A SEAL ASSEMBLY

The present invention relates to a seal assembly for use in between two components between which there is relative rotational movement, particularly but not exclusively, a shrouded axial flow turbine rotatably mounted within a turbine housing in a turbomachine, such as an axial flow turbine expander of a waste heat recovery system.

An engine assembly typically comprises an internal combustion engine and may further comprise a turbocharger. A waste heat recovery system may be used to recover heat from the engine assembly and convert the recovered heat into usable power. The waste heat recovery system may be used to recover heat from engine exhaust gas and from an engine charge air cooler. Power derived from the waste heat recovery system may be used to generate electricity and/or to augment power output from the internal combustion engine.

A conventional waste heat recovery system uses a refrigerant fluid which is pumped around a closed loop. A heat exchanger is used to transfer heat from the engine (e.g. from exhaust gas) to the refrigerant, which is initially in liquid form. This heat causes the refrigerant liquid to vaporise. The refrigerant vapour passes to an expansion turbine (hereinafter referred to as a “turbine expander”) and drives a turbine wheel or rotor of the turbine expander to rotate. Power is derived from the rotation of the turbine rotor. The refrigerant vapour passes from the turbine expander to a condenser which is configured to cool and condense the refrigerant so that it returns to liquid form. The refrigerant liquid is then passed to the heat exchanger to complete the loop.

The turbine rotor of the waste heat recovery system is rotatably mounted on a turbine shaft within a housing. In some cases the turbine shaft may be provided with a gear or other power transfer coupling to augment the output of an internal combustion engine to which the waste heat recovery system is connected. In some cases the turbine shaft may be directly or indirectly connected to a further shaft, upon which may be mounted the gear or power transfer coupling.

The turbine rotor typically incorporates an array of blades which extend radially outwards from a central hub which is mounted on the turbine shaft. The radially outer edge of each blade is often connected to a circumferentially extending ring or shroud; as a result, a turbine rotor of this kind is typically referred to as a “shrouded turbine”. In a shrouded turbine, the radially outer circumferentially extending surface of the shroud rotates at high speed adjacent to a wall of the housing, which is of course stationary. The shroud and the housing are designed so that the shroud rotates in close proximity to the housing wall to minimise the possibility of fluid being lost through the radial clearance between the shroud and the housing rather than flowing through the blades of the turbine as intended. It is common practice to provide a conventional seal, such as a labyrinth seal, in the space between the shroud of the turbine and the housing wall to minimise fluid leakage flow around the turbine wheel.

Much of the recent work aimed at improving performance beyond the level of conventional labyrinth seals has focused on designs that employ adaptable physical barriers to reduce leakage e.g. brush, leaf and finger seals etc. Improved sealing arrangements have also been developed that are based upon gas-curtain type fluidic jets. The jets create a blockage in the form of a cross-flow fluid ‘curtain’ which establishes a static pressure drop within the leakage path across the curtain. This pressure drop acts to effectively reduce the pressure gradient in the inlet region of the seal, thereby reducing the amount of leakage flow entering the seal. Despite recent advances in mechanical and aerodynamic sealing arrangements, a significant need still exists for a simple, robust, yet effective means of providing a seal between a pair of components between which there is relative rotation, such as a shrouded axial flow turbine and a turbine housing.

It is an object of the present invention to obviate or mitigate one or more problems associated with existing seals and/or axial flow turbines. Moreover, it is an object of the present invention to provide an improved or alternative seal assembly and/or an improved or alternative axial flow turbine.

According to a first aspect of the present invention there is provided an axial flow turbine comprising

a turbine rotor mounted within a housing for rotation about a turbine axis,

a fluid flow inlet passage upstream of said turbine rotor arranged to direct a first fluid towards the turbine rotor in a substantially axial direction,

a fluid flow outlet passage downstream of said turbine rotor and

a seal assembly provided in a fluid leakage cavity defined between the turbine rotor and the housing, wherein the seal assembly comprises

a fluid jet outlet configured to admit a second fluid into the fluid leakage cavity in an upstream direction which is inclined to the turbine axis,

a first flow restriction located downstream of the fluid jet outlet and at a location such that second fluid admitted from the fluid jet outlet would impinge on the first flow restriction once the second fluid has turned to flow in an axial direction, and

a second flow restriction located downstream of the first flow restriction.

Detailed investigations into the flow characteristics of fluid jet seals has led to the realisation that the kinetic energy of the fluid jet can, unless carefully controlled, negatively influence the sealing capability of the seal. It has surprisingly been found that combining a fluid jet with at least two downstream flow restrictions, the most upstream of which is positioned at a specific, predetermined position relative to the fluid jet, results in a sealing arrangement that exhibits significantly improved performance as compared to conventional sealing arrangements including aerodynamic and/or physical seals.

The fluid jet outlet is preferably configured to admit the second fluid into the fluid leakage cavity in an upstream direction which is inclined to the turbine axis so as to partially oppose the flow of the first fluid through the fluid leakage cavity. That is, in a direction transverse to the turbine axis such that, upon ejection from the outlet, a component of the flow of the second fluid initially opposes the flow of the first fluid through the fluid leakage cavity, which will be in a generally axial direction, disregarding any swirl, and a component of the flow of the second fluid is initially perpendicular to the flow of the first fluid through the fluid leakage cavity. The second fluid may be ejected so as to be initially admitted into the fluid leakage cavity in an upstream direction having an axial component and a radial component. The direction in which the second fluid initially enters the fluid leakage cavity may be at an angle of around 20 to 70 ° to the turbine axis, 30 to 60 ° to the turbine axis, 40 to 50 ° to the turbine axis or around 45 ° to the turbine axis.

The fluid jet outlet may be defined by a wall of the housing which lies radially outboard of the turbine rotor. The sealing assembly may comprise one, two, three or more fluid jet outlets. Where a plurality of fluid jet outlets are provided, they may be arranged in a regular or irregular array. For example, a plurality of fluid jet outlets may be provided in a regular circumferential array surrounding the radially outer periphery of the turbine rotor. By way of a further alternative, the fluid jet outlet may be provided in the form of one or more continuous or discontinuous channels extending circumferentially around the wall of the housing which faces into the fluid leakage cavity. It will be appreciated that in order for the second fluid to be able to be admitted into the fluid leakage cavity against the flow of the first fluid through the fluid leakage cavity so that it can function acceptably as an element of the seal assembly, the pressure of the second fluid exiting the fluid jet outlet must be greater than the pressure of the first fluid passing the fluid jet outlet. The fluid jet outlet may be connected to a conduit extending through the housing to a source of pressurised fluid, that is, a fluid with a pressure greater than the expected pressure of the first fluid as it flows along the fluid leakage cavity passed the fluid jet outlet. If the pressure of the second fluid leaving the fluid jet outlet is too low the second fluid will not traverse a sufficient distance across the radial width of the fluid leakage cavity to adequately disrupt the leakage flow of the first fluid. The pressure of the second fluid preferably exceeds a threshold required to cause the second fluid to cross at least around half of the radial width of the fluid leakage cavity before its momentum has been fully turned into the direction of the first fluid flowing along the fluid leakage cavity.

The turbine rotor is preferably provided with a cylindrical ring or ‘shroud’ which extends circumferentially around the radially outer periphery of the turbine rotor. Hereinafter, such a turbine rotor will be described as a ‘shrouded turbine rotor’.

An axial section of the fluid leakage cavity may be a radial clearance defined between the radially outer periphery of the turbine rotor, e.g. the shroud of a shrouded turbine rotor, and a circumferentially extending section of an inner wall of the housing which lies radially outboard of the radially outer periphery of the turbine rotor.

A flow-controlling element, such as a stator supporting one or more vanes, may be provided in between the fluid flow inlet passage and the turbine rotor to pre-condition or pre-swirl, the first fluid before it impinges on the turbine rotor to optimise turbine efficiency.

