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HEAT TRANSFER AND FLUID FLOW IN A CONSTRUCTAL HEAT EXCHANGER

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ABSTRACT

Numerical simulation is carried out for heat transfer and fluid flow for constructal heat exchanger for water and air as operating fluids. The Computational Fluid Dynamics software “FLUENT” has been used to investigate the fluid flow and heat transfer in the constructal heat exchanger. For operations with air, straight fins are used to enhance thermal conductance. Even though the flow is laminar, the heat transfer coefficient is high due to two reasons: first, flow is thermally developing and second, increased compactness due to small hydraulic diameter. These results were compared with heat exchangers with fully developed ideal plug flow without any flow maldistribution. It was found that the effectiveness, for the flow rates considered, was 10% greater than that of ideal cross flow heat exchanger. This result is very much encouraging due to the fact that in presence of three dimensionality of flow and maldistribution, the effectiveness is expected to be lower than the ideal cross flow heat exchanger. The effectiveness for the second construct which will be an assembly of first constructs is expected to be still higher.

INTRODUCTION

Laminar forced convection in the entrance region of rectangular duct is of significant importance in the application of compact heat exchangers, as discussed in Fraas (1989). The analysis of heat transfer in rectangular duct is more complex than that of circular duct. For example, determination of the local Nusselt number in rectangular duct requires three-dimensional analysis in contrast to the usual two-dimensional axi-symmetric analysis for developing circular pipe flow.

One of the most interesting boundary conditions, defined in Hartnett, J.P and Costic, M (1989), is the well-known T condition, characterized by equal and uniform wall temperature. Wibulswas, P (1966) solved the thermal entrance length problem for the T boundary condition for negligible fluid axial conduction, viscous dissipation, and thermal energy sources. Chandraputla, A.R and Sastri, V. M. K (1977) analysed the T thermal entrance length problem for square ducts. Aparecido, J.B., and Cotta, R.M (1990) solved the problem of thermally developing flow in rectangular ducts. Their study provides a set of results for local Nusselt numbers in the entrance region, which can be used to validate the numerical solutions.

The constructal theory was first stated in 1996, in the context of optimizing the access to flow between a point and an area, with application to traffic, Bejan. A, (1996) and the cooling of electronics ,Bejan. A, (1997). The flow path was constructed in an “atomistic” sequence of steps that started with the smallest building block (elemental area) and continued in time with larger building blocks (assemblies or constructs). The mode of transport with the highest resistivity (slow flow, diffusion, walking, and high cost) was placed at the smallest scales filling completely the smallest elements. The constructal method of optimizing flow architectures has been applied in several areas: the method and its published applications are reviewed in Bejan. A, (1997)

The subject of this paper is the constructal architecture that gives a heat exchanger with the ability to pack maximum heat transfer in a fixed volume. It reveals the modular, hierarchical architecture with the most dense heat transfer packing while keeping the pressure drop to its minimum.

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This is the main inspiration for the present work. In the present study, we present the numerical study of the entrance region problem of laminar heat transfer in the single construct module of a constructal heat exchanger. The momentum and the energy equations are solved using FLUENT 6.1 software and parameters indicating heat exchanger performance are extracted from the post processing of the numerical data.

The concept of constructal heat exchanger was first proposed by Bejan[9]. The central idea of this type of heat exchanger is using laminar flow but keeping the flow length in each module smaller than the entry length and thus ensuring a higher heat transfer rate with a lower pressure drop. However Bejan's paper (2002) was limited in presenting the basic concept along with a scaling analysis which needs to be investigated in terms of performance with respect to the accepted heat exchanger parameters. The present work aims at taking a step in this direction.

CONSTRUCTAL HEAT EXCHANGER

In a constructal heat exchanger as shown in fig.1 the flow structure has multiple scales. The smallest scale which is analysed here consists of parallel plate channels the length of which lies within the thermal entrance length of the small stream that flows through the channel. This feature has two advantages: first, it eliminates the longitudinal temperature increase (flow thermal resistance) that would occur in the fully developed flow, and second, it increases the heat transfer coefficient associated with fully developed laminar flow.

