



GLOBAL JOURNAL OF RESEARCHES IN ENGINEERING: A  
MECHANICAL AND MECHANICS ENGINEERING

Volume 17 Issue 7 Version 1.0 Year 2017

Type: Double Blind Peer Reviewed International Research Journal

Publisher: Global Journals Inc. (USA)

Online ISSN:2249-4596 Print ISSN:0975-5861

## Leakage and Thermal Characterizations of Canted Labyrinth Seals

By R Mohana Rao & Dr. Manzoor Husain

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*GJRE-A Classification:* JEL Code: 091399p



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# Leakage and Thermal Characterizations of Canted Labyrinth Seals

R Mohana Rao <sup>α</sup> & Dr. Manzoor Husain <sup>σ</sup>

**Abstract-** Labyrinth seals are the simple and most commonly used rotating seal in cold (compressor) and hot (turbine) regions of gas turbine engines. In order to achieve successful engine design estimating accurate leakage flow rates, windage heating and heat transfer coefficients are critical for rotor dynamic stability. The windage effect inside the flow creates direct loss of power also increase fluid temperature. In this paper numerical simulation of a canted stepped labyrinth seal is performed using commercial Fluent CFD code with 2D axisymmetric is used to study leakage flow and windage heating for various seal clearances for solid (smooth) stator land geometry. The 2D model is subsequently used to determine average heat transfer coefficients on the rotor and stator for heating and cooling situations.

**Keywords:** CFD, labyrinth seal, windage effect, gas turbine.

**Abbreviations:** RANS – Reynolds-Averaged Navier-Stokes.

## Nomenclature

$a = \sqrt{kRT_{static, in}}$  = speed of sound (m/s)  
 $C_D = \dot{m}/\dot{m}_{id}$  = discharge coefficient  
 $C_p$  = specific heat capacity (J/Kg K)  
 $H = C_p \cdot \dot{m} \cdot \Delta T_{total}$  = windage heating power (kW)  
 $K = V_t/U$  = preswirl ratio  
 $\dot{m}$  = mass flow rate (kg/s)  
 $Mu = U/a$  = circumferential March number  
 $n$  = rotor speed (rpm)  
 $p$  = static pressure (Pa)  
 $P_t$  = total pressure (Pa)  
 $R = 287.2$  = specific gas constant (J/Kg K)  
 $R_m$  = averaged rotor radius (m)  
 $Re_x = \dot{m}/\mu_m \pi R_m$  = Axial Reynolds number  
 $T$  = temperature (K)  
 $U = \omega \cdot R_m$  = rotor circumferential velocity (m/s)  
 $V_t$  = fluid tangential velocity (m/s)  
 $\varepsilon$  - Dissipation of turbulent kinetic energy  
 $\kappa$  - Turbulent Kinetic energy  
 $s$  - Tooth pitch, mm  
 $A$ : Area [m<sup>2</sup>]  
 $L$ : Step height [mm]  
 $h$ : Teeth height [mm]  
 $N$ : Number of teeth  
 $s$ : radial clearance

**Author α:** Post graduate from BITS Pilani, India.

e-mail: mohan.ramanadham@gmail.com

**Author σ:** PhD, Professor, JNTU Hyderabad, Telangana, India.

e-mail: manzoorejntu@gmail.com

## Greek Letters

$\Delta$  = difference between inlet and outlet  
 $\kappa$  = ratio of specific heats  
 $\mu$  = dynamic viscosity (N s/m<sup>2</sup>)  
 $\pi = P_{in}/P_{out}$  = pressure ratio  
 $\pi$  = effective pressure ratio  
 $\rho$  = density (kg/m<sup>3</sup>)  
 $\sigma = 2C_p \Delta T_{total}/U^2$  = windage heating number  
 $\omega$  = rotor angular velocity (rad/s)

## Subscripts

in/out = inlet/outlet condition  
 $m$  = arithmetic average of inlet and outlet  
 $r/t/x$  = radial/tangential/axial direction  
static/total = static/total value

## I. INTRODUCTION

As the airline industry is growing OEMs are working improved and efficient Aero Engine Technologies to offer to their customers. As per ACARE 2020 norms Fuel burn, Noise and NOx emissions are to be improved significantly, in order to achieve this either new technologies that radically improve performance or identify the current design leakages so that gaps can be filled & improved.

