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Technical article

Rotary Disc Heat Engine Concept

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ABSTRACT

Waste heat to power systems are an essential part of the sustainable future, allowing otherwise wasted heat to produce useful electricity. Depending on system scale, these may be steam Rankin cycles, Stirling engine, or more commonly Organic Rankin Cycle (ORC) systems using turbines, pistons, scroll type mechanical expanders. A new rotary reverse Brayton cycle heat pump with expander isentropic efficiencies of >98% offers the potential of a new type of waste heat to power system mechanical system. The basics of this rotary heat pump, operated in reverse, is modelled as a heat engine. This simple system allows ORC and transcritical ORC heat engine system in a single system, allowing the system to adapt and be optimized for varying heat supply. A system model is developed, and the effects of various parameters are assessed. The rotary system with one moving part is modelled with the effects of RPM, radius, geometry parameters, and heat flow. The rotary heat exchanger geometry is modelled and optimized to provide optimum heat exchange with the radius allowing changing pressure within the heat exchanger and approach temperatures of 2C. Modelling indicates the system will offer superior performance compared to current heat to power systems with second order efficiencies of 50% or greater. Initial assessment of potential mechanical arrangements and future modelling and development is discussed.

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1. INTRODUCTION

With increasing global warming due to unmitigated CO_2, the 60% of electricity generated using fossil fuels must be minimized. A proven alternative method of generating electricity is making use of waste heat in heat to power systems. This energy would otherwise not be utilized. Waste heat sources can be divided into three main categories according to their temperature ranges: high temperature (> 650°C), medium temperature (230–650°C) and low temperature (< 230°C), where more than 50% of industrial waste heat and renewables are within the low-temperature range. This lower temperature range is unsuited for Rankin cycle systems, the highly efficient ubiquitous steam turbine systems. About 100 TWh/year of waste heat at temperatures lower than 200°C exists in the industrial sector in the EU alone. This lower temperature low-grade waste heat recovery for power generation is generally relegated to the Organic Rankine Cycles (ORC) which uses fluids with critical points well below the water critical point.

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The Organic Rankin Cycle (ORC) is the de facto state of the art of for low grade heat to power systems. These systems can be of the mega-watt scale, recovering heat from industrial processes, solar-thermal, and geothermal heat sources. Smaller system, down to 10kW scale, are used to recover energy from vehicles, smaller solar thermal systems, small scale industrial waste, and many other sources. These systems use piston, scroll (Lavernia, 2018), screw, and turbine expanders to generate electrical power (Lecompte, 2015), variations of the cycle include the Trilateral Cycle (TFC) which uses expansion within the vapour dome. Newer developments include supercritical CO_2 (sc CO_2) with higher performance at larger scales (Ge, 2018). The basic ORC cycle arrangement is shown in Figure 1 (Càceres, 2017) where the working fluid, as a liquid is pumped to higher pressure, where it then enters the heat exchanger (evaporator), then the expander. More complex systems include additional heat exchangers as recuperators. Variations also include supercritical cycles and Trilateral cycles, where expansion is within the vapour dome, requiring an expander suitable for wet expansion.

