COMPARATIVE STUDY AND THE GEOMETRY EFFECTS ON THE THERMAL PERFORMANCE OF OFFSET AND LINEAR THERMOSYPHONS

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ABSTRACT

An experimental study on the effect of geometric parameters on the thermal performance of both offset and linear water closed thermosyphons was performed. The cooled-heated length ratio Γ_1 has been varied from 1 to 3, and the heated length-diameter ratio Γ_2 from 8 to 24.It was found that the offset thermosyphon exhibits lower thermal performance when compared with the linear system of the same geometric length and heat flux density. At the same time, it operates with higher Rayleigh number. Analysis of the experimental data revealed the weak effect of the ratio Γ_1 on the thermal performance in case of the linear thermosyphon. This effect becomes stronger when an offset system is tested, for which Nusselt number increases with the ratio Γ_1. Also, it was proved that the Nusselt number is in an inverse function of the heated length-diameter ratio Γ_2 for both configurations. The mechanism explaining the thermal behaviour of both thermosyphons is described. Two empirical correlations exploring the effects of geometry and Rayleigh number on the heat transfer were deduced. These correlations support the suggestion of the stated mechanism and allow prediction of the heat transfer performance for both systems.

Keywords: Thermosyphon Offset linear.

Nomenclature

a	thermal diffusity	(m^2/s)
d	inner diameter,	(m)
D	outer diameter.	(m)
g	gravitational constant	(m/s^2)
L	length	(m)
h	heat transfer coefficient	(W/m^2 °C)
k	thermal conductivity	(W/m.°C)
v	working fluid volume	(ml)
Nu	Nusselt Number	
Ra	Rayleigh number	
Q	Input power,	(W)
q	heat flux	(W/m^2)
T	Temperature,	°C

Greek Symbols

ν	kinematic viscosity	(m^2/s)
β	thermal expansion coefficient	(1/K)
Γ_1	L_c/L_H, cooled - heated length ratio	
Γ_2	L_H/d, heated length to inner diameter ratio.	

Subscripts

a	adiabatic
c	condenser, cooled
f	reference
H	heated.

INTRODUCTION

A thermosyphon is a loop system in which fluid circulates due to the natural convection caused by heat generated usually in the lower part of the loop and cooled from above, which then establishes unstable density gradient. Utilizing the buoyancy forces on the fluid, the lighter hot fluid rises and due to the gravity, the heavier cold fluid falls and the fluid is said to flow due to natural convection.

Thermosyphons are of interest in industry because they have been successfully applied to a variety of technological problems concerning the transfer of heat without pumps. Most of these applications are above the ground, such as in solar hot systems, cooling of transformers, car internal combustion engines, heating of fish farming tanks, and emergency cooling of nuclear reactors.

Underground applications include permafrost protection , geothermal energy highway de-icing and heating the soil under cold stores, etc.

Many of the early and current applications are based upon circular toroidal thermosyphons [1-6] , and linear shaped loops heated at one end and cooled at the other [7-12] . This simple arrangement limits the use to situation in which the heat source and sink may be thus connected. In the last few years situations appeared in which the size, shapes and locations of the source and sink demanded the use of nonlinear loops. Thus , the urgent need for additional configurations for these uncommon and new applications has rapidly increased.

To date , the most constructive effort in this subject has been made by Lock and his group. In this concern Lock and Ladoon [13-14] studied the influence of the tilt angle on natural convection in a water filled right-angled elbow thermosyphon for Rayleigh number less than 10^7. The obtained data indicated the existence of fully mixed and impeded regimes of flow . Here , again , the designer is restricted to simple common circumstances in which the stationary heat source lies immediately adjacent and largely beneath the heat sink. The cranked or offset tubular thermosyphon is one example of a variety of different configurations of highly promising flexibility for various applications . In such a configuration the cranked thermosyphon consists of two parallel sections- one heated and the other cooled separated by an adiabatic section.

