An Experimental Investigation Of Inclined Thermosyphon Fluid Flow Within An Enclosed Irregular Hexagonal Cavity With Heat Transfer Via Pin-Fins.

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Abstract- Evacuated Tube Solar Water Heaters (ETSWHs) have superior thermal performance in Northern Maritime Climates compared with Flat Plate Solar Water Heaters (FPSWHs) delivering 5% to 15% more energy per annum. ETSWHs with heat pipe absorbers are especially suited for use in Northern Maritime Climates due to their freeze tolerance and self-regulating thermal diode characteristics. ETSWHs have a higher capital cost than FPSWHs and have yet to be extensively tested and optimised under Northern Maritime climatic conditions. The adoption of thermosyphon fluid circulation in conjunction with ETSWHs has the potential to reduce capital and maintenance costs whilst increasing reliability, without affecting thermal performance. Previous research has reported that inclining thermosyphons influences flow structure and only focused on inclined cylindrical manifolds with a single orifice, and the inlet and outlet located at the same end of the inclined manifold. Other research into enclosed natural convection has focused on inclined rectangular cavities. This paper describes the construction of a truncated, 5-pin fin system, laboratory model of a proprietary ETSWH manifold. This manifold has separate orifices for the inlet and outlet, with an irregularly shaped 6-sided, rather than cylindrical or rectangular manifold cross section. This was incorporated into a collector loop and storage tank mimicking an operational thermosyphon solar water heating system. A range of existing empirical relationships for natural convection within enclosed horizontal cavities was compared to the experimental data. It was found that the best agreement was obtained for the case of natural convection from inclined horizontal smooth cylinders.

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1 INTRODUCTION

In 2004, 90.5% of primary fuel needs in the United Kingdom were met from the combustion of fossil fuels, (DTI Energy Statistics, 2005). Fossil fuel combustion releases carbon dioxide and other air pollutants, increasing their concentration in the atmosphere. This anthropogenic influence on the concentration of atmospheric carbon dioxide has the potential to increase mean global surface temperatures by 2.4 to 5.4K by the end of this century, (Murphy et al, 2004). Solar thermal technologies can be used to displace fossil fuels, hence avoiding their inherent problems, as the use of solar thermal technology does not affect the global climate nor emit pollutants during use, leading to air pollution problems such as acid rain or ozone depletion. Solar water heaters suitable for generating hot water in northern maritime climates can employ either flat plate or evacuated tube absorbers. Evacuated tube solar water heaters (ETSWH) have a higher efficiency as they have lower thermal losses than flat plate solar water heaters (FPSWH), (Morrison et al., 1984). Heat loss by conduction and convection to the external environment in an ETSWH is prevented, by enclosing the solar absorber in an evacuated, (10^{-5} bar), glass envelope. Coating the copper absorber plate of the ETSWH with a solar selective surface reduces radiative heat loss. Heat pipe ETSWHs utilise the unique thermo physical properties of heat pipes to transfer solar gain to the circulation fluid, can withstand freezing, have no moving parts, require no external pumping power and also act as thermal diodes, (Mathioulakis and Belessiotis, 2001). The system is completely self-regulating as the evaporation/condensation cycle within the heat pipe occurs more frequently at higher insolations and less frequently at lower insolations. Temperatures in excess of 453K may be achieved, especially in non-domestic applications requiring process steam. Under current economic conditions heat pipe ETSWHs have a higher capital cost compared with either FPSWHs or fossil fuel systems. Previous studies have reported that thermosyphon solar water heaters operate as effectively as pumped systems, but with lower initial and running costs, (Norton et al, 2001). For domestic use it was reported that thermosyphon solar water heaters were more reliable than equivalent pumped systems, (Harding and Zhiqiang, 1985). Thermosyphon systems have no moving parts and continue to operate when electricity supplies are interrupted. Thermosyphon flow arises from hydrostatic pressure differentials existing within the same collector loop, which initiate and maintain fluid circulation throughout the system. A number of investigations into thermosyphon fluid flow

