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Research Article

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Transient Convection Thermal Analysis of Heat Exchanger Using Ansys Software Package

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Abstract A heat exchanger (cooling fin) made of an aluminum alloy with thermal properties of 170W/mK isotropic thermal conductivity, 870J/kgK specific heat and 2800kg/m^3 density was analyzed with Transient Thermal system from ANSYS software package. Additional conditions were ambient temperature and Film coefficient of 20^{0}C and 30W/m^2 ^{0}C respectively. The initial (room) temperature of the alloy was set at 20^{0}C, heat source temperature of 90^{0}C, step end time 200s, initial time step 2s, minimum time step 0.2s and maximum time step at 20s. As the cooling process continues, it was observed that while at a time of 33.537s the temperature at fin-base juncture (probe 2) assumed a steady state and becomes uniform from 132.17s until the maximum time (200s) where the temperature increased to 89.114^{0}C, the tip of the base without fin (probe 3) assumed a steady state at a time of 37.83s and becomes uniformly distributed from 152.17s until the maximum time where the maximum temperature was 89.321^{0}C. It was also observed that no other part attained or exceeded 90^{0}C.

Keywords Heat Exchanger, Cooling Fin, Plate Fin, Transient Thermal Analysis, Ansys, Film Coefficient, Cross Flow, Counter-Flow, Cross-Counter Flow. Heat Flux, Temperature Distribution

Introduction

The transient response of fins is important in a wide range of engineering devices including heat exchangers, clutches, motors and so on [1]. The authors' interest in this topic grew out of analysis of transient data for determination of surface heat transfer coefficients on Aluminum heat exchanger, in which the outer face of the plate is treated as a fin. Next a review of literature is given in the areas of transient fins and in transient experiments for estimation of heat transfer coefficients. Heat sinks are objects that absorb and dissipate heat from another hotter object by thermal contact either by direct conduction transfer or radiation transfer. Cooling fins are projections designed to increase the surface are from which heat can be radiated away from a device.

Plate fin exchanger is a type of compact heat exchanger where the heat transfer surface area is enhanced by providing extended metal surface, interfaced between two fluids and is called the fins [2]. Out of the various compact heat exchangers, plate fin heat exchangers are unique due to their superior construction and performance. They are characterized by high effectiveness, compactness, low weight and moderate cost. As the name suggests, a plate fin heat exchanger (PFHE) is a type of compact exchanger that consists of a stack of alternate flat plates called parting sheets and corrugated fins brazed together as a block.

A plate fin heat exchanger can have few or multiple streams which may flow in parallel or perpendicular directions to one another.

There exist three main configurations for plate fin heat exchangers [3]. These are: (a) cross flow, (b) counter-flow and (c) cross-counter flow.

The cross flow heat exchanger configuration, as shown in figure 1, involves fluids that flow normally to each other. This type of configuration comprises only two streams and the largest structural temperature difference occurs at the corner side of the entering hot and cold fluids. The header tanks in this type of arrangement are located on all four sides of the heat exchanger core. This arrangement makes this type cheap and simple. If high effectiveness is not necessary and if the two fluid streams have widely different volume flow rates or if either one or both streams have constant temperature, the cross flow arrangement should be preferred. Typical application for cross flow configuration type of PFHE include automobile radiators and some aircraft heat exchangers.

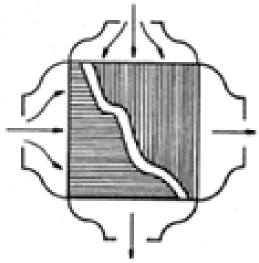

Figure1: Cross flow heat exchanger

The counter flow PFHE provides the most thermally effective arrangement for recovery of heat or cold from process streams because the two fluids flow parallel to each other in opposite direction. Although, counter flow arrangement demands proper design because of its complex geometry of headers, it produces the highest temperature change in each fluid compared to any other two fluid arrangements for a given overall thermal conductance, fluid flow rates and fluid inlet temperatures. Application of this type of arrangement involves cryogenic refrigeration and liquefaction equipment utilizes this geometry almost exclusively. Figure 2 shows counter flow configuration of PFHE.

