Design optimization and test of the novel FeTu ‘compander’, utilising organic fluids within a closed cycle for HVACR applications

J Subert1, J P Fenton1,2, K Hinchliffe1 and I M Arbon2

1 FeTu, The Wharf, Gas Works Lane, Elland. HX5 9HH, UK
2 Engineered Solutions, Merkland, Pinmore, Girvan, KA26 0TE, UK
3 Author to whom any correspondence should be addressed: info@fetu.co.uk

Abstract: HVACR performance is influenced by the efficiency of the compression and expansion process. The ability to operate oil-free; with non-toxic, inert, natural working fluids such as air or CO2 holds high environmental value.

This paper presents data of the FeTu™ compander operating in a closed cycle, for means of heating and cooling. A novel system, where compression and expansion is a continuous and simultaneous process; inherently regenerating energy to assist the compression process by expanding within a positive displacement chamber. This study documents the oil-free performance with organic working fluids pertaining to HVACR applications.

Commercial 1D modelling software; GTSuite (GTS) is used to optimise cooling capacity (Q) and Coefficient of Performance (CoP). Different volume ratios (VR), pre-charge pressures, temperature inputs (ΔT) and running speeds are analysed. A multi-faceted test rig is built to compare and validate all performance data against the GTS fluid model which, once calibrated, can then be used with a high degree of certainty in predicting the performance of other applications and ΔTs. Increasing the system pre-charge pressure has a proportional impact on capacity, so this phenomenon is tested to consider the conflicting impacts of higher pressure against increased viscosity, with respect to volumetric efficiency.

The CoP of the system ranges from 1 to 5 across typical HVACR ΔTs. Low VR offers highest CoP but lowest Q, so the optimal trade-off is explored. The strengths & weaknesses are then comparatively documented against incumbent vapour cycle systems.

1. Introduction

1.1. Literature Review
The use of fluorinated refrigerants (F-gases) specifically Chlorofluorocarbons (CFC’s), hydrofluorocarbons (HFC’s) and hydrochlorofluorocarbons (HCFC’s) have a known and significant detrimental effect on the environment. Under the 1987 Montreal Protocol, both CFC’s and HCFC’s have been phased out in developed countries - January 1996 and 2020 respectively [1].

HFC’s have been widely used as a replacement refrigerant for HCFC’s, as they do not contain chlorine atoms, it has no effect on the O-Zone layer. However, whilst HFC’s have zero O-Zone Depletion Potential (ODP), their Global Warming Potential (GWP) is very high. Furthermore, in comparison to chlorine containing refrigerants, HFC’s last longer in the atmosphere thus their usage is regulated via the Kyoto Protocol [2]. R134A, a widely used HFC, has a GWP 1430 times that of CO2.
[3]. Chen et al [4] states the ‘leakage of R134A in automotive air conditioners can reach up to 6%-9% of the total charge’. Refrigerant leakage associated with use in supermarkets is estimated between 9% and 25% although these rates have been reducing [5]. HFC’s directly contribute to global warming, their phase out coupled with The Paris Agreement and the United Kingdom’s own legislation to achieve net zero by 2050 are driving the demand for natural refrigerant alternatives.

Emani and Mandal [6] state the basic properties necessary to be considered an effective refrigerant. Characteristics include being harmless, non-flammable, non-explosive, non-corrosive and non-toxic. The refrigerant should demonstrate a high Coefficient of Performance (COP) within the working temperature range. The pressures used should enable practical sizing of both the pipes and the compressor. Critically, any ideal working fluid should be zero GWP and very low ODP.

Air is one such fluid, free and abundant, it is non-toxic, non-flammable, zero ODP and zero GWP. Yang and Yang [7] describe the widespread use of air cycle machines as part of an aircraft’s environmental control system, citing its high reliability and low maintenance workload. Shengjun, Zhenying and Lili [8] agree that their ease of maintenance, high reliability and no leakage concerns are distinct advantages. However, low energy efficiency has prevented wide uptake in the refrigeration and air conditioning sector up to date.

