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FLUIDIC JET BARRIERS FOR SEALING APPLICATIONS

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ABSTRACT

The turbine industry is continually looking for new developments to improve thermodynamic performance and sealing has received significant attention over the years. Fluidic seals employ aerodynamic flow features to create blockage/loss and reduce leakage, rather than relying on physical barriers to flow such as brush seal bristle packs etc. They are also potentially cheaper to implement than contacting seal technologies such as brush seals.

The fundamental mechanism by which fluid jets inclined in an upstream direction produce blockage and reduce the flow along leakage channels are examined in the paper. Computational Fluid Dynamics is used to quantify the net gain in leakage performance that can be achieved in simple channel flow for various operating conditions and jet configurations. These results are used to guide further CFD calculations in which the potential for leakage reduction from adapting conventional labyrinth turbomachinery seal designs to include fluidic jets is investigated. Calculations are carried out for operating conditions that are typical of gas and steam turbine applications, in order to demonstrate the potential of new seal designs of this generic type. The device considered in the paper is essentially a conventional labyrinth seal design which is modified to include internal flow channels within the structure supporting the labyrinth fins, to supply the fluidic jets. The new technology is therefore a modification to an existing component with potential for application in existing turbine designs, requiring no/minimal changes outside of the seal design space to implement.

NOMENCLATURE

A Leakage Channel Cross-Sectional Area (m^2)
f_1, f_3 Wall Friction Factor Upstream and Downstream of the Jet entry point (Figure 3)
HP Typical High Pressure Steam Turbine Conditions (Table 1)
h Leakage Channel Height (m)
h_0 Stagnation Enthalpy (J/kg)
LP Typical Low Pressure Steam Turbine Conditions (Table 1)
L_1, L_3 Leakage Channel Lengths Upstream and Downstream of the Jet Entry Point (Figure 3)
m Mass Flow Rate (kg/s)
PR Pressure Ratio (= p_{02}/p_{01} , or equation 1)
p_0 Total Pressure (bar)
p Static Pressure (bar)
V Flow Velocity (m/s)
W Leakage Channel Width (m)
ρ Density (kg/m^3)

Subscripts
1 Seal Inlet/Inlet Plenum
2 Jet Inlet/Jet Supply
3 Seal Outlet/Exit Flow

INTRODUCTION

Conventional gas and steam turbine designs make extensive use of labyrinth type seals to minimize unwanted leakage flows between stationary and rotating components. Labyrinth seals are non-contacting aerodynamic seals traditionally used in turbomachines to reduce unwanted leakage flows between stationary and rotating components. They are by far the most commonly used technology for turbine stage and turbine shaft gland sealing. Figure 1 shows a typical design for a steam turbine spring backed shaft gland seal (rotor not shown). The seal consists of a series (in some applications many 10's) of tight restrictions between stationary and rotating surfaces. These are formed by fins on the stationary and in some designs also the rotating components, with small (usually order 1mm) clearances between the tips of the fins and the adjacent sealing surface. The leakage flow across each restriction forms a jet which expands into the relatively large volume between the sealing fin forming the jet and the next fin downstream. The rapid acceleration of the leakage flow to form the jet under each restriction, followed by the uncontrolled expansion into the cavity immediately downstream of the fin (a process during which the kinetic energy of the leakage jet is lost), is a very tortuous path for the flow to follow. Consequently, a large pressure drop is required to force relatively small quantities of flow through the labyrinth. This technology therefore offers a relatively simple and low cost device for restricting leakage flow, that can operate successfully as a rotating aerodynamic seal in the challenging environments found in gas and steam turbines.

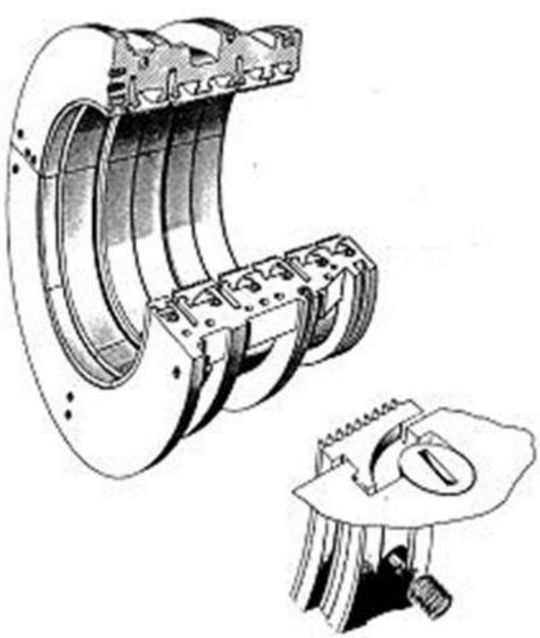

Figure 1: Typical Industrial Labyrinth Type Seal Arrangement used on a Steam Turbine Shaft Gland (the turbine rotor is not shown).

Recently (i.e. over the last 2 or 3 decades), researchers have been attempting to develop new types of contacting seal designs such as brush seals (Figure 2), in order to further reduce leakage flows and thereby improve turbine efficiency [1,2].

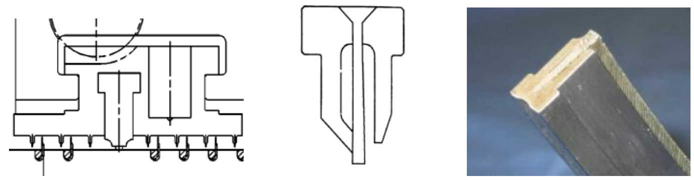

Figure 2: Brush Seal Development for Turbine Applications.

There have been many studies that show very encouraging performance of brush seals (and other contacting seal types such as leaf seals [3], finger seals [4] etc.) in test rigs. Much of the service feedback to date from applications using contacting seal technologies has shown that durability of seals can be a barrier to successful long-term application in real gas and steam turbine environments. Real turbine phenomena such as high velocity impacts from solid particles carried by the flow, shaft vibration and high levels of swirl in the approaching flow, have been identified as potential sources of damage to contacting seals, causing bristles/leaves to break-off in service resulting in a rapid degradation in sealing performance.

