Experimental Investigation of Natural Convection Heat Transfer from Vertical Grooved Plates

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***Abstract*— The areas such as microelectronic cooling, especially in central processing units, macro and micro scale heat exchangers, gas turbine internal airfoil cooling, fuel elements of nuclear power plants, and bio medical devices here heat transfer enhancement concept is used .In this work experimental investigation of natural convection heat transfer from vertical roughened surfaces has been carried out. The effect of variation in surface roughness and heat input on convective heat transfer coefficient and Nusselt number has also been evaluated. Natural convection from heat transfer from vertical roughened plates has experimentally been compared to that of plain plate. Three different roughened surfaces are made by varying pitch and by varying the depth of roughness. With increase in surface roughness the heat transfer rate, convective heat transfer coefficient and Nusselt number increases compared with plain plate.**

***Keywords-* Crossed groove, Natural Convection, Heat Transfer, Vertical Plates, Heat transfer coefficient, Nusselts number.**

I. INTRODUCTION

Natural convection represents a limit on the heat transfer rates and this becomes a very important consideration for problems in which other modes are either not possible or not practical. This free convection mostly used in electronic cooling system where the circuit boards represents a naturally enhanced free convection situation .In other application where the heat dissipating surface is normally smooth, it may be necessary to enhance the surface to achieve the desired temperature level for this achievement two methods we can employed first one add vertical fins and second one roughening the surfaces. The work by M. J .Sable[1] taking Plain vertical plates vertical fin array, V fin array of different sizes and carried out experimentation. and he concluded heat transfer coefficient for V fin array plates is more than rest of plates.

