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Natural-Convection Phenomenon from a Finned Heated Vertical Tube: Experimental Analysis

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Abstract

In this work, an experimental analysis is made to predict the thermal performance of the natural-convection phenomenon from a heated vertical externally finned-tube to surrounding air through an open-ended enclosure. Two different configurations of longitudinal rectangular fin namely, continuous and interrupted are utilized with constant thickness, different numbers, and different heights are extended radially on the outer surface of a heated tube. The tube is heated electrically from inner surface with five varied power input magnitudes. The effect of fins configuration, fins number, fins height, and heat flux of the inner tube surface on the thermal performance of natural convection have been studied and analyzed experimentally. Obtained results show that the tube with twelve interrupted longitudinal fins gives the best natural-convection thermal performance in terms of average Nusselt number, about 20% greater than that for the tube with continuous fins. Experimental correlations to predict the average Nusselt number for the heated tubes with continuous and interrupted longitudinal fins are proposed. The present data are compared to previous study and good convergence is noticed.

Keywords: *Vertical tube, continuous, interrupted, finned, natural-convection.*

1. Introduction

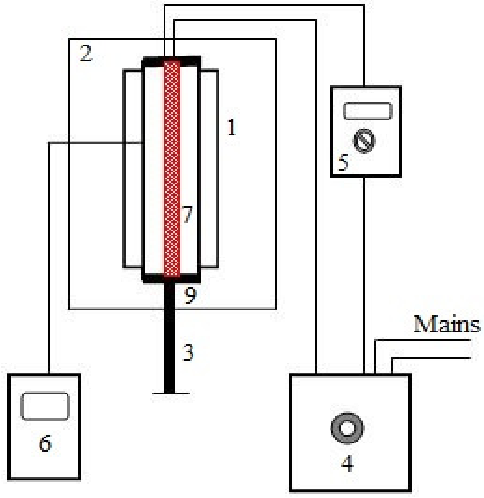
The natural-convection heat transfer from a vertical, horizontal, and inclined heated finned tube in different fin positions to surrounding air occurs in a many industrial and engineering applications. The importance applications are heat exchangers, evaporators, condensers, oil cooling systems, solar collectors, systems of energy storage,…and etc. The main purpose of these systems or equipments is to increase of heat dissipation rate from tube surface by utilizing different configuration of fins with varying heights and numbers, and high thermal conductivity materials.

Morgan [1] published a review of articles for the natural heat convection from outside surface of a heated smooth circular cylinder to surrounding air. He correlated the available experimental data to predict the Nusselt number values based on the diameter of cylinder for a wide range of Rayleigh numbers. Stewart and Verhulst [2] experimentally investigated the two dimensional natural-convection from a two heated copper horizontal tubes to cooled rectangular-section enclosure at constant temperature condition to simulate the natural-convection heat transfer in systems of the underground thermal distribution. They correlated the experimental data to determine the Nusselt number for Rayleigh numbers ranging from 2.1×10^{8} to 4.8×10^{9}, and found that the natural-convection heat coefficients computed for underground or utility corridors case with air as working fluid smaller compared with other experimental correlations for concentric tubes. Wu and Tao [3] numerically simulated the heat transfer by laminar natural-convection from a heated horizontal compound cylinder with longitudinal external fins. They utilized the conjugated computational approach with primitive variables, and found the optimum fins number at a fixed fin height and the optimum fin height for a constant fins number. Kadhim and Abdulhussain [4] performed an experimental investigation for natural-convection from external surface of finned cylinder to surrounding air. They used two configuration of an external fins namely, triangular and rectangular with different inclination angles of cylinder and heat fluxes, and noted that the heat transfer from the triangular finned-cylinder is maximum at an inclination 90^o (vertical position). Dogan et al. [5] presented a numerical investigation for natural-convection heat transfer from annular fins on a heated horizontal cylinder. They studied the effect of fin spacing, fin diameter, and temperatures difference between the base and the surrounding on the performance of heat transfer. They suggested a correlation to obtain the optimum fin spacing as function of fin diameter and Rayleigh number. Totala et al. [6] presented an experimental investigation for natural heat convection from a heated vertical cylinder to surrounding air. They evaluated experimentally the local natural convection heat transfer coefficient along the height of cylinder and compared the results of them with obtained results from a theoretical part and they found a good convergence between of them. Also, they found the maximum values of local heat transfer coefficient at the beginning and gradually reduces in the upward direction. Niezgoda-Zelasko and Zelasko [7] experimentally investigated the natural and forced-convection from longitudinally finned cylinder in vertical position. They used two air flow velocities, parallel and cross flow to the cylinder axis and computed experimentally convection heat transfer coefficients values for all cases. They found that the convection heat transfer coefficients for case of perpendicular flow 22% greater than case of parallel flow for cylinder axis and developed experimental correlations to compute heat transfer coefficient for natural and two cases of forced-convection. Lee et al. [8] experimentally investigated the natural heat convection from vertical cylinder with two different types of plate-fins namely, radial and inclined. They studied many parameters like number of fins, inclined angle of fins, and base temperatures on the convection heat transfer rate. They observed that the optimal heat resistance of cylinder with inclined plate-fins is 30% less with that of cylinder with radial plate-fins.

