DEVELOPMENT AND TESTING OF A FREE PISTON LINEAR EXPANDER FOR ORGANIC RANKINE CYCLE BASED WASTE HEAT RECOVERY APPLICATION

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ABSTRACT

The global community has agreed to limit global warming well below 2ºC (UNFCCC. Conference of the Parties (COP) 2015). Automotive internal combustion engines (ICE) generally convert 40% of fuel energy into useful power and discharge the remaining to the environment. Disruptive ideas are needed to achieve the set goals for improvement of energy conversion efficiency. Organic Rankine cycle systems appear to be favorable for waste heat recovery from the automotive exhaust, but the technology readiness levels are still low for mobile applications particularly due to non-availability of suitable and cost-effective expansion machines in mini-scale (<20 kW) power scale.

The work is focused on the development and experimental testing of linear free piston expander. Free piston expanders are suitable for small size application and have smaller leakage losses. The expander also has a passive inlet port which controlled by bounce pressure due to small recompression and thus complex valve mechanisms can be avoided. The generator also encompasses a linear electrical machine on the piston of the expander to generate electricity directly. Multi-physics models were used to understand the gas expansion characteristics, valve dynamics, leakage losses and heat losses using Mathworks Simscape. The models helped to size the equipment (ports/plenum/chambers etc.) and also provided the control schemes for the control of the expansion machine

The work also presents details of adaptation/retrofitting of an organic Rankine cycle (ORC) rig originally used for testing of a turbo-generator. The current proposed expander is based on R1233zde so the balance of plant was re-iterated along with the specific expander integration equipment details. The expander was tested at 8 bar inlet pressure and 1 bar discharge pressure, the machine under investigation is the downsized version to generate 2.5 kWe power.

The results concluded that the proposed expander is an effective solution for waste heat recovery applications for 10 kWe scale output. Despite, lower efficiencies compared to a turbo-expander, the design and shape permits integration of pump and high-temperature heat exchanger to the expander to form a compact unit, thus saving cost and weight.

1. INTRODUCTION

The ever-growing world population requires the development and extended exploitation of renewable and conventional energy sources. The global energy requirements can be met by introducing a larger share of renewables and increasing the efficiency of current energy conversion processes. The complexities further arise when it has to be assured that in future, more energy will be produced but emissions have to be kept lower than the current state to keep global warming under check. The transportation sector accounts for approximately one-third of the global CO2 emissions (Alshammari *et al.* 2018a). Transportation sector as of today is still heavily dependent on internal combustion engines (ICE). ICE engines, even with advanced technologies like, exhaust gas recirculation (EGR), turbocharging, variable valve controls etc. can still convert only 40% of the fuel energy to useful power and release the remaining 60% to the ambient. Disruptive ideas are needed to further enhance the efficiency of the ICEs. In recent years Organic Rankine cycle based power systems have gained popularity and are considered a viable solution for waste heat recovery application (Tchanche *et al.* 2011). Organic Rankine cycle based system operates in a similar manner to the Rankine cycle based power systems but employee organic working fluid which allows the flexibility to recover heat even below 100 ^{O}C. Figure 1 presents the major components of an ORC based waste heat recovery system. Exhaust gas stream from ICE can be passed through the evaporator of ORC to transfer heat to the pressurized working fluid which can expand through an expander to generate mechanical/electrical power. The low-pressure vapor is then condensed in the condenser which could be supplied with the automotive radiator fluid to reject heat at a lower temperature. The fluid is pumped again and the cycle continues to operate. The figure also presents the temperature-entropy diagram of the process. The added advantage of ORCs based heat recovery system is that, they are autonomous, require low maintenance, can recover heat from heat sources which are at a lower temperature to be suitable for a Rankine cycle system. In general, ORCs also have simple architecture and less number of components.

