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A brief review on mixed convection heat transfer in channel flow with vortex generator for electronic chip cooling

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ABSTRACT
In an effort to increase processor speeds, 3D IC architecture is being aggressively pursued by researchers and chip manufacturers. This architecture allows extremely high level of integration with enhanced electrical performance and expanded functionality, and facilitates realization of VLSI and ULSI technologies. However, utilizing the third dimension to provide additional device layers poses thermal challenges due to the increased heat dissipation and complex electrical interconnects among different layers. The conflicting needs of the cooling system requiring larger flow passage dimensions to limit the pressure drop, and the IC architecture necessitating short interconnect distances to reduce signal latency warrant paradigm shifts in both of their design approach. Additional considerations include the effects due to temperature non-uniformity, localized hot spots, complex fluidic connections, and mechanical design. This paper reviews the advances in electronic chip cooling in the last decade and provides a vision for code signing integrated cooling systems. For various heat fluxes on each side of a chip acting as discrete heat source, the current single-phase cooling technology is projected to provide adequate cooling, albeit with high pressure drops. Effectively mitigating the high temperatures surrounding local hot spots remains a challenging issue. Various forms of tabulators above the chips, different geometric arrangements of the chips positioned top and bottom wall of the duct serves very well in the heat augmentation technique with better performance.
Keywords – Convection heat transfer, chip cooling, channel, vortex generator

I. INTRODUCTION

The rapid advances in the computer industry have resulted in an increased need for reliable and efficient cooling technologies. Almost all of the electrical energy consumed by electronic devicesappears as heat, the power density that must be dissipated by individual chips called discrete heat sources. Advances in electronic devices have yielded increasing power dissipation per chip and resulted in increased heat flux densities. The subject of electronic cooling has therefore generated increased interest in the analysis of fluid flow and heat transfer in discrete heating situations.To avoid unacceptable temperature rises in electronic devices,an industrial system and metallurgy to prevent thermal problems, improvements in cooling methods are obligatory. Heat transfer enhancement is pertinent to the design of compact heat exchangers as automotive industry, electronic cooling, spacecraft, and aircraft applications.

II. CHANNEL WITH DISCRETE HEAT SOURCE

A summary of the important numerical investigations of discrete heat sources in different array form representing electronic chips for a laminar flow is presented in Table 1. The buoyancy effected secondary flow created by the discrete heat sources and the onset of instability are responsible for enhancement of heat transfer rate. This fact also leads to the reduced flow rate, which in turn reduces power consumption. Several researchers have studied the effect of discrete heat sources in array form onto mixed convective the heat transfer augmentation experimentally using air as working fluid considering laminar flow. T. Pirasaci [1]worked with 8×4 protruded heat sources with uniform heat flux on the lower and upper wall of a channel at various Reynolds number (Re), modified Grashof number (Gr), Richardson number (Ri) and height/width ratio (H/W) and have found that Row averaged Nusselt number (Nu_{Dh}) increases with increase of Ri. A. Dogan[13] worked with 8×4 flush mounted heat sources positioned on the top and bottom wall with AR=6, and with different Gr and Re values and observed that the surface averaged Nusselt number 1st decreases than increases with the increases with the row number.S. Chen, Y. Liu [15] attempted to find the optimum spacing ratio between the heated elements to get better heat transfer rate with an improved performance.

Table. 1 Summary of important investigations of discrete in array form in laminar flow

