Validation of performance for a novel compressor-expander concept

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ABSTRACT

A novel mechanism that can be adapted for use as a compressor or as an expander is described. The results of detailed modelling of performance in both modes are presented in the form of graphs of isentropic and volumetric efficiency plotted against pressure ratio and pressure difference. These are compared and contrasted with equivalent graphs for conventional types of compressor. This comparison is used to demonstrate how the novel concept combines the benefits of both reciprocating and screw compressors and expanders. The model results are validated against performance data measured on a test cell designed to enable rapid testing across a range of operating conditions and the comparison is used to fine tune the model, enabling other configurations of compressor and expander to be simulated. A detailed description of the test cell is provided and proposals for future development of this work are suggested, with some possible new applications for the novel concept.

This paper is part of a series of presentations and extends work presented at previous conferences, including the Herrick Compressor Conference in 2022. This paper includes performance data that has not previously been published and the discussion explores some explanations for the differences between the novel concept and conventional compressors.

Keywords: compressor, expander, efficiency, performance model, validation.

1. INTRODUCTION

The FeTu turbine is described in a previous paper presented to the Herrick Compressor Conference in 2022 (Pearson *et al*, 2022) so the mode of operation is not repeated in detail here. The present paper presents progress to date in a series of validation exercises which have been in progress since the previous paper, and which have produced some useful insights into the detailed design of the machine. A test cell exploring the use of the turbine as an expander in two different modes of operation is described and test results in an ORC circuit are presented.

A series of simulations of compressor performance for a range of compressor variants are presented. It had been hoped to validate them with some test cell performance figures but at the time of writing this work has not been undertaken. It is hoped to be able to present some test results in July, or in a later paper to the Herrick Conference.

The FeTu turbine combines some beneficial features of reciprocating compressors with some beneficial features of screw compressors. These are identified in the analysis of the simulation and indicate where this machine might be particularly useful in practice.

The FeTu machine comprises a relatively thick blade (22) mounted on a shaft (18) and constrained within a spherical rotor (16) inside a housing (12a and 12b). As the blade rotates the sphere oscillates alternately increasing and reducing the four chambers that exist between the blade and the sphere. This creates two flow paths through the sphere, each comprising two chambers, analogous to a two cylinder double-acting reciprocating machine. An exploded view of the machine, showing two of the four chambers (34a1 and 34a2), is shown in Figure 1. When used as a positive displacement pump the full area of each chamber on the surface of the sphere is open to inlet and outlet ports. When used as a gas compressor the inlet port is the full area of the chamber but the area of the outlet port is restricted. This creates a volume ratio between the maximum volume on the suction side of the compressor, which denotes the swept volume of the machine, and the volume at the point the discharge port opens to the outlet pipe allowing compressed gas to be discharged from the chamber. When the machine is used as a pressure reduction device it can be configured with full sized ports on inlet and outlet or with a restricted port area on the inlet. The former, called a turbine, is suitable when a liquid is used to drive the machine and the latter, called an expander, is suitable when gas is expanded to provide the power output.

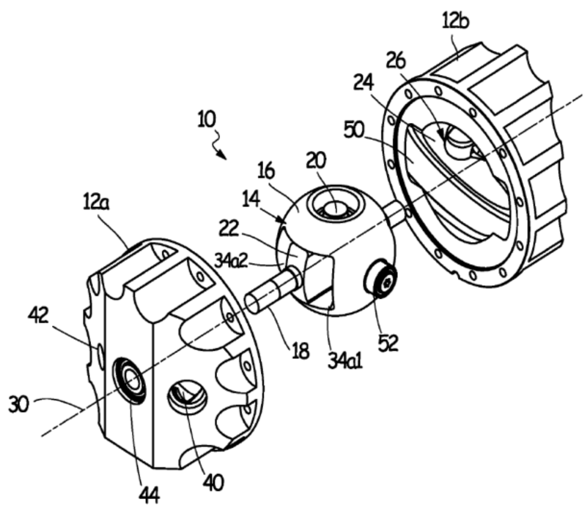

Figure 1: an exploded view of the turbine assembly (Fenton, 2021)

One of the unresolved questions about the use of this machine for power output from pressure reduction is whether it should be configured as a turbine (no inlet porting) or an expander (with inlet porting) when saturated liquid is fed to the inlet. As a turbine the entire volume of the chamber fills with liquid and is then instantaneously sealed from the high pressure side just at the point that it opens to the low pressure side, causing the liquid to flash down to a low pressure gas/liquid mixture. Motive force to the drive shaft is provided by the fact that the blade has high pressure liquid acting on the sphere on one side and low pressure gas/liquid on the other side. As an expander the chamber is sealed from the inlet when it is only partly formed, with the ratio of the inlet volume to the full volume determined by the shape of the port in the housing. This causes flash gas to be created during the rest of the ingestion stroke, enabling some enthalpy to be extracted during the expansion process, but reducing the mass flow capability of the machine by reducing the volume of liquid ingested on each revolution in comparison to the swept volume of the chamber.