The elemental channels of hot fluid are placed in cross flow with the elemental channels of cold fluid. The elemental channel pairs are assembled into sequentially large flow structures (first construct), which have the purpose of installing (spreading) the elemental heat transfer as uniformly as possible throughout the heat exchanger volume. At length scales greater than the elemental, the second construct is formed by the assembly of four first constructs as shown in fig.2, where the streams of hot and cold fluid are arranged in counter flow. Each stream bathes the heat exchanger volume as two trees joined canopy to canopy. One tree spreads the stream throughout the volume (like a river delta), while the other tree collects the same stream (like a river basin). The fig.3 illustrates the formation of counterflows during the assembly of second construct.

The main objective of this work is to analyse the thermally developing fluid flow and heat transfer in the first construct and to make a comparative study with the fully developed ideal cross flow heat exchanger.

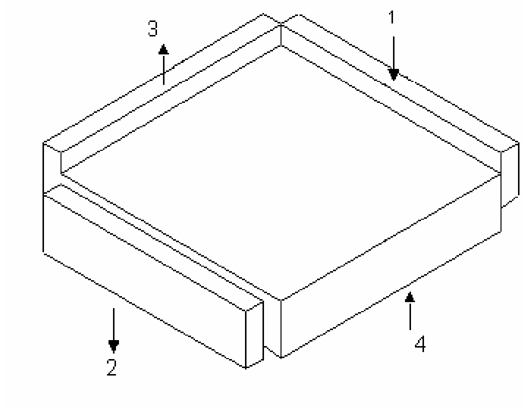


Fig.1 First construct

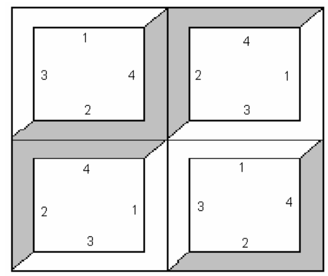


Fig.2 Second Construct (top view) Containing four first constructs.

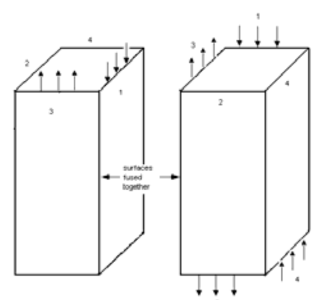


Fig.3. The formation of counter flows during the assembling of the second construct shown in fig.2.

GRID INDEPENDENCE AND VADILATION

In order to assess the effect of the number of the mesh points on the accuracy of the results, a grid independence study was conducted. The result of such study is shown in the table.1 below. As the result shows, a minimum number of 50x50x20 grid points were necessary for each channel before a satisfactory and accurate enough result could be obtained. To validate the present numerical strategy the results for a fully developed rectangular duct flow was compared to available standard results in Holman, J.P. The results in table.2 show a reasonable accuracy for the simulation.

Table.1 Grid Independence

Grid Points	Hexahedra l cells	h, W/m^2-K	Nu
40x40x15	24000	2002.4462	12.81
50x50x15	37500	2468.2522	15.79
60x60x15	54000	2462.542	15.75
40x40x20	32000	2451.49	15.68
50x50x20	50000	2439.7185	15.60
50x50x25	62500	2425.8628	15.52

Table.2 Validation for computation

Duct aspect Ratio	Nu (Laminar Fully developed)	Nu (Fluent)
1/8	6.49 (q=const)	6.54
1/6	5.137 (T=const)	5.16
1/4	4.439 (T=const)	4.47
Circular pipe	3.66 (T=const)	3. 63

NUMERICAL PROCEDURE

Modelling Method

The steady-state governing equations are solved using a segregated solver, which means that temperature and flow fields are segregated from each other and are solved separately using FLUENT 6.1 software.

