MTU Aero Engines

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THE MTU BRUSH SEAL DESIGN

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ABSTRACT

This report deals with the principles of design and fabrication of the MTU Brush Seal which use a unique manufacturing technique. While other brush seal suppliers around the world use a fabrication process which entails the 0.07 mm thin wires being secured by a welding process, MTU has developed and patented a method which uses only mechanical joining techniques, such as clamping and swaging. In doing so, the welding process required to fix the bristles has completely been eliminated.
In the following the advantages inherent in the MTU seal design are explained and an insight is given into the current range of applications. Additionally some results obtained by rig and engine testing are presented which confirm the functionality, performance and life of MTU Brush Seals.
Particular attention is paid to the most significant drawbacks brush seals have to cope with, like blow down effect, hang-up effect, high operational bristle stiffness, and it is shown that the MTU seal design has either resolved or reduced them to a negligible degree.
The paper concludes that the development undertaken so far with MTU Brush Seals paves the way for making the brush seal a highly competitive element for a variety of future applications.

INTRODUCTION

1. History

Back in 1983 MTU Munich commenced to manufacture brush seals initially applying welding techniques of the “conventional” seal design. After a series of welding problems became apparent, a new method for bristle retention was sought in order to delete welding processes.
After numerous attempts, a promising new method to fix the bristles was developed and established. This method was applied for patent in 1985. The first MTU Brush Seals were rig tested in 1986 and engine testing commenced in 1992. Based on successful engine testing the number of MTU Brush Seal applications in both military and civil engines have increased steadily. Meanwhile MTU Brush Seals are validated at challenging positions where brush seals are successfully applied the first time.
Testing of MTU Brush Seals for industrial gas turbine and compressor applications started in 1996. Since 1998 MTU Brush Seals are delivered for flight and industrial production engines.

2. The Conventional Brush Seal Design

Fig. 1 shows the conventional brush seal design consisting of a backing plate, retaining ring and bristle pack. The brush seal is formed by squeezing the bristle pack between the backing plate and the retaining ring followed by circumferential welding to join these three elements to one unit. For more details see Ref. 1. Basically, the philosophy that is pursued by the conventional seal design is to fix the bristles at the outer end via the welding seam, whereas the inner end contacts the mating runner.

The following requirements have to be fulfilled:

- Safe retention of every single bristle (bristle diameter usually 0.07 mm (.0028 in))
- No coning of backing plate
- No distortion of side plates
- Constant penetration depth of weld material into the bristle pack

Conventional Brush Seal

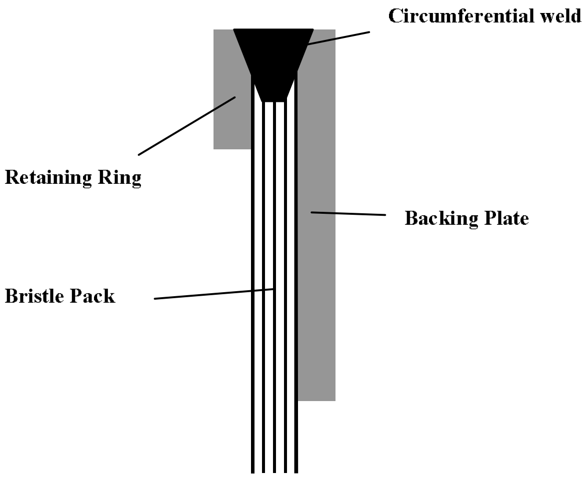

Fig.1: Conventional brush seal design (schematic)

The welding process itself is demanding and can only be used for metallic brush materials. Obviously it must be re-optimised for each design change (e.g. change of bristle pack thickness, bristle diameter, plate thickness, material).
Furthermore, the bristle material characteristics may change significantly at the heat affected zone, causing for example embrittlement and undercut of bristles respectively. Both effects jeopardise bristle retention, thus making the seal susceptible for loss of bristles during operation.

