Automatic Test Stand to Impose Operating Suction and Discharge Conditions on Compressors

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ABSTRACT

This work describes the control algorithms used for automating a test stand to impose suction and discharge pressures and suction temperature on reciprocating compressors, allowing the operation to be evaluated under different conditions within the compressor envelope. Since the objective is to impose conditions on the compressor, a superheated gas cycle was considered. In addition, an auxiliary circuit was considered, which is responsible for cooling the fluid as it passes through the reservoir in the intermediate region of the main circuit. This auxiliary system is particularly important to achieve low discharge pressure conditions even after the compressor under test has been operating for a long time. To reach the entire operating envelope of several compressor models, an automatic procedure is proposed to insert and remove the refrigerant fluid from the test stand. For this purpose, a reservoir tank was connected through solenoid valves to both the discharge and service lines, making it possible to control the volume of fluid in the main circuit. The dynamics of the variables of interest were identified using experimental data and the resulting models were used to tune automatic controllers for imposing the compressor operating condition in terms of pressures and temperature. The current version of the test stand is able to automatically impose any condition inside the operating envelope of a family of compressors and keep the compressor operating around this point with tolerances of less than 1 °C.

1. INTRODUCTION

Refrigeration compressor manufacturers invest large amounts in research and development to continuously improve their products and also for quality control purposes. One of the activities carried out in this context is compressor testing. Some of the tests are regulated by international standards, such as EN 13771-1 (DIN, 2017) and ANSI/ASHRAE 23 (ASHRAE, 2005), which outline test methods, operating conditions, and maximum acceptable measurement uncertainties for the instruments. A relevant aspect for the technological development of refrigeration compressors is the ability of testing them in numerous operating conditions, in a reliable way.

Numerous factors contribute to defining the operating condition of a compressor. These include the thermodynamic state of the fluid entering and exiting the compressor, the temperature of the compressor housing, forced ventilation condition, rotation speed, among others (Cengel & Boles, 2014). During testing procedures, most of these factors are typically kept constant, thereby the operating condition becomes primarily defined by the conditions imposed at the compressor inlet and outlet ports. In this case, each operating point can be described by the suction temperature and the suction and discharge pressures. As in typical refrigeration systems those pressures are good approximations of the evaporator and condenser pressures, except for the pressure drop in the piping, it is a common practice in industry to define the operating points in terms of saturation temperatures instead of pressures.

In test stands, the operating condition of the compressor is typically imposed by adjusting valve openings for the pressure control loops, as well as the electrical power in heating elements and the rotational speed for coolers in temperature control loops. Traditionally, these variables are managed using two or more single-input and single-output controllers (SISO), regardless of whether the test stand variables exhibit multiple-input and multiple-output (MIMO) characteristics, nonlinearities, or time delays (Goyal et al., 2019).

In general, refrigeration tests that require feedback control use classical techniques, such as proportional-integral-derivative (PID) controllers. Some examples of works which made use of PID controllers in the context of refrigeration are: the test rig for measuring isentropic and volumetric efficiencies in variable-speed compressors (Vetsch et al., 2016); the control of the degree of superheat at the outlet of the evaporator (Maia et al., 2010); the controller to improve the energy efficiency ratio of refrigeration systems (Franco et al., 2022); and temperature and heat transfer control of a thermoelectric cooler (Kherkhar et al., 2022). This kind of control solution is used mainly because it is simple and presents good closed-loop behavior for a wide class of processes. However, for processes with significant time delay, nonlinearity, or coupling between variables, the performance of PID controllers may not be satisfactory, and advanced control techniques can provide an advantage over them. Some examples of advanced control techniques used in refrigeration systems are: a predictive controller with a suboptimal nonlinear model used to control the suction and discharge pressures of compressors being tested in a test rig (Schwedersky et al., 2018) and the development of a predictive controller to minimize the energy consumption of condensing units while maintaining the cooling setpoint temperature (Ra et al., 2023).

This work proposes a control strategy for a test stand exhibiting MIMO behavior, aiming to achieve a wide range of operating conditions inside a certain operating envelope. For this purpose, a superheated gas cycle test rig was designed and built to control suction and discharge pressures and suction temperature, as well as other variables that can affect test results. An auxiliary circuit responsible for cooling the refrigerant fluid as it passes through the reservoir within the intermediate region of the main circuit was installed, enabling to reach low discharge pressure conditions even after the compressor under test has operated for a long time. An automatic procedure was also proposed to insert and remove refrigerant fluid from the test rig, by implementing a reservoir tank that is connected through solenoid valves to both the discharge and service inlet lines. To reach conditions in which both the suction and discharge pressures are close to each other, a bypass valve between the compressor inlet and outlet ports was added, which, if fully opened, would equalize the two pressures. Although such modifications enabled a substantial increase in the testing capabilities, they complicated the control problem to be solved. With the proposed method, based on a discrete supervisory controller associated with PID loops, it was possible to tune SISO controllers for the MIMO system.

