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UPDATE OF THE SCO_2-TEST RIG AT CRANFIELD UNIVERSITY

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ABSTRACT

Since 2018, there is an experimental supercritical carbon dioxide (sCO_2) facility operating at Cranfield University. The purpose of this rig is to enable the exploration of supercritical carbon dioxide as a working fluid for future bottoming power cycle applications and, more recently, for thermal management applications. The core of the rig is a transcritical closed loop, which has recently been upgraded. The upgrades include an increase in the number of measurement stations, changes to the types of measurements taken, as well as the addition of a new, dedicated data acquisition system. A summary of some of the lessons learned from different test campaigns conducted from 2018 to 2021 is provided, along with a discussion on the measurement upgrades performed. The experience obtained with this rig, as recounted in this paper, could be relevant to similar test rigs or future power cycles applications.

Keywords: supercritical carbon dioxide, test rig, temperature measurement, thermal management, uncertainties, heat exchanger, waste heat recovery, transcritical CO_2 cycle regulation

NOMENCLATURE

COP	Coefficient of performance
DAQ	Data acquisition
EVD	Expansion Valve Driver
$h, \delta h$	Enthalpy, enthalpy error
HVAC	Heating, ventilating and air conditioning
HPV	High-pressure valve
MHEX	Main heat exchanger
PCHE	Printed circuit heat exchangers
MPaG	Mega Pascal gauge
P&ID	Piping and instrumentation diagram
PI	Proportional and Integrative control
Popt	Optimal high pressure
Q_{evap}	Heat exchanged at the evaporator
RTD	Resistance temperature detectors
T_{GC}	Outlet temperature of the gas cooler
VSD	Variable speed driver
W_{comp}	Work of the compressor

1. INTRODUCTION

The supercritical CO_2 rig at Cranfield University was developed during the years 2015 to 2018, as part of Innovate UK project 101982, titled “Supercritical CO_2 Waste Heat Recovery for Marine Applications” [1]. This project was joint research undertaken between Rolls-Royce PLC, Heatric (Division of Meggitt PLC), and Cranfield University.

For the design of the rig, a modular approach was adopted (the so-called “pencil parabolic” pattern [2]). The main requirement driving this approach was to isolate major equipment components (i.e., heat exchangers and compressors) or control loops (i.e., flow split and bypass valves) for testing.

Figure 1 shows the modular and staged development process followed. At present, the rig comprises three stages or ‘loops’, referred to as ‘Gas triangle’, ‘Stage 1A’, and ‘Stage 1B’. The stages are interconnected through a manifold (not shown). This allows redirecting the CO_2 stream, at supercritical state, to the stage under testing, while bypassing the remaining stages. The basic principle for testing is to maintain a transcritical circulation loop, using different expansion devices to adjust the intermediate/low-pressure level and mass flow in each branch. The circulation loop is a modification of a transcritical CO_2 refrigeration system (shown in Figure 2, often referred to in this paper as refrigeration pack), which can be adjusted to provide a reliable source of carbon dioxide in a supercritical state.

The circulation loop comprises the following main elements:

- Two semi-hermetic compressors of 45 kW, one of which can be regulated in frequency via an inverter (segment 7-1 in Figure 2)
- One gas cooler of 200 kW (segment 1-2)
- One brazed plate heat exchanger of 95 kW (segment 5-6)

- One liquid receiver of 60 L (between 3-4-5)
- Electronic expansion valves for
    - Gas expansion valve (segment 2-3)
    - Liquid expansion valve (segment 3-5)
    - Flash gas valve (segment 3-4)
- Flow, pressure and temperature regulation are achieved using an industrial controller which regulates operating modes of compressors and fans (on/off, speed) and valve positioning.

At the evaporator, liquid refrigerant is vaporized by means of a water circulation loop coming from a cooling tower. The flowrate is, on average 5 L/s, whereas its temperature depends on the ambient temperature and whether the cooling tower is used for other purposes in the test area facility. The dependency on the weather has motivated further rig modifications that will be discussed in Section 4.

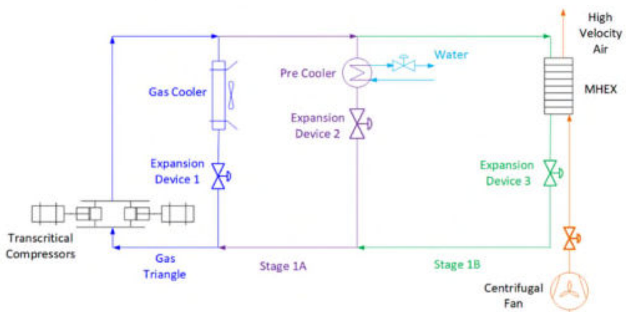

Figure 1: SCHEMATIC OF THE CRANFIELD UNIVERSITY sCO_2 RIG

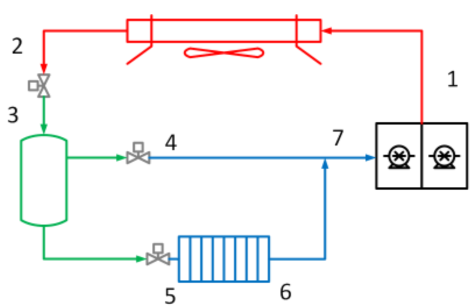

Figure 2: TRANSCRITICAL CO_2 PACK - CIRCULATION LOOP

As indicated in [1], the initial objectives of this experimental program were to de-risk and demonstrate the robustness of a closed-loop system, as well as to prove the function and performance of individual components and various measurement and control modules. The modular approach followed for the rig's design and construction allowed pursuing the test campaigns previously described as: Gas triangle (operation of the transcritical circulation loop in a pure gaseous state), Stage 1A (demonstration of PI control for operating PCHEs), and Stage 1B (de-risk testing of novel main heat exchanger) [2].

On the other hand, the manufacturing, commissioning, and testing of a bespoke supercritical centrifugal compression system (Stage 2A), has been postponed until further notice. Instead, the rig has subsequently been used in a few projects with Rolls-Royce PLC. It is currently being used in Innovate UK project 113263, titled 'Powerplant Integration of Novel Engine Systems (PINES)', mainly for the verification and validation of numerical models for the use of carbon dioxide in closed loops.

