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Heat Transfer Enhancement of Automobile Engine Fin by Thermal Boundary Layer Disruption

Solanki Dhaval T[1], Prof.Vivek Joshi[2]

[1]*Mechanical department, Parul institute of engineering and technology* [2]*Mechanical department, Parul institute of engineering and technology*

Abstract-*Present research paper suggests geometrical modification methods to improve heat transfer rate of automobile air cooled engine fins by disruption of thermal boundary layer. Thermal boundary layer plays vital role in heat transfer from fins and act as barrier in convective heat transfer. So many methods and designs of geometry available to break thermal boundary layer, use of vortex generator, perforation on fin surface, and use of wavy fins are some of examples. Review of different research paper presented on thermal boundary layer disruption and heat transfer enhancement are presented here. Effect of louvered shape fin on heat transfer and effect of different geometrical parameters and integration with vortex generator are also reviewed.*

Keywords-*Thermal Boundary layer, Disruption, Vortex generator, louvered shape, pitch, angle of attack, heat transfer co-efficient.*

I. INTRODUCTION

Automobile sector grows rapidly and subjected with an expected problem of fossil fuel crises, as a result of such problems researches rush towards alternative power trains and alternative fuels and its development in the existing technology is also that much important. To reduce fuel consumption rate and to increase performance rather than diverted towards the new power trains and fuels we are still able to develop conventional Internal combustion engines. Due to its compactness and light weight, air cooling system is still used in automobile engines. Using air as a cooling media and convection heat transfer phenomenon, it is cheaper to use and less maintenance then liquid cooling system which is comparable complex and bulky. Still there are limitation regarding lower cooling rate of air cooled engine in high speed bikes and vehicles. Fins are not that much effective as liquid cooled system, to improve heat transfer rate of air cooled engine and to overcome problems regarding heat transfer rate researchers have done so many researches with fin materials and geometry.Thermal boundary layer is one of the barrier which affect heat transfer enhancement and as thermal boundary layer growth and its thickness increases heat transfer rate decreases. Besides of fin material this is most affecting parameters, it is necessary to break thermal boundary layer formed on the surface of fin in order to enhance heat transfer rate. Present review paper shows different methods to disrupt thermal boundary layer used by different researchers. Vortex generator, perforation on the surface of fin, sudden disturbances provide by sudden geometrical changes as in louvered fin shape, use of louvered shape with vortex generator, use of wavy shaped fins and staggered arrangements rather than inline are effective methods to enhance heat transfer of fin.

II. EFFECT OF THERMAL BOUNADARY LAYER ON CONVECTIVE HEAT TRANSFER

The region of the fluid next to the surface in which energy exchange is occurring is known as thermal boundary layer. Thermal boundary layer develops if the free stream and surface temperatures differ. At the leading edge, the temperature profile is uniform, with $T(0,y) = T_\infty$. Fluid particles that come into contact with the plate achieve the plate's surface temperature, T_s. In turn, these particles exchange energy with those in the adjoining layer, and the temperature gradients develop in the fluid.

The region of the fluid in which these temperature gradients exist is the thermal boundary layer, and its thickness δt is typically defined as the value of y for which the ratio $[(T_s-T)/(Ts-T_\infty)] = 0.99$. With increasing distance from the leading edge, the effects of heat transfer penetrate further into the free stream, and the thermal boundary layer grows in a similar manner as does the hydrodynamic boundary. At any distance x from the leading edge, the local heat flux may be obtained by applying Fourier's law to the fluid at y =0 in terms of the thermal conductivity of the fluid, k, and the temperature gradient at the surface.

$$q"_s = -k\frac{\partial T}{\partial y}\bigg|_{y=0}$$

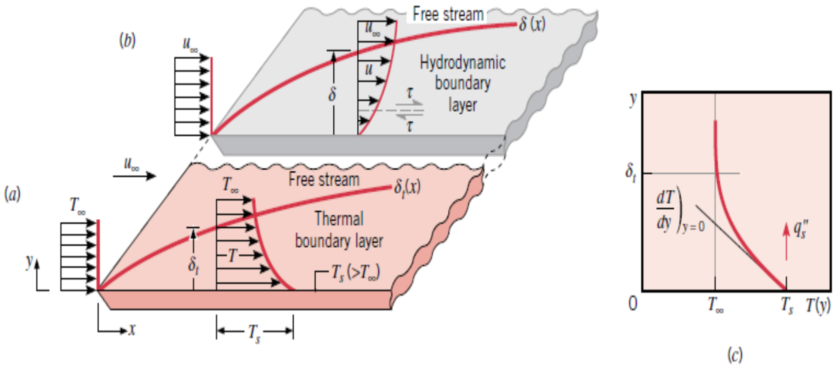


Figure 1. Fluid with the uniform free stream velocity u_∞ temperature T_∞ in laminar flow over a flat plate with uniform temperature T_s(T_s>T_∞).

This expression is appropriate because at the surface, the fluid velocity is zero (no-slip condition) and energy transfer occurs by conduction. Recognize that the surface heat flux is equal to the convective flux, which is expressed by Newton's law of cooling.By combining the foregoing equations, we obtain an expression for the local convection coefficient

$$q''_s = q_{con} = h_x(T_s - T_\infty)$$

$$h_x = \frac{-k\partial T/\partial y|_{y=0}}{T_s - T_\infty}$$

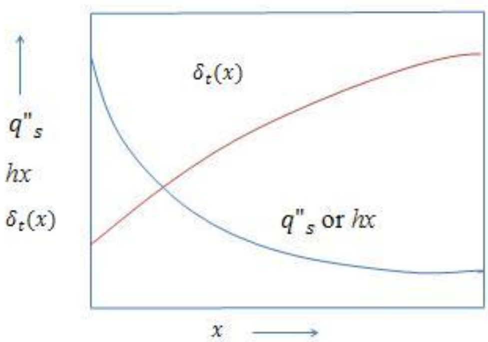


Figure 2. Heat transfer co-efficient, heat flux, and boundary layer thickness varies with longitudinal distance x.

The structure of the flow in the hydrodynamic boundary layer can undergo a transition from laminar flow near the leading edge to turbulent flow. The thermal boundary has flow characteristics and temperature profiles that are a consequence of the hydrodynamic boundary layer behaviour. In the laminar region, fluid motion is highly ordered and characterized by velocity components in both the x- and y-directions. The velocity component v in the y-direction (normal to the surface) contributes to the transfer of energy (and momentum) through the boundary layer. The result in temperature profile changes in a gradual manner over the thickness of the boundary layer.

