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"An experimentally investigate the fin thermal performance to the different fin spaces by natural convections"

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Abstract:

In oil and gas industries there are a lot off heat transfer devices used for different purposes. These devices are widely used in various industrial, transportation, or domestic applications such as heat exchangers thermal power plants, means of transport, heating and air conditioning systems, electronic equipments and space vehicles. In all these applications, improvements in the efficiency of heat exchangers can lead to substantial cost, space and materials savings. The research work summarized in this articles presents an experimental investigation on the effect of fin space (s) and aluminum materials on the fin performance using rectangular fins. The steady-state natural convection heat transfer from vertical rectangular fins extending perpendicularly from horizontal square base was investigated experimentally at new range not found in the previous works, this range of temperatures from 50 to 150 C^o . The effects of fin space parameter and base-to-ambient temperature difference on the heat transfer performance of fin arrays were observed and the environmental condition were determined. Five fin space settings,(22, 27, 30, 35 and 38 mm) with a constant fin height is 50mm for all types of configuration are presented in this work were employed under free convection heat transfer conditions. This range of fin space not found on previous study or research. The heat transfer area was kept the same. The performance of the fin expressed in terms of fin efficiency, effectiveness and thermal resistance as a function of the ambient temperature and fin space parameters has been study in this work. The dimensionless parameter Biot no. on the locally variable environmental condition is examined for different fin spaces to the fin heat transfer rate. Also, the effect of environmental condition is study. Experimental results show that the effect fin space on fin performance is more significant.. The maximum increase in convection heat transfer coefficient value obtained is about 22 percent. The increase in heat transfer coefficient value is also manifested by a corresponding decrease in the fin base temperature. Also, it is concluded from the experimental results that the performance of heat transfer rate increase with decreasing the fin space in respect of heat transfer coefficient, thermal resistance ,overall efficiency and effectiveness.

Key Word: (Fin space, Rectangular Fin , Natural Convection, Heat Transfer Performance) .

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دراسة عملية لقياس الاداء الحراري للزعانف في حالة تغير المسافات بينهما بواسط أنتقال الحرارة الطبيعي.

الخلاصة :

التطور الكبير في صناعات النفط والغاز أدى الى أستخدام الكثير من أجهزة أنتقال الحرارة لأغراض متعددة . وكذلك تستخدم هذه الأجهزة على نطاق واسع في مختلف التطبيقات الصناعية ، ووسائط النقل ، أو ضمن الصناعات المحلية مثل المبادلات الحرارية ومحطات الطاقة الحرارية ، وأنظمة التدفئة وتكييف الهواء ، واجهزة الحاسوب والمعدات الإلكترونية والمركبات الفضائي. وتلعب الزعانف دور كبير في الحفاظ على هذه الاجهزة الحرارية وعملها المستمر لذلك ازدادت البحوث في هذا المجال . في جميع هذه التطبيقات ، يمكن أن تؤدي التحسينات في كفاءة المبادلات الحرارية إلى توفير كبير من التكاليف والفضاء والمواد. يقدم بحثنا هذا تلخيصه تجريبيًا حول تتحسين نقل الحرارة بواسطة الزعانف والمصنعة من معدن الألومنيوم بشكله المألوف المستطيل.

أنتقال الحرارة المنتظم للحمل الحراري الطبيعي من زعانف مستطيلة عمودية تمتد بشكل عمودي من قاعدة مربعة أفقية تم فحصها عمليا في نطاق جديد غير موجود في الأعمال السابقة ، وهو المدى الكبير لدرجات الحرارة من 50 إلى 150 مئوي, وكذلك الفراغ بين الزعانف حيث تم أختبار تأثير المسافة بين الزعانف المصطفة على القاعدة. حيث تم استخدام خمس مسافات مختلفة بين الزعانف المصطفة وهي (22 ، 27 ، 30 ، 35 و 38 ملم) بارتفاع ثابت للزعانف 50 مم وكذلك ثبات مقدار العرض والسمك لجميع أنواع الزعانف المستخدمة في عملنا هذا عند ظروف نقل الحرارة بالحمل الحر. كما تم دراسة تاثير النتائج في حالة الظروف العمل اليومي (نهارا وليلا) اي تاثير درجة حراة المحيط الجوي. كل ماذكر أعلاه تم أختباره عمليا على الاداء الحراري لعمل الزعانف. ويتم قياس الاداء الحراري بواسطة القياسات العملية الدقيقة لكل من كفاءة الزعنفة وفعاليتها ومقاومتها الحرارية كدالة في درجة الحرارة المحيطة وتحت المساحة السطحية والعدد الثابت للزعانف . أيضا تم حساب معامل عدد بيوت وهو معامل بدون أبعاد Biot .

