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A numerical study of the R744 primary cooling system for ATLAS and CMS LHC detectors

Étude numérique du système de refroidissement primaire au R744 pour les détecteurs ATLAS et CMS LHC

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ABSTRACT

A R744 (CO_2) refrigeration system has been designed to cool down the Large Hadron Collider (LHC) silicon detectors ATLAS and CMS, located at CERN, Switzerland. The silicon detectors are subjected to high radiation levels. The system is composed of a pri- mary CO_2 trans-critical booster vapor compression loop operated with piston compressors, and an oil-free liquid pumped loop on the evaporation side, to preserve the detectors. To ensure the system's reliability, the cooling facility is designed to operate under a parallel operation mode of several modular 70 kW plant units providing evaporation temperature as low as -53 °C. This layout, is also useful in case of components failure and maintenance. A numerical model is developed using a dynamic simulation software Dymola that is based on the open source Modelica modelling language. The simulation results are proven on a first demonstration plant (System A) experimentally to explore the systems control logic and to validate the reliability of the system before it is built on the detectors side. In this paper the models development is explained and the results of the experimental validation of the numerical model are shown.

1. Introduction

"The Large Hadron Collider (LHC) at CERN[1] is the world's largest and most powerful particle accelerator" (CERN, 2022). It is situated at 100 m underground and consists of two separate beam pipes, scaling 27 km each where high-energy particle beams are circulated at opposite directions, close to the speed of light, before collision. These particle beams are guided by a strong magnetic field maintained by super-conducting electromagnets (CERN, 2022).

The collision of the beams takes place at four locations of the accelerator ring, where the particle detectors ATLAS, CMS, ALICE and LHCb are located. The main purpose of the accelerator and its experiments is to explore and understand the first instants of the universe creation. "The "A Toroidal LHC ApparatuS" (ATLAS) and "Compact Muon Solenoid" (CMS) experi- ments are general-purpose detectors, designed to see a wide range of particles produced in LHC collisions and to search for new phenomena, including the Higgs boson, supersymmetry and extra dimensions" (Rocca, 2014).

1.1. History of detectors thermal management

Since 2000, CO_2 two-phase evaporative cooling has been successfully used for the thermal man- agement of tracking detectors (Tropea, 2014). First, the Alpha Magnetic Spectrometer (AMS-02) Tracker and LHCb vertex detector (Velo) were upgraded with R744 refrigeration technology to cool down their silicon detectors to avoid thermal runaway. Both CO_2 systems, for AMS-02 and LHCb Velo, use the 2-Phase

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[1] European Organization for Nuclear Research; an intergovernmental organization with 23 Member States, located in Geneva, Switzerland

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Acronyms

LHC	Large Hadron COllider
CERN	European Organization for Nuclear Research A
ATLAS	Toroidal LHC ApparatuS
CMS	Compact Muon Solenoid
CO2 LP	Carbon Dioxide Low
HP MP	Pressure
CS	High Pressure
R744	Compressor Slice
CO2	Carbon Dioxide
ODP	Ozone Depletion Potential Global
GWP	Warming Potential
2PACL	2-Phase Accumulator Controlled Loop Hot
HGBP	Gas Bypass Valve
OD	Opening Degree Continuous
CPC	Process Control
DEMO	Demonstration Plant
SIM	System Information Manager

