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Integrated Modeling and Control Strategy for a Novel Rotor Design of a Rotation Heat Pump

Michael Lauermann[1], Reinhard Jentsch[1], Adam Buruzs[1], Mustafa Kalfa[1], Andreas Längauer[2], Bernhard Adler[2]

[1] AIT Austrian Institute of Technology, Center for Energy, Giefinggasse 2, 1210 Wien, Österreich, michael.lauermann@ait.ac.at
[2] ecop Technologies GmbH, IZ NÖ-Süd, Ricoweg Obj. M372351 Wr. Neudorf

Abstract

A novel design concept for rotation heat pumps (RHP) proposes the use of microchannel diffusion bonded heat exchangers (MCDBHX) as rotating elements. Compared to conventional rotors, this approach aims to simplify manufacturing, increase modularity for different cycle configurations, and enhance scalability across various capacity ranges. Additionally, higher rotational speeds become feasible. The freedom in defining arbitrary flow channel geometries allows these components to be precisely tailored to thermodynamic process requirements, e.g. steam generation.
The modelling methodology follows a multi-level approach. Detailed Computational Fluid Dynamics (CFD) simulations provide insight into local flow phenomena and guide the parameterization of simpler models. On the system level, one-dimensional finite volume models are implemented using Dymola/Modelica with the TIL library to capture the thermodynamic behaviour of the rotary heat pump across various operating conditions. With the help of these physics-based models, surrogate models are further developed to support fast and efficient controller design and optimization tasks.
Throughout the modelling process, several technical challenges must be addressed, including the selection and handling of accurate fluid property data, appropriate discretization strategies, simplifications of fluid dynamics, and the formulation of geometry design as an inverse problem. This hierarchical modelling framework - from high-fidelity CFD to 1D simulations and surrogate models - enables a comprehensive understanding of the system and supports both detailed component design and efficient control strategy.

Keywords: Rotation heat pump; Control strategy; Computational Fluid Dynamics; 1D fluid simulation; Surrogate Modeling

1. Introduction

1.1. Motivation

The required energy transition from fossil fuels to climate-neutral renewable energy—both in general and particularly in industry [1] represents a major opportunity for the Austrian and European economies. It enables inclusive growth and employment, reduces energy costs for consumers, decreases Austria's and the EU's dependency on gas imports, and redirects investments toward sustainable infrastructures [2].

Globally, industry is one of the largest energy consumers (28% [3]). Approximately 75% of industrial energy demand is used for process heat, 37% of which requires temperatures below 200 °C [4]. It is estimated that global heat demand will increase from about 50,000 TWhth in 2020 to 56,600 TWhth by 2050, with industrial process heat being the main driver of this growth [5].

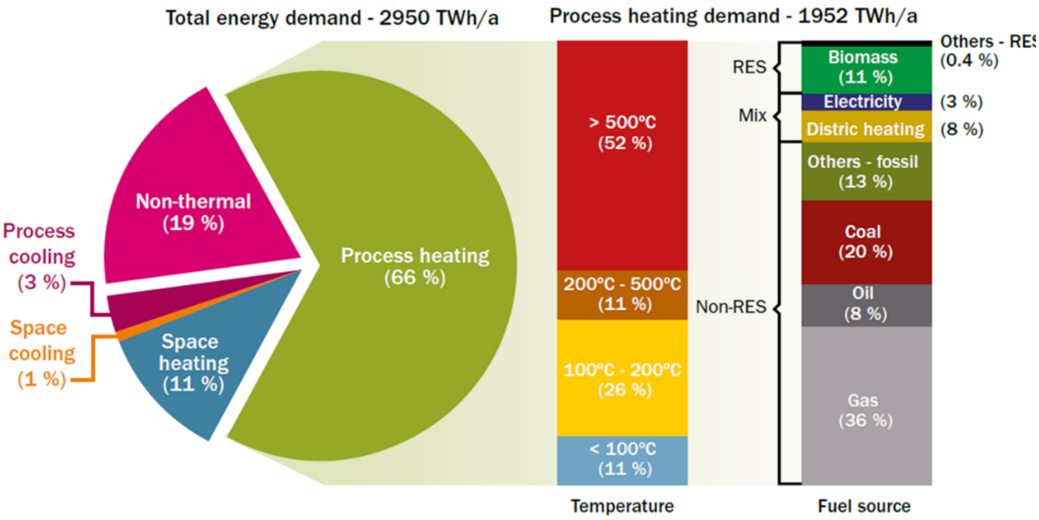

Figure 1: Overview of required process heat demand [4].