At some distance from the leading edge, small disturbances in the flow are amplified, and transition to turbulent flow begins to occur. Fluid motion in the turbulent region is highly irregular and characterized by velocity fluctuations that enhance the transfer of energy. Due to fluid mixing resulting from the fluctuations, turbulent boundary layers are thicker. Accordingly, the temperature profiles are flatter, but the temperature gradients at the surface are steeper than for laminar flow. Consequently, the local convection coefficient is larger than for laminar flow, but to decrease with x as shown in figure.

Thus from above discussion it is clear that thermal boundary layer is as its thickness increases reduces convective heat transfer co efficient which overall decrease heat transfer rate and act as an obstacle in convective heat transfer. To increase heat transfer in convective viscous flow it is necessary to break thermal boundary layer for the given surface flow. In this study different methods for thermal boundary layer disruption are explained.

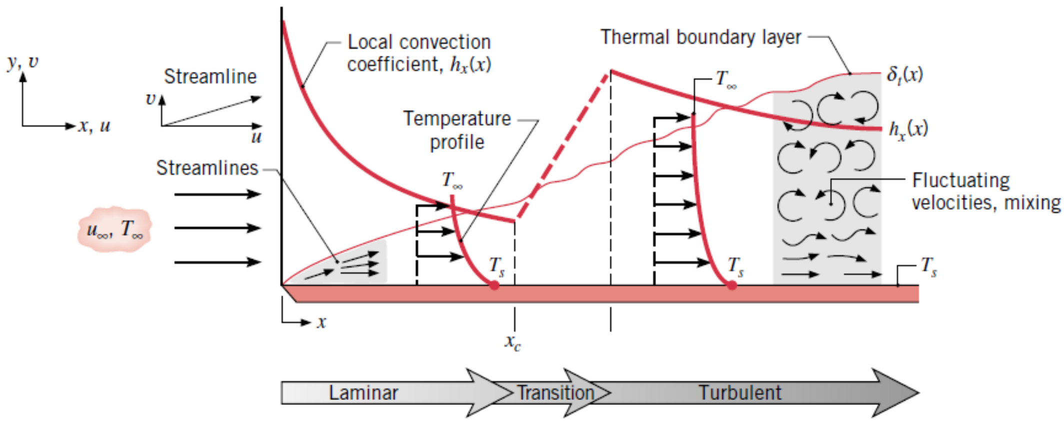


Figure 3. Thermal boundary layer development on flat plat showing changes in fluid temperature profile and local convection co efficient in laminar and turbulence flow regions.

III. PAST RELEVANT STUDIES

Fengming Wang et al [1] have studied heat transfer and pressure drop of different shaped pin fins in rectangular channel and they have found that Heat transfer enhancement of drop-shaped pin fin is weaker than of circular pin fin. The reduction in average Nusselt number between the drop shaped and circular pins was about 24% , 26%, 27% for three drop shape fins having reducing thickness respectively. Thus results shows that drop shape fins are more advantageous aerodynamically but circular pin fin shows more heat transfer and pressure drop penalty.

J.Cui and D.K.Tafti[2] found that with three dimensional multilouvered fin geometry with three different region angle louvered, transition region and flat region. Heat transfer enhancement shows increase 50% at the junction due to strong acceleration, when compared to the geometry in which angled louver-extends all the way to the tube surface.

Octavio A. Leon et al [3]found that staggered arrangement is aerodynamically less efficient than standard inline arrangement for same incoming velocity. They have expressed aerodynamic efficiency as the ratio of removed heat and the energy spent to drive the coolant flow through fins. For the equal cooling rat, the aerodynamic efficiency of rounded staggered cooling fin can be a factor 2.80 higher than for rectangular in-line cooling fins. Staggered rounded arrangement has maximum surface heat flux 596.43 W/m^2 at pressure drop of 11.278 Pa.

Xiaoze Du, LiliFeng et al [4] studied longitudinal vortex generator punched on wavy finned flat tube Heat exchanger. They have used four types of longitudinal vortex generator and studied numerically with experimental verification. The delta winglet pair is the best longitudinal vortex generator for air side heat transfer enhancement among the four configurations. The average PEC (Performance Evaluation Criteria) of the delta winglet pair with angle of attack 25^0 can reach to 1.23 while the inlet air velocity varies from 1 m/s to 5 m/s.

Junxiang Shi, Jingwen Hu et al [5] found that vortex induce vibration can be effectively used to disrupt thermal boundary layer to enhance heat transfer rate. Analytical study of channel in terms of vortex dynamics, disruption of thermal boundary layer, local and average Nusselt number and pressure loss. Four channels without cylinder, with stationary cylinder, stationary cylinder with plat, and a stationary cylinder with flexible plat are studied. A channel with flexible plat cylinder was performing well compare to others and enhances the heat transfer by thermal boundary layer disruption and found that Nusselt number is higher for the channel number four having cylinder with flexible plat at Reynolds number 327.7, mean velocity 4 m/s we find out maximum Nusselt number 4.65.

Li Li, Xiaoze Du et al [6] studied heat transfer for flat fins, flat plat continues, and continues wavy fins, result shows that for Reynolds number between 440 and 2800, which corresponds to face inlet velocity between 1 m/s and 6 m/s, the Nusselt numbers are shown to increase by 3-22% for multi period convergent divergent channel for of wavy fins compare to straight channel.

Wu Xuehong, Zhang Wenhui et al [7] carried out study on composite fin based on advantages of longitudinal vortex generator and slit fin respectively. The performance of air side heat transfer and fluid flow is investigated by numerical simulation for Reynolds number ranging from 304 to 2130. Conclusion shows that composite fin, while compare with plain fin the performance of a heat transfer of composite fin can improve 77.16% -90.21% and Nusselt number increase with increasing of Reynolds number. The overall performance of composite fin can be improved 26.18%-50.89% and 45% -14% respectively for composite fin while compare with plain fin and slit fin.

B Ameel, J Degroote, H Huisseune et al [8]In this study, a round tube and fin geometry with individually varying louver angles is analysed. A laminar and steady calculation was performed, with symmetric boundary conditions for the Reynold number on the hydraulic diameter (ReDh) of 535. It is shown that the use of individually varying louver angles allows increasing the Colburn j factor by 1.3% for the same friction factor, with respect to the optimal uniform louvered fin configuration.