In this paper the leakage & windage thermal characterization behaviors of Lab seals are worked, the ability to accurately model lab seals leakage flows is extremely important, as leakages can significantly affect latter stages of compressor and turbine efficiencies that cause undesirable hot gas ingestion. In order to meet the stringent norms the component material durability studies in heat generated boundary conditions by viscous dissipation or windage flow environment. Lab seals are exposed to high flow velocities contributing to elevated temperatures further enhance heat transfer coefficients.

## II. LITERATURE REVIEW

Lab seals windage heating effect was performed initially with experimental methods subsequently correlations are derived. The first of this experiment was prepared by Stocker H[3] subsequently Tipton [8] conducted test that resulted temperature profile by variable factors of pressure ratio, clearance and rotor speed. He measured the total temperature increase in the seal and reported that the total

temperature difference between the seal outlet and inlet total can reach upto 19.4K. McGeehan and Ko[4] experimented to investigate the windage heating effect and derived a correlation equation to predict the windage power loss in the labyrinth seal. The influence of the mass flow rate, rotor speed, inlet preswirl, and friction factor, the wall surface has taken into consideration in their equation.

Wróblewski [6] analyzed radiation affects that intensifies heat transfer leading temperature differences in the casing elements caused by radiation, are much smaller than those resulting from different thermal conductivity coefficient much greater than the one resulting from convection. Piotr et al [9] investigated tip leakage off-design conditions the effect of tip leakage flow rate and the tip leakage swirl angle was found to bring up to a 1.5% decrease in enthalpy losses. D. Frączek et al [11] worked out geometry for rubbing effect by modifying the fin tip inclination geometry resulted by mass flow reduction of 13%.

Denecke et al. [1] used scaling & dimensional methods to analyze the leakage and windage heating of rotating seals. By adopting the Buckingham-theorem, they found that the discharge behavior, exit swirl, and windage heating in the contact less seal can be uniquely expressed by several non-dimensional numbers. They also presented a rule to scale the engine seal to the laboratory conditions. Moreover; they measured the total temperature increase in as tepped lab yrinth seal with smooth stator.

### III. PRESENT WORK

In this work, combination of parameters to improve the seal design, the experimental case of seal configuration reported by J. Denecke, K. Dullenkopf, S. Witting and H.-J. Bauer in [1] is considered as baseline case for smooth land. In the present work, the authors conducted a CFD investigation on different configurations of canted teeth as Design 1, canted with Wavy surface as Design 2 and canted with hat step and way surface as Design 3. The design evolved while conducting a parametric study on teeth height, teeth tip thickness, stepped teeth, inclined teeth etc. The baseline configuration is a simple sharp teeth labyrinth seal. The results obtained for the baseline configurations using this methodology were validated by comparing against 2D and 3D experimental data with smooth land.

#### a) *Baseline Geometry: Sharp Teeth*

Fig. 1 shows labyrinth seal geometry with dimensions. The labyrinth seal test section consists of an upper part, stepped, and a lower part with teeth. In a real engine, the upper and lower parts correspond to the stationary and rotating parts, respectively, have 4 teeth with sharp teeth. The seals have almost similar teeth dimensions. Geometry definition: Pitch – 8 mm; step

height – 2 mm; Knife height – 4 mm; Teeth included angle 20 deg; and seal canted angle 50 deg.