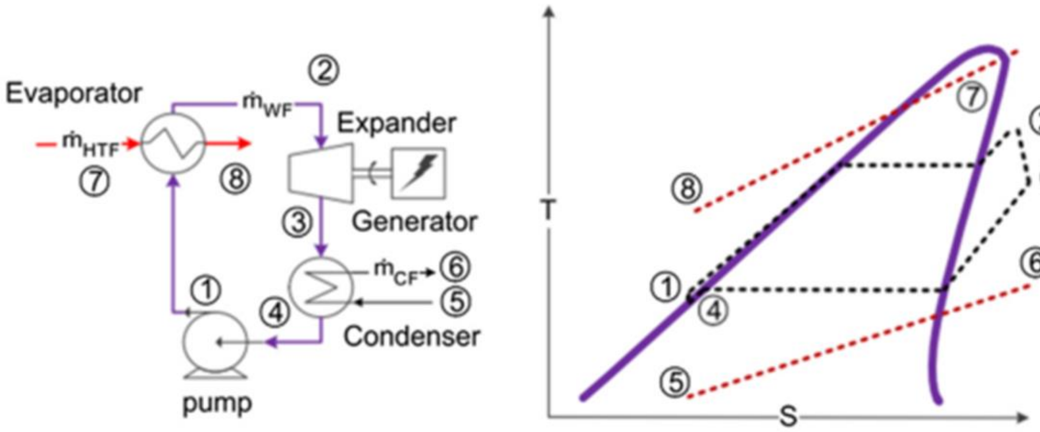

Figure 1: Schematic of Organic Rankine Cycle and Ts diagram (Park et al. 2018)

Although the ORCs have acquired higher technology readiness levels for stationary applications but the installation and its economic viability for automotive application require further developments. The added weight of equipment is one of the major concern along with the packaging volume. The expansion machines for mini-scale application are still under-development. Turbo-machines are well-known for the higher efficiencies compared to the volumetric types but the supersonic flows and small tolerances within the turbo-expanders contribute to larger costs. In order to address these issues, a new type of linear free piston expander is being investigated in this work which can accommodate, the expander, pump, generator and a heat exchanger within a single unit to provide a very compact and a robust solution.

2. MATERIAL AND METHODS

The proposed gas expander is being developed primarily by Libertine FPE Ltd ("Gas expander development - Libertine") who have the expertise to develop linear actuators and linear piston engines. The gas expander being developed comprises of an electrical machine with two opposing expansion chambers at each end. Figure 2 presents the CAD of the proposed machine which is also housing an electrical machine directly connected to the expansion pistons.

Figure 3 presents a cross-section of one half of the gas expander. A metallic rod with permanent magnets (translator) traverses through electromagnetic coils while generating power during expansion stroke at one end of expansion chamber and while recompressing fluid to activate the passive transfer valves before that end goes into expansion mode. The organic working fluid enters the machine through a gas inlet port.