Accumulator Controlled Loop (2PACL) design developed at Nikhef in the late nineties (Verlaat, 2009). The 2-phase accumulator is used in capillary pumped loops to control the evaporator pressure with the benefit for High Energy Physics (HEP) detectors to have low material costs by e. g. reusing already insulated transfer pipes (in case of CMS) and to be very stable. The 2PACL method "uses only passive compo- nents inside the detector volume, and it is easy to control" (Verlaat,2009). This means, most of the active elements can be placed far from the detector in an accessible place using coaxial pipelines. This is an important argument since the particle detectors are located in highly irradiated areas. In 1999, the LHCb Velo R744 micro-channel refrigeration system was assumed to operate in a temperature range between −25 °C and 10 °C [O. Postma (1999)]. It is a 150 W system (Verlaat,2009). −20 °C was confirmed in 2014 for the LHCb VELO upgrade (de Aguiar Francisco, 2015), which was successfully scaled up to a 1.5 kW system (Verlaat, 2009). Then, the ATLAS Insertable B-Layer (IBL) detector and the CMS Pixel phase 1 (CMS Pix-Ph1) tracker (Zwalinski, 2015) have followed. The ATLAS IBL detector operates at −35 °C and needs two 3.3 kW plants (Zwalinski, 2015) while the CMS Pix-Ph1 tracker requires two 15 kW plants (Marzoa, 2016). The successful operation of these cooling systems has inspired the HEP community to consider R744 cooling as a credible candidate for future tracking detectors" (Verlaat,2009) instead of conventionally used hydrofluorocarbons (HFCs).

1.2. F-gas regulations and R744 as suitable alternative

Due to the Montreal Protocol (1987) and the London and Copenhagen amendments (1990, 1992), alternative refrigerants to the synthetic ones which are characterized by high ozone-depletion (ODP) and high global warming potential (GWP) are needed to reduce the usage of fluorinated gasses. The use of carbon dioxide (R744) as a natural refrigerant represents a number of bene- fits amongst them: non- flammable, nontoxic, and characterized with excellent thermodynamic and transport properties at low temperatures (Bansal, 2012). R744 is also characterized by a large density in the liquid phase, allowing the use of small components and the inclusion of compact systems particularly in confined areas. Although R744 cooling technologies are very popular in commercial and industrial refrigeration applications, its strong radiation hardness, non-conductive and non-corrosive properties make this working fluid a suitable refrigerant can- didate for the cooling of experiments in high-energy physics (Paola Tropea, 2014).

R744 system compactness is convenient to be integrated in a limited area 100 m underground, where access is restricted, just like cooling the detector. It is possible to place the evaporators underground and bridging the height difference between underground and surface within a single system. This way, the produced heat can be rejected on surface.

1.3. Future plans of detectors thermal management

In 2025, the LHC will undergo further exploration studies. It will be shut down to have major upgrades on the accelerator and its experiments (Pater, 2019). A complete replacement of the tracking detectors, together with trigger and calorimeter upgrades of the ATLAS and CMS experiment, with all new silicon-based detectors is planned. The new system will need to handle a detector occupancy (irradiation) of up to 10 times higher than before, which means lower evaporation temperature of the refrigeration system of −40 °C (Rocca, 2014). The CMS experiment currently operates with a C_4F_{10}-water system, while the ATLAS inner detector is cooled by C_3F_8-water (Verlaat, 2009). Both refrigeration loops will be replaced by a primary CO_2 two-stage compression system plus a secondary CO_2 refrigerant cooling system: the oil-free, pumped low-pressure 2PACL on the evaporation side, and a primary, trans-critical cycle on the high pressure side operating with piston compressors. These two R744 loops are interfaced by a heat exchanger acting like a condenser for the 2PACL and like an evaporator on the primary cycle. This way, it is assured that no drop of oil ever passes the pipelines, which would degrade under high radiation. This furthermore means, that the already existing concentric transfer pipes (green, horizontal line underground) of the CMS 2PACL can be reused and recycled respectively (ATLAS) (Verlaat, 2009).

