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Effect of Dimple Arrangement on The Thermal Performance of Dimpled Stepped Plate Heat Sink

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ABSTRACT

Thermal management of electronic devices is the biggest problem in the electronic industry today. Heat sinks are one such devices that are used to cater this need and addition of dimples to these devices are proved to improve the thermal performance further. In the present study, computational analysis is carried out for comparing the stepped fin heat sinks with inline and staggered dimple arrangements having same number of dimples on each fin plate at velocities in the range of 0.5m/s to 4.5 m/s for 9 cases. The results of the computational study are validated by comparing it with the experimental results of previous literature. The outcomes reveal that there is negligible difference in thermal performance of the two heat sinks at lower velocities and this gap widens with increase in velocity. Out of the two-heat sink model, heat sink with inline arranged dimples proved to have better thermal performance at all the velocities considered.

Keywords : Heat Sink, Base Temperature, Thermal Resistance, Dimples

I. INTRODUCTION

One of the biggest problems in the electronic industry today is the thermal management of electronic devices. These electronic devices are subjected to very high thermal load because of the performance requirement and operating conditions they are put in. This situation has become even more worse with the miniaturisation of electronic components and the metallurgical limitations in the materials used for electronic components. Each and every electronic component has certain level of capability to withstand the heat generated and operating beyond that limit may deteriorate the performance or may even fail the electronic component. Hence proper thermal management of electronic component is necessary for smooth and optimal performance. One such device which is used to cool the computer processor or CPU is heat sink. Heat sink (HS) are devices that act as heat transfer devices for transferring heat from the source to the surroundings. HSs are of many kinds and are used depending upon the application area. Plate fin HS, foam fin HS, radial HS, plate fin HS with additional attachments, pin fin HS etc; are some to name. Demand for the need of better cooling technology in electronic industry has attracted many researchers, in this regard there are lots of work that are being carried out in order to improve the cooling performance using HS. Some of the works are discussed below:

S N S K et. al. [1] investigated HS geometries with and without the dimples under forced convection environment in the Reynolds number span of 2000 to 30000. The results showed that inclusion of dimples to HS proved to be worthy at higher velocities and had about 21 % more heat transfer than the normal HS. Mao-Yu et.al. [2] conducted experimental and numerical analysis of aluminium fin HSs to study the effect of creation of holes at the base of the HS. Two HSs were created, one with the hole and the other without the hole and tested under natural convection environment. The results show that the thermal performance of hallow HS was much better than unhallow HS and the performance increased with increase in fin height and hole diameter with maximum heat transfer co-efficient recorded when the porosity value is less than 0.262. Awasarmol et.al. [3] conducted experimental studies on HS with perforations and holes. Analysis was carried out for different hole diameters and inclination angles under natural convection environment. The results show that HS with 12 mm perforation diameter and the inclination angle of 45^{o} had about 32 % more heat transfer coefficient than the regular fin HS with 30% savings in material. Maiti et.al. [4] investigated three HS designs computationally under forced convection environment for finding the thermal performance and pressure drop. The designs include kidney fin, solid cylinder and slotted cylinder fin geometries. Analysis was carried out at Reynolds number range of 2000 till 11000. The outcomes reveal that kidney fin HS showed better thermal performance than the other two designs. Dafedar et al. [5] experimentally studied the effect of dimpled surfaces for heat transfer augmentation. The result showed that triangular dimple had higher heat transfer rate and heat transfer coefficient in the longitudinal direction with apex of triangular dimple facing the air flow. Katkhaw et al. [6] conducted experimental analysis of flat surface with 10 different ellipsoidal dimple arrangement. The results showed that there was about 15.8% increase in heat transfer coefficient for staggered arrangement when compared with smooth surface and about 21.7% when compared with inline arrangement. Kanokjaruvijit et al. [7] compared two dimpled plates, one with hemispherical shape and the other with cusped elliptical shape. The results showed that there was no much difference in the thermal performance between the two plates. Authors suggested to use hemispherical shape based on the fabrication cost and pressure loss parameters. Performance of HSs depend on several factors such as HS design, cooling fluid used, fan capacity and the cooling fluid speed used for creating force convention environment, material used for HS fabrication etc. [8].