The first flow restriction is preferably located downstream of the fluid jet outlet and at a location such that, in use, second fluid admitted from the fluid jet outlet would impinge on the first flow restriction once the second fluid has turned to flow in an axial direction under the influence of flow of the first fluid through the fluid leakage cavity. The first flow restriction is positioned at this location to disrupt the axial flow of the second fluid through the fluid leakage cavity and thereby prevent the kinetic energy of the second fluid from undesirably influencing the leakage flow of the first fluid through the fluid leakage cavity. In the absence of the first flow restriction the kinetic energy of the second fluid admitted by the fluid jet would accelerate the first fluid passing along the fluid leakage cavity beyond the second flow restriction located further downstream, thereby negatively affecting the performance of the sealing assembly as a whole.

The first flow restriction may be spaced from the centre of the fluid jet outlet by a distance that is approximately one half to twice the radial width of the fluid leakage cavity. That is, where the spacing of the first flow restriction to the centre of the fluid jet outlet is x, and the radial width of the fluid leakage cavity is y, x may be equal to or greater than 0.5 y or it may be equal to or less than 2 y. Furthermore, the relationship between x and y may be closer to parity, for example, $0.75y \leq x \leq 1.5y$, or the first flow restriction may be spaced from the centre of the fluid jet outlet by approximately the same distance as the separation between the section of the wall of the housing which radially overlies the turbine rotor (the radial width of the fluid leakage cavity), i.e. $x \approx y$.

The first flow restriction may be located on an opposite side of the fluid leakage cavity to the side from which the second fluid is admitted into the fluid leakage cavity via the fluid jet outlets. The first flow restriction may be provided on the radially inboard side of the fluid leakage cavity while the fluid jet outlet may be provided on the radially outboard side of the fluid leakage cavity. For example, the first flow restriction may be a projection extending radially outwards from the radially outer periphery of the turbine rotor or the shroud of a shrouded turbine rotor.

A single first flow restriction may be provided, or a plurality of flow restrictions may be provided. It may be desirable to provide a first flow restriction in register with each fluid jet outlet. Alternatively, it may be desirable to associate two or more first flow restrictions of similar or dissimilar size and/or shape with each fluid jet outlet. As a further alternative, the first flow restriction may be provided as a continuous or discontinuous rim extending radially outwardly around the circumference of the radially outer periphery of the turbine rotor or the shroud of a shrouded turbine rotor. It will also be appreciated that any of the aforementioned alternative configurations may be used in any desirable combination.

The first flow restriction may extend across any desirable amount of the radial width of the fluid leakage cavity provided it can effect its intended function of causing second fluid admitted from the fluid jet to impinge upon its surface once the second fluid has turned to flow in an axial direction. The first flow restriction may extend across up to around 90 % of the radial width of the fluid leakage cavity, up to around 70 %, up to around 50 %, or up to around 30 % of the radial width of the fluid leakage cavity. The first flow restriction may extend across at least 5 % of the radial width of the fluid leakage cavity, at least 15 % or at least 25 % of the radial width of the fluid leakage cavity.

The or each first flow restriction can be of any desirable shape, again, provided it can perform its intended function. It may have a regular axial cross-sectional profile, or it may be irregular. It may incorporate surfaces which extend axially and/or radially, and/or which are inclined to the axial or radial directions. Any non-axial or non-radial surfaces may be linearly inclined, non-linearly inclined, concave or convex. The first flow restriction may have any desirable axial extent; it may extend across 5 % or less of the axial length of the fluid leakage cavity, or it may be larger and extend across 10 to 20 % of the axial length of the fluid leakage cavity. Alternatively, the first flow restriction may be defined by a section of the turbine rotor or turbine rotor shroud downstream of the fluid jet outlet at a location to ensure that the second fluid, when ejected from the fluid jet outlet, impinges on that section of the turbine rotor or turbine rotor shroud before it has turned completely to flow in the axial direction. In an embodiment described below with reference to Figure 6, the first flow restriction is in the region of a downstream corner of the turbine rotor shroud at the connection of the axially extending region of the fluid leakage cavity and the radially extending outlet region of the fluid leakage cavity. This region around the corner serves to prevent the second fluid being ejected from the fluid jet outlet in such a way that the kinetic energy of the second fluid carries it into the outlet region of the fluid leakage cavity without the second fluid having been turned to flow axially by interaction with the first fluid leakage flow. As discussed in more detail in the Examples, the spacing of the fluid jet outlet from the corner of the shroud to avoid this undesirable effect is a function of the pressure differential between the first and second fluids, the axial width of the conduit through which the second fluid passes to the fluid jet outlet and the angle to the turbine axis at which the second fluid is initially directed into the oncoming first fluid leakage flow.

The second flow restriction downstream of the first flow restriction may take any convenient form. It may be similar in one or more respects, e.g. size, shape, radial extent, axial extent, etc, to the form of the first flow restriction or it may be different in all respects. The second flow restriction may be a conventional aerodynamic or physical seal, such as a radially extending fin, a labyrinth seal, a brush seal, a leaf seal, an abradable seal, etc. Alternatively, the second flow restriction may be defined by a section of the wall of the housing downstream of the first flow restriction which lies radially and/or axially closer to the turbine rotor than the section of the wall of the housing which defines the fluid jet outlet. In an embodiment described below with reference to Figure 6, the second flow restriction is an outlet region of the fluid leakage cavity defined as a relatively tight axial clearance between a radially extending section of the wall of the housing and a radial surface of the downstream end of the turbine shroud.

While the second flow restriction may take any convenient form as described above, in a preferred embodiment, it extends radially inwards from the same side of the fluid leakage cavity from which the second fluid is admitted via the fluid jet outlet. Moreover, in an embodiment where the first flow restriction extends radially outwards it is preferred that the second flow restriction extends radially inwards from the opposite side of the fluid leakage cavity from which the first flow restriction extends. Taking the embodiments described below with reference to Figures 4 and 5, the fluid jet outlet and second flow restriction may be associated with the housing wall which lies radially outboard of the turbine shroud with which the first flow restriction is associated.

In order for the second flow restriction to be considered as being ‘downstream’ of the first flow restriction, it will be appreciated that all that is required is for the most upstream feature of the second flow restriction to be downstream of the most upstream feature of the first flow restriction. That is, the first and second flow restrictions can radially overlie one another and the second flow restriction still be considered as being downstream of the first flow restriction provided the first flow restriction extends axially further upstream than the second flow restriction. A non-limiting, exemplary embodiment of such an arrangement is described below with reference to Figure 5.

A second aspect of the present invention provides a method for sealing a fluid leakage cavity in an axial flow turbine, the turbine comprising

a turbine rotor mounted within a housing for rotation about a turbine axis,

a fluid flow inlet passage upstream of said turbine rotor arranged to direct a first fluid towards the turbine rotor in a substantially axial direction,

a fluid flow outlet passage downstream of said turbine rotor and

a seal assembly provided in said fluid leakage cavity defined between the turbine rotor and the housing,

wherein the method comprises

directing the first fluid towards the turbine rotor in a substantially axial direction, a portion of the first fluid flowing through the turbine rotor and a further portion flowing through the fluid leakage cavity

admitting a second fluid from a fluid jet outlet into the fluid leakage cavity in an upstream direction which is inclined to the turbine axis,

providing a first flow restriction downstream of the fluid jet outlet and at a location such that second fluid admitted from the fluid jet outlet impinges on the first flow restriction once the second fluid has turned to flow in an axial direction after contacting the further portion of the first fluid flowing through the fluid leakage cavity, and

providing a second flow restriction downstream of the first flow restriction to restrict the flow of a mixture of said first and second fluids further through the fluid leakage cavity.