Manjunath S N, S.S.Mahesh Reddy, Saravana Kumar [11]In this study the effect of louver angle and louver pitch geometrical parameters, on overall thermal hydraulic performances of louvered fins is studied. The effects of geometric parameters have been investigated by considering heat transfer and friction loss of louver plate fins in terms of Stanton number and friction factor. The study is performed within the range of Reynolds number from 100 to 1000. Good agreement is observed as far as Stanton number with the deviation in heat transfer of 1% up to 12%. Maximum heat transfer co-efficient and minimum pressure drop achieved at angle 20 degree of louvered fin.

Shivdas S. Kharche1, Hemant S. Farkade[12]have used notch in centred of the fin. Copper is used as a fin material for the experimental work. From the experimental study it is found that the heat transfer rate in notched fins is more than the un-notched fins. The average heat transfer coefficient for without notched fin is 8.3887 W/m2K and for 20% notched fins it is 9.8139 W/m^2K.

Paul A. Sanders, Karen A. Thole[15]The experimental study presented in this paper concentrates on augmenting the heat transfer along the tube wall of the compact heat exchanger through the use of winglets placed on the louvers. The experiments were completed on a 20 times scaled model of an idealized louvered fin exchanger with a fin pitch to louver pitch ratio of 0.76 and a louver angle of 27^0. The Reynolds numbers tested, based on louver pitch, were between 230 and 1016. Heat transfer augmentation increased with increasing angle of attack, increasing winglet size (decreasing aspect ratio in this case), and with decreasing winglet distance from the wall.

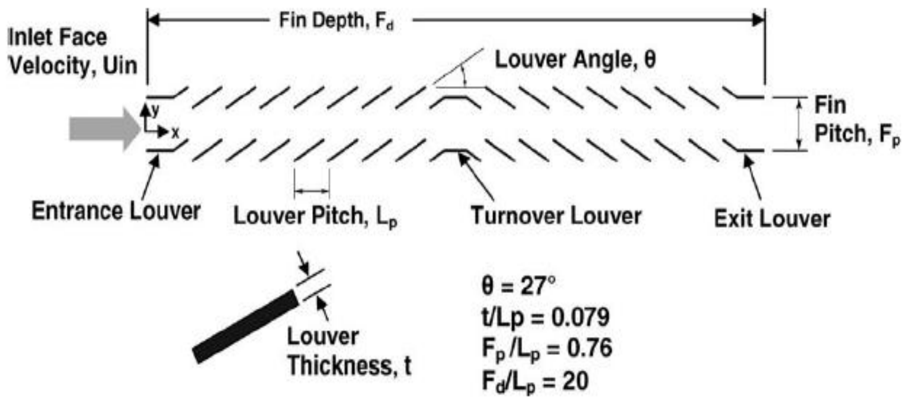


Figure 4. Louvered array geometry and definitions used in this study

Geometrical parameters and configuration of louvered fins are showed in figure 4 and placement of winglet on surface of fin and its configuration are showed in figure 5.For an angle of attack of 20^0, the results showed little augmentation; however, angles of attack greater than or equal to 30^0 performed very well. An aspect ratio of 3 did not provide much benefit while an aspect ratio of 1.5 produced high augmentations. Doubling the surface area of a winglet to make rectangular winglets increased the heat transfer augmentation. The best heat transfer augmentation was found with rectangular winglets giving 38%, 36%, and 3% at Reynolds numbers of 1016, 615, and 230, respectively

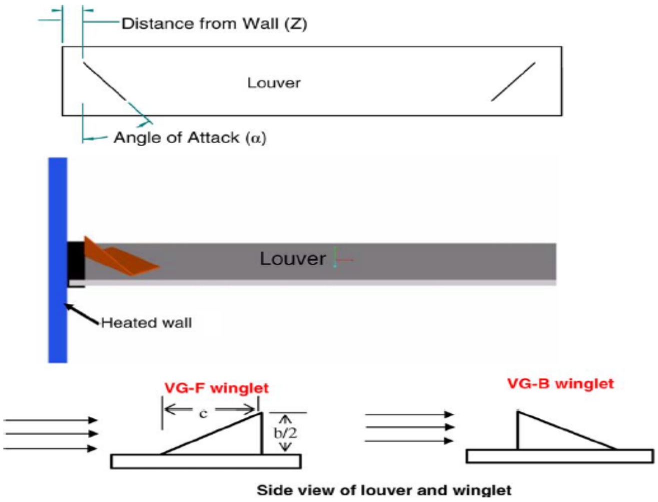


Figure 5. Definition various winglet parameters

IV. NUMERICAL INVESTIGATION

Numerical simulation has been carried out on four louvered fin geometry models made in Creo parametric 2.0. Four models include straight rectangular fin, straight circular fin, Louvered fin array, and louvered fin array with vortex generator. Numerical simulation has been performed on these models with the help of ANSYS 15.0 Fluent tool. Instead of full engine assembly, half cylindrical shape with two fin array was used for simulation in order to decrease computational burden on hardware. As shown in figure louvered fin array consists one flat fin at upstream flow and one at downstream flow in order to stabilize flow and domain selected for air has been extended up to 200 mm for rear and front to avoid air recirculation. Louvered array consists one turn over louver in middle region of array which allows the fluid flow to change its direction. Total four flat fins, eight louvered fins, and two turnover fins are used in simulation work in two fin array.

4.1 Governing Equations

To examine the mathematical consequences of the boundary layer concept we begin with the governing equations for a simplified case. We make the following assumptions: (1) steady state, (2) two-dimensional, (3) laminar, (4) uniform properties, (5) no dissipation, and (6) no gravity. Based on these assumptions the continuity, momentum, and energy equations are

Continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0$$

Momentum equation for x direction:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial u}{\partial x}+\vartheta\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)$$

Momentum equation for y direction:

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial v}{\partial x}+\vartheta\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)$$

Energy Equation:

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)$$

These equations will be simplified using the boundary layer concept.

4.2 Geometrical Models

Geometry of louvered shape fin array on semi-circular cylindrical shape is created in ANSYS geometry module using design parameters list down in table, all parameters are in mm and geometry contains basic louvered shape fin array on semi cylindrical shape with vortex generator on its surface. Vortex generators are of delta winglet type.

