

Effect of Dimple/ Protrusion Shape on The Cooling Performance of The Turbine Blade Using CFD

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ABSTRACT

It is well understood that one way to improve a gas turbine engine's power output and thermodynamic performance is to increase the temperature of the turbine inlet (TIT). The inlet temperature should be gradually elevated to higher targets to pursue greater control. However, with the rise in blade inlet temperature, the heat applied to the blade increases, and the permissible melting temperature of materials increases at a slower rate. This implies that the inlet temperature of the turbine blade will exceed the melting temperature of the material by more than 500°C. Thus, cooling turbine blades for a safe and long-lasting operation is important. To lower the temperature of the blade content below its melting point, different internal and external cooling techniques are employed. In recent years, in the tip region of the turbine blade, multiple augmentation devices such as fins, ribs, pins, and dimples / protrusions have gained a lot of attention to enhance heat transfer. With increasing inlet temperature of the turbine, pin fin arrays cannot satisfy the need of the cooling alone. To be paired with the pin fin arrays, several more methods are also added. With slight loss of pressure, dimples and protrusions are fine options. In this present work, the numerical approach is used to investigate the influence of dimples or protrusions shape on the cooling of the turbine blade. The ANSYS 17.0 simulation software was used. The results of this study show that the, due to flow acceleration, increase in impingement area and shrinkage of the flow recirculation region within the dimple, pin fin-dimple wedge duct with triangular shape dimples/protrusions provides improved heat transfer enhancement.

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KEYWORDS: Gas turbine, Turbine inlet temperature, turbine blade cooling, internal convective cooling, Pin fin cooling, dimples/protrusions shape, and CFD

I. INTRODUCTION

Turbines play a very important role in the technologically sophisticated industries today. To cope up with the current demands of higher power generation, improved thermal efficiency and to eventually improve the overall performance of the turbine, we need to operate on very high turbine inlet temperature (TIT). Studies show that only a decrease of 30°C – 40°C will affect the overall turbine output. However, the disadvantage of the very high inlet temperature is that it limits the life of blades and valves and often reaches the blade melting point. Diverse other factors besides the melting point of a blade steel, such as the creep strain, centrifugal and rotational stress, produced while the turbine blade operates in practise, bound TIT.

The continuous distribution and physical distortion under constant load (creep) increases the rotational stresses induced by a blade breakdown when the blade is in operation. It is found that in recent years, the turbine blades and vanes have been shielded from such high TIT in the context of modern cooling concepts. The purpose of the turbine blade cooling is to reduce the blade temperature without reducing the turbine's efficiency and output.

Air or liquid may either be used as a means for cooling. It looks very enticing with the use of a liquid coolant like water because the water's thermal potential is high and the chances of an evaporative refreshment are minimal, but it can leak and cause corrosion, shock and further decrease the life of the blade. As a cooling medium, the new cooling systems use cold air from the compressor unit as a way of mixing quickly with the main flow of hot gas in the turbine.

In gas turbine blades, several cooling methods are employed; convection, film, transpiration cooling, cooling effusion, pin fin cooling, etc. fall under the divisions of internal and external cooling. Although every approach has its variations, they both work to eliminate the heat from the turbine blades using colder air (often bleeding from the compressor).

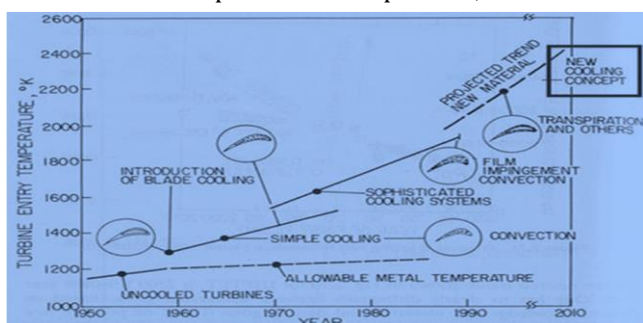


Figure 1. Development during the last 60 years of turbine cooling system.

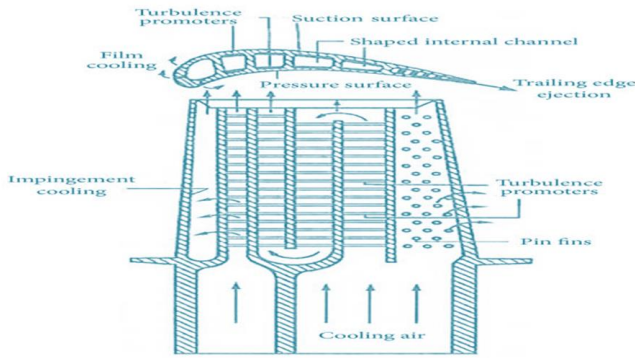


Figure 2. Typical gas turbine cooling structure.