In the situations where the computer processors are exposed to surroundings, it’s not possible to consider any other fluid as the cooling medium. Its known fact that increasing the fan speed increases the mass flow and in turn the heat carrying capacity of the fluid. But, creating high speed cooling environment requires large fan size, which makes overall design of the electronic component bulky. Hence, these situations make way for improvement in cooling performance of HS by design changes and is the motivation for present study. In the present study, computational analysis is carried out for comparing the stepped fin HSs with inline and staggered dimple arrangements having same number of dimples on each fin plate.

II. COMPUTATIONAL METHODOLOGY

In this study, a CFD model was used to analyse the thermal performance of HSs with inline and staggered arranged dimples. The CFD analysis technique is divided into three steps. Pre-processing is the first stage, which entails creating the necessary geometrical model and generation of grid for the same. Solver execution is the next stage, which entails providing appropriate boundary conditions, making essential solver settings, and enabling the solver to complete the analysis. The findings are presented in the post-processing step, as expected.

A. Geometrical details

The dimensional details of the geometries used in the present study is shown below. All the geometries are modelled using ANSYS SpaceClaim modelling software. Figure 1 shows the geometrical model used for validation purpose. It is the standard HS model having base dimensions of 80 mm width and 80 mm length. The thickness of the base plate is 6 mm and the thickness of the fin plates are 0.7 mm. the spacing between the plate fins are maintained at 7.3 mm and the height of each fin is limited to 50 mm.

Further for the present study, geometrical models as shown in Fig. 2 and 3 are considered. Fig. 2 represents stepped flat plate HS with inline dimple arrangement and Fig. 3 represents stepped flat plate HS with staggered dimple arrangement. Both the HS has same base plate dimension of 80 mm width and 80 mm length with 6 mm thickness. The maximum height and minimum height of plates at the front and back side are kept at 60 mm and 52.5 mm respectively with three 2.5 mm steps in-between. Both the HSs have 4 mm diameter dimples on the plate fins. Inline arranged dimpled HS has dimple arrangements in order, while staggered arranged dimples HS has dimple arrangements in disorderly manner. Total number of dimples on each plate, for both the HSs are kept same and the number being equal to 60.

