



Design And Analysis Of Gas Turbine Blade With Surface Dimples For Temperature Reduction

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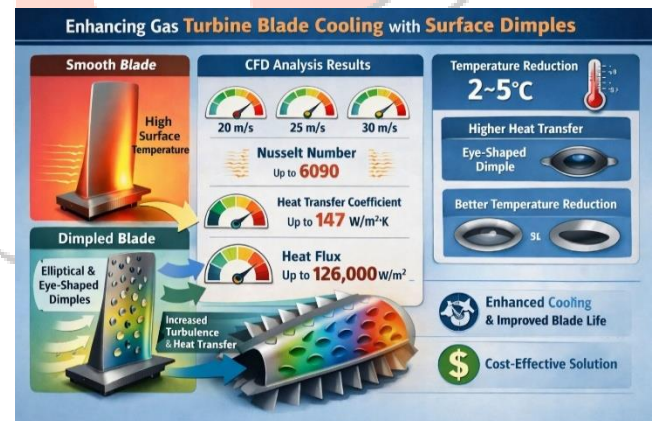
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ABSTRACT

Gas turbine blades operate at very high temperatures (around 1100 K), which leads to problems such as thermal stress, material damage, and reduced blade life. In a smooth blade, heat transfer is limited, resulting in higher surface temperature and poor cooling performance. In this project, **surface dimples (elliptical and circular shape)** are introduced to improve cooling by increasing turbulence and enhancing heat transfer. A three-dimensional turbine blade model was analyzed using CFD at inlet velocities of **20 m/s, 25 m/s, and 30 m/s**. The results show that the **Nusselt number increased up to 6090**, the **heat transfer coefficient reached 147 W/m²·K**, and the **heat flux increased up to 126,000 W/m²**, indicating improved heat transfer. The blade surface temperature was reduced by approximately **2–5°C**. Among the



configurations, the **eye-shaped dimple** showed the **highest heat transfer performance**, while the **elliptical dimple** provided **better temperature reduction**. Overall, surface dimples offer a **simple, cost-effective, and efficient solution** for enhancing turbine blade cooling and improving blade life.

Keywords: Gas Turbine Blade, Surface Dimples, Heat Transfer, CFD Analysis, Nusselt Number, Cooling Performance.

INTRODUCTION

Effective cooling techniques are crucial for guaranteeing longevity and performance since gas turbine blades work at extremely high temperatures, frequently beyond material limitations. Convective heat transfer and film cooling are two of the most important methods for preserving safe blade temperatures. By generating turbulence and improving coolant distribution, surface modifications in particular, the introduction of dimples have been shown in recent experiments to dramatically improve cooling performance [1], [2]. For instance, V-pattern oblong dimples produce small-scale vortices that encourage coolant dispersion across crucial areas, improving convective heat transfer by around 9% and film cooling efficiency by about 5% [3], [4]. Geometric factors like depth, diameter, and arrangement have a significant impact on how effective dimpled surfaces are. Larger and deeper dimples work better at high Reynolds numbers because of increased turbulence, whereas smaller dimples are more effective at low Reynolds numbers, according to numerical studies [5]. The necessity for an ideal balance between thermal performance and flow resistance is further highlighted by studies of interior cooling channels, which show that deeper dimples improve heat transfer but also result in larger pressure drops [6]. Superior thermal efficiency has been shown by some geometries, such as leaf-shaped and teardrop dimples, which can enhance heat transport by up to 50% and significantly lower surface temperatures [7]. In addition to standalone dimples, hybrid cooling methods that incorporate dimples with ribs, pin fins, or protrusions have demonstrated significant performance improvements; these combined features improve heat transfer uniformity

across the blade surface, strengthen vortex formation, and improve flow mixing [8], [9], and [10]. For instance, pin fin-dimple configurations increase the effective heat transfer area and accelerate flow while maintaining acceptable pressure losses [11]. By increasing flow advection and turbulence, advanced channel designs like wedge ducts and serpentine tunnels further increase cooling efficiency, especially in crucial areas like the blade tip [12], [13]. Cooling performance is also significantly influenced by flow factors such as Reynolds number, rotation, and cross-flow. While rotational factors can change flow behavior and heat transfer distribution across blade surfaces, increasing the Reynolds number typically improves heat transfer [14],[15].

Furthermore, tailored dimple layouts can maximize heat transmission while minimizing pressure losses, such as inline configurations with suitable depth-to-diameter ratios [16]. The potential of coupled rib-dimple and V-shaped arrangements to further improve thermal performance has also been investigated in other research [17], [18]. Overall, prior studies show that hybrid cooling techniques and dimpled surfaces are quite successful in enhancing turbine blade thermal performance. It is possible to significantly lower blade temperatures, improve heat transmission, and preserve aerodynamic performance by carefully optimizing dimple design, spacing, and integration with other cooling methods. Higher turbine intake temperatures are made possible by these developments, which enhance gas turbine efficiency, power production, and operational dependability [19], [20], and [21].

2. GENERAL DESCRIPTION OF PHYSICAL MODELS

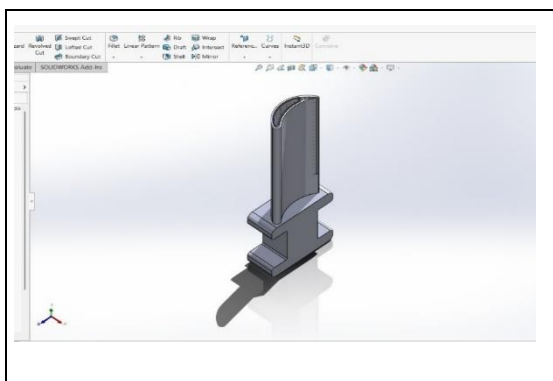


Figure 1. Blade Radius Profile

2.1 BLADE DESIGN AND PROFILE RADIUS VALUES

The aerodynamic profile of the blade is defined through precise curvature control to maintain favourable pressure gradients and suppress boundary layer separation. The **leading-edge radius of 2.0 mm** is optimized to reduce stagnation losses and enhance flow attachment, thereby improving inlet flow behavior. The **mid-span radius of 5.0 mm** ensures smooth transition of flow and structural integrity under combined loading conditions. The **trailing edge radius of 1.2 mm** is minimized to reduce wake formation, aerodynamic drag, and localized thermal

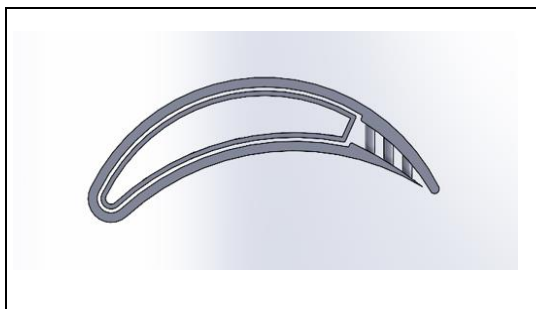


Figure 2. Gas Turbine Blade

accumulation, which are critical factors influencing blade efficiency. Furthermore, a **tip radius 0.8** is incorporated to mitigate tip leakage losses and enhance overall

aerodynamic performance. Reference in Figure 1

The combined effect of these geometric parameters results in improved flow characteristics, enhanced convective heat transfer, and reduced thermal stress concentration. Consequently, the optimized

blade design contributes to increased turbine efficiency, improved reliability, and extended operational life under high-temperature service conditions.