II. LITERATURE REVIEW

Recent advances in increasing turbine inlet temperature were accomplished by improved turbine blade cooling and better knowledge of turbine passage heat transfer mechanisms. Several publications can be found reviewing heat transfer from gas turbine and studies into cooling technologies [1-5].

The blade should be cooled in all regions, which are subject to high temperature gas and thermal load. The tip of the blade is an area for high-pressure turbines in particular. The heat drain raises the thermal load at the tip of the blade, resulting in a high local temperature. The tip of the turbine blade and the area at the end are therefore very critical to cool. The blade tip works between the spinning blade and stationary box in an area where the intense thermal fluid pressures inside the turbine are encountered [6-7]. The tip will to some degree be cooled, if it profits from the effects of impinging and turning. Advanced methods of improving internal convective cooling in the tip area are required to increase forced convection. The longevity and trustworthiness of the blade is enhanced.

2.1. Previous work

Much research work has shown that pins, dimples and protrusions in confined low aspect ratio channels have a large improvement in heat transfer. While ribs and pins are known to dramatically increase thermal transfer through primary or smooth surfaces, the increase is often influenced by the high additional pressure loss.

For circular staggered pins with a high Reynolds counting but a coefficient of friction increase of about 70 compared with that of a completely built smooth flow channel, the **Chyu et al. (1990)** experimentally tested mass/heat transfer improvements of up to 4.0 [8].

Bailey and Bunker et al. (2003) the improvement in heat transfer for ribbed channels with a very high blocking ratio of 27.5 – 47.5 per cent for a smooth channel without ribs was achieved up to 3.6 per cent. This improvement was therefore followed by an increase of the friction coefficient of 65. Through comparing multiple heat transfer types, increase techniques for heat/mass transmission increases [9].

Goldstein et al. (1994) the circular pin-fin array of stepped diameter supported a larger transfer coefficient and a smaller pressure loss than the circular pin-fin array of a single diameter [10].

Chyu et al. (1997) the heat transfer was experimentally tested using a transient liquid crystal system on staggered

dimpled surfaces. They reported that the heat transfer ratios across the dimple surface were higher than those of the smooth channels, and the main factor in growing heat transfer was over 2.5 below the dimples [11].

Moon et al. (2000) to research rectangular channels with dimples imprinted on one wall using similar experimental method. Results demonstrate that a 2.1 time's improvement in heat transfer with a 1.6–2.0 times reduction in pressure can be obtained relative to smooth surfaces [12].

Moon and Lau (2002) Average heat transfer coefficients measured on a concave dimples in a square channel on a wall. Experimental reporting. They observed that the cylindrical dimple supported better heat transfer with a lower pressure drop than the concave dimples of the same diameter and depth [13].

Park et al. (2004) Predicted turbulent structure above a dimpled surface, using the $k-\epsilon$ model without wall functions, by way of FLUENT. They submitted that the vortex pair contains an improved eddy diffusivity for momentum and heater due to the reattachment and recirculating flow in the pliable holes as well as to the efficient secondary flow and mixtures [14].

Hwang et al. (2008) used the TLC technique for calculating local coefficients for heat transfer on pale or protrusion walls with low Reynolds numbers ranging from 1000 to 10,000 for transient thermochromics. They found that the best average heat transfer rate with the highest-pressure drop was a double protrusion cord. The heat transfer change factor is up to 14 at low Reynolds numbers of 1000. The double protrusion wall canal gave poorer output than other cases in the broad Reynolds range: double dimple wall, single protrusion wall and basic dimple wall [15].

Furthermore, **Zhou and Catton (2011)** The flow and heat transfer were measured by a numerical method in a plate-pin fin sink with the various cross pin section (square, circular, elliptical and NACA profile), concluding that the square cross section pin-fin had the best heat transmission increase for all the pin fins tested [16].

Later, **Siw et al. (2012)** the thermal transfer efficiency of the triangular, semi-circular and circular pin fins was studied through a detailed experiment. The results measured with hybrid liquid crystal imaging showed that the greatest improvement in heat transfer was the triangular pin fin area, but also the highest friction factor [17].

