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F class gas turbine blade cooling calculation

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Abstract

Gas turbines play a vital role in today's industrial environment. They are used for demand power and energy sources. There are two type of gas turbines; aero derivative and industrial types. Aero derivative gas turbines are from aero plain engines and derivate for use in power manufacturing. The other and most common ones are industrial type. Industrial types are huge size and produce great power output like 250-275 MW less or more. These units are called F class, H class. In this papper we are going to analyse the cooling process of the turbine blades. Cooling process is directly effecting the gas turbine performance. That is why it is so important. Before doing that we are going to introduce components of the gas turbine and their running prienciple. This will guide us for understanding better of calculation and importance of calculation.

Keywords: gas turbine, blade, cooling calculation, TBC.

1. Introduction

F class gas turbines consists of IGV section, compressor section, combustion chamber section, turbine section and exhaust section. These are the general component and used for every type of gas turbines. Below figure you can see the section of the gas turbine. For understanding the running prienciple of gas turbines we should know each sections listed above. Each sections have different duty and running conditions.

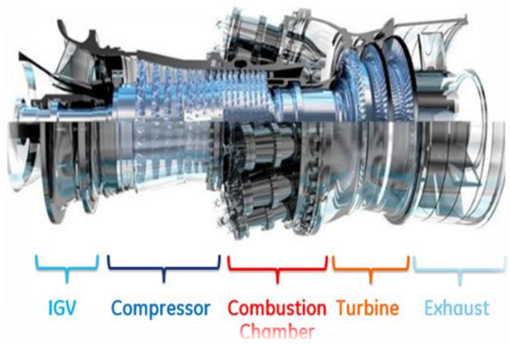

Figure 1. Gas turbine section

1.1. IGV (inlet guide vane) section

The air which came from air intake and passed from filters enter the unit from the IGV section. The air should be clear and without particules and should have sufficient temperature and pressure. Temperture and pressure direclty effect the power output. IGV blades can be rotate and adjust the air flow direction. Blades are controled by hydraulic control unit. All blades are connect to a pinion gear or rotable mechanism for control and movement. The air quantity can be adjust by open or close the blades. We adjust the air quantity for power output. If we need more power means that we need more air and open the IGV blades. If we need less power we just close the IGV blades. All these activiets are done under control of turbine logic. On the other hand IGV are used for safty of the unit. If unit trip for any reason IGV direclty close and cancel the air for damage the blades or other component of the unit. Following figres are about IGV section.

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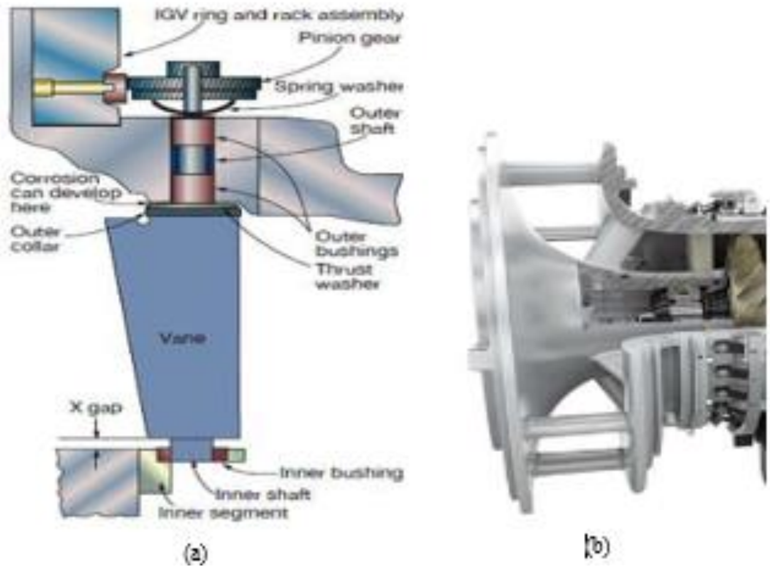


Figure 2. IGV blade position (a) and IGV view (b)

Chart above (Source of TURKSTAT-Turkish Statistical Institute)show that during period of 2003-2018 average annual growth rate in Turkey realized as 5.5%. Unfortuanlty economic growth started to decrease since 2018 and reached to deep point in 2019. When we analyzed the sectoral growth it is show that the economiy may start to growth within first quorter of 2020 as shown in Figure 3.