أظهرت النتائج التجريبية أن هناك تأثير واضح في درجات الحرارة وكذلك المسافة بين الزعانف على أداء الزعانف الحراري والتي اثبتت المسافة او الفراغ أكثر أهمية في أنتقال الحرارة. بلغت الزيادة القصوى في قيمة معامل انتقال الحرارة الحراري حوالي 22٪. تتجلى أيضًا الزيادة في قيمة معامل نقل الحرارة بانخفاض مماثل في درجة حرارة القاعدة. أيضا ، تبين من النتائج التجريبية أن أداء معدل نقل الحرارة يزداد مع تقليل المسافة بين الزعانف والذي يعتمد بشكل مباشر على ، المقاومة الحرارية ، الكفاءة والفعالية للزعانف بشكل عام.اما تاثير العمل نهارا او ليلا فوجد هناك فرق بسيط في الاداء الليلي لعمل الزعانف بسبب امخفاض حراة الليل عن النهار وهذا الفرق ممكن أهماله لانه بسيط جدا.

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I. Introduction:

In the free-convection cooling of electronic and thermoelectric devices, as well as in improving the heat transfer in radiators for air conditioning and in other heat exchangers, finned surfaces are extensively used. Compared to a bare plate, a finned surface increases the heat transfer area. However, with the fins the flow rate reduced. Hence, if not properly designed it is possible that no improvement achieved in terms of overall heat transfer. Therefore, only if the fins properly designed, they are very attractive for these applications, since they offer an economical, trouble-free solution to the problem. Extended surfaces, which are popularly known as fins, are extensively used in air-cooled automobile engines and in air-cooled aircraft engines. Fins are also used for the cooling of computer processors, and other electronic devices. Fins are used in the cooling of oil carrying pipe line which runs several hundreds of miles. Heat pipes are also used along with fins to enhance cooling rate. A great deal of research effort has been developed for developing apparatus and performing experiments to define the conditions under which an agentive technique will improve heat transfer. The more effective and feasible techniques have graduated from laboratory to full-scale industrial equipment.
Starner and McManus [**1**] study experimentally very early in the heat transfer performance for arrays of rectangular fins by natural convection. They used four sets of fins array at different position (horizontal, 45 degree and vertical) based on the main heater at constant temperature 40 ^{o}C. They found the heat transfer coefficient for vertical position less than others by 10 to 30%.**Leung and prober, [2]** did another experimentally investigate the effect of fin height to the fin space for optimum ratio at two rectangular fins array positions (vertical and horizontal) . The results for the range used from 20 to 40 ^{o}C , shows the optimum fin spacing value were 9 to 11 mm . It was also found that not affect orientation considering to the change of fin height and base-to-ambient temperature difference.

Leung, Probert and Shilston [3] carried out experimental work for rectangular fins array at three different cases: vertical based on horizontal fins ,vertical based on vertical fins and horizontal based on vertical base . This work for a temperature range from 40 to 80 ^{o}C at three different heights, namely 32mm, 60 mm and 90 mm. There result showed no affect of fin height to the change of position, but the fin space is most effective for vertical fins based on vertical base. The effects of changing. fin length from 250 to 375 mm on the rate of heal transfer and the optimum fin spacing of vertical rectangular fins protruding from a horizontal or a vertical rectangular base have been investigated by **Leung. Probers and Shilston [4]** experimentally except fin length, other geometric parameters of several fin configurations were kept fixed for considered orientations. There result concerned at a constant base temperature,40°C above that of the ambient environment. The experimental measurements for vertical base showed that the increase in fin length caused reduction in the rate of heat dissipation per unit base area from the fin array. In addition, the optimal fin spacing rose from 10 ± 1mm to 11 ± 1 mm as a result of fin length increase. On the other hand, with horizontal base, large reduction in the rate of heat transfer per unit area occurred when the fin length was increased. The optimal fin. spacing of horizontally based fin array increased from 11 ± 1 mm to 1 4 ± 1 mm as the fin length was increased from 250 mm to 375 mm. All these consequences revealed that the effect of fin length on heal transfer performance of fin arrays is significant.**Walunj,.Daund,and .Palande**