Lan et al. (2013) Mixed ribs, dimples and protusses and five different variations have been tested in the Reynolds scale of 10,000 to 60,000. The results revealed that the combination of rib and protrusion strategies in a rectangular tube was able to increase heat transfer penalty for low-pressure falls. [18].

Xie et al. (2013) further experiments have taken place on the function of internal dimple protrusion. The principal design parameters were the location of the protrusion in the clear cavity in the direction of the stream. This analysis resulted in a major improvement in the fluid flow zone in the internal dimple area. The most increase in the heat transfer was attributed to the internal protrusion mounted at the central position of the dimple [19].

Besides, **Xie et al. (2015)** they studied a rectangular channel with teardrop bumps or protrusions using a numerical method. The study revealed an increased heat transfer efficacy with a smaller number of Reynolds compared to a hemispheric dimple / protrusion. [20].

Hwang et al. (2010) the transient TLC (the liquid crystal thermochromics) technique on periodic dimple-protrusion plates was investigated for local thermal transfer and thermal production. The results showed that for all the tested plate the thermal efficiency was identical in a given Reynolds number [21].

Lan et al. (2011) Mixed ribs, dimples and protrudes and five different variations have been tested in the Reynolds scale of 10,000 to 60,000. The results revealed that the combination of friction and protrusion technology in a rectangular tube was able to improve the heat transfer penalty for low-pressure dropping [22].

Luo et al. (2017, 2018) computational methodology was used for the study of the channel/duct flow structure and heat transfer properties with dimples / protrusions. According to the findings, converging angle channels showed a greater heat transfer improvement but also an increased pressure loss. Compared with the dimple cases, the protrusion case resulted in easier heat transfer and more friction loss [23, 24].

Wang et al. (2019) the dimpled / protrude pin fin wedge duct heat transfer properties were examined for multiple converging blade angles. According to their results, because of flow acceleration, increase in impingements, and shrinkage of the flow field, the larger converging angle pin end-dimple wedge canal is able to boost the heat transfer, but also a much larger friction factor. A larger converging angle pin finish wedge canal invokes increased thermal transport when the stream speed is increased and the protrusion is affected more extremely, but at a higher-pressure penalty. [25].

Multiple heat-transfer devices such as fins, ribs, pins, and protrusions have received much interest over the past few years in the tip region of the turbine blade. The pin fin arrays cannot fulfil the cooling criteria alone with the rising inlet temperature of the turbine. More approaches are also added to be combined with the pin fin arrays. Dimples and protrusions are good solutions with minor loss of pressure. Dimples and protrusions are generated on a smooth surface to achieve high turbulence kinetic power (TKE) flow and facilitated the thermal transfer. The research therefore aims to examine the heat transfer characteristics of a different shaped dimpled/protrusioned for turbine blade cooling with ANSYS 17.0 software.

III. METHODOLOGY

In this portion, the research uses the CFD model to analyse the heat transfer physiognomies of a wedge duct, with the result of various shapes of dimples or protrusions placed on a turbine blade's heated end wall surface. There are three key phases involved in the CFD analysis. The first step requires the development of the desired model's geometry and mesh generation, while the effects in the last step are

shown as anticipated. The boundary conditions are fed into the model in the execution of the solver (middle stage).

The steps that have been taken to accomplish the job's goals.

- 1) First of all on Workbench of ANSYS 17.0 Program, we build the wedge duct model with different form of dimples or protrusions for turbine blade.
- 2) It is moved to ANSYS for CFD analysis after the model is developed.
- 3) On the CFD pre-processor, meshing of the model and name selection is performed.
- 4) The boundary conditions are applied to the model and the solver is used to determine the numerical solutions.
- 5) In solving the problem, the approach of finite volume is used.
- 6) (6) The solution is determined by giving iterations to the model-applied mathematical and energy equations.
- 7) The effects can be visualised by CFD post processor in the contours and graphs of the form.
- 8) Use of calculations to calculate the convective heat transfer and coefficient of heat transfer.
- 9) Review of result.

3.1. Computational model geometric descriptions

The geometry for performing simulation analysis is taken from Wang et al. (2019) [25], a detailed scale research scholar. Table 1. displays the specifics of the conceptual model of traditional architecture.