The remainder of this paper is organized as follows. The description of the test rig and the initial evaluation are presented in Section 2. In Section 3, the modifications to the initial test rig are proposed, presenting the arguments for the choices and possible impacts. The results of the proposed modifications are shown in Section 5, where the proposed controllers are presented, as well as the final achievable conditions considering the operating envelope. Finally, the conclusions are presented in Section 6.

2. INITIAL EVALUATION

The test rig considered in this paper is illustrated in the piping and instrumentation diagram (P&ID) presented in Figure 1a. This test rig, previously developed in Schwedersky et al. (2018), is used in the refrigeration industry to emulate the operating conditions to which refrigeration compressors are subjected in real cooling systems. Its operation is based on a superheated gas cycle, in which the refrigerant fluid experiences no phase change. This concept is used in some methods to assess compressor performance (ASHRAE, 2005) and compressor sound power level and vibration (AHRI, 2005).

The test rig contains two valves and a buffer tank, which is used to partially decouple the discharge and suction pressure lines. During testing procedures, the operating conditions become primarily defined by the regulation of the pressures at the compressor inlet and outlet, defined respectively as suction pressure p_s, and discharge pressure p_d. To acquire data related to the process, the test rig is instrumented with pressure transducers on the discharge and suction lines, as well as a temperature sensor on the compressor housing.

The test conditions shown in Figure 1b are used as a case study in this paper and were obtained based on the application map of a hermetic reciprocating compressor used in an ice cream display freezer. R-290 (propane) is used as the refrigerant fluid and the nominal cooling capacity of the compressor is 965 W, considering ASHRAE low back pressure condition. In this figure, the blue squares represent the reference values, the blue exes represent the actual pressure measurement points, and the red exes represent the conditions that the test stand was not able to impose to the compressor under test.

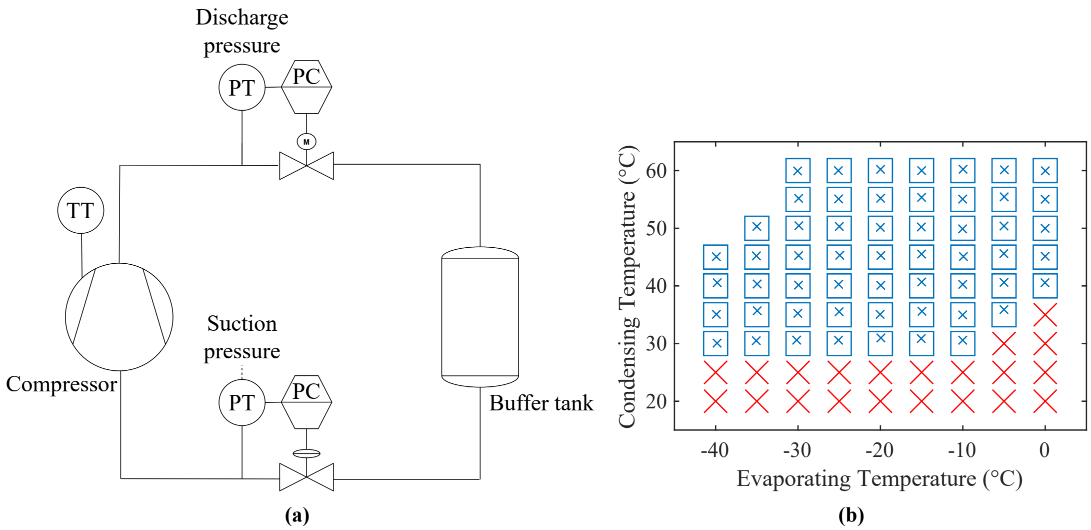

Figure 1: Initial test rig setup: (a) P&ID of the automatic test rig; (b) Condition map with the initially achievable conditions marked in blue.

The original formulation of test rig was able to impose operating conditions in the range of 111 kPa to 474 kPa for p_s, and 1079 kPa to 2116 kPa for p_d. In terms of evaporating temperature, T_{ev}, the range goes from about −40 °C to 0 °C, while the range for the condensing temperature, T_{co}, goes from about 30 °C to 60 °C. As can be seen, the lower condensing temperatures are not achieved during the test and they represent about 28% of the desired test conditions. It is worth mentioning that if the low condensing temperature conditions are all measured as soon as the compressor is switched on, it could be possible to achieve practically the whole map. However, this strategy is often not feasible, since typical tests requires other variables to be settled, such as the temperature of the compressor housing.

