



Conceptual mechanical design of a rotating alpha-type stirling cryocooler for superconducting motor cooling[☆]

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ABSTRACT

Superconducting electric motors are an enabling technology for the goal to electrify aircraft propulsion. A major challenge in achieving this goal is the cooling of superconducting rotor coils. Previous rotor cooling systems have used cryogenic thermosyphons or gaseous helium circuits connecting the rotor to a refrigeration plant, commonly using Gifford McMahon cryocoolers. To enable the transfer of fluid from the stationary to the rotating side, sometimes ferrofluidic seals are used, which are best suited to slow shaft speeds. For higher speeds, axisymmetric Stirling and pulse tube cryocoolers mounted on the rotor have been proposed for direct conductive cooling of the rotor. However, the suitability of pulse tube cryocoolers for commercial aircraft propulsion is questionable, as their efficiency depends on orientation during operation. In contrast, Stirling cryocoolers provide higher efficiency than pulse tube cryocoolers with orientation independence, particularly at low temperatures. They therefore seem to be a promising option for superconducting rotor cooling. However, commercially available Stirling cryocoolers have not been designed for high-speed rotation. This paper presents a new alpha-type Stirling cryocooler that is specifically designed for high-speed rotation and superconducting rotor cooling. An overall axisymmetric design is proposed in which both pistons are supported by non-contact gas bearings and driven by stationary linear motors which are outside the rotating gas circuit, using the magnetic field to bridge the stationary-rotating interface.

1. Introduction

Commonly, the rotor coils of superconducting electric motors are cooled by cryocoolers, of which three types are considered in this paper: Stirling, pulse tube, and Gifford-McMahon (or G-M in short) cryocoolers. All of them have been used for the cooling of superconducting rotor coils but all the prior art solutions come with disadvantages that limit their use, especially for high-speed applications considered in this paper in the range of 4500–6000 rpm.

G-M cryocoolers provide a high cooling capacity at moderate cost, however, their efficiency is comparatively low due to irreversible expansion through the valves. Due to the usage of PTFE-based rubbing seals for the displacer piston in the cold head, they require frequent maintenance with intervals between 9000 and 18,000 h of operation [1,2]. In addition, they are commonly connected to a pipeline that provides compressed working fluid by a standalone compressor. This makes G-M cryocoolers impractical for being mounted on a rotating

system at high speeds. Nevertheless, G-M cold heads have been mounted on a low-speed rotor of a superconducting wind turbine generator for the SUPRAPOWER project [3]. In this case, the cold heads were operating with pressurized helium between 25 and 18 bar, that was supplied by a rotary helium union specifically designed for that purpose. The rotary helium union used 132 stages of ferrofluid seals to overcome the pressure difference which resulted in a length of almost one meter. It was tested up to 33 bar and 150 rpm without measurable leakages. A similar approach was used for another superconducting wind turbine generator project called EcoSwing [4]. This time, however, the rotary helium union was a slightly modified standard industrial component that was designed for a maximum rotational speed of 25 rpm at 24 bar while using a PTFE composite slip ring seal [5]. The leak rates were low enough to allow operation for about a year without refilling helium. Stationary G-M cryocoolers in combination with cryogenic thermosyphons were also used for superconductive rotor cooling systems. In this case, only the working fluid of the thermosyphon needs to be sealed by a

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rotary joint at a much lower pressure compared to the working fluid of the cryocooler. Typically, neon is used as a working fluid for the thermosiphon and ferrofluidic seals for the rotary joint [6,7]. Even though high pressure is not a problem here, the ferrofluidic seal must be isolated from the cryogenic temperature of the working fluid, which still results in a large overall length of the rotary joint. As all the mentioned rotor cooling solutions utilizing G-M cryocoolers have only been tested at low rotational speeds up to 230 rpm, they are not expected to be suitable for commercial aircraft propulsion. Nevertheless, a publication by Nick et al. [8] indicates that a fully superconducting generator can operate at 3600 rpm when using the combined G-M / thermosiphon cooling even though no details about the rotor cooling have been published.