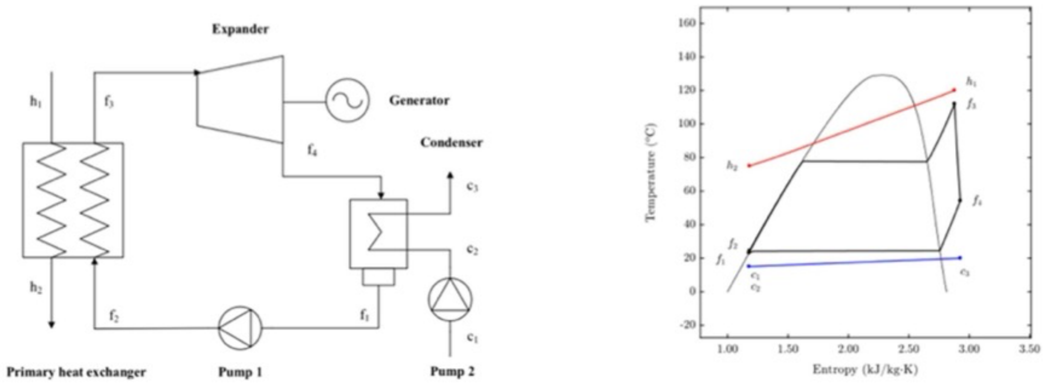

Figure 1. Typical ORC cycle diagram and Ts Plot (Cáceres et al., 2017).

2. HYDROSTATIC COMPRESSION

Hydrostatic compression is the compression due to the sum of the force of the column of fluid above, normally due to the force of gravity; it is the mechanism for air pressure on the surface of the earth (Figure 2) and pressures in the depths of the ocean. The proposed heat engine uses hydrostatic compression in a rotating frame where the centrifugal force compresses the fluid in a rotating tube as shown in Figure 3. With gases such as Argon filling the tube and rotating at 5000 rpm in a 50cm radius, 200°C and 200bar are 50,000 times the force of gravity is easily achieved. Incompressible fluids also greatly increase in pressure in this rotation yet with temperature rise of 2-5°C. Equations 1-3 show forces and hydrostatic compression.

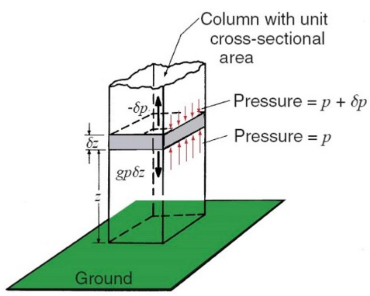

Figure 2. Hydrostatic compression of air column.

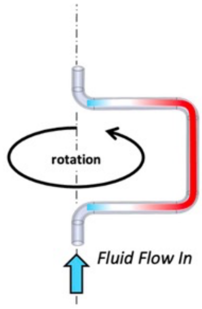

Figure 3. Rotating tube and hydrostatic compression of fluid.

Force due to rotation:

$$F = m\,\omega^2\,r \tag{1}$$

Angular acceleration (Centrifugal Force):

$$(r) = \ \omega\text{^}2\ r \tag{2}$$

Where:
F : Acceleration force due to rotation, centrifugal force [Nm/s]
m: Mass [kg]
ω : Angular velocity [rad/s]
r : Radius [m]
a : Angular acceleration [m (rad/s)^2]
The pressure at the maximum radius of a rotating system (Hydrostatic Compression):

$$P(r) = P_{start} + \int_{r_{min}}^{r_{max}} a(r)\,\rho(r)\,dr \tag{3}$$

Where:
P : Pressure [Pa]
P_{start} : Inlet Pressure [Pa]
r_{min} : Inlet radius [m]
r_{max} : Maximum radius [m]

A commercial heat pump system utilizing hydrostatic compression is produced by ecop GmbH. This Joule cycle heat pump achieves greater than 80% heat pump efficiency (0.80 COP_{Carnot} Second order efficiency). Importantly, it has demonstrated >98% compressor and expander isentropic efficiency (Längauuer, 2021), (Alder, 2014), (Adler, 2017). Figure 4 shows their commercial heat pump system. This large 700kWt system (8.5m x 2m x 2.5m) consists of pairs of rotating heat exchangers with the working fluid of Nobel gases circulated between them and is able to achieve a temperature to 150°C.