Due to the nature of non-disclosure agreements with our industrial partners, this paper does not discuss any results of the test campaigns performed. It instead elaborates on the lessons learned to overcome testing challenges of the test campaigns and discusses the improvement of the measurement techniques.

The first part of the paper will expound on the experiences gained from performing different test campaigns in the rig and the influence of its service facilities. It will specifically also discuss elements of post-commissioning and operability.

The second part will describe upgrades to the rig, focusing on those related to the measurement of temperature and data acquisition. Discussion and findings about the time response and the accuracy of different types of sensors will be presented to propose a path for selecting sensors suitable for highly transient power cycles. Finally, the way ahead and the planned test campaigns for the key components will be described.

2. WASTE HEAT RECOVERY TEST CAMPAIGNS

The objective of this section is to share with the readers some experience gained by Cranfield University during the execution of different waste heat recovery test campaigns in the rig. This could be relevant to other test rigs or future power cycles applications.

2.1 Stage 1A

In Cranfield's experience, measurements techniques for CO_2 systems have three associated, known challenges: high-pressure (> 1 MPa, even on the low-pressure side of the cycles), two-phase flow, and fast time response. Depending on their function in the cycle, components operating in trans- or supercritical cycles also face high temperatures (> 550 °C), variable density, and significant changes in fluid momentum.

Developing a control system around a suitable and affordable set of instruments is challenging, given the natural compromises of matching measurements requirements, such as range, levels of accuracy, invasive/non-invasive tradeoffs, and response time, together with a wide set of operational conditions. This is already known in the refrigeration and HVAC industry because their plants operate in non-stationary conditions. Such conditions result from low and high frequency (i.e., seasonal, daily) environmental changes, combined with two additional challenges: variable customer demands and non-linear systems [3]. It is expected that extending the use of R744 control systems for waste heat recovery applications will add extra layers of complexity. This was the case with our Stage 1A test campaign.

Stage 1A was already described in [2]. The pre-cooler assembly for this campaign can be seen in Figure 3. Although this assembly was designed to be coupled with a future full-cycle demonstrator (including centrifugal compressor and turbine), the first task of the arrangement was to characterize the pre-cooler and expansion valve assembly (CV01) and demonstrate the control capability of the real-time target computer.

Tests were divided into two categories: steady-state and transient. For each category test matrices were proposed as two-factor experiments. As an independent variable, for the expansion valve CV01 and the water control valve WVC1, the percentage of the valve's opening was selected. The test matrices recorded measurements of pressure and temperature at each station at combinations of opening positions of between 40% and 100% for valve CV01 and WVC1. For transient testing, the latency response of the rig at different valve timings needed to be characterized, ensuring that knowledge was gained on how the rig responds to different transient conditions. A description of the sensors installed is included in Table 1.

Using Simulink-Real Time®, it was possible to develop open- and closed-loop controls to set desirable values of pressure at PT70, by adjusting the output of CV01 (similarly to the temperature at TT61 and the opening of the water valve in another control loop). The gains for each PI controller were tuned at each step to explore the trade-off between oscillation and tracking accuracy. Reasonable stability was obtained for both closed-loop controllers (CO_2 gas and water). It was possible to establish two compatible control loops – one for pressure and another for temperature, which couples the performance of the heat exchanger (D8906) and the expansion device (CV01). Temporal separation was used in the controllers (one controller running faster than the other). Unsurprisingly, these achievements came after overcoming several challenges.

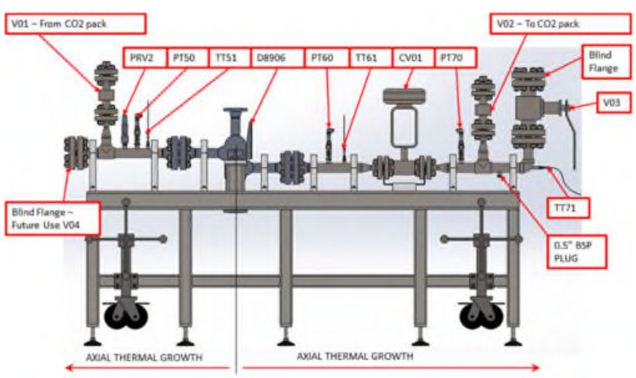

Figure 3 STAGE 1A CO_2 ASSEMBLY AND ITS INSTRUMENTATION LABELLING

Table 1 MEASUREMENT DEVICES INCORPORATED IN STAGE 1A AT CRANFIELD UNIVERSITY

Parameter	Device	Range	Accuracy
	CO_2 Stream		
Pressure	Piezo resistive	0 – 20MPa	±0.25%
Temperature	RTD Pt100	-20 – 250°C	±0.2°C
	H_2O Stream		
Pressure	Piezo resistive	0 – 1MPa	±0.25%
Temperature	RTD Pt100	0 – 100°C	±0.15°C
Volumetric flow	Electromagnetic	0 – 10lps	±0.5%

2.1.1 Lessons learned from Stage 1A test campaign

The assembly was designed and instrumented for de-risking components and their operability. However, its first task was for developing a PI control system, and this subsequently portrays the type of challenges that had to be overcome.

First, matching inputs/outputs was imperative. The real-time target computer was selected with Stage 2A in mind. Its I/O module can handle instruments with high-frequency sampling (kHz), whereas the selection of instruments for Stage 1A operated at 250 Hz.

More critical than this waste of resources were procurement delays due to reissuing specifications. A last-minute check identified a mismatch between output signal instruments (initially in mA) and I/O module inputs (voltage ranges: ±10 V, ±5 V, ±3.3 V, 0-10 V, 0-5 V). The lesson learned here was to produce and keep a single list of I/O and persevere with it – for the whole project.