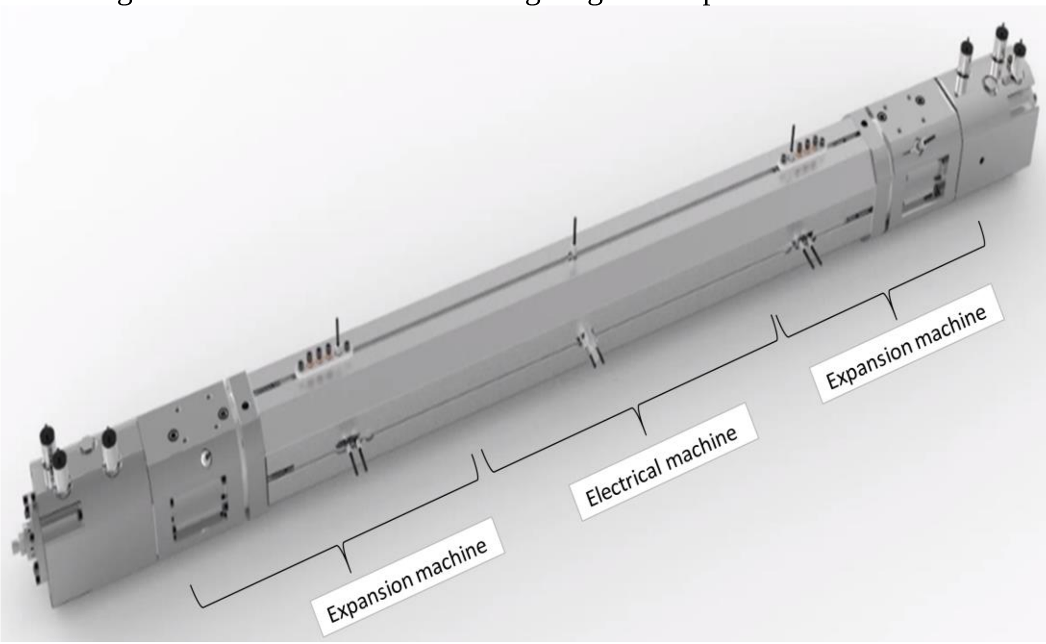

Figure 2: CAD of the investigated linear free piston gas expander with integrated electrical machine