The main advantages of pulse tube cryocoolers are their high reliability and low vibration levels due to the absence of moving parts in the cold head. However, there are several disadvantages associated with pulse tube cryocoolers that make their suitability for cooling superconducting rotor coils questionable. Firstly, according to the cryocooler survey of Ladner, even the best performing pulse tube cryocoolers reach efficiencies between 10 % to 15 % of Carnot in the temperature range of 50 K to 65 K while Stirling cryocoolers reach about 25 % [9]. Secondly, pulse tube cryocoolers work best when operating vertically, and a change in position can lead to instabilities in the gas column that reduces efficiency. Swift and Backhaus demonstrated experimentally and analytically that these instabilities can be suppressed with high operating frequencies and low aspect ratios (diameter/length) of the pulse tube [10]. However, the frequency of 100 Hz they used in their experiments will be detrimental for the efficiency due to a significantly reduced regenerator performance above 60 Hz [11]. Despite that, a pulse tube cryocooler is currently being developed at NASA's Glenn Research Center to cool the superconducting rotor coils of a High Efficiency Megawatt Motor (HEMM) for electrified aircraft propulsion [12]. Experimental validations of the general feasibility of rotor-mounted pulse tube cryocoolers have been published [13,14].

Stirling cryocoolers, as the last considered candidate, offer higher efficiency compared to the G-M and pulse tube type with orientation independence, particularly at low temperatures. With an actively driven piston and displacer, Stirling cryocoolers can further be tuned to remain efficient over a wide variety of operating conditions [15]. Free-piston Stirling cryocoolers (FPSCs) are particularly suitable for being mounted on a rotating system at high speeds as the absence of any mechanical linkage between power piston and displacer allows an overall axisymmetric design. As another advantage, FPSCs can be operated without any contact between moving parts by utilizing precisely aligned clearance seals or self-centering gas bearings. This design feature enabled a free-piston Stirling generator to demonstrate an accumulated maintenance-free operation of 16.1 years (140,611 h) during an ongoing lifetime test, which is considered as a world record [16]. The low reliability often attributed to Stirling cryocoolers [1,2] is mainly due to insufficient differentiation between crank-driven and free-piston machines. As a result of their rubbing piston seals and their higher operating frequencies compared to G-M cryocoolers, crank-driven Stirling cryocoolers require even more frequent maintenance intervals of 6000 h [17], making them inadequate for cooling superconducting power systems [18]. In previous projects on superconducting rotor cooling, commercial off-the-shelf FPSCs have been used [19] Available product families include relatively small beta-type FPSCs which are overall axisymmetric and air cooled, as well as larger gamma-type FPSCs with a split configuration (meaning that compressor and displacer have a separate housing) and water cooling [20]. Commercially available beta-type FPSCs can be operated on a rotor, but they do not provide sufficient cooling capacity for the intended superconducting rotor cooling that requires at least 20 W at 50 K (see next section). For example, the cooling capacity of the Sunpower CryoTel® GLT cryocooler is limited to 8 W at 50 K [21]. Commercially available gamma-type FPSCs such as the Sunpower DS30 are besides their insufficient cooling capacity of about 12 W at 50 K [20] not suitable for rotary operation due to their

unbalanced masses and the need for cooling water supply. Nevertheless, a stationary Sunpower DS30 cryocooler has been mounted on the shaft of a superconducting rotor to simultaneously cool and drive the flow of a helium cooling loop [22]. Due to the relative motion between the cryocooler and the rotor, this arrangement acts like a pump. Even though the helium cooling loop is at low pressure, a dynamic seal is required, which is prone to leakage and heat generation. Ultimately, FPSCs appear to be a promising option for direct superconducting rotor cooling by heat conduction that prevents the need for dynamic seals. However, none of the commercially available FPSCs are designed for a high cooling capacity at high-speed rotation. This paper presents a new mechanical design concept for an alpha-type FPSC that is specifically designed for high-speed rotation up to 6000 rpm and superconducting rotor cooling.

2. Cryocooler requirements

As a boundary condition for the alpha-type FPSC to be developed, a fully superconducting motor concept according to Jeong et al. is considered [23]. This concept aims to provide a 4500 rpm air-core 3 MW aircraft propulsion motor. Multifilamentary MgB₂ cables are considered for the AC carrying stator coils that are to be cooled by liquid hydrogen. It is assumed that liquid hydrogen will be available as it can serve as coolant and energy source simultaneously. The DC carrying rotor coils shall use REBCO (Rare-earth barium copper oxide) coated conductors that need to be cooled to 50 K. The original concept specified a cooling capacity for the rotor coils of 100 W at 50 K, but recent studies suggest that 20 – 50 W at 50 K will be sufficient [22]. A rotor-mounted cryocooler that eliminates the need for a rotating seal or gas coupling therefore represents a technically feasible option for rotor cooling. A schematic view of such a cryocooler mounted on the rotor of a superconducting motor is shown in Fig. 1.