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[5] studies various experimental have been made to investigate effect of fin height, fin spacing, fin length and fin thickness over convective heat transfer. Effects of thermodynamic properties like heat input, base-to-ambient temperature difference are also studied by many researchers. Some investigators make known sets of correlations screening the relation between various parameters of heat sink. Experiments are taken by some researchers for upward and downward facing rectangular fins. Also, trivial investigation has been carried out for different angle of inclination of the heat sink. The sensitivity of inclination over geometric parameters found to be great importance

Welling and Wooldridge [6] performed another experimental study to compare actual rectangular fin experiments with those of vertical plate, enclosed duct and parallel plate data from previous studies. During the tests, guard heater plate was utilized to minimize the heat losses from the sides and rear of the set-up. Data obtained from experiments showed that with closely spaced fins, the heat transfer coefficients were smaller compared to wider fin spacing's, because of boundary layer interference, which prevents air inflow. It was observed that the heat transfer coefficients of finned arrays were smaller than those of vertical plate and greater than either those of enclosed ducts or those of parallel plates. For a given base-to-ambient temperature difference, an optimum H/s (fin height to fin spacing) ratio at which heat transfer coefficient is maximum was determined from the considered fin configurations. **Mi sandar Mon, Ulrich Gross** [7] reported that the effect of fin spacing on four row annular finned tube bundles in staggered and inline arrangements are investigated by 3D numerical study. To investigate the velocity and temperature distribution between fins. The flow behavior of the developing boundary layer, the horse shoe vortex system, and thermal boundary layer developments in the annular finned tube banks will be visualized.

Azimifar A., Payan S [8] study the optimization of characteristics of an array of thin fins using PSO algorithm in confined cavities heated from a side with natural convection. **Hossein Z. and et.al. [9],**investigate natural convection of a Nano fluid in an enclosure with an inclined local thermal non-equilibrium porous fin considering with respect of fin spaces. However **Emel Evren S. and et.al . [10] experimentally** study the effect of fins space on the transition to oscillating laminar natural convection in an enclosure. **Khalil K., Abdalla AlAmiri [11]**reported that the effect of fin space at laminar natural convection heat transfer in a differentially heated cavity with a thin porous fin attached to the hot wall. **Ilker T., Mehdi M [12]** studies various experimental have been made to investigate effect of fin height, fin spacing and fin length for inclined position over convective heat transfer. Recently **Rishikesh and Kiran [13]** presented the characterization of radial curved fin heat sink with the effect of fin heights.

II. Experimental methods:

The experimental apparatus is comprised of a rectangular fin as cross-section to the U-heater shape direction in an open loop. The former is used to control the base temperature from 50 to 150 ^{o}C, and the latter sets the fins surface temperature. Figure 1. shows a schematic representation of the test rig, which is divided into the lower part, where the tests are carried out by different space and the upper part, where the power supply and temperatures gauges are work together. The rectangular aluminum table has a dimensions (90 cm x 60 cm) prove the base of the

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fins on the heater. So that the base of the fins has a constant area for five sets of fins and this base seek directly on the heater so that the transmitted heat conduction from the heater to the base of fins that contains a row of fins is working to expel heat to the surrounding environment, with an insulator between the heater and the base of the apparatus and demonstrate electrical panel containing gauges and switches in the front of the base of the apparatus as observed in figures 1 .

Figure (1) Overall view of the experimental rig and associated instrument.

Five aluminum plates size (250 mm) * (250 mm) ,has thermal conductivity (233w/m²k) installed . On each one set of aluminum fins each fin height and width are constant (50 mm &250 mm) has the same surface area, number and the same specifications as the plates. The five sets of plate contains (6 fins) separated by a distance (22 mm), and the second plat separated by a distance (27 mm), the third base has distance 30mm, fourth plate has space equal to 35mm, while the fifth plat separated by a distance (38 mm) . Under each one fixed heater to gives temperature and thermocouple to measure temperature.

Figure (2) Base of Aluminum plate with rectangular fins.

The three horizontal electrical U-heaters are placed on 70 mm above the experimental table to avoid ground effect. Electrical heating coil with 2.25 kW capacity is kept inside the tube. Thermal conductivity of aluminum is 236 W/mK.Heat transfer coefficients is important fin parameters measured from the following formula: The heat transfer coefficient (h, w\m^2 .K)) can be estimated from the following equation: [14,15]

$$h = \frac{Q_{fin}}{A_t * (T_s - T_\infty)} \qquad (1)$$

Where: Q_{fin} is the heat transfer from the fin surface at T_s,
A_t is the total fin surface area,
T_∞ is the ambient temperatures.