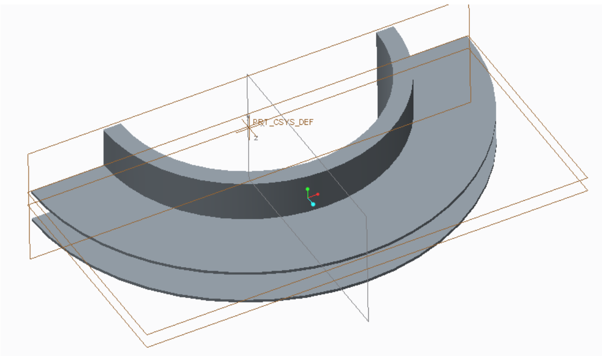


Figure 6. Circular Fin Model

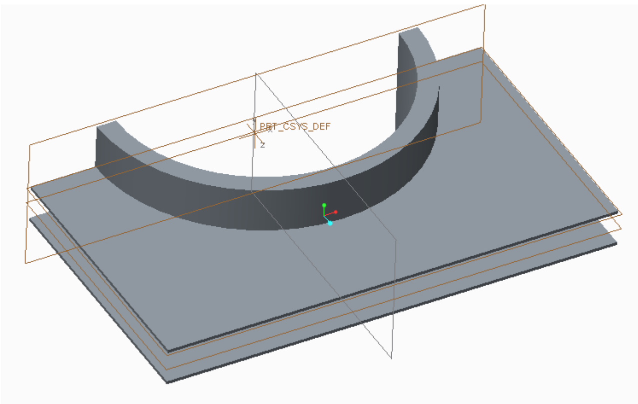


Figure 7. Rectangular Fin Model

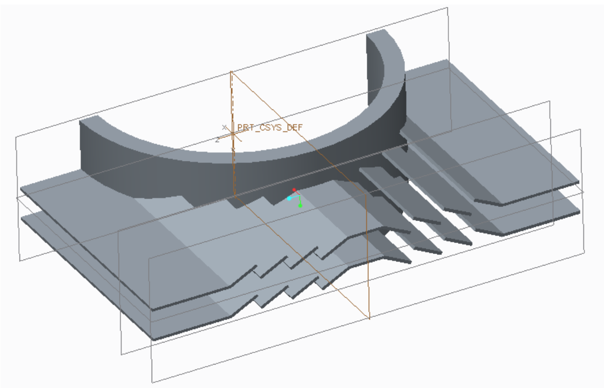


Figure 8. Louvered Fin Model (Without vortex generator)

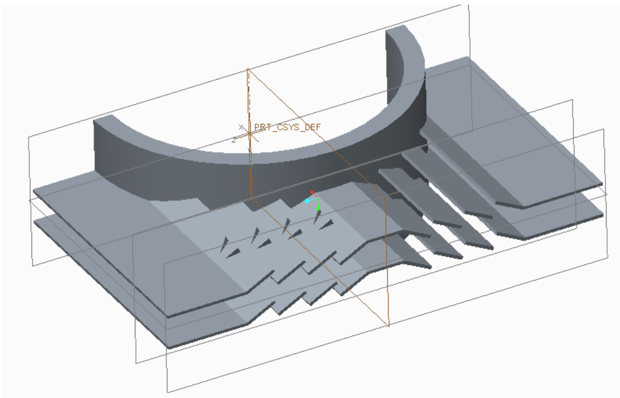


Figure 9.Louvered fin Model (With vortex generator)

Shape	**Parameters**	**Value**
Semi cylinder	Inner Diameter (mm) Test section Height (mm)	Bore-88 mm Base-80 mm
Louvered Fin	Length (l) Angle of Twist of louvered (θ) No of Fins (n) Fin Pitch (Fp) Louvered Pitch (Lp)	82.02 mm From center plane (For t=1mm and 2 mm) 20^0-30^0 louvered angle With 2^0 step 6 Louvered fins+ 1Turnover Fin 7.6 mm 12.65 mm
Delta Winglet pair	Length (c) mm Height (b) mm Angle of Attack (α)	Aspect ratio 1.5,2,3 are defines as ratio 4b/c. 20^0-40^0 With Step 5^0

Table 1. Geometrical Parameters of Fin and Winglet

4.3 Boundary Conditions and Meshing

In this case tetrahedron mesh is selected which contains unstructured mesh elements for open air domain which encloses the geometry as well as fin geometry. Care should be taken to use the unstructured tetrahedron cells to an extent as high as possible. Meshing of all four models have been done with tetrahedron patch conforming method with proximity and curvature which generates 29,35,394 elements and 44,86,294 nodes. And with maximum skewness 0.85, which is within the limit of 0.98. Statistic of elements and skewness shows a good quality of mesh.

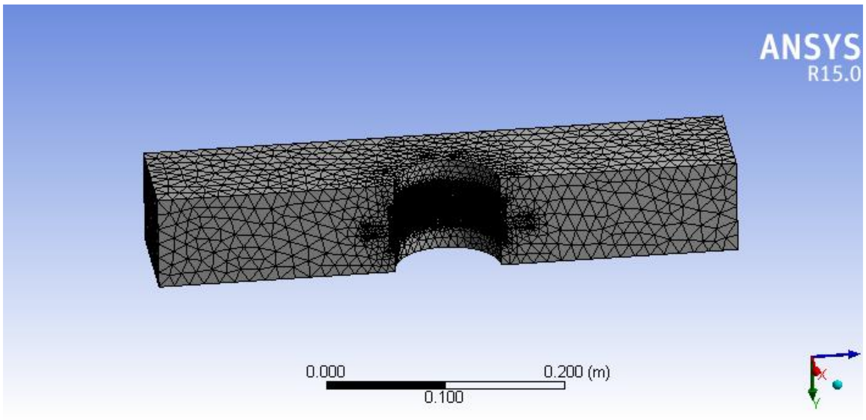


Figure 10. Meshing of domain and Fin Assembly

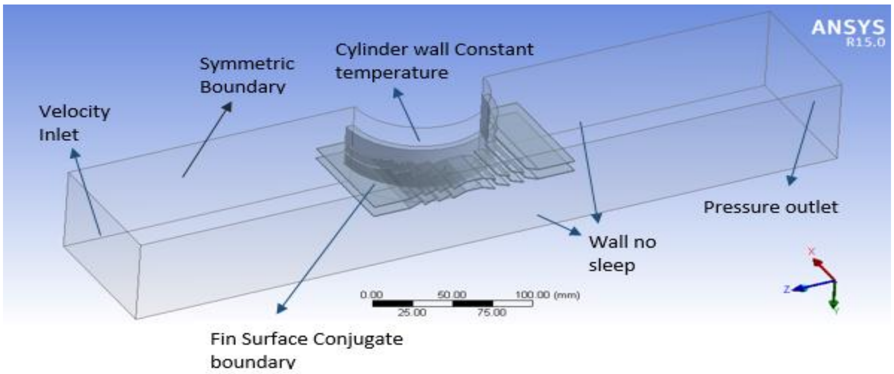


Figure 11.Boundary condition Domain and fin

For the given problem inlet and outlet conditions are given as per problem requirement of forced convection. Boundary conditions given to encloser and geometry wall are as listed in bellow table.

