Interdisciplinary Design of a High-Speed Drivetrain for a Kinetic Compressor in a High Temperature Heat Pump

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The authors would like to express their gratitude to Business Finland (grant decision 6430/31/2019, High Temperature High Efficiency Oil-free Heat Pump) for their financial support.

ABSTRACT

Bearingless motor (BM) technology offers a competitive solution to active magnetic bearings (AMBs) in high-speed applications. A bearingless drivetrain (BDT) working as a part of a kinetic compressor in a heat pump exceeding 0.5 MW and 25 000 rpm is open for questions: how should the geometry, materials and parameters be chosen without exceeding electromechanical and thermal limits; what rotor configuration suits the heat pump; how can the working fluid of the heat pump be used in the motor cooling? This paper presents the interdisciplinary design process of a BDT as a part of the compressor of a high temperature heat pump (HTHP). The motor — as the core driving the compressor of the heat pump, each interacting with the other, bringing about various design challenges — design process is summarized in a flowchart and subsequently elaborated. Dimensioning rules for the bearingless structures are presented. Finite element analysis (FEA) is used in evaluating the considered surface and interior permanent magnet BM structures. Different stack materials are considered and preliminary electromechanical optimization by a genetic algorithm is carried out. To ensure structural rigidity, FEA of electromagnetic, mechanical, thermal, and rotor dynamic performance is conducted. FEA of drive losses is carried out and cooling alternatives are considered, based on which temperature distributions and thermal stresses with the chosen cooling arrangements are calculated again with FEA. Power and performance of the structures are discussed. As a result of the analyses, a suitable hermetic motor design is chosen for the HTHP compressor drive.

INDEX TERMS

Bearingless motor, interdisciplinary, multiobjective optimization, electromagnetic analysis, high-speed, high-temperature heat pump application, kinetic compressor, rotor dynamics, thermal analysis, thermal stresses

I. INTRODUCTION

In recent years efficient and ecological heat production has followed a global uptrend of sustainable energy. A very efficient and ecologically sound solution in heat production is offered by heat pumps [1], which can provide low-temperature heat for buildings, factories as well as high-temperature heat for manufacturing processes of industrial goods. However, there are only a few specific heat pump solutions available, producing high-temperature heat above 100 °C [2].

Moreover, an efficient and environmentally safe high-temperature heat pump needs an environmentally acceptable working fluid, a compressor able to overcome high pressure ratios [3] with an oil-free design: bearings not lubricated by oil, unable to contaminate the working fluid and degrade heat transfer performance [4]. This solution can be achieved by incorporating a working fluid with ultra-low global warming potential (GWP) [3] into the process and integrating an oil-free high-speed compressor with a magnetically suspended rotor into the system.

In comparison to a traditional drive with an active magnetic bearing (AMB), a bearingless motor (BM) can offer a more compact structure, as torque and suspension are created with the same windings and iron [5]. With the same windings used for torque and suspension force production, an even more compact design can be achieved with a possibility of increased efficiency [6], [7]. Bearingless machines can be categorized based on the number of degrees of freedom (DOFs) controlled [8]. Different rotor topologies, e.g. the induction [8], homopolar [6] and permanent magnet [10], [11], [12], have been investigated. A structure with power of 60 kW e.g., in [11] is an example of a topology with permanent magnets. With the 4DOF configuration, shown e.g., in [10], the BM is
a suitable replacement for applications where AMB solutions are used.

In high-speed and high-power applications, such as turbo compressors, blowers, turbines and high-temperature heat pumps, the length of the rotor can become a limiting factor, as the critical speeds can hamper achieving the maximum speed of the application [13]. BM configurations are mostly tested in small power applications, but rarely above 100 kW [8] [10].

The design of high-speed applications is a multidisciplinary process, incorporating mechanical, electromagnetic, control, fluid dynamics, and thermal aspects, to achieve a feasible solution for the application [13],[17]. Multiphysical design, showing electromagnetic and thermal analyses of a 1.5 kW 5 000 rpm bearingless switched reluctance motor, was presented in [15], and coupled thermal and structural analysis of a permanent magnet motor supported by AMBs was reported in [16]. Furthermore, in [17] multiphysical design and optimization of an ultrahigh speed BM with thermal, structural and rotor dynamic analyses were carried out with simplified models. However, interdisciplinary design of a heat pump with a kinetic compressor with a bearingless drivetrain has not been reported in literature.

The main contribution of this paper is to introduce a multidisciplinary design procedure for a heat pump with a kinetic compressor, incorporating an industrial size bearingless drivetrain (BDT), and to provide information to assist in the dimensioning of the drive. By achieving greater power, bearingless machines could be used in industrial scale the way AMB supported machines are used today. The steps of the design process, involving multiphysics, are presented for a high temperature heat pump (HTHP) application, driven with a BM, not earlier reported in literature. The conceptual design of the motor windings is based on [14] [18].

The paper is structured as follows. The second section presents a multidisciplinary flow chart of the design: a BDT inside a compressor of a heat pump. It elaborates on the necessary design steps and provides guidance on how to overcome various challenges one could meet in the design of a compressor intended for high-speed operation in an HTHP and that of the drive within. Furthermore, the third section focuses on the general dimensioning rules of a BDT for a heat pump.

The fourth section presents the design of an industrial heat pump process and that of the used compressor. The fifth investigates bearingless drivetrain topologies in detail. SPM and IPM motor structures are considered, and preliminary optimization is carried out with the application in mind. Based on the optimization results and scaling rules, case studies segregating different structural design choices are established for comparative purposes.

The sixth section applies the guidance from section two and investigates the cases from section five by interdisciplinary design steps where FEA is used. This section presents a BDT concept that could be used e.g., for an HTHP. The results, incorporating multiphysics, are presented and the achieved power range of the BDT is discussed.

II. A HEAT PUMP APPLICATION – CHALLENGES AND DESIGN CHOICES

The design of the heat pump covers the process, compressor and drive design. The system in its complexity combines several fields of engineering, e.g., mechanical, electromagnetic, thermal, fluid dynamics and control. When high–temperature, high-power and high–speed operation comes into question, such design involves various interlinked challenges.

The design can be divided into three main design tasks, which are iterated until the performance of the system satisfies the requirements. Figure 1 depicts the overall design task. Furthermore, to answer the interdisciplinary challenges and to obtain a feasible design, a systematic approach is proposed and presented in Fig. 2, along with a breakdown of the key aspects.

The results of the heat pump process calculations define the requirements for the motor power and the operational speed ranges of the motor and the compressor. The required pressure rises as well as the mass flow rate are input parameters needed to proceed with the design of the compressor. The iteration of the motor and compressor designs involves the acceptable pressure drop and maximum rotational speed. The aerodynamic performance of the compressor and its mechanical strength and stresses constitute a checkpoint — the outputs of the design. Fulfilling the requirements, follows the possibility of iterating the electrical motor design. As regards a BM, capability of producing the required torque and suspension force allows further progress in the design. The cooling arrangement and rotor dynamic analysis are a key checkpoint when it comes to designing the BM further. The heat pump process, motor, and compressor designs are evaluated by checking the coefficient of performance (COP), mechanical and electrical power, and losses and efficiency.

FIGURE 1. The general flow of the design work from beginning to completion.
A. HEAT PUMP PROCESS
The heat pump design process is comprised of the following phases to meet the requirements of the targets of the application.

At the beginning of the heat pump design process, different heat pump cycle layouts, including one- and two-stage compression processes, as well as different working fluids are studied. Within two-stage compression processes, different options for intercooling are explored.

Suitable design values are evaluated based on the given data, typically including the power range as well as the heat source temperature levels in the evaporator and heat sink temperature levels in the condenser (Step A.1). In many cases, a wide operating range is beneficial so that the heat pump can be operated at various conditions. Thus, the operation and performance of the heat pump both in design and off-design conditions are considered. One of the most important choices is to select the appropriate design values for the heat pump process cycle and for the heat pump components to meet the requirements in both the design and off-design operating conditions. Typically, the condenser heat rate as well as the evaporating temperature and the condensing temperature are key design parameters.

In the early phase of the design process, the selection of a suitable working fluid is a major issue; the working fluid has a significant impact on the performance of the heat pump (Step A.2). The selection of a suitable working fluid depends on the power scales and operating conditions, including the temperature levels, as well as on the design of the heat pump components. In addition to the thermodynamic and thermophysical properties, the aspects related to ozone depleting potential (ODP) and GWP must be considered in the selection of the working fluid, along with other properties, such as toxicity, flammability, availability, price, and the compatibility with materials [22].

Thermodynamic computations of the heat pump cycle (Step A.2.1) result in the power need and speed range of the electric motor of the compressor unit as well as the COP for heating, an often used and essential indicator for describing the heat pump performance. The computations also result in the necessary data that defines the design values of the compressor unit, e.g., mass flow rate and the required pressure rise, including the rotational speed as well.

B. COMPRESSOR DESIGN
A two-stage compressor operates as a part of the heat pump system. The boundaries for the compressor design are set by the heat pump process, including the required pressure rise and mass flow rate of the working fluid, yielding the maximum rotational speed (Step B.1.1). Investigation of different compressor cooling arrangements is closely linked to the electric motor design while having a significant impact on the mechanical design (Step B.1 – Step B.2) as well. If a vaporeous working fluid is used as a cooling medium in the electric motor, enabling a hermetic design, the heat transfer properties, the pressure drop as well as the gas friction and gas flow losses are issues of import that are investigated computationally in the design phase [19] [36] (Step B.4).