According to a third aspect of the present invention there is provided a turbomachine having an axial flow turbine according to the first aspect of the present invention as defined above. The turbomachine may be an axial flow turbine expander of a waste heat recovery system.

A fourth aspect of the present invention provides a waste heat recovery system comprising an axial flow turbine according to the first aspect of the present invention.

A fifth aspect of the present invention provides a seal assembly for restricting fluid leakage flow through a fluid leakage cavity defined between a first component and second component that is rotatable about an axis relative to the first component, the seal assembly comprising

a fluid jet outlet configured to admit a fluid jet into the fluid leakage cavity in an upstream direction which is inclined to the axis,

a first flow restriction located downstream of the fluid jet outlet and at a location such that the fluid jet admitted from the fluid jet outlet would impinge on the first flow restriction once the fluid jet has turned to flow in an axial direction, and

a second flow restriction located downstream of the first flow restriction.

It will be appreciated that the seal assembly of the fifth aspect of the present invention shares the same basic features of the seal assembly of the first aspect of the present invention described above, however it is not limited to use in any particular machine. Accordingly, preferred and optional features of the seal assembly of the first aspect of the present invention may also be applied in any technically compatible combination to the seal assembly according to the fifth aspect of the present invention. Moreover, it will be appreicated that the seal assembly of the fifth aspect of the present invention and the method according to the sixth aspect of the present invention defined below may be employed in any machine where a seal is required between two components between which there is relative rotational movement, examples of which include turbines of many different types and geometries, such as gas turbines and steam turbines.

A sixth aspect of the present invention provides a method for sealing a fluid leakage cavity defined between a first component and second component that is rotatable about an axis relative to the first component, the method comprising

directing a first fluid towards the second component in a substantially axial direction, a portion of the first fluid flowing through the second component and a further portion flowing through the fluid leakage cavity

admitting a fluid jet into the fluid leakage cavity in an upstream direction which is inclined to the axis,

providing a first flow restriction downstream of the fluid jet and at a location such that the fluid jet impinges on the first flow restriction once the fluid jet has turned to flow in an axial direction after contacting the further portion of the first fluid flowing through the fluid leakage cavity, and

providing a second flow restriction downstream of the first flow restriction to restrict the flow of a mixture of said first fluid and said jet fluid further through the fluid leakage cavity.

Specific embodiments of the present invention will now be described, by way of example only, with reference to the accompanying drawings in which:

Figure 1 is a schematic sectioned view of a prior art single stage axial flow turbine incorporating a shrouded turbine wheel (bearing system omitted for clarity);

Figure 2 is a detailed view of a section of Figure 1 focusing on an upper region of a labyrinth seal adjacent to the shroud of the turbine wheel;

Figure 3 is a schematic representation of an alternative prior art sealing arrangement for a shrouded turbine wheel in which a fluidic-jet replaces the labyrinth seal of the Figure 2 arrangement;

Figure 4 is a schematic representation of a first embodiment of a sealing arrangement for an axial flow turbine incorporating a shrouded turbine wheel according to the present invention;

Figure 5 is a schematic representation of a second embodiment of a sealing arrangement for an axial flow turbine incorporating a shrouded turbine wheel according to the present invention;

Figure 6 is a schematic representation of a third embodiment of a sealing arrangement for an axial flow turbine incorporating a shrouded turbine wheel according to the present invention;

Figures 7a and 7b are respectively, a CFD Mesh, and a pair of contour plots of the flow for a typical fluidic seal CFD calculation;

Figure 8 is a schematic illustration of the fluidic seal design optimisation parameters;

Figure 9 is a graph of CFD leakage flow predictions for a fluidic jet with a predetermined configuration;

Figure 10 is a graph of net power gain for a fluidic jet having the same configuration as that which provided the results presented in Figure 9;

Figure 11 is a graph of net power gain from the fluidic seal as a percentage of turbine power for a fluidic jet having the same configuration as that which provided the results presented in Figure 9;

Figure 12 is a graph of the effect of jet angle, *A*, on power output for a fluidic jet of narrower width, *W*, than that which was employed to obtain the results presented in Figure 9; and

Figure 13 is a graph of the effect of distance from corner, *D*, on power output for a fluidic jet with the same configuration as that which provided the results presented in Figure 12.

Figure 1 illustrates a prior art a single stage axial flow turbine 1 suitable for use in a waste heat recovery system of an internal combustion engine. High pressure fluid, such as refrigerant vapour generated by thermal contact between liquid refrigerant and waste heat recovered from the engine, is applied to an inlet scroll 2 defined by a housing 3 of the turbine 1. The path of the vapour is shown in Figure 1 using three arrows. The vapour is turned axially in the inlet scroll 2 before it expands across a stator 4, which in this specific embodiment consists of twelve vanes arranged to introduce a swirl angle of 80° into the vapour. In this embodiment, the turbine 1 is designed to operate with a very high pressure ratio (> 5.0) and so the flow through the stator 4 is choked. It will be appreciated that any number and arrangement of vanes, and/or choked or unchoked flow conditions, may be used to produce any desired swirl angle, or indeed the stator 4 may be omitted altogether if appropriate for a particular application. As shown in Figure 1, the flow of swirling vapour exiting the stator 4 then passes to a turbine wheel or rotor 5. In this embodiment, the rotor 5 incorporates thirty-four blades, although any desirable number or arrangement may be used. A shroud 6 extends circumferentially around a radially outer periphery of the blades of the rotor 5.

Exemplary, non-limiting, parameters for a typical axial flow turbine of the kind shown in Figure 1 are set out below in Table 1.

Rotor Diameter	63 mm
Nominal Operating Speed	50,000 rpm
Rotor Blade Height	4.125 mm
Shroud Axial Length	8.0 mm

Table 1: Turbine Design Parameters

The shroud 6 and the housing 3 are designed so that the shroud 6 rotates in close proximity to a wall 7 of the housing 3 which radially overlies the shroud 6 to minimise the possibility of vapour being lost through a radial clearance between the shroud 6 and the housing wall 7 rather than flowing through the rotor 5. In the embodiment shown in Figure 1 a conventional labyrinth seal 8 is provided in the radial clearance between the shroud 6 and the housing wall 7 to minimise vapour leakage flow around the rotor 5 via the radial clearance and therefore maximise the efficiency of the turbine 1.

A more detailed view of the labyrinth seal 8 is shown in Figure 2. Vapour leaking around the rotor 5 does so with the high levels of swirl momentum that are induced in the vapour by the stator 4. Arrow 9 in Figure 2 represents vapour leaking into the radial clearance between the shroud 6 and the housing wall 7, while arrow 10 represents vapour exiting the radial clearance. The tangential velocity component of this swirling vapour leakage flow is typically greater than the rotational speed of the shroud 6.

After passing around the rotor 5, the leakage flow re-enters and mixes with the main flow of vapour immediately downstream of the rotor 5, before being exhausted from the turbine 1 in the axial direction.

The particular design of labyrinth seal 8 shown in Figures 1 and 2 features three radially extending labyrinth restrictions 8a, 8b, 8c, although any number or design of restrictions may be used. In addition to the radial restrictions 8a, 8b, 8c, the sealing arrangement depicted in Figures 1 and 2 also comprises a first relatively tight axial clearance between an upstream radial surface 11 of the shroud 6 and a wall 12 of the housing 3 which defines an inlet to the radial clearance between the shroud 6 and the housing wall 7, and a second relatively tight axial clearance between a downstream radial surface 13 of the shroud 6 and a wall 14 of the housing 3 which defines an outlet to the radial clearance. The radial clearance and two axial clearances together define a fluid leakage cavity. The sealing arrangement serving to restrict the undesirable flow of vapour around the rotor 5 through the fluid leakage cavity is therefore made up of the three radial restrictions 8a, 8b, 8c, of the labyrinth seal 8 together with the tight axial clearances at the inlet and outlet of the radial clearance. It is known that density effects mean that the greatest pressure drop within the fluid leakage cavity is carried by the last or most-downstream restriction in any multi-labyrinth seal system and so, in the arrangement shown in Figures 1 and 2, it is the tight axial clearance at the outlet of the fluid leakage cavity that exerts the greatest influence on the leakage flow.

