Maintenance free gas bearing helium blower for nuclear plant

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Abstract. This paper describes the design, testing and operation of novel helium blowers used to recirculate the helium blanketing gas in the nuclear reactor used as a neutron source at the Institut Laue Langevan, Grenoble, France. The laser sintered shrouded centrifugal wheel operates at speeds up to 45000 rpm supported on helium lubricated hydrodynamic spiral groove bearings, and is driven by a sensorless permanent magnet motor. The entire machine is designed to keep the helium gas (polluted by a small amount of D2O) out of contact with any iron or copper materials which would contribute to the corrosion of parts of the circuit. It is designed to have zero maintenance during a lifetime of 40,000 hours of continuous operation. This paper will describe the spiral groove journal and thrust bearings. Design and manufacture of the 1 kW motor and centrifugal wheel will be explained including their CFD and FEA analyses. Measurements of rotor displacement will be presented showing the behaviour under factory testing as well as details of the measured centrifugal wheel and motor performances. Two machines are incorporated into the circuit to provide redundancy and the first blower has been in continuous operation since Jan 2015. The blower was designed, manufactured, assembled and tested in the UK using predominantly UK suppliers.

1. Background
The nuclear plant in Institut Laue Langevan, Grenoble is a small heavy water cooled high flux reactor of only 58 MW thermal power used to produce neutrons for research purposes [1]. It operates 220 days a year producing neutrons for research focused primarily on fundamental science in a variety of fields: condensed matter physics, chemistry, biology, nuclear physics and materials science.

The plant has been in operation since 1971 and is due to continue operating for at least another 10 years which may be extended depending on demand. The heavy water is covered (blanketed) by a layer of Helium gas that has to be continuously cooled and treated to remove any heavy water ions. This helium gas must not contain any copper or iron ions as their presence would contribute to corrosion of parts of the reactor vessel, nor is it acceptable to have oil contaminating the circuit which, amongst other issues, would foul the catalytic converters and heat exchangers.

The original helium circulators were manufactured in the early 1970s but were overhauled 15 years ago, with the replacement of the inverter drives. A recent review of safety in French nuclear industry resulted in a requirement to provide modern helium blowers to replace the original ones – even though these latter were still functioning perfectly well on the original gas bearings. The original gas bearings, supporting a regenerative blower, were of the spring preloaded tilting pad type that were common in many nuclear circulating pumps and blowers in the 1960s and 70s. Decker [2] reviews 22 different installations in USA, Germany and England.
OFTTech Ltd are specialists in designing and using spiral groove gas bearings which have many advantages compared to tilting pad ones.

The basic requirement of the blower is as follows:
- Flow rate: 20 m³/hr
- Pressure rise 30 mbar
- Electrical power input 1 kW

The flow and pressure requirement only require tens of Watts but there is a client requirement that the motor be a 1 kW one for safety considerations. The final design is shown in figure 1. The helium inlet and outlet ports are at the top and an air cooling jacket provides cooling of the stator winding.

2. Blower wheel

The maximum rotational speed of the blower was fixed at 40000rpm to enable a design that could be easily assembled and replaced in service. This leads to a low specific speed (0.21 Balje’s definition) and hence a large diameter radial impeller with small blade heights – the impeller diameter is 110mm with blade height varying from 3mm down to 1.2mm. At this low specific speed the fluid friction and disc windage losses are high, leading to relatively low efficiency – however, the shaft power required at design point was only of the order of 50W and, given the 1kW motor capability, high efficiency was not a prime requirement.

The aerodynamic design of the impeller was carried out iteratively using Ansys-CFX with the turbomachinery enhancements BladeGen and TurboGrid. The design philosophy was to employ highly backswept blades (outlet angle -65° from the radial direction) in order to achieve a very stable operating characteristic. The general view of the impeller and design point velocity triangles are given in figure 2.
A shrouded design was employed primarily to minimise the axial thrust of the impeller, and an axial labyrinth seal employed at the inlet to control the leakage flow. Figures 3 and 4 show predicted pressure contours and blade loading at design point – the loading diagram shows the smooth velocity distribution obtained, with minimal over-acceleration and modest diffusion levels. For simplicity, the splitter vanes were specified exactly halfway between the main blades and with the same blade angle distribution – this resulted in lower loading on the splitter, as indicated by the two innermost lines in figure 4. The final design had nine full and nine splitter blades.