3. PROPOSED TEST RIG

To overcome the limitations presented in the tests described in Section 2, modifications were made to the test stand according to the P&ID diagram shown in Figure 2. The modified version of the test rig is divided into two main parts for the sake of clarity: the main circuit, described in Subsection 3.1, and the auxiliary system, described in Subsection 3.2.

3.1 Main circuit

Compared to the original test stand described in Section 2, the basis of operation is essentially the same. The main modification in the main circuit is the addition of the pressure-controlled diaphragm equalization valve, which aims to partially equalize the pressures between discharge and suction, providing another degree of freedom for the control of the system condition. The core idea behind this addition is the possibility of reaching conditions where p_s and p_d are close in terms of pressure values, as presented in the lower right corner of Figure 1b.

Additionally, a suction temperature control loop was added to track a desired value during the tests. The suction temperature is measured with a Pt100 sensor and the controller output is implemented using heating and cooling elements.

Another proposed addition is a reservoir tank that allows for control over the volume of fluid in the system. This process is controlled by two solenoid valves in the reservoir tank inlet and outlet. When the valve between the discharge outlet and the service inlet is open, the tank is pressurized and the overall volume of refrigerant fluid in the main circuit is decreased. This enables the test stand to reach conditions with lower suction and discharge pressures. On the other hand, when the solenoid valve between the reservoir tank and the service inlet is open, the pressurized refrigerant is released into the main circuit and there is an increase in the system pressures, enabling higher evaporating and condensing temperatures to be reached.

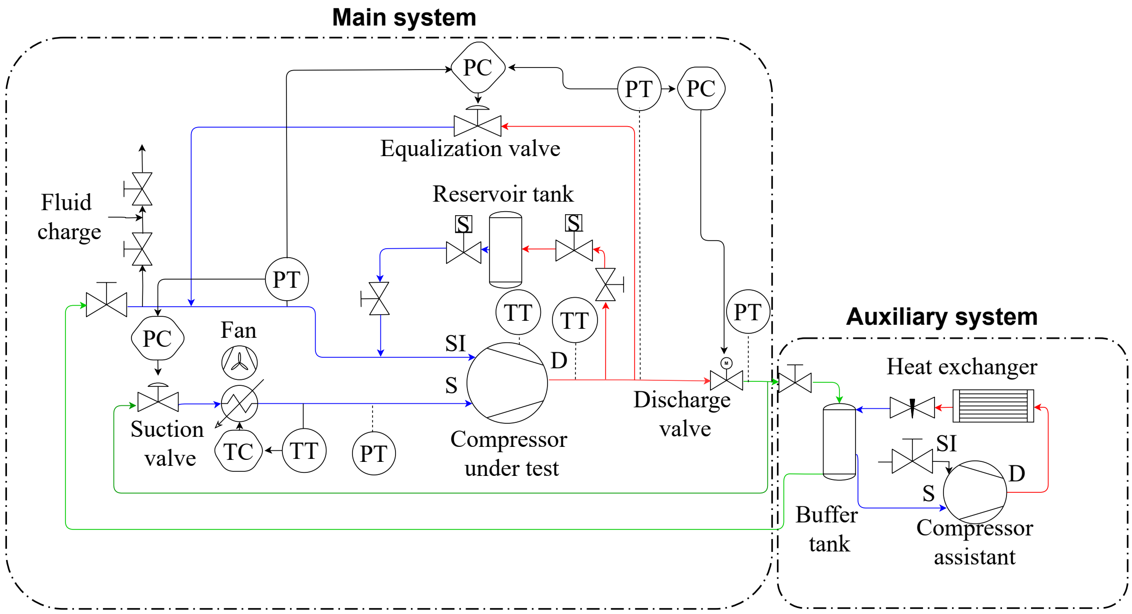

Figure 2: P&ID of the proposed automatic test rig.

3.2 Auxiliary system

In the operating envelope, there are 76 points to be achieved, each one of them representing ordered pairs of T_{ev} and T_{co}. During the continuous operation of the compressor under test subjected to different conditions, the average refrigerant fluid temperature tends to increase, which affects the ability of the test rig to reach low p_{d} and consequently low T_{co} conditions.