The turbine industry now fully recognizes the challenges associated with developing cost effective and robust contacting type turbomachinery seal technologies and although development work in this area has far from ceased, some workers are now turning attention to other potential new seal types. One such seal type, recognized for many years, but only now starting to receive increased attention for the reasons given above is fluidic seals.

In fluidic type seals, aerodynamic flow features are employed to create loss along a leakage path rather than physical barriers to flow such as bristle packs. The purest form of fluidic seal relies on the jets alone to form a 'curtain' of fluid across the leakage channel that creates blockage and forms the seal [5]. Clearly there is a trade-off for designs of this kind because some flow (which could otherwise be expanded through the turbine blades to produce useful work) has to be taken from upstream of the seal to form the fluidic sealing jet. So, the improvement in shaft output resulting from the better sealing performance due to the fluidic jets, must be greater than the loss in shaft output due to the extraction of some upstream flow to form the fluidic jets, for seal designs of this type to be worthwhile. This is obviously a critical criteria of fundamental importance to all types of fluidic seal design.

In the work described in this paper, the performance of simple 'air-curtain' type fluidic seals is explored first. These results are then used to benchmark advanced fluidic seal designs for turbomachinery applications. Finally, a complete new type of fluidic sealing system is proposed which can essentially be applied within the same design space as conventional labyrinth seals, that has the potential to deliver improved sealing performance without the cost and durability issues associated with contacting (e.g. brush & leaf) type seals.

PERFORMANCE OF 'AIR CURTAIN' TYPE FLUIDIC SEALS.

The geometry used to investigate the performance of simple 'air-curtain' type fluidic seals is shown in Figure 3 (cross-flow jets in a uniform leakage channel). The geometry chosen consisted of a leakage channel that was 180 mm in length with a height of 1mm. An inlet plenum was included in the calculations at the channel inlet so that a uniform total pressure inlet boundary condition could be applied away from the region where the flow is accelerating towards the channel inlet. The outlet boundary was positioned at the channel exit where a static pressure boundary condition was applied. The fluidic cross-flow jets were 1mm diameter with a centerline spacing of 4mm. The fluidic jets were supplied (same fluid as the main leakage channel flow) from a second plenum, allowing a uniform total pressure boundary condition to be applied in the jet supply plenum. Jet fluid supplied from the plenum flowed through 3 mm long jet channels before issuing into the main leakage channel flow. The 3-D calculation domain included 3 jets (as shown by the solid model in Figure 3). Symmetry boundary conditions were applied on either side of the leakage channel.

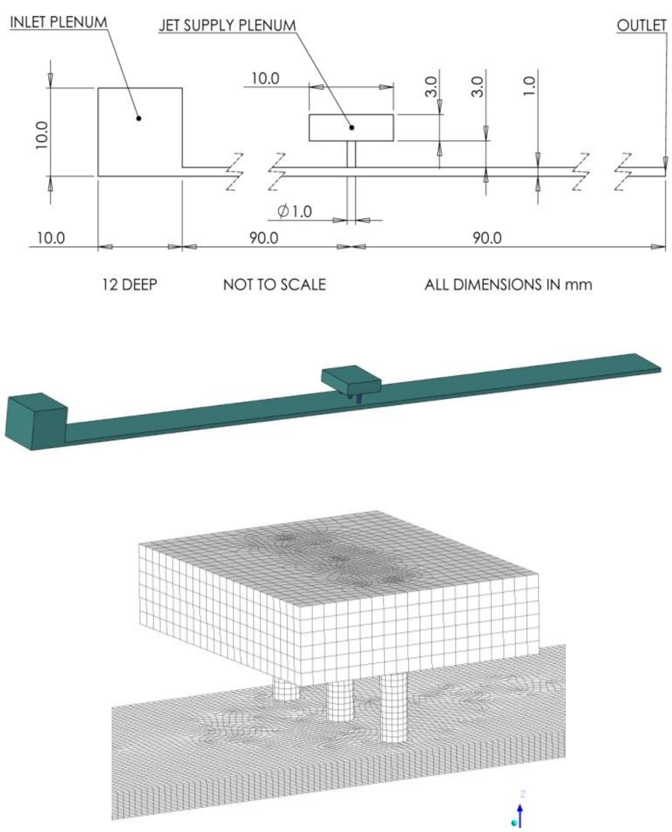

Figure 3: Geometry used in the Calculations to Investigate the Performance of 'Air-Curtain' Type Fluidic Seals.

3-D steady CFD calculations were carried out using the Fluent commercial CFD package. A section of the unstructured mesh in the region surrounding the jets generated using Gambit is also included in Figure 3. The mesh size was approximately 500.000 hexahedral cells. Grid dependency was checked by carry out calculations with half this number of cells. The results were found to vary by less than 1%. Turbulence was modeled by the k-ε model with a 5% turbulence kinetic energy boundary condition applied at all inlets. Wall functions were used for the near wall treatment and y-plus values were maintained in the range 20-250 in the calculations. Solutions were obtained using 2nd order upwind discretization. The calculation results were validated against those from a simple analytical model based on the global conservation equations for mass, momentum and energy for compressible flows (see Appendix A). The leakage flows predicted by both the CFD and the analytical model showed good agreement.

Calculations were carried out in air. Initial calculations used perpendicular jets. The total pressure in the leakage channel inlet plenum was set to 2.0 bar and the static pressure at channel exit was 1.5 bar. Calculations were carried out for a range of values for the fluidic jet supply plenum total pressure. These were set according to the ratio of jet supply plenum to inlet plenum total pressure, PR = 1.0 to 2.0 in steps of 0.2 i.e. the jet supply plenum total pressure was varied from 2.0 bar to 4.0 bar in the calculations. The inlet boundary condition for total temperature of the air in both plenums was 20^{0}C. The results of these calculations are shown in Figure 4. As has already been explained, the calculation domain consisted of three equally spaced jets. The flow around each jet is symmetric about planes that bisect the jets. The mass flow values in Figure 4 (and those shown later in Figure 6), are based on the results of the calculations with 3 jets scaled up to 25 jets in total, in order to increase the mass flows to values that are more convenient to plot.