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The fin efficiency of a any fin ,η_{fin}, is defined as:

$$\eta\text{fin} = \frac{qfin}{qfin\ max} = \frac{Acual\ heat\ ftrnsfer\ rate\ from\ the\ fin}{Ideal\ heat\ transfer\ rate\ from\ the\ fin}) \quad ....(2)$$

This relation enables us to determine the heat transfer from a fin when its efficiency is known. But the overall fins efficiency is express by the following formula:

$$\eta o = 1 - ((\frac{Afin}{At})(1 - \eta fin) \quad ....(3).$$

The performance of the fins judged on the basis of the enhancement in heat transfer relative to the no-fin case . The performance of fins expressed in term of the fin effectiveness ϵ_f is defined as :

$$\epsilon_f = \frac{q_f}{hA_{c,b}\theta_b} \quad ....(4)$$

Fin thermal Resistance (R_{fin}) is defined as temperature rise per unit of power, analogous to electrical resistance, and is expressed in units of degrees Celsius per watt (°C/W). If the device dissipation in watts is known, and the total thermal resistance is calculated, the temperature rise of the die over ambient can be calculated as express in the following formula: [14,15]

$$R_{fin} = \frac{1}{h * Af * \eta f} \quad ....(5)$$

This equation may be used to expression for the thermal resistance of a fin array. A small value of thermal resistance indicates a small temperature drop across the heat sink, and thus a high fin efficiency. R_o is an effective resistance that account of heat parallel flow paths for conduction-convection in the fins and by convection from the prime surface. Equation 10 represent the overall effective resistance R_o . The governing equation for one dimensional conduction with convection is applicable to systems in which the lateral conduction resistance is small relative to the convection resistance. Under these conditions the temperature profile is one dimensional. The conditions for which Eq. (6) is valid are determined from the following criterion:

$$Bi = \frac{hlc}{K} < 0.1 \quad ....(6)$$

Where Bi is the Biot number based upon the maximum half thickness of the fin profile.
The fin Biot number is simply the ratio of the lateral conduction to lateral convection resistance: [14]

$$Bi = \frac{R\ conduction}{R\ convection} \quad ....(7)$$

III. Results and Discussions:

The mean point in our work is study the effect fin space to the fin thermal performance at very important range base temperatures from 50 to 150 ^{o}C. This range of temperatures has a huge application in the oil and gas industries and mechanical industries especially the heat exchanger ,internal combustion engine and power station . To examine the fin s heat performance for this wide range of temperatures, we need to estimate a many parameters such as heat transfer coefficient, fin effectiveness, fin efficiency, thermal resistance ,Biot number and heat transfer of

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fin surface. It decide to fixed the five sets of array of rectangular fin at a cross direction to the heater (power supply) as shown in the figure **3.** There is an approximately linear relationship between the heat transfer coefficient (h) for all base temperatures range used with fins space . As the fin space increase the heat transfer coefficient decrease due to low distance between the arrays of fins. The convection heat transfer rates from fin arrays and the vertical flat plate are plotted as a function of base-to-ambient temperature difference for fin spacing, s_1 = 22 mm(no.1), s_2 = 27 mm (no.2), s_3 = 30 mm (no.3), s_4 = 35 mm (no.4) and s_5 = 38 mm (no.5)and for fin a constant lengths, L = 250 mm , height [H = 50 mm] and thickness [1mm] as illustrated in figure 1 and table 1 respectively. The results for examine the effect the fins space (s) is observed in table.1 and figure.2 for morning time Am, but no big difference at the night Pm as found in the measurements.

Table 1 The heat transfer coefficient for different space with the base temperature at Am.