Very little information is available on the cranked or offset thermosyphons. The most interesting studies are those done by Lock et.al [15-17]. They studied, the effect of the tilt angle, Rayleigh number, and flow regimes on natural convection in a cranked water filled single - phase thermosyphon. In their research, the heated - cooled length ratio was kept constant ($L_c/L_H = 1$), and the flow regime describing the thermal behaviour was based on studies given in references [18 and 19]. The present study extends the above - mentioned works to an offset two phase thermosyphon. The objective of this work is to obtain experimental information on the interactive influences of the Rayleigh number and the geometric effects, in the form of cooled-heated length ratio and the heated length-diameter ratio, on the heat transfer of an offset two phase closed thermosyphon. For the purpose of comparison, additional experiments have been conducted using a linear configuration thermosyphon having the same geometry and operating at similar test conditions.

EXPERIMENTS

Test Facility

A Schematic of the test rig which was built in the Heat Transfer Laboratory of Mechanical Power Engineering Department at Zagazig University is shown in Figure (1).The experiments have been conducted using 20mm ID and 26mm OD copper tubes joined in linear and offset configurations, both of the same geometric length. The tested thermosyphon consisted of condenser (cooled), adiabatic, and evaporator (heated) sections. In fact, the lengths of the condenser and adiabatic sections were fixed at 48 and 16 cm, respectively. The evaporator sections had the length of 16,32 and 48 cm and was simply varied by attaching or detaching additional lengths using threaded coupling pieces.

A 48 cm long PVC tube which has outer and inner diameters of 60 and 54 mm , respectively was set on the condenser section simulating a cooling jacket using the water as a coolant to remove the heat from the thermosyphonic system. Two inlet and outlet PVC tubes of 2cm outer diameter were directed tangentially to the inner surface of the cooling jacket. The cooling jacket was fed with water from the building water main via a water regulating valve by means of which the water flow rate was adjusted. The heating of the evaporator was provided directly by a constant and uniform heat flux being created by 500 Watt nickel chromic resistive heater. The evaporator in its maximum length (0.48 m) - which consisted of two parts whose lengths were 16 and 32 cm.- had been heated by two heaters connected in parallel, thus giving independent control over the local heat flux density (q) and the temperature along the tube wall.