All temperature-dependent properties of fluids, i.e, water and air, such as density are evaluated at local temperature by using the least-square fit equations derived from thermodynamic data compilations taken from Holman, J. P.

The intial guesses for velocity, temperature and viscosity fields were set to constant values over the entire computational domain. To obtain the flow and heat transfer solutions, the solver undertakes iteration until the convergence criterion is satisfied, which employs scaled residuals of the modified variables in the governing equations as the measure. In addition, the averaged fluid temperature was examined explicitly for convergence(to less than a 0.01% variation between iterations). Finally the energy balance was ensured by computing the differences in enthalpy between inlet and outlet for each fluid.

Typical computational time with 570,800 grid points is about 10 hours on one CPU of a Compaq system with 512 MB RAM.

RESULTS AND DISCUSSION

Convection heat transfer of thermally developing flow in a constructal heat exchanger was simulated numerically using the same FLUENT 6.1. The test section consists of seven rectangular channels stacked one over the other with inlet and outlet ports as shown in fig.1. For this geometry 570,800 grid points were required for grid independent result.

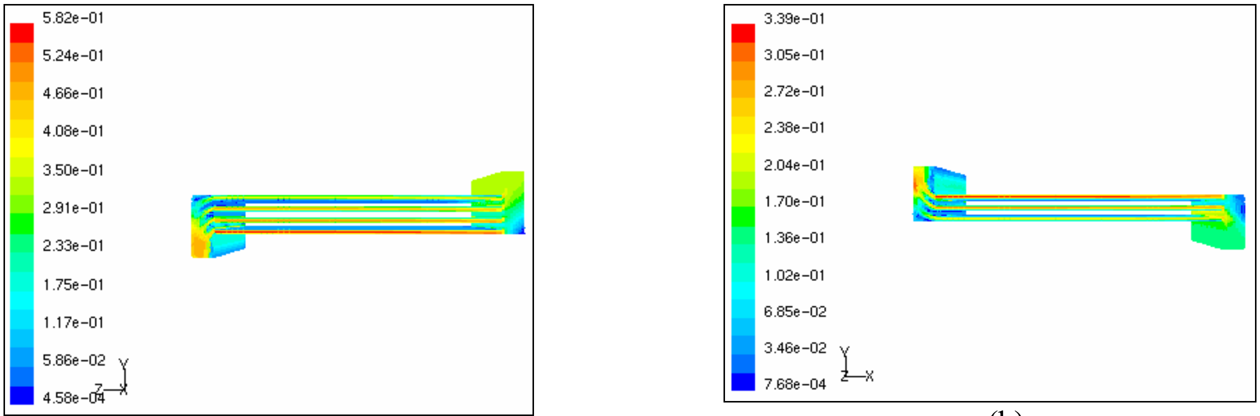
Figure. 4 illustrates the flow distribution in the constructal heat exchanger for Re_{ch} =1500 with water as the working fluid. It is observed that there is flow maldistribution in the channels. The channels closer to the exit port experience higher velocities. Figure. 5 illustrates the temperature distribution in the heat exchanger for the same case. Since the channels closer to the port exit experience higher velocities, the temperature change is lower. The channels closer to port inlet experience higher temperature change as a consequence.

Figure. 6 illustrates the flow distribution in the constructal heat exchanger for Re_{ch}=200 on hot water side and Re_{ch}=2000 on the cold air side. It is observed that flow velocities vary very little from channel to channel. This is because at lower Reynolds number channel flow resistance is higher (higher friction factor) and as a result the maldistribution is small. It is clearly observed that the channels closer to the port exit experience relatively higher velocities.

Figure. 7 illustrate the temperature distribution for the same case with channels closer to port inlet experiencing higher temperature change.