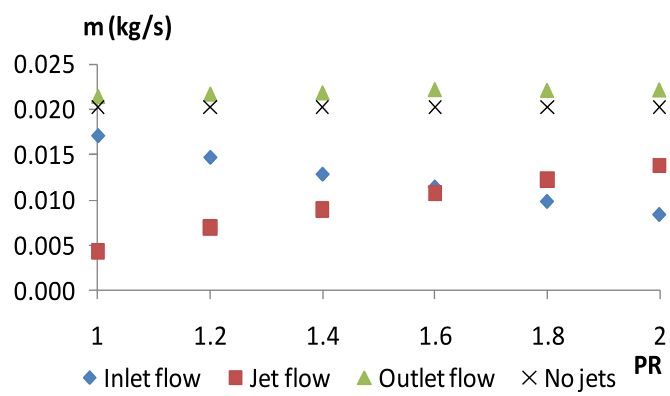

Figure 4: Predicted Leakage Flow for Air Curtain Seal with Perpendicular Jets.

Changes in the leakage jet mass flow (square symbols) and the flow that enters the leakage channel from the inlet plenum (Inlet flow – diamond symbols) with jet supply plenum pressure ratio are shown in Figure 4. The Reynolds number (based on jet diameter) of the jets shown in Figure 4 lie in the range 12000 to 40000. It is clear that as the pressure in the plenum supplying the fluidic jets and therefore jet mass flow is increased, blockage of increasing strength is created within the channel, resulting in a reduction in the leakage flow entering the channel (Inlet flow).

The cross-flow jets create blockage because the jet fluid has to gain momentum in the direction of the main leakage flow after it enters the channel. This process is illustrated in the velocity distribution shown in Figure 5. This example shows a perpendicular jet entering the leakage channel for PR=1.4. The jet flow enters the channel initially with just momentum in the cross-flow direction. As the jet flow enters the channel it is accelerated in the direction of the main channel flow, which can be seen in Figure 5 by the curvature of the jet flow in the downstream direction within the channel flow. A force must be applied to the jet flow to accelerate it in the direction of the channel flow. This forces comes from a static pressure drop in the main leakage channel across the jet entry region. The higher the value of PR, the greater the jet mass flow and therefore amount of fluid to be accelerated, hence the greater the drop in static pressure across the jet. The static pressure in the leakage channel immediately downstream of the jet mixing region is governed by the channel exit flow static pressure boundary condition and the losses due to wall-friction in the downstream section of the leakage channel. These losses will depend on the outlet mass flow, which can be seen in Figure 4 to remain almost constant over the range of PR values calculated. So, increasing PR increases the jet mass flow rate which increases the static pressure drop in the channel across the jet mixing region. The pressure immediately downstream of the jet mixing does not vary significantly with PR. Therefore, the static pressure in the channel immediately upstream of the jet entry region must increase with PR. Inlet plenum total pressure is constant in the calculations. Increasing the static pressure in the channel immediately upstream of the jets in this way, results in a reduction in the pressure drop driving the channel Inlet flow. Hence the reduction in Inlet flow with PR shown in Figure 4.

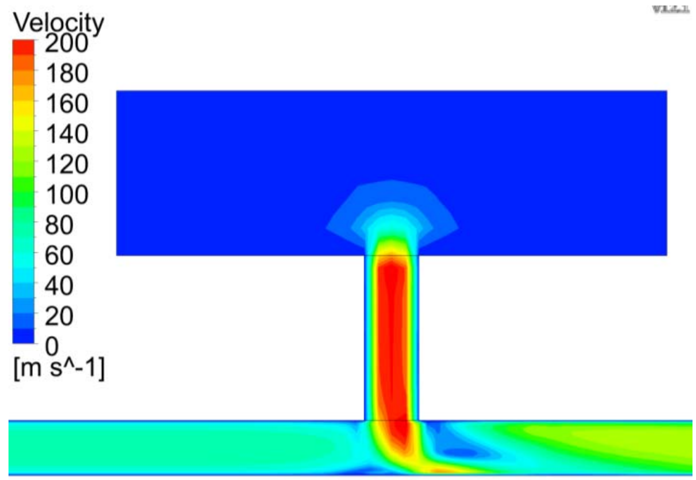

Figure 5: Velocity Distribution showing Mixing between a Perpendicular Leakage Jet and the Main Leakage Channel Flow (PR=1.4 in the example shown).

Outlet flow (triangular symbols in Figure 4) is the total leakage flow exiting the leakage channel i.e. the sum of the Inlet flow and the Jet flow. It has already been noted that the Outlet flow does not change significantly with PR in Figure 4. The predicted flow through the leakage channel in the absence of any fluidic jets is shown by the crosses in Figure 4 at each value of PR for comparison. This shows that even though the fluidic jets are successful at creating blockage, the total leakage flow exiting the channel is always greater than that with no jets present. In fact, the results in Figure 4 show that the increase in total leakage flow (Outlet flow) above the 'No jet' leakage gets slightly greater as PR is increased. The conclusion is that it is not possible to improve leakage performance overall by using an air curtain formed from perpendicular jets.

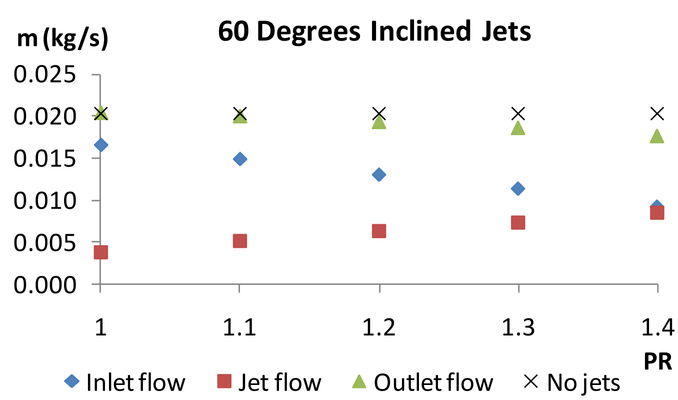

Figure 6: Predicted Leakage Flow for Air Curtain Seal with Jets inclined 60° from perpendicular in the upstream direction.