Figure 3 illustrates a fluidic jet aerodynamic sealing arrangement applied to a turbine expander of the same general type discussed above in relation to Figures 1 and 2. Figure 3 is a similar close up view of the sealing arrangement as shown in Figure 2 but in which the labyrinth seal 8 of Figure 2 has been replaced with a fluidic jet seal as will now be described. In Figure 3, the shroud 6 again rotates in close proximity to the wall 7 of the housing 3 but now the flow of vapour through the radial clearance defined between the shroud 6 and the wall 7 is hindered by a pressure curtain defined by the ingress of a high pressure fluid jet which is admitted into the leakage flow path in the direction of arrows 15 and 16 from a jet conduit 17 via one or more orifices 18 defined in the wall 7 which surrounds the shroud 6. Arrow 16 illustrates how the fluidic jet is turned and mixes with the vapour leakage flow passing along the fluid leakage cavity in the direction of arrows 19 and 20 from an inlet 21 to an outlet 22.

The jet enters the fluid leakage cavity about half way along the axial length of the shroud 6 in a direction which opposes the leakage flow. In the embodiment shown in Figure 3 the jet conduit 17 is orientated at an angle of around 45° to the turbine axis, which is typical in fluidic jet seals. As the incoming high pressure jet mixes with the prevailing flow of vapour along the fluid leakage cavity, a static pressure drop is required to turn the jet flow to the same direction as that of the vapour leakage flow. This causes a sudden drop in static pressure across the fluid leakage cavity in the region where the jet flow enters the fluid leakage cavity via the orifice(s) 18. The presence of a fluidic jet acts to increase static pressure in the fluid leakage cavity upstream of the jet orifice(s) 18 and to reduce it downstream of the orifice(s) 18 compared to a turbine which incorporates neither the labyrinth seal of Figures 1 and 2, nor the fluidic jet seal of Figure 3. The higher pressure upstream of the fluidic jet reduces the pressure gradient in the inlet 21 to the fluid leakage cavity, and so less vapour leakage flow enters the inlet 21 leaving a greater proportion of the vapour to flow through the rotor 5 as intended.

As mentioned above, the presence of the fluid jet acts to increase the pressure in the fluid leakage cavity upstream of the jet and decrease it downstream of the jet, compared to the ‘no-jet’ case. This will reduce the pressure gradient in the region of the outlet 22 of the fluid leakage cavity and so the leakage flow exiting from the fluid leakage cavity must be reduced compared to the equivalent no-jet case. As long as the fluidic jet seal is not ‘over-blown’ (discussed in more detail below), the flow exiting from the outlet 22 of the fluid leakage cavity must be the sum of the flow into the fluid leakage cavity inlet 21 plus the fluidic jet flow. So, there must always be a net overall leakage reduction benefit for any jet flow up to the point that the fluidic jet seal becomes over-blown.

If the fluid jet supply pressure is too low then the fluid jet will not traverse a sufficient distance across the radial width of the vapour leakage flow between the shroud 6 and the wall 7 of the housing 3 to create the ‘fluid curtain’ required to provide effective sealing. To provide an effective seal, it is generally desirable for the fluid jet supply pressure to exceed a threshold required to cause the fluid jet to traverse at least around half of the radial width of the radial clearance between the shroud 6 and the housing wall 7 (denoted by the dotted line in Figure 3) before its momentum has been fully turned into the direction of vapour flowing along the fluid leakage cavity.

If the fluid jet supply pressure is too high it is possible to reverse the vapour leakage flow. Under these conditions, the jet fluid entering the clearance between the shroud 6 and wall 7 of the housing 3 splits, with a proportion of it flowing back upstream along the fluid leakage cavity, exiting the fluid leakage cavity through its inlet 21 into the inlet flow of the rotor 5. Operation of an aerodynamic seal under these conditions is described as being ‘over-blown’. As would be appreciated to the skilled person, operation of a fluidic jet seal under over-blown conditions has a negative impact on turbine performance since the jet fluid that is being forced back out through the inlet of the fluid leakage cavity mixes with the vapour immediately upstream of the rotor 5.

Consequently, the over-blown condition represents a performance threshold for aerodynamic seals of the kind shown in Figure 3, akin to stall of an aerofoil.

In general in fluidic jet seals, as the inlet pressure, and hence the mass flow of the jet fluid, is increased, the vapour leakage mass flow through the inlet to the leakage flow path will be reduced. This situation continues until the momentum of the fluidic jet is just sufficient to prevent any flow from entering the leakage flow path via its inlet. Any further increase in jet supply pressure will result in the seal becoming over-blown. It is therefore conventional to design fluidic seals with jet conditions that allow some margin before the seal becomes over-blown, akin to the to stall margin applied during the design of aerofoils.

Referring now to Figure 4, there is shown a first embodiment of a sealing arrangement according to the present invention which can be used in place of the labyrinth seal 8 of Figures 1 and 2, and the fluidic jet of Figure 3. The shroud 6 and wall 7 of the housing 3 have the same general arrangement as shown in Figures 2 and 3 with the prevailing direction of leakage fluid flow being in the direction of arrow 23. The sealing arrangement shown in Figure 4 comprises a fluidic jet 24 disposed upstream of a labyrinth seal 25, which includes one radially extending leakage flow restriction (shown in solid lines), but which may, if desired, include multiple radially extending leakage flow restrictions (shown in dotted lines). The sealing arrangement shown in Figure 4 also incorporates an intermediate flow restriction 26 in between the fluid jet 24 and the labyrinth seal 25. The intermediate flow restriction 26 is disposed on the opposite side 27 of the fluid leakage cavity to the side 28 upon which the fluid jet enters the fluid leakage cavity via one or more orifices 29 in the housing wall 7. While the fluid jet may be admitted via one or more discrete orifices 29, it will be appreciated that the outlet for the fluid jet may take any desirable form and may, for example, be a continuous or discontinuous channel extending circumferentially around the wall 7 of the housing 3 which faces into the fluid leakage cavity.

In this specific embodiment, the intermediate flow restriction 26 is connected to the radially outer periphery of the shroud 6. The intermediate flow restriction 26 may be formed as a separate component to the shroud 6 and then connected to the shroud using any desirable means of connection, such as brazing, or the intermediate flow restriction 26 may be formed integerally with the shroud 6 such that the intermediate flow restriction 26 and the shroud 6 are produced as a single, unitary structure. The distance over which the intermediate flow restriction 26 extends radially from side 27 of the fluid leakage cavity is selected to ensure that it presents a restriction to the high pressure jet fluid as it turns to flow in the same direction as the prevailing vapour leakage flow. In this way, the intermediate flow restriction 26 prevents the kinetic energy of the jet fluid from undesirably influencing the vapour leakage flow encountering the downstream labyrinth seal 25 at arrow 30. In the absence of the intermediate flow restriction 26, even if the fluidic jet was operated at a jet pressure beyond the theoretical threshold required to provide an effective seal, as described above, the kinetic energy of the fluidic jet would accelerate vapour leakage flow passed the downstream labyrinth seal 25, thereby negatively affecting the performance of the sealing arrangement as a whole. The skilled person may previously have been of the opinion that supplementing a conventional mechanical seal, such as a labyrinth seal, with an aerodynamic seal, such as fluidic jet, to provide what might have been considered a ‘blown mechanical seal’, would provide an enhanced sealing arrangement as compared to either type of seal in isolation. However, the devisors of the present invention have determined that the opposite is in fact true; simply augmenting a mechanical seal with an aerodynamic seal, or *vice versa*, results in a more complicated and costly sealing arrangement with reduced performance benefit, and may in fact result in a less effective sealing arrangement than could be achieved using either type of seal separately.