Table 2. Boundary Conditions (Thermal and Hydrodynamic)

Boundary condition	**Parameter**	**Surface/ Direction**	**Value**
Inlet	Velocity,	Flow stream direction. Air inlet	10,15,20,25 m/s
	Temperature		27^0C (300K)
		Air Domain	
	Turbulence		5% Medium
Outlet	Gauge Pressure	Outer face	0 Pa
Wall	No sleep	Top, side, bottom and geometry surface	-

Base temperature	Temperature	cylinder wall	727^0C (1000K)

4.4 Problem Setup

The type analysis is forced convective heat transfer type. The velocity and pressure are changed to absolute and heat is transfer from constant temperature semi cylinder wall to fins by conduction and from fins to it transfer to surrounding air by forced convection.

Energy is set - ON position in Fluent Setup. As we prefer to analyse turbulent flow, so the viscous model is selected: "k-ε" model (2 equations).

Material properties have been changed to the properties of Aluminum alloy 6061 which is used as a fin material in engines in most of two wheelers. Specific heat remains constant and evaluated at mean air temperature and bouncy and radiation effect is neglected. Governing equations are solved based on finite volume technique. Second order upwind scheme used to calculate convective flow on boundary surface of control volume. This second order scheme is least sensitive to mesh structure.

Table 3. Material Used for Fin

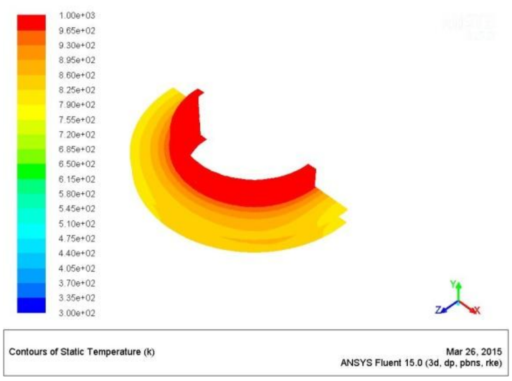
Material	**Composition Wt %**	**Thermal Conductivity**	**Density Kg/m^3**	**Film Co-efficient W/mm^2**	**Ultimate Tensile Stress Mpa**	**Poisson's Ratio**	**Specific Heat kj/kg^{0}C**	**Yield Stress Mpa**
All Alloy 6061	Al 95.8-98.6 Mg 0.8-0.2 Si 0.4-0.8 other each max 0.005	167	2700	0.000083	310	0.331	1.256×10^3	276

V. RESULTS AND DISCUSSION

Simulation work has been carried out for two fin arrays which are mounted on semi-circular base which here act as an engine base. Simulation work has been done on conventional straight rectangular and circular fin model at varying wind velocity from 10 m/s to 25 m/s which is equivalent to 70- 80 km/h speed of bike. Simulation work carried out on louvered fin model without vortex generator for varying louvered angles between18^0-33^0 for given velocity range as mentioned above. Model with vortex generator are tested for varying angle of attack of winglet pair between 25^0-45^0 for aspect ratio 1.5, 2, and 3 for winglet pair.

5.1 Results Comparison for Varying Louvered angles With flat Rectangular and Circular Fins

Graph shows that up to velocity of 20 m/s all louvered angle gives same heat transfer increment, but after 20 m/s velocity all angle gives change in heat transfer rate which is in lower range of deviation with each other. Among these entire angles 30^0 Louvered fin shows maximum deviation and gives heat transfer rate maximum from cylinder wall. This is 33% augmentation on rectangular fin, 43.53 % over circular fin and 16.17% over other louvered angle fins.



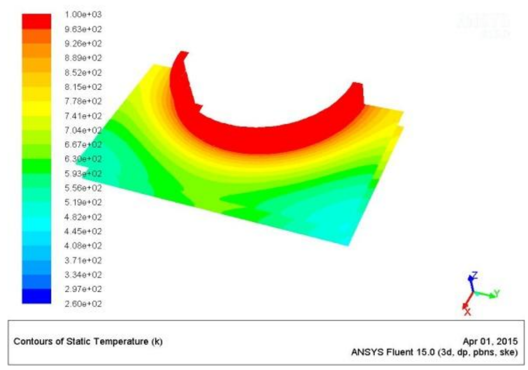


Figure 12.Temperature contours for rectangular and circular fin array

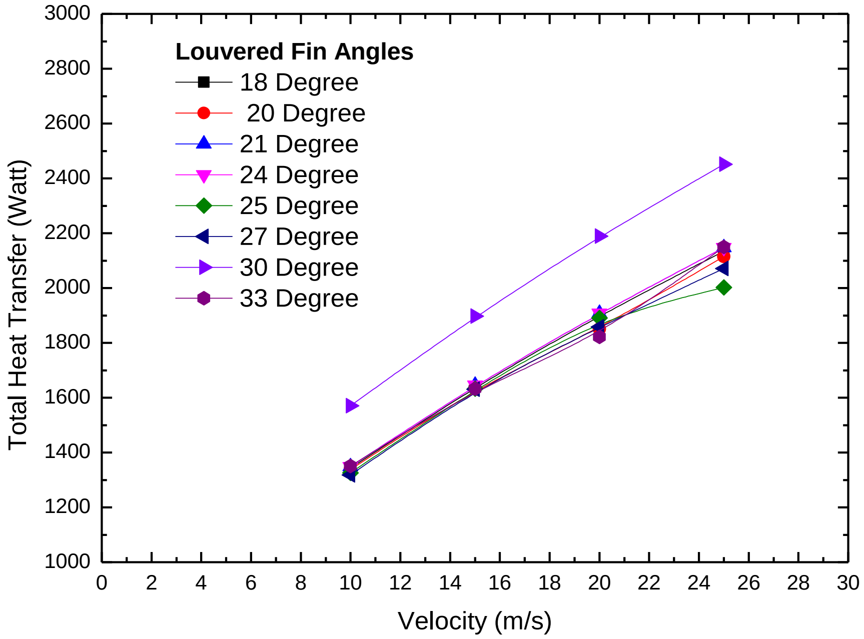


Figure 13. Total Heat transfer V/S Velocity for given louvered angles.