The objective of the present work is to present an experimental analysis for the natural heat convection from vertical externally finned-tube to surrounding air inside an open-ended enclosure utilizing two different configurations of external longitudinal fin, continuous and interrupted, with different numbers, heights, and regular distributed radially on the external surface of the heated tube. The influence of fins shape, fins number, fin height, and surface heat flux on the thermal performance of the natural-convection phenomenon is taken into consideration.

2. Experimental-Setup

The experimental-setup is designed and manufactured to cover the experiments of the present work as shown in Figure (1). It consists of externally-finned tube vertically assembly inside an open ended square-section enclosure, voltage regulator, digital multi meter, and twelve channels data logger thermometer. The finned-tube assembly is fixed vertically into an acrylic sheet enclosure opened from top and bottom utilizing a plastic stand. It consists of tube with external diameter (D) of 28 mm, inner diameter (d) of 20 mm and length (L) of 200 mm and made from polished aluminum. Two configurations of longitudinal rectangular fin, continuous and interrupted are made from the same material of tube with thickness (t) of 2 mm and fixed radially on the tube outer surface with different numbers and heights. The interrupted fin consists of three equal segments with length (L_{int}) of 60 mm and the blank (b) between two segments is 10 mm as illustrated in Figure (2). The inside surface of tube is heated electrically using heating coil of 500 W capacity and putted in a Pyrex glass pipe with outer diameter of 20 mm to fit into the externally-finned tube and joined to a voltage regulator model SAKO-TDGC$_2$ size of 0.5 KVA with rating of current 2 A and voltage from 0 to 250 V to control the supplied voltage for the electrical heater. A digital multi meter type VICTOR-VC890C$^+$ having voltage range from 2 V to 750 V and current range from 20 mA to 20 A is utilized to gauge the supplied voltage and alternating current intensity passing to the heating coil. Two Teflon covers of 25 mm thickness are utilized to insulate and minimize heat lost from tube ends. The outer surface temperatures of tube are recorded using ten type-K calibrated thermocouples are distributed axially along the height of tube with equal distances as shown in Figure (3) also two the same type-K calibrated thermocouples are utilized to read the temperatures difference through Teflon covers in the two ends of the tube to estimate the heat conduction lost, and connected to a twelve channels data logger thermometer model BTM-4208SD of resolution 0.1 ºC and rating from -100 ºC to 1000 ºC for type-K thermocouple prob. Another digital type-K calibrated thermocouple is used to measure the surrounding air temperature inside an enclosure.

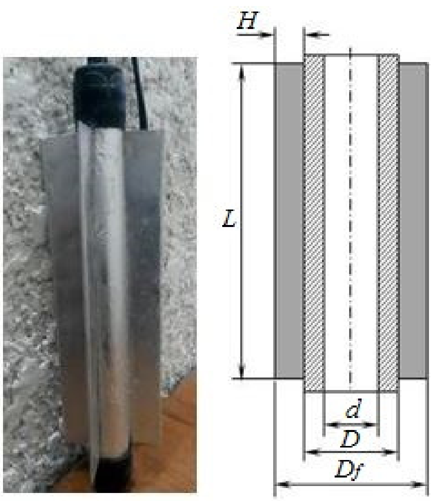


a. Sketch of an experimental-setup

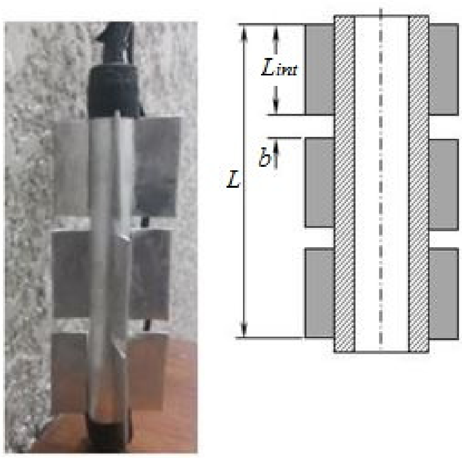
b. Photos of an experimental-setup

1. Vertical externally-finned tube 2. Open ended enclosure 3. Stand 4. Voltage regulator 5. Digital multi meter 6. Data logger thermometer 7. Heater 8. Thermocouple wires 9. Teflon covers 10. Longitudinal rectangular fins.

Fig. 1. Experimental-setup.



a. Tube with continuous fins



b. Tube with interrupted fins

Fig. 2. Front views of longitudinal rectangular fins configuration.

