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INVESTIGATION THE EFFECT OF ASPECT RATIO ON THE FREE HEAT CONVECTIVE INSIDE A HORIZONTAL HEATED TUBE

Saad Najeeb Shehab*

Lecturer, Mechanical Engineering Department, Al-Mustansiriyah University, Baghdad, Iraq.

Abstract: In this work, an experimental investigation has been done for steady-state free heat convection into a heated externally horizontal open ends circular tube with and without internal ring fins. The tests are carried out for five different aluminum tubes in aspect ratios. Another four tubes are used providing internal ring fins with varying fin spacing and constant aspect ratio . Wide levels of the wall heat fluxes (254 - 2267) W/m^2 are used. The influence of aspect ratio , heat flux , ring fins , fin spacing on the free-convection through a heated tube are analyzed and discussed. The experimental results indicate that the local Nusselt number increases with decreasing the aspect ratio. Also the results appear that the free heat convection performance through the tube in terms of local Nusselt number improved with internal fins about 25% higher than without fins. An experimental correlations of local and average Nusselt numbers as a function of Rayleigh number for a wide range of aspect ratios under uniform wall heat flux condition are concluded.

Keywords: *Aspect ratio , horizontal tube , free-convection , internal ring fins , experimental investigation.*

دراسة تأثير نسبة الطول الى القطر على الحمل الحراري الحر داخل أنبوب أفقي مسخن

الخلاصة: في هذا البحث أجريت دراسة عملية للحمل الحراري الحر في الحالة المستقرة داخل أنبوب دائري مسخن من الخارج مفتوح النهايتين بوجود وبدون وجود زعانف حلقية داخلية . أجريت الأختبارات لخمسة أنابيب من الألمنيوم مختلفة في نسبة الطول الى القطر. كما أستخدمت أربع أنابيب أخرى تحتوي على زعانف حلقية داخلية بمسافات مختلفة للزعانف ومتساوية بنسبة الطول الى القطر. أستخدمت عدة مستويات للفيض الحراري لجدار الأنبوب (254 - 2267) واط/م^2. تم تحليل و مناقشة تأثير كل من نسبة الطول الى القطر و الفيض الحراري ووجود الزعانف الحلقية و المسافات البينية بينها على الحمل الحر داخل الأنبوب المسخن . تشير النتائج العملية بأن رقم نسلت الموقعي يزداد مع نقصان نسبة الطول الى القطر. كذلك بينت النتائج أن أداء الحمل الحراري الحر خلال الأنبوب بدلالة رقم نسلت الموقعي بوجود الزعانف الحلقية الداخلية يزداد بحدود 25% مقارنة بعدم وجود الزعانف. أستنتجت علاقتين أرتباطيتين عمليتين لرقم نسلت الموقعي والمعدل كدالة لرقم ريليه لمدى واسع من نسب الطول الى القطر بثبوت الفيض الحراري.

1. Introduction

The cooling and heating by free-convective heat transfer has many industrial and thermal engineering applications like cooling of the electrical and electronic apparatuses , heat pipes , passive cooling systems of the nuclear reactors ,

* saad_najeeb16@yahoo.com

engineering of thermal insulation , furnaces , solar collectors , compact heat exchangers , electrochemical processes , refrigeration systems , oil recovery processes , systems of energy storage , ground thermosyphons , ceramic processes and many others.

Dong and Ebadian [1] analyzed numerically using finite difference method (FDM) the combined free and forced convection through vertical semi-circular duct with and without radial internal longitudinal fins. They noted that the Nusselt number and friction factor in internally finned duct increase as Rayleigh number increases and heat transfer rate is enhanced dramatically for internally finned duct. Al-Sarkhi and Abu-Nada [2] presented a numerical analysis of forced heat convection from a vertical internally finned cylinder using the finite volume method (FVM) to solve the governing equations.

They deduced that the velocity and temperature distributions into the cylinder depend on the height and number of the internally rectangular fins. Salman and Mohammed [3] performed experimental investigation for laminar natural convective into an externally heated vertical open-ends cylinder under constant heat flux with restriction type effects in cylinder entrance.

They used two different configurations of restrictions namely, sharp edge and bell mouth. They showed that the Nusselt number values are higher for bell mouth restriction type and developed a general empirical correlation joined all studied cases. Ali [4] presented an experimental investigation for steady state free-convection in vertical ducts under constant heat flux. He used square and rectangular cross sections of ducts. He correlated average Nusselt numbers with modified Rayleigh numbers and area ratios for laminar regime. Ali [5] studied numerically the natural-convection through square enclosure with thin fins to improve heat transfer rate. His study used the finite difference method (FDM) to solve the governing equations and studied three cases namely, single , two and three of rectangular thin fins for different Rayleigh numbers in range of 10^3 to 10^4. The results observed that the Nusselt number and temperature gradient increases by fins number decreasing and Rayleigh number increasing.

Bakhti and Siameur [6] studied numerically using finite volume method (FVM), the laminar mixed-convective inside an inclined thick circular duct with uniform outer surface heat flux. They used wide range of Grashof numbers and inclined angles with horizontal position from 0^o to 60^o. They concluded that the heat transfer by convection is better for higher values of Grshof number and for large inclination angles.

Roul and Nayak [7] investigated experimentally the heat transfer by natural convection inside a heated vertical tube with constant heat flux. They used a wide ratios of length to diameter of tube and different discrete rings inside the tube. They found that the higher values of temperatures wall and convection heat transfer coefficient are obtained with higher heat flux and higher ratio of length to diameter. They also noted that better heat transfer rate can be obtained with tube including discrete rings. El-Sayed et al. [8] studied experimentally the convective heat transfer through an internally finned circular tube.They utilized three configurations of internally fins, staggered and in-line distributions of longitudinal continuous and interrupted. They used two fins numbers, 6 and 12 and all tests carried out at uniform surface heat flux. They noted that the Nusselt number values for staggered and in-line arrangements when number of fins equal 6 are greater than that for fins number equal 12. Also, they concluded that the convective heat transfer coefficient values for circular tube contains continuous fins are higher than that the circular tube with interrupted fin.