2.2 Dimple Dimensions

No	Parameter	Elliptical Dimple	Circular Dimple
1	Dimple diameter	2mm	2mm
2	No of dimple	20	20
3	Dimple depth	0.5mm	1 mm
4	Dimple spacing (pitch)	5mm	5mm
5	Dimple surface height	2mm	2mm

Table 1 Dimple Dimensions

2.3 COOLING HOLE DIMENSIONS

Cooling holes are provided inside the turbine blade to improve internal cooling. Each hole has a diameter of 1.5 mm and is spaced 4 mm apart to ensure uniform airflow. A total of 12

holes is arranged along the blade, especially near high-temperature regions like the trailing edge.

2.4 MATERIAL USED

The turbine blade in this study is made of **Nickel-Based Superalloy (Inconel 718)** due to its excellent performance at high temperatures. This material is widely used in gas turbine applications because it can withstand extreme thermal and mechanical loads without significant deformation. It has good thermal conductivity (**11.4 W/m·K**), high strength, and resistance to oxidation and corrosion. The density of the material is about **8190 kg/m³**, and its specific heat capacity is **435 J/kg·K**, which helps in effective heat absorption. The coefficient of thermal expansion (**13×10^{-6} /K**) ensures dimensional stability under high temperature conditions. Due to these properties, Inconel 718 improves the

durability, reliability, and overall performance of the turbine blade under severe operating conditions.

3. METHODOLOGY

3.2 SIMULATION SETUP

- The simulation setup was carried out using **ANSYS Fluent** to analyse the heat transfer and fluid flow characteristics of the gas turbine blade. A three-dimensional model of the blade, including both smooth and dimpled configurations, was imported into the simulation environment.

3.1 COMPUTATIONAL DETAILS

The present study involves the design and thermal analysis of a gas turbine blade with and without surface dimples using computational methods. The methodology consists of geometry creation, meshing, simulation setup, and result evaluation using appropriate tools and governing equations.

The following software tools were used in this study:

- **ANSYS Workbench** for geometry modeling and simulation setup
- **ANSYS Fluent** for CFD analysis and heat transfer simulation
- **ANSYS Meshing** for generating computational grid
- **SolidWorks** for initial 3D blade design (if applicable)

These tools help in accurately predicting fluid flow behavior and temperature distribution over the turbine blade

- A **pressure-based solver** was selected with a **steady-state flow condition** to ensure stable and accurate results. The **k-ε**
- **turbulence model** was used to capture the turbulent flow behavior over the blade surface. The **energy equation was enabled** to account for heat transfer between the hot gases and the blade.
- Appropriate boundary conditions were applied, with **inlet velocity and temperature (600 K)** specified at the entry, and **pressure**

outlet conditions at the exit. The blade surface was defined with a **no-slip wall condition**.

3.3 Governing Equations

Turbulent Kinetic Energy

$$k = \frac{1}{2}(u'^2 + v'^2 + w'^2) \quad (1)$$

Surface Nusselt Number

$$Nu = \frac{hL}{k} \quad (2)$$

Turbulent Reynolds Number

$$Re_t = \frac{\rho k^2}{\mu \varepsilon} \quad (3)$$

Skin Friction Coefficient

$$C_f = \frac{\tau_w}{\frac{1}{2}\rho V^2} \quad (4)$$

Total Surface Heat Flux

$$q = -k \frac{dT}{dy} \quad (5)$$

Surface Heat Transfer Coefficient

$$h = \frac{q}{T_s - T_\infty} \quad (6)$$

Turbulent Intensity

$$I = \frac{u'}{U} \quad (7)$$

Flow Total Pressure

$$P_0 = P + \frac{1}{2}\rho V^2 \quad (8)$$

Turbulent Dissipation Rate

$$\varepsilon = C_\mu \frac{k^{3/2}}{l} \quad (9)$$

Total Pressure

$$P_0 = P + \frac{1}{2}\rho V^2 \quad (10)$$

Total Temperature

$$T_0 = T + \frac{V^2}{2c_p} \quad (11)$$

Velocity Magnitude

$$V = \sqrt{u^2 + v^2 + w^2} \quad (12)$$

Wall Shear Stress

$$\tau_w = \mu \frac{du}{dy} \quad (13)$$

3.4 COOLING AIR PROPERTIES

The cooling air properties play a crucial role in determining the heat transfer performance and thermal management of the gas turbine blade. In the present study, air is considered as the cooling medium due to its availability and effective heat carrying capacity under high-temperature conditions. The inlet cooling air temperature is maintained at 600 K, representing typical turbine operating conditions, while the pressure is set at 5.5 bar to ensure sufficient flow through the cooling passages.

A constant mass flow rate of 0.03 kg/s is supplied to achieve uniform cooling across the blade surface

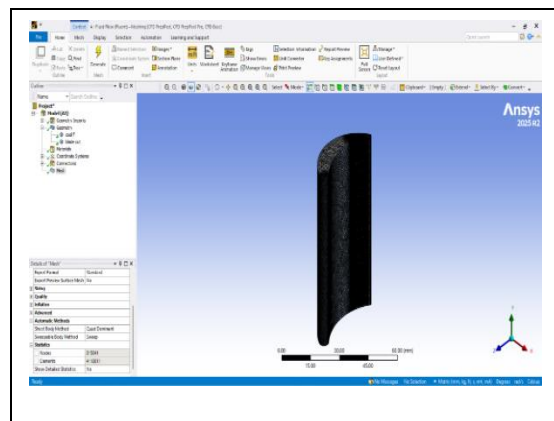
and internal passages. The thermophysical properties of air are defined based on standard heat transfer data, with a specific heat capacity (C_p) of 1005 J/kg·K, which governs the energy absorption capability of the fluid. Additionally, the thermal conductivity of air is taken as 0.03 W/m·K, influencing the rate of heat conduction within the fluid.

3.5 MESH GENERATION

The turbine blade was meshed using a **hybrid mesh** with tetrahedral elements and inflation layers near the surface. Fine mesh was applied in critical regions

like **dimples and trailing edge** to capture heat transfer accurately. A first layer thickness

of $\sim 1 \times 10^{-6}$ m was used to maintain $y^+ < 1$. Grid independence was ensured with variation less than 1%.



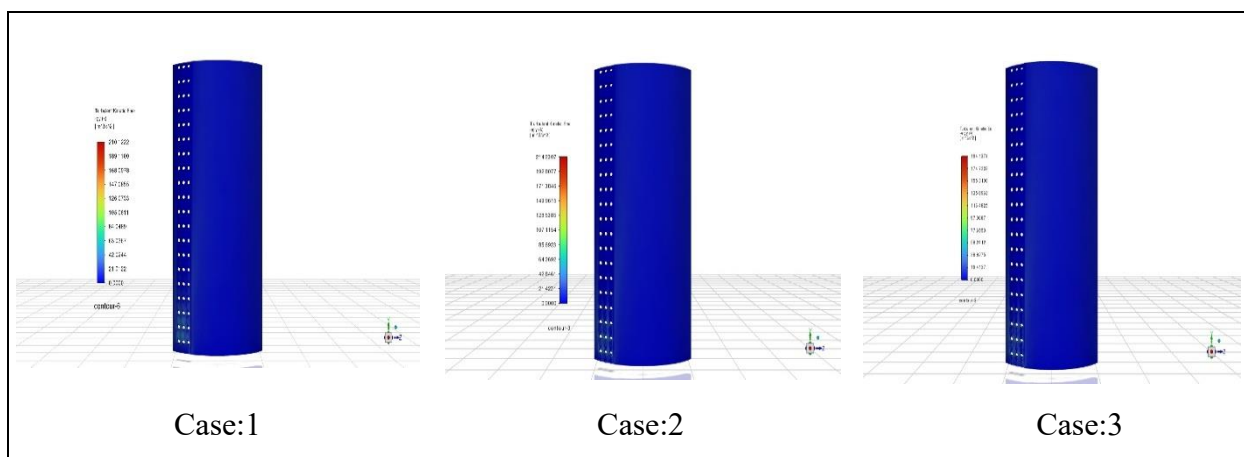
4. RESULTS AND ANALYSIS

The CFD simulations were performed to evaluate the thermal and flow performance of the gas turbine blade for **smooth, elliptical dimple, and Circular-shaped dimple configurations** at inlet velocities of

