



Internal pump method for convective cooling of the rotor of a superconducting motor

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ABSTRACT

Electrification has been proposed as a route to decarbonizing air travel. Conventional electric motors are too heavy to achieve the power densities required for aviation so superconducting motors for aircraft are being developed. Superconductors however, work at cryogenic temperatures, which indispensably require a reliable and efficient cooling mechanism. The Robinson Research Institute is developing a 3 MW superconducting motor using an HTS (High Temperature Superconductor) rotor operating at 50 K. The motor is intended to run at a shaft speed of 4500–6000 rpm to directly drive a ducted fan. A key challenge is the removal of heat from a cryogenic spinning rotor to a stationary refrigerator so it can be rejected at ambient temperatures. Previous motors have utilized externally pumped or thermo-syphoned cryogens to cool the rotor, which require complex rotary seals and a heavy external cold box for the refrigerator. The aircraft application demands a compact and low weight solution. In a synchronous superconducting motor, a popular configuration is to have the rotor with direct current (DC) field coils and the stator with alternating current (AC) coils. This configuration significantly reduces cooling load on the rotor as DC superconducting coils have few losses. Most of the heat to be lifted to ambient from the rotor comes from conduction down the shaft(s), especially if the DC coils are energised with a flux pump. However, the rotor is rotating, so the heat needs to be transferred across an interface to the non-rotating (stationary) world. This work proposes using a novel concept using a stationary cryocooler's cold heat exchanger as part of a pump to circulate gaseous helium inside the rotor and to use the rotor's spinning action to perform the pumping. In this way the rotating interface for heat transfer becomes the pump. A rotating gas seal is still necessary, but this can be at near ambient temperature and pressure. Computational fluid dynamics (CFD) analysis has indicated that the system will work with acceptable losses and requires less than 20 W of cooling to keep a motor's rotor at 50 K. This paper presents the CFD modelling, results of the proof-of-concept experiments that validated the CFD model, and will present further improvements to the concept, demonstrating a feasible cooling method and its application to a superconducting rotor.

1. Introduction

1.1. Electric aviation

Electrification of air travel is a proposed route for reducing carbon emissions for aviation, which in 2019 amounted to 3% of the world's CO₂ emissions [1]. Small aircraft have been able to use batteries and permanent magnet motors but for large aircraft the power density is too

high for current battery technology, and liquid hydrogen has been proposed as an energy carrier. The hydrogen would either be used in a gas turbine driven generator or converted to electricity with a fuel cell. The electric drive train offers the possibility of greater aerodynamic efficiency, up to 20%, via distributed propulsion [2].

An electric motor for aviation would require a very high power-density to compete with gas turbines. Conventional electric motors are too heavy to achieve the power densities required for aviation, so

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superconducting motors have been proposed as they are light and compact [3]. Superconductors however, work at cryogenic temperatures, which indispensably require a light, reliable and efficient cooling mechanism to be implemented on an aircraft. A key challenge is the removal of heat from a spinning cryogenic rotor to a refrigerator so it can be rejected at ambient temperatures. Previous superconducting motors have utilized cryogenic coolants in the form of pumped gaseous helium [4–7] or thermo-syphoned cryogenics [8,9]. Transferring a cryogen to a spinning rotor requires rotary seals, usually ferro-fluidic seals [8] as they have a very low leak rate. The aircraft application demands a robust, compact, and low weight solution. Moreover, aircraft fans run at high shaft speeds. Ferrofluidic seals are not compact and generate significant amounts of heat at high shaft speeds, needing additional cooling and thermal isolation from the cryogenic environment. Ferrofluidic seals can only seal a pressure difference of up to 1 bar, so sealing a high-pressure cryogenic fluid, for example helium gas at 30 K and 10–20 bar, would require many stages of sealing, making the motor too large and heavy for aviation. For example, the helium gas supply for a set of Gifford McMahon cryocoolers on a wind turbine generator [10] had 132 stages of ferrofluidic seals for a 16–18 bar pressure differential.

Jeong [11] and Dyson [12] proposed rotating cryocoolers for cooling the rotor. Jeong proved experimentally that spinning about its axis did not affect the performance of a pulse tube. However, potential difficulties with spinning the cryocooler involve the addition of a large spinning mass to the rotor, transferring power to the cryocooler, and removing the heat from the cryocooler. The advantage of rotating the cryocooler is that heat transfer across the rotating/stationary interface would happen at an ambient temperature, the disadvantage is that the amount of heat to be transferred across the rotary interface is much greater as it is in the order of 1 kW rather than the 20 W at the cold head.

This work proposes using a novel concept using a stationary cryocooler's cold heat exchanger as part of a pump to circulate gaseous helium inside the rotor and to use the rotor's spinning action to perform the pumping.

1.2. A superconducting aircraft motor

The Robinson Research Institute (RRI) is developing a 3 MW superconducting motor using an HTS (High Temperature Superconductor) rotor operating at 50 K with the AC stator coils being either cooled copper, MgB_2 running at near 20 K, or HTS coils operating at around 35 K. The motor is intended to run at shaft speeds of 4500–6000 rpm to directly drive a ducted fan as an aircraft propulsor. RRI will first build a 100 kW prototype to test key aspects of the concept, being the power electronics, the rotor and its cooling, and the stator with its cooling. The 100 kW prototype design has a rotor that is 200 mm in diameter and 400 mm long, the same physical size as the planned 3 MW motor design. RRI's calculations for both the 100 kW and 3 MW motors are that the rotor will nominally require 20–50 W cooling at 50 K. It is envisioned that the aircraft will be fuelled by liquid hydrogen, providing a large heat sink for the stator and rotor cooling, but there may be intermediate steps that involve a non-superconducting stator in which case mechanical refrigeration, a cryocooler, would be required for the rotor. The heat loads on the rotor come from a combination of heat sources, which were initially calculated and are listed in Table 1. The outer cylinder of the rotating cryostat was designed to function as a shield for the magnetic field harmonics from the stator, thus dissipating the energy at ambient temperature instead of adding to the heat load on the rotor.

1.3. Modes of operation

This paper considers two modes of operation for the superconducting motor's rotor. These are referred to in this paper as rotating and stationary.

When the motor is in use, driving the aircraft's propulsion, the rotor

Table 1

Sources of heat in the rotor and initial estimates of their values.

Heat source	Approximate magnitude	Mechanism and comment
Drive shaft conduction	15 W	Calculation of conduction for a nominal shaft between 300 K and 50 K is 7.5 W each side
Radiation and insulation losses	3.4 W	Bare vacuum radiation assumed for a 200 mm diameter cylinder 400 mm long between 300 and 50 K, polished metal surfaces combined emissivity 0.03.
Coils	3.75 W	Estimated Joule heating of HTS joints, from the coils and flux pump design by RRI.
Coolant circulation	0.5 W	Estimated viscous losses from the helium circulation internal to the rotor. A conservative estimate based on twice the velocity head and flow rate available from the pumping action.
Total	22.65 W	

is rotating. In the rotating mode the motor is producing shaft power through the rotor's drive shaft to a fan or propeller. Inside the rotor, the rotor will be spinning relative to the stationary cold heat exchanger producing the cooling flow. The superconductor coils will be energised with a direct current (DC) so will have very low losses. Most of the heat from the coils will be the small amount from the solder joints to the flux pump.

The other important mode of operation is when the motor is not spinning, and the rotor is stationary. For the motor to start, the superconducting coils must be charged with a current via the flux pump and therefore need to be cold enough to sustain superconductivity before the flux pump is activated. For aircraft that do multiple flights per day, the motors will need to be kept cold on standby, with the coils energised, between flights.

1.4. The proposed solution

The proposed concept uses advection to transport the heat from its sources around the rotor to the cold heat exchanger of a cryocooler. For the purposes of this study, a shortened rotor was used for the prototype to prove the pump concept. The rotor maintains the 100 kW motor's diameters and end details to test the pumping and cryocooler heat transfer. The concept is shown in Fig. 1. There is a stationary (non-rotating) cryocooler internal on the left-hand side of the rotor. The cryocooler's warm end is on the left and cold end to the right. The cryocooler is in helium gas which is around 0.2 bar above atmospheric pressure. Lip seals separate the gas from the atmosphere and bearing. The cold end of the cryocooler is attached to a heat exchanger with fins and channels for flow and heat exchange. The 3D printed rotor has cooling channels inside it that direct the cold gas to flow around the rotor (orange arrows), from the centre left-to-right, through the pump impeller vanes on the right then out to the circumference and back towards the left, then to the centre through the cold heat exchanger. The vacuum space provides insulation, and an axial flexure is used to minimise stress from contraction of the cold parts relative to the warm cryostat.

The difference in motion between the stationary cold heat exchanger and the rotor's impeller is used to form the two sides of a pump, with the rotor imparting energy to the flow and the heat exchanger blades converting the flow's rotational velocity to pressure at the centre. The pressure differential drives a flow that can transport heat from around the rotor to the cold heat exchanger.

Two more assumptions of the design are required to make it work. Firstly, the heat leak down an annulus of stationary helium gas adjacent to the cryocooler's cold finger is low, and secondly if the circulation gas is kept slightly above ambient pressure, then simple and compact seals

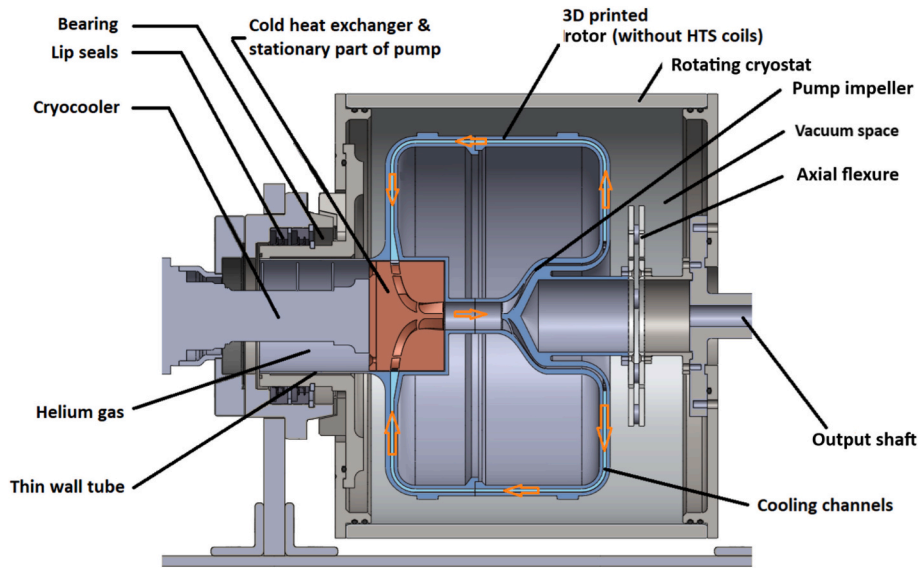


Fig. 1. Proposed cooling system for a rotor, shown without the coils in place. The rotor has been shortened from the 100 kW motor design for test purposes, the diameter is the same as the 3 MW and 100 kW designs.

can be used with very little leakage and heat generation.

An additional benefit of the concept is that, if a cryogen flow is available on the motor, from cooling the stator coils for example, then the cryocooler could be replaced with a stationary heat exchanger cooled by a high-pressure cryogen flow. The advantage of a cryogen-fed heat exchanger is that it would be lighter than a cryocooler and would not have the moving parts of the cryocooler. This system would allow use of a high-pressure helium circuit to transport the heat from the rotor to a liquid hydrogen heat exchanger, whilst maintaining a low-pressure gas inside the rotor thus reducing the load on the rotary gas seals.