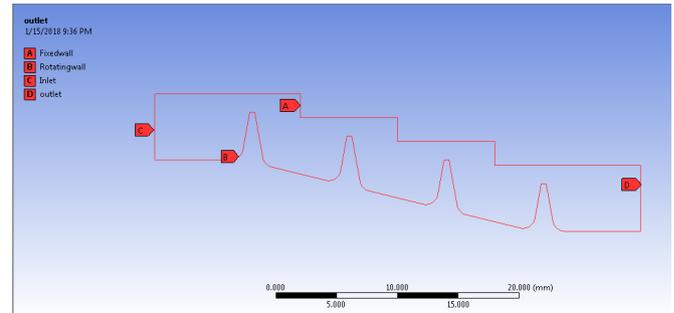


Fig. 1: Base line Design

#### b) *New Geometry: Canted Advanced Teeth*

In this work there are 3 computational geometries considered.

Design 1 – canted teeth.

Design 2 – canted teeth with wavy surface.

Design 3 – canted teeth with wavy surface having hat step.

Fig. 2 depicts the seal sharp teeth are modified to inclined to 50 Deg, rest of parameters are kept same as original design.

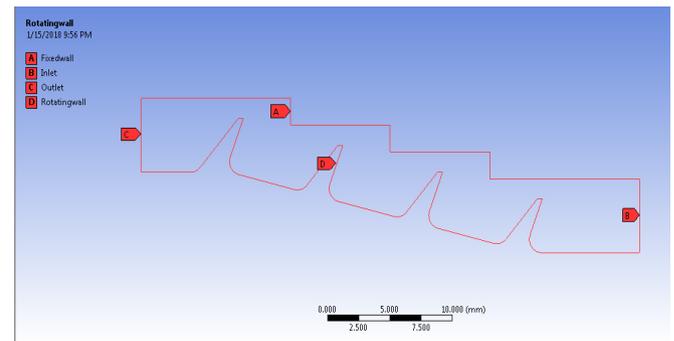


Fig. 2: Design 1 – 50 Degree canted teeth

Fig. 3 depicts the seal sharp teeth are modified to inclined to 50 Deg, with 2 wavy surfaces rest of parameters are kept same as original design.

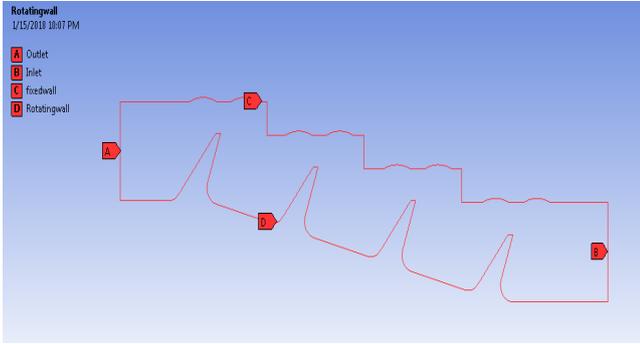


Fig. 3: Design 2 – canted teeth with wavy surface

Fig.4 depicts the seal sharp teeth are modified to inclined to 50 Deg, with 2 wavy surfaces with step 1 x 0.7 mm rest of parameters are kept same as original design.

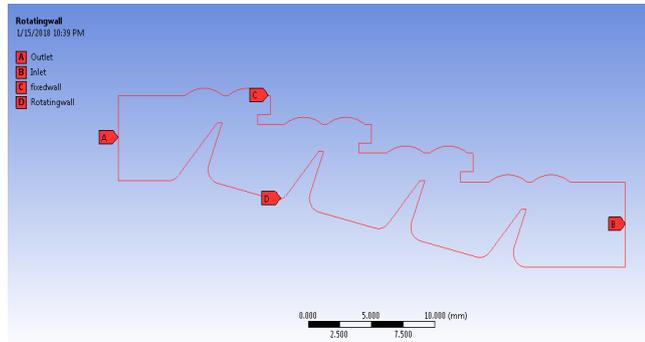


Fig. 4: Design 3 – canted teeth with wavy surface having step

c) Performance Parameters

The performance of a seal can be described by the relation between the pressure ratio and a flow parameter. The most common flow parameter is the following flow function:

$$C_d = \frac{\dot{m} \sqrt{T_{0,in}}}{A_c P_{0,in}}$$

The discharge coefficient depends on the pressure ratio, number of used seals, leakage mass flow rate. The flow through an ideal labyrinth seal.

