HEFA2014
10th International Conference on Heat Transfer, Fluid Mechanic and Thermodynamics
14-16 July 2014
Orlando, Florida

NEW CORRELATIONS FOR THE AVERAGE NUSSELT NUMBER IN SQUEALER TURBINE BLADE TIP

Ghenai C.*, Madani, M. and Janajreh I.**
*Author for correspondence
*Ocean and Mechanical Engineering Department
College of Engineering and Computer Science
Florida Atlantic University
Boca Raton, Florida, USA
E-mail:cghenai@fau.edu
**Masdar Institute of Science and Technology, Abu Dhabi, UAE

ABSTRACT

Numerical simulations of the flow field and heat transfer of squealer blade tip are performed in this study. The effects of the Reynolds number, the clearance gap to width ratios (C/W = 5% - 15%) and the cavity depth to width ratios (D/W = 10% - 50%) on fluid flow and heat transfer characteristics are obtained. The temperature and velocity distributions inside the cavity, the local heat transfer coefficients, and the average Nusselt numbers for the pressure and suction sides of the turbine blade tip are determined. This paper presents the results of the effects of Reynolds number, clearance gap and width ratios on the Nusslet number for the pressure and suction sides of squealer turbine blade tip. The results show a good agreement with the experimental data obtained by Metzger and Bunker. New correlations for the average Nusselt numbers for turbine blade tip pressure and suction sides are presented.

INTRODUCTION

Turbine blade tips are subject to higher temperature gases and higher aero thermal loading [Bunker, 2006]. The blade tips are subjected to the highest heat transfer rate and there is a durability problem associated with the generated thermal stresses that result in structural damage to the blade tips. In fact, turbine blade damages could reduce its overall performance significantly. In order to reduce thermal stresses within the blade material and avoid a durability issue which is due to high operating temperature of the turbine, different criteria must be taken into account when designing engine cooling system. The three important factors involved in cooling technology of a turbine are optimizing the cooling technique, operating conditions, and also turbine blade geometry [Je-Chin et al., 1999]. Once of the most common approaches utilized to reduce this issue is to reduce the heat transfer and tip flow by grooving a single rectangular cavity along the blade tip. By doing so, flow resistance will increase that results in reduction of the flow across the tip; therefore, flow reduction will cause a decrease in heat transfer rate. The clearance gap between the tip of an axial turbine blade and the adjacent stationary shroud provides a narrow flow passage between the pressure and suction side of the blade. Although the resulting leakage flow is undesirable, it is impractical to eliminate the gap entirely because the clearance must accommodate centrifugal growth of the blade as well as differential thermal expansion between the blade and shroud through a variety of operating conditions [Henneke, 1984]. In general, blade tip leakage flow, which is a pressure driven flow, is associated with degradation in the blade aerodynamic performance and ads up to convective heat-transfer load of the blade. The blade tip leakage flow could affect the heat transfer on the pressure near the tip at entrance and also on the suction surface near the tip at the exit. Ameri et al., [1998] performed numerical study to simulate the tip flow and heat transfer on the General Electric (GE-E) first stage turbine. Two-dimensional cavity problem was calculated using the k- ω turbulence model. Areas of large heat transfer were identified on the blade tip and the mechanisms of heat transfer enhancement were discussed. Ameri and Rigby [1999] performed numerical simulation to predict the distribution of convective heat transfer coefficient on a blade tip with cooling holes. Numerical flow visualization showed that the uniformity of wetting of the surface by the film cooling jet is helped by the reverse flow due to edge separation of the main flow. Thorpe et al. [2005] performed experimental study on blade tip heat transfer and aerodynamics in a transonic turbine stage test facility. The data reveal the effect of vane-rotor interactions on the unsteady heat transfer along the blade tip mean camber line.

The principal objective in this study is to predict the average Nusselt number at the blade tip pressure and suction sides and to develop correlations of the Nusselt number that take into account the Reynolds number (based on the clearance gap between the blade tip and stationary shroud), the clearance gap between the blade tip and stationary shroud and the cavity depth. In this study, the shear-stress transport (SST) $k-\omega$ model was used. The SST $k-\omega$ model is similar to standard $k-\omega$ model but the definition of the turbulent viscosity is modified to account for the transport of the turbulent shear stress.