As a solution, the use of an auxiliary system to support the main circuit is proposed, being it responsible for cooling the refrigerant fluid through a heat exchanger placed in the buffer tank, thus preventing system overheating. For this purpose, a compressor was added to the test rig, composing a separated refrigeration system operating with R-600 (isobutane) refrigerant fluid in a conventional refrigeration cycle.

4. TEST RIG CONTROL

Due to the addition of new actuators, the control of the test rig in the configuration presented in Section 3 becomes significantly more complex than in its initial configuration. In order to accommodate these changes, the control framework of the test rig, which previously consisted of two independent feedback controllers C_{s} and C_{d} for controlling the suction (u_{s}) and discharge (u_{d}) valve openings, respectively, was updated to the one presented in Figure 3, with controllers C_{e} and C_{aux} added for controlling the equalization valve opening (u_{e}) and the on/off state of the auxiliary system (u_{aux}). In the framework diagram, r_{ev}, r_{co}, r_{s}, and r_{d} are the setpoints for the evaporating temperature, condensing temperature, suction pressure, and discharge pressure, respectively, and G is the test rig plant, which can be expressed as:

$$y_{\mathrm{s}} = f(u_{\mathrm{s}}, u_{\mathrm{d}}, u_{\mathrm{eq}}, u_{\mathrm{aux}}),$$
$$y_{\mathrm{d}} = g(u_{\mathrm{s}}, u_{\mathrm{d}}, u_{\mathrm{eq}}, u_{\mathrm{aux}}),$$

with u_{e} and u_{aux} being the opening of the equalization valve and the state of the auxiliary system, with time dependency omitted for the sake of clarity, and $f(\cdot)$ and $g(\cdot)$ being two nonlinear and unknown functions. The system as a whole presents nonlinear dynamics, but some factors can be considered when designing the controllers:

- dynamics related to u_{s} are faster than those related to other manipulated variables, and y_{s} is more affected by changes to u_{s} than by other variables;
- y_{d} is more affected by changes to u_{d} than by other variables;
- dynamics related to u_{aux} are over 100 times slower those related to other manipulated variables;

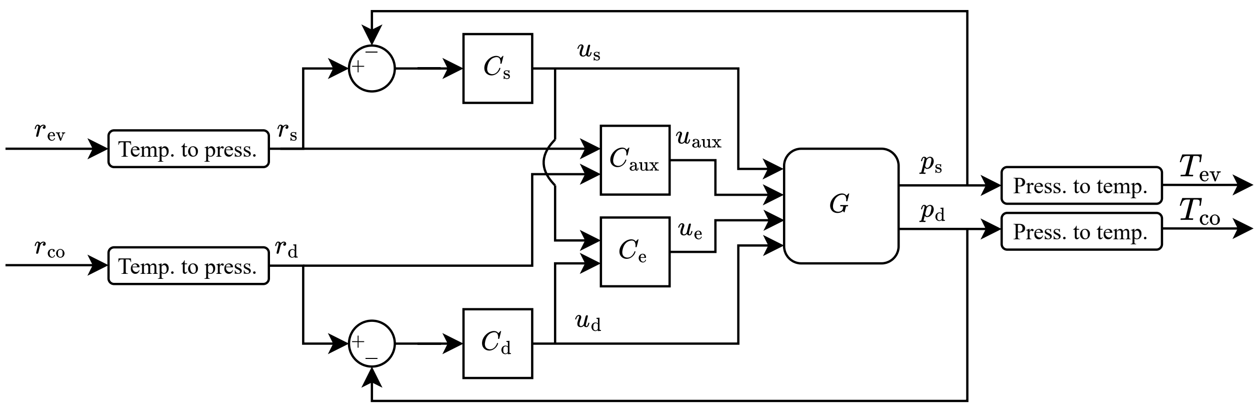

Figure 3: Block diagram of the test rig control framework.

- for fixed u_e and u_{aux} conditions, the dynamics of f and g are approximately linear;
- due to the buffer tank in the auxiliary system, the effect of u_d over y_s is negligible.

Considering the system behavior and the possible simplifications, controllers C_d and C_s were designed as classical SISO feedback controllers with sampling periods of 1.5 s and 300 ms, respectively. For these controllers, a simplified first-order plant was identified, considering the approximately linear dynamics for closed equalization valve ($u_e = 0$) and auxiliary system turned off ($u_{aux} = 0$). The identified system can be expressed as:

$$\begin{bmatrix} y_s \\ y_d \end{bmatrix} = \begin{bmatrix} \dfrac{4.00}{1.75s+1} & 0 \\ \dfrac{37.40}{4.10s+1} & \dfrac{32.30}{24.80s+1} \end{bmatrix} \begin{bmatrix} u_s \\ u_d \end{bmatrix}. \tag{1}$$

Since the dynamics of u_s are faster than the dynamics of u_d, the faster loop keeps its associated variable close to its reference while the other loop adjusts its process variable. As a consequence, the dynamics in the secondary diagonal are mitigated, and interaction between the loops is almost eliminated (Skogestad & Postlethwaite, 2005).

