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AN INTERNAL COOLING TEST BENCH FOR GAS TURBINE RESEARCH

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ABSTRACT

A test bench for experimental investigations of turbine internal cooling applications has been designed and commissioned. The test bench mainly consists of an air tunnel including test section, a radial fan, data acquisition system and an infrared camera. Lower wall of the test section is covered with a thin sheet heater to provide uniform heat flux perpendicular to the streamwise direction. Surface temperature distribution is measured with infrared camera for ribbed and unribbed configurations to determine the surface local convective heat transfer coefficients. The paper mainly focuses on the tunnel description and some validation measurements at different Reynolds numbers. The data has been compared with the literature.

INTRODUCTION

Cooling of turbine blades for modern gas turbine engines is essential due to high combustor exit temperature. Cooling air flowing through the internal channels of blades removes heat to keep the metal temperature within desired limits. Internal cooling channels can be one or multi-pass, flat or ribbed surface depending on the cooling requirements. Heat transfer augmentation inside cooling channels is achieved by using repeated ribs as turbulence promoters.

Detailed description of the flow physics in a turbine cooling channel is extensively explained by [Han et al., 2000]. The periodic ribs break the laminar sublayer and create local wall turbulence due to flow separation and reattachment between the ribs, greatly enhancing the heat transfer [Srinath et al., 1997]. The effect of number of ribbed walls is also an area of interest. [Chandra et al., 1997] showed the heat transfer characteristics of rectangular channel with transverse ribs on one, two and four walls. Wall temperature distribution and uniformity of rib-roughened cooling channels are experimentally presented by [Çakan, 2000]. [Casarsa, 2003] presented the combination of heat transfer and highly three-dimensional flow behavior by conducting aerodynamic measurements. [Han, 1988] carried out an experimental study to investigate the effect of the channel aspect ratio on the distribution of the local heat transfer coefficient in rectangular channels with two opposite ribbed walls for Reynolds number in the range of 10000 to 60000. [Han and Park, 1988] also studied the heat transfer and friction factor effects for different aspect ratios. He concluded that square channel perform better than rectangular ones.

Rib spacing is investigated in many research campaigns. [Han et al.,1978] studied the effects of rib spacing on pressure drop, flow separation and reattachment. [Taslim and Spring, 1994] indicated the optimum rib spacing for very high blockage ratios whereas [Zhang et al.,1994] did so for very small blockage ratios. [Han et al.,1985] showed that the highest heat transfer rates and pressure drops are obtained with 75° ribs in square cross-sectioned channel. He concluded that 30° - 45° ribs are optimum for highest heat transfer rate with desired pressure drop. [Park et al., 1992] showed that 45° - 60° ribs are appropriate with favorable pressure drop for five different rectangular channels. [Lau et al.,

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1992] studied the use of parallel or crossed ribs on two opposite walls by indicating that two counter rotating vortices produced by parallel angled ribs produces more heat transfer enhancement rather than crossed angled ribs. [Taslim and Kercher, 1994] studied on V-shaped ribs and stated that 45° V-shaped ribs with apex facing downstream performs better than those with apex facing upstream with large pressure drop. [Han et al., 1992] compared 60° V-shaped ribs with 60° parallel, 60° crossed and 90° ribs and concluded that 60° V-shaped ribs performed better than the latters. [Lau et al. 1992] concluded that parallel discrete ribs bring higher ribbed wall heat transfer than parallel full 45° and 60° ribs, but lower heat transfer at smooth side walls. [Lio et al., 2003] experimentally investigated the heat transfer and fluid flow characteristics inside a rotating two-pass duct with detached 90 degrees ribs. [Egge et al., 2012] used a transient technique with infrared thermography for the heat transfer measurements inside a two-pass internal cooling channel connected by a 180 degree bend. [Jenkins et al., 2008] carried out an experimental study to determine the effect of ribs and tip wall distance on heat transfer inside a two-pass ribbed internal cooling channel.