Calibration is not a one-off activity, which is troublesome when working with high-pressure vessels. Pressure sensors were firstly tested with a portable pressure calibrator at ambient temperature. After connecting the sensors to Stage 1A assembly, nitrogen was used to generate a set of five testing pressures up to 14 MPaG. A linear static calibration was derived from these five points and applied to the DAQ. However, once the assembly was filled with CO_2, it was harder to revisit pressure calibration curves, given the implications of isolating, depressurizing, and repressurizing an assembly that has a capacity of 20 L.

The suitability of some instruments also could be debatable at times. For example, as Table 1 shows, the range of the pressure sensors was selected with a broader functionality in mind (including a future testing of a novel centrifugal sCO_2 compressor), whereas the task in Stage 1A demanded lower pressure targets. Measurement resolution was then affected.

Finally, there were aspirations to produce back-to-back comparisons among instruments in the same measurement station. However, each measurement station only has one port Ø½" BSP. The lessons learned here were:

1) Provide isolation valves for calibrating, repairing, or substituting a pressure sensor in high-pressure pipes or vessels.

2) Static pressure measurements should be taken from at least two stations (ideally four), spaced at 90 degrees in the same plane as that of the pipe

3) Budget for a few sensors with different specifications for the same assembly.

Temperature measurement was based on the application, and not on general expectations. The selection of RTDs for Stages 1A and 1B was related to the level of accuracy desired. Platinum resistance thermometers, however, have a medium time response (1 to 50 s) [4]. This imposed a learning period to identify and reduce the phase lag between the sensor output and the fluctuating temperature of the assembly. Once the fluctuating nature of the process was under control, it was possible to estimate a time constant for the sensor (in each measurement station) and tune the desired PI controller. To convert the signal to a voltage output, a 330 Ω potential divider to 12 V was used.

Here, a challenge arose when unexpected disturbances and variations in the readings were detected. It turned out that a defective power supply was the cause. The lesson learned here was that each test campaign requires a specific assessment of the measurement techniques suitable for it, as agreed with the final customer of the test campaign.

Concerning the influence of modelling on rig performance, the execution of the Stage 1A test campaign was, in principle, equivalent to switching the gas cooler for a water-cooled condenser system in the refrigeration pack. In the controller, the regulation variable (in transcritical mode) is the output temperature from the gas cooler. For Stage 1A, it was not obvious how to select the regulation variable. Firstly, this was because there was no certainty of keeping transcritical conditions and, secondly, because of the lack of experience in modelling and in using PCHE. Across the test campaign, that uncertainty was resolved. However, it was obvious that the control output for the water supply needed readjustment.

The water leaving the precooler (D8906) was piped to a recirculated water system, which can have a strong influence on its temperature. However, the biggest influence on the process variability was coming from the water control valve. Despite its equal percentage flow characteristics, the valve was failing to accomplish its performance, as it was oversized. A second control valve, in parallel, measuring half of the size of the first one, managed to deliver the expected results of the "condenser" functionality. The lesson to be learned is that it is worth reviewing assumptions in modelling that could influence the selection of equipment.

Due to health and safety policy considerations, the refrigeration pack kept its original control system (a pR300T). This controller manages the functionality of the refrigeration pack and the gas cooler in terms of a refrigeration cycle – subcritical or transcritical. To pursue the test objectives of Stage 1A, a better understanding of the logic behind this controller would have been required and keep a stable performance at transcritical operation mode. However, this activity has been delayed by problems getting access to the controller software.

As a closing remark, the control system developed in Stage 1A was the first step to attempting what is called "HPV valve management system" in R744 refrigeration [5]. Such a system separates the high-pressure part of the system from the medium pressure part, determining the transcritical and subcritical operation mode of the plant. In waste heat recovery applications, a similar approach would be required to finely control the critical pressure and temperature of the CO_2 at the inlet of the compressor unit [6]. This system should be able to cool or heat the CO_2 stream near an isobar line sat in the critical pressure, like the Integrated System Test functionalities in the Knolls Atomic Power Laboratory [7].

2.2 Stage 1B

Whereas the measurement techniques for control are based primarily on-time response, the measurement techniques for performance tests are developed around accurate measurements that can ensure expected uncertainty ranges for performance parameters. In the case of single-phase heat exchangers, these performance parameters are the overall heat transfer coefficient, the heat transfer rate, and the pressure drop. To achieve the targeted overall uncertainty for these parameters, consideration for instrument accuracy, repeatability, and sensitivity were the criteria for their procurement approach. The development of Stage 1B demonstrated that there were more elements to be assessed.

The initial objective for this stage was to test the performance of the main heat exchanger (MHEX D8908) as a cooler ("cold runs": CO_2 vs air) [2]. For this purpose, the gas cooler of the transcritical CO_2 refrigeration pack was bypassed using a valve manifold, whereas the CO_2 was directed to the MHEX. A centrifugal fan provided the required air to cool down the supercritical CO_2 at the MHEX, before being expanded and returned to the transcritical compressors.

As shown in Figure 4, the MHEX was connected in a counter-flow arrangement, with measurement equipment installed for the validation of the energy balance between the hot and the cold side. Platinum RTD sensors, of "Class A" (±0.2 K), were installed at the inlet and outlet of MHEX for both the CO_2 and air sides, according to the operating temperature range. The same applies to the pressure transducers (which are piezo-resistive).

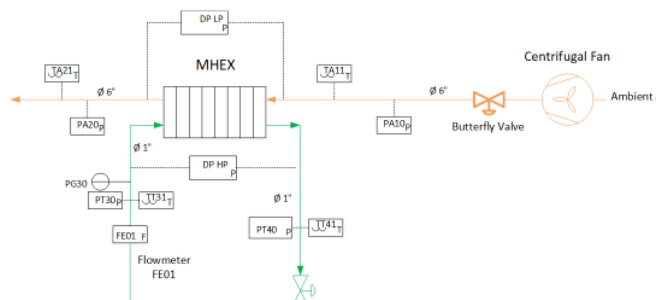

Figure 4 P&ID STAGE 1B

The carbon dioxide temperature and pressure are measured at 10 cm (that is approximately 3 tube diameters), respectively upstream of the inlet and downstream of the outlet of the MHEX. A differential pressure gauge was attached to the airside, to measure the pressure drop over the cold side of the heat exchanger. The mass flow rates of both sides are controlled by valves, while a Coriolis mass flow meter (with an error of ±0.5%) was installed to measure CO_2 mass flow rate. A hot-wire anemometer (error of ±3% + 0.3 m/s) was employed to measure air velocity at a specified position in the air duct. This could be used to estimate air mass flow rate.