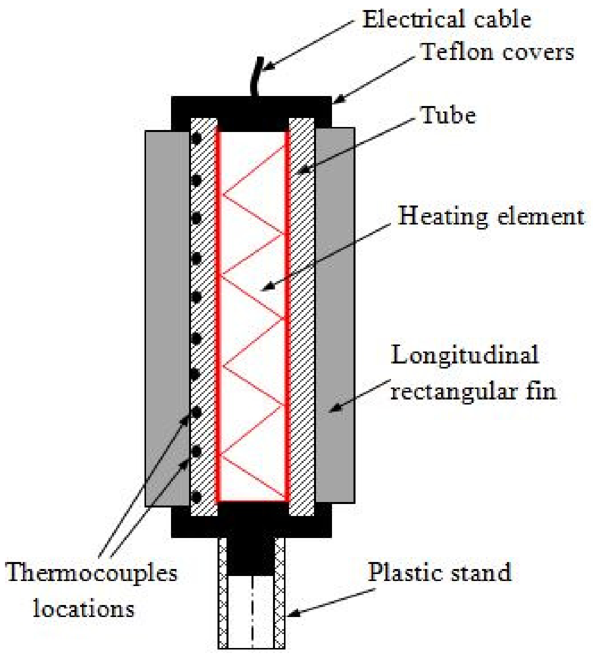


Fig. 3. Schematic diagram of front-section for vertical externally-finned tube assembly with thermocouples locations.

3. Tests Procedure and Computation

The present work is carried out for thirty different finned tubes. Two types of external longitudinal fin, continuous and interrupted are utilized with fixed length (L) of 200 mm and thickness (t) of 2 mm for all fins. The interrupted longitudinal fin consists of three segments of fins with constant length (L_{int}) of 60 mm and constant blank (b) of 10 mm, (L_{int}/b= 6). Five different numbers of each continuous and interrupted fins (N) namely, four, six, eight, twelve, and sixteen are fitted radially on the outer surface of heated tube with three different heights (H) 10, 20, and 30 mm. Five varying values of surface heat fluxes (q) namely, 796, 1990, 3981, 5971, and 7962 W/m^2 are utilized.

The steady-state conditions for all experiments of the present work are attained from 90 to 120 minutes depending on the power input. The thermocouples readings have been registered when the difference of two readings of temperatures is $\pm \leq$ 0.5 ^{o}C within 15 minutes approximately. Then, the tube surface temperatures, surrounding air temperature inside an enclosure, input voltage, and current intensity passed to electrical heater are registered.

The power input (Q_{in}) to the electrical heater into tube is computed as product of voltage (V) and input current (I) with power factor ($cos\,\phi$):

$$Q_{in} = V\,I\,cos\phi \quad ...(1)$$

It's transformed to heat energy and transferred from finned-tube to air surrounding by natural-convection (Q_{conv}) and thermal radiation (Q_{rad}), moreover the thermal conduction lost (Q_{cond}) from ends of vertical finned-tube. Then,

$$Q_{in} = Q_{conv} + Q_{rad} + Q_{cond} \quad ...(2)$$

The loss of thermal radiation between outer surfaces of finned-tube and surrounding air inside an enclosure is calculated as follows [9]:

$$Q_{rad} = \sigma\,\varepsilon\,F A_s\,(T_{sav}^4 - T_a^4) \quad ...(3)$$

where,

ε= 0.05 is the emissivity of polished aluminum,

F is the shape factor, T_a is the surrounding air temperature, and T_{sav} is the average outer surface temperatures of vertical tube. For interval from 0 to tube height (L) is computed as follows [9]:

$$T_{sav} = \frac{1}{L}\int_0^L T_{sx}\;dx \quad ...(4)$$

where, T_{sx} is the local surface temperature.

The heat conduction loss (Q_{cond}) is computed as thermal potential difference to thermal resistance. It's found less than 3% of power input (Q_{in}) because the two thick covers of Teflon are utilized in the finned-tube ends therefore can be neglected. Then the heat transfer by natural-convection (Q_{conv}) can be computed as follows:

$$Q_{conv} = V\,I\,cos\phi - Q_{rad} \quad ...(5)$$

It's also can be calculated from Newton's equation of cooling as follows [10, 11]:

$$Q_{conv} = h_{av}\,A_s(T_{sav} - T_a) \quad ...(6)$$

The average natural-convection heat transfer coefficient (h_{av}) can be calculated as:

$$h_{av} = \frac{V\,I\,cos\phi - Q_{rad}}{A_s(T_{sav} - T_a)} \quad ...(7)$$

A_s is the surface area of finned-tube and computed as follows:

- tube with continuous longitudinal rectangular fins,

$$A_s = (\pi DL - NtL) + 2NHL \quad ...(8)$$

- tube with interrupted longitudinal rectangular fins,

$$A_s = (\pi DL - 3NtL_{int}) + 6NHL_{int} \quad ...(9)$$

where,

(D) is the outside diameter of tube, (L) is the height of tube or fin length, (H) is the fin height, (N) is the fins number, (t) is the fin thickness, and (L_{int}) is segment length of interrupted fin.