Tb(K)	h1(w/m²k)	h2(w/m²k)	h3(w/m²k)	h4(w/m²k)	h5(w/m²k)
323	189.8	160.33	136.36	113.12	91.46
343	105.4	88.91	76.52	60.65	49.5
363	82.86	69.95	59.75	47.14	37.13
383	57.91	47.01	39.57	33.51	28.85
403	46.6	38.05	32.05	27.04	23.77
423	37.4	30.81	25.6	23.01	20

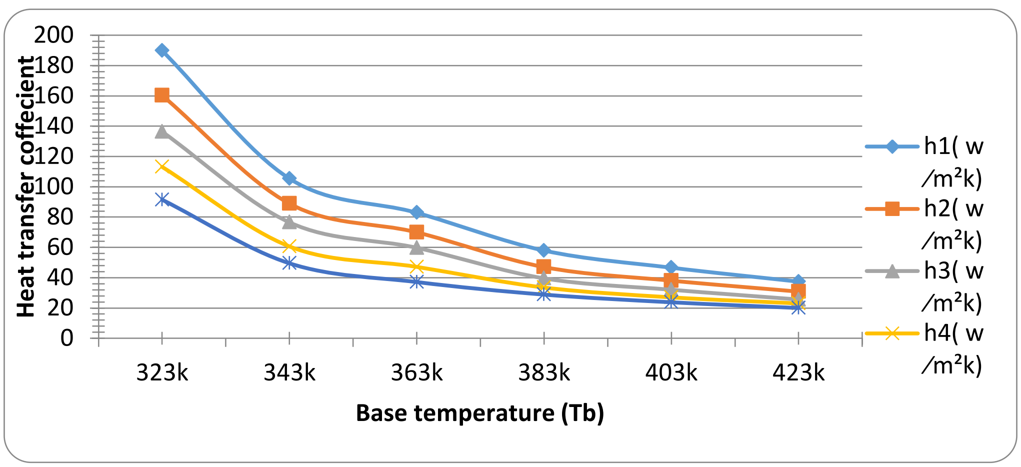

Figure 3. The effect of fin space with heat transfer coefficient for different base temperature at Am

It is seen that the convective heat transfer coefficient from the fin arrays increases with decrease the base temperature. The effect of extending the fin space from 22 mm to 38 mm results in higher steady-state convective heat dissipation from the fin arrays. However, the curves demonstrating the behaviors of fin spaced show decreasing the heat transfer coefficient with increasing the fin space for all base temperature range. The heat transfer coefficient measured from five fin spaced are close to each other at high base temperature whereas at low base temperature, the heat transfer rates tend to diverge with the fine space. For increasing the heat

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transfer rate we need the extends surface known a fin .Also, increasing the temperature difference between the fins and environmental .The best equation to calculate the heat transfer rate for rectangular fin is :[15]

$$\mathbf{Q_{fin} = M * \tanh(mL)} \quad \text{.................(8)}$$

Where: $M = \sqrt{h * p * k * Ac} * (Tb - T\infty)$

$$mL = \sqrt{\frac{h*p}{k*Ac}} * L$$

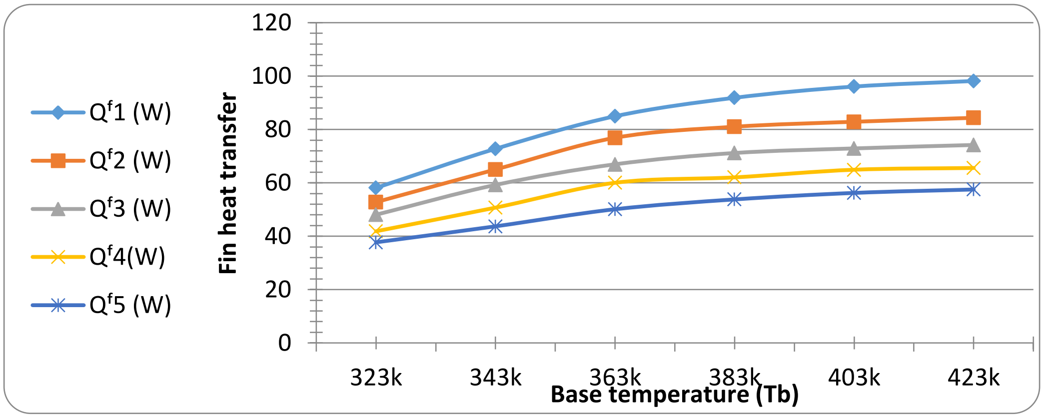

Figure 4. Fin heat transfer rate for five fin spaces at the base temperatures range.