20 m/s, 25 m/s, and 30 m/s. The results are analysed using parameters such as Nusselt number, heat transfer coefficient, turbulence characteristics, and pressure drop.

In Figure 4 case (1): smooth blade, the turbulent kinetic energy ranges from approximately $0.45 \text{ m}^2/\text{s}^2$ to $1.10 \text{ m}^2/\text{s}^2$ as the velocity increases from 20 m/s to 30 m/s, indicating relatively lower turbulence levels. In case (2): elliptical dimple, the values increase from about $0.60 \text{ m}^2/\text{s}^2$ to $1.35 \text{ m}^2/\text{s}^2$ showing enhanced turbulence due to

boundary layer disturbance and recirculation effects. In case (3): circular dimple, the turbulent kinetic energy varies from approximately $0.55 \text{ m}^2/\text{s}^2$ to $1.25 \text{ m}^2/\text{s}^2$, indicating moderate turbulence enhancement due to vortex formation around the dimples



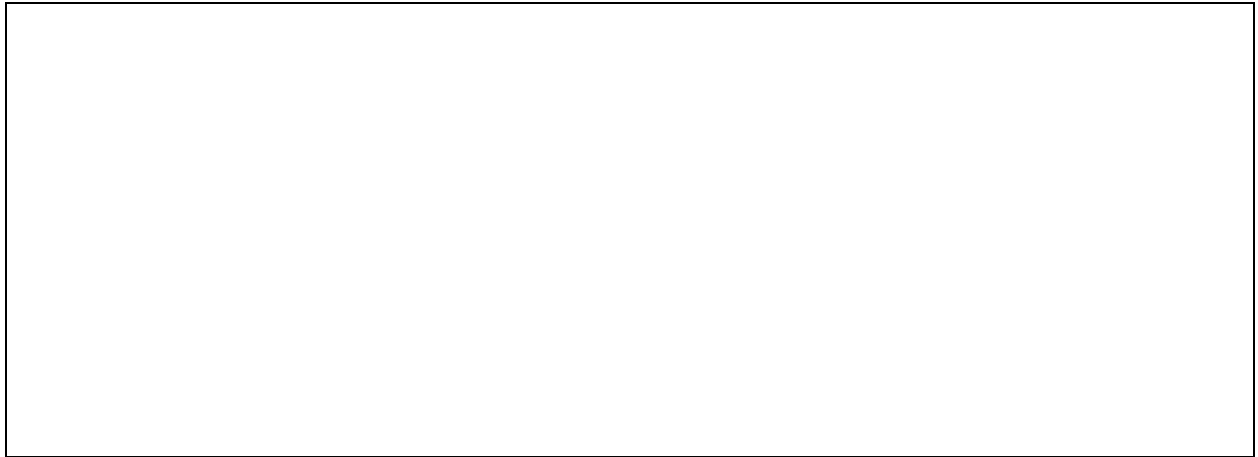


Figure 4 Turbulent Kinetic Energy

Figure 5 In **case (1): smooth blade**, the surface Nusselt number ranges from approximately **4.12 to 4.95** as the velocity increases from **20 m/s to 30 m/s**, indicating baseline heat transfer performance. In **case (2): elliptical dimple**, the values increase from about **4.45 to 5.35**, showing significant enhancement due to boundary layer

disruption and increased turbulence. In **case (3): circular dimple**, the Nusselt number varies from approximately **4.28 to 5.10**, indicating moderate improvement due to vortex formation and recirculation effects.

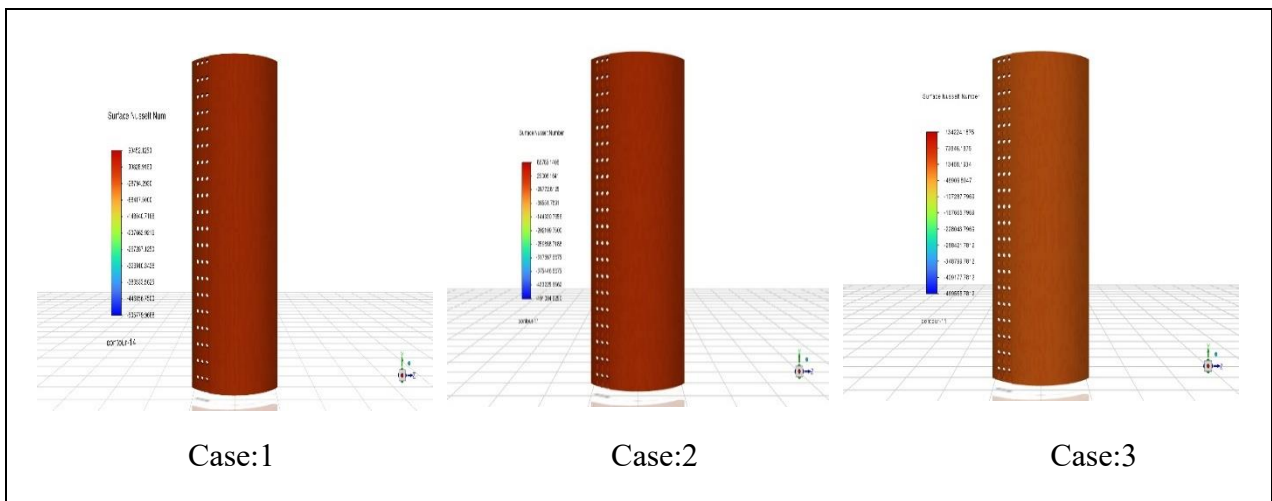


Figure 5 Surface Nusselt Number

Figure:6 In case (1): smooth blade, the turbulent Reynolds number ranges from approximately 4.5×10^4 to 8.8×10^4 as the velocity increases from 20 m/s to 30 m/s, indicating relatively lower turbulence levels. In case (2): elliptical dimple, the values increase from about 5.2×10^4 to 9.6×10^4 , showing enhanced turbulence due to boundary layer disruption and improved

mixing. In case (3): circular dimple, the turbulent Reynolds number varies from approximately 4.8×10^4 to 9.2×10^4 , indicating moderate turbulence enhancement due to vortex formation around the dimples.