The advantages of the solution are that it eliminates the need for rotating a large cryocooler and it makes possible the use of a cryogen-cooled heat exchanger instead of a cryocooler. The challenges are that it requires a rotary gas seal and the maintenance of pressure in a closed volume that will inevitably cool from ambient to a cryogenic temperature and warm up again.

1.5. Flow rate

For the concept to work, the circulation gas's temperature rise must be kept within an acceptable level, 2 K for example. The helium flow rate, $\dot{m}$, required to remove 22.65 W (Table 1) of heat at 50 K with a 2 K temperature rise is described by:

$$\dot{m} = \frac{25W}{h_{52K} - h_{50K}}$$

In this case the enthalpy for helium at atmospheric pressure and 52 K, h_{52K} , is 275.38 kJ/kg and at 50 K it is 264.98 kJ/kg (source RefProp [13]). A mass flow of 2.2 g/s is then needed to transport 22.65 W of heat away from the solid rotor to the cryocooler heat exchanger. The density of helium at 52 K is 0.92 kg/m³, so the flow rate would be 2.4 l/s. For context, this flow rate in a 20 mm diameter tube would equate to a velocity of 8 m/s. Comparing this solution to conduction in a copper link, with a nominal 300 mm conduction length from the cryocooler to the heat source (the 100 kW motor's rotor is 400 mm long and 200 mm diameter), copper with a RRR of 1000 at 50 K has a conductivity of 1200 W/m.K, so the diameter of a copper bar required for a 22.65 W and 2 K temperature difference for between its ends is 60 mm, and would weigh 7.6 kg.

1.6. Rotary seals

A useful concept is that all moving seals leak, some more than others and some very slowly, but they all leak. The aim of a seal is to reduce the leakage to an acceptable level. Ferro-fluidic seals 'seal' very well and could be used for the pumped rotor but are bulky and can generate significant heat at high rotational speeds and large diameters. One hypothesis in this concept is that lip seals, which are a very compact and mature technology, can keep leakage to a small amount when the pressure difference is small. If the working gas in the rotor was just above ambient, it could ensure any leakage was small but outwards, so contaminants will not enter. The low-pressure differential means the lip seals would not have a high contact force, minimising heat generation and a compact design could be achieved. The leakage rate will have to be experimentally measured to determine if the lip seal concept can work. Ferro-fluidic type seals can be kept as a backup if the lip seals do not work well enough. The system would need a helium reservoir to provide makeup gas for leakage, but mostly to keep the pressure uniform during cooling down. A pressure relief would be required for when the rotor warms up. It is envisaged that the cooldown time for the cryogenic system is long enough to warrant keeping the motor continuously cold, only warming up for maintenance.

1.7. Circulating pump

Centrifugal pumps take a non-rotating flow and impart energy on it by turning it with an impeller. Pressure is developed through a combination of centrifugal force and diffusion of the tangential velocity from the impeller by the stator section of the pump. The rotor of an electric motor lends itself to spinning a fluid and creates the impeller. A non-rotating heat exchanger can be configured to perform the function of the stator section, completing the pump. Without the stationary heat exchanger section, the fluid would just spin with the rotor and not circulate. The cold heat exchanger returns the rotating fluid from the rotor to non-rotating, generating pressure. Re-entering the rotor, the pump impeller imparts kinetic energy to the fluid. A flow of a desired speed can be created by selecting the blade angles appropriately for the rotation speed and diameter. The pumping pressure is determined by the diameter of the heat exchanger and the rotational speed. In this case, the pressure generated will be small as it only needs to overcome the fluid losses from circulation. The final design consideration is the number of

impeller and heat exchanger blades, firstly to keep the pressure loading on the blades low enough to achieve smooth flow without separation, and secondly to provide enough area for heat exchange with a sufficiently low temperature difference between the heat exchanger and the fluid.

1.8. Thermal contraction

The design has a cold rotor inside a warm cryostat with the constraint of a small gap between the stationary cryocooler heat exchanger and the spinning rotor. For the 100 kW motor design, the rotor length is 400 mm and the axial contraction of the cold rotor could be as large as 1.2 mm, which would represent a high level of strain and stress in the rotor unless a flexible segment is introduced. An axial flexure, shown in Fig. 1, that was stiff radially and torsionally but soft axially was introduced to the design on the driveshaft side, which allowed the cryocooler side to maintain its relative clearance with the rotor. A finite element analysis was performed to ensure that the flexure could withstand the motor torque and thermal contraction without dangerous vibration modes. An important function of the flexure bearing was that it added a significant conduction length and contact resistances to the driveshaft end, further reducing the heat leak in from that source.

2. CFD model of concept

2.1. Test rig design

A test rig, shown in Fig. 1, was designed to test the pumping concept with the same 200 mm diameter rotor as the proposed 100 kW motor, and a shortened rotor length. The main effect of shortening the rotor was a reduction in the thermal load from radiation. The end sections and diameter of the rotor were kept the same as for the proposed 100 kW motor, so the pumping action and heat transfer to the cryocooler would be representative for the 100 kW motor. The test rig used a SunPower DS30 cryocooler [14] to provide refrigeration. The test rotor was spun by a separate electric motor as it was only for testing the cooling method and did not include the rotor coils or a stator section.

2.2. CFD model of system

A CFD model using ANSYS® CFX was used to check the test rig design. The model included the gas space and the solid parts of the rotor

that terminate at the ambient temperature parts of the shafts. The whole 360 degrees of the rotor geometry was modelled as shown in Fig. 2 because the stationary condition involved complex three-dimensional flows. The cryocooler and heat exchanger were not modelled as solids, but as surfaces to the gas with temperature boundary conditions.

Boundary conditions:

- The cold heat exchanger walls were set to 50 K and the walls of the cryocooler were set to adiabatic as they would share a similar temperature gradient to the helium gas surrounding them.
- The warm surfaces of the bearing housing that contact the gas were modelled as boundaries at 300 K.
- The gas connection to the seals was cut short and modelled as an opening at 300 K and 1 bar absolute pressure. The opening condition allowed the model to draw gas in while cooling down and maintain a constant internal pressure.

A nominal heat load of 2 W was added to the outer diameter of the rotor solid to simulate a heat load from radiation and insulation losses. This load reflects the test rig that has been shortened and does not have the coil or flux pump mentioned in Table 1.

Domains:

- The solid rotor and the internal gas were modelled with a rotating domain. CFX allows a domain to be given a rotating frame of reference where the angular speed is specified. CFX then calculates the appropriate Coriolis and centrifugal momentum terms for the total energy equations for the fluid domains. The results can be viewed as either relative to the domain rotation, or relative to the non-rotating frame of reference.
- The gas around the cryocooler was modelled as a stationary domain.
- The interface between the stationary and rotating gas domains is defined in CFX as a 'domain interface' which calculates the transfer of flow, momentum and heat across the interface.
- The Redlich-Kwong real gas model for helium from the CFX materials library [15] was used as the working gas.
- The k-epsilon turbulence model was used as the Reynolds number in the heat exchange and central parts of the model indicated turbulent flow.
- The following polynomials were used to define temperature-dependent conductivity and heat capacity of the stainless steel specified for the solid domain:

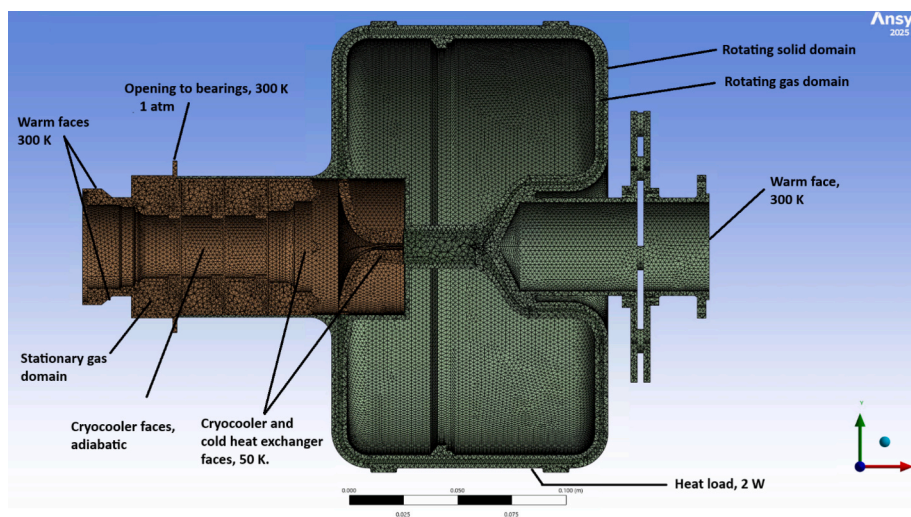


Fig. 2. CFD model of the test rig with the cryocooler showing the mesh and boundary conditions for the model set up. The rotating domains are green and the stationary gas around the cryocooler is brown. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

$$\text{Conductivity } K = 0.03 T + 6.4678 \frac{W}{m.K}$$

$$\text{Heat Capacity, } C_v = 3 \times 10^{-5} T^3 - 0.0224 T^2 + 6.196 T - 156.18 \text{ J/kg}$$

Mesh:

- Fig. 2 shows the mesh.
- Meshing was performed in ANSYS Mesh®.
- Tetrahedral elements were used throughout.
- The general mesh size was 2 mm which was sufficient to represent the geometry well. This was a compromise between computational time (due to the full 3D model) and the fidelity of the result. A mesh dependence study showed that finer elements than the 2 mm mesh used did not affect the results significantly.

Solver:

- The solver was set to a steady state solution
- The timescale for the solver was set to automatic, which was 2×10^{-4} s for the gas and 1×10^3 for the rotor solid (which had a much larger heat capacity than the gas).
- The convergence criteria were set to residual RMS values of 10^{-5} . Steady state was determined manually by monitoring the heat flows in the model so the overall energy balance was below ~ 0.2 W. This was because the time to cool the solid portion meant the RMS residuals reached a low value before the temperatures had reached steady state.

2.3. CFD to optimize the pumping effect

Conventional geometric blading calculations from the Pump Handbook [16] were used to determine the inlet and outlet blade angles of the stator and rotor sections of the pump to achieve a smooth flow path for the desired gas volume flow rate at the design rotor speed. The impeller had straight vane radial blades to match channels around the rotor. The stator blades took flow radially from the rotor channels so the tangential speed at 30 mm radius resulted in an inlet angle for the stator (the cryocooler's heat exchanger) of 81.5 degrees to the radial (facing into the direction of rotation). The outlet angle of the heat exchanger vanes was 0 degrees to remove the rotating component of the flow. A cubic spline from inlet to outlet defined the blades surface. The CFD model was then iterated to determine the number of blades and channels required for a smooth flow pattern and good heat transfer.

The pumping pressure was primarily dependent on the outer diameter of the cold heat exchanger relative to its outlet to the rotor. Flow in a channel outside the cold heat exchanger's outer diameter played no part in the pumping as any pressure developed while moving the flow to a larger diameter was equally countered by its moving back towards the centre.

The channels in the rotor ensured that the flow rotated with the rotor. The channels would impart angular momentum on the flow as it moved out radially, then would remove the angular momentum as the flow returned to the centre and the cold heat exchanger. As the flow moved to the circumference of the rotor, it compressed due to the centrifugal forces and expanded again on return to the centre.

The fluid friction losses in the rotor circuit were very low due low velocities as the flow slowed down when it moved to a larger diameter.