Further calculations were carried out over a range of other jet angles in which the jets were inclined such that some of the jet momentum opposed that of the flow through the leakage channel. The mechanism described previously by which cross-channel jets create blockage can be enhanced by inclining the jets so that the jet fluid enters the channel with a component of momentum in the opposite direction to the leakage channel flow. This increases the force required to accelerate the jet fluid in the direction of the leakage flow and therefore increases the blockage effect from the jets. Figure 6 shows a similar plot to Figure 4 except in these calculations the jets have been inclined so that they are introduced against the main channel flow at an angle of 60^o to the perpendicular jet direction i.e. 30^o from horizontal when the geometry is viewed as in Figure 3. Calculations are shown in Figure 6 for jet supply plenum pressure ratio PR = 1.0 to 1.4 in steps of 0.1. The results in figure 6 show that total channel leakage can be reduced compared to the no fluidic jet case, when the jets are inclined against the channel leakage flow. The leakage reduction due to the presence of the fluidic jets is seen to increase with PR for the inclined jet case, but, even for the highest pressure ratio calculated, PR = 1.4, the reduction is modest in the region of 5% to 10% of the total leakage flow.

The overall conclusion from this first set of calculations was that using fluidic jets to create simple ‘air curtain’ type blockage across leakage channels has only limited potential for reducing leakage flow.

PERFORMANCE OF LABYRINTH SEALS WHEN AUGMENTED BY FLUIDIC JETS.

One alternative to the simple ‘air curtain’ type fluidic seal described in the previous section is to combine fluidic jets with conventional labyrinth seal technology for improved sealing performance. Figure 7 shows one embodiment of a seal of this type from a patent by Turnquist et.al.[6]. In this design of fluidic seal, a jet of flow from upstream of a labyrinth is introduced part-way along the seal leakage path in an upstream direction, in order to promote additional blockage against the leakage flow. The labyrinth design shown has inclined fins and the aim of introducing the jet is to improve the performance of the labyrinth seal beyond that of the labyrinth without the addition of the fluidic jet.

In the present study, leakage flow predictions were carried out for the geometry shown in Figure 8. The geometry consisted of a 3 fin ‘see-through’ labyrinth seal with the central fin inclined by 45^o against the oncoming leakage flow. Steady 2-D RANS CFD calculations were carried out on this geometry using Fluent. An unstructured mesh consisting of 175.000 triangular cells was created with Gambit. A section of the mesh in the region of the central inclined fin is shown in Figure 8. The clearance between the tips of all three fins and the opposing sealing surface was set to 1 mm in all calculations. The sealing surfaces were stationary in the calculations (i.e. no surface rotation was included). In general, surface rotation is not expected to have a significant effect on labyrinth seal leakage flow. The k-ε turbulence model was used with a 5% turbulence kinetic energy boundary condition applied at all inlets. Wall functions with y-plus values maintained within the range 20-300 were used for the near wall treatment. Solutions were obtained using 2nd order upwind discretization.

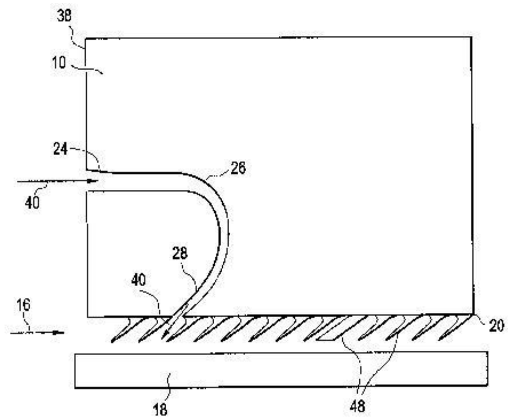

Figure 7: Fluidic Seal Design Consisting of a labyrinth Seal Augmented by an inclined Fluidic Sealing Jet (Figure taken from GE Patent No. US2009/0297341 A1).

Fluidic jets were introduced into the geometry by injecting fluid into the calculation domain along the upstream face of the central inclined labyrinth fin as indicated in the geometry shown in the lower part of Figure 8. The jets were modeled as a slot 0.5 mm wide in the 2-D calculation domain as shown in the figure.

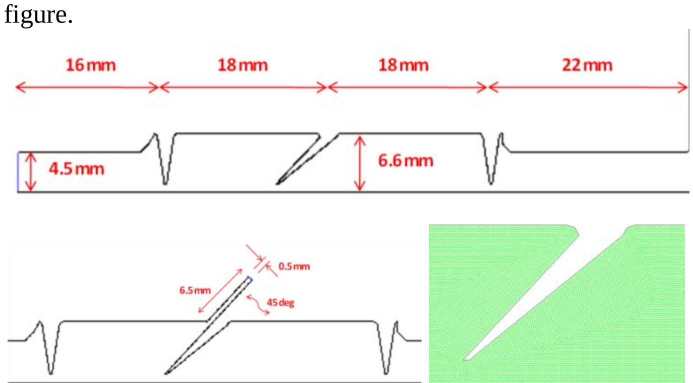

Figure 8: Idealized Labyrinth Seal Geometry and Section of Mesh used to CFD Study.

Leakage predictions were made for two sets of boundary conditions. These were selected to be representative of typical turbine stage sealing conditions in high pressure (HP) and low pressure (LP) steam turbine cylinders.

	HP Conditions	LP Conditions
Inlet Total Pressure, p_{01} (bar)	100	95
Exit Static Pressure, p_3 (bar)	5	4.75

Table 1: Boundary Conditions used in HP and LP CFD Predictions

The seal inlet and exit boundary conditions used in the HP and LP calculations are given in Table 1. Calculations were carried out with air as the working fluid and the temperature boundary conditions were selected to give fluid densities in the calculations that were representative of steam at the two sets of boundary conditions calculated.