Industrial facilities also generate large quantities of industrial waste heat. Due to their comparatively high temperature levels—up to around 100 °C and above—these heat streams represent a valuable exergy source and offer very high ecological savings potential. However, they remain largely unused to date [6, 7, 8]. According to [6], around 300 TWhth per year of industrial waste heat in the EU is still not exploited - equivalent to the annual energy consumption of the entire Italian industrial sector.

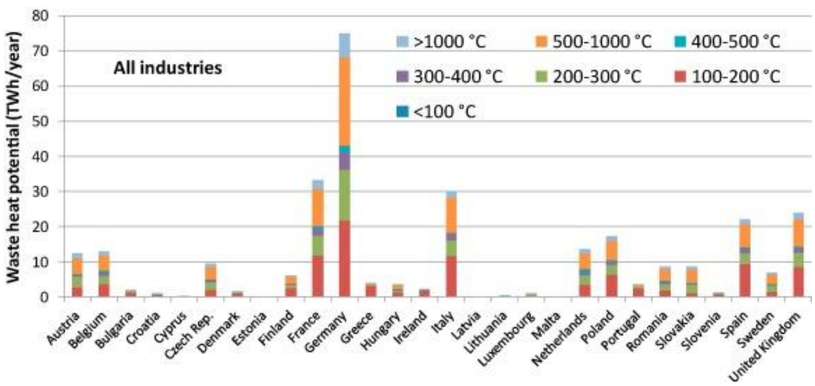

Figure 2: Waste heat potential in each EU country per temperature level [6].

From an economic perspective, industrial companies are also very attractive customers for district heating suppliers, as they require large, steady amounts of heat throughout the entire year (not only during winter) and usually operate with high base load and long operating hours [9]. Industrial heat demand is also highly predictable.

However, due to the relatively low temperature level of available waste heat or district heat (around 100 °C), industrial facilities cannot be supplied directly. This situation can be improved by using high-temperature heat pumps (HT-HPs), which elevate the temperature level of waste heat or district heat. This leads to increased energy efficiency and substantial CO_2 reductions in industrial applications [10, 11].

By employing a temperature-flexible heat pump, a significant portion of natural gas consumption—and thus CO_2 emissions—can be saved across nearly all industrial sectors. Based on CO_2 emission factors for gas and electricity (Environmental Agency 2018) and typical seasonal performance factors (SPF ≈ 2.5), at least 63% of all CO_2 emissions associated with process heat generation up to 200 °C can be eliminated. Applied to the EU, this corresponds to approx. 85 million tonnes of CO_2 savings per year—with an even higher potential if SPFs increase and the electricity mix includes a larger share of renewables.

However, for industrial heat supply up to 200 °C (with temperature lifts of around 100 K), processes typically require steam, high-pressure hot water, or thermal oil. Therefore, HT-HPs must be sufficiently flexible to provide both latent and sensible heat. Achieving these Key Performance Indicators (200 °C and 100 K lift) is the core objective of the Rotation Heat Pump (RHP).

According to [12], international research currently focuses on HT-HPs up to approx. 180 °C. Most of these HT-HPs use multi-stage and cascade compression heat pumps (vapor compression cycles VCHP, Perkin/Evans cycle). Their disadvantages include the use of refrigerants with:

- high Global Warming Potential (GWP),
- PFAS substances,
- flammability (e.g., R600a, R601), or
- toxicity (e.g., R717),

as well as their thermodynamic limitations for sensible heat sources:

The approach pursued in a RHP enables an overcritical heat pump cycle using a non-flammable, non-toxic, fully PFAS-free and GWP-free gas mixture. Compression and expansion occur very efficiently in a rotational field at high pressures and rotational speeds. The core technology is the Micro-Channel Diffusion-Bonded Heat Exchanger (MCDBHX), which allows new cycle configurations and higher speeds, making it far better suited for HT-HP requirements than all existing alternatives.

1.2. Innovation

In a conventional compression heat pump (Perkin/Evans cycle), a refrigerant evaporates at low pressure by absorbing heat from the source, is compressed by a compressor and brought to high pressure, where it releases heat in the condenser to heat the sink. The pressure is reduced in an expansion valve back to low pressure, and the refrigerant flows again into the evaporator.

In contrast, the RHP uses a noble gas mixture as working fluid ("refrigerant"), which is compressed and expanded by centrifugal forces. This generates a heat-pump cycle very similar to the reverse Joule cycle. Both operate entirely in the single-phase region. The RHP, however, operates at much higher pressures, such that the process takes place more in the supercritical region of the fluid rather than in the gaseous region (hence the process is also referred to as a reverse supercritical cycle). Heat absorption (low-pressure heat exchanger) and heat rejection (high-pressure heat exchanger) occur with a temperature glide.

In the reversed Joule Brayton cycle, a compressor (usually a multistage axial or radial compressor) compresses the gas to high pressure; heat is discharged isobaricly (neglecting pressure losses), the gas is expanded in a turbine (usually a multistage axial turbine), and heat is then taken up again isobaricly from the source before the gas re-enters the compressor.

In contrast, in the RHP, the actual compression and expansion take place within the centrifugal field—at significantly higher isentropic efficiencies compared to the reversed Joule Brayton cycle. Only a fan is required, which must compensate for pressure losses and the divergence of the isobars (between low and high pressure) . Another advantage of the RHP over the reversed Joule Brayton cycle is that the process is operated at significantly higher pressures in the supercritical region, where the divergence of the isobars is smaller than in the classic gas region. As a result, a substantially higher coefficient of performance (COP) can be achieved in the RHP than in the reversed Joule Brayton cycle. Furthermore, the fan has a much lower pressure ratio than the compressor of the reversed Joule Brayton cycle, enabling fewer stages and higher isentropic efficiencies.

Additionally, a RHP offers significantly lower exergy losses in the low-pressure heat exchanger compared to the evaporator of the VCHP, significantly lower compressor-(fan-) outlet temperatures, and - depending on the working fluid - lower exergy losses in the condenser of the VCHP (especially when using a drying-compression fluid and the specific heat capacity of the vapor is low). A RHP also does not suffer from lubrication issues nor from problems associated with refrigerant flammability, toxicity, global-warming potential (GWP) or persistent and harmful chemicals. If the temperature glide in the high- and low-pressure heat exchangers is small and matches the temperature spread of the external heat-transfer media on the sink and source side, exergy losses during heat transfer can be minimized. This process is particularly well suited for finite heat sources such as industrial waste heat or district heating and for supplying process heat using high-pressure water or thermal oil.

Thus, the RHP technology constitutes an ideal high-temperature heat pump (HT-HP), as it also resembles a Lorenz cycle. The "IEA HPT Annex 58: High Temperature Heat Pumps" shows, however, that the market demands solutions that can supply process heat up to 200 °C (and partially up to 250 °C) for different heat-transfer media (steam, thermal oil, high-pressure hot water) with temperature lifts up to 100 K. This is necessary for broad market penetration and to make a decisive contribution to the energy transition. This is why a new rotor design using a MCDBHX, integrating the high- and low-pressure heat exchangers (see Figure 3) is introduced.

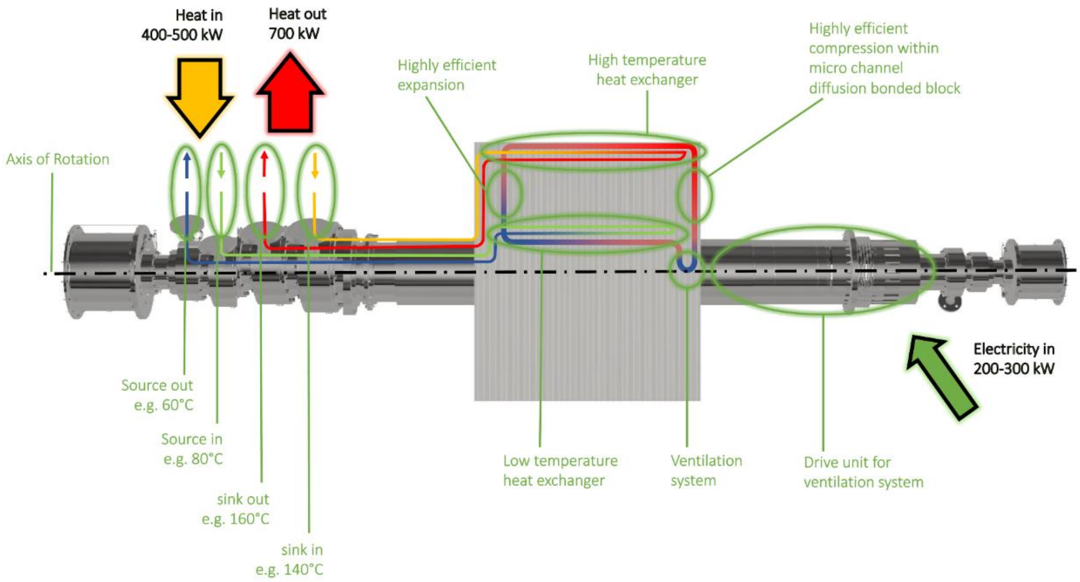

Figure 3: Rotor section with gas circuit and sink and source routing.

The MCDBHX is a type of plate heat exchanger in which the flow-channel profiles (micro-channels) are etched into the plates. After etching, the plates are precisely stacked and diffusion-bonded in an autoclave at high temperature (≈90% of the melting temperature) and pressure. The resulting joint displays a homogeneous metallic structure according to the manufacturer.

This offers two advantages:

1. Significantly higher mechanical strength compared to welded heat exchangers, enabling higher rotational speeds, higher pressures, and larger pressure ratios—thus enabling higher temperatures and temperature lifts.
2. Freedom in channel design for the gas mixture and the water (heat-transfer medium). This allows flexible process configurations - for example, radial outward flow of the gas during heat rejection, enabling isothermal behaviour, which yields a substantial efficiency advantage when generating steam via a downstream flash process.

This new rotor design therefore represents a breakthrough for high-temperature heat pumps by enabling higher temperatures and temperature lifts. Additional significant benefits are expected in maintenance and manufacturing. However, the freedoms in the design also present challenges: plate thicknesses and numbers of plates in the MC-DB-HX stack can be chosen freely, as can the arrangement of channels (e.g., radial heat exchangers) and the hydraulic diameters, see Figure 4.

Figure 4: Example image of a cross-section of the M-DB-HX from manufacturers (including micrograph with grain boundaries) [13].

2. Challenge

The essential distinguishing features of an RHP compared with a conventional heat pump, in terms of basic principle, design and mode of operation, require the development of fundamental knowledge and a detailed analysis of the individual components. Control systems and strategies used for commercial heat pumps cannot be transferred, since additional control parameters are available and further, non-comparable complex interrelationships have to be taken into account. An important aspect here is the operating limits, which in conventional heat pumps are determined by the compressor. In an RHP, these limits are partly determined by the fan characteristics. This means that certain changes in operating conditions can lead to a stall at the fan and bring the system to a standstill.

For controlling conventional compression heat pumps, numerous publications, theoretical principles and commercial controllers are already available. Here, a well-tuned system can be provided in advance using "default settings" for e.g. the expansion valve. To determine the main differences between an RHP and a conventional compression heat pump in terms of basic principle, design and operation, basic knowledge must be acquired, and the individual components be analyzed in detail. Control systems and strategies for commercial heat pumps cannot be transferred easily, as additional control parameters are available and further non-comparable complex correlations must be taken into account. Due to the lack of literature, the associated behavior of an RHP regarding the unusual physical relationships of its components needs to be explored. The current approach is based on flow simulations of the main components (heat exchangers, fan, pipes) for the entire operating range with variable boundary conditions. Out of these considerations, three major challenges in regard of controlling an RHP were identified:

- **Challenge 1:**
  Due to the rotation the component's operating behavior differs from available literature data. Furthermore, measurements in the system are difficult to establish or even impossible for both, pressure and mass flow sensors, due to high loads on the outer diameter of the rotor, which are problematic for the transducers, and would require complex wiring. Therefore, the modeling approach was chosen and the model is tried to be validated on a global basis (energy balance, efficiency, etc.) for further control development.
- **Challenge 2:**
  The fan like every turbomachine has a characteristic curve with certain operating limits caused by flow phenomena. Both, the fan characteristics and the system curve of the refrigeration cycle furthermore change with the rotor speed of the RHP. Experiments have shown that there are combinations of fan speed and rotor speed, where the fan given the system curve cannot build up enough pressure difference to push the refrigerant mass flow, causing the refrigeration cycle to stop. With the help of CFD for flow phenomena and further modelling, those combinations can be identified and can be considered for controlling and successfully operating the RHP.
- **Challenge 3:**
  The current state of the art in terms of control is a fixed/rigid pre-setting of the fan speed and the rotor speed to achieve a certain operating state. This results in a simple but non-optimal control, especially regarding energy efficiency. A Digital-Twin of the heat pump offers the possibility to carry out the controller development in a time- and resource-efficient way. It requires correlations of the thermodynamic behavior of the components.

3. Method

3.1. General implementation

Modelling with CFD necessitates a comprehensive understanding of the underlying physical mechanisms, including heat transfer processes and inlet–outlet interactions within channel flows, particularly in rotating reference frames. While CFD simulations enable the derivation of reduced surrogate models, conducting high-fidelity simulations of the complete rotor remains prohibitively complex and computationally impractical for most applications. In contrast, a one-dimensional approximation provides a reduced-order representation that is well suited for system-level analyses, extensive parameter studies, and the identification of optimal geometric configurations. Such models must achieve an adequate balance between predictive accuracy and computational efficiency. For the implementation of Model Predictive Control (MPC), a model specifically adapted to real-time control requirements is essential, with strict emphasis placed on computational speed and deterministic execution to ensure reliable operation. Figure 5 shows our approach.

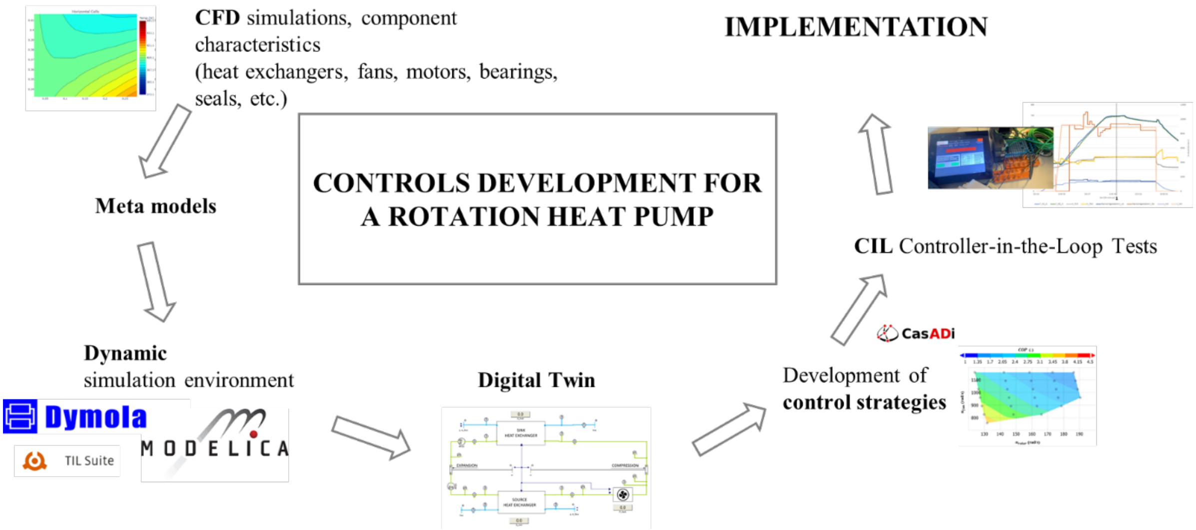

Figure 5: General Implementation.

3.2. Rotating Domain Fluid Dynamics

For conservation equations for mass, momentum and energy:

$$\frac{\partial A\rho}{\partial t} + \nabla \cdot (A\rho\vec{u}) = 0 \tag{1}$$

$$\rho\frac{\partial \vec{u}}{\partial t} + \rho\vec{u} \cdot \nabla\vec{u} = -\nabla p - f_D \frac{\rho}{2d_h}\vec{u}|\vec{u}| + \rho\vec{f} \tag{2}$$

$$\frac{\partial \rho h}{\partial t} + \nabla \cdot \vec{u}\rho h = \nabla \cdot Ak\nabla T + \dot{Q}_{wall} + f_D \frac{\rho A}{2d_h}|\vec{u}|^3 + \frac{\partial p}{\partial t} + \vec{u} \cdot \rho\vec{f} \tag{3}$$

for the case of the rotating domain, the volume forces are:

$$\vec{f} = \nabla\Phi \tag{4}$$

$$\Phi = \frac{\omega^2 r^2}{2} \tag{5}$$

For the rotor case, the analysis is based on several simplifying assumptions. Thermal diffusion within the fluid domain is neglected, such that heat transfer is considered to be dominated by convective mechanisms. Furthermore, rotationally induced apparent forces beyond the centrifugal contribution - specifically the

Coriolis and Euler forces - are omitted from the formulation. These assumptions reduce the model complexity while preserving the dominant physical effects relevant for the intended analysis.

3.3. 1D-Approximation

- Challenge: 1D fluid dynamics in a 2D domain (symmetry in 3. Dimension)
- According to 1, a flow can be approximated 1-dimensionally if:
    - Speed profile is fully formed
    - Shocks are neglected
    - Pressure loss due to channel curvatures negligible compared to wall friction
    - Speeds normal to the channel are assumed to be 0

The vector equations can be defined by the tangential velocity u with $\vec{u} = u\vec{e}_t$ and the unit bar tangential vector $\vec{e}_t$ defined with $\vec{e}_t = (e_{t,x}, e_{t,y}, e_{t,z})$ can be transformed into the following equations:

$$\frac{\partial A\rho}{\partial t} + \nabla_t \cdot (A\rho u\vec{e}_t) = 0 \tag{6}$$

$$\rho\frac{\partial u}{\partial t} = -\nabla_t p \cdot \vec{e}_t - \frac{1}{2} f_D \frac{\rho}{d_h}|u|u + \rho\nabla_t\Phi \cdot \vec{e}_t \tag{7}$$

$$A\frac{\partial \rho h}{\partial t} + A\nabla_t \cdot u\vec{e}_t\rho h = \dot{Q}_{wall} + \frac{1}{2} f_D \frac{\rho A}{d_h}|u|u^2 + \frac{\partial p}{\partial t} + u\vec{e}_t \cdot \rho\nabla_t\Phi \tag{8}$$

∇_t stands for the tangential derivative, i.e. the Cartesian components of a tangential projection of the gradient

$$(\nabla f)_t = (\vec{I} - \vec{n}\vec{n}^T)\nabla f \tag{9}$$

3.4. 1D pressure field estimation

The high-speed rotation creates centrifugal force (see Eq. (4)), that can be quantified by the rotational potential (Eq. (5)). If the angular speed is high enough, the centrifugal force is the main contribution of the pressure increase. In the channels filled by the noble gas, the density strongly depends on the pressure, so a pressure field estimation can be obtained by solving a corresponding differential equation.
By neglecting the friction forces, the stationary solution (by constant angular speed ω) of the momentum equation can be converted to the simple form of an ordinary differential equation:

$$\frac{dp}{dr} = \rho(p(r)) \cdot \omega^2 \cdot r \tag{10}$$

The density ρ of the compressible noble gas depends on the pressure, and enthalpy. To calculate the thermodynamic state dependent density, we have used the Coolprop library [14]. As the rotational potential only depends on the radius of a point, the resulting pressure field has axial symmetry.
We can use this formula to get a first approximation of the pressure field of the noble gas filled channels for the case when the whole structure is rotating with the constant angular speed ω.
In the part where there is no heat transfer between the gas and the water channels, we can assume isentropic compression. As initial condition the pressure and temperature at the rotation axis $r = 0$ or at a given section ($r = r_0$) is specified. From this point (point in radius, but a cylinder shape in 3D) the pressure field at any radius can be obtained as a solution of an initial value problem. The density is calculated from Coolprop with the constant entropy S_0 :

$$\rho_{isen} = \rho_{Coolprop}(p, S_0) \tag{11}$$

We can solve the initial value problem (Eq. 10) numerically with the Python scipy library [15] (function scipy.integrate.solve_ivp function). The upper part of the high pressure subsystem contains the heat transfer from the high temperature compressed gas to the heat sink containing flowing water. In the design case, this heat transfer is performed through isothermal gas states (see Chapter 1.2), so in this case we can model the density with isothermal Coolprop calls:

$$\rho_{\mathrm{isoT}} = \rho_{Coolprop}(p, T_0) \tag{12}$$

We have used the 4-th order Runge Kutta method to obtain the pressure profile, plotted in Figure 6.

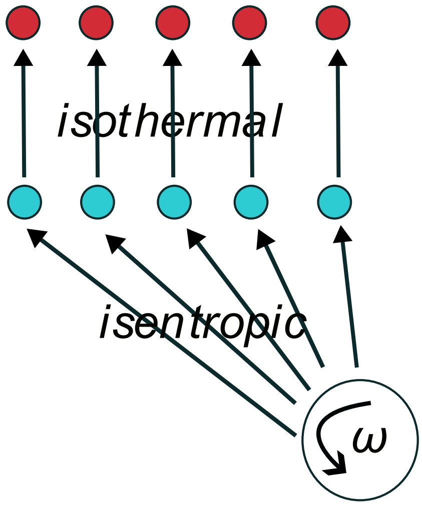

Figure 6 Sketch of the 1D pressure calculation principle for the high-pressure gas ports. The arrows show the direction of noble gas flow, the blue circles are the lower, the red circles are the top ports.

So the calculation of the pressure of the ports is performed in 2 steps: first the pressure states are calculated in the lower ports (the 5 blue circles on Figure 6) by using the isentropic solution of Eq. 10, then from these states as initial values the pressure states in the top ports are obtained as an isothermal solution. This two- stage calculation is performed for all the 5 port-pairs.

The simplified model is compared with the detailed CFD simulations. The stationary angular speed of 356 rad/s was selected as comparison example. For solving the initial value problem of the pressure differential equation described in Chapter 3.4 we have used initial temperature of 370K, radius $r_0 = 7cm$, pressure $p_0 = 76bar$ according to the CFD simulation setting. The obtained pressure levels at the ports are shown in Figure 7. The lower port pressures show a deviation of 3 bars from CFD (2.5% difference), while the top ports only 1.4 bar (0.9% deviation) off the CFD calculated values. This can be considered as very good agreement, regarding the very fast (1-2sec) calculation time compared to the multiple hours simulation with the Fluent CFD Software.

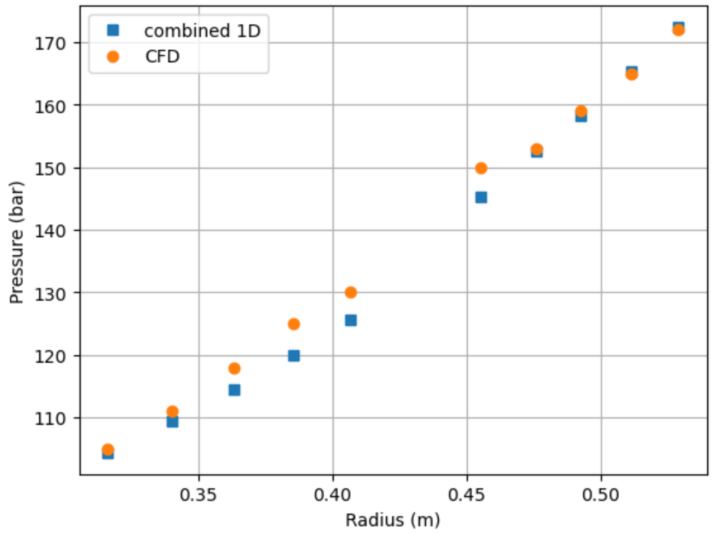

Figure 7: Simulation results for pressure at the gas ports. CFD vs simple 1D differential equation solution.

3.5. MCDBHX – Discretization

The modelling approach is based on the TIL Library. The fluid domain is discretized using an arbitrary intersection of horizontal and vertical channels, resulting in 4 n x m matrices containing the coordinate tuples for the corresponding inlet and outlet positions of the horizontal and vertical channel segments. Each channel is linearized piecewise, such that the horizontal and vertical sections are represented by discrete steps Δx and Δy for horizontal and vertical channels respectively.
From these increments, path vectors

$$\vec{l} = \begin{pmatrix} \Delta x \\ \Delta y \end{pmatrix} \tag{13}$$

are defined, and the discrete path length is computed as

$$\Delta l = \sqrt[2]{\Delta x^2 + \Delta y^2} \tag{14}$$

The resulting discretized geometry is subsequently modelled using finite volume methods, enabling the spatially resolved analysis of the relevant thermofluid-dynamic phenomena, see also Figure 8.

This approach forms the basis for modelling the rotary heat pump (RHP) in Dymola/Modelica, where all components are integrated into a unified system representation. The model encompasses the complete process chain—from the fan, through the distribution and collection plates, to the high-pressure and low-pressure heat transfer zones—thus ensuring a consistent and fully coupled depiction of the thermo-fluid dynamic behaviour.

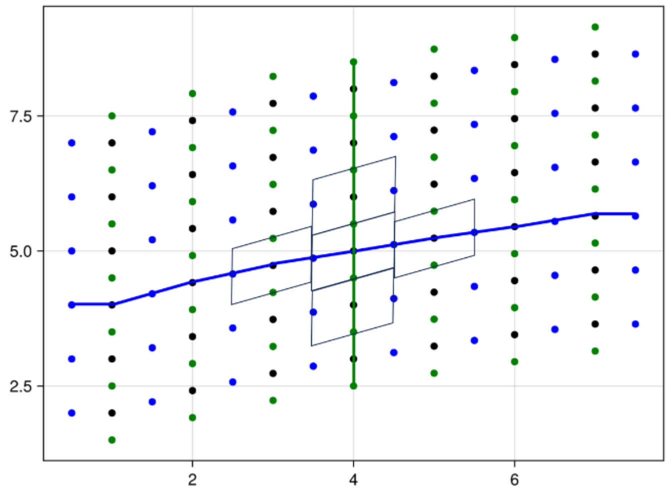

Figure 8: 2D discretization of the heat exchanging channels.

4. Control strategy

4.1. General principles

With the help of the validated full RHP model we develop control strategies in the direction of optimal control. Table 1 shows a list of the different control variants including their advantages and disadvantages. In this table we present out initial thoughts and expectations as it is given in the literature. Until the conference date in May 2025, we will tap this and investigate it for the application of MPC for the RHP.

Table 1: Comparing control concepts.

PI controller	ESC	Model-based look-up table	MPC
+ Proven technology + Reach a wide range of operation + Easy to implement ~ Must be adjusted precisely - Negatively influencing each other (rotor / fan) - Does not necessarily reach an optimal operating point - Undesired operating points can also be reached (e.g. stall)	+ Optimal COPs possible + Adapts to unknown system model ~ Range of operation has to be defined experimentally ~ Thermal inertia of the system can reduce impact based on the controller oscillation principle - slow optimization process - Local minima operation possible	+ Optimal COPs possible + High variety of operation points + Low complexity during operation ~ Moderate development effort ~ Transitions between loads must be smoothed - Models must represent reality very accurate (high modelling effort) - Deviations to reality must be additionally considered (stability)	+ Optimal COPs possible + Any operation point within validity range can be covered + Predictions allow faster response to changes - Models should represent reality very accurate (very high modelling effort) - Very high development effort - Undetected disturbances can destabilize the system

5. Summary and conclusion

In this paper, essential steps were described to model a digital twin of an RHP and prepare it as a basis for controller development. We modelled the RHP in the Dymola/Modelica environment with the TIL Library.

The RHP model was validated based on CFD. First, the pressure and temperature increase based on the centrifugal forces of the simplified model was compared with the detailed CFD simulations. The stationary angular speed of 356 rad/s was selected as comparison example. For solving the initial value problem of the pressure differential equation, we have used an initial temperature of 370 K, $r_0 = 7cm,\ p_0 = 76\,\mathrm{bar}$, pressure according to the CFD simulation setting. The lower port pressures show a deviation of 3 bar from CFD (2.5 % difference), while the top ports only 1.4 bar (0.9 % deviation) off the CFD calculated values. This can be considered as very good agreement, regarding the very fast (1-2 sec) calculation time compared to the multiple hours of simulation with the Fluent CFD Software.

Secondly, we described our discretization strategy in order to have a sound full system model of the RHP which is ready for deployment as a basis for the model reduction and subsequently for the controller development.

6. Outlook

6.1. MPC formulation with CasADi

In this approach, the Dymola/Modelica model is first employed to perform detailed physical modelling and validation of the system. Based on this high-fidelity representation, a reduced-order model is subsequently derived that retains the dominant system dynamics while eliminating unnecessary computational complexity. This reduction can involve aggregating physical volumes, applying quasi-steady assumptions, simplifying thermodynamic relationships, or reformulating the original DAE system into a more tractable set of ODEs.

Once the reduced model has been established, it is implemented explicitly in CasADi, allowing the full benefits of algorithmic differentiation and symbolic optimisation to be exploited. As a result, the model becomes highly suitable for nonlinear optimal control, real-time nonlinear model predictive control (NMPC), trajectory optimisation, and parameter estimation. Compared to an FMU-based black-box coupling, this method offers significantly improved computational performance and determinism, as CasADi can make use of exact analytical derivatives and generate highly efficient solver code.

Although this workflow demands a greater initial modelling effort—particularly in deriving and reformulating the reduced-order model—it provides the most powerful and flexible integration between Dymola-based physical modelling and CasADi-based optimisation.

Nomenclature

Acronyms

CFD	computational fluid dynamics
COP	coefficient of performance
ESC	extreme seeking control
FMU	functional mockup unit
HTHP	high temperature heat pumps
HP	high-pressure
LP	low-pressure
MPC	model predictive control
RHP	rotation heat pump
VCHP	vapor compression heat pump

Roman symbols

c_p	specific heat, J/(kg K)
$\dot{m}$	mass flow rate, kg/s
n	rotation speed, rad/s
$\dot{Q}$	heat capacity, W

T	temperature, °C
r	radius, m

Subscripts and superscripts

fan	fan of the heat pump
in	into the system
lift	temperature lift
meas	measured
rotor	rotor of the heat pump
out	out of the system
sink	sink side of the heat pump
source	source side of the heat pump

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