3. Cryocooler conceptual design

Two design options are considered for the FPSC, both based on a patent by Beale [24]. His disclosure shows a similar alpha-type machine with a concentric piston arrangement, but the mechanical structure is merely indicated by a simple combination of blocks representing the main components. Therefore, several extensions and some deviations were necessary to obtain the concepts proposed here, the first of which is shown in a schematic cross-sectional view in Fig. 2. The term 'alpha-type' might be confusing here since Kirkley originally defined alpha-type machines by a "working space [that] is divided between two separate cylinders one of which forms the hot space and the other the cold space." [25]. However, Kühl gave a more general definition of alpha-type machines by stating "that for each temperature T_i there exists only one cylinder volume V_i in which the entire required volume change is caused by the movement of a single, one-sided acting piston" [26]. In other words, the distinguishing criterion is that in the Stirling machine shown in Fig. 2, both pistons contribute to the compression and expansion of the working gas. In contrast, in beta-type machines only

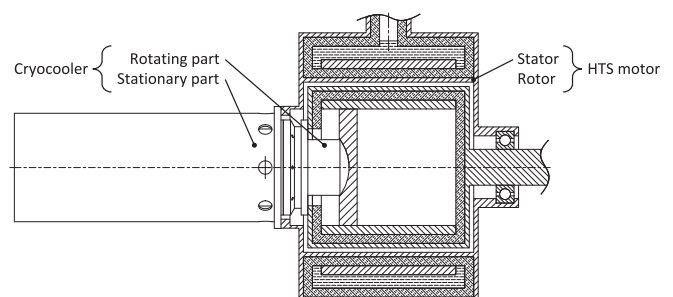


Fig. 1. Cryocooler mounted on the rotor of a superconducting motor.

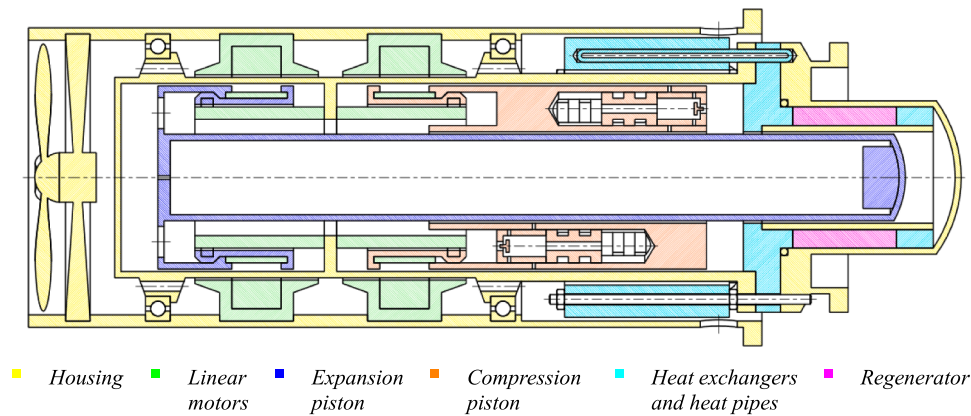


Fig. 2. Design option for the rotor-mounted cryocooler with linear motors for compression and expansion piston.

the power piston contributes to the compression and expansion of the working gas, while the displacer merely shifts the working gas back and forth between the compression and expansion space (the influence of the relatively thin displacer rod is considered to be negligible in this case).

The alpha-type FPSC in Fig. 2 is mainly characterized by an inner expansion piston that consists of a continuous cylinder and is slidably mounted in the central bore of a hollow cylindrical compression piston. Both pistons are guided by self-centering aerostatic gas bearings, whose pressure is supplied internally by the compression pressure of the pistons and buffered in a cavity within the compression piston by the use of miniature ball check valves. By releasing the pressure through throttles in both bearing gaps, the pressure directly behind the throttles becomes a function of the gap height, which creates a self-centering force. The expansion piston has a much larger projected bearing area than the thin displacer rod of beta-type FPSCs and can therefore accommodate additional forces due to possible imbalances during rotation. The risk of rubbing on the displacer of conventional beta-type FPSCs when simultaneously oscillating and rotating at cryogenic temperatures was already pointed out by Dyson et al. [12]. This is mentioned as one reason why they prefer a pulse tube cryocooler to a more efficient Stirling type for the rotor cooling of their High Efficiency Megawatt Motor (HEMM).