Fakoor-Pakdaman and Bahrami [9] developed an analytical model to predict an unsteady-state laminar forced heat convective into a circular tube under a step surface heat flux condition. They noted that the air bulk-temperatures along the channel length are a function of time for initial times only, while it resides for each axial locations until arrival steady state condition. Also, they deduced that the values of Nusselt number for initial times are greater than that of steady state. They found the relative difference between the present analytical model and the numerical results obtained using COMSOL software is maximum 9.1%. Hasobee and Salman [10] studied experimentally the heat transfer by natural convection through an externally heated inclined tube with uniform heat flux. They used four different positions of inclinations namely, 0^o (horizontal) , 30^o , 60^o and vertical for wide range of wall heat fluxes at fixed diameter and length of tube.

They deduced different experimental correlations of all inclinations for local and average Nusselt numbers as a function of Rayleigh number. Ahmed et al. [11] investigated experimentally free convection through smooth and rough triangular channel for different inclinations from 15^o to 90^o (vertical position) at constant wall heat flux. They found that the average Nusselt number in rough triangular channel is bigger than that of smooth triangular channel with 10.0% for vertical position and 8.1% for inclined position at 45^o. They concluded empirical correlations for smooth and rough triangular channels at vertical and different inclined positions. Satyanarayansa et al. [12] presented a numerical solution using CFD techniques to simulate the natural-convection into a vertical tube with and without internal helical fins. They used single fin with multi-coils and multi-fins with single coil.

They showed that the multi-fins with single turn is more efficient and higher in heat transfer rate than other type. Also Rani [13] used a CFD analysis and ANSYS 13.0 software to predict the free-convective heat transfer into a vertical internally finned pipe. He utilized three pipes with same dimensions, the first pipe without fins, the second pipe has one helical fin and the third pipe has ten helical fins with equally spaced. He noted that the third pipe is more effective and has more values of convection heat transfer coefficient than another two pipes.

The objective of this work is to present an experimental investigation for natural-convection heat transfer inside an externally heated horizontal open-ends tube with wide range of aspect ratios namely, 11.6 , 13.7 , 15.8 , 17.9 and 20 to find the empirical correlations for Nusselt number as a function of Rayleigh number. The effect of aspect ratio , wall heat flux , internal ring fins and fin spacing on the characteristics and process of natural heat convective are taken in consideration.

2. Experimental Test-Rig

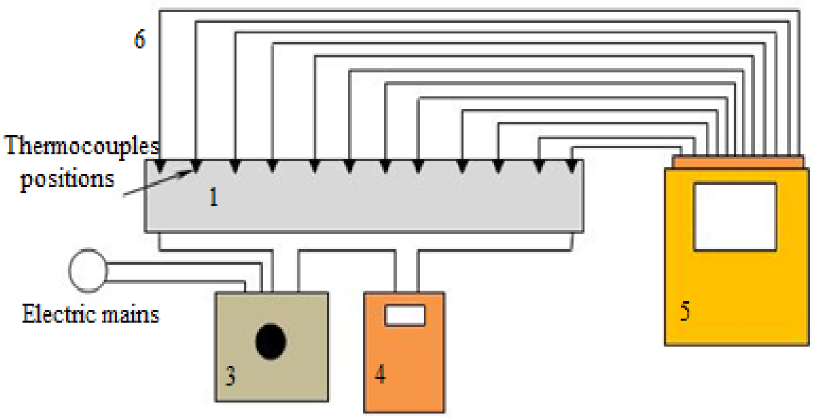
The overall experimental test-rig is shown graphically and photographically in "Fig. 1". It is especially manufactured to covering the tests of present work and located in a small room to ensure the good conditions of free-convection. It consists of a test-section placed horizontally on the table by two wooden stands shape-V to ensure stability of the tube and maintain it in the horizontal position. A control and measurement devices namely digital voltage regulator , a digital multi-meter and a multi-channels data logger thermometer.

The test-section consists of open-ended tube of constant inner diameter (D) of 38 mm and constant wall thickness (t) of 2 mm is made from polished aluminum. Five tubes with different lengths (L) are used namely 440 , 520 , 600 , 680 and 760 mm to change the aspect ratios (R). Another four tubes with constant length (L= 440 mm) including an internal circular-section ring fins with constant dimensions, outer diameter (d_o= 38 mm) , inner diameter (d_i= 32 mm) and thickness (t_r= 2 mm) and varying in fins number (n) and spacing (s) as shown graphically in "Fig. 2".

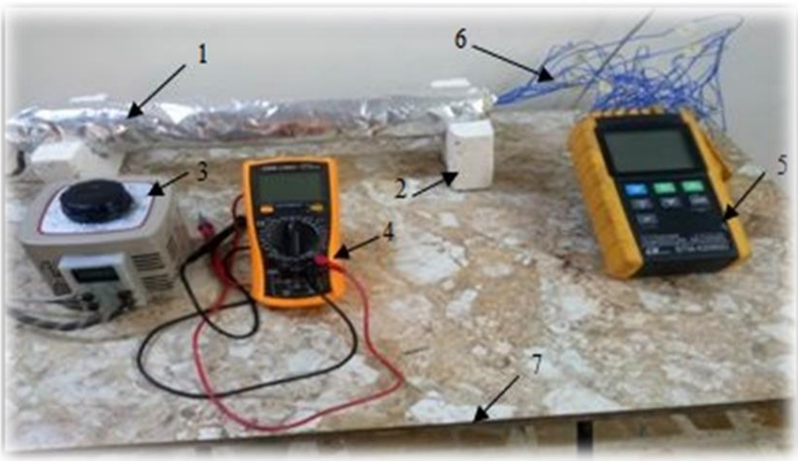
The inner surface temperatures of tube are measured using twelve K-type calibrated thermocouples are distributed axially along the tube with equal spaces. The thermocouples are joined by multi-channels of data logger thermometer model BTM-4208SD with range of temperatures from -100 to 1300 ^{o}C for type-K thermometer and accuracy ± (0.4% + 0.5^{o}C). Additional same K-type calibrated traversing thermocouple is used to record the axially air bulk temperatures inside the tube at the same locations of surface thermocouples. The outer surface of tube is insulated by thin layer of 10 mm of asbestos and it’s electrically heated by varying power input levels using nickel-chrome electrical wire heater of 0.3 mm diameter with power up to 800 W is coiled helically and uniformly around the asbestos layer and then the asbestos rope is wrapped above the heater coil to close the spaces between wire heater coiling pitches. Another asbestos layer of thickness 30 mm is used to cover the heating element and then the test-section is covered by a thick layer of glass wool up to 50 mm thickness with aluminum foil to reduce the radiation heat loss to surrounding. Furthermore, two Teflon rings of 25 mm thickness are used to minimize heat conduction loss from both ends of tube as illustrated schematically in "Fig. 3".