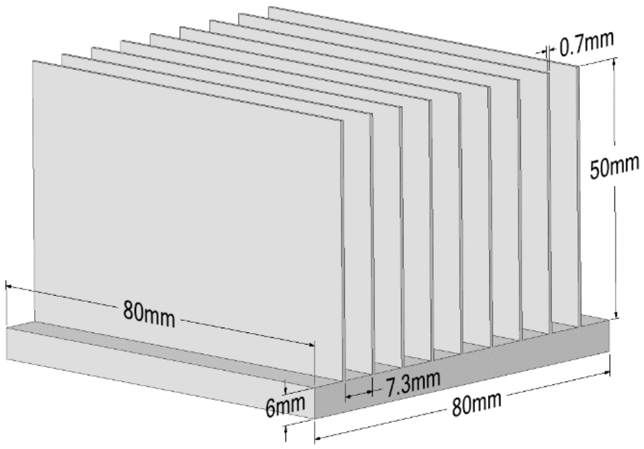

Figure 1: Standard HS

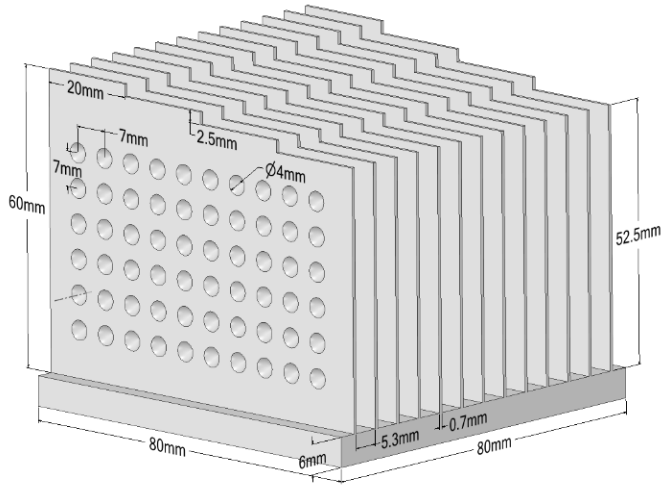

Figure 2: Stepped flat plate HS with inline dimples

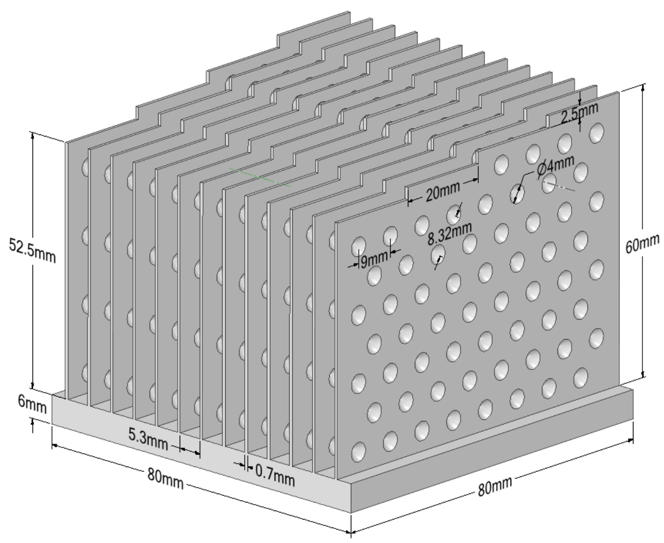

Figure 3: Stepped flat plate HS with staggered dimples

B. Meshing

Automatic 3-dimensional tetrahedral mesh is generated during the process of discretization using ANSYS AUTODYN software. Fine mesh is created at the walls of geometry and course mesh is created away from the wall and near the boundary. For the purpose of checking the accuracy of the generated mesh, orthogonal quality is used as the parameter. Every care is taken, such that the quality of orthogonality is not less than 0.1. Grid independency test is also carried out to show that the values of the results are independent of grid size. Total of 16998172 elements and 3190453 nodes are created and the minimum element size is set to 2.2e-3 after the grid independency test.

C. Boundary conditions

In the current work constant heat load of 18450 W/m² is added to HS base. The study is conducted at 9 different velocities. The intake of the test section has changing inlet velocity between 0.5 m/s to 4.5 m/s. The outlet of the test domain is given pressure outlet boundary condition and the domain outside walls are treated as adiabatic. Aluminium and air are chosen as the materials for HS and cooling fluid respectively with default solver values for density, specific heat and thermal conductivity for both the materials.

D. Governing equations

The following governing equations are solved for predicting the results of the present study.

Continuity equation is given by:

$$\nabla(\rho\vec{U}) = 0 \quad (1)$$

Momentum equation in *x, y* and *z* directions is:

$$\nabla(\rho\vec{U}u) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \quad (2a)$$

$$\nabla(\rho\vec{U}v) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} \quad (2b)$$

$$\nabla(\rho\vec{U}w) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} \quad (2c)$$

Energy equation is given by:

$$\nabla(\rho h\vec{U}) = -p\nabla\vec{U} + \nabla(\mathrm{k}\nabla\mathrm{T}) + \emptyset + S_h \quad (3)$$

Where U is fluid velocity (m/s), K is Thermal conductivity (W/m·K), h is Aggregate enthalpy, ϕ is Dissipation term and S_h is source term.

E. Calculation procedure

Thermal resistance is given by

$$R_{th} = \frac{1}{\bar{h}A_T} \quad (4)$$

Where A_T is the total cooling area and $\bar{h}$ is the average heat transfer co-efficient.

Average heat transfer co-efficient is given by

$$\bar{h} = \frac{Q}{A_T(T_b - T_m)} \quad (5)$$

Where Q is the amount of heat transferred to the cooling medium i.e., air, T_b is the base temperature of HS and T_m is the mean temperature of air.

Mean temperature of air is given by

$$T_m = \frac{(T_{avg} + T_b)}{2} \quad (6)$$

Where T_{avg} is the average temperature of air measures with respect to inlet and outlet temperature.

Average temperature of air is given by

$$T_{avg} = \frac{(T_{out} + T_{in})}{2} \quad (7)$$

Where T_{out} and T_{in} are the temperatures measured at the outlet and inlet of the test section respectively.

Amount of heat transfer to the cooling medium is given by

$$Q = \dot{m_a} C_{pa}(T_{out} - T_{in}) \quad (8)$$

Where $\dot{m_a}$ and C_{Pa} refers to mass flow rate and the specific heat of air respectively.

It is also important to calculate the pressure drop across the HS in order to evaluate the additional pumping power required.