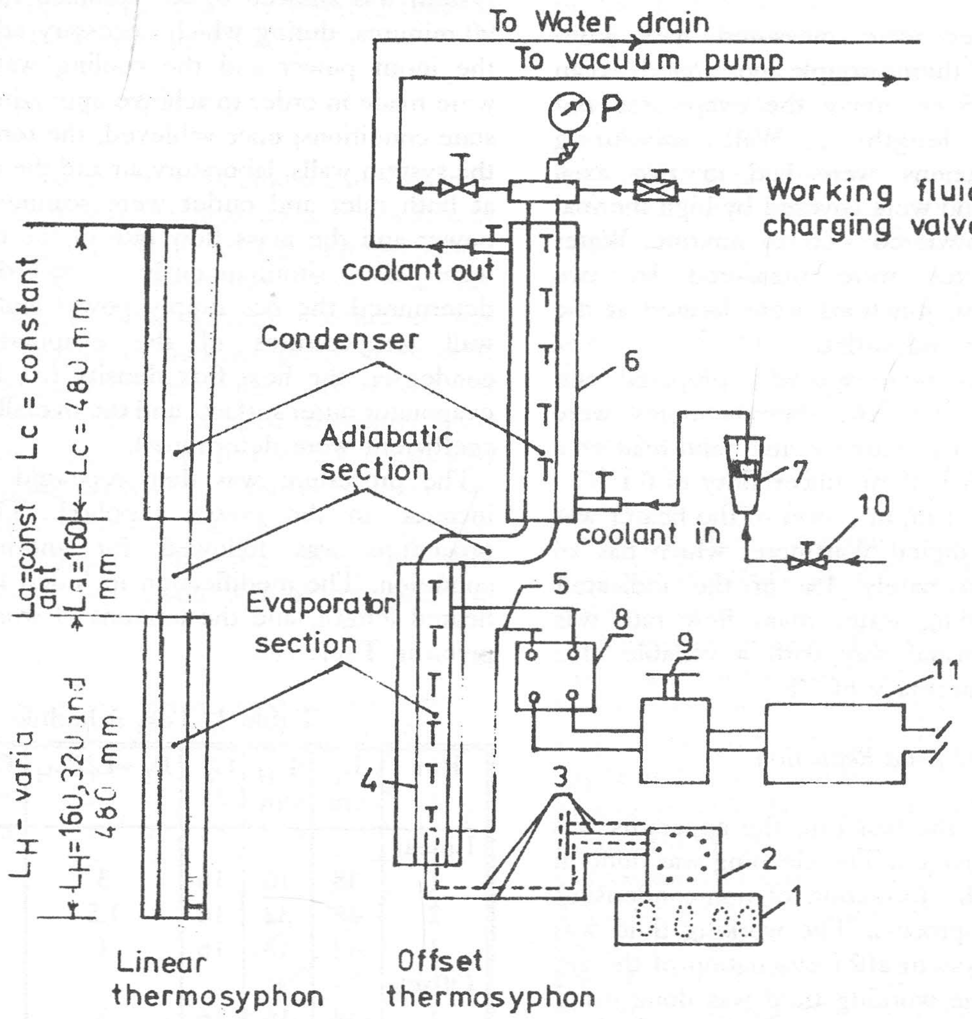

Figure (1) A Schematic diagram of the experimental set-up

1. Digital thermometer
2. Selecting switch
3. Thermocouple leads
4. Thermosyphon tube
5. Heater leads
6. Cooling jacket
7. Rotameter
9. Variac
10. Coolant flow control valve.
11. A.C. Power source.
P. Pressure gauge.
T. Thermocouple

Chromic electrically resistant wire tapped inside small ceramic rings which were knotted and uniformally wrapped circumferentially around the entire evaporator length. A reflective aluminum foil sheet was set to enclose the evaporator with the heater to ensure maximum reflection of the heat generated by the heater to the evaporator surface. A 50 mm thickness of glass-wool thermal insulation followed by additional reflective aluminum foil sheet were then wrapped. This construction ensured the minimum heat loss to - or gained from - the surrounding atmosphere. The adiabatic section for offset configuration was horizontally joined to the vertically aligned condenser and evaporator sections by means of 20 mm ID bend copper elbows. It is worth noting that the cooling jacket, the elbow pieces and the offset adiabatic section were also wrapped in thick insulating layer to approximate an adiabatic condition.

Measurements

Wall temperatures were measured with iron-constantan type J thermocouple positioned at an equal interval of 5 cm along the evaporator and condenser wall length . Wall measuring thermocouple junctions were laid in two axial opposite grooves, and were covered by high thermal resistive copper powdered - epoxy mixture. Water cooling temperatures were measured by two thermocouple whose junctions were located at the cooling jacked inlet and outlet.

All thermocouples were specially prepared and calibrated. Signals from the thermocouples were channeled through a selecting switch and read on a digital thermometer with an uncertainty of 0.1 °C.

The gross electrical input power of the heater was measured using a digital Wattmeter which has an accuracy of approximately 1% of the indicated readings. The cooling water mass flow rate was measured in the usual way with a variable area rotameter with an accuracy of 2% .

Test Procedure and Data Reduction

Before beginning the test run, the apparatus was first cleaned and charged. The cleaning was done in order to prevent the formation of non-condensing gases in operation process. The working fluid was injected into the system after evacuation of the air. The injection of the working fluid was done using hypodermic syringe in order to ensure adequate amount of charged fluid. In the present study, distilled water was chosen as the working fluid , since it is compatible with copper and safe to work with.

Prior to undertaking the main series of the experiments some preliminary experiments were conducted in order to estimate the heat loss to the atmosphere. This loss was found to be close to 5-7% of the input heater power depending on the evaporator wall and laboratory air temperature difference. The main test procedure was as follows: having chosen the selected configuration and the evaporator length the apparatus was filled with an adequate amount of the working fluid as given in Table (1).