It should be borne in mind that a high density, vaporous cooling medium can cause significant gas friction and gas flow losses in a system with a high-speed rotor, influencing the performance of the heat pump.

Concerning the high-speed compressors, the management of the compressor leakages from the impeller outlets to the space of the lower pressure level surrounding the shaft and the minimization of the residual axial force of the compressor shaft [23] must be considered in the design of the labyrinth seals and in the layout design (Step B.4 – Step B.5a & Step B.5b).

With regard to the materials, the impellers of the compressor should be light, maintain rigidity, and withstand the process temperature (Step B.5a). To improve the performance of the compressor, the rotational speed and impeller type and size can be reconsidered.

The above-mentioned decisions have significant impact on the design and performance of its components and the heat pump process cycle.

C. MOTOR & DRIVETRAIN DESIGN
At the earliest stage of motor design, the motor type, operational speed, and power are defined. The required motor power is defined by thermodynamic computations of a heat pump cycle. Furthermore, those computations along with the maximum rotational speeds of the motor define the boundaries for the compressor design.

The positioning of the drive, vertically or horizontally, has great bearing on the dynamics and required suspension force capacity, as vertical placement removes the need to counter the rotor weight with a current generated suspension effect [19]. However, a vertically oriented drive requires more from the axial AMB, having to support the drive, a higher force capacity is required. Whereas the design and placing of the axial bearing, back-up bearings, axial and radial sensors are based on the physical dimensions of the rotor of the compressor [49]. Provided that the drive is placed vertically, it is supported using a fixed number of supports, in contrast to horizontal placement, where the drive could be bolted on the floor across its entire length. In that case, the end shields support greater weight, resulting in lower axial critical speeds, leading to control challenges. Furthermore, depending on the drive orientation, the housing can have different vibration modes, the frame modes being excited by electromagnetic forces or loads present in the application. More challenging housing dynamics, studied in the section about the analysis of rotor dynamics, are present when the system is positioned vertically.

In general, a controllable and mechanically rigid BDT with high suspension force capacity, torque density and a wide operational speed range is desired for compressor applications.
FIGURE 2. Flowchart of the design process of a heat pump, involving a compressor with a bearingless drivetrain, incorporating multiple engineering disciplines.

VOLUME XX, 2021
However, such design objectives are met with corresponding design challenges, such as controllability, mechanical limitations, dimensioning of torque and suspension force, thermal and cooling design, and rotor dynamics. The relevant objectives and challenges are discussed together in the following subsections.

1. TORQUE AND SUSPENSION FORCE DIMENSIONING
A high-speed BM with high power density should be dimensioned by optimizing the ratio of rotor length to its diameter; dynamics prefer a shortest rotor, whereas low stresses a smallest diameter. By dimensioning the drivetrain well, high efficiency and torque density as well as sufficient suspension force capacity can be achieved.

Scaling the drive to equal or greater power can be done by maintaining a constant volume — replacing the materials or varying the rotor length and diameter inversely — or by increasing the overall volume by changing one or more dimensions.

Good suspension performance, characterized by low force error angle [25] and vector amplitude error as well as high force capacity, can be established by optimizing the motor geometry. Such optimization can involve finding the smallest possible air gap, optimal magnet arc in relation to the thickness of the magnet and size of the slot opening or skewing the rotor or the stator [12]. Mechanical performance limits the design choices that can be made in the course of the optimization, ruling out unfeasible designs [16]. Guidance to overcoming various design challenges is presented in the following subchapters.

1.1) AIR GAP
Decreasing the air gap width and shortening the rotor can lead to an increase in tangential stress, improving the suspension force capabilities of the motor. However, a small air gap, that goes hand in hand with small mechanical drive clearances, can cause problems with the unbalanced magnetic pull (UMP) and make the cooling more challenging. These, thermal challenges, are elaborated in detail in section 2.

1.2) MAGNET AND SLOT DIMENSIONS
Balancing the magnet arc against its thickness can decrease the force error angle, force vector amplitude error and torque ripple, improving controllability, (addressed in detail in section 4). The sizing of the slot opening should be thought out carefully; with small slot openings the stator stack can be fractured, but small slot openings make fitting the windings challenging. The slot shape should also be considered beforehand to avoid manufacturing issues and so that the windings can be packed with a high fill factor.

1.3) WINDINGS
Different solutions can be employed to produce torque and suspension force, e.g., either by two separate sets of windings or one set of shared windings, that can be configured distributedly or concentrically. The use of concentric, shared windings can decrease the length of the end windings and simplify manufacturing. However, even if distributed windings are used at the cost of increased length of the end windings, the waveform of the air gap flux is more sinusoidal, leading to lower eddy current losses on the rotor side. The number of turns, proportional to the voltage induced in the windings, should be chosen carefully; a greater number of turns results not only in higher voltage, but also improves the filtering of higher harmonics.

1.4) MATERIALS
Proper selection of the materials [27] is important for the optimization of the drive as a whole. Low magnetic steel should be used for the shaft, whereas materials with thin laminations should be used for the stacks, both to decrease iron losses. Silicon-iron steel, e.g., SuraNo20 or 10JNEX900, and cobalt-iron steel, e.g., Vacoflux, result in low core losses [26]. Low saturation levels follow low iron losses, which result in increased suspension performance, e.g., by decreasing force error angle, that come with the costs associated with oversizing, however.

Rotor losses increase when suspension current is supplied into the windings, the rate of increase being significant if high suspension current is required. To avoid high eddy current losses in the magnets, especially concerning the SPMs, unshielded from the external magnetic field, the segmentation of the magnets is recommended [26]. The downside is the magnetic field strength of the magnets decreases by some percentages, depending on the process and number of segments as well as the mechanical strength and cost.

Optimization is an iterative process, involving both mechanical and electromagnetical constraints, such as structural rigidity and acceptable stresses or suspension force and force error angle. Analytical calculations can be used in the beginning, and to speed up the iteration process, a genetic algorithm can be used during FEA.

2. THERMAL ANALYSIS AND COOLING DESIGN
When the drive is intended to work in an HTHP, different types of cooling arrangements can be considered; use of compressor suction from the evaporator of the heat pump, internal fan cooling with ambient air flowing from one end of the drive, internal cooling fan with flow from the radial holes in the middle of each BM, or an external cooling fan.

Electric motor cooling based on the compressor suction can be seen as a viable alternative: gas entering each motor from the inlets, located close to the ends of the motors, and leaving them from the middle of the compressor after cooling the motors. The temperature of the outlet flow from the evaporator defines the temperature of the coolant. When the cooling is designed, the properties of the coolant need to be considered in detail; the flow of the coolant can produce significant flow losses, as well friction losses that depend on dynamic viscosity and fluid density.

Additional cooling for the stator, i.e., a water jacket, can be employed if necessary.

The flow of the coolant through the clearances of the structure and the motor air gap needs to be considered. From
the electromagnetic point of view, a small air gap is desirable to increase suspension force capacity. However, effective flow requires width for the air gap. Computational Fluid Dynamics (CFD) analysis should be carried out to predict if a satisfactory flow rate is achieved, the pressure drop between the inlet and the outlet should remain low.

Even with very efficient cooling, losses, i.e., heat sources, should be minimized in design by choosing low loss stacks, materials with low bulk conductivity and segmenting the magnets.

After thermal analysis is carried out using the chosen coolant, the effect of the maximum temperature of the coils, their insulation and that of the magnets on suspension force production and motor performance can be seen. Material resistivity tends to grow with increasing temperature, and the effect should be noted in the loss calculations. Furthermore, the field strength of the magnets decreases as temperature rises, possibly decreasing the torque density of the drive.

The obtained temperatures of the drive components coupled with mechanical stresses enable the estimation of thermal stresses. These stresses can cause structural deformation, and so, losses should be kept low with proper design and materials with high mechanical strength be favored.

3. ANALYSIS OF ROTOR DYNAMICS

Rotor dynamics should be studied thoroughly to ensure dynamical performance at operating speed, especially concerning high-speed machines [30]. Typically, under critical performance is targeted as supercritical performance requires additional steps to go beyond the critical speeds. In subcritical operation, the critical speeds (excluding rigid body modes) above the operating speed with a safety margin (SM) of at least 10% is required [29]. API Standard 617 suggests the separation margin to be 17–27% [30]. A smaller safety margin results in the first bending mode starting to excite, which increases the overall vibration of the machine.

When it comes to the compressor of the HTHP, equipped with impellers, the impact on rotor dynamic performance should be analyzed, since the impellers act as overhang masses. Analysis of rotor dynamics is meant to assess the influence of the masses of the system parts, i.e., motors, impellers, sensors, and bearings, on the structural stiffness of the shaft and eventually the critical speeds.

As the impellers affect the dynamics greatly, sensitivity analyses with different impeller masses should be carried out. The position stiffness of the motor is close to the stiffness of the radial AMB, i.e., in the magnitude of 1·10^6 N/m, which means that the rotor geometry typically defines the dynamical behavior when such motors are used in large high-speed machines. The bearing stiffness achieved with electromagnetics is relatively loose compared to journal bearings or rolling element bearings.

Based on the initial analysis of rotor dynamics, preliminary control, actuator, and sensor locations can be planned. The housing dynamics should be analyzed, to be incorporated in control design. The housing design could be iterated to increase the stiffness of the housing, thereby increasing the critical speeds of the housing. It is challenging, however, to produce a complete rotor design where the required parts are included, yielding a dynamically feasible solution.

The rotor dynamic analyses can be performed with an FE-based method, and later a model-based controller for the system can be designed [32]. Here, the length of the rotor and the impeller sizes as overhang structures should be used as they are the most limiting factors from the perspective of dynamics. If adequate dynamical performance is not achieved, the sizes of different parts have to be reconsidered.

4. CONTROLLABILITY OF A BDT

Control of a BM entails the lift up of the rotor and centering when displaced during operation due to system disturbances. Control of the system is directly connected to rotor dynamics, as it forms the framework where the system operates. The drive should be able to work throughout the whole speed range of the application, and if needed cross the critical speeds with suitable ascent across the speed range.

Different current injection control strategies, introduced in [18], involving e.g., split windings and an equivalent six-phase system, could be considered. In particular, a current injection control strategy based on [24], where the same windings were used for suspension force and torque production, could be implemented.

Suspension force characteristics i.e., force error angle, amplitude error and bandwidth, need to be established so that controllability can be estimated. The first step to ascertaining controllability is to calculate the force error angle of the suspension force, which must not exceed 15° during lift-up or operation [25]. One way to ensure the controllability of the motor is to minimize the force error angle by optimization of the geometry. A broad force bandwidth for the BL motors and the AMB is desired, and the suspension force can most effectively be controlled when the angle between the electromagnetic force components in a Cartesian coordinate system is minimal.

When the control is in the development stage, the system delay should be minimized, and the control algorithm should be designed to account for application specific disturbances, i.e., the external forces from impellers, manufacturing eccentricities and the possible UMP. The control system, operating in the discrete domain, should include estimation of the rotor position and manipulate suspension current to produce the necessary force to suspend and center the rotor.

Parts of the system also used for control, such as position sensors, are dimensioned according to the parts of the rotor. Both axial and radial position sensor should be employed so that dynamics can be controlled, i.e., nodal positions of the modes need be avoided. The challenge in placing the position sensors, however, is that they need to measure absolute displacement so as to ensure dynamical performance and controllability. Low noise and uncertainty levels for actuators and feedback signals are necessary to receive accurate measurements.
III. DIMENSIONING OF THE HEAT PUMP WITH A BDT

A. DIMENSIONING OF A HEAT PUMP & COMPRESSOR UNIT

The scaling and dimensioning of a heat pump process can be achieved by using multiple heat pumps connected in parallel, without the need for re-designing the process, compressor, and motor. Another way for scaling the process is to combine multiple parallel compressor units with low- and high-compressor pressure stages, each with their own e-motor, to achieve the higher heat rate. However, here the aim is to achieve a compact solution with no additional heat pump units or compressors.

A variety of means can be employed to scale the heat pump process to a higher heat rate.

The mass flow rates of the working fluid, the volume of the heat exchangers and flash intercooler, as well as the process piping including the process components, e.g., valves, related to it can be increased in proportion to the scaling. Using the approach of unaffected process nodes, the method affects the size of the heat exchangers, valves, as well as the process piping.

However, in case higher mass flow rates of the working fluid are required, scaling the compressor becomes necessary. To that end, the size of the impellers could be increased, the impeller diameters in particular requiring upsizing to a great extent; the efficiency of the compressor can be improved by decreasing the rotational speed. The scaling of the compressor can be achieved based on the fundamental equations described in [47]. Pressure rise, isentropic enthalpy rise, and the volume flow rate are important when re-dimensioning of the compressor is attempted. [47] However, when upscaling the heat pump process without changing the unaffected nodal values of the cycle, the volume flow rate is important, i.e., there is no need for a higher pressure and isentropic enthalpy rise.

The required motor power is defined by the process simulation of the heat pump. The further focus of the paper is dedicated to detailed dimensioning of the BM that should be capable of producing power to provide the targeted heat rate for the heat pump.

B. DIMENSIONING AND SCALING RULES OF A BDT

This chapter aims to propose rules for scaling and dimensioning a BDT along with the impellers attached to the ends of the shaft. Changing the dimensions can be required for various reasons, e.g., falling short of the overall performance targets set for the motor or unsatisfactory performance in terms of rotor dynamics. Scaling can entail achieving the same power density and force capacity per volume with a different motor length to diameter ratio or increased power density and force capacity. Figure 3 presents some significant structural design choices, which can be made to scale the drivetrain in case of unsatisfactory electromechanical performance.

1) FIRST RULE

Whenever a change in drive dimensions is required, e.g., the performance of rotor dynamics is not satisfying yet the same power density and force capacity per volume is expected, the drive can be shortened while simultaneously increasing the diameter. So, the first scaling rule proposes to keep the same number of turns and current and to increase shaft, rotor and stator yoke diameters along with the outer arcs of the slots, and the rotor sleeve thickness by a percentage-based factor \( \alpha_1 \), while the length of the motor is shortened by that same factor. The first scaling rule aims to shorten the rotor by the same factor as the diameter was increased.

As a result of the scaling, the magnets are shifted outward and elongated in proportion. So, the magnet height needs to be increased in relation to its arc angle, the factor for increasing the magnet height derived based on a simplified 2D magnet structure presented in Fig.4.

\[
h_{m1} = h_m \cdot \alpha_1 + h_m
\]  

(1)
The correction factor $\alpha_2$ is derived by

$$\alpha_2 = \frac{2\pi (r_2 + r_2 \cdot \alpha_1) \left( \frac{\theta}{360} \right) - 2\pi (r_1 + r_1 \cdot \alpha_1) \left( \frac{\theta}{360} \right)}{2\pi (r_1 + r_1 \cdot \alpha_1) \left( \frac{\theta}{360} \right)} \quad (2)$$

which results in the final scaled height of the magnet $h_{m2}$ by

$$h_{m2} = h_{m1} \cdot \alpha_2 + h_{m1} \quad (3)$$

Furthermore, when it comes to segmented magnets embedded in the IPM structure, the thickness of the magnets needs to be increased so that the magnets retain the same volume and thus the same field strength as before the segmentation of the magnets, for the lamination bridges take up space between the segments.

2) SECOND RULE

The second scaling rule aims to increase the produced power by increasing the rotor length, however not significantly increasing the friction losses. Since, rotor dynamics are highly dependent on the rotor length, an increase in length changes the critical speed. As a consequence, a sensitivity analysis with different rotor lengths and impeller loads should be carried out. In addition, a dynamical performance with different rotor lengths and impeller masses should be analyzed to find the limitations of the design.

3) THIRD RULE

The third rule of scaling aims to increase the power density of the BDT, like the second rule, but also involves changing the stator and rotor core materials. The case examples can involve silicon-iron steel 10JNEX900, possessing higher silicon content than Sura NO20, and cobalt-iron steel Vacuflux 50 [51] as stack materials that were investigated in [26] and compared to Sura NO20. The considered lamination materials have different knee points of the BH curve, above which the flux density that can be generated by the magnetic core have different range of values.

IV. DESIGN OF A HEAT PUMP WITH A KINETIC COMPRESSOR

A. REQUIREMENTS FOR THE DESIGN OF THE HTHP

The requirements of the HTHP application are based on industrial need and defined in Table II. The design procedure and analyses, leading to the manufacture of a prototype meeting the objectives set for the design, is elaborated in the following sections.

$$\begin{array}{|c|c|c|}
\hline
\text{Parameter} & \text{Symbol} & \text{Value} \\
\hline
\text{Evaporating temperature} & T_1 \ [\degree C] & 10-90 \\
\text{Condensing temperature} & T_c \ [\degree C] & 80-120 \\
\text{Condenser heat rate} & Q \ [\text{kW}] & \geq 0.4 \\
\text{Minimum LP Compressor efficiency} & \eta_{LP} \ [%] & \geq 75 \\
\text{Minimum HP Compressor efficiency} & \eta_{HP} \ [%] & \geq 73 \\
\text{Rated speed} & \omega_n \ [\text{rpm}] & 17,000 - 30,000 \\
\text{Maximum peak voltage} & U_p \ [\text{V}] & 600 \\
\text{Minimum required rated point efficiency} & \eta \ [%] & 96 \\
\text{Rated electric power of the motordrive (min-max)} & P_{input} \ [\text{kW}] & 125-300 \\
\text{Power density of the motor} & P_{D} \ [\text{W/kg}] & \geq 1.5 \\
\text{Current density in the windings} & I_{d} \ [\text{A/mm}^2] & \leq 5.5 \\
\text{Coefficient of performance} & COP & \geq 3 \\
\hline
\end{array}$$

A. HEAT PUMP PROCESS

The heat pump process is designed based on the targeted condenser heat rate. The aim is to upgrade heat from low-temperature heat sources and provide heat for high-temperature applications which results in low evaporating and high condensing temperatures, leading to a high compressor pressure ratio. Because of the high-pressure ratio, a two-stage heat pump cycle was selected.

Two options of two-stage heat pump cycles were studied: a two-stage cascade cycle [22] and a two-stage cycle with a flash intercooler, suggesting that either the compressor low- and high-pressure stages are on the same shaft or two separate low- and high-pressure compressors are used.

In a two-stage cascade cycle, low- and high-pressure circulation loops are separated by using an intermediate heat exchanger called a cascade heat exchanger. If two individual low- and high-pressure compressors are used, a two-stage cascade cycle provides a possibility to use different working fluids in the low- and high-temperature circulation loops while enabling separate control of the low- and high-pressure compressors.

In the two-stage cycle incorporating a flash intercooler, the low- and high-pressure circulation loops are in fluid communication with each other. Due to the lack of a cascade heat exchanger with a temperature difference between high- and low-temperature working fluids, the cycle permits higher efficiency in the design point as compared to a two-stage cascade cycle.

Good efficiency and the possibility to use only a single compressor with the rotors of the low- and high-pressure...
compressors as well as the rotor of the electric motor all coupled on the same shaft were considered important prospects during the design. So, the two-stage cycle with a flash intercooler was selected.

In the evaporator, heat is transferred from the fluid of the heat source to the working fluid — the process schematic presented in Fig. 5 and Fig. 1 [19]— whereas in the condenser, heat is transferred from the working fluid to the actual medium used in the heating application. Between the low- and high-pressure compressor stages and the low- and high-pressure expansion valves, there is a flash intercooler, in there the saturated vapor-liquid equilibrium and a particular liquid level.

The computation of the process cycle is governed by the mass and energy balances of the process components as well as by the boundary condition, stating that the rotors of both the compressors and that of the electric motor are attached to the common shaft and therefore have the same rotational speed. In the design, the mass flow rates in the low- and high compressor stages differ from each other.

For the selection of a suitable working fluid, potential alternatives were studied by conducting thermodynamic cycle computations. In particular, the working fluids R1336mzz(Z), R1233zd(E), R1234ze(Z), and R1224yd(Z) as well as the natural working fluids pentane (R601), isopentane (R601a), butane (R600), and isobutane (R600a) were investigated.

A widely used refrigerant R245fa was used as a reference fluid in the comparison of the potential working fluids, the properties of which are listed in Table III. However, R245fa was not considered an actual candidate for the working fluid of the heat pump because of the high global warming potential (GWP). As a result of the investigation, R1233zd(E) was selected as it performs adequately in the heat pump application, is well available and used in industry.

The other above-mentioned natural working fluids were omitted because they are highly flammable and require high rotational speeds for the compressor stages. Based on the results of the heat pump design work and the related investigation of different options, an evaporating temperature of 25°C, a condensing temperature of 90°C, as well as a motor power of 130 kW were established and used as a basis for the process cycle design. A two-stage heat pump cycle was selected. The reason was to enable off-design operation of the heat pump at higher pressure ratios and mass flow rates than in the design point as well as to achieve a condenser heat rate of 0.5 MW. To get a more accurate evaluation of the performance of the heat pump, the values related to the heat pump process cycle can be updated with computed and measured process component and piping values. In doing so, also the e-motor losses, the compressor gas friction and gas flow losses, the process component, piping, and e-motor cooling vapor pressure losses, as well as the values related to the compressor stages can be updated.

### Table III: Properties of the Studied Working Fluids

| working fluid | R245fa | R1336mzz(Z) | R1233zd(E) | R1234ze(Z) | R1224yd(Z) | pentane R601 | isopentane R601a | butane R600 | isobutane R600a |
|---------------|--------|-------------|------------|------------|------------|-------------|-----------------|-------------|----------------|
| **Molar mass, g/mol** |        | 134.05      | 164.06     | 130.50     | 114.04     | 148.49      | 72.15           | 72.15       | 58.12          | 58.12         |
| **Critical temperature, °C** |        | 153.86      | 171.35     | 166.45     | 150.12     | 155.54      | 196.55          | 187.20      | 151.98         | 134.66        |
| **Critical pressure, bar** |        | 36.51       | 29.03      | 36.24      | 35.31      | 33.37       | 33.68           | 37.78       | 36.29          |                |
| **Triple point temperature, °C** |        | -103.15     | -90.50     | -78.00     | -10.15     | -129.68     | -160.50         | -138.26     | -159.42        |                |
| **Triple point pressure, kPa** |        | 0.0119      | 0.0181     | 0.2733     | -          | 35.16       | 0.000078        | 0.00000009  | 0.000067       | 0.000028      |
| **Normal boiling point temperature, °C** |        | 15.05       | 33.45      | 18.26      | 9.73       | 14.62       | 36.06           | 27.83       | -0.49          | -11.75        |
| **Vapor pressure at 20 °C, bar** |        | 1.23        | 0.60       | 1.08       | 1.49       | 1.24        | 0.57            | 0.77        | 2.08           | 3.02          |
| **Freezing point temperature, °C** |        | -107        | -90.5      | -107       | N/A        | -115        | -130            | -160        | -138           | -159          |
| **Global Warming Potential (GWP)** |        | 1030        | 2          | 1          | <1         | <1          | 4 ± 2           | 4 ± 2       | 4              | 3             |
| **Ozone Depletion Potential (ODP)** |        | 0           | 0          | 0.00034    | 0          | 0.00012     | 0               | 0           | 0              | 0             |
| **Flammability** |        | Non-flammable | Non-flammable | Non-flammable | Low flammability | Non-flammable | Highly flammable | Highly flammable | Highly flammable | Highly flammable |
| **ASHRAE Safety Group** |        | B1          | A1         | A1         | A2L        | A1          | A3              | A3          | A3             | A3             |
Similarly as the heat pump process and motor design, the compressor design, dedicated to direct-driven and high-speed application, relies on the best practices, general knowledge, and recommendations based on literature and in-house experience [48]. The designs of low (LP) and high pressure (HP) compressors receive the boundaries for the inlet temperature, inlet pressure, mass flow rate and the required pressure rise from the heat pump process design, and for rotational speed and power from the motor design.

The preliminary design of the compressors performed with LUT’s in-house code give the main dimensions, operating conditions and performance of the compressors [48]. These values can be used to estimate the impeller manufacturability already at the early design phase. The impeller tip speed and outlet temperature indicate what kind of material is needed, and the required material information allows the estimation of manufacturability and costs.

The preliminary design is followed by design of 2D and 3D geometry, where the aim is to achieve sufficient diffusion (pressure rise) and compressor performance. The 3D models of the LP and HP compressors are created and analyzed within the commercial ANSYS software. In both compressors, splitter blades are used to increase the flow passage area in the inducer sections. Both compressors are designed unshrouded due to issues related to the combination of the small geometry and manufacturing capabilities. The blade thicknesses are based on experience, and their suitability is confirmed by stress analysis. To enable a wide operating range for the compressors, vaneless diffusers are chosen. The volutes are designed for a free-vortex flow and the outlet cone geometries are based on the design principles of [23].

The 3D geometry is used in parallel to check both the aerodynamic performance with Computational Fluid Dynamics (CFD) and the mechanical strength and stresses with Finite Element Method (FEM). After the convergence of the CFD simulations, the performance parameters and velocity triangles at the impeller outlet are analyzed, and the profiles of mean relative velocity, aerodynamic loading, and mean relative Mach number are checked. The LP and HP impeller structures are presented in Fig.6. Furthermore, the compressor maps are presented in Fig.7.

After analyzing the design point, the performance of the compressors can be studied numerically at off-design conditions.

The final design of the compressor is still verified after the performance of the motor, e.g., rotational speed and power, and the mechanical strength of the compressor are established.

Based on the design of the compressor stages the evaluation of the performance of the heat pump process is carried out. Figure 8 presents the heating COP as a function of condensing and evaporating temperatures.
FIGURE 8. Coefficient of performance for heating as a function of condensing and evaporating temperatures.

C. DRIVETRAIN

Bearingless drivetrain incorporated into an HTHP system should have the operating range, in terms of speed and power, defined based on the process simulation. Since the design and optimization of the BM is an iterative and interdisciplinary process, the sections V–VI are dedicated to describing the motor topology, its dimensioning and optimization procedure along with iterations between different domains, i.e., electromagnetics, mechanics, thermodynamics, and control. The following sections correlate to the motor design steps presented in the flowchart of Fig.2.

V. TOPOLOGY OF A BDT INTEGRATED INTO A KINETIC COMPRESSOR

A. MOTOR SELECTION OF A BDT FOR AN HTHP APPLICATION

The operational drive parameters, shown in Table II, are based on the requirements set by the HTHP and the power grid. The drive consisting of two identical motors is intended as a part of an oil-free, high-speed two-stage compressor of the HTHP. A cross section of the compressor, showing the bearingless drivetrain, is presented in Figs. 9–10.

The heat rate of the condenser of the heat pump defines its power rate. The desired thermal coefficient of performance (COP) is 3.125, which results in a power range at least of 0.5 MW for the HTHP, and targeted power range at least of 160 kW for the drive. It is chosen to place the drive vertically [19] to minimize the use of suspension current during operation.

B. INVESTIGATION OF MOTOR GEOMETRIES

In this section, concerning the Step C.2 of the flow chart, permanent magnet rotor topologies chosen for the motor are investigated. Interior permanent magnet (IPM) and surface permanent magnet (SPM) rotors with different stator slot shapes and slot openings have been investigated in [12] and the results suggest that parallel and trapezoidal tooth shapes merit further investigation. The geometry with shoe-shaped slots leads to an unacceptable force error angle and is thus not investigated further [12].

While the trapezoidal tooth shape has been found only to improve the performance of a cobalt-iron steel SPM motor, parallel tooth shapes are studied further in an investigation of SPM and IPM motors. The stator geometry investigated in this paper is presented in Fig.11 and the rotor geometries in Fig.12.
C. PRELIMINARY OPTIMIZATION OF THE GEOMETRY WITH A GENETIC ALGORITHM

Optimization for the SPM and IPM rotor structures presented in Fig. 12 was carried out. The rotor structures were considered to have carbon-fiber sleeves shrink fitted on the structures to ensure structural rigidity at high operating speed. Similarly, the rotor stack was envisioned to be shrink fitted on the shaft but to transfer generated torque. The retaining band with a radial fit of 0.1 mm was found to stay in compression at all operating conditions in a pre-analysis done in Ansys. The actual optimization, a two-stage optimization of the rotor structures and electromagnetics, using a genetic algorithm — related to Step C.4 of the flow chart — was carried out in JMAG.

Introducing many objective functions in a genetic algorithm at one time leads to a very long optimization process. For that reason, the rotor structure was optimized first to find the mechanical limits and a second round of optimization, using the same geometry, but aiming to achieve optimal control and electromagnetic performance, was conducted. A more detailed investigation of the mechanical performance is presented later in section VI.C.

The limits of the rotor structures with desired speed range and maximum torque were investigated with coupled stress and electromagnetic FEA. A multiple objective function $MOF(x)$ [20], representing the candidate solutions of the optimization, was defined as

$$MOF(x) = \frac{1}{T_N} \sigma_B \sigma_M \sigma_L \sigma_S$$

(4)

in which $x$ is a vector denoting the design variables: magnet arc angle, magnet thickness and slot opening, the simultaneous variation of which result in different motor designs, the values concerning the optimization comprising torque $T_N$, and von Mises Stresses on the retaining band, magnets, lamination, and shaft $\sigma_B, \sigma_M, \sigma_L, \sigma_S$, constituting the objectives of the optimization — maximize torque and minimize the stresses.

The genetic algorithm was carried out for 30 generations. The individuals, aka chromosomes, in each generation were compared to the previous population and only those with high fitness were transferred to the population of the next generation, ultimately converging on the Pareto optimum solution.

The solutions for each rotor part, meeting the minimum stress criterion, were compared to the maximum material strengths of the specific parts, so as to verify the validity of the results. The material properties used in the analysis are presented in Table IV.

Based on the optimization, the minimum thickness for the retaining band of the SPM and that of the steel between the magnets and the sleeve for the IPM structure were established.

The results of the analysis suggest that the magnets in the IPM structure can be segmented if the stresses on them and the outer lamination bridge are higher than their material strength.

The electromagnetics were optimized the same way as the limits of the rotor structure, but using

$$MOF'(x') = \frac{1}{T_{N'}} E_A T_{rip} E_{VA}$$

(5)

in which $x'$ is a vector denoting the design variables: the magnet arc angle, magnet thickness and slot opening, whereas the objectives of the optimization: torque, force error angle $E_A$, torque ripple $T_{rip}$ and force vector amplitude error $E_{VA}$, with an additional boundary condition: $T_{rip} < 15\%$ and $E_{VA} < 30\%$ imposed. The goal of the optimization was to maximize torque and minimize the rest of the objectives.

The genetic algorithm was likewise carried out for 30 generations. The correlations between the design variables and objective functions were obtained by calculating the Pearson correlation coefficients (PCCs) in the JMAG software.

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**TABLE IV**

| MATERIAL PROPERTIES USED IN MECHANICAL ANALYSIS |
|-----------------------------------------------|
| Density [kg/m³] | Young’s Modulus [GPa] | Poisson’s Ratio | Yield strength [MPa] |
|-----------------|------------------------|----------------|---------------------|
| Vacoflux 50     | 8120                   | 185            | 0.3                | 9.44                | 9.44               | 9.44               | 250               |
| 10JNEX900       | 7490                   | 185            | 0.3                | 12                  | 12                 | 12                 | 604               |
| L Type magnets  | 7600                   | 160            | 0.24               | 7                   | -1                 | -1                 | 285               |
| Carbon fiber    | 1790                   | 141            | 0.3                | 30                  | -0.063             | 30                 | 1400              |
| Sura NO20       | 7650                   | 210            | 0.3                | 12                  | 12                 | 12                 | 370               |
| 42CrMo4         | 7850                   | 210            | 0.3                | 11.15               | 11.15              | 11.15              | 500               |

*CTE – coefficient of thermal expansion*
The PCC is a measure of linear correlation between two sets of data, calculated by dividing the covariance of two variables by the product of their standard deviations, i.e., by

$$PCC = \frac{COV_{ab}}{\sigma_a \sigma_b}$$  \quad (6)$$

where $COV_{ab}$ represents the covariance of variables $a$ and $b$ and $\sigma_a$ and $\sigma_b$ the standard deviations of variables $a$ and $b$ respectively.

A correlation matrix for the investigated SPM and IPM rotor structures was generated as a result of the optimization, shown in Table V, intended to provide intuitive understanding on how the design variables are related to the objectives of the optimization. Based on Fig.13, it can be seen how the design variables, pertaining to the geometry, can be tuned to achieve desired performance. Considering the SPM motor for instance, when simultaneously varying the design variables, i.e., the magnet height, arc angle and the slot opening, the change in relation to force error angle is 0.46, 0.79, -0.01 respectively. This means that variation of magnet arc angle has the strongest effect on force error angle, whereas the effect of changing the slot opening is negligible.

The contours of force error angle and torque ripple with simultaneous variation of the height of the magnets as well as the arc angle were calculated, shown only for the SPM structure in Fig. 13. The contours display the effects of varying the dimensions of the magnets on key indicators of controllability.

The starting parameters and dimensions of a BL motor unit — where permanent magnet height, tangential length and slot opening are results of the optimization — with good performance in terms of controllability and power within the expected range are presented in Table VI.

### TABLE V

**CORRELATION MATRIX OF THE DIMENSIONAL PARAMETERS AND THE PERFORMANCE OF THE SPM AND IPM MOTORS**

| Motor Type | Torque [N] | Torque ripple [%] | Force error angle [°] | Force vector amplitude [N/m] |
|------------|------------|-------------------|----------------------|-----------------------------|
| SPM        | 0.80       | 0.39              | 0.94                 | 0.46                        |
| IPM        | -0.22      | 0.17              | 0.35                 | -0.16                       |

| Motor Type | Torque [N] | Torque ripple [%] | Force error angle [°] | Force vector amplitude [N/m] |
|------------|------------|-------------------|----------------------|-----------------------------|
| SPM        | 0.83       | 0.02              | -0.96                | 0.80                        |
| IPM        | 0.19       | 0.36              | -0.43                | -0.42                       |

| Motor Type | Torque [N] | Torque ripple [%] | Force error angle [°] | Force vector amplitude [N/m] |
|------------|------------|-------------------|----------------------|-----------------------------|
| SPM        | 0.35       | -0.18             | 0.85                 | -0.88                       |
| IPM        | -0.89      | -0.11             | -0.08                | -0.18                       |

| Motor Type | Torque [N] | Torque ripple [%] | Force error angle [°] | Force vector amplitude [N/m] |
|------------|------------|-------------------|----------------------|-----------------------------|
| SPM        | -0.32      | 0.16              | -0.18                | 0.32                        |
| IPM        | 0.88       | -0.11             | -0.87                | 0.60                        |

| Motor Type | Torque [N] | Torque ripple [%] | Force error angle [°] | Force vector amplitude [N/m] |
|------------|------------|-------------------|----------------------|-----------------------------|
| SPM        | 0.85       | 0.16              | -0.93                | 0.23                        |
| IPM        | 0.85       | 0.08              | -0.87                | -0.44                       |

### TABLE VI

**MAIN OPERATING PARAMETERS AND DIMENSIONS OF BLM**

| Motor and levitation ratings | Symbol [unit] | Value |
|-----------------------------|---------------|-------|
| Rated speed                 | $n_{nc}$ [rpm]| 30 000|
| Number of poles             | $p$           | 4     |
| Rated line-to-line voltage  | $U$ [V]       | 400   |
| Rated motor current         | $I_{max}$ [A] | 83    |
| Rated electric power        | $P_S$ [kW]    | 80    |
| Rated torque                | $T_S$ [Nm]    | 25.4  |
| Staror resistance           | $R_{ac}$ [Ω]  | 0.0037|
| Magnets remanence at 100 °C (N42UH) | $B_{r,22}$ [T] | 1.18  |
| Maximum nominal current density | $I_{n}$ [A/mm²] | 5     |
| Maximum levitating current  | $I_{max}$ [A] | 83    |
| Levitation force to overcome gravity | $F_L[N]$ | 230   |
| Maximum mean force capacity of the radial force at nominal rotor position | $F_R[N]$ | 800   |

| Motor and levitation ratings | Symbol [unit] | Value |
|-----------------------------|---------------|-------|
| Stator slot count           | $Q$           | 6     |
| Stack iron length           | $L_s$ [mm]    | 90    |
| Stator and rotor core material |           | SURA NO20 |
| Stator outer diameter       | $D_{st}$ [mm] | 265   |
| Mechanical air gap length   | $\delta$ [mm] | 1.5   |
| Rotor outer diameter        | $D_r$ [mm]    | 91    |
| Permanent magnet height     | $l_{mp}$ [mm] | 5     |
| Permanent magnet tangential length | $l_{max}$ [°] | 69.7  |
| Slot opening                | $s_{op}$ [mm] | 4.6   |
| Winding connection:         | Star connection |       |
| Stator coil-turns in series per phase | $N_s$ | 15    |

### D. CASE STUDIES OF THE BDM

The first scaling rule aimed to shorten the rotor by the same factor as the diameter was increased. The SPM and IPM motor structures, their dimensions presented earlier in Table VI, implemented using parameterized geometry in the FEM software, were scaled by the factor $\alpha_1$, save for the magnet height scaled by $\alpha_2$ . The results of the electromagnetic analyses, including suspension force with
different currents, torque, tangential stress, computed friction losses and surface velocity, force error angle with a skewed rotor with respect to the scaling factors are presented in Figs. 14 a-d.

The first scaling rule shows that the motoring and suspension performance behaves nearly linearly with respect to the degree of scaling when volume is kept constant. However, the friction losses of the drive at axial velocity, also containing the losses at the end plates, calculated for two motor units with an assumed coolant, R1233zd(E) at 60 °C and 2.1 bar [40], increased significantly. Furthermore, increasing the rotor diameter by a factor higher than 0.3 causes the surface velocity to rise to almost 200 [m/s], which can cause very high stresses on the parts and challenge the rigidity of the structure [34]. So, it is proposed to scale the motor by not more than 15% to avoid a significant increase in friction losses and high surface velocity.

The second scaling rule, along with comparisons to the unscaled structures, pertaining to the values of Table VI, was investigated with the help of case studies. Study 0 represents the unscaled structures (Table VI), whereas Study 1 the first rule of scaling, involving a scaling factor of 10%. Study 2 entails increasing the rotor diameter according to scaling rule 1 and, in addition, the motor length is increased by the same scaling factor, instead of decreased like in Study 1. Study 2 aims to increase the power produced by the motor, bearing in mind different impeller loads in the HTHP application while not significantly increasing the friction losses. To analyze the rotor dynamic performance, sensitivity analysis with different rotor lengths and impeller loads was carried out. The behavior of rotor length and the corresponding influence on critical speed is summarized in Table VII. Relative decrease in critical speed with respect to impeller mass is shown in Fig. 15.

The third rule of scaling aims to increase the power density of the drive, like the second rule, but also involves changing the stator and rotor core materials.

When the stack materials are changed, the magnet arc angle needs to be optimized again. So, the same optimization process described in Section C is carried out, and through the related electromagnetic and structural analyses, the optimum arc angle is chosen for 10JNEX900 and Vacoflux 50 steels. The results are presented in Table VIII.

| Study | Rotor Length \[mm\] | Impeller 1 [g] | Impeller 2 [g] | Critical speed 30 000 rpm FW [rpm] | Critical speed 30 000 rpm BW [rpm] |
|-------|---------------------|----------------|----------------|-----------------------------------|-----------------------------------|
| Study 0 | \(L_0\) | 189 | 493 | \(\omega_{0FW}\) | \(\omega_{0BW}\) |
| Study 1 | \(L_{11}\) = \(L_0\alpha_1\) | 189 | 493 | \(\omega_{1FW} = \omega_{0FW} + 18\%\omega_{0FW}\) | \(\omega_{1FW} = \omega_{0FW} + 18\%\omega_{0FW}\) |
| Study 2 | \(L_{21}\) = \(L_0\alpha_2\) | 189 | 493 | \(\omega_{2FW} = \omega_{0FW} + 5\%\omega_{0FW}\) | \(\omega_{2FW} = \omega_{0FW} + 5\%\omega_{0FW}\) |

**FIGURE 14.** First scaling rule and motor performance.
The magnets

Starting point. Dimensions of the SPM and IPM motor presented in Table VI.

I

With basis from Study 0, the shaft, rotor and stator yoke diameters, rotor sleeve height are increased by a percentage factor \( \alpha_1 = 10\% \), while the length of the motor is shortened by the same factor. The magnets are scaled by the factor \( \alpha_2 \).

Aim: Keeping the volume constant, achieving similar power density (W/kg) and improving dynamical performance.

II

With basis from Study 0, the shaft, rotor and stator yoke diameters, rotor sleeve height and the rotor length are increased by a percentage factor \( \alpha_1 = 10\% \). The magnets are scaled by the factor \( \alpha_2 \).

Aim: Increasing the produced power without significant increase in friction losses and surface velocity.

III

With basis from Study II, the stator and rotor stack materials are changed in addition.

Aim: Increasing the produced power density.

Studies 0-III, based on the values of the unscaled structure (Table VI), summarized in Table IX along with the scaling rules involved, are investigated further from the perspectives of multiple engineering disciplines in Section VI. Studies 0-III are divided into Cases 1-10, listed in Table IX, so as to distinguish between the different motor structures: SPM and IPM, and the different stack materials of Study III. In Study III, the same scaling factor of 10\%, used in Study II, is also used to scale the motorizing and suspension currents while the stack materials are changed to 10JNEX900 and Vacoflux 50 in turn. However, current density should not exceed 5 A/mm² when 20\% of the maximum suspension current is utilized. The slot area should be increased if that limit is exceeded — the slot arc can be increased by \( \alpha_3 \). The flux densities should be kept in the recommended ranges, shown in Table I to avoid saturation of the stacks and increasing the iron losses.

VI. INTERDISCIPLINARY ANALYSES OF THE BDT TOPOLOGIES

A. CASE STUDIES — MULTIDOMAIN ANALYSES

The performance of SPM and IPM motors is investigated with case studies, pertaining to Section V and summarized in Table IX. The multi-physical investigation of the drive performance is divided into sub-feasibility FEAs, i.e., mechanical, electromagnetic, rotor dynamic, and thermal feasibility studies are presented, and an investigation of the design limits is carried out, (as shown in the BDT design loop in the flowchart of Fig.2).

B. FEASIBILITY OF ELECTROMAGNETICS & CONTROL (STEP C.4)

The motor and levitation windings are intended to be supplied with four three-phase inverters: two for motor windings and two for suspension windings. The current injection method [18] is intended to be used for the control of a double 3-phase system with shared motor and suspension windings. The double 3-phase system has the advantage of lower failure rate than a traditional 3-phase design [52]. Suspension force is assumed to be controlled based on the position of the rotor in the air gap, including a feedback loop with a position estimator. Direct torque control could be used for motor control.

Simulations were carried out in the 2D FEM software JMAG, using an ideal sinusoidal current supply. Mesh sizes of 1 mm, 2 mm and 0.2 mm were used for the magnets and the retaining band, stator and rotor cores and the air gap.
respectively. Skin depth was factored in for the magnets, retaining band and shaft [19].

Details of the used electromagnetic design methodology [7][18][43] and performance factors necessary in the evaluation of a BM motor [5][14][56] were presented, whereas the analysis methods are extensively elaborated in [19]. Details of the electromagnetic performance, i.e., the flux distribution, when a) only motoring current, b) only suspension current as well as c) nominal motoring current and maximum suspension current is employed, are presented in Fig.16. Furthermore, the suspension force production with different suspension currents at 30 000 rpm are shown in Fig.16 d)

Loss analyses were carried out, and joule losses and hysteresis losses of the stator and rotor stacks were obtained, using the equations and methods described in [37]. A frequency range of 50-10 000 Hz and 27 peak magnetic flux density values, corresponding to the range of frequencies, were included in the table containing the iron loss data in the FEM software. Eddy currents were allowed in the magnets, retaining sleeve and shaft. As the eddy currents are a 3D phenomenon, they were accounted for in [19] a method for decreasing the magnet losses presented in [26] was applied, and the 3D phenomenon rendered 2D was simulated [19].

The friction losses were estimated analytically based on the torque coefficients calculated with respect to Reynolds numbers by equations for high-speed machinery presented in [19], [33], [35]. The AC winding losses, taking into account the properties of Litz wire and the temperature related change in resistivity, were calculated according to [19].

The force error angle of the motor under study was calculated as in [25]. The results of the modeling show a force error angle reduction of 18% as a result from skewing the rotor by 15°.

The motoring currents $I_m$ and the suspension currents $I_s$ have been recalculated for Cases 7-10, involving the third rule of scaling with changed stack materials. The saturation ranges of the materials, and limits for current density and voltage 5 A/mm$^2$ and 400 V rms respectively were borne in mind. The height of the stator slot was increased by 2/1 in relation to the scaling factor $\alpha$, so as not to exceed current density of 5 A/mm$^2$ while the current was increased by steps of 10%.

In Cases 1–6 where Sura NO20 was used, the reference current was $I_n=83$ A and $I_s=0.2I_m$. Cases 7 and 9, implemented using the stack material 10JNEX900, came with the possibility of increasing current by $\alpha=20\%$, resulting in the values $I_{m1}=I_n+0.2I_m$ and $I_{s1}=0.2I_n$. Correspondingly for Cases 8 and 10 where Vacoflux 50 was used, the low magnetic saturation (corresponding to the lower bound presented in Table I) allowing an increase in current by $\alpha=40\%$, the values used were $I_{m2}=I_n+0.4I_m$ and $I_{s2}=0.2I_n$.

The case studies summarized in TABLE IX were simulated at 30 000 rpm, assuming utilization of 20% of the force capacity, i.e., suspension current. The results, namely torque, torque ripple, force error angle, power density and radial force are presented in Fig.17 and Fig.18, whereas the results of the loss analyses, are presented in Fig.19. Power density, depicted in Fig.18, was calculated by computing the masses of the motor parts, e.g., shaft, stator and rotor stacks, magnets, and coils, and dividing the produced power by the sum.

Figure 17 shows corresponding values for torque, torque ripple and force error angle when comparing Studies 0 and I, indicating that scaling according to the first rule seemed to meet its goal of retaining performance. Likewise, in Fig.18 it can be seen that power density and radial suspension force remain almost constant.

An increase in values can be seen in Fig. 17 and Fig.18 when comparing the values of Study II to Study 0, showing that scaling according to the second rule, leads to improvement in power density, increase in torque and the radial force capacity. By this approach, the increase in friction losses can be kept relatively low in comparison to the gain in motor performance. So, the second rule of scaling seems to deliver on its targets.

When comparing the values of Study III to Study 0 of Fig. 17 and Fig.18, a significant increase in power density and torque production can be observed, meeting the goal of increased power laid out for the third rule of scaling; an analysis of IPM and SPM structures with 10JNEX900 and Vacoflux 50 shows that a motor of the same volume but

![FIGURE 16. Flux distribution when the BM operates with different current conditions; a) nominal motoring current and $I_{max} = 0$ is utilized; b) $I_{max}$ is utilized; c) nominal motoring current and 20% of the maximum suspension current capacity is utilized, whereas d) pictures suspension force production with only different suspension current levels. All computations are conducted at 30 000 rpm.](image-url)
constructed with different lamination materials can be scaled
to over 140 kW (Cases 8 & 10), the power of the drivetrain
scaling correspondingly to over 280 kW.

The iron losses of SuraNO20 are 1.1% of the 80-kW motor
power (SPM). Silicon steel, 10JNEX900, on the other hand,
offers a motor with power of 125-140 kW with an iron loss
of 0.37%. An SPM motor above 140 kW with cobalt–iron
laminations and an increased nominal current has almost five
times as high losses in the magnets as its IPM variant.

Though the IPM rotor structure shields the permanent
magnets with an iron bridge, eddy current losses also occur
in interior magnets, because of the high motor operational
frequency of 1 kHz. Although high-quality steels improve
motor performance, the drawbacks are their high price and
low availability on the markets.

C. MECHANICAL FEASIBILITY STUDY (STEP C.4)
The 3D geometries were analysed in detail with finite
element method (FEM), see e.g., [44] with Ansys
Workbench 2019 R3 software. A quarter model of the
bearingless rotor was used, and conditions of symmetry were
applied to it. Frictionless contacts were used between the
parts, so that they could be separated, but not able to
penetrate between the other parts in normal direction. The
parts were free to be moved or rotated in the tangential
direction. This allows the shrink fit and PM stress evaluation
at all conditions (no speed, full speed, both cold and hot
conditions).

The carbon fibre sleeve is shrink fitted on the rotor
surface, whereas the rotor lamination stack is shrink fitted on
the shaft. This guarantees the rigidity of the structure at all
conditions. The magnets are assumed to be free in the slots,
i.e., in the IPM topology the lamination and the sleeve hold
the magnets in place, while in the SPM topology, only the
carbon fibre sleeve holds them in place.

In the feasibility study, fine mesh was utilized to avoid
singularities in single nodes and thus unrealistic stress
behavior. The results represent the maximum stress in worst
case conditions. The stress values are thus comparable with
the different structures.

The results of the mechanical feasibility study, namely the
Von Mises stresses without thermal loads, are shown in
Fig. 20. Furthermore, Fig. 21 shows their distribution for
Cases 7 and 9. The results show that the sleeve experiences
high stresses, due to the centrifugal loads generated in the
laminations and magnets. In addition, at the top corners of
the magnet surface in IPM laminations, high stresses are
present, mainly due to the loads generated by the magnets in
the slots. The SPM structure experiences the highest stresses
about the inner diameter zone of the lamination, but well
below the material strength. In addition, the magnets
experience loads due to deformations of the slots, whereas
the shaft is exposed only to minor stresses.
FIGURE 20. Von Mises stresses of the rotor parts for the SPM and IPM motors at 30,000 rpm in the room temperature.

FIGURE 21. Von Mises stresses [MPa] of the SPM (scale 41) and IPM (scale 53) scaled rotor structures of Cases 7 and 9 respectively.

D. FEASIBILITY STUDY OF ROTOR DYNAMICS (STEP C.7)

The study of rotor dynamics aims to compare the performance of the initial structure, indicated by Study 0, to that of the structures obtained with different scaling rules in Section V. Impeller loads vary with the desired power of the HTHP, and thus a sensitivity analysis with different impeller masses was carried out.

The analysis of the structures presented by case studies was carried out in RoBeDyn, a Matlab based FE-software, which bases on Timoshenko beam elements, therefore allowing rapid iterations and parameter studies [44],[45]. The rotor FE model, consisting of 56 Timoshenko beam elements, is depicted in Fig. 22. Impellers and laminations were modelled as mass points, and stiffnesses with spring elements.

In nodes 2 and 55 there are impellers weighing 0.45 kg and 1.148 kg respectively. In nodes 10 and 47 there are 1 kg point masses, that represent the sleeve collars. The nodes 18 and 39 represent the two BM units, where the laminations and magnets are modelled as mass points.

The motors are capable of producing a radial suspension force, which acts like a radial AMB. The position stiffness of the BM is 1·10^6 N/m, modelled as a spring element.

The corresponding critical speeds in horizontal and vertical directions are in pairs of backward and forward whirling (BW and FW) modes. The FW is in the direction of rotation and is excited, e.g., by unbalance, therefore leading to significant resonance when rotation speed is close to it. The severity of BW mode is not as significant as FW, as it is rotating in the opposite direction to the shaft and excited by the asymmetric bearing stiffnesses.

The critical speeds of the modes 1-6 with different impeller loads were analyzed with respect to the studies presented in Table VII, where the rules for changing the rotor length of the BL motor are introduced. The critical speeds of those modes are presented in TABLE X.

Figs. 23-24 depict the Campbell diagrams for the configurations of Studies I and II, which correspond to the shortest and the longest rotor structure respectively, showing the critical speeds for modes 3–6 in addition. The free-free modes of Studies I and II are shown in Figs. 25-26, while the Ansys model, showing the deformation for the first bending mode, is presented in Fig. 27.

Figure 28 shows the influence of increasing impeller mass on the 1st bending FW mode for Studies 0-II. The sensitivity study with starting masses (at 0%) of the impellers in nodes 2 and 55 are 0.493 kg and 0.189 kg respectively.

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The analysis shows an acceptable decrease in critical speed, allowing the scaling of the motor structures to greater power. The increase in shaft thickness by the scaling factor of 10% in both Studies I and II results in an SM of 33.2% (mode 6) for the configuration of Study I and 16.7% (mode 6) for Study II when the drive operates at 30 000 rpm (500 Hz mechanical frequency). Furthermore, Fig. 28 shows the sensitivity of the critical speed to impeller mass, which is linear and comparable to Fig. 15. The increase in the SM is achieved with the chosen aluminum impellers, weighing 0.493 kg and 0.189 kg respectively.
E. THERMAL FEASIBILITY STUDY (STEP C.10)

Cooling of the BMs [19] is arranged so that the axially flowing vapor, R1233zd(E), entering the motors from their far ends and exiting from the middle of the compressor where the axial AMB is located, i.e., between the motors, cools the units while passing through. The temperature of the coolant entering the compressor is 60 °C, and its properties [40] are listed in Table XI.

Thermal analysis was based on the estimation of heat transfer coefficient [42], [48] depending on the contact surface and the properties of the coolant. These coefficients \( h_c \) are obtained based on the ratio of the calculated Nusselt number \( Nu \) and thermal conductivity \( k \) of the coolant to the hydraulic diameter \( d_h \) by

\[
h_c = \frac{Nu \cdot k}{d_h}
\]

(6)

The Nusselt numbers of the boundaries [48], i.e. the air gap, the stator, the rotor, the axial AMB, the shaft end, the end windings, the cooling water jacket and the machine frame were found based on the range of Reynolds numbers [42] [48], used for high-speed motors.

Thermal resistances for forced convection and thermal contact resistances, with an insulator placed between the parts, were included in the model.

Transient FEA of electromagnetics was coupled with 3D thermal analysis; motor losses presented in Fig. 19 at 30 000 rpm, acting as thermal loads, were passed to the thermal model, along with winding losses, bearing in mind their dependency on temperature, and the losses in the carbon-fiber sleeve. Friction losses were assigned on the rotor outer and stator inner surfaces as additional heat sources.

The results of the analysis, comprising the temperatures of the motor parts for all ten cases and the temperature distributions of different motor parts for Case 9, are shown in Fig. 29-30 respectively.

The first scaling rule, represented by Cases 3 and 4, involving an increase in diameter by the scaling factor \( \alpha = 10\% \) and a corresponding decrease in motor length, shows that the temperature of the motor parts is elevating mainly due to increasing friction losses. However, with the same cooling arrangement, the rotor temperature still remains below 115 °C for both the SPM and IPM structures, the winding temperature within the permissible range, i.e., less than 130 °C, and the temperature of the insulation under 155 °C. Increasing the motor length, pertaining to scaling rule II, with an aim to achieve higher power, does not increase temperatures significantly.

The third rule of scaling, involving upgraded stack materials along with scaled dimensions, aims to increase the power of the drivetrain. Cases 7 and 8 correspond to the SPM topologies with 10JNEX900 and Vacoflux 50 respectively. With such rotor arrangements, the main concern was the permanent magnets, their temperatures rising past 140 °C with silicon steel and past 200 °C with cobalt–iron. With such temperatures, the motor design had to be analyzed again, but the revised design showed that a power of 125-135 kW is achievable with 10JNEX900. However, the SPM topology with cobalt–iron was only able to provide 104 kW, and therefore, requires better cooling to provide more than 140 kW per motor, which can be observed in the electromagnetic study before a temperature related drop in performance. On the other hand, the IPM structure with Vacoflux 50 can deliver in excess of 140 kW. The permanent magnet material used in the study was once laminated, high-density N42UH, with operating temperature up to 220 °C.

Case 9, representing a silicon steel IPM motor, provides 125 kW with power density of 2.60 kW/kg with the

| TABLE XI | PROPERTIES OF R1233zd(E) ENTERING THE MOTORS. |
|---|---|---|---|---|
| Pressure [bar] | Density [kg/m³] | Dynamic viscosity [µPas] | Thermal conductivity [W/mK] | Isobaric heat capacity [J/kgK] |
| 2.9200 | 14.9800 | 11.5600 | 0.0135 | 902.0000 |

FIGURE 29. Temperature distribution of SPM and IPM structures for cases 1–10.

FIGURE 30. Temperature distribution of the simplified BM, case 9.
temperature of the magnets under 115 °C, whereas Case 10, representing a cobalt-iron IPM motor offers likewise more than 140 kW and 2.99 kW/kg. Such IPM motor structures could be used to power a twin motor BDT of 250-280 kW, operating in a 0.780-0.875 MW heat pump with a COP of 3.125. Though it should be borne in mind that the COP tends to increase with operating power, so a heat pump with greater power could possibly be achieved. The permanent magnets of an IPM structure are protected by lamination from external fields and the associated eddy currents, resulting in only 10% losses in the magnets compared to an SPM structure.

**F. THERMAL STRESS FEASIBILITY STUDY (STEP C.10)**

To ensure long operational lifespans for the BM, thermal stress analysis was carried out in 3D using FEM. The temperatures of motor parts from the thermal analysis were used as sources in the thermal stress analysis.

Rotor models, with their entire length, were analysed in order to account for thermal expansion, and it was assumed that the shrink fits are in compressions at all conditions to ensure the balance and rigidity of the machine. Thermal forces, acting on the rotor, were calculated at standstill and rated speed, with an aim to study the thermal expansion of the SPM and IPM structures under thermal loading.

The results of the thermal feasibility study for Cases 1–10 are shown in Figs. 31-32. Figure 31 depicts nominal operation when the machine is warm, and Fig. 32 a situation where the machine operates at nominal speed, and is then stopped, remaining warm. Fig. 33 shows the thermal stress distributions of the SPM and IPM motor structures.

The results of the thermal stress feasibility study show that thermal stresses of the structures with SuraNO20 and 10JNEX900 are within the permissible range of the material strength. However, the cobalt-iron steel, though promising from the electromagnetic perspective, its yield strength was shown to have the smallest safety margin in comparison to other silicon steels in the conducted study when temperature and thermal stresses were accounted for.

**FIGURE 31.** Thermal stresses of the rotor parts of the SPM and IPM motors at 30 000 rpm under thermal loading.

**FIGURE 32.** Thermal stresses of the rotor parts of the SPM and IPM motors at standstill under thermal loading.

**FIGURE 33.** Von Mises stresses [MPa] of the SPM (scale 35) and IPM (scale 34) structures of Cases 7 and 9 respectively.

**G. ASSESSMENT OF THE CHOSEN BDT DESIGN FOR A HEAT PUMP EXCEEDING 0.5 MW**

The final dimensions of the BM for a heat pump of at least 0.5 MW and above were achieved by analyses of the SPM and IPM structures involving multiple branches of physics and engineering. Based on the dimensions obtained by the preliminary optimization presented in Table VI, scaling principles and inter loop of the motor interdisciplinary analysis and design — part II.III of Fig.2 — feasible dimensions and parameters were obtained. The SPM motor depicted by Case 3, and the IPM motor represented by Case 9 can be applied to a heat pump with power greater than 0.5 MW. The designed dimensions and parameters of these two cases are presented in Table XII.

As a side note, it has to be noted that Case 10 can provide higher power than Case 9. However, the mechanical strength of silicon steel, its low core losses, and better thermal properties favorize the material over cobalt-iron steel.
Concluding the results of the interdisciplinary analyses, the BDT with better performance in terms of rotor dynamics and power of 160 kW, consisting of two identical 80 kW BM units, is decided to be manufactured with the intention to power the compressor of the heat pump prototype.

### VII. CONCLUSION

Interdisciplinary design of the heat pump, including a centrifugal compressor and a bearingless drivetrain was presented. The challenges in designing such a complicated system where the working fluid of the heat pump flows also through the electrical motors were elaborated.

The heat pump process defined the boundary conditions for the motor and compressor design. The power and speed range were taken as targets for the motor design, whereas temperature, pressure, mass flow rate, and the required pressure rise governed that of the compressor. A hermetic solution for the system was designed; the most significant interconnection between the heat pump process and the BDT was the same fluid that acted both as the working fluid in the process and the cooling agent of the drive. This fact emphasizes the importance of choosing a working fluid with good cooling properties and low enough density that allowed the friction losses to stay within acceptable range.

The maximum speed needs had to be compromised with the design of the compressor. As the inputs from the process simulation were defined, the compressor design followed the path presented in the flowchart. The flow rates obtained in the process design were iterated for the impeller design.

Within the internal drivetrain design loop, the motor design had to be compromised with rotor dynamics. Aluminum impellers were chosen at a point, and motor length had to be shortened in comparison to the preliminary design. The steps were taken to increase the critical speed of the drive, to improve the mechanical performance and efficiency of the compressor.

Furthermore, a design incorporating multi-physics with iterations for a bearingless drivetrain of industrial size were carried out, adhering to the methodology involved, leading to a feasible motor design for an HTHP. Multi-objective optimization using a genetic algorithm was used to dimension the motor geometry.

An investigation of PM motor structures was done, starting from assessment of structural feasibility. A matching structure, employing undivided magnets, functional as an SPM structure, could not be used for an IPM motor, because of stresses exceeding the flexural strength of the magnets. Therefore, the magnets of the IPM structures had to be divided in the circumferential direction. The assessment of feasibility of the structures with promising electromagnetic performance was continued; a drivetrain with two units of identical motors with power of 80 kW required, with such basis for design, a power density of almost 1.95 kW/kg was achieved, using Sura N020 as stack material. Changing the stack material to 10JNEX900 resulted in a power density of 2.60 kW/kg, the drivetrain power of 250 kW, owing to low core and magnet eddy current losses, as well as low saturation levels, allowing increased current as concerns the IPM structure. Vacofflux 50 was likewise a useful material for scaling motor power, but structural analysis showed an SPM rotor not desirable in terms of stresses; the yield
strength of the steel approaching the limit.

The study of rotor dynamics showed: the best design is obtained when the motor is scaled according to the first scaling rule, provided an increase in power density is not desired. However, if greater power is targeted, increasing the motor length by 10% still leads to a feasible design; a critical speed related safety margin of 16.7% was achieved.

Thermal investigations showed that the magnets, windings, and the winding insulations of both the SPM and IPM motors with power of 80 kW had permissible temperatures with the designed cooling. However, the calculated thermal stresses were in favor of the IPM structure. Should 10NEX900 and cobalt–iron steel be employed to increase power, the SPM motor requires more from cooling than its IPM variant, chiefly due to the eddy current losses in the magnets. Should the SPM structure with Vacoflux 50 be used, better cooling of the structure is required, the power dropping because of high temperature in the magnets. The smallest safety margin of the core strength was obtained for Vacoflux 50 i.e., the motor with cobalt-iron steel had worse mechanical performance compared to the structure with silicon steel.

As results of the design process, drive topologies were established for an industrial heat pump with a power range over 0.5 MW. The 160-kW drive, consisting of two 80 kW BM units, with rotor length decreased according to scaling rule I, is a clear solution from electromagnetic, mechanical, rotor dynamic, control, and thermal perspectives. It can be considered an end-result of the design, comprising series of interdisciplinary analyses, and it is decided to be manufactured and used in the HTHP. When aiming for a drive with higher power, according to scaling rule III, changing the stack materials while increasing the length by 10% leads to a feasible solution, however, IPM topology, having smaller eddy current losses, should be used instead of the SPM that was decided to be used for the 160-kW drive.

ACKNOWLEDGMENT

The authors would like to express their gratitude to Eero Schermer for sharing the compressor structure drawing.

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