Thermal Performance Exploration of Air Foil Shape of Pillars using Impinging Jet in Heat Sink

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Objectives: The current investigation introduces the concept of heat sink with combination of jet impingement, micro – channel and air foil shaped pillars. A numerical model is designed to explore the thermal performance of jet impingement with constant heat flux. The steady state conditions are assumed for the laminar and incompressible flow. For the purpose of study dimensionless variables are formed. The performance of jet impingement was predicted in terms of different parameters like temperature rise, drop in pressure and coefficient of heat transfer. Augmentation in pitch diameter ratio, leads to increase in temperature for a particular value of height diameter ratio. Also the heat transfer coefficient gets lowered with the increase in pitch diameter ratio. So proper selection of dimensionless parameters to increase the heat dissipation is of utmost importance.

Key words: Thermal, Jet Impingement, Pitch diameter ratio, Heat sink, Pillar, Height diameter ratio, CFD
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INTRODUCTION

There has been a quick growth in the field of electronics in the recent years. The various parameters of electronics components like efficiency and effectiveness are truly dependent on the temperature. And hence thermal management has been a critical issue in the electronics components. High temperature may result in the severe loss of the electronics components. A.Husain et al. [1] forwarded a heat sink of hybrid nature with jet, piller and micro-channel. Performance was studied and the high h was noticed in the case when ratio of jet pitch and diameter of jet was low. Also the pumping power was analyzed. A.Husain et al. [2] carried out parameter and optimization based studied for cooling of electronic components. Jet impingement heat sink based on silicon was used to analyze the thermal properties for laminar flow conditions in three dimensions. Study was performed using Navier-Stokes equations for steady state incompressible conditions. The optimization was done using two design variables. First variable was taken as the ratio of channel height and diameter of nozzle and the second variable was the ratio of diameter of the nozzle and spacing in the jets. For a heat flux of 1000 KW/m^2 a temperature rise of 339 K was obtained at a pressure drop of 24 KN/m^2. For the conservative analysis convection and radiation heat transfer is not considered at the outer walls. Jian Wang et al. [3] predicted the heat as well as flow properties in thermal sink with finned foam of copper experimentally. The cooling was provided with the help of slot jet impingement as well as with the help of axial fan. Comparative study was made for the heat sink of conventional type and with foam. It was observed that the finned foam type heat sink was having better performance in contrast to the heat sink without foam. The height of the thermal sink also affects the achievements in terms of temperature and heat. Kim, SY, & Webb, RL [4] Studied the phenomenon of convective thermal resistance fie different types of heat sink. Heat sinks taken for study are of plate fin type, heat sink with offset strip fin and sink of round pin fin type. It was found from the results that higher thermal resistance which was of the convective nature was obtained in thermal sink with pin fin of round shape. Also when compared heat sink of plate fin type shows more thermal convection resistance than offset strip heat sink. There was a decrease in the thermal performance as and when there was a decrease in the offset strip length. Chandratilleke, T. T. et al. [5] extended a novel idea in a heat sink which was based on the micro channel to increase the removal of heat. Concept of synthetic jet was introduced for the numerical simulation. This concept of synthetic jet gave enhanced thermal properties. It was observed from the results that this technique gives almost sixty times more rate of heat transfers as compared to simple natural convection base sink. Also in differentiation to channel based flow it gives almost 4 times of Q. So this type of technique is very useful in electronic components which requires high rate of cooling. Seo Young Kim et al. [6] performed experimental study in case of multi jet impingement. Investigation was performed by taking heat sink with aluminum foam. The effect of various parameters like pore density, velocity of jet etc. was investigated on the nusselt number. Comparative study was also made for single and multi-jet impingement. When the thermal performance was studied for one jet impingement it was found that performance is high in thermal sink with aluminum foam than the heat sink with plate fin. The value observed to be eight to thirty-three percent higher. Also in case of multi jet impingement the value is two to twenty-nine percent more. It was also observed that with the increase in spacing between the jets the performance in terms of cooling decreases with aluminum foam. It was also focused that the single jet impingement shows less performance in comparison to multi jet impingent heat sink. Hadad, Y. et al. [7] studied that there was small drop in pressure in thermal sink of water cooled impingement type. Due to this reason these are preferred more as compared to thermal sink of parallel flow type. To study and analyze the performance of heat sink a numerical model was initiated. Both the distributor and collector were also included in the model. This was because of their effect on the performance of heat sink. Design and response parameters were selected. Design parameters were based on the geometry. Hydraulic resistance and thermal resistances were taken as response parameters. It was predicted from the results that geometry of channel has more effect on the drop in pressure in the sink as compared to the geometry of impingement. Sensitivity based investigation was also performed. Ariz, M. et al. [8] Executed numerical analysis in 3 dimensions with low values of Reynolds number. The analysis was performed in case of coupled heat sink of copper based. Diameter of jet, diameter of effusion hole etc were taken as design variables and the end result of these parameters were studied with the heat transfer rate. It was found from the results that with higher value of the ratio of standoff and diameter of jet provide low value of thermal resistance. Also the interrelation between the power of pumping and overall thermal resistance was established in the study. Lee, D.-Y., & Vafai, K. [9] presented a relative analysis of cooling in case of impinging jet and micro channel. The study was performed for the target dimension which was taken as prime parameter. It was analyzed from the studies that for smaller target dimensions' micro channel heat transfer is suitable while in case of larger target dimensions' jet impingement is most suitable. In case of jet impingement technique with a less drop in pressure there is large requirement of rate of coolant flow while in case of micro channel there was a small requirement of rate of coolant flow with a large drop in pressure. Li, H.-Y et al. [10] studied sink attributes using confined impinging numerically. The effect of diverse parameters was explained on R_{Thr}. As the value of Re surges, R_{Thr} gets decreased. Deduction from end effects was that with the surge in the fin height, thermal accomplishment gets improved. Also the rate of decrease of R_{Thr} deteriorate with the increase in height of fin. Also it was observed that the optimal width of the fin is a function of Re. As Re increases it also increases. R_{Thr} is inversely proportional to thermal conductivity. The presence of the plate upper side increases R_{Thr}. A. Husain et al. [11] studied the cooling system with jet impingement with the help of a finite volume solver. The aim was to dissipate the heat in case of Light emitting diodes used as arrays. The analysis was performed numerically in three dimensions for the turbulent flow conditions with steady state. Different combinations of array were selected to analyze and evaluate the drop in pressure and resistance. When the rate of flow was high, lower value of thermal resistance was obtained. For optimization four jet array combinations was selected and the design variables were analyzed. Li, H.