The cooling water was turned out and the power to the heater tap was set at a convenient value. The system was allowed to be stabilized for a period of 90 minutes, during which necessary adjustments in the input power and the cooling water flow rate were made in order to achieve approximately steady state conditions, once achieved, the temperatures of the system walls, laboratory air and the cooling water at both inlet and outlet were scanned. The gross power and the mass flow rate of the cooling water were also simultaneously recorded. Having determined the net supply power and the average wall temperatures of the evaporator and the condenser, the heat flux density (q), based on the evaporator outer surface, and the overall heat transfer coefficient were determined.

The procedure was then repeated following an increase in the power supplied. The complete procedure was followed for another modified condition. The modification included the change in heated length, and the amount of working fluid as given in Table (1).

Table 1. Test Schedule

Run	L_c cm	L_H cm	L_a	$\Gamma_1 = L_c/L_H$	$\Gamma_2 = L_H/d$	v ml
Linear						
1	48	16	16	3	8	15
2	48	32	16	1.5	16	30
3	48	48	16	1	24	45
Offset						
1	48	16	16	3	8	15
2	48	32	16	1.5	16	30
3	48	48	16	1	24	45

Table (1) gives a complete survey of the geometric variables. In addition, the amount of working fluids was varied from 15 to 45ml corresponding to the fill ratio charge volume to inner volume of evaporator zone of 33%, as outlined by Lock any Jialin [17]. The heat flux density was varied from 2000 to 12000 W/m^2.

The physical properties of water in non-dimensional Nusselt and Rayliegh numbers:

$$Nu = q.\ d/k\ (t_H - t_C),\ Ra = \beta\ g\ (t_H\ t_C)\ d^3/\nu.a$$

were taken based on the reference temperature t_f, which is defined as an average of the evaporator and the condenser wall temperatures. Hence, it can be considered as the temperature of working fluid as recommended by Kanji and Teruo [11].

DISCUSSION OF RESULTS

The primary analysis of the obtained data shown in Figure (2) reveals that the heat transfer coefficient has the same trend for all the tested thermosyphons, it increases with the heat flux. The relatively maximum value of heat transfer coefficient was found to the belong to thermosyphon of shortest heated length irrespective of its configuration. At constant heat flux, linear shaped thermosyphons exhibit higher values of heat transfer coefficient compared to those of offset configuration. These can be briefly attributed to the greater evaporator-condenser temperature difference associated with offset thermosyphons which will be explained in detail later.

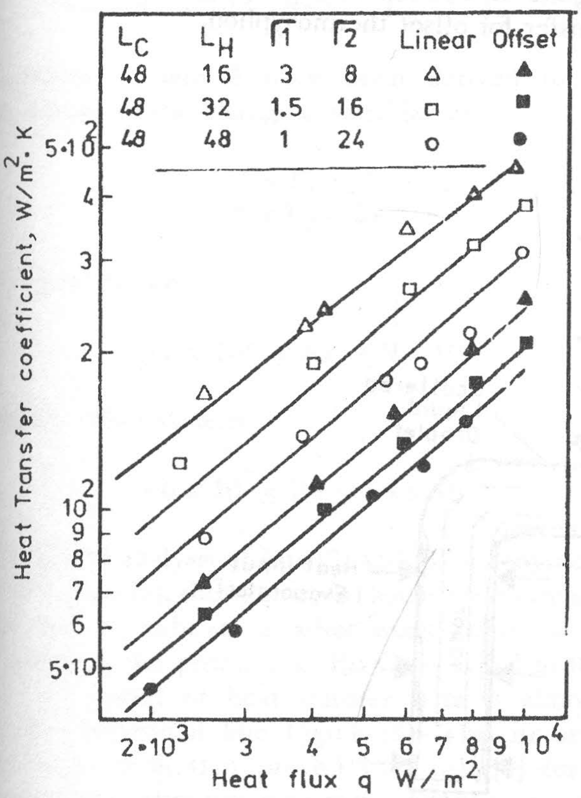

Figure 2. Heat transfer coefficient versus the input heat flux for tested thermosyphons.