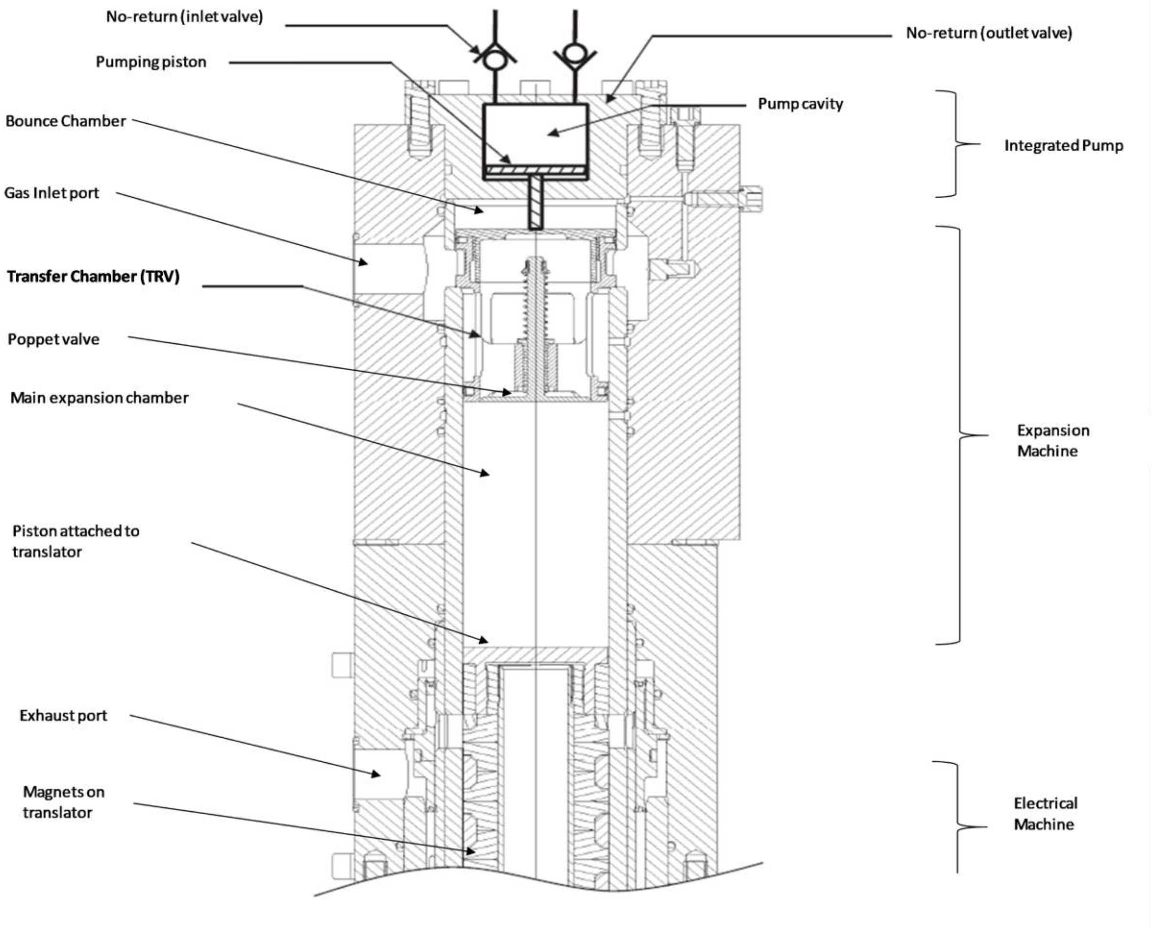

Figure 3: Cross-Section through one half of the gas expander

When one end of the expander is undergoing the expansion stroke the alternating end recompresses the residual gas to a pressure higher than inlet port. The higher pressure in the main chamber due to recompression causes a transfer valve to offset from its seating position and fill the high-pressure gas into a transfer chamber. The poppet valve in the transfer chamber opens to release the high-pressure gas into the main chamber which pushes the piston to move in the opposite direction. The force exerted on the piston generates motion of translator relative the electromagnetic coils generate electrical