1.2. Compressor section

Compressor is consist of rotor and stator components. Rotor is rotating at about 3000 rpm and stator is guiding air for laminar flow. This section is compress air for more power output. The dimention of the blades that are used on the rotor and stator are different from each other. Each stage blades dimensions are different from each other. They star from the big dimentions and drops related to the stages. Number of stages can differ form each other related to the power of the unit. Most of them have 13-14 or 17 stages. Figure 3 show the compressor section.

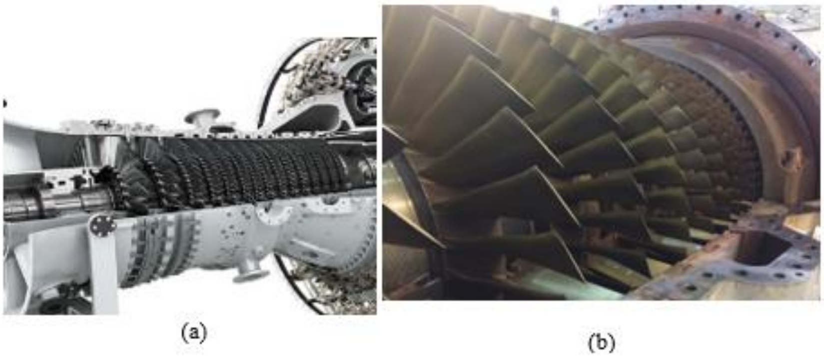
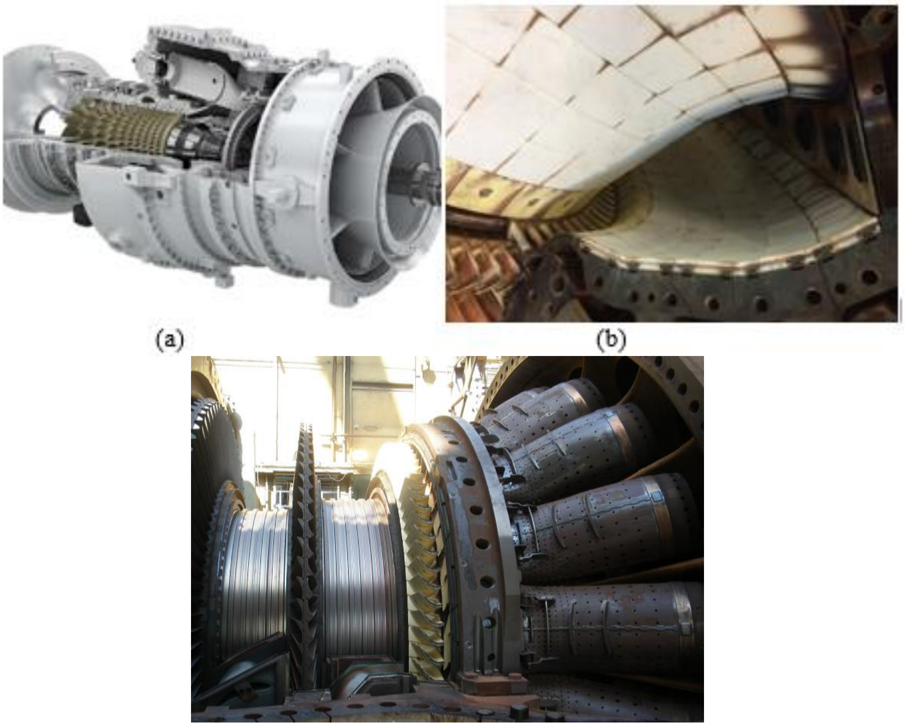


Figure 3. Compressor section (a), (b)

1.3. Combustion chamber section

The flame presents in the compustion chamber. The relevant quantity of air and fuel mixe in the combustion and ignited by electrical arc. There are two type of combustion chamber; annular and can annular. Annular means we have just one combustıon chamber around the unit. Can-annular means we have many combustion chmabers. Some one has 12, some one has 18 and so on. It is depend on the design.