GOVERNING EQUATIONS

The time averaged gas phase equations for steady turbulent flow are:

$$\frac{\partial}{\partial x_j}(\rho u_i \Phi) = -\frac{\partial}{\partial x_i}\left(\Gamma_\Phi \frac{\partial \phi}{\partial x_i}\right) + S_\Phi \qquad (1)$$

Φ is the dependent variable that can represent the velocity u_i, the temperature T, the turbulent kinetic energy k, and the dissipation rate of the turbulent kinetic energy ε. The governing equations are:

Continuity:

$$\frac{\partial \overline{\rho u_i}}{\partial x_i} = 0 \qquad (2)$$

Momentum Equation:

$$\frac{\partial\left(\overline{\rho u_i u_j}\right)}{\partial x_j} = -\frac{\partial \overline{P}}{\partial x_i} + \frac{\partial\left(\overline{t_{ij}} + \overline{\tau_{ij}}\right)}{\partial x_j} \qquad (3)$$

Where $\overline{t}_{ij}$ is the viscous stress tensor defined as:
$\overline{t}_{ij} = \mu\left[\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right) - \frac{2}{3}\frac{\partial \overline{u_k}}{\partial x_k}\delta_{ij}\right]$

$\delta_{ij} = 1 \ \ if \ \ i = j \ \ and \ \ \delta_{ij} = 0 \ \ if \ \ i \neq j$

$\overline{\tau}_{ij}$ is the average Reynolds stress tensor defined as:
$\overline{\tau}_{ij} = -\rho\,\overline{u_i' u_j'}$

$\overline{\tau}_{ij} = \mu_t\left[\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right) - \frac{2}{3}\frac{\partial \overline{u_k}}{\partial x_k}\delta_{ij}\right] - \frac{2}{3}\left(\overline{\rho k\, \delta_{ij}}\right)$

Where k is the average turbulent kinetic energy defined as:
$\overline{k} = \frac{1}{2}\overline{u_i' u_i'}$

μ_t is the turbulent eddy viscosity expressed as: $\mu_t = c_\mu \frac{\overline{\rho k^2}}{\overline{\varepsilon}}$

Where C_μ is constant (C_μ = 0.09) and $\overline{\varepsilon}$ is the average dissipation rate of the turbulent kinetic energy and defined as:
$\overline{\varepsilon} = \nu \overline{\frac{\partial u_i'}{\partial x_j}\frac{\partial u_i'}{\partial x_j}}$

Turbulent kinetic energy (k) equation:

$$\frac{\partial\left(\overline{\rho\, k\, u_i}\right)}{\partial x_i} = \frac{\partial\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \overline{k}}{\partial x_j}\right]}{\partial x_j} + G_k - \overline{\rho\, \varepsilon} \qquad (4)$$

Where σ_k = 1 and G_k is the production of the turbulent kinetic energy defined as:

$$G_K = \mu_t\left[\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right)\right]\frac{\partial \overline{u_i}}{\partial x_j} - \frac{2}{3}\frac{\partial \overline{u_i}}{\partial x_j}\delta_{ij}\left[\mu_t \frac{\partial \overline{u_k}}{\partial x_k} + \overline{\rho\, k}\right]$$

Specific dissipation rate (ω) equation:

$$\frac{\partial\left(\overline{\rho\, \omega\, u_i}\right)}{\partial x_i} = +\frac{\partial\left[\Gamma_\omega \frac{\partial \overline{\omega}}{\partial x_j}\right]}{\partial x_j} + G_\omega - Y_\omega + D_\omega + S_\omega \qquad (5)$$

In these equations, G_ω represents the generation of ω , Γ_ω represent the effective diffusivity of ω, Y_ω represent the dissipation of ω due to turbulence, D_ω is the cross-diffusion term, and S_ω is the user-defined source term [Wilcox, 1998].