Figures (3) and (4) show the experimental results of the tested thermosyphons represented in the usual form of Nu vs Ra. From these figures it is evident that for all tested cases, Nusselt number Nu increases with Rayleigh number. It has been found that, for high Rayleigh numbers, the buoyancy forces are sufficiently intense near the wall, so that boundary layer regime is obtained [18]. In such regime, an annulus of fluid flows from both heated and cooled ends toward the junction region. At the same time, the central cores flow in the opposite direction replenishing the annuli with working fluid, thus preventing or minimizing the dryness of evaporator section. This results in higher values of Nusselt number.

As the heat fluxes are reduced, the buoyancy forces are less and the effects of wall shear forces becomes significant throughout the tube. Hence, compared with the boundary layer flow, the flow is impeded resulting in low Nusselt number.

The global judging of Figures (3) and (4) reveals that offset shaped thermosyphons exhibit rather lower values of Nusselt number compared to linear shaped ones at fixed geometric length. This can be explained by understanding the flooding phenomenon of both thermosyphon configurations, Figure (5). In the case of linear thermosyphon, the evaporator end acts as a cavity and the liquid does not form pools as the liquid drops were scattered on the wall. The liquid drops were distributed from one end to the center of the heating region and evaporated. In the condenser, the condensate returns by gravity to the evaporator forming some rather thickened rivulets, which ensure the continuity of the heated surface wetness. When the offset thermosyphon was tested, the rate of scattered liquid droplets on the horizontal adiabatic section was increased, thus creating a stable stratified liquid layer on its lower surface. The result is the minimum liquid supply to the heated section in the form of thin rivulets, and finally dryness may be obtained. This causes significant evaporator-condenser temperature difference which in turn leads to the deterioration in heat transfer. Moreover the scattered liquid droplets in adiabatic section which evaporated by the heat transferred by conduction leads to the partially blocking the vapor pass to the condenser causing an additional increase in evaporator-condenser temperature difference. This in turn leads to considerable deterioration in heat transfer. Also due to significant values of $\Delta t_{H\text{-}C}$ associated with an offset thermosyphon, it operates at relatively higher Rayleigh number compared with linear configuration system.

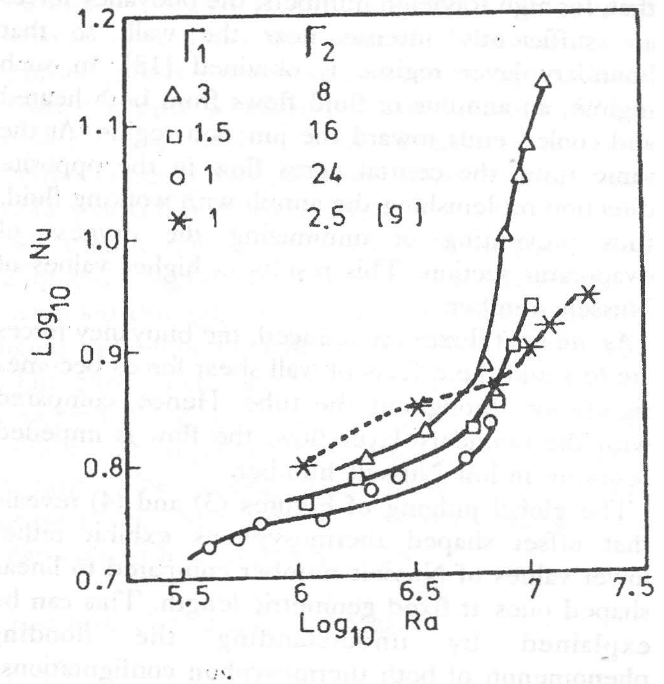

Figure 3. The effect of Rayleigh number on heat transfer for linear thermosyphon.

