DIPLOMA THESIS

CFD-Simulations for Advanced Turbomachinery Sealing Technologies: Brush Seals

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ABSTRACT

Thermodynamic cycles of aeroengine gas turbines are constantly driven to higher pressure ratios, bypass ratios and turbine inlet temperatures. Because of these airflows in the internal air system and resultant thermodynamic cycle losses increase. Reducing internal airflows while increasing thermodynamic efficiency puts greater emphasis on improvements to the internal flow system.

Although the labyrinth seal technology is very well developed, leakages are still high. The need for a sealing technology with good leakage sealing performance and compact size has risen. Brush seals fulfilled such conditions. The technology was present for decades, but the accelerated development can be seen only over past twenty years. The manufacturing technologies, as well as computing capabilities had to reach a satisfactory level.

One must take into consideration that along with drastically improved leakage sealing, decrease of cooling appears. Brush seals are designed to contact the rotor. It is possible to minimise the interference between bristle tips and the shaft surface to zero. But the problem of overheating will appear. So during the design process it is generally advisable to adjust bristle interface to the point where cooling will be efficient. Also wear occurring, as bristle tips contact rotating parts is a particular problem. These questions are addressed in the following chapters.

Sealing technologies undergo constant development and improvement. The progress is unstoppable. On one hand labyrinth seals are shown as expiring technology. Efforts are made to replace these types of seals with more efficient ones. On the other hand, new technologies are being developed. Finger seals show somewhat better test results and seem to be more predictable than brush seals. It is a never-ending battle between technologies shortcomings and human needs. In light of recent fuel price fluctuations and inevitable reaching the end of fossil fuels deposits on Earth, the energy put into decreasing parasite leakages and therefore increasing turbomachines efficiency should never be underestimated.

A general overview of the technology supported by literature research and computer simulations performed at Vienna University of Technology is presented.

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NOTATION

A	[m ²]	Total inlet face area: $A = (R^2 - r^2)\Pi$
ĉ	[-]	Inertia coefficient
F_{fact}	$\left[\sqrt{K}\cdot s\right]$	Flow factor
g	$[m/s^2]$	Acceleration of gravity
Н	[m]	Total head
h	[m]	Elevation height
h	[m]	Piezometric head
h_{bf}	[m]	Bristle free height
$h_{_{fh}}$	[m]	Fence height
h_1	[m]	Inlet height
h_{1}, h_{2}	[m]	Lengths measured with respect to arbitrary (horizontal) datum level
(h_1-h_2)	[m]	Difference of piezometric head across the filter length L
K	[-]	Conductivity coefficient
• m	[kg/s]	Mass flow rate
р	[Pa]	Static pressure
\overline{p}	[Pa]	Mean pressure
P_a P_d	[Pa] [Pa]	Absolute pressure Pressure downstream
p_{nd}	[-]	Dimensionless pressure
p_u	[Pa]	Pressure upstream
R_p	[-]	Pressure ratio

T_{avg}	[K]	Average inlet temperature
$\ \boldsymbol{u}_i\ $	[m/s]	Magnitude of the velocity
V	[m/s]	Flow velocity
V_{S}	[m/s]	Surface velocity
x	[m]	Distance in x direction
Y	[-]	Normalised radial coordinate
у	[m]	Distance in y direction
γ	[N/m ³]	Specific weight
ε	[-]	Porosity coefficient
ĸ	[-]	Permeability coefficient
$\overline{\mu}$	$[kg/(m\cdot s)]$	Dynamic viscosity
V	[m ² /s]	Kinematic viscosity
ρ	[kg/m ³]	Density
ϕ	[°]	Lay angle, cant angle
Ψ	[-]	Stream function

SUPERSCRIPTS

* Non-dimensional value

SUBSCRIPTS

e Effective property of the value of which must be prescribed

1 Introduction

Thermodynamic cycles of aeroengine gas turbines are constantly driven to higher pressure ratios, bypass ratios and turbine inlet temperatures. Because of these airflows in the internal air system and resultant thermodynamic cycle losses increase. Reducing internal airflows while increasing thermodynamic efficiency puts greater emphasis on improvements to the internal flow system.

Due to the very high rubbing speeds and temperatures existing in gas turbine air system seal positions, finned labyrinths have been used almost exclusively since the invention of the gas turbine. Development over the years has reduced their leakage flow to the ultimate, but leakage is very much dependent on the clearance, which, in case of labyrinth seals, is a significant factor during determining the leakages.

Although the labyrinth seal technology is very well developed, leakages are still high. The need for a sealing technology with good leakage sealing performance and compact size has risen. Brush seals fulfilled such conditions. The technology was present for decades, but the accelerated development can be seen only over past twenty years. The manufacturing technologies, as well as computing capabilities had to reach a satisfactory level.

To design a good brush seal that will not go quickly out of service or, in worst case, damage the rotor, many aspects have to be taken into consideration. Environment, materials used, weight, bristle behaviour, etc. In the following chapters the majority of encountered aspects of design, as well as problems encountered, are described. Brush seals are a potential replacement for air-to-air labyrinth seals in gas turbine engines. They are passive seals, as well as labyrinth, abradable and leaf seals. They are now the subjects of intensive international research. Their behaviour is far from being understood. With drastically improved sealing capabilities many new aspects during the design process have risen. Leakage rates in different seals, from honeycomb through cloth, to labyrinth, are relatively large. Turbomachinery designers utilised that additional air was wasted for cooling. One must take into consideration that along with drastically improved leakage sealing, decrease of cooling appears. Brush seals are designed to contact the rotor. It is possible to minimise the interference between bristle tips and the shaft surface to zero. But the problem of overheating will appear. So during the design process it is generally advisable to adjust bristle interface to the point where cooling will be efficient. Also wear occurring, as bristle tips contact rotating parts is a particular problem. These questions are addressed in the following chapters. A general overview of the technology supported by literature research and computer simulations performed at Vienna University of Technology is presented.

2 Overview of Sealing Technologies

In this chapter an overview of most commonly used sealing technologies is presented. Their construction, operating parameters and applications are shown. In **Table 2.1** an overview of engine seal capabilities is presented. The values are taken from Steinetz et al. **[37]**.

Seal	Pressure (MPa)	Temperature (K)	Surf. Speed (m/s)	Materials
Face	1.034	811	145	Carbon
				Ni
Labyrinth	1.724-2.758	978	457	Superalloy
				Teeth
				+
				Abradable
Danah	0.551-0.689/	079	205	Cobalt
DIUSII	stage	978	303	Superalloy

 Table 2.1 Turbine engine seal technology [37]

Comparison in **Table 2.1** is a basis for description of sealing technologies. Many factors are not taken into consideration here. For example, the axial dimensions of seals. A factor that is crucial in demanding construction specifications.

2.1 Carbon Face Seals

Face seals applications include bearing locations in turbine engines and auxiliary power units. Carbon Graphite face seals have high corrosion resistance and natural lubricity. They also have low leakage and can seal pressures up to 1 MPa. Surface speeds up to 145 m/s are acceptable in terms of friction and wear rates. According to Steinetz et al. [37], low costs of face seals make them an alternative to more expensive labyrinth seals – especially in advanced aircraft engines. The selection of carbon face seals is shown in **Fig. 2.1**



Fig. 2.1 Selection of carbon face seals [45]

There are two ways of inserting carbon into mechanical seal. One is called a monolithic face seal, seen in **Fig. 2.2**.



Fig. 2.2 Carbon/graphite seal face (monolithic)

This design has become popular in recent years. Though its popularity, another design type has advantages over monolithic type. It is called a composite face seal (**Fig. 2.3**).



Fig. 2.3 Carbon/graphite face seal inserted into a metal holder

When using composite seals, carbon part can have a smaller cross section. It is important because of material homogeneity. It is easier to impregnate smaller part and eliminate unwanted air pockets and making it better conductor of heat. Carbon is stronger in compression then in tension. Metal holder keeps the carbon face in compression. Metal holder acts as a heat sink and carries unwanted heat away from the face seal. Anti-rotation channels and pins work best when metal is contacting metal. Drawbacks of using a carbon/metal composite include differences in thermal

Drawbacks of using a carbon/metal composite include differences in thermal expansion of materials. The carbon part can loosen in the holder and start to leak or spin. Low expansion metals have to be used:

- Carpenter Low Expansion 42® Alloy It is a nickel-iron alloy that has thermal expansion rate almost equal to carbon. Low expansion rate is sustained at temperatures up to 616 K. Coefficient of thermal expansion is defined as the fractional increase in length per unit rise in temperature. Values for carbon face seal elements are presented in Table 2.2.
- Carpenter Invar 36® Alloy It is a 36% nickel-iron alloy possessing a rate of thermal expansion approximately one-tenth that of carbon steel at temperatures up to 477 K

Material	Mean Coefficient of Thermal Expansion [10 ⁻⁶ /K]
Carpenter Low Expansion 42® Alloy	6.3 - 7.2
Carpenter Invar 36® Alloy	1.3 – 7.6
Carbon	2.34 - 2.7

Table 2.2 Mean coefficients of thermal expansion fo different materials

A problem encountered while using face seals is coking and blistering. Coking appears during sealing oil environments. Problem of coking is explained in detail in **Chapter 3.5**. Another problem during usage of carbon face seals appear when large diameters are to be sealed. Tolerance control makes the costs rise significantly, as depicted by Carlile et al. [7]. Honeywell managed to replace carbon face seals with ceramic ring seal, overcoming coking problem and eliminating oil odour in the cabin. Seal operating life was also significantly increased.

Also keeping the carbon face flat after it has been installed in the metal holder causes troubles.

For more details on carbon face seals, refer to [44].

2.2 Labyrinth Seals

It is the most commonly used flow path seal over turbine engine history. The labyrinth seal consists of multiple knife edges (typically 5) run in close clearance $(2.5 \cdot 10^{-4} - 5.08 \cdot 10^{-4} \text{ m.})$, depending on location. Labyrinth seals are clearance type seals and therefore have high leakage rates. Leakage increases over time. Clearances open when shaft excursions force the labyrinth teeth into the adjoining rub strips. Labyrinth seals are used as shaft seals, turbine rim seals, and as inner air seals – sealing the vane-to-drum inter-stage locations. An example of labyrinth seal is shown in **Fig. 2.4**.

The basic principle of a labyrinth seal is based on the geometric shape of the labyrinth seals which causes some turns of the contaminant on its way to penetrate the seal. The major factor for the efficiency of a non-contact seal is the centrifugal force caused by rotation and applied to the contaminant to throw it radially away before penetrating the seal. At higher peripheral speed an air barrier is built inside the seal keeping different types of contamination (e.g. dust or liquids) out.

There is no chance for a labyrinth seal to protect against higher liquid levels and against a pressure gradient between both sides of the seals. Pressure gradients may be reduced but not be sealed.





Fig. 2.4 Labyrinth seals – different designs

A detailed comparison of labyrinth seal technology with brush seals is presented in **Chapter 3.6.1**

2.3 Film-Riding Seals

Film riding seals rely on a thin film of air to separate the seal faces and show promise of reducing wear and leakage to its practical limit. Film riding face seals can be designed to operate at the high pressures and temperatures anticipated for nextgeneration gas turbine engines. There are two classes of film riding seals being developed for gas turbines: hydrostatic and hydrodynamic seals.

Hydrostatic face seals port high pressure fluid to the sealing face to induce opening force and maintain controlled face separation. Changes in the design clearance results in an increase or decrease of the opening force in a stabilizing sense. Converging faces are used to provide seal stability. Hydrostatic seals are not applicable to lower pressure differential applications. Hydrostatic face seals suffer from contact during startup, requiring faces made of rub-tolerant materials.

Hydrodynamic or self-acting face seals incorporate lift pockets to generate a hydrodynamic film between the two faces to prevent seal contact. Hydrodynamic seals operate on small ($<1.27 \cdot 10^{-05}$. nominal) clearances resulting in very low leakage compared to labyrinth or brush seals. Because rubbing occurs during start-up and shutdown, seal faces are made of rub-tolerant materials [**37**].