Energy equation:

$$\frac{\partial\left(\overline{(\rho\, E + p)\, uj}\right)}{\partial x_j} = \frac{\partial\left[(k_{eff})\frac{\partial \overline{T}}{\partial x_j} - \sum_j h_j J_j + \left(\overline{\tau}_{eff} u_j\right)\right]}{\partial x_j} + S_h \qquad (6)$$

Where E is the total energy ($E = h - p/\rho + v^2/2$ where h is the sensible enthalpy), k_{eff} is the effective conductivity ($k_l + k_t$: laminar and turbulent thermal conductivity), J_j is the diffusion flux of species j, and S_h is the term source that includes the heat of chemical reaction and any other volumetric heat sources.

GEOMETRY, BOUNDARY CONDITIONS AND MESH GENERATED

Figure 1 shows a schematic of the two dimensional section of the three dimensional geometry used in this study with the corresponding boundary conditions. The gas is introduced from the left boundary at certain velocity Vin (uniform velocity) and temperature Tin (air temperature was 140 F). The top and bottom boundaries are walls (no slip condition). A temperature of 90 F was imposed at the walls. These two temperatures were selected based on the conditions of the experimental study performed by Metzeger et al. [1989]. The numerical results will be compared to the experimental data for the validation of the turbulence model. The right boundary is an outflow. Three values for cavity depth-to-width ratios (D/W = 0.1, 0.2 and 0.5) were used for each clearance-to-width ratios (C/W= 0.05, 0.10 and 0.15). The geometry cavity width was kept constant at W= 50.8 *mm*. A total of nine different geometries were created. For each geometry, three Reynolds number (Re = 10000, 20000 and 40000) were tested. A total of 27 cases were tested in this study: (1) Reynolds numbers: Re = 10,000, 20000, 40000, (2) clearance to cavity width ratio: C/W = 5%, 10% and 15% and (3) the depth to cavity width ratio: D/W = 10%, 20%, and 50%.

High quality mesh was generated for the nine geometries. The mesh has quadrilateral shape. The resolution of the mesh is greater in regions where greater computational accuracy was needed, such as the region near the cavity floor and the blade tips. Figure 2 shows the mesh generated for the blade cavity model with C/W = 0.05 and D/W = 0.1. The numbers of grids for each of the geometries vary based on the values for cavity depth-to width ratios and clearance-to-width ratios used for each case; however, the number of grids for the blade tips and the cavity floor in all the geometries are consistent and total of 100 grids on the tips and 540 for the cavity floor was used. A grid independent study was performed to make sure the solution does not dependent on the number of grids.

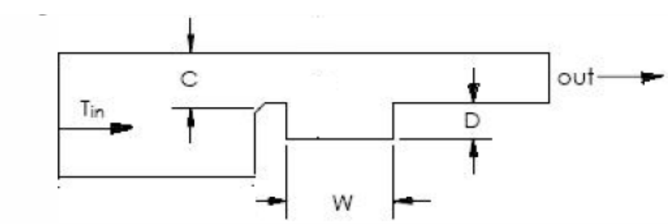

Figure 1 Schematic of the 2-D section of the three dimensional geometry

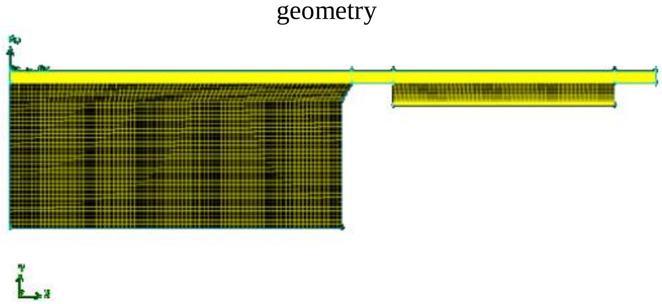

Figure 2 Mesh generated for C/W = 5% and D/W = 10%

NUMERICAL METHOD

A control volume based finite difference method is used in order to solve a system of partial differential equations governing the conservation of mass, momentum, energy, and turbulent flow parameters. The SIMLER algorithm [Patankar, 1980] is used to solve explicitly for the velocity and pressure fields. The numerical method utilized for the simulation had a density based solver with implicit formulation. A coupled implicit solver and an external compressible flow model for the turbulence were used in this study.