It is very clear from the above figure increasing of fin heat transfer rate with decreasing the fin space. This is because increasing the different between the surface fin temperatures with base temperatures and increasing the heat transfer coefficient as shown in the previous figure .Fin performance can be described in three different ways. The first is fin effectiveness (equation no.4). It is the ratio of the fin heat transfer rate to the heat transfer rate of the object if it had no fin and the ratio of the fin heat transfer rate to the heat transfer rate of the fin if the entire fin were at the base temperature as defined as fin efficiency. Fin efficiency will always be less than one. This is because assuming the temperature throughout the fin is at the base temperature would increase the heat transfer rate. The third way fin performance can be described is with overall surface efficiency as described in equation no.3. We used equations 3 to 8 for our calculation to examine the fin thermal performance. Figure 5 represent the results of fin effectiveness with different fine space to the a wide range of base temperature.

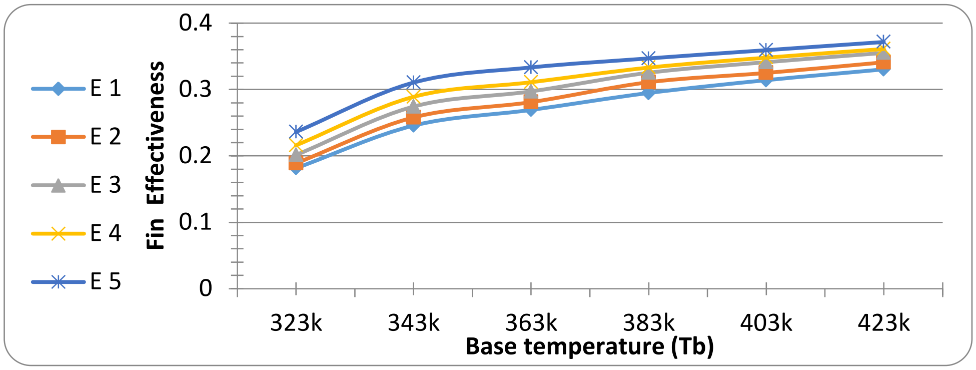

Figure 5 Fin space with Effectiveness' at different base temperature in the morning condition

It is clear from this figure, fin effectiveness is increase with increasing the base temperatures and fin space for rectangular shape. The results of fin effectiveness at the high temperatures more than 100^{o}C are very closed for five fin spaces**.** The second way for measure the performance is calculate the efficiency for different space of fin at the range of temperatures used as shown in figure 6 and table 2 at morning condition**.**

Table2. The results of efficiency for different fin space

Tb(K)	$\eta^{f}1$	$\eta^{f}2$	$\eta^{f}3$	$\eta^{f}4$	$\eta^{f}5$
323	0.4376	0.468	0.5048	0.544	0.5894
343	0.5593	0.589	0.6273	0.665	0.7142
363	0.6103	0.641	0.6777	0.719	0.7658
383	0.6839	0.716	0.7551	0.778	0.8081
403	0.7261	0.755	0.7898	0.808	0.8333
423	0.7645	0.791	0.8231	0.838	0.8588

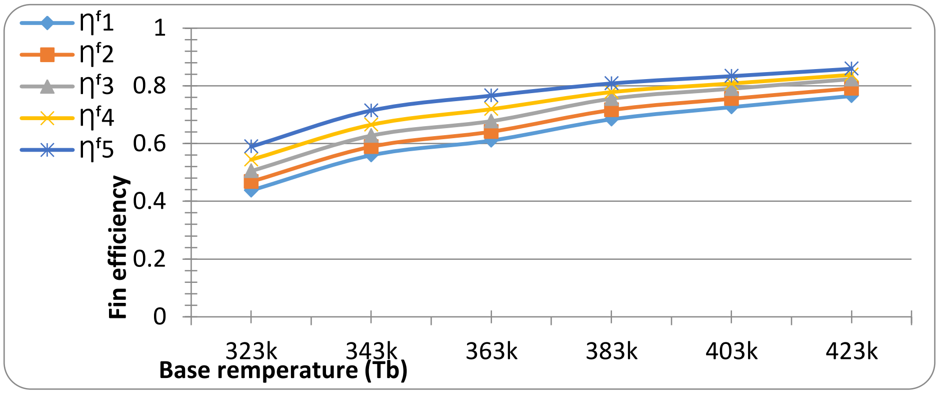

Figure 6. Efficiency of fin with different space and base temperatures.

It is obvious from the above figure, the fin efficiency increase with the increasing the base temperatures and the spaced fin. . In morning heat transfer coefficient is lesser than in the

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evening and efficiency will change according to this parameter. Efficiency of different space at the night condition is 2.7% higher than the day condition. It shows that the efficiency of the fins is changed according to the environmental condition. Figure 7. represent the overall efficiency for five space between the two fins. The relationship is the same second way of measured the performance for all spaces to the range of base temperatures from 50 to 150 ^{o}C.