-Y et al. [12] explored the performance of heat sink with the help of infrared thermograph. The design variables selected were Reynolds number, fin width, fin height, nozzle and fin tip distance and heat sink type. Thermal resistance was taken as response variable. Results showed that thermal resistance is inversely proportional to Reynolds number. Thermal performance can be increased with an increase in the width of fin but corresponding to a suitable Reynolds number. When comparison was made between fin width and height, it was observed increase in the width of fin is more prominent in decreasing the thermal resistance as compared to increase in the height of the fin. As far as geometric dimensions are concerned, Reynolds number plays an important role. At a lower value of Reynolds number, the effect of geometric dimensions is more on the thermal performance as compared to the higher values of Reynolds number. X. Song et al. [13] Computational fluid dynamics and optimization technique based on surrogate model were implemented in case of a thermal sink with plate fin to study the thermal achievements in the form of thermal resistance as the objective function. The validation of Computational fluid dynamics results was done with the help of experimental values. The interrelation between the design variables and the temperature was pointed out. The gap width was noticed to be the most prominent variable among all the design variables. Also the combination of two techniques is detected to be a promising tool in case of jet impingement heat sink design. Abo-Zahhad, E. M et al. [14] Compared the jet impingement cooling with micro channel to simple jet impingement without micro channel. The study was made in case of photovoltaic structure of high concentration in both with and without micro channel. Five different designs of heat sink were analyzed for cell temperature. From the findings it was concluded that the hybrid scheme is more effective in contrast with simple jet impingement in terms of cell temperature reduction as well as in the uniformity of temperature. A temperature reduction of around 13 ^{0}C was observed for one of the case and in another case it was 20 ^{0} C when the rate of mass flow was increased at the inlet. Also there was an improvement in the efficiency of cell. Naphon, P., & Wongwises, S. [15] investigated experimentally thermal properties in CPU of computer. Heat sink with mini rectangular fin was taken for analysis. Real operating conditions were taken. Study was performed under no load and full load cnditions. Comparative study was performed for conventional liquid cooling system and liquid jet impingement system. It was found from the results that jet impingement system is more efficient than the conventional system. But there is a disadvantage that it leads to increase in consumption of energy. Also the results are of the real life importance for the design of computers. De Oliveira, P. A., & Barbosa, J. R [16] presented the concept of heat sink with two phase jet. The end results of the design parameters were analyzed on the objective parameters. The diameter of orifice, applied load and the stroke of piston of the compressor were taken as parameters of design. It was explored that orifice with diameter 0.0005 m gives more thermodynamic performance as compared with 0.0003 m diameter orifice. This was obtained when the length of jet and temperature of high temperature reservoir was fixed. In case of high diameter orifice pressure ratio obtained was low. And due to this reason there was less power consumption in compressor. For the 0.75 and 1 stroke of the compressor IInd law ratio was quite higher. A.Husain et al. [17] Different flow schemes were compared in case of jet impingement stem of cooling. Heat sink with different array combinations with the ratio of height of channel and diameter of nozzle as 2 was analyzed in terms of temperature increase; drop in pressure, thermal resistance and heat transfer coefficient. Lesser values of pressure drop and higher conformities were found in case of unconfined flow. In case of flow extraction h was found to be higher but low value of thermal resistance. A trend of increasing uniformity in temperature and decreasing performance was observed with increase in Reynolds number. As far as higher uniformity in temperature is concerned, the unconfined flow scheme seems to be better as compared with the other two. Kim, T. H et al. [18] proposed the sink concept using plate fin along jet impingement of uniform air for performance analysis. The proposed relations and the available experimental results were compared. From the previous studies it was found that these studies were based on the non-uniformity. And hence uniformity at the inlet plays an important role on the performance. These correlations can be helpful in determining the performance deviations during actual and ideal conditions. Also using these correlations, the end result of height of fin on the design parameters were studied. Positive relation was seen between the height of fin and the performance of heat sink. As the height is increased from 0.005 m to 0.025 mm, the performance also increased around 60 percent. Wiriyasart, S., & Naphon, P. [19] Presented the jet impinging technique with several fin shapes. Taking circular, rectangular and conical fins parametric study was performed. The inlet temperature of the coolant, mass flow rate of liquid was also taken for parametric analysis. Circular fin was observed to be giving high thermal performance as compared to the other two. For rectangular fin it was 12 percent higher and 25 percent for the conical fin. So this study can be implemented which taking care of design of thermal systems using different fin shapes. Chapman, C. L et al. [20] Tested heat sink of aluminum with fin which was extruded and comparative study was made with rectangular pins which were of the cross cut type and pins of elliptical shape. The testing was performed for different factors like drop in pressure, thermal conductivity, surface area and formation of boundary layer. In case of elliptical pin, the vortex was found to be reduced and hence no boundary layer formation. Extruded shape found to be performing better than the other two. Zunaid, M. et al. [21,22] studied heat characteristics using different heat sink materials. Study was performed for different Reynolds numbers.

Depending on the availability of literature, a novel air foil shape is used for the pillars to enhance the dissipation of heat with impingement and channel flow.

GEOMETRIC REPRESENTATION OF IMPINGING JET HEAT SINK

The arrangement regarding impinging jet heat sink with airfoil shaped pillars is exhibited in figure 1. The measurement of the heat based sink as base of substrate is taken as 1 cm X 1 cm. The height of channel (H_{ch}) is taken as 0.4 cm. Substrate base thickness is taken as 0.2 cm. A plate of thickness 0.4 cm is taken on which nozzles are fitted is known as nozzle plate and thickness of this plate is named as nozzle plate thickness (t_{nz}). Total five design dimensionless variables are selected for design and analysis. Dimensionless variables are ratio of diameter of jet (D_j) and diameter of pillar (D_{pl}) named as diameter ratio, proportion of width of channel (W_{ch}) and width of wall of channel (W_{chw}) named as width ratio, proportion of width of channel (W_{ch}) and diameter of jet (D_j) names as width diameter ratio, ratio of height of channel (H_{ch}) and diameter of jet (D_j) named as height diameter ratio and ratio of pitch of jet (P) and diameter of jet (D_j) named as pitch diameter ratio. For analysis diameter ratio and width ratio are assumed to be constant and the value of height diameter ratio, width diameter ratio and pitch diameter ratio are jumbled.

