ISSN 2395-1621

# Investigation of Jet Impingement on Flat Plate Using Triangular and Trapezoid Vortex Generators

#1Sonali S Nagawade, #2Prof. S Y Bhosale, #3Prof. N K Chougule

1Sonalinagawade1@gmail.com
3nkc.mech@coep.ac.in

#1Mechanical Department, PES's Modern College of Engineering, Pune – 05
#2Proffesor, Mechanical Department, PES's Modern College of Engineering, Pune – 05
#3Asso.Proffesor, Mechanical Department, College of Engineering, Pune – 05

**ABSTRACT**

**This experiment is to investigate fluid flow and heat transfer augmentation of air jet impingement on flat plate surface with vortex generators. The flat plate of 100 mm × 100 mm is taken for study with constant heat supply. Reynolds number is varied from 15,000 to 25,000 based on nozzle exit conditions along with variation in jet to plate spacing from1, 2 and 4 times the nozzle diameters. The vortex generators used for the study include six equilateral triangles of side 5mm, trapezoids with height 3.33 mm. At Z/D= 2, surface with triangular and trapezoid vortex generators gives 15-16 % enhancement in average Nusselt number over smooth surface. At Z/D= 4, surface with triangular and trapezoid vortex generators gives 6-7 % enhancement in average Nusselt number over smooth surface. The maximum local Nusselt number value of secondary peak is observed at radial distance of r/D =1 for surface with triangular and trapezoid vortex generators at Z/D=1, 2 and 4. It is more pronounced at Z/D= 1 and Z/D= 2. Triangular vortex generators shows 55% enhancement at Z/D= 1 and 73% enhancement at Z/D= 2 and 36% enhancement at Z/D= 4 over smooth plate at secondary peak point for same Reynolds number.**

**Keywords— Jet impingement, Nusselt number, Reynolds number, Vortex generators.**

ARTICLE INFO

**Article History**

Received :18th September 2015

Received in revised form :

22nd September 2015

Accepted : 1st October , 2015

**Published online :**

**5th October 2015**

**Nomenclature**

| | |
|---|---|
| D | Diameter of nozzle, mm. |
| L | Length of nozzle, mm. |
| Nu | Local Nusselt number |
| $Nu_{avg}$ | Average Nusselt Number |
| r/D | Dimensionless radial distance on surface |
| Re | Reynolds number |
| Z/D | Dimensionless jet to target plate spacing |

## I. INTRODUCTION

Jet impingement is widely used where high convective heat transfer rates are required. The rapid development in microelectronics and the simultaneous drive to reduce the size and weight of electronic products have led to increased importance of the thermal management issues in industry.

Jet impingement systems provide an effective means for the enhancement of convective processes due to the high heat and mass transfer rates that can be achieved. The range of industrial applications that impinging jets are being used in today is wide. In the annealing and tempering of materials, impinging jet systems are finding use in the cooling of hot metal, plastic, or glass sheets as well as in the drying of paper and fabric. For all the studies performed it has been shown that, for a constant jet diameter, heat transfer increases with increasing Reynolds number [1]. It has been shown that, for a constant Reynolds number, decreasing the jet diameter yields higher stagnation and average heat transfer coefficient. This indicates that higher velocities are created by the smaller nozzles.The thermal and hydraulic characteristics of various impinging heat sinks have

© 2015, IERJ All Rights Reserved 

extensively studied by many researchers. Luis A. Brignoni and Suresh V. Garimella evaluated experimental optimization of confined air jet impingement on a pin fin heat sink [2]. Four single nozzles of different diameters and two multiple-nozzle arrays were studied at a fixed nozzle-to-target spacing, for different turbulent Reynolds numbers ($5000 \leq Re \leq 20\ 000$). The bare surface experiments showed that for a given Reynolds number, higher heat transfer coefficients were obtained with the smaller diameter nozzles which may be attributed to the associated higher jet velocities. In contrast, the larger single nozzles performed better when the heat sink was present at a given Reynolds number.