THE MTU BRUSH SEAL

1. Basic Design

In order to avoid the above mentioned welding problems, the ideal solution was to dispense with the welding process at all. This approach led to the foundation of the alternative MTU Brush Seal design which is characterised by a completely different manufacturing process.
The idea to eliminate the welding process for fixing the bristles was further developed and means sought as to mechanically secure the bristles in place. The final solution of this development process is shown in Fig. 2.
The MTU Brush Seal unit is exclusively formed by applying mechanical securing techniques such as clamping and swaging.
In this way a second important design feature was established, the manufacture of the brush seal unit from individual and independent elements, namely the core element and the casing, i.e. support plate and cover plate. This separation allows for a simple adaptation, should the brush seal be applied as a retrofit solution to an already existing interface. The brush seal side plates can be scaled in dimension to match with the seal carrier without changing the interior configuration of the seal. This means the geometry which essentially determines seal function remains unchanged.

2. Manufacturing Process

The manufacturing process is described in detail in Ref. 2. The main process steps are briefly explained in the following.

2.1. Core Element

The brush seal core element is made from

i) the core wire
ii) the bristle pack
iii) the clamping tube

The first step to produce the core element is to wind the metal thread over the core wire which is situated twice on a special spindle. These core wires are arranged in parallel and spaced apart from each other. The winding process produces a densely packed thread pack of approximately oval cross section.

Pack thickness can be varied to produce core elements of 100, 200, 300 or any intermediate number of bristles per mm. The winding process is followed by the clamping procedure which serves to effectively fix the bristles around the core wire. Two clamping tubes are pushed over the thread pack at the core wire positions.
After clamping the thread pack is cut in a section parallel to the spindles, such that two opposite straight semi-finished brush products of about equal bristle length are produced.
Fig. 3 shows such a longitudinal brush seal strip which is the basic product for the next working steps to follow.

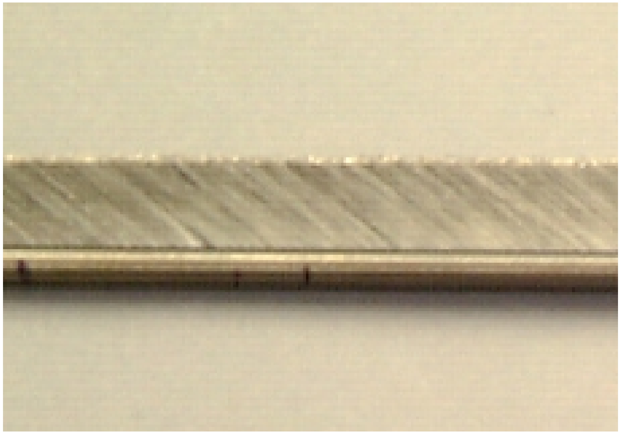

Fig. 3: Semi-finished Brush Seal strip

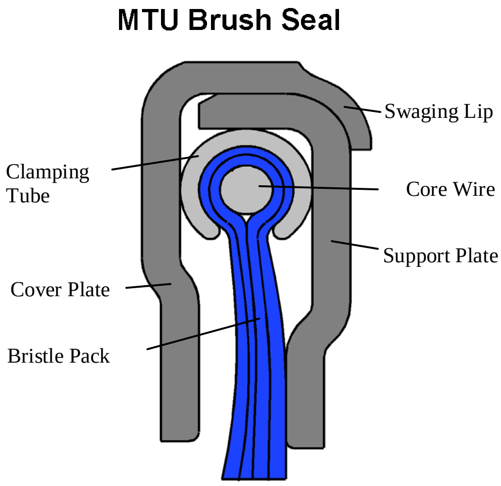

Fig. 2: Cross section of MTU Brush Seal (light weight version)

This manufacturing technique ensures safe retention of every single bristle. Unlike the conventional seal design, the bristles are clamped at half the wire length, i.e. both wire ends exit in a free state from the clamping tube. As the clamping provides a form-fit, loss of bristles other than by rupture when applying excessive force is excluded.