The computational region for the problem is shown in the figure 1. It consists of heat sink with jet of water. The inlet and outlet of channel are shown. Pillars are used to increase the heat exchange rate.

MATHEMATICAL FORMULATION

The flow domain is shown in the figure 1. The governing equations used for the solutions are based on the conservation of mass, conservation of momentum and conservation of energy. The numerical simulation is performed with the help of computational fluid dynamics. Water is taken as a coolant which flows through the channel and jet.

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \quad (1)$$

$$\frac{\partial(\rho u)}{\partial t}+u\frac{\partial(\rho u)}{\partial x}+v\frac{\partial(\rho u)}{\partial y}+w\frac{\partial(\rho u)}{\partial z}=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}[\lambda \nabla . V+2\mu\frac{\partial u}{\partial x}]+\frac{\partial}{\partial y}[\mu(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y})]+\frac{\partial}{\partial z}[\mu(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x})]+\rho f_x \quad (2)$$

$$\frac{\partial(\rho v)}{\partial t}+u\frac{\partial(\rho v)}{\partial x}+v\frac{\partial(\rho v)}{\partial y}+w\frac{\partial(\rho v)}{\partial z}=-\frac{\partial p}{\partial y}+\frac{\partial}{\partial y}[\lambda \nabla . V+2\mu\frac{\partial v}{\partial y}]+\frac{\partial}{\partial x}[\mu(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y})]+\frac{\partial}{\partial z}[\mu(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y})]+\rho f_y \quad (3)$$

A constant value of heat flux is taken and is initiated as source of heat at the bottom side of substrate. Reynolds number is given by

$$Re = \frac{\rho v d}{\mu} \quad (4)$$

Local heat transfer coefficient is

$$h = \frac{Q}{A\Delta T} \quad (5)$$

Nusselt number is given by

$$Nu = \frac{hd}{k} \quad (6)$$

RESULTS & DISCUSSION

The numerical model has been tested for grid independence test. The three dimensional model was validated with the help of available literature. The results obtained are in straight accordance with experimental outcomes available [1,2,23,24]. Computational fluid dynamics analysis has been accomplished to control thermal attributes of model. The results were found using parameters of rise in temperature, drop in pressure, thermal resistance, coefficient of heat transfer and Nusselt number

Figure 2 shows the comparative analysis of modification in the temperature rise of fluid $(\Delta T)_m$ with P/D_j corresponding to H_{ch}/D_j = 4, 3, 2 and 1. For H_{ch}/D_j =1, It is observed from the graph that as the P/D_j is increased the rise in maximum temperature also increases. This trend is observed up to pitch diameter proportion is 12. Also corresponding to H_{ch}/D_j =2, It is observed from the graph that as P/D_j is increased the rise in maximum temperature also increases. This trend is observed up to P/D_j is 12. In case of H_{ch}/D_j = 3, increasing trend is observed in the temperature profile in contrast to surge in P/D_j. From graph the minimum value of temperature rise is observed to be 27 k at P/D_j = 6 and the peak value to be 38 k at P/ D_j = 12 for of H_{ch}/D_j = 4. The maximum rise in temperature is observed at H_{ch}/D_j = 1 and P/ D_j = 12. So it can be concluded that the as the P/D_j increases, the $(\Delta T)_{m\ also}$ increases. The augmentation in $(\Delta T)_m$ is due to the less number of impingements with augmentation in P/D_j.

Figure 3 shows the comparative analysis of variation in the overall pressure drop (OP) with the pitch diameter ratio (6, 8, 10 and 12) corresponding to height diameter proportion (H_{ch}/D_j) = 1, 2, 3 and 4. P/D_j is varied from 6 to 12 and the variation in pressure drop is analyzed. Corresponding to H_{ch}/D_j = 1, It is observed from the graph that there is an inverse relation between P/D_j and overall drop in pressure. The minimum and maximum value of OP is observed to be 3508 and 4811 respectively which are corresponding to P/D_j = 12 and P/D_j = 6. In case when the height diameter ratio (H_{ch}/D_j) is kept constant and is equal to 2 and pitch diameter ratio is varied from 6 to 12, It is observed from the graph that OP decreases with increase in P/D_j. The minimum and maximum value of OP is observed to be 5401 and 8069 respectively which are corresponding to P/D_j = 6 and P/D_j = 12. For H_{ch}/D_j = 3 which is kept constant, the results show the minimum and maximum value is at P/D_j = 12 and P/D_j = 6. Large pressure drop is found from the results at P/D_j = 6 and P/D_j = 12 at H_{ch}/D_j = 4. Form the results it is concluded that Large values of OP are found corresponding to H_{ch}/D_j = 4.

This is a fact that coefficient of heat transfer plays an important role in case of heat transfer. Coefficient of heat transfer is found using heat flux = 100 X 10^4 W/m^2. Figure 4 depicts the comparative analysis of development in the heat transfer coefficient (h) with the pitch diameter ratio (6, 8, 10 and 12) corresponding to H_{ch}/D_j = 1, 2, 3 and 4. The curve in figure 4, in case when the height diameter proportion (H_{ch}/D_j) is kept constant and is equal to 1shows the modification in coefficient of heat transfer (h) with (P/D_j). P/D_j is varied from 6 to 12 and the variation in h is analyzed. The minimum and maximum value of h is observed to be 11111 and 14027 respectively which are corresponding to P/D_j = 12 and P/D_j = 6. For H_{ch}/D_j = 2, h minimum is 14285 at P/D_j =12. Also the maximum value of h is found to be 20000 at P/D_j = 6. One of the curve in figure 4, depicts the variability of h with the P/D_j for H_{ch}/D_j = 3. The minimum value of h is observed to be 22578 at P/D_j =12. Also the maximum value of h is found to be 31625 at P/D_j = 6. Decreasing trend is observed from the results for h with an increase in pitch diameter ratio. Also for H_{ch}/D_j = 4, minimum value of h is observed to be 26288 at P/D_j =12 and maximum value of h is found to be 36900 at P/D_j = 6. Decreasing trend is observed from the results for h with an increase in P/D_j. Now from the figure 4 it was concluded that h is found to be maximum in case of H_{ch}/Dj = 4 and P/D_j = 6. Larger value of heat transfer coefficient will help in higher heat transfer rate so this case is preferred over the other ones. With a surge in P/D_j, number of impingements decreases which results in diminishing of h.