Pressure drop is given by

$$\Delta P = P_{in} - P_{out} \quad (9)$$

Where P_{in} and P_{out} refers to inlet and outlet pressure respectively.

III.RESULTS AND DISCUSSION

Present analysis tends to compare stepped fin HS with inline and staggered dimple arrangements with same number of dimples and the outcomes of the same is summarised in the below sections.

A. Validation of computational model

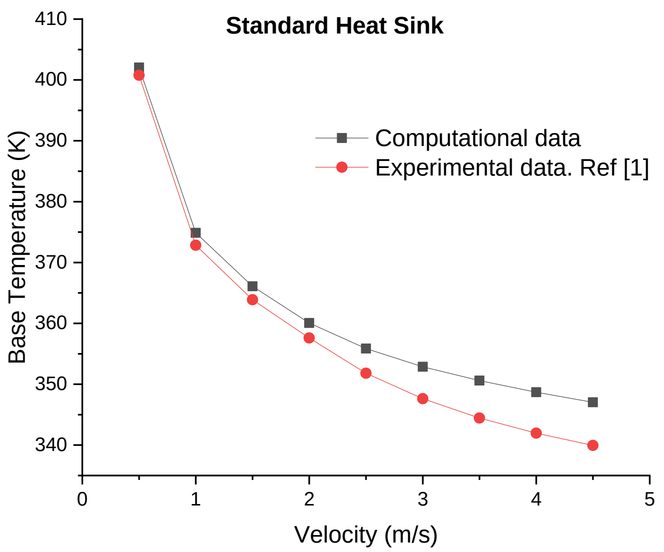

Figure 4: Validation with respect to Base Temperature

For the purpose of validation, the geometric profile (Fig. 1), boundary conditions and other operational conditions were considered exactly similar to the experimental work carried out in Ref [1]. Validation is done at different velocities by considering decrease in base temperature and thermal resistance as the performance factors for standard HS geometry. The validation results show that the authors are able to obtain the results for base temperature and thermal resistance with differences within 9% when compared with published experimental work. The possible reason for the difference in results can be because of simulation error. Since the difference in values between the two approaches is acceptable depending upon the complexity involved in simulating the experimental conditions, it can be assumed that the developed computation model is quite accurate and hence can be used effectively for the present analysis. The thermal performance validation results are shown in the Fig. 4 and 5. It can be seen that in both the figures the base temperature and thermal resistance values decreases with increase in velocity and the reasons for the same is discussed in the coming section. Also, it is observed that the deviation in results between the two approaches increases with increase in velocity. The probable reason for this is due to increase in measurement uncertainty with increase in velocity.

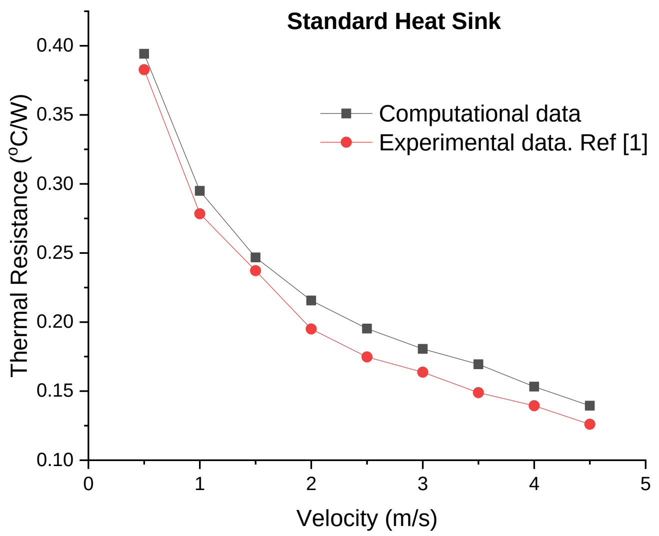

Figure 5: Validation with respect to Thermal Resistance

B. Effect of dimple arrangement on Base Temperature

Fig. 6, 7, 8 and 9 shows the base temperature contours for inline and staggered dimpled HS at 0.5 m/s and 4.5 m/s. The maximum surface temperature for both the geometries is found to be around 376 K at 0.5 m/s and 327 K at 4.5 m/s and is found to be present at the base of the HS, since it is in contact with the heat source. It is also observed that the value of surface temperature drops from base to the tip of HS. Since the cooling air first hits the front side of the HS, these surfaces experience less temperature values than the surfaces at the back side of the HS. Temperature contours show that there no much changes in temperature values between HS with inline and dimpled arrangements.