Another two same K-type calibrated thermocouples are inserted in an asbestos and glass wool layers to read the temperatures and estimate the insulations thermal resistance. Furthermore, one thermocouple is used to measure the ambient temperature.

The digital voltage regulator model SAKO-TDGC$_2$ (0.5 KVA , maximum current 2.0 A) is used to control the supplied voltage input to the heating element. Also a digital multi-meter model VICTOR-VC890C$^+$ with range of voltage from 2 V to 750 V and accuracy ± (0.8% + 5) and range of current from 20 mA to 20 A and accuracy ± (2.0% + 5) is used to measure voltage and alternating current passed through the heating element.



a. Schematic diagram for the experimental test-rig .



b. Photograph of experimental test-rig

1. Test-section 2. Wooden stand 3. Digital voltage regulator 4. Digital multi-meter 5. Multi-channels data logger thermometer 6. Thermocouple wires 7. Table

Figure 1. The experimental test-rig

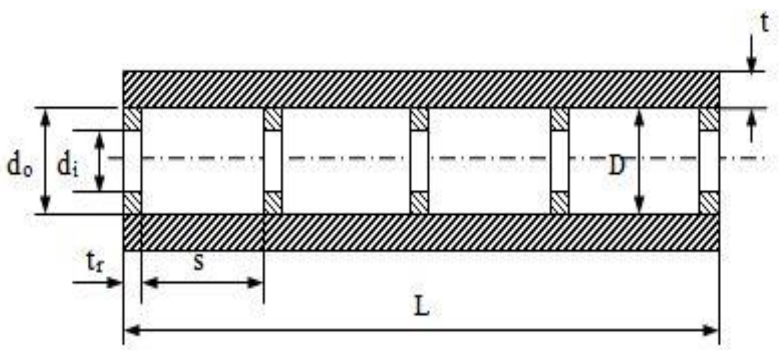


Figure 2. Schematic diagram of longitudinal section for the tube with internal ring fins

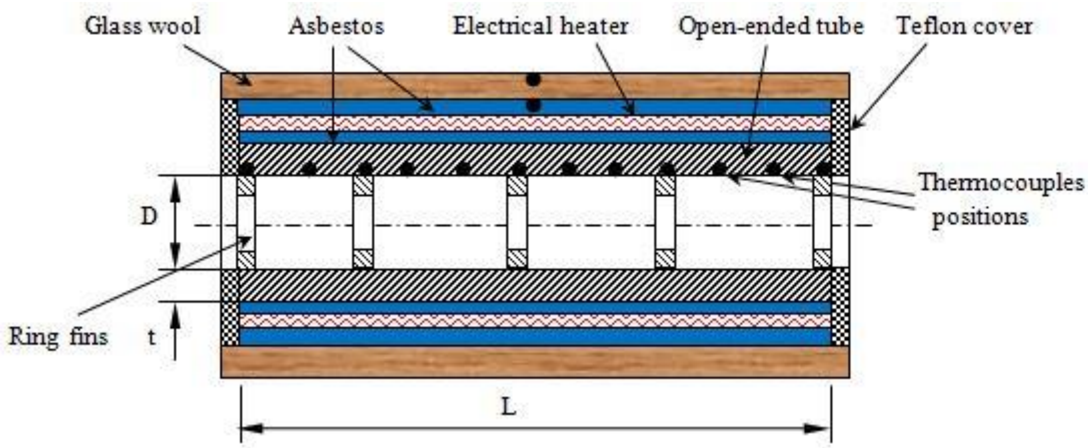


Figure 3. Schematic diagram of longitudinal section for the test-section with thermocouples positions

3. Test Procedures and Computations

Five tubes are used in the tests with constant inner diameter (D_i) of 38 mm and wall thickness of 2 mm while varying in lengths (L) to obtain a wide range of aspect ratios (R=L/D) as listed in "Table 1". Another four tubes with constant aspect ratio (R= 11.6) providing four different numbers of internal ring fins as listed in "Table 2". Different values of the heat fluxes (q) from 254 to 2267 W/m^2 are used.

Table 1. Lengths of tested tubes with aspect ratios for D= 38 mm

Length (L) , mm	440	520	600	680	760
Aspect ratio (R)	11.6	13.7	15.8	17.9	20

Table 2. Number of internal fins with fin spacing for tube of aspect ratio R= 11.6

Number of fins (n)	3	5	7	9
Fin spacing (s) , mm	217	107.5	71	52.75

The steady-state condition for all tests of present work is achieved from 120 to 150 minutes approximately depending on the electrical power input. The readings of thermocouples have been recorded when the difference between two temperature readings not more than (±0.5 ^oC) within 15 minutes. Then the temperatures, supplied voltage and current to the heating element are recorded under steady-state condition.

The power input to the electrical heater coiled around the tube is transformed to heat energy inside the open-ended tube by free-convection furthermore, heat losses from outer surface of tube through insulations layers and outer surface of insulation across aluminum foil to surrounding in addition, heat lost from tube ends through Teflon rings then [14],

$$Q_i = Q_{cv} + Q_l \quad (1)$$

The power input (Q_i) is product of input current intensity (I) and voltage supplied (V) to wire electrical heater with the power factor (cos ϕ) as [14]:

$$Q_i = I\,V\cos\phi \quad (2)$$

The heat losses (Q_l) can be computed as follows:

$$Q_l = Q_r + Q_{cd} \tag{3}$$

The heat radiation lost (Q_r) is evaluated as [14]:

$$Q_r = \varepsilon\, \sigma\, A_s\ \Delta T_r^4 \tag{4}$$

where, ΔT_r is the temperatures difference between aluminum foil surface covering the test section and ambient air temperature.