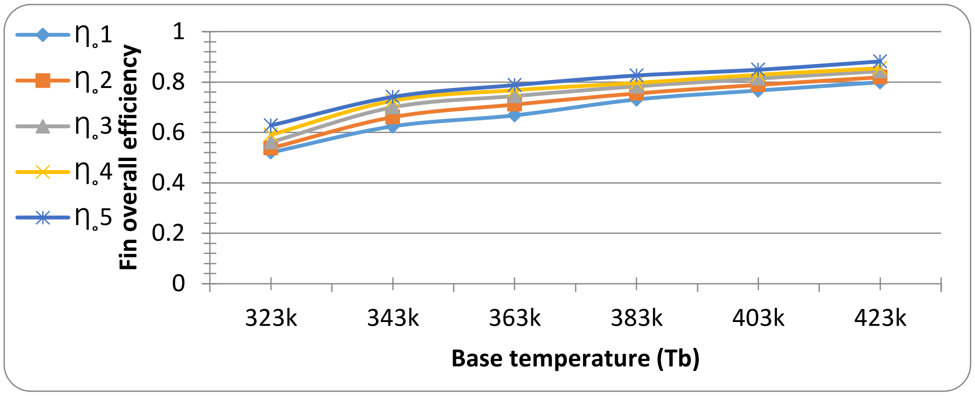

Figure 7. The overall efficiency for different fin space at Am condition.

The results for three types of ways to examine the fin performance to the heat transfer is very confirm, that's increasing of fin performance with increasing the fin space. Also, the fin performance increase with increasing the base temperatures. Thermal resistance is defined as temperature rise per unit of power, analogous to electrical resistance, and is expressed in units of degrees Celsius per watt (°C/W). If the device dissipation in watts is known, and the total thermal resistance is calculated, the temperature rise of the die over ambient can be calculated as express in formula no. 10.Tthe most important think is the calculation of overall thermal resistance R_{to}:[15]

$$R_{t.o} = \frac{1}{h * At * \eta_{o}} \qquad \text{.........(10)}$$

The results for effects the fin space with thermal resistance is illustrate in figure 8 .

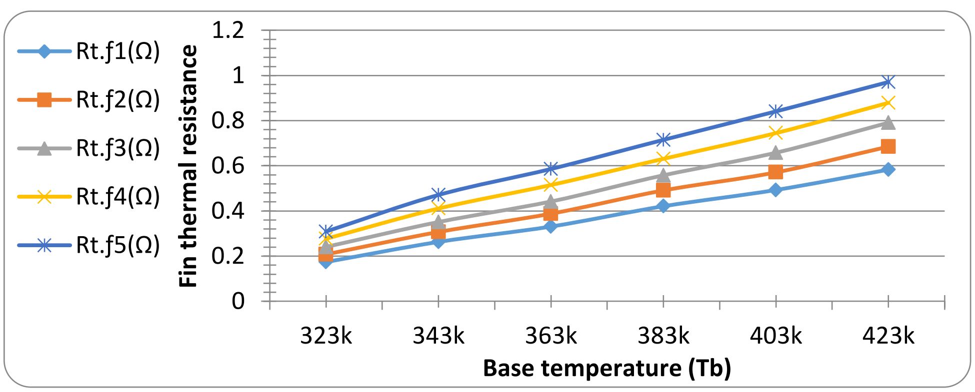

Figure 8. Fin thermal resistance against the base temperate for five fin space.

It is clear from this figure there is a linear relation between the base temperatures with fin thermal resistance for different fin spaces. Also, there is increasing of fin thermal resistance with increasing the fin space due to the decreasing the heat transfer coefficients of fins. The same

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relationship it's found for overall thermal resistance with different five fin space as shown in figure 9 and table 3 with very closed results for all the range of base temperatures used from 50 to 150 ^{o}C.