regimes have been undertaken, Zhiqiang et al (1985), Behnia and Morrison (1991), Gaa et al (1996), Morrison et al (2004), Morrison et al. (2005). These studies were undertaken on non-evacuated glass-tubes, used to simulate all glass evacuated solar water heaters, with only one orifice, similar to a flow through absorber. The manifold of proprietary heat pipe ETSWHs differ from such inclined thermosyphon systems in that there are separate orifices for fluid inlet and outlet, and thermal energy is transferred from the condenser of the heat pipes to the heat transfer fluid. Previous investigations have studied the natural convective heat transfer regimes existing within other enclosed cavities such as inclined rectangular profiles with heated lower surfaces and cooled upper surfaces, Lee and Goldstein (1988) Lacroix and Joyeux (1996), Cesini et al (1999) Sezai and Mohamad (2000), Tso et al., (2004). This paper investigates a different heat transfer phenomenon where the distance between hot and cold surfaces is not constant, the cavity shape is irregular and six-sided and heat transfer is accomplished via an array of pin-fins which increase resistance to fluid flow through the manifold chamber producing additional interactions between the heat transfer fluid, cavity walls and pin-fins. A schematic of the 6-sided manifold and pin-fin arrangement is shown in Figure 1. There has been no published literature investigating natural convection within heat pipe ETSWH manifolds. This paper presents experimental data investigating the effect of manifold inclination on thermosyphonic flow and heat removal within an enclosed irregular shaped six-sided cavity.
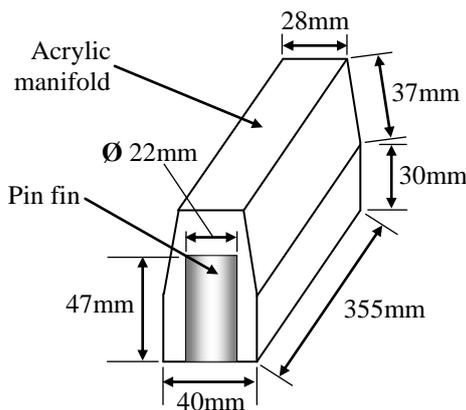

Figure 1 Schematic of Acrylic manifold and Pin fin

2 EXPERIMENTAL METHODOLOGY

The influence of the inclination of an ETSWH manifold on the thermosyphon fluid flow regime was examined by fabricating and testing a transparent laboratory model of a heat pipe ETSWH manifold chamber, under strictly controlled conditions. The heat emitted from the heat-pipe condenser, resulting from incident solar radiation, was simulated using electric pin-fin heaters, which provided the thermal energy input required to initiate and sustain thermosyphon flow. Simulating an equivalent incident solar radiation of $1000Wm^{-2}$ required a thermal input of 70W per pin-fin. The condenser in a propriety ETSWH had an exposed surface area in contact with the circulation fluid of 3.63×10^{-3} m^{-2} corresponding to a surface heat flux of $19284W/m^2$. The pin-fins used to simulate the condensers were constructed from a solid copper cylinder around a heating element. They were considered as pin-fins as heat transfer internally is via conduction and not through latent heat transfer processes. A physical model with the same profile as a proprietary manifold was constructed. This prototype acrylic manifold was fabricated by heating clear acrylic sheet and moulding it around a thin wire to the required shape and geometry as shown in Figure 2.
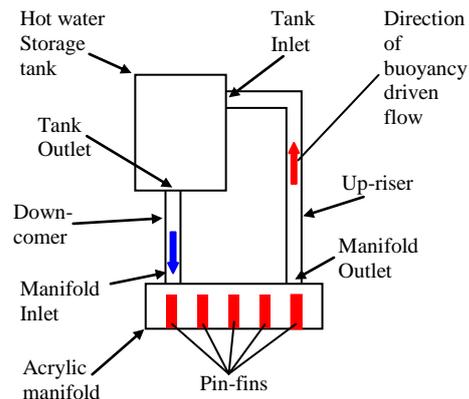
Figure 2 Thermosyphon flow circuit

The acrylic manifold was fabricated using five separate components, which were then bonded together using liquid acrylic adhesive, (methyl methacrylate), to produce the final manifold. The pin-fins were machined from solid copper bar. A platform was fabricated from medium density fibreboard (MDF) to enable the acrylic manifold to be supported and accurately inclined from the horizontal plane in two dimensions. The limitations and physical characteristics of acrylic sheet resulted in a number of minor material differences between the acrylic manifold and the proprietary ETSWH manifold. The ETSWH manifold was constructed from folded 316L stainless steel of thickness 8×10^{-4} m. The acrylic fabrication process generated larger internal

radius angles when folded and were not as sharp as those of the stainless steel ETSWH manifold. In addition, the fabrication process resulted in an acrylic manifold, which was not completely straight along its length.