2. EXPANDER TEST CELL

A special machine housing was created with open porting on one side and shaped porting on the other, resulting in one half of the machine working as a turbine and the other half working as an expander with an expansion volume ratio (Vi) of 2:1. A test cell was constructed to feed this machine with superheated gas as a first step of testing but it was quickly found that the turbine flowpath needed a regulating valve to reduce the inlet pressure because the mass flow rate through the turbine was much higher than through the expander since the ingested volume is larger through the open porting than the shaped porting. This test gave some encouraging results but it was realized that this mixed mode operation was not feasible and so the machine was substituted for one with a 2:1 expander on both flow paths. A schematic of this test cell is shown in Figure 2.

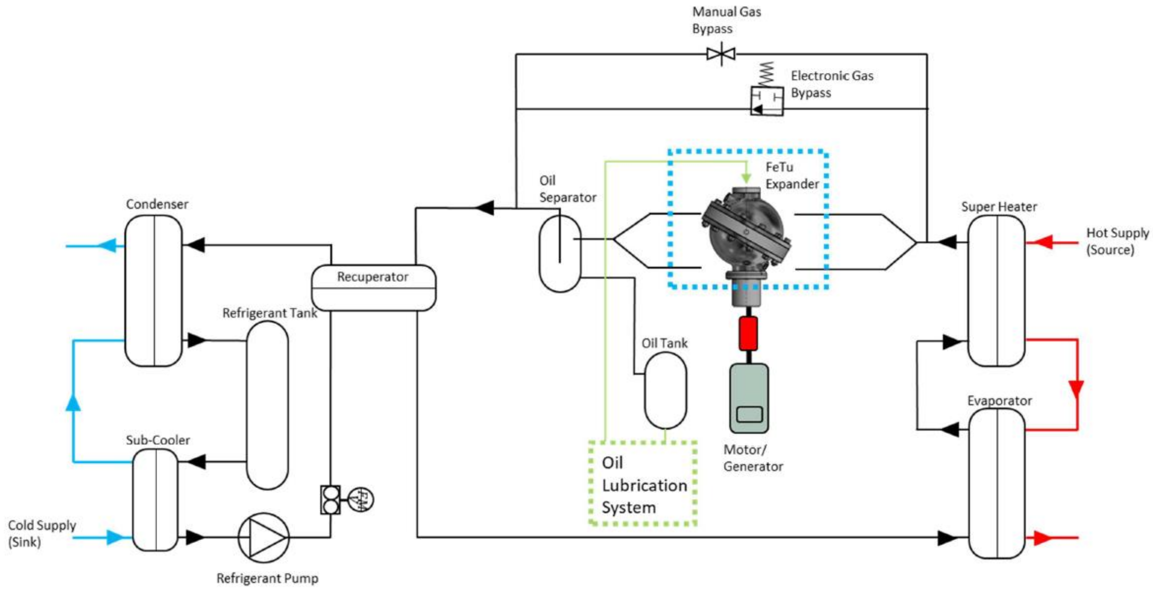

Figure 2: Test cell schematic for dry gas expander operation

Tests for this setup were run with a fixed cold water temperature of 10°C and hot water temperatures of 45°C to 75°C in 5 K steps. The high pressure was measured on the pipe connected to the expander inlet and the low pressure was measured on the refrigerant tank. The internal pressures were recorded using an array of probes built into the body of the expander.