Author	Parameters	Observations	Remarks
T. Pirasaci[1]	➢ 32 protruded heat sources in 8×4 array form ➢ H/W =1/2, 1/4, 3/20 ➢ Re= 2150, 1450, 850 ➢ Gr= 3×10^8, 2×10^8, 1×10^8, ➢ Richardson number has been obtained between 0.02 and 12.5. ➢ Walls are maintained insulated and adiabatic. ➢ Rectangular channel	➢ For all heater rows Nu_{Dh} number increases with the increase in Ri_{Dh} number. ➢ For bottom heaters differences in Nu_{Dh} numbers decrease for the first four rows of heaters with the increase in Ri_{Dh} number (the same not observed for other rows of **bottom** heaters and for all rows of the **top** heaters) ➢ The fact that the heat transfer enhancement is largest for low Reynolds numbers, suggests that heat transfer may be enhanced due to buoyancy-induced flow by reducing the flow rate and hence the ventilation power requirements. ➢ The increase in Re_{Dh} number improves the heat transfer, $(Nu_{Dh})_{avr}$ for all values of H/W. ➢ Heater temperature decreases importantly by increasing Re_{Dh} number. ➢ For top heaters high values of temperatures are obtained at low values of Re_{Dh} number.	➢ Buoyancy affected secondary flow is more effective at the greater values of H/W ratios. ➢ Convection heat transfer effects are more important for the first four rows of bottom heaters. ➢ A device placed on the top wall will realize temperatures much higher than that of the lower wall. ➢ Electronic components with the greatest power dissipation should be placed on the first rows at the bottom and top walls. ➢ For the conditions of this study, top heaters are more affected by the forced convection flow. ➢ Some important effects of the secondary flow can be seen for small Reynolds numbers and low aspect ratios, these effects vanish for high values of aspect ratios.
A. Dogan et al.[3]	➢ 32 flush-mounted heat sources in 8×4 array form. ➢ Channel, AR=6 ➢ $955\leq Re_{Dh}\leq 2220$ ➢ Gr=1.7×10^7 to 6.7×10^7 ➢ Every walls are maintained insulated and adiabatic	➢ Surface temperatures increase with increasing Grashof number ➢ The row-averaged Nusselt numbers first decrease with the row number and then increase towards the exit as a result of heat transfer enhancement. ➢ As the Reynolds number is decreased for a given Grashof number, heat transfer enhancement is obtained for the last rows of the channel ➢ Nusselt number variations for the first rows show a forced convection thermal entry region characteristic.	➢ Buoyancy affected secondary flow and the onset of instability causes heat transfer augmentation above forced convection limit ➢ Electronic components with the greatest power dissipation should be placed at the inlet and outlet sections of the channel. ➢ Low power dissipation components should always be placed around the middle section. ➢ Heat transfer augmentation due to buoyancy-induced flow reduces the flow rate followed by ventilation power consumption.

		➢ Heat transfer enhancement is largest for low Re numbers	
S. Chen, Y. Liu [15]	➢ 9 heated resistors in 3×3 array form. ➢ Spacing ratio ➢ Rectangular chennel	➢ Better thermal performance could be obtained when the center-to-center distances between the resistors follow a geometric series. ➢ At Re = 800, when the spacing ratio (s_3/s_2) among the heated resistors is 1.8, the highest temperature decreases by 8.24% and the temperature difference among resistors can reduce about 27.62% compared to that of the equi-spaced arrangement.	➢ The temperature distribution strongly depends on the spacing arrangements.

III. CHANNEL WITH DISCRETE HEAT SOURCE TRIANGULAR SECTIONAL BAR AS VORTEX GENERATOR

A summary of the important numerical investigations of discrete heat sources in different array form representing electronic chips for a 2-D laminar flow with Triangular cross sectional bar in a plain duct is presented in Table 2. Several researchers studied numerically to improve the heat transfer rate through a rectangular channel with discrete heat sources of uniform heat flux inserting a triangular cross-sectional bar to create turbulence in the flow using air as working fluid. S. Alahyari Beig [2] applied Genetic algorithm combined with Gaussian Process to find the optimum location of vortex generator and tried to maintain a uniform heat transfer rate above each of the discrete heat sources, considering steady state forced convection heat transfer. Hakan F. Oztop [5] also performed similar task with different Re but did not performed optimization and obtained the position of triangular cross sectional bar where higher heat transfer rate has been obtained.