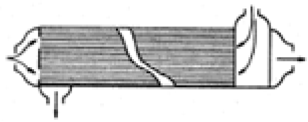

Figure 2: Counter flow heat exchanger

The cross counter flow arrangement is a hybrid of counter flow and cross flow geometries. This makes the thermal effectiveness of counter flow PFHE with superior transfer properties of the cross flow arrangement. In this configuration, one of the streams flows in a straight path while the second stream follows a zigzag path which is normal to that of the first stream. While moving along the zigzag path, the second fluid stream covers the length of the heat exchanger in a direction opposite to the direct stream. This type of configuration is shown in figure 3 and is suited for the applications where the two streams have considerably different volume flow rates, or permit significantly different pressure drops. Here, the fluid with the highest volume flow rate or that with the smaller value of allowable pressure drop is made to flow through the straight channel while the other stream follows zigzag path.

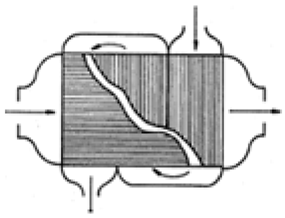

Figure 3: Cross counter flow heat exchanger

For instance, in a liquid-to-gas heat exchanger, the gas stream with a large volume flow rate and low allowable pressure drop is allowed to follow the straight path, while the liquid stream with a high allowable pressure drop flows normal to it over a zigzag path. This configuration optimizes the overall geometry.

In a separate development, Cole et. al. [3], working on analysis of flux-base fins for estimation of heat transfer coefficient opined that traditional fin analysis describes a long, thin, high conductivity body with a specified temperature one end and a convection condition over the surface. The temperature distribution in the fin depends on the competing effects of conduction along fin and convection from the surface of the fin. Tseng et al [5] also analyzed transient heat transfer but in two-dimensional straight fins of various shapes subjected at their base to a decayed exponential function of time in heat flux. Saha and Acharya [6] also conducted a detailed parametric analysis of unsteady three-dimensional flow and heat transfer in a pin-fin heat exchanger which was motivated by their desire to enhance the performance of compact heat exchangers which are designed to provide high heat transfer surface area per unit volume and to alter the fluid dynamics to enhance mixing. There have been other numerical studies of transient fns combined with complicating factors such as natural convection [7-8], phase change materials [9] and spatial arrays of fins [10-11].

Fins, as earlier said, are used in a large number of applications to increase heat transfer process from surfaces. This paper aims to study transient heat transfer in an aluminum alloy fin. The fin is exposed to a flowing fluid (air in this analysis), which cools or heats it, with the high thermal conductivity allowing increased heat being conducted from the wall through the fin.

Methodology

Mathematical Analysis

In this section, transient temperature in flux-base fins is analyzed mathematically. Considering a straight fin initially in equilibrium with the surrounding fluid environment at temperature T_e, with a constant cross-sectional area, but may be of rectangular, pin or any shape. For $time\ (t) > 0$, steady heat flux is applied to the base of the fin. The temperature satisfies the following equations:

$$\frac{\partial^2 T}{\partial x^2} - m^2(T - T_e) = \frac{1}{\alpha}\frac{\partial T}{\partial t}; \qquad 0 < x < L \qquad (1)$$

$$at\ t = 0, \qquad T(x, 0) - T_e = 0 \qquad (2)$$

$$at\ x = 0, \qquad -k\frac{\partial T}{\partial x} = q_o \qquad (3)$$

$$at\ x = L, \qquad K_2\frac{\partial T}{\partial x} + h_2(T - T_e) = 0 \qquad (4)$$

The quantity m is the fin parameter given by $m = (\frac{hA_h}{KV})^{0.5}$. The boundary condition at $x = L$ is a general condition that represents one of three different tip conditions for the fin. For a tip condition of the first kind, setting $K_2 = 0$ and $h_2 = 1$ represents a specific end temperature at $(T_e = T)$. For a tip condition of the second kind, setting $K_2 = K$ and $h_2 = 0$ represents an insulated end condition. For a tip condition of the third kind, setting $K_2 = K$ represents convection at $x = L$. It is not necessary that the convective coefficient at the end of the fin be the same as that along the sides of the fin (i.e. $h_2 \neq h\ in\ general$), but often $h_2 = h$ is used.

Analysis with Ansys software

An alloy of aluminum with thermal properties as shown in table 1 and figure 4 was analyzed. The initial (room) temperature was set at 293.15K, heat source temperature of 90^0C, step end time 200s, initial time step 2s, minimum time step 0.2s and maximum time step at 20s as shown in figure 5. This initial temperature serves as a reference temperature of the aluminum fin to start with. The end time was set at 200s so that there will be enough time for the convection process to take place. Ambient temperature and Film coefficient of 20^0C and $30W/m^2\ ^0C$ respectively. In addition to the analysis, two points to be analyzed are temperature probes 2 and 3 as shown in figure 6.