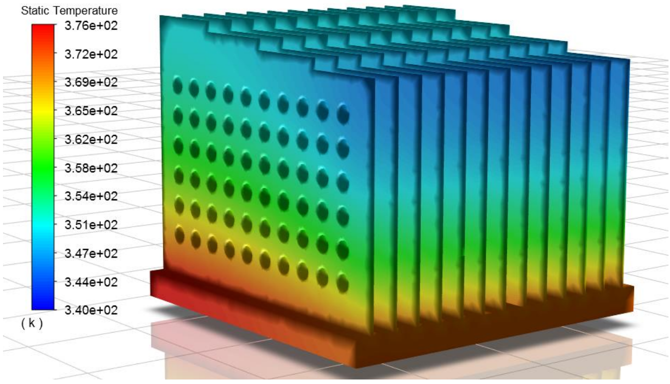

Figure 6: Base Temperature contours of inline dimpled HS at 0.5 m/s

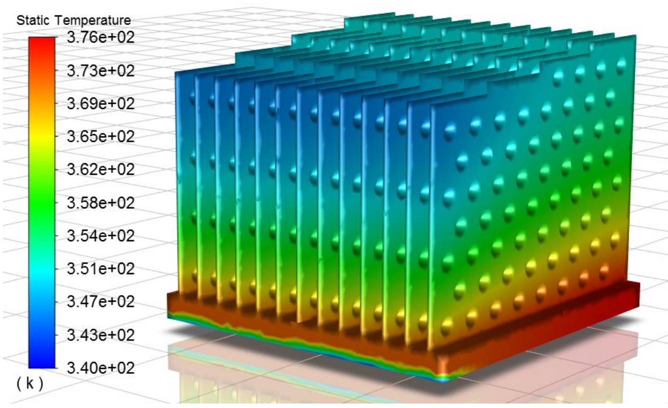

Figure 7: Base Temperature contours of staggered dimpled HS at 0.5 m/s

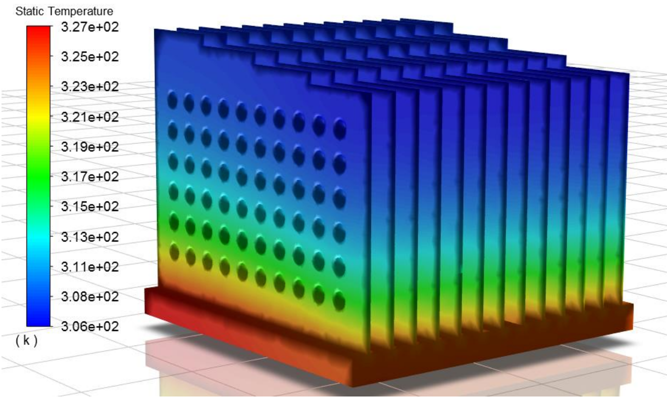

Figure 8: Base Temperature contours of inline dimpled HS at 4.5 m/s

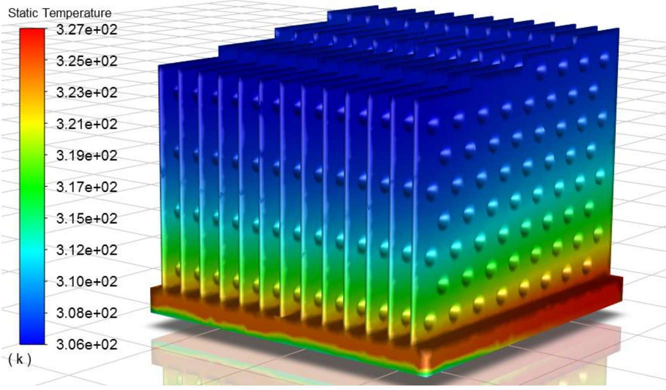

Figure 9: Base Temperature contours of staggered dimpled HS at 4.5 m/s

Fig. 10 depicts the variation of base temperature with velocity. It can be observed that for both the cases the value of base temperature decreases with increase in velocity. This is because as the velocity increases, mass flow rate of the cooling fluid also increases which in turn increases the heat carrying capacity of the cooling fluid and hence reduces the base temperature of the HS. Further it can be seen that the difference in base temperature values between the inline and staggered arranged HSs for the same number of dimples at different velocities is negligible with maximum value of difference in base temperature value being only 0.55178 K at 1.5 m/s and the percentage variation is only 0.159 %. Next, it can also be observed that, for the same number of dimples, inline arranged HS produces slightly less base temperature values than staggered arranged HS. The probable reason for the above difference is that, since the dimples are arranged in staggered manner, slight pile up of air at the boundaries of the staggered arranged dimples takes place which might be due to suppression of air