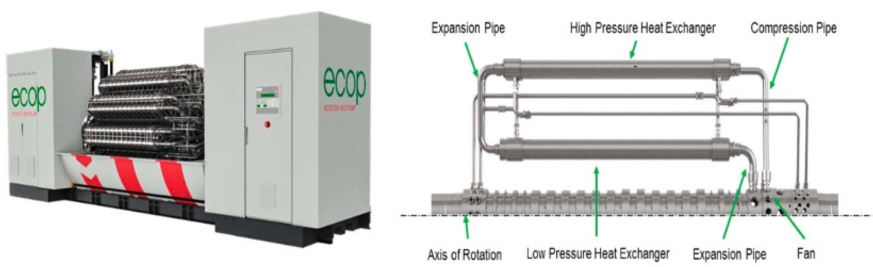

Figure 4. Ecop K7 heat pump system.

The ecop system demonstrates hydrostatic compression and expansion of a supercritical fluid (gas); such a heat pump appears limited to gas cycles. The rotational forces likely make any compression and expansion of wet fluids, those of partial liquid and vapour, impossible due to the extreme difference of densities between the vapour and liquid. For this reason, hydrostatic compression would be limited to pure gas or liquid compression, that is increasing of pressure, and dry vapour or gas expansion, meaning transcritical and vapour compression cycles unsuited for such systems.

Ecop is developing a new system where the rotating tubes shown are replaced by plates of a Printed Circuit Heat Exchanger (PCHE, Figure 5) which allows a much more compact and powerful system with temperatures to 250C and power to 10MWt. PCHE (Figure 5) are proven technology that allows heat exchangers with as low at 1K approach temperatures (Chai, 2023) in a very compact form, able to withstand temperatures to 800°C and pressures of 800bar. In these, plates with chemically etched channels are diffusion bonded together to form a single unit. Heat exchange wall conduction distances as small as 1mm are common (Sui, 2022). Such plates are used by ecop to create discs with the compressor, heat exchangers, and expander all on a single disk, which are then stacked together. Their new system offers higher performance in a much more compact form.

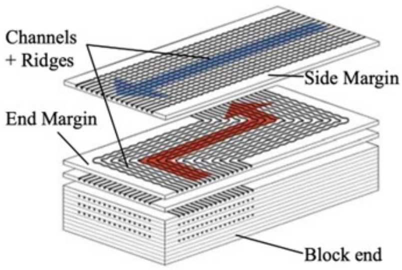

Figure 5. Printed Circuit Heat Exchanger (PCHE).

3. ROTARY HEAT ENGINE CONCEPT

A rotary heat engine concept is described below. In this, the rotary compression concept described above is applied to the pump of an ORC cycle system as shown in Figure 6. This disk also combines the pump and heat exchanger on a single disk, with the working fluid path shown in green. It enters axially and then the radial outflow of the pump, and then fluid then enters the heat exchanger, spiralling counter to the direction of rotation. The fluid then flows axially to the expander disk, where it then flows radially inward, expanding as the centrifugal forces decrease with decreasing radius. The system modelled is of plate 20cm radius and 2mm-10mm thick. They may be PCHE using chemical etching or water jet cut plates. The flow of the high temperature heat transfer fluid (HTF) into the reverse side of the heat exchanger is not show for brevity. This HTF would be heated by the waste heat source then routed axially into the rotary system. The path of the incompressible HTF (generally a mineral oil) is similar to the pump outward and then with a crossing flow of the working fluid, creating a counter flow heat exchanger.

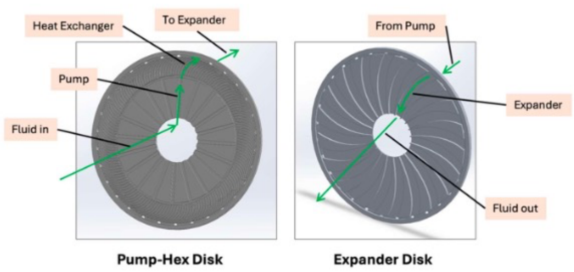

Figure 6. Pump-Hex and Expander Disks.