Table. 2 Summary of important investigations of discrete heat source in array form in 2-D laminar flow

Author	Parameters	Observations	Remarks
S. Alahyari Beig et al. [2]	➢ 3 blocks attached to the bottom wall (i.e. electronic chips) ➢ Re = 100, 400, 800,1200 ➢ Triangular bar location is varied, Four cases (X = 4.6, Y = 0.36 and X = 9.85, Y = 0.44) with Re = 400, 1200 ➢ All the walls are	➢ The optimization results show that the greater value of the standard deviationmultiplier, the more uniform Nusselt numbers. ➢ The optimum location of vortex generatoris seen to be above the first block for which uniformity is neglected ➢ The optimal locations of triangular bar for different value ofReynolds number are almost the same. ➢ Hypotenuse of the triangular bar plays an important role to create flow vortices.	➢ A well trained GP can accurately predict the Nusselt number of each block separately which matches very well with data obtained from the outputs of Navier–Stokes solver. ➢ GP could better manage missing data ➢ GP needs less amount of training information for a particular amount of error, in comparison to other prediction tools such as Artificial Neural Network. ➢ The optimal position of the vortex generator is independent of the Reynolds number.

	maintained isothermal		➢ The structure of flow pattern is not related to Reynolds number, it is just a function of the position of vortex generator.
Hakan F. Oztop et al. [5]	➢ Triangular cross-sectional bar position ➢ 400≤ Re ≤ 1300 ➢ Both the top wall of the channel and the bar were isothermal	➢ Insertion of a triangular cross-sectional bar enhances the heat transfer for all Reynoldsnumbers ➢ Best heat transfer was observed for the position of the bar with y=3.5. ➢ Insertion of a triangular bar affects the flow and temperaturedistribution and the flow deviates to the top wall of the channel andblocks for the position of y=3.5. ➢ When it is located to the top wall of the channel, the flow onlyimpinges on the blocks and the heat transfer at the channel exitincreases.	➢ Location of a block to the top surface of the channel makes abetter effect at the left vertical surfaces of the blocks. ➢ The bar can be used as a passive element to control heattransfer in heated blocks inserted into channel.

IV. CHANNEL WITH DISCRETE HEAT SOURCE AND OBLIQUE PLATE AS VORTEX GENERATOR

A summary of the important numerical investigations of discrete heat sources representing electronic chips for a flow through a duct inserting oblique plates at different angles using air as working fluid is shown in table.3. Horng-Wen Wu et al. [6] accomplished a numerical investigation on heat transfer enhancement of mixed convective flow in a horizontal block-heated channel by installing an oblique plate in cross-flow above an upstream block for internal flow modification induced by vortex shedding at various oblique angles, Re, Gr in unsteady state, with air as working fluid and obtained a fruitful consequence of the mixed convective heat transfer through the channel. H. W. Wu at al. [7] also performed the same task and obtained some outputs in modified form regarding heat transfer rate.

Table. 3 Summary of important investigations of discrete heat source in array form inserting oblique plate at different angles

Author	Parameters	Observations	Remarks
6. Horng-Wen Wu et al.	➢ Oblique angle varied between 30^0 to 90^0 ➢ Reynolds number=260, 400 and 530 ➢ Grashof numbers=0, 8000 and 3200000 ➢ Pr=0.7	➢ Installation of an oblique plate in cross-flow above an upstream block can effectively enhance the heat transfer performance of mixed convection in the horizontal channel row ➢ Themaximum local Nusselt number for a given block occursat the front corner and the minimum value occurs at the groove between two blocks ➢ For three Gr/Re^2values at a fixed value of Reynolds number, the maximum increase in time-meanoverall average Nusselt number is 39.5% when theoblique angle is 60^0 with Gr/Re^2 =20	➢ Installing an oblique plate can effectively improvethe heat transfer characteristics through the modification of the flow pattern ➢ Coupling the buoyancy effects and vortex shedding has profound influences in determining the unsteady fields and heat transfer characteristics. ➢ At Gr/Re^2 = 20 the strong buoyant upflow along thevertical surfaces of the blocks interacts with the wave flows and strengthens