The CFD model also indicated the need for baffles in the gas space next to the cryocooler's regenerator to stop gravitational density (buoyancy) driven flows transporting heat axially from the cold heat exchanger to the warm faces of the cryocooler.

Once the design had been completed, the CFD model was used to calculate heat transfer in the system to test the hypothesis that integral

pumping could be used to effectively remove heat from the rotor, plus the heat leaking into the system from the gas adjacent to the cryocooler was sufficiently small.

Fig. 3 shows the streamlines inside the rotor as calculated by the CFD model. The figure shows flow in the local frame of reference, meaning that the frame of reference is rotating for the rotor sections and is stationary for the cold heat exchanger. The high-speed flow exiting the heat exchanger is shown in the centre (green streamlines). The flow slows (turns blue) as it moves radially outward through the rotor's pumping section and on to the rotor's circumference, where the rotor coils would be.

2.4. Heat leak calculations

Fig. 4 shows steady-state temperatures of the helium gas and solid walls with a scale from 50 K to 300 K. It shows the conduction gradients in the drive shaft (right hand side) and the gradients in the shaft and gas adjacent to the cryocooler (left). The effect of the flexure is evident as it intercepts the shaft heat flow producing a longer conduction path. Fig. 5 is an enlargement of the same plot with the temperature scale between 50 K and 60 K to show the temperature of the rotor in more detail. It shows that a significant amount of heat is added near the cryocooler heat exchanger inlet, which enables the coil part of the rotor (outer diameter) to remain cooler than the final gas temperature entering the cryocooler.

The energy balance calculated by the CFD model is shown in Table 2. It shows that at 50 K the largest heat load of 10.19 W is conduction down the thin wall tube at the cryocooler end. In comparison, the drive shaft conduction, with the axial flexure increasing the length of the conduction path, is only 2.47 W. Conduction in the helium gas adjacent to the cryocooler's regenerator is calculated as the sum of the heat leak into the 'Seal end warm faces in contact with the gas' at that end plus heat flow through the 'Seal end gas opening to the seals', totalling only 1.6 W.

The cryocooler absorbed 16.43 W. The heat flow into the coil section of the rotor represents radiation and insulation losses and was a boundary condition for the cylindrical surfaces of the rotor set at 2 W. The final balance of 0.17 W includes viscous work plus numerical errors in the calculation.

The mass flow of helium around the circuit was calculated by the CFD model to be 1.1 g/s, which should give a temperature rise of 2.83 K in the helium for 16.4 W of heat input. However, the modelling showed that the 10.19 W from the thin-walled tube to the seal end was intercepted by the flow entering the cold heat exchanger and therefore did not heat the section where the HTS coils would be mounted. The temperature rise experienced at the coils is only the driveshaft heat leak plus the heat into the coil section, totalling 4.47 W, which should only increase the helium temperature by 0.8 K. The CFD model predicted a temperature difference between the cold heat exchanger and the gas

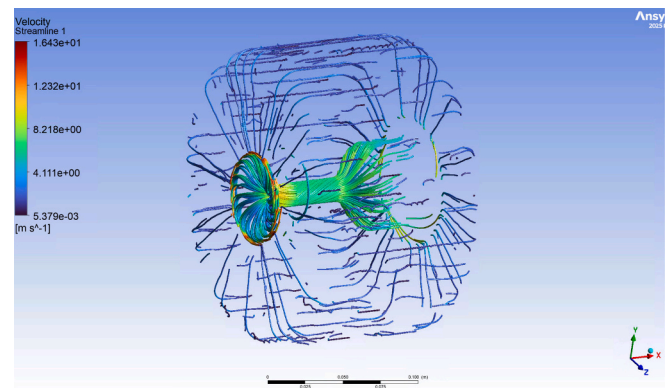


Fig. 3. Streamlines of the flow through the heat exchanger and around the rotor. The streamlines are displayed in the rotating frame of reference of the spinning rotor and stationary frame of reference for the heat exchanger.

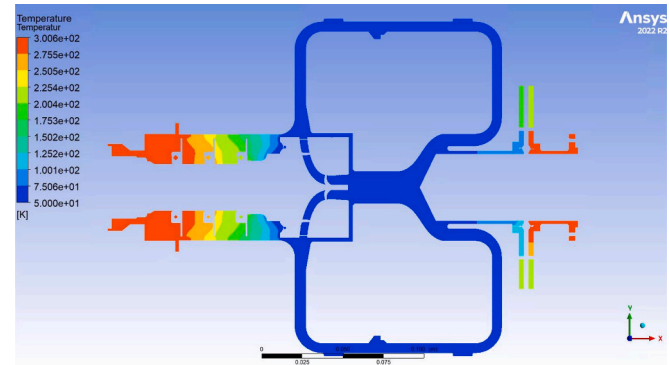


Fig. 4. Steady state temperature profile for the cold parts of the rotor (gas and solid) showing the gradients in the shaft ends and helium gas space around the cryocooler.

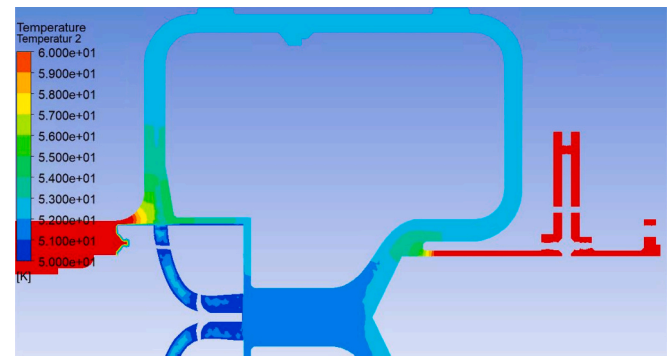


Fig. 5. Steady state temperatures of gas and solid parts with the scale expanded to show temperature gradients in the 50-60 K range in the rotor and around the cold heat exchanger.

Table 2

Energy balance calculated by the CFD model of the prototype.

Component	Heat flow into the system
Cryocooler heat exchanger	-16.43 W
Seal end warm faces in contact with gas	0.75 W
Seal end gas opening to seals	0.85 W
Seal end thin wall tube conduction	10.19 W
Drive shaft end conduction	2.47 W
Heat flow into the coil section of the rotor	2.0 W
Energy Balance in model	-0.17 W

exiting of 1.7 K. The temperature of the helium re-entering the cold heat exchanger was 53 K, a 1.3 K rise, indicating that the back of the heat exchanger was doing a significant amount of the cooling work, which is expected as most of the heat to be removed comes from the adjacent thin wall tube.

The temperature difference between the cold heat exchanger and the gas exiting it was 1.7 K so the calculated temperature difference between the cryocooler and the rotor section was a total of 2.5 K. Overall, the expected 16 W of cooling required by the DS30 to maintain steady state is more than the ~10 W available at 50 K so it is expected that the experiment would settle with the cryocooler at around 60 K instead of 50 K.

2.5. Heat leak through the gas next to the cryocooler

One unknown with the concept was whether the heat leak from conduction in the gas next to the regenerator was significant. The CFD model calculated this leak and in early models showed the potential for

density driven convection in this gas space leading to excessive heat leak as in Fig. 6.

Three baffles, the number that could be physically fitted to the cryocooler's regenerator tube whilst being thermally isolated from each other, were modelled on the cryocooler regenerator shaft and the CFD model showed that a steady temperature gradient could be established when flow in the axial direction was disrupted. The presence of the conduction driven temperature gradient in the thin tube wall from the bearing to the rotor, plus the natural temperature gradient of the regenerator wall, helped hold the gas' temperature gradient in place, resulting in conduction being the only heat transfer mechanism through the gas from warm to cold. The net heat leak through the gas was measured as the heat flow needed to maintain the ambient surfaces at 300 K, plus any heat flow across the opening (which had zero mass flow once steady state was achieved). The baffles reduced the heat load through the gas to 1.6 W confirming the design concept.

The effect of the helium gas pressure on the heat flows is shown in Fig. 7. It shows that with increasing pressure, there is a small increase in the heat to be removed, 1 W, which comes from an increase in the heat leaked to the opening and in the bearing end's thin wall tube. This shows that if the gas pressure is kept between 1.2 and 1.5 bara during experimentation, the gas pressure will not dominate the results.

2.6. Cooldown transient

The CFD model was modified to run a transient to predict how the rotor would cool down with the gas circuit and a cryocooler capable of the target temperature as its only means of refrigeration. The transient cooldown model was expected to give an indication of the cooldown time, any temperature transients the parts might have to withstand and how the gas would flow during cooldown (buoyancy driven flows may aid cool-down heat transfer). As a superconducting motor would not be able to run until it was fully cooled, the transient was calculated with the rotor stationary.

Fig. 8 shows the transient model part way through cool-down. It shows the gas adjacent to the cold heat exchanger being close to 50 K and falling under gravity (—Y direction) to cool the lower part of the rotor first. The gas warms up as it goes around the bottom part of the rotor and continues to rise to the top. The gas and wall temperatures are close to each other during most of the circulation indicating good heat transfer between the walls and the gas. The temperature gradient from 300 K to 50 K along the cryocooler shaft is clear on the left side of the figure. The transient model took approximately one month of computer time to run with 36 cores, simulating 11 h of cooling the model with a timestep size of 1 s. A timestep of 0.1 s was used for numerical stability at startup but after setting up the initial flow pattern, the model responded very slowly due to the high heat capacity of the stainless steel rotor solid

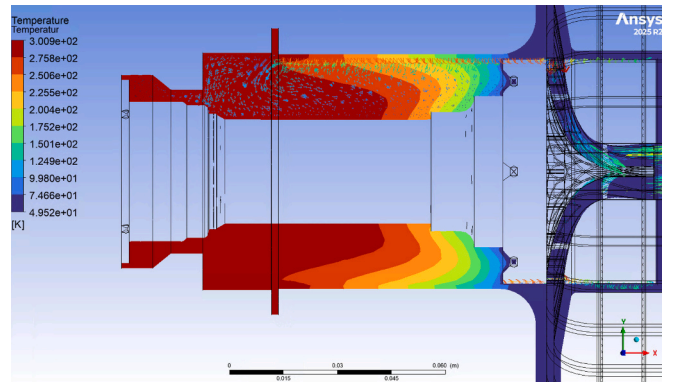


Fig. 6. CFD model of the gas next to the cryocooler without baffles, showing axial flow transporting heated gas towards the cryocooler end. This run had a heat load on the cryocooler of 22.17 W compared with 16.43 W with baffles.

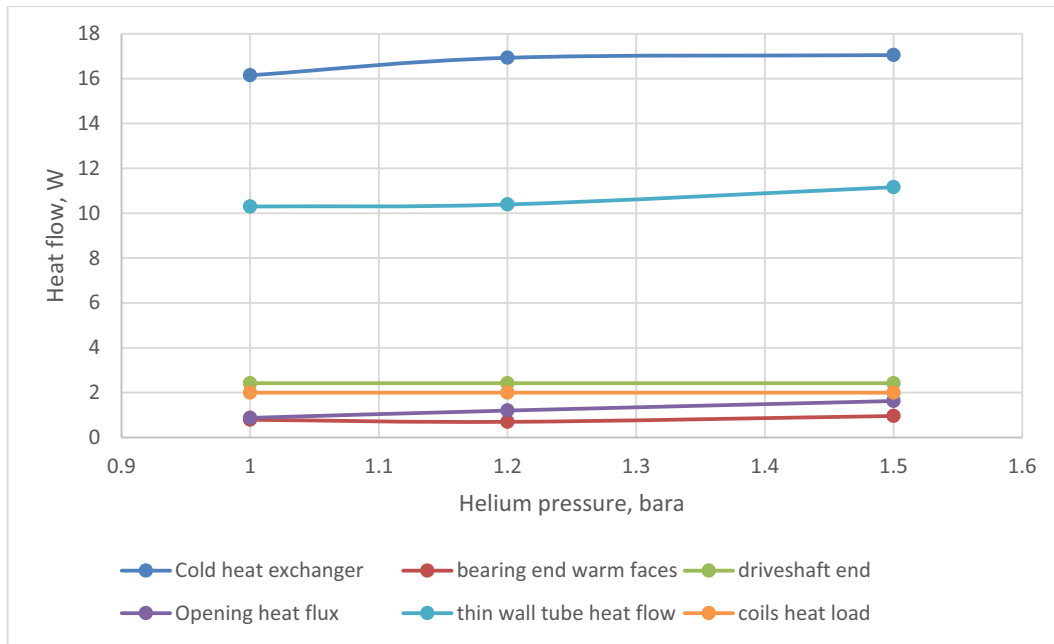


Fig. 7. Effect of Helium gas pressure on the energy flows in the CFD model.