The basic disk shown above would be layered into a stack that is then diffusion bonded (essentially baked together), creating a single rotating unit performing the pump, heat exchanger, and expander functions. The stack, the layering of the specialized disk, is shown in Figure 7. In this, the working fluid is shown in green and the HTF shown in red. The disk stack, from left to right, consists of pump end plate (dark grey), pump-heat exchanger plate, HTF routing plate, the expander plate, and the expander end plate. These are shown as single disk, yet these may be stacks of disks to allow for the desired mass flow and system power. The routing of the working fluid and HTF in and out of the axial area is not show for brevity. A number of engineering studies of fluid path options and required seals have been undertaken. A 20cm radius rotor and 70cm total rotor length allows a 324kW heat engine at 1 kg/s mass flow (R1234zf).

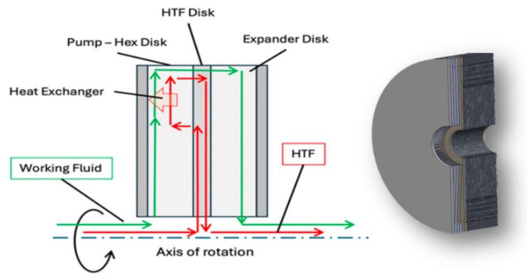

Figure 7. Disk stack fluid flow.

In the concept, the fluid enters the disk axially (Figure 6, left) and flows radially outward with hydrostatic compression acting as a pump; these channels will generally be 30mm x 40mm (depending on system design power). The fluid then enters the rotary heat exchanger with the small channels of the counter flow heat exchanger. The heat exchanger will either be laser jet or water jet cut aluminium sheets, allowing 3mm x 4mm channels and 2-3mm heat exchange wall thickness, or PCHE with 2mm x 1mm channels and 1mm wall thickness. CFD and analytic studies of optimum channel sizing, taking into account heat flow, wall thickness, pressure drop and fluid flow, and cost have been ongoing.

In this initial scoping of the rotary heat engine potential, total heat energy (kW) into the rotating heat exchanger was used, evenly distributed over the annulus area of the heat exchanger area of the disk. Future studies of the heat exchanger design will include channel shape optimization for fluid flow in the very steep force gradient rotating frame. The dynamic nature of the fluid flow though the pump, heat exchanger, and then expander dictate a 1D dynamic model to fully understand the fluid behaver and optimum heat exchanger parameters. This will be critical for the operations from sub-critical to supercritical, where the fluid behaviour is less well understood (Sui 2022) A 0D model was created in Mathematica using fluid models of CoolProp (Bell, 2014), the cycle diagram is shown in Figure 8 with rotating elements shown in grey. As shown above, the working fluid and HTF enter the rotating system axially. The meter pump is used to control the mass flow into the rotating pump.

3.1. ROTARY EXPANDER AND WET EXPANSION

It is assumed that the rotary expansion should be constrained to remain vapour, outside of the vapour dome. This may require a dry or isentropic fluid or require superheat to insure this. It is conjectured that the formation of liquid with huge density compared to vapour would not exit the expanded, instead reversing flow due to centrifugal force of rotation. While this would then just enter the rotary evaporator and then be revaporised, the actual liquid droplet formation radial location along the steep pressure gradient and effects are unknown, possibly damaging the system. In the rapidly rotating system, asymmetrical formation of liquid would cause a dynamic imbalance and potential damage the system.

4. ROTARY HEAT ENGINE MODEL

The cycle arrangement is shown in Figure 8. In this the rotating elements are shown in the grey shaded area. The working fluid enters the rotating element axially, then radially outwards, into the pump (Pts 2-3), through the heat exchanger (Pts 3-4), and radially inwards in the expander (Pts 4-5). While the HTF fluid (Pts 8-9) is shown to the side, the HTF is routed into and out of the rotating element axially. It is depicted as such for clarity. The meter pump (Pts 1-2) would be used in at least the development system to meter the flow of working fluid into the rotary system.

