Analysis of Constructal Two Phase Micro-Channel Heat Exchanger with Water and R 134a

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***ABSTRACT* - Investigative study on constructal two phase model has been performed with a given stem diameter. Three different fluids, i.e. water and R 134a, are used as working fluid in the model. Calculations to estimate the maximum pressure drop, heat transfer coefficient, wall temperature and pumping power have been carried out. Acceleration, friction and junction merging losses have been taken into account to find out the maximum pressure drop. A comparative study has been made based on the thermo-fluid performances of the various working fluids. The pressure drop characteristics and the wall temperature difference curves are plotted representing the advantages and disadvantages of using refrigerants over water in the constructal model. The comparative study concludes that the constructal tree shaped heat exchanger with R 134a as a working fluid can be used as a highly efficient and environment friendly electronic cooling system.**

Keywords: Constructal theory; Heat exchanger; Microchannel; One-dimensional; Separated flow; Electronic cooling

I. INTRODUCTION

There has been a fast development in the field of computing, which leads to the need of high data storage and handling. The data in computers is handled in the form of electrical signals and stored using lasers or optical storage devices. In all these processes, tremendous amount of heat is generated which needs to be dissipated. Electronic cooling is thus a major requirement of the electronics and IT industries. The higher is the computer‟s processing capability; more is its heat production rate. The heat if not dissipated may cause the integrated circuit chips and components to melt. Thus there is a need of better and efficient heat removal method to permit faster computing technologies. Thus, it can be inferred that the present advancements in the computing methods are limited by their ability to dissipate out the heat generated. At present, conventional heat exchangers are used for electronic cooling. A hot fluid is used to absorb the heat produced from the micro-processors and transfered it to a cold fluid (or coolant) from where the heat is force-convected out using a blower fan. These methods have limitations on, the area coverage and the amount of heat transferred or removed.

The principle of constructal theory is a summary of common observations, that if a flow system is endowed with sufficient freedom to change its configuration, then the system exhibits configurations that provide progressively better access routes for the currents that flow [2]. This principle was formulated in 1996 as the constructal law of the generation of flow configuration. Bejan [2] stated the constructal law as “For a flow system to persist in time to survive it must evolve in such a way that it provides easier and easier access to the currents that flow through it.” The optimal structure is constructed by optimizing volume shape at every length scale, in a hierarchical sequence that begins with the smallest building block, and proceeds towards larger building blocks (which are called “constructs”) [3].

Bejan [4] explained the constructal route to the conceptual design of a two-stream heat exchanger with maximal heat transfer rate per unit volume and gave the advantages of the tree-like (vascularized) heat exchanger structure over the use of parallel small-scale channels with fully developed laminar flow. Chen and Cheng [6] proposed a fractal tree-like micro-channel net heat sink for the cooling of electronic chips. The micro-channel net was designed to have a top and a bottom circulation pattern in a wafer. The study showed that this type of heat sink had better heat transfer characteristics and required less pumping power than traditional parallel nets. Zamfirescu and Bejan [25] investigated constructal tree-shaped two-phase flow for cooling a surface. They studied the optimal structure of the phase change with convective heat transfer. Bonjour J., Rocha L.A.O., Bejan A. [5] investigated the heat exchange process with counter flows in two coaxial pipes, and optimized the fin set between the two pipe walls.

It can be inferred that there is an advent need for electronic cooling techniques. The conventional heat exchangers mainly suffer in design consideration, lesser heat transfer from circuit to fluid and higher degree of losses due to friction. With the increasing miniaturization of electronic chips and increasingly larger heat dissipation rates, better designs of cooling system are needed. Since natural systems often give the best solution for many problems, the structure of the living organism may provide the inspiration for the design of an effective micro-cooling system for electronic chips. Two-phase micro-channels, having hydraulic diameters ranging from 10 to 1000 μm, with forced convection boiling of a liquid through circular or noncircular tubes, results in extremely high heat transfer rates.

Comparison of a constructal heat exchanger and normal heat exchanger is analyzed by Manjunath and Kaushik [15] using second law analysis. Analysis is carried out by considering the three irreversibilities due to heat transfer, pressure drop and production of the materials and the construction of the heat exchanger. Based on constructal theory, entropy generation minimization and second law efficiency equations are formulated by Manjunath and Kaushik [16] for tree-shaped counter flow imbalanced heat exchanger for fully developed laminar and turbulent fluid flow. Entropy generation number, rational efficiency and effectiveness behavior with respect to changes in number of pairing levels and different tube length-to-diameter ratios of constructal heat exchanger are analyzed analytically.

The hydrodynamic performance of the network, composed of a series of rough ducts, for both laminar and turbulent flow regimes was studied by Miguel [17]. Transient response of internal fluid pressure is also modeled and analyzed. Kim Y., Lorente S., Bejan A [12] showed numerically how the geometric configuration of the tubular flow structure controls the global performance of a cross-flow heat exchanger.