flow. This piling up of air causes reduction in heat carrying capacity of the air. For better heat transfer to happen there should be no suppression of cooling fluid flow, so that it can carry maximum amount of heat. Hence, inline arranged dimpled HS produced slightly better performance than staggered arranged dimpled HS for the same number of dimples.

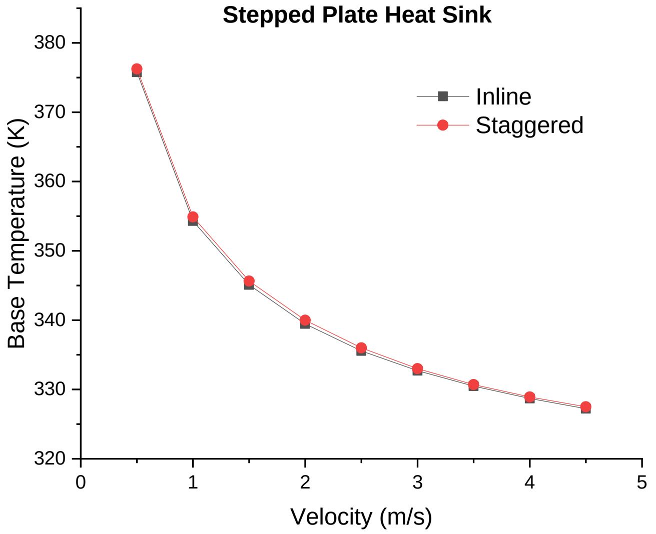

Figure 10: Variation of base temperature with velocity

C. Effect of dimple arrangement on Pressure Variation

Fig. 11 shows the variation of pressure drop with velocity. It can be seen that the pressure drop value increases with increase in velocity for both the cases. This is because as the velocity increases energy loss also increases which happens due to friction. This loss in energy results in drop in pressure. Further it can be seen that the pressure drop difference value is negligible at lower velocities and the pressure drop gap widens with increase in velocity. The maximum difference is to be found at 4.5 m/s and the corresponding pressure drop difference is around 7.65 %. Next it can be seen that inline arranged dimpled HS experiences slightly less pressure drop than staggered arranged dimpled HS. The probable reason is that staggered arranged dimpled HS supresses the flow of cooling fluid more when compared with inline arranged dimpled HS. This flow suppression results in more pressure drop and hence inline arranged dimple HS experiences less pressure drop as compared to staggered arranged dimpled HS for the same number of dimples.