The 2PACL loop will be built at the same level as the detectors. It will absorb the heat from the detectors and keep the temperature below −40 °C to avoid thermal runaway. The heated fluid (2-phase flow) is then pre-cooled, condensed and sub-cooled in a heat exchanger (primary evaporator). The sub-cooled liquid CO_2 is pumped through a 100 m long, concentric transfer line, pre-cooling the returning 2-phase flow. While the evaporators are situated in the experimental pit caverns, the active components such as the gas cooler are placed in a separate, shielded service cavern connected by the concentric transfer lines. Furthermore, 2PACL is connected to a CO_2 liquid accumulator on surface with a controlled pressure level. The primary 2-stage system is responsible for condensing vapor at the setpoint temperature in the return line of the 2PACL system. The amount of circulated cold liquid is controlled by the 2PACL system as explained by Verlaat (2019). Long vertical pipelines (about 86 m) connect the two units on surface with the pumped loop underground. This configuration leads to evaporation temperatures of the primary cycle to below −50 °C.

ATLAS and CMS are expected to dissipate several hundred kilowatts of heat (up to 550 kW) (Pardinas, 2020). Scaling up a refrigeration system to these large cooling powers is challenging. As a result, it is planned to be built to an intermediate power system of about 50–70 kW and use several identical plants in parallel. "This approach is easier to prototype and has advantages for operation" as well as maintenance reasons by swapping or spare cooling units (Verlaat, 2009).

1.4. Primary refrigeration cycle project

A series of prototypes has been built and tested before the final system will be launched. The reason is the necessity of the cooling plant to be 100% stable and reliable because the detectors are worth several million euros. Moreover, a smooth integration of the final system is targeted. A first demonstration plant (System A) is built to prove each component of the system regarding its functionality, to test the reliability of the control logic and to investigate the high power demand (up to 50 kW per unit) at a very low operational temperature (below −50° More details about system A is given in 2.1. The purpose of the second chiller prototype (System B) is to evaluate the behavior of the refrigerant inside the long vertical and horizontal pipelines as well as the oil management. A third demonstration plant will be used as production-ready unit to support the commissioning of the CMS Inner detector. The final ATLAS and CMS R744 primary units will correspond to twenty single units. In parallel, simulation models for each prototype will be developed to validate the experimental results retrieved after testing the systems.

In this paper, the configuration of the primary refrigeration system (Section 2) and, particularly, system A is explained (Section 2.1) in detail. Furthermore, the control logic of system A is presented in section Section 2.2. Then, the development of the simulation model of system A is written in Section 3.1 as well as its control logic in Section 3.1.2. Some results of the simulation model using averaged input parameter can be found in Section 4.1, using experimental input parameter in Section 4.2 and experimental validation in Section 4.3.

2. The primary refrigeration cycle

The main function of the primary refrigeration system is to extract the heat from the 2PACL cycle, which will cool down the detectors of the ATLAS and CMS experiments of the Large Hadron Collider (LHC) at CERN, Geneva. To avoid thermal runaway, the detectors must be kept below -35 °C. The primary loop has been designed to provide a stable and reliable temperature of -53 °C (Verlaat, 2019) under different operating conditions (during one operational cycle: low & high load, enabling & disabling of compressor slices) all year around to ensure fully compliance with the requirements of the detectors and to assure the pumps of the second cooling cycle (2PACL) a sub-cooling of 10 K.

R744 has a triple point at -56.5° and 5.19 bar, thus, being together with its favorable properties described in 1, is a suitable refrigerant for the primary refrigeration loop.

In Fig. 1, the concept design is shown consisting of parallel compressor slices (green boxes), the common equipment unit (top grey box) and the underground plant-box (bottom grey box). Fig. 2 shows the corresponding R744 ph-diagram. The values in the thermodynamic cycle are illustrative and may vary depending on the performance of compressors and gas coolers.

The Common Equipment unit includes the water-cooled gas cooler (19–1), which decreases the gas temperature from about 35 °C down to 30 °C taking care of the final temperature stabilization, the high-pressure (HP) valve (1–2, CV01), which is responsible for maintaining the trans-critical pressure of about 78.8 bar (pressure set-point is a function of gas cooler outlet temperature), and the liquid receiver (2–3) with its function to provide cooling for the 2PACL loop (underground plant box) as well as on the inlet of the LP and HP compressors to improve efficiency (Barroca, 2020).