The current research paper presents establishment of an internal cooling channel test setup at TUSAS Engine Industries Inc. The tunnel is designed to study turbine cooling related topics such as internal cooling channel, U-turn performance in the serpentine channel, film cooling. Wide range of Reynolds number can be investigated in the designed channel. The paper describes the tunnel characteristics and shows some validation studies.

DESCRIPTION OF THE EXPERIMENTAL SET-UP

Experimental set-up mainly consists of an air tunnel, a radial fan with speed controller, a data acquisition system and an infrared camera. The sizing of the tunnel is done in such a way that the internal cooling channel of real blades is simulated for a Reynolds number range of 20000-80000. Air tunnel has 2000 mm length divided into two sections. The flow reaches to fully developed stage in the frontal flat part of the channel. Then, it enters to the test section where the roughness elements and heater are located. The cross section of the tunnel is 100x166 mm. The tunnel is running in suction mode by a radial fan located at the exit of the tunnel. The fan speed is adjusted via a controlling unit which allows running tests at different Reynolds numbers. The inlet of the tunnel is equipped with a honeycomb to enable flow uniformity before test section. The velocity distribution of the mid-channel along channel height before test section is depicted in Fig. 1.

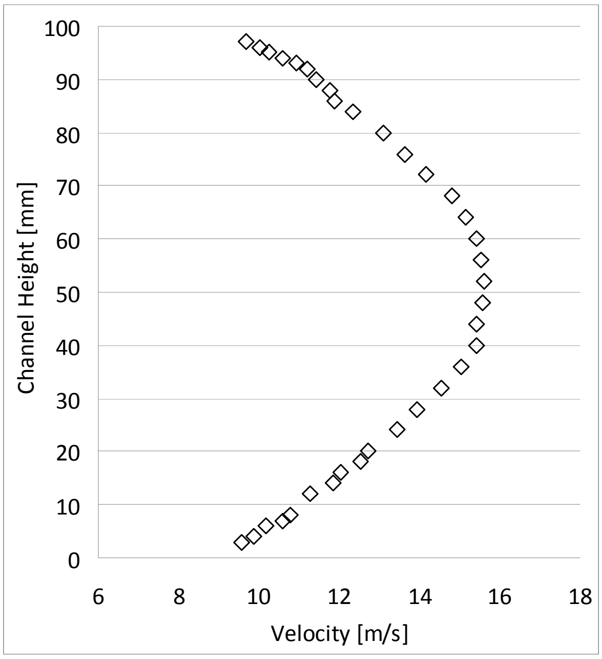


Figure 1 Velocity Profile on the symmetry plane

The surface temperature is measured by using infrared thermography. The resolution of the camera is 640x480 pixels which allow two pixels per millimeter. The wall material is not transparent for the infrared wavelength so a special windowing applied to the test section. The heating amount is simply computed based on joule heating by multiplying the voltage and current (Eq. 1). To increase the accuracy, the voltage measurement is performed between the two legs of the heater very close to the tunnel boundaries.

$$\dot{q} = VI \tag{1}$$

The tunnel is insulated from the bottom wall in order to minimize the heat losses. The bottom wall is instrumented with thermocouple to compute the heat loss amount. All thermocouples are calibrated in a constant temperature bath and the calibration is checked before the test using the ambient temperature against any drift. By using thermocouple readings, heat loss from the bottom wall is computed as 1.5% of the total heat generated.

The flow conditions are measured at the inlet section of the tunnel. The gas bulk temperature is obtained with T-type thermocouple whereas the air velocity is measured by using pitot probe at the channel mid-section. The velocity uncertainty of the measurements is about 1.5 m/s. The schematic view and the photograph of the test bench are shown in Fig. 2.