In this context it is worth-mentioning that the winding process is not confined to special thread materials but can be applied to every thread or fibre material that is ductile enough not to break when being wound around the core wire. In this way brush seals consisting of various metal threads as well as ceramic and plastic fibres have been successfully fabricated.
The process itself is generally independent of the most significant brush seal parameters such as brush seal size, bristle pack thickness and bristle diameter. Furthermore, reproducibility is excellent.

Depending on the desired brush seal size, the longitudinal strip is mechanically formed by rolling to become a closed ring which completes the core element manufacturing process, Fig. 4.

2.2 Brush Seal Casing

The brush seal casing is formed by two separate parts, the support plate and the cover plate.
These plates are either produced by a deep-drawing process or by turning. The former uses pre-cut round blanks which are formed to the respective shape via special-to-type die tools. Usually this type of fabrication of side plates is chosen, if either seal weight plays an important role (for example in aero engine applications) and/or the quantity of required seals is high to balance additional tooling costs by reduced production time. A brush seal of this type is shown in fig. 2. Producing the side plates by a turning process saves the die tooling costs and give flexibility to the design process as the seal outer dimensions (for example wall thickness) can be adapted to match with the corresponding seal carrier.

Any change in seal geometry during the development process can easily be introduced, hence this manufacturing process is usually selected for prototype seals.
An additional field of application for this type of seal is with non-aero engine applications (e.g. gas and steam turbines, industrial compressors), where seal weight does not play an important role. Fig. 5 shows a typical cross section of a MTU Brush Seal having side plates machined on a lathe. The turning process is currently being optimised and standardised to reduce costs.

Fig. 4: Photograph of a Brush Seal core element

2.3 Completion of Brush Seal Unit

Once the side plates and the core element have been finished, the brush seal unit is formed by putting the core element inside the support plate and the cover plate. Subsequently these parts are squeezed together and the swaging lip is rolled inwards to close the seal unit. Thereby the seal housing is encapsulated which ensures that the core element is axially and radially clamped avoiding rotation within the seal housing. In special cases the support plate and the cover plate can alternatively be joined by a welding process. This may be beneficial if due to space constraints butt-welding of the side plates is required to further reduce radial seal height (compare for example fig. 6). Furthermore a welding joint is often applied to single prototype seals to keep tooling costs down. In this way, fabrication is highly flexible and capable of quickly supporting varying customer needs.
Once the seal unit is complete, the inner bore diameter is finally machined. Upon customer requirement the brush seal can be cut into segments to enable for example installation into machines with split housings.

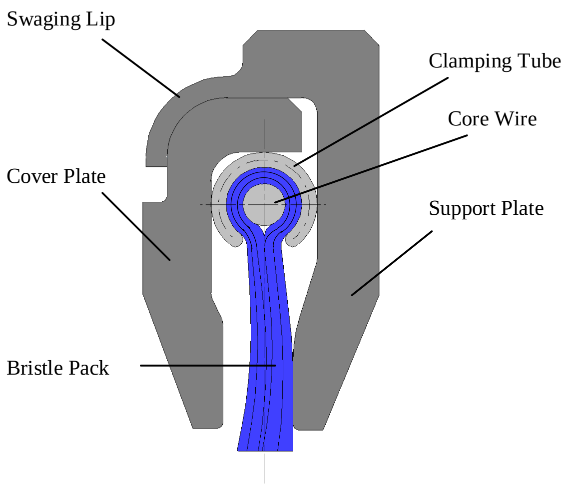

Fig. 5: Cross section of MTU Brush Seal with machined side plates

3. Special Design Features

3.1 Shape of Support Plate and Cover Plate

The support plate and the cover plate are both of a cranked shape. In addition to the important functional aspects which are set out in section 3.6 this allows a reduction of the wall thickness of the support plate without compromising the structural capability as advantage is taken from the stiffening effect (applies of course to the light weight version only). The front plate serves to protect the bristle pack against handling damage and also against high energy swirl during operation. Additionally it helps to reduce the blow down effect.