An important aspect of the suggested alpha-type FPSC is the heat removal from the ambient side heat exchanger. According to initial simulations by Sage, this heat could be in the range of one kilowatt, which will be challenging to dissipate. Commercial cryocoolers with a high cooling capacity commonly use a water-cooling jacket, however, this solution would be impractical for a rotating cryocooler. Convection cooling could be enhanced by rotation but the cooling capacity must also be guaranteed when the shaft is at rest in order to maintain the operational readiness of the aircraft on the ground. To solve this problem, the current mechanical design suggests heat pipes to transfer heat from the relatively small ambient heat exchanger to a large area convection cooled heat sink. It has already been reported that rotating heat pipes can be used advantageously for cooling the rotor of electrical machinery [27]. These heat pipes are typically installed directly in the shaft and have a conical inner wall instead of a wick to transport the liquid back to the evaporator through centrifugal force. However, since the alpha-type FPSC does not have a central shaft, the heat pipes need to be placed at a radial offset. This configuration has already been successfully tested by Groll et al. in a rotor cooling application with speeds of up to 5000 rpm [28]. The heat pipes were mounted parallel to the rotational axis with a radial offset of 10 cm and the liquid transport was caused by minor level differences of the liquid puddle between condenser and evaporator. In order to provide a pumping force even at rest, capillary grooves in the heat pipes were necessary. A similar cooling method was presented by Anderson, in which standard wick heat pipes are embedded slightly tilted and radially offset on a rotating shaft [29]. This solution is considered to be a more cost-effective and higher-performance

alternative to a traditional rotating heat pipe. Therefore, it represents the most promising option for heat removal from the alpha-type FPSC's ambient heat exchanger. After the heat was transferred to the radial fin heat sink by the heat pipes, it can be removed convectively by a fan inside a flow pipe. The flow pipe serves as stationary housing for the rotating part of the alpha-type FPSC at the same time. Fig. 3 shows the proposed alpha-type FPSC in a different color scheme indicating rotating and stationary part.

In the first design option, it is intended that both pistons are driven and controlled separately by moving magnet-type linear motors. However, thermodynamic modeling in Sage revealed that while maintaining a proper phase shift the expansion piston's linear motor would partially act as an alternator during the working cycle. This can be explained by the fact that the expansion piston absorbs or rather cushions the compression work in the working gas. As a result, work is performed on the expansion piston and not consumed. This does not affect technical feasibility, but the expansion piston's linear motor must be able to alternately switch from motor to alternator operation. The electrical energy generated must be fed back to the compression piston's linear motor. Compared to a single linear motor design, this operation involves additional lossy energy conversions, which must be minimized by appropriate tuning of the piston movement. Another challenging aspect of using separate linear motors for each piston is that the permanent magnets of the expansion piston need to be placed at an axial distance from the lower piston portion. This shifts the center of gravity of the expansion piston out of the area supported by the gas bearing (in the horizontal orientation) and causes a high leverage force on the gas bearing. In order to keep the center of gravity approximately in the middle of the supported area, it would be necessary to attach a counterweight to the tip of the expansion piston.

The linear motors not only provide (or recover) the expansion work, they also act as a stationary-rotating interface for energy transfer by placing the outer stators and coils outside the rotating gas circuit. Previous solutions for rotor-mounted cryocoolers required additional means for the energy transfer such as slip rings or rotational transformers [12,14,19]. However, the suitability of slip rings providing several kilowatts input power at up to 6000 rpm is questionable due to the friction, while rotational transformers will increase the system weight due to the high density of copper coils. Another advantage of placing the outer stators and coils of the linear motors outside the rotating gas circuit is that the buffer space volume can be reduced to approximately the same volume as the working space. This will strongly reduce the shift of the piston's center position during operation. As described in detail by Walker and Senft [30], this shift is caused by unequal pressure amplitudes in the buffer and working space, resulting in a net unidirectional leakage gas flow in the piston gap. It is assumed that the pressure equalization paths used to overcome this problem in commercial FPSCs [31] can be omitted in the design concept presented