5.2 Results Comparison for Vortex Generator and its Angle of Attack

Comparison for vortex generators varying angle of attack shows no major deviation in heat transfer rate for all three aspect ratio 1.5, 2, 3 . At the same time if we compare the results of vortex generator louvered array and without vortex generator louvered array we found increase in heat transfer after placing vortex generator, this is due to disturbances generation in boundary layer flow by vortices generation which as a result cause thinning of thermal boundary layer, which in normal condition act as a barrier in convective heat transfer.

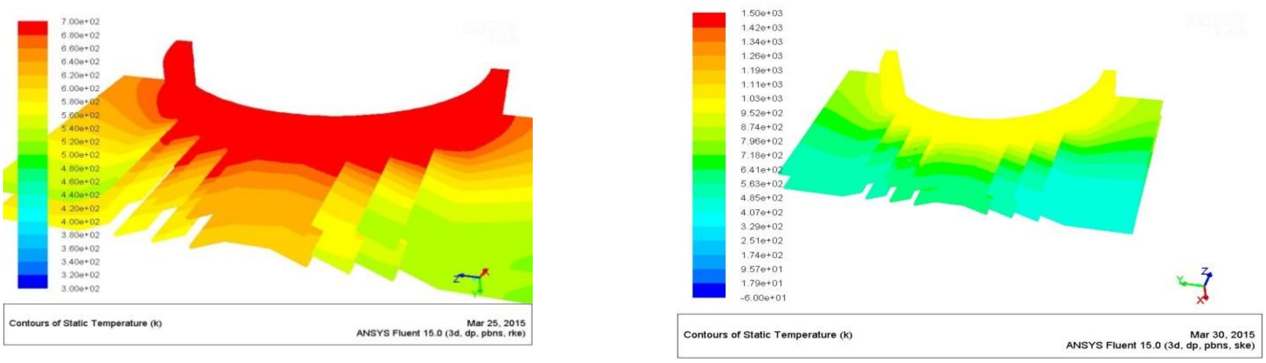


Figure 14.Temperature contours for louvered fin array with vortex generator

Fins with winglet pair having aspect ratio 2 and 3 shows 7.21% higher heat transfer from base as compare to having aspect ratio1.5. But aspect ratio 2 and 3 shows same results for all given velocities and angle of attack.And louvered fin with vortex generator shows 7.45 % augmentation over louvered fins without vortex generator at 25 m/s air velocity.

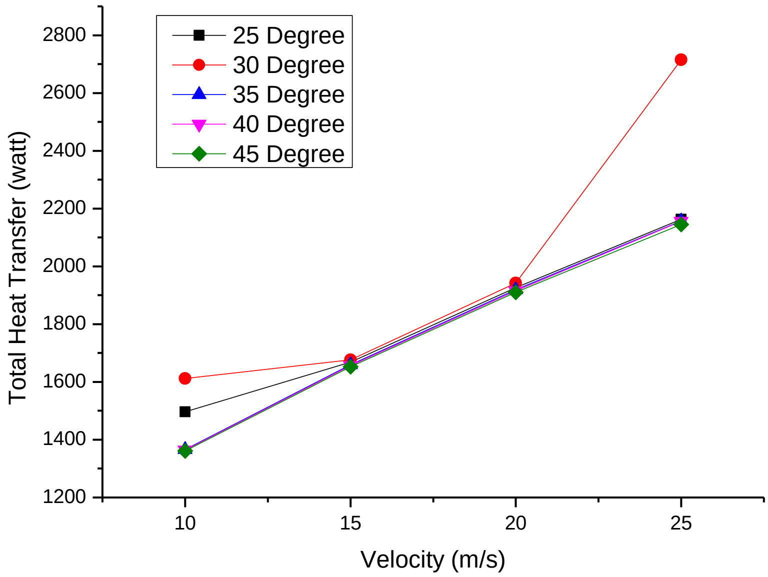


Figure 15. Total heat transfer verses velocity for 25,30,35,40, and 45 Degree angle of attack of VG **aspect ratio 1.5.**

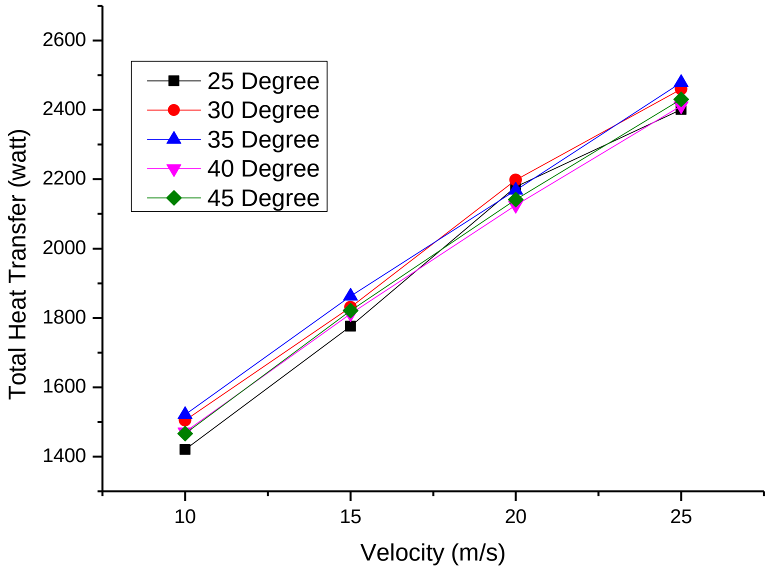


Figure 16. Total Heat Transfer for 25,30,35,40 and 45 angle of attack of VG having Aspect Ratio 2