Table 1: Properties of the Aluminum alloy

Engineering Data	Values
Density	$2800kg/m^3$
Isotropic Thermal Conductivity	170W/mk
Specific Heat	870 J/kgk

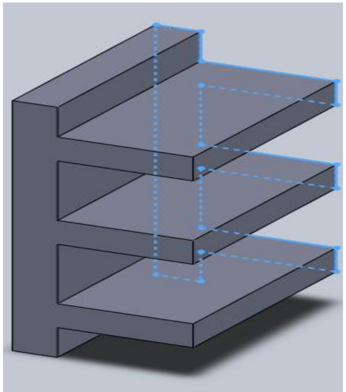

Figure 4: Aluminum fin

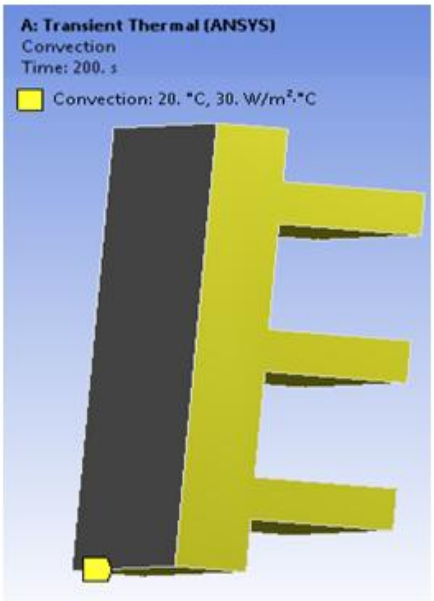

Figure 5: Convection process input values

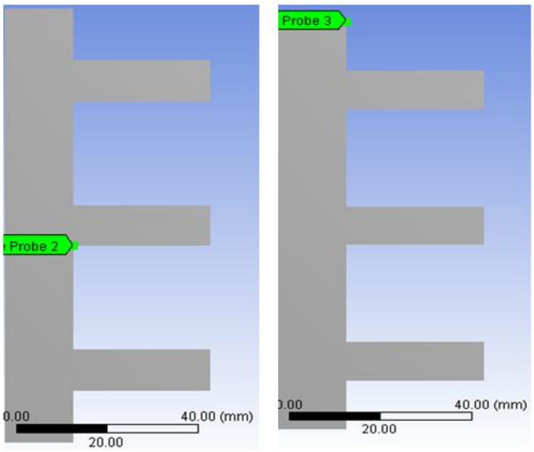

Figure 6: Temperature distribution probe points

Results and Analysis

1. Temperature changes at probes 2 and 3. From the different temperature probe results at vertices 2 and 3 respectively, the plots shown in figure 6 have been displayed by the Ansys simulation result.

It shows that, from the beginning of the analysis, the temperatures at both vertices increase from 20^0C but stopped at different temperatures. While the maximum temperature for Vertex 2 is at 89.114^0C that of Vertex 3 is at 89.321^0C. The little difference might be because of convection and conduction of heat from Vertex 2 since one fin is joined to the point at 2. This fin conducts heat from the Point hence reducing the temperature at this point by conduction while the convection process also takes place. For Vertex 3, there is no fin attached to it, hence at the maximum time, the heat at that point has only convection process to dessipate heat to the environment. From the graph in figure 7 and table 2, it has been displayed that the temperature changes with time. The temperature is seen to have rising from 20^0C. This is because the given initial temperature of the model was set at 20^0C. As the cooling process continues, it is shown in table 2 that while at a time of 33.537s the temperature at probe point 2 assumes a steady state and becomes uniform at 132.17s until the maximum time (200s) where the temperature is 89.114^0C, that of probe point 3 assumes a steady state at a time of 37.83s and becomes uniformly distributed at a time of 152.17s until the maximum time where the maximum temperature is 89.321^0C.