A centrifugal fan (1000 ft^3/min nominal capacity) was used to provide cold air to the heat exchanger. Air mass flow was adjusted by throttling a butterfly valve. In Table 2, the accuracy information of the instrumentation used is summarized.

Table 2 MEASUREMENT DEVICES EMPLOYED FOR STAGE 1B TESTING AT CRANFIELD UNIVERSITY

Parameter	Device	Range	Accuracy
CO2 Stream			
Pressure	Piezo resistive	0 – 20MPa	±0.25%
Temperature	RTD Pt100	-20 – 250°C	±0.2°C
Mass flow	Coriolis	0 -1kg/s	±0.5%
Air Stream			
Pressure	Piezo resistive	0 – 0.1MPa	±0.25%
Diff. press	Piezo resistive	0 – 7.5kPa	±0.3%
Temperature	RTD Pt100	0 – 150°C	±0.2°C
Speed	Hot-wire Anemometer	0 – 40m/s	±3%

The testing setup and procedure were developed following guidelines from the ASME PTC 12.5 [8]. However, specific details of the MHEX geometry design could not be shared by Heatric, and therefore the heat transfer area could only be estimated. From a literature review, Cranfield and Thessaly universities developed an approximated methodology to explore the performance of the MHEX for cold runs (steady and dynamic). This methodology is reflected in the collaborative work of [9], where results for four (4) experimental cases are discussed in detail.

It was possible to execute a low fidelity performance test for the MHEX, acquiring the low-pressure losses of both gas streams of the new technology. There were some experiences worth reflecting upon.

2.2.1 Lessons learned from the Stage 1B operation

Achieving stable process conditions for executing the Stage 1B test campaign implied another learning curve in the operation of the rig.

The longer the experiment lasts, the larger the drift in the temperature of the "cooling" air. A temperature difference between ambient and discharge of the centrifugal fan of up to 15 °C was registered. This arose from an unaccounted-for inefficient fan.

As the MHEX was acting as the bypassed gas cooler of the refrigeration pack, the pack itself started to increase temperature and pressure at each level. The obvious consequence was a hotter system and higher pressures, which could cause a safe shutdown of the compressor, due to high-pressure at the suction or the discharge of the compressors. A side consequence was the limited margin to operate the throttle valve at the discharge of the fan. The lesson learned here was to always run isolated performance tests of services equipment to verify their process delivery conditions.

The load control of the MHEX was achieved using a combination of different industrial solutions to select the expansion device and its controller. This proved to be not only pragmatic but also rigid. The controller was set using the same principles of controlling a transcritical CO_2 gas cooler. In transcritical mode, the EVD offers a simple PID algorithm [10] which follows formula (1):

$$P_{GCset} = A \times T_{GC} + B \tag{1}$$

Where:

P_{GCset} CO_2 gas cooler pressure setpoint

T_{GC} Gas cooler outlet temperature

A, B Linear coefficients, between -100 to 800, user's choice

The control is direct: as the pressure increases, the valve opens. This was a solution that worked, if the difference between the gas cooler outlet temperature and the setpoint remained within ±2.5 °C. In this experiment, the controller was monitoring the rapid change of the MHEX outlet temperature; therefore, setting manually the linear coefficients was an urgent activity to keep the percentage of the valve aperture in place. Lesson learned: there is always a trade-off between control solutions bought from the shelves and the time and effort invested in developing tailored control applications.

Measuring techniques in Stage 1B also provided opportunities for learning and analysis. These related to addressing inadequate installation practices to revisiting assumptions in the uncertainty analysis.

The calculation of the heat transfer in the printed-circuit heat exchanger accounted for the uncertainty in the mass flow rate, temperature, and pressure measurements. The equation for the heat transfer uncertainty is expressed as follows in equation (2) (see [11]):

$$\frac{\delta \dot{Q}}{\dot{Q}} = \left[\left(\frac{\delta \dot{m}}{\dot{m}}\right)^2 + \left(\frac{\delta h_{in}}{h_{out} - h_{in}}\right)^2 + \left(\frac{\delta h_{out}}{h_{out} - h_{in}}\right)^2\right]^{1/2} \tag{2}$$

Where enthalpy error can be calculated by (3):

$$\delta h = \left[\left(\frac{\partial h}{\partial T}\delta T\right)^2 + \left(\frac{\partial h}{\partial P}\delta P\right)^2\right]^{1/2} \tag{3}$$

Therefore, from the measured values and the accuracy of sensors for the mass flow rate, temperature, and pressure, the uncertainty of both hot and cold sides could be calculated. The partial derivatives for the accuracy estimation of enthalpy were estimated using the NIST Refprop database [12].

However, calculation of uncertainty performed before the test (pre-test) conducted by [13] showed that estimating the different properties of state near the CO_2 critical point, from the point of view of uncertainty propagation, was superior by using density and pressure measurements rather than using temperature and pressure. As is known, near the critical point, the properties change more abruptly with temperature – changes that would affect the estimation of enthalpy. Therefore, instead of using equation (3), equation (4) can be used

$$\delta h = \left[\left(\frac{\partial h}{\partial \rho}\delta \rho\right)^2 + \left(\frac{\partial h}{\partial P}\delta P\right)^2\right]^{1/2} \tag{4}$$

In this parametric study [13] and in the original work [14], it was also shown that PCHE's operating near the critical point or when the temperatures between the cold and hot side were small (less than 5 K), a large propagation of temperature uncertainty was expected from the temperature sensors. A situation that would impose a more demanding requirement for instrument selection. Alternatively, density measurement via a Coriolis flowmeter would provide a way to estimate thermodynamics properties.