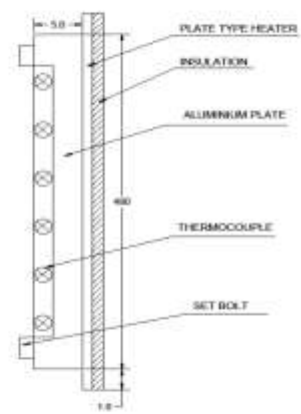
Heat transfer by free convection is the scope of this research work, this convection is present in both nature and man- made engineering system such on oceanic currents, wind formation oversea and in the rising plum of hot air from fire. Combine effect of conduction and free convection it appeared in many practical and industrial devices , such as fin heat transfer , electronic cooling, building insulation, hot wall heat exchanger used in small refrigerator and in solar energy collector , heat removal from spent nuclear fuel bundles and cooling of nuclear reactor after loss of coolant accident etc. are examples of engineering application. Heat transfer by natural convection phenomenon has been investigated by different researcher for various geometry. Firth [2] conducted the research on heat transfer performance as well as friction factor of these three different types of artificially roughened surface such as…Square transverse ribbed surface, Helically ribbed surface ,Trapezoidal transverse ribbed surface . The key result was that enhancement in heat transfer is achieved at the expense of higher friction factor. Also the result showed that heat transfer was enhanced along the rough region by great amount. Singh [3] result showed Nusselt number and friction factor for roughened duct compared to smooth duct were enhanced by 51.4% and 26.5% respectively. Abdullah H. A [4] study investigated the gain in fin area and perforation dimension, the perforated fin enhances heat transfer. The magnitude of enhancement is proportional to the fin thickness and its thermal conductivity. Also, the extent of heat dissipation rate enhancement for perforated fins is a function of the fin dimensions, the perforation geometry and the fin thermo physical properties. Linhui [5] experimentally studied the laminar natural convection phenomena from a vertical plate with discrete heat sources. They found the constants of the general correlation between Nusselt and Rayleigh numbers, as a quadratic equation in terms of d / L and computed the values of A and n of equation. Mohammad Mashud [6] research, a solid cylindrical fin and two other cylindrical fins with circular grooves and threads on their outside surface are investigated experimentally. The heat input to the fin is varied such that the base temperature is maintained constant under steady state. Based on a study of effect of pressure reduction, using available resources, the chamber is designed for a vacuum of 680 mm Hg. The experimental result shows that for cylindrical fin with circular grooves (depth 3.5mm) heat loss is a maximum. The grooved cylindrical fin loses approximately1.23 times greater heat per unit area, compared to the threaded cylindrical fin, and 2.17 times greater heat per unit area, respectively compared to the solid pin fin at a pressure lower than atmospheric pressure. As pressure decreases heat loss reduces and contribution of radiation heat transfer on total heat loss increases. Christopher L. Chapman[7] had work on comparative thermal tests have been carried out using aluminum heat sinks made with extruded fin, cross-cut rectangular pins, and elliptical shaped pins in low air flow environments. The elliptical pin heat sink was designed to minimize the pressure loss across the heat sink by reducing the vortex effects and to enhance the thermal performance by maintaining large exposed surface area available for heat transfer. The performance of the elliptical pin heat sink was compared with those of extruded straight and cross-cut fin heat sinks, all designed for an ASIC chip. The results of the straight fin were also compared with those obtained by using software Sauna T M, a commercially available heat sink modeling program developed based on empirical expressions. In addition to the thermal measurements, the effect of air flow bypass characteristics in open duct con-figuration was investigated. As expected, the straight fin experienced the lowest amount of flow bypass over the heat sink. For this particular application, where the heat source is localized at the center of the heat sink base plate, the overall thermal resistance of the straight fin was lower than the other two designs mainly due to the combined effect of enhanced lateral conduction along the fins and the lower flow bypass characteristics. Enhanced surfaces increase heat transfer due to increased surface area and the increased fluid turbulence that result from the surface design this work done by David J. Kukulka[8].Dharma Rao [9] investigated the problem of laminar natural convection heat transfer from a fin array containing a vertical base and horizontal fins is theoretically formulated by treating the adjacent internal fins as two fin enclosures. The governing equations of mass, momentum and energy balance for the fluid in the two fin enclosure together with the heat conduction equations in the fins are numerically solved using ADI method. The heat transferred to the ambient fluid from the two end fins is also computed separately. Heat transferred by radiation is considered in the analysis. The numerical results are compared with the experimental data available in literature. The effects of system parameters such as base temperature, fin height, fin spacing on heat transfer rate from the fin array are studied. They concluded the following points

- With decreasing of fin spacing heat transfer rates are increasing sharply.
- Convective heat transfer rates from a fin array are increasing with increasing of base temperature of the fin array at all fin spacing's.
- Average Nusselt number for short fins more than long fins.

Naylor and Tarasuk [10] have investigated the natural convection phenomenon by numerical as well as experimental techniques using a vertical channel divided by a heat generating wall. They used, the finite element method based upon CFD code FIDAP to solve the Navier-Stokes equations and the laser interferometry, an experimental technique to observe the isotherm patterns. They also compared their numerical and experimental results and found that the average Nusselt numbers trend is similar for both techniques but the experimental values of Nusselt number were 10% lower than the numerical predictions. D.N Ryu [11] conducted an investigation to analyze the resistance in turbulent flow channels roughened by two-dimensional ribs (square ribs, triangular ribs, semicircular ribs, wavy wall) and three-dimensional blocks. And he concluded that highest average Nussel number, is co-related to that of maximum resistance coefficient for ribs. Tanda [12] have investigated natural convection phenomena from a staggered array of horizontal plates and found that heat transfer performance of middle and upper plates were affected by changing the position of the plate. They found that the convective interactions among the plates were identified by examining the heat transfer coefficient. A.E. Bergles [13] have been work on surface roughness technique belongs to passive techniques according to the research give the conclusion that the artificial surface roughness enhances the heat transfer by triggering the thermal layer breakup as well as facilitating intensive turbulent mixing. This phenomenon, however, is accompanied by unexpected raise of frictional resistance. The research by Prasolov (14), Heya et al (15), Fujii et al (16), Bhavnani and Bergles (17), investigated that the roughness element whose height is less than the boundary layer thickness will have no appreciable influences on the heat transfer of natural convection.