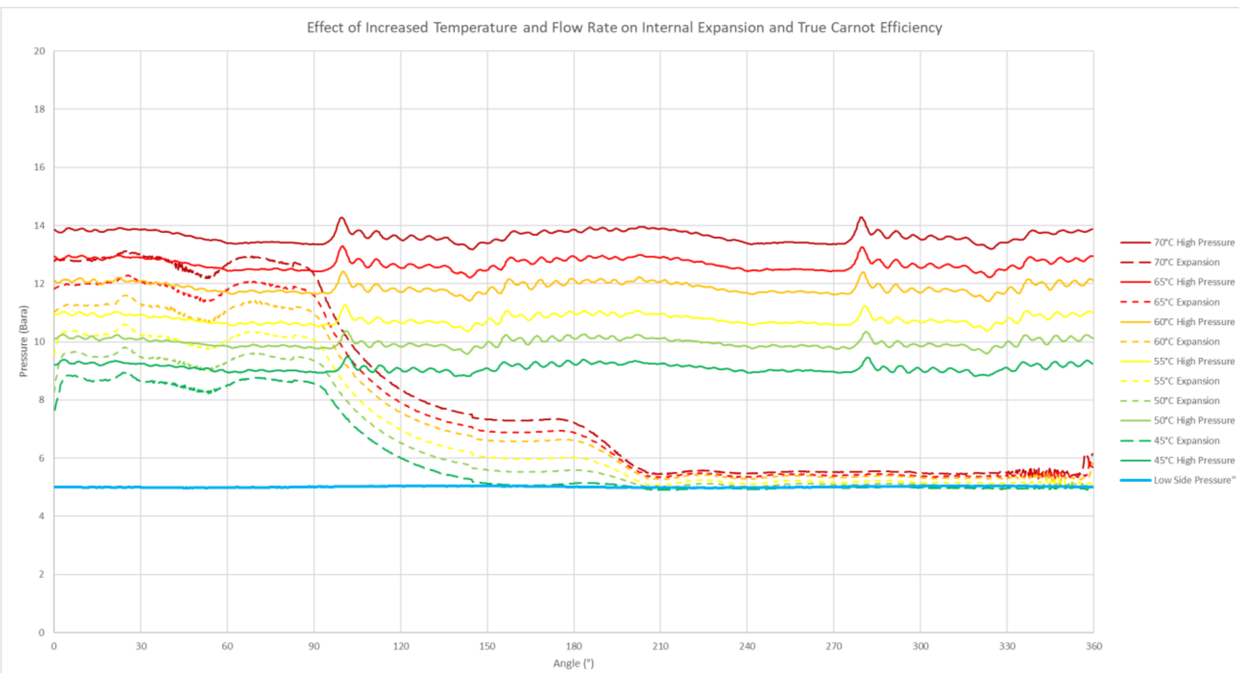

Figure 3: External and internal pressures

The results of these tests are summarized in Table 1, where CCR is the Carnot conversion ratio (sometimes called "Carnot efficiency") based on the cold water supply (heat sink) and the hot water supply (heat source) temperatures.

Table 1: Theoretical and actual heat to power conversion ratios

Nominal	Sink	Source	CCR	Work/Heat	Nett Conv	True %	Nett %	Nett/True
45°C	283.7K	318.0K	10.8%	2.49%	1.84%	23.0%	17.1%	74.1%
50°C	283.2K	323.0K	12.3%	2.66%	1.93%	21.6%	15.7%	72.5%
55°C	282.4K	328.1K	13.9%	2.77%	2.12%	19.9%	15.2%	76.6%
60°C	281.8K	332.3K	15.2%	3.03%	2.29%	19.9%	15.1%	75.6%
65°C	281.2K	337.4K	16.6%	3.32%	2.41%	20.0%	14.5%	72.5%
70°C	281.0K	343.1K	18.1%	3.62%	2.52%	20.0%	13.9%	69.7%

The percentage given as Work/Heat is the output from the expander shaft divided by the heat input from the hot water circuit to the evaporator of the test circuit. The Nett Conversion is when the power input to the refrigerant pump and oil pump are deducted from the shaft power output. The columns True % and Nett % express these conversion ratios as a percentage of the Carnot Conversion Ratio (CCR). The column headed Nett/True shows the percentage of work output available for export after the auxiliary loads are taken into account.

Some further aspects of the results shown in Figure 3 are worth noting.

- The solid lines are the pressure in the pipes leading to the machine and in the refrigerant tank, and the dotted lines are internal pressures within the compression chamber on one side of the machine
- The external pressure traces repeats exactly after 180° rotation showing that the tests are very repeatable
- The internal pressure traces only show the pressure on one side of the blade and the same pressure profile is repeated on the other side 180° out of phase
- The expansion from nominal 45°C (the dark green lines) gives the closest match of pressure ratio to the built-in volume ratio of 2:1 and so the pressure at the end of expansion most closely matches the downstream pressure maintained by the heat sink. This also gives the highest True % (ratio of actual work output to Carnot Conversion Ratio) although the work output from these conditions was the lowest of the six tests
- The cold water feed temperature was arranged to maintain a 5K temperature differential between condensing temperature and refrigerant pump inlet temperature, in order to prevent the pump from cavitating. Future configurations may enable this differential to be reduced, further improving the heat to power conversion ratio
- The internal pressure traces show a pressure drop at the inlet, with as much as 1.5 bar differential between the pressure in the pipe leading to the machine and the internal pressure at the start of expansion. This was found to be caused by several unnecessary pressure drops in the pipe which have subsequently been eliminated so the efficiency will be better than the figures presented here from the original configuration
- It is evident that the internal pressure drops by about 0.5 bar at about 55° shaft angle, then rises again. The inlet port comprises two separate openings in the housing and this increase in internal pressure corresponds to the point at which the second port opens. This pressure drop indicated that efficiency could be improved by some redesign of the inlet chamber
- The external pressure traces (the solid lines in Figure 2) show a high frequency pulsation in the inlet pipe starting just after the inlet port closes at 90° shaft angle. At this point the inlet pipe is isolated from the chamber so this pulsation is not related to the expansion inside the machine. The pressure pulsation starts at about 100° with a single large peak but is followed by a gradually decaying ripple which persists through to 210°. The same pressure ripple is observed 180° later when the second chamber is active. The machine speed is 500rpm so a single cycle accounts for 0.12s. There are 16 cycles from 100° to 191° - one cycle every 5.69°. Each ripple therefore accounts for 0.001896s, a frequency of 527.5 Hz. Since the frequency of the ripple is the same in each test it is not related to the gas velocity in the inlet pipe at the point that the inlet port closes. It may be a resonance effect related to the geometry of the pipe, in which case it will be affected by machine speed and can probably be designed out by careful configuration of the pipe assembly.