3.2 Radial Seal Height
The MTU Brush Seal (light weight version) typically measures only about three quarters of the height of a conventional brush seal. The reduced installation height opens the possibility to fit MTU Brush Seals at positions where conventional brush seals cannot be installed.

3.3 Maximum Pressure Loading
Meanwhile the MTU Brush Seal is well proven at differential pressures up to 1200 kPa (175 psid) across a single element, provided the bristle pack overhang beyond the support plate is moderate. The capability to cope with such high pressure loads offers the advantage to use a single brush seal element were formerly multiple brush seal arrangements were required.
Thus the problems known to exist with multiple seal arrangements are avoided and, as a side effect, weight reduces significantly due to lower number of seals and correspondingly shortened runner length.

3.4 Overall Weight Saving
Typically at aero engine applications, the light weight version (see fig. 2) is selected. When compared with the conventional seal design, this MTU Brush Seal type offers significant weight savings due to:
- thinner side plates,
- reduced radial seal height enabling reduced seal carrier diameter at a given shaft size,
- use of single seal elements instead of double or triple seals, hence, shortened runner lands

Obviously the benefit multiplies with the number of seal positions per engine. This is of special interest for example at military engine projects where high sealing performance at minimum seal weight is a requirement.

3.5 Extraction Lip
As detailed in this report, the individual brush seal casing elements allow for simple integration to existing interfaces or specific customer needs, e.g. to design the seal with an integral extraction lip, as shown in fig. 6.
This seal is situated such that inspection is possible on modular strip of the engine. The previously used labyrinth seal, however, required complete module tear down to substitute the seal having significant cost and time impact to the engine overhaul process. In order to remove this extra work and to expedite maintenance, the customer expressed the wish to remove and replace the seal in a simple manner with access only from the rear (right hand side in the figure) and with the main shaft in place.
This requirement was fulfilled by adding an integral extraction lip to the brush seal housing enabling seal removal by means of a simple puller tool.

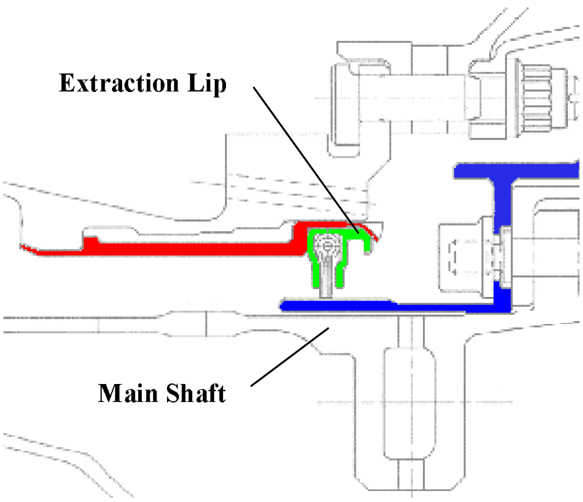

Fig. 6: MTU Brush Seal with integral extraction lip

3.6 Low Hysteresis and Low Stiffness Design

The performance and reliability of highly loaded brush seals is significantly compromised by bristle blow down, pressure stiffening and hysteresis (hang up) effects. These phenomena inherent in the conventional brush seal design are well documented in numerous publications, e.g. Ref. 3, 4 and 5.

Blow Down Effect

Blow down is mainly driven by the pressure differential acting across the seal. For a given seal design, this effect intensifies with increasing Δp. Typically the blow down effect produces two distinct wear patterns:

i) chamfering of the upstream bristle rows

ii) uneven circumferential wear mainly in a saw tooth pattern

As a result, the bristle pack suffers from premature wear which extends partially beyond the backing plate. The blow down effect of the MTU Brush Seal is very low even at high Δp conditions and, hence, wear problems are negligible. For the front bristle rows this may be attributed mainly to the shape and position of the cover plate which protects the bristles from swirl and aerodynamic effects. Brush seal inspections performed on numerous engines neither showed chamfering nor uneven wear (fig. 7).