Also, the thermal conduction lost (Q_{cd}) is computed from temperatures difference between insulation layers (ΔT_{cd}) divided by insulation thermal resistance (R_{th}) as follows [14]:

$$Q_{cd} = \frac{\Delta T_{cd}}{R_{th}} \tag{5}$$

The free-convective heat transfer rate (Q_{cv}) into tube is calculated as cooling Newton's equation [14,15]:

$$Q_{cv} = h\, A_i (T_s - T_b) \tag{6}$$

and the free-convection heat flux (q) calculated as:

$$q = \frac{Q_{cv}}{A_i} = \frac{I\,V - Q_l}{A_i} \tag{7}$$

Then, the local convection heat transfer coefficient (h_x) can be computed as :

$$h_x = \frac{q}{(T_{sx} - T_{bx})} \tag{8}$$

and the local Nusselt number (Nu_x) based on the inner tube diameter (D) as characteristics length can be computed as follows [14,15]:

$$Nu_x = \frac{h_x\, D}{k} \tag{9}$$

where, T_{sx} and T_{bx} are the local surface and air bulk temperatures inside the tube respectively.

Define the local Rayleigh number as [15,16]:

$$Ra_x = Gr_x\, Pr \tag{10}$$

where, the local Grashof number (Gr_x) based on the tube diameter is calculated as [15,16]:

$$Gr_x = \frac{g\, \beta (T_{sx} - T_{bx})\, D^3}{\nu^2} \tag{11}$$

Equation (10) can be rewrite as follows:

$$Ra_x = \frac{g\,\beta(T_{sx} - T_{bx})\,D^3\,Pr}{\nu^2} \quad (12)$$

The physical properties of air are calculated corresponding to local film temperature (T_{fx}) an arithmetic mean of local surface temperature (T_{sx}) and air bulk temperature (T_{bx}) as follows [15,16]:

$$T_{fx} = \frac{T_{sx} + T_{bx}}{2} \quad (13)$$

where, T_{sx} and T_{bx} in K.
The average Nusselt number (Nu_{av}) based on the inner tube diameter (D) as characteristics length can be computed as follows [14,15]:

$$Nu_{av} = \frac{h_{av}\,D}{k} \quad (14)$$

and average Rayleigh number becomes:

$$Ra_{av} = \frac{g\,\beta(T_{sav} - T_{bav})\,D^3\,Pr}{\nu^2} \quad (15)$$

where, h_{av} is the average convection heat transfer coefficient , and can be calculated as follows:

$$h_{av} = \frac{q}{(T_{sav} - T_{bav})} \quad (16)$$

T_{sav} and T_{bav} are the average surface and air bulk temperatures respectively. They are evaluated as follows:

$$T_{sav} = \frac{1}{L}\int_0^L T_{sx}\,dx \quad (17)$$

and,

$$T_{bav} = \frac{1}{L}\int_0^L T_{bx}\,dx \quad (18)$$

The air properties are calculated based on the average film temperature (T_{fav}) as follows [15,16]:

$$T_{fav} = \frac{T_{sav} + T_{bav}}{2} \quad (19)$$

Define the aspect ratio (R) as a proportion between the tube length (L) and internal diameter of tube (D):

$$R = \frac{L}{D} \tag{20}$$

The percentage error (e%) is the percentage relative between absolute error (e_{ab}) and expected value (v_{ex}) as following:

$$e\% = \frac{e_{ab}}{v_{ex}} \times 100\% = \left|\frac{v_{ex} - v_m}{v_{ex}}\right| \times 100\% \tag{21}$$

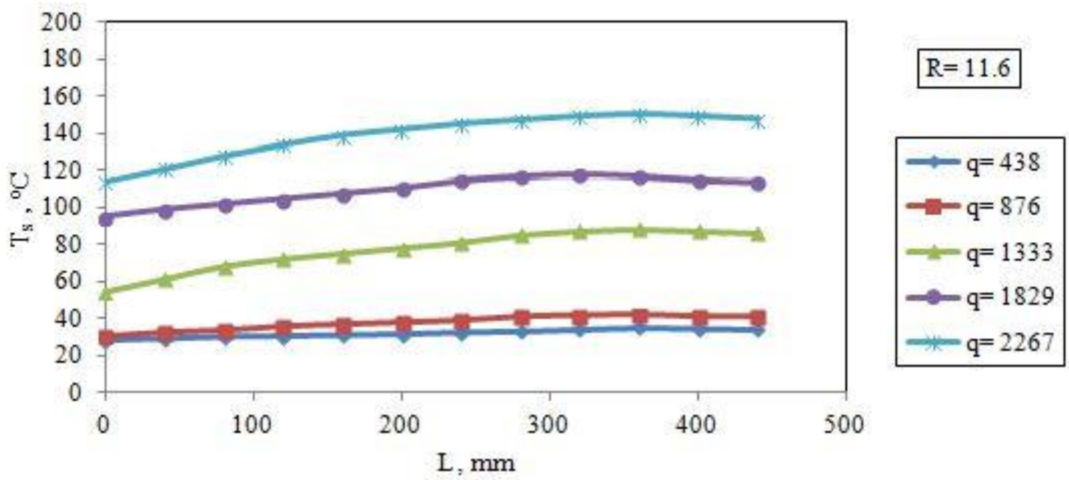
where, the absolute error (e_{ab}) is the absolute difference between the expected value (v_{ex}) and measured value (v_m) as:

$$e_{ab} = |v_{ex} - v_m| \tag{22}$$

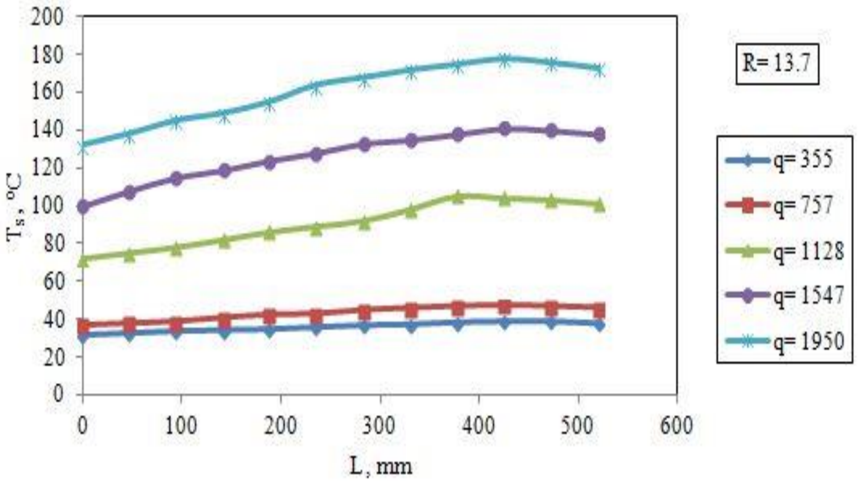
4. Results and Discussion

The present work investigates experimentally the influence of aspect ratio (R), internal ring fin, fin spacing on the performance natural-convective heat transfer into a heated tube in the horizontal position for different wall heat fluxes.

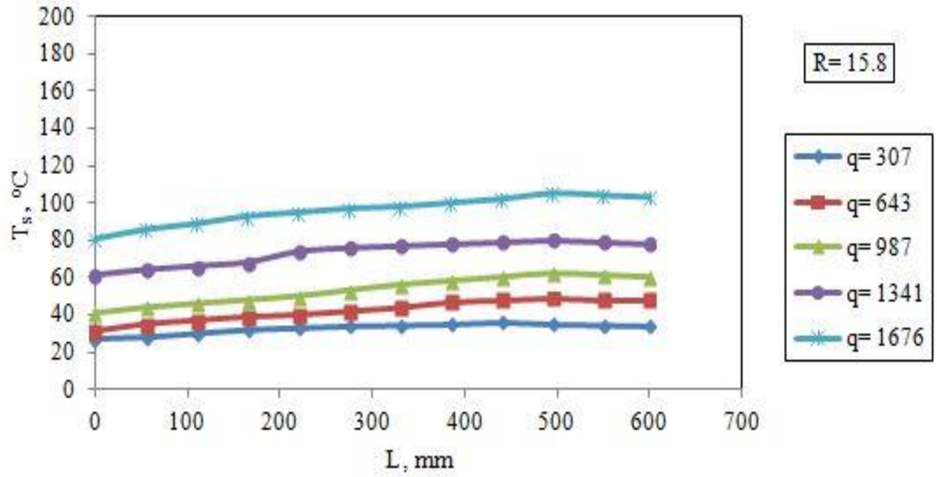
"Figs. 4(a)" to "4 (e)" show the distribution of axial inner surface temperature along the horizontal tube for different aspect ratios and heat fluxes. It is gradually increase with increasing the axial distance along the tube and then lightly decreases forward the end tube because the conduction heat lost across tube end. At the tube inlet, the boundary layer thickness is begin from zero and increases to reach maximum and fills the whole tube channel caused increasing in surface temperatures and then slightly decreases near tube end because increasing in heat transfer rate at tube end. Also, it is clear that the surface temperature increases as the heat flux increases for all different aspect ratios because the faster increasing of thermal boundary layer resulted from increasing of wall heat flux.



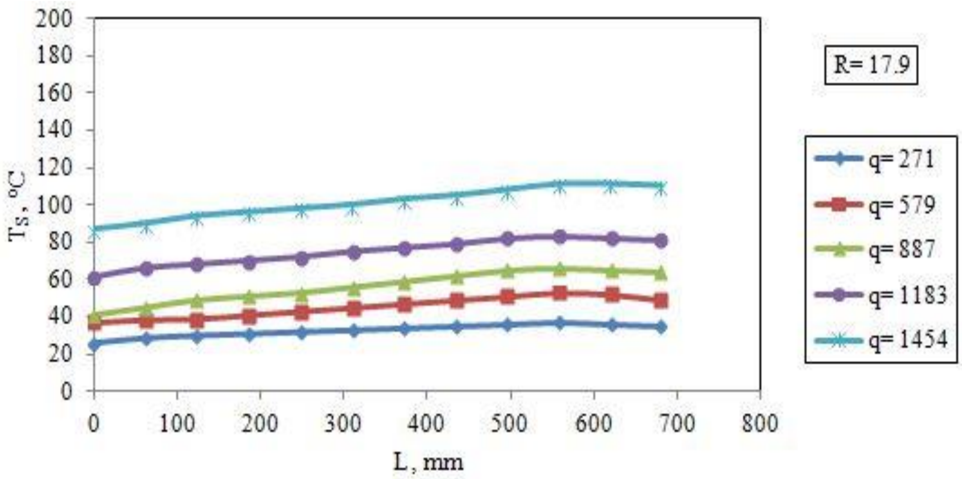
(a) Aspect ratio R= 11.6



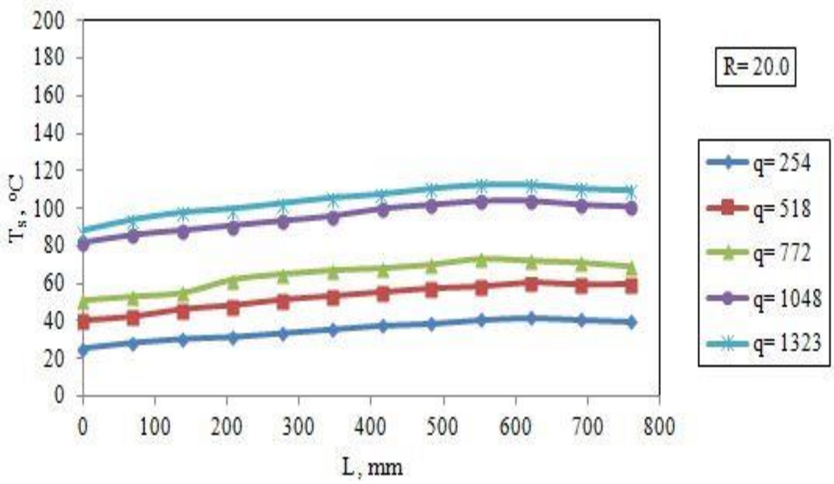
(b) Aspect ratio R= 13.7



(c) Aspect ratio R= 15.8



(d) Aspect ratio R= 17.9



(e) Aspect ratio R= 20.0

Figure 4. Variation of the surface temperatures via axial distance along the tube for different heat fluxes and aspect ratios.