II. EXPERIMENTATION

Figure 1 (a) and (b) shows schematic diagram and actual photograph of experimental set up. The experimental system includes the duct has dimension (800 mm × 150 mm × 150 mm) to maintain the environment for pure natural convection. The duct is kept open from bottom and top side. The 4 test plates are made up of Aluminum material having dimension (480 mm × 50 mm) held vertically in the enclosure with the help of hangers which are supported at top. The plate type heater (480 mm × 50 mm × 5 mm) is attached to test plates with the help of nut and bolt arrangement. The amount of heat supplied to heater is regulated by connecting the dimmer stat (specification 230V for input and 270 Ohm resistance) in between the heater and main electric supply. With the help of dimmer stat the voltage across the heater can be regulated which regulated the electric power supplied to heater. The six holes are drilled in the front sides to insert the temperature sensors (T type Thermocouple) which are further connected to the digital temperature indicator (0 – 300 ^{0}C) to indicate temperature. One temperature sensor is immersed in the enclosure to measure the ambient temperature.



(a)

(b)

Figure 1 (a) Experimental set up, (b) Photographic view of experimental set up

II.I TEST SECTION

For the test we are made up four plates of aluminum material has dimension 480mm X 50mm X 5mm .Out of four first is plain surface plate Schematic diagram and actual photograph as shown in figure (2) .second plate is crossed grooved surface having pitch 5mm, depth 0.3mm and lip angle 30 ^{0}. Schematic diagram and actual photograph as shown in figure (3). Third plate is crossed grooved surface having pitch 5mm, depth 1mm and lip angle 30^0. Schematic diagram and actual photograph as shown in figure (4). And forth plate is diamond knurled surface having pitch 10mm, depth 1mm and lip angle 30^0. Schematic diagram and actual photograph as shown in figure (5).

Table 1 Geometrical parameters of the test section

Sr. No.	Test Surface	Depth (mm)	Pitch (mm)	Angle (Degree)
1	Plain	0	0	0
2	Cross Grooved	0.3	5	30
3	Cross Grooved	1	5	30
4	Diamond Knurled	1	10	30

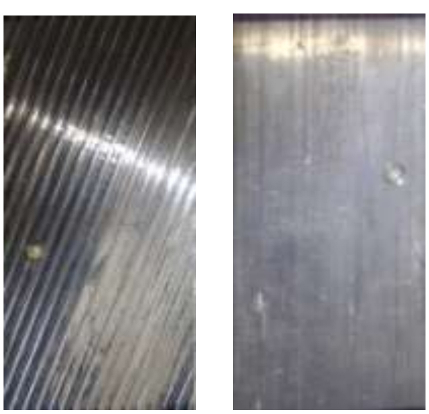


FIGURE 2 PHOTOGRAPHIC VIEW OF TEST PLATES

II.II EXPERIMENTAL PROCEDURE

In order to Experimental Evaluation of Natural heat transfer coefficient through vertical heated roughened surface, it has been decided to vary the heat input from 25 W to 100 W in the step of 25 W. Readings are taken at steady state. Heater is placed behind the test plate. Whole test set up is mounted vertically. Voltage supplied is varied with the help of dimmer stat.

1. Switch on the supply and adjust the dimmer stat to obtain the required heat input (25W, 50 W, 75 W, 100 W).
2. Monitor the temperature T1 to T5 every five minutes till steady state is reached.
3. Wait till the steady state is reached. This is confirmed from temperature readings (T1 to T5). If they remain steady and do not register a change of more than 1oC per hour.
4. Measure the surface temperature at various points (T1to T5).
5. Note the ambient temperature, T6.
6. Repeat the experiment for different heat inputs (25W, 50 W, 75 W, 100 W). by varying dimmer stat position.