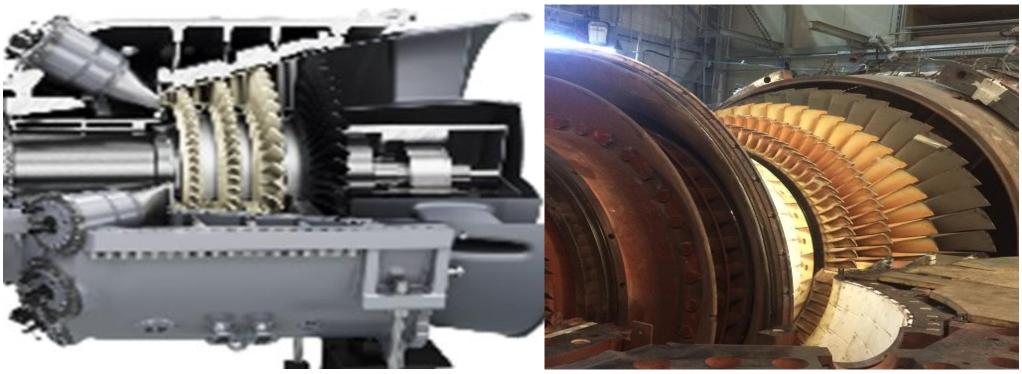


(c)

Figure 4. Combustion chamber section (a), (b),(c)

1.4. Turbine section

After combustion chamber hot air is directed to turbine section. It convert mechanical energy to electrical energy. That is why the efficiency of the turbine is very important for power output. The stage of the turbine is depend on the unit type. Some units have 4 stages and some units have 3 stages and so on. The blades of turbine section have TBC (Thermal Barier Cooting) for protect metal blades from hot air. They have also internal cooling holes. Otherwise the blades start to melt under hot air.



(a) (b)

Figure 5. Turbine section (a), (b).

1.5. Turbine section

After the turbine section we have exhaut section which consist the last bearing of the unit. Exhaust section guide the hot gas to the athmoster (simple cycle) or guide it to the boiler (combine cycle). It also contain tmpereture indicator for measure of the exhaust temperature to control the unit for logic.

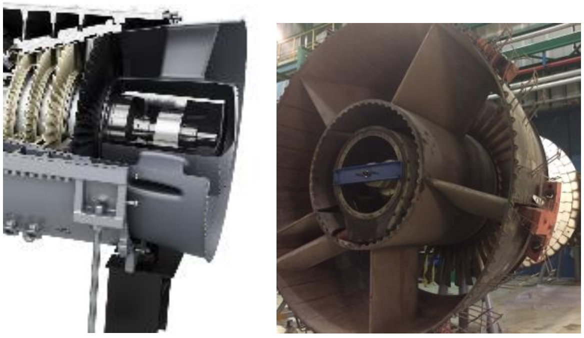


Figure 6. Exhaust section.

2. Turbine blades cooling calculation

The efficiency and power output of a gas turbine increases with higher turbine inlet gas temperature. Modern gas turbine vanes and blades are exposed to gas with temperatures which far exceeds the melting point of the component material. Thus, the blades and vanes have to be cooled in order to lower the temperature. When cooling the component it is important to know the correct boundary conditions, to avoid creating too large temperature gradients. Large temperature gradients cause thermal stresses and significantly decrease the component life.