For the first-order models employed, PI controllers were considered to ensure zero steady-state error tracking, with a tolerance of ±1 °C adopted for each condition. For tuning controller C_s, the following specifications were considered: overshoot less than 10% and settling time equal to 4.2 s, 20% less than in open loop. The transfer function of the controller designed for suction pressure is presented in equation 2, with gains $K_c = 3.78$ and $T_i = 1$ s:

$$C_s(s) = \frac{3.78(s+1)}{s}. \tag{2}$$

As for controller C_d, the PI controller was specified with overshoot less than 20% and the closed-loop system time constant is 24.8 s, three times faster than in open loop. It is observed that, even by speeding up the response, the closed-loop settling time is significantly slower than that observed for the suction pressure. Again, this is done to prevent coupling between the outputs. The transfer function of the controller designed for discharge pressure is:

$$C_s(s) = \frac{3.50(3.33s+1)}{3.33s}, \tag{3}$$

with gains $K_c = 3.50$ and $T_i = 3.33$ s. For C_e, some preliminary identification tests were performed to evaluate the dynamic characteristics of the test rig plant in four different conditions for the suction and discharge valves. The results of these tests are presented in Figure 4.

For each condition, a series of additive steps was applied to the equalization valve opening. The identification tests showed that the relationship between the equalization valve opening and the temperatures is strongly nonlinear, varying greatly in order, gain, and dead zone even in a single suction and discharge valve condition. Due to the high complexity of the system, and to the fact that most reference points are reachable at fully closed equalization (Section 2), C_e was