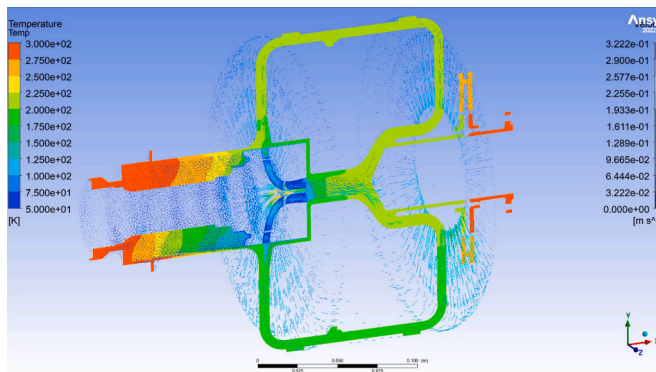


Fig. 8. Transient in progress showing temperature contours and velocity vectors of the buoyancy driven flow.

so the timestep was increased to 1 s. The longer timestep increased the Courant number but since the flow was essentially steady state, the model was stable and converged every timestep.

The model showed the cryocooler quickly cooling the adjacent gas to 50 K and creating a buoyancy driven flow around the lower part of the rotor, which circulated up the driveshaft end and return dropping into the heat exchanger. Importantly for experimentation, the transient model showed that during a stationary cooldown, the gravity driven convection currents would produce a measurable temperature difference, in the order of 10–20 K, between opposite corners of the rotor, that is positions R8 and R2 on Fig. 9.

3. Experimental validation

3.1. Hypothesised behaviour to prove the concept

It was hypothesised that, with the rotor stationary the temperature pattern in the rotor will be like the transient CFD model with a large temperature difference between the cryocooler cold head and the rotor, and a temperature difference of 10–20 K across the rotor between the bottom (positions R8 and R4) and top of the rotor (positions R2 and R7). When the rotor starts spinning, the CFD modelling predicts that the circumferential unevenness will disappear, and the pumping action will increase flow through the cold heat exchanger, which in turn will

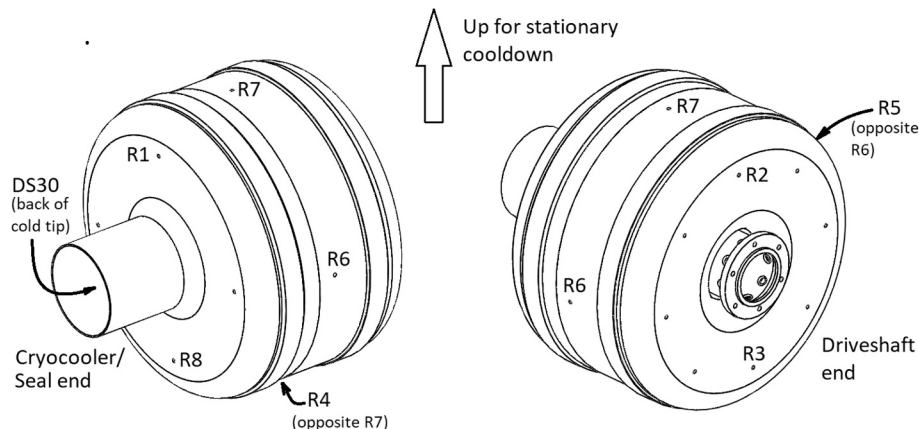


Fig. 9. Placement of the temperature sensors in the model and on the experimental rotor.

increase heat transfer and warm the cryocooler up whilst cooling the rotor further. Eventually the temperature difference between the cryocooler and the rotor will reduce to around 2.5 degrees when spinning at 4500 rpm. The flow direction through the centre of the rotor will mean that the coldest part of the rotor will be at the centre on the driveshaft end (positions R2 and R3), and the warmest part will be near where the gas re-enters the cold heat exchanger (positions R1 and R8).

If the transition of temperatures is observed when the rotor starts spinning, then it is evidence of the pumping action and proves the concept.

The CFD model's heat flow calculations can be validated by comparing the observed warm-up rate of the cold rotor's 7.8 kg of stainless steel, giving the heat ingress to the model to achieve the transient. The cryocooler's cooling curve provides the heat removed from the cryocooler exchanger at a given temperature so can be compared with the model's heat flow predictions in Table 2.

3.2. Experimental set up

The shortened rotor experimental rig is shown in Fig. 10. The drive shaft was extended to allow it to be turned by an electric motor via a toothed belt. Fig. 11 shows the 3D printed rotor and PT100 sensors prior to fitting the cryostat tube.

The rotor itself was a welded assembly of 3D printed parts. The circulation channels around the rotor needed to be narrowed to avoid overhangs for the 3D printing. The mass of the rotor increased, but the flow area of the circulation channels was maintained. The cold heat exchanger and rotor blading were 3D printed as per the original design. Eight PT100 temperature sensors were mounted around the rotor at positions shown in Fig. 9, and another PT100 was mounted on the cryocooler's cold head. The temperature readings were transmitted from the rotating shaft to a stationary computer via an infrared diode, of the same type used for consumer remote controls. The data acquisition on the shaft was battery powered and could run for two to three days at a time depending on the frequency of data transmission.

3.3. Calibration of the temperature sensors

Prior to fitting to the rotor, the eight PT100 temperature sensors (R1 to R8) were calibrated by fitting all the sensors to one block mounted onto the cryocooler cold head. The values of the temperature sensors within 1 K of each other down to 50 K, either using a LakeShore 218 temperature monitor or the rig's electronics.

3.4. Cryocooler characterisation

The refrigeration power of the SunPower DS 30 Cryocooler [10] to be used in the experiments was measured so the temperature of the

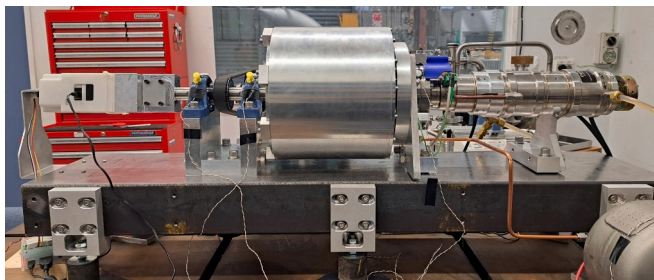


Fig. 10. The rotor test rig with guarding removed. From left to right: data acquisition and IR transmission (white), vacuum pumpout port (capped), belt drive between blue bearing blocks, cylindrical rotor cryostat, DS30 cryocooler. At the bottom right is the helium buffer tank connected by a copper tube. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

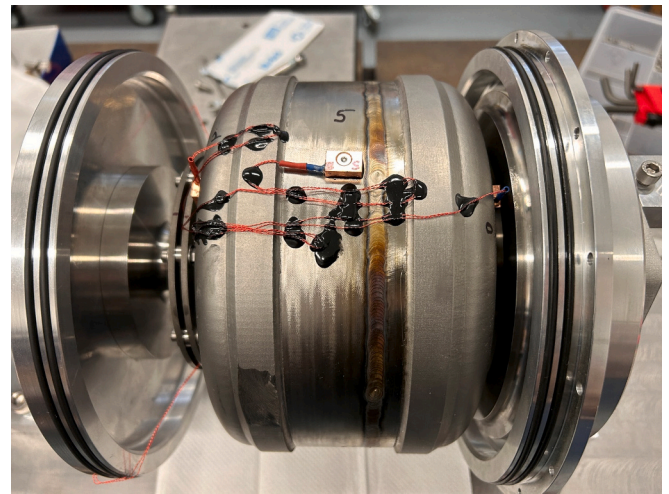


Fig. 11. The 3D printed rotor with temperature sensors mounted, prior to fitting the outer cryostat tube.

cryocooler could be used to infer the amount of heat being removed from the rotor. A heater was mounted onto a copper block on the cold face of the cryocooler and a range of heating powers applied whilst the temperature was measured. The characterisation was carried out with the cryocooler horizontal, in the same orientation as it would be used in the rotor tests. The results in Fig. 12, show that the DS30 cryocooler would be expected to reach 60 K with 16 W of heat removal from the rotor. The ultimate low temperature of the cryocooler was 35 K as expected from SunPower's cooling curve [14] but the 80 K cooling power was only 27.5 W, as opposed to the claimed 30 W. The discrepancy could be due to contact resistances between the cryocooler cold tip and measurement block in the experimental set up. The result, however, is a good simulation of the heat flow and temperature characteristic that would be expected of the cold heat exchanger in the experiment.

3.5. Stationary cool-down experiment

Fig. 13 shows the temperature vs time plot for a cooldown and warmup cycle with the rotor stationary. The DS30 cooled to a low of 69.2 K over 8 h, corresponding to 22 W of heat removal according to Fig. 12. The eight temperature sensors on the rotor cooled down as predicted with the coldest (R8) being immediately below the cold head and warmest (R7) being the opposite corner. The temperature distribution is consistent with the CFD modelling, and the temperature difference between warmest and coldest reduced with temperature as the cooling rate decreased. Additionally, when the cooler was turned off, the rotor's temperatures homogenised. These effects are a good indication that the temperature distribution was due to the cold heat exchanger producing buoyancy driven convection currents in the helium space.