			these flows across the blocks
7. H. W. Wu et al.	➢ H/W=2.5 ➢ L/W=25 ➢ d/w=0.45 ➢ h/w = 0.5 ➢ Oblique angle =300, 600, 900 ➢ Re=260 to 530 ➢ Gr= 0 to 32,00,000 ➢ Pr=.7 ➢ Time increment =.0008	➢ When Gr/Re^2 is less than 0 wave flows generated by vortex shedding behind the oblique plate pasts stronger and faster across the first to the third block, then weaker and slower over the subsequent two blocks. ➢ The maximum increase in time-mean overall average Nusselt number is 39.5% when the oblique angle is 60^0 with $Gr/Re^2 = 20$ ➢ The value of average time-mean Nussel number along the block increases with increasing Grashof number for the oblique plate ➢ At $Gr/Re^2 = 20$ the strong buoyant upflow along the vertical surfaces of the blocks interacts with the wave flows and strengthens these flows across the blocks. ➢ The value of average time-mean Nusselt number along the block increases with increasing Reynolds number for the oblique plate as well as for no oblique plate	➢ Strong buoyant upflow along the vertical surfaces of the blocks strengthen the wave flows behind the oblique plate. ➢ Installing the oblique plate locally accelerates the flow past the passageway between the plate and the first block but generates different patterns of wave motion induced by vortex shedding and assist the heat transfer along the block. ➢ The strong buoyant upflow enlarges the size of both recirculation zones. On the whole, the wave flows can improve heat transfer along the block

V. INSERTING NUMBER OF HEATED CHIPS

A summary of the important investigations of discrete heat sources representing electronic chips for a flow through a duct using air as working fluid is shown in table.4. Several researchers have studied the effect of discrete heat sources mixed convective heat transfer augmentation numerically as well as experimentally using air as working fluid. Y. Luis et al. [4] numerically investigated to fin the optimum spacing for four heated chips rested on a convective substrate in a channel and solved by splitting pseudo-time-stepping finite element method for a two dimensional mixed convection around 4 heat sources mounted on a thermally conducting substrate in a channel. A. Mazloomi et al. [11] numerically studied the conductive cooling of 1 rectangular chip heated from the bottom surface, connected to a heat sink and different configurations of a highly conductive material embedded in the chip, distributed in the lower thermal conductivity media are investigated and an optimal configuration for transferring heat to the heat sink has been achieved. The respective chip which they used for study is shown in fig.1 along with the dimensions.

Satish Kumar Ajmera et al.[13] studied experimentally mixed convection heat transfer in multiple ventilated rectangular enclosure with 3 numbers of discrete heat sources at bottom. Each of the 3 heat sources has been flush mounted at the enclosure bottom and subjected to uniform heat flux.They also proposed different correlations for Nusselt number within the range of parameters considered in the study.S. Chen et al. [16] investigated experimentally the effect of different arrangements of obstacles on cooling of simulated electronic package. They used a channel formed by two parallel plates, bottom plate is attached with 5 identical electrically heated square obstacles, perpendicular to the mean airflow.

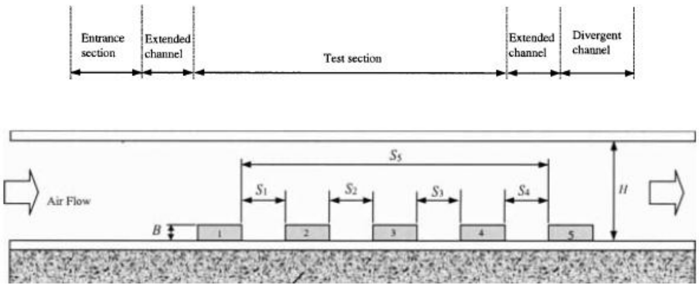

Fig.1 Test geometry of the simulated electronic package used by S. Chen et al.[16]

Table. 4 Summary of important investigations of discrete heat source in a rectangular duct both experimentally and numerically