Further testing with this circuit is planned to check the effect of running at different speeds and then with different volume ratios. Changing the volume ratio requires the housing of the machine to be machined differently. This is described in more detail in section 7 of this paper.

3. COMPARISON WITH TRADITIONAL COMPRESSORS

Performance data points for a typical multicylinder reciprocating compressor, a scroll compressor and a typical twin screw compressor operating on R-134a were compiled from the manufacturer's selection software and the isentropic and volumetric efficiencies calculated. The efficiencies for the reciprocating compressor are shown in Figure 4.

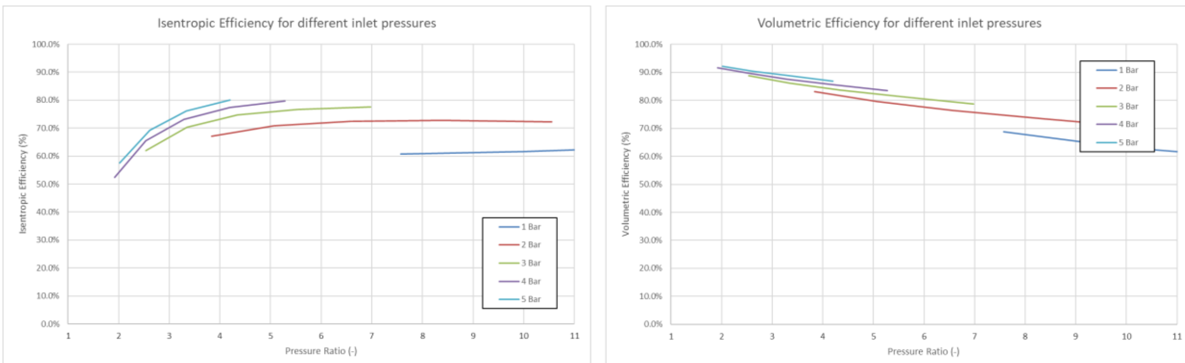

Figure 4: Efficiency of the Bitzer 6F.2Y at 1450rpm – nominal swept volume 151.6 m^3h^{-1}

The reciprocating compressor shows high volumetric efficiency at low pressure ratios, because the amount of gas re-expanding from the clearance volume is relatively small at these conditions. The volumetric efficiency reduces as the pressure ratio increases and re-expansion becomes more significant. The isentropic efficiency is low at low pressure ratios but is relatively flat at ratios above 5:1. However the isentropic efficiency is low for lower inlet pressures, about 60% across the range, because of higher losses in the inlet valves to the compression chamber. The pressure ratio that can be reached with higher suction pressures is limited by the discharge pressure limit which is set at 70°C saturated discharge pressure, giving a pressure ratio of 4.2:1 for an inlet pressure of 5 bar a. The maximum isentropic efficiency for the reciprocating compressor, of 80.1%, is achieved at this condition.

The screw compressor efficiencies are shown in Figure 5. This machine is an open drive compressor running on R-134a, which was selected for this comparison to avoid any issues of motor efficiency distorting the results. The compressor is the K version, suitable for higher back pressure, with a built in volume ratio (Vi) of about 2.6 (calculated from the pressure ratio at the peak efficiency point). The isentropic efficiency is at a maximum when the pressure ratio corresponds to the built in volume ratio so that the compressor is neither over- nor under-compressing the gas. As the required pressure ratio increases beyond the ideal, the efficiency declines due to the inrush of gas back into the compressor when the discharge port opens

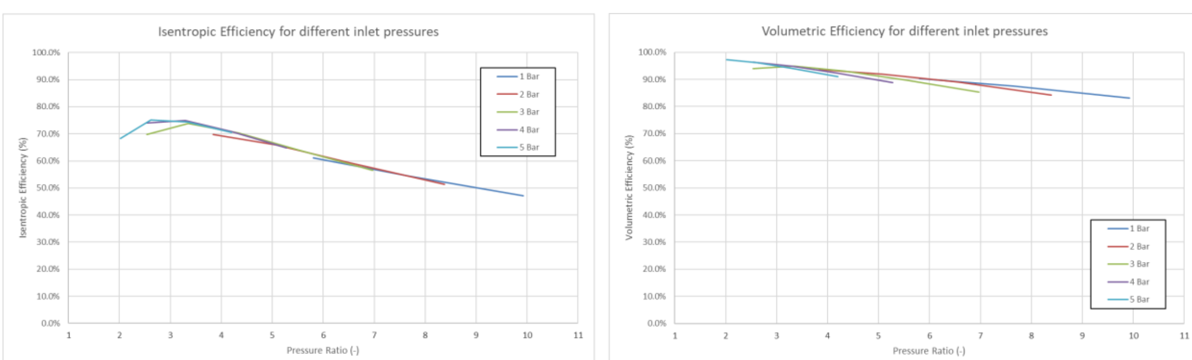

Figure 5: Efficiency of the Bitzer OSK8561 at 2900rpm – nominal swept volume 359 m^3h^{-1}