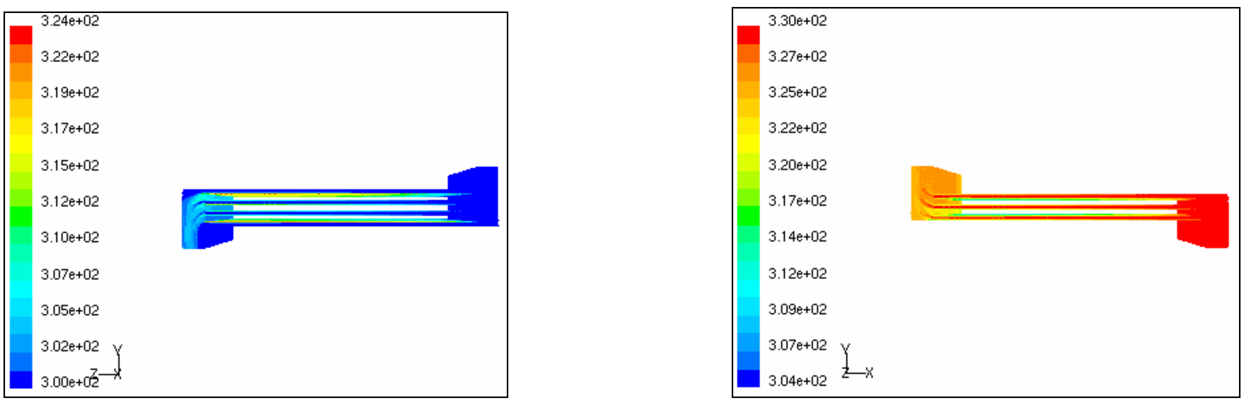
Figures. 8 (a) and (b) present the numerical results for the local heat transfer coefficient for water Re_{ch} =1500. It is observed that the flow is thermally developing. The heat transfer coefficients are found to be higher on the cold water side than that of the hot water side. From fig. 8 (a) and (b) it is observed that the heat transfer coefficient varies along the length of the channel. From this it is clear that the flow is thermally developing in the channels. Due to flow maldistribution the heat transfer coefficient is different for each channel. This variation is found to increase as Re and number of channels increases. It also observed that there is a sudden jump in the heat transfer coefficient at the channel outlet. This is due to axial conduction from the channel outlet. This sudden jump is notfound at very low Reynolds numbers. As Reynolds number increases the magnitude of the jump also increases. This jump is more in the channels closer to the port outlet than the other channels. This is because the velocity is found to be higher in the channels closer to the port outlet.

Figures.9 (a) and (b) present the local heat transfer coefficients for hot water Re_{ch}=100 and cold air Re_{ch}=1000.



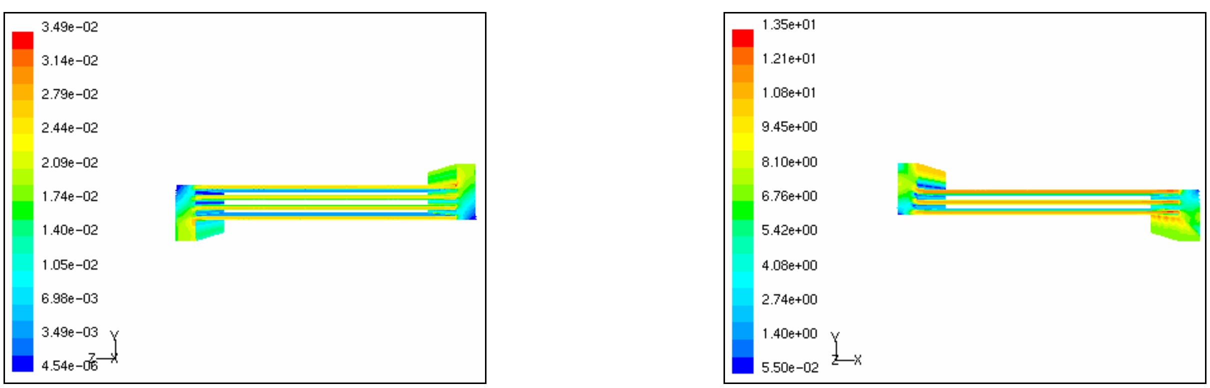
(a) (b)