RESULTS AND DISCUSSION

Three turbulence models were used first to validate the numerical model. The Reynolds stress model (RSM), k-epsilon, and the k-omega models were tested in this study. A comparison between the numerical results obtained with the three turbulence models and the experimental results are presented in figure 3. The experimental results are from Metzeger et al. [1989]. The goal is to make sure the model is performing well before we start any parametric study. The results in Figure 3 show the distribution of the local Nusselt number versus the horizontal distance. As shown in Figure 3, the $k - \omega$ model is in good agreement with the experimental data with less than 10 % deviation. The Nusselt number represents the ratio of the convective and conductive heat transfer across the boundary (The Nusselt number $Nu = h.L/k_f$, where h is the heat transfer coefficient, L is the characteristics length scale and k_f is the thermal conductivity of the fluid).

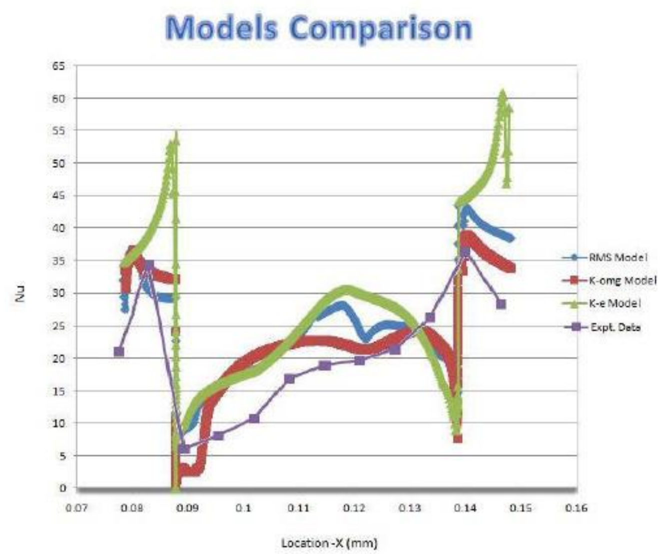

Figure 3 Local Nusselt numbers: Comparison between the experiments and numerical data

It is noted from Figure 3 that the peak local Nusselt number are located at the pressure and suction sides of the blade tip. The Nusselt number at the floor of the cavity is smaller compared to the blade tip pressure and suction sides. A mean Nusselt number at the blade tip pressure and suction sides were calculated. The mean Nusselt number on the pressure and suction sides of the blade tips and for each Reynolds number are summarized in the tables 1 to 3. As shown in these tables, the heat transfer rate is higher on the suction side of the blade tip than that of the pressure side for all the cases tested in this study. It is also noted that the Reynolds number, D/W and C/W are affecting the directly the average heat transfer rate or Nusselt number.

Table 1 Average Nusselt Numbers: Re = 10,000

D/W	C/W	Pressure side	Suction Side
0.1	0.05	37.20	44.20
0.1	0.10	45.21	53.13
0.1	0.15	53.11	64.92
0.2	0.05	31.24	40.36
0.2	0.10	37.20	48.00
0.2	0.15	42.75	52.61
0.5	0.05	27.60	37.13
0.5	0.10	32.16	46.20
0.5	0.15	43.20	51.19

Table 2 Average Nusselt Numbers: Re = 20,000

D/W	C/W	Pressure side	Suction Side
0.1	0.05	65.90	72.10
0.1	0.10	79.49	78.31
0.1	0.15	86.32	106.24
0.2	0.05	58.89	66.71
0.2	0.10	70.83	73.98
0.2	0.15	76.36	95.27
0.5	0.05	58.89	64.47
0.5	0.10	68.48	73.56
0.5	0.15	74.21	89.20

