EXPERIMENTAL CHARACTERIZATION OF A HIGH-TEMPERATURE BRAYTON HEAT PUMP TEST RIG WITH OFF-THE-SHELF-COMPONENTS

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ABSTRACT

This paper evaluates the performance of two commercial centrifugal compressors operating in parallel within a High-Temperature Brayton Heat Pump (HTHP) test rig. Utilising small-scale, off-the-shelf centrifugal compressors can foster the widespread adoption of Brayton HTHPs for decarbonising heat demand in small-scale industrial sectors that require compact and flexible machinery. In this context, assessing the effectiveness and feasibility of adapting components developed for other applications to the specific requirements of a Brayton heat pump becomes crucial.

INTRODUCTION

High-temperature heat pumps (HTHPs) play a crucial role in decarbonising thermal energy demands in the industrial sector [1]. Among the various configurations, heat pumps based on the reverse Brayton cycle have gained increasing attention, particularly for applications delivering heat above 150°C – 200 °C [2].

This work aims to assess the real operations of the Brayton High-Temperature Heat Pump test rig designed and developed at the University of Pisa, described in [3]. Moreover, the idea is to investigate potential operational issues arising from adapting off-the-shelf compressors to an HTHP cycle, which if from one side can foster the widespread of this technology thanks to the high readiness of the components on the other side can limit the real operation due to the different field of application.

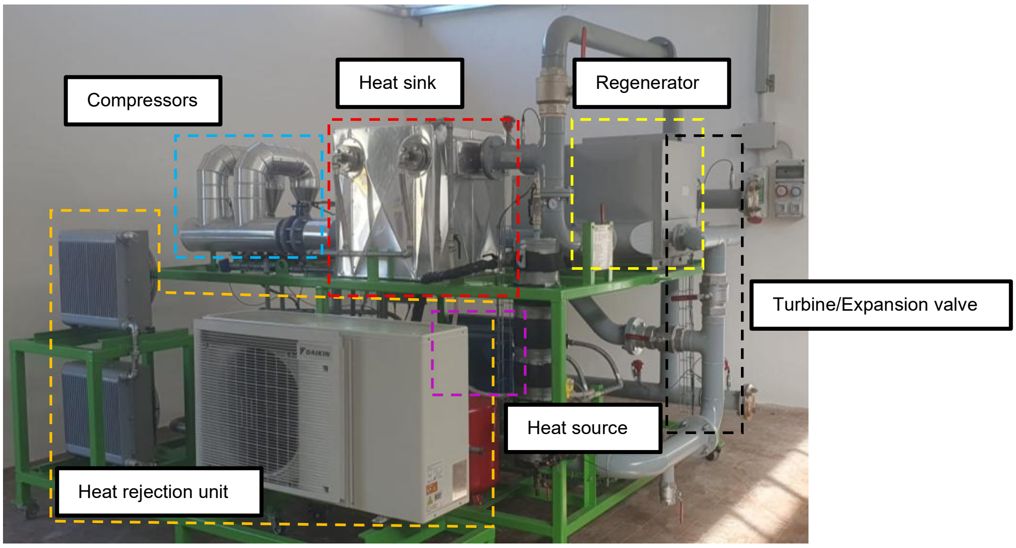

Figure 1 – Prototype developed at the University of Pisa

RESULTS AND DISCUSSION

To evaluate the real operational behavior of the Brayton HTHP prototype, the parallel compressors were experimentally characterized to reconstruct their actual performance maps. This assessment is crucial for identifying discrepancies between real-world operation and nominal manufacturer data. Since the test rig integrates two compressors in a parallel configuration [3], deriving the map from experimental data is essential to capture non-ideal effects and flow maldistributions, which are otherwise neglected when simply doubling the corrected mass flow rate of the manufacturer's single-compressor curves.