3.6. Seal leakage

The leakage of the lip seals while stationary was manageable, the cooldown was conducted at a pressure between 1.1 barg and 0.8 barg, requiring topping up every two hours as the gas both cooled down and leaked out. The temperature of the cryocooler, the DS30 line on the plot, can be seen to increase during each top up. The reason being primarily due to better heat transfer as the rotor heat capacity dominated the warm gas entering the system. Once the rotor was cold, the gas only needed topping up twice per day to maintain a pressure between 0.2 barg and 0.5 barg. The total gas volume including the warm buffer was 5.6 l, split between 3 l of cold gas in the rotor and 2.6 l of warm gas in the buffer tank. The leakage rate, as calculated by the change of mass in the system whilst cold, averaged 0.6 standard litres per hour. At this rate a G

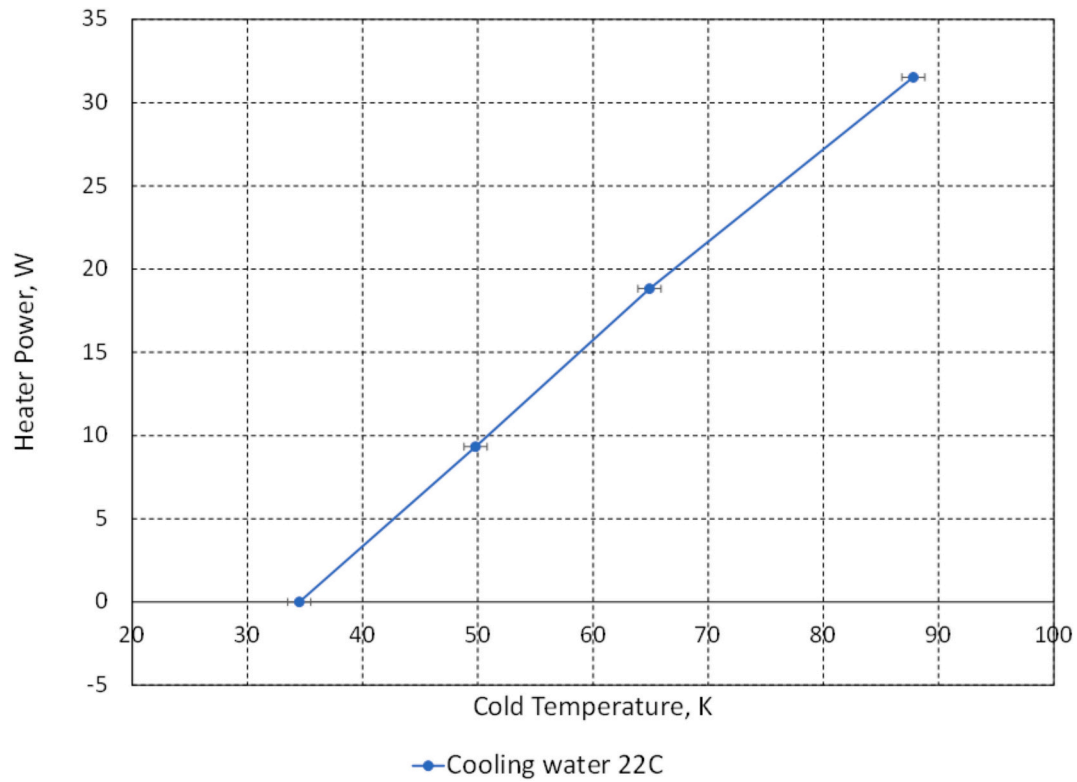


Fig. 12. The cooling power vs temperature characterisation performed on the DS30 cryocooler using a heater.

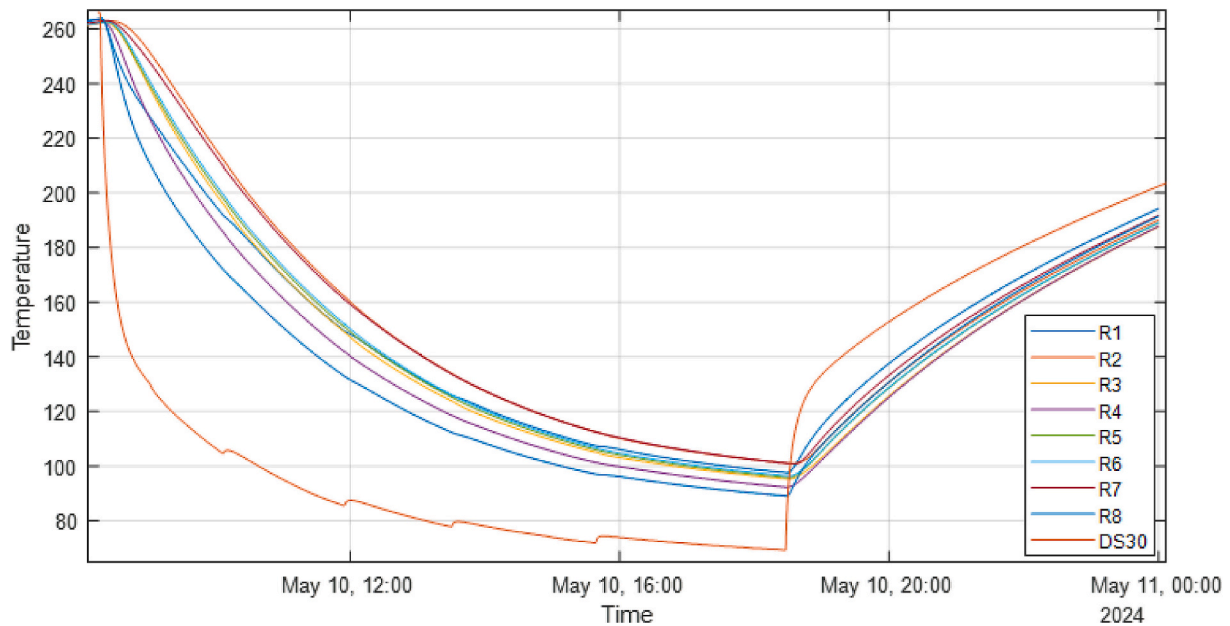


Fig. 13. Temperatures for a cooldown test with the rotor stationary (not rotating). R1 to R8 are the temperatures around the rotor and DS30 is the temperature measured on the DS30 cold head.

sized helium gas bottle would last 1.5 years.

3.7. Confirmation of heat flow calculations

Fig. 14 shows the rotor average temperature vs net warming power. The warming power was calculated from the change in internal energy of the rotor, being a function of the heat capacity and temperature of the rotor. On warmup (positive values), heat was added to the rotor from

shafts, gas conduction, insulation, and heat leak through the cryocooler when not refrigerating. The warming power was a linear function with temperature as can be expected as conduction was the main warming mechanism. Extrapolating the warming power to the target 50 K, indicates that 19 W of cooling would be required to maintain the rotor at 50 K, as compared with the CFD prediction of 16 W. Extrapolating the cooldown rate line (negative values) to the zero crossing indicates that with the DS30 cryocooler, the average rotor temperature will eventually

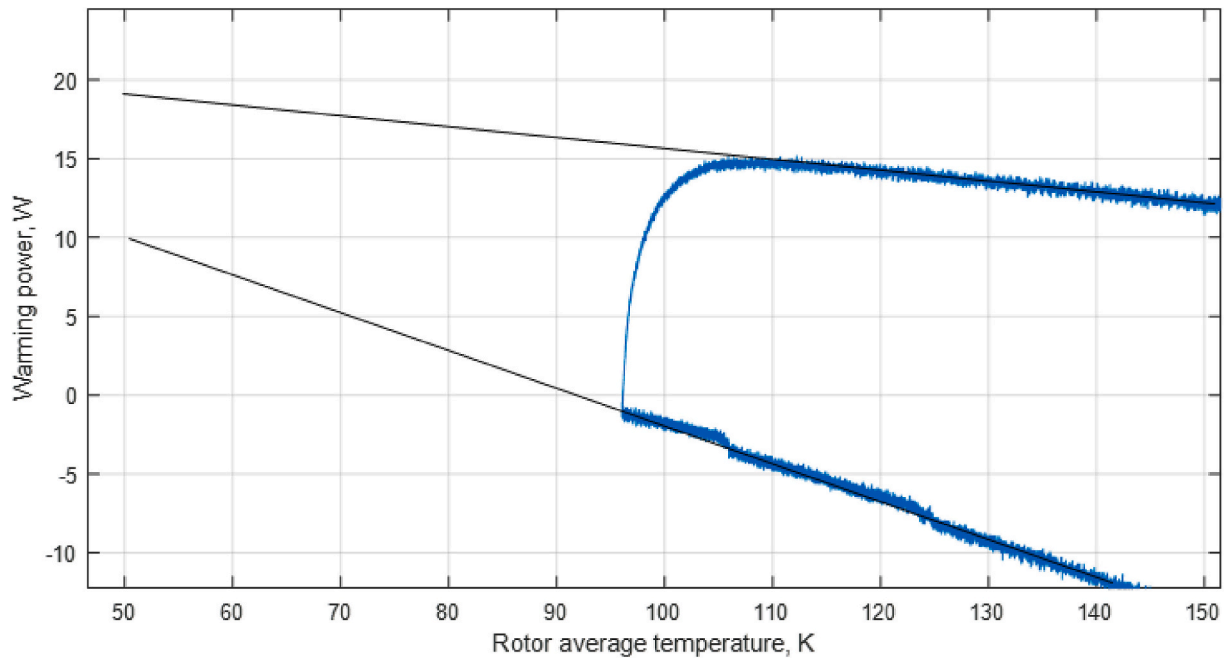


Fig. 14. The net heat entering the rotor (warming power) as calculated from its heat capacity and temperature transient, for a stationary cool-down.

reach 91 K and require 16 W of refrigeration. From Fig. 12, the DS30 temperature at 16 W of load is 61 K. In a later test, the cryocooler stabilised at 62 K with the rotor average temperature at 88 K.

With the cryocooler removing 16 W from the system at 62 K, and the rotor at 88 K, the conductance of the heat exchange would be 0.6 W/K. That is, for every 1 W of heat lift needed, the cryocooler would have to be 1.67 K lower in temperature than the target rotor temperature.

3.8. Effect of the helium presence for heat transfer

An experiment was performed to determine the whether the majority of the heat transfer observed was due to the gas in the rotor or due to conduction through the metal. The heat exchanger was replaced with a component that contacted the rotor, and the helium space inside the rotor was evacuated with a vacuum pump. Thereby the only heat transfer mechanism in the rotor was conduction. Two stationary cool-downs were performed, one with the vacuum and one with helium

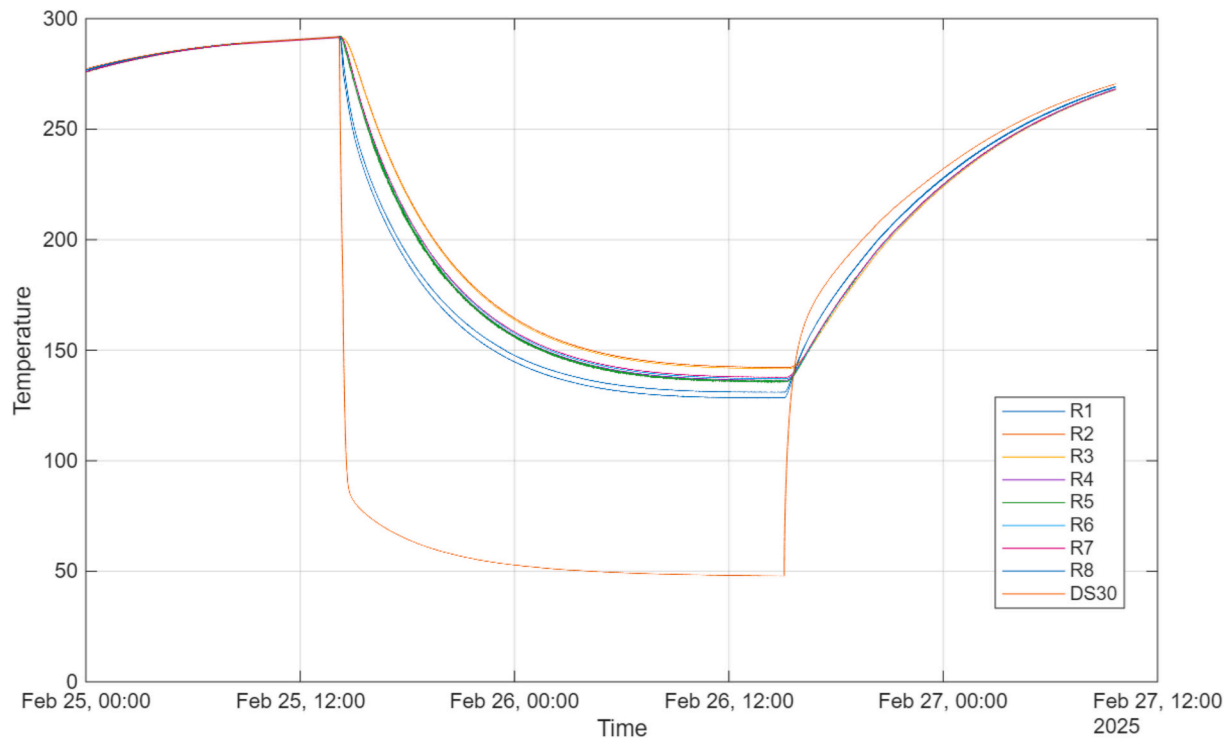


Fig. 15. Cooldown with the rotor stationary and a vacuum in the gas space. R1 to R8 are the rotor temperature sensors and DS30 is the temperature sensor on the cryocooler's cold head.

inside the rotor. The cooldown temperatures for the vacuum test are shown in Fig. 15 below.