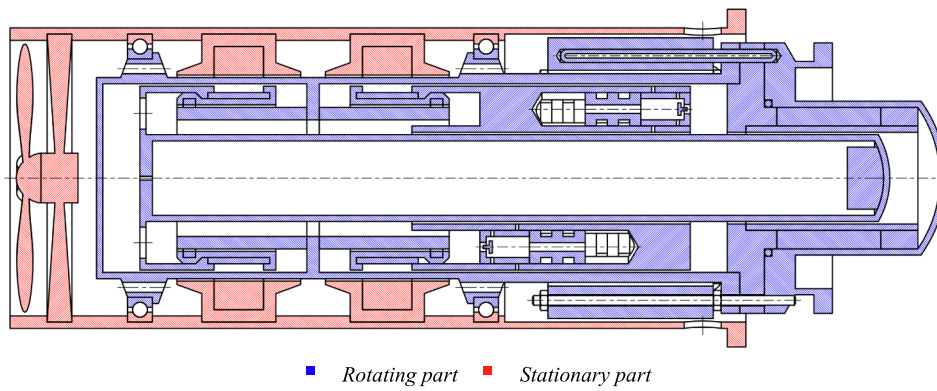


Fig. 3. Rotor-mounted cryocooler from Fig. 2 in a color scheme indicating rotating and stationary part.

here because similar pressure amplitudes will occur in the buffer and working space. Small disturbances in the pressure amplitudes can be compensated by the cogging force of the linear motor, which acts like a self-centering spring [32]. The outer stators of the linear motors are preferably made of laminated sheet metal blocks arranged around the circumference. This arrangement automatically creates gaps that allow the cooling air flow to pass through easily.

The first design option initially appeared promising, but the original idea of distributing the required drive power between two linear motors while simultaneously being able to control the optimal phase shift between compression and expansion piston proved to be unrealistic. Therefore, a second design option was elaborated, which is shown in Fig. 4.

The main difference compared to the first design option is that only one linear motor for the compression piston is used while the expansion piston is solely connected to a mechanical spring, which provides support and a defined starting position. In commercial FPSCs, planar flexure springs are commonly used to avoid side forces on the pistons. However, these springs typically require a large diameter that would be detrimental for the concept proposed here, as the buffer space volume and the centrifugal forces would increase. Instead, double start machined springs could be used to resolve all internal moments in the spring itself, as already suggested in a patent of Pellizzari et al. [33]. The compression piston does not require an additional spring because it is held in a starting position by the cogging force of the permanent magnets. The second design option is still considered an alpha configuration since the volume changes in the compression and expansion spaces are entirely caused by the respective pistons according to Kühl's criteria cited above. In contrast, the compression space volume of beta-type FPSCs is affected by the displacer as well, leading to differences in the operational

characteristics. As already pointed out in the Beale patent [24], this design uses a gas-spring coupling between the two pistons for indirect transfer of PV power from expansion space to compression space rather than providing mechanical transfer via a displacer as is the case of a beta configuration. However, after initial Sage simulations, it proved difficult to tune the model for an appropriate phase shift and to prevent overstroke of the free expansion piston. It seems that the elimination of the commonly used piston rod also means that an important design parameter for tuning the free piston movement is missing. Nevertheless, the overstroke problem might be overcome by using a nonlinear spring with higher stiffness at the end of the intended free piston movement. Although the second design option has the potential to simplify the mechanical design and control of the linear motor, the thermodynamic design proved to be more challenging compared to the first design option.

4. Gas bearing mechanical design

An essential aspect of the suggested alpha-type FPSC is the gas bearing, which is identical in both design options. It is located in the compression piston and enables a lightweight piston design, even when using stainless steel, as well as simple, high-precision manufacturing. An exploded view of the compression piston assembly is shown in Fig. 5.

An internal circumferential cavity, which serves as a high-pressure reservoir, is created by axial blind bores on both sides of the compression piston in a rotationally offset manner relative to one another, whereby they connect inside. In addition, each axial blind bore is connected to a tiny radial through bore. The axial blind bores are filled from both sides with spacer and throttle inserts. The spacer inserts act as a depth stopper for the throttle inserts and adjust the cavity volume to a

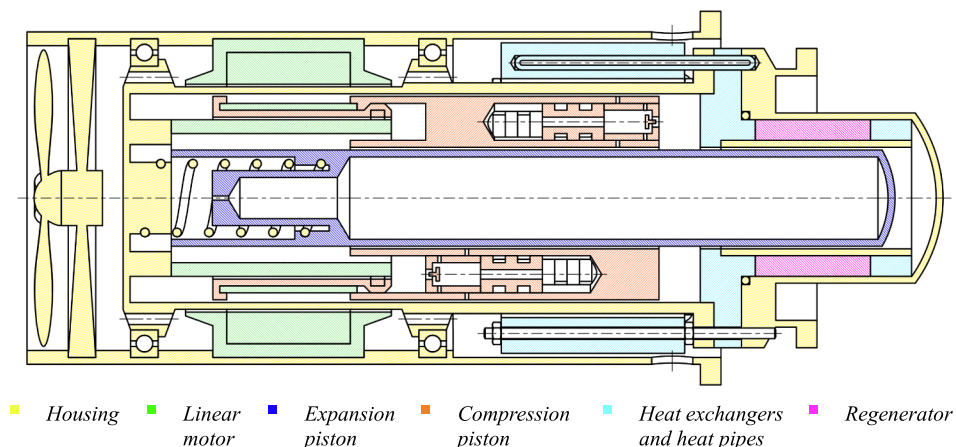


Fig. 4. Design option for the rotor-mounted cryocooler with linear motor only for compression piston.