In the specific embodiment shown in Figure 4, a particularly preferred arrangement is depicted in which the intermediate flow restriction 26 is located so that its most upstream feature is provided a distance x upstream of the centre of the orifice(s) 29, which approximately matches the radial width y of the radial clearance between the shroud 6 and the housing wall 7. It is generally preferred that the most upstream feature of the intermediate flow restriction 26 is located upstream of the centre of the orifice(s) 29 by a distance which is greater than or equal to around one half of the radial width of the radial clearance between the shroud 6 and the housing wall 7and less than or equal to around double the radial width of the radial clearance between the shroud 6 and the housing wall 7, i.e. $0.5y \leq x \leq 2y$.

The flow restriction in between the fluidic jet and the downstream flow restriction, e.g. the intermediate flow restriction 26 of the Figure 4 embodiment, may take any convenient size and shape provided it can function to prevent the kinetic energy of the jet fluid from undesirably influencing the vapour leakage flow encountering the downstream flow restriction. While the embodiment of the intermediate flow restriction shown in Figure 4 has a simple rectangular cross-section, it may include one or more ramped and/or arcuate surfaces facing upstream, downstream, or both upstream and downstream. The ramped or arcuate sections may be arranged to direct incident fluid radially outwards if the downstream flow restriction (e.g. labyrinth seal 25) extends radially inwards from the wall 7 of the housing 3 as shown in the Figure 4 embodiment. If the downstream flow restriction takes some other form, then the form of the intermediate flow restriction can be adapted to ensure that it achieves the optimum level of 'pre-conditioning' of the vapour leakage flow after it has passed the fluidic jet and before it impinges on the downstream flow restriction to achieve the desired degree of sealing.

Figure 5 illustrates an alternative embodiment of the sealing arrangement of the present invention to that shown in Figure 4, in which the downstream flow restriction is located more upstream than in Figure 4. In Figure 5, the shroud 6 and the wall 7 of the housing 3 remain unchanged from the arrangements shown in Figures 1 to 4. Components similar to those illustrated in Figure 4 take the same reference numbers in Figure 5 save for being increased by 100.

In Figure 5, the general arrangement of the fluid jet 124 and the intermediate flow restriction 126 is the same as in Figure 4, the axial spacing *x'* of the intermediate flow restriction 126 from the centre of the fluid jet orifice(s) 129 again being between around a half and double the radial width *y'* of the radial clearance between the shroud 6 and the housing wall 7. In Figure 5, however, the downstream flow restriction 125 is now composed of a first radially extending flow restriction 125a which axially overlies the intermediate flow restriction 126, and optionally, one or more second radially extending flow restrictions 125b, which may define, for example, a conventional labyrinth seal. Despite the first component of the downstream flow restriction 125a now being located more upstream than the corresponding component 25 in Figure 4, it will be appreciated that the first component of the downstream flow restriction 125a is still 'downstream' of the most upstream feature or edge of the intermediate flow restriction 126. In this way, the intermediate flow restriction 126 still pre-conditions the vapour leakage flow before it impinges upon the most upstream of the downstream flow restrictions 125a, 125b at arrow 130.

Figure 6 illustrates a further embodiment of a sealing arrangement according to the present invention. The general form of the sealing arrangement shown in Figure 6 is similar in some respects to the conventional fluidic jet seal described above with reference to Figure 3, however, certain key parameters of the design of the fluidic jet seal shown in Figure 6 have been optimised as a result of the realisation of the factors which influenced the design of the embodiments of the sealing arrangement described above in relation to Figures 4 and 5. That is, an appreciation for the first time of the importance of controlling the potentially negative influence of the kinetic energy of the fluid jet on the effectiveness of a seal incorporating an aerodynamic feature has enabled the development of a more effective fluidic jet seal.

In the sealing arrangement of Figure 6, the layout of the shroud 6 and the wall 7 of the housing 3 again remain unchanged from earlier embodiments. Components similar to those illustrated in Figures 3 and 4 take the same reference numbers in Figure 6 save for being increased by 200. A fundamental difference in the seal shown in Figure 6 compared to the conventional fluidic jet seal shown in Figure 3 is the optimisation of the axial displacement *D* of the downstream edge of the or each fluid jet orifice 229 from the radially extending wall 231 of the housing 3 that defines the downstream end of the radial clearance between the shroud 6 and the housing wall 7. Optimisation of the displacement *D* has taken account of three other seal parameters; the fluid jet pressure ratio *PR* (the ratio of the total pressure of the fluid supplying the jet relative to the rotor exit static pressure, to the total to static pressure drop across the rotor), the axial width *W* of the fluid jet conduit 217, and the angle of inclination *A* of the fluid jet conduit 217 to the turbine axis. The parameters *D*, *A* and *W* were optimised to achieve the best overall performance of the fluidic jet seal.

The vapour leakage flow entering the fluid leakage cavity is swirling strongly having just passed through the stator 4. It will be apparent that the swirl velocity must be greater than the rotational speed of the shroud 6 otherwise the turbine 1 would be operating below the zero work-line. The shear forces on the shroud 6 will act to decrease the swirl in the vapour leakage flow to approximately half the speed of the rotor 5 since the fluid leakage cavity has one rotating wall (the shroud 6) and one stationary wall (the wall 7 of the housing 3) of similar scales. In order to achieve this, the swirl kinetic energy that must be removed from the vapour leakage flow is transferred to mechanical energy in the rotor 5, through the shear forces acting on the rotating shroud 6. Thus, the vapour leakage flow contributes some positive energy into the rotor 5 via this mechanism, although the overall total effect of the vapour leakage flow must, of course, be to increase losses and reduce the output power of the rotor 5. The presence of a fluidic jet seal reduces or reverses the positive effect of the shroud seal forces. Firstly, the fluidic jet reduces the vapour leakage inlet flow and so less swirl momentum is carried into the vapour leakage path. Secondly, unless the fluidic jet itself is pre-swirled, the jet fluid enters the vapour leakage path with no swirl. The jet fluid therefore has to gain swirl momentum as it mixes with the vapour leakage flow, removing energy from the rotor 5. This latter effect can be minimised by positioning the fluid jet orifice(s) towards the downstream end of the radial clearance between the shroud 6 and the housing wall 7.

Detailed investigations into *D*, *PR*, *W*, *A* and the effect of shroud shear have concluded that it is advantageous to position the fluid jet orifice(s) 229 towards the downstream end of the radial clearance between the shroud 6 and the housing wall 7 but that the or each orifice 229 should not be so close to the corner 226 of the shroud 6 that the fluid jet flow fails to impinge on the shroud 6, and instead mixes with the vapour leakage flow exiting radially from the fluid leakage cavity, thereby significantly reducing the effectiveness of the seal. An understanding of the need to control the kinetic energy of the fluid jet has also influenced the design of the sealing arrangement shown in Figure 6. As discussed above in relation to Figures 4 and 5, it is advantageous to disrupt the flow of the jet fluid as it turns to flow in the same direction as the main leakage flow along the leakage flow path. In the embodiment shown in Figure 6, the disruption is provided by the downstream corner 226 of the shroud 6. That is, the axial displacement D has been selected to ensure that the fluid jet impinges on the corner 226 of the shroud as it turns to flow in the same general direction as the main leakage flow. The corner 226 of the shroud 6 is therefore functioning in a similar manner to the intermediate flow restrictions 26 and 126 discussed above in relation to Figures 4 and 5. Moreover, in the arrangement shown in Figure 6, the tight axial clearance 225 defined between a radial surface 213 of the shroud 6 and a wall 214 of the housing 3 is analogous to the flow restrictions 25 and 125 present in the arrangements shown in Figures 4 and 5. Accordingly, by careful selection of D, taking into account the various other parameters mentioned above, what may appear to be a conventional fluidic jet sealing arrangement is actually a highly optimised sealing arrangement incorporating both the intermediate and downstream flow restrictions whose importance is discussed above in detail in relation to the embodiments shown in Figures 4 and 5.