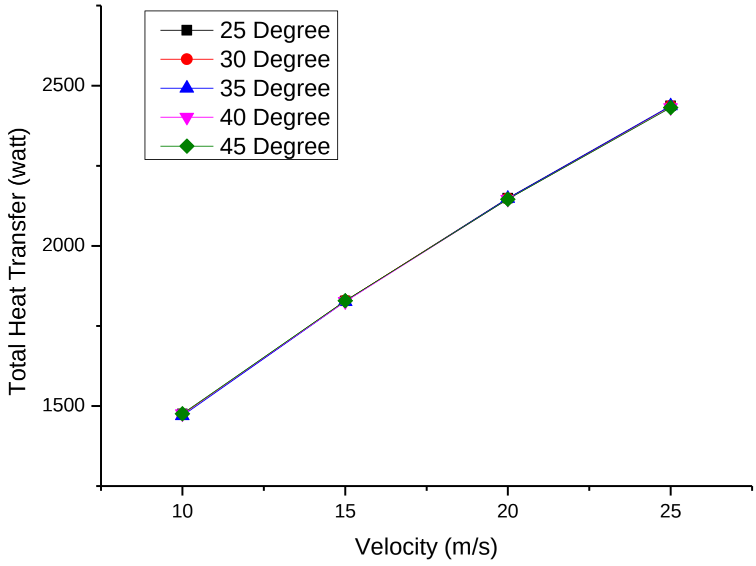


Figure 17. Total Heat Transfer for 25,30,35,40 and 45 angle of attack of VG having Aspect Ratio3

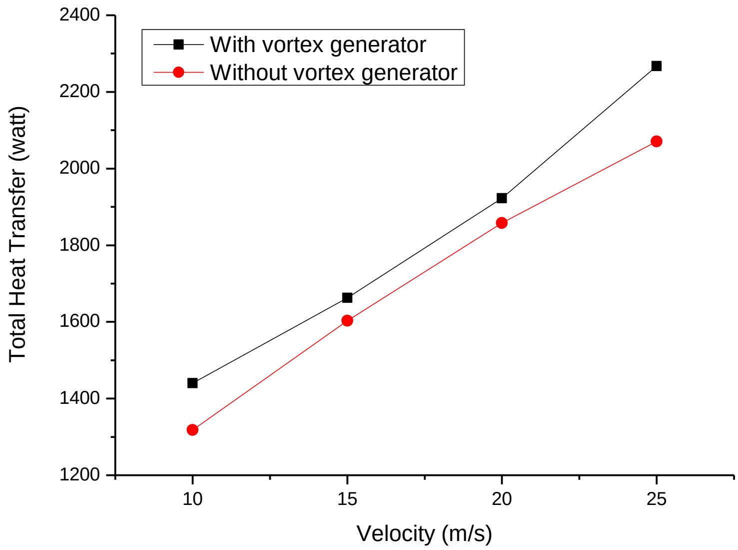


Figure 18. Total heat transfer verses velocity for louvered fin with VG and without VG.

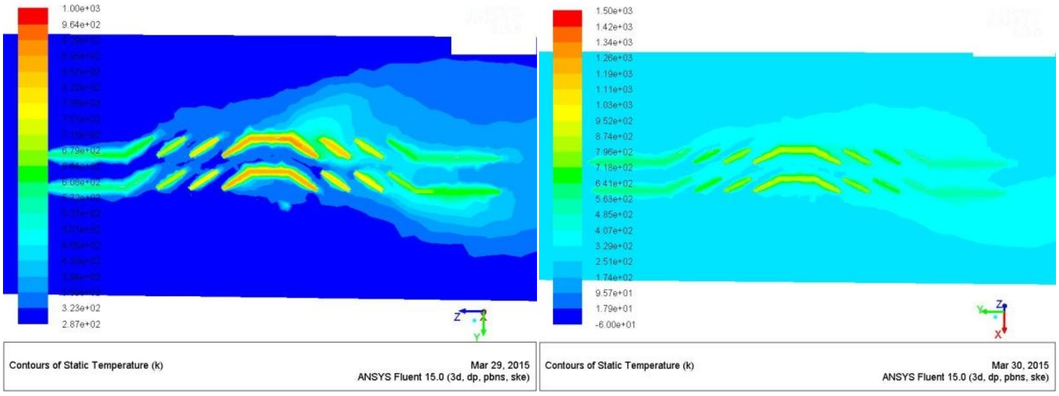


Figure 19. Heat dissipation temperature contours with vortex generator

5.3 Results comparison for Aspect ratio 1.5, 2 and 3

Results for vortex generator as discussed above for different angle of attack it shows no much variation, but same results if we compare for aspect ratio 1.5, 2 and 3, aspect ratio 2 and 3 gives 7.21 % heat transfer augmentation over aspect ratio 1.5, Which is showed in bellow.

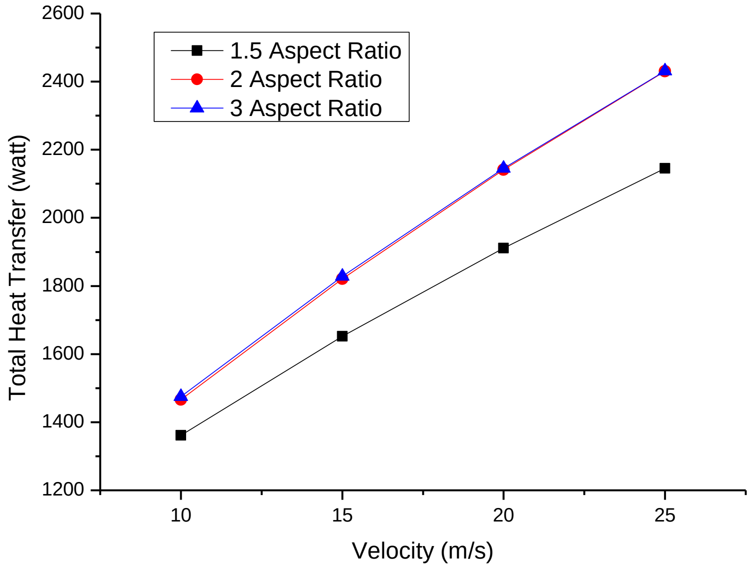


Figure 20. Total heat transfer for aspect ratio 1.5, 2 and 3

5.4 Results Comparison for Variation in Heat Transfer Co-efficient in Stream Wise Direction

Graph for rectangular fin shows constant heat transfer co-efficient over the stream wise direction and shows peaks at the leading and trailing edge, which is the results of strong attachment and separation of flow.If we look out heat transfer co-efficient for louvered fin in stream wise direction it shows large valleys and peaks instead of constant value as in rectangular fin. This behaviour of heat transfer co-efficient shows phenomenon of thermal boundary layer disruption and boundary layer disturbances.