power. At the end of the stroke, the expanded gas moves out of the machine by the exhaust port, which if controlled actively, can make a variable expansion ratio machine.

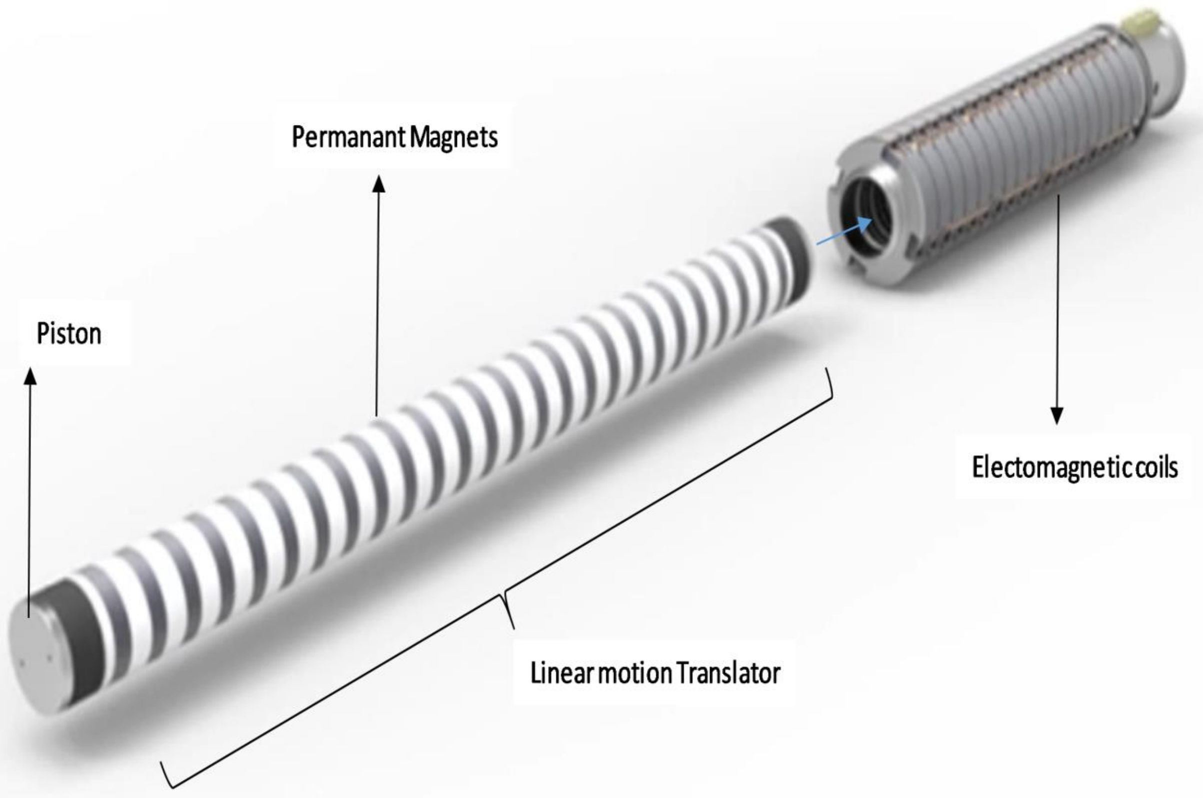

Figure 4: Translator with electrical coils arranged in the electrical machine with ends fitted with pistons for gas contact

Figure 4 presents the arrangement of the linear piston and the electrical coils which together make up the electrical machine, the ends of the translator act as piston and interact with the organic fluid gas. The pumping chambers can be added to each end of the expansion chamber with a system of no-return valves to act as volumetric (piston type) pumps to pressurize the fluid.