Table 1. Specifics of the conceptual model of traditional architecture

Parameters	Dimension
Extended length of inlet segment (L_i)	20 mm
Extended length of outlet segment (L_o)	20 mm
Converging channel length (L_c)	160 mm
Width of the channel (W)	30 mm
Streamwise distance between two pin fins (S_y)	30 mm
Dimples or protrusions are centrally positioned with a span-wise gap between the pin fins (S_x)	15 mm
The circular shaped diameter of the pin fins (d)	10 mm
Diameters of the hemispherical dimple (d_m)	10 mm
Diameters of the hemispherical protrusion (d_p)	10 mm
Ratio between dimple depth and dimple diameter	0.2
Ratio between protrusion height and protrusion diameter	0.2

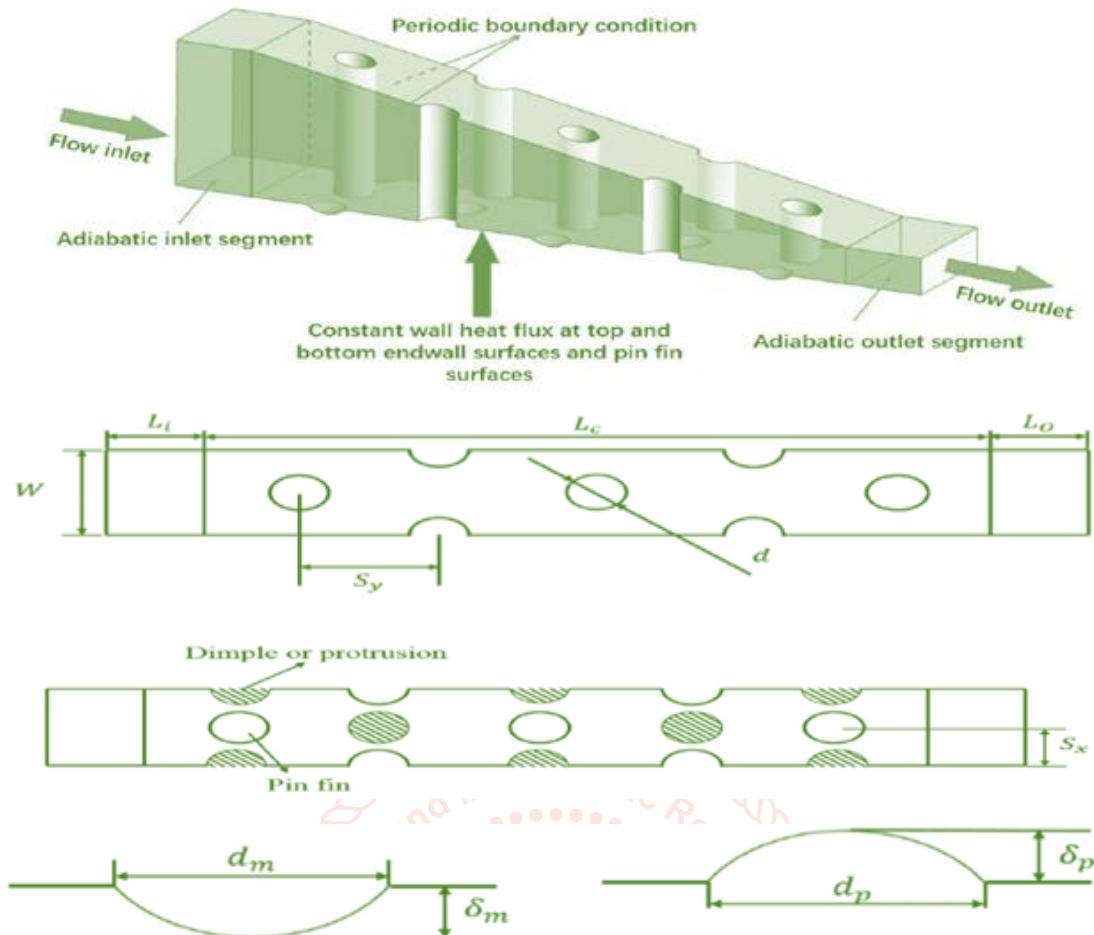


Figure 3. Details schematics of the computational model.