Windage Heating:

The total temperature increase due to the internal losses in adiabatic flow is called windage heating,

$$H = C_p \dot{m} \Delta T_{total}$$

The windage heating number  $\sigma$  is defined as

$$\sigma = \frac{2C_p \Delta T_{total}}{U^2}$$

IV. ANALYSIS

a) Numerical Analysis

Computational Fluid Dynamics (CFD) is extensively used because its capability to analyze a large number of design configurations and parameters in a relatively short period of time. Therefore, with the development of commercial codes, the use of CFD analysis has been increasing rapidly in recent years in 2 or 3 dimensional analysis. The test cases adopted in this work are two-dimensional with number of different operating conditions are analyzed.

b) Boundary Conditions

A commercial finite volume code, Ansys Workbench with Fluent 16.1v [10] is used. This commercial tool has wave linking geometry that eliminates loss of geometry while importing from CAD model to Fluent, manages entire problem in project charter. It was assumed that air was an incompressible ideal gas and the flow was steady and adiabatic. Various turbulence models available in Fluent [10] were considered for the current simulations.

Table 1

Average rotor radius	0.253m
Inlet total temperature	300K
Outlet static pressure	200,000Pa
Turbulence model	Realizable k-epsilon,
Fluid	Air ( ideal gas )
Wall Properties	Adiabatic, smooth surface

Fig. 5, 6 & 7 shows an example of generated computational grids. The grid density in the clearance area was refined to locate sufficiently large number of meshes, this is done using inflation of smooth transition with 8 ~ 10 layers.

c) Mesh

Design 1: Canted with 1.3 ~1.9 gap, Mesh .02 mm to .035 max

Inflation: smooth with 10 layers. 192625 mixed cells.

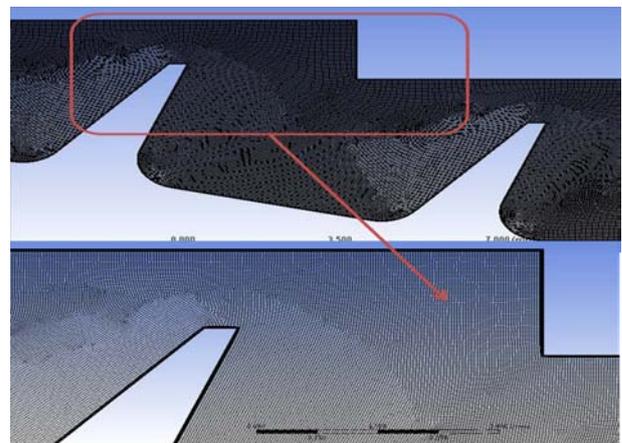


Fig. 5: Mesh Details of Design 1

Design 2: Canted with 1.3 ~1.9 gap, Mesh .025 mm to .035 max. Inflation: smooth with 10 layers. 214923 mixed cells.

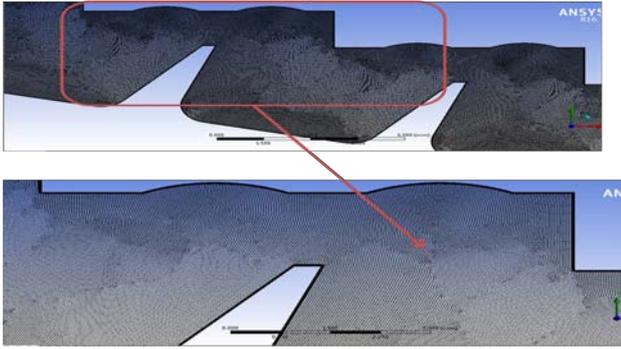


Fig. 6: Mesh Details of Design 2

Design 3: Canted with 1.3 ~1.9 gap, Mesh .025 mm to .035 max.