The heat flux (q) practiced on the finned-tube is computed from power input to heating element dividing by inner surface area of tube (A):

$$q = \frac{Q_{in}}{A} = \frac{VI\,cos\phi}{\pi dL} \quad ...(10)$$

The average Nusselt number (Nu_{av}) based on the external diameter of finned-tube (D_f) as characteristics length can be simply computed as:

$$Nu_{av} = \frac{h_{av}\, D_f}{k} \quad \text{... (11)}$$

Also average Rayleigh number (Ra_{av}) is computed from:

$$Ra_{av} = Gr_{av}\, Pr \quad \text{... (12)}$$

where, (Pr) is Prandtl number and (Gr_{av}) is the average Grashof number (Gr_{av}):

$$Gr_{av} = \frac{g\beta\, D_f^3 (T_{sav} - T_a)}{\nu^2} \quad \text{... (13)}$$

and

$$\beta = 1/T_f \quad \text{... (14)}$$

Air properties utilized in the experimental calculations are taken at film temperature (T_f). It is computed as the arithmetic mean between the surrounding air temperature (T_a) and average surface temperature (T_{sav}) of the heated finned-tube in (K).

The percentage error ($e\%$) is evaluted as the percentage relative of absolute error (e_{ab}) to experimental value (V_{exp}):

$$e\% = \frac{e_{ab}}{V_{exp}} \times 100 = \left| \frac{V_{exp} - V_{cor}}{V_{exp}} \right| \times 100 \quad \text{... (15)}$$

where, the absolute error (e_{ab}) is the absolute difference between experimental value (V_{exp}) and correlated value (V_{cor}) as follows:

$$e_{ab} = |V_{exp} - V_{cor}| \quad \text{... (16)}$$

The mean relative quadratic error ($\bar{e}$) is computed as:

$$\bar{e} = \left[\sum_{i=1}^{n} \left(\frac{V_{exp} - V_{cor}}{V_{exp}} \right)^2 \Big/ (n-1) \right]^{1/2} \quad \text{... (17)}$$

where, n is the data points number.

4. Results and Discussions

The present work investigates experimentally effect of the fin configurations (continuous and interrupted), fin height, fin numbers, and the surface heat flux on the thermal performance for a heated finned-tube in vertical position into an enclosure with open ends.

Figures (4) to (7) illustrate the effect of number of fins on the temperatures difference ($\Delta T = T_{sav} - T_a$) and average heat-transfer coefficient at different surface heat fluxes ranging from 796 to 7962 W/m^2 and fin height (H= 30 mm) for continuous and interrupted fins. They are clear that the temperatures difference (ΔT) and average heat-transfer coefficient (h_{av}) sharply increases as surface heat flux (q) increases. Also, the average heat-transfer coefficient (h_{av}) increases with increasing fins number (N) up to N= 12 and then decreases because the average surface temperature (T_{sav}) decreases with fins number increasing up to N= 12 and then increasing when the thermal dissipation by natural convection for tubes with 16-fins becomes poor. Additionally, Figures (6) and (7) appear the values of average natural-convection heat coefficient for interrupted fins are larger than those continuous fins because the interruptions in fin help to dissipate the growth of thermal boundary layer and interrupting it.

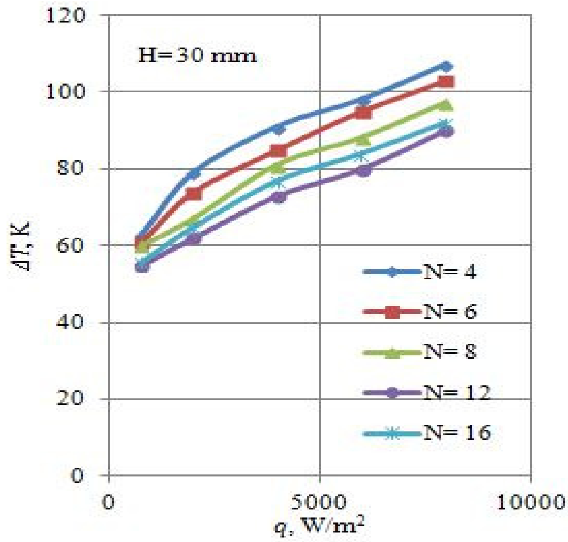


Fig. 4. Effect of fins number on the temperatures difference and surface heat flux for the continuous longitudinal fins.

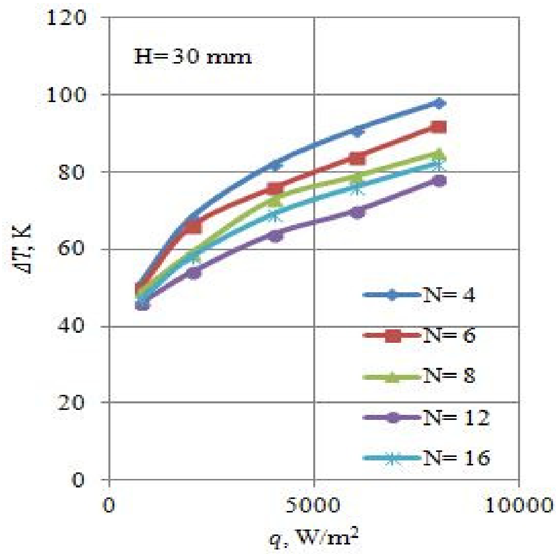


Fig. 5. Effect of fins number on the temperatures difference and surface heat flux for the interrupted longitudinal fins.