Hajmohammadi M.R., introduced new patterns for the highly conductive pathways, called „fork-shaped‟ configurations in which two types of fork-shaped pathways, „F1‟ and „F2‟ were considered. Numerical results demonstrated that, these two configurations of highly conductive pathways, the „peak‟ temperature could be reduced by 41% and 46% by using fork-shaped pathways of „F1‟ type, and „F2‟ type, respectively, when compared with the base X-shaped inserts. HajmohammadiM.R, studied the influence that the size of adiabatic spacing in an array of heat sources exerts on the heat transfer performance. Two thermal boundary conditions for the heat sources are employed: constant heat flux condition or constant temperature. It was shown that in the first case, the „hot spot‟ temperature was reduced by 11% for two optimized-size heat sources with uniform heat flux articulated with optimum adiabatic spacing. Additionally, for two heat sources with uniform temperature, a level of maximum heat transfer enhancement of 7.7% was achieved.

Flow boiling in a constructal tree-shaped minichannel network was numerically investigated using a one-dimensional model, taking into consideration the minor losses at junctions by Zhang C., Chen Y., Wu R., Shi M. [26]. Hajmohammadi M.R. et al. proposed a simple type of cavities, consisting of N branches of equal cavities intruding into the symmetrical and equal divisions of a rectangular/trapezoidal heat generating body. Optimization process was done to seek for the optimal „„„N-branch‟‟‟ cavities, which minimize the peak temperature of the body with respect to the constraint of fixed volume occupied by all cavities. They showed that, by utilizing the so-called cavities, which are not of a complex type at all, the heat generating body operates at the lower level of the peak temperature when com- pared with those cavities of complex type and recommend that for practical engineering applications, the so-called complex cavities must be replaced by cavities of simpler type, such as the N-branch shaped cavity, which is studied in the present work.

The new approach of constructal theory has been employed to design shell and tube heat exchangers by Azad and Amidpour [1]. The results of design using constructal theory are heat exchangers with in-series sections which are called constructal shell and tube heat exchangers. Constructal entropy dissipation rate minimization of a round tube heat exchanger cross-section was studied by Wei S., Chen L., Sun F. [23]. Kim Y., Lorente S., Bejan A. [12] developed analytically the constructal design of steam generators with a large number of tubes. The main features of a steam generator are determined based on the method of constructal design. Lee J. worked on the minimization of global pressure drop in a comb-like channel network with self-healing and cooling functionalities implementing the concept of constructal design.

Adewumi O.O. carried out a numerical study on three dimensional forced convective heat transfer through a microchannel heat sink with micro heat sinks for both fixed and relaxed axial lengths. The result showed that asBejan number increases, the minimized peak temperature decreases.

This work intends to compare the thermo-fluid performance of various working fluids by using two phase constructal model. The pressure drop, heat flux, number of transfer units (NTU), and thermal resistance are compared for the different fluids. The results will be useful for design of electronic heat exchangers.

II. GEOMETRY

The flow architecture for the given model is as given in Fig. 1. This structure was developed on the constructal methodology [2.4]. It has many advantages, such as, it is well constructed, the flow path length is the same for all the source-sink paths, and the source-sink pressure drops and flow rates are the same for all the paths, given that the same working fluid is being used. The constructal tree of Fig. 1 distributes a stream (e.g., $\dot{m_n} = \dot{m_2}$) uniformly over a square shaped surface.

A balanced counter-flow heat r has two identical constructs, one for the hot fluid and the other for the cold fluid which are mated perfectly, so that tube of the hot tree is parallel to and in excellent

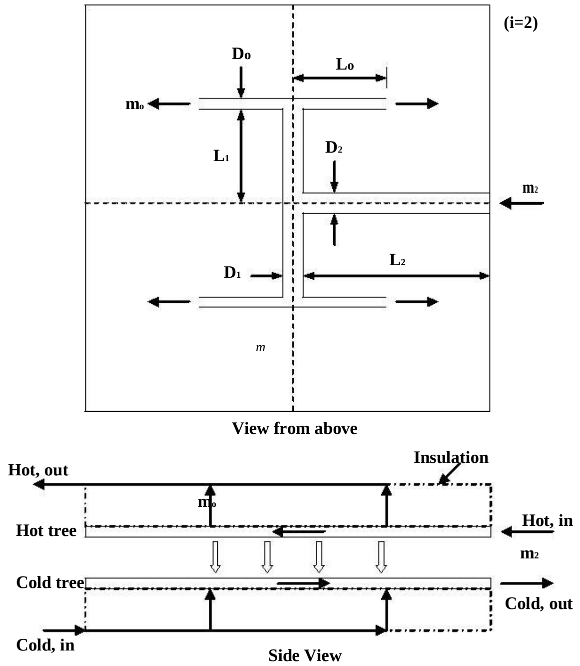

FIG. 1 Counter flow of tree shaped streams distributed over a square area

thermal contact with the corresponding tube of the cold tree. In the second part of Fig. 1, the hot construct distributes the single stream (m_n) over the square area occupied by the structure and the cold construct collects a large number of mini streams (m_0) and forms a single cold stream (m_n).