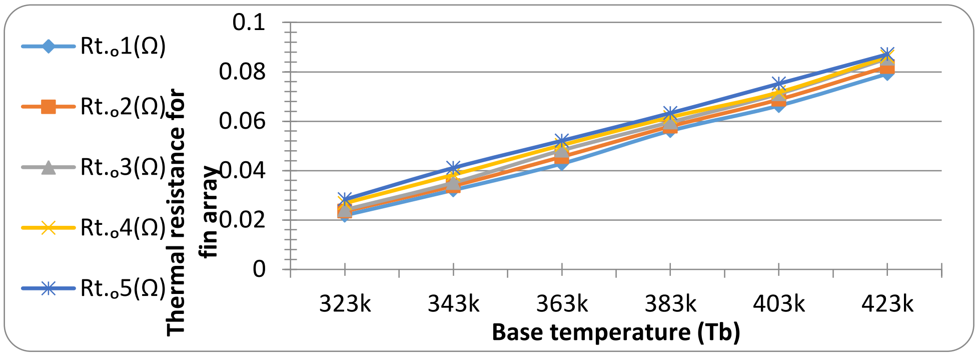

Figure 9. Overall thermal resistance at five fin space with base temperatures.

Table 3 Overall thermal resistance according to different base temp. for five fin space.

Tb(K)	$Rt._o1(\Omega)$	$Rt._o2(\Omega)$	$Rt._o3(\Omega)$	$Rt._o4(\Omega)$	$Rt._o5(\Omega)$
323	0.0219	0.0235	0.0241	0.0268	0.0283
343	0.0321	0.0338	0.0351	0.0383	0.0411
363	0.0428	0.0456	0.0482	0.0503	0.0521
383	0.0561	0.0579	0.0596	0.0616	0.0633
403	0.0663	0.0688	0.0711	0.0717	0.0752
423	0.0792	0.0821	0.0854	0.0861	0.0871

Biot no. is the ratio determines whether or not the temperatures inside a body will vary significantly in space, while the body heats or cools over time, from a thermal gradient applied to its surface. In general, problems involving small Biot numbers (much smaller than 1) are thermally simple, due to uniform temperature fields inside the body. Biot numbers much larger than 1 signal more difficult problems due to non-uniformity of temperature fields within the object. The Biot number has a variety of applications, including transient heat transfer and use in extended surface heat transfer calculations. In this work the Biot number is examined for different fin space to the big range of base temperatures from 50 to 150 ^{o}C as illustrated in figure 10 and table 4 for morning condition.

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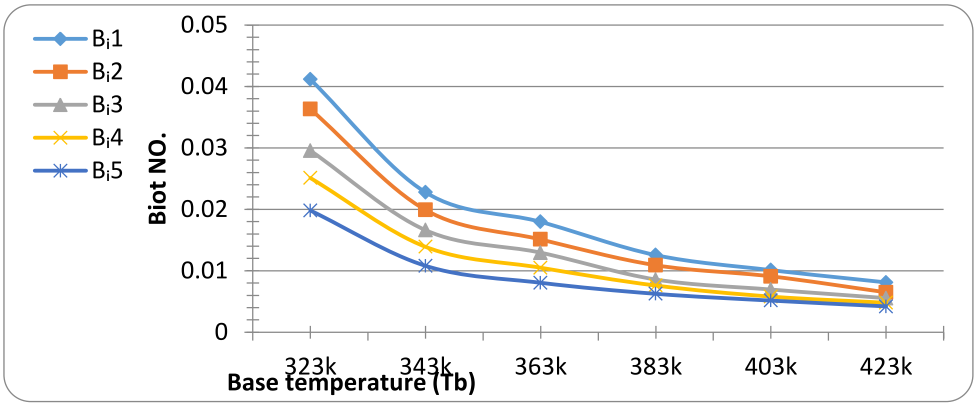

Figure 10. Biot no. at different base temperatures for many fin space at Am

.

It is clear from the above figure, that's Biot no. very small and less than 1 for all space of fin used. This mean uniform distributions of surface fin for rectangular shape. Also, it's found there is an inverse relation between the base temperatures and Biot no. for all space used. There is a closed results of Biot no. at high temperature up than 100^{o}C .

Conclusion:

The following conclusion can deduced from the present work:

1) It is found the heat transfer coefficient depends upon the space, temperatures and types of material. If there are changes in environmental conditions, there is a small changes in heat transfer co-efficient and efficiency also.

2) Average heat transfer increase with decreasing the fin space in natural convection mode.

3) The range of Biot no. for all fin spaces is from 0.004 to 0.043,this mean a good distribution in the surface of rectangular fin .

4) The range of effectiveness for different fin spaces is from 0.27 to 0.33 less than one.

5) As the fin spaces increase the overall thermal resistance, overall fin efficiency and fin effectiveness increase also.

6) No optimum fin space found in this research at the range from 22 to 38 mm.

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