EXAMPLES

A series of CFD-based experiments were conducted to investigate the relationship between *D*, *PR*, *W*, *A* and shroud shear.

CFD Method Used To Design The Fluidic Seal

Figure 7a shows a typical 2-D axisymmetric CFD mesh used in the design calculations for a shrouded turbine geometry using the parameter values set out above in Table 1. The domain is the baseline geometry with a single inclined fluidic jet (continuous circumferential fluid sheet) applied through the casing wall. The jet is angled so that it enters the fluid leakage cavity with a component of its momentum opposing the leakage flow through the seal. In all of the CFD calculations, uniform values for total pressure and total temperature were applied at the turbine rotor inlet flow and jet inlet flow boundaries (the values used are given later in Table 2). The calculation domain was extended radially part way across the rotor flow in the direction of the turbine hub end-wall, in the rotor inlet and outlet regions. The flow through the rotor blades was not calculated (as illustrated in Figure 7a). A zero axial gradient boundary condition was applied at the outlet from the calculation domain.

The CFD calculations were carried out using Ansys Fluent. All calculations used the SIMPLE pressure-correction solver and a high quality structured mesh consisting of 35,000 cells. Mesh dependency checks showed that this mesh density was sufficient to ensure mesh-independent results in these 2-D calculations. The k-ε turbulence model was used throughout. The turbulence fields are dominated by the turbulence created in the region in and around the fluidic jet close to where it enters the fluid leakage cavity. Mean flow velocity gradients are much larger in this region than they are in any other area of the flow. The calculation results were therefore essentially insensitive to any sensible boundary condition values for k and ε at the flow inlets.

Using Fluid Jets To Reduce Leakage Losses

Figure 7b shows contours of total pressure and absolute flow velocity from a typical fluidic jet seal CFD prediction, with the boundary conditions shown in Table 2 and a jet inlet total pressure of 5.0 bar. Only the section of the flow in the region of the calculation domain that has been blown-up in Figure 7a is shown. The velocity contours illustrate how the fluidic jet is turned and mixes with the main leakage flow in the fluid leakage cavity, before exiting from the fluid leakage cavity. If the jet supply pressure is too high an ‘over-blown’ jet condition will exist, as described above, which has a negative impact on turbine performance and so it is desirable to design fluidic seals with jet conditions that allow some margin before the seal becomes over-blown.

Rotor Inlet Total Pressure	4.67 bar
Rotor Inlet Total Temperature	412.7 K
Rotor Inlet Swirl Angle	80 degrees
Rotor Outlet Static Pressure	4.14 bar
Rotor Speed	5236 rad/s
Rotor inlet turbulent k.e.	5% of mean flow k.e.
Rotor inlet turbulent length scale	0.096 mm

Table 2: Turbine Operating Parameters

The pressure contours shown in Figure 7b illustrate how the fluid jet acts to reduce fluid leakage flow. The jet enters the fluid leakage cavity in a direction opposing the fluid leakage flow. A static pressure drop is needed to turn the jet flow to the same direction as the leakage flow as mixing occurs. This causes the sudden drop in static pressure across the fluid leakage cavity in the region where the jet flow enters the fluid leakage cavity shown in Figure 7b. The presence of the jet acts to increase static pressure in the fluid leakage cavity upstream and to reduce it downstream of the point at which the jet flow enters, compared to the ‘no-jet’ condition. The higher pressure upstream of the fluidic jet reduces the pressure gradient in the inlet region of the fluid leakage cavity and therefore less fluid leakage flow enters through the inlet to the fluid leakage cavity from the main fluid flow to the rotor. For the geometry tested, the net overall leakage flow will be reduced, as compared to the ‘no-jet’ condition, so long as the fluidic jet seal is not over-blown. The fluidic jet seal also acts to reduce the pressure immediately downstream of the fluidic jet entry point. This reduces the pressure drop across the downstream axial restriction in the outlet region of the fluid leakage cavity, thereby reducing flow through the fluid leakage cavity outlet. Since the outlet flow is the sum of the inlet flow and the jet flow, the overall fluid leakage flow is reduced.

As discussed above, the effect of the presence of the fluidic jet on shroud shear, as well as fluid leakage mass flow, must be taken into account in assessing the overall turbine stage loss reduction from a fluidic seal. When a fluidic jet is added, two things happen that causes the positive effect of the shroud seal forces to be reduced or reversed. Firstly, the fluidic jet reduces the seal inlet flow and so less swirl momentum is carried into the fluid leakage cavity. Secondly, unless the fluidic jet itself is pre-swirled, the jet fluid enters the fluid leakage cavity with no swirl. The jet fluid therefore has to gain swirl momentum as it mixes with the fluid leakage flow and this will take energy out of the rotor. This latter effect can be minimized by positioning the jet close to the downstream end of the rotor shroud (as is the case in the geometry shown in Figures 7a and 7b). This is most likely to also have an advantageous effect on the mixing losses where the fluid leakage flow re-enters the main stage flow, because of the low swirl levels in the absolute rotor exit flow. The CFD calculations show that the shroud shear effect is important when assessing the performance benefit from fluidic seals.

Optimisation Of The Fluidic Seal Design

The fluidic seal design was optimised by investigating the effect on turbine power output of the parametric changes shown in Figure 8. The pressure ratio (PR) of the fluidic jet supply is defined according to equation 1.

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

PR is the ratio of the total pressure of the fluid supplying the jet (p_{02}) relative to rotor exit static pressure (p_3), to the total to static pressure drop across the rotor (p_{01}-p_3). Optimisation calculations were carried out for values of PR starting from 1.0 and increasing in steps of 0.4 until the fluidic seal reached the over-blown state. The pressure drop driving leakage flow through the fluid leakage cavity inlet meant that a PR value of unity still resulted in some jet flow entering the fluid leakage cavity.

A) Leakage Mass Flow Reduction

Seal leakage mass flow rate predictions for jet angle = 45° and distance from corner = 0.500 mm, are shown in Figure 9, for all jet widths (*W*) and jet supply pressure ratios

(*PR*) calculated. In each case, the maximum value of *PR* shown in Figure 9 is the value just before the over-blown fluidic jet seal condition is reached. The leakage mass flow rate values in the figure are the sum of the leakage flow through the inlet into the fluid leakage cavity from upstream of the rotor, plus the mass flow of the fluidic jet. The results show that the narrower the fluidic jet, the greater the reduction in total fluid leakage flow, i.e. leakage into the fluid leakage cavity plus jet flow, that can be achieved; there is higher jet flow momentum for a given mass flow rate as the jet is narrowed. The figure also shows how narrowing the fluidic jet results in a higher jet pressure ratio being required in order to achieve a given reduction in leakage flow. It is also evident from the results that greater leakage flow reductions can be achieved with narrower jets before the over-blown seal condition is reached. For the narrowest jet examined (jet width = 0.03536 mm), the results indicate that a reduction in total fluid leakage mass flow compared to the no-jet case of approaching 50 % is possible for this jet angle and axial position.

B) Overall Net Power Output Improvement Due To Reduced Leakage Flow And Taking Into Account The Impact On Shroud Shear

The impact on shroud shear is significant in quantifying the performance benefit from the improved sealing. Figure 10 shows the total rotor power output improvement due to the fluidic seal, extracted from the same calculation data set used for Figure 9.