Evaporators (6–7), expansion devices (5–6, CV06) and post-heat exchanger (7–8) are units of the underground plant box.

The required cooling load is covered by several evaporators (6–7) located underground in the plant box, which operate in parallel. Their internal post heat exchangers use the supplied ambient temperature liquid (3–4) from the surface to post-heat the cold gas (7) of the evaporator to ambient temperature conditions. Thus, supplied liquid (3) and return gas lines (8) do not need insulation, saving cost and integration space. The transport of warm gas reduces the losses on the return pressure drop by static height due to the low density of the warm gas. The evaporators capacity is controlled by expansion valves (CV06) regulating the super-heating at the evaporator outlet, while keeping the saturation temperature stable at −53°

With the low critical temperature of CO_2 (31.1 °(C)), the vapour-compression system is designed for trans-critical cycles, where the temperature of the raw primary water during summer can be above 31.1° (Barroca, 2020). Having a high critical pressure of 78.76 bar and a low evapo- ration temperature of 6 bar means a pressure ratio between 10 and 15. This requires a configuration with two stages of compression.

The compressor slices of the 2-stage, inter-cooled compression system consists of three heat exchanger, two compressors, several control valves (CVxx) and on/off solenoid valves (EVxx). The warm up-streaming gas is pre-cooled by mixing a portion of the pre-cooled high-pressure discharge gas (9–10) of the LP compressor (11–12) and a pre-cooler heat exchanger (10–11) before being compressed by a LP compressor to reduce the compression losses. The hot gas is cooled by a portion of the high-pressure discharge gas (12–13) of the HP compressor (17–18), a water-cooled interstage cooler (13–14) and two cold 2-phase

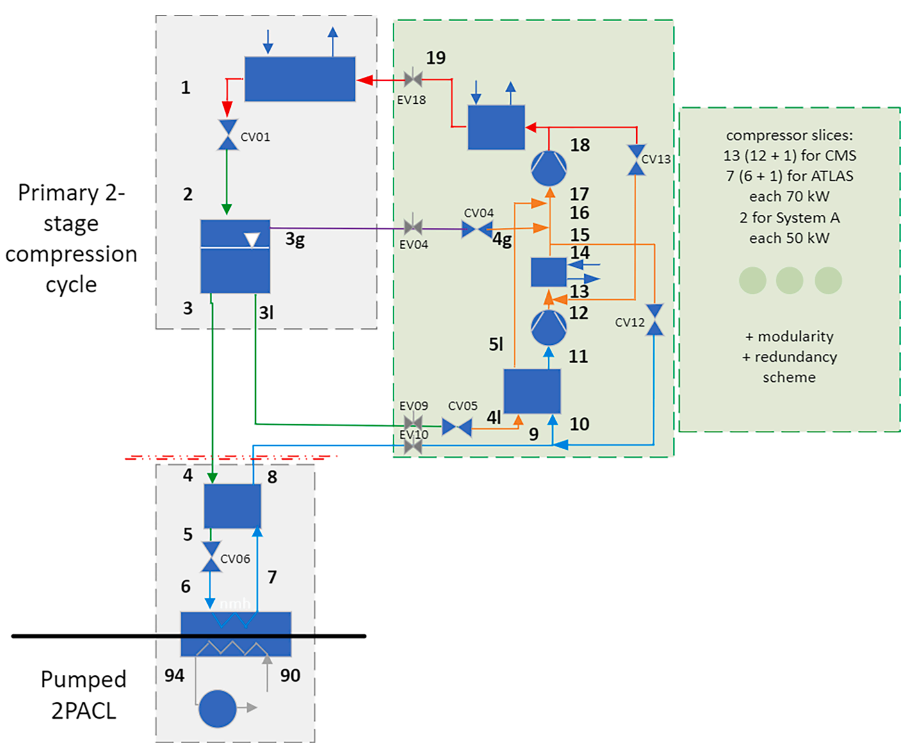

Fig. 1. Concept design of the primary refrigeration cycle having multiple compressor slices in parallel to achieve an adjustable cooling load. Each compressor slice is constructed the same.