During Stage 1B test campaign, there were concerns about operating the MHEX equipment near the critical point. Apart from setting the controllers of the circulation loop in "transcritical mode only", it was required to monitor the change of the CO_2 density upstream of the heat exchanger itself, therefore, the Coriolis flowmeter was placed at the inlet of the MHEX.

Estimating the properties of the airside was less rigorous and complete than the original aspirations. Important concerns about offering a relative low-pressure air-gas stream delayed the acquisition of humidity and flow rate measurement (with low permanent loss pressure) equipment during the duration of the project. The hot-wire anemometer proved to be useful mostly for the range of design temperatures, but installation (hermicity, steady position) was still challenging.

To conclude this section, the operations and measurements followed in Stage 1B represented an important effort to execute a performance test. Achieving the operational conditions to keep the test under stable conditions represented an important but time-consuming task. Achieving an acceptable set of measurement techniques, which guaranteed the uncertainty targets in the performance parameters, was a long-term task that required a few practical iterations of modelling and testing.

3. UPGRADES OF THE RIG – THERMAL MANAGEMENT

Lessons learned during the execution of the test campaigns started to point out that the measurement techniques of the rig required reconsidering. Most of the instruments alone were selected under the premise of achieving proof-of-the-concept or "performance-like" testing. However, elements such as sensitivity, traceability, repeatability, and error propagation were not fully addressed. Also, most of the initial measurements were not appropriate for transient characterization.

The first section of the rig to be upgraded was the transcritical CO_2 pack (circulation loop). During the execution of the test campaigns, it was obvious that having a better understanding and predictability of the pack performance were required. Results from different numerical models of the rig [15,16] started to show incongruencies with experimentation during the verification and validation procedures. On top of the challenges of biphasic modelling, measurements in the refrigeration pack were insufficient, delayed, and even misplaced for the single-phase legs. Fortunately, the pack itself can offer room for testing different CO_2 (or R744) technologies: for refrigeration, heat pumps, and HVAC. This justified further investment in its instrumentation. Effectively, in 2019, measurements of pressure and temperature for the circulation loop were doubled. By 2020, it had an independent DAQ system, which was developed in LabView.

Fortunately, the entire rig could also facilitate research activities within a new Innovate UK project, called Powerplant Integration of Novel Engine Systems (PINES). This was specifically for thermal management and will last until 2023.

Thermal management studies are becoming increasingly more important in civil aviation research. This is because of higher thrust and power generation demands on gas turbine engines, which results in increased heat generation. This leads to hotter fluids and higher component temperatures. Thermal management technologies will utilize engine fluids, refrigerants, or thermally neutral heat transfer fluids to transfer excess heat from the engine waste heat sources, such as bearings, pumps, generators, batteries, and transmission systems, to the main aircraft heat sinks (ram air and fuel).

The following sub-sections will describe the main measurements upgrades, reversed engineering findings, and lessons learned in this new version of the transcritical CO_2 pack.

3.1 Choice of temperature sensors

A common practice in the HVAC industry is to use thermistors placed on the surfaces or recesses of pipes. The advantages of this kind of sensors are that they are quite robust and accurate. However, they usually have a time constant of around 10 s [4], which is very high. This is acceptable regarding the slow evolution regimes associated with industrial HVAC systems, such as food cooling chambers. However, when considering systems facing faster dynamics, such as airplane thermal management systems, this time constant is too high to allow a more responsive regulation. Indeed, the heat load to be dissipated can vary quite swiftly, depending on the flight conditions, and temperature limits can be very strict. This has motivated the selection of thermocouples with much shorter time constants (below 1s for diameters under 3mm).

Figure A1 of the Annex shows a diagram of the circulation loop of the sCO_2 rig at Cranfield, detailing the layout of the different components and the location of the sensors. Measurements have been performed at the inlet and the outlet of the main elements of the rig to enhance the description of their thermodynamic behavior. The additional sensors are tagged as follows: letters 'P' and 'T', for the measurement of pressure and temperature, respectively. After these letters follows an identification number, which can range between 10 and 70, depending on the position of the sensor in the rig. Table 3 shows the characteristics of the instrumentation, including details about the time response of the sensor.

General guidelines for minimizing the sources of error when making temperature measurements in the rig were taken. Instrument installation tries to assure that conduction (immersion) errors, radiation errors and recovery errors were negligible. Where possible, insulation of the pipe wall near the thermocouple was installed. The immersion length of the probe extended a sufficient distance into the fluid stream to minimize the conduction of the heat. Tips of probes were set in the middle of the stream, avoiding stagnant areas, and not touching the pipe walls.

Table 3 MEASUREMENT DEVICES APPLIED IN THE TRANSCRITICAL CIRCULATION LOOP AT CRANFIELD UNIVERSITY

Para-meter	Device	Range	Accuracy	Time res-ponse
Rig instrumentation upgrade				
Pressure	Piezo resistive	0 – 6MPa	±15kPa	1ms
	Piezo resistive	0 – 10MPa	±25kPa	1ms
	Piezo resistive	0 – 16MPa	±40kPa	1ms
Temper ature	K-type TC – Ø 3mm	-40 – 1100°C	±1.5°C	0.8s
	K-type TC – Ø 0.75mm	-40 – 1100°C	±1.5°C	90ms
Mass flow	Coriolis flowmeter	0 – 1kg/s	±0.5%	n/a
Original instrumentation				
Pressure	Piezo resistive	0 – 15MPa	±0.15 to 0.6MPa	10ms
Temper ature	NTC 10kΩ	-50 – 105°C	0.3 to 1°C	10s
	NTC 10kΩ	-50 – 105°C	1°C	50s
	NTC 50kΩ	0 – 150°C	1 to 1.9°C	30s

Following guidelines from ASME PTC 19.3 [17], calculations of the aforementioned bias errors were also made. Conduction and radiation errors were negligible, which was expected given the magnitude of the heat transfer coefficient between the CO_2 (gas state) and the thermocouple sheath (stainless steel 304) of 1700 [$W.m^{-2}K^{-1}$]. Calculations of recovery errors (also known as the aerodynamic heating effect) were also low, in the range of ±0.04 °C.