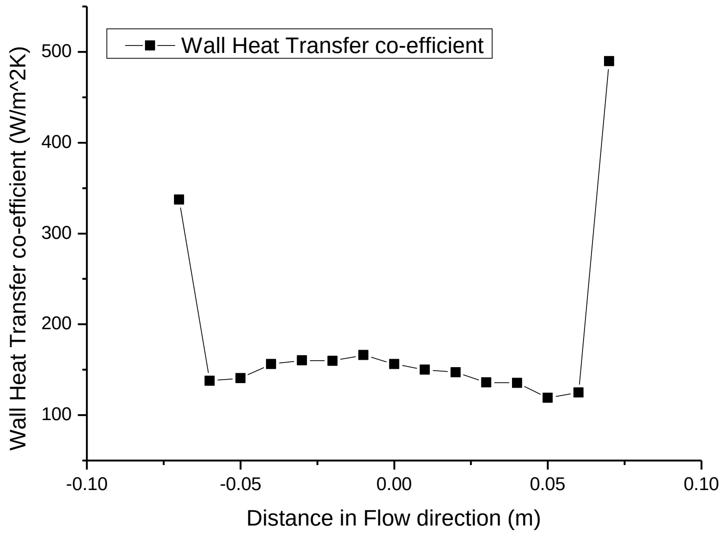


Figure 21. Heat transfer co-efficient in stream wise direction for Rectangular fin

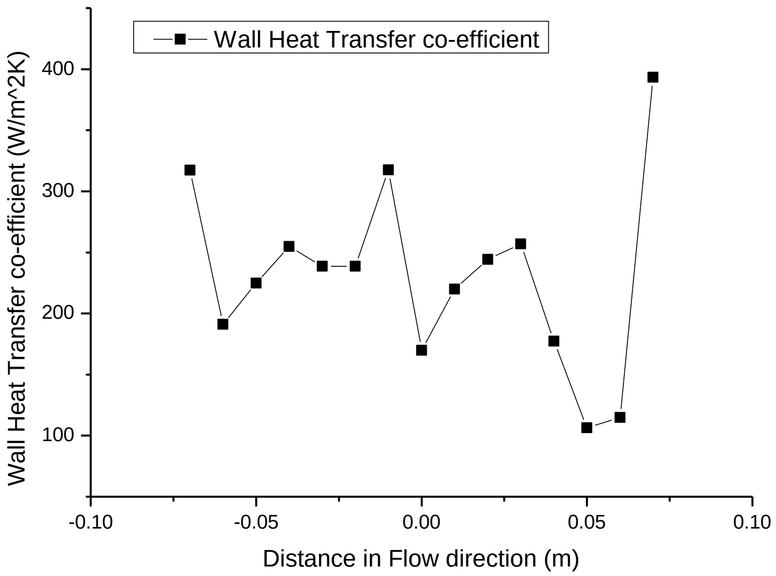


Figure 22. Heat transfer co-efficient in stream wise direction for louvered fin

5.5 Comparison of obtained simulation results with practical results

While comparing baseline results (without vortex generator) with the baseline results of Paul A.Sendars as shown in graph it gives very huge deviation at leading and trailing edge of fin and moderate deviation at middle part. Difference in fin thickness, fin material, number of fins, base design from flat to the circular as well as change in air velocity for inlet are the responsible factor for these deviation in Nusselt number.

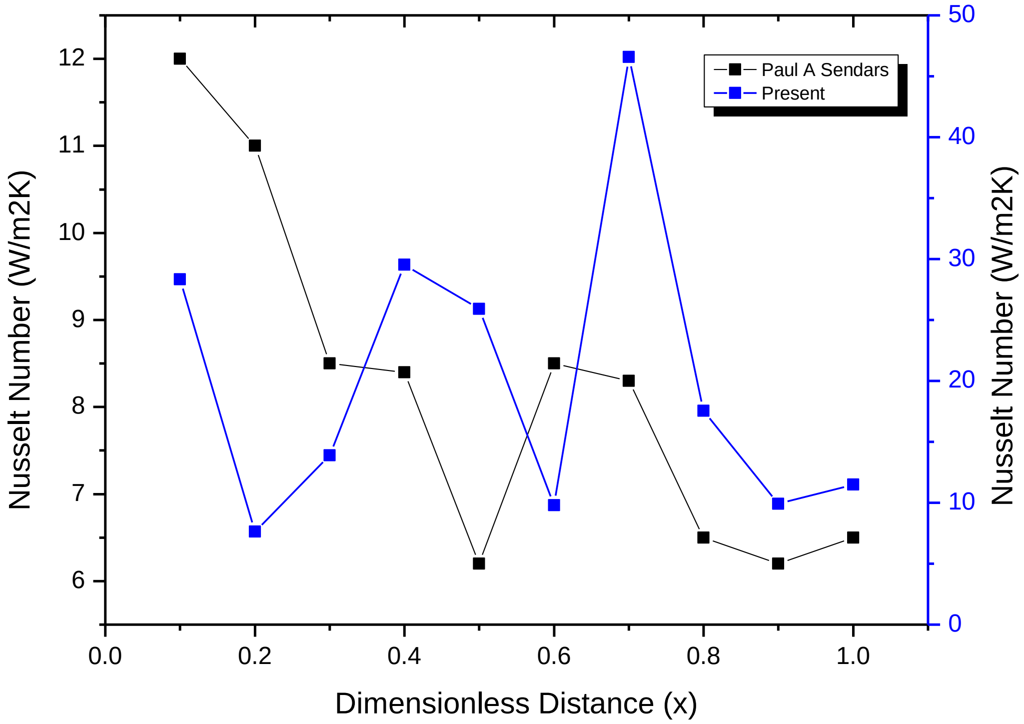


Figure 23. Baseline Result Comparison with Results of Paul A.Sendars

VI. CONCLUSION

From above study we can say that heat transfer enhancement can be achieved at promising level by thermal boundary layer disruption method. There are so many other methods are available to disrupt boundary layer including wavy fins, perforation on fins, slit fins, and many other disturbances methods which can generates vortices and eddies as well as methods like perforation and slit fins can suck down the generated boundary layers, thus by creating more mixing, disturbances, and as a result thinning thermal boundary layer we can improve heat transfer rate.

VII. REFERENCES

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