The FeTu open cycle ‘compander’ discussed by Fenton et al [9] sees high COP, overcoming the performance limitations historically associated with conventional air cycle HVACR systems. However, it is predicted that closed cycle holds a distinct advantage, giving the ability to pre-charge the system with pressure, scaling the capacity for the same geometric sized machine.

This paper investigates the closed cycle performance using air, for typical HVACR applications, testing different sized machines, Built In Volume Ratios (BIVR), external temperature differentials (ΔT) and pre-charge pressures. The results are simulated using a mathematical approach and a GT Suite model which validate each other. A closed cycle, proof of concept test is also completed in order to verify the real-world capability of the machine.

1.2. FeTu Concept

The FeTu™ Rotoriculating™ unit is a positive displacement ‘compander’ in which compression and expansion occur simultaneously with two pairs of opposed double-acting chambers. Driveshaft rotation operates a spherical rotor which defines the four chambers and allows continuous flow of fluid via the suction and discharge ports. Each of the four chambers reduce in volume by 250:1 [9]. The device is designed with micron level internal clearance, and although lubricant free, precision manufacturing allows high volumetric efficiency, recorded at up to 97% in compressor tests of Zhang [10].

By designing a BIVR between Chamber A and B of Figure 1 and 2, rotation of the driveshaft allows the ‘compander’ principle to operate by creating a positive or negative pressure ratio in the flow path which joins the chambers, creating a change in temperature. A Heat eXchanger (HX) is utilised in the flow path for heat addition or extraction, forming a simple HVACR system with only two moving parts. The ‘compander’ system can run open or closed cycle, is scalable, modular and at variable speed to invoke analogue heat transfer rates (Q), dependant on demand. [9].

![Figure 1: Internal Operation [9].](image1)

![Figure 2: Showing Volume Ratio of A – B.](image2)
2. Test Methodology

2.1. Proof of Concept Test
The first ‘comander’ prototype that was manufactured was an 84mm, BIVR:1.464 unit. To verify that the closed cycle HVACR system is operational in a real-world application, a small, in house test rig was designed with two HX, four Omega pressure transducers (0-10bara, ±0.05%) and four type K thermocouples (0 to 200°C, ±0.4%), in Figure 3 and 4. Data was recorded using an Omega data capture unit.

![Figure 3: Closed cycle test rig.](image)

![Figure 4: Closed cycle schematic.](image)

2.2. Mathematical Approach
The process followed in this analysis is broken into four discrete instances in time within a 360deg shaft rotation, schematised by Figure 5 below. There are two compressions (HX2) and two expansions (HX1) each revolution, as well as two instances of four locked volumes (top dead centre -TDC). The Volume, Pressure, Temperature and Mass are calculated at each moment based on Equations 1-37. \( Q \), \( W \) and COP are then calculated. The knowns and unknowns of the method are summarised in Table 1:

![Figure 5: Mathematical Approach - Cycle Analysis.](image)
Table 1: Knowns and Unknowns pre-process.

| Inputs (Knowns)                          | Outputs (Unknowns)                  |
|-----------------------------------------|------------------------------------|
| Gas Properties: $R$ and $C_v$           | Polytopic index of compression and expansion: $n$ |
| Motor Speed $\omega$ (r/min) & Frequency: $F = \frac{2 x \omega}{60}$ (s$^{-1}$) | System Pressures (Pa) and Pressure Ratio: $PR$ |
| Volumes of chambers and HXs: $V_L$, $V_s$, $V_{HX}$ (m$^3$) | System Temperatures and Temperature Ratio: $TR$ |
| Pre-Pressure in the system: $P_{pre}$ (Pa) | Stabilised Mass: $m_T$ (kg) and Mass flow $\dot{m}$ (kg/s) |
| Temperature at Pre-charge: $T_{pre}$ (K) | Work in/out: $W$ (J) |
| External Temperatures: $T_{HX, External}$ (K) | Cooling/Heating Power in/out: $Q$ (J) |
| Initial Total Mass after Pre-charge: $m_{T1}$ (kg) | Coefficient of Performance: $COP$ |