The cold stream is collected in a common chamber and is then send to the condensing unit to be condensed back to liquid state before circulating it back to the evaporator. The mixing of the hot mini streams inside the manifold is not a source of irreversibility because the m_0 streams arrive at the manifold not only at the same pressure but also at the same temperature. Similarly, on the back side of the cold tree, the cold stream m_0 is distributed by a manifold as a large number of mini streams m_0, which feed the canopy of the cold tree. The cold manifold is insulated with respect to the cold tree flow structure. The arrows that indicate the flow of fluid (not heat) in Fig. 1 could be reversed, but the arrangement itself (the drawing) and its thermo-fluid performance do not change.

One tree is made of many tubes of (n+1) sizes. One tube has the length L_i and internal diameter D_i, where i =0, 1... n. The number of tubes of type i is n. The construction of the entire flow architecture starts with the smallest tube scale (L_0, D_0), which inhabits the smallest square area element (called elemental system). Larger constructs are made by pairing smaller constructs. For example, the first constructs (i=1) consist of joining two elemental systems, and joining the two elemental streams into a first-construct stem of size (L_1, D_1). In the second construct (i=2), the stem (L_2, D_2) is twice as long as L_1, or L_0. Tube lengths double after two consecutive construction steps. When there are many construction levels, we may express the length-doubling rule by writing approximately [3]:

$$L_{i+1} = 2^{1/2} L_i \quad (\mathrm{i} = 0,1,\ldots,\mathrm{n}) \quad (1)$$

Pairing at every construction level means that the tube numbers and flow rates are ordered as:

$$n_i = 2^{n-i} \quad (\mathrm{i} = 0,1,\ldots,\mathrm{n}) \quad (2.a)$$

$$m_0 = 2^i m_0 \quad (\mathrm{i} = 0,1,\ldots,\mathrm{n}) \quad (2.b)$$

There are n construction stages, and in the end a single stream of flow rate m_n flows into or out of the largest (nth) construct. The tree structure bathed by m_n has (n +1) length scales, which are organized hierarchically.

The tree-shaped network configuration provides an optimized access to connect area-point flow. However, the free circulation of the fluid is difficult to be realized by a single layer tree-shaped network. To solve this problem, the constructal tree-shaped channel network is configured to have two layers. The lower layer channel network is of the same distribution as the upper one except that the inlet on the upper layer and the outlet on the lower one layout towards the opposite direction. The highest branching level channels (i = 6) of the upper layer communicate with the highest branching level channels of the lower one. The cross section of each branching level channel is circular, the diameter of the 0th channel D_0 = 4 mm, and the length of the 0th channel L_0 = 50 m. A total number of branching levels of six is assumed in the construct of tree-shaped network due to the superior flow heat transfer performance in heat and fluid flow [22]. And there is no daughter branches touch mother stems in the finite iteration.

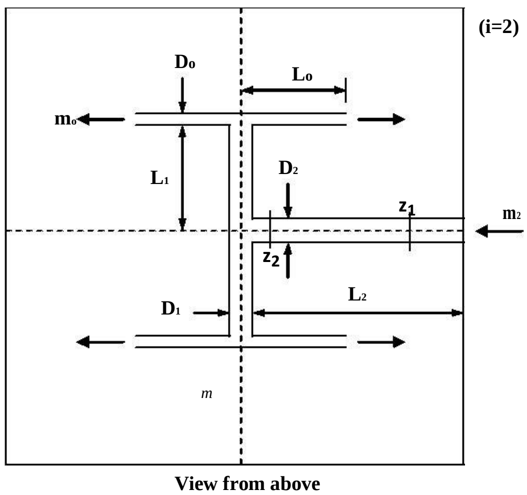

FIG. 2 Counter flow of tree shaped streams representing the positions z_1 and z_2

III. MATHEMETICAL MODELLING AND FORMULATION

The flow boiling is considered to be a one dimensional (1-D) model with water and refrigerant R134a as working fluid. The constructal model is utilized to model the cooling system in the current investigation.

CONSTRUCTAL TWO PHASE MODEL:

The following assumptions are considered for flow boiling model [22]:

1. Saturated liquid enters the channel at inlet. It undergoes a phase change in the constructal network such that the fluid at the outlet is in the state of saturated vapour.
2. The pressure of outlet fluid is considered to be atmospheric pressure.
3. The mass flow rate is uniformly distributed in each junction.
4. The heat flux is assumed to be constant.

The above given assumptions of uniform heat flux throughout the construct and of flow distribution at each junction may not be practically possible and may lead to some losses. However, the model being a theoretical one designed only to analyze the heat transfer and pressure drop characteristics, the above given assumptions was considerable.

The pressure drop of flow boiling occurred in the constructal network includes the acceleration pressure drop, frictional pressure drop and the junction pressure drop.

$$\Delta P = \Delta P_{acc} + \Delta P_{fric} + \Delta P_{junction} \quad (3)$$

Where, ΔP is the total flow pressure drop, ΔP_{acc} is the acceleration pressure drop, ΔP_{fric} is the frictional pressure drop and $\Delta P_{junction}$ is the junction pressure drop. The acceleration and friction pressure drops are calculated based on the two phase methodology and parameters like void fraction and Martinelli parameter are considered.