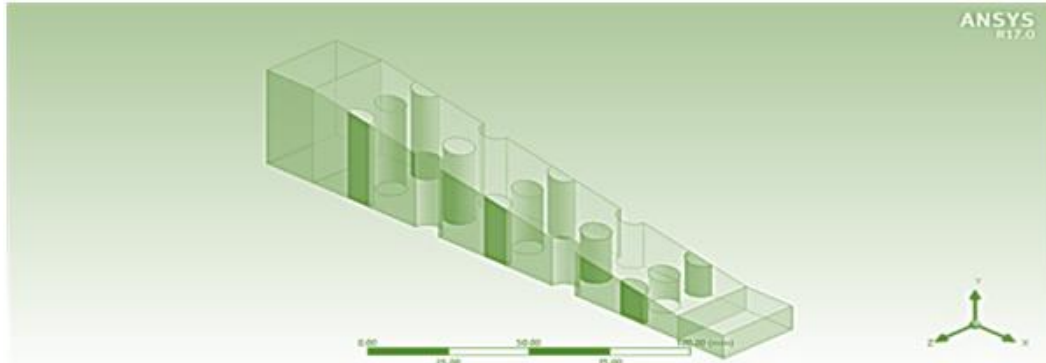


Figure 4. Conventional Design Geometrical Model (having hemispherical shape dimple/protrusion).

The triangular form dimple/protrusion is used in the cooling channel in the proposed designs. A part of the model developed for the programme ANSYS (fluent) workbench 17.0.

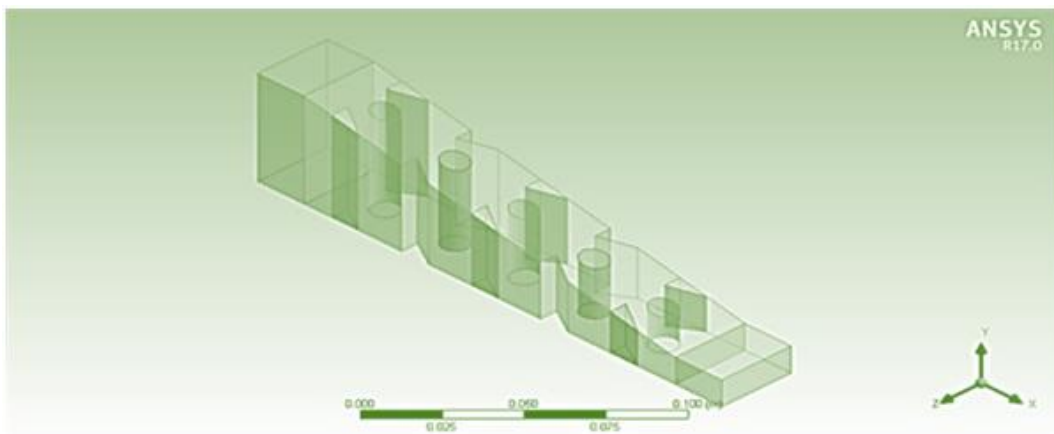


Figure 5. Proposed Design Geometrical Model (having triangular shape dimple/protrusion).

3.2. Grid details and Mesh independence

A three-dimensional discretionary model was built in the pre-processor phase of ANSYS FLUENT R 17.0. Although the grid types are related to simulation results, the full configuration is by necessity, discretized in the finite volume; ANSYS software generates a gross mesh. Mesh comprises mixed cells per unit area with triangular boundary faces (ICEM-Tetrahedral cells). The mesh used in this analysis is a mesh metric with a relatively smooth curvature. As seen in the figure, the mesh form produced tetrahedral meshing.

Table .2. Meshing detail of Conventional and proposed various models

S. No.	Parameters	Hemispherical dimple/protrusion (Conventional design)	Triangular dimple/protrusion (Proposed design)
1	Number of nodes	22509	152918
2	Number of elements	18254	561249

3.3. Model Selection and Solution Methods

Fluent 17.0 was used for numerical estimation. The manner in which the governing equations is divided is a finite element in physics. The scientists used a simpler algorithm for this convective term, the SIMPLE, for the relation with the pressure velocity.

K-ε Model for steady-state measurement of turbulent three-dimensional flow, and the transfer of heat. The heat transfer equations and composition of the fluid flow include mass, momentum and energy conservation. Equations are as follows:

The mass conservation:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0$$

The momentum conservation:

$$\frac{\partial(\rho \bar{u}_i \bar{u}_j)}{\partial x_j} = -\frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\mu + \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \right)$$

The energy conservation:

$$c_p \bar{u}_i \frac{\partial(\rho \bar{T})}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\lambda \frac{\partial \bar{T}}{\partial x_i} \right) - c_p \frac{\partial}{\partial x_i} \left(\frac{\mu_t}{Pr_t} \frac{\partial \bar{T}}{\partial x_i} \right)$$

3.4. Material Property

There are thousands of materials accessible in the ANSYS environment and if possible, a library is not available in ANSYS repositories, a new material directory can be generated as needed. With any kind of research properties, the most important items to described before further analysis can be transferred.