III. DATA REDUCTION

Following data reduction process is adopted for an analysis. Heat supplied to the system calculated by Eq. (1)

$$Q = V \times I \qquad (1)$$

Where, Q is heat supplied in watts, V is the voltage, I is the current.

And heat transfer coefficient determined by Eq. (2)

$$hexp = Q'/A \times [(Tsavg) - Tamb] \qquad (2)$$

Actual heat supplied Q' is calculated By Eq. (3)

$$Q' = Q/2. \quad (3)$$

Because Heater is sandwitched between heated plate and insulating material

Where A is the area of test plates, (Tsavg) is the average surface temperature calculated by Eq. (4)

$$(Tsavg) = (T1+T2+T3+T4+T5+T6)/6 \quad (4)$$

Tamb = ambient temperature

Tf = Film temperature Calculated by Eq. (5)

$$Tf = ((Tsavg) + Tamb)/2 \quad (5)$$

Properties of Aluminum at Film Temperature (Tf):

Density (ρ) in kg/m3, Specific heat (Cp) in J/ kg.K, Dynamic Viscosity (μ) in kg /m s, Thermal Conductivity (K) in W/m. K, β = 1/Tf in / K, Prandtl No. (Pr)

Rayleigh Number (Ra) is calculated by Eq. (6)

$$\text{Rayleigh Number (Ra)} = \frac{g \times \beta \times \Delta T \times L^3 XPr}{v^3} \quad (6)$$

Nusselt number calculated by Eq. (7)

$$Nu = hL/k \quad (7)$$

IV. RESULT AND DISCUSSION

Prior to starting the experimentation, Temperature difference (Tsurface – Tamb.) is found to be repeatable at deviation up to 1.1 °C for given heat flux .

4.1 Heat transfer coefficient comparison

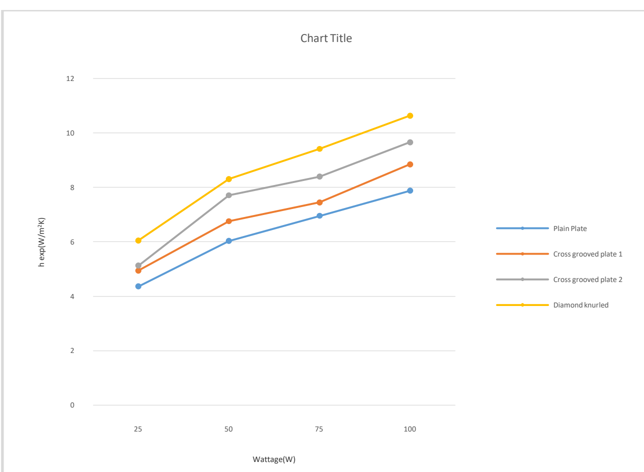


Figure 3 Heat transfer coefficient Vs Heat supplied for all plates

Figure (3) shows the variation of heat transfer coefficient with heat supplied. Heat transfer coefficient increases as the heat supplied increases. Because increase the temperature of boundary layer which result into increase in exchange of the heat between fluid and surface due to large temperature difference .This is attributed to the grooveness to the surface then this additional surface interrupt the temperature profile . As a result, then boundary layer breaks and is redeveloped repeatedly at leading edge of each fins. This flow always remaining in developing stage in regime of the fins. Redevelopment of the thermal boundary layer provides better mixing of the fluid and improves heat transfer rate. As the surface grooveness increases then the heat transfer coefficient also increases because as in case of plain surface plate there is less surface area (18% less as compared to first grooved surface plate, 21% less as compared to second grooved surface plate and 28% less as compared to diamond knurled surface plate) as compared to the remaining grooved plates so that heat transfer coefficient for the plain surface plate is less as compared to other plates.