Fig.4 Velocity profiles (m/s) for (a) cold water and (b) hot water for Re_{ch} = 1500



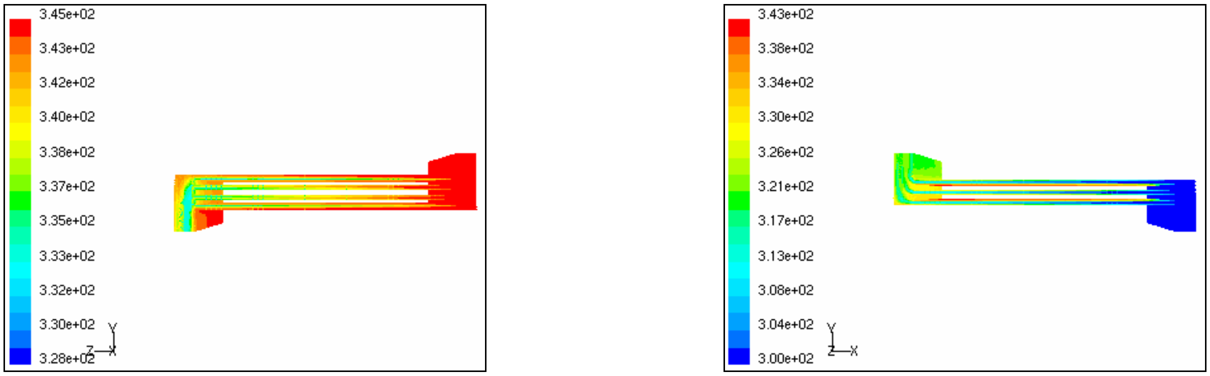
(a) (b)

Fig.5 Temperature profiles (in K) for (a) cold water and (b) hot water for Re_{ch} = 1500



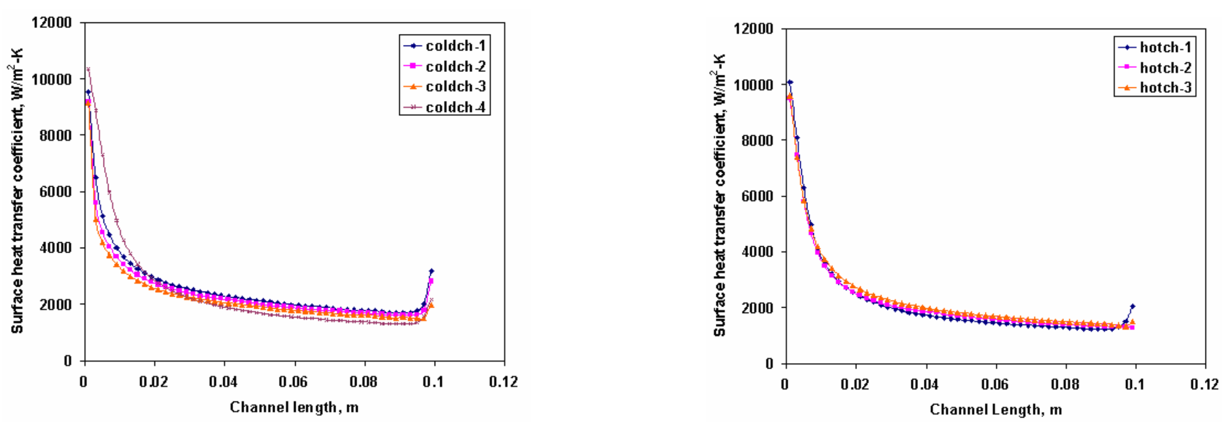
(a) (b)

Fig.6 Velocity profiles (in m/s) for (a) hot water for Re_{ch}=200 and (b) cold air for Re_{ch} = 2000