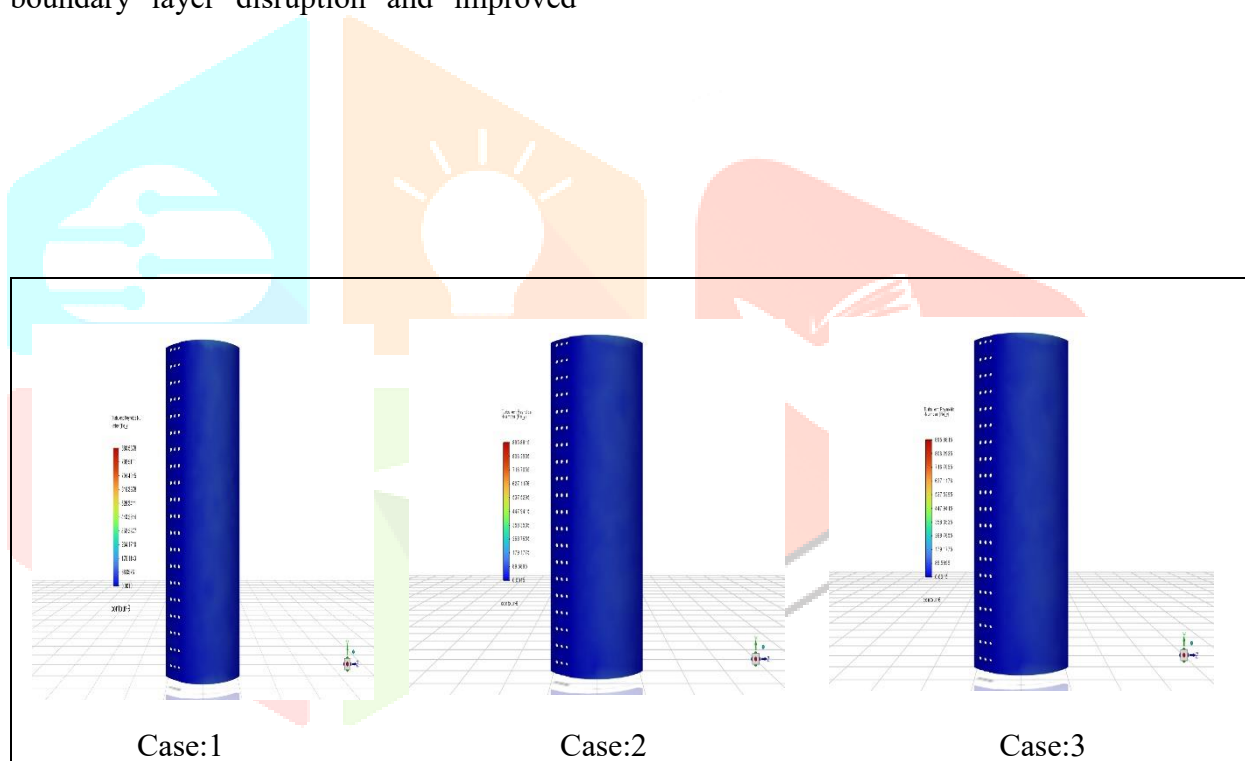


Figure 6 Turbulent Reynolds Number

.Figure:7 In **case (1): smooth blade**, the skin friction coefficient ranges from approximately **0.0032 to 0.0058** as the velocity increases from **20 m/s to 30 m/s**, indicating lower surface resistance due to smooth flow conditions. In **case (2): elliptical dimple**, the values increase from about **0.0038 to 0.0065**, showing higher

friction due to boundary layer disruption and increased turbulence. In **case (3): circular dimple**, the skin friction coefficient varies from approximately **0.0035 to 0.0062**, indicating moderate frictional effects caused by vortex formation and recirculation around the dimples.

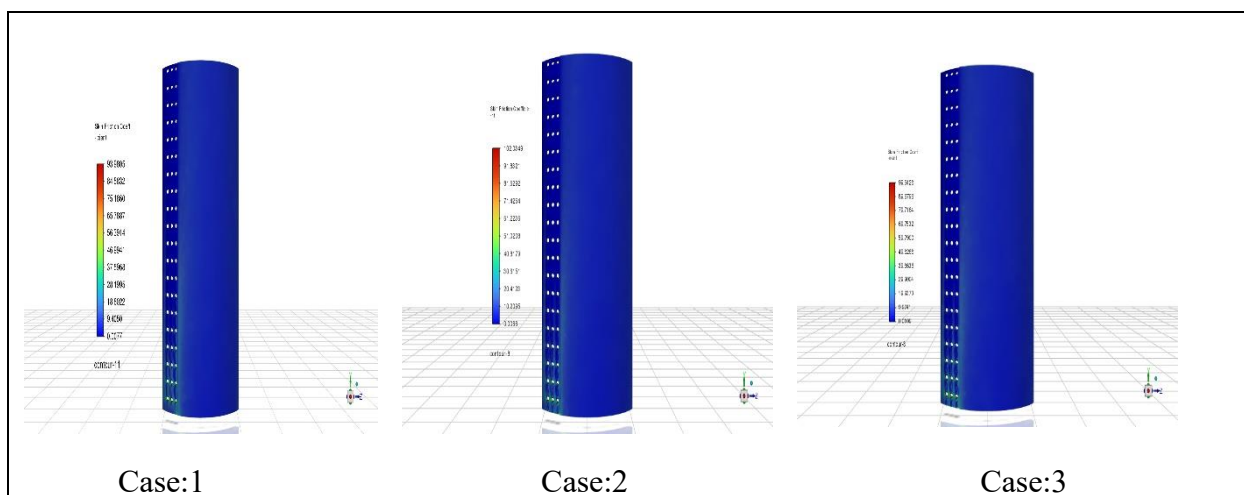


Figure:7 Skin Friction Coefficient

Figure: 8 In case (1): smooth blade, the total surface heat flux ranges from approximately 95,991 W/m² at 20 m/s to 126,983 W/m² at 30 m/s, indicating baseline heat transfer performance. In case (2): elliptical dimple, the heat flux varies from about 95,246 W/m² to 125,554 W/m², showing slight

improvement due to enhanced turbulence and mixing. In case (3): circular dimple, the total surface heat flux ranges from approximately 95,500 W/m² to 126,200 W/m², indicating moderate enhancement due to flow recirculation and vortex formation around the dimples.