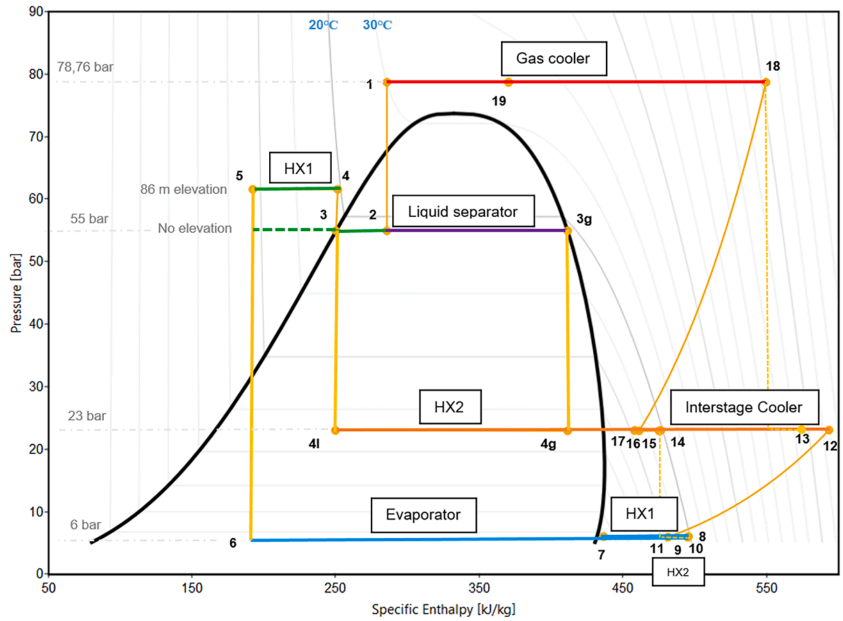

Fig. 2. Ph-diagram for the R744 trans-critical two-stage compression cycle.

flows (3–4l, 3 g-4 g) coming from the liquid separator (2–3). The HP compressor inlet temperature (17) is kept stable at about 5° and is maintained by CV05 to a superheat of 20 K. The suction pressures are maintained by the openings of CV12 and CV13, respectively, which is 6 bar for the LP compressor and 23 bar for the HP compressor. An air-cooled gas cooler (18–19) cools down the discharge gas of the HP compressor to a temperature of 35 °C, which depends on the oil separator efficiency as a function of temperature. Two oil separators arranged in series are located downstream of the HP compressor. The lubricant is collected in a reservoir connected to the oil separators. On demand, each compressor is able to recharge oil from this accumulator. In wintertime, the temperature in Geneva can reach −10 °C. Therefore, the rpm of the fan can be controlled and the air-cooled gas cooler can be bypassed in this refrigeration system. Operating the system sub-critical is an alternative, which was canceled due to the following reasons: Heat recovery is planned for the future, and currently energy efficiency is not a relevant milestone yet. All what counts for the detector is a reliable cooling. Furthermore, sub-critical operation must be limited to the corresponding 20 °C outlet temperature (2). Otherwise point 3 can not be reached and outside condensation along the pipe will occur as the wall temperature drops below the dew point of the air. The combination of the air-and water-cooled gas cooler makes the system independent of water failure and maintenance. The liquid receiver pressure is controlled by CV04 maintaining a set-point of 55 bar to have warm liquid of 20 °C entering the underground. All CV04s share the same PID controller output. The technical details of the first prototype built of the primary refrigeration system is described in Barroca (2021).