Nusselt number is an important parameter in case of heat transfer A comparative study was made for the Nu and pitch diameter ratio for different height diameter ratio (1, 2, 3, and 4) and is shown in the figure 5. The alteration of Nusselt number (Nu) with pitch diameter ratio for height diameter ratio = 1 is shown in the one of the curve in figure5. The value of Nu is found to be maximum at pitch diameter ratio = 6 and minimum value of Nu is found from the results at pitch diameter ratio = 12. Corresponding to H_{ch}/D_j = 2, After a value of pitch diameter ratio = 10 no significant change in the value of Nusselt number is observed in the results. Also from the results it can be concluded that the maximum value of Nu is at P/D_j = 6. The minimum value of Nu is observed at pitch diameter ratio = 12. For a constant value of height diameter ratio = 3, Inverse relation is predicted from the results in between the two variables. The maximum value of Nu is predicted at pitch diameter ratio = 6 and minimum value at P/D_j = 12. As is seen from the graph with an increase in the value of pitch diameter ratio from 6 to 12, there is a decrease in the value of Nu. So the case with P/D_j = 6 is preferred when H_{ch}/D_j = 4. From the results it is clear that the same type of trend is observed in case of height diameter ratio = 3 and 4. The maximum value of Nusselt number is found in case of height diameter ratio = 1 and pitch diameter ratio =6.

Thermal resistance (R_{Thr}) is the resistance to the flow of heat transfer. Higher thermal resistance will result in lesser heat transfer rate. Differentiating analysis between the thermal resistance and pitch diameter ratio for different values of height diameter ratio (1, 2, 3 and 4) is shown in the figure 6. From the curve in figure 6, Lesser R_{Thr} is obtained from the results at pitch diameter proportion = 6 and H_{ch}/D_j = 1. Maximum value of R_{Thr} is found at P/D_j = 12 corresponding to H_{ch}/D_j = 1. At a constant value of H_{ch}/D_j = 2, the modification in thermal resistance with the change in the pitch diameter ratio is shown in the figure 6. Higher R_{Thr} is obtained from the results at P/D_j = 12. This implies that at this condition less rate of heat transfer will be there as compared to the heat transfer rate at pitch diameter proportion = 8. Corresponding to height diameter proportion = 3, as the value of pitch diameter proportion increases the thermal resistance also increases which is not a favorable condition for the rate of heat transfer. The maximum R_{Thr} is observed from the results at pitch diameter proportion = 12. At a height diameter proportion = 4, an increasing trend is detected in R_{Thr} with an increase in the value of pitch diameter ratio. Maximum R_{Thr} is found from the results at P/D_j = 12. So it was concluded from the figure 6 that highest R_{Thr} is predicted at proportion of height and diameter =1 and proportion of pitch and diameter = 12. Also the lowest value of thermal resistance is predicted at H_{ch}/D_j = 4 and P/D_j = 6. The surge in the R_{Thr} is due to increased velocity, which further increases the frictional losses.

Pumping power (PR) is power which is needed for the fluid to flow through the heat sink. Comparative exploration of PR with pitch diameter proportion for different values of height diameter proportion as Hch/D_j = 1, H_{ch}/D_j = 2, H_{ch}/D_j = 3 and H_{ch}/D_j = 4 has been shown in figure 7. At a height diameter ratio = 1, from the results maximum value of PR is found at a pitch diameter ratio of 6 and minimum value is at a pitch diameter ratio of 12. In the case H_{ch}/D_j = 2, maximum value of PR is found at a pitch diameter ratio of 6 while the minimum value of PR is observed at a pitch diameter ratio of 12. Higher pumping power is not desirable. For a constant value of height diameter ratio = 3, From the results it has been noted that as the pitch diameter ratio is increased from 6 to 12, Pumping power gets decreased. The minimum value of pumping power is observed corresponding to pitch diameter ratio of 12 while the highest value is obtained at 6. for H_{ch}/D_j = 4, lowest observation of PR is obtained corresponding to pitch diameter ratio =12 while the highest value is corresponding to 6. So It was concluded that the lowest pumping power is predicted in case of the combination with pitch diameter proportion = 12 and height diameter proportion = 2.

Figure 8(a) depicts the distribution of temperature in impingement of jet in a heat sink with the air foil pillars. Temperature contours are shown in the figure in case of height diameter proportion = 1 and pitch diameter proportion = 6. The distribution of temperature for the fluid domain shows the minimum and maximum values of temperature. In the figure 8(b) distribution of pressure is depicted with the help of simulation using CFD.

Figure 8(c,d) shows the flow patter in the form of velocity vector at $\frac{H_{ch}}{D_j} = 1$ and pitch diameter proportion = 6. Also the streamlines are shown in the figure 8(e, f). Vortices are noticed behind the impingement of jet. The end result of jet is noticed to be somewhat destroyed due the flow in the channel.

CONCLUSION

A numerical model has been constructed to study and analyze the outcomes of air foil shaped pillars on the accomplishments of a heat based sink with jet impingement technique and micro channel. From the results it has been found that rise in maximum temperature was obtained with height diameter ratio 1 and pitch diameter ratio 10. The drop in pressure was found lowest in case of height diameter ratio 1 and pitch diameter ratio 12. Heat transfer coefficient is found highest in case of height diameter ratio 4 and pitch diameter ratio 6. Also the value of Nusselt number is highest with height diameter ratio 1 and pitch diameter ratio 6. Thermal resistance is predicted minimum in case of height diameter ratio 4 and pitch diameter ratio 6. Pumping power has been observed minimum with height diameter ratio 2 and pitch diameter ratio 6. So highest temperature rises and lowest pressure drop has been found with increasing pitch diameter ratio. The channel flow results in the reduction of the jet end results. So a proper selection of parameters is required to augment the heat dissipation.