2.4 Outer Air/Blade Tip Seals

Better management of blade tip leakage improves engine designs in several ways. Reduced compressor blade tip leakage improves compressor efficiency and improves stall/surge margins, improving engine operability. Maintaining tighter clearances over the life of the engine addresses a key observation that 80-90% of engine performance degradation is caused by blade tip clearance increase. In a limited number of commercial engines, blade tip clearance control is used. Blade tip clearance control is performed by preferentially cooling the turbine case during cruise operation. This has been successful in greatly reducing turbine blade clearances in the PW4000 series of engines and has resulted in turbine efficiency gains [37].

2.5 Finger Seals

The finger seal is composed of a series of finger elements sandwiched between aft and forward spacers and cover plates. Each finger element has been machined to create a series of slender curved beams or fingers around its inner diameter. The finger elements are alternately indexed so that the fingers of one element cover the spaces between the fingers on the adjacent element. The flexible fingers can bend radially to accommodate shaft excursions and relative growth of the seal and rotor resulting from rotational forces and thermal mismatch. The seal is made of sheet AMS5537, a cobaltbase alloy that has good formability, excellent high temperature properties, and displays excellent resistance to the hot corrosive atmospheres encountered in jet engine operations [32].

Detailed description and comparison with other sealing technologies is presented in **Chapter 3.6**.

2.6 Abradable Seals

One type of improved sealing being incorporated into turbines is abradable seal to reduce the blade-tip clearances. An abradable material is placed on the stationary shroud or casing opposite the rotating blade tips to reduce clearances with minimum risk to the turbine components during rubs. Also, applying an abradable material further reduces effective clearances for often-encountered casing out of roundness and rotor lateral movement. A thermally sprayed coating is applied to stage 1 E-Class gas turbine shrouds to reduce tip clearances and improve turbine performances up to 0.8%. Honeycomb is used as an abradable seal in the stage 2 and/or 3 of E-Class turbines with performance benefits up to 0.6%. Efforts are continuing to develop abradable materials for these turbine locations with increased service life.

Fig. 2.5 shows a schematic of where the abradable materials are placed. As the name suggests, an abradable material is worn-in by the rotating blade during service. These materials applied to the casings or shrouds of gas and steam turbines decrease clearances to levels difficult to achieve by mechanical means. Abradable seals are gaining appeal in gas turbines as a relatively simple means to reduce gas-path clearances in both the compressor and turbine. They offer clearance reductions at relatively low costs and minor engineering implications for the service fleet. Abradable seals have been in use in aviation gas turbines since late 1960's/early 1970's. However, they have been used less in land based gas turbines for power generation, primarily because of the long cycle times the materials are in service. With increasing fuel prices and advances in materials to allow extended service periods, abradable seals are gaining popularity within the power generation industry.



Fig. 2.5 Schematic of a abradable material for blade tip sealing [14]

Abradable seal materials are used to decrease the operating blade tip clearances. Without abradable seals, the cold clearances between blade (or "bucket") tips and shrouds must be large enough to prevent significant contact during operation. Use of abradable seals allows the cold build clearances to be reduced with the assurance that if contact occurs, the sacrificial part will be the abradable material and not the blade or bucket tips. Also, abradable seals allow further tightening of effective clearances with commonly encountered shroud or casing out-of-roundness or lateral movement of the rotor relative to the casing shroud. For these situations, the shroud material is worn away locally rather than wearing all the rotor blade tips during interference[14].

Abradable seals can be classified according to their temperature capability:

- Low temperature, usually for LP compressors— ambient to 673 K.
- Mid-range for LP and HP compressors—ambient to 1033 K.
- High temperature for HP turbines—1033 K to 1423 K.

Alternative, abradable seals can be also characterized by their method of application:

- Castings for polymer based abradable materials.
- Brazing or diffusion bonding for honeycomb and/or fiber metals (porous fiber metal structures).
- Thermal spray coatings for a large range of powdered composite materials [14].

For detailed analysis of abradable sealing technologies, see Chupp et al. [14] and [16].

3 Description of Brush Seals

3.1 History

The first attempt to replace labyrinth seals with brush seals was done in 1955 with General Electric (GE) J-47 engine. This turbojet engine was developed by GE from the earlier J35 engine and was first flight-tested in May 1948. It replaced J35 engines in North American XF-86 "Sabre". Unfortunately, the application of brush seals at that time turned out unsuccessful. Rolls Royce managed to apply brush seals in the 80' in demonstrator engines. Afterwards, in 1987, RB-199 engine was produced with installed brush seals. IAE V2500 is an engine certified in 1987. It was, for several years, the only production engine with brush seals. Allison has conducted many tests with usage of brush seals. In T800 brush seals mounted at compressor interstage discharge locations and in the engine hot section. Allison has came up with conclusions that brush seals reduced leakage flow up to an order of magnitude over labyrinth seals, and are tolerant to transient clearance changes.

3.2 Construction

A brush seal consists of front plate, bristle pack and a back plate. A typical brush seal element is shown in **Fig. 3.1**.



Fig. 3.1 Brush seal: Cross section

Some seals, especially the older design, consist also of welding that holds bristles together, see Chapter 3.4.1. The air enters the seal from the side of the front plate, and exits at the other side of the seal. Bristles are mounted to the seal in different ways, as explained in Chapter 3.4. The bristles are supported from the back by the backing plate. The purpose of that plate is to prevent bristles from deflections in axial direction. Clearance between backing plate (also referred to as back plate) and the rotor shaft is high enough to accommodate any excursions due to vibrations or centrifugal and thermal growth. Decreasing the clearance improves pressure difference capability. But it also increases the risk of rubs. Increasing the clearance, on the other hand, makes bristles deform at inner diameter. It also leads to increased leakage. The evident fact is that a reasonable compromise must be obtained to properly design this part of the brush seal. Front plate, which is placed at the front of the seal, directs the stream outwards from the top of the seal and leads it to the bottom, where the bristles will affect it most effectively. Front plate is one of many features of the seal that can be altered to obtain satisfactory results. Bristles have usually an interface between their tips and the shafts surface. Alternating it affects the stream behaviour. It is also possible to diminish the interface at all, which it is the case considered in the thesis's simulation. Although, as mentioned in Chapter 1, zero clearance in not desirable. The bristles themselves are densely packed in a circumferential length of the seal. The thickness of the pack is described by parameter B. The usual amount is 1200 bristles per centimetre circumferential of the seal. The material used for bristle production is a super alloy Haynes 25, Hastealloy X or non-metallic materials, as seen in Chapter 3.5. Bristles are characterised by their length, diameter and packing density (which is connected with diameter). Also the free length of bristles is important. It is a distance where bristles are not protected by a front plate against the flow. The most interesting feature is the lay angle. It is an angle at which bristles are tilted towards the shaft, seen in Fig. 3.2. The standard value is $\phi = 45^{\circ}$ in the direction of shaft rotation. Bristles are pushed away from the shaft when a contact with rotor occurs. They are designed to bend. The angle influences the bristle wearing resistance and improves the sealing itself. It also prevents bristles from buckling, which would appear if the bristles were aligned radially.



Fig. 3.2 Brush seal: Front view

Typical values for brush seal construction are as follows:

Bristle diameter	$\sim 7.0 \cdot 10^{-6} - 5.0 \cdot 10^{-5} \ m$
Number of bristles	$\sim 100 - 200 \ 1/mm$
Bristle length	$\sim 0.007 - 0.01 \text{ m}$
Backing plate gap	0.001 – 0,002 m
Lay angle	$\sim 45^{\circ}$

Table 3.1 Typics	al geometric values	of brush seals
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There are two possible methods of installing a brush seal:

- Conventional brush seal is fitted into existing labyrinth packing ring.
- New design the pack is welded to side rails and the strip is slid into a slot in the packing ring. The side of the slot serves as the backplate. It can be rolled to diameter for cycle time savings.

Typical locations for brush seals at rotor shaft are end packing locations and interstage shaft seal. They are also selectively placed at bucket tips. A typical high pressure section may have 8 - 12 turbine stages. The most common brush seal application would be one brush at each interstage location, and 3 - 6 brush seals at end packing locations. As seen in **Table 3.2** the performance benefit of brush seals in steam turbines makes them a significantly worthwhile investment in the majority of utility and industrial units.

	Utility ST	Industrial ST
Interstage	0.5 - 1.2 % HP section efficiency	0.2 - 0.4 % efficiency
_	0.1 - 0.2 % unit heat rate	
End Packing	0.1 - 0.2 % unit heat rate	0.4 - 0.8 % efficiency
Bucket Tip	0.5 - 1.0 % HP section efficiency	0.7 - 1.1 % efficiency
	0.1 - 0.2 % unit heat rate	

Table 3.2 Performance benefits [40]

Utility steam turbine is typically rated at 400 - 800 MW and industrial steam turbine at 50 - 150 MW.

Average values of pressure drop across interstage seal are between 0.68 MPa -2.7 MPa. Currently work is being conducted to work the way to handle up to 13.8 MPa at inlet end of steam turbine.

Rotordynamics is a very important consideration in how many seals are applied, at which locations and with what level of assembly clearance/interface. By increasing assembly clearance the impact on rotordynamics is reduced. Contacting seals at the middle of rotor affect start-up and influence behaviour below first bending critical speed. Contacting seals at rotor ends influence behaviour below second critical speed. They also affect stability at running speeds. After mounting seals, turbine can be started and operated normally with no special considerations. The steam turbine solid, flexible shaft is sensitive to rub-induced heating and possibility of resultant rotor "bow", which results in rotor vibrations. When a brush seal is assembled with clearance, blowdown results in minimal contact of bristles to rotor, but a significant performance improvement is observed. High-pressure endpacking leakage feeds low pressure endpacking – it seals the low pressure end. It means that unit must remain self-sealing and that the number of brush seals at each end must be optimised. End packing brush seals must be integrated into the overall unit sealing system. The system must be balanced.

During the service brush seal usually contacts the rotor. The drag factor on the engine spools is inevitable. This fact has to be taken into the consideration. Brush seals, although easier to mount than labyrinth seals, can also be damaged. Most failures occur due to exceeding the design specifications and during assembly. There are situations in which brush seals are unsuitable. When the dynamic or differential movements are particularly severe, the seal may not be able to cope with them completely. They would have the leakage much higher than their optimum. A finned labyrinth seal, on the other hand, in the same conditions would have to be run with the higher clearance, along with higher leakages. The experience of seal developers have shown, however, that even in such conditions brush seal seals the system better than the finned labyrinth seal in the same position.

It is worth mentioning that brush seals can and are installed in series. It is possible to put more then one seal next to each other to obtain specific work parameters. These include improving of reliability of critical engine components, distribution of the pressure drop per brush and mitigation of wear. In the past it was a common solution for high-pressure application. A stage consisted of two or three brush seals among which the pressure drop was shared. It was found out that the pressure distribution in dual brush seal configuration is not even. About 40% of the total pressure drop across

the system of seals occurred over the first brush and 60% across the second. The consequence of such state is the necessity of using stiffer bristles in the second seal. That leads to faster wear process of the bristles. The system undergoes uneven wear process. Dual brush seals leakage factor is approximately 2.5 times smaller than in comparable labyrinth seal (see article [27] for further details). When a dual brush seal system is subjected to increasing pressure, it tends to pack. Due to that leakage flow decreases to some point (0.83 MPa in [27]) and after reaching that border, increases. This configuration decreases specific fuel consumption around 4% with comparison to labyrinth seal.

It is not always possible to predict a pressure break down between the stages. With time the percentage of pressure drop across each stage can change, and shift, causing one stage to take the majority of pressure drop and fail prematurely. This in turn shifts the burden of the high-pressure drop to another stage accelerating its failure.

Addition of lubricant to the bristles reduces leakage by 2.5 times when compared with non-lubricated bristles (Carlile et al. [7]). An example of lubricant can be halocarbon grease. Halocarbon 25-5S is silica-thickened chlorotrifluoroethylene grease designed for use in contact with powerful oxidizers. It is the ideal lubricant for valves, seals, and gear lubrication for high-pressure oxygen, chlorine, 90% hydrogen peroxide, 100% nitric acid, red fuming nitric acid, and other oxidizers and aggressive chemicals. The silica is, of course, subject to attack by hydrogen fluoride and other related compounds. In those cases, polymer-thickened greases (25-10M) should be used. In applications where self- thickened chlorotrifluoroethylene greases are more desirable, the usage of Halocarbon 25-10M is recommended [42]. It should be put into the bristle pack and along the inside diameter.