When possible, temperature measurements were performed with intrusive probes into the CO_2 flow, using compression fittings and Ø3 mm K-type thermocouples (Figure 5– right). These had an associated time constant of around 1s and enough robustness to withstand the high pressures of the rig (up to 12 MPaG). Otherwise, when installing intrusive probes could not be possible, thermocouples attached to the surface of the pipes were used. In this case, a narrow K-type thermocouple of Ø 0.75 mm and a time constant of 0.09 s (see Figure 5 – left) were selected. Thermocouples were attached to the pipe with thermal paste and covered with a layer of 13 mm thick lagging (with a conductivity of 0.033 $W.m^{-1}.K^{-1}$ at 0 °C). All sensors are connected via screened and earthed extension wires to reduce electromagnetic perturbation from other equipment.

Figure 5: NARROW THERMOCOUPLE (Ø0.75MM) FOR SURFACE MEASUREMENT BEFORE BEING COVERED WITH PIPE LAGGING (LEFT) THERMOCOUPLE INSERTED WITHIN THE PIPE (RIGHT).

To compare the differences between inline and surface measurements, an additional surface thermocouple was placed next to an inline one. The results for these two sensors at the discharge of the compressor, after the oil separator, have been plotted in Figure 6. This graph shows low-frequency oscillations of the discharge temperature on which a delay of around 2 s for the surface measurement and an offset of 0.15 °C on the mean value can be noticed. This provides a reasonably good agreement between surface and direct flow measurement, especially in the case of steady acquisitions.

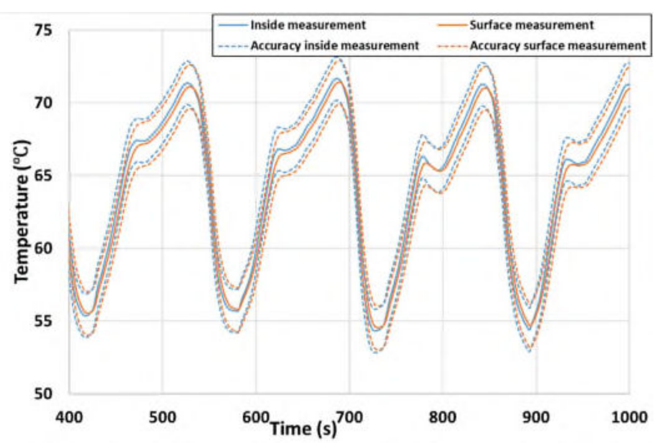

Figure 6: COMPARISON BETWEEN DIRECT FLOW (T21) AND SURFACE MEASUREMENT (T210) AT THE DISCHARGE LOCATION OF THE COMPRESSOR

Pressure measurements were performed using transducers with 100 ms of response time and between 15 and 40 kPaG of accuracy, depending on the range of measurement. The Coriolis flowmeter previously used for testing the MHEX (Stage 1B) was moved and re-installed at the discharge of the compressors. This was done downstream of the oil separator to measure CO_2 in the gas phase. The accuracy of the measurement of mass flow is 0.5% of the maximum mass flow. Attempts to measure CO_2 in liquid phase were unsuccessful using an ultrasonic flowmeter. In principle, the installation of the ultrasonic flowmeter would be far simpler than with a Coriolis one. However, such non-intrusive units require clean liquids (they must contain less than 10% suspended solids or entrained air/gas) in full pipes of between 10 mm to 3000 mm in diameter. The rig layout installation could not guarantee a single-phase liquid stream.

3.2 First results and discussion

Figure 7 shows an example of measurements from the start-up of the liquid valve opening to the shutdown in transcritical operation.

The ambient temperature was 13°C on average, the suction pressure set to 3.8 MPaG, and the outlet temperature of the gas cooler to 38°C. The first observation on these measurements is that, in these conditions, a steady-state regime is obtained in approximately 800 s and maintained until the shutdown of the liquid valve. Three levels of pressure and temperature stand out: high pressure and temperature (9 MPaG and 85 °C – above the 7.28 MPaG of the CO_2 critical point) at the discharge of the compressor; medium level (5 MPaG and 50 °C) at the liquid separator level; and low pressure and temperature (3.8 MPaG and between 0 and 18 °C) at the evaporator stage.

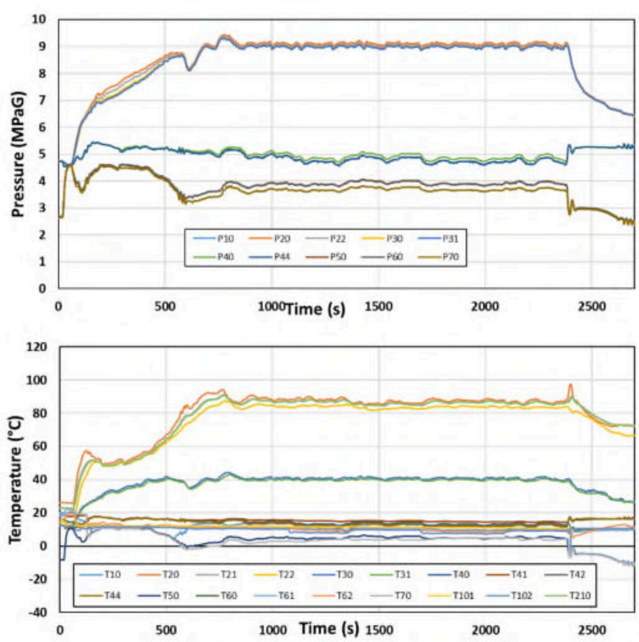

Figure 7: MEASUREMENTS OF PRESSURE (±40 kPaG), TOP, AND TEMPERATURE (±1.5 °C), BOTTOM, DURING TRANSCRITICAL OPERATION OF THE RIG – GAS COOLER OUTLET TEMPERATURE OF 38 °C AND SUCTION PRESSURE OF 3.8 MPaG

The initial compression work seems to operate over two time periods: the first, between 70 and 120 s, where the temperature and the pressure of CO_2 increase quickly up to the critical pressure, and the second, where CO_2 is brought to the requested pressure and temperature. This change of behavior corresponds to the overcoming of the CO_2 critical point.