Table 2: Temperature distributions at vertices 2 and 3 respectively

	Time (s)	Temp at Probe 2 (Vertex A) (^{0}C)	Temp. at Probe 3 (Vertex B) (^{0}C)
1	0	20	20
2	2	44.639	51.543
3	2.6667	52.844	61.426
4	3.2288	58.57	67.867
5	3.7909	63.041	72.514
6	4.7527	68.232	77.284
7	6.2023	73.101	81.082
8	8.5679	77.595	83.945
9	12.176	81.398	85.932
10	16.379	84.059	87.156
11	20.66	85.793	87.911
12	24.953	86.921	88.392
13	29.245	87.669	88.709
14	33.537	88.162	88.918
15	37.83	88.488	89.055
16	42.122	88.702	89.146
17	46.414	88.843	89.206
18	50.707	88.935	89.245
19	54.999	88.996	89.271
20	59.291	89.037	89.288
21	72.168	89.084	89.308
22	92.168	89.105	89.317
23	112.17	89.111	89.32
24	132.17	89.114	89.32
25	152.17	89.114	89.321
26	172.17	89.114	89.321
27	192.17	89.114	89.321
28	200	89.114	89.321

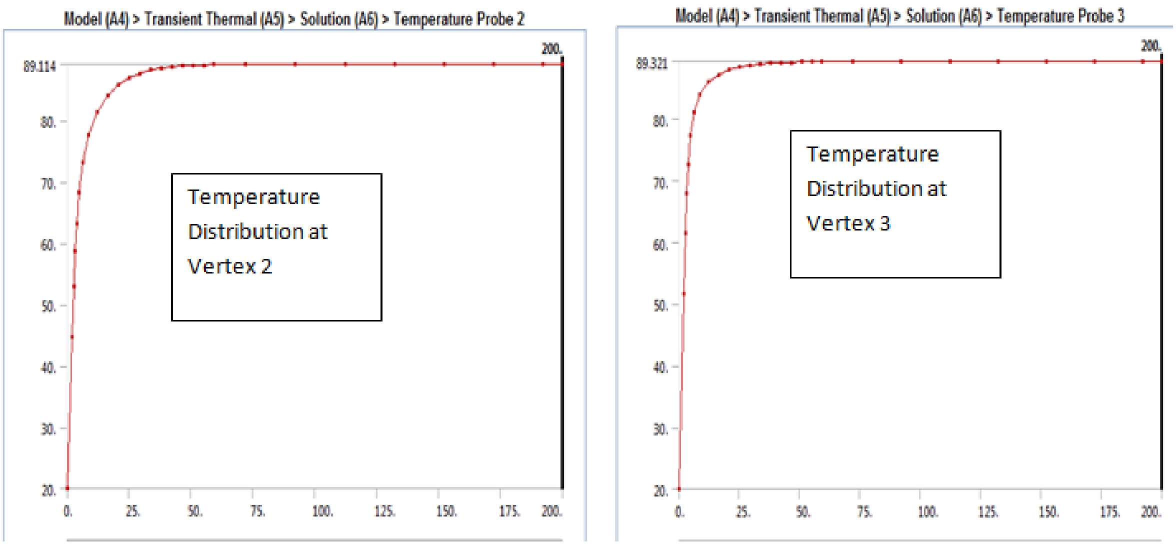

Figure 7: Temperature distribution plots at vertices 2 and 3 respectively

2. Temperature distribution over the fin

The plot for the minimum and maximum temperature distribution over the fin is shown in figure 8 and table 3. It shows that over a period of time, the temperature is distributed over the body with the base having the maximum temperature while the fins display minimum temperature. This enables engineers to know the maximum length of material to be used for construction of cooling fins. It has been seen that temperature distribution increases rapidly with time at initial stage.

Table 3: Change in temperature with time across the fin

	Time (s)	Minimum Temp. (^{0}C)	Maximum Temp. (^{0}C)
1	2	23.006	90
2	2.6667	24.461	90
3	3.2288	26.041	90
4	3.7909	27.948	90
5	4.7527	31.859	90
6	6.2023	38.317	90
7	8.5679	48.002	90
8	12.176	59.448	90
9	16.379	68.689	90
10	20.66	74.967	90
11	24.953	79.184	90
12	29.245	81.952	90
13	33.537	83.773	90
14	37.83	84.971	90
15	42.122	85.76	90
16	46.414	86.279	90
17	50.707	86.621	90
18	54.999	86.846	90
19	59.291	86.994	90
20	72.168	87.168	90
21	92.168	87.247	90
22	112.17	87.27	90
23	132.17	87.277	90
24	152.17	87.279	90
25	172.17	87.279	90
26	192.17	87.279	90
27	200	87.279	90