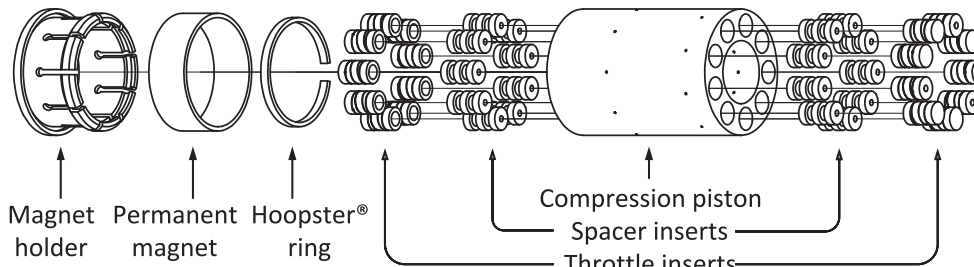


Fig. 5. Exploded view of the compression piston assembly.

desired value. The throttle inserts sealingly close the bores by a press fit and provide fluid connection paths between the high-pressure cavity and the bearing gaps of both pistons. One of these fluid connection paths that includes a tiny helical throttling groove at the outer circumference of the throttle insert is illustrated in Fig. 6. Gas bearings in commercial FPSCs use similar tiny throttling grooves located at the interface between two press-fitted sleeves that form the piston [34]. This leads to a relatively bulky piston design compared to the solution presented here and requires aluminum as the piston material for weight reduction. However, when using aluminum for the piston, it needs to be used for the cylinder liner and the displacer rod (or rather the expansion piston) as well in order to keep the tight tolerances for the gas bearing even at small temperature differences. Using aluminum would be impractical for the design presented here, as it would cause an increased heat leakage path in the expansion piston and a reduced magnetic flux transmission of the linear motors through the cylinder liner, which would then need to be thicker.

A patent of Cale and Stanley indicates that the cylinder liner of the gas bearing is very sensitive to deformation caused by internal pressure [35]. This is due to the very small clearance gap of gas bearings for Stirling machines, which typically have a height of about 15 μm [36] and must be maintained over the full bearing length. Since the cylinder liner in the proposed concept is also the outer wall of the pressure vessel, deformations caused by the internal pressure have a direct effect on the gas bearing gap. Calculations have shown that the outer wall of the pressure vessel made of stainless steel must have a thickness of 5 mm in the gas bearing area to limit the gap height increase to 3 μm when the internal pressure is 30 bar. This expansion corresponds to 20 % of the nominal gap height and appears to be manageable since it can already be taken into account during tolerancing. As an alternative solution, the heat sink of the ambient heat exchanger could be made of copper with a thermal expansion almost the same as stainless steel and used to

reinforce the pressure vessel whose outer wall could then be thinner.

5. Gas bearing analytical design

Although the mechanical design of gas bearings for Stirling machines is well described in the patent literature [34,37,38,39], only little literature is available on the analytical design. Either proprietary codes are reported to be used [40], or the calculations presented are based on experimental data [41] and therefore cannot be used for the design of new machines. Consequently, a Simulink model was developed to dynamically simulate the load capacity of the gas bearing. The original version of the Simulink model is described in detail in the corresponding author’s Master’s thesis [42]. This original version was slightly modified for the cryocooler design presented here. Fig. 7 shows the top-level of the adapted Simulink model. Basically, the Simulink model represents a fluidic network in which bearing gaps and throttling grooves between volumes are considered as resistances. All varying volumes are calculated from the sinusoidal piston movements. The pressures corresponding to these volumes are calculated according to the ideal gas law and the summation of all incoming and outgoing mass flows. The load capacity of the gas bearing arises from the force component in the direction of the eccentric deflection and can be obtained by integrating the pressure profile in the bearing gap. In addition to the load capacity, the Simulink model continuously calculates the overall mass of the working gas as a control criterion. Since the overall mass needs to stay constant at all times, a deviation would imply an error in the Simulink model.