The void fraction can be estimated by Zivi correlation:

$$\alpha = \frac{1}{1 + \left[\left(\frac{1-x}{x}\right)\left(\frac{\upsilon_l}{\upsilon_v}\right)^{\frac{2}{3}}\right]} \quad (4)$$

Where, α is the void fraction, x is vaporquality, ν_l and ν_v are specific volume of liquid and vapor respectively.

The density can be calculated by a derived format of Zivi's correlation method:

$$y = AG^3\left[\frac{x^3}{(\alpha\rho_v)^2} + \frac{(1-x)^3}{((1-\alpha)\rho_l)^2}\right] \quad (5)$$

$$\rho = \sqrt{\frac{mG^2}{y}} \quad (6)$$

Where, G is the mass flux, and M the mass flow rate.

The steam quality can be calculated by the energy balance equation and between two adjacent points (point1 and point2) along the flow direction is given by [5, 8]:

$$qA_w = mr(x_2 - x_1) \quad (7)$$

Where, q is the heat flux on the channel wall, Aw is lateral surface area between points z_1 and z_2, m is the mass flow rate through point 1 and point 2, r is latent heat of vaporization, and x_1, x_2 is the vapour quality or dryness fraction at z_1, z_2, respectively as shown in figure.

Thus, the acceleration pressure drop between point 1 and point 2 along the flow direction can be estimated by:

$$\Delta P_{acc\ 1-2} = G^2\left(\frac{1}{\rho(z_2)} - \frac{1}{\rho(z_1)}\right) \quad (8)$$

The frictional pressure drop gradient of the two-phase flow during flow boiling can be expressed as:

$$\frac{dP_{fric}}{dz} = \phi^2{}_{lo}\frac{dP_{flo}}{dz} = \phi^2{}_{lo}\lambda_{lo}\frac{G^2{}_l\nu_l}{2d} \quad (9)$$

$$G_l = (1 - x)G \quad (10)$$

Where, Gl is mass flux of pure fluid flow, dPflo/dz is the frictional pressure drop when liquid phase is assumed to flow alone in the channel, λlo is the liquid only friction factor, d is the channel diameter, ϕ2lo is the two-phase multiplier. Lockhart and Martinelli proposed the two phase multiplier $\phi^2{}_{lo}$ as a function of Lockhart–Martinelli parameter X^2[9]:

$$\phi^2{}_{lo} = 1 + \frac{C_{LM}}{X} + \frac{1}{X^2} \quad (11)$$

Where, the values of the Lockhart–Martinelli parameter X2 and the phase interaction parameter CLM depend upon whether each phase is flowing laminar or turbulently. A turbulent – turbulent model is considered.

When the fluid is R 134a then the friction parameter is considered depended on the equivalent Reynolds number [29]. It can be calculated as:

$$Re_{equi} = \frac{Gd}{\mu_l} * \left[(1 - x) + x * \left(\frac{\rho_l}{\rho_g}\right)^{0.5}\right] \qquad (12)$$

The friction factor for two phase flow is thus determined by the correlation:

$$f_{tp} = 0.11 * Re_{equi}{}^{-0.1} \qquad (13)$$

The pressure drop gradient of the two-phase flow for R 134a can be expressed as:

$$\frac{dP_{fric}}{dz}_{R\,134a} = f_{tp}\frac{G_l{}^2\nu_l}{2d} \qquad (14)$$

Thus, the frictional pressure drop between two adjacent points 1 and 2 along the flow direction can be estimated by the integration of dP_f:

$$\Delta P_{fric\,1-2} = \int_{z1}^{z2}\frac{dP_{fric}}{dz}dz \quad (15)$$

The junction pressure drop can be calculated in terms of junction loses as:

$$\Delta P_{junction} = \frac{f_T\upsilon_l G^2}{2}\left\{1 + x\left(\frac{\upsilon_v}{\upsilon_l} - 1\right)\right\} \quad (16)$$

Where, fT is the junction loss factor of fluid flow. In the constructal structure, there are splitting and merging flows at junctions.

For merging flow, used in model, from (k + 1)th to kth channel, the junction loss factor is:

$$f_t = 1 + (\frac{d_k}{d_{k+1}})^2 + 3(\frac{d_k}{d_{k+1}})^2\left[\left(\frac{Q_{k+1}}{Q_k}\right)^2 - \left(\frac{Q_{k+1}}{Q_k}\right)\right] \quad (17)$$

Where, Q_k and Q_{k+1} are volumetric flow rates along the kth and (k+1)th channel.

The power can be estimated by:

$$P_p = m\upsilon_l\Delta P \quad (18)$$

Along with, the pressure drop, the wall temperature distribution along the flow direction is also needed to be estimated to analyze heat transfer characteristics of the two - phase in constructal network. The channel wall temperature t_w is estimated by:

$$q = h(t_w - t_b) \quad (19)$$

Where, t_b is the bulk fluid temperature, which is determined by the vapor pressure of fluid at that saturation temperature and h is the convective heat transfer coefficient for water can be calculated by Yu et al. correlation:

$$h = 6400000(Bo^2 We_l)^{0.27}(\frac{\upsilon_v}{\upsilon_l})^{-0.2} \quad (20)$$

Where, the Boiling number (Bo) and Weber number (Wel). They are given as follows:

$$Bo = \frac{q}{G * r} \quad (21)$$

$$We = \frac{G^2 d\upsilon_l}{\sigma} \quad (22)$$

Where, σ is the fluid surface tension.