Brunel University London is responsible for the development of a multiphysics dynamic model for the proposed machine to investigate the thermo-fluid and heat transfer interactions within the expansion and recompression cycles within the machine to validate the design and support in the development of the controller for the actuation of the machine. The model will provide a suite of analytical and developmental tools for the development of the machine. The model will support the testing of alternative working fluids, variable expansion ratios, operation at different frequencies, characterize the friction and heat losses, and operation at different pressure and temperature conditions.

A 1D model for the machine was developed in Mathworks Simscape ("MathWorks Matlab") programming environment which supported physical connections, multi-physics, parametrization, acausal modelling approach and fluid properties could be incorporated by using look-up tables and NIST database ("NIST Standard Reference Database 23").
The overall model was composed of various sub-models to represent the overall machine characteristics. The scope of this paper is limited to the thermos-fluid interactions while the electrical machine was modelled by Libertine FPE and both models (the thermos-fluid model and electrical machine model) to represent the overall characteristics. Both models interact with each other based on the velocity profile along with translator position with respect to the position encoder data.

For initial testing purpose, one expansion chamber was modelled only and the other expansion chamber was replaced with a bottom dead centre bounce chamber to ensure the electrical machine does not cross current limits for initial testing.

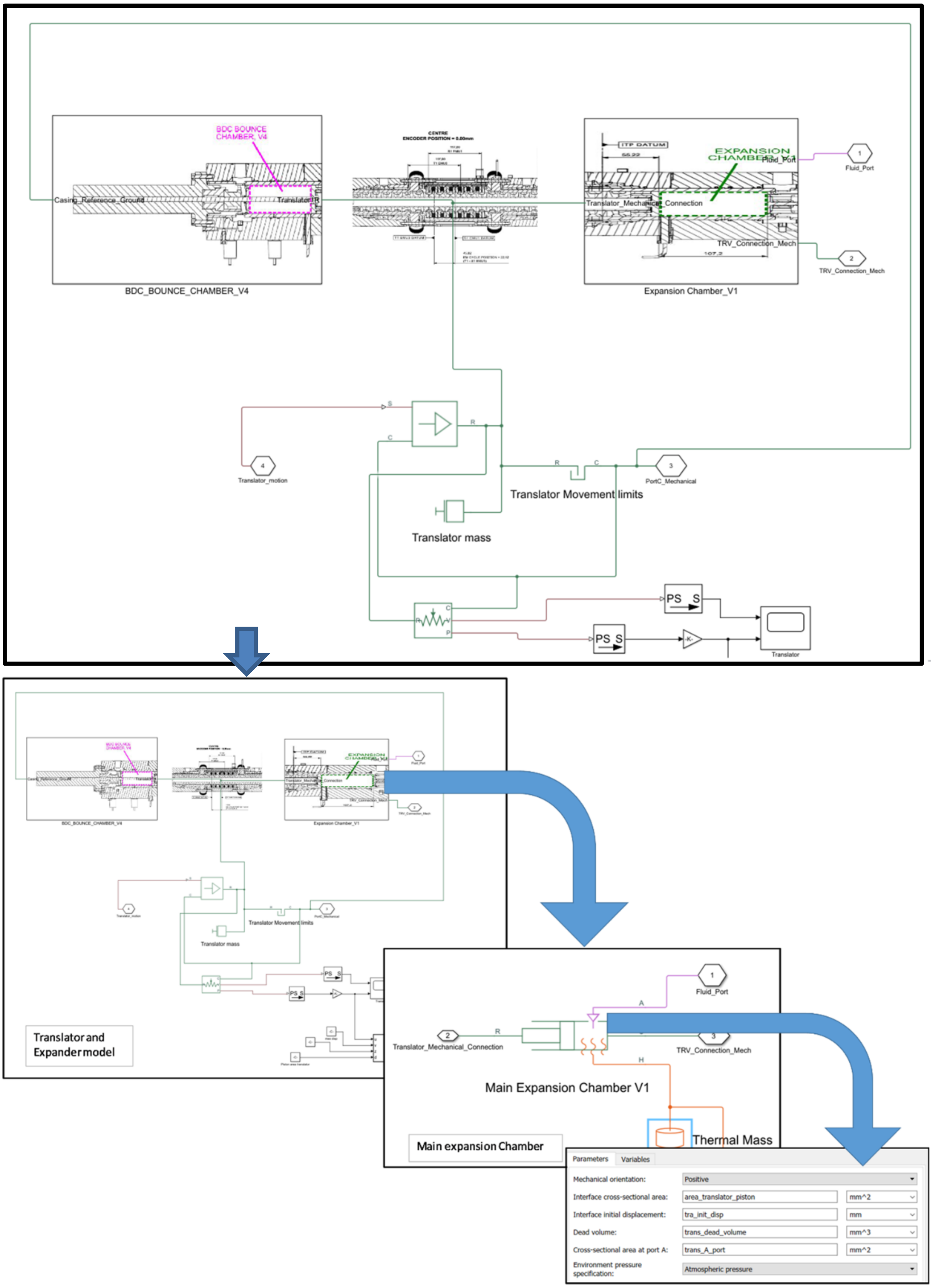

Figure 5. Hierarchy of the expander model in Simscape

Figure 5 presents the hierarchy of the expander model and its sub-model components and how the submodels were parameterized. An initialization file generated with all the parameters to be tuned

with minimal model details requirement for easier tuning and validation with the experimental data.
The mechanical translational mechanical converter in Simscape is an interface between a gas network and a mechanical translational network. The block converts mechanical force intor pressure and vice versa. Port A in Fig 5 is gas convserving port associated with converter inlet, port H is associated with temperature of gas inside the converter and port R and C are the mehanical translational conserving ports associated with moving interface and converter casing.
Following set of equations are solved to evaluate results.

Mass balance:

$$\frac{\partial \mathrm{M}}{\partial p}.\frac{\partial \mathrm{p}_1}{\mathrm{dt}} + \frac{\partial \mathrm{M}}{\partial T}.\frac{\partial \mathrm{T}_1}{\mathrm{dt}} + \rho_1 \frac{\mathrm{dV}}{\mathrm{dt}} = \dot{m}_A \qquad \text{Eq. 1}$$

Where
$\frac{\partial M}{\partial p}$ is the partial derivative of the mass of the gas volume with respect to pressure at constant temperature and volume.
$\frac{\partial M}{\partial T}$ is the partial derivative of the mass of the gas volume with respect to temperature at constant pressure and volume.
p_1 is the pressure of the gas volume. Pressure at port A is assumed equal to this pressure, pA = pI.
T_1 is the temperature of the gas volume. Temperature at port H is assumed equal to this temperature, TH = TI.
ρ_1 is the density of the gas volume
V is the volume of gas
t is time
$\dot{m}_A$ is the mass flow rate at port A. Flow rate associated with a port is positive when it flows into the block

Energy Balance:

$$\frac{\partial \mathrm{U}}{\partial p}.\frac{\partial \mathrm{p}_1}{\mathrm{dt}} + \frac{\partial \mathrm{U}}{\partial T}.\frac{\partial \mathrm{T}_1}{\mathrm{dt}} + \rho_1 h_1 \frac{\mathrm{dV}}{\mathrm{dt}} = \phi_A + Q_H \qquad \text{EQ. 2}$$

Where,
$\frac{\partial U}{\partial p}$ is the partial derivative of the internal energy of the gas volume with respect to pressure at constant temperature and volume
$\frac{\partial U}{\partial T}$ is the partial derivative of the internal energy of the gas volume with respect to temperature at constant pressure and volume
ϕ_A is the energy flow rate at port A.
Q_H is the heat flow rate at port H.