4.2 Heat transfer coefficient Vs Temperature Difference

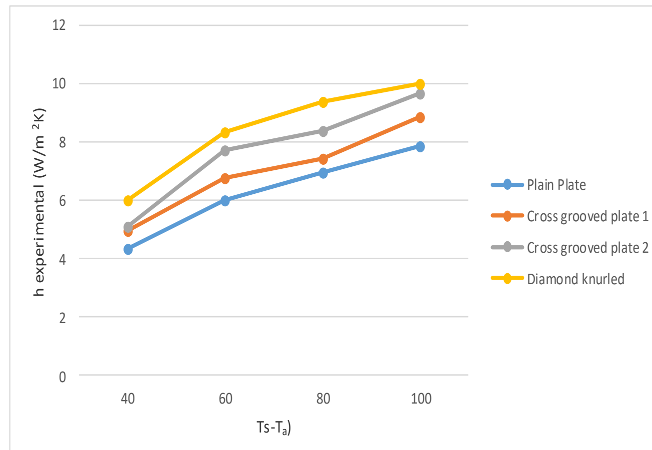


Figure 4 Heat transfer coefficient Vs Average surface temperature difference for all plate

Figure (4) shows the variation of heat transfer coefficient with Temperature difference. As the Temperature difference increases then the heat transfer coefficient also increases because as the heat supplied increases then temperature also increases so due to that of boundary layer which result into increase in exchange of the heat between fluid and surface. In case of grooved surface there will be better redevelopment of the thermal boundary layer and its results better mixing of fluid ultimately increases the heat transfer coefficient.

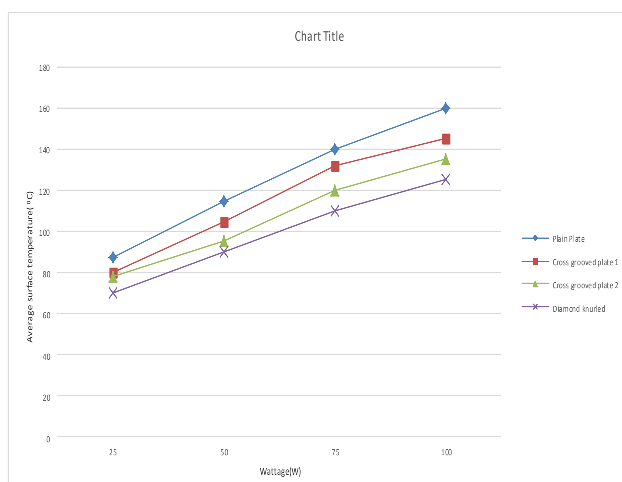


Figure 5 Average surface temperature Vs Heat supplied for all plates

4.3 Average surface temperature Vs Heat supplied

Figure (5) shows the variation of Average surface temperature with heat supplied. As the heat supplied increases then the temperature also increases for all plates. For plain surface plate average surface temperature is more as compared to remaining plates because surface area of that plate is less (18% less as compared to first grooved surface plate, 21% less as compared to second grooved surface plate and 28% less as compared to diamond knurled surface plate) compared to remaining plates so that this plate suddenly heated

4.4. Comparison of temperature between the Tested plates

Figure (6) shows the variation of surface temperature (T in 0 C) with height of the temperature sensor (L in cm) from bottom of the plates. for the plain surface plate there will be high surface temperature than the other plates reason behind this already discuss in Figure (3). The temperature difference between plain surface plate and first crossed grooved surface plate is 100^0C. The temperature difference between first crossed grooved surface plate and second crossed grooved surface plate is 20^0C because in that two plates are made up of crossed grooved surface only change is depth. In diamond knurled surface plate there will be 80^0C difference with the second cross grooved plate. But in diamond knurled surface plate there will be the more surface area so the temperature is reduced compared with other plate surface area.