2.2.1. Mass flow in HX1 and HX2 are equal at stabilisation.

\[
\dot{m}_{HX1} = \dot{m}_{HX2}
\]
\[
\dot{m}_L = \dot{m}_s
\]
\[
PV = mRT \quad \therefore m = \frac{PV}{RT} \text{ (gas law)}
\]
\[
\frac{p_LV_L}{RT_L} = \frac{p_SV_S}{RT_S}
\]
\[
\frac{p_S}{p_L} = \frac{T_S}{T_L} \times \frac{V_L}{V_S}
\]
\[
P_L = P_4, T_L = T_3, P_2 = P_2, T_x = T_2
\]
\[
\frac{P_2}{P_4} = \frac{T_2}{T_4} \times \frac{V_2}{V_4}
\]
\[
P_2 = P_4 \times \frac{T_{HX2, External}}{T_{HX1, External}} \times BIVR = P_4 \times TR \times BIVR
\]

Assume that the Heat Exchanger transfers all heat available to the system.

\[
T_{HX1, External} = T_4
\]
\[
T_{HX2, External} = T_2
\]

2.2.2. Conservation of Mass - the total mass at rest is equal to the total mass at stabilization. Total mass before operation:

\[
m_{T1} = \frac{\dot{m}_{PC} \times V_{Total}}{R \times T_{PC}}
\]

Total mass after stabilisation:

\[
m_{T2} = m_{HX1} + m_{HX2} + m_L + m_S
\]
\[
m_{T2} = \frac{p_LV_{HX1}}{RT_4} + \frac{p_SV_{HX2}}{RT_2} + \frac{p_LV_L}{RT_2} + \frac{p_SV_S}{RT_2}
\]
\[
m_{T2} = m_{T1}
\]
\[
R \times m_{T1} = P_2 \left( \frac{V_{HX2} + V_S}{T_2} \right) + P_4 \left( \frac{V_{HX1} + V_L}{T_4} \right)
\]

Sub in Equation 13 into 18:

\[
P_2 = P_4 \times TR \times BIVR
\]
\[
R \times m_T = P_2 \left( \frac{V_{HX2} + V_S}{T_2} \right) + P_4 \left( \frac{V_{HX1} + V_L}{T_4} \right)
\]
\[
P_4 = \frac{R \times m_T}{T_{HX, BIVR} \times \left( \frac{V_{HX2} + V_S}{T_2} \right) + \left( \frac{V_{HX1} + V_L}{T_4} \right)}
\]

Finally, calculate $P_I$ with Equation 13.
2.2.3. **Internal Energy Balance.** There is a mixing of two locked volumes at stage 1-2 and 3-4. This occurs at a constant internal energy (U) since it is instantaneous with zero compression or expansion.

\[
U_3 = U_{HX1} + U_f
\]  
\[
U = m \times C_v \times T
\]

\[
m_3 \times C_v \times T_3 = (m_{HX1} \times C_v \times T_4) + (m_4 \times C_v \times T_2)
\]

\[
T_3 = \frac{(m_{HX1} \times T_4) + (m_4 \times T_2)}{(m_{HX1} + m_4)}
\]

\[
\therefore T_1 = \frac{(m_{HX2} \times T_2) + (m_L \times T)}{(m_{HX2} + m_L)}
\]

### Table 2: Summary of Equations.

|   | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| V | V_{HX2} + V_L | V_{HX2} + V_f | V_{HX1} + V_f | V_{HX1} + V_L |
| P | \frac{m_1 R_1}{V_1} | \frac{P_4 \times TR \times BIVR}{V_1} | \frac{m_3 R_3}{V_3} | \left(\frac{TR \times BIVR \times \left(V_{HX1} + V_f\right)}{V_2}\right) \times \left(\frac{V_{HX1} + V_L}{V_4}\right) |
| T | \frac{m_{HX2} T_2 + m_L T_2}{m_4} | \frac{m_{HX1} T_1 + m_2 T_2}{m_3} | \frac{T_{HX1 \text{ External}}}{m_2} | \frac{p_4 V_4}{R_2 V_4} |