The power gains shown in Figure 10 are calculated assuming that a 1 % reduction in turbine stage flow leaking through the fluid leakage cavity, will yield a 1 % improvement in turbine stage power, minus the power reduction due to shroud shear force effects described previously. Figure 10 shows that the conditions that produce the largest leakage mass flow reductions in Figure 9, result in a net increase in rotor power of around 450 W.

The data shown in Figure 10 is plotted as a percentage of total turbine stage power in Figure 11. This shows that the net power gain of 450 W translates to an improvement in output power of approximately 3.5 % for the turbine incorporating a fluidic jet sealing arrangement according to an embodiment of the present invention.

All of the CFD calculations described so far were for a jet angle, *A*, of 45° and distance from corner, *D*, of 0.500 mm. Calculations were carried out for other values of these parameters during the optimization of the seal design, as indicated below in Table 4.

Width, *W* (mm)	0.03536	0.04419	0.05303	0.06187	0.07071
Angle, *A* (°)	30	45	60		
Distance, *D* (mm)	0.21	0.355	0.500		

Table 4: Fluidic Seal Design Parameters Tested

The effect of the jet angle, *A*, on turbine stage power output is shown in Figure 12 for the narrowest jet width calculated (0.03536 mm) positioned 0.500 mm from the corner of the shroud cavity. Results are shown for the three values of jet angle from the axial direction, listed in Figure 8. As the angle reduces, the proportion of jet momentum directed against the fluid leakage flow increases. This increases the effectiveness of the jet in creating blockage, reducing leakage mass flow rate and consequently increasing turbine power output. In general, for any given jet pressure ratio, the improvement in power output from decreasing the jet angle from 60° to 45° is greater than that for the change from 45° to 30°. The results indicate diminishing returns for the benefits to be gained by designing seals with low jet angles, which are also likely to present greater manufacturing challenges. The fluidic seal becomes over-blown at lower values of *PR* as the jet angle reduces. The data in Figure 12 shows that it is possible to achieve turbine power outputs improvements of 3.5 % with both 30° and 45° jet angles. So, moving to the lower jet angle does not improve the absolute performance benefit, it allows that improvement to be achieved at lower jet supply pressures. In the present model of a turbine expander, the fluid supplying the jet is extracted from upstream of the turbine stage and throttled down to the desired jet supply pressure. So, the maximum jet supply pressure (p02) that is available in the current system is the turbine stage inlet pressure, i.e. no throttling of the flow supplying the jet. Under these conditions, *PR*, as defined by equation 1, becomes approximately equal to the reciprocal of the turbine stage reaction. The turbine is a low reaction impulse design and so it is possible to achieve values for *PR* that are several times greater than the maximum value of *PR* = 5.5 shown on the horizontal axis in Figs. 9 - 13. The principal seal operating constraint for this system is the jet supply pressure at which the seal becomes over-blown, rather than the value of *PR* required for optimum fluid leakage reduction. For these reasons, a jet angle of 45° was selected as the optimum angle for the present embodiment. This means that a slightly higher value of *PR* will be needed to achieve best overall leakage reduction, compared to that for a lower angled jet with a greater counter-leakage flow velocity component.

Figure 13 shows the influence of varying the axial distance of the jet entry point to the downstream corner of the shroud seal cavity (see Figure 8). The results shown in Figure 13 are all calculated for the narrowest jet width considered in the study (W = 0.03536 mm) and a jet angle, A, of 45°. Three jet positions are compared in Figure 13. The variation of output power increase with pressure ratio (*PR*) is similar for the cases where the jet is positioned 0.355 mm and 0.5 mm axial distance, D, from the corner. Quite different results can be seen when the jet is positioned closer to the corner (D = 0.210 mm). The performance improvement from the fluidic seal is much reduced in this latter case. At low values of *PR* the fluidic seal has a negative impact on performance compared to the no-jet case. At higher values of *PR* there is a net benefit from the fluidic seal, but much higher values of *PR* are required to achieve performance gains compared to the other jet axial positions. Also, the maximum achievable performance gain is significantly lower with the seal in the closest position to the corner compared to what is possible at the other jet locations. The CFD results revealed the principal difference between the flow structure in the D = 0.210 mm case and the other cases. The predictions showed that if the jet is positioned too close to the end of the shroud, the deflection of the fluidic jet in the fluid leakage flow is such that the jet flow does not impinge on the shroud, and instead mixes with the fluid leakage flow exiting radially from the fluid leakage cavity. An example of this flow regime is shown in the velocity contour plot inserted into Figure 13. It is important that the fluidic jet impinges onto the surface of the shroud in order for the jet to create maximum blockage and to achieve the best performance gains. As long as this is achieved, the performance gains are similar, as shown by the 0.355 mm and 0.5 mm axial distance results in Figure 13. It is desirable to position the fluidic jet axially closer to the outlet of the fluid leakage cavity than the inlet in order to minimize the work done by the shroud shear forces in accelerating the swirl velocity component of the jet fluid. There will be some variation in relative axial positioning between the rotor and the casing due to manufacturing and assembly tolerances and during operation. This needs to be taken into account when designing the seal. For these reasons, the axial position of 0.500 mm from the corner of the shroud seal cavity was selected as the optimum for the current embodiment.

In summary, the optimization process has resulted in the selection of the seal parameters shown in Table 4 for the fluidic seal design being developed in this study.

Jet Width	0.03536 mm
Jet Angle	45°
Jet Axial Distance from Corner	0.500 mm

Table 4: Optimised Fluidic Seal Design Parameters

The jet is supplied with fluid bled from upstream of the turbine stage and throttled to a supply pressure ratio *PR* = 3.0. This will allow some margin before the seal becomes over-blown. Under these conditions, the CFD simulations indicate that the optimised fluidic seal will improve the output power from the turbine by just over 3 %.

Conclusions

The design of a fluidic jet aerodynamic seal for an application as a turbine rotor tip seal on a small, high-speed, single stage axial flow turbine has been described. The results of a CFD-based investigation show that it may be advantageous to make the jet width as narrow as possible within manufacturing constraints, that the performance gains yield diminishing returns as the jet angle becomes increasingly acute, 45° being a good compromise, and that it is advantageous to position the jet axially towards the downstream end of the fluid leakage cavity, but not so close that the deflected jet fails to impinge on the rotor shroud. The CFD calculations have also shown that the effect of shroud shear forces is significant and should be taken into account when evaluating the net performance gain from the sealing.

The described and illustrated embodiments are to be considered as illustrative and not restrictive in character, it being understood that only the preferred embodiments have been shown and described and that all changes and modifications that come within the scope of the inventions as defined in the claims are desired to be protected. It should be understood that while the use of words such as “preferable”, “preferably”, “preferred” or “more preferred” in the description suggest that a feature so described may be desirable, it may nevertheless not be necessary and embodiments lacking such a feature may be contemplated as within the scope of the invention as defined in the appended claims. In relation to the claims, it is intended that when words such as “a,” “an,” “at least one,” or “at least one portion” are used to preface a feature there is no intention to limit the claim to only one such feature unless specifically stated to the contrary in the claim. When the language “at least a portion” and/or “a portion” is used the item can include a portion and/or the entire item unless specifically stated to the contrary.

CLAIMS

1. An axial flow turbine comprising

a turbine rotor mounted within a housing for rotation about a turbine axis,

a fluid flow inlet passage upstream of said turbine rotor arranged to direct a first fluid towards the turbine rotor in a substantially axial direction,

a fluid flow outlet passage downstream of said turbine rotor and

a seal assembly provided in a fluid leakage cavity defined between the turbine rotor and the housing,

wherein the seal assembly comprises

a fluid jet outlet configured to admit a second fluid into the fluid leakage cavity in an upstream direction which is inclined to the turbine axis,

a first flow restriction located downstream of the fluid jet outlet and at a location such that second fluid admitted from the fluid jet outlet would impinge on the first flow restriction once the second fluid has turned to flow in an axial direction, and

a second flow restriction located downstream of the first flow restriction.