Hung-Yi Li,S.M. Chao, G.L. Tsai studied the thermal performance of heat sinks with confined impingement cooling is measured by infrared thermography[3]. They found that increasing the Reynolds number of the impinging jet reduces the thermal resistance of the heat sinks consistently. Increasing the fin height to enlarge the area of heat transfer also decreases the thermal resistance, but the effects are less conspicuous than those on altering the fin width. Hung-Yi Li, Kuan-Ying Chen utilized the infrared thermography technique to investigate the thermal performance of plate-fin heat sinks under confined impinging jet conditions [4]. They concluded that the thermal resistance of the heat sink apparently decreases as the Reynolds number increases; however, the decreasing rate of the thermal resistance declines with the increase of the Reynolds number. An increase of the fin width reduces the thermal resistance initially. Increasing the fin height can increase the heat transfer area which lowers the thermal resistance. Moreover, the influence of the fin height on the thermal resistance seems less obvious than that of the fin width.

Colin Glynn, Tadhg O'Donovan and Darina B. Murray studied jet impingent cooling, it has been shown that when Z/D increased from 1 to 4, Jet diameter varied from 0.5mm to 1.5 mm and found that Area average heat transfer increases with decrease in jet diameter because of increase in velocity[5]. H. A. El-Sheikh and S. V. Garimella investigate Heat transfer from pin-fin heat sinks under multiple impinging jets [6]. The results for the multiple jets were compared to single jets, both at a fixed orifice diameter (d=12.7 mm) and for the same total orifice area (single jet with d= 25.4 mm). At a fixed Reynolds number, the heat transfer coefficient decreased by 10% as the jet-to-jet spacing was decreased from of 3 to 2 for the large, unpinned heat sink.

However, an increase was observed for the pinned heat sinks when S/D was decreased from 3 to 2 largely due to the unique design of the heat sink in this study. Single jets yielded lower heat transfer coefficients than multiple jets of the same nozzle diameter for all heat sinks tested at a fixed Reynolds number.

P.M. Nakod, S. V. Prabhu, R. P. Vedula, [7] experimentally investigated the effect of the finned surfaces and surfaces with vortex generators on the local heat transfer coefficient between impinging circular air jet and flat plate. Reynolds number is varied between 7000 and 30,000 based on the nozzle exit condition and jet to plate spacing between 0.5 and 6 nozzle diameters. The augmentations in the heat transfer for the surfaces vortex generators are higher than that of the finned surfaces. The heat transfer augmentation in case of vortex generator is as high as 110% for a single row of six vortex generators at a radius of 1 nozzle diameter as compared to the smooth surface at a given nozzle plate spacing of 1 nozzle diameter and a Reynolds number of 25,000 at extreme radial location.

## II. Description Of Physical Problem

*A. Description of Physical Model*

A plat plate heat sink of 100×100 × 6 is taken for study which is subjected to constant heat flux of 50 W from bottom. The shape of Vortex generators are in the form of equilateral triangle of 5 mm side and trapezoids of height 3.33 mm with base 5 mm are arranged in a circular pattern at a radius of 1 times the nozzle diameter on two different smooth plate. The material of heat sink is aluminium. Nozzle diameter is 10 mm. Reynolds number is varied between 15000 to 25000 in turbulent range to study the effect of Re on Nusselt number.

## III. Experimental Details

A schematic diagram of the experimental set up is shown in Fig 1.It is an Air flow bench, which provides controlled and measurable flow of air through nozzles or jet plate directed towards the target plate. It consists of a blower, air straightener (air box), contraction section, and structure and Data

Acquisition System (DAQ) to measure temperature, pressure. There is honeycomb structure inside the air box in order to provide streamlined flow prior to impinging on the heat sink and a butterfly valve used in order to regulate the discharge from the centrifugal blower. The blower draws air from the atmosphere and delivers it along a pipe to an air box, which is above the test area. The air flow bench structure is made to incorporate other devices such as DAQ system, Power supply, pressure measurement devices