The maximum length of the prototype acrylic manifold was 0.355m, limited by the fabrication apparatus. This truncated, laboratory model of the manifold enclosed 5 pin-fins and was connected to a storage tank of 50 litres. Temperatures were recorded using type-T thermocouples at the manifold inlet and outlet, at selected locations in the hot water storage tank, the back and sides of the acrylic manifold and the ambient air. The experimental apparatus was located within an internal windowless room to minimise fluctuations in ambient temperature from solar gains. Thermosyphon mass flow rates were measured using a magnetic induction flow meter. The experimental apparatus was arranged, as shown in Figure 2, and filled with mains tap water. After the system was filled it was allowed to settle for 24hrs until its internal temperature was within +/- 1K of ambient conditions. The electric pin-fin heaters in the acrylic manifold were then switched on and measurements commenced. Data was recorded using a multi-channel data logger. The thermosyphon flow rate was first measured when the manifold was in the horizontal position. The experiment was repeated 3 times to obtain a mean value and to ensure repeatability. This process was repeated for inclination angles of 0°, 5°, 10°, 15° and 20° from the horizontal. At 0° from the horizontal it was observed that the flow direction was reversed and that heat removal was at an optimum at 5° so the experiment was repeated for manifold inclinations of 0.5°, 0.7°, 1.0°, 2.5°, 6°, 6.5°, and 8° from the horizontal to determine the critical angle of inclination from the horizontal for these two phenomena.

3 RESULTS ANALYSIS

Heat removal from the acrylic manifold was calculated using Equation 1, (Cengel, 1998).

$$Q = mC_p(T_o - T_i) \qquad (1)$$

The calculated values for heat removal and angle of inclination are depicted in Figure 3. The optimum heat removal from the manifold occurs at inclination angles of 0.7° to 2.5°. The heat removal ranged from 0.97 to 0.78, decreasing at inclination angles of greater than 2.5° from the horizontal.
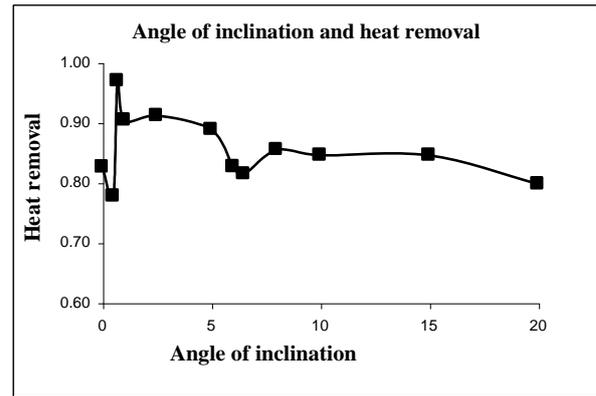
Figure 3 Angle of Inclination v heat removal

The convective heat transfer coefficients for the acrylic manifold at each angle of inclination were calculated using Equation 2, (Cengel, 1998).

$$Q = hA(T_s - T_\infty) \qquad (2)$$

To calculate the Rayleigh and Nusselt numbers a characteristic length had to be assigned to the acrylic manifold. For horizontal and inclined rectangular enclosures the characteristic length is defined as the distance between the hot and cold surfaces, (Cengel, 1998). This example of enclosed natural convection heat transfer differs from horizontal and inclined rectangular enclosures in that the cross section is an irregular six-sided shape and the distance between the hot and cold surfaces is not constant. The selected characteristic length will affect the calculated Grashof, Rayleigh and Nusselt numbers so two characteristic lengths were chosen for this example: the hydraulic diameter, (L1), and the average distance between the pin-fins and the manifold walls, (L2). The hydraulic diameter was calculated, using Equation 3, (Incropera and DeWitt, 1996), as 0.048m, the average distance between the pin-fins and manifold wall was 0.01425m. All fluid properties were evaluated at the average fluid temperature.

$$\delta = \frac{4A_c}{Wp} \qquad (3)$$

The Grashof number was calculated using Equation 4, (Incropera and DeWitt, 1996).

$$Gr = g\beta(T_s - T_\infty)\delta^3 / \upsilon \qquad (4)$$

The Rayleigh number was calculated using Equation 5, (Incropera and DeWitt, 1996).