The volumetric efficiency is not as badly affected by high pressure ratios as the reciprocating compressor because there is no re-expansion from a clearance volume. Volumetric efficiency for the twin screw is more affected by pressure difference increasing the leakages from discharge to suction through various paths within the compressor including over the tip seals of the rotors and through the "blowhole"; the gap between the rotors. The scroll compressor isentropic efficiency profile was similar to the screw but about 5% lower at the peak and the operating envelope was more constrained, particularly at low suction pressure.

4. COMPRESSOR SIMULATION

Performance of the new machine configured as a compressor was modelled in GT Suite for six different built-in volume ratios ranging from 1.86 to 5.67. The six variants are named "A" machine to "F" machine, corresponding to pressure ratios of 2 to 7. Figure 6 shows the results for the "B" machine, with a volume ratio of 2.66 which corresponds most closely to the sample screw compressor shown in Figure 5.

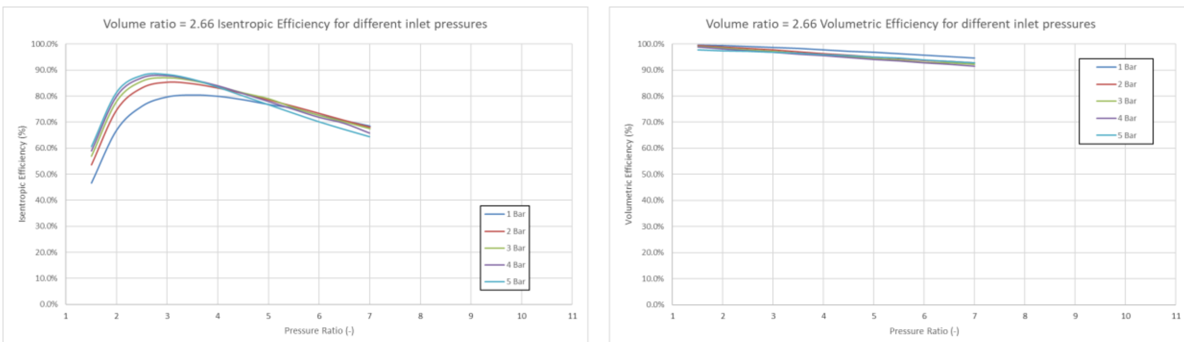

Figure 6: Efficiency of the FeTu compressor with Vi = 2.66 at 1000rpm – nominal swept volume 15.9 m^3h^{-1}

The isentropic efficiency is generally good across a range of inlet conditions, showing similar variation to the screw compressor, but with a peak efficiency of 88.3% at a pressure ratio of 3:1 compared to 75% for the screw. The isentropic efficiency was noticeably lower for the lowest inlet pressure of 1 bar a, possibly because of pressure loss effects in the inlet pipe, which are most evident with the lower density gas. The volumetric efficiency remains high across the full range of pressure ratios, dropping from about 99% at a pressure ratio of 1.5:1 to about 90% at a pressure ratio of 7:1. The volumetric efficiency is slightly lower for the higher suction pressure cases, indicating that it is influenced by pressure difference, like the screw compressor, whereas the reciprocating compressor showed the highest volumetric efficiency for the higher suction pressures.

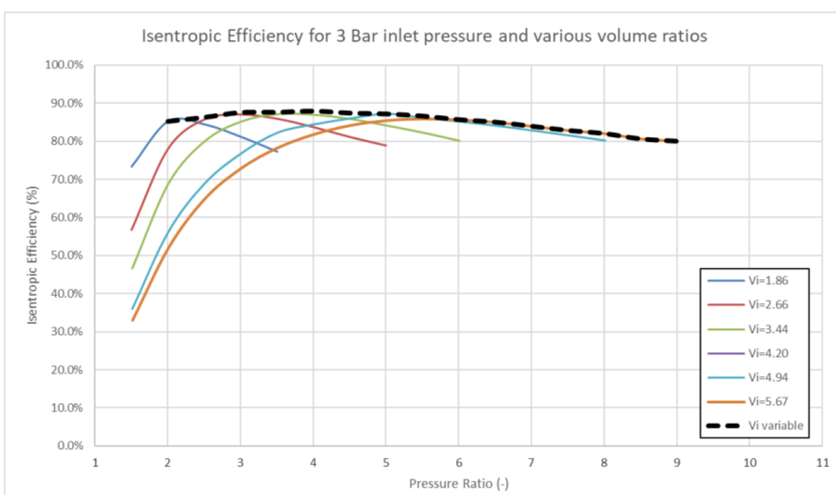

Figure 7: Isentropic Efficiency of the FeTu compressor with variable Vi at 1000rpm

Similar results were achieved for all six built-in volume ratios modelled and the results are aggregated in Figure 7 to show the result that would be achieved by developing a mechanism that would enable the port geometry to be adjusted in operation to optimize the performance. This figure shows the results for the six machines "A" to "F" with the dotted line indicating the variable Vi efficiency that would be expected.