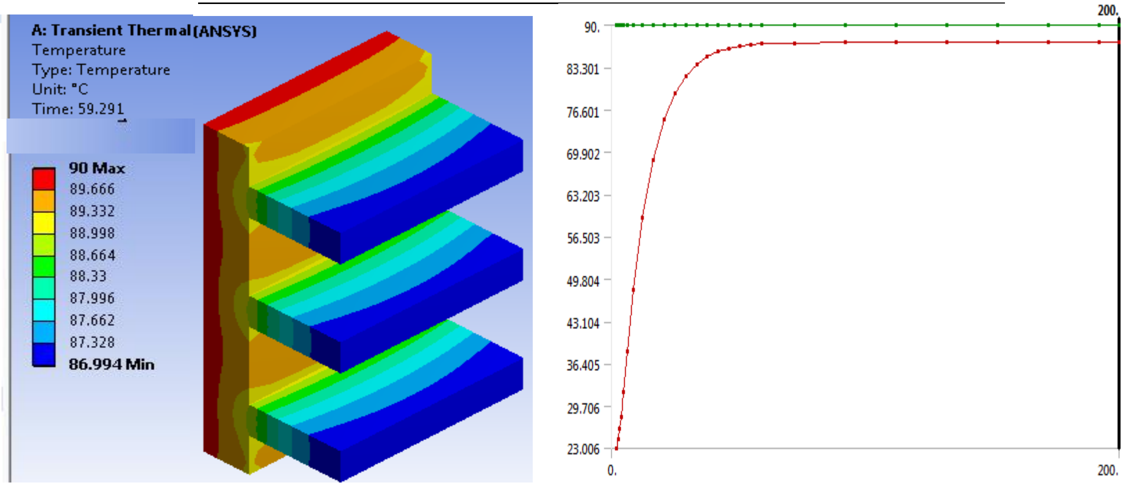

Figure 7: Temperature distribution plot over the fin

Then slowly increases until it reaches a steady state where the distribution becomes uniform as time increases. The cooling continues until 46.414s where it seems to assume a steady state. But finally, at a time of 152.17s the temperature becomes constant. Hence, this time is the time at which steady state occurs where the temperature is constant at 87.279^{0}C. At this stage, increasing the length of fin or time, will not change the temperature. So we can use this medium to identify the length of material needed and the cooling time. It is seen that the maximum temperature which is 90^{0}C is the temperature value of source of heat. The maximum temperature on any other part cannot exceed 90^{0}C because at any given point in time, the heat is transferred from the source to the point concern. So any point on the fin can experience heat transferred to it but cannot exceed or attain the source temperature value which is 90^{0}C in this analysis.