2.2.4. **Polytropic Index Calculation.** To find \( Q \) capacity, the polytropic index each side is calculated. This will be influenced by the amount of heat transfer during the compression and expansion.

\[
\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^{n_3-4}
\]

\[
\ln \frac{p_3}{p_4} = n_{3-4} \ln \frac{V_4}{V_3}
\]

\[
n_{3-4} = \left(\frac{m_3 p_3}{m_2 p_2}\right) \left(\frac{V_4}{V_3}\right)
\]

\[
\therefore n_{1-2} = \left(\frac{m_1 p_1}{m_2 p_2}\right) \left(\frac{V_1}{V_2}\right)
\]

2.2.5. **Work.** Note: for rate of Work (W) multiply by \( F \) (s⁻¹)

\[
\Delta W_{3-4} = \int_{V_3}^{V_4} p \, dv = \int_{V_3}^{V_4} \frac{p_4 V_3^2}{V_4^{n_3-4}} \, dv = \frac{P_4 V_4}{V_1} \left[ V_1^{1-n_{3-4}} \right]^{V_4}_{V_3}
\]

\[
\Delta W_{3-4} = \left(\frac{P_4 V_4}{V_1} \right) \left(\frac{V_4^{1-n_{3-4}}}{V_3^{1-n_{3-4}}} \right)
\]

\[
\therefore W_{1-2} = \left(\frac{P_4 V_4}{V_1} \right) \left(\frac{V_2^{1-n_{1-2}}}{V_1^{1-n_{1-2}}} \right)
\]

\[
W_{\text{Total}} = W_{3-4} + W_{1-2}
\]

2.2.6. **Cooling/Heating Capacity.** Note: for rate of Q (W) multiply by \( F \) (s⁻¹)

**1st law:** \( \Delta Q_{3-4} = \Delta U_{3-4} + \Delta W_{3-4} = m_{HX1} C_v (T_4 - T_3) + \frac{P_3 V_3 - P_4 V_4}{m_{HX1} V_{HX1}} \left(\frac{T_4 - T_3}{n_{3-4}}\right)
\]

\[
Q_{3-4} = m_{HX1} \times \left( C_v + \frac{R}{1-n_{3-4}} \right) \times (T_4 - T_3)
\]

\[
\therefore Q_{1-2} = m_{HX1} \times \left( C_v + \frac{R}{1-n_{1-2}} \right) \times (T_2 - T_1)
\]

2.2.7. **COP.**

\[
COP_{HX1} = \frac{Q_{3-4}}{W_{Total}}
\]

\[
COP_{HX2} = \frac{Q_{1-2}}{W_{Total}}
\]
2.3. GTSuite Model
FeTu have currently manufactured three unit sizes: 84mm, 120mm, 215mm. A GTSuite model has been designed to replicate each sizes performance, utilised within a closed ‘compander’ cycle.

The model has four chambers, bound by a common shaft, which oscillate with set volume curves seen on Figure 6 (215mm), each 180°. All chambers expand and fully evacuate fluid twice every 360° rotation. The top and bottom chamber pairs operate 180° antiphase. The bottom two chambers are lesser in volume to the top two by a factor of BIVR. The volume flow dictated by the curves are emitted though suction and discharge ports of given opening areas, again plotted though 180° as seen in Figure 6.

The HXs have a given volume and set so they can accept/reject max Q to a given input temperature, replicating idealised performance and validating the method in Section 2.2. Other inputs include: speed (r/min), leakage, dead zones volumes and internal heat migration - set to zero for idealised comparison.

3. Results

3.1. Proof of Concept Test Results
Figures 7 and 8 show real test data, with proof of concept, using the 1.46 BIVR, 84mm test rig (Figure 10). When run at 800r/min, HX1 becomes low and HX2 becomes high pressure and temperature.

3.2. Validation of Mathematical and GT Model
The results of the GTSuite simulations of Section 2.3 and the mathematical model of Section 2.2 agree entirely as seen in Figures 9 – 12 which plot stabilised $P$, $m$, $Q$ and $COP$ over different BIVRs (1-3).