The total pressure inlet boundary condition for the fluidic jet flow (p_{02}) was again set on a pressure ratio basis. A different definition of pressure ratio, PR, was used for the labyrinth seal calculations compared to the definition used in the 'air curtain' fluidic seal calculations described in the previous section.

The definition of PR used in these calculations was:

$$PR = \frac{p_{02} - p_3}{p_{01} - p_3} \qquad (1)$$

So, a value of PR=1.0 corresponds to the situation were the total pressure of the flow supplying the leakage jet is equal to the total pressure of the flow at the inlet to the labyrinth seal. This would be the case for the geometry shown in Figure 7. Values for PR that are greater than unity correspond to situations where the jet supply total pressure is greater than the seal inlet total pressure, by some proportion of the total to static pressure drop across the seal. CFD calculations were carried out for values of PR in the range 1.0 to 1.4. So, at the highest value of PR, the total pressure of the fluid supplying the fluidic jet was set to be 40% of the total to static pressure drop across the labyrinth, greater than the total pressure at the seal inlet. For HP and LP conditions this corresponds to p_{02} = 102 bar and p_{02} = 5.1 bar respectively. It should be noticed that these are in effect only very modest increases in absolute pressure level above the conditions at the seal inlet.

Figure 9 shows the predicted leakage flows from the 2D CFD calculations at both HP and LP conditions per meter length of seal. The format of the plots is the same as that described in the previous section in relation to figures 4 & 6 showing the 'air curtain' seal leakage predictions. Comparing the results shown in Figure 9, with the earlier figures, illustrates much greater leakage reduction potential when fluidic jets are used to augment labyrinth seals, compared to what can be achieved with the jet flow alone ('air curtain' type seal). The data in Figure 9 shows that for PR=1.4, total seal leakage flow reductions (comparing triangular symbols with crosses in the figure) of around 25% are predicted for the geometry calculated at both HP and LP conditions. The Reynolds number (based on slot width) of the jets shown for the HP conditions in Figure 9 lie in the range 20000 to 35000. The Reynolds number of the jets under LP conditions is an order of magnitude lower lying in the range 3000 to 6000, although the proportional reduction in leakage due to the jets is similar to that at HP conditions. When comparing this result with those discussed in the previous section, it should also be borne in mind that, PR=1.4 defined according to equation 1 translates to a proportionally much lower value for p_{02} in the labyrinth seal calculations, then the value of p_{02} corresponding to PR=1.4 under the different definition of pressure ratio (PR = p_{02}/p_{01}) that was used in connection with the 'air curtain' seal calculations.

The velocity distribution for PR=1.2 is shown in Figure 10. This illustrates the mechanism by which the presence of the jets modifies and acts to reduce the leakage flow through the labyrinth seal. Higher values of flow velocity can be seen in the leakage flow through the restriction formed by the central inclined fin compared to the flow underneath the perpendicular final restriction closest to the seal exit. Both restrictions have the same 1.0mm clearance and the leakage mass flow rate is the same for both of these restrictions. The higher flow velocity (> 160 m/s) in the flow underneath the inclined fin is caused by the momentum of the jet along the upstream surface of the fin impinging on the leakage flow as it passes through the restriction, increasing the radial momentum (compared to the no jet case) of the flow as it passes through the restriction. This will strengthen the vena contracta of the leakage jet, resulting in an effective reduction in leakage flow area and therefore an improvement in sealing performance. The vena contracta effect is not as great for the downstream conventional labyrinth restriction and lower velocities are seen in the flow through this restriction.

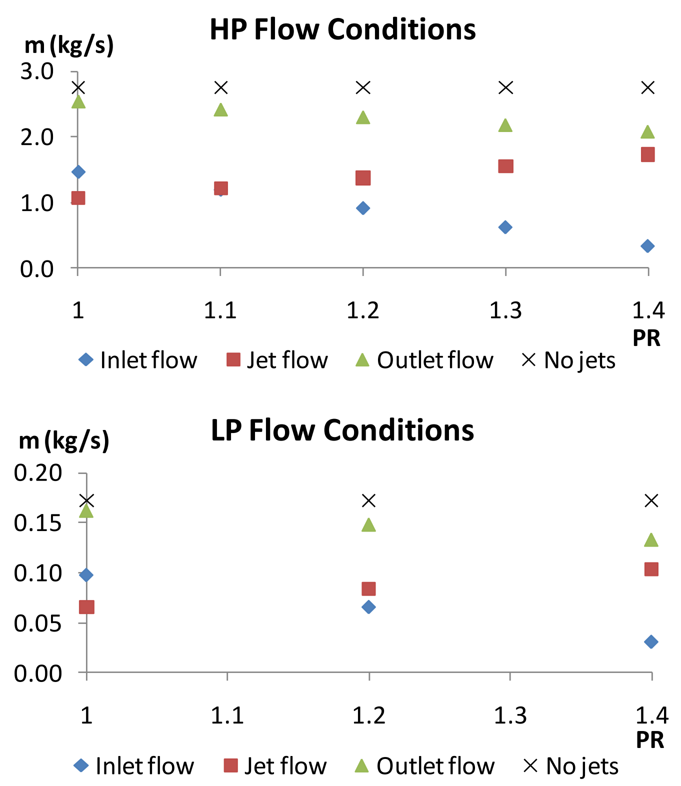

Figure 9: Effect of Fluidic Jet Strength on Predicted Leakage through the Idealized Labyrinth Seal Geometry for HP and LP Flow Conditions.

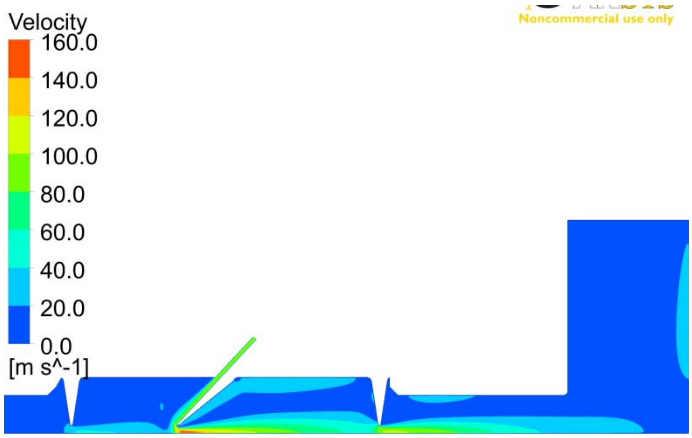

Figure 10: Velocity Distribution showing how Jet Momentum reduces the Vena Contracta of the Leakage Flow across the Inclined Central Labyrinth Restriction. (PR=1.2 in the example shown).