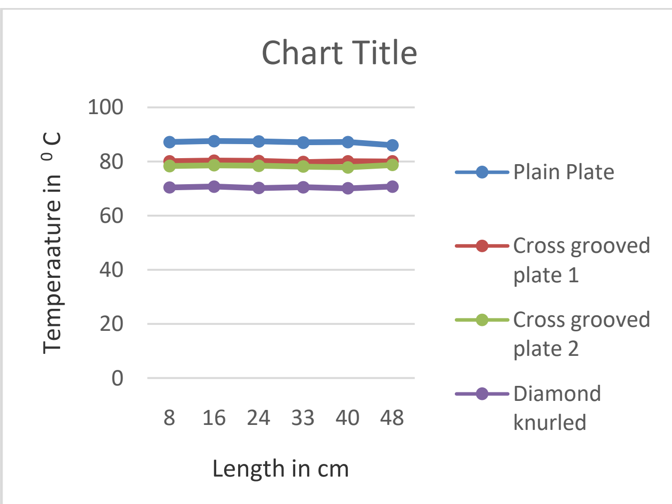


Figure 6 Comparison of temperature between the Tested plates

4.5 Nusselt number Vs Rayleigh number for all plates

Figure (7) shows the variation of Nusselt number with Rayleigh number. Nusselt number increases as the Rayleigh number increases for the all plates because increament of nusselt number is depend on the heat transfer coefficient and increament. Rayleigh number depend on the temperature difference but the relation between heat transfer coefficient and temperature difference already discuss in figure (4) however from we conclude that as the Nusselt number increases as the Rayleigh number increases. Also same reason for the graph line for plain surface plate is at lower position as compared to other plates.

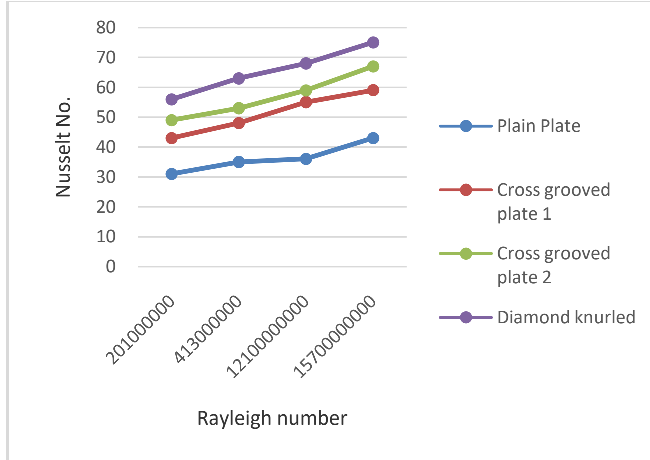


Figure 7 Variation of Nusselt number Vs Rayleigh number for all plates.

V. Conclusion

In this work experimental investigation of natural convection heat transfer from vertical roughened surfaces has been carried out. The effect of variation in surface roughness and heat input on convective heat transfer coefficient and Nusselt number has also been evaluated.

1. With increase in surface roughness (18% more surface area of first grooved surface plate as compared to plain surface plate ,21 % more surface area of second grooved surface plate as compared to plain surface plate and 28 % more surface area of diamond knurled surface plate as compared to plain surface plate) the convective heat transfer coefficient (13.58% more heat transfer coefficients of first grooved surface plate as compared to plain surface plate ,17.70 % more heat transfer coefficients of second grooved surface plate as compared to plain surface plate and 28 % more heat transfer coefficients of diamond knurled surface plate as compared to plain surface plate) is increases.
2. With increase in surface roughness (18% more surface area of first grooved surface plate as compared to plain surface plate ,21 % more surface area of second grooved surface plate as compared to plain surface plate and 28 % more surface area of diamond knurled surface plate as compared to plain surface plate) the Nusselt number (13.43% more Nusselt number of first grooved surface plate as compared to plain surface plate ,17.55 % more Nusselt number of second grooved surface plate as compared to plain surface plate and 28.73 % more Nusselt number of diamond knurled surface plate as compared to plain surface plate) increases compared with plain plate.