- Rotor D: 120mm
- Speed: 3000r/min
- $T$: HX1 & HX2 = 20°C
- $V$: HX1 = 4L, HX2 = 4L
- Pre-charge: 10bara
3.3. Pre-charge Pressure Investigation

An investigation was completed to observe the effect of pre-charging the system with different pressures. $PR$, $Q$, $W$ and $COP$ were simulated across a range of BIVRs in Figures 13-16 respectively:

- Rotor D: 120mm
- Speed: 3000 r/min
- $T$: HX1 & HX2 = 20°C
- $V$: HX1 = 4L, HX2 = 4L
- Pre-charge: 1, 5, 10 bar

![Figure 9: Pressure vs BIVR – Comparison.](image1)

![Figure 10: Mass Flow vs BIVR – Comparison.](image2)

![Figure 11: $Q$ and $W$ vs BIVR – Comparison.](image3)

![Figure 12: $COP$ vs BIVR – Comparison.](image4)

![Figure 13: Pressure vs BIVR – Different Pre-charge.](image5)

![Figure 14: Temp vs BIVR – Different Pre-charge.](image6)
3.4. Δ Temperature Investigation

An investigation was completed to observe the effect of different ΔT externally in Figures 17 and 18.

Note: ΔT = T_L External – T_R External. Test Conditions:

- T: HX1 = 15, 20, 25°C
- T: HX2 = 25, 20, 15°C
- Rotor D: 120mm
- Speed: 3000r/min
- Precharge: 1, 5, 10 bar
- V: HX1 = 4L, HX2 = 4L

![Figure 15: Work vs BIVR – Different Pre-charge.](image)

![Figure 16: COP vs BIVR – Different Pre-charge.](image)

3.5. Rotor Size Investigation

The three currently manufactured units (84, 120, 215mm) of FeTu have been simulated across the BIVR range of 1-3 in Figures 19 and 20, plotting mass flow and Q. Case:

- Rotor D: 84, 120, 215mm
- Speed: 3000r/min
- Precharge: 10 bar
- V: HX1 & HX2 = 20°C
- V: HX1 & HX2 = 4L, 4L, 10L

![Figure 17: Q vs BIVR – Different ΔT.](image)

![Figure 18: COP vs BIVR – Different ΔT.](image)

![Figure 19: Mass Flow vs BIVR – Different Rotor D.](image)

![Figure 20: Q vs BIVR – Different Rotor D.](image)

The temperature pre- and post-exchanger is a good indication of the min/max temperature attainable in the external cooling and heating. Figures 22 shows the temperature at probe points: A, B, C, D for each rotor size for the case above based on Figure 21 test arrangement.
4. Discussion

4.1. Validation (Section 3.1, 3.2)
The FeTu closed-cycle ‘componder’ is a new derivation of the TRL6 proven FeTu oil-free compressor. Although basic, the proof-of-concept test in Section 2.1 successfully proves that the theorised cycle behaves as expected; from start-up HX1 drops in pressure and temperature, and in contrast HX2 rises.

Figures 9-12 clearly show the agreeing results of both calculation methods - Section 2.2 and 2.3, giving complete validation. Although GTSuite provides a more extensive output than the mathemetic approach, the more complex input settings and long simulation time means that it is less time efficient.

Running a BIVR of 1.0 creates a fluid pump system with zero pressure change, this is exhibited in Figure 9 where the HX1 and HX2 maintain the pre-charge pressure state (10bara). As BIVR trends away from 1.0 this promotes the ‘componder’ principle, lowering HX1 and raising HX2 in pressure. Greater BIVR leads to a larger PR. When there is temperature addition of ambient 20°C (as the case denotes), the rise and drop in pressure is of equal magnitude about the initial pre-charge pressure condition.

Increasing the BIVR means keeping $V_i$ the same but reducing $V_i$ to a smaller volume, because of this, the total stabilised mass flow of the system reduces, as seen in Figure 10. It is important to know that in the GTSuite model at stabilised conditions, the mass flow through HX1 and HX2 are equal.