In turbines, the seal pressure drop is much lower than the absolute pressure level in almost all sealing applications. The leakage flow predictions that have been described in this section show that large reductions in leakage (order 25% demonstrated for the geometry calculated) can be achieved with relatively small increases in absolute jet supply pressure above seal inlet pressure. The increase in jet supply pressure above seal inlet pressure as a proportion of the pressure drop across the seal (equation 1) is an important pressure ratio.

PROPOSAL FOR A PASSIVE FLUIDIC SEALING SYSTEM FOR TURBOMACHINERY APPLICATIONS.

The idea of elevating the total pressure of the fluid supplying the jets in fluidic seals is not a new one. Figure 11 is from a GE patent by Auyer [7], registered in 1954. In this design, jet supply fluid that is bled from the compressor in a gas turbine is used to supply a fluidic 'air curtain' type turbine blade tip seal.

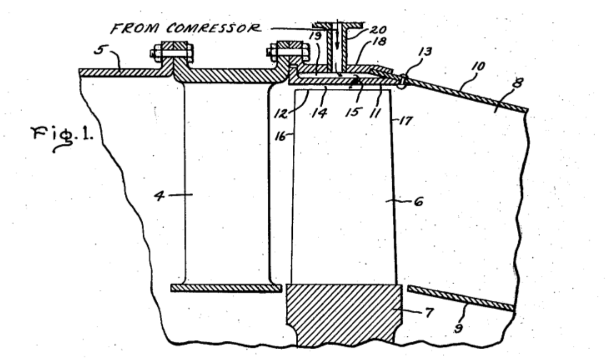

Figure 11: Externally Pressurized (Flow Bled from Compressor) Air Curtain Type Fluidic Seal Design (Figure taken from GE Patent No. 1017525.5).

This type of arrangement would be very challenging to implement in a real machine in a way that delivers an overall evaluated benefit. The system would be complex to implement, expensive, difficult to maintain and any useful work that the fluid bled from the compressor could have done prior to its re-injection downstream of the turbine stage, will be lost. What is required is a simpler and less expensive, in terms of both cost and loss of work potential of the fluid supplying the jets, method for supplying fluid at elevated pressures to the sealing jets.

Figure 12 shows a schematic diagram of a turbine stage, with sealing systems where fluidic jets are being used to augment labyrinth seals. In the sealing systems shown, the fluidic jet supply pressure is raised to levels in excess of the seal inlet pressure, by recovering some kinetic energy from swirl in the flow upstream of the seal. This system is described in detail in UK Patent Application No. 1017525.5 [8]. In both the turbine blade shroud tip seal and the diaphragm glade seal in Figure 12, the fluidic jets augmenting the labyrinth seals are supplied from reservoirs immediately above the labyrinths. The reservoirs are fed through ports communicating with the flow on the upstream side of the labyrinth in the seal inlet region. The flow upstream of the seal will contain swirl either by virtue of the turning of the flow through the turbine stage nozzles in the case of the tip seal, or by virtue of the rotation of the turbine shaft in the case of the diaphragm gland. Devices are included at the inlet to the reservoir supply ports such that their inlet plane is normal to the circumferential direction and facing towards the swirl component of the flow. The flow is turned between the entry to the supply ports. The summed cross-sectional area of the supply ports is greater than that of the fluidic jets and so velocities within the reservoir chamber are low. The flow therefore essentially stagnates in the inlet of the supply ports and some of the kinetic energy from the swirl in the flow entering the ports will be recovered as pressure. The recovery of swirl kinetic energy will cause the pressure in the reservoir to be increased to the extent that the effective value of PR (as defined in equation 1) will be raised above unity for the seal.

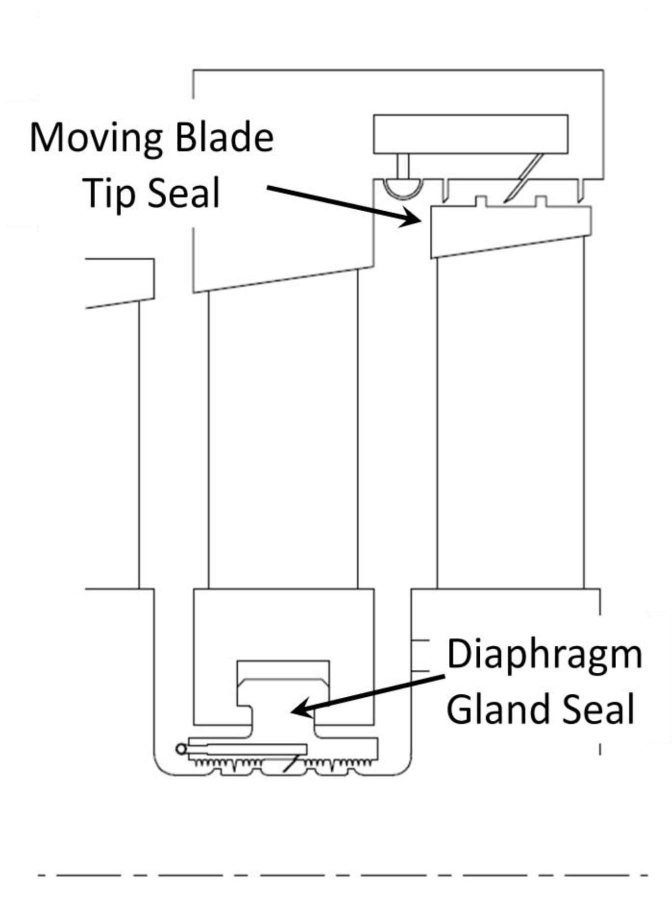

Figure 12: Schematic Diagram of a Turbine Stage showing a passive system for supplying fluidic jets to augment the labyrinth seal at supply pressure above the pressure at seal inlet.