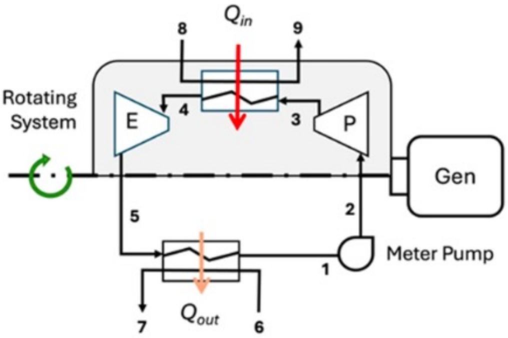

Figure 8. Rotating heat engine diagram.

Hydrostatic compression as a pump and in the opposite of mass flow, expansion of Equation 3 and the OCR cycle equations presented below apply to the rotary heat engine system. It is assumed that the expander and pump isentropic efficiency are 0.95. While ecop has validated expander efficiency to be greater than 0.98, this is with supercritical fluid of the Joule cycle. It is unknown if the expander performance will differ significantly expanding from either super critical or from vapour to lower energy vapour. The fluid behaviour of hydrostatic compression is shown in Figure 9.

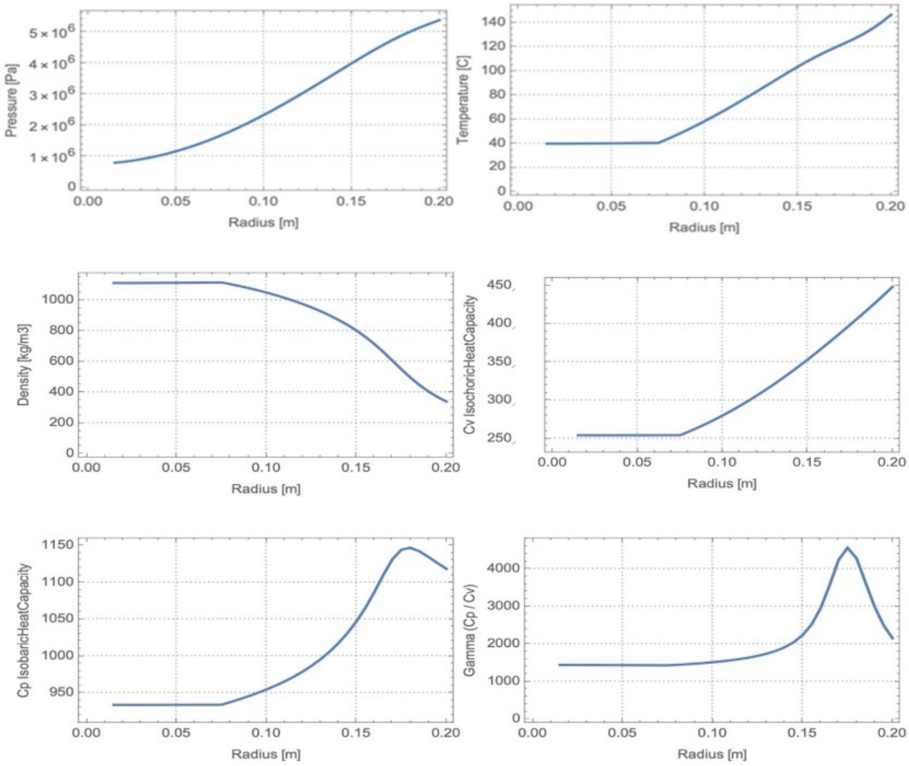

Figure 9. Fluid behaviour in pumping and heat exchanger.