Definition of the brush seal inner bore diameter usually considers some initial rub-in of the seal in order to compensate for build tolerances, such as eccentricity, actual size of mating parts, etc. In this way the brush seal is capable of adapting itself to the actual build situation. Once this rub-in process has come to an end, seal wear should basically stop for steady state running. Fig. 8 presents a typical MTU Brush Seal wear characteristic given as change of the inner bore diameter vs. running time.

The graph shows an average wear trend that is based upon mechanical inspection results of eight engines of a military project after extensive development and certification testing. The dotted lines depict the tolerance band for new seals. The graph clearly indicates that after some running in, taking up to 50 hrs., the initial wear rate declines and levels off beyond 250 running hours to give constant seal performance. Brush seal wear may then be driven only by extreme flight manoeuvres involving high g-loads or high gyro loads.

Fig. 7: MTU Brush Seal after 356 hrs. flight testing in a military project

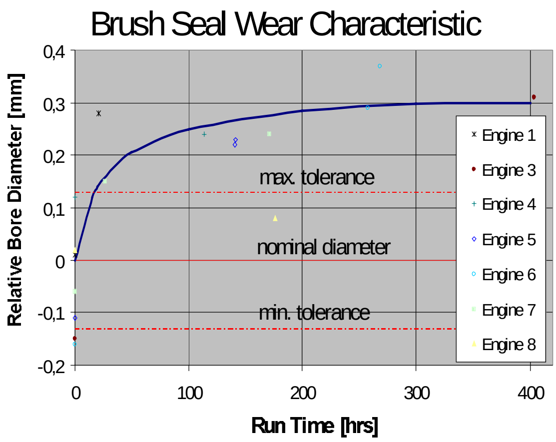

Fig. 8: Wear characteristic vs. running time of MTU Brush Seal of a military project

Pressure Stiffening Effect

The pressure stiffening effect occurs when the bristle pack is compressed and pushed against the backing plate. The increased friction between bristle to bristle and bristles to back plate reduce the seal flexibility drastically. In the first approximation, this effect grows proportionally with the pressure differential applied across the seal, i.e. it is particularly pronounced at high pressure loading. During rotor excursions, this creates high bristle tip forces which in turn leads to wear out of the bristles.. At its worst the frictional heat generated at the bristle tips may melt the bristle material causing either build-up of deposit on or damage to the runner.

In order to suppress this effect, brush seal design must aim at generating a pressure-balanced bristle pack. Ideally, if this was achievable the operational stiffness would be independent of the pressure loading. In reality, however, as the bristle pack is compressed by the pressure drop and the bristles are supported by the backing plate, friction and thus seal stiffness inevitably increases. Therefore, the design objective is to minimise this increase in stiffness. This was accomplished with the MTU design and proven on dedicated static rig tests, i.e at zero speed condition.
Fig. 9 presents a graph of brush seal stiffness versus differential pressures up to 500 kPa (73 psid). Ref. 3 reports that the stiffness ratio of conventional brush seals increase by more than an order of magnitude over a pressure range of 550 kPa (80 psid), whereas an advance development of a so called 'low hysteresis seal 1', shows a rise in stiffness ratio from 1 at zero load condition to only 4 at 345 kPa (50 psid).

The stiffness ratio is defined as

$$\frac{\text{Bristle stiffness at a given } \Delta p}{\textit{Bristle stiffness at a zero } \Delta p}$$

It is evident from the curve in fig. 9, that the pack stiffness of the MTU Brush Seal steadily increases with increasing pressure drop. At a pressure differential of 500 kPa (73 psid) seal stiffness measures approx. 3 times that at a zero load condition, and compares well with the 'low hysteresis seal 1' mentioned above.
When looking at the rise of stiffness ratio it can be seen that the graph tends to reduce from pressure differentials of 200 kPa (29 psid) onwards. This indicates that seal stiffness may increase moderately only at pressure differentials above 500 kPa (73 psid).
Rig measurements conducted to date have been under static conditions only (i.e. zero speed), and it is assumed, that stiffness values may reduce somewhat under dynamic conditions. To investigate this behaviour further, dedicated rig tests are scheduled for later this year.