Table 3. Thermo-physical Properties of Air

Air	Density (Kg/m ³)	Specific Heat (J/Kg-K)	Thermal conductivity (W/m-K)
	1.225	1006.43	0.0242

3.5. Boundary Conditions

In this analysis, the inlet and outlet portions are adiabatic. A constant heat stream of 3280 W/m² is applied on the end wall and on the pin fin surfaces. The end wall surface and surfaces are referred to as slip-free limits. Based on the linear temperature, the ideal gas air is a fluid with thermal conductivity and viscosity. The inlet airflow temperature is 297.55 K and the turbulence intensity is 5%; the latter walls of the figure contain a pair of periodic conditions for the storage of device electricity. In this study, the numeric errors are reduced using high-resolution numbers of turbulence and a high-resolution advection method.

IV. RESULTS AND DISCUSSIONS

The segment discusses the physiognomies of the heat transfer of a wedge duct, with the result of various shapes of dimples or protrusions placed at different Reynold numbers on the heated end wall surface of a turbine blade.

4.1. Validation of numerical computations

The analysis done by Wang et al. (2019) [25] is cited and the conclusions are integrated into this report. Computations corresponding to the work by Wang et al. (2019) [25] have been done to validate the precision of the CFD findings. The geometry used to verify numerical computations was taken into account in the same manner as the geometry seen in Fig. 3.

➤ **For Re = 10000**

For this case fluid is flowing at Re =10000. The temperature, velocity, pressure contours and value of Nusselt number is shown below:

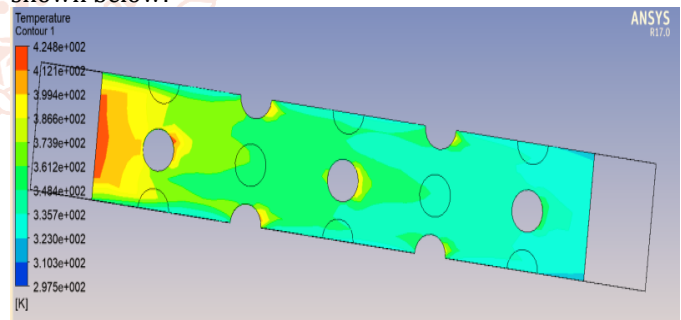


Figure.6. Temperature field of hemispherical dimple/protrusion at Re=10,000.

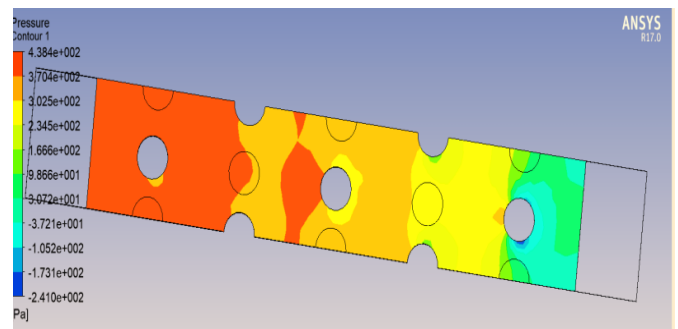


Figure.7. Pressure field of hemispherical dimple/protrusion at Re=10,000.

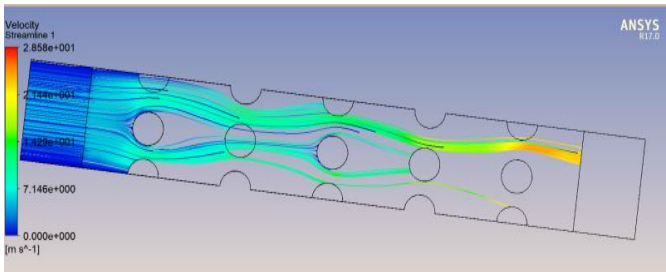


Figure.8. Velocity field of hemispherical dimple/protrusion at Re=10,000.