The OCR powers and efficiencies are found using Equations 4-9, with subscript points of Figure 8.
Power of compressor:

$$Pwr_p = m(h_3 - h_1) \tag{4}$$

Power of expander:

$$Pwr_e = m(h_4 - h_5) \tag{5}$$

Heat inflow:

$$Q_{in} = m(h_3 - h_4) \tag{6}$$

Power of system:

$$Pwr_{HE} = Pwr_e - Pwr_p \tag{7}$$

Heat engine efficiency:

$$\eta_{HE} = \frac{Pwr_e}{Q_{in}} \tag{8}$$

Second order efficiency:

$$2nd\ Order\ \eta_{HE} = \frac{\eta_{HE}}{\eta_{HE\ ideal}} \tag{9}$$

Where:

m : mass flow of working fluid [kg/s]
Pwr_i : Power of compressor, expander, or Heat Engine [W]
Q_{in} : Heat flow from HTF into heat engine [J/s]
h_4 : Enthalpy [J/kg]
η_{HE} : Heat engine efficiency [-]

5. OPERATIONAL PARAMETERS: RPM AND HEAT FLOW

With increasing rotor rpm, the resultant pump pressure rapidly increases to allow pressures greater than the fluid critical point. This is shown in Figure 10 with increasing RPM, allowing a single system to adjust to source heat temperatures to achieve optimum expansion rations. The transcritical cycle also allows for higher exergy due to the improved temperature matching of the heat exchanger (Woodland, 2020).

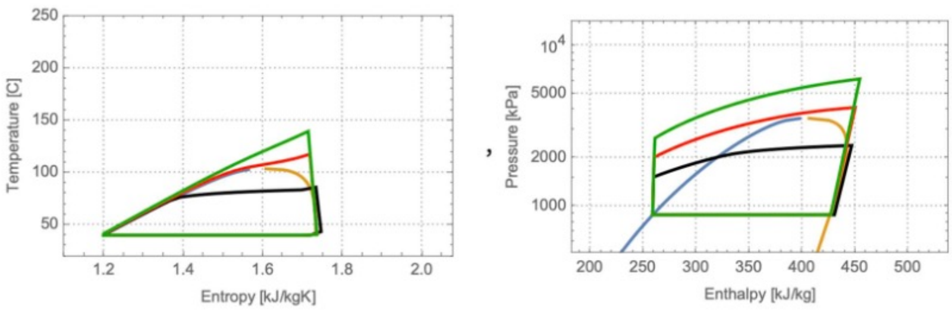

Figure 10. Operations at 3300, 4400, 5500 RPM.

The optimum RPM increases with source temperature, with the tcORC becoming optimum with temperatures above 140C for R1234zf. The optimum RPM for the source temperature of 110C is shown in Figure 11 (also in black of Figure 10).

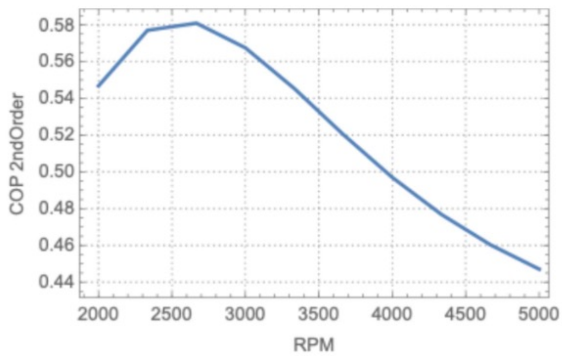

Figure 11. 2nd Order efficiency as function of RPM

The design heat flow into the system also shows an optimum, depending on ORC and tcORC. The design optimisation would need to consider the design point and off design point efficiencies for the anticipated heat source for the system.

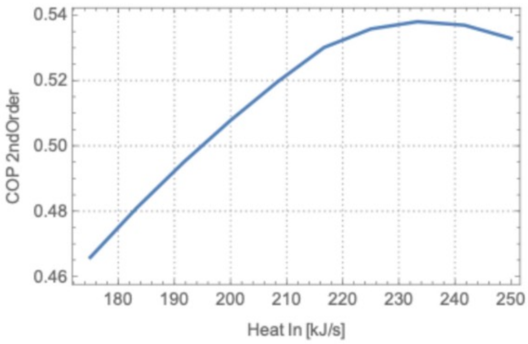

Figure 12. 2nd Order efficiency as function of heat flow.