Table 3 Average Nusselt Numbers: Re = 40,000

D/W	C/W	Pressure side	Suction Side
0.1	0.05	109.70	123.60
0.1	0.10	115.69	144.91
0.1	0.15	134.69	178.36
0.2	0.05	104.36	115.20
0.2	0.10	110.03	138.53
0.2	0.15	119.82	160.83
0.5	0.05	103.00	114.30
0.5	0.10	108.60	137.20
0.5	0.15	118.00	162.10

Figures 4 to 9 show the variation of the average Nusselt number versus the depth to width cavity ratio (D/W) for different Reynolds number (Re), tip clearance to width ratio (C/W) for both pressure side and suction sides. The results show: (1) the average Nusselt number increases by increasing the clearance to cavity width ratio (C/W) from 5% to 15%; (2) the Nusselt number slightly decreases by increasing the cavity depth to width ratio (D/W) from 10% to 50 %, and (3) the average Nusselt number increases by increasing the Reynolds number Re from 10000 to 40000. It could be observed that by increasing the cavity depth (D/W) and keeping the Reynolds number and tip clearance to width ratio (C/W) constant, the heat transfer is reduced due to reduction in the flow across the tip. It is also noted that the local heat transfer coefficient increases with increasing gap clearance-to-cavity width ratio (C/W), while the cavity depth was held constant. The reduction of the clearance gap will greatly reduce heat transfer load to the blade tips.