For R 134a the convective heat transfer coefficient can be calculated by [27]:

$$h_{R\,134a} = 30 * Re^{0.857} Bo^{0.714} \frac{k}{d} \quad (23)$$

The number of heat transfer units (NTU) can be calculated by [28]:

$$NTU = \frac{UA_w}{c_p} \quad (24)$$

The acceleration and friction pressure drops of a control volume are solved using trapezoidal integration method, and then combined with junction losses if present to determine the local pressure drop at a position z in the direction of flow. It should be noted that, in the numerical calculation, the mass flow rate is determined by the heat flux for a given tree-shaped network under the assumption

4; the bulk fluid temperature is estimated as the saturation temperature corresponding to the local static pressure. The latent heat and surface tension are considered to be constant.

IV. RESULTS AND DISCUSSION

The above given modeling and formulations were based on the constructal two phase separated flow model. Comparisons were made between two different operating fluids that is water and R134a in the constructal two phase model. The main criteria used in comparison are pressure loss of the piping system, pumping power required and the heat transfer characteristics, including the heat transfer units, heat transfer rate and wall temperature difference, between inlet and outlet. The data developed is plotted for comparison based on change in mass flow rate and geometric parameters. Software like MatLab 2013Ra and EES are used to develop the mathematical model. The results thus obtained are classified in following cases.

A. Case 1 (Water):

Water is used as operating fluid in the constructal two phase model. The working fluid (water in this case) is allowed to change its phase from saturated liquid at inlet to saturated vapour at outlet. The model is developed based on Martinelli's friction correlation and Yu's Heat convection correlations. This model compares the thermo-fluid properties based on variation in stem diameter and mass flow rate.

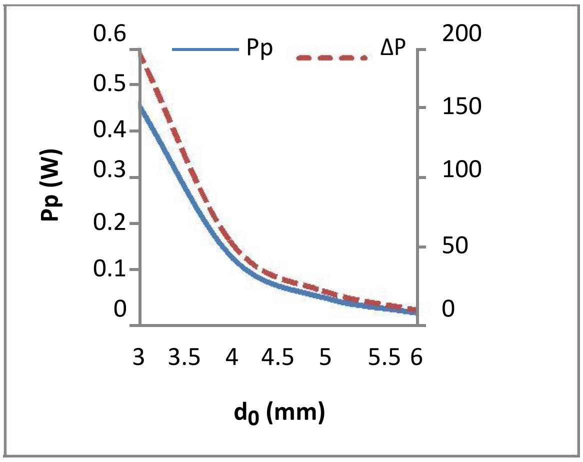

FIG. 3 Pressure drop and Pumping power versus Initial diameter for water

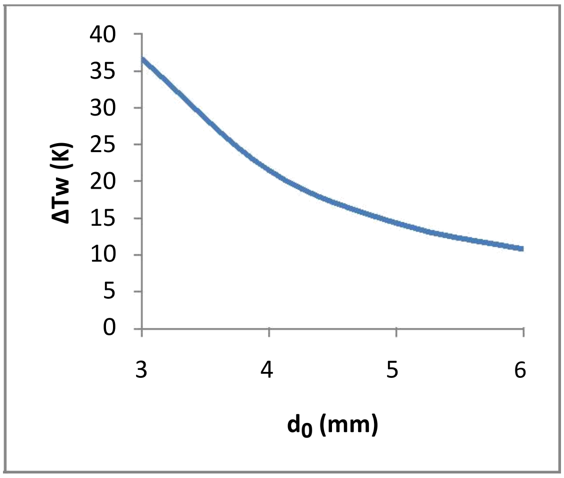

FIG. 4 Wall Temperature difference versus Initial diameter for water

Figure [3] represents the variation of total pressure drop in the given constructal channel under various initial diameter conditions. The pressure drop characteristic determines the pumping power required for operating the system as per Equation (16). It can be clearly deduced that with the increase in diameter the pressure drop gets reduced dramatically. Thus, with increase in diameter of the constructal model, less pumping power is needed for the same mass flow rate.

Figure [4] represents the variation of Wall Temperature difference in the given constructal channel under various initial diameter conditions. The ΔT_W graph signifies the temperature uniformity and heat transfer characteristic. It represents the difference of wall temperature from inlet to outlet. As the diameter increases the difference is getting reduced showing more temperature uniformity between inlet and outlet. It is also indirectly related to the pressure drop which is used to determine the saturation temperature used in Equation (14).