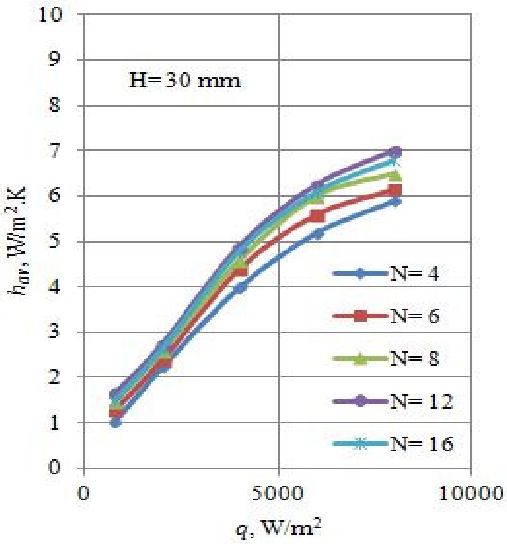


Fig. 6. Effect of fins number on the average heat transfer coefficient and surface heat flux for the continuous longitudinal fins.

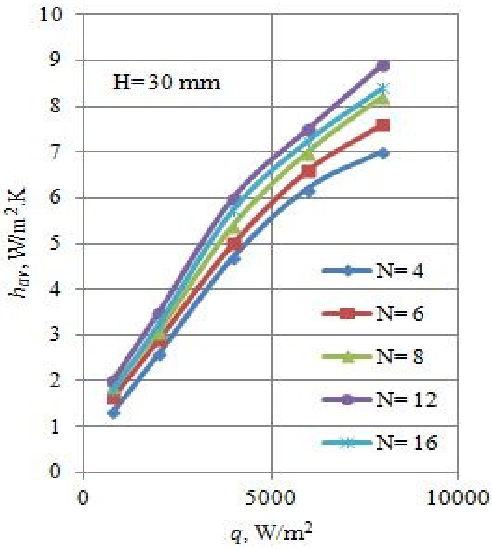


Fig. 7. Effect of fins number on the average heat transfer coefficient and surface heat flux for the interrupted longitudinal fins.

Figure (8) shows the behavior of average Nusselt number (Nu_{av}) against fins number (N) for continuous and interrupted longitudinal fins at three varying fin heights (H= 10, 20, and 30 mm) and different surface heat fluxes. It is clear that the average Nusselt number as a function of fins number at first increases up to a maximum value with the increase of fins number and then decreases. The optimum fins number (N_{opt}) at maximizing average Nusselt number is (N_{opt}= 12) for all cases because the average surface temperature (T_{sav}) decreases down to a minimum value with fins number increasing up to (N= 12) and then increases. This leads to bad natural-convection thermal performance at fins number (N= 16). Also it is clear that the values of average Nusselt number for interrupted longitudinal fins are greater than those of continuous fins. The interrupted fins cause higher values of natural-convection heat transfer coefficient because the interruptions help to dissipate the growth of thermal boundary layer along the height of tube and interrupting it, and this leads to a higher natural-convection heat transfer rate and improvement of natural-convection thermal performance. Finally, it is clear that the average Nusselt number enhanced when surface heat flux increases because the average convection heat transfer coefficient increases as surface heat flux increasing for all studied cases.

Figure (9) appears two correlations between average Nusselt number and Rayleigh number for two configurations of fin, continuous and interrupted at constant fins number (N= 12) and fin height (H= 30 mm) with a wide range of heat fluxes from 796 to 7962 W/m^2. It is clear that the average Nusselt number increases with increasing average Rayleigh number. The following two experimental correlations for continuous and interrupted fins respectively; are concluded:

$$Nu_{av} = 6 \times 10^{-22}(Ra_{av})^{3.4691} \quad ...(18)$$

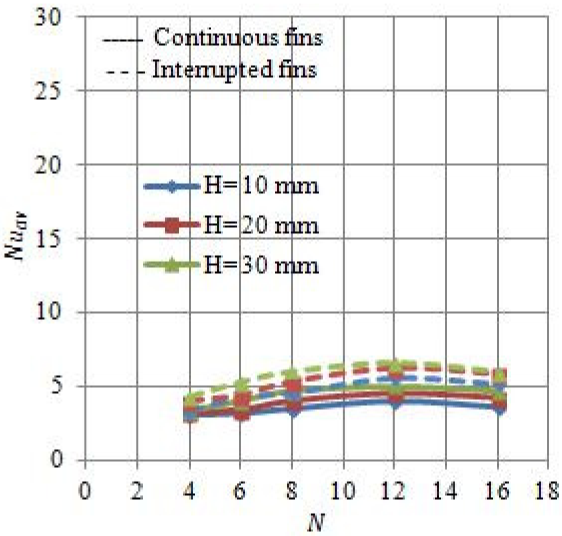
$$Nu_{av} = 5 \times 10^{-18}\,(Ra_{av})^{2.905} \quad ...(19)$$

Figures (10) and (11) show comparisons of the correlated average Nusselt numbers by Eqs. (18) and (19) with those experimental of present data for tubes with continuous and interrupted longitudinal fins. They are clear that the percentage error ($e\%$) of the experimental values located between +8% and -8% of correlated values for continuous fins and between +10% and -7% of correlated values for interrupted fins. Also the mean relative quadratic errors ($\bar{e}$) are 0.168 and 0.119 for continuous and interrupted fins respectively.