Author	Parameters	Observations	Remarks
Y. Luis et al. [4]	➢ Rectangular cross-sectional duct=L×0.5L ➢ Inlet length=10l ➢ Test section=9L ➢ Outlet=21L ➢ Different geometry arrangements • S_1, S_5 (distance of centre of 1st& last block from inlet & outlet of test section respectively)=1.5 ,0 • S_2, S_3, S_4(spacing between each blocks)=0.618,0.71,1,1.2,1.618,1.8,2 ➢ 4 blocks	➢ Conventional equi-spaced arrangement is not an optimum option for mixed convection situation. ➢ An optimum thermal performance can be obtained when the center-to-center distances between the chips follows a geometric series. ➢ Maximum temperature and the maximum temperature difference can be decreased significantly for the ratio of 1.2 compared to 1.0 ➢ When the ratio is greater than 1.2 the maximum temperature and temperature difference have no dominant variation.	➢ Spacing between heat generated elements are important in reducing heat accumulated damage to computer chips and high density circuitry design. ➢ Temperature distribution strongly depends on spacing arrangements ➢ When the center to center distance of an element follows geometric series the accumulated heat has more space to dissipate and a better thermal performance can be achieved. ➢ Same methodology can be used to improve the TEF such as high voltage transmission cables and heat exchanger, heat sink, cooling fan within the chip set as well as PCB layout design
A. Mazloomi [11]	➢ Thickness,t=1.5 to 3.8 ➢ X_1= 2 to 8 ➢ X_2= 2 to 11 ➢ W_2=0.5 to 2 ➢ Other geometrical configurations are also considered. ➢ 1 block	➢ A considerable decrease in the maximum temperature of the rectangularchip can occur by applying an efficient configuration of the conductivechannels. ➢ The maximum temperature was reached to50.5 °C, with the assumption of constant volume fraction of k_h material,i.e. ϕ=0.11. ➢ By usinghalf of the cooling material with the optimal configuration, the chipdoes not experience a great increase in temperature.	➢ Optimal configuration can be obtained by using side branches, parallel with the main channel, and also increasing the thickness of the main channel. ➢ ➢ The application of conductive cooling may be justified more at small scales (micro- and nanoscales) where the application of convective cooling becomes inefficient. ➢ Using channels distributed in parallel with the main channel, and finally increasing the thickness of

			cooling channels the optimum performance may be achieved.
Satish Kumar Ajmera[13]	➢ Flow velocity ➢ Heat flux, 277.78 to 4444.44 W/m^2 ➢ AR (L/H)=1 ➢ $270 \le Re \le 6274$ ➢ $7.2\times10^6 \le Gr \le 5.5 \times 10^7$ ➢ Richardson number obtained in the range 0.201–571 ➢ All the walls in the enclosure are insulated and considered adiabatic. ➢ 3 block	➢ The heater nearest to enclosure inlet (heater-1) subjected lowest surface temperatures atall Reynolds numbers while the surface temperatures of heater-2 and heater-3 are almost same untilGrashof number attains a critical value. ➢ Nusselt numberis a strong function of Reynolds number while it increases sharplyat lower Richardson number but this increase is slow at higherRichardson number.	➢ The component dissipating highest amount of warmth should beplaced near the enclosure inlet. ➢ The element with the lowestheat dissipation should be placed opposite to the enclosure inletfor higher Reynolds number and all values of Grashof number. ➢ For the lower Reynolds number and Grashof number, it shouldbe placed in between.
S. Chen et al. [16]	➢ Different side to side distances of square obstacles (in mm) • S_1=19.05,8.23,5.08,11.08,8.16 • S_2=19.05,13.17,10.16,17.72,16.33 • S_3=19.05,21.08,20.32,28.35,32.66 • S_4=19.05,33.724 0.64,19.05,19.05 ➢ 5 block	➢ At Re=800, the highest temperature of the optimum arrangement could be reduced by 12% compared to equi-spaced arrangement and the maximum temperature difference among the 5 obstacles is lower than the equi-spaced arrangement by 32.1%. ➢ If the ratio is 1.6, the maximum temperature and temperature difference among 5 obstacles could be decreased significantly. ➢ When the ratio is further increased to 2, an opposite trend is observed	➢ Better thermal performance could be obtained when the side to side distances between the obstacles followed a geometric series.

VI. INSERTING DISCRETE HEAT SOURCES IN RECTANGULAR DUCT USING DIFFERENT FLUIDS AND OSCILLATING CYLINDER

A summary of the important investigations of discrete heat sources representing electronic chips for a flow through a duct not using air as working fluid and an oscillating cylinder is shown in table.4. Aziz Koyuncuoglu et al. [8] uses monolithic liquid for cooling the electronic chip in a Novel CMOS compatible micro channel heat sink experimentally under various heat flux and coolant flow rates. Wu-Shung Fu [9] numerically investigated the effect of oscillating cylinder on the heat transfer from heated blocks in a channel flow and arbitrary Lagrangian–Eulerian kinematics description method is adopted to describe the flow and thermal fields. A penalty consistent finite element formulation is applied to solve the governing equations. Mohammad Hemmat Esfe et al. [12] investigated numerically the laminar mixed convection flow of Al2O3/water nanofluids in a horizontal adiabatic channel where two hot obstacles are mounted on the bottom wall. Three thermophysical models including temperature-dependent and temperature-independent relations are selected for the study.