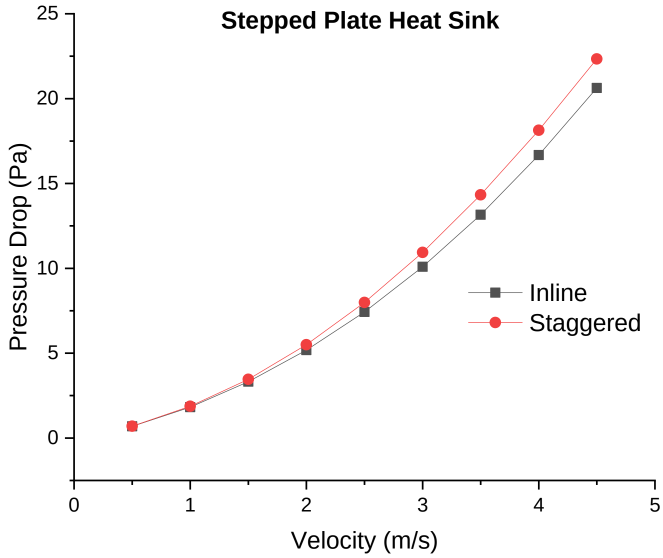

Figure 11: Variation of pressure drop with velocity

D. Effect of dimple arrangement on Thermal Resistance

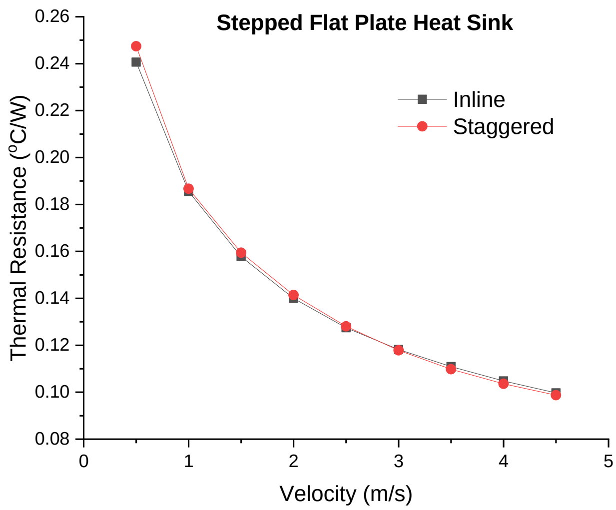

Figure 12: Variation of thermal resistance with velocity

Fig. 12 depicts the variation of thermal resistance with velocity. It can be seen that the thermal resistance value decreases with increase in velocity for both the cases. This is because of the increase in heat transfer with increase in velocity. Further it can be observed that the difference between the thermal resistance for both the cases is negligible at lower velocity and the difference increase as the velocity increases with maximum value being equal to 0.98 % at 4.5 m/s. Also, it can be observed that inline arranged dimpled HS has less thermal resistance than its counterpart. The probable reason is that, inline arranged HS produces more heat transfer when compared with staggered arranged HS.

IV.CONCLUSION

In the present analysis the thermal performance of the stepped HS with inline and staggered dimpled arrangements having same number of dimples are analysed computationally and the computational model is validated by comparing the computational results with the experimental results available in the literature. The following conclusions can be drawn from the above study.

V. REFERENCES

[1]. S N, S.K., S, S., H R, P. (2021). Heat transfer analysis of plate fin heat sink with dimples and protrusions: Investigation of new designs. International Journal of Heat and Technology, Vol. 39, No. 6, pp. 1861-1870. https://doi.org/10.18280/ijht.390621.

[2]. Wen, MY., Yeh, CH. Natural convective performance of perforated HSs with circular pin fins. Heat Mass Transfer 51, 1383–1392 (2015). https://doi.org/10.1007/s00231-015-1504-8

[3]. Umesh V. Awasarmol, Ashok T. Pise, An experimental investigation of natural convection heat transfer enhancement from perforated rectangular fins array at different inclinations, Experimental Thermal and Fluid Science, Volume 68, 2015, Pages 145-154, ISSN 0894-1777, https://doi.org/10.1016/j.expthermflusci.2015.04.008.

[4]. Maiti A., & Prasad, B. (2017). Alternative HS to enhance thermo-hydraulic behaviour of an array of short pin fins. Proceedings of the World Congress on Mechanical, Chemical, and Material Engineering. doi: 10.11159/htff17.121

[5]. Dafedar, Mohammad. (2013). Heat Transfer Augmentation In Different Geometries Of Dimpled Surface Under Natural Convection An Experimental Approach. International journal of research in engineering and technology. 02. 937-940. 10.15623/ijret.2013.0206004.

[6]. Nopparat Katkhaw, Nat Vorayos, Tanongkiat Kiatsiriroat, Yottana Khunatorn, Damorn Bunturat, Atipoang Nuntaphan, Heat transfer behavior of flat plate having 45° ellipsoidal dimpled surfaces, Case Studies in Thermal Engineering, Volume 2, 2014, Pages 67-74, ISSN 2214-157X, https://doi.org/10.1016/j.csite.2013.12.002.

[7]. Koonlaya Kanokjaruvijit, Ricardo F. Martinez-botas, Jet impingement on a dimpled surface with different crossflow schemes, International Journal of Heat and Mass Transfer, Volume 48, Issue 1, 2005, Pages 161-170, ISSN 0017-9310, https://doi.org/10.1016/j.ijheatmasstransfer.2004.08.005.

[8]. SN, S. K. (2019). Recent developments in the design of heat sinks for heat transfer enhancement–A review. International Journal of Science, Technology, Engineering and Management-A VTU Publication, 1(4), 1-8. http://ijesm.vtu.ac.in/index.php/IJESM/article/view/464

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