Gas Volume:

$$V = V_{dead} + S_{int}.x_{int}.\varepsilon_{int} \qquad \text{EQ. 3}$$

Where
V_{dead} is the dead volume.
S_{int} is the interface cross-sectional area.
x_{int} s the interface displacement.
ε_{int} is the mechanical orientation coefficient. (positive=1 negative=-1)

Force Balance:

$$F_{int} = (P_{env} - P_1)S_{int}.x_{int}$$

Where
F_{int} is the force from port R to port C
P_{env} is the environment pressure

The developed machine was tested with pressurized air and its motion characteristics were being monitored, the experimental data was used to calibrate the frictional characteristics and match the motion profile. The testing conditions used of the model are presented as following:

Table 1: Key parameters for testing (initial conditions)

Key Parameters	Units	Values
Main chamber pressure	Bar (abs)	1.17
Transfer chamber pressure	Bar (abs)	0.94
Inlet manifold pressure	Bar (abs)	24.27
Bounce chamber pressure	Bar (abs)	24.8
Exhaust manifold pressure	Bar (abs)	1
Main chamber temperature (estimated)	C	30
Cyclinder diameter	mm	45
Working medium	Compressed Air	

3. RESULTS

The validated model result presented in Figure 6 reports a re-compression of the residual gas volume to a 60 bar peak. The main chamber pressure values utilized to acquire the forces on the piston and resultant velocity profile was iterated with electrical machine model to provide the overall motion profile of the machine. It can be seen from the figure 6, that during the compression phase the main chamber pressure rises beyond the inlet manifold pressure which is at same pressure as bounce chamber. The higher pressure triggers the movement of transfer chamber which was initially at lower pressure (0.94 bar), the movement cause the chamber to become in contact with the inlet manifold and filled by the high pressure working fluid. This is the reason for increase in pressure from time 0.093 to 0.097. The next even is expansion stroke, as the main chamber pressure start droping the from peak pressue and becomes slightly lower than transfer chamber pressure, a poppet valve in transfer chamber opens and release the pressurized working fluid to mainchamber, from that point onwards throughout the expansion stroke is the energy recover (power production zone) and main chamber pressure remains same as transfer chamber.

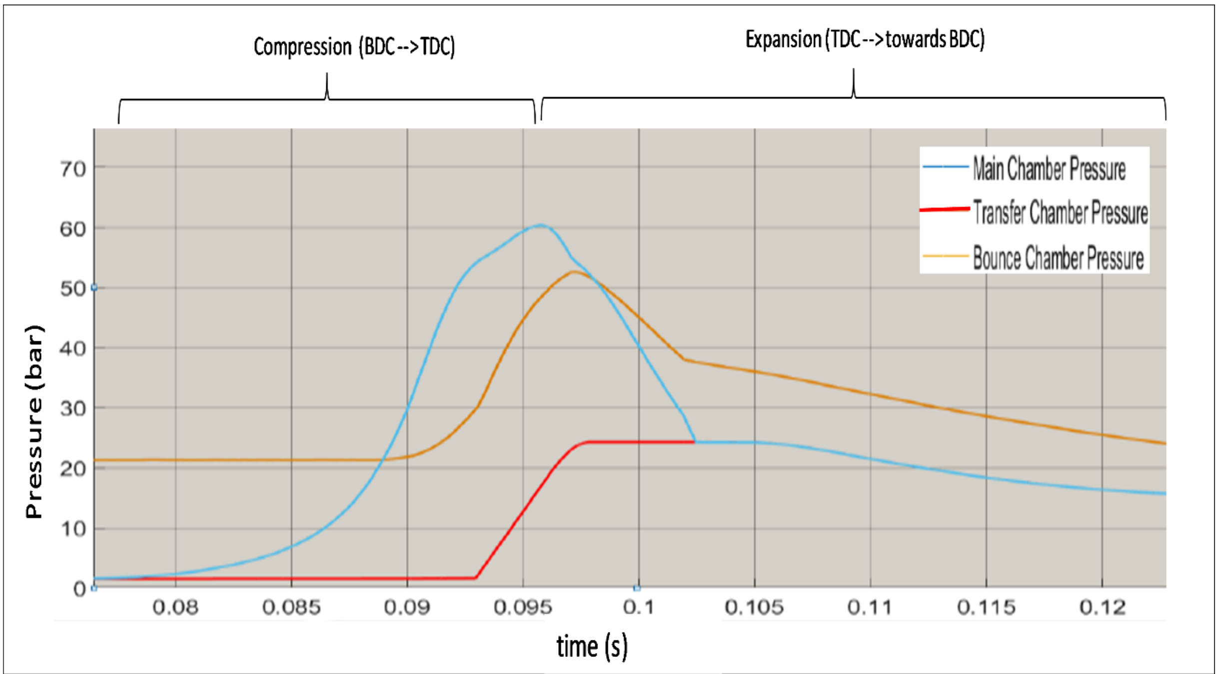

Figure 6: Resulting pressure profile against single re-compression and expansion cycle for a given motion profile