"Fig. 5" illustrates the variation of surface temperature with axial distance along the length of tube for different aspect ratios and fixed heat flux (q= 985 W/m^2). It is seen that the surface temperature increases as aspect ratio increases for same wall heat flux. That due to from faster increasing in thermal boundary layer for surface of channel tube.

"Fig. 6 (a)" and "6 (b)" illustrate the distribution of surface temperatures with axial distance along the tube channel for five different cases namely, smooth tube channel (n= 0), tube channel with 3 , 5 , 7 and 9 internal ring fins for two different wall heat fluxes at fixed aspect ratio. It is clear that the surface temperature decreases with increasing number of internal ring fins up to a certain value of fins number (n= 7), while the surface temperature increases at fins number (n= 9) because the number of fins increases oversteps a specific limit, and thus temperature gradient in the boundary layer decreases and consequently the heat transfer rate from channel surface to air flows inside the tube decreases.

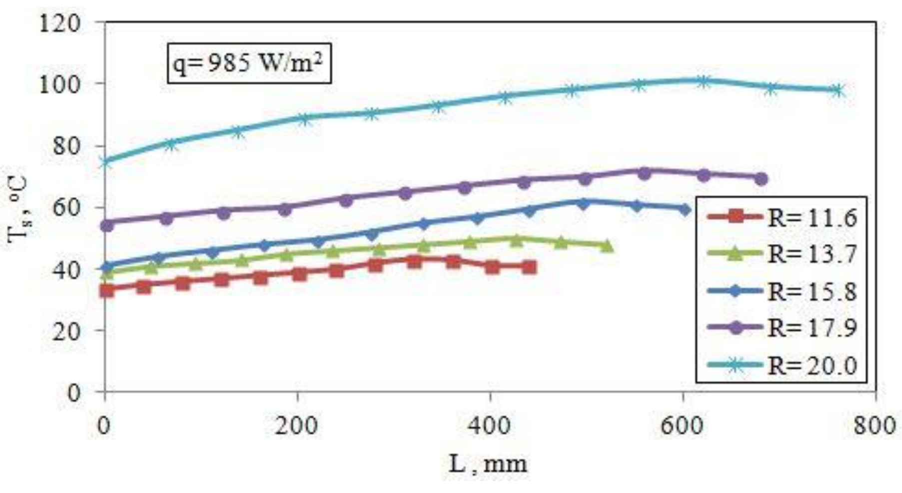
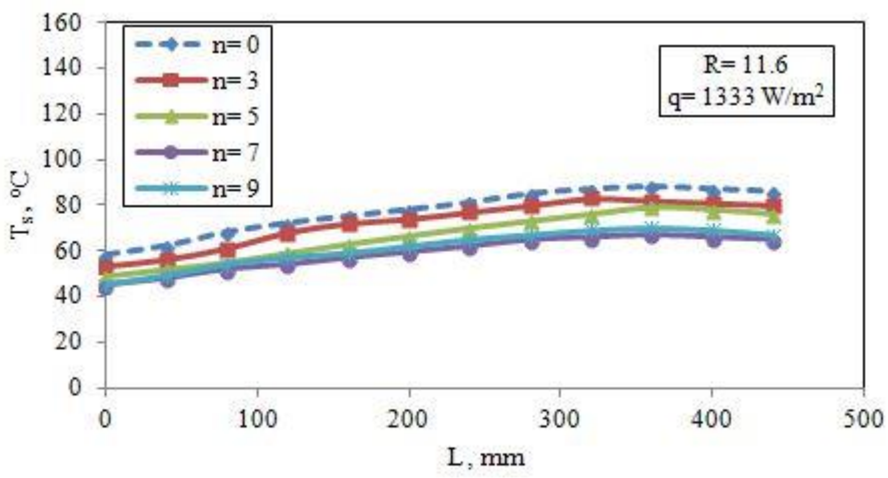
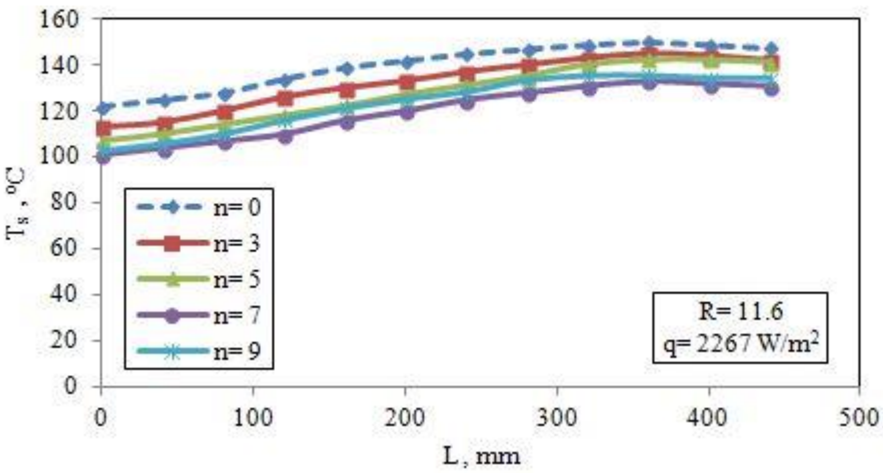


Figure 5. Variation of the surface temperatures via axial distance along the tube for different aspect ratios and fixed heat flux



(a) heat flux, q= 1333 W/m^2



(b) heat flux, q= 2267 W/m^2

Figure 6. Variation of the surface temperatures via axial distance along the tube for different number of ring fins and fixed aspect ratio

In "Fig. 7 ", variation of the local Nusselt numbers versus axial distance along the tube for two different channels, without ring fins and with fins number (n=7) and two values of heat fluxes q= 438 , 2267 W/m^2 at constant aspect ratio (R= 11.6).