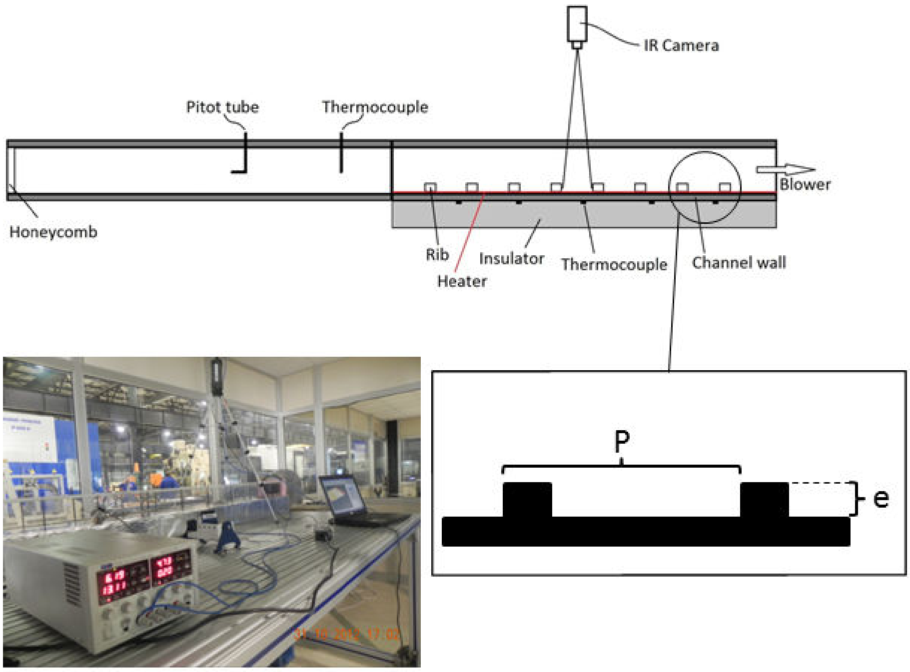
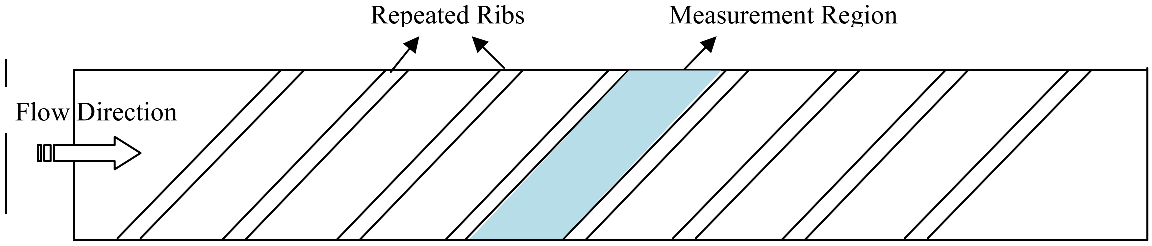
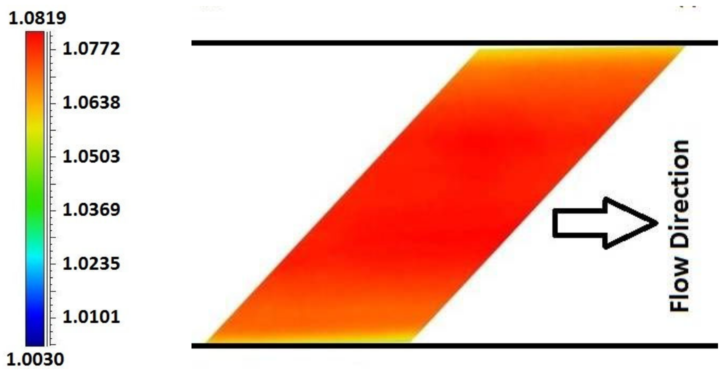


Figure 2 Schematic View of the Test Bench (upper), Test Bench (lower–left) Main dimensions (lower–right)