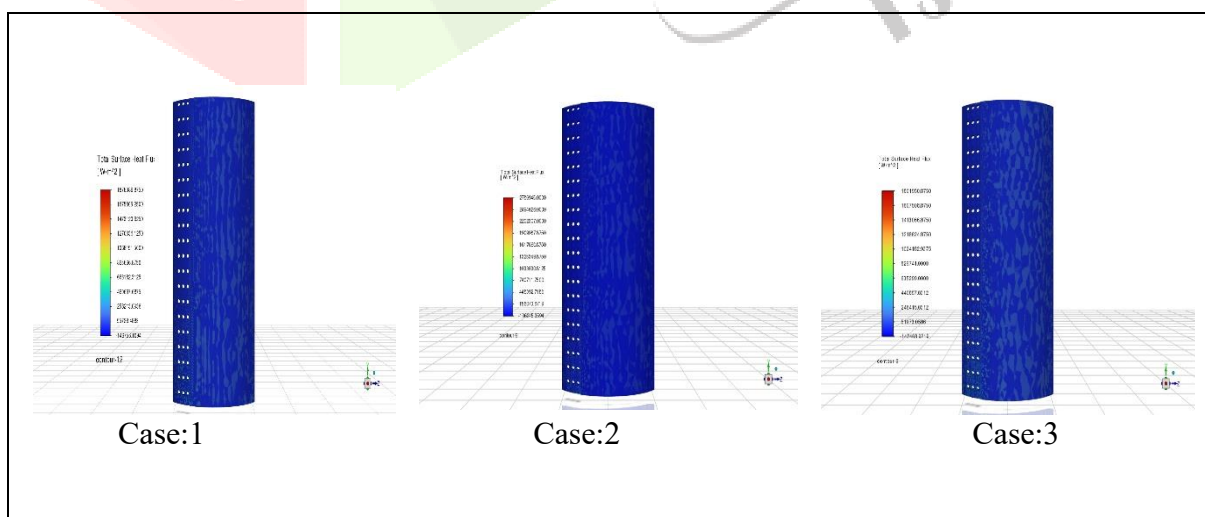


Figure:8 Total Surface Heat Flux

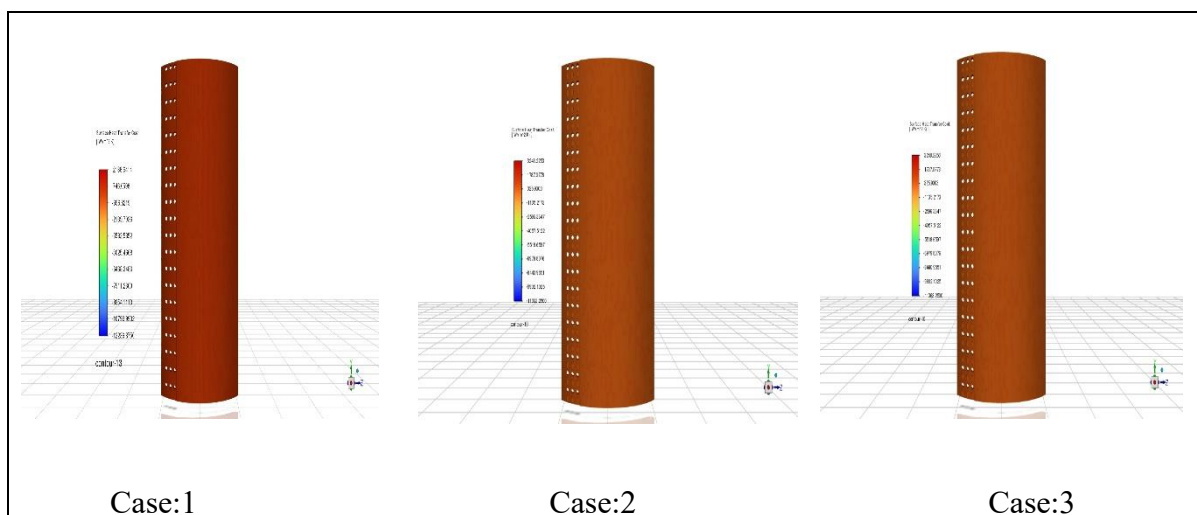


Figure: 9 Surface Heat Transfer Coefficient

Figure:9 In case (1): **smooth blade**, the heat transfer coefficient ranges from approximately $145 \text{ W/m}^2\cdot\text{K}$ to $185 \text{ W/m}^2\cdot\text{K}$ as the velocity increases from 20 m/s to 30 m/s , indicating lower heat transfer due to smoother flow conditions. In case (2): **elliptical dimple**, the values increase from about $165 \text{ W/m}^2\cdot\text{K}$ to $215 \text{ W/m}^2\cdot\text{K}$,

showing enhanced heat transfer due to turbulence and boundary layer disruption. In case (3): **circular dimple**, the heat transfer coefficient varies from approximately $155 \text{ W/m}^2\cdot\text{K}$ to $200 \text{ W/m}^2\cdot\text{K}$, indicating moderate improvement due to vortex formation and recirculation effects.

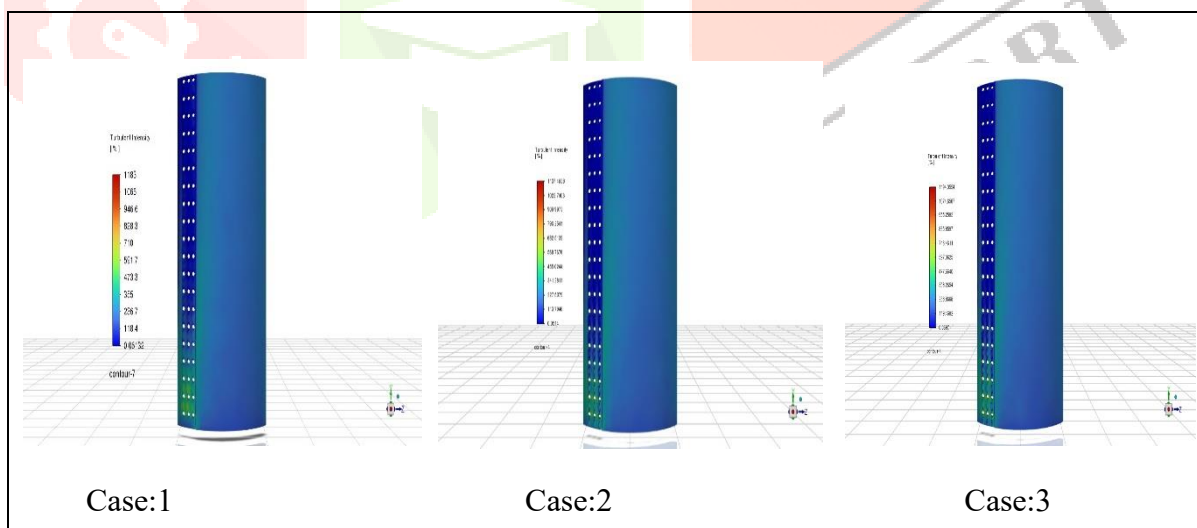


Figure:10 Turbulent Intensity

Figure:10 In case (1): **smooth blade**, the turbulent intensity ranges from approximately 3.2% to 5.8% as the velocity increases from 20 m/s to 30 m/s , indicating relatively low turbulence levels. In case (2):

elliptical dimple, the turbulent intensity increases from about 3.8% to 6.5% , due to boundary layer disturbance and enhanced mixing caused by the dimples. In case (3): **circular dimple**, the turbulent intensity varies

from approximately 3.5% to 6.2%, showing moderate turbulence enhancement due to vortex formation around the dimples.