$$Ra = Gr\,Pr \qquad (5)$$

The Nusselt number was calculated using Equation 6, (Cengel, 1998).

$$Nu = \frac{h\delta}{k} \qquad (6)$$

The calculated results were compared with an empirical correlation for natural convection within an enclosed horizontal cavity as depicted in Figure 4. The empirical correlation was calculated using Equation 7, (Cengel, 1998).
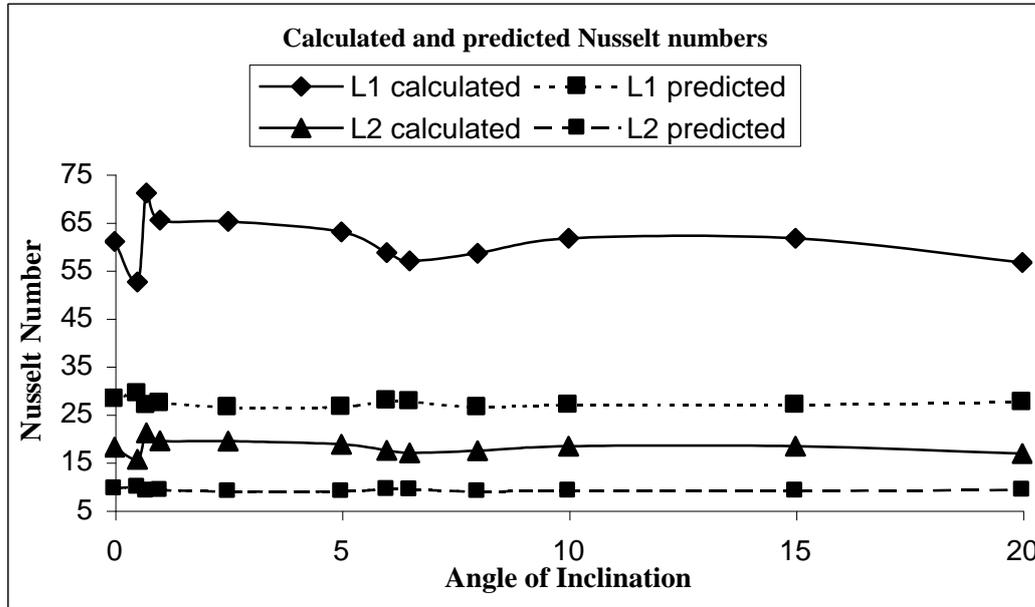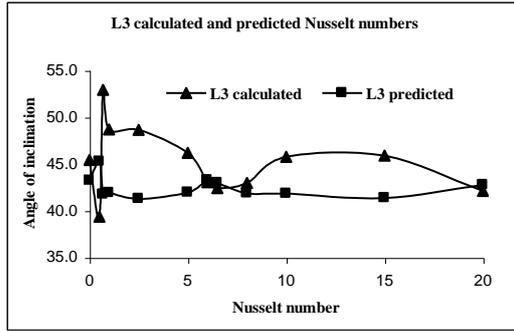

Figure 4 Calculated and predicted values

$$Nu = 0.13\,Ra^{0.3} \qquad (7)$$

$$Nu = 0.125\,Ra^{0.333} \qquad (8)$$

Using the hydraulic diameter, (L1), as the characteristic length of the acrylic manifold the difference between the calculated and predicted Nusselt numbers ranged from 44% to 62%. For characteristic length, (L2), the difference between the calculated and predicted Nusselt numbers ranged from 37% to 57%. Using L1 as the characteristic length produced Rayleigh numbers in the range 4.91×10^7 to 6.94×10^7. In this range Tso et al, 2004 reported a Nusselt number of 9. Characteristic length L2 produced Rayleigh numbers in the range 1.29×10^6 to 1.82×10^6. For this range Tso et al, 2004 reported a Nusselt number of 6.2. Average local Nusselt (Nu_L) numbers were calculated for each angle of inclination. Two characteristic lengths were considered, the diameter of the pin fin divided by the cosine of the inclination angle of the cylinder, (L3), and the diameter of the pin fin, (L4). The calculated values for L3 were compared with an empirical correlation for natural convection from an inclined horizontal cylinder (Equation 8, Morgan, 1975), fluid properties were evaluated at the film temperature.