6. DESIGN PARAMETERS

The primary geometry design parameter for the rotating disk is the portion of the disk as a pump and portion as heat exchanger. Effectively, the heat exchanger is an extension of the pump, with the hydrostatic fluid pressure still creating a force radially outward. In the heat exchanger the working fluid undergoes heating, either increasing the latent energy for sub-critical cycles, or from liquid to supercritical liquid to supercritical vapour. Figure 13 shows the heat exchanger radius starting at 40%, 60%, and 80% of the total disk radius. The effect is very apparent in the P-h plot, where the pressure is increasing during heat exchange. The significant changes in heat exchanger pressure are not readily possible in other systems. Figure 14 shows the second order efficiency of the system with increasing start of the heat exchanger radius, increasing from 0.40 to 0.90, with optimum near 0.7, or 70% of the disc radius. This changes for the design point, with the optimum being less for sub-critical ORC, and to the right for tcORC systems.

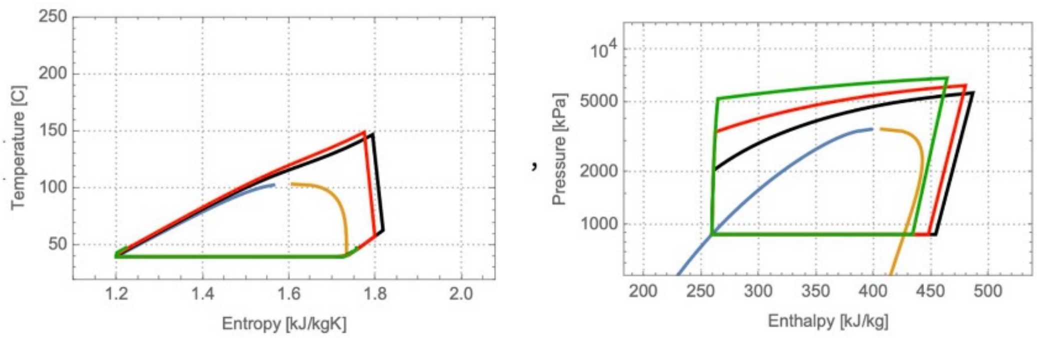

Figure 13. Hex radius part starting at 40%, 60%, and 80% of the total disk radius.

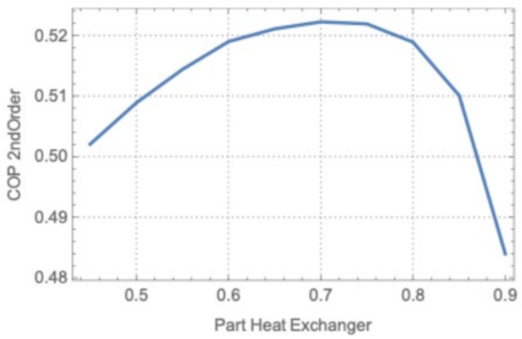

Figure 14. 2nd Order efficiency as portion of pump-heat exchanger disk as pump.

7. ROTATIONAL FRICTION

While the potential pump and expander efficiencies of the proposed system indicate a highly efficient heat engine system, the rotational friction of the seals and bearings have yet to be accounted for in this initial concept modelling. With the rotation rates in the range of 2500-4500 rpm, readily available, low-cost bearings may be used; friction from these are minimal. Initial engineering concepts indicate that two of the rotational seals will be required to be non-contact labyrinth seals, and two contact type seals (likely PTFE lip seals) with seal velocity <20 m/s, meaning off the shelf, readily available, off the shelf commercial seals will be suited for the R&D and potential commercial systems. The system arrangement allows from minimal pressure (<3 bar) across the seals indicate minimal rotational friction. No attempt to include the seal and bearing friction into this initial concept modelling was made. Angular momentum of the working fluid and heat transfer flow as it flows radially outwards and inwards is completely recovered if the inlet and outlet diameters are the same. The optimum arrangement of the fluid inlets and outlets will be part of the fuller system design. The fluid friction moving through the system would result in pumping power requirements, thus torque and loss of generator power.