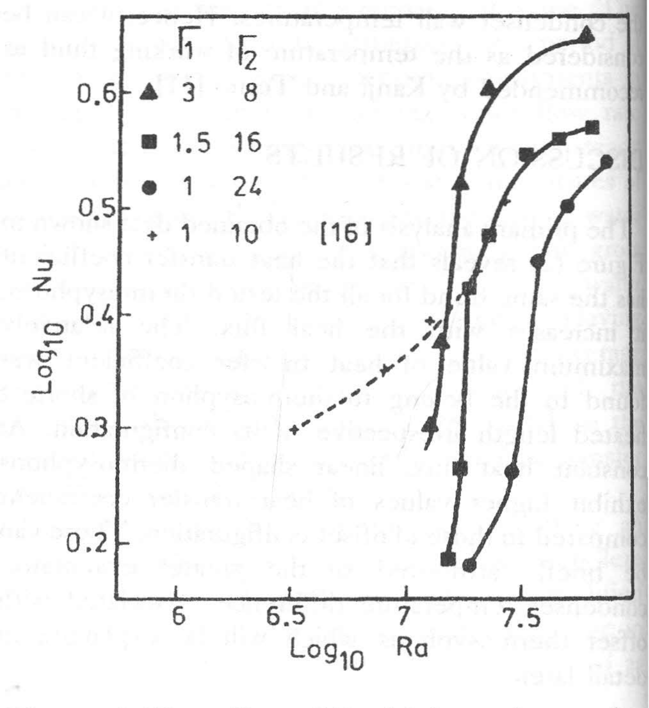

Figure 4. The effect of Reyleigh number on hea transfer for offset thermosyphon.

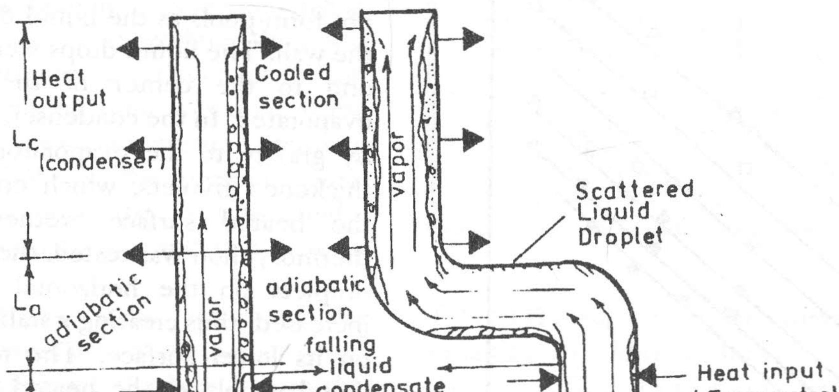

Figure 5. Approximated mechanism in the closed thermosyphons (coordinate system not to scale).

The effect of geometry on heat transfer can be clarified by dividing it into two major parts: the effect of cooled-heated length ratio and the effect of heated length-diameter ratio which are denoted by Γ_1 and Γ_2 respectively. Before attempting to interpret the influence of the ratios Γ_1 and Γ_2 on the heat transfer, it is worthwhile to correlate the experimental data obtained for various ratios Γ_1, and Γ_2 incorporated directly into Rayleigh number Ra that is:

$$Nu = CRa^n \cdot \Gamma_1^{\,m} \cdot \Gamma_2^{\,k} \tag{1}$$

Ignoring the transition data and following the least square method we have:
for linear thermosyhons:

$$Nu = 0.71 \cdot Ra^{0.25} \cdot \Gamma_1^{\,0.024} \cdot \Gamma_2^{\,-0.57} \tag{2}$$

for offset thermosyphon:

$$Nu = 5.2 \times 10^{-7} \cdot Ra^{0.91} \cdot \Gamma_1^{\,0.25} \cdot \Gamma_2^{\,-0.11} \tag{3}$$