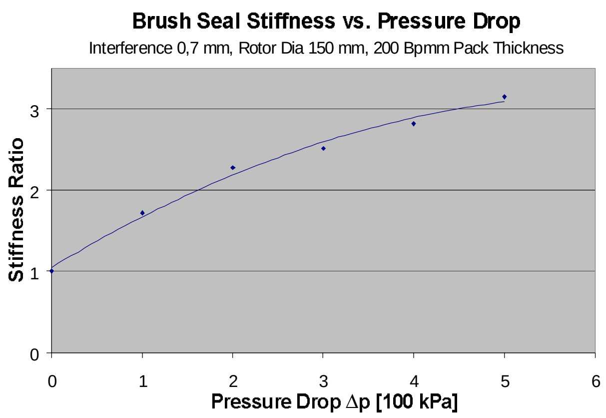

Fig. 9: MTU Brush Seal stiffness ratio vs. pressure differential

Hysteresis Effect

The hysteresis effect of a brush seal causes the bristles to get stuck in a displaced position, for example after a rotor excursion, and do not drop back to the runner surface creating an enlarged gap which results in higher leakage flow.
The MTU Brush Seal hardly shows such hysteresis. This is mainly a result of the pressure-balanced design along with the position and the width of the contact area where the bristle pack rests against the support plate.

Fig. 10 depicts a graph taken from rig testing. The intent of the test was to demonstrate seal performance and life under simulated maximum conditions of a military engine, exposing the seal to radial offsets up to 0.50 mm (.020 in). During the test campaign, when the test rig was operated at 500 °C (930° F), the offset mechanism unintentionally drifted away from the pre-set value causing radial offsets up to 0.80 mm (.031 in).
With the rig set at about 170 m/s (558 ft/s) rotational speed, inlet air temperatures up to 500°C (930 °F) and pressure drops up to 600 kPa (87 psid) across the brush seal, the housing was radially displaced downwards and held eccentric for 30 sec before returning it to a concentric position (illustrated as vertical arrows on fig. 10).