The test section has been heated from the bottom wall using a thin sheet heater. The heater generates a uniform heat flux over its surface. Test section involving heated surface and repeated 45 degree angled ribs is shown in Fig. 3 (a). Measurement is taken from the region between 4th and 5th ribs after the flow reaches periodic conditions in the streamwise direction. Temperature data is taken from the measurement region for flat surface with no ribs and ribbed surface to determine the heat transfer enhancement. The temperature distribution of the surface for the flat channel (no rib is equipped) is shown in Fig. 3 (b). About 95% of the surface area provides uniform heat flux whereas 5% of the surface has non-uniformities due to side wall effects. Those regions are not taken into account for the post-processing.



(a)



(b)

Figure 3 (a)Test Section with 45° Rib Configuration and and (b) Temperature Distribution of the Flat Surface (No Ribs)

The primary focus of the test setup is to obtained heat transfer effectiveness for different cooling schemes. The heat transfer effectiveness is computed by the ratio of Nusselt numbers (ribbed channel over flat channel). By measuring the surface temperature distribution, surface convective heat transfer coefficient (h) can be determined by Eq. 2.

$$h = \frac{\dot{q}}{T_s - T_\infty} \tag{2}$$

Where $\dot{q}$ is the heat flux, T_s is local surface temperature and T_∞ is mainstream gas temperature.

Following the calculation of the heat transfer coefficient, dimensionless parameter of the heat transfer, Nusselt number (Nu), is obtained by Eq. 3

$$Nu = \frac{hD_h}{k} = \frac{\dot{q}\,D_h}{(T_\infty - T_s)k} \tag{3}$$

Where "k" is the thermal conductivity of air and D_h is the hydraulic diameter of the channel.

A more convenient dimensionless parameter is defined as enhancement factor (EF) which is independent of the geometric dimensions and boundary conditions to provide comparability with the literature. Enhancement factor is calculated by Eq. 4

$$EF = \frac{Nu}{Nu_0} \tag{4}$$

Here Nu_0 is the Nusselt number of a pipe which is calculated by Eq. 5 which is provided by [Dittus and Boelter, 1930].

$$Nu_0 = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^n \tag{5}$$

RESULTS and DISCUSSIONS

The first measurement campaign is conducted in order to understand the Reynolds number and pitch-to-height ratio effects on the EF. Square cross section roughness elements (rib) are placed to the channel. The inter-rib space (distance between two ribs) is defined as six and eight times the height of the rib. The angle between the flow direction and the normal of the rib surface is set to 45 degree. The experiments are done for a wide Reynolds number range between 35000 to 70000.Thermal camera images are processed by an in-house code. The code is able to filter the image then corrects the orientation and finally computes the enhancement factor considering the all thermal losses (Fig. 4).