Conflicts of Interest Disclosure Statement

The authors declare no conflict of interest in the authorship or publication of this work. The authors declare no sponsored financial sources for the undertaken study.

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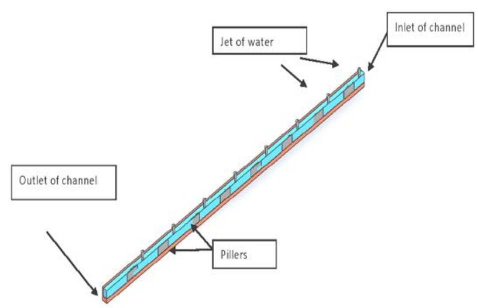


Figure – 1: Computational Region

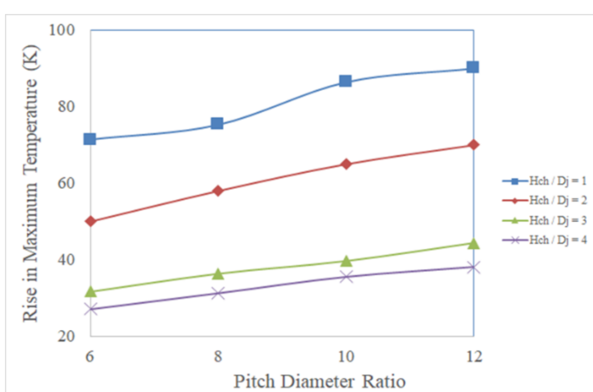