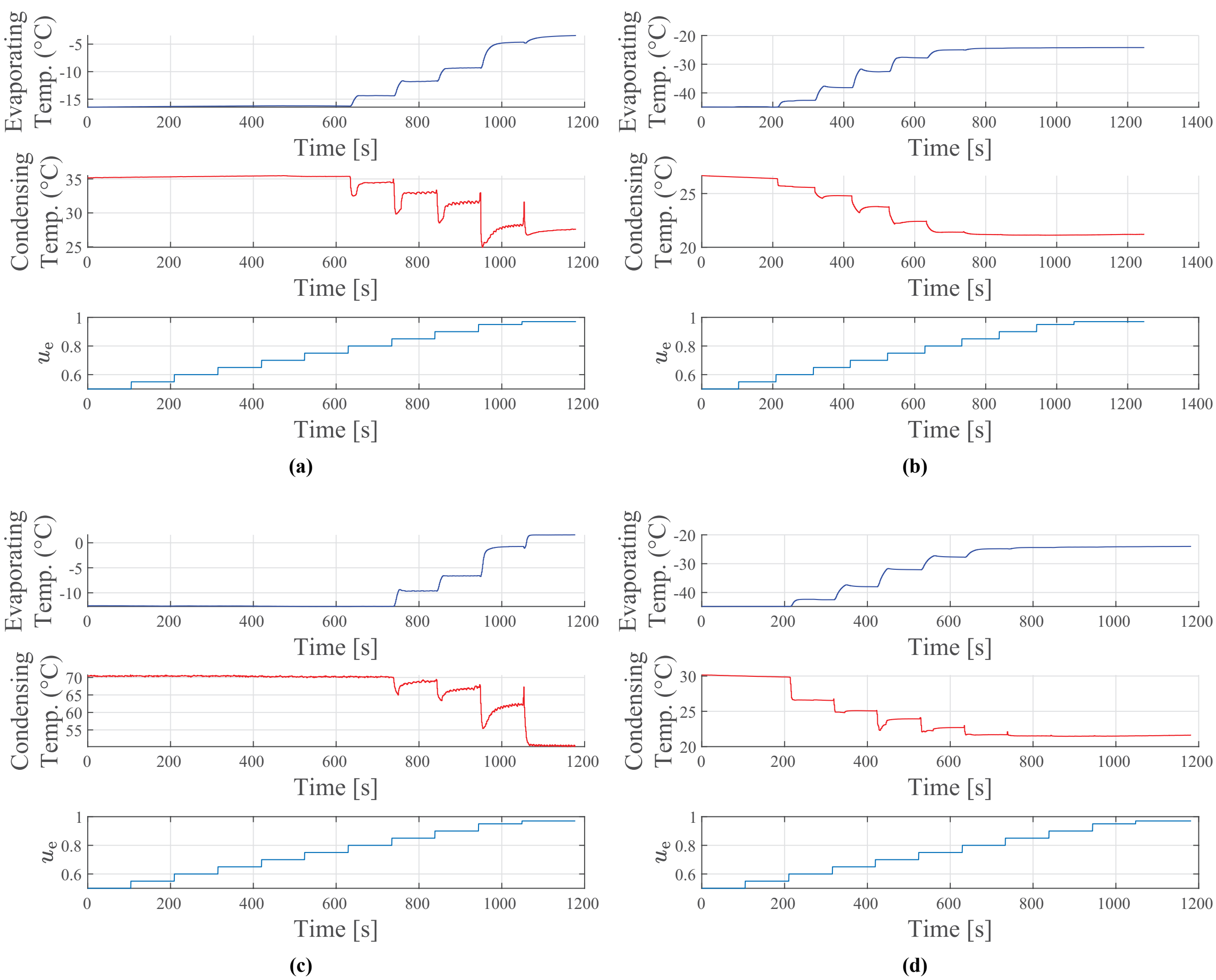

Figure 4: Equalization valve identification tests: (a) u_s = 1, u_d = 1; (b) u_s = 0, u_d = 1; (c) u_s = 1, u_d = 0; (d) u_s = 0, u_d = 0.

designed not for reference tracking, as the controllers C_s and C_d, but for increasing condition reachability only when necessary. For this purpose, the controller was designed as a discrete supervisory system, illustrated as a timed automaton in Figure 5. It operates at a lower sampling period (30 s) than C_s and C_d based on the state of other manipulated variables.

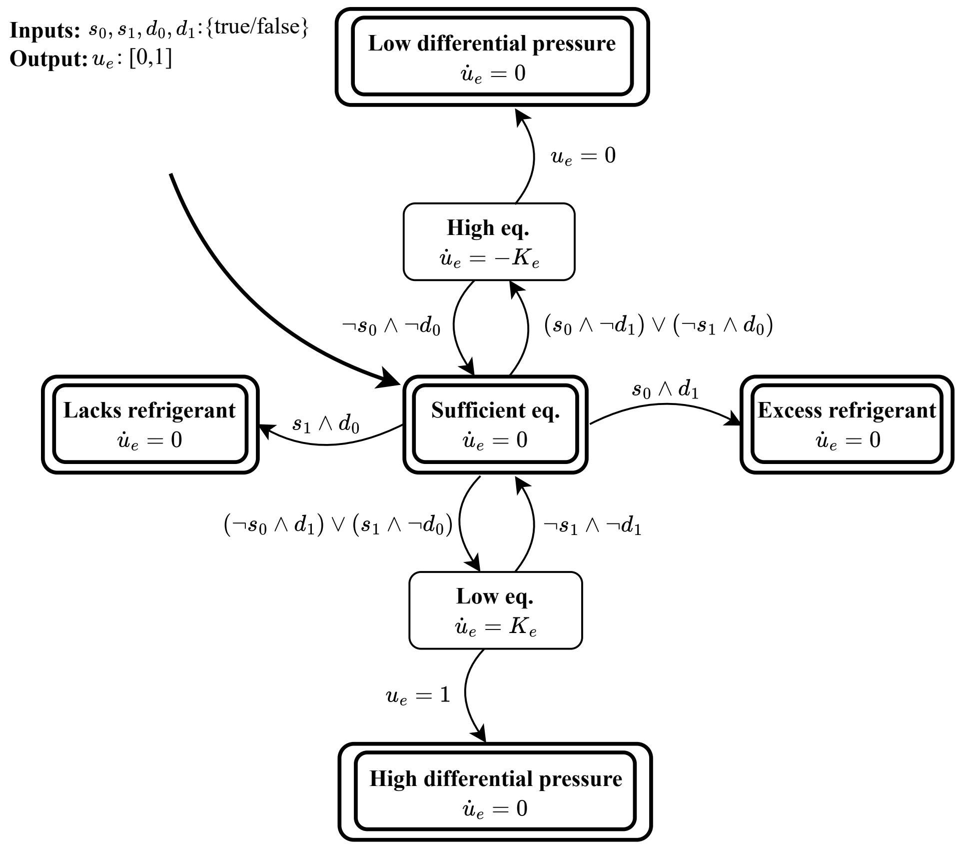

Figure 5: Discrete supervisory controller logic for the equalization valve (time dependency omitted for the sake of clarity).