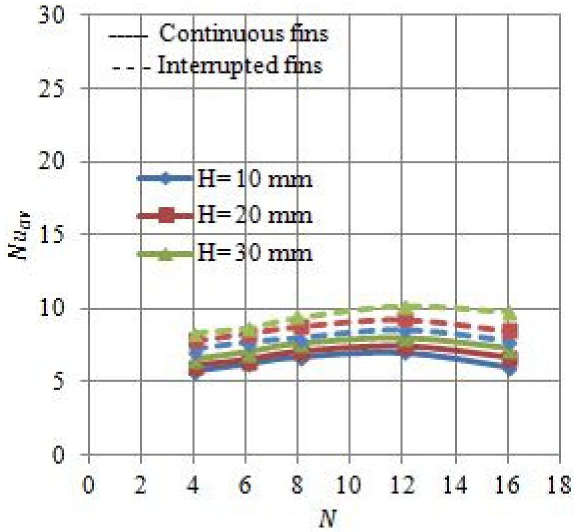
Moreover, the experiment is carried out on a heated vertical smooth tube with same conditions to compare than Morgan correlation:

$$Nu_D = 0.48\,(Gr_D\,Pr)^{0.25} \quad ...(20)$$

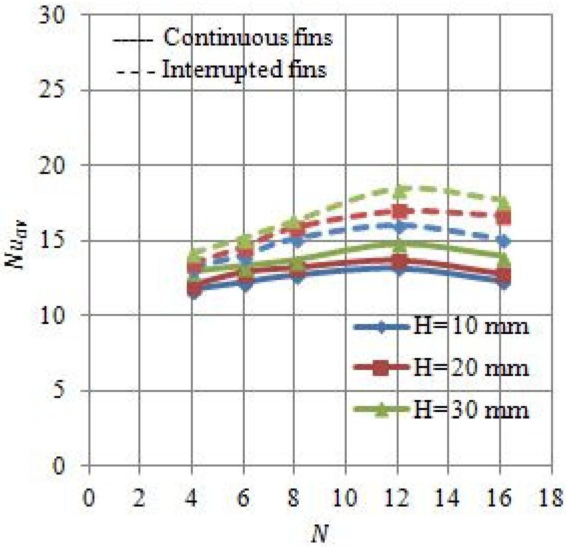
This is valid for values of Rayleigh number between $10^2 \leq Ra \leq 10^7$ as shown Figure (12). It can be seen a good agreement in behavior.



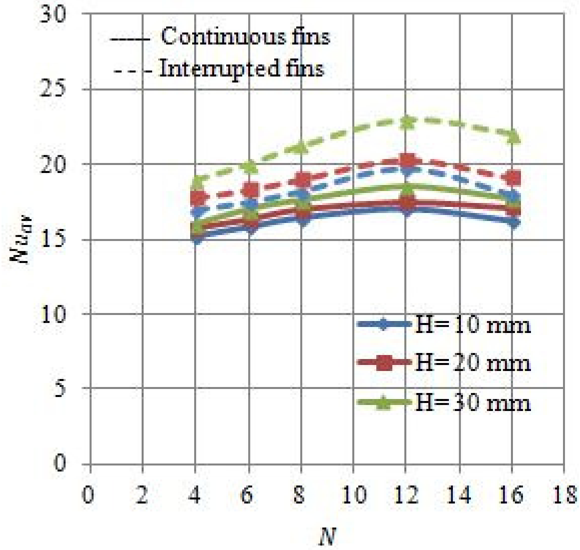
a. Surface heat flux, q= 796 W/m^2



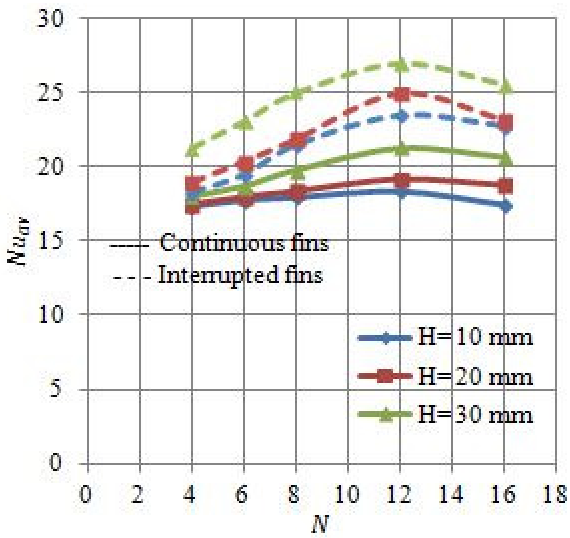
b. Surface heat flux, q= 1990 W/m^2



c. Surface heat flux, q= 3981 W/m^2



d. Surface heat flux, q= 5971 W/m^2



e. Surface heat flux, q= 7962 W/m^2

Fig. 8. Behavior of verage Nusselt number against fins number for different fin heights and surface heat fluxes.