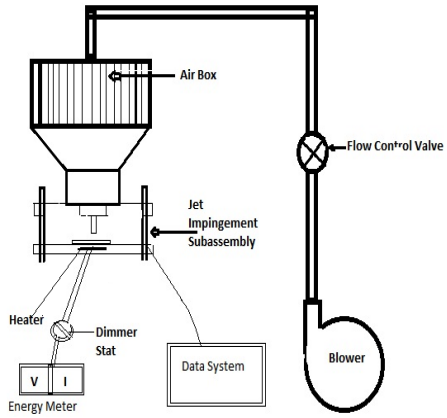



Fig. 1: Experimental set up

and various controls. Micro-manometer is used to measure the static pressure of air at the outlet of the jet at different locations.Thermocouples are used to measure the temperature Average of all the readings is taken and jet velocity is calculated. To measure the temperature at the base of the fin, K type thermocouples are mounted on base through 1 mm diameter hole.

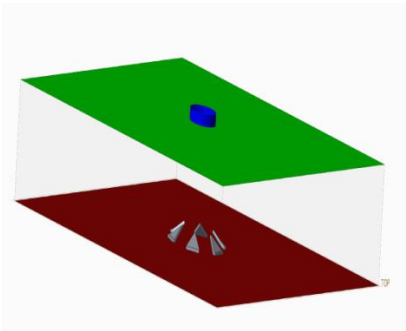
© 2015, IERJ All Rights Reserved 

## IV. Mathematical Modeling

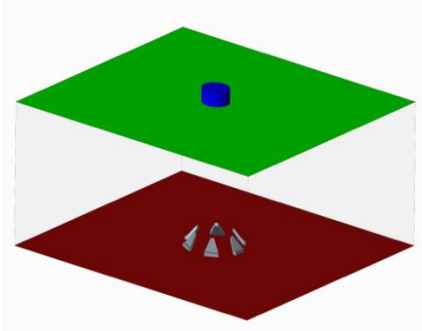
### A. Geometry and Meshing

The geometry is created in Creo 3.0 and meshing is done in ICEM CFD. As the geometry is small, we consider complete geometry for the analysis. The solution domain is filled with stagnant air. The computation domain is fluid domain (air).

It is assumed that heat is generated inside the heat sink at uniform rate and can be represented by constant heat flux from bottom surface of hear sink. To reduce the temperature of heat sink at stagnation point vortex generators are provided. Air flow at high velocity passes through a round jet with length = 40 mm and diameter = 10 mm.



(a) Flat plate with 6 triangular vortex generators at 30 degree



(b) Flat plate with 6 trapezoid vortex generators at 30 degree

Fig. 2: Computational and physical domain of jet impingement

### B. Numerical Solution

The fluid flow and heat transfer is governed by three fundamental equations continuity equation, conservation of momentum and conservation of energy equation In this case, the numerical solution were carried out using commercial CFD solver Fluent version 14.5.

SST (Shear Stress transport) Turbulence model is used because some authors have already found in previous study that SST K-ω model is in better agreements with the internal heat transfer coefficient measured inside a cooling device similar to that of present study. In the fluid domain the inlet boundary condition is specified the measured velocity and static temperature (300K) of the flow were specified at the inlet of the nozzle. No-slip condition was applied to the wall surface. In fluid domain there is also opening boundary condition in which flow regime is subsonic, relative pressure is 0 Pa with operating temperature (300K) and the turbulence intensity of 5%.

The variation of thermal and physical properties of air with temperature is neglected. The flow field was numerically examined by use of Fluent (ANSYS v14.5), assuming the steady-state flow.

### C. Solver

A geometry and mesh object is imported into Fluent CFD software environment for solving governing equations. The flow and turbulence fields have to be accurately solved to obtain reasonable heat transfer predictions. Second order scheme is used for all terms that affect heat transfer. Standard scheme is used for the pressure; second order upwind discretization scheme is used for momentum, turbulence kinetic energy, specific dissipation rate, and the energy. Flow, turbulence, and energy equations have been solved. To simplify the solution, the variation of thermal and physical properties of air with temperature is neglected. The simulation type is steady state condition, convergence criteria are specified as 10E-06 residuals.