Both internal and external cooling is used in turbine blades and vanes. The cooling air is extracted from the compressor. The cooling affects the gas turbine in two ways. First, less mass flow is available for combustion in the combustion chamber. Second, the trailing edge thickness has to be increased, which creates a larger wake behind the trailing edge which affects the aerodynamics negatively. The extraction of air decreases the efficiency of the turbine, since less air is available for power generation. Maximum cooling with minimum cooling air is therefore desired.

This work concerns internal cooling of turbine blades 1 and 2 on F type big size gas turbines. The main prienciple and runing conditions are semeler to all F type units. These components are situated next to the combustor chamber, and are therefore exposed to the highest temperatures.

If the cooling process is not enought for blades the TBC (Thermal Barier Cooting) start to demange and after then the blade materials start to melt. That is why cooling of the blades are critical and very important for life cycle and turbine performance. Below you can see figures of blade that melted under bad cooling condition.

figure 7. Melted blades under bad cooling condition.

Blades and vanes are cooled by internal channels, through which the cooling air flows in different schemes and configurations. The cooling air decreases the channel wall temperature by convective cooling. A number of cooling methods are applied to different part of the vane or blade. To make the cooling systems more efficient and spend a minimum of air, the cooling systems nowadays usually include features that increase the heat transfer coefficient and/or increasing the heat transfer surface area. The heat transfer coefficient is increased by enhancement of the flow turbulence and by breaking the flow boundary layer. The penalty paid for the increased heat transfer is higher pressure loss.

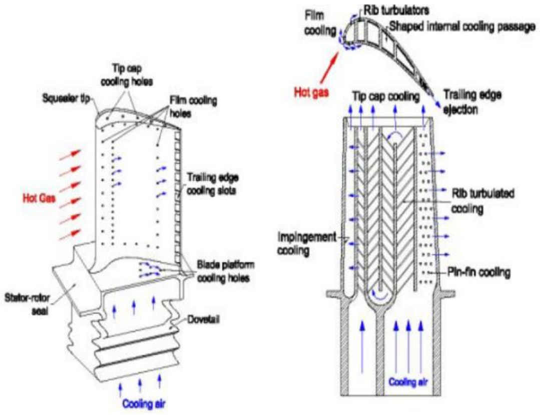


Figure 8. Cooling holes.

Many methods exist in theory, but only a handful are widely used in practice. Three are many cooling methods are available. Some of them are called rib turbulated cooling, matrix cooling and impingement cooling. Before starting to calculate the enginnering cooling on blades we should remember the main prienciple of heat transfer methods.

3. Flow and heat transfer basics

Much of the heat transfer in blades internal cooling systems takes place by convective cooling, where heat is transferred from the hot wall to the cooling air. The heat transfer coefficient - α is defined by equation

Qconv =α· A·(Tw −Tair)

TW : Metal Temperature (K)

Tair : Air Temperature (K)

A : Aria (m2)

α = heat transfer coefficient [W/(m2 K)]

To generalize heat transfer correlations, it is common to use non-dimensional parameters. The heat transfer coefficient is often made non-dimensional by the Nusselt number, defined in equation below

$$Nu = \frac{\alpha \cdot L_c}{\lambda}$$

Nu : Nusselt number (dimensionaless)

Lc : Charcteristic length according to table below (m)

λ = thermal conductivity of fluid [W/(mK)]

α = heat transfer coefficient [W/(m2 K)]

The characteristic length Lc varies for different geometries and some examples are given in Table 1.

Table 1. Examples of characteristic lengths.