3. The heat transfer coefficients and Nusselt number obtained from the roughened plate are higher than that of the plain plate due to boundary layer breakup.
4. For a given heat input the average plate temperatures continuously increases.

REFERENCES

[1]. M.J. Sable, S.J. Jagtap, P.S. Patil, P.R. Baviskar, S.B. Barve "Enhancement of Natural Convection Heat Transfer on Vertical Heated Plate by Multiple v-fin array", IJRRAS 5 (2) November 2010

[2]. Firth, R.J. and Meyer, L., 1983, "A comparison of the heat transfer and friction factor performance of four different types of artificially roughened surface", Int. Journal of Mass and Heat Transfer, vol. 26, no. 2, pp. 175-183.

[3]. Singh, S., Chander, S., Saini, J. S., 2011, "Heat transfer and friction factor correlations of solar air heater ducts artificially roughened with discrete V-down ribs", Energy, Volume 36, Issue 8 , 5053–5064.

[4]. Abdullah H. AlEssa, Ayman M. Maqableh and Shatha Ammourah "Enhancement of natural convection heat transfer from a fin by rectangular perforations with aspect ratio of two", International Journal of Physical Sciences Vol. 4 (10), pp. 540-547, October, 2009

[5]. Linhui, T. Huaizhang, L. Yanzhong and Z. Dongbin, 2006, Experimental study on natural convection heat transfer from a vertical plate with discrete heat sources mounted on the back, Energy Conversion and Management, Vol. 47, no. 1, pp. 3447-3455.

[6]. ohammad Mashud, Md. IliasInam, ZinatRahmanArani and AfsanulTanveer "Experimental Investigation of Heat Transfer Characteristics of Cylindrical Fin with Different Grooves", International Journal of Mechanical & Mechatronics Engineering IJMME Vol: 9 No: 10

[7]. Christopher L. Chapman, and Seri Lee Bill L. Schmidt "Thermal Performance Of An Elliptical Pin Fin Heat Sink", Tenth IEEE SEM1-THERM

[8]. David J. Kukulka, Kevin G. Fuller "Development of an Enhanced Heat Transfer Surface", 20th European Symposium on Computer Aided Process Engineering – ESCAPE20

[9]. Dharma Rao, S.V. Naidu, B. GovindaRao and K.V. Sharma "Combined Convection and Radiation Heat Transfer from a Fin Array with a Vertical Base and Horizontal Fins", Proceedings of World Congress on Computer science 2007 WCECS 2007, october 24-26, 2007, San Francisco, USA

[10]. D. Naylor and J. D. Tarasuk, 1993, (a) Natural Convection Heat Transfer in a Divided Vertical Channel: Part-I Numerical Studies, Journal of Heat Transfer ASME Series C, Vol. 115, no. 1, pp. 377-387

[11]. D. Ryu, D. Choi, V. Patel, 2007, "Analysis of turbulent flow in channels roughened by 1321 two-dimensional ribs and three-dimensional blocks": part II: heat transfer, Int. 1322 J. Heat Fluid Flow 28, pp. 1112–1124.

[12]. G. Tanda, 1993, Natural Convection Heat Transfer from a Staggered Vertical Plate Array, Journal of Heat Transfer; Transactions of the ASME Vol. 115, no. 1, pp. 938.

[13]. A.E. Bergles, A.R. Blumenkrantz, J. Taborek, 1974, "Performance evaluation criteria for enhanced heat transfer surfaces", in: Proceedings of the Fourth International Heat Transfer Conference vol. 2, pp. 239-243.

[14]. R.S. Prasolov, "The effect of surface roughness of horizontal cylinder on heat transfer to air".Inzh-fiz (In Russian) vol.4 pp.3 1961.