It is important to know the effect of reversing the pressure drop across the seal. It results in leakage performance drop making it comparable to labyrinth seal. Such conditions may happen after improper seal installing or during adverse engine operation. However a situation when leakage flow changes directions is sometimes inevitable. It takes place in wave rotors. In such a case a bidirectional brush seal was used to solve the reverse flow problem. For details on wave rotor and it's sealing, see Hendricks et al. [29].

3.3 Design Considerations

During the design process the behaviour of bristles has to be taken into account. The knowledge of this part of the seal is still insufficient. As mentioned in **Chapter 6.1** bristle simulations are most computer time consuming and proper codes are more in development then in use. The known problems can somehow be dealt with.

During designing of brush seal its wear down has to be accounted for. After bristles will wear due to, for example, rotor excursions, leakage performance will decrease. When implementing a brush seal into an engine by replacing an existing annular seal, changes in cooling air and engine dynamics appear due to modification of secondary flow path.

A significant problem is a blow down effect. It is driven by the pressure differential acting across the seal. It usually intensifies with increasing pressure difference. It

appears due to the airflow moving radially downward through the bristle pack causing the bristles to be driven towards the rotor. Blow down tends to "straighten" the bristles and pushes them towards the rotor surface. During that process bristles flutter. The consequences of such a phenomenon include chamfering of the upstream bristle rows and uneven circumferential wear mainly in a saw tooth pattern. To deal with this problem, it is possible to incorporate a flow deflector upstream of the bristle pack, as seen in **Fig. 3.3**. This solution is described in detail in the article by Short **[36]**.

Another problem is pressure-stiffening effect. It takes place when the bristle pack is compressed and pushed against the backing plate. Friction between bristles themselves increases as well as between bristles and backplate. Seal flexibility is reduced drastically. Effect grows proportionally with the growth of pressure difference. In the worst case the bristles melt because of a friction heat and build up of deposit on the runner occurs. It is important to design a pressure-balanced bristle pack. It is possible to deal with this problem by adding a relief in the back plate, as seen in the **Fig. 3.3**. This solution is described in detail in the article by Short **[36]**.



Hysteresis is an effect that also decreases brush seal efficiency. In this case bristles are stuck at displaced position after, for example, rotor excursion. After that they do not return to their previous position. It creates a gap resulting in higher leakage flow.

3.4 Design Types

As stated in **Chapter 3.2**, the bristles are arranged at an angle to both the radial and circumferential directions. The angle defined by a radial vector and the bristles is known as the cant angle. Thanks to these arrangements bristles are flexible and withstand rotor contact. It is difficult to maintain those precise parameters while manufacturing the seal because of the small diameter of bristles – ranging $\sim 7.0 \cdot 10^{-6} - 5.0 \cdot 10^{-5}$ m depending on material. Turbomachinery manufacturers

develop the process to make it simpler and to improve the seal parameters – bristle resistance, brush seal weight, maintenance simplicity. In this chapter the most common designs will be presented.

3.4.1 Standard Brush Seal Design

In conventional brush seal design bristles in the seal are joined together using welding process. A circumferential welding seam is applied, as seen in **Fig. 3.4**. The brush seals elements – front plate, back plate and the bristles – are joined together in that way.



Fig. 3.4 Standard brush seal with visible welding

To create a seal using welding techniques some important requirements need to be fulfilled. Every single bristle needs to be safely retained. Side plates cannot be distorted. Weld material constantly penetrates the bristle pack. Also, only metallic bristles can be welded. The major drawback of welding is that bristle material characteristics may change significantly at the heat affected zone. Bristles can undergo embrittlement and after that be undercut. This means loosing bristles during operation. Solutions to these problems can be found in the next sections describing manufacturing techniques without using welding process.

3.4.2 General Electric Fabrication Method

The brush seal patented by General Electric (**Fig. 3.5**) consists of bristles that are not welded by the end tips together (the numbers in parentheses represent part numbers shown in **Fig. 3.5** to **Fig. 3.7**). They extend along opposite surfaces (44) and (46). The tips (48) extend beyond the edge (50) of the carrier (42). The bristles are fitted into the grooves (52) and (54) which are formed during manufacturing process.



Fig. 3.5 A cross-sectional view illustrating a brush seal constructed in accordance with a preferred embodiment of patent US 6505835 [31]

The tips of bristles are in contact with a sealing surface (shaft, for example). Bristles are not supposed to extend along both sides of the carrier (42). Nevertheless, it is possible to extend them on both sides – that makes the seal bidirectional – making sealing of both positive and negative leakages possible.

In **Fig. 3.6** the process of fabricating of brush seal is illustrated. In **Fig. 3.6A** the carrier (42) is in the form of an arcuate segment. A pair of knurling wheels (60) and (62), with oblique cutting ribs (64) and (66), are pressed into contact with the opposite surfaces (44) and (46) of the carrier to form tiny grooves (52) and (54). The width of each groove is less than a millimeter.



Fig. 3.6 Schematic perspective views illustrating various steps of brush seal fabrication [31]

The angle formed between the groove and the radius relative to the axis is the desired cant angle ϕ . It is also desired that the oblique cutting ribs form grooves at corresponding cant angles on opposite sides of carrier.

The discrete bristles are wrapped about the carrier and laid into the grooves. Various types of machines are available for wrapping or bending the bristles about the carrier and locating the bristles in the grooves.

In Fig. 3.5 the grooves (52) contain multiple bristles (45). The grooves are defined in part by ribs (70) that terminate in tips (72), seen in Fig. 3.7, which project beyond the bristles contained in the grooves. Means are provided for joining together (securing) the bristles and the carrier. For example, the tips of the ribs may be bent over to substantially close the grooves, locking the bristles from movement within the grooves. To accomplish this, pair of smooth rollers (74) and (76) (Fig. 3.6C) are applied to the tips to deform them in the same direction, as illustrated in Fig. 3.7.



Fig. 3.7 Cross-sectional view of the grooves containing bristles. Tips of the ribs are deformed to retain the bristles [31]

By flattening the tips of the ribs, the bristles are clamped into the grooves. As shown in **Fig. 3.5**, it is also possible to apply epoxy (78) over the deformed teeth and bristles in the groove to facilitate retention of the bristles in the grooves. Epoxy may also be applied over the entire surface of the carrier, filling in the grooves to retain the bristles, without deforming the rib tips.

As seen in (**Fig. 3.6D**), the bristle-free radial height is present (84), also referred to as the "pinchpoint". It affects the bristle stiffness and seal flexibility and can be adjusted by machining the surfaces of carrier, back from the edge (50), to a greater or lesser extent.

3.4.3 MTU Brush Seal Design

MTU Aero Engines calls this design "unique". It allows creating a brush seal by joining its components without using of neither welding nor gluing – processes influencing chemical and physical structure of the product. The process is using only mechanical techniques as clamping and swaging to join the parts together.



Fig. 3.8 Cross section of MTU Brush Seal light weight version (left) and with machined side plates (right) [22]

The brush seals core element, as seen in **Fig. 3.8**, consists of a core wire, bristle pack and a clamping tube. The metal thread is winded over the core wire. Wires are arranged in parallel and spaced apart from each other. It is possible to control the pack thickness to obtain a satisfactory number of bristles per millimeter. After the wire is threaded, it has to be fixed effectively 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 core wires, so two opposite, straight semi-finished brush products of about equal bristle length are produced. This method of wire application makes breaking of wires during operation almost impossible. The wire is continuous through the core wire and extends evenly on both sides.

This method of production enables to use a variety of materials for bristle production, as long as it will not break during the threading process. In this way MTU was able to produce brush seals with metal, ceramic and plastic bristle fibers.

The core element is then put into casing. It is made from support and cover plate. MTU produces both light casings, for aero engines applications, as well as standard, less costly constructions used in gas and steam turbines, industrial compressors. Brush seal is formed by putting the core element inside support plate and the cover plate. Swaging lip is rolled inwards to close the seal unit. After construction finalising it is even possible to cut the brush seal into segments to enable, for example, installation into machines with split housing. In some cases support and cover plate can be welded together. It can be useful when low radial height of the seal is required. Butt-welding is used at that case. Such a radial height reduction makes possible to use MTU brush seals in places where conventional brush seals did not fit. Military aero engines are of particular interest for such light seal usage. The weight plays a significant role in designing that type of engines. The requirements are high sealing performance at minimum seal weight.

3.5 Materials

Tribology of brush seal is of considerable interest due to the continuous influence of bristle wear on long term seal performance and life. This issue concerns directly the choice of materials used for fabrication of bristles – part of the brush seal most directly exposed to hazardous environment. Results of experiments done at NASA Lewis Research Center show the influence of different material compositions on bristle durability.

	Co	Ni	Cr	Fe	W	Others(< 6 wt.%)
H25	51	10	20	3	15	Mn, Si, C
H214		75	16	3		Mn, Si, Al, C, B, Zr, Y
IX750	0-1	70	14-17	5-9		Ti, Al, Nb, C

 Table 3.3 Chemical composition of wire sampled (wt.%) [19]

The experiment described above tested the following superalloys:

Haynes 214 is a Nickel-Chromium-Aluminium-Iron alloy that is principally intended for use at temperatures of 1228 K and above. It exhibits resistance to oxidation that far exceeds virtually all conventional, heat-resistant wrought alloys at these temperatures.

The most popular material used for bristle manufacturing, Haynes 25. This alloy is a Cobalt-Nickel-Chromium-Tungsten alloy that combines excellent high temperature strength with good resistance to oxidizing environments up to 1253 K for prolonged exposures. It also has excellent resistance to sulphidation.

The reports conclusions were that IX750 composition was the most suitable. It is a Nickel-Chromium alloy made precipitation hardenable by additions of Al and Ti, having creep-rupture strength at high temperatures to about 973 K. It is widely used for high temperature conditions. Without the addition of Co it is ideal for nuclear reactors. Applications include nuclear reactors, gas turbines, rocket engines, pressure vessels and aircraft structures [41].

All versions of H214 even did not survive till the end of the test – they inflated before the completion. This does not mean that H214 is not suitable for manufacturing bristles from it. The experiment conditions were much more severe then actual operation ones. It is a sign, though, to solve the long life problem of brush seals bristles. The coating of bristles has also an effect on their performance. Coatings used by NASA are as follows: Plasma sprayed nickel-chrome bonded chrome carbide, HVOF nickel-chrome bonded chrome carbide, and plasma sprayed zirconia. The proper match of tuft and coating materials give the overall lowest system wear. For more detail analysis of the subject see [19].

Carlile et al. **[7]** mentions the possibility of usage of Hastealloy X, which is a Nickel-Chromium-Iron-Molybdenum alloy with an exceptional combination of oxidation resistance, ease of fabrication and high temperature strength. It has also been found to be exceptionally resistant to stress corrosion cracking in petrochemical applications. Applications include gas turbine engine components, industrial furnace applications, chemical processing and petrochemical industry.

Non-metallic brush seals are also a subject of investigation. They are to be used as an oil seal for use in turbomachinery. This type of sealing is typically required around bearings. Brush seals for oil sealing have been used for a short time. The requirements are that tight clearance is preserved to avoid oil contamination of the downstream turbine components. The traditional sealing technology in these areas utilized labyrinth seals. As mentioned in Chapter 3.6.1, labyrinth seals are designed with a large radial clearance to avoid any contact with rotor surface, which would result in overheating and damaging of rotor. Carbon circumferential seals were a solution for many years. They are effective as vapor oil seals. They do not generate abrasive particles that could cause damage to turbomachinery components. Problems appear during tolerance control in large diameter applications and the seal costs itself create a challenge. Carbon deposits building up on critical surfaces may also cause the seal to hang up. The problems during designing a brush seal as an oil seal include coking – the term refers to the carburization of oil particles at excessively high temperatures. The temperature at which oil starts to coke depends on its chemical composition. The phenomena may result in generation of carbon deposits that stick on the blades of the compressor causing performance degradation as well as increase in maintenance costs. It was found out that adding a lubricant to the bristles improves its performance. This conclusion was presented by Carlile et al. [7] and Hendricks et al. [26] during investigating brush seals leakage performance using liquid helium.