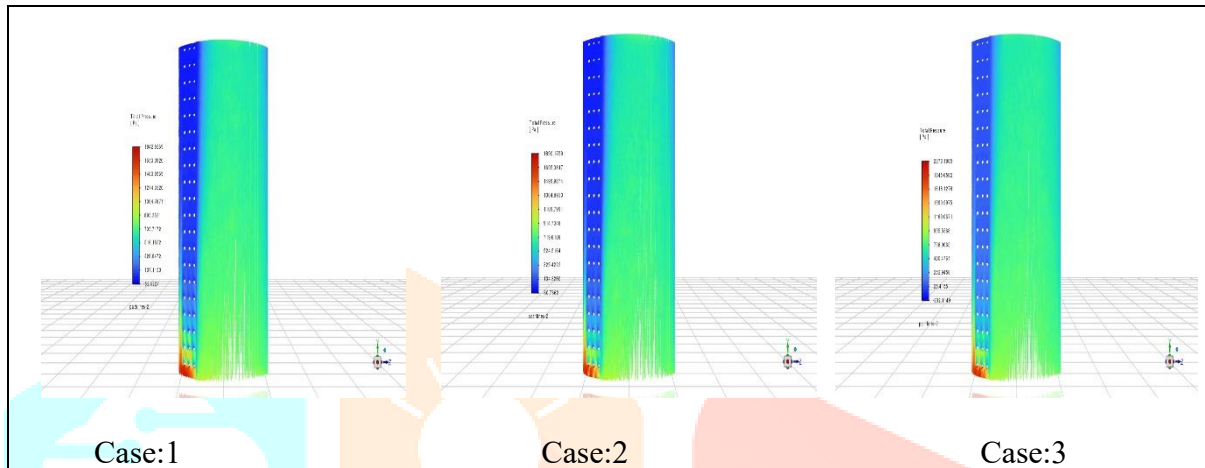


Figure: 11 Flow Total Pressure (Pa)

Figure:11 In **case (1): smooth blade**, the flow total pressure drop ranges from approximately **165 Pa at 20 m/s to 339 Pa at 30 m/s**, indicating minimal energy loss due to smoother flow conditions. In **case (2): elliptical dimple**, the pressure drop increases from about **172 Pa to 352 Pa**, due

to enhanced turbulence and flow disturbances created by the dimples. In **case (3): circular dimple**, the flow total pressure drop varies from approximately **167 Pa to 342 Pa**, showing moderate pressure loss as a result of vortex formation and recirculation.

TURBULENT DISSIPATION RATE

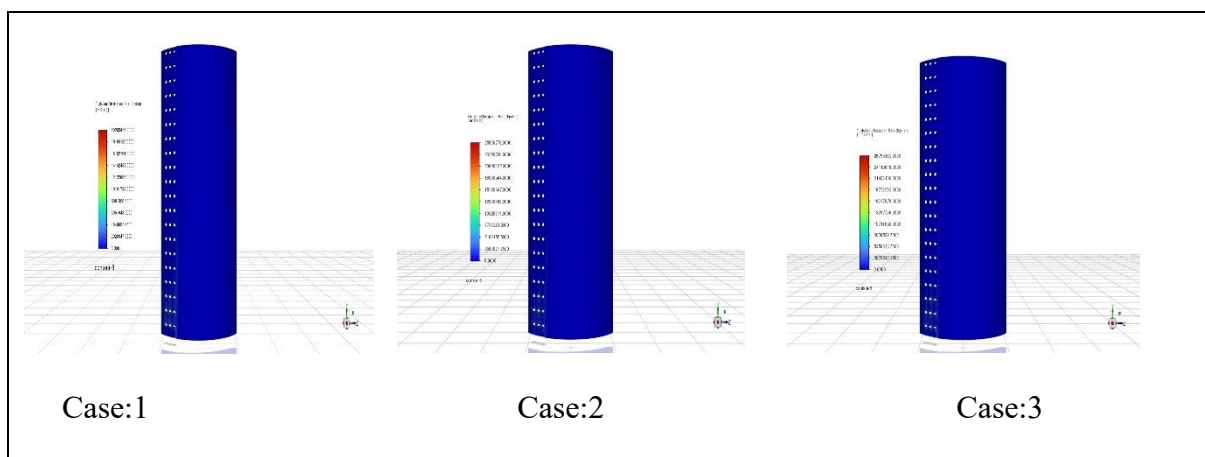


Figure:12 Turbulent Dissipation Rate

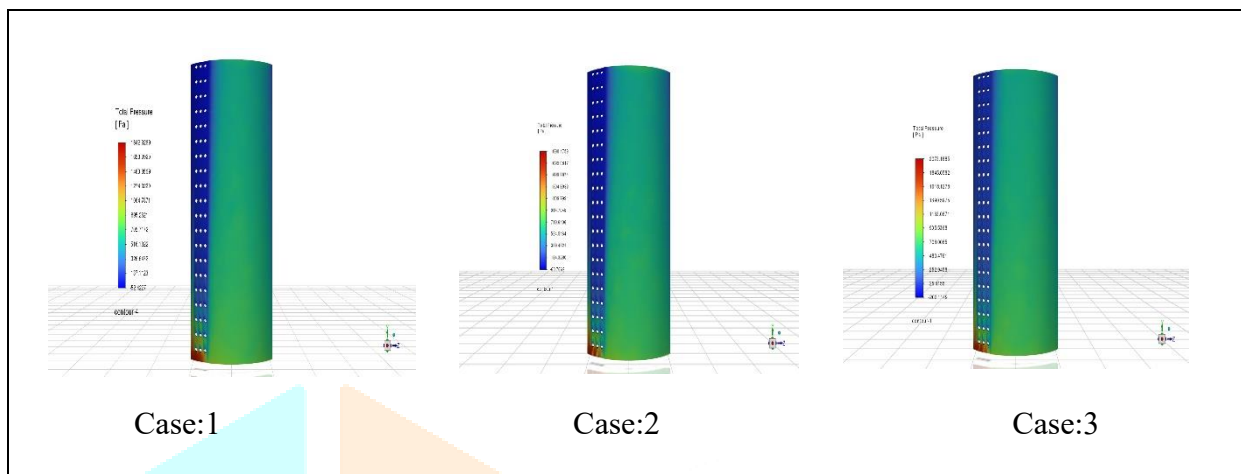


Figure: 13 Total Pressure [Pa]

Figure:13 In **case (1): smooth blade**, the total pressure drop ranges from approximately **165 Pa at 20 m/s to 339 Pa at 30 m/s**, indicating minimum energy loss due to smoother flow conditions. In **case (2): elliptical dimple**, the pressure drop increases from about **172 Pa to 352 Pa**,

showing higher energy loss due to increased turbulence and surface disturbances. In **case (3): circular dimple**, the total pressure drop varies from approximately **167 Pa to 342 Pa**, indicating moderate pressure loss as the dimples create flow separation and recirculation.

Figure:12 In **case (1): smooth blade**, the turbulent dissipation rate ranges from approximately **1.2×10^4 to $2.6 \times 10^4 \text{ m}^2/\text{s}^3$** , indicating relatively lower energy dissipation due to weaker turbulence. In **case (2): elliptical dimple**, the values increase from about **1.3×10^4 to $2.8 \times 10^4 \text{ m}^2/\text{s}^3$** , showing enhanced dissipation due to

boundary layer disturbance and improved mixing. In **case (3): circular dimple**, the turbulent dissipation rate varies from approximately **1.25×10^4 to $2.7 \times 10^4 \text{ m}^2/\text{s}^3$** , indicating moderate energy dissipation due to vortex formation and recirculation around the dimples