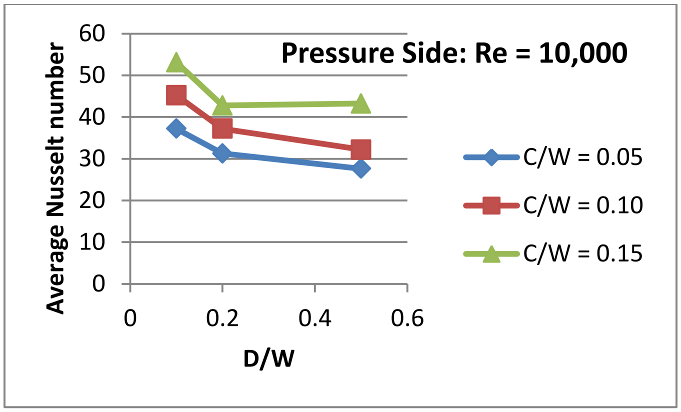

Figure 4 Average Nusselt numbers: Pressure Side, Re = 10000

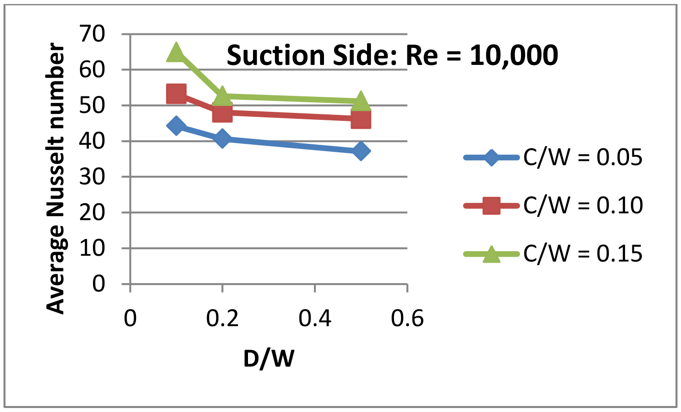

Figure 5 Average Nusselt numbers: Suction Side, Re = 10000

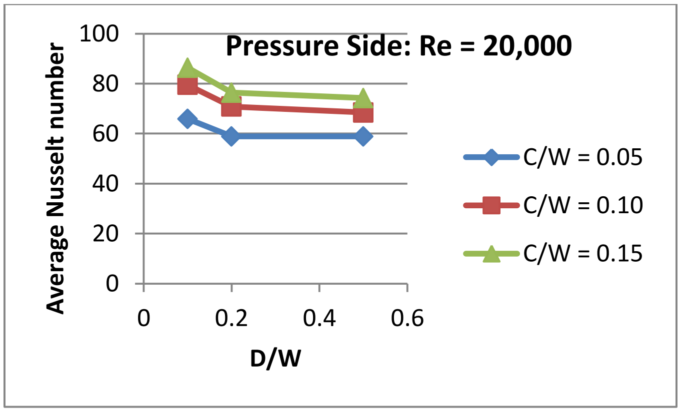

Figure 6 Average Nusselt numbers: Pressure Side, Re = 20000

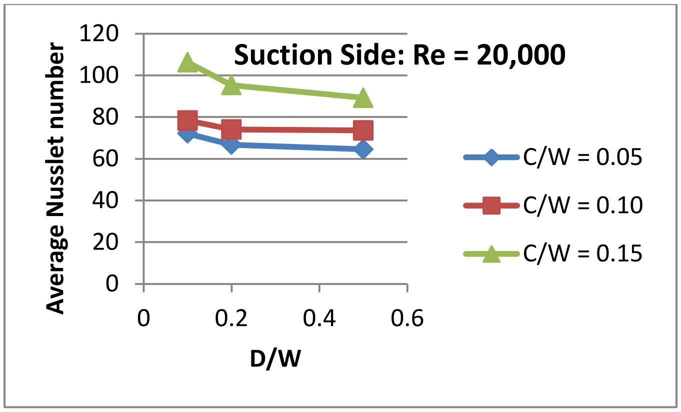

Figure 7 Average Nusselt numbers: Suction Side, Re = 20000

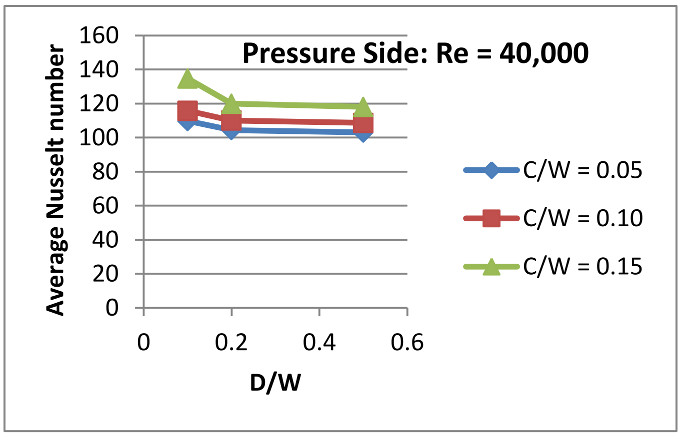

Figure 8 Average Nusselt numbers: Pressure Side, Re = 40000

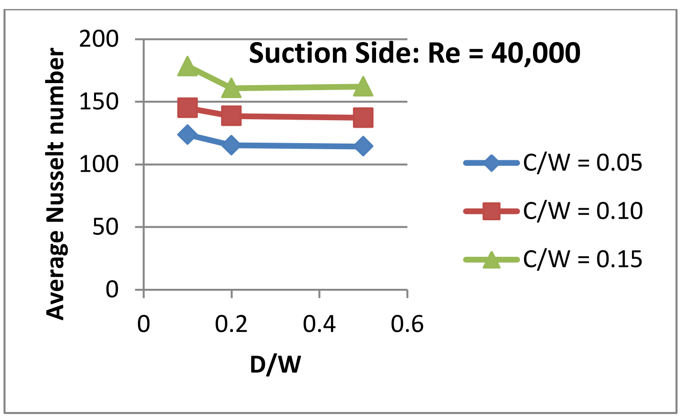

Figure 9 Average Nusselt numbers: Suction Side, Re = 40000

Figures 10 and 11 show the average Nusselt number respectively for the blade tip pressure and suction sides for all the conditions tested in this study. The results show the variation of the average Nusselt number versus D/W for different Reynolds number and tip clearance to width ratio (C/W). Based on these results, new correlations are developed for the blade tip pressure and suction sides. The new proposed correlations will take into account the flow speed or Reynolds number (Re), the tip clearance to width ratio (C/W) and the cavity depth to width ratio (D/W). Figure 12 and 13 show the variation of the average Nusselt number using the new proposed correlations. The new proposed correlations for the average Nusselt number at the blade tip pressure and suction sides are given by:

Average Nusselt Number: Blade Tip Pressure Side:

$$N_u = 0.10\left[R_e^{0.7}\left(\frac{c}{W}\right)^{0.22}\left(\frac{D}{W}\right)^{-0.1}\right] - 3.14 \tag{7}$$

Average Nusselt Number: Blade Tip Suction Side:

$$N_u = 0.13\left[R_e^{0.7}\left(\frac{c}{W}\right)^{0.22}\left(\frac{D}{W}\right)^{-0.1}\right] - 10.87 \tag{8}$$

The results show a linear variation of the average Nusselt number versus the Reynolds number Re, the tip clearance to width ration and the cavity depth to width ratio (D/W). It is also noted from Figure 12 and 13 that the slope for the blade tip pressure and suction sides is slightly different. The proposed new correlations will be used for the prediction of the average heat transfer during the design process of gas turbine blade.

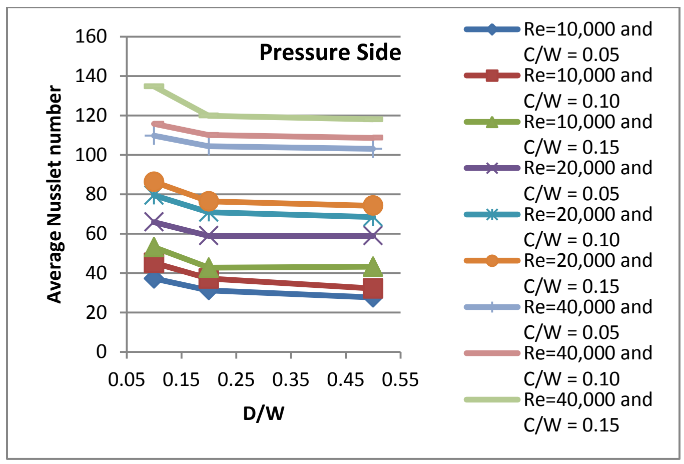

Figure 10 Average Nusselt numbers - Pressure Side

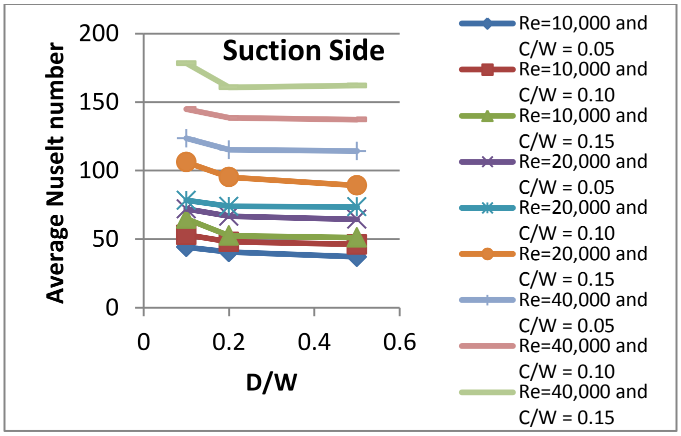

Figure 11 Average Nusselt numbers – Suction Side

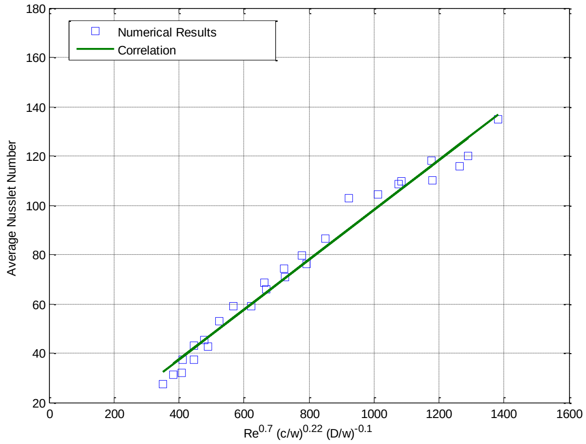

Figure 12 Correlation for the average Nusselt numbers - Pressure Side

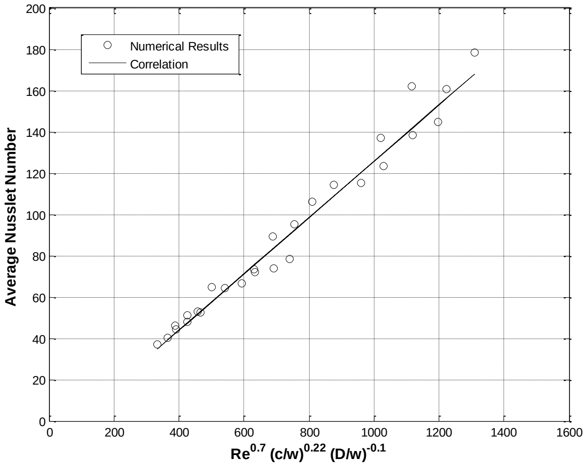

Figure 13 Correlation for the average Nusselt numbers – Suction Side

CONCLUSIONS

Computational fluid dynamics analysis of the flow and heat transfer in squealer blade tip was performed in this study. The k-omega turbulence model was used in this study. The effect of Reynolds number Re, the clearance gap between the blade tip and stationary shroud and the cavity depth on fluid flow and heat transfer characteristics were determined. A total of 27 runs were tested: Three Reynolds numbers Re = 10,000, 20,000, and 40,000, three tip clearance to width ratios C/W = 5%, 10% and 15% and three cavity depth to width ratios D/W = 10%, 20% and 50%. For all the cases tested in this study, the temperature of the gas and the walls of the cavity were kept constant respectively to 140oF and 90oF. The temperature and velocity distributions inside the cavity, the local heat transfer coefficients, and the average Nusselt numbers for the pressure and suction sides of the turbine blade tip were obtained. The numerical results obtained with the k-ω turbulence model show a good agreement with