It is emphasized that such a mechanism does not yet exist. This development is discussed further in section 7. If implemented, this would offer a compressor with an isentropic efficiency in the range 80% to 90% across a wide range of operating conditions.

5. DISCUSSION OF RESULTS

The machine shows some of the beneficial characteristics of a screw compressor. The compression space is continually swept in a forward rotary motion with induction and discharge of gas being enabled by the covering and uncovering of openings in the casing. This avoids the penalty of re-expansion that affects reciprocating compressors at high pressure ratios. Unlike a screw compressor the built-in volume ratio can be as low as 1:1 (as is the case with the turbine version of the expander). The casing can also accommodate higher volume ratios than the maximum of 5.66:1 that was modelled here, but in practice that would result in relatively small discharge ports, which might compromise the machine efficiency. The machine also shares some of the beneficial characteristics of a reciprocating compressor. The leakage paths past the rotating blade are across a plane surface, like the sides of a piston, not a line contact like the tip of a screw rotor, so leakage from discharge to suction can be kept very low. The low leakage rate is achieved by running parts in close proximity with minimal clearance rather than by using piston rings.

The system was modelled with an oil injection system where the oil is introduced to the compression space through small galleries in the blade, ensuring that the oil does not adversely affect the volumetric efficiency. The flowrate and temperature of the oil were fixed for all of the tests.

Figure 6 shows low efficiencies when the pressure ratio is less than the optimum for a given port configuration, because the higher built-in volume ratio causes the gas to be overcompressed and then to re-expand into the discharge chamber. Efficiencies are also low when the volume ratio is too low. In this case, as described earlier, the port opens before the internal pressure reaches the external pressure, causing a backflow of gas which then has to be recompressed to discharge it from the compressor. This is like a screw compressor, whereas the reed valves in a reciprocating compressor do not open until the internal pressure exceeds the internal, so there is no possibility of backflow. However the reed valves themselves impose a slight penalty requiring a little overpressure to overcome their resistance.

One of the observations from early test runs was that the internal temperature of the compressor was very steady and tended to sit midway between the suction gas temperature and the discharge temperature. The modelling software gives the opportunity to examine temperatures during the compression process and these are shown by the dotted lines in Figure 8.

It is noticeable that the in chamber and discharge temperatures are very similar when the pressure ratio is close to the optimum for the volume ratio, but are divergent at other conditions. For example with a volume ratio of 1.86 the temperatures are almost the same at the ideal pressure ratio of 2.0 but as the actual pressure ratio increases the discharge temperature rises more rapidly than the in-chamber temperature. However for higher volume ratios the crossover occurs at higher temperatures, but slightly below the optimum. For example with a volume ratio of 5.67 the ideal pressure ratio would be 7.0 but the temperature lines crossover at 6.0. This is probably because the system is modelled with the surroundings at room temperature so radiated heat losses from the compression process increase as the internal temperature rises. This theory is supported by the observation that the dotted lines get gradually closer together as the volume ratio increases, suggesting that the heat losses also increase. The close match of internal temperature to the ideal temperature rise suggests that there will be some merit in exploring the effect of oil cooling on the compression efficiency. It is not yet clear whether this unusual temperature behavior will be a benefit or a hindrance to more efficient operation.