Step Height: 1x.7.

Inflation: Smooth with 10 layers. 220485 mixed cells.

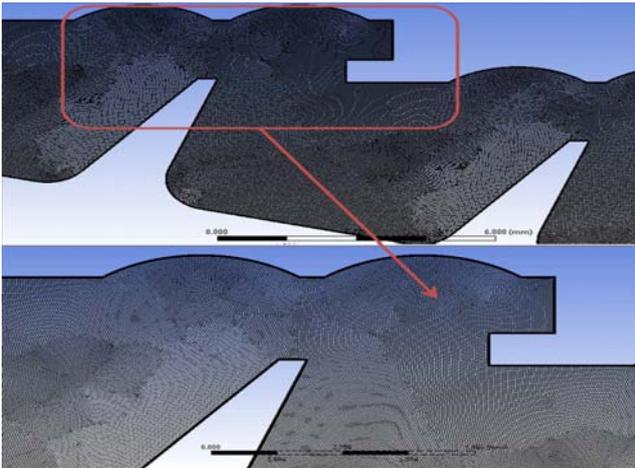


Fig. 7: Mesh Details of Design 3

d) Mesh Sensitivity

Grid dependence was checked to produce sufficiently converged solutions according to mesh size. grid dependence in the case of the Base geometry seal, ranges from 26,000 to 220,000 considered variation of pressure ratio with the number of meshes for a given mass flow rate. Convergence plots for basic design and design 1 are shown below.

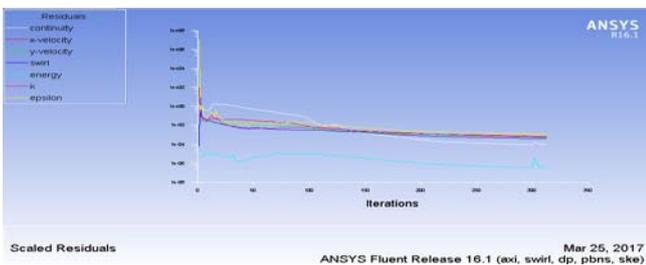


Fig. 8: Convergence plot for designs for double precision- Basic Design

The convergence plots are done for mass flow 0.15 m/s. number of iteration for basic design is very less compared to Design 1 and others.

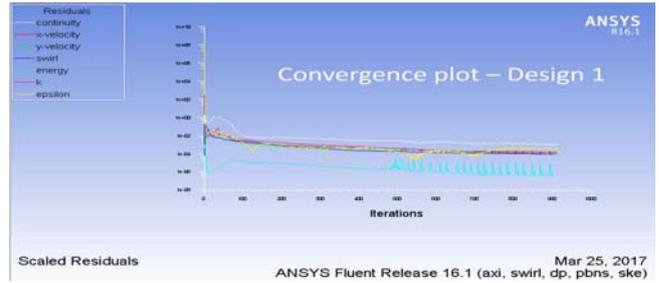


Fig. 9: Convergence plot for designs for double precision- Design 1

Analysis is carried out with basic design with Inlet and outlet conditions as per the reference[1] with Convergent and divergent labyrinth seal. The conditions are matched with the experimental resulted. Solution is imposed values for convergence factors equal to  $10^{-2}$  exception is the energy equation with factors of convergence  $10^{-3}$