The cryocooler reached 48 K, and the rotor settled between R1 and R8, nearest the cryocooler, at 128 K and R2 and R3, furthest from the cryocooler, at 141 K. There was very little temperature variation between sensors the same distance from the cryocooler end (R4, R5, R6, R7) which confirms conduction as the main heat transfer mechanism. A second cooldown, with the same set up and refilling the rotor with helium to gas settled with the cryocooler at 67 K and the rotor at an average temperature of 84 K, confirming the enhanced heat transfer from the presence of helium in the rotor.

3.9. Rotating test demonstrating proof of concept

The proof-of-concept temperatures for a rotating test are shown in Fig. 16. The CFD modelling predicted two different heat transport mechanisms in the rotor with different temperature profiles on the rotor. When the rotor was stationary, the heat transfer was predominantly via buoyancy driven flow, making the bottom two temperature sensors, R8 and R4, the coldest, and the top sensors, R2 and R7, the warmest. When rotating, the pumping action started and delivered the coldest gas to the driveshaft end making R2 and R3 the coldest and the gas returning to the cryocooler heat exchanger was the warmest making R1 and R8 the warmest positions. To show that the pumping action was happening, this transition in temperature profiles would be observed. To show that the pumping action was producing better cooling than the buoyancy currents, the temperature difference between the rotor and the cryocooler heat exchanger would reduce. Initially the cryocooler temperature would rise as it draws more heat from the rotor to cool it down more.

3.10. The rotating experiment

Referring to Fig. 16, the rotating test started after steady state was reached during a stationary cooldown with 1.37 bara helium pressure inside the rotor. The rotor was at an average temperature of 92.8 K and the cryocooler at 63 K. The temperature patterns across the rotor were

consistent with convection from density-driven currents with the temperatures R8 and R4 the coldest and R2 and R7 the warmest, referring to Fig. 9 for the sensor positions. The vacuum pump was then disconnected, and rotation was started at 1000 rpm. Rotation continued for approximately 80 min. Initially the rotor temperature reduced and the cryocooler temperature rose, indicating improved conductance between the cryocooler and the body of the rotor. After 47 min, the rotor temperature started to rise with the cryocooler temperature also rising, indicating an increased heat input with continuing levels of conductance between the cryocooler and the rotor. The rotation was then stopped with the cryocooler continuing to run. The rotor continued to warm due to the increased heat input and reduced conductance whilst stationary. Finally, the cryocooler was turned off and the system warmed up quickly.

3.11. Interpretation of results

As hypothesised, Fig. 16 shows that whilst stationary, the coldest temperatures of the rotor were R8 and R4, the lowest two sensors, which were cooled by buoyancy driven convection currents. Conversely, the warmest sensors, R2 and R7, were at the top of the rotor. When rotation started, the temperature distribution changed to the CFD model prediction, with the flow of cold gas down the centre of the rotor from the heat exchanger to the driveshaft end of the rotor, making the sensors R2 and R3 the coldest. The warmest sensors were R1 and R8 at the cryocooler end of the rotor where the gas re-entered the heat exchanger after picking up the heat from the rotor. This transition of temperatures confirms the pumping action. Additionally, the cryocooler temperature increased while the rotor temperatures decreased, indicating a higher heat load on the cryocooler as it first cooled the rotor then started working against an increased heat load from the deteriorating vacuum. At the best point, the rotor was cooled to an average temperature of 90.5 K.

At the start of the test, the average rotor temperature was 92.8 K with the cryocooler at 63 K. The heat load from the cryocooler's cooling curve, Fig. 12, was 17 W resulting in a conductance of 0.57 W/K. When rotation started, the rotor average temperature dropped to 90.5 K from a cryocooler temperature of 76.1 K. The heat load, as calculated from the

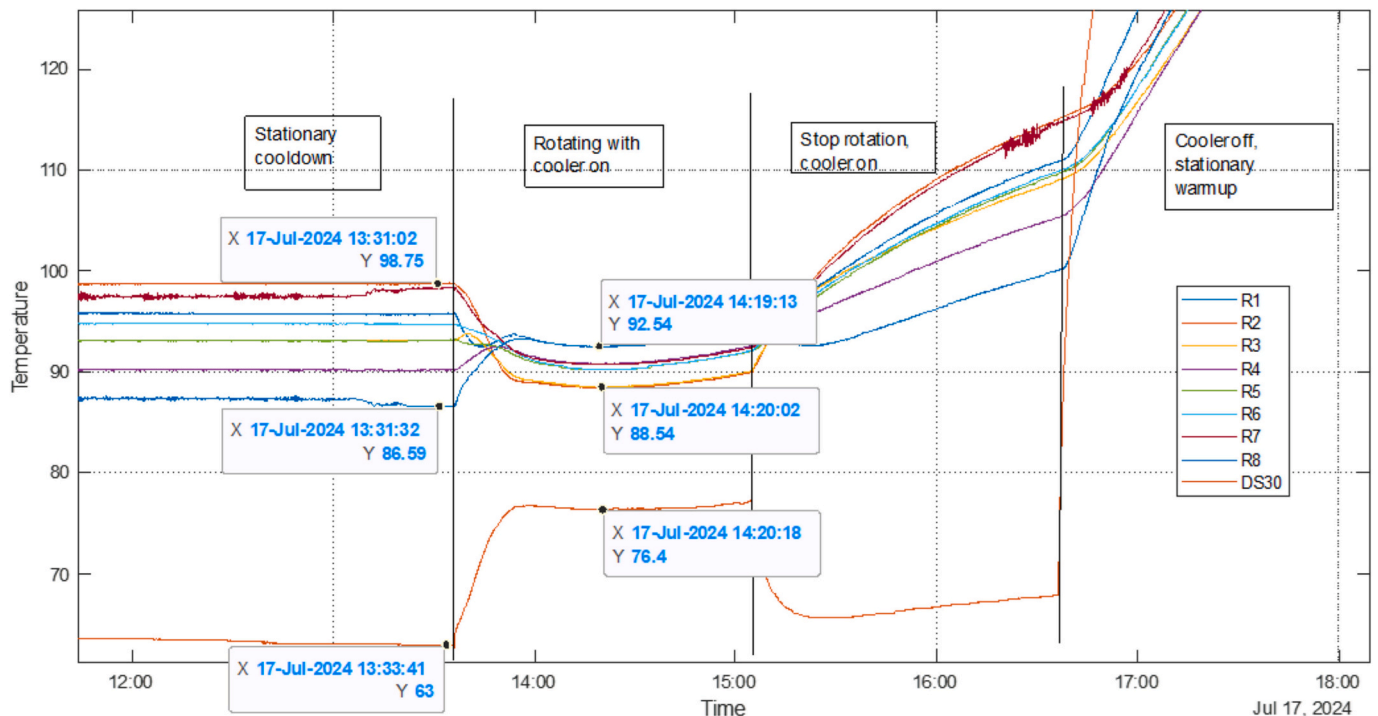


Fig. 16. The rotor and cryocooler temperatures during the rotating experiment.

cryocooler's cooling curve, increased to 25 W. The resulting conductance was 1.76 W/K and demonstrates improved heat transfer from the pumping effect.

The heat capacity of helium at 1.2 bara and 85 K is 5.19 kJ/kg.K. A temperature rise of 14.4 K in the gas (warmest part of rotor to gas in cold heat exchanger) for 25 W of heat load implies a mass flow of 0.3 g/s. The CFD model calculated 1.1 g/s calculated at 4500 rpm and this test was at 1000 rpm. Scaling the measured mass flow up with rpm implies a mass flow of 1.3 g/s at 4500 rpm, and the conductance will be similarly higher and more in line with the CFD prediction of 6 W/K.

The rotating test was not long enough to determine whether the lip seals leaked significantly more while rotating than stationary. The frictional resistance of the lip seals was more than expected, in the order of 1 Nm torque, which translates to 471 W at 4500 rpm and would result in higher than anticipated seal temperatures. A reduced maximum rotational speed of up to 1200 rpm was chosen for the tests to reduce the risk of seal failure before the proof-of-concept could be achieved. The seal housing reached a steady state temperature of 55 °C from the heat generated by the lip seals.

3.12. Vacuum in the cryostat

The increased heat load on the rotor was most likely due to the vacuum in the cryostat deteriorating. The effect of the seal heating was ruled out as the conductance down the thin-wall-tube to match the extra 8 W of heat load that the cryocooler was experiencing, would mean a seal temperature over 195 °C at the seal housing, where only 55 °C was observed.

The small volume of the rotor cryostat provided a challenge for maintaining a sufficient level of vacuum for insulation. Calculation of the bare vacuum heat transfer for the rotor indicates that the heat leak increases from approximately 1 W at a vacuum level of 10^{-4} mbar to 10 W at 10^{-3} mbar, a vacuum level which is consistent with the extra heating levels that were observed during the testing. The rotor was initially pumped out to 10^{-5} to 10^{-6} mbar, providing good insulation. It was observed on one vacuum leakage test that in 30 min the vacuum deteriorated from 4×10^{-6} mbar to 1.6×10^{-4} mbar, the level where the heat leak increased to over 1 W. This leakage corresponds to a rate of 3.5×10^{-7} mbar l/s, which is consistent with an O-ring sealed assembly with a data feed-through.

More work is required to improve the vacuum tightness of the rotor cryostat to allow longer rotating tests to be conducted to determine the ultimate rotor temperature achievable with the DS30 cryocooler. It is expected that welding the cryostat to seal it would provide a sufficiently hard vacuum for long term operation, but experimental work requires occasional disassembly, so welding is not practical. At the time of writing, a solution was being designed with a ferro-fluidic seal to allow constant vacuum pumping whilst rotating. While the constant vacuum pumping is not appropriate for aircraft use, it will allow development to continue without a welded cryostat.

3.13. Improvements to the cold heat exchanger

The effective conductance of the original design whilst stationary was 0.6 W/K. For 20 W of cooling load, this would result in an unacceptable temperature difference of 33 K between the cold heat exchanger and the rotor, meaning the cryocooler would have to remove heat at 18 K to achieve 50 K on the rotor. The original shape of the heat exchanger followed the ideal gas flow path for a pump component. This gave a smooth flow but limited the heat exchanger area for cooling while stationary. An alternative heat exchanger, shown in Fig. 17, was designed and manufactured to increase the heat exchange area by filling the available cylindrical space in the rotor whilst keeping the same inlet and exit angles for smooth gas flow when rotating. The larger surface area would enable the gas to reach a colder temperature before circulating around the rotor.