Equations 2 and 3 have been derived for the conditions of the changing variables as:

$$1 \le \Gamma_1 \le 3$$
$$8 \le \Gamma_2 \le 24$$

for linear system

$$5.2 \times 10^5 \le Ra \le 9 \times 10^6$$

and for offset system

$$1.6 \times 10^7 \le Ra \le 5.4 \times 10^7$$

The global analysis of Eqs (2) and (3) in conjunction with Figure (6), shows that, for linear thermosyphon, the ratio Γ_1 exhibits a rather weak effect on heat transfer, as the power n in Eq (2) is equal to 0.024 and the trend of heat transfer data is almost a straight horizontal line Figure (6). This finding is consistent with that proved by Lock [9] for the linear thermosyphons who suggested that it can be expanded to the offset one. However by way of contrast to the Locks suggestion the ratio Γ_1 was found to have a rather stronger effect when offset thermosyphons were tested. The power n in Eq (3) is equal to 0.25 and trend of heat transfer increases with ratio Γ_1. In fact, the varying of $\Gamma_1 = (L_c/L_H)$ by changing the heated length - at constant L_c - is done simply to change the performance of the reservoir that supplies cold fluid to the heated section. So as, Γ_1 increases the descending cold flow-condensate - in the cooled section will be thicker than the heated ascending one (due to the increased capability of the condenser), thus causing relatively reasonable feeding by working fluid to the heated section. On the contrary, as Γ_1 decreases at relatively greater heated length, the cold flow should be thinner than the ascending one, and the scattering of the liquid on the horizontal adiabatic section should be significant. This leads to the dryness of the evaporator section which can be viewed as a tendency toward increasing the overall thermal impedance of the system.

The effect of the heated length-diameter ratio Γ_2 on heat transfer was found to be more dominant in linear thermosyphons and to have less significant in offset thermosyphons as can be judged from Figures (3),(4) and (7). These figures show monotonic decrease in heat transfer with an increase of the heated length-diameter ratio. This briefly can be attributed to the damping effect expected with longer tubes.

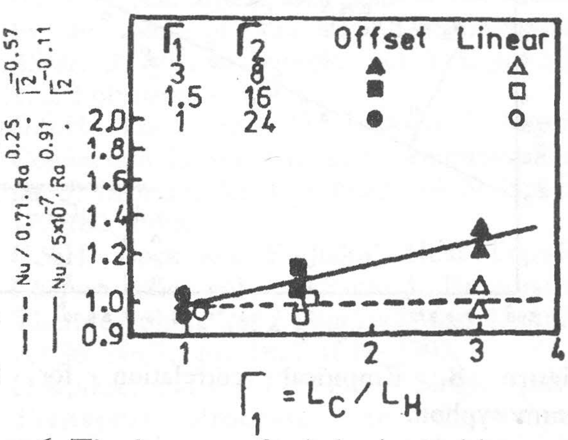

Figure 6. The heat transfer behaviour with respect to the cooled-heated length ratio $\Gamma_1 = L_c/L_H$.

Figures (8) and (9) show the correlation of the experimental data. In these figures, all the obtained data has been replotted according to Eqs (2) and (3).

From Figures (8) and (9) it is evident that the equations (2) and (3) fit the majority of the experimental data with average error of 10-20 % which is acceptable for such conditions.

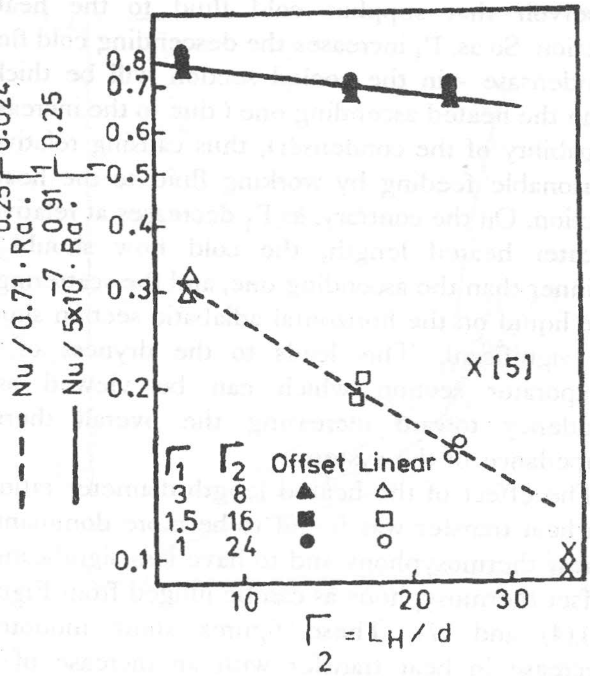

Figure 7. The heat transfer behaviour with respect to the heated length-diameter ratio $\Gamma_2 = L_H/d$.