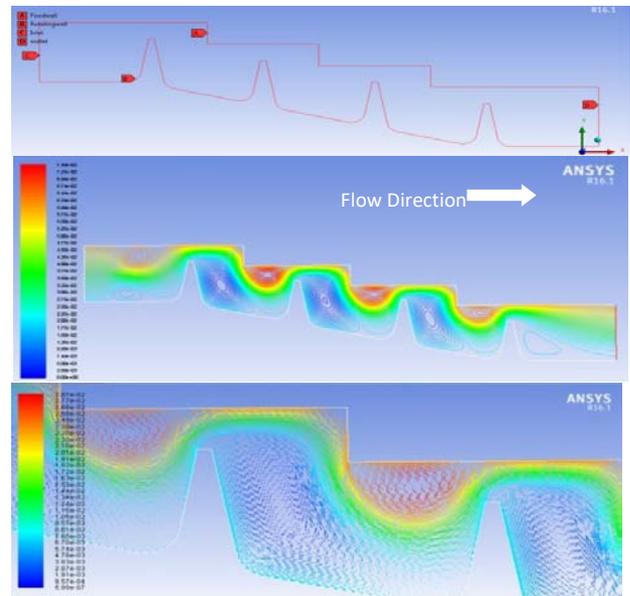


Fig. 10: Basic Design

The flow directions is reversed [Basic design] and analysis is carried out with various mass flow conditions at speed of 20,000 r.p.m., the delta temperature difference for basic design is gradually reduced as the mass flow increases whereas as the basic design with reverse flow the delta temperature is almost study. With clearance of 1.5 the temperature rise is high until mass flow of 0.15 then gradually reduced whereas for clearance 1.7 & 1.9 the temperature rise is study, Refer fig. 11 for the graph.

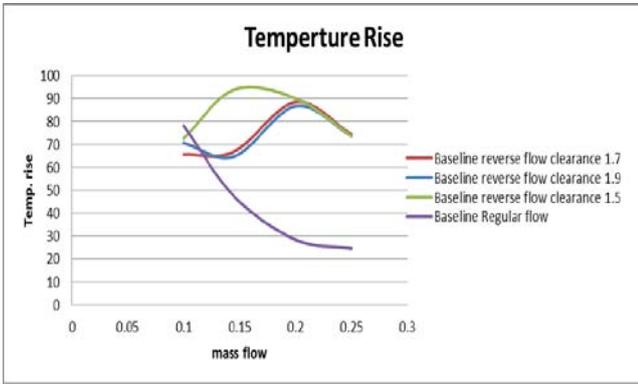


Fig. 11: Basic Design flow vs delta temperature rise

The flow in the labyrinth seal is mainly dominated by pressure difference in the axial direction. The study of interest is on reverse flow direction. The Models of basic design and Design 1 to 3 are run through by varying mass flow with boundary condition as per table 1.

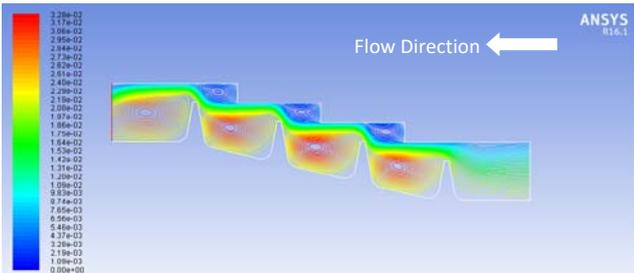


Fig. 12: Basic Design velocity contours

The Wall shape change introduced in the fixed wall refer design 1 to design 3 , two things will happen either wall shear increase or decrease and second one will be flow circulations. Design 1 will be canted shape and fixed wall is straight. Design 2 the seal shape is canted and fixed wall is having 2 dimple shapes. Design 3 is having canted seal with fixed wall having dimple shape with hat projection.

The analysis is run with Table 1 boundary condition and the mass flow is varied with .1 to .25 kg/s for basic design and design 1 to 3 with flow direction as shown in figs 12 to 15. The jet stream in design1 continues its full contact with fixed wall at exit.