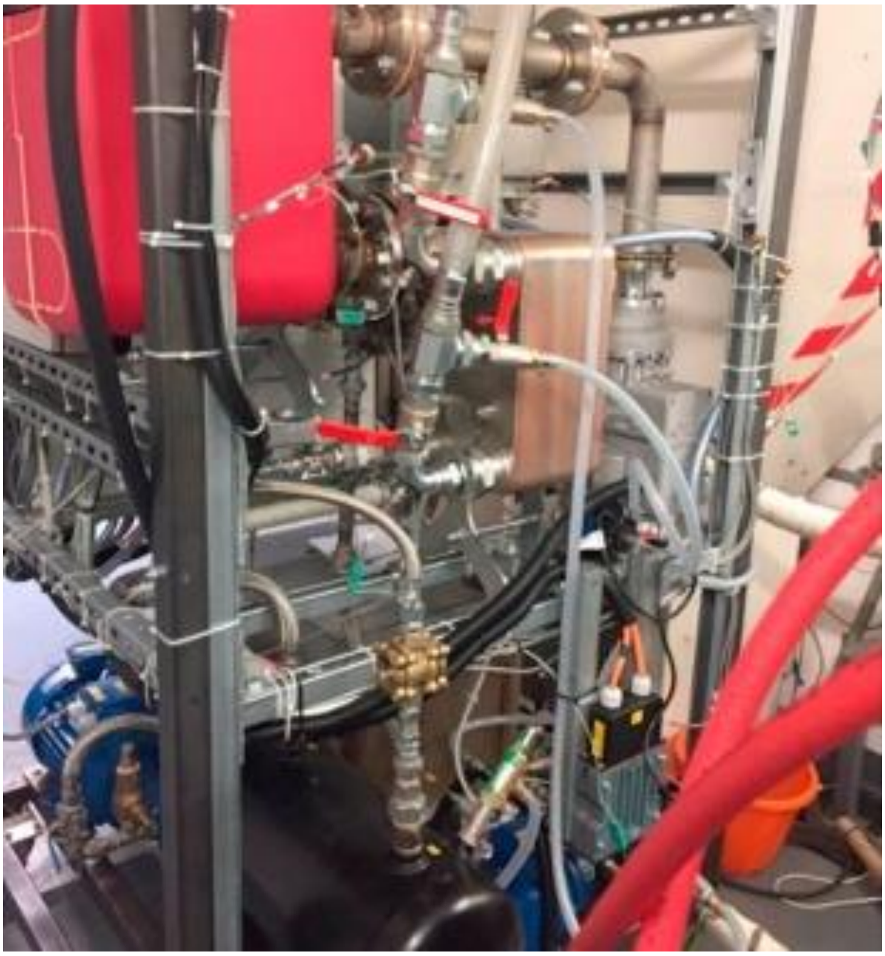

Figure 7: ORC rig at Brunel University London

4. CONCLUSION

The simulation results conclude that although the expander might not surpass the efficiencies values of a turbo-expander its lower cost and its ability to integrate major ORC waste heat recovery components in a single package can bring the cost of the systems well below £2,000/kWe if the system is produced at commercial scale.

The future work includes testing the machine at Brunel University London. Figure 7 presents the ORC rig at Brunel University London which was developed by Entropea labs ("Entropea Labs") and Brunel for the testing of waste heat recovery system using a heavy-duty diesel engine (Alshammari *et al.* 2018b). The test rig was originally designed for testing of a turbo-expander based on Novec649 working fluid but it is being retrofitted to test the free piston gas expander at much smaller power scale and using R1233zde working fluid.

NOMENCLATURE

$\dot{m}$	Mass flowrate
HTF	Heat transfer fluid

Subscript

wf	Working fluid

cf Cooling fluid

T Temperature

s Entropy

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