Figure 2: $(\Delta \text{ T})_m \; Vs \; (\frac{P}{D_j})$

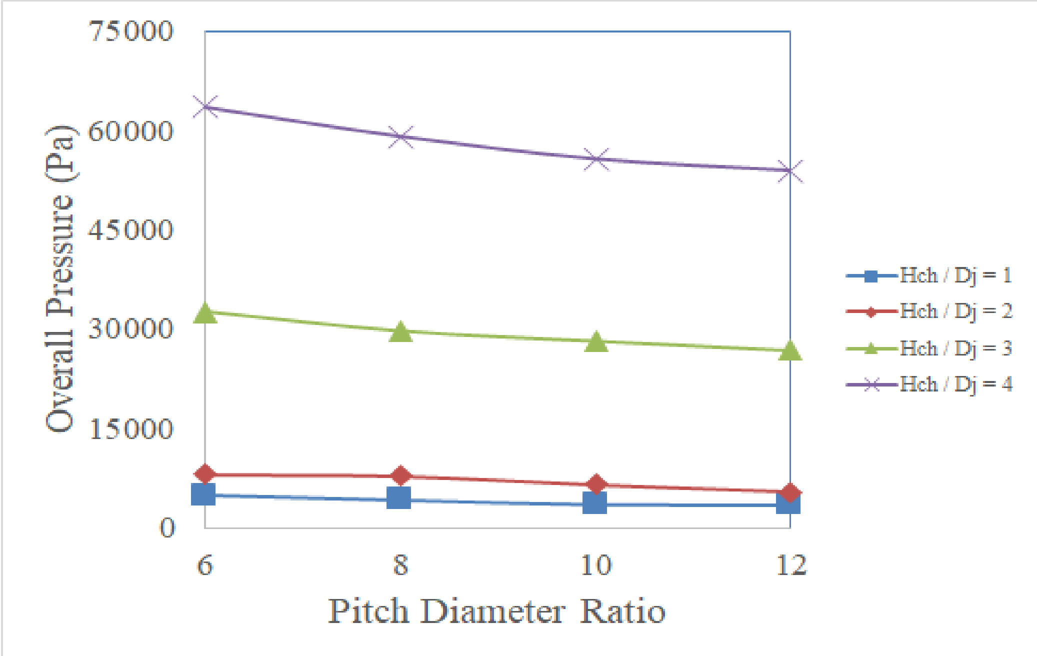


Figure 3: OP $Vs\ (\frac{P}{D_j})$

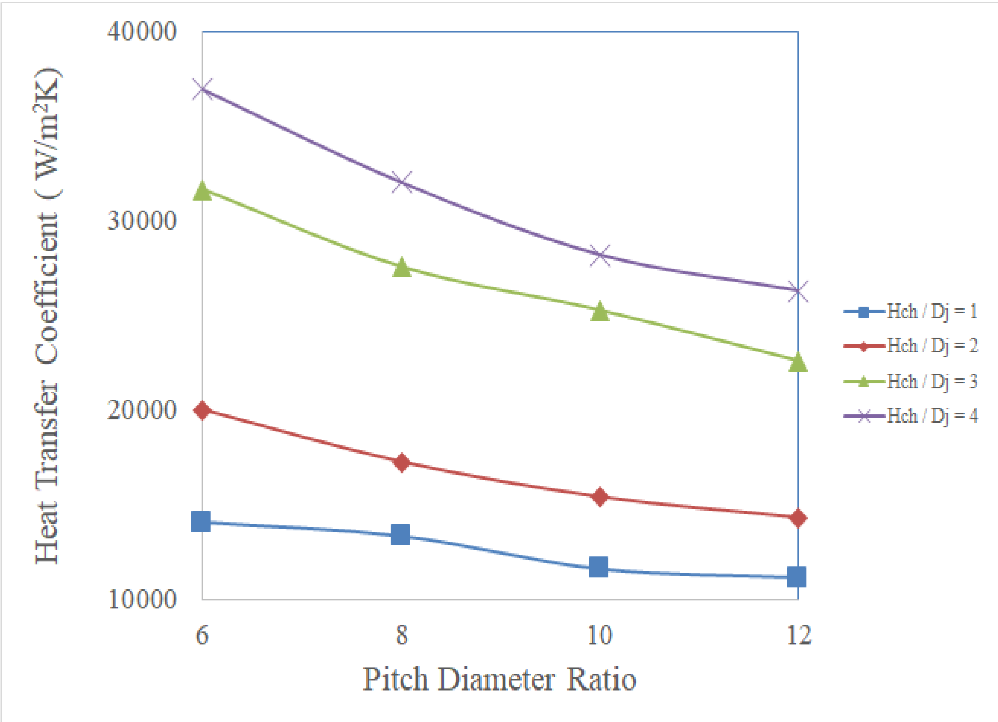


Figure 4: h $Vs\ (\frac{P}{D_j})$

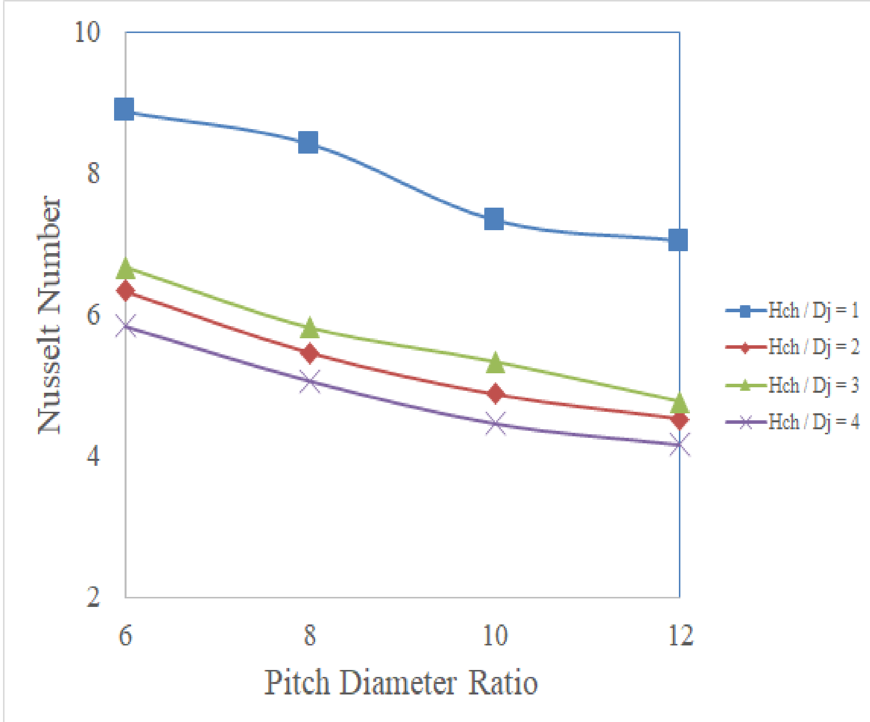


Figure 5: Nu $Vs\ (\frac{P}{D_j})$

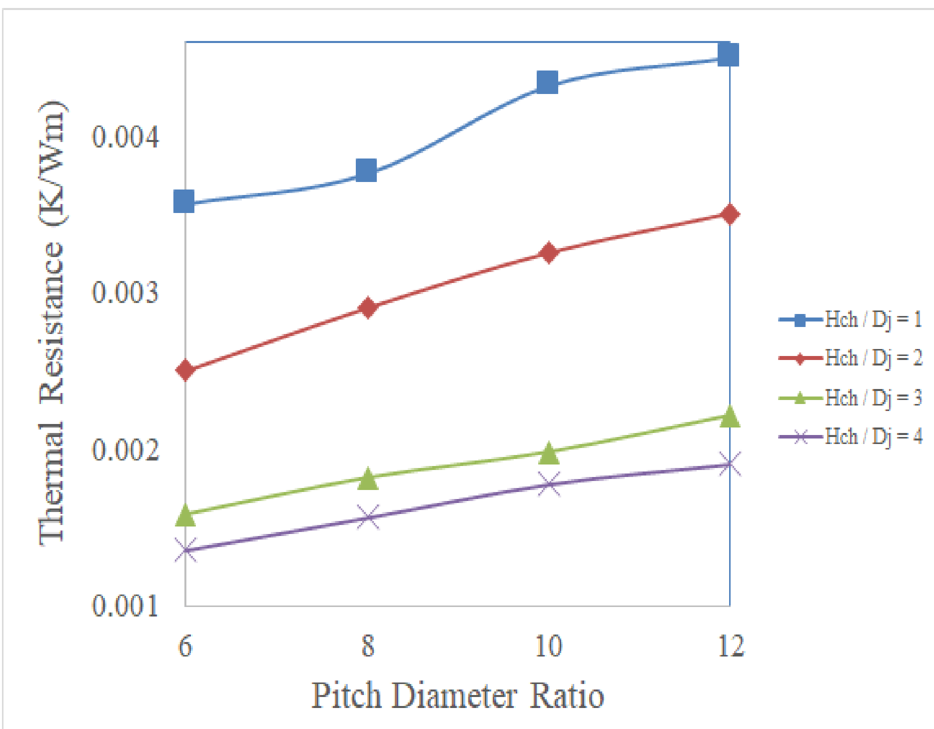