The Simulink model divides the bearing gaps into segments whose number depends on the number of throttle inserts. As a simplifying assumption, these segments are considered as rectangular channels with parallel flow from the throttle outlet to the working or buffer space and between the upper and lower throttle outlets (in axial direction). However, the parallel flow assumption is only true when the piston is exactly centered in radial direction. In reality, the piston weight causes eccentricity, which is further amplified by linear motor side forces [43]. This eccentricity leads to a circumferential flow from the side with a decreased bearing gap (and increased pressure) to the side with an increased bearing gap (and decreased pressure). To account for this circumferential flow, the resulting load capacity for the parallel flow is multiplied with an empirical correction factor that was originally defined by Shires and Pantall [44].

The Simulink model of the gas bearing includes a simplified isothermal model of the Stirling process. However, it was found by comparison with Sage simulations that the isothermal model underestimates the pressure amplitude of the Stirling process. This behavior was already observed in earlier publications [45,46]. According to an explanation of Hoegel, the Sage model considers “adiabatic effects [that] heat and cool the working fluid beyond the temperatures of the heat exchanger walls. The higher temperature amplitude results directly in a higher pressure amplitude. In the isothermal model, the gas temperature is always constraint by the temperature of the heat exchangers and the pressure amplitude is thus smaller for otherwise identical conditions.” [45]. The underestimated pressure amplitude leads to the load capacity of the gas bearings being underestimated as

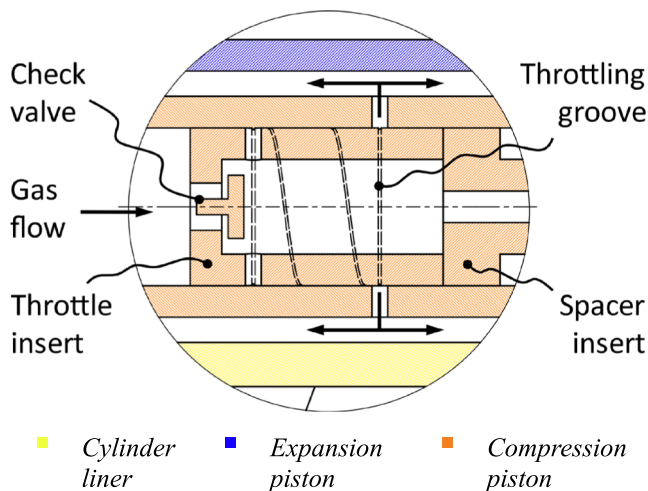


Fig. 6. Detailed view of a throttle insert when installed in the compression piston.