One of the reasons of choosing a non-metallic bristle brush seal is a problem of particle generation. It is highly advised to avoid any debriefs in oil environments. Metal bristles are therefore inadequate for the oil environment.



Fig. 3.9 Wear test results for Aramid and Haynes 25 tufts against Ni-Cr-Mo-V. Data are normalized with wear of Haynes 25 bristles at 423 K [4]

The same problem concerns ceramic fibres. Abrasive nature of wear eliminates it from oil environment usage. Any metallic or ceramic particles in oil can be hazardous.

The solution is organic fibers. They, however, have low temperature resistance and shrink when the temperature rises. And when bristles shrink, leakage increases. In oil or oil mist environment temperatures reach 423 K in bearing cavities.

Aramid (Kevlar[®]) is an organic polymer of high strength and density. Its operating temperature reaches 423 K. The materials shrinkage is negligible, as well as moisture absorption. Aramid strength reaches 2482 MPa and an average 3.2 % strain was observed at failure. Experiments have shown that at 423 K strength decreases to 2413 MPa. For 21 days the loss is not significant. After that time the loss in strength is accelerated, to reach 2300 MPa after 35 days. When the temperature was increased to 533 K, the loss became significant. But this temperature exceeds operating conditions in oil environment. Aramid shows good creep properties at 423 K. As seen in Fig. 3.9, the material reveals better wear performance then commonly used Haynes 25. Friction coefficients are comparable between these two materials at 423 K. Aramid bristles have smaller leakage than Haynes 25 ones. It is due to the bristle packing characterisation. Aramid bristles are smaller then Haynes 25. Their diameter is $1.17 \cdot 10^{-5}$ m. In that way bristles are denser packed. Seal porosity is also reduced. During static condition tests, organic polymer seal baseline leakage is less then half of the metal one. There also have been to traces of oil particles coming through aramid fibres, as long as bristle tips maintained contact with the shaft surface. The bristles present an obstacle for oil particles. Aramid is being investigated for a short time – it is a rather new concept in sealing technology. Many issues and problems have to be resolved. One of them is the amount of heat generated by the seal - it is within the

range of oil-coking temperature. The more detailed analysis of using Aramid as a bristle material can be found in article written by Bhate et al. [4].

Usual material used for coating the rotor surface with which bristles are in contact, is ZrO_2 or CrC, which are ceramic materials, highly polished.

3.6 Comparison with Competing Turbine Engine Seals

It is important to realise the overall difference between competing turbine engine seals. In this chapter brush seal is compared with labyrinth seal – the most popular and known sealing technology. Brush seal supremacy is striking – several times better leakage performance, weight reduction and space requirements. On the other hand, a new sealing technology developed by NASA is put in comparison. It seems that finger seal described in **Chapter 3.6.2**, a rather new construction – exhibits better leakage performance to brush seal. The image of constantly changing and developing area of engineering is shown.

3.6.1 Labyrinth Seals

It is important to know that one of characteristic features of labyrinth seals is intrinsic clearance. It can be tight at the beginning of seals operation, but during exploitation it increases mostly because of shaft excursions and thermal growth. This increment results in parasitic leakages and engine performance losses. According to Ludwig and Bill (1980) clearance increment can result in even 17 percent loss in power and 7.5 percent increase in specific fuel consumption (SFC).



Fig. 3.10 Sealing performance of brush seal compared with various five-finned labyrinth seals at a pressure ratio of 2.0 [21]

Fig. 3.10 shows the enormous sealing potential of brush seals, in comparison to different types of labyrinth seals. The values were obtained at Rolls-Royce plc laboratories and published in the article [21]. A vertical finned seal, with 0.00075 m radial clearance, is used as the basis for comparison with a single element brush seal. At large clearances, it can be seen that finned seal leakage can be reduced to just under half by using inclined step up seals. When the clearance is reduced however, the effect on fin geometry also decreases until it has only a small effect. These data are applicable to seal in a range of diameters from approximately 0.1-0.7 m. Surface speeds can vary from around 60-300 m/s. The comparison shown in Fig. 3.10 is based on empirical values. In one case, during obtaining the data, a 3% increase in thrust was demonstrated in comparison with finned labyrinth seals. Many other authors report a significant sealing improvement. Carlile et al. [7] reports leakage reduction up to 9.5 times in comparison with labyrinth seal. As mentioned in Chapter 3.2, brush seal is constructed in such a way that it accommodates any transient radial deflections without significant increase in seal leakage. The finned labyrinth seal, on the other hand, increases its leakage in direct proportion to the increased clearance opened by a transient rub. Due to the geometry of the brush seal, it responds to the pressure difference by the bristles moving in towards the rotor. If the clearance between rotor and the brush bore exists, it will tend to close it. The advantages of brush seals are also visible when used as mainshaft bearing chamber seals. Compared to finned labyrinth seals, brush seal can maintain a much higher pressure difference. The outcome is a much more stable and balanced sealing system that is unlikely to generate leakages.

Testing has shown the heat generation of a brush seal is also much lower then of labyrinth seals. This state is valid though a brush seal is in rubbing contact. According to Ferguson [21], when compared with a five-finned labyrinth seal, at a pressure ratio of 1.3, the heat generated by the brush seal was approximately ¹/₄ that of the labyrinth seal. Brush seal can also result in less heat transfer to the engines lubricating oil, due to windage effects. This occurs due to lower air density within the bearing chambers as a result of larger pressure drop across brush seal. Flexibility of the bristles allows radial differential movements. However, the cost is higher leakage. But when compared to finned labyrinth seal, leakage is still considerably lower. Another interesting advantage is that even though rubbing between bristle tips and rotor surface occurs, the heat generated is lower then one created by windage in finned labyrinth seal.

Brush seals offer simple repair procedures. Due to construction method, elements of the seal can be easily replaced. The rotor coating, when worn, can also be relatively easily resprayed. The costs of maintenance are considerably lower when compared to labyrinth seals.

3.6.2 Finger Seals

The finger seal is composed of a series of finger elements sandwiched between aft and forward spacers and cover plates. Each finger element has been machined to create a series of slender curved beams or fingers around its inner diameter. The finger elements are alternately indexed so that the fingers of one element cover the spaces between the fingers on the adjacent element. The flexible fingers can bend radially to accommodate shaft excursions and relative growth of the seal and rotor resulting from rotational forces and thermal mismatch. The seal is made of sheet AMS5537, a cobaltbase alloy that has good formability, excellent high temperature properties, and displays excellent resistance to the hot corrosive atmospheres encountered in jet engine operations [32].

Finger seal is the newest sealing product – patented just in 2002 by AlliedSignal Engines. It was formerly tested at NASA Glenn Research Center. Tests of the seal presented here can be examined in more detail in article by Proctor and Delgado [32],



Fig. 3.11 Finger seal design [32]

presented during ASME conference in June 2004. The article represents the most up to date research concerning sealing technologies. An interesting equation was devised in

the article. A so-called flow factor (3.1). It enables to compare the leakage performance of seals with different diameters and with different operating conditions. The accuracy of such a measured flow factor is ± 1.5 %.

$$F_{factor} = \frac{i}{m} \sqrt{T_{avg}}}{p_u \times D_{seal}} \left[\frac{kg \cdot \sqrt{K}}{MPa \cdot m \cdot s} \right]$$
After rearrangement: $\left[\sqrt{K} \cdot s \right]$ (3.1)

The test results show finger seals supremacy over brush seal in 700 K performance test. The flow factor ranged from 60 to 75 percent of the brush seals. At maximum test conditions (T=922 K, V_s =366 m/s, R_p = 4.9) finger seal showed only slightly better performance than brush seal (11.5 percent improvement). Endurance tests have shown overall better finger seal performance.

4 Application of Brush Seals

An example of wide range applications of brush seals is cryogenic turbomashines used in space shuttle main engine (SSME) seen in **Fig. 4.1**. The Space Shuttle Main Engine is the most reliable and highly tested large rocket engine ever built. The SSME is a reusable, staged-combustion cycle engine. Using a mixture of liquid oxygen and liquid hydrogen, the SSME can attain a maximum thrust level (in vacuum) of 256,475 tons which is equivalent to greater than 12,000,000 horsepower. The regeneratively cooled engine also features high performance fuel and oxidizer turbopumps that develop 69,000 horsepower and 25,000 horsepower respectively. Ultra-high-pressure operation of the pumps and combustion chamber allows expansion of all hot gases through a high-area-ratio exhaust nozzle to achieve efficiencies never previously attained in a production rocket engine. These advantages allow a heavier payload to be carried without increasing the launch vehicle size.



Fig. 4.1 Block II Space Shuttle Main Engine

The engines components are exposed to extreme conditions. Cryogenic turbomachines used in SSME class have short run times, and the operating conditions and fluid environment is hostile. Speeds range up to 300 m/s, pressures to 55 MPa, temperatures from 20 K and nonequilibrium fluid mixtures.

Experimental data concerning leakage performance using gaseous working fluids is also available. Air, helium and carbon dioxide were investigated by Carlile et al. [7]. Influence of adding a lubricant was also investigated. The comparison with annular seal gave the improvement range from 3.5 to 9.5 times. Lubricating a seal improved the leakage rate up to 2.5 times less than that of the nonlubricated seal. It indicates that the lubricant held the bristles in place and blocked the porosity through the seal making it more effective. Lubricant was more effective in the static case.

Another rocket engine using brush seals is RS-68. The RS-68 engine is the first new large liquid-fuelled rocket engine to be developed in the United States in 25 years. Designed for the Boeing Delta IV family of evolved expendable launch vehicles (EELV), the bell-nozzle RS-68 is a liquid hydrogen - liquid oxygen booster engine utilizing a simplified design philosophy resulting in a drastic reduction in parts compared to current cryogenic engines. This design approach results in lower development and production costs. RS-68 Turbopump requires that the hydrogen seal will separate the pump from the turbine with extremely low levels of leakage and be contained in small packages. Brush seals used in RS-68 turbopump give lower leakages and greater rotor dynamic stability. Low leakage enhanced tight radial tolerance allows for lower fence height. Cross-coupled stiffness is eliminated because brush seal behaves like a swirl brake, removing circumferential energy from the fluid. Disadvantages of brush seals in RS-68 are mostly heat generation and material wear. But the cool liquid hydrogen passing through the seal removes the heat generated by the bristles rubbing on the shaft. Minimizing the interface reduces the magnitude of normal force between the bristle and the shaft, thus minimizing heat generation. Brush seal is designed to operate in this extreme environment, with shaft rotating at 145 Hz producing 165.6 MPa of pump discharge pressure and 0.63 m^3/s of flow. Turbine inlet temperature is 777 K and the pressure of 4.5 MPa. Volume flow rate is equal to 0.001 m^3/s . But the turbine is not designed to last. Because it is designated for ELV application, its lifetime is around 800 seconds. For further details see [30].

General Electric is one of the leading developers of brush seals. Its engines run successfully with that type of sealing. An example of investigation of brush seal appliance can be T-700, see article [27]. It was designed for the U.S. Army's UH-60A Black Hawk as a result of lessons learned from helicopter operations in Vietnam. The engine was tested to use brush seals as compressor discharge seals. Rub runner was coated with CrC, chromium carbide. Brush seal usage implies better secondary airflow distribution, better engine performance (3 % at high pressures to 5 % at lower pressures).

Till September 2001, General Electric had 70 brush seals operating in 9 machines in the field. The fleet leader operated 32000 hours until that time. Unit inspection after three years resulted in replacement of ten seals – although they looked not damaged. Another inspection was conducted after 1.5 years of service – seals had a minimal brush wear and were returned to operation.