Figure 6: $(\mathrm{R})_{Thr}\ Vs\ (\frac{P}{D_j})$

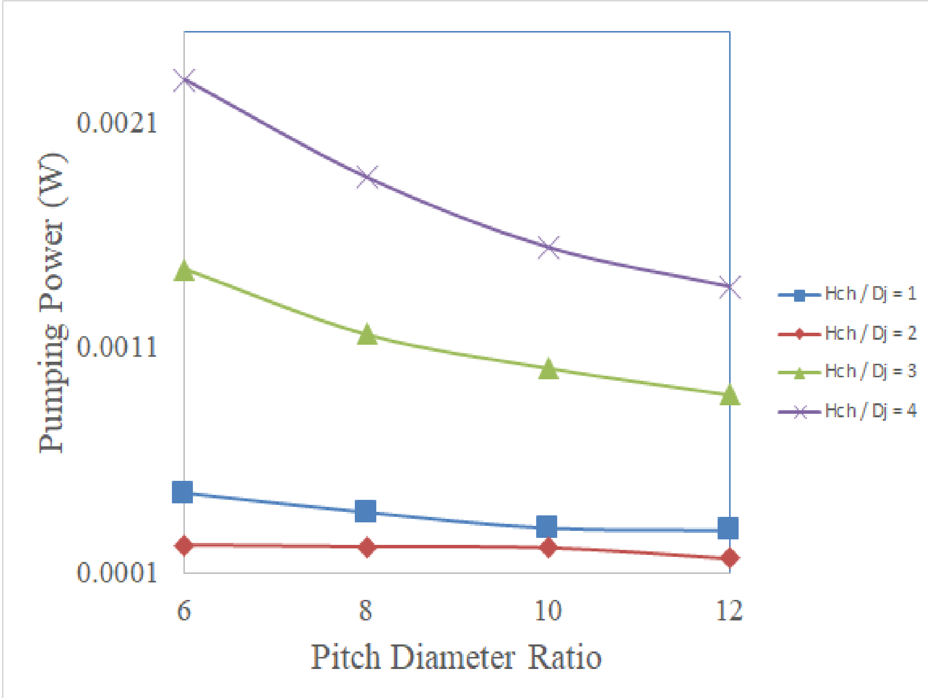


Figure 7: PR $Vs\ (\frac{P}{D_j})$

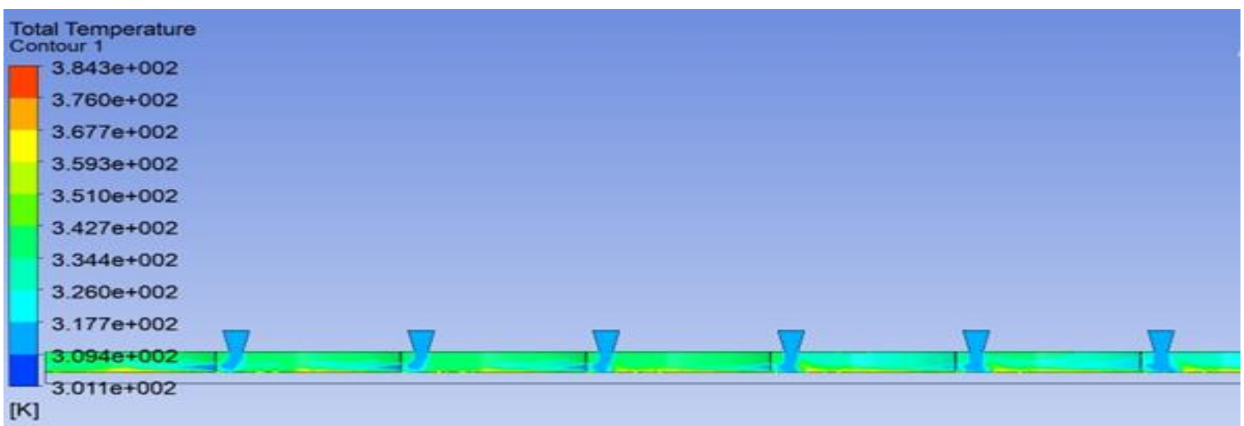


Figure 8(a): Temperature Distribution at $\frac{H_{ch}}{D_j} = 1, \frac{P}{D_j} = 6$

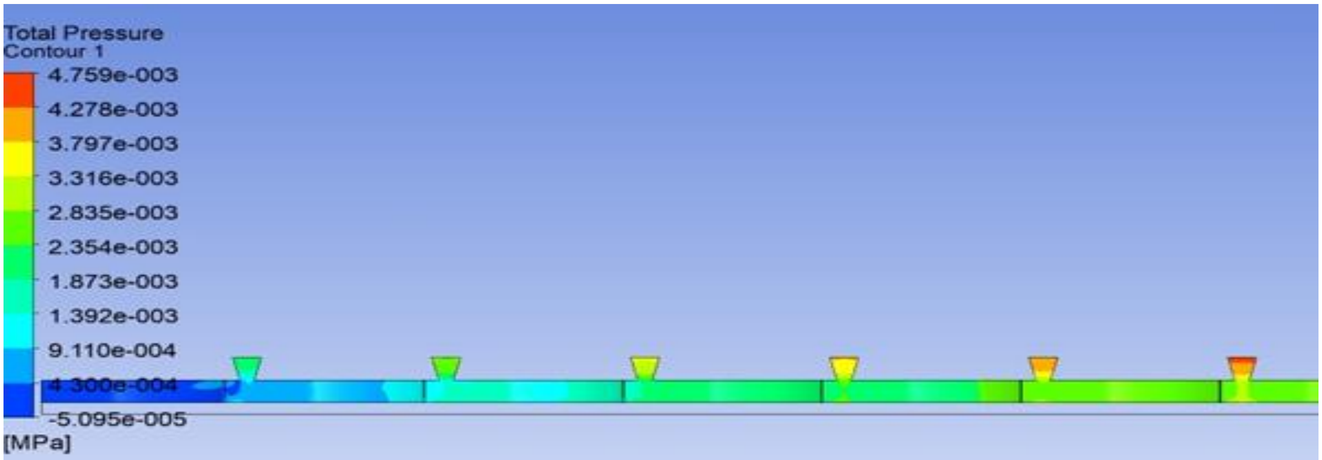
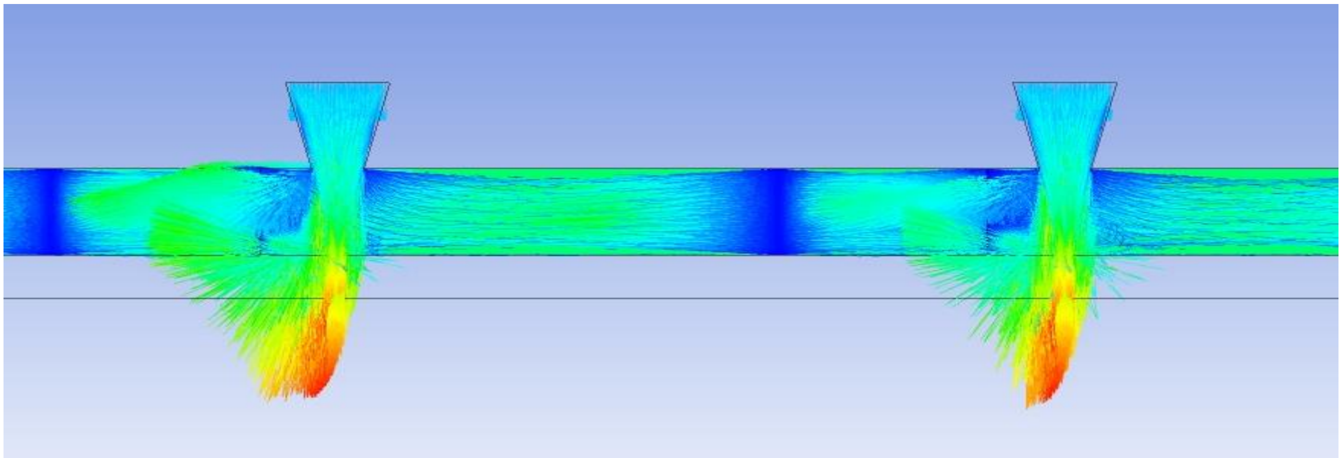
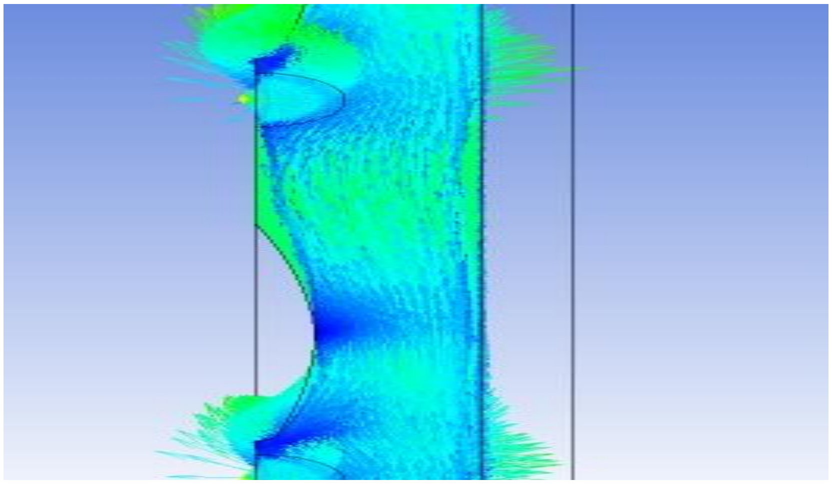