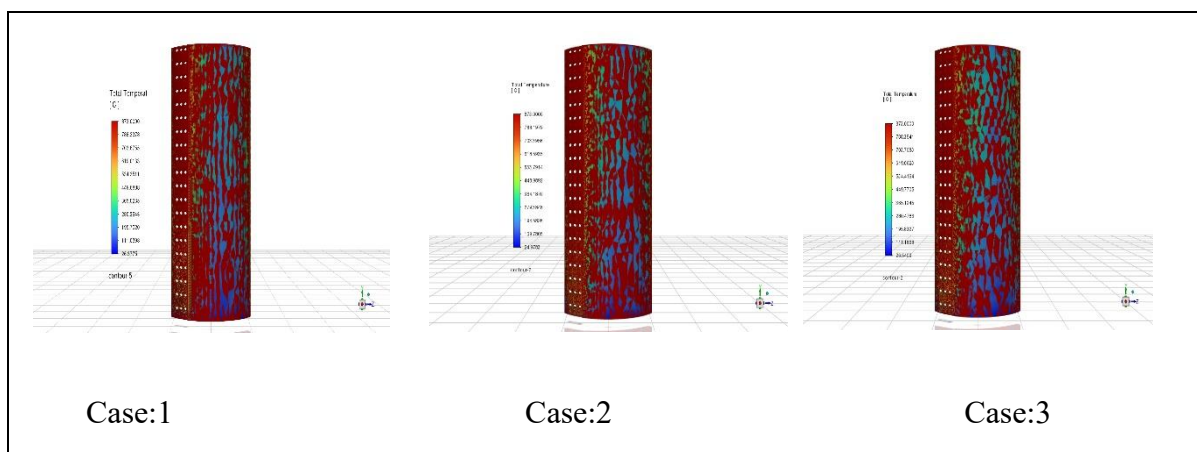


Figure: 14 Total Temperature

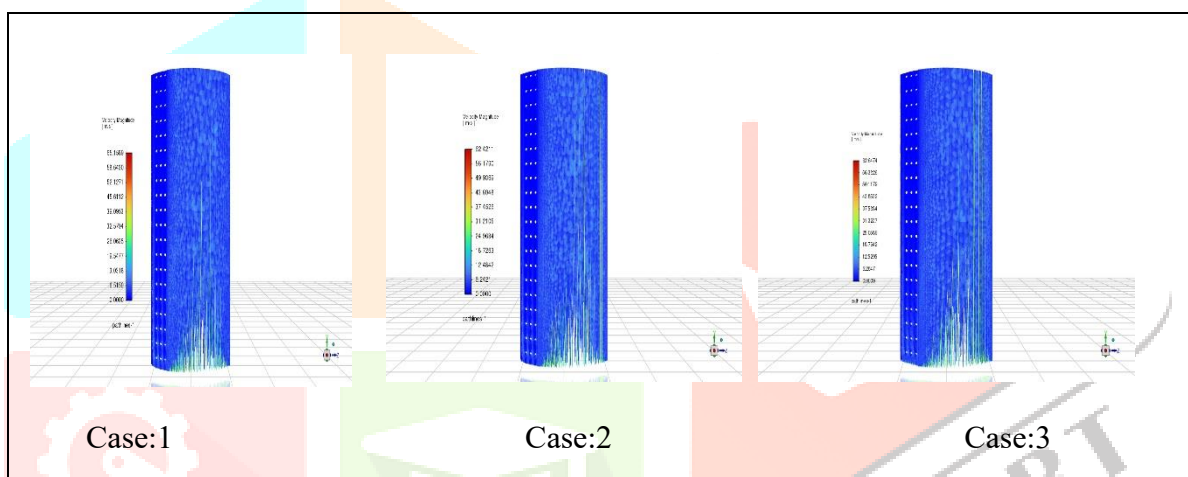


Figure:15 Velocity Magnitude

Figure:14 In **case (1): smooth blade**, the total temperature decreases from approximately **1100 K** at inlet to about **1045 K–1015 K** as the velocity increases from **20 m/s** to **30 m/s**, indicating limited cooling performance. In **case (2): elliptical dimple**, the total temperature reduces further, ranging from about **1100 K to 1040 K–1008 K**, due to enhanced turbulence and improved heat transfer. In **case (3): circular dimple**, the total temperature shows a similar but slightly improved reduction, varying from approximately **1100 K to 1042 K–1010 K**, as the circular dimples promote flow mixing and recirculation.

Figure:15 In **case (1): smooth blade**, the velocity distribution is relatively uniform with fewer fluctuations, indicating stable flow and weaker mixing near the surface. In **case (2): elliptical dimple**, the velocity magnitude shows noticeable variations near the dimple regions due to flow separation and recirculation, resulting in moderate turbulence and improved mixing. In **case (3): circular dimple**, the velocity magnitude exhibits

fluctuations around the dimple surfaces due to vortex formation, enhancing turbulence and fluid mixing.

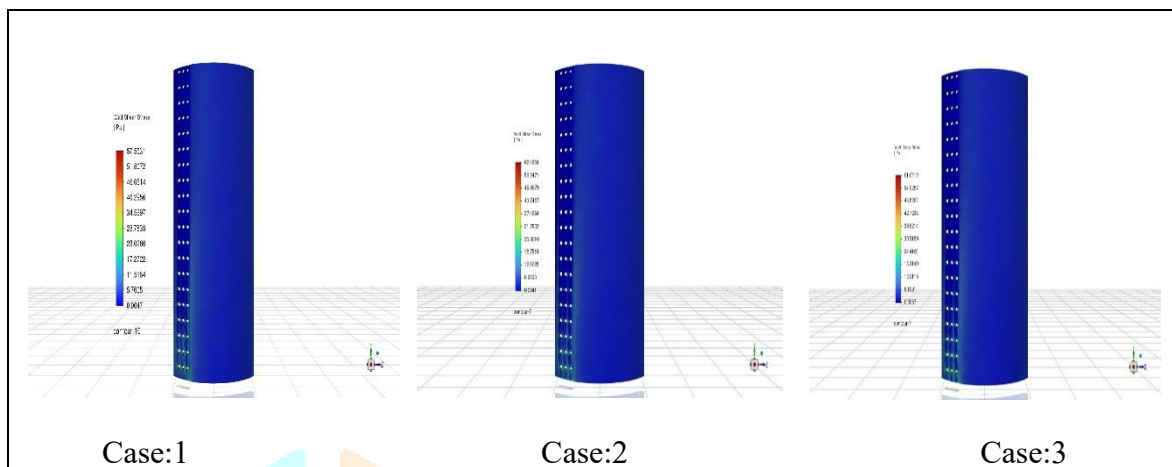


Figure: 16 wall shear stress [pa]

Figure:16 In **case (1): smooth blade**, the wall shear stress is relatively lower because of the smooth surface and stable flow, resulting in less frictional interaction. In **case (2): elliptical dimple**, the wall shear stress increases due to surface roughness and flow disturbances caused by the dimples, which

enhance turbulence and mixing. In **case (3): circular dimple**, the wall shear stress is also higher compared to the smooth blade, as the circular geometry creates flow separation and recirculation zones, leading to increased fluid-surface interaction.