Figure 11 shows that the $Q$ for both heating and cooling tends to 0 at BIVR=1, this is because there is zero compression and expansion. From BIVR=1-1.5, there is a sharp increase in absorbed heat ($Q_{HX1}$) and rejected heat ($Q_{HX2}$). $Q_{HX1}$ then starts to plateau after 1.5 meaning that for a cooling application, there is no longer an advantage of increasing the Volume/Pressure Ratio past BIVR=1.5. $Q_{HX2}$ however does not plateau, therefore there appears to be no peak BIVR for heating. Using the 1st law of thermodynamics in a lossless system; $Q_{HX2} - Q_{HX1} = W$, dividing through by $W$ and using Equation 36 and 37 give: $\text{COP}_{HX2} - \text{COP}_{HX1} = 1$. This law is upheld in both modelling techniques as seen on Figure 12 that shows Heating COP$_{HX2}$ exactly 1 higher than Cooling COP$_{HX1}$. Figure 11 shows that the $W$ tends to 0kW at low BIVR which leads to inordinately high COP (Figure 12). Higher BIVR sees the lowest COP values.

4.2. Investigations (Section 3.3-3.5)
The investigation of Section 3.3 shows that the pre-charge pressure increase proportionally scales the internal density, mass-flow, and heat capacity (Figure 14). The cost of this being extra input work as seen in Figure 15. Figure 16 shows that the thermodynamic COP is unaffected by internal pre-charge pressures.

The $\Delta T$ in HVACR systems is critical to the system performance. A large $\Delta T$ in favour of the operational HX typically results in higher levels of performance. This is seen in Figure 17 which shows larger cooling and heating potential at elevated $\Delta T$. Changing external temperature subtlety, as investigated, has minimal impact on TR and as such PR (Equation 8). Because of this, $W$ doesn’t change as much as $Q$ and therefore larger $\Delta T$ results in a greater COP (Equation 36 and 37) - seen in Figure 18.
The rotor Diameter increases the volumetric capacity of the machine, due to the spherical design: by a cubic factor. Larger machines running at the same speed will therefore exhibit a greater internal flow rate (Figure 19), resulting in an increase in $Q$ (Figure 20). The $W$ to compress and expand the fluid increases proportionally keeping $COP$ constant with rotor size. Figure 22 shows that the larger rotor sizes produce a larger cooling/heating potential in terms of temperature change.

5. Summary
The paper explores two comparative modelling approaches, which agree, furthermore the application is verified by physical test. All objectives of the study were met. The success of this foundation work shows encouraging potential to function well within a HVACR scheme. Whilst open-cycle systems using air as the working fluid are commonplace to aviation; the ability to pre-charge the FeTu closed-cycle system, significantly increases the cooling and heating capacity for a given size. Due to its unique operation, high volumetric efficiency and small number of flow stages, the ‘compander’ seems to overcome $COP$ limitations of air cycle HVACR, able to achieve a $COP$ in excess of 5.

A complementary, mechanical GTSuit Model, which considers physical losses such as heat migration, port geometry, leakage and bearing friction is in development. Whilst not yet validated, simulations show interesting trends: As pre-charge pressure increases, the bearing loads remain constant but $Q$ increases vastly, meaning that mechanical losses become a smaller factor, elevating the $COP$. However, larger pressures become port sensitive which limits $COP$, meaning each rotor diameter has an optimal pre-charge value for a given set of operational conditions. Furthermore, the extra $W$ due to real losses, removes the unrealistically high $COP$ at low $BIVR$ (Figure 20), typically peaking around $BIVR$:1.2-1.6 and diminishing toward $BIVR$:1.0, depending on conditions.

Although successful, the data supporting Section 2.1 is scarce, so a large amount of performance data for the closed-cycle operation will be collected throughout 2021. Using this to calibrate the combined GTSuit thermo/mechanical model will allow FeTu to predict and optimise a wide range of applications.

Other natural working fluids will be tested, Supercritical CO$_2$ is of specific interest (Heat-to-Power).

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