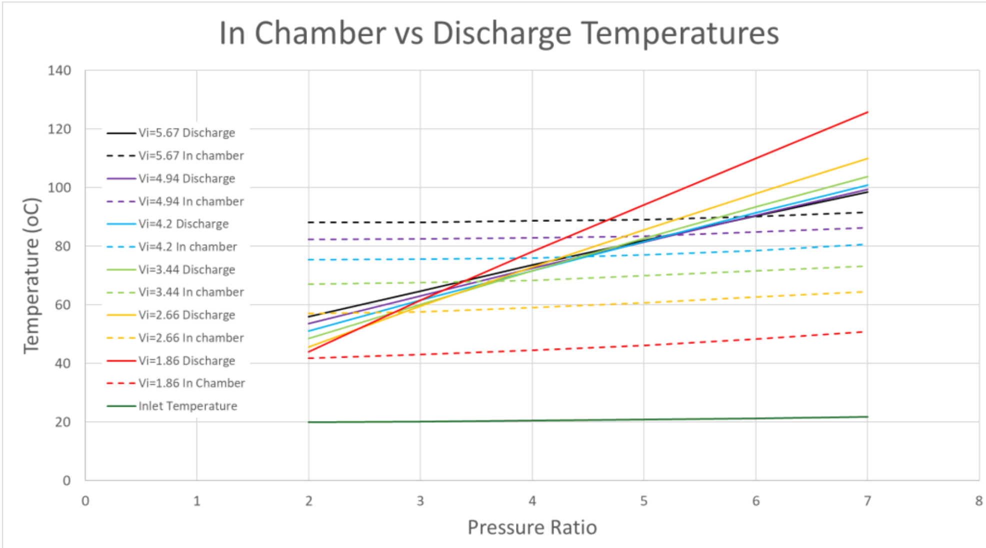

Figure 8: Comparison of in chamber and discharge temperatures, all based on 1000rpm and 3 bar a inlet

The compressor size that was modelled was significantly smaller than the comparison reciprocating and screw compressors. This was for two reasons. It was matched in size to the machines currently in use in the expander test cell so is relatively small diameter and hence small displacement. The reciprocating and screw compressors used in the comparison were chosen from the upper end of the size range to give as favourable figures for efficiency as possible. However, they were run at speeds used in typical commercial applications whereas the modelled compressor was run at 1000rpm, further reducing the displacement. In future it is expected that the speed of the machine can be increased. A brief study of the effect of speed on efficiency was conducted and is shown in Figure 9. This was only for the 3 bar a inlet pressure case, which would be suitable in a water chilling application.

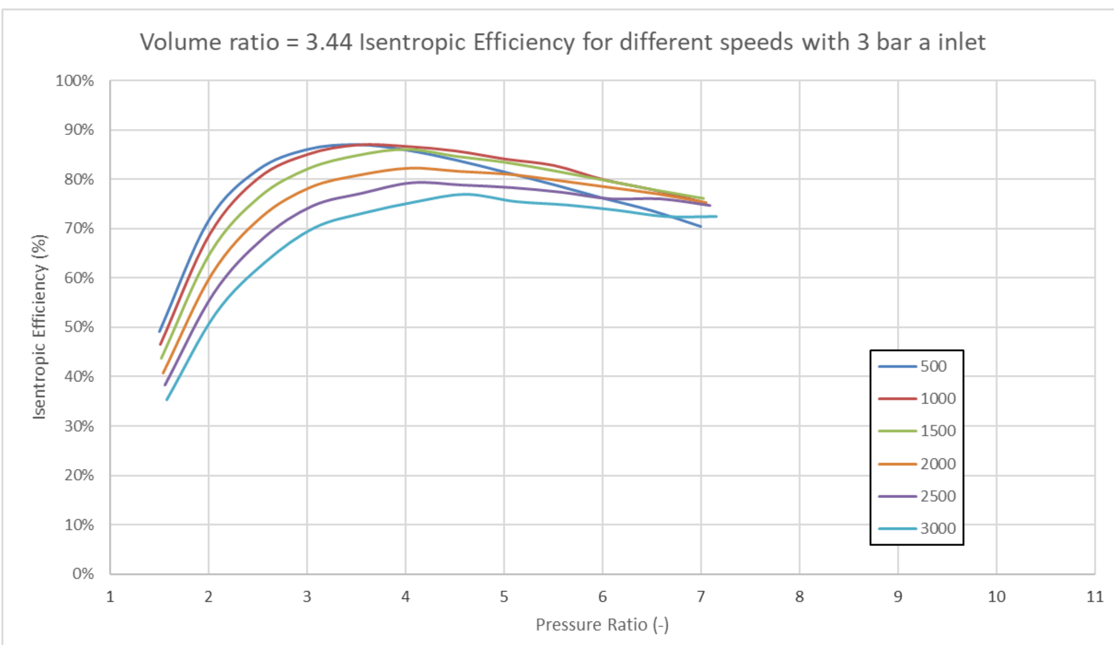

Figure 9: Isentropic efficiency for Vi = 3.44 (machine “C”) at different speeds and 3 bar a inlet

It is evident that the lowest speed, 500 rpm, causes the efficiency to drop disproportionately at high pressure ratios. The ripples in the characteristic for the highest speeds are likely to be resonance effects being picked up by the model that can probably be eliminated be careful design of the associated pipes. For example it can be seen that the efficiency for the 3000 rpm simulation dips slightly for pressure ratios from 4.5 to 6.5, whereas at 2500 rpm the dip is between pressure ratios of 4.0 and 6.0 This effect is not evident at lower speeds. The general trend of lower efficiency at higher speeds is expected since pressure losses through the inlet and outlet ports are higher and the bearing losses increase with the square of the speed.