Brush seals are not a remedy for all sealing problems. Sometimes, after testing, it turns out that they are not suitable for particular application. This is the case of LM2500+ industrial aeroderivative gas turbine [25]. Some of the problems that made the seal unsuitable for this application were solved by MTU and described in [22]. These problems included the concerns over bristle durability and weld damages during construction. It is a good example of how a brush seal design influences possible customers decisions. Also the weight of the seal tested was too high in comparison to other seals. Although advantages were evident – lower leakage rates, extremely good performance at irregular surfaces, easy absorption of thermal motions and deflections – they were not sufficient to overcome the drawbacks, which also included price. \$7000 for a brush seal compared with \$650 for a labyrinth seal.

During the design of a jet engine, many factors have to be taken into consideration – factors that are not especially of interest for turbomachinery designers. Weight is the determining factor. It has to be kept in specific boundaries. If the weight is too high, all benefits coming from the sealing improvement will vanish due to higher power required to operate a heavier aircraft. This problem is addressed in **Chapter 3.4.3**, as well as in article by Gail and Beichl **[22]**. The size also matters. Some aircraft engines areas might be not available for brush seals if they will not be smaller – both in radial and in axial direction.

Unlike stationary turbomachinery, aircraft turbines undergo a varying operating cycle. A representative limited-life engine cycle has an initial maximum power condition (100 percent engine speed) for ten minutes, corresponding to launch at altitude. It is followed by a cruise power condition (85 percent engine speed) for 35 minutes, corresponding to cruise operation at sea level. For more details on limited-life engines operations see the article by Chupp and Dowler [13].

5 Operating Conditions of Brush Seals

The research to understand the brush seal behaviour and improve its parameters has intensified in the past years. It has all started from a pioneer article of Ferguson [21] who described the outstanding brush seal parameters. Since that time brush seals successively replace labyrinth seals (see **Chapter 3.6.1**). The most obvious reason for choosing brush seals is their leakage sealing performance. As the development continues, the operating conditions change. As seen in **Table 5.1** brush seal applications tend to be more demanding.

Date	$R_p[-]$	V_{s} [m/s]	T [K]	source
1988	?	300	923	[21]
1993	3.20	47.6	588	[13]
1993	6.98	?	91	[6]
1994	?	160	680	[27]
1996	4.66	?	527	[15]
1996	8.99	?	294	[15]
1997	?	24.0	923	[19]
1997	?	?	777.62	[30]
1997	?	?	293	[39]
1998	?	24	923	[20]
2002	?	<500	<1273	[17]
2003	?	400	973	[22]
2004	4.9	366	923	[32]

 Table 5.1 Brush seal parameters over the years

Due to the different goals expressed in the articles, not all parameters were available. The most interesting are pressure ratio, surface velocity and temperature. In some cases values extracted from the literature had to be converted to ones seen in the table. Pressure ratio values were possible to calculate using data available only in five articles. Still it is visible that from 1993 to 1996 the pressure sealing capabilities have increased. In [32] the goal was not to show the extreme operating conditions but to compare sealing technologies in common applications. That is why R_p is not the maximum that brush seal can withstand. The article by Carlile et al. [6] has almost twice as good pressure ratio value than [13]. It is because of an extreme brush seal application. It is used in a part of space shuttle engine where sealing capabilities are crucial. In this case liquid nitrogen was used as a flow medium. Also the temperature of brush operation is appropriate to the medium – minus 455 K.

Still, excluding such extreme applications, brush seals gradually increase its operating conditions. It does not mean that conditions described by Ferguson [21] do not exist. It means only that new applications for brush seal implementation are found. It is enough to see **Chapter 3.5** to understand that materials used for brush seal components can withstand severe conditions. Space industry has discovered brush seals outstanding sealing performance relatively earlier and used them for environments even ten years later considered as extreme. It is visible how versatile a brush seal system is.

6 Numerical Approach for Brush Seals

6.1 Full Computational Fluid Dynamics Calculation

It is the most complex way of simulating brush seals. Bristles under pressure load behave in various ways. They might concentrate in one area, while in another an interface might appear. Some regions of brush seals are characterised by river jetting, vertical or crossflow formations that exist upstream, downstream or within the seal. Effects like local recirculation, reverse and lateral flows between the rows of the brush, or downstream of the brush zone, appear to play major role in the sealing process.

Although the position of bristles in real life is random, flow analysis through a uniform column of bristles gives useful information – mainly about flow and pressure fields in the bristle pack in axial direction. That kind of analysis is two-dimensional and neglects the radial direction, which is important for blow-down, hang-up, hysteresis, flutter, etc. As mentioned in **Chapter 1**, wear occurring as bristle tips contact rotating parts is a particular problem. Understanding bristle behaviour in such situations is beneficial. 3D bending behaviour of bristles is important to understand.

Designing a 3D model of a bristle pack includes considerations not present in other approaches, like porous medium approach. An algorithm must be applied describing bristles behaviour and reaction to forces. Guardino and Chew [23] describe in general such an algorithm, used in SUBSIS (Surrey University Brush Seal Iterative Simulator) code, created for simulation of bristle behaviour:

- 1. Compute the total forces (i.e. aerodynamic plus reaction forces) on all the bristles and determine their deflections.
- 2. Calculate resulting deflection, orientation and deformed lay angles. Apply periodicity conditions.
- 3. Determine the required corrections to all bristle-bristle, rotor and backing ring reaction forces for all bristles.
- 4. Compute residuals and determine which reaction forces to adjust by comparing all the computed reactions forces throughout the brush seal.
- 5. Scan for and adjust the bristle pair with the greatest reaction force surfaceangle error (i.e. bristles for which the reaction forces do not point exactly normal to the corresponding bristle surfaces).
- 6. Apply Newton's third law.

7. Go back to Step 1 and repeat until either the maximum of all the residuals is smaller than a given tolerance, or until the calculated reaction forces and displacements have converged.

The complete algorithm and the procedures for determining which reaction forces to correct are presented in more detail in [24].

6.2 Bulk Flow Model

It is a semi-empirical approach based on results of crossflow through bristles. Bristles are represented as non-dimensional tube bundles. The model contains of effective brush thickness, which is defined as a measure of the compactness of the bristle pack, pressure drop in flow across disturbed tube bundles. The tubes can be packed hexagonally or randomly.

The method estimates leakages mainly as a function of seal geometry and operation parameters, and is useful for the initial design iteration.

6.3 Porous Medium Approach

In this approach the whole bristle pack is treated as porous medium with set flow resistance. In industrial processes the movement of gases and liquids through porous medium is very common:

- Distillation and absorption columns
- Filters
- Porous-media approximation to model pressure drop due to automobile radiator or a fan
- Flow distributors
- Packed beds

6.3.1 The Basis

The basic for porous medium investigation is an experimental law of Darcy. In 1856 Henry Darcy investigated flow of water in vertical homogenous sand filters, used for supplying water to fountains in French city of Dijon.



Fig. 6.1 Darcy's Experiment [3]

The conclusions from the experiment were that the rate of flow (volume per unit time) Q is

- proportional to the constant cross-sectional area A
- proportional to $(h_1 h_2)$
- inversely proportional to the length *L*.

The relations mentioned above are compiled in a Darcy formula,

$$Q = \frac{KA(h_1 - h_2)}{L}$$
(6.1)

The head is obtained by dividing each term of the Bernoulli Equation with the specific weight.

 $\gamma = \rho g \tag{6.2}$

Bernoulli Equation (constant along a stream line):

$$p + \frac{1}{2}\rho v^2 + \gamma h = const.$$
(6.3)

Bernoulli Equation transformed to express the head:

$$\frac{p}{\gamma} + \frac{v^2}{2g} + h = const. = H$$
(6.4)

The total head is constant along the streamline. It can be measured by the stagnation pressure using a pitot tube.

Piezometric head is the sum of pressure head $\frac{p}{\gamma}$ and elevation head *h*. The piezometric head in a flow can be measured through a flat opening parallel to the flow. It describes (in terms of head of water) the sum of pressure and potential energies of fluid per unit weight.

Therefore we can interpret $\frac{(h_1 - h_2)}{L}$ as hydraulic gradient $\frac{(h_1 - h_2)}{L} = J$ and $\frac{Q}{A}$ as specific discharge $\frac{Q}{A} = q$ (discharge per unit cross-sectional area normal to the flow direction). These substitutions allow representing Darcy's formula in another form:

 $q = KJ \tag{6.5}$

6.3.2 Mathematical Representation

Suppose in the domain of interest, Ω , exist two distinct regions: V and Ω_f . Their description is as follows:

Region	Description
V	Contains rigid, porous material saturated with viscous
	incompressible flow
$\Omega_{ m f}$	Contains only fluid

If two regions share a common, permeable interface, saturating fluid in V is identical to that in Ω_f . Otherwise, different fluids are possible in regions. If we assume that:

- Porous medium is homogenous
- Fluid and solid are in thermal equilibrium

The equations describing fluid motion and energy balance in region V are as follows. Within V, let V_f represent the volume occupied by the fluid and V_s represent volume occupied by solid, where

$$V = V_f + V_s \tag{6.6}$$

The porosity of porous medium is defined by:

$$\varepsilon = \frac{V_f}{V} \tag{6.7}$$

To define the porous flow equations, two averaged quantities are introduced:

(6.8)

$$\overline{a} = \frac{1}{V} \int_{V_c} a dV$$

and

$$\hat{a} = \frac{1}{V} \int_{V} a dV \tag{6.9}$$

where *a* is any quantity (scalar, vector or tensor). The parameters \overline{a} and \hat{a} are referred to as pore average and volume average, respectively, of the quantity *a*. They are related by the equation:

 $\overline{a} = \varepsilon \hat{a} \tag{6.10}$

In particular, the volume-averaged velocity \hat{u}_i of the fluid is

$$\hat{u}_i = \frac{1}{V} \int_V u_i dV \tag{6.11}$$

Similarly, the volume-averaged pressure and temperature are defined by

$$\hat{p} = \frac{1}{V} \int_{V} p dV \tag{6.12}$$

$$\hat{T} = \frac{1}{V} \int_{V} T dV$$
(6.13)

After performing the volume average of the momentum, mass and energy equations (and dropping the $^{\wedge}$ from the u_i , p, and T), yields:

Equation	Volume-Average Form	
Mass	$\frac{\partial \rho}{\partial t} + \left(\rho u_i\right)_{,i} = 0$	(6.14)
Species	$\rho \left(\frac{\partial c}{\partial t} + u_j c_{,j} \right) = \left[(\rho \alpha)_e c_{,j} \right]_{,j} + q_c + R$	(6.15)
Momentum	$\frac{\rho}{\varepsilon}\frac{\partial u_i}{\partial t} + \left(\frac{\rho \hat{c}}{\sqrt{\kappa_i}} \ u_i\ ^m + \frac{\mu}{\kappa_i}\right) u_i = -p_{,i} + \left[\overline{\mu} (u_{i,j} + u_{j,i})\right]_{,j} + \rho f_i$	(6.16)
Energy	$\left(\rho c_{p}\right)_{e}\frac{\partial T}{\partial t}+\rho c_{p}u_{j}T_{j}=\left(k_{e}T_{j}\right)_{j}+H$	(6.17)

Subscript *e* is related to fluid and solid matrix properties by the relations:

$$(\rho c_p)_e = \varepsilon \rho c_p + (1 - \varepsilon)(\rho c_p)_s$$
$$k_e = \varepsilon k + (1 - \varepsilon)k_s$$
$$(\rho \alpha)_e = \varepsilon \rho \alpha$$

Where the subscript *s* refers to solid matrix properties. Properties without the subscript are those of the fluid.

(6.14) to (6.17) represent generalisation of standard Darcy equations for nonisothermal flow in a saturated porous medium. System also referred to as Forchheimer-Brinkman model of porous flow.

By selecting coefficients of (6.14) to(6.16) results in any of several standard flow models:

 $\hat{c} = 0$ - Brinkman model $\overline{\mu} = \hat{c} = 0$ - Standard Darcy formulation

Porous medium flow equations are similar in form to the viscous fluid flow. The difference is that convection terms in the momentum equation are replaced by the Darcy-Forchheimer term in (6.14).