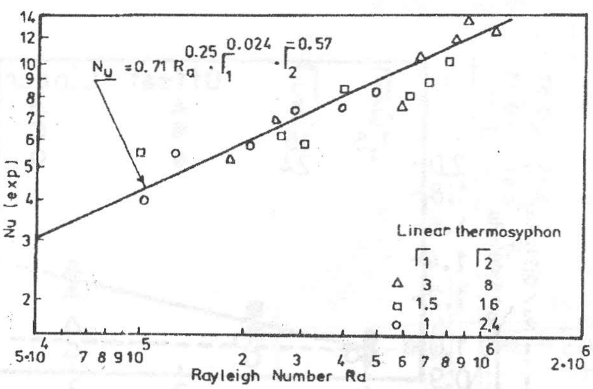

Figure 8. Empirical correlation for linear thermosyphon.

Comparison with various results obtained by previous investigators [9] and [16] on linear and offset shaped thermosyphons of approximately similar geometric characteristics is presented in Figures (3) and (4) respectively. The difference between the present data and those of previous works is likely to be due the systematic physical effect resulting from dissimilarity of geometric characteristics of the studied thermosyphons. Also, can be attributed to the experimental error.

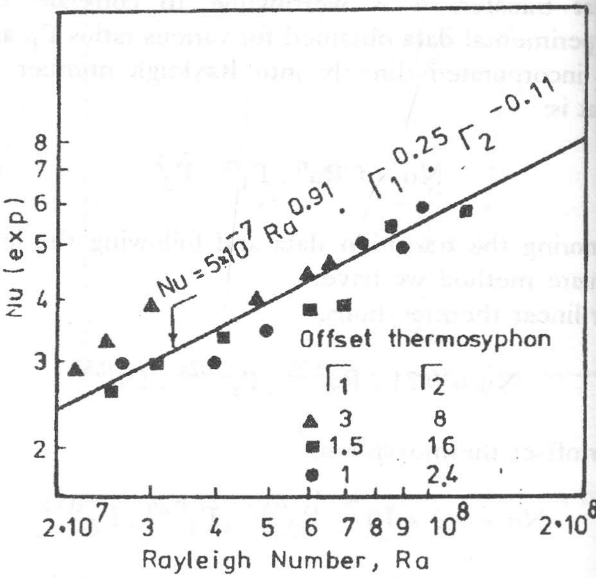

Figure 9. Empirical correlation for offset thermosyphon.

CONCLUSIONS

An experimental study was conducted on the effect of geometric characteristics on the thermal performance of linear and offset two-phase closed thermosyphons having the same geometry and operating at the same heat flux density. Water was used as the working fluid. The results are as follows:

- An offset thermosyphon exhibits a rather lower value of Nusselt number compared to a linear system of the same geometry length. The offset system operates at relatively higher Rayleigh number necessary to attain any given heat flux level.

- Analysis of the experimental data showed that the cooled-heated length ratio has a weak effect on heat transfer in case of linear thermosyphons. This effect was found to be stronger when an offset system was tested. In this case, the Nusselt number increases proportionally with this ratio.

- Nusselt number has been proved to be in an inverse function of the heated length diameter ratio for both configurations.
- The mechanism describing the behaviour of both tested thermosyphons with respect to the variable geometry and changing Rayleigh number have been approximately stated.
- In the light of the obtained experimental data, two empirical correlations have been derived. The obtained correlations support the suggestion of the stated mechanism and allow prediction of the Nusselt number with reasonable accuracy.
- In spite of the relatively lower thermal performance of the offset thermosyphon compared to the linear system, it gives the designer great flexibility for a wide variety of applications.

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