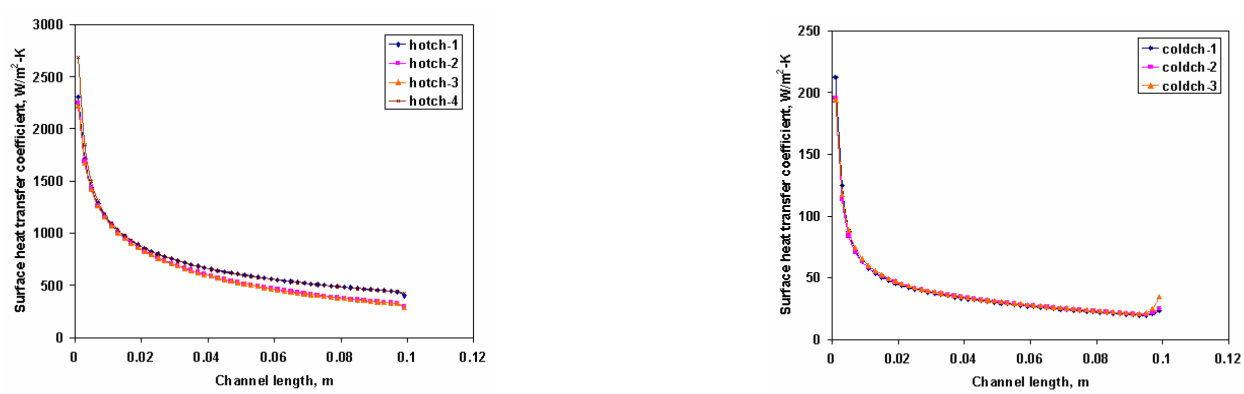
(a) (b)

Fig.7 Temperature profiles (in K) for (a) hot water for Re_{ch}=200 and (b) cold air for Re_{ch} = 2000



(a) (b)

Fig.8 Variation of surface heat transfer coefficient along the channel length for (a) cold water Re_{ch} = 1500 and (b) hot water Re_{ch}=1500



(a) (b)

Fig.9 Variation of surface heat transfer coefficient along the channel length for (a) hot water Re_{ch} = 100 and (b) cold air Re_{ch}=1000

It is observed that the heat transfer coefficient varies along the length of the channel. From this it is clear that the flow is thermally developing on both the air and water side. Due to flow maldistribution the heat transfer coefficient is different for each channel on the water side. Whereas on the air side all channels experience almost same heat transfer coefficient because maldistribution is almost non-existant.

From the above observations it can be concluded that in the first construct

a) flow is thermally developing.

b) heat transfer coefficient is higher in the channels closer to the port exit: bit lower in the channels closer to the port inlet: still lower in the mid channels.

c) Jump in heat transfer coefficient is relatively more on the cold side compared to the hot side for the same Re.

d) Jump is relatively higher in the channels closer to the exit port than the other channels.

e) At lower Re_{ch} (<300), for water, there is even a slight decrease in the heat transfer coefficient in the exit region of the channel.

In fig.10 the effectiveness of the single construct is compared to an equivalent crossflow heat exchanger with fully developed flow. It is observed that effectiveness reduces with Reynolds number for both thermally developing and fully developed flow. This is expected because increase in Re reduces the NTU particularly in the laminar regime. This is also due to the fact that at lower flow (hence lower Re_{ch}) rates bulk mean temperature change in the fluid is larger. Since effectiveness is proportional to temperature change the effectiveness is higher at lower Re. For thermally developing flows bulk temperature change is higher than fully developed flow. With increase of Re the residence time of fluid in the channel decreases and hence bulk mean temperature change will be less resulting in a decrease in effectiveness. This decrease in effectiveness dominates over the increase in effectiveness caused by higher heat transfer coefficient associated with higher Re. Effectiveness in case of thermally developing flow is higher as compared to fully developed flow because of higher heat transfer coefficient associated with developing flow.