## V. Results And Discussions

### A. Validation with Experimental Results

Comparison between experimental and CFD results are shown in Table 1. To validate CFD results, the detailed experimentation were carried out on flat plate without vortex generators, Flat plate with triangular vortex generators, and flat plate with trapezoid vortex generators. The average temperature obtained from the experimental data is used to validate the computational work.

To simulate the above experimental conditions in context of CFD analysis, the same geometry, boundary conditions are applied and also temperature monitoring points are located at the same position where thermocouples are physically located. Mostly within all the range of parameters, it is observed that CFD results are in good agreement with experimental results. Average Nusselt number increases by 15-17 % in flat plate with vortex generators over flat plate without vortex generators at Z/D= 2 and Z/D = 4 when Re = 25000.

Table I

Comparison Of Experimental And CFD Results at Re 25000

| ***Configuration of plate*** | ***Z/D*** | ***Experimental Results*** $Nu_{avg}$ | ***Numerical Results*** $Nu_{avg}$ | ***% Error*** |
|---|---|---|---|---|
| Flat Plate without vortex generators | 1 | 109.072 | 95.3 | 12.62 |
| | 2 | 95.176 | 81.41 | 14.46 |
| | 4 | 100.892 | 87.162 | 13.61 |
| Flat Plate with Triangular vortex generators | 1 | 108.978 | 92.966 | 14.69 |
| | 2 | 110.4898 | 95.216 | 13.82 |
| | 4 | 109.164 | 92.336 | 15.42 |
| Flat Plate with Trapezoid vortex generators | 1 | 111.804 | 93.034 | 16.79 |
| | 2 | 106.432 | 94.506 | 11.21 |
| | 4 | 110.532 | 94.37 | 14.62 |

© 2015, IERJ All Rights Reserved 

*B. Temperature contours of vortex generators on surfaces.*

The temperature contours of smooth plate without vortex generators, surface with triangular and trapezoid vortex generator for Re= 25000 at Z/D= 1, 2 and 4 is shown below (Fig. 3) as higher Nusselt number is observed at this Reynolds number. From the temperature distribution it is clear that at smaller Z/D ratio (Z/D= 1) the temperature on target surface is lower and uniform. The temperature variation in vortex area is in the range of 301K to 317 K.

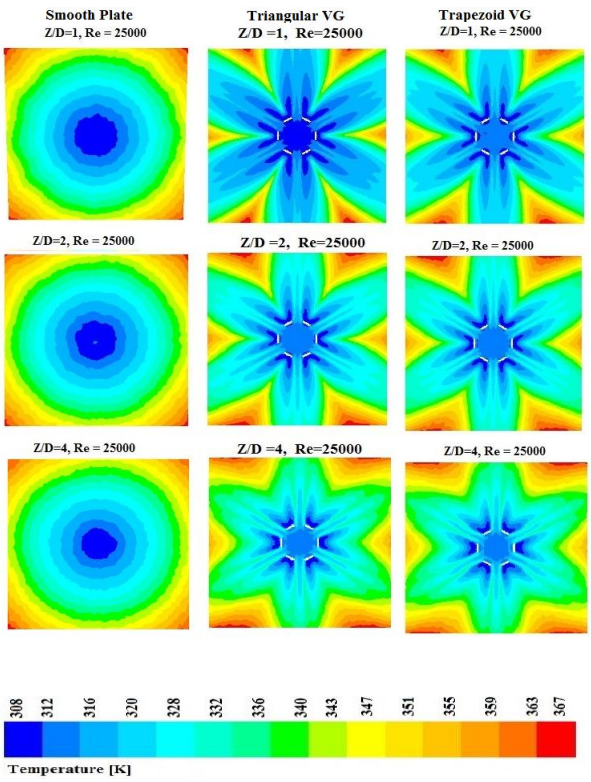



Fig. 3 Temperature contours (Re= 25000)

These temperature contour plots show that for higher Reynolds number, average temperature of plate surface is lower due to localized cooling.Fig.3 shows the distribution of average temperature on flat plate with triangular and trapezoid vortex generators at different Z/D. It is also observed that the portions of flat plate below nozzle area are cooled intensively than other area. Minimum temperature is obtained at stagnation point.