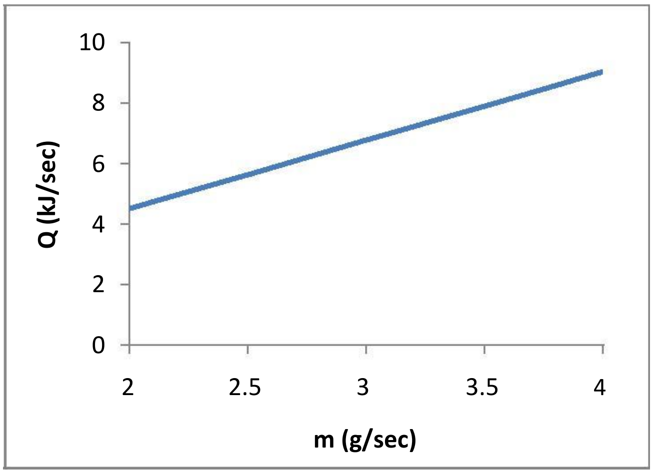

FIG. 5 Heat transfer rate versus mass flow rate for water

The heat transfer rate increases with the increase in the mass flow rate of water in the construct. The figure [5] depicts a linear increment in Q with m which concludes that with the increament in the mass flow rate more and more heat can be dissipated thus increasing the heat handling capacity of the heat exchanger.

B. CASE 2 (R134a):

In this case, refrigerant R134a is used as operating fluid in the constructal two phase model. The working is allowed to change its phase from saturated liquid at inlet to saturated vapor at outlet.

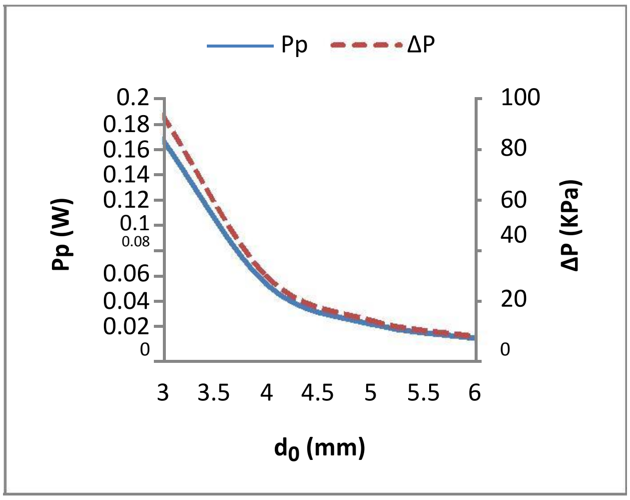

FIG. 6 Pressure drop and Pumping power versus Initial diameter for R134a

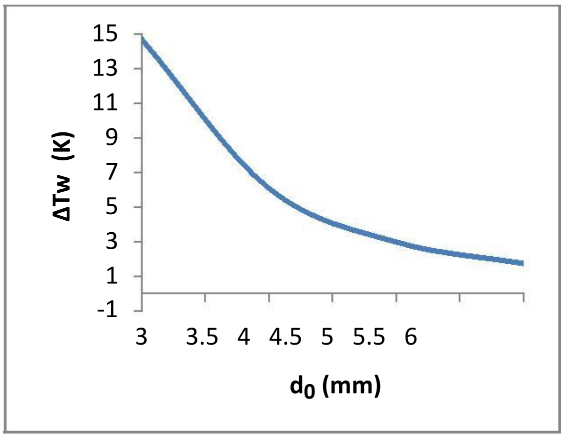

FIG. 7 Wall Temperature difference versus Initial diameter for R134a

Figure [6] represents the variation of net pressure drop in the given constructal channel under various initial diameter conditions. It can be visualized that with the increase in diameter the pressure drop gets reduced rapidly, thus requiring lesser pumping power need.

Figure [7] represents the variation of Wall Temperature difference for R134a under various initial diameter conditions. As the diameter increases the difference in Wall Temperature is getting reduced showing more temperature uniformity between inlet and outlet.