It is straightforward to include a porous medium and a general flow region in a given flow simulation (*V* regions may exist in conjunction with Ω_f). To maintain consistent boundary conditions at fluid and porous medium boundaries, the equations in the fluid and porous medium are all solved in terms of volume-averaged quantities.

6.3.3 Anisotropic Permeability

Standard porous medium equations treat the region as isotropic and homogenous. Isotropic medium is one whose certain property is independent on direction within the medium. Otherwise it is anisotropic. Homogenous medium is one when a certain property of medium is independent of position within the medium. Otherwise it is heterogeneous. Bristles in brush seal, represented as porous medium, are anisotropic. It is so because of the lay angle ϕ that ranges around 45°. Relation between specific discharge $q(q_1, q_2, q_3)$ and the gradient $J(J_1, J_2, J_3)$ in the general case of anisotropic medium, can be written as:

$$q_{x} = K_{xx}J_{x} + K_{xy}J_{y} + K_{xz}J_{z}$$

$$q_{y} = K_{yx}J_{x} + K_{yy}J_{y} + K_{yz}J_{z}$$

$$q_{z} = K_{zx}J_{x} + K_{zy}J_{y} + K_{zz}J_{z}$$
(6.18)

Or in more compact form:

$$q = KJ$$
 or $q_i = K_{ij}J_j$ $(i, j = 1, 2, 3 \text{ or } x, y, z)$ (6.19)

In (6.19) Einstein's summation convention (or double-index summation convection) is implied. According to this convection, in any product of terms, a suffix (subscript or superscript) repeated twice (and only twice) is held to be summed over its range of values. In this way the second equation of (6.19) should be understood as:

$$q_{1} = K_{11}J_{1} + K_{12}J_{2} + K_{13}J_{3}$$

$$q_{2} = K_{21}J_{1} + K_{22}J_{2} + K_{23}J_{3}$$

$$q_{3} = K_{31}J_{1} + K_{32}J_{2} + K_{33}J_{3}$$
(6.20)

The nine components K_{ij} in a three-dimensional space, or four in a two-dimensional space, define a viscous conductivity tensor. They can be written in a compact matrix form:

$$K = \begin{bmatrix} K_{11} & K_{12} & K_{13} \\ K_{21} & K_{22} & K_{23} \\ K_{31} & K_{32} & K_{33} \end{bmatrix} \text{ or } K = \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix}$$
(6.21)

In the case of brush seal, the tensor is supposed to represent the resistance in the bristles. The resistance force can be represented by the following equation:

$$\vec{F}_r = -A \cdot \eta \cdot \vec{v} - B \cdot \rho |v| \cdot \vec{v}$$
(6.22)

The viscous resistance tensor A is symmetric with principal axes in the directions normal to the bristles in the $r-\theta$ plane, parallel to the bristles and parallel to the axial directions. The basic form of that vector is as follows:

$$A = \begin{bmatrix} a_{rr} & a_{r\theta} & a_{rz} \\ a_{\theta r} & a_{\theta \theta} & a_{\theta z} \\ a_{zr} & a_{z\theta} & a_{zz} \end{bmatrix}$$
(6.23)

A pressure gradient aligned with any of the principal axes will produce motion in that direction only. In any anisotropic porous medium, the velocity may not be in the direction of the pressure gradient. Defining a_n , a_s and a_z to be resistance coefficients in the principal directions, the elements of A (denoted a_{ij}) have the following form in the natural cylindrical coordinate system (r, θ, z) :

$$A = \begin{bmatrix} a_{s} \sin^{2} \phi + a_{n} \cos^{2} \phi & (a_{n} - a_{s}) \sin \phi \cos \phi & 0\\ (a_{n} - a_{s}) \sin \phi \cos \phi & a_{s} \cos^{2} \phi + a_{n} \sin^{2} \phi & 0\\ 0 & 0 & a_{z} \end{bmatrix}$$
(6.24)

The assumption is that the inertial resistance tensor *B* has the same form as *A*. Therefore there are six resistance coefficients a_n , a_s , a_z , b_n , b_s and b_z to be defined. It is assumed that $a_n = a_z$, $b_n = b_z$ and that all coefficients are uniform through a bristle pack and do not vary with pressure difference across the seal. The influence of parameters on results is important to know. Mass flow rates are principally controlled by the axial resistance a_z and b_z . Backing ring (back plate) pressures depend mainly on the level of anisotropy. The following parameters, based on predictions from Chew and Lapworth [11] model were chosen:

$$a_z = a_n = 60a_s = 5,317 \times 10^{11} m^{-2},$$

 $b_z = b_n = 1,998 \times 10^6 m^{-1},$ $b_s = 0$

To incorporate the viscous resistance tensor A and inertial resistance tensor B into FIDAP calculations, the command of the following form had to be used (for tensor shown in (6.23):

$$\begin{aligned} &PERMEABILITY(SET = "a", ACOEF, CONSTANT = 1., X = a_{rr}, Y = a_{\theta\theta}, \\ &Z = a_{zz}, XY = a_{r\theta}, XZ = a_{rz}, YZ = a_{\thetaz}) \end{aligned}$$

The term *ACOEF* represents the coefficients of Darcy term. The second command is written with *BCOEF* term, representing coefficients of Forchheimer term. The exact code listing for the simulation is submitted in the **APPENDIX B**.

After calculating all tensor parameters, the following values were submitted to the program (here still in dimensional form, see **Chapter 7.2** for details): For *ACOEF* :

$$\widetilde{A} = \begin{bmatrix} 2.703 \times 10^{11} & 2.659 \times 10^{11} & 0\\ 2.659 \times 10^{11} & 2.703 \times 10^{11} & 0\\ 0 & 0 & 5.317 \times 10^{11} \end{bmatrix}$$

And for BCOEF:

$$\widetilde{B} = \begin{bmatrix} 9.990 \times 10^5 & 9.990 \times 10^5 & 0\\ 9.990 \times 10^5 & 9.990 \times 10^5 & 0\\ 0 & 0 & 1.998 \times 10^6 \end{bmatrix}$$

The resistance coefficients vary with bristle packing density. The following formulas that have been deduced for flow in packed beds are adapted to the brush seal problems:

$$-\frac{dp}{dz} = 5\alpha \frac{S^2}{\varepsilon^3} \mu V + \frac{\beta}{8} \frac{S}{\varepsilon^3} \rho V^2$$
(6.25)

Porosity ε is equal to the volume of voids divided by total volume. *S* is the wetted surface per unit volume and α and β are constants. The medium is treated here as an assembly of tubes.

In the case of brush seal with a lay angle ϕ to the tangential direction, *S* and ε may be expressed in the following way:

$$\varepsilon = 1 - \frac{V_{bristle}}{V_{solid}} = 1 - \frac{\pi d^2 N}{4B\sin\phi}$$
(6.26)

$$S = \frac{\pi dN}{l\sin\phi} = \frac{4(1-\varepsilon)}{d}$$
(6.27)

After algebraic manipulations of (6.25) to (6.27) and definitions of resistance coefficients *a* and *b* the following equations arise:

$$ad^{2} = \alpha \frac{80(1-\varepsilon)^{2}}{\varepsilon^{3}}, \quad bd = \frac{\beta}{2} \frac{(1-\varepsilon)}{\varepsilon^{3}}$$
 (6.28)

Where d is a bristle diameter.

Values of α and β were determined empirically by Kay and Nedderman (1974), and presented by Chew and Hogg [10] as satisfactory for a wide range of Reynolds number. Values obtained by using (6.28) give results varying from the assumed values of resistance coefficients. It is because experimental results are the basis for CFD models. Therefore, all coefficients are then adjusted to fit the experimental results as much as possible.

7 Numerical Simulation

The possibility of using commonly available CFD software for brush seal simulations was examined. The aim was to model the 2D brush seal environment with porous medium as bristles. Then conduct simulations and validate the model by comparing results with ones available in the literature. The software chosen for this purpose was FIDAP 8.7.0. It was developed by Fluid Dynamics International, a company later acquired by FLUENT. FIDAP is a flow-modeling tool with Finite Element Method based solver. The FEM method yields extremely accurate spatial resolution of flow details. Shape functions in the linear and quadratic elements give a description of the flow solution not only at the nodal points of the mesh but also at all locations inbetween. FEM-based solutions suffer very little numerical diffusion (the "smearing" of the solution when the flow is not aligned with the grid) [43]. The hardware used was Compaq AlphaServer DS10 with 600MHz processor and 1GB RAM.

7.1 Brush Seal Configuration

The mesh created to simulate the brush seal and duct in which it was placed, was based on Chew et al. [11]. The detailed dimensions are shown in **Fig. 7.1**.



Fig. 7.1 Schematic of brush seal (dimensions in meters)

All points used for creating the mesh are listed in **Appendix A**. Based on dimensions given a mesh was created. It consists of 23031 nodal points and 23460 elements.



Fig. 7.2 Mesh of brush seal system placed between the rotors surface and casing. Mesh was created in FIDAP

Mesh is denser in the brush seal area and near the walls. Also, the interface between the back plate and the rotor surface (also known as fence height) was meshed more densely **Fig. 7.3**. The code for mesh generation and setting all simulations values was written and is listed in **Appendix B**. It is more practical to use text mode instead of GUI, because of possibility of controlling all parameters. When using, GUI not everything is so clear and visible. It was also much easier to conduct series of simulations for different mass flow rates – the usefulness of knowing FIDAP commands in text mode showed its power during that process.



Fig. 7.3 Zoomed area of brush seal. Denser grid surrounding the seal and close to walls surfaces is visible

It is important to notice the fact that such a dense mesh was possible to create only due to non-dimensionalisation of the whole system. Distance between mostly dense packed nodes was so small that in dimensional system FIDAP generated errors – it treated such closely placed nodes as one node, making the mesh faulty (the units of such small distances reached the power of minus twelve). The process of non-dimensionalisation is explained in detail in the next chapter.

7.2 Description of Simulation

To simulate airflow through a bristle pack, porous medium approach was used (**Chapter 6.3**). The bristles were represented by an anisotropic permeability tensor. Bristles in brush seal, represented as porous medium, are anisotropic. It is so because of the lay angle that ranges around 45°. The aim of simulations was to compare the results with existing ones, available in the literature. The most common and significant ratio is mass flow versus pressure ratio. It is one of defining characteristics of brush seals and can be found in almost every source dealing with CFD simulations of brush seals. The simulations were based on different values of mass flow rate.

The gas used was air at temperature of 294 K. The following parameters were set for all simulations, in non-dimensional form:

Density	1
Viscosity	1/Re (outside bristles)
Viscosity	1/100*Re (inside bristles)

The nomenclature for all simulation sets is as follows:

brushx	– first set of simulations
brushxn	– second set of simulations
high_brushxn	- third set of simulations

where x = (1, 2, 3, 4)

Each set of simulations consisted of four single simulations conducted for four different values of mass flow rate:

 $\dot{m} = 0.005 \ kg/s$ $\dot{m} = 0.008 \ kg/s$ $\dot{m} = 0.010 \ kg/s$ $\dot{m} = 0.015 \ kg/s$

First set of simulations was brushx. The model was dimensional. In this set setting of mass flow rate was done in the following way: First, from equation

$$\dot{m} = \rho \cdot V \cdot A \tag{7.1}$$

the velocity value was calculated. That value was put into the simulation code as constant value c_1 . Unfortunately, the results, as well as convergence time were unsatisfactory. For brush1 it took 1315 iterations for solution to converge. Also the obtained values were different than those from the literature. The problem was solved by non-dimensionalising of all values. Each software has some limitations concerning the size of values. One must take into consideration, that during brush seal simulations

extremely small dimension values appear, in contrast to very high-pressure difference and airflow speed. Diameter of a single bristle is about $7.0 \cdot 10^{-6}$ m, compared to ca 0.007 m in length. The gap between shaft rig surface and bristle tips is also measured in fraction of millimeter. By non-dimensionalising the dimension values, all problems with values handling by the software were gone. Also the problems with mesh generation were gone. Before, creation of a fine mesh necessary for brush seal simulation was impossible due to extremely small grid size. After nondimensionalisation the effect was imminent. Result values started to reassemble those from literature, and convergence time for the solution dropped down dramatically. From 1315 iterations in brush1 calculations to 998 in brush1n. The iterations amount difference is tabulated in **Table 7.1**.