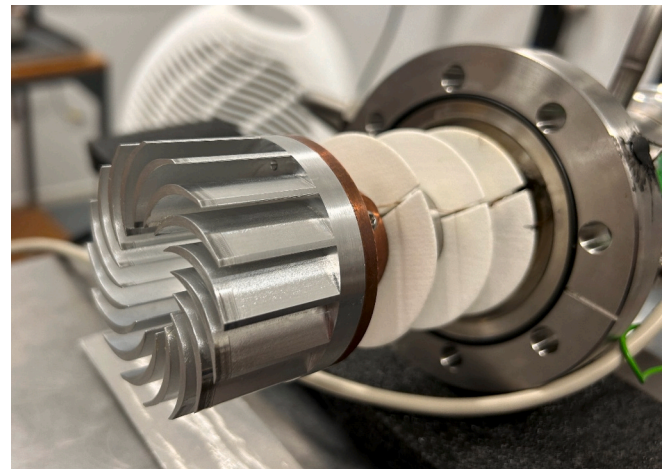


Fig. 17. The improved cold heat exchanger mounted on the cryocooler cold head.

The CFD model of the new design is shown in Fig. 18, and predicted a lower over-all temperature of the rotor whilst stationary. An addition to the CFD model was to include the gas inside the body of the rotor as it plays a part in the buoyancy driven circulation and heat transfer whilst the rotor is not rotating. The model predicted an average rotor temperature of 68.5 K from a cryocooler at 50 K removing 18.4 W, equalling a conductance of 1 W/K as compared to the previous 0.6 W/K.

Fig. 19 shows the CFD model prediction of the flows with the rotor rotating. The heat transfer and flow path through the improved heat exchanger was very similar to the previous design.

Fig. 20 shows the average rotor temperature and cryocooler temperature for a stationary cooldown test with the improved heat exchanger. The average rotor temperature was 80.9 K with the cryocooler 65.3 K, and corresponds to a conductance of 1.125 W/K which is an improvement on the previous 0.6 W/K.

Fig. 21 shows a rotation test with the new heat exchanger. The experiment also tested and confirmed the hypothesis that vacuum was the main source of increased heat during rotation. The test started from steady state while stationary, the vacuum was disconnected and rotation started at 1000 rpm, increasing to 1200 rpm. Rotation was briefly reduced to 600 rpm before returning to 1200 rpm. The rotation was then stopped, and the rotor temperatures continued to rise whilst the cryocooler temperature decreased. As soon as the vacuum pump was reconnected and pumping resumed, the rotor and cryocooler temperatures both dropped.

The conductance of the heat transfer system is shown in Fig. 22. At

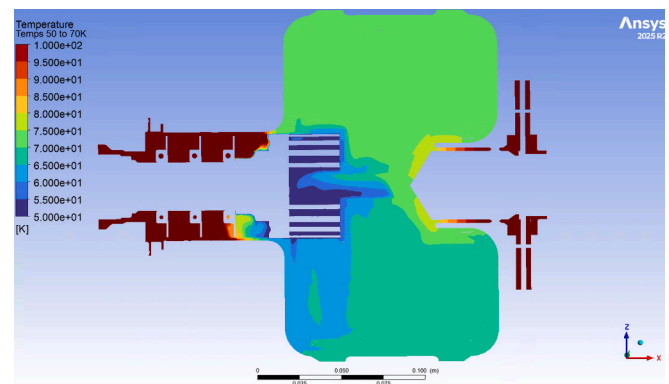


Fig. 18. CFD model of the rotor with the improved heat exchanger design. The cold heat exchanger is set to 62 K, Enhanced buoyancy-induced convection currents in the rotor showing an increased flow.

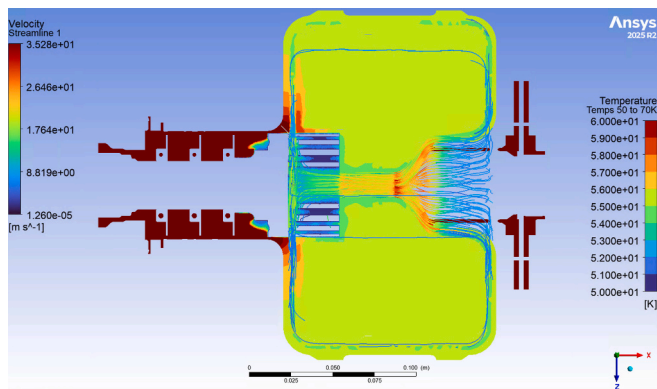


Fig. 19. CFD model of the temperature and flow patterns within the improved heat exchanger design whilst rotating.

1000 rpm, the conductance was 1.9 W/K, an improvement on the previous 1.76 W/K, possibly due to the greater area on the cold heat exchanger for heat transfer. At 1200 rpm the conductance increased to about 2.2 W/K the first time then 2.0 W/K the second time, an indication of the repeatability of the data.

3.14. Prediction of performance at full rpm

The rotating tests were performed at a maximum of 1200 rpm where a conductance of 2 to 2.2 W/K was achieved. The full speed of the operating motor is intended to be 4500 rpm. The pumping has a mass flow rate that is approximately proportional to the rotational speed. Fitting a curve to the data points for 600, 1000 and 1200 rpm, predicts a conductance of 9.2 W/K for 4500 rpm, which for 20 W of heat load,

would correspond to a temperature difference of 2.2 K between the cold head of the cryocooler and the rotor average temperature (temperature at the coils). This compares very favourably with conduction cooling which must conduct heat across multiple contact resistances and over a significant distance, in the order of 0.3 m for a motor of that length.

4. Discussion and conclusion

The novel concept of a self-pumping gaseous helium circulation loop inside the rotor of a superconducting motor has been validated by CFD calculations and experimentation as being able to transfer heat from a rotating rotor to a stationary cold heat exchanger. This concept places the stationary-to-rotary interface in the gas between the stationary heat exchanger and the rotating rotor. Having a self-contained circulation loop in the rotor means that the gas pressure can be maintained close to ambient to reduce the pressure across the rotary gas seals, reducing leakage and heat generation. The design has a temperature gradient from the cold heat exchanger to the cryocooler's mounting flange and the seals, resulting in seals that are at room temperature and isolated from the cryogenic environment in a similar manner to the seals in a cryogenic bayonet fitting. Like the bayonet fitting, the concept relies on the conduction through the gas adjacent to the cryocooler's regenerator being small compared with conduction down the thin-walled tube to the seals.

The CFD modelling showed that most of the heat ingress to the rotor came from the thin-walled tube at the cryocooler end, making up 74% of the heat to be lifted by the cryocooler. This heat was intercepted by the cold heat exchanger at the end of the gas flow path, meaning that the coil section of the rotor was relatively closer to the cryocooler in temperature. Improvement to the system could be achieved through extending the conduction length of the thin-walled tube to reduce the heat ingress. Alternatively, the tube could be replaced with a G10 epoxy composite

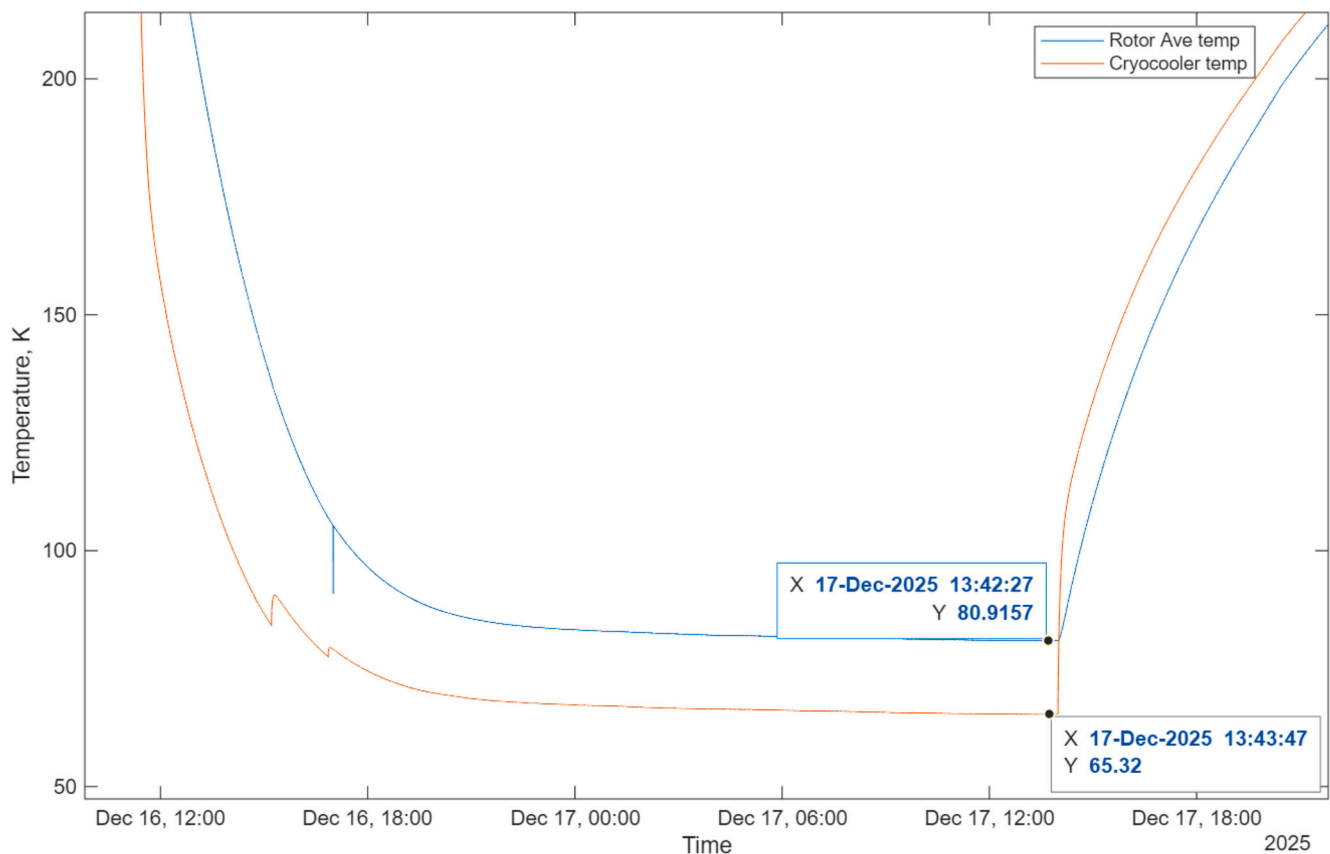


Fig. 20. Rotor average and cryocooler temperatures for a stationary cool down with the new cold heat exchanger.