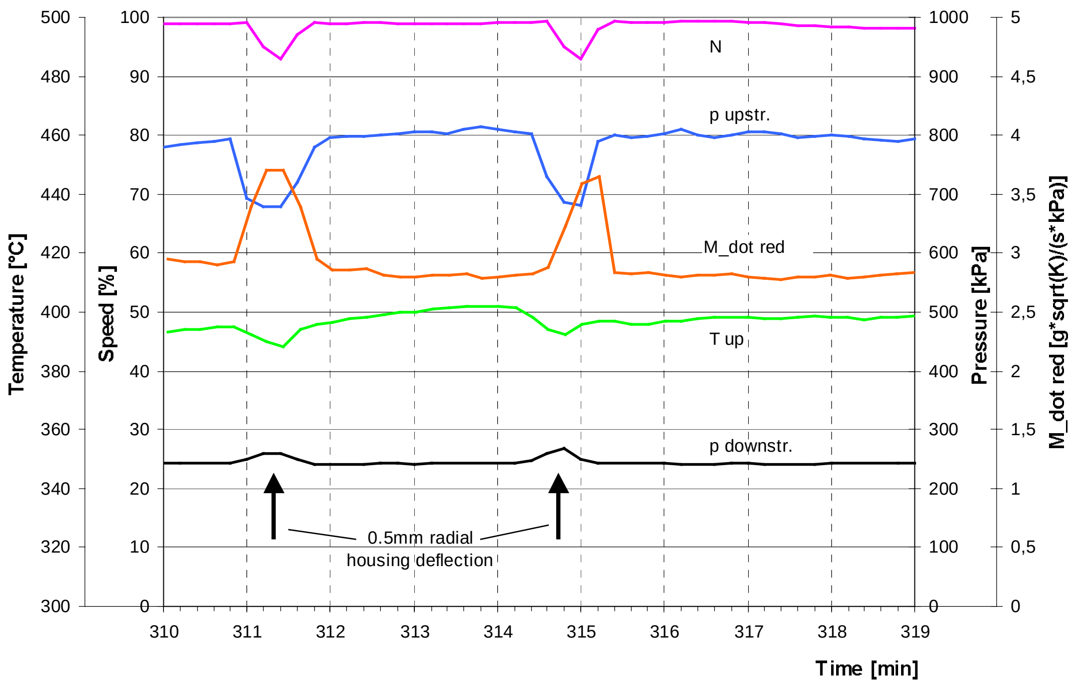

Fig. 10: Test results of MTU Brush Seal under radial housing displacement

During each cycle the rig setting was maintained constant, i.e. changes of the respective curves are due only to the radial housing movement.
Since the testing was endurance the scan rate was reduced somewhat not to overflow the data storage capacity. For this reason most of the peaks are cut off (flat, inclined or declined “peak”) and hence do not represent the true maximum values reached.
In order to get representative results, a total of 300 cycles were performed; all of them giving consistent seal behaviour.
As can be seen from the traces, the leakage flow increased with the radial displacement of the housing, because a sickle-shaped gap opens at the lower half of the seal. This leads to a decrease of the upstream and an increase of the downstream pressure. When looking at the rotational speed it is apparent that during housing deflection the runner speed reduces. This behaviour can be traced back to the rig drive motor, being an air turbine. At a given driving pressure turbine revolutions reduced due to increased friction when the brush seal was pushed down towards the runner.
After returning the housing and thus the brush seal to concentric position, pressures, rotational speed and seal leakage recover completely to their original value. This shows that the bristles follow the housing movement without visible hysteresis.

4. Range of Application

In principle MTU Brush Seals can operate in gas to liquid and gas to gas environments. Currently the following maximum operating conditions have been tested (per brush seal element):

Absolute Pressure: up to 3500 kPa (500 psi)

Differential Pressure: up to 1200 kPa (175 psid)

Temperature: up to 700 °C (1300 F)

Sliding Velocity: up to 400 m/s (1300 ft/s)

Meanwhile the applicable pressure range is well-proven. With regard to temperature and sliding velocity engine testing has successfully demonstrated up to 650 °C (1200 F) and 270 m/s (900 ft/s). The maximum values given above have been demonstrated on rig testing and will be subject to engine testing in the near future.
The manufacturing process applied by MTU enables fabrication of brush seals in a big range of size. At present MTU Brush Seal are produced from 50 mm (2 in) to 650 mm (25in). It is planned, however, to extend brush seal size to 1 m (39 in) and above to allow installation at the main gas path of large industrial gas and steam turbines.

5. Experience

The MTU Brush Seal design has been substantiated through various test activities, e.g. rig, aero and industrial turbo engines. The experience gained is briefly summarised in the following sections.

5.1 Rig

MTU operates three individual brush seal rigs, viz.:

- The Segment Rig
- The Small Rig (rotor dia 60 mm to 170 mm)
- The Large Rig (rotor dia 170 mm to 350 mm)

Additionally, a separate test rig is currently being designed to investigate multiple seal arrangements. These rig facilities allow the test specimen to be exposed to the following conditions:

- Upstream pressure up to 3000 kPa (450 psi)
- Downstream pressure up to 1000 kPa (150 psi)
- Temperature up to 600 °C (1100 °F)
- Sliding velocity up to 400 m/s (1300 ft/s)
- Radial housing deflection up to 0.8 mm (.032 in)

Rig testing commenced in 1986. Since then comprehensive experience has been gathered in the fields of

- Coating of runner lands
- Single and multiple seal arrangements
- Bristle/fibre materials (metallic and non-metallic)

Numerous test sequences have been conducted to understand and optimise seal performance and life when subjecting the seal to extreme conditions found within military engines. To date, in excess of 1500 hrs. rig testing has been performed.