The calculated values for L4 were compared with the correlation suggested by Zhukauskas (1972), (Equation 9), for a cylinder subjected to cross flow, using fluid properties evaluated at the average fluid temperature. The relationship between the calculated and predicted values of the average local Nusselt number for characteristic lengths L3 and L4 are presented in Figures 5 and 6 respectively. The empirical relationship for L4 was calculated using Equation 9. Under the prescribed experimental conditions the values of C, m, and n used were, 0.51, 0.5 and 0.37 respectively.

$$Nu_L = C\,Re_D^{m}\,Pr^{n}\left(\frac{Pr}{Pr_s}\right)^{1/4} \qquad (9)$$

L3 calculated and predicted Nusselt numbers

Figure 5 Calculated and predicted values

L4 calculated and predicted Nusselt numbers
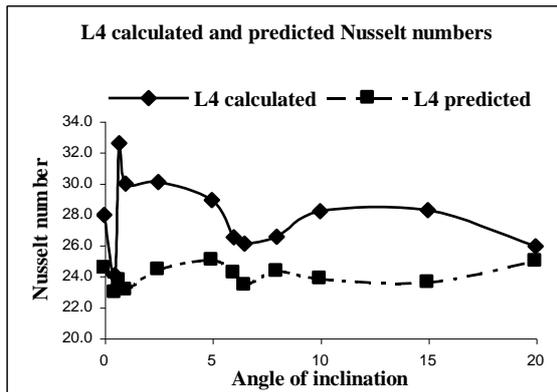
Figure 6 Calculated and predicted values

As depicted in Figure 5 the difference between the calculated and predicted average local Nusselt numbers for L3 using Equation 8, ranges from 0.5% to 21%. The largest discrepancies between the calculated and predicted Nusselt numbers occur between inclination angles 0.7° to 2.5° (14% to 21%). The difference between the predicted and calculated values varies by no more than 10% for the other inclination angles considered. As depicted in Figure 6, the difference between the calculated and predicted average local Nusselt numbers for L4 ranges from 4% to 27%. The largest discrepancies between the calculated and predicted average local Nusselt numbers occur at inclination angles 0.7° (27%) and 1° (23%). The predicted and calculated values vary by no more than 19 % for the other inclination angles considered.

4 CONCLUSIONS

It was experimentally determined that inclining the acrylic manifold at 0.7° to 2.5° from the horizontal produced the optimum heat removal for the angles examined. As the acrylic manifold's inclination from the horizontal increases above 2.5°, thermal losses increased.

The calculated Nusselt numbers for the acrylic manifold differed significantly from those predicted by the available correlations for enclosed natural convection within an enclosed horizontal cavity with a heated lower surface and cooled upper surface. This suggests that the pin-fins enhance heat transfer within cavities and that a new correlation derived from experimental data is required to accurately predict the average Nusselt number for irregular six sided manifolds. The correlation suggested by Morgan (1975) was more accurate in predicting average local Nusselt numbers, (average difference 9%), when compared to the correlation suggested by Zhukauskas (1972) predicting the average local Nusselt numbers for cylinders subjected to cross flow, (average difference 14%). A detailed thermofluid analysis validated by Particle Imaging Velocimetry (PIV) will be required. A 3-dimensional PIV analysis will provide a visual record of the instantaneous fluid flow regime providing detailed experimental data. The use of a hot water storage tank, with a more advantageous height to diameter ratio than the current hot water storage tank used in this experiment will promote increased thermal stratification and provide more accurate results due to reduced variation in inlet temperature.

NOMENCLATURE

A_c Area (m^2)
β Coefficient of expansion (K^{-1})
C_p Specific heat (J kg^{-1}. K^{-1})
D diameter (m)
δ Characteristic length (m)
g Gravity (m s^{-2})
Gr Grashof number
h Convective heat transfer coefficient (W/m^{2}K)
k Thermal conductivity (W/mK)
Wp Wetted perimeter (m)
m Mass flow rate (kg s^{-1})
Nu Nusselt number
P Density (kg m^{-3})
Pr Prandtl number
Q Heat Flux (W)
q Heat flux density (W/m^2)
Re Reynolds number
Ra Rayleigh number
T Temperature (K)
v Kinematic viscosity (m^{2}s^{-1})

SUBSCRIPTS

c	cross sectional
D	Diameter
i	Inlet
L	Local
0	Outlet
p	Pressure
s	Surface
∞	Free stream conditions

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