In the following table the naming of the components used for System A is listed:

Naming	Component
CV01	Gas cooler control valve
CV04	MP flash gas valve
CV05	MP liquid valve
CV06	Expansion valve
CV12	Low pressure hot gas by-pass valve
CV13	High pressure hot gas by-pass valve
EV04	Gas supply on/off valve
EV05	Liquid supply on/off valve
EV09	Gas return on/off valve
EV10	Suction gas on/off valve
EV18	Discharge gas on/off valve
2–3	Liquid separator
7–8	Post-heat exchanger
6–7	Evaporator
10–11	Pre-cooler heat exchanger
11–12	LP compressor
13–14	Water-cooled interstage cooler
17–18	HP compressor
18–19	Air-cooled gas cooler
19–1	Water-cooled gas cooler

(*continued on next column*)

(*continued*)

It is necessary to keep the cooling system reliable and stable in evaporation temperatures while coping with large load variations (0–100 %).

Furthermore, components failure or maintenance must be considered. Therefore, a modular primary cooling loop based on parallel operation of modular units is developed. The modular configuration is described in Fig. 3. It is a baseline scheme for systems operating with sev- eral compressor slices (green boxes) including one or more extra (back-up) slices (yellow box) following the redundancy scheme. For ATLAS seven compressors slices (green boxes), each one achieving 70 kW, are designed together with two extra slices (yellow box) of 70 kW. Like this, a cooling capacity of 630 kW can be achieved through the primary refrigeration system. For CMS eleven compressor slices plus two back-up slices are planned, all together achieving 910 kW.

In order to examine the control logic of the primary refrigeration system a first prototype, System A is built and tested. It consists of two compressor slices, each achieving 50 kW. System A is explained in the following section.

2.1. System A

To operate the R744-R744 system as the next generation of LHC detectors cooling plant un- der optimal conditions, experiments are first run at pilot scale. Its aspiration is to achieve a maximum cooling capacity of 100 kW consisting of two interconnected compressor slices (CS), each providing a maximum cooling capacity of 50 kW. One CS consists of a low-pressure (LP) and high-pressure (HP) compressor from the manufacturer DORIN, while the other slice con- sists of compressors from the manufacturer GEA Bock. To check its stability and reliability, the system is connected to a dummy load system. This means, no elevation difference accrues here. To provide 50 kW of cooling capacity

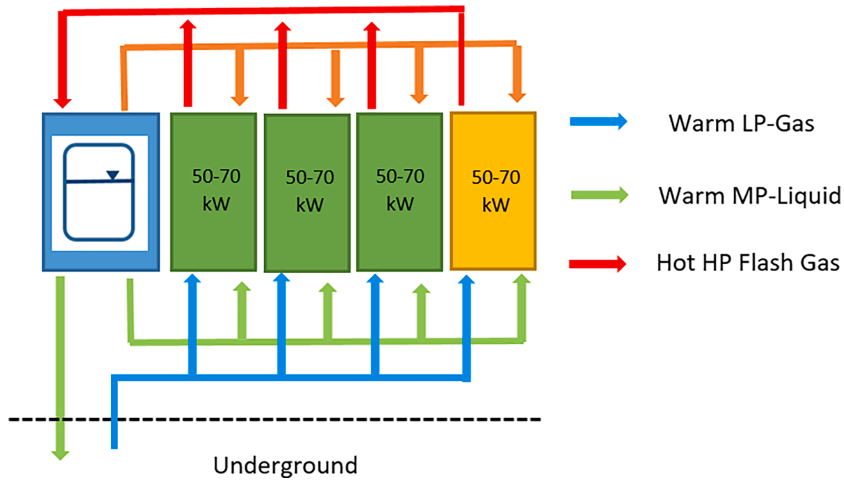

Fig. 3. Modularity scheme for systems with one or several compressor slices (green boxes) and $n + 1$ redundancy scheme (modified Figure of Barroca (2020)).

per slice, the LP and HP compressors need to be selected accordingly, with a maximum mass flow rate of 210 g/s for the LP and 364 g/s for the HP compressor. Each compressor slice has a target capacity of 50 kW at -53°, which is controlled by CV12.