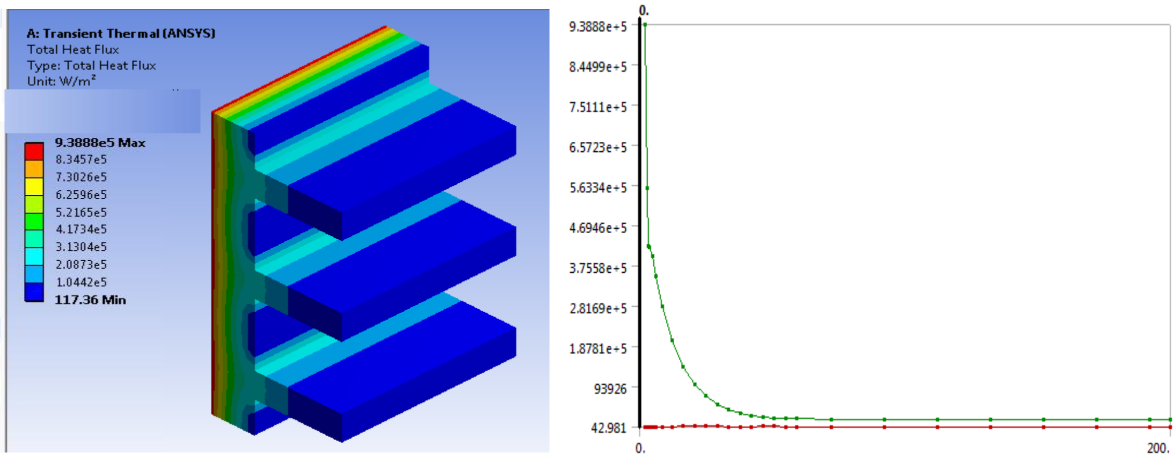

Figure 9: Heat flux distribution across the fin

Table 4: Heat flux distribution over time

	Time (s)	Minimum Heat flux. (W/m^2)	Maximum Heat flux. (W/m^2)
1	2	117.36	9.3888 e+005
2	2.6667	143.63	5.5755 e+005
3	3.2288	121.19	4.2296 e+005
4	3.7909	42.981	4.1913 e+005
5	4.7527	172.38	3.9803 e+005
6	6.2023	497.83	3.5105 e+005
7	8.5679	910.53	2.8178 e+005
8	12.176	1287	2.0381 e+005
9	16.379	1552.3	1.4202 e+005
10	20.66	1737.4	99947
11	24.953	1824.9	71942
12	29.245	1903.8	53336
13	33.537	1494.8	41062
14	37.83	1407.3	32980
15	42.122	1486.8	27664
16	46.414	1592	24168
17	50.707	1604.8	21870
18	54.999	1434.1	20359
19	59.291	1339.1	19366
20	72.168	1250.1	18202
21	92.168	1219	17674
22	112.17	1211.2	17519
23	132.17	1209.1	17474
24	152.17	1208.3	17461
25	172.17	1208.2	17457
26	192.17	1208.2	17456
27	200	1208.1	17455

Considering the heat flux plot shown in figure 9, it is seen that the heat reduces rapidly in the early stage of the analysis. As time continues the heat becomes uniformly distributed thereby changing the shape of the graph. Initially the slope is high. But at a time of 152.17s where the minimum flux is 1208.3W/m^2 and the maximum heat flux is 17461W/m^2 as shown in table 4; the slope is approxmately zero indicating that the heat flux does not change any longer.

Conclusion

Transient thermal analysis has been used to analyze heat transfer through an aluminum alloy heat exchanger. The analysis has been used to determine the time and length of material needed to design heat exchanger at a given temperature.

The heat distribution over the fins tells how fast different areas absorb and give out heat to the environment. Areas where heat is not affected at all can be cut off. This analysis helps in material management.

Reference

[1]. Tejpratab Singh, Sanjeev Shrivastava and Harbans Singh Ber (2013). Analysis of transient heat conduction through fins wth an enhancement of quasi-theory. [Online]. Available from: www.confabjournals.com

[2]. Sidramappa Alur (2012). Experimental studies on plate fin heat exchangers. [online]. Available from: ethesis.nitrkl.ac.in/4426/

[3]. Maiti, D.K.,and Sarangi.S.K. (2002). Heat Transfer and Flow Friction Characteristics of Plate Fin Heat Exchanger Surfaces- A Numerical Study PhD Dissertation, Indian Institute of Technology, Kharagpur

[4]. Cole K.D., Tarawneh C., and Wilson B. (2009). Analysis of flux-base fins for estimation of heat transfer coefficient. [Online]. Available from: www.elsevier.com/locate/ijhmt.

[5]. Tseng J. K., Shih C. L. and Wu W. J. (1993). Analysis of transient heat transfer in straight fins of various shapes with its base subjected to a decayed exponential function of time in heat flux, *Comput. Struct.* 47 (2) (1993), pp. 289–297

[6]. Saha A. K. and Acharya S.(2003). Parametric study of unsteady flow and heat transfer in a pin-fin heat exchanger, *Int. J. Heat Mass Transf.* 46 (20) (2003), pp. 3815–3830

[7]. Hsu T. H. and Chen C. K. (1991), Transient analysis of combined forced and free-convection conduction along a veritcal cicular fin in micropolar fluids, *Numer. Heat Transf. A* 19 (2) (1991), pp. 177–185.

[8]. Benmadda M. and Lacroix M. (1996). Transient natural convection from a finned surface for thermal storage in an enclosure, *Numer. Heat Transf. A* 29 (1) (1996), pp. 103–114

[9]. Tafti D. K., Zhang L. W. and Wang G., (1999). Time-dependent calculation procedure for fully developed and developing flow and heat transfer in louvered fin geometries, *Numer. Heat Transf. A* 35 (3) (1999), pp. 225–249

[10]. Saha A. K. and Acharya S. (2004). Unsteady simulation of turbulent flow and heat transfer in a channel with periodic array of cubic pin-fins, *Numer. Heat Transf. A* 46 (8) (2004), pp. 731–763.

[11]. Tutar M. and Akkoca A. (2004). Numerical analysis of fluid flow and heat transfer characteristics in three-dimensional plate finand-tube heat exchangers, *Numer. Heat Transf. A* 46 (3) (2004), pp. 301–321.