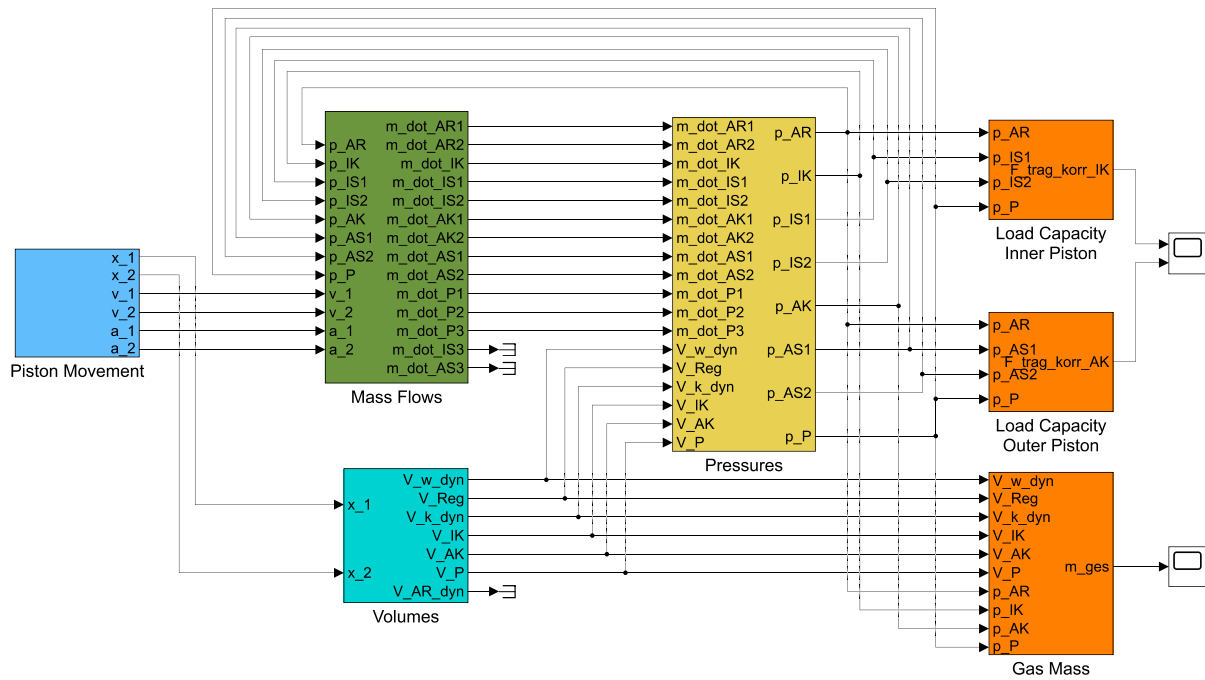


Fig. 7. Top-level of the Simulink model for the gas bearing.

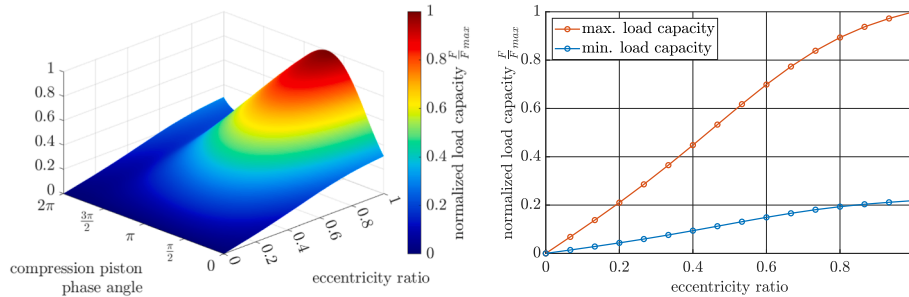


Fig. 8. Left: 3D surface plot of the gas bearing load capacity depending on phase angle and eccentricity ratio. Right: 2D plot of maximum and minimum gas bearing load capacity depending on eccentricity ratio.

well. Consequently, the isothermal model will be replaced by Sage outputs once they are finalized.

Fig. 8 shows a typical output of the Simulink model for the load capacity of the gas bearing. Since there will be changes to the current design, absolute values for the load capacity are not meaningful at this time. Therefore, the load capacity is normalized by the maximum values. The load capacity curve looks very similar for both pistons, but the expansion piston has 58 % smaller absolute values than the compression piston. These lower absolute values can be explained by the smaller projected bearing area and a less favorable length/diameter ratio of the expansion piston gas bearing. Simulations performed so far with the Simulink model suggest that the gas bearing design is capable of carrying both pistons with a large safety factor regarding load capacity. Further optimization of the gas bearing will focus on ensuring a sufficient load capacity while simultaneously minimizing the pressure loss in the Stirling process. The current design of the gas bearing reduces the pressure amplitude of the Stirling process by about 5 % (based on the isothermal model).

6. Conclusion

Existing off-the-shelf cryocoolers have never been intended for operation under high rotational speeds. Therefore, it is recommended to

adapt the cryocooler design to the particular application of superconducting rotor cooling to ensure efficient and reliable operation. A key for reliable operation is the avoidance of rubbing seals on the rotor. This is comparable to the development of commercial Stirling cryocoolers where the introduction of contact-free clearance seals combined with planar flexure springs lifted the specified lifetime from typically 0.5 to more than 5 years [47]. Two options are presented for a new rotor-mounted alpha-type FPSC that inherently avoid dynamic seals. To ensure reliable operation, both pistons are guided by contact-free gas bearings, for which a new mechanical design as well as a new analytical Simulink model is presented. Further work on the thermodynamic design of both concepts will be necessary in order to finally evaluate their potential.

CRediT authorship contribution statement

Bruce Fischer: Writing – original draft, Conceptualization. **Michael Gschwendtner:** Writing – review & editing, Supervision, Funding acquisition. **Alan Caughley:** Writing – review & editing, Supervision.

Declaration of competing interest

The authors declare the following financial interests/personal

relationships which may be considered as potential competing interests: [Bruce Fischer reports financial support was provided by New Zealand Ministry of Business, Innovation and Employment (MBIE). Bruce Fischer has patent Rotierende Stirling-Maschine in Alpha Anordnung pending to assignee. If there are other authors, they 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

Data will be made available on request.

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