5.2 Aero Engines

Today, MTU Brush Seals are validated and cleared for use in two military projects. These two projects comprise a total of seven seal positions, all being shaft seals, single elements and designed to a seal pack width of either 100 or 200 Bristle per mm circumferential length. The brush seals are situated at High and at Low Pressure Turbine areas and serve to control the secondary air system of the engine.
Since 1995 extensive bench and flight testing has been conducted to support engine validation and certification. Up to now a total of 17,000 bench hours and 3,000 flight hours have been accumulated within military projects. The highest flight time achieved on a single seal to date is 6oo hrs., and the seal is still running on. Since this was accomplished on a military engine being subjected to a number of engine cycles during a typical flight mission, i.e. frequent changes of forward speed, altitude, shaft speed, g- and gyro loading it multiplies by an order of magnitude when being compared to the flight profile of civil aero engines.
The success gained on military projects has laid the groundwork to expand the field of application in order to include civil aero engines.
Today, MTU Brush Seals are operated at high duty seal positions such as forward HPT disc as well as at inner air seal position between adjoining turbine stages. At both positions engine validation tests are well underway and substantiation is very close to completion. Clearance for production use is expected imminently.

5.3. Industrial Turbo Engines

In 1997 MTU Brush Seals were first time installed into industrial gas turbines and compressors to demonstrate their functional benefit over the existing labyrinth seal configurations.
With industrial compressors the leakage flow reduced to one third of the previous labyrinth seal and was proven to be constant over thousands of hours of operation at service. As a result, MTU Brush Seals were given clearance for series production beginning of the year 2000.
On stationary gas turbines the performance demonstration was similarly successful, but endurance testing is still ongoing to prove seal life. It is expected, that life substantiation work will be completed by the end of the year and clearance for series production is planned for the beginning of 2001.
Since November 1998 MTU Brush Seals have successfully run in an operational steam turbine. Additional tests are being performed to investigate brush seal performance in a multiple element arrangement at the balance piston position.
To date in excess of 120,000 hrs. have been accumulated with industrial turbo engine applications. The highest time achieved on a seal amounts to approx. 14,000 hrs , and the seal is still running on at a sealing performance almost unchanged from the beginning.

CONCLUSION

The MTU Brush Seal design and manufacturing technique has been presented and shown, that it differs fundamentally from the conventional design. The advantages inherent in the MTU design have been explained and rig and engine test results have been discussed. These confirm that the functional problems known from conventional brush seals are either resolved or reduced to an acceptable level.
Since the beginning of initial MTU Brush Seal development in 1986 significant experience has been collected mainly with single stage seals. Whilst validation and certification work has generally been completed, service experience is now being gathered.
Development work is continuing at MTU including basic research studies of seal design as well as future applications, such as multiple seal arrangements and alternative seal materials.
The MTU Brush Seal design has reached a high level of maturity which is considered a sound basis for making the brush seal a highly competitive element for a variety of applications in the years to come.

REFERENCES

1. J. Ferguson, ASME Paper No. 88-GT-182, 1988, „Brushes as High Performance Gas Turbine Seals“

2. European Patent No. 0211275, MTU Motoren- und Turbinen-Union GmbH

3. P. Basu, A. Datta, R. Johnson, R. Loewenthal; J. Short AIAA Paper No. 93-1996, 1993, “Hysteresis and Bristle Stiffening Effects of Conventional Brush Seals”

4. G. Berard, J. Short, AIAA Paper No. 99-2685, 1999, “Influence of Design Features on Brush Seal Performance”

5. R. Chupp, P. Nelson, AIAA Paper No. 90-2140, 1990, “Evaluation of Brush Seals for Limited-Life Engines”