Moreover, each slice works independently. Therefore, they can be activated depending on the required load, while the last activated slice is fine-controlling the capacity by the use of CV12. Configured like this, the system aims to operate stable from zero to full load. The control logic is explained in detail in Section 2.2.

2.2. Control logic

It is necessary to keep the detectors temperature stable during LHC experiments as well as in times of no operation. The system is designed to operate running at least one compressor slice (CS) with a maximum opening degree (OD) of the low-pressure (LP) bypass valve (CV12), while the pipeline (10) from the evaporator to the second compressor slice (CS2) is disconnected.

Hot gas by-pass valves (CV12) control the capacity of its CS by maintaining the suction pressure of the LP compressor. As soon as the

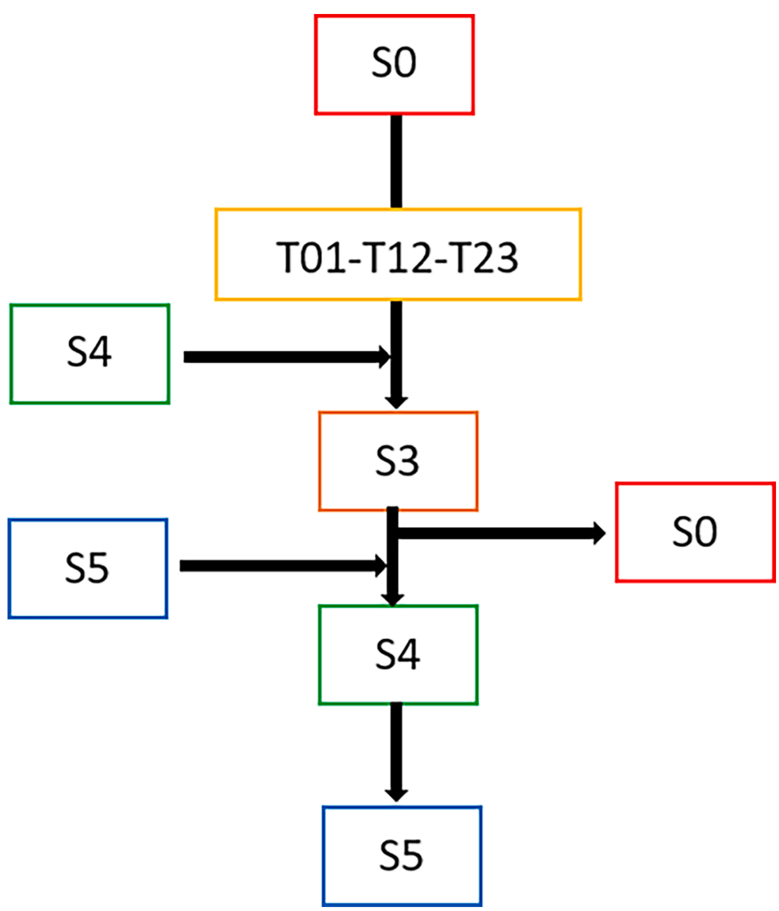

Fig. 4. Sequence of states of the control logic of system A.

detector is operated, the thermal load increases and the OD of CV12 decreases. Hitting the upper limit (45 % of maximum OD), CS2 gets ready to operate by receiving an enabling signal followed by a course of events inside the compressor slice to be activated. The sequence of the control logic is shown in Fig. 4 and explained in Table 1. To1-T12-T23: A three-way valve is opened connecting the gas cooler to the system. At a certain OD, the on/off valves EV04 and EV18 open, connecting the slice to the common system. If the suction pressure of the HP compressor is within a certain range, both compressors get the start signal, followed by the opening of EV05. CS2 now is in stand-by (S3) mode, ready to be activated.

Once the second CS is ready, the OD of CV12 is closed and