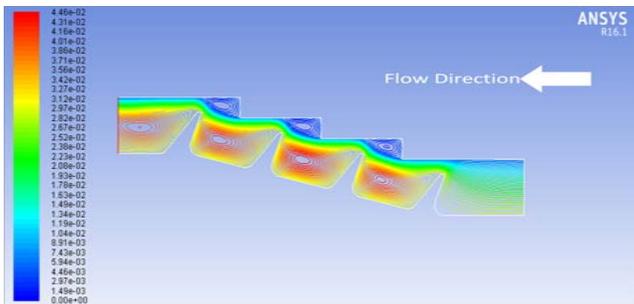


Fig. 13: Design1 velocity contours

In Design 2, the jet stream slides over the concave surface of fixed wall , the velocity vectors hits down ward path by not sliding over the remaining straight surface of fixed wall as shown in Fig. 14.

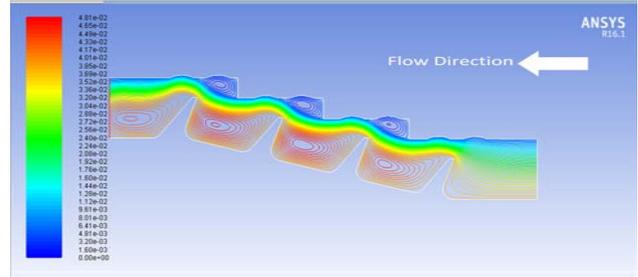


Fig. 14: Design 2 velocity contours

In Design 3, the velocity vectors hit the walls of stationery surface of concave with double step creating static current that obstruct the main jet shown in Fig. 11, at the exit the jet diffuse compared to Design 2.

In design 3 the jet stream hits the concave surface of fixed wall and reflects back to downwards their by losing its velocity. In this the jet is like jig jag pattern. The hat shape cavity at fixed wall creates a seal lock hence reduce leakage.

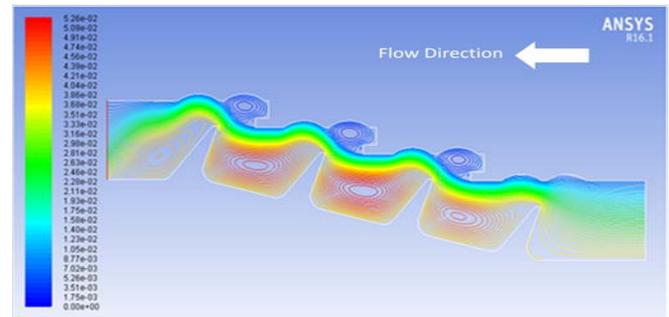


Fig. 15: Design 3 velocity contours

The results are plotted with flow rate with pressure ratios windage heat, swirl velocity and temperature variation. Refer figs 16 to 19.