Only with the first of the kind system will the potential of the concept become apparent. While the modelling above suggests second order efficiencies on the order of 0.50, frictions are expected to reduce this somewhat. In the actual system, the efficiency will be determined via Equation 10, where seal and bearing friction would reduce the W.

$$\eta_{ORC} = \frac{W_{out}}{Q_{in}} \tag{10}$$

Where:
η_{ORC} : Efficiency of ORC system
W_{out} : Work output of system [J/s]
Q_{in} : Heat into the system [J/s]

8. OTHER FLUIDS

An initial assessment of additional fluids are shown below. The simplicity of the rotary heat engine system should allow adaptation to many of the fluids with a single system, allowing and extensive research and development program with the same system, requiring little more that seal material consideration.

Table 1. Assessment of potential working fluids in the rotary heat engine, Source at 110C.

Working Fluid	$\eta_{ORC\ model}$	**2nd Order η**
R1243zf	7.07%	50%
R245fa	7.13%	35%
R1233zd(E)	5.95%	27%

9. FURTHER WORK

The rotary heat engine is fundamentally a dynamic system with close coupling of the pump, heat exchanger, expander, and condenser. To further develop a full understanding of the system operation, a 1D model must be developed. Specific Modelica models are being develop, modifying the dynamic pipe models to allow for hydrostatic compression. This allows modelling of the pump, heat exchanger, and expander, and also the heat transfer fluid flow in a dynamic model to assess normal operation, off design point operations, as well as start-up and shut-down sequences.

The model of the heat exchanger would include latent heat in the evaporator of an ORC and sensible heat in the supercritical region of the tcORC operations. 1D models of supercritical CO_2 PCHE have been developed by Sui xiv and others. These have allowed assessment of the heat exchangers in steady state, off design point, and transients conditions, to include start up and shut down. Such a model will need to be expanded to include operation in the vapour dome. Additional aspects to be investigated include;

- Sealing: labyrinth seals to allow non-contact, long-life seals

- PCHE, water jet or other cut plates to be diffusion bonded and form the rotary system.
- Heat exchanger design to include channel depth and width, and conduction wall distance.
- Expander geometry and the option to include a Radial Outflow Turbinexv allowing greater expansion and energy recover of the expanding working fluid. This would allow the fluid to expand radially inwards, then radial outwards, creating a compact, simple arrangement.
- Evaluating possible working fluids to include HCFO, HFO, Hydrocarbons, and other potential fluids.
- Full implications of off design point operation
- System scaling studies (10kWe to 2MWe systems)

10. CONCLUSION

A new concept of a rotary heat engine is presented. The system would be very simple in operation with only one moving part. The initial modelling includes 0D scoping models to understand the basic behaviour of the system. The results show the concept has the potential to allow high efficiencies with optimal fluids, on the order of 50% second order efficiency. Uniquely, the system potentially allows sub-critical and supercritical operations with the same mechanical system by adjusting the RPM, thus resulting pumping pressure, tuned to the waste heat provided and heat engine needs. The effects of varying RPM, heat flow, and design geometry parameters are shown.

The modelling points to future modelling and development needs. The rotary system and unique arrangement will require extensive modelling to fully understand the optimum design. The mechanical arrangement will also require extensive modelling and design considerations of fluid inflow and outflow of the rotary system arrangement and sealing to then be progressed to a mechanical system for research and development.

AUTHOR CONTRIBUTIONS

Chris Benson: Conceptualization, conducted research, data analysis, writing, and revised the manuscript.

DISCLOSURE STATEMENT

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

DATA AVAILABILITY STATEMENT

The data presented in this paper is available upon request to the author.

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