*C. Effect of Z/D on Local Nusselt Number*

Fig 4 shows the effect of Z/D ratio on local Nusselt number at Reynolds number of 25000 on smooth plate with triangular vortex generators. With decreasing Z/D ratios the Nusselt number increases accordingly at secondary peak point at radial distance r/D = 1 but Local Nusselt number at stagnation point increases at Z/D= 4. The triangular vortex generators at Z/D= 1 shows 26 % enhancement over Z/D= 4 at secondary peak point for same Reynolds number. The Z/D= 4 shows 17-18 % enhancement over Z/D= 2 and Z/D= 1 at stagnation point for same Reynolds number. It is observed that at smaller Z/D ratios the jet strikes directly on target surface without losing its much momentum with surrounding flow, this result in the jet reaching the target surface with higher momentum, this increases the heat transfer.

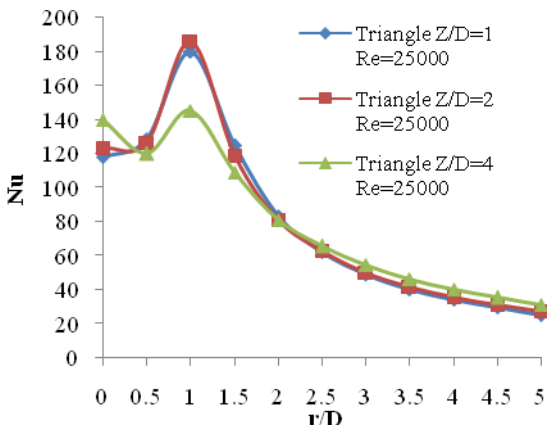



Fig. 4 Comparison of the Nusselt number for smooth surface with triangular vortex generators at Reynolds number of 25000 for different Z/D's

*D. Effect of Different Configurations on Nusselt Number Distribution*

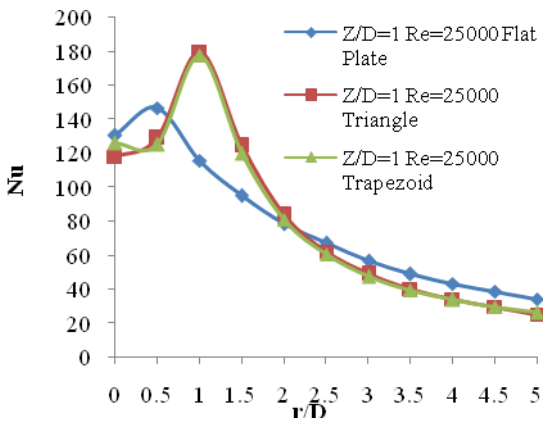



Fig. 5 Comparison of the Nusselt number for smooth surface without vortex generators and with vortex generators at Reynolds number of 25000 and Z/D= 1.

Fig 5 shows the effect of different configurations on local Nusselt number at Z/D= 1 and Reynolds number of 25000 on smooth plate without vortex generators, smooth plate with triangular and trapezoid vortex generators. In this case secondary peak point of Nusselt number for smooth plate without vortex generators is observed at radial distance r/D = 0.5 whereas secondary peak point for smooth plate with triangular and trapezoid vortex generators is observed at r/D=1.The Triangular vortex generators shows 55% enhancement over smooth plate at secondary peak point for same Reynolds number.