• m	Dimensional system	Non-dimensional system
0.005	1315	998
0.008	1433	993
0.010	2235	988
0.015	6336	974

Table 7.1 Iterations number for two types of models

Another change was in the way of obtaining different mass flow rates for simulations. Instead of calculating velocity from (7.1) the Reynolds number became the input factor, as seen in **Appendix B**. The following equation was used:

$$\operatorname{Re} = \frac{V \cdot h}{V}$$
(7.2)

As previously mentioned, all values are non-dimensional. They were calculated in the following way:

To obtain non-dimensional distance in x and y direction, one has to divide dimensional value by inlet height. Inlet height was chosen as a reference dimension. Inlet height is equal to 0.015345[m]. The transformation of dimensional distances into non-dimensional was done in the following way:

$$x^* = \frac{x}{h_1} \tag{7.3}$$

$$y^* = \frac{y}{h_1} \tag{7.4}$$

To obtain a non-dimensional viscosity value (calculated in the simulation code, see **Appendix B**) the following equation was used:

$$\mu^* = \frac{1}{\text{Re}} \tag{7.5}$$

To obtain non-dimensional resistance coefficients a and b:

$$a^* = a \cdot h_1^2 \tag{7.6}$$

$$b^* = b \cdot h_1 \tag{7.7}$$

The third set of simulations was calculated for different values of resistance coefficients. It simulated denser bristle packing. The values of non-dimensional resistance coefficients used in two series of simulations are tabulated in **Table 7.2**.

	Tenso	r matrix A	Tensor matrix B		
	Low density	High density	Low density	High density	
ZC	$1.252 \cdot 10^{8}$	$1.752 \cdot 10^{8}$	$3.066 \cdot 10^4$	$4,297 \cdot 10^4$	
RC	$6.365 \cdot 10^7$	$8.905 \cdot 10^{7}$	$1.533 \cdot 10^4$	$2,148 \cdot 10^4$	
THETA	$6.365 \cdot 10^7$	$8.905 \cdot 10^{7}$	$1.533 \cdot 10^4$	$2,148 \cdot 10^4$	
YZ	$6.155 \cdot 10^7$	$8.759 \cdot 10^{7}$	$1.533 \cdot 10^4$	$2,148 \cdot 10^4$	

Table 7.2 Values of non-dimensional resistance coefficients for two densities of bristle p	pack
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7.3 Results

The results obtained from the simulation are originally given in non-dimensional form. The necessity of using a non-dimensional system came up during the creation of the mesh. The number of intervals during simulation was set to 10000. It took approximately seven hours to converge. After changing all the values to non-dimensional, the time of single simulation decreased to about three hours. Apart from obtaining faster simulation times, the manipulation of results is simpler. It is often necessary to express results in different units than obtained during simulation.

7.3.1 Mass Flow Rate

The mass flow rate is a quantity characterising leakage of the brush seal. In **Fig. 7.4** a graph comparing values for different brush seal parameters is shown. The meaning of brushxn and high_brushxn is explained in **Chapter 7.2**. The values of Bayley and Long have been extracted from Chew et al. **[11]**. They are experimental data.



Fig. 7.4 Comparison of simulation results with literature data

It is visible that present CFD calculations match the experimental values of Bayley and Long [2]. The values from simulations high_brushxn seem to be the most accurate. It is visible that for lower pressure ratios the predicted results match the experimental ones. With the increment of pressure ratio, the measured flow rate exceeds the predicted value. Simulation high_brushxn represent brush seal with tighter bristle pack. Such tightening appears under increased pressure load. It implies that permeability coefficients are not constant during experiment. However, the present simulation is based on constant permeability coefficients. As seen, the use of the same set of permeability coefficients within considered range of pressure ratio gives a very good leakage agreement with the experiments.

The way of implementing mass flow values into the CFD code prepared for simulations is explained in **Chapter 7.2**. The way of presenting mass flow rate data versus pressure ratio was chosen in that way to obtain a graph of the same scale and with same units as it is found in the literature.

The way of obtaining pressure ratio is shown in (7.8). But it is written for dimensional values.

$$R_{p} = \frac{p_{u}}{p_{d}} = \frac{p_{d} + \overline{p}}{p_{d}} = 1 + \frac{\overline{p}}{p_{d}}$$
(7.8)

For non-dimensional quantities, the following equation as used:

$$R_p = 1 + \frac{\overline{p}^* \cdot \rho V^2}{p_d} \tag{7.9}$$

Where

$$\overline{p}^* = \frac{\overline{p}}{\rho V^2} \tag{7.10}$$

Concluding, the final equation used to obtain pressure ratio is:

$$R_p = 1 + \frac{\overline{p}^* \cdot \rho V^2}{1bar}$$
(7.11)

The values of pressure ratios versus mass flow rates for each simulations are tabulated in **Table 7.3**. Also values of mean non-dimensional pressure values are listed.

	Mean Pressure	Pressure Ratio	Mass Flow
brush1n	1.228E+05	1.580	0.005
brush2n	9.420E+04	2.140	0.008
brush3n	8.467E+04	2.601	0.010
brush4n	7.194E+04	4.060	0.015
high_brush1n	1.616E+05	1.764	0.005
high_brush2n	1.217E+05	2.473	0.008
high_brush3n	1.084E+05	3.049	0.010
high_brush4n	9.061E+04	4.854	0.015

Table 7.3 Simulation results needed for pressure ratio calculations

Values of mean pressures are extracted from post processor. Density is constant: $\rho = 1,21 \ kg/m^3$. Velocities are calculated from mass flow rate equation, see **Chapter 7.3**.

7.3.2 Axial Pressure Distribution

Axial pressure distribution on rotor surface is a second parameter that shows the results of CFD simulation in comparison with literature, experimental results presented by Bayley and Long [2]. The brush seal is subjected to high axial pressure load. The pressure drops down drastically over the bristle pack from upstream to downstream side (Fig. 7.5). The dimensionless pressure p_{nd} is plotted against axial coordinate z. The values of p were obtained from the following equation:

(7.12)

$$p_{nd} = \frac{p_a - p_d}{p_u - p_d}$$



Fig. 7.5 Comparison of axial pressure distribution on rotor surface

The experimental results by Bayley and Long [2] do not change significantly with pressure ratio. The gradient resulting from CFD simulation illustrates the linearity of pressure drop. The scatter is obviously not present in CFD results. The scatter observed by Bayley and Long [2] is a result of circumferential variations in the axial deflections of the brush seal. Other source, Braun et al.. (1990), not related to Bayley and Long [2] confirms this almost linear pressure gradient. CFD result roughly resembles the experimental results. It is also approximately 0.0004 m off from the trend line of experimental results. In CFD model bristle pack is assumed to remain in its original position withought any axial deflection during operation. In practical applications, bristles in fence height region are free to bend in axial direction depending on balance of aerodynamic, elastic, and frictional forces.

7.3.3 Radial Pressure Distribution

In **Fig. 7.6** radial pressure distributions are compared. The parameters used in this chart are dimensionless pressure distribution p_{nd} and normalised radial coordinate Y. This coordinate was calculated using following equation:

$$Y = \frac{y}{h_{bf}}$$
(7.13)

It is a non-dimensional value. Bristle free height is a distance of bristle from its tip to the bottom of front plate – part of bristle that is exposed to the stream withought any protection.

In experimental results little scatter is observed. The plots presented are for maximum and minimum values of pressure ratios used in present CFD simulations. It is visible that the most pressure drop occurs in the region of Y < 0.3 for CFD simulations. The real life experiment shows a significant drop from Y = 0.4. The pressure drops until the end of backing plate (region described as fence height, see **Chapter 8** for details). The fence height, in normalised radial coordinate, equals to $h_{fh} = 0.130841$. The dimensional value is equal to $h_{fh} = 0.0014$ m. The amount of experimental measure points is limited to five because of small dimensions of measuring area. Pressures were measured at four different circumferential locations.

Bristles under pressure load compress and form a very tight structure. The main leakage flow goes through fence height region. At the same time a strong inward radial flow develops in the upper region of the seal as the flow diffuses into voids among bristles **[18]**. The present CFD results are in a good agreement with experimental data. However, to simulate brush seal properly, the model requires adjustments. As mentioned before, this is not the aim of current work.



Fig. 7.6 Comparison of radial pressure distribution on backing plate

Both experimental and CFD results show that the pressure on backing plate for the upper region is almost constant and nearly equal to upstream pressure. The pressure drop occurs in the region of backing plate. Downstream pressure is reached there. Such a pressure distribution indicates a pressure gradient from the upper regions to fence height area directing the flow radially inward. The pressure gradient is shown in **Fig. 7.7**. In real life this pressure causes bristles to move towards the rotor surface, causing a blow-down effect.

7.3.4 More Details on Flow Behaviour

As mentioned in **Chapter 7.3.3**, a pressure gradient exists in fence height region. It is depicted in **Fig. 7.7**. It is visible that a pressure drop occurs directly in the bristle pack at the fence height region. The gradient is very high on the small distance.



Fig. 7.7 Pressure gradient in fence height region

The more general view of flow behaviour is shown in **Fig. 7.8**. It is a streamline distribution of air in the entire model. Lines of constant stream function ψ are streamlines of the flow. It means that they are everywhere tangential to the local velocity vector [34].



Fig. 7.8 Streamlines distribution in the model

As the stream enters seal area, the streamlines are altered. After leaving the fence region the stream velocity is highest right behind the backing plate. It is shown in more detail in **Fig. 7.14**.



Fig. 7.9 Velocity vector field in brush seal region with highlighted parts described in detail (general view)

In **Fig. 7.9** velocity vector field in the brush seal region is shown. The flow approaches the front part of the brush seal. When it gets nearer to the bristle pack it directs downwards to pass the fence height region. The bristles in fence height region block the flow. When the stream passes the bristle pack having lower pressure it extends to the down-stream cavity.

In **Fig. 7.10** a bristle pack region is shown. Porous medium region, simulating bristle pack, is highlighted with red frame. Theoretically the stream should significantly redirect its path when entering the bristle pack at the higher regions of the seal. The bristles reinforce that process – stream should advance almost parallel to the bristles in radial direction. In present CFD model it is not so strongly developed. However, stream changes its direction and close to the front part of backing plate it is practically parallel with the bristles in radial direction. Velocity is very low there, therefore this effect is hardly visible. The stream behaviour in the fence height region is illustrated in **Fig. 7.11**.

The stream accumulates at the backing plate corner and enters the cavity between backing plate and the rotor cover surface. The flow velocity reaches its maximum in both radial (**Fig. 7.12**) and in axial (**Fig. 7.13**) direction. In both cases flow velocities have positive values. The inward radial flow from the upper region joins with the axial flow in the fence height region and enters downstream cavity along the rotor surface. The downstream face of the bristle pack in fence height region is subjected to relatively higher axial velocities. This velocity tends to pull the bristles from the last columns of the pack towards the downstream part of the fence height region. Bristles tend to flutter because of this. Fluttering depends on balance of forces. A recirculation region is observed in literature, which is located under the backing plate. This effect was not observed in present CFD model.



Fig. 7.10 Velocity vector field with magnified bristle pack region (Part 1)



Fig. 7.11 Velocity vector field with visible behaviour in the fence height region (Part 2)



Fig. 7.12 Contour plot of radial velocity



Fig. 7.13 Contour plot of axial velocity

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Fig. 7.14 Velocity vector field. Zoomed area shows stream behaviour behind backing plate (Part 3)

The behaviour of stream behind the backing plate is visible in **Fig. 7.14**. Recirculation in that part of the channel is due to narrowing the channel's diameter by installing brush seal. Bristle pack itself may not have an influence on that effect. However, this effect can cause the stream recirculation under the backing plate. It is visible in the literature presentations of the models. The lack of that particular recirculation needs

further investigation. Its presence influences bristle behaviour. It may reinforce bristles from behind and prevent them from higher deflection. Still, none of the available literature sources deal with that problem in detail.