It is observe that the local Nusselt number increases with increasing axial distance and it increases as increasing wall heat flux because the influence of secondary flow which increases with increasing heat flux and then enhances the natural-convection heat transfer process.

This increasing in heat flux caused the air near of the inside surface of tube becomes hotter and slighter in the tube core leading to two forward currents flow along the inner surface while air near the tube center flows backward, and therefore occur increasing in convection heat transfer coefficient and local Nusselt number.

Also, it is seen that the tube of seven internal ring fins (n=7) gave better natural-convection heat transfer performance in term of local Nusselt number than same tube without internal fins (n=0), so the optimum fin spacing is 71 mm.

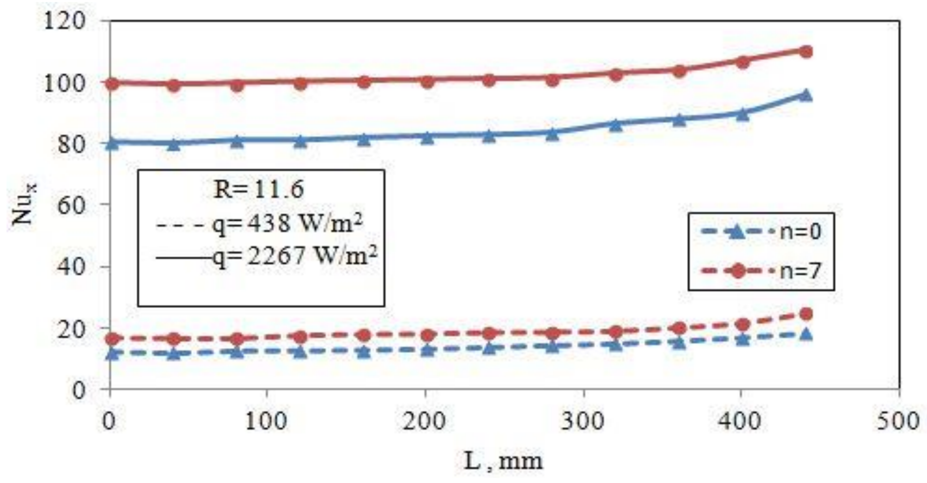


Figure 7. Variation of the local Nusselt numbers via axial distance along the tube without ring fins and with fins number (n=7) at fixed aspect ratio

"Fig. 8" shows the variation of local Nusselt number with axial distance along the length of tube for five different aspect ratios and fixed heat flux (q= 985 W/m^2). It is seen that the local Nusselt number decreases as aspect ratio increases at same wall heat flux. This due to the heat transfer rate increases with the decrease in length of tube.

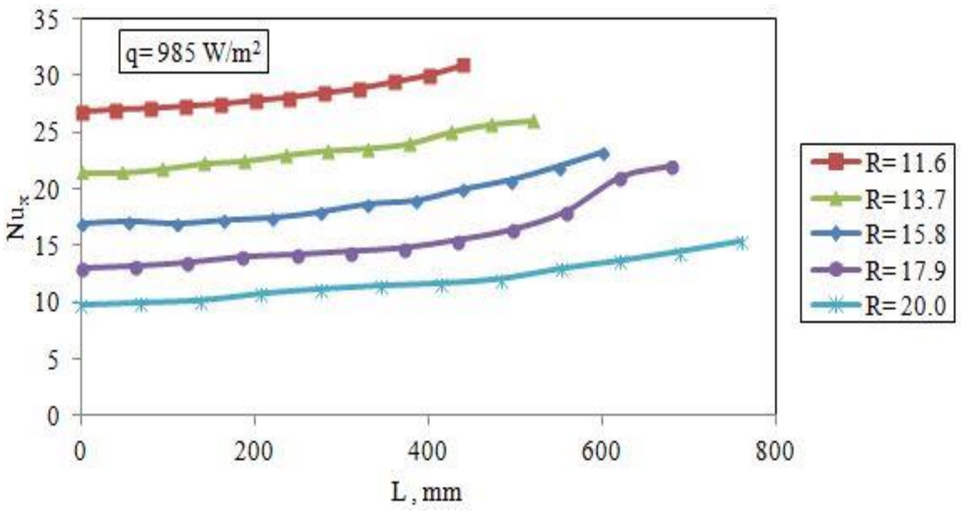


Figure 8. Variation of the local Nusselt numbers via axial distance along the tube for different aspect ratio at fixed heat flux

"Fig. 9" illustrates a correlation between local Nusselt number and local Rayleigh number for an aspect ratios of $11.6 \leq R \leq 20.0$ at constant heat flux (q= 985 W/m^2). It is clear that the local Nusselt number increases with increase in local Rayleigh number. The following experimental correlation was resulted:

$$\mathrm{Nu_x} = 0.0408\,(\mathrm{Ra_x})^{0.4699} \tag{23}$$

The squared correlation coefficients (R^2) is 85.5 %. The percentage error (e%) between the resulted correlation, "Eq. 23" and experimental data is (e% $\leq$ 17%).

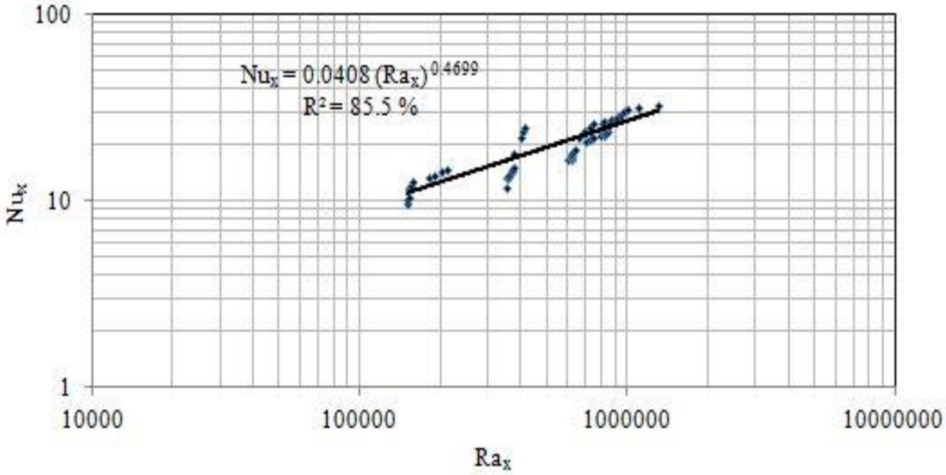


Figure 9. An experimental correlation between local Nusselt number against local Rayleigh number for all different aspect ratios at constant heat flux.