The extent to which PR can be elevated depends upon the available swirl kinetic energy in the seal inlet region. For a typical HP shrouded blade tip seal operating at the pressures given in table 1, a reasonable estimate for the swirl component of velocity at seal inlet would be approximately 200 m/s. This compares to a rotational shroud speed in the region of 150 m/s. Steam density in this region might typically be around 35 kg/m^3. These values correspond to a swirl velocity component dynamic head of 7.0 bar. If all of this swirl kinetic energy was recovered the value of PR for the fluid in the jet supply reservoir would be:

$$PR = \frac{p_{02} - p_3}{p_{01} - p_3} = \frac{107 - 95}{100 - 95} = 2.4 \qquad (2)$$

Under these conditions, less than one third of the available swirl kinetic energy would need to be recovered in order to achieve PR =1.4, which gave the 25% leakage reduction for the seal geometries that were described in the previous section.

The leakage reduction potential for seal applications under the LP flow conditions in Table 1 is even greater than for HP conditions. The low absolute value of pressure in this region means that much smaller increases in absolute reservoir pressure level above seal inlet conditions are required to generate high values of PR. The flow in the seal inlet region of an LP tip seal would have much lower density than for HP conditions due to the lower pressure levels, but flow velocities would be similar to those for HP conditions. As dynamic head depends on the square of velocity, the potential for creating increased values of reservoir pressures ratio, PR, through swirl kinetic energy recovery is even greater for LP conditions than it is for HP conditions.

Figures 13 and 14 show two 3-D schematics of the system described above applied to a shrouded moving blade tip seal and a fixed blade diaphragm gland seal respectively.

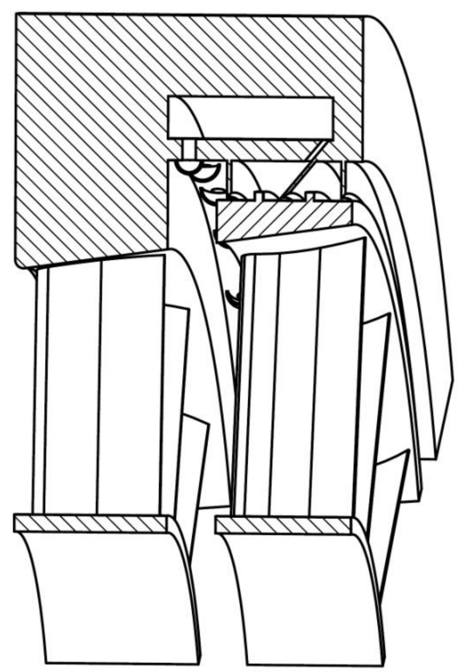

Figure 13: Example of a labyrinth seal on a Turbine Moving Blade Tip Seal Augmented by Fluidic Jets Supplied from a Reservoir with a Swirl Kinetic Energy Recovery System.

Figure 13 shows a turbine shrouded blade tip seal consisting of a 3 fin non-see-through labyrinth seal. Fluidic jets are applied on the upstream side of the inclined central fin. The jets are supplied from a reservoir. The ports that connect the reservoir with the tip seal inlet region have quarter spheres at their inlets. These inlet cups are orientated to recover swirl kinetic energy in the flow that enters the ports, resulting in the elevated reservoir pressures described previously.

Clearly there are many different types of geometry that could be applied at the inlets to the ports feeding the reservoir that would achieve some level of swirl kinetic energy recovery. One alternative is shown in Figure 14 applied to a diaphragm gland seal. In this design fluidic jets are applied on the upstream face of two inclined long fins within the labyrinth seal. The ports supplying the jet reservoir have tubes inserted in their inlets. The tubes are bent through 90^0 in the labyrinth seal inlet region to achieve swirl kinetic energy recovery in the flow entering the reservoir.

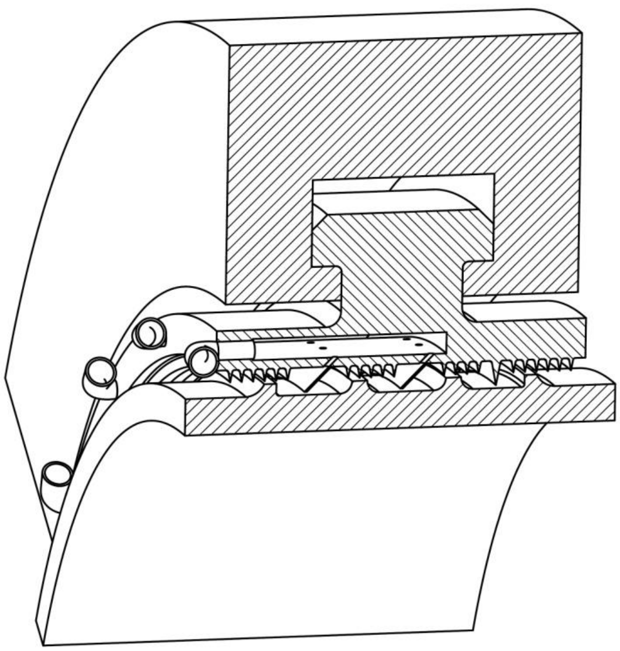

Figure 14: Example of a Diaphragm Gland Seal on a Turbine Shaft Augmented by Fluidic Jets Supplied from a Reservoir with a Swirl Kinetic Energy Recovery System.

The sealing system described in this section will have additional benefits in relation to improving rotordynamic stability. When swirling flow enters conventional labyrinth seals in turbomachinery applications it can produce de-stabilizing radial forces on the rotor, which can lead to rotordynamic stability problems. With conventional labyrinth seal designs, this is often countered by putting ‘swirl brakes’ (plane radially aligned plates to stop the swirl velocity) at the seal inlet to remove the swirl energy from the flow before it enters the seal. The new seal technology described in this paper will automatically perform the function of reducing swirl within the leakage flow to levels lower than what they would be for a conventional labyrinth seal without swirl brakes. Therefore, in addition to reducing leakage flows, the new technology is also expected to be beneficial for rotordynamic stability compared to conventional labyrinth designs for the same duty.