Initial studies using simple one dimensional methods suggested that the addition of a volute at the impeller exit would be of marginal benefit as the friction losses in the small passage would cancel out any static pressure recovery due to flow deceleration. Therefore the design was specified with no diffusion system – this also enabled the use of the existing inlet and outlet port positions so that the machine could comply with the need to fit to existing pipe connections.
A full performance characteristic was predicted using CFX, and the results corrected for disc windage and labyrinth seal leakage (figure 5). It can be seen that the design was predicted to achieve higher than target pressure rise at 40000rpm – this was deliberate given the nature of the development program and some of the uncertainties in achievable clearances and roughness effects. It can also be seen that converged CFD solutions were obtained at very low flow rates, which suggested that the required stable operating characteristic should be achieved. It can be seen that the predicted characteristic indicated that the target flow and pressure rise should be achieved at 37000rpm, which gave a useful speed margin.

The mechanical design of the impeller was carried out alongside the aerodynamic design. Stainless steel 316L was specified by the customer. The disc taper and shroud geometry was carefully optimised to minimise deflections and stress levels, and to ensure that disc natural frequencies were well above the running speeds. As the design featured a shroud, it was decided to adopt Direct Metal Laser Sintering (DMLS) to ‘print’ the impeller and hence achieve a shrouded design without the need for joining. One downside of using this process is the difficulty of achieving smooth internal surfaces. However, the passage Reynolds Number (based on passage hydraulic diameter and relative velocity) is between 3000 (inlet) and 1200 (exit), so the flow is arguably laminar, and roughness effects should be small. It was therefore decided to not finish the internal passages (by abrasive flow machining) unless testing indicated a performance shortfall.

The final peak stress levels in the impeller were at the junction of the blades and the inner diameter of the shroud (figure 6). Due to the lack of previous experience of using the DMLS process the design was conservatively engineered to have a peak stress at 40000rpm of just 65% of the manufacturer’s quoted yield stress.

![Figure 6. Peak Stress Levels at Blade / Shroud Intersection](image)

**3. Permanent Magnet Motor**

A permanent magnet synchronous motor topology was chosen. To ensure that the helium is not contaminated by iron or copper from the motor active components, a 0.2 mm thick stainless steel (316L) can was used to enclose the stator. The canned stator was potted in epoxy inside a finned stainless steel housing. The magnets were glued inside the tungsten carbide shaft tube; an ASI4010 (EN19) bolt passes through the centre of the magnets.
A KEB (KEB Automation KG) sensorless control PWM drive is used to drive the motor. A sine filter is used to reduce current ripple and associated losses in the machine. It also helps with improving EMC (ElectroMagnetic Compatibility)

Placing the magnets inside a tungsten carbide tube, and using a conducting stator can made the design of the machine challenging. The former meant the effective airgap of the machine was relatively large: the magnet OD is 20 mm and the effective airgap is 6.5 mm (tungsten carbide has the same permeability as air and hence the effective airgap includes both the clearance gap and the tungsten carbide shaft tube). As a result, there is a significant amount of leakage flux as shown in Figure 8, which shows flux distribution obtained using a finite element analysis model of half the machine.

Figure 7. Finite Element 3D Model

Figure 8. Flux distribution at no-load.

Transient three-dimensional FEA including the PWM drive circuit (see Figure 8) and rotation was used to analyse the machine, estimate losses and quantify its performance. To reduce core losses, the flux density in the laminations was kept well below saturation, at about 1T. This resulted in core losses of about 30W. The copper losses are about 3.5 W. However, these are dwarfed by losses in the can which needs to be as thin as possible as the losses are proportional to its thickness. The eddy current losses in the 0.2 mm can used in the final design are about 130 W (Figure S3). Manufacturing and fitting such a thin can is challenging. Care needed to be taken that the can does not short circuit the laminations and that the gaps between the can and the teeth are filled with thermally conducting epoxy to ensure heat removal.
The Synthet-PM propriety software [3] was used to conduct the initial motor sizing studies as well as carrying out thermal analysis of the final machine design (using FEA loss results). The thermal analysis estimated that the maximum winding temperature is about 140 degrees C, which means that using class H insulation (180 degrees rating) will have a theoretical life of about 320,000 hours, which is well above the target.