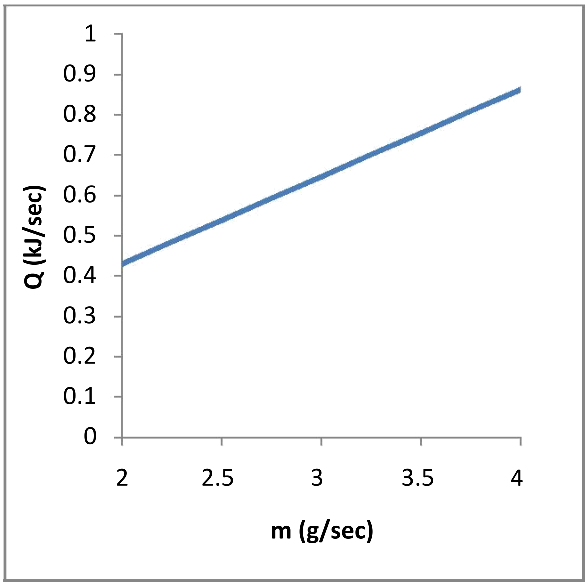

FIG. 8 Heat transfer rate versus mass flow rate for R 134a

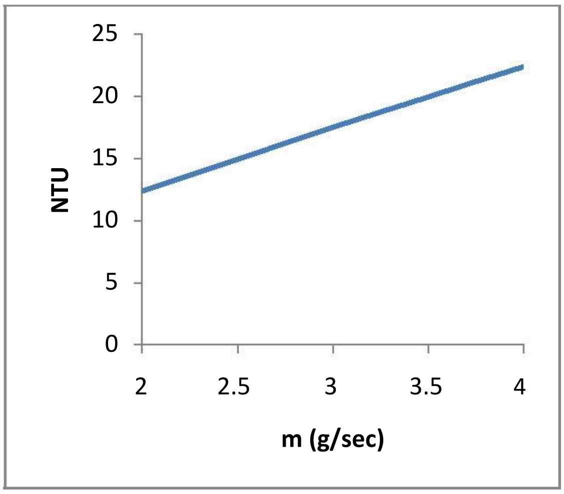

FIG. 9 Heat transfer rate versus mass flow rate for R 134a

The figure [8] depicts a linear increment in Q with m which depicts that with the increment in the mass flow rate more and more heat can be dissipated thus increasing the heat dissipating capacity of the heat exchanger. The nature of curve is similar to that of figure [5].

The figure [9] represents the change in NTU with the mass flow rate of R 134a. It shows that by increasing the flow rate of fluid in the construct the NTU increases. By the ϵ NTU method, the effectiveness of the heat exchanger increases with increment in NTU. Thus by increasing the flow rate the effectiveness also gets increased.

Figure [10] represents the lowering of the entropy towards minima which is the needed condition for optimization. So, at d_0 equal to 4.2 mm at 0.04 kg/sec flow rate the performance of the heat exchanger is the best due to minimum Ns.

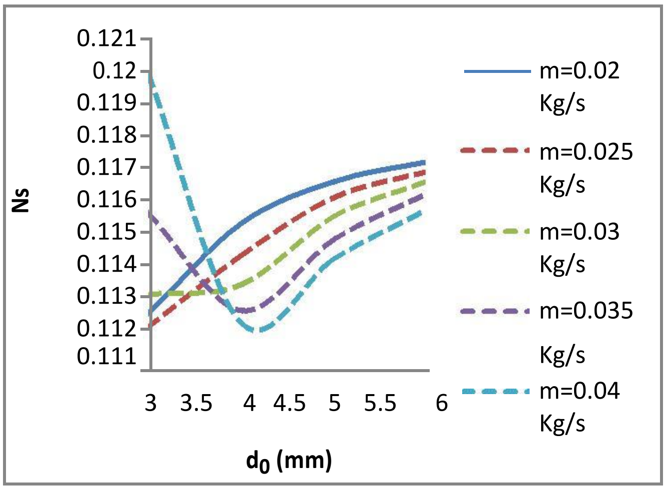

FIG. 10 Entropy Generation Number versus Initial diameter for R 134a

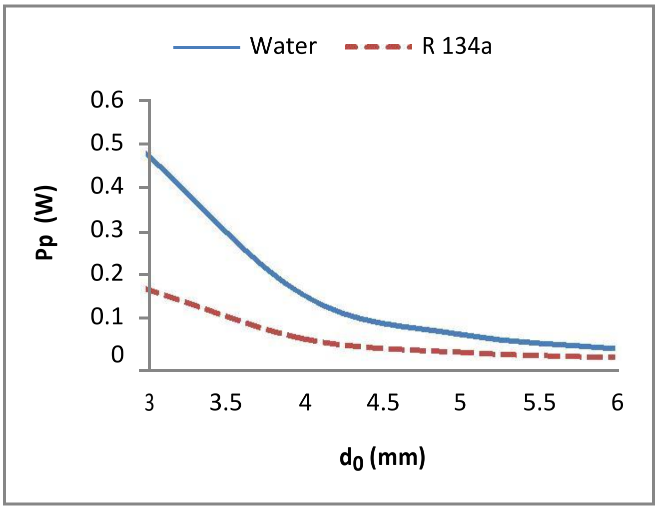

FIG. 11 Pumping power versus Initial diameter for water and R134a

C. Comparison between Water and R 134A

When the three fluids are compared on the same scaled graph, then their thermo-fluid properties can be compared and the best fluid characteristics can be determined thus forth.

Figure [11] represents a comparative graph showing the variation of pumping power with change in stem diameter in the given model using water and R134 refrigerant. It represents that water needs very high pumping power as compared to the refrigerant. Thus, using refrigerant over water as working fluid allows much better fluid flowing characteristics and requiring much less pumping power.