In figs.11 and 12 show the relation between NTU and effectiveness. Effectiveness for hot water and cold air (fig.12) is higher when compared to hot water and cold water (fig.11) for the same NTU as heat capacity ratio is lower for the former case. Effectiveness is higher for developing flow as compared to fully developed associated with developing flow

The results show clearly that there is an increase in the local heat transfer coefficient. The effectiveness for the constructal heat exchanger was found to be around 10% higher than that of heat exchanger with fully developed flow conditions. One reason for this high effectiveness is that the flow is thermally developing. The second reason is that the fluid flows in small channels (h=k/D) giving higher compactness.

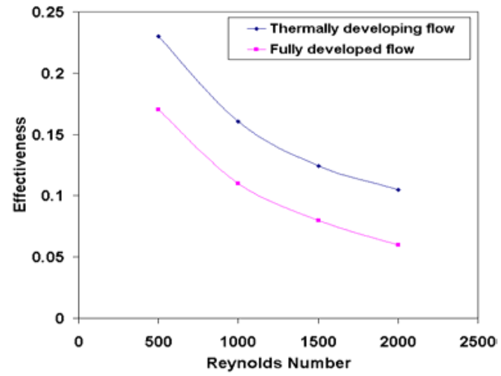


Fig. 10 Effectiveness Vs Reynolds Number.

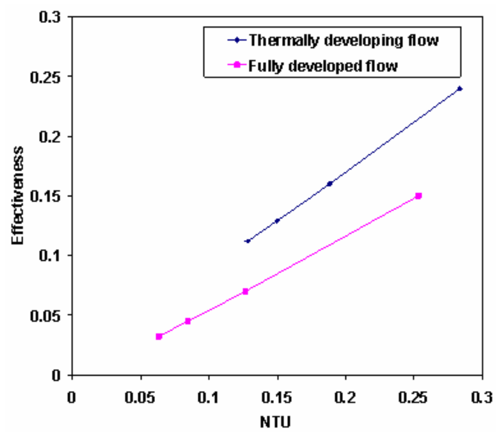


Fig. 11 Effectiveness Vs NTU for water on both hot and cold sides

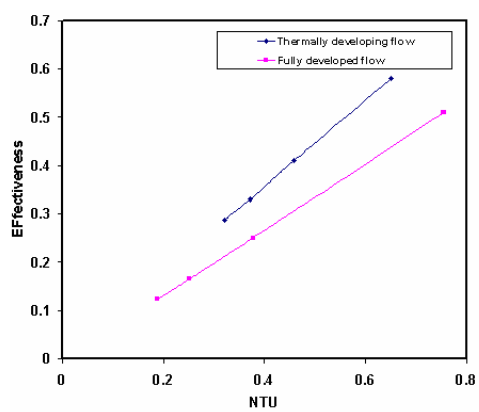


Fig. 12 Effectiveness Vs NTU for hot water and cold air as working fluids.

It is encouraging to see that the constructal heat exchanger gives better performance even for a single construct. Obviously when these single constructs are combined into a multiple scale complete heat exchanger, due to heat transfer through port walls the effectiveness is expected to be further enhanced.

NOMENCLATURE

coldch-1	first cold channel
coldch-2	second cold channel
coldch-3	third cold channel
coldch-4	fourth cold channel
hotch-1	first hot channel
hotch-2	second hot channel
hotch-3	third hot channel
NTU	Number of Transfer Units
Re	Reynolds number
Re_{ch}	Channel Reynolds number
1	cold fluid inlet
2	cold fluid outlet
3	hot fluid outlet
4	hot fluid inlet

CONCLUSION

In this study, numerical solutions were obtained for constant property, laminar fluid flow in the thermal entrance region of rectangular channels stacked one over the other. Results of forced convection heat transfer in the fully developed flow were compared with the thermally developing flow. It was found that the heat transfer coefficients were higher due to small channel spacing and developing laminar flow. Also, there is a sudden and sharp increase in the heat transfer coefficient at the exit region of the channels due to axial conduction. The increase in effectiveness is found to be around 10% for the first construct. It can be concluded that in a dendritic constructal heat exchanger higher effectiveness is achievable compared with conventional cross flow heat exchangers.

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