Figure 9. External circuit of the PWM drive used in the 3D FEA model

![Diagram](image_url)

Figure 10. Core and can losses calculated using FEA.

4. Gas Bearings

Spiral groove gas bearings are one of the types of aerodynamic (hydrodynamic, self acting) bearings that has proven successful in the market place. The others being foil bearings (which have lower radial
stiffness and wear/fatigue issues) and spring loaded tilting pad bearings (require individual tuning and often have higher starting torque and poor low speed performance). OFTTech has concentrated on spiral groove bearings since the early 1980s. [4].

The key feature of spiral groove bearings are the shallow grooves (of the order of 2 to 4 times greater than the clearance) that create the pressures inside the bearing due to rotation of one of the surfaces (either surface can rotate, it’s the relative motion that is needed). The spiral groove thrust bearing is thought to have been invented in the UK during the 1940s but was classified by the AERE [5] for many years.

Figure 11. Journal and thrust bearing bearing grooves.

The two patterns may or may not meet in the middle depending on the optimal performance in the application. These grooves are sometimes called herringbone groove bearings. The right-hand image shows thrust bearing grooves for an inward pumping thrust bearing. Similarly, as for the journal bearings a second set of grooves can be added on the inside that acts to limit or prevent through flow – the choice comes down to the application requirements.

These bearings are usually made of hard materials to ensure low wear and good structural performance in order to keep the surfaces parallel under the small clearances usually necessary for gas bearings. Tungsten carbide and silicon nitride are two of the most common materials now used, boron carbide was the primary choice for the successful navigation gyroscopes used for many years [6]. Clearances are typically of the order of 5 to 15 µm radially, with grooves 10 to 40 µm. The use of cylinders and flat disks makes the manufacture easier than if the complex conical or hemispherical forms are used. Modern grinding techniques allows direct grinding to size although for prototypes the classic finish lapping method can be used.

Surface coatings are often used but only two have proven successful: MoS2 (sputtered) and DLC (amorphous carbon); both with thicknesses of 2-4 µm applied after final sizing providing low friction during start up and overload.

The grooves can be made by many methods: photo etching, grinding, ECM, EDM, ION beam, pressing, grit blasting or by laser. The latter is the only solution we now use as the groove geometry is excellent and relatively low cost.

Recent examples of the use of OFTTech spiral groove gas bearings are in laser blowers [7] and a Brayton cycle cryogenic space cooler [8], which has been in operation on spiral groove gas bearings since 2002: and a new application will involve a turbo expander cryocooler on Neon lubricated spiral groove bearings [9].

The design of aerodynamic gas journal bearings is much more difficult than bearings lubricated with liquids because the compressibility of the gas makes the bearing behaviour frequency dependant. This means the designer has to examine all ranges of operating speeds and frequencies of excitation to ensure the bearings are stable under any excitation. The usual form of instability seen with gas bearings is half speed whirl seen at frequencies between 20 and 45% of the synchronous running speed, and is usually fatal with the bearing crashing within seconds. The parameter usually used to
describe the stability of gas journal bearings is the critical mass [10], defined simply as that mass the bearing will support before experiencing half speed whirl. In addition to the gas bearing stability we have to examine the rotodynamic behaviour to check system stability. The analysis of gas bearing behaviour was performed using a numerical method following the narrow groove theory [11, 12]; this design method has been under continual improvement and checking at OFT Tech since its origins at the University of Southampton [13].

The journal bearing performance was optimised using a simple hill climbing procedure (although a genetic algorithm is available for simply designs) resulting a journal bearings with a 30 mm diameter, 30 mm long and grooves 17 \( \mu \)m deep.

The basic performance of the journal bearings is shown in figure 12 showing the stability (critical mass, kg), power (Watts), stiffness (steady state, MN/m) and damping (synchronous speed, Ns/m) with variation of radial clearance. Here one sees an optimal clearance around 9.5 \( \mu \)m radially. Increasing the clearance to 11 \( \mu \)m reduces the stability by a factor over 100: but even with this larger clearance the stability margin is still ~5 (critical mass/actual rotor mass per bearing).