In "Fig. 10", an empirical correlation between average Nusselt number and Rayleigh number for an aspect ratios from 11.6 to 20.0 and constant heat flux. It is seen that the average Nusselt number increases with increase in average Rayleigh number. The following experimental correlation was deduced:

$$Nu_{av} = 0.0968\,(Ra_{av})^{0.4056} \qquad (24)$$

The squared correlation coefficients, R^2= 89.0 %. The percentage error (e%) between the deduced correlation, "Eq. 24" and experimental present data is less than 10%. To verify the present results of tube without internal ring fins, a comparison is done with Hasobee correlation:

$$Nu_{av} = 0.02115\,(Ra_{av})^{0.43148} \qquad (25)$$

It is valid for steady state natural-convective through a heated externally open ends circular tube without internal fins in horizontal position under uniform surface heat flux condition as shown in "Fig. 10".

It is seen that approx similarity in behavior, but the present results gave values of average Nusselt number higher than that done in Hasobee correlation because the higher values of heat fluxes using in present work.

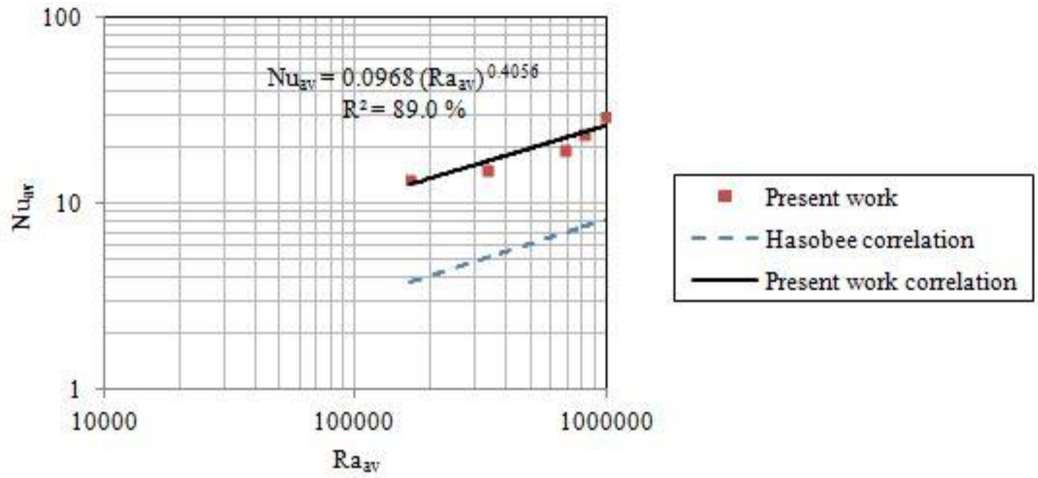


Figure 10. An empirical correlation between average Nusselt number versus average Rayleigh number for different aspect ratios with comparison than Hasobee correlation

5. Conclusions

The heat transfer by free-convective in steady-state condition through a heated externally horizontal tube with open ends is investigated experimentally. Many parameters like aspect ratio, wall heat flux and internal ring fins number are taken in consideration and studied. The following conclusions can be drawn:

- As the wall heat flux increases, the local Nusselt number increases.
- The local Nusselt number increases with decreasing the aspect ratio.
- Heat transfer rate increases as the internal fins number increases up to n= 7 , hence the optimum value of fin spacing is about 71 mm.
- The performance of free-convective heat transfer into a tube channel as local Nusselt number improves with internal ring fins about 25% higher than without fins (smooth tube channel).
- An experimental correlations are suggested to predict the local and average Nusselt numbers for aspect ratios $11.6 \leq R \leq 20.0$.

Nomenclature

A_i	inner surface area of tube, (m^2)
A_s	surface area of radiation heat transfer , (m^2)
D	internal diameter of tube (channel diameter), (m)
g	gravitational acceleration, (m/s^2)
Gr	Grashof number, (-)
h	convection heat-transfer coefficient, ($W/m^2.K$)
I	input current intensity, (A)
k	thermal conductivity of material, (W/m.K)
L	length of the tube , (m)
n	number of internal ring fins, (-)
Nu	Nusselt number, (-)
Pr	Prandtl number, (-)
q	wall heat flux, (W/m^2)
Q_{cd}	conduction heat transfer rate, (W)
Q_{cv}	convection heat transfer rate, (W)

Q_i	electrical power input, (W)
Q_l	heat losses, (W)
Q_r	radiation heat transfer rate, (W)
R	aspect ratio, (-)
Ra	Rayleigh number, (-)
R_{th}	thermal resistance, ($^oC/W$)
s	Spacing between internalfins, (m)
T_b	air bulk temperature, (^oC)
T_f	film temperature, (K)
T_s	surface temperature, (^oC)
V	voltage supplied, (V)
ΔT_{cd}	temperature difference to estimate thermal conduction lost, (^oC)
ΔT_r	temperature difference to estimate heat radiation lost, (^oC)
β	volumetric coefficient of thermal expansion, (1/K)
cos ϕ	power factor, (-)
ε	emissivity of the surface, (-)
σ	Stefan-Boltzmann constant , ($\sigma = 5.67 \times 10^{-8}$ W/ $m^2.K^4$)
$\boldsymbol{\nu}$	kinematic viscosity of the air, (m^2/s)

Subscript Symbols

av	Average
x	Local

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