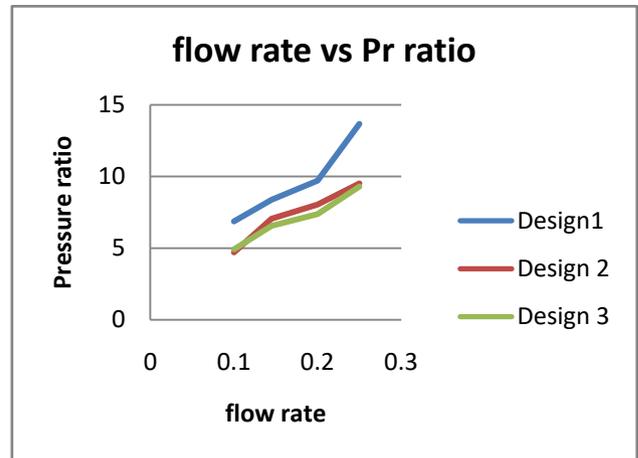


Fig. 16: Mass flow vs Pressure ratio for various shapes

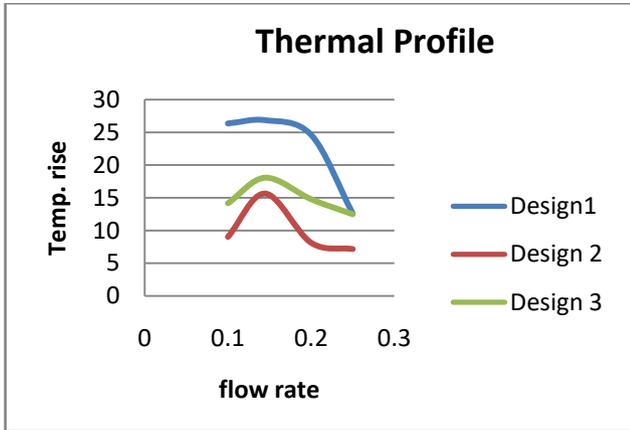


Fig. 17: Thermal profile of various designs

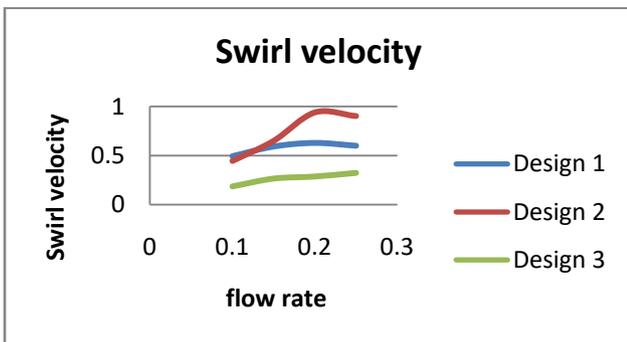


Fig. 18: Swirl Velocity

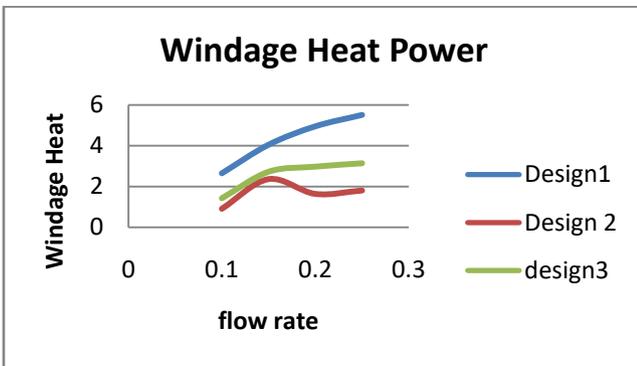


Fig. 19: Windage Heat Power

## V. RESULTS AND DISCUSSION

### a) Turbulence Model Study

2D axisymmetric analysis of the Labyrinth Seal geometry (with rotation speed of 20,000 rpm) was first carried out. The objective was to establish baseline capability to run lab seal CFD analysis using Fluent and validate the experimental results.

### b) Analysis of Results with Different Seal Configurations

As observed in Figures 13 to 15 the fluid flow into the seal and accelerated towards the clearance gap. The jet separates the flow into two counter rotating vortices causing the pressure to increase, it disrupts

flow entering the clearance region and increases flow blockage thereby reducing the exit pressure at each tooth.

Refer figures 16 to 19, as the flow rate increases the pressure ratios is also increased thus linear in nature for all the 3 designs. Its observed that the design 1 is having higher pressure ratios than other two designs. The temperature rise in Design 1 is higher than the other two designs. Design 3 shows that less swirl velocity compared to two other designs. Design 1 and 2 are starting with same swirl velocity but as it progresses design 2 has steep increase. The windage heat power is higher in design 1 compared to two other designs.

The seals can further be studied with honeycomb fixed wall by various speed parameters. It may further studied by introducing air injection between knife edges and swirl interrupting steps that would create turbulence flow in the flow path that restricts movement.

## VI. CONCLUSIONS

A new model of canted teeth with different wall shapes for labyrinth seals has been studied numerically and compared to base line sharp teeth with smooth land. The temperature rise and windage heat power is high in Design 1. Analyses indicate that the design 3 is capable of reducing overall seal leakage by generating more turbulence and friction flow.

## ACKNOWLEDGMENT

The authors wish to thank kondala Rao N and Suresh B for his guidance during case studies.

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