The rules for transitioning between controller states are based on the fully opened and fully closed state of the other valves, with s_0 and d_0 being the fully closed state of the suction and discharge valves, respectively, and the fully opened state of these valves being represented as s_1 and d_1. The only adjustable parameter of the controller is the gain K_e, with higher gains speeding the valve opening, but possibly leading to oscillatory behaviour. The supervisory system also accounts for 4 terminal states in which no control action can further increase the reachability of the system: the states of too much or too little refrigerant fluid, in which the reservoir tank must be filled or emptied in order to reach the setpoint; the low differential pressure state, in which the compressor is not able to provide enough work to decrease suction and increase discharge pressures; and the high differential pressure state, in which the opening of suction and equalization valves is not enough to bring closer the suction and discharge pressures.

Last, the controller C_aux was designed to activate the auxiliary system on low setpoints. The controller framework consists of a simple decision tree based on r_ev and r_co, such that if either one of the setpoints is lower than its predefined threshold, the auxiliary system is turned on. This threshold is defined heuristically according to the lower reachable conditions of the test rig without the auxiliary system.

The results of the implementation of the whole control framework in the proposed test rig are presented in Section 5.

5. RESULTS

This section describes the results obtained after implementing the proposed test rig (Section 3) and controllers (Section 4). To define the thresholds for the controller C_{aux}, the system was first evaluated with the auxiliary system turned off and the results are presented in Figure 6. The condition map presented in Figure 6a confirms the effectiveness of using the equalization valve. With this additional valve and the proposed control strategy, it was possible to achieve 90% of the operating envelope, in contrast to the initially achievable conditions, presented in Figure 1b (Section 2), which amounted to only 72% of the necessary conditions. The equalization valve control action u_e is presented in Figure 6b. A high u_e is mostly seen on higher evaporating temperatures and some lower condensing temperatures. The explanation for the high control action when dealing with the operating conditions on the right side of the condition map is the physical limitation of the suction valve opening, compensated by the controller with a higher equalization output. A suction valve with a larger flow coefficient would probably result in a smaller equalization opening for those conditions.

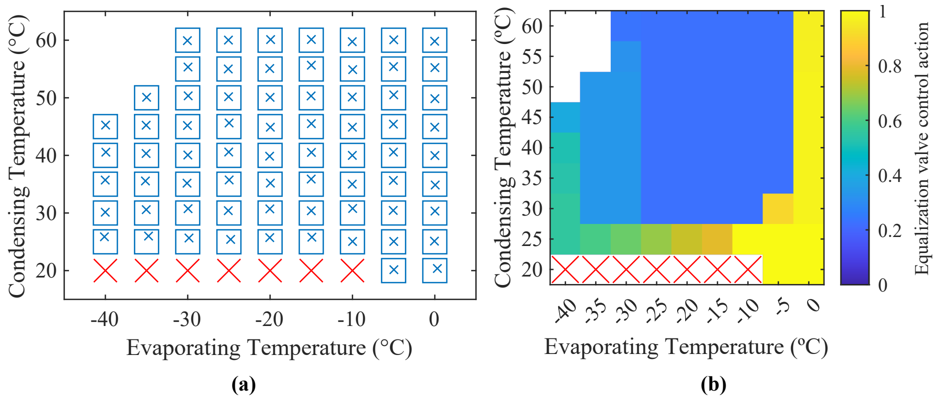

Figure 6: System behavior with controlled equalization valve and no auxiliary system. Condition notations are the same as in Figure 1: (a) Achievable conditions; (b) Equalization valve control action.

For a final evaluation of the complete control system, the auxiliary system supervisory control was implemented, acting when the condensing temperature setpoint was set at 20 °C. The results obtained in this case in which all the proposed system controllers are used are presented in Figure 7.

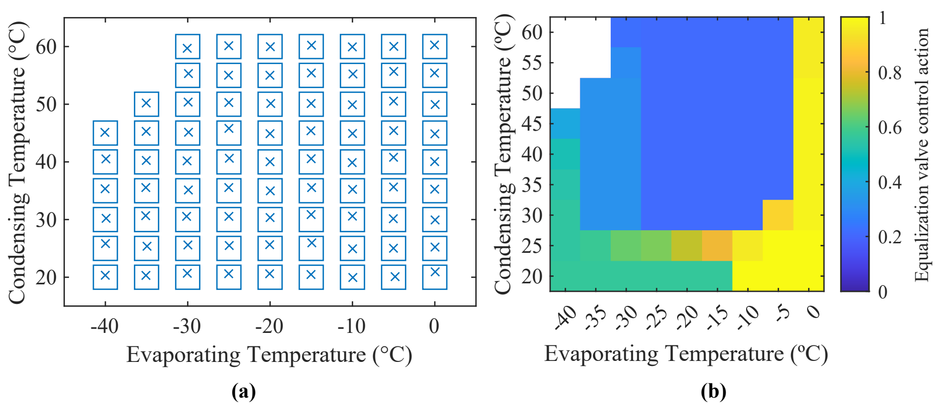

Figure 7: System behavior with controlled equalization valve along with the auxiliary system. Condition notations are the same as in Figure 1: (a) Achievable conditions; (b) Equalization valve control action.