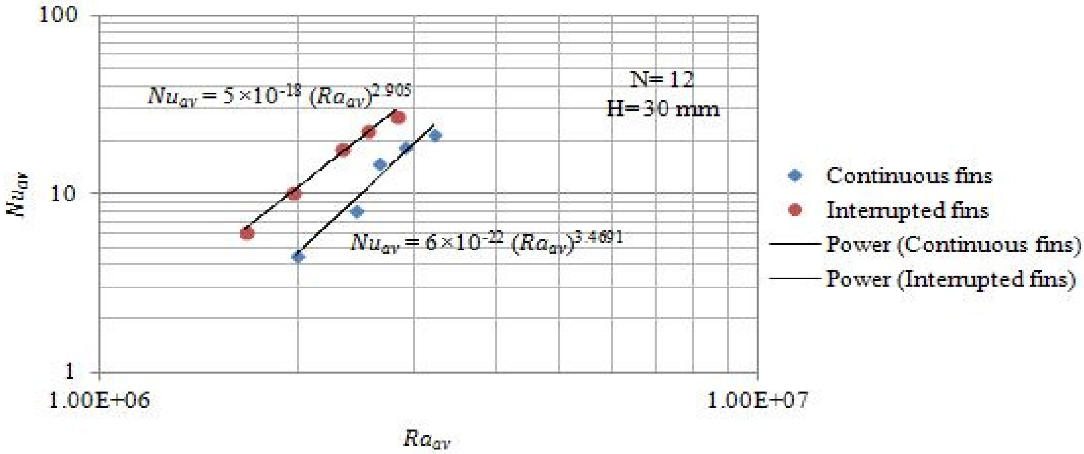


Fig. 9. Experimental analysis of present data for continuous and interrupted longitudinal fins with concluded correlations.

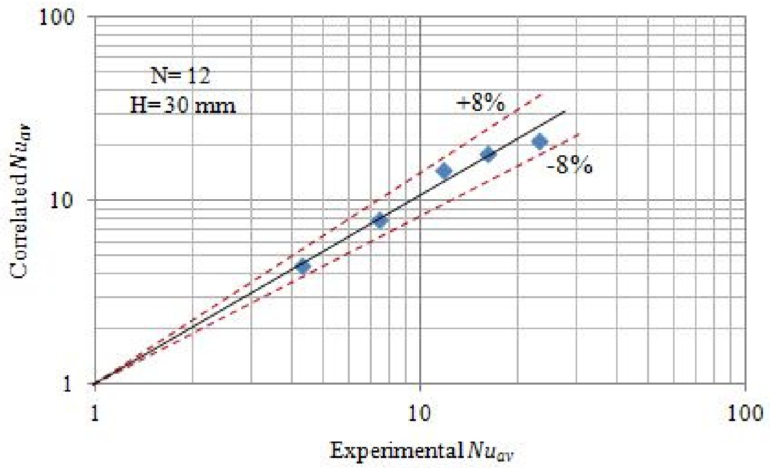


Fig. 10. Comparison of correlated average Nusselt numbers with those of the present experimental work for continuous longitudinal fins.

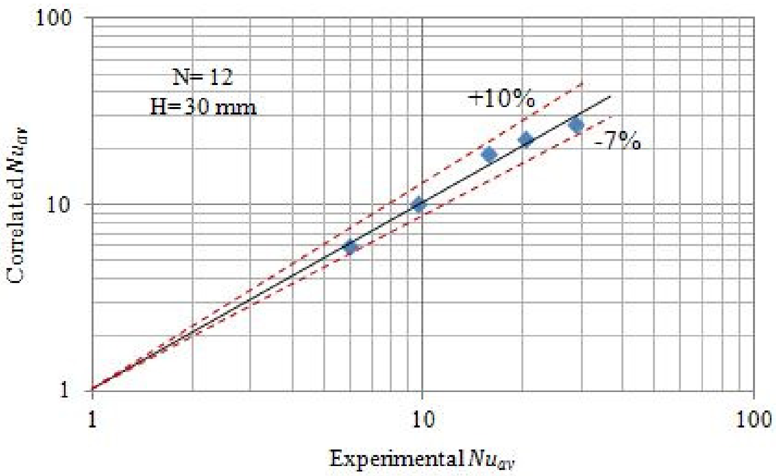


Fig. 11. Comparison of correlated average Nusselt numbers with those of the present experimental work for interrupted longitudinal fins.