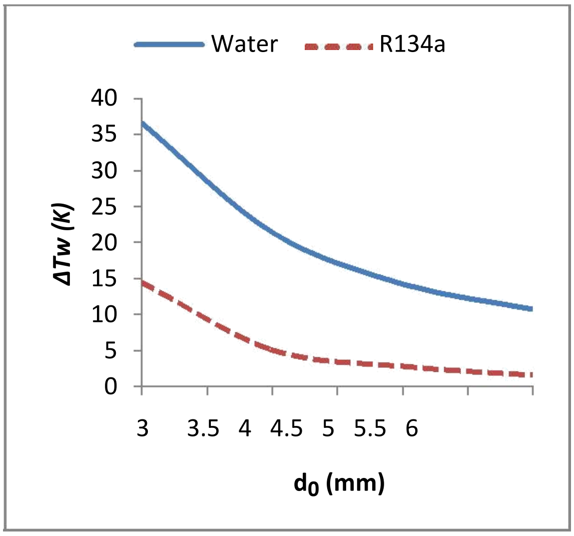

FIG. 12 Wall Temperature difference versus Initial diameter for water and R134a

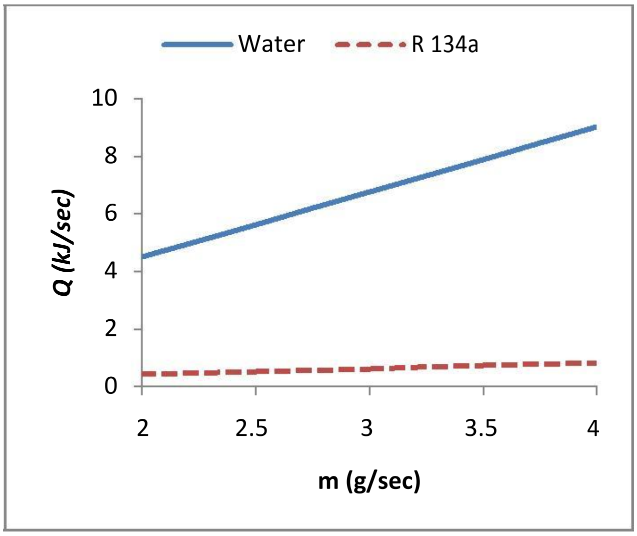

FIG. 13 Heat transfer rate versus mass flow rate for R 134a

Figure [12] represents a comparative graph showing the variation of Wall Temperature difference in the given constructal channel under various initial diameter conditions using water and R 134a refrigerant. As the diameter increases the difference is getting reduced showing more temperature uniformity between inlet and outlet. It can be seen that water shows higher wall temperature difference characteristics than R 134a refrigerant. Thus water can absorb more heat that the other refrigerants. When the comparison is based in actual wall temperature then, the refrigerant works under much lower temperature than water, which is much below 0^{O}C. Furthermore, Figure [13] represents the variation of Heat Transfer Rate in the given constructal channel with change in mass flow rate of water and R 134a refrigerant. The figure shows that water can absorb more heat than R 134a. Thus, R 134a can be preferred where amount of heat to be dissipated is less but there is more need of a lower temperature cooling systems.

V. CONCLUSION

It is already understood that there is a need for effective and efficient electronic cooling technique, due to the rapid advancements in computing technology. Thus the paper focuses on the objective of applying the constructal approach to micro-channel heat exchanger.

The constructal two phase model has been considered in this paper. In this paper, the heat exchanger is modeled for a mass flow rate (m_0=0.0025 kg/s), initial length (L_0=0.05m), and initial diameter (D_0=0.004m). The thermo-fluid performances have been studied by changing the working fluid in the given model under various geometrical parameters and mass flow rates. A comparative study has been made between the various working fluids that are water and R 134a in constructal two phase model. The following conclusions have been drawn from the comparative study:

1. The fluid performance of a constructal model with water as working fluid shows much poorer characteristics than with the refrigerant. The study showed that water had much higher pressure drop nature than the refrigerant when flowing through the same model. Thus, using water as working fluid would require higher pumping power than R 134a.
2. The thermal performance of water in the constructal two phase model is better than that of the refrigerant. Water shows higher wall temperature difference than the other refrigerants. Moreover, water shows higher heat transfer rate than R 134a refrigerant. This proves that water can absorb more amount of heat produced than refrigerants. But R 134a refrigerant can maintain the working temperature much below 0^{O}C. This allows very low temperature cooling characteristics than water.

Thus, the comparative study between the various working fluids in constructal two phase model concludes that the heat exchanger designed with refrigerant as working fluid shows better performance than water in aspect of lower pumping power requirement, considering that there is a need of low wall temperature but with lower heat absorption rate. Thus, the constructal shaped heat exchanger with R 134a as a working fluid can be used as a highly efficient and effective cooling system.

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Table 1 Nomenclature of variables and symbols

Nomenclature		t	temperature
A_w	wall surface area	We	Weber number
Bo	boiling number	x	steam quality
C_{LM}	two-phase interaction parameter	Greek symbols	
c_p	specific heat of fluid	α	void fraction
d_i	branch diameter of the ith level	λ	friction factor
f_T	junction loss factor	ρ	density
G	mass flux	σ	surface tension
h	convective heat transfer coefficient	ϕ^2_{lo}	two-phase multiplier
k_{lo}	pressure loss coefficient of only liquid flow through a tube bend	v	specific volume
L_i	branch length of the kth level	X^2	Lockhart–Martinelli parameter
$\dot{m}$	mass flow rate	Subscripts	
NTU	number of transfer units	i	branching position
P	pressure	l	liquid
P_p	pumping power requirement	lo	liquid only
Q	volumetric flow rate	junction	local junction position
q	heat flux	v	vapor

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