In terms of regulation, in subcritical operation, the discharge pressure is directly linked to the condensation temperature, considering or not an eventual subcooling [5]. Another parameter has to be found above the critical point. A common approach to determine the optimal discharge pressure for transcritical cycles is to evaluate the COP (Equation (4)) of the installation. The high pressure will then be chosen to maximize this COP [18]. To do so, theoretical cycles (compression – gas cooling – expansion – evaporation) have been modeled for different discharge pressures at a fixed gas cooler outlet temperature (T_{GC}) and led to Figure 8, which plots the COP as a function of the discharge pressure for different cases of T_{GC}. Due to the inflection of the isotherms above the critical point, the COP increases to a maximum before falling back as the compression does not provide additional assistance in retrieving a significant amount of energy.

$$COP = \frac{Q_{evap}}{W_{comp}} \quad (4)$$

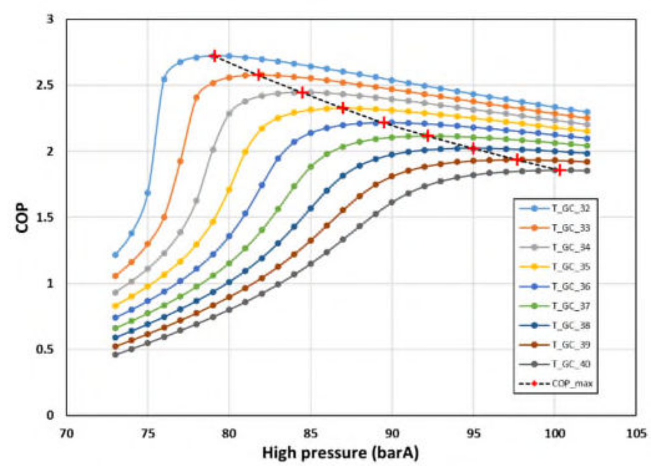

Figure 8: COP OF A THEORETICAL TRANSCRITICAL CYCLE AS A FUNCTION OF THE DISCHARGE PRESSURE FOR TGC BETWEEN 32 AND 40°C. THE RED PLUS CURVE CORRESPONDS TO THE MAXIMUM COP FOR EACH CURVE

From this, an optimal pressure, corresponding to a maximum COP, can be defined as a function of T_{GC} (2). A set of measurements were performed for settings of T_{GC} between 32 °C and 40 °C on the refrigeration pack to acquire the discharge pressure set by the controller. Results are plotted in Figure 9. The evolution of the pressure follows quite accurately the slope given by Equation (5), despite a negative offset of around 0.3 MPaG. This offset can have different explanations, such as the calculation of the ideal COP, which can be affected by the choice of the suction pressure and the superheat.

$$P_{opt} = 2.645T_{GC} - 5.5422 \quad (5)$$

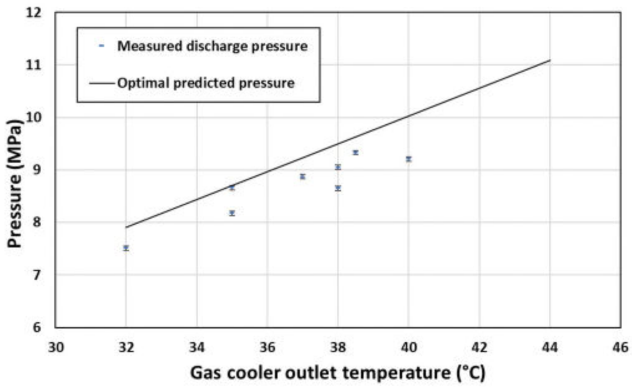

Figure 9: DISCHARGE PRESSURE DEPENDENCE ON T_{GC} IN THE TRANSCRITICAL ZONE: MEASURED (WITH ERROR BARS) AND CALCULATED (WITH (5)

In addition to these measurements, the signal of the Coriolis flowmeter is acquired at the same frequency (through the same interface) and monitoring of compressor 1 data are recorded in parallel directly from its drive.

4. EVOLUTION OF THE RIG

As stated in Section 2, the functioning of the rig is highly dependent on ambient conditions. The temperature of the air is imposing to work mostly in transcritical mode in summer when the external temperature exceeds the critical temperature of CO_2 (31 °C). In winter, the most limiting parameter is the temperature of the water for the evaporation of the CO_2. Although the capacity of the water tank is not negligible (3500 L), the high capacity of the evaporator, can drop the ambient temperature of the water circulation up to 2.6 °C in 20 minutes (in Autumn). With the large temperature difference between the evaporator inlet and outlet, the water can quickly reach the freezing point in winter conditions. This has motivated two future evolutions to make the rig less weather dependent. The first is to control the temperature of the water in a small loop equipped with an immersion heater, offering a variable heat load. The second is to use the assembly of Stage 1A as a variable heat sink.

With improved control and monitoring, of the heat exchange at the evaporation and the condensation/gas cooling level, the rig will transform from a refrigeration cycle into a thermal management concept.

4.1. Variable heat load

This upgrade intends to divert the water circulation used at the evaporation level to a smaller loop equipped with an immersion heater (Figure 10). This loop will have a capacity of 2500 L of water, stored in a polypropylene tank covered with expanding insulating foam (25 mm in thickness and thermal conductivity of 0.042 W/(m·K)), and instrumented with temperature sensors to monitor the water circuit temperature. This temperature will be modulated with a 50 kW immersion heater in circulation heating configuration (flow going through the heater). The water will be driven to the evaporator by a 4 kW centrifugal pump, to attain a flow rate up to 8 L/s.