the experimental data obtained by Metzger and Bunker. The results show that the peak local Nusselt numbers are located at the pressure and suction sides of the blade tip. The Nusselt number at the floor of the cavity is smaller compared to the blade tip pressure and suction sides. The results also show that average Nusselt number increases by increasing the clearance to cavity width ratio (C/W), decreases by increasing the cavity depth to width ratio (D/W), and increases by increasing the Reynolds number Re. New correlations for the average Nusselt number for both the blade tip pressure and suction sides are proposed in this study. These new correlations take into account the Reynolds number, the tip clearance to width ratio, and the cavity depth to width ratio. These correlations will be used for the prediction of the average heat transfer during the design process of gas turbine blade.

REFRENCES

Ameri, A..A, Seinthorsson, E., and Rigby, D.L., Effect of Squealer tip rotor heat transfer and efficiency, ASME Journal of Turbomachinery, Vol. 120, No4, Oct. 1998, pp. 753 – 759

Ameri A.A., and Rigby, D.L., A numerical analysis of heat transfer and effectiveness on film cooled turbine blade tip models, NASA/CR 1999 – 209165

Bunker, R.S., Axial Turbine Blade Tips: Function, Design, and Durability, Journal of Propulsion and Power, Vol. 22, No. 2, March–April 2006

Henneke, D. K., "Heat Transfer Problems in Aero-Engines, "Heat and Mass Transfer in Rotating Machinery, D. E. Metzger and N. H. Afgan, eds., Hemisphere, Washington, DC; 1984, pp 353-379.

Je-Chin, H., Sandip, d., Srinath, E., Taylor and Francis Group, 1999

Metzger, D.E., Bunker, R.S., and Chyu, M.K., 1989, "Cavity Heat Transfer on a Transverse Grooved Wall in a Narrow Flow Channel", *J. of Heat Transfer*, Vol. 111, pp. 73-79.

Patankar, S. V., Numerical Heat Transfer and Fluid Flow, McGraw- Hill New York, 1980.

Thorpe, S.J., Yoshino, S., Thomas, G.A., Ainsworth, R.W., and Harvey, N.W., Blade tip heat transfer in a transonic turbine, Proc. IMechE vol. 219 Part A: J. Power and Energy, 2005, pp. 421 – 430

Wilcox, D.C., Turbulence Modeling for CFD, DCW Industries, Inc., La Canada, California, 1998.