Theoretically, two identical compressors operating in parallel at the same rotational speed should process equivalent mass flow rates and achieve identical pressure ratios. In practice, however, flow maldistributions and geometric effects can lead to performance deviations. To reconstruct the performance maps of the parallel compressor configuration, the following experimental procedure was implemented:

1. A valve position is set to impose a certain pressure drop across the compressors
2. A certain motor speed is set for both the inverters that drive the compressors

3. The speed is maintained constant, waiting for the system to reach the steady state (i.e. when the mass flow rate and the pressure are stable)
4. The data are recorded for some minutes

Steps 2 to 4 are repeated for increasing rotational speeds up to the maximum allowable limit of the motors. Subsequently, the valve position is modified to vary the system pressure drop, and the entire cycle (Steps 1–4) is repeated to map the entire operating range.

Figure 2a compares the ideal parallel operation (derived from the manufacturer's data) with the measured performance, while Figure 2b represents the percentage difference of pressure ratio and corrected mass flow rate for every compressor speed.

The experimental performance map exhibits a lower pressure ratio for a given corrected mass flow rate and corrected speed compared to the manufacturer's baseline. This discrepancy can be attributed to several factors. While the manufacturer's map refers to a single compressor – which was ideally doubled in terms of corrected mass flow rate for this comparison – the two units operating in parallel may not share the fluid load perfectly. Consequently, the combined mass flow rate is lower than the theoretical double of a single-compressor operating point. This discrepancy is exacerbated at higher rotational speeds, increasing in both absolute and relative terms; a similar degrading trend is observed for the pressure ratio.

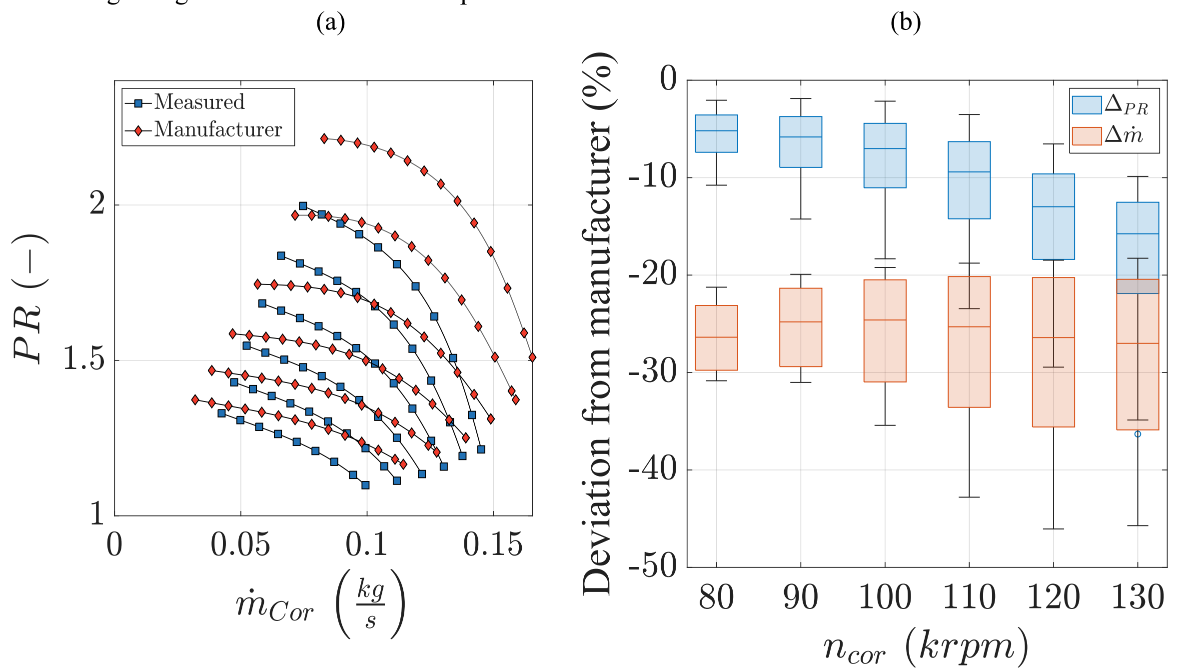

Figure 2 – Comparison between the compressor map provided by the manufacturer (red diamond) and the one measured within the test rig (blue square); the results are reported for different compressor speeds, from 80 krpm to 120 krpm every 10 krpm (a). Percentage difference between the measured data in terms of corrected air flow rate and pressure ratio for each compressor speed (b).

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