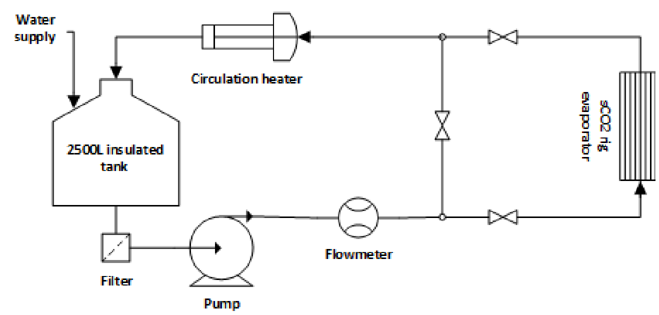

Figure 10: VARIABLE HEAT LOAD ARRANGEMENT

With a simple calculation, using typical values of heat exchange at the evaporator (≈90 kW), and considering an initial water temperature of 20 °C and an average flow rate of 5 L/s, one can expect 45 to 60 minutes of running above the freezing point of the water.

An additional Coriolis flowmeter is planned to be installed on the liquid line, upstream of the liquid expansion valve to complete the measurement of CO_2 mass flow along the circuit. This is to provide more precise data for numerical validation and to quantify more precisely the transfer that occurs through the evaporator. Still, there are risks of inaccurate measurement, given that the installation favors gas in the liquid line.

5. CONCLUSION

After summarizing and self-reflecting on the main activities of Cranfield's sCO_2 rig until the end of 2021, this paper described the changes in criteria for instrumentation selection. This was motivated by a will of approaching transient behavior with faster, but less accurate, sensors. Lessons learned through the test campaigns highlighted that the selection of measurement techniques for CO_2 cycles should be guided by accuracy, properties near the critical point, and time response characteristics. The first results provided access to useful insights on transcritical regulation methods. Additionally, the lessons learned already gave direction to new upgrades for 2022. These include 'common sense' upgrades, but ones that are nonetheless important, as they would allow for isolation of sensor connections for maintenance and replacement, as well as planning for extended ranges for sensors and actuators sized via models.

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ANNEX A

DIAGRAMS AND SCHEMES

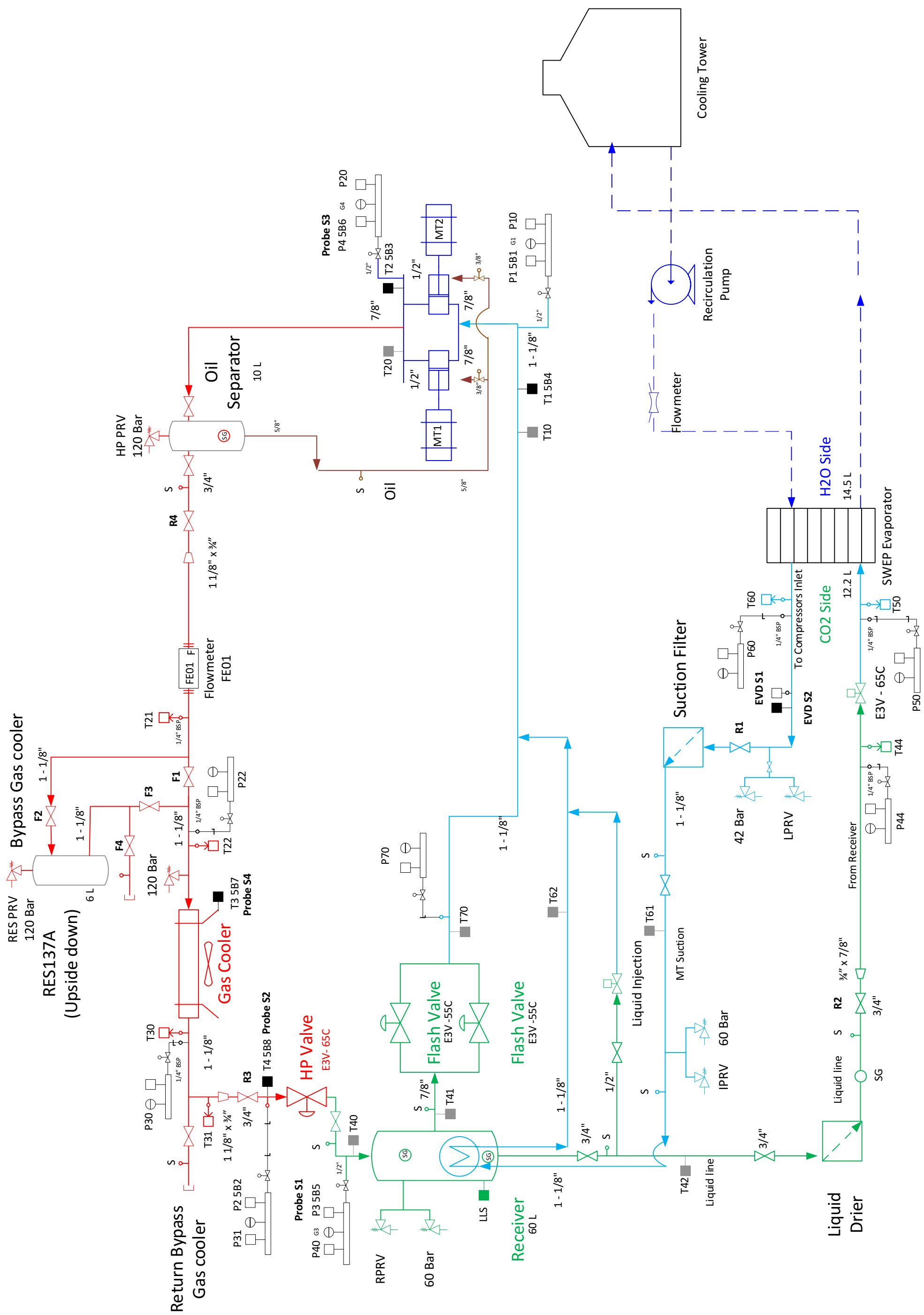

Figure A1 – Updated P&ID of the transcritical circulation loop (bypassing manifold after RES137A not shown)

Update of the sCO2-test rig at Cranfield University

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2022-10-28

Anselmi E, Belleoud P, Roumeliotis I, Pachidis V. (2022) Update of the sCO2-test rig at Cranfield University. In: ASME Turbo Expo 2022: Turbomachinery Technical Conference and Exposition, 13-17 June 2022, Rotterdam, Netherlands

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