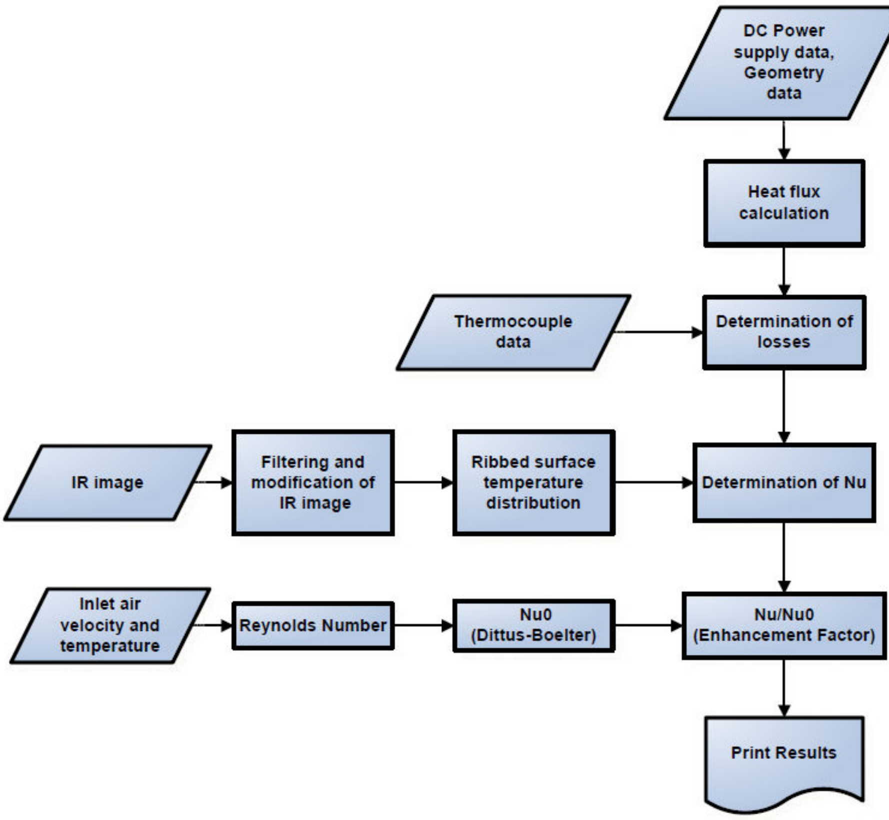


Figure 4 Flowchart of post processing

The normalized surface temperature (T_s/T_∞) distribution is given in Fig. 5 for P/e (periodic distance over rib height) ratios of both configurations at three different Reynolds numbers. Three regions appear on the measurements. Several temperature zones is clearly seen downstream of the rib as depicted in Fig. 5 (Region A, B and C). The flow field is analyzed in details numerically by using commercial CFD software in order to understand the flow physics. The velocity magnitude and the surface temperature of the numerical results are presented in Fig. 6. A strong vortex core is captured in Region A. The boundary layer developed upstream of the rib and on the lateral wall moves over the rib and roll up just behind the rib structure to create this vortex region. It moves along the rib span up to 70% of the rib span. At this location a second vortex structure is captured by numerical model. The flow is sustained in the Region B like a dead zone behind the backward facing step. Hence, the flow is heated up by the surface. The higher flow temperature cause a lower heat transfer rate as well as a higher wall temperature. The vortex tube changes the path and moves more downstream when the vortex tube of Region A meets with the Region B. The lowest surface temperature is captured downstream of the Region A similar to the study of [Cakan, 2000]. He stated that a vortex generated in-between two ribs sweeps the surface around 1.5-2 rib height downstream of the rib. Another low temperature region is also observed just upstream of the rib (Region C). The boundary layer developed inter-rib space interacts with the rib, a pressure gradient occurs at the stagnation region due to the pressure gradient of the boundary layer. The pressure gradient creates a corner vortex at this region. This vortex region is also stated by [Magi, et al., 2004]. When the inter-rib space is increased, the length after impingement line is higher so the boundary layer is much thicker when it arrives to the rib. The thicker boundary layer results in much stronger pressure gradient hence a stronger corner vortex. Therefore, the surface temperature is lower at the rib upstream for higher inter-rib spaces. The average surface temperature reduces when the Reynolds number increases for the both cases.