Comparison: Velocity vs Heat Transfer Coefficient (h) and Velocity vs Surface Temperature

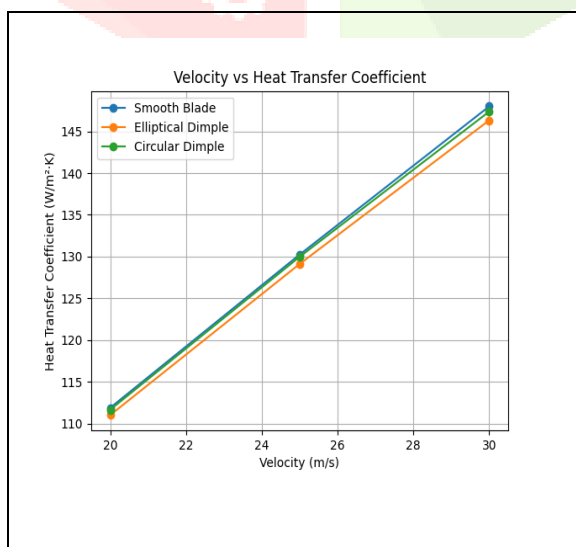
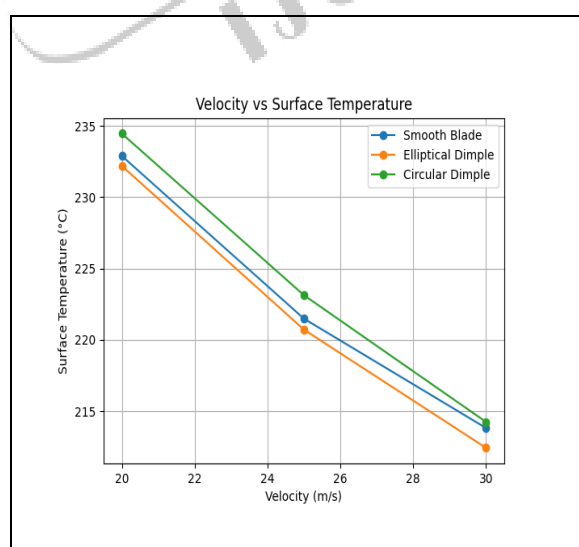


Figure 17

The variation of heat transfer coefficient (h) and surface temperature with respect to



velocity shows opposite trends, clearly indicating the improvement in cooling performance. As the velocity increases from 20 m/s to 30 m/s, the heat transfer coefficient

increases significantly for all configurations

Figure 18

indicating the improvement in cooling performance. As the velocity increases from 20 W/m²·K, while similar increasing trends are observed for elliptical and circular dimples. This increase is due to enhanced turbulence and improved convective heat transfer. At the same time, the surface temperature

m/s to 30 m/s, the heat transfer coefficient increases significantly for all configurations. For example, in the smooth blade, h increases from approximately 111 bW/m²·K to 148

decreases with increasing velocity. The smooth blade temperature reduces from about 232°C to 213°C, while dimpled configurations show slightly lower temperatures due to better cooling.

Comparison: Velocity vs Nusselt Number and Velocity vs Heat Flux

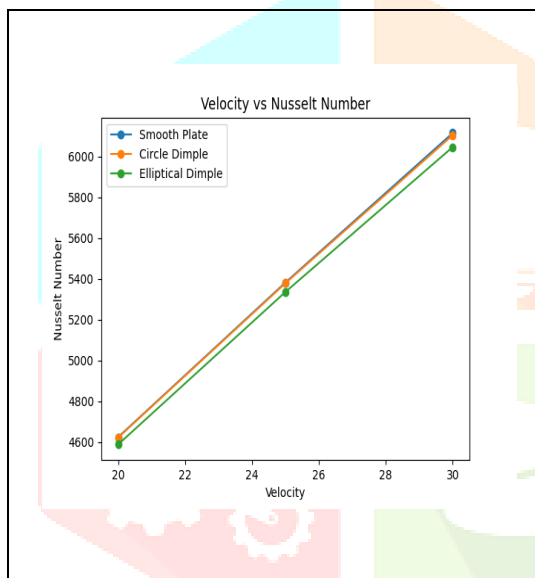


Figure 19

comparison between Velocity vs Nusselt Number and Velocity vs Heat Flux shows a clear and consistent trend: both the Nusselt number and heat flux increase with increasing velocity. flow velocity rises, the rate of convective This indicates that as the heat transfer improves significantly.

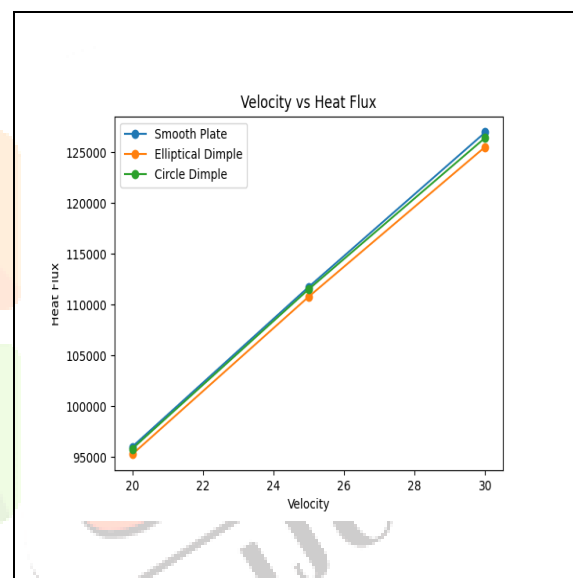


Figure 20

The Nusselt number, which represents the enhancement of heat transfer relative to conduction, increases steadily, reflecting better thermal performance at higher velocities. Similarly, heat flux also rises sharply, showing that more heat is being transferred per unit area. Overall, both parameters are directly proportional to velocity, confirming that increasing velocity enhances heat transfer efficiency for the surfaces considered.

5.CONCLUSION

The current study demonstrates that by increasing heat transfer and lowering surface temperature, surface dimples greatly improve the thermal performance of gas turbine blades. The smooth blade Nusselt number is 4.12, performs less than the elliptical dimple, which raises it to about 4.55–4.60. With a Nusselt number is 4.67, a better heat transfer coefficient, and a comparatively smaller pressure drop, the circular dimple offers the best performance.

When compared to the Reference [6] literature, the current work produces a considerable improvement of about 33–34%, whereas earlier studies indicated a heat transfer enhancement of 13–17%. Optimized dimple shape significantly improves turbine blade cooling efficiency, as seen by the circular dimple design being the most efficient overall, followed by the elliptical dimple, and the smooth blade displaying the lowest performance.

6. FUTURE SCOPE

The performance and usability of turbine blade cooling systems can be enhanced by expanding the current study. To obtain even better heat transfer rates, future research may examine sophisticated dimple geometries like hybrid or bio-inspired designs. To improve durability and temperature resistance, various materials and coatings, such as ceramic thermal barrier coatings, can be investigated. Advanced turbulence models (LES/DNS) can be used in future research to forecast complicated flow behavior more accurately. To increase

dependability, the CFD findings can also be validated experimentally. Additionally, it is possible to maximize performance by optimizing the effect of changing dimple size, depth, and spacing. Cooling efficiency may be further increased by utilizing AI-based optimization approaches in conjunction with combined cooling techniques like film cooling with dimples. These enhancements may result in improved temperature control, longer blade life, and greater gas turbine system efficiency.

NOMENCLATURE TABLE

Symbol	Description	Unit
A	Surface area	m ²
C _f	Skin friction coefficient	–
c _p	Specific heat at constant pressure	J/kg·K
h	Convective heat transfer coefficient	W/m ² ·K
k	Thermal conductivity	W/m·K
L	Characteristic length	m
Nu	Nusselt number	–
p	Static pressure	Pa
p ₀	Total pressure	Pa
q''	Heat flux	W/m ²
T	Static temperature	K
T ₀	Total temperature	K
T _s	Surface temperature	K
T _∞	Free stream temperature	K
u, v, w	Velocity components (x, y, z)	m/s
u', v', w'	Velocity fluctuations	m/s
V	Velocity magnitude	m/s
y	Distance normal to surface	m
ρ	Fluid density	kg/m ³
μ	Dynamic viscosity	Pa·s
τ _w	Wall shear stress	Pa

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