When describing velocities and pressure gradient on **Fig. 7.7** to **Fig. 7.14**, no values have been used. The aim is to visualise the flow behaviour in the brush seal. Detailed numerical values are post processed and included in **Chapters 7.3.1** to **7.3.3**.

8 Summary and Conclusions

Simulation results are comparable with experimental values found in [11]. Porous medium function found in FIDAP, after being adapted to anisotropic permeability conditions, works fine with bristle simulations. For denser bristle packing the results are more accurate when compared with experimental data. Experimental results were introduced only as a reference point. The idea was not to create the most accurate CFD model of a brush seal. The aim was to check the usability of commercial CFD software, withought altering its code. The model needs readjustments. Problems with lack of recirculation under backing plate need to be solved. Axial pressure distribution also needs further work. The linear pressure drop is too unrealistic. CFD simulations by Dougu [18] show results more resembling experimental results. Another problem is the porosity values inputted into the FIDAP code. It seems the value itself has completely no influence on model behaviour. Obviously, porosity value is already included in the tensor matrix. By changing tensor matrix values, the porosity of the system is also altered.

The fact is that bristle pack modelling is so unique that no software is specially adapted to it. There may appear problems due to the uniqueness of the application. Porous medium functions were created to simulate filters or soil areas (see Bear [3]). Bristle pack simulations are rather different then usual porous medium applications. Nevertheless, the results look promising. To create a CFD model that would replace the real life experiments, one simply needs a test stand to validate the model. It is crucial to copy the exact parameters of the system that will be simulated. Of course, such detailed data is not commonly available. It is a secret of manufacturers. Still, basing on experiments and simulations available in the literature it was possible to generate a basic porous medium model of the brush seal. Adjusting it to existing parameters is a different matter that needs further development and studies.

Sealing technologies undergo constant development and improvement. A seen in **Chapter 3.6**, the progress is unstoppable. On one hand labyrinth seals are shown as expiring technology. Efforts are made to replace these types of seals with more efficient ones. On the other hand, new technologies are being developed. Finger seals show somewhat better test results and seem to be more predictable than brush seals. It is a never-ending battle between technologies shortcomings and human needs. In light of recent fuel price fluctuations and inevitable reaching the end of fossil fuels deposits on Earth, the energy put into decreasing parasite leakages and therefore increasing turbomachines efficiency should never be underestimated.

Appendix A

Point #	Х	у	Point #	Х	у
1	-0.06	0.06088	11	0.0016	0.07158
2	-0.06	0.06228	12	0.00221	0.06088
3	-0.06	0.07158	13	0.00221	0.06228
4	-0.06	0.076225	14	0.00221	0.07158
5	0	0.06088	15	0.00381	0.06088
6	0	0.06228	16	0.00381	0.06228
7	0	0.07158	17	0.00381	0.076225
8	0	0.076225	18	0.06	0.06088
9	0.0016	0.06088	19	0.06	0.06228
10	0.0016	0.06228	20	0.06	0.076225

Listing of points used for mesh generation:

Appendix B

Code listing from brush2n.FDREAD file. It is a code written for brush seal simulation. All parameters stay unchanged except for Reynolds number.

```
TITLE
Brush-Seal, m_dot = 0.008 kg/s
11
FI-GEN( ELEM = 1, POIN = 1, CURV = 1, SURF = 1, NODE = 0, MEDG = 1, MLOO = 1,
MFAC = 1, BEDG = 1, SPAV = 1, MSHE = 1, MSOL = 1)
WINDOW(CHANGE= 1, MATRIX )
    1.000000
               0.00000
                            0.000000
                                        0.00000
    0.000000
               1.000000
                            0.000000
                                        0.000000
    0.00000
               0.000000
                            1.000000
                                        0.000000
    0.000000
              0.000000 0.000000
                                      1.000000
    -0.06300
                0.06300
                            -0.00914
                                         0.08536
                                                     -0.12600
                                                                  0.12600
             45.000000 45.000000
                                      45.000000
   45.000000
11
UTILITY ( TOLERANCE = 1.0E-6 )
11
//**Reynoldszahl:
11
$Re1 = 1016.0
11
//**Widerstandstensor A [-]:
11
       = 1.0/1.252E8
Śzca
     = 1.0/6.365E7
$rca
t = 1.0/6.365E7
     = 1.0/6.155E7
Śvza
11
//**Widerstandstensor B [-]:
11
       = 1.0/POW(3.066E4, 2.0)
$zcb
srcb = 1.0/POW(1.533E4, 2.0)
\theta = 1.0/POW(1.533E4, 2.0)
       = 1.0/POW(1.533E4, 2.0)
$yzb
11
//**Porositaet:
11
por = 1.0
11
//**Eingabe der Punkte:
11
POINT( ADD, COOR, SHOW, X = -2.000000, Y = 3.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = -2.000000, Y = 4.058651, Z=0.0 )
POINT( ADD, COOR, SHOW, X = -2.000000, Y = 4.664712, Z=0.0 )
POINT( ADD, COOR, SHOW, X = -2.000000, Y = 4.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.000000, Y = 3.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.000000, Y = 4.058651, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.000000, Y = 4.664712, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.000000, Y = 4.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.104268, Y = 3.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.104268, Y =
                                            4.058651, Z=0.0
11
POINT( ADD, COOR, SHOW, X = 0.104268, Y = 4.664712, Z=0.0 )
POINT( ADD, COOR, SHOW, X =
                             0.144021, Y =
                                            3.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.144021, Y = 4.058651, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.144021, Y = 4.664712, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.248289, Y = 3.967416, Z=0.0 )
11
POINT( ADD, COOR, SHOW, X = 0.248289, Y = 4.058651, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 0.248289, Y = 4.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 3.910068, Y = 3.967416, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 3.910068, Y = 4.058651, Z=0.0 )
POINT( ADD, COOR, SHOW, X = 3.910068, Y = 4.967416, Z=0.0 )
11
//**Definition der Linien:
11
POINT( SELE, ID, WIND = 1 )
```

```
5
 9
12
15
18
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
2
 б
10
13
16
19
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
 3
 7
11
14
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
 4
 8
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
17
20
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
 1
 2
 3
 4
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
 5
 б
 7
8
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
9
10
11
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
12
13
14
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
15
16
17
CURVE( ADD, LINE, SHOW )
POINT( SELE, ID, WIND = 1 )
18
19
20
CURVE( ADD, LINE, SHOW )
//
//**Definition der Meshedges:
11
CURVE( SELE, ID, WIND = 1 )
1
 б
11
14
MEDGE( ADD, SUCCESSIVE, INTE = 60, RATI = 0.9 )
CURVE( SELE, ID, WIND = 1 )
 2
 7
12
MEDGE( ADD, SUCCESSIVE, INTE = 40, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
CURVE( SELE, ID, WIND = 1 )
 3
 8
```

```
13
MEDGE( ADD, SUCCESSIVE, INTE = 30, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
CURVE( SELE, ID, WIND = 1 )
4
9
MEDGE( ADD, SUCCESSIVE, INTE = 40, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
CURVE( SELE, ID, WIND = 1 )
5
10
15
MEDGE( ADD, SUCCESSIVE, INTE = 50, RATI = 1.17 )
CURVE( SELE, ID, WIND = 1 )
16
19
22
24
26
28
MEDGE( ADD, SUCCESSIVE, INTE = 40, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
CURVE( SELE, ID, WIND = 1 )
17
20
23
25
MEDGE( ADD, SUCCESSIVE, INTE = 60, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
CURVE( SELE, ID, WIND = 1 )
18
21
MEDGE( ADD, SUCCESSIVE, INTE = 50, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
CURVE( SELE, ID, WIND = 1)
27
29
MEDGE( ADD, SUCCESSIVE, INTE = 60, RATI = 1.2, PCEN = 0.5, 2RATIO = 1.2 )
11
//**Definition der Meshfaces:
11
CURVE( SELE, ID, WIND = 1 )
1
19
 б
16
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
CURVE( SELE, ID, WIND = 1)
б
20
11
17
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
CURVE( SELE, ID, WIND = 1 )
11
21
14
18
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
CURVE( SELE, ID, WIND = 1 )
2
22
 7
19
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1)
CURVE( SELE, ID, WIND = 1 )
7
23
12
20
MFACE(WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1)
CURVE( SELE, ID, WIND = 1 )
3
24
8
22
MFACE (WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1)
CURVE( SELE, ID, WIND = 1 )
8
25
13
23
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
```

```
CURVE( SELE, ID, WIND = 1 )
 4
26
 9
24
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
CURVE( SELE, ID, WIND = 1 )
5
28
10
26
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
CURVE( SELE, ID, WIND = 1 )
10
29
15
27
MFACE( WIRE, EDG1 = 1, EDG2 = 1, EDG3 = 1, EDG4 = 1 )
11
//**Vernetzung:
11
ELEMENT( SETD, QUADRILATERAL, NODES = 4 )
11
MFACE( SELE, ID, WIND = 1 )
 1
 2
 3
 4
 5
 8
 9
10
MFACE( MESH, MAP, ENTI = "Fluid" )
MFACE( SELE, ID, WIND = 1 )
 б
 7
MFACE( MESH, MAP, ENTI = "Brush" )
11
ELEMENT( SETD, EDGE, NODES = 2 )
11
MEDGE( SELE, ID, WIND = 1 )
16
22
26
MEDGE( MESH, MAP, ENTI = "Eintritt" )
MEDGE( SELE, ID, WIND = 1)
21
29
MEDGE( MESH, MAP, ENTI = "Austritt" )
MEDGE( SELE, ID, WIND = 1 )
 1
 5
 8
11
13
MEDGE( MESH, MAP, ENTI = "Rotor" )
MEDGE( SELE, ID, WIND = 1 )
 4
27
 7
10
25
12
28
15
MEDGE( MESH, MAP, ENTI = "Stator" )
11
END ( )
11
FIPREP
11
PROBLEM ( CYLINDRICAL, STEADY, TURBULENT, NONLINEAR )
11
EXECUTION ( NEWJOB )
11
ENTITY ( FLUID, NAME = "Fluid", MVISC = "Fluid" )
ENTITY ( POROUS, NAME = "Brush", MVISC = "Brush", MAPERM = "A", MBPERM = "B" )
ENTITY ( WALL, NAME = "Rotor" )
```

```
ENTITY ( WALL, NAME = "Stator" )
               NAME = "Eintritt" )
NAME = "Austritt" )
ENTITY ( PLOT,
ENTITY ( PLOT,
1
DENSITY ( CONSTANT = 1.0 )
11
VISCOSITY ( SET = "Fluid", CONSTANT = 1.0/$Re1, MIXLENGTH = 0.091235/2.0, CLIP = 1.0E6 )
11
VISCOSITY ( SET = "Brush", CONSTANT = 1.0/(100.0*$Re1) )
11
PERMEABILITY ( SET = "A", ACOEF, CONSTANT = 100.0, ZC = $zca, RC = $rca, THETA = $thetaa, YZ =
$yza, POROSITY = $por )
11
PERMEABILITY ( SET = "B", BCOEF, CONSTANT = 1.0, ZC = $zcb, RC = $rcb, THETA = $thetab, YZ =
$yzb, POROSITY = $por, POWER = 2.0 )
11
PRESSURE ( PENALTY = 1.0E-9, DISCONTINUOUS )
//PRESSURE ( MIXED = 1.0E-6, CONTINUOUS )
11
SOLUTION ( SEGREGATED = 10000, VELCONV = 0.0001, SCHANGE = 0.0 )
11
RELAXATION
0.3 0.3 0.3 0.3 0 0 0.3
                                     0.3
11
OPTIONS ( UPWINDING )
11
UPWINDING ( STREAMLINE )
11
EXTRAPOLATE ( OFF )
11
ICNODE ( UX, CONSTANT = 1.0, ENTITY = "Fluid" )
11
BCNODE ( UX, CONSTANT = 1.0, ENTITY = "Eintritt" )
11
BCNODE ( VELOCITY, ZERO, ENTITY = "Rotor" )
11
BCNODE ( VELOCITY, ZERO, ENTITY = "Stator" )
11
RENUMBER (PROFILE)
11
END
CREATE(FISOLV)
```

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