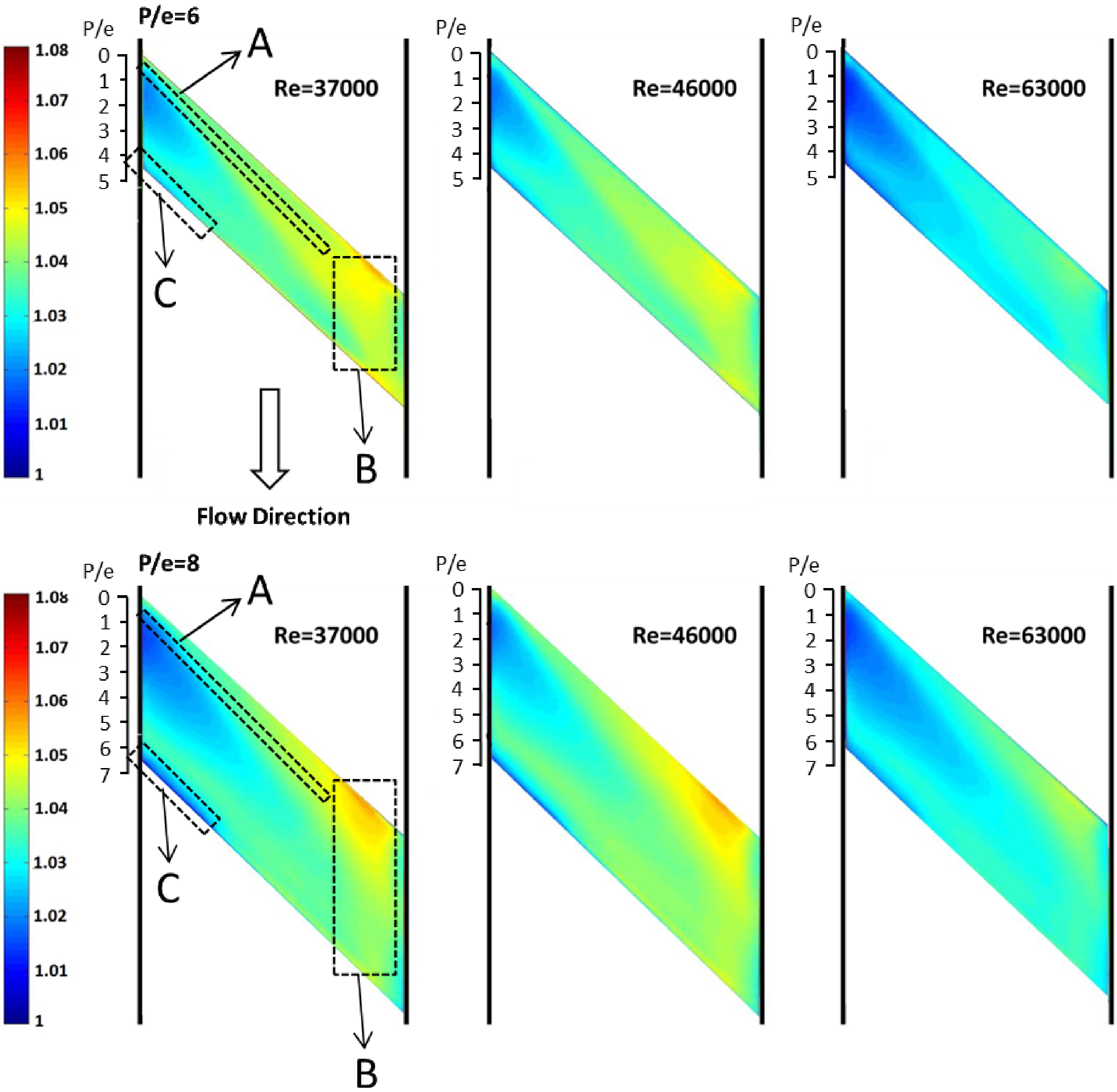


Figure 5 Normalized Surface Temperature Distributions for Different Reynolds Numbers