Figure 8(b)：Pressure Distribution at $\frac{H_{ch}}{D_j} = 1, \frac{P}{D_j} = 6$

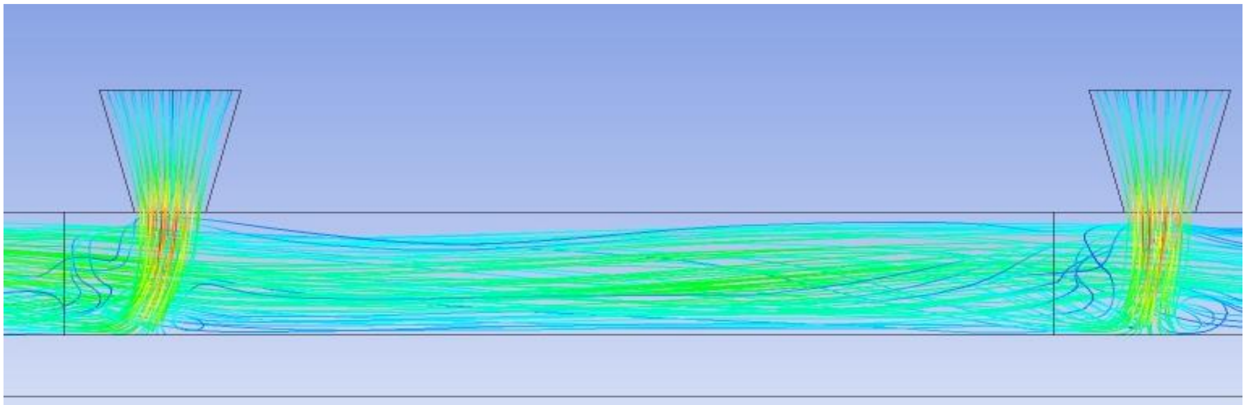


(C)

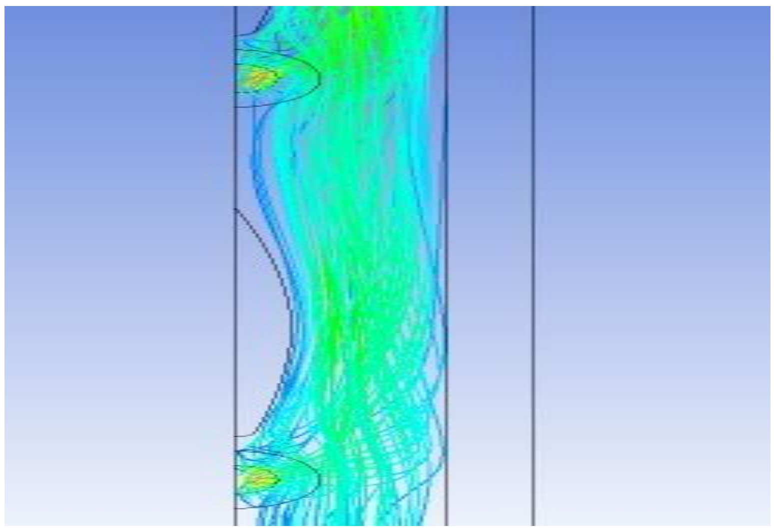


(d)

Figure 8 (c, d): Flow Pattern at $\frac{H_{ch}}{D_j} = 1, \frac{P}{D_j} = 6$



(e)



(f)

Figure 8(e, f): Stream lines at $\frac{H_{ch}}{D_j} = 1, \frac{P}{D_j} = 6$

Nomenclature

D_j	Diameter of Jet	μ	Dynamic Viscosity
D_{pl}	Diameter of Pillar	$\frac{P}{D_j}$	Pitch diameter Ratio
h	Coefficient of Heat Transfer	$\frac{H_{ch}}{D_j}$	Height Diameter Ratio
H_{ch}	Height of Channel	ρ	Density of Fluid
K	Thermal Conductivity		
Nu	Nusselt Number		
P	Pitch of Jet		
PR	Pumping Power		

OP	Overall Pressure
Q	Rate of Heat Transfer
R_e	Reynolds Number
R_{Thr}	Thermal Resistance
$(\Delta\ \mathrm{T})_m$	Rise in Maximum Temperature
t_{nz}	Thickness of Nozzle plate
W_{ch}	Width of Channel
W_{chw}	Width of wall of channel