6. FUTURE WORK

The expander testing focused on the creation of a test cell that gave meaningful results with a phase change fluid in an Organic Rankine Cycle system. This testing was carried out with R-134a but in the near future it is intended to repeat the tests with R-290 as the working fluid. Previous modelling of expander performance suggested that this should give a slight performance improvement. The objective of the tests was to determine whether the expander was particularly suitable for low input temperature heat to power, using the Rankine cycle test cell as the benchmark. Future work will include other working fluids and other single phase cycles to identify which gives the best conversion ratio. A significant amount of additional work is required to explore the best configuration of expander to use with saturated liquid inlet or even with two phase inlet fluid.

The compressor modelling was set up to explore some of the anomalous behavior related to internal temperatures that had been observed in early tests. It also allowed a greater understanding of the effect of the built in volume ratio on the compressor performance. The next step in this process will be to construct a test cell for one of the configurations, probably model "B" or "C", to validate the simulation and demonstrate performance in a typical chilling system, most likely a water chiller. Up to now, manufacture of prototype units has been subject to long lead times from third party machine shops. With a recent investment in a bespoke 5-axis CNC manufacturing facility for in-house manufacture of the rotors and housings it is intended to increase the availability of variants for testing, and then to move on to the design of a variable volume ratio mechanism.

The compressor simulation also showed that there are significant losses in the peripheral bearing when the machine runs above 1500rpm. Future work will include alternative drive and bearing arrangements to enable higher speed running without incurring such large losses. The initial modelling of different compressor speeds showed that higher speed reduced the peak efficiency, but also that it moved the peak towards higher pressure ratios. This is shown in Figure 9 and suggests that there are two effects that tend to cancel each other out, meaning that the optimal Vi for higher speed machines might be less than would be suggested by the ideal pressure ratio. This would be a problem for a screw compressor casing where the port dimensions are restricted. Further work is required to investigate the relationship between speed and volume ratio to give peak efficiency.

For the oil injected machine a clearance of 75 microns is required between the blade, the sphere and the housing. This gap is sealed with oil in the expander test cell and the compressor modelling. However there is scope for manufacturing to tighter tolerances once the in-house machining capability is established. This gives the opportunity to achieve higher efficiencies with oil free operation unlike the reciprocating compressor which requires oil to lubricate the crankshaft and connecting rod bearings or the screw compressor which requires oil for tip sealing and cooling, as well as for bearings. If testing proves that internal temperatures are not excessive then an oil free variant with tighter clearances could be manufactured to take advantage of the unusual behavior of the machine. Removal of the oil from the compressor will reduce the drag required to move the oil around the machine and raise it to discharge pressure. If this can be done without compromising the volumetric efficiency then there could be scope for a further improvement in the isentropic efficiency. On the other hand, if the oil can be used beneficially to regulate the compression temperature then there may be benefit in retaining it, but controlling the temperature and flowrate to achieve the desired effect.

Set up of the machining center will also enable rapid prototyping of more variants, particularly to exploit the behavior as an expander and to create a variable port mechanism for the compressor in order to enable the optimal efficiency indicated in Figure 7 to be achieved. Since the port is only cut into the housing, not the rotor, and the base case is with no port at all, giving a volume ratio of 1:1 there is greater scope for low pressure lift applications, for example with carbon dioxide in booster applications.

In the longer term once the machining center is established a design will be required for larger machines running at higher speed and maintaining their efficiency. This is not a trivial task and there are many issues to be addressed, but there is no fundamental reason why the design should not be scaled up to match the size of the comparison compressors used in this study.

7. CONCLUSIONS

The expander test cell gave valuable experience in the design, construction and operation of a facility and the lessons learned will be applied for compressors and companders as well as expanders and turbines.

The compressor modelling provided further insight into the operation of the machine as a dry gas compressor, using different port geometries to match operating parameters with performance characteristics. These results gave several insights of possible improvements to be made to the machine. Careful design of the gas inlet chamber could provide a saving in suction side pressure drop, enabling better performance to be achieved for both the expander and the compressor.

The comparison with other compressor types showed that the new design, operating as a compressor, shares some of the beneficial features of each type of conventional compressor while avoiding their shortcomings. This suggests that improvements in operating efficiency will be possible when the design is refined, particularly for higher speed operation.

The improvements in isentropic efficiency that are suggested by this work could have a significant effect on the efficiency of refrigeration and air conditioning systems. An air conditioner with a SEER rating of 13 may consume 3.6kW when it runs and could be in operation for 3,000 hours per year. Raising the compressor isentropic efficiency from 60% to 90% would reduce the consumption to 2.4kW, saving 3,600kWh per year. At an electricity cost of 2¢per kWh this is a saving of $72 per year. That does not sound like a lot, but there are over 100 million homes in the USA with air-conditioning and domestic air conditioning accounts for about 6% of all the electricity produced in the USA, costing homeowners about $29 billion per year (US DoE, 2024). This level of reduction applied to all the air-conditioners in the USA would save about 360TWh per year and could save $7 billion in operating cost as well as freeing up the equivalent of over 100 gas turbine power plants.

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