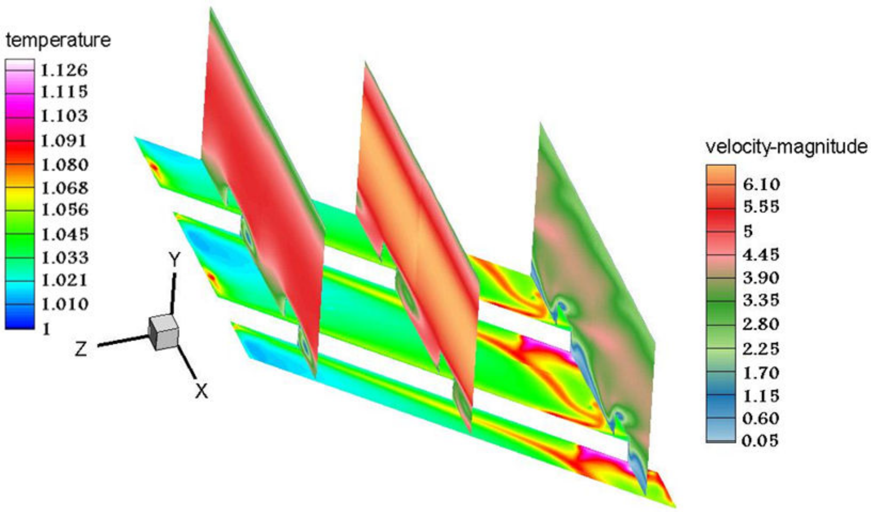


Figure 6 Flow structures vs temperature distribution on the ribbed wall

Figure 7 summarizes the results together with the experiments of Chen et al. the results of new wind tunnel shows similar trend with the Chen et al. study. However, the results obtained have higher values then the literature. Such difference can be explained with the effect of the aspect ratio and the slightly lower blockage ratio. Also the TEI wind tunnel has a rectangular cross section whereas the Chen et al. used a square cross section wind tunnel. EF for smooth wall (unribbed configuration) is also presented in Figure 7.

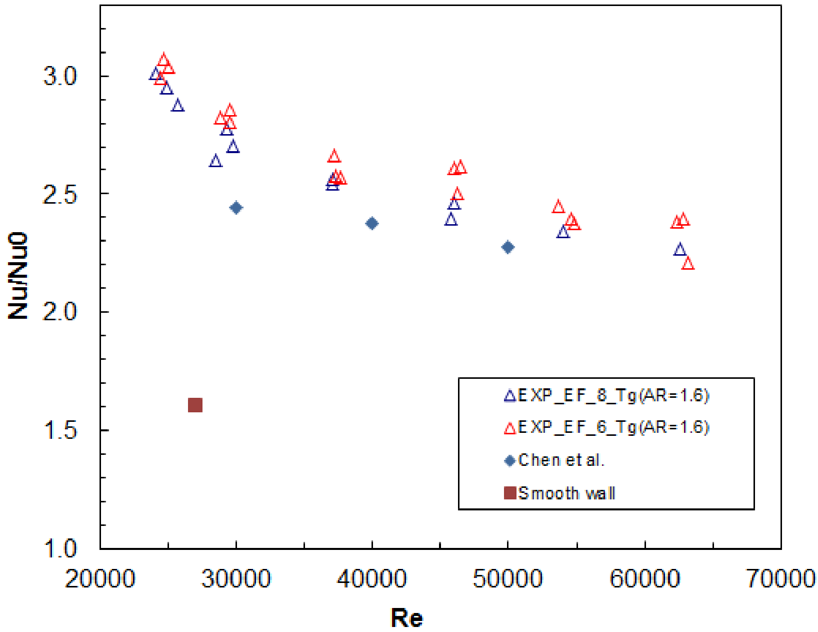


Figure 7 Enhancement Factor (EF) for Different Reynolds Numbers

CONCLUSION

An experimental set-up is designed and installed for turbine internal cooling investigations. Heat transfer measurements are carried out for flat and ribbed surfaces to obtain cooling enhancement factor of the ribbed configurations. Infrared thermography technique is used to measure the surface temperature distribution. A code is constructed to evaluate the enhancement factor from raw temperature and boundary condition data. A number of measurement campaigns are conducted and validity of the data obtained from measurements is proved against literature. Experiments are carried out for two pitch-to-height ratios. Reynolds number is kept between 35000 and 70000 which is a typical range in cooling channels for real applications. By the results of the current work, the validity of the TEI Internal Cooling Test Bench is proved.

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