![Figure 12: Journal bearing theoretical performance (40,000 rpm, helium lubricant at 1 bar, groove depth 17 \( \mu \)m).](image)

\( Krr \) = resulting radial stiffness at steady state conditions.
\( Crr \) = resulting damping at the bearing natural frequency.

The thrust bearing has also been optimised with the result shown in figures 13 and 14. Figure 13 shows the variation of thrust bearing performance with speed and figure 14 shows the force generated with variable clearance. The result is that the thrust bearing will operate at an axial clearance of ~24 \( \mu \)m. when supporting the shaft mass.
Figure 13 Thrust bearing theoretical performance (40,000 rpm, helium lubricant at 1 bar, groove depth 34 µm). Kzz = axial stiffness at steady state conditions. Clearance = axial lift under shaft mass, µm

Figure 14: thrust bearing theoretical axial force capability (40,000 rpm, helium lubricant at 1 bar, groove depth 34 µm)
Figure 15 shows the internal view of the assembly showing the two journal bearings and single sided thrust bearing at the top under the wheel. The blower is mounted vertically such that a single sided thrust bearing can be used.

- a) Blower wheel
- b) Thrust bearing
- c) Motor stator and rotor
- d) Cooling fins
- e) Lower journal bearing

The design is such that all moving parts can easily be replaced from the top. The motor is air cooled with fins being machined into the housing. There are no welds on this design avoiding any need for Non Destructive Testing and has been designed according to the European pressure vessel design standard EN13445-3:2009 including a pressure test to 5 bar which is a factor of safety in excess of 100 compared to the system nominal pressure.

There are no parts requiring maintenance or service and it is expected that the units will run for the 40,000 hrs design life unless contaminated (for example water or other fluids).

5. Test Results

The rotordynamic analyses undertaken shows that the system is stable for all operating speeds. Figure 16 shows the first critical speed at 7762 rpm with the wheel motion being the primary action.

Figure 15. The internal layout of the helium blower

Figure 16. Theoretical calculation of the critical speed – the conical rigid mode at 7762 rpm.
The most important behaviour to consider to enable correct balancing is the response to out of balance. Figure 17 shows the response to 2 gmm on the wheel and 2 gmm on the other end of the shaft. As can be seen the peak displacements are only 3 µm – much less than the bearing radial clearance.

![Response to Out of Balance](image)

Figure 17. Response to Out Of Balance (OOB). The theoretical predictions use an OOB of 2 gmm at each end of the shaft. The rotor was balanced to <2 gmm at each end. The displacements, both theoretical and experimental, are RMS values.

The grooves are manufactured using laser machining and results in excellent quality grooves with uniform depth, accurate angle and width ratio. The profile is shown in figure 18. The sloping sides of the grooves are intentional to ensure correct laser machining.

![Profile across the thrust bearing grooves](image)

Figure 18. Profile across the thrust bearing grooves machined by laser. Axes are mm. The mean groove depth is 33.5 µm with tiny variations.
The final performance of the blower was tested by constructing a closed-loop helium-filled circuit with appropriate valves, orifice plate and coolers. Full characteristics were obtained right down to zero flow, with no evidence of increased vibration levels due to stall / surge. Figure 19 shows the test results at 37000rpm for the two delivered units, overlaid with the CFD-derived prediction. It can be seen that the agreement with prediction is good at design flow, but the tests indicate a slightly steeper slope to the characteristic.

![Figure 19. Test Results showing the predicted and experimental performance of the blower wheel (test at 37,000 rpm, 1 bar pressure, helium lubricant).](image)

6. Concluding Remarks
A helium circulator running at 40,000 rpm on spiral groove helium lubricated gas bearings has been in continual operation for over a year with no downtime and no detectable increase of vibration or power required to drive it.

The spiral groove gas bearings have proven to be robust and reliable and the tested performance matches well with the computer predictions.

A permanent magnet motor mounted inside the tungsten carbide gas bearing shaft with a stainless steel can to isolate the motor winding from the helium has shown performance close to the prediction and temperatures measured near the can correspond well with FEA prediction.

The laser sintered shrouded blower wheel has demonstrated performance within a few% of the CFD predictions.

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