Table. 5 Summary of important investigations of discrete heat source in a rectangular duct using nano fluid and oscillating cylinder

Author	Parameters	Observations	Remarks
Aziz Koyuncuoglu et	➢ Heat flux(W/cm^2) • 7.86 to 8.04 (100µm 10-	➢ The micro channel heat sinks were able to extract up to 127 W/cm^2 heat flux	➢ 50 W/cm^2 heat flux in steady state continuous operation from the entire

al. [8]	channel device) • 13.1 to 13.7 (200 µm 10-channel device) • 44.8 to 49.9 (100 µm single channel device) ➢ Coolant flow rate(ml/min) • 1 to 2.8(100 µm 10-channel device) • 1 to 4 (200 µm 10-channel device) • 100 to 400 (100 µm single channel device)	froma hot spot. ➢ Heat flux values up to 50 W/cm^2were successfully removed from the entire chip surface. A singlechannel heater, simulating a hot spot on a CPU was operatedsuccessfully up to 127 W/cm^2 heat flux. ➢ For both the 100 mmand 200 mmwidechannels, the friction factors estimated by the laminar theory arelower than the experimental results.	heated surface. ➢ slightly higher hydraulic diameters may be fabricated for lower thermal resistance values ➢ The overall performance of the micro channel heat sinks is comparable with the suggested correlations from the literature and it can be improved further by optimizing the channel dimensions. ➢ The fabricated micro channel heat sinks are capable of cooling high heat flux electronic devices such as CPUs.
Wu-Shung Fu [9]	➢ Reynolds Number = 100, 250, 500 ➢ Oscillating amplitude= 0.05, 0.1, 0.2, 0.4 ➢ Oscillating frequency= 0.1,0.2,0.4	➢ Heat transfer from heated blocks is enhanced remarkably as theoscillating frequency of the cylinder is in lock-in region. ➢ The influence of oscillating amplitude on the heattransfer rate is not obvious under the lock-in region.	➢ The heat transfer rate is increased when the Reynolds number increases.
Mohammad Hemmat Esfe [12]	➢ Richardson number= 0.1,1,5,10 ➢ Rayleigh number= 10^3 to 10^5 ➢ Nanoparticles volume fraction = 0 to 0.05 ➢ Different aspect ratios of obstacles (h/H)=0.1,0.15,0.2,0.25 (h=block height, H= duct height) ➢ 2 blocks	➢ The difference between average Nusseltnumbers obtained from the three sets of thermo physical models does not exceed 3%. ➢ With increasing the nanofluid concentration from 0% to 5%, the average Nusselt number overthe obstacles increases less than 10%. ➢ The effects of various thermo physical models of nanofluids on the predicted average Nusselt number are insignificant, even for high concentration of 5%.	➢ The predicted average Nusselt number increases slightly withan increase in nanofluid concentration. ➢ The predicted average Nusselt numbers for both obstacles increase with a decrease in Richardson number for a fixed Rayleigh number. ➢ The predicted average Nusselt number decreases with an increase in height or width of the obstacles.

VII. CONCLUSION

Several researchers have work on heat transfer augmentation passive techniques both numerically and experimentally using discrete heat sources in various array forms inserting triangular cross-sectional bar and oblique plate and oscillating cylinder as vortex generator and air or nanofluids in water as working fluid. And the following conclusions are drawn from the above survey.

- Buoyancy affected secondary flow and the onset of instability causes heat transfer augmentation above forced convection limit also reduces the flow rate followed by ventilation power consumption.
- The optimum location of triangular cross sectional vortex generator is seen to be above the first block and that too does not depend upon Reynolds number
- The maximum time average Nusselt number is obtained for oblique plate vortex generator angle 60^0.
- Optimum thermal performance and uniform higher heat flux from each blocks acting as electronic chip can be obtained when the center-to-center distances between the chips follows a geometric series.
- The heat transfer rate from the electronic chip also depends upon the frequency and amplitude of the vortex generator as well as volume fraction of the nano particles in water acting as working fluid.

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