© 2015, IERJ All Rights Reserved 

*E. Effect of Reynolds Number on Local Nusselt Number*

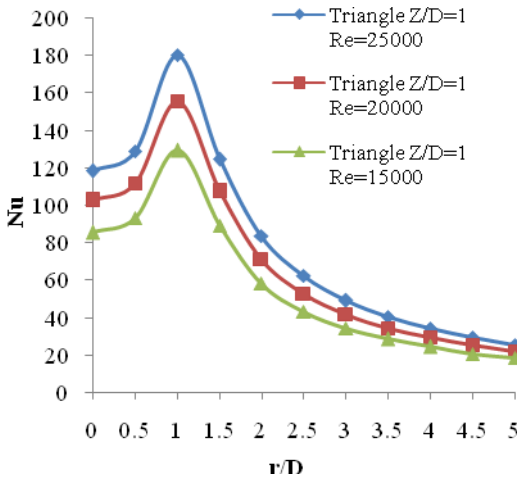



Fig.6 Comparison of the Nusselt number with radial distance for smooth surface with triangular vortex generators at a Z/D= 1 for different Reynolds number.

Fig.6 shows average Nusselt number variation with respect to Reynolds number on flat plate with triangular and trapezoid vortex generators. It is observed that the Nusselt number is directly proportional to Reynolds number.
At higher Reynolds number, its effect on Nusselt number is more pronounced. Secondary peak value of Nusselt number is observed at radial distance of r/d =1 for Z/D= 1 and 2 is more compared with Z/D= 4.

## VI. CONCLUSION

Z/D ratio has a significant impact on heat transfer. At smaller Z/D ratios with higher Re, there is vortex flow near the target surface in wall jet region, causing increase in Nusselt number. At Z/D= 2, surface with triangular and trapezoid vortex generators gives 15-16 % enhancement in average Nusselt number over smooth surface. At Z/D= 4, surface with triangular and trapezoid vortex generators gives 6-7 % enhancement in average Nusselt number over smooth surface.
The maximum local Nusselt number value of secondary peak is observed radial distance of r/D =1 for surface with triangular and trapezoid vortex generators at Z/D = 1, 2 and 4. It is more pronounced at Z/D= 1 and Z/D= 2. Triangular vortex generators shows 55% enhancement at Z/D= 1 and 73% enhancement at Z/D= 2 and 36% enhancement at Z/D= 4 over smooth plate at secondary peak point for same Reynolds number.
At Z/D= 1 and Re=25000 comparatively favorable results are observed for flat plate with trapezoid vortex generators.
The CFD result shows good agreement with experimental results (82 – 86% agreement).

## ACKNOWLEDGEMENT

I deeply thank Dr. N.K. Chougule for his guidance and providing experimentation support which has made this study possible. I am also grateful to Prof. Bhosale S.Y. for his helpful suggestions and criticism

## REFERENCES

[1] N. Zuckerman, N. Lior, “Impingement heat transfer: correlations and numerical modeling”, J. Heat Transfer 127 (2005) 544–552.
[2] L.A. Brignoni, S.V. Garimella, “Experimental optimization of confined air jet impingement on a pin fin heat sink”, IEEE Transactions on Components and Packaging Technologies 22 (3) (1999) 399 - 404.
[3] H.Y. Li, S.M. Chao, G.L. Tsai, “Thermal performance measurement of heat sinks with confined impinging jet by infrared thermography”, International Journal of Heat and Mass Transfer 48 (2005) 5386 - 5394.
[4] H.Y. Li, K.Y. Chen, “Thermal performance of plate-fin heat sinks under confined impinging jet conditions”, International Journal of Heat and Mass Transfer 50 (2007) 1963 - 1970.
[5] Colin Glynn, Tadhg O’Donovan and Darina B. Murray, “Jet Impingement Cooling”, from CTVR (Centre for Telecommunications Value-Chain-Driven Research), Department of Mechanical and Manufacturing Engineering, Trinity College, Dubli
[6] H. A. El-Sheikh and S. V. Garimella, “Heat transfer from pin-fin heat sinks under multiple impinging jets”, IEEE transactions on advanced packaging, vol. 23, no. 1, (2000)
[7] P. M. Nakod, S. V. Prabhu, R. P. Vedula, Heat transfer augmentation between impinging circular air jet and flat plate using finned surfaces and vortex generators, Experimental thermal and fluid science 32(2008) 1168 - 1187.

© 2015, IERJ All Rights Reserved 