By employing both the equalization valve control and the auxiliary system, it was possible to achieve 100% of the operating envelope conditions, as presented in Figure 7a. The equalization valve control action is presented in Figure 7b, where it is possible to observe that it presented a similar behavior to that observed in Figure 6b. For the conditions not achieved previously in Figure 6a, the ones with higher evaporating temperature were reached with fully opened equalization valve, while the other conditions only needed about half of the control action due to the refrigerant fluid cooling in the buffer tank, which resulted in lower discharge pressures and consequently lower condensing temperatures.

6. CONCLUSION

In order to evaluate the operation of a model of compressor for a wide range of operating conditions inside its operating envelope, a test rig was built upon a previous version. Although the modifications enabled a substantial increase in the testing capabilities, they complicated the control problem to be solved. This paper proposed a control method for the system based on a discrete supervisory controller associated with PID loops. As a result, it was possible to keep the controller tuning very simple, since it is just necessary to tune SISO controllers for the MIMO system.

To broaden the range of achievable conditions, a set of auxiliary elements were employed on the test rig. To reach conditions where the suction and discharge pressures are close to each other, a control valve was added between suction inlet and discharge outlet, enabling the fluid bypass to approximate both pressures. Additionally, an auxiliary circuit responsible for cooling the refrigerant fluid was installed, mitigating the effects of overall refrigerant fluid temperature increase due to continuous compressor operation. With the auxiliary circuit, it was possible to achieve lower discharge pressure conditions even after several hours of continuous operation. A reservoir tank was also added to enable the insertion and removal of refrigerant fluid.

After the modifications in the test rig, the dynamics of the variables of interest were studied and controlling strategies were proposed. For imposing the compressor operating condition in terms of pressures and temperatures, a feedback control strategy was tuned for the suction and discharge pressures. The equalization valve and refrigerant fluid introduction strategy were devised using a discrete supervisory control approach, which relied on the controller outputs from both the suction and discharge closed-loop controllers. The auxiliary system activation was also based on a discrete supervisory control approach that considered in which region of the operating envelope the reference is situated. By combining the proposed strategies, the test rig was able to automatically impose any condition inside the operating envelope, while keeping the compressor operating around the desired references within a tolerance of less than 1 °C.

Future work related to this topic will explore the use of advanced control techniques when dealing with the suction, discharge, and equalization control.

NOMENCLATURE

S	Suction inlet	
SI	Service inlet	
D	Discharge outlet	
C_{aux}	Auxiliary refrigeration system controller	
C_{d}	Discharge valve controller	
C_{e}	Equalization valve controller	
C_{s}	Suction valve controller	
G	Test rig plant	
d_0	Discharge valve fully closed	(true/false)
d_1	Discharge valve fully opened	(true/false)
K_{c}	Controller gain	
K_{e}	Equalization controller gain	
p_{d}	Discharge pressure	(kPa)
p_{d}	Suction pressure	(kPa)
r_{co}	Condensing temperature setpoint	(°C)
r_{d}	Discharge pressure setpoint	(kPa)
r_{ev}	Evaporating temperature setpoint	(°C)
r_{s}	Suction pressure setpoint	(kPa)
s_0	Suction valve fully closed	(true/false)
s_1	Suction valve fully opened	(true/false)

T_{co}	Condensing temperature	(°C)
T_{ev}	Evaporating temperature	(°C)
T_i	Integration time	(s)
u_{aux}	Auxiliary refrigeration system control variable	(on/off)
u_d	Discharge valve control variable	(-)
u_e	Equalization valve control variable	(-)
u_s	Suction valve control variable	(-)

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ACKNOWLEDGMENT

This work was supported in part by the Brazilian National Agency of Petroleum, Natural Gas and Biofuels (ANP) under the Human Resource Training Program (PRH), in part by the Brazilian National Council for Scientific and Technological Development (CNPq) under Grant 315546/2021-2, in part by the Coordenação de Aperfeiçoamento de Pessoal de Nível Superior – Brasil (CAPES) – Finance Code 001, and in part by Nidec Global Appliance.