Flow Case	Lc =
flat plate, local value	x
flat plate with length L, the hole plate	L
cylindrical pipe	d
non-cylindrical pipe	Dh

Another parameter that describes convective heat transfer is the Stanton number, see equation

St = Re/(Nu·Pr)

Where:

Re: Reynolds number

Nu: Nusselt number

Pr: Prandtl number

3.1. Friction

The fanning friction factor is defined from the pressure loss according to equation.

$$\Delta p = \frac{f}{4} \cdot \frac{L}{D_h} \cdot \frac{\rho \cdot \bar{v}^2}{2}$$

Δp: Pressure Loss,

f: friction factor,

L: length (m)

Dh:hydraulic diameter in rebber channel (m)

ρ: density (kg/m3)

ν: kinematic viscosity of fluid (m2/s)

The Darcy friction factor

$$\Delta p = f_D \cdot \frac{L}{D_h} \cdot \frac{\rho \cdot \bar{v}^2}{2}$$

The relationship between the fanning and the Darcy friction factor is displayed in equation.

$$f = \frac{f_D}{4}$$

4. Flow

Flow can be turbulent, laminar or in the transitional region between laminar and turbulent. Laminar flow occurs for non-disturbed flow with relatively low velocities and is characterised by even velocities and orderly motions. The opposite, turbulent flow, occurs at higher velocities and is characterised by velocity fluctuations and disordered motions. The Reynolds number describes the flow regime, see equation below. For example, flow in a pipe is laminar for Re < 2300 and turbulent for Re > 4000 approximately.

The Reynolds number is defined as the ratio of the inertia forces to the viscous forces in the fluid. The inertia forces depend on the flow kinetic energy and are a function of the fluid density and the square of flow velocity. The viscous forces depend on the fluid viscosity and the flow velocity. The characteristic length is the same as for the Nusselt number, see Table 1.

$$\mathrm{Re} = \frac{\rho \cdot \bar{v} \cdot L_c}{\mu}$$

Where:

ρ: density (kg/m3)
Lc : Charcteristic length according to table 1 (m)
ν = kinematic viscosity (velocity) of fluid [m2/s]
μ = dynamic viscosity of fluid [kg/(ms)]

The Prandtl number describes the thickness of the boundary layer, see equation.

$$\mathrm{Pr} = \frac{\nu}{\lambda/(\rho \cdot C_p)}$$

Where:
Cp : Specific heat at constant pressure (J/kgK)
λ: Thermaş conductivity of fluid (W/mK)

The definition of mass flow is important for correlations. Mass flow is defined in equation
m= ρ· A· ν
where:
A: aria (m2)
ν: velocity (m/s)
ρ: density (kg/m3)

A discharge coefficient is often used for mass flow definitions in correlations. It is defined in equation as.

$$C_d = \frac{\dot{m}_{real}}{\dot{m}_{ideal}}$$

5. Heat transfer calculation on blades during real running condition

In this stage we will calculate heat transfer on the first stage of blade during turbine real running condition. All calculated values have been taken from turbine DCS server. After then we will try to understand if our cooling is sufficient or not. If the cooling is not enought the TBC (thermal barrier coating) start to burne and after then blade inconal materil going start to melt. Below figures show the melted blades related to bad cooling condition.

Figure 9. Melted blades under bad cooling condition.

5.1. Objective of study

In this Study the cooling holes dimentions have been measured from the real turbine blades. The temperature and pressure have been taken from DCS when the unit is about 260 MW which quate near the full power. The other values have been taken from referance books. We will calculate the heat transfer during normal running condition by using engineering formula taken from refence hand books.

Below figures show that how a turbine blades cool.