2. An axial flow turbine according to claim 1, wherein the fluid jet outlet is configured to admit the second fluid into the fluid leakage cavity in an upstream direction which is inclined to the turbine axis so as to partially oppose the flow of the first fluid through the fluid leakage cavity.

3. An axial flow turbine according to claim 1 or 2, wherein the second fluid is ejected from the fluid jet outlet so as to be initially admitted into the fluid leakage cavity in an upstream direction having an axial component and a radial component.

4. An axial flow turbine according to claim 1, 2 or 3, wherein the direction in which the second fluid initially enters the fluid leakage cavity is at an angle of around 20 to 70 ° to the turbine axis or around 45 ° to the turbine axis.

5. An axial flow turbine according to any preceding claim, wherein the fluid jet outlet is defined by a wall of the housing which lies radially outboard of the turbine rotor.

6. An axial flow turbine according to any preceding claim, wherein the pressure of the second fluid exiting the fluid jet outlet is greater than the pressure of the first fluid passing the fluid jet outlet.

7. An axial flow turbine according to any preceding claim, wherein the pressure of the second fluid exceeds a threshold required to cause the second fluid to cross at least around half of the radial width of the fluid leakage cavity before it has turned to flow in an axial direction.

8. An axial flow turbine according to any preceding claim, wherein the turbine rotor is provided with a shroud extending circumferentially around the radially outer periphery of the turbine rotor.

9. An axial flow turbine according to any preceding claim, wherein the first flow restriction is located downstream of the fluid jet outlet and at a location such that, in use, second fluid admitted from the fluid jet outlet would impinge on the first flow restriction once the second fluid has turned to flow in an axial direction under the influence of flow of the first fluid through the fluid leakage cavity.

10. An axial flow turbine according to any preceding claim, wherein the first flow restriction is spaced from the centre of the fluid jet outlet by a distance that is (i) approximately one half to twice the radial width of the fluid leakage cavity, or (ii) approximately the same distance as the radial width of the fluid leakage cavity.

11. An axial flow turbine according to any preceding claim, wherein the first flow restriction is located on an opposite side of the fluid leakage cavity to the side from which the second fluid is admitted into the fluid leakage cavity via the fluid jet outlet.

12. An axial flow turbine according to any preceding claim, wherein the first flow restriction is provided on the radially inboard side of the fluid leakage cavity and the fluid jet outlet is provided on the radially outboard side of the fluid leakage cavity.

13. An axial flow turbine according to any preceding claim, wherein the first flow restriction is a projection extending radially outwards from the radially outer periphery of the turbine rotor.

14. An axial flow turbine according to any preceding claim, wherein the first flow restriction extends across up to around 90 % of the radial width of the fluid leakage cavity and/or at least 5 % of the radial width of the fluid leakage cavity.

15. An axial flow turbine according to any preceding claim, wherein the first flow restriction is defined by a section of the turbine rotor downstream of the fluid jet outlet.

16. An axial flow turbine according to any preceding claim, wherein the second flow restriction downstream of the first flow restriction is an aerodynamic seal or a physical seal, such as a radially extending fin, a labyrinth seal, a brush seal, a leaf seal, an abradable seal.

17. An axial flow turbine according to any one of claims 1 to 15, wherein the second flow restriction is defined by a section of the wall of the housing downstream of the first flow restriction which lies radially and/or axially closer to the turbine rotor than the section of the wall of the housing which defines the fluid jet outlet.

18. An axial flow turbine according to claim 17, wherein the second flow restriction is an axial clearance between a radially extending section of the wall of the housing and a radial surface of the downstream end of the turbine shroud, the axial clearance being of smaller dimension than the radial width of a section of the fluid leakage cavity between the housing and the radially outer periphery of the turbine rotor.

19. An axial flow turbine according to any one of claims 1 to 16, wherein the second flow restriction extends radially inwards from the same side of the fluid leakage cavity from which the second fluid is admitted via the fluid jet outlet.

20. An axial flow turbine according to any one of claims 1 to 16 and 19, wherein the first flow restriction extends radially outwards and the second flow restriction extends radially inwards from the opposite side of the fluid leakage cavity from which the first flow restriction extends.

21. An axial flow turbine according to any one of claims 1 to 16, 19 and 20, wherein the fluid jet outlet and second flow restriction are associated with the housing wall which lies radially outboard of the turbine shroud with which the first flow restriction is associated.

22. A turbomachine comprising an axial flow turbine according to any preceding claim.

23. A waste heat recovery system comprising an axial flow turbine according to any one of claims 1 to 21.

24. A method for sealing a fluid leakage cavity in an axial flow turbine, the turbine comprising

a turbine rotor mounted within a housing for rotation about a turbine axis,

a fluid flow inlet passage upstream of said turbine rotor arranged to direct a first fluid towards the turbine rotor in a substantially axial direction,

a fluid flow outlet passage downstream of said turbine rotor and

a seal assembly provided in said fluid leakage cavity defined between the turbine rotor and the housing,

wherein the method comprises

directing the first fluid towards the turbine rotor in a substantially axial direction, a portion of the first fluid flowing through the turbine rotor and a further portion flowing through the fluid leakage cavity admitting a second fluid from a fluid jet outlet into the fluid leakage cavity in an upstream direction which is inclined to the turbine axis,

providing a first flow restriction downstream of the fluid jet outlet and at a location such that second fluid admitted from the fluid jet outlet impinges on the first flow restriction once the second fluid has turned to flow in an axial direction after contacting the further portion of the first fluid flowing through the fluid leakage cavity, and

providing a second flow restriction downstream of the first flow restriction to restrict the flow of a mixture of said first and second fluids further through the fluid leakage cavity.

25. A seal assembly for restricting fluid leakage flow through a fluid leakage cavity defined between a first component and second component that is rotatable about an axis relative to the first component, the seal assembly comprising

a fluid jet outlet configured to admit a fluid jet into the fluid leakage cavity in an upstream direction which is inclined to the axis,

a first flow restriction located downstream of the fluid jet outlet and at a location such that the fluid jet admitted from the fluid jet outlet would impinge on the first flow restriction once the fluid jet has turned to flow in an axial direction, and

a second flow restriction located downstream of the first flow restriction.

26. A method for sealing a fluid leakage cavity defined between a first component and second component that is rotatable about an axis relative to the first component, the method comprising

directing a first fluid towards the second component in a substantially axial direction, a portion of the first fluid flowing through the second component and a further portion flowing through the fluid leakage cavity

admitting a fluid jet into the fluid leakage cavity in an upstream direction which is inclined to the axis,

providing a first flow restriction downstream of the fluid jet and at a location such that the fluid jet impinges on the first flow restriction once the fluid jet has turned to flow in an axial direction after contacting the further portion of the first fluid flowing through the fluid leakage cavity, and providing a second flow restriction downstream of the first flow restriction to restrict the flow of a mixture of said first fluid and said jet fluid further through the fluid leakage cavity.

ABSTRACT

A SEAL ASSEMBLY

A seal assembly for restricting fluid leakage flow through a fluid leakage cavity defined between a first component and second component that is rotatable about an axis relative to the first component, the seal assembly comprising a fluid jet outlet configured to admit a fluid jet into the fluid leakage cavity in an upstream direction which is inclined to the axis, a first flow restriction located downstream of the fluid jet outlet and at a location such that the fluid jet admitted from the fluid jet outlet would impinge on the first flow restriction once the fluid jet has turned to flow in an axial direction, and a second flow restriction located downstream of the first flow restriction. An axial flow turbine comprising a turbine rotor mounted within a housing may incorporate such a seal assembly in a fluid leakage cavity defined between the turbine rotor and the housing.