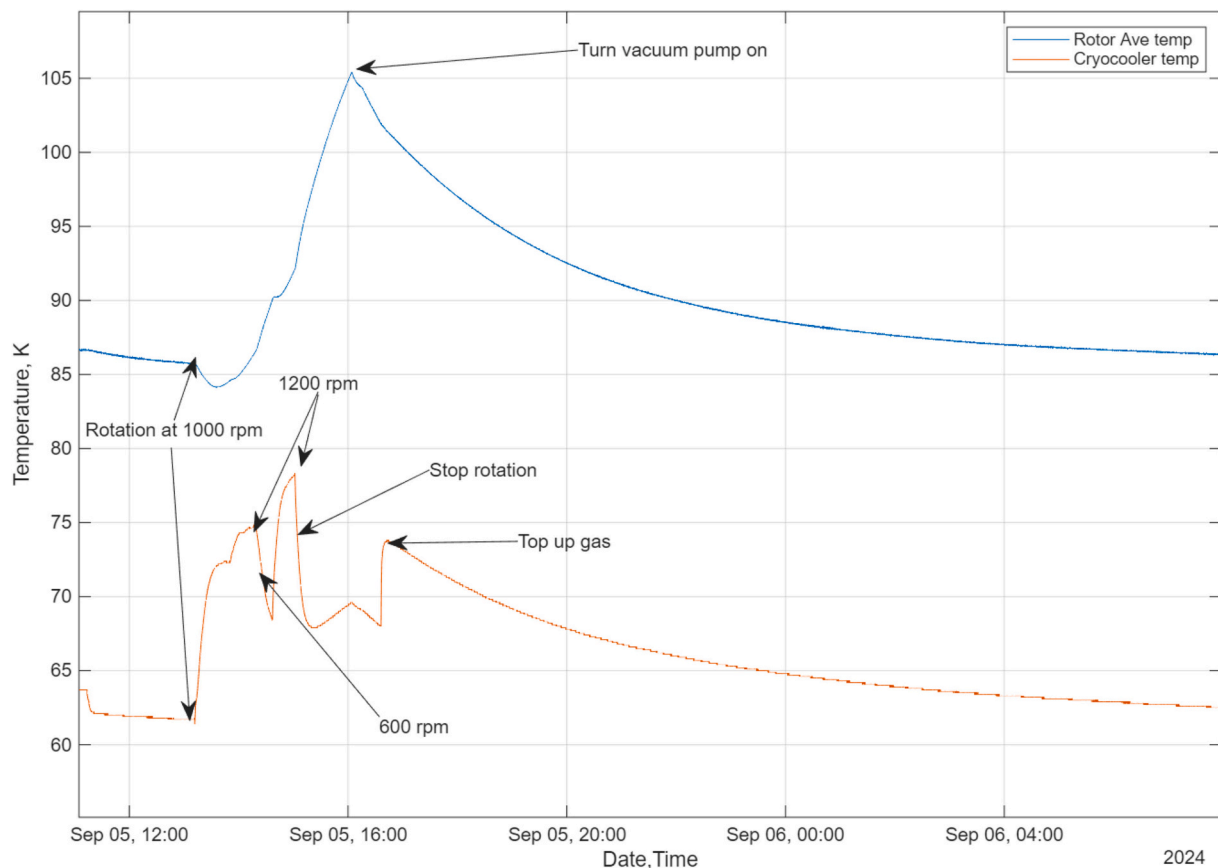


Fig. 21. Rotation test with new heat exchanger showing cryocooler temperature and rotor average temperature for a variety of rotation speeds and with the vacuum pump connected while stationary.

tube which has a much lower conductivity.

The stationary cooldown experiments showed that gravitationally induced convection in the helium space was effective in cooling the rotor with a temperature difference of 16 K between the cryocooler and the rotor. The temperature patterns were consistent with the CFD model of the stationary cooldown. The measured warm-up rate of the rotor was consistent with the heat leak calculation by the CFD model. Additionally, the cryocooler's cooling capacity, at its measured temperature, was close to the measured and predicted heating of the rotor via conduction, radiation, and convection through the gas adjacent to the cryocooler. Increasing the thermal efficiency of the rotor design to reduce the heat load so a cryocooler such as the SunPower DS30 or smaller can be used is desirable from a weight perspective on an aeroplane, however the cooling power available becomes small when compared with the thermal heat capacity of the rotor, which results in a very long cool-down time. The cool-down time can be optimised by reducing the weight of the cold rotor but is still expected to be many hours. An active commercial aircraft would need to keep the refrigeration running continuously to keep the rotor cold and motor ready for use, only warming up if un-operational for several days. The advantage of the pumped rotor over a conduction cooled system is that the helium itself has a low heat capacity compared with an equivalent thermal link and will therefore speed up the cooldown time.

The main limitation of the concept is that it has a gas space to maintain with a moving seal. Leakage from the lip seals was manageable for this set of experiments indicating that with optimisation and development, there is potential for a sealing system that is simpler than ferrofluidic seals. Ferrofluidic seals remain a backup option should an acceptable helium leakage rate not be achieved. Maintenance of the gas pressure on a mobile application would require a buffer tank and a small high-pressure tank to make up for any leakage.

The concept of incorporating a circulating pump and helium gas to the centre of a rotor has been shown to be a compact and effective way to transfer heat from the rotor to the cold heat exchanger. The drawbacks of this concept are that a rotary gas seal is required, and that cooling is not as effective while stationary as it is while rotating. However, the advantages of the concept are:

- The ability of the helium to transport heat a long distance with little temperature rise is of great use for large rotors where there is a long distance between the coils and the cold heat exchanger.
- Adding more cooling power capacity can be done by increasing the flow area for the pumping, which is lighter than adding more cross-sectional area for conduction.
- The direction of the gas flow can mean that the heat loads from the cryocooler end are intercepted at the end of the flow path, leading to a lower coil temperature for a given over-all heat load.

With a more powerful cryocooler, capable of more than 20 W at 50 K, the rotor coils of this design would be able to be pre-cooled and maintained at below 75 K whilst stationary. This temperature is sufficient for superconductivity to occur with a low current in the coils enabling the motor to be started. Once rotation starts, the rotor would then cool down further, enabling a full power current in the coils.

The pumped rotor concept lends itself to replacing the cryocooler with a heat exchanger cooled by a circulated cryogen, such as liquid hydrogen cooled high-pressure helium gas as might be used to cool the motor's stator windings in aerospace applications. This method retains the ability to have a high-pressure cryogen with a low-pressure in the rotor, reducing the pressure difference across the rotating seal, enabling lip or ferrofluidic seals to be used. An added advantage of utilizing the stator's cooling circuit is that a liquid hydrogen cooled stator would run

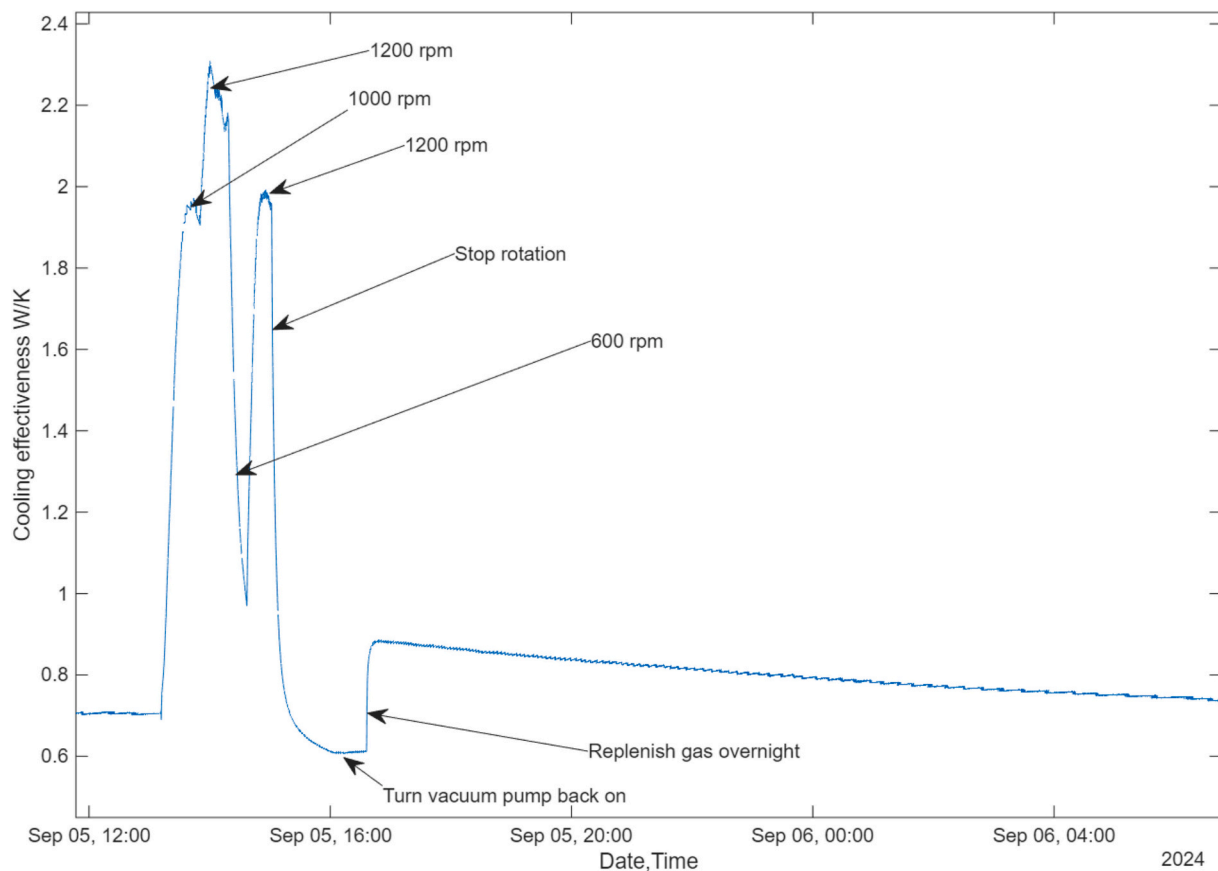


Fig. 22. Conductance between cryocooler and rotor average test during rotation test with new heat exchanger. These figures were calculated using the rotor and cryocooler temperatures plus the measured cooling curve of the test DS30 cryocooler.

at a lower temperature, around 30–35 K. This makes the stationary (startup) case more feasible as the rotor could reach at 50–55 K with 20 W of heat leak with 1 W/K conductance, which is sufficient for high current in superconducting coils.

4.1. Future work

The current prototype has achieved its purpose of showing that the pumped rotor concept can aid heat transfer around the rotor and has highlighted areas for improvement towards a design that can keep the superconducting coils sufficiently cold. The highest heat load in the experimental design was from conduction in the thin-walled tube at the cryocooler end. The second highest heat load was conduction at the driveshaft end. Replacement of the driveshaft and thin-walled tubes with a section of composite tube will reduce the heat ingress to the cold section of the rotor. Both the modelling and experimental work used the poor conductivity of stainless steel to highlight the temperature distribution in the pumped helium gas. In a superconducting motor's rotor, it is desirable to have a high conductivity rotor material to achieve a homogenous temperature for mounting the coils. With the thermal insulation provided by using G10 composite for the shafts at either end of the rotor section, the use of a more conductive material, such as Aluminium or copper, for the rotor becomes possible. The next iteration planned for the rotor development is to make a rotor long enough to mount coils, have an aluminium rotor body, a composite driveshaft and composite thin wall tube at the cryocooler end. A preliminary design analysis indicates a total heat load of less than 10 W which will allow the DS30 cryocooler to achieve a temperature close to 50 K and cool the coils to around 55 K. Once a functioning rotor with energised superconducting coils has been demonstrated, then the next steps would be to optimize the concept for the operational conditions of an aircraft motor.

CRediT authorship contribution statement

Alan Caughley: Writing – original draft, Visualization, Validation, Supervision, Software, Project administration, Methodology, Investigation, Formal analysis, Conceptualization. **Grant Lumsden:** Writing – review & editing, Supervision, Resources, Project administration, Methodology, Funding acquisition, Conceptualization. **Michael Gschwendtner:** Writing – review & editing, Supervision, Methodology, Investigation, Conceptualization. **Sangkwon Jeong:** Writing – review & editing, Supervision, Methodology, Investigation, Conceptualization. **Nic Rogers-Rehn:** Writing – review & editing, Validation, Methodology, Conceptualization. **Ryan McKinlay:** Writing – review & editing, Validation, Supervision, Methodology.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

The authors do not have permission to share data.

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