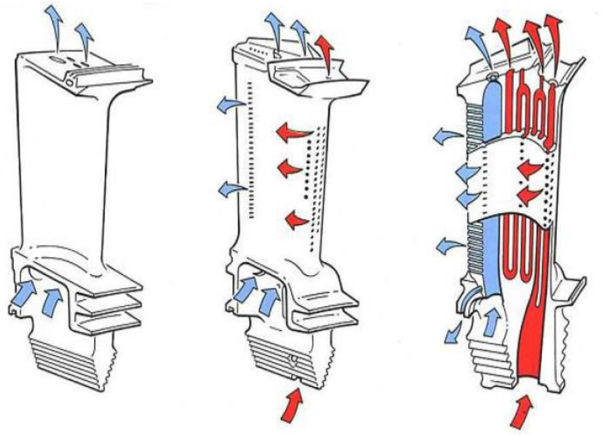
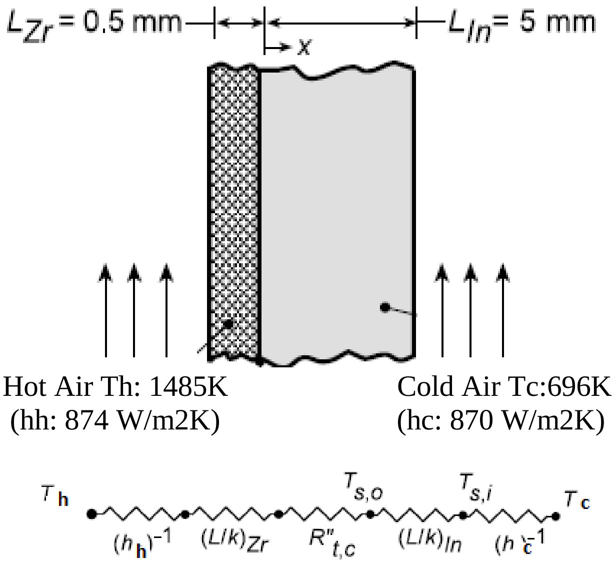


Figure 10. Blades cooling process.

The performance of gas turbine unit is improve by increasing the tolerance of the turbine blades to hot gases to emerging from the combustion chamber. One approach to achieving high operating temperatures involves application of a thermal barrier coating (TBC) to the exterior surface of a blade while passing cooling air, which came from compressor section, through the blades cooling holes. Typically the blades are made of high-temperature super alloy, such as Inconel (Ti-Si,Mo,Al…..and so on)(k=20.5 W/m2K). TBC material is basely ceramic, such as Zirconia (k=22.6 W/m2K).

In normal running condition, our combustion chamber exhaust temperature is about 1485 K. Cooling air temperature, which is coming from the compressor, is about 696 K. We consider that the TBC thick is about 0.5 mm and is attached at about 5 mm thick of Inconel blade wall by means of metallic bonding agent, which provide an interfacial thermal resistance of R’’t,c=10-4m2K/W. We will check if the Inconel material can be maintain at a temperature that is below its maximum allowable value of 1200 K. We will neglect the radiation effect and we will assume the turbine blade as a plane wall. We will check both situation with and without TBC coating.



Assumptions:

1. One-dimensional,
2. Steady State conduction in a composite plane wall,
3. Constant properties,
4. Negligible radiation,

Analysis:

We will analyse the situation with TBC and after then without TBC coating. By doing these we will understand the importance of TBC during running condition. As you know, TBC is protecting row material from hot air. TBC material is Zirconia and k: 22,6 W/MK).

Calculation with TBC coating:

R”tot=(1/hh)+(Lzr/kzr)+R’’t,c+(LIn/kIn)+(1/hc)
R’’tot= (1/874)+(0,0005m/22.6 W/m2K)+10-4m2K/W)+(0.005m/20.5 W/m2K)+(1/870 W/m2K)
R’’tot=2.66 10-3 m2K/W
Heat flux:
q’’=(Th-Tc)/R’’tot
q’’=(1485-696)/2.66 10-3 =29.7 104 W/m2
the inner surface temperature of the Inconel is:
Ti=Tc+(q’’/hc)
Ti=696+(29.7 104/870)=1037 K
the outer surface temperature of the Inconel is:
Ti=Tc+((1/hc)+(LIn/kIn))*q’’=696+((1/870)+(0.005/20.5))*29.7 104=1109 K

Calculation without TBC coating:

R”tot=(1/hh) +(LIn/kIn)+(1/hc)
R’’tot= (1/874)+(0.005m/20.5 W/m2K)+(1/870) W/m2K)
R’’tot=2.54 10-3 m2K/W
Heat flux:
q’’=(Th-Tc)/R’’tot
q’’=(1485-696)/2.54 10-3 =31.1 104 W/m2
the inner surface temperature of the Inconel is:
Ti=Tc+(q’’/hc)
Ti=696+(31.1 104/870)=1053 K
the outer surface temperature of the Inconel is:
Ti=Tc+((1/hc)+(LIn/kIn))*q’’=696+((1/870)+(0.005/20.5))*10.75 104=1129 K

As we can see that, the temperature without TBC is quite near the melting point. This means that long term run, thermal expansion, and extent TBC coating damage create big problem on the surface of the blades. After a long period, blades start to burn first and after then start to melt. Please see figures below show burned and melted blades and vanes.

Figure 11. Melted blade and vanes.

If we keep the blade cool enough we will not face of such problem. That is why the cooling process of the unit must be followed very well.

3. Conclusion

- TBC coating process is very important for protect the row materils from the high temperature aria,
- TBC coating process is high technology requirement and is used for all gas turbines and normal aircraft engines,
- Cooling process must be keept enough for protect TBC coating as well as row materials.

References

[1] Fank P. Incropera and David P. DeWitt, “Fundamentals of Heat and Mass Transfer,” 4. Edition.
[2] SE Jenny Sundberg, Heat Transfer Correlations for Gas Turbine Cooling LITH-IKP-EX--05/2313—
[3] Gorelov, Y. G. Ways of Further Enhancement the Convective Cooling of Vortex-Matrix Rotor Blades in High-Temperature Turbines of Gas Turbines Engines Thermal Engineering, Vol. 51, No 11, 2004
[4] Acharya, Zhou, Lagrone, Mahmood & Bunker Latticework (Vortex) Cooling Effectiveness Part 2: Rotating Channel Experiments Proceedings of ASME Turbo Expo 2004, Power for Land, Sea and Air, June 14-17, Vienna, Austria

[5] Van Treuren, Wang, Ireland, Jones & Kohler Comparison and Prediction of Local and Average Heat Transfer Coefficients under and Array of Inline and Staggered Impinging Jets ASME 96-GT-163

[6] Hollworth & Berry Heat Transfer from Arrays of Impinging Jets with Large Jetto-Jet Spacing Journal of Heat Transfer, May 1978, vol 100

[7] Tabakoff & Clevenger Gas Turbine Blade Heat Transfer Augmentation by Impingement of Air Jets Having Various Configurations Journal of Engineering for Power, January 1972

[8] Huber & Viskanta Effect of Jet-Jet Spacing on Convective Heat Transfer to Confined , Impinging Arrays of Axisymmetric Air Jets Int. J. Heat Mass Transfer, Vol. 37, No 18, 1994

[9] Gillespie, Wang, Ireland & Kohler Full Surface Local Heat Transfer Coefficient Measurements in a Model of an Integrally Cast Impingement Cooling Geometry Journal of Turbomachinery, Vol. 120, January 1988

[10] Gau & Chung Surface Curvature Effect on Slot-Air-Jet Impingement Cooling Flow and Heat Transfer Process Journal of Heat Transfer, Vol. 113, Novemeber 1991

[11] Korotky & Taslim Rib Heat Transfer Coefficient Measurements in a RibRoughened Square Passage Journal of Turbomachinery, Vol. 120, April 1998

[12] Kiml, Mochizuki & Murata Effects of Rib Arrangements on Heat Transfer and Flow Behavior in a Rectangular Rib-Roughened Passage: Application to Cooling of Gas Turbine Blade Trailing Edge Journal of Heat Transfer, Vol. 123, August 2001

[13] Hong & Hsieh Heat Transfer and Friction Factor Measurements in Ducts with Staggered and Inline Ribs Journal of Heat Tranfer, Vol. 115, February 1993

[14] Taslim & Wadsworth An Experimental Investigation of the Rib Surface-Average Heat Transfer Coefficient in a RibRoughened Square Passage Journal of Turbomachinery, Vol. 119, April 1997