ$, but since symmetry is expected about $\alpha = 45^\circ$, this behaviour evidently extends over $0^\circ < \alpha < 90^\circ$. This implies flexibility in design. For negative tilts, a dramatic reduction in heat transfer rate was observed. This took place largely within the first 20° of negative tilt, and brought the heat transfer rate closer to values associated with pure conduction. Again there is a practical lesson for designers. From a fundamental standpoint, the negative tilt data strongly suggest that turbulence was absent, but they did not exhibit the local maximum previously found at $\alpha = -10^\circ$ for laminar flow with much lower Rayleigh numbers.

The positive tilt data also stood in contrast with previously obtained laminar flow data which show a local minimum in the heat transfer rate at $\alpha = 45^\circ$. This difference has also been attributed to turbulence in the experimental data at the higher Rayleigh numbers encountered. Corroborating this suggestion were data exploring the effect of tube geometry on heat transfer.

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