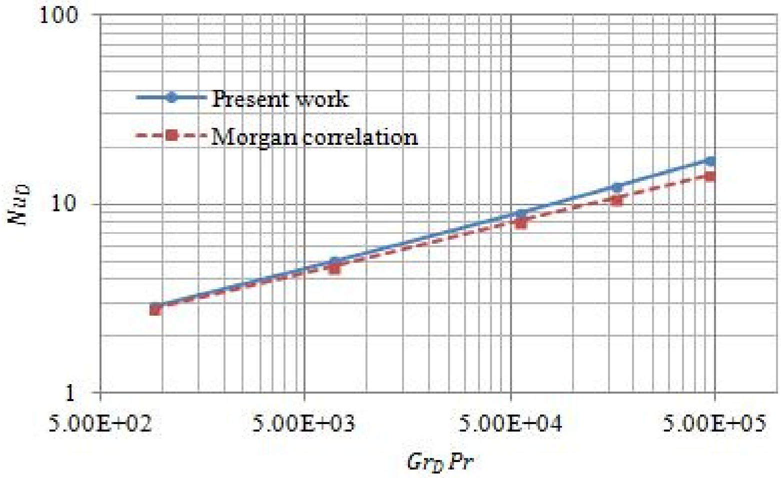


Fig. 12. Comparison the present experimental data for a heated smooth tube with Morgan [1] correlation.

5. Conclusions

The present work has performed an experimental analysis of a heated vertical externally-finned tube inside an opened enclosure exposed to natural-convection. The tube is electrically heated under constant surface heat flux and steady-state conditions. Continuous and interrupted longitudinal rectangular fins are utilized with different heights and numbers. The main conclusions can be drawn:

1. The average Nusselt number is strongly a function of fins number, fin height, and surface heat flux.
2. The optimum value of fins number is (Nopt= 12) for interrupted and continuous fins when maximizing average Nusselt number.
3. The average Nusselt number improves with increasing fin height and reaches to maximum values at (H= 30 mm).
4. The tube with twelve-interrupted longitudinal fins gave best natural-convection thermal performance in terms of average Nusselt number about 20% higher than those continuous longitudinal fins.
5. Two experimental correlations are concluded to expect the values of average Nusselt number for interrupted and continuous longitudinal finned-tubes.

Nomenclature

A	internal surface area of finned-tube, (m^2)
A_s	surface area of heat transfer, (m^2)
b	interrupted length, (m)
d	internal diameter of tube, (m)
D	external diameter of tube, (m)
D_f	external diameter of finned-tube, (m)
F	shape factor
g	gravitational acceleration, (m/s^2)
Gr	Grashof number
h	convection heat transfer coefficient, ($W/m^2.K$)
H	height of fin, (m)
I	input current, (A)
k	thermal conductivity, (W/m.K)
L_{int}	length of fin segment, (m)
L	height of tube or fin length, (m)
N	number of fins
Nu	Nusselt number
Pr	Prandtl number
q	surface heat flux, (W/m^2)
Q_{cond}	rate of heat conduction loss, (W)
Q_{conv}	rate of natural-convection heat transfer, (W)
Q_{in}	power input, (W)
Q_{rad}	heat radiation loss, (W)
Ra	Rayleigh number
t	thickness of fin, (m)
T_a	surrounding air temperature, (^oC)
T_f	film temperature, (K)
T_s	surface temperature, (^oC)
V	voltage, (V)

Greek Letters

ΔT	Temperatures differences, (K)
$\cos\phi$	power factor
β	volumetric coefficient of thermal expansion, (1/K)
ε	surface emissivity
ν	kinematic viscosity, (m^2/s)
σ	constant of Stefan-Boltzmann, (5.67×10^{-8} W/ $m^2.K^4$)

Subscript Symbols

av	average
D	based on tube external diameter
opt	optimum
x	local

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ظاهرة الحمل الطبيعي من أنبوب عمودي مسخن مزعنف: تحليل تجريبي

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الخلاصة

يتناول هذا البحث دراسة تجريبية للتنبؤ بالأداء الحراري لظاهرة الحمل الطبيعي من أنبوب عمودي مزعنف خارجياً الى الهواء المحيط داخل مغلف مفتوح النهايتين. أستخدم نوعان من الزعانف ، مستطيلة طولية مستمرة ومستطيلة طولية متقطعة بسمك ثابت وأرتفاعات وأعداد مختلفة وزّعت بشكل شعاعي على السطح الخارجي للأنبوب المسخن. تم تسخين الأنبوب كهربائيا من السطح الداخلي بخمسة مستويات مختلفة للقدرة الداخلة. تم دراسة تأثير شكل الزعانف وعدد الزعانف وارتفاعهاوالفيض الحراري لسطح الأنبوب على الأداء الحراري للحمل الطبيعي عمليا. أظهرت النتائج المستحصلة أن الأنبوب ذا الأثنتي عشرة زعنفة طولية متقطعة يعطي أداء حرارياً أفضل للحمل الطبيعي بدلالة معدل رقم نسلت بنسبة أكبر بحوالي ٢٠% للأنبوب نفسه بزعانف مستمرة. أستنبطت علاقات عملية للتنبؤ بمعدل رقم نسلت لأنابيب بزعانف طولية مستمرة ومتقطعة. وقورنت النتائج المقدّمة مع دراسة سابقة وأظهرت تقارباً جيداً.