CONCLUSIONS

The performance of different types of fluidic seal design have been compared. ‘Air Curtain’ type seals formed purely by using cross-flow fluidic jets to create blockage against leakage flows were shown to have only limited potential for reducing leakage. Significantly greater potential exists for improving seal performance when fluidic jets are combined with labyrinth type seals. It has been shown that for an idealized 3 fin see-through labyrinth seal geometry, supplying fluidic jets at a pressure level increased above seal inlet pressure by an amount equal to 40% of the seal pressure drop, reduces leakage by approximately 25%. A new sealing technology has been proposed which uses labyrinth seals augmented by fluidic jets, where the jets are supplied with fluid from the seal inlet region at elevated pressure levels, by employing a passive system for recovering some of the swirl kinetic energy in the flow supplying the jets. The new technology is relatively low cost (compared to brush seals etc), easy to implement and is expected to bring benefits in leakage reduction and improved rotordynamic stability.

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REFERENCES

1. Ferguson, J.G., 1988, ‘Brushes as High Performance Gas Turbine Seals’, 33rd ASME Gas Turbine Conference, Amsterdam, Paper No. 88-GT-182.

2. Neef, M., Sulda, E.,Surken, N. & Walkenhorst, J., 2006, ‘Design Features and Performance Details of Brush Seals for Turbine Applications’, ASME Turbo Expo 2006, Barcelona, Paper No. GT2006-90404.

3. Jahn, I.H.J., Owen A.K., Franceschini G., Gillespie D.R.H.. 2008, ‘Experimental Characterisation of the Stiffness and Leakage of a Prototype Leaf Seal for Turbine Applications’, Proceedings of the ASME Turbo Expo 2008, Vol 4, Parts A & B, pp. 1657-1666.

4. Proctor, M.P., Kumar, A. & Delgado, I.R., 2002, ‘High-Speed, High-Temperature Finger Seal Test Results’, AIAA-2002-3793, NASA/TM-2002-211589.

5. Curtis, E.M., Denton, J.D., Longley, J.P., Rosic, B., 2009, 'Controlling Tip Leakage Flow over a Shrouded Turbine Rotor using an Air-Curtain', ASME Turbo Expo 2009, Orlando, Paper No. GT2009-59411.

6. Turnquist, N.A., Bruce, K.L., Cerretelli, C. & Tourigny, J.E., 2009, 'Fluidic Sealing for Turbomachinery', United States Patent US2009/0297341 A1, Assignee General Electric Company, USA.

7. Auyer, E.L., 1954, 'Dynamic Sealing Arrangements for Turbomachines', United States Patent 2685429, Assignee General Electric Company, USA.

8. Hogg, S.I., 2010, 'Sealing Device for Reducing Fluid Leakage in Turbine Apparatus', UK Patent Application No. 1017525.5, Assignee University of Durham.

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APPENDIX A:

VALIDATION OF 'AIR-CURTAIN' CFD CALCULATIONS WITH PERPENDICULAR JETS AGAINST AN ANALYTICAL MODEL.

An analytical model was written on an MS Excel spreadsheet to solve the steady compressible flow through the geometry shown in Figure 3. The model was based on equations for the conservation of mass (equation 3) energy (equation 4) and momentum (equation 5), applied to the jet and channel flows. The equation set was closed by standard equations for the isentropic flow of an ideal gas (air in this case) together with Blasius formula for wall shear stress to account for the effects of wall friction on the leakage channel flow.

$$m_3 = m_1 + m_2 \tag{3}$$

$$(m_1 \cdot h_{01}) + (m_2 \cdot h_{02}) = (m_3 \cdot h_{03}) \tag{4}$$

$$\begin{aligned}(p_1 \cdot A) - (p_3 \cdot A) - \left(f_1 \cdot \frac{1}{2} \cdot \rho_1 \cdot V_1{}^2 \cdot 2 \cdot (W+h) \cdot L_1\right) \\ - \left(f_3 \cdot \frac{1}{2} \cdot \rho_3 \cdot V_3{}^2 \cdot 2 \cdot (W+h) \cdot L_3\right) \\ = (m_3 \cdot V_3) - (m_1 \cdot V_1)\end{aligned} \tag{5}$$

In Figure 15, leakage flow predictions from the analytical model (open symbols) are compared with a sub-set of the CFD results shown previously in Figure 4 (closed symbols). The results from the analytical model with Blasius formula show good agreement with the CFD predictions. The total leakage flow rates (Outlet flow) predicted by the two methods differ in value by only a small percentage thus validating the CFD predictions.

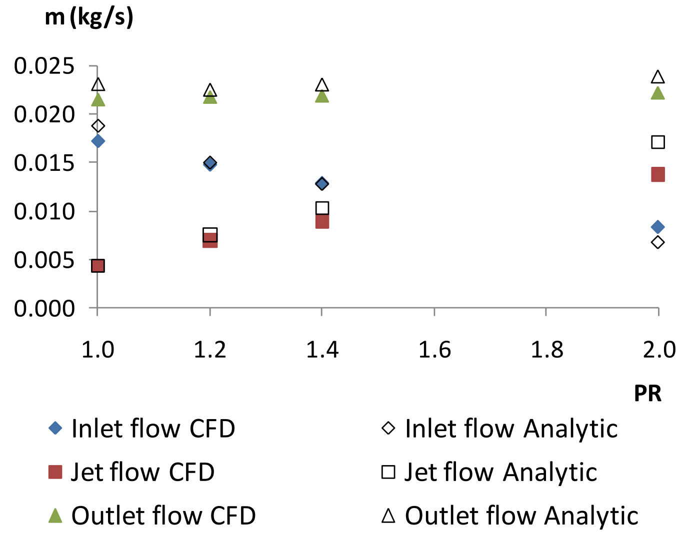

Figure 15: Comparison between Leakage Mass Flow Predictions from CFD and Analytical Model.