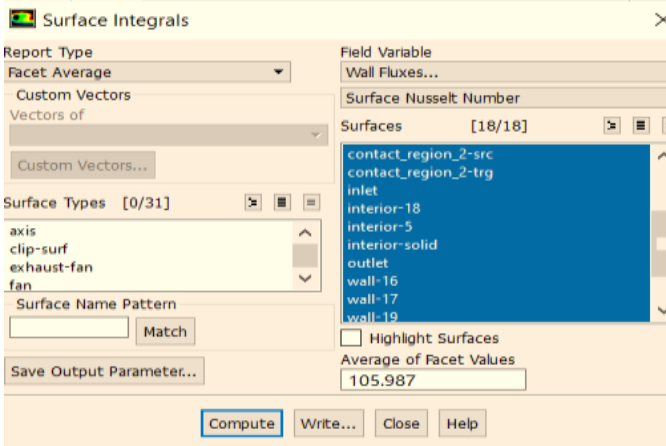


Figure.9. Value of Nusselt number of hemispherical dimple/protrusion at Re=10,000.

The value of the Nusselt was calculated at a particular Reynold's number by way of CFD analysis. In contrast to values derived from analyses by Wang et al. (2019) [25], the values of the CFD modelling estimates in the Nusselt numbers were compared.

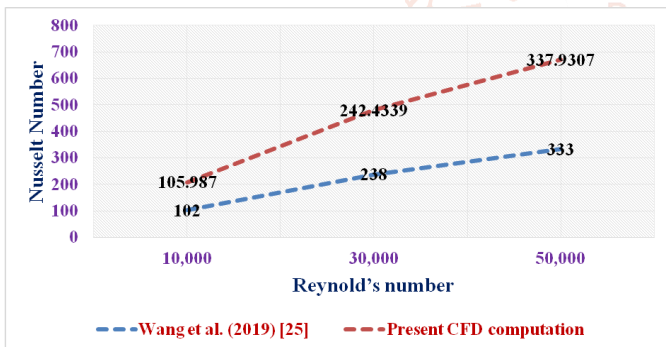


Figure 10. Comparison of the numerical Nusselt number values of CFD results with the results of Wang et al. [25].

In the current work, the computations were carried out under the same conditions as the experiment [25], and for the validity of the model, comparisons of the Nusselt number values with those of the experimental results were carried out. It is noted that their variance patterns are qualitatively compatible when comparing the numerical findings with the experimental evidence. This show that the computational model on the Nusselt number estimation has good precision.

4.2. Effect of triangular shape dimples/protrusions on the temperature field, Pressure field, Velocity field, and on Nusselt number at different Reynold's number

It is seen from the numerical findings and prior work that variance patterns in the Nusselt number values are qualitatively stable. We therefore take three Reynold numbers, i.e. 10,000, 20,000, and 30,000, to examine the influence of triangular form dimples/protrusions on the

temperature field, pressure field, velocity field, and on Nusselt number at various Reynold numbers. The boundary conditions were similar.

➤ For Re = 10000

Fluid moves at Re = 10 000 in this scenario. Below are the temperature, velocity, pressure contours and value of the Nusselt number:

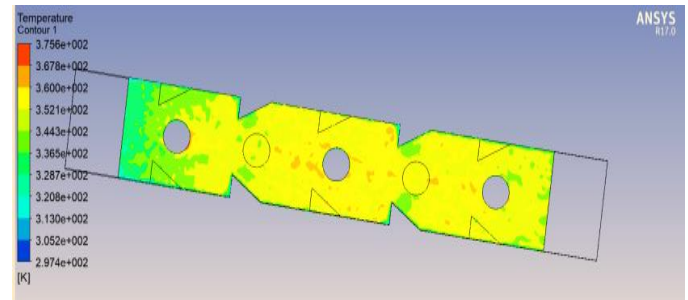


Figure.11. Temperature field of triangular shaped dimple/protrusion at Re=10,000.

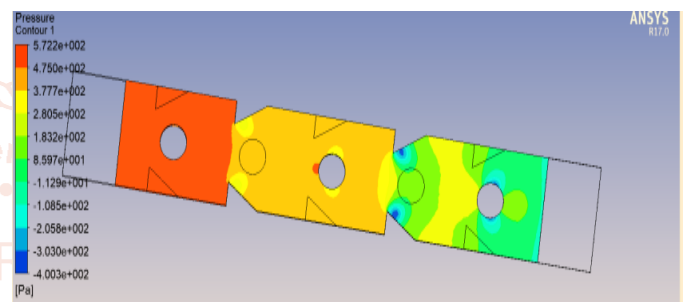


Figure.12. Pressure field of triangular shaped dimple/protrusion at Re=10,000.

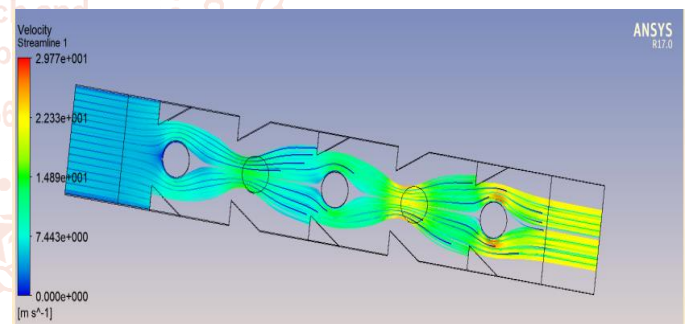


Figure 13. Velocity field of triangular shaped dimple/protrusion at Re=10,000.

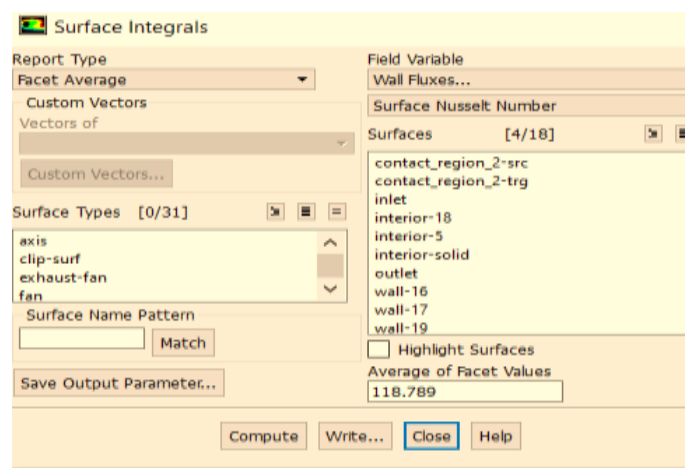


Figure.14. Value of Nusselt number for triangular shaped dimple/protrusion at Re=10,000.

4.3. Comparison between hemispherical and triangular shaped dimples/protrusions

Different dimples or protrusion shapes for turbine blades are contrasted in this segment to enhance previous understandings and to differentiate the contribution of dimples or protrusion shapes to the overall thermal efficiency of the proposed system.

Table 4. Comparison of the value of the Nusselt number for hemispherical and triangular shape dimples/protrusions

S. No.	Reynold's number	Nusselt number	
		Hemispherical dimple and protrusion (Conventional design)	Triangular dimple and protrusion (Proposed design)
1.	10,000	102	118.789
2.	30,000	238	298.408
3.	50,000	333	355.841

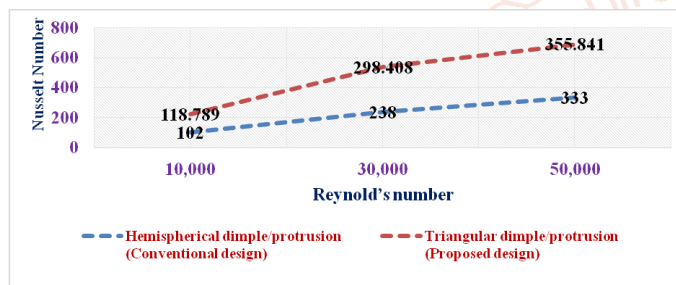


Figure 15. Comparison of the value of the Nusselt number for hemispherical and triangular shape dimples/protrusions.

V. CONCLUSIONS

The cooling performance of turbine blades with the effect of various shapes of dimples/protrusions placed on the heated end wall, i.e. hemi-spherical and triangular-shaped dimples/protrusions, was numerically investigated in this research. Within the Re range of 10,000-50,000, computational investigations were conducted to analyse the flow structures, heat transfer distributions, and Nusselt number in detail. The following is the specific conclusions:

- ❖ Due to flow acceleration, increase in impingement area and shrinkage of the flow recirculation region within the dimple, pin fin-dimple wedge duct with triangular shape dimples/protrusions provides improved heat transfer enhancement.
- ❖ Since improving the dimple/protrusion systems, the channel's cooling efficiency increased.
- ❖ The Nusselt number for the proposed solution was 12.34 percent higher than the standard design at various Reynold number values, as the overall Nusselt number increased.
- ❖ For Reynold's number = 50,000, the normalised area-averaged Nusselt number indicates the strongest improvement in heat transfer.

The solid surface temperature decreases substantially with the configuration of the shape of triangular dimples/protrusions relative to the shape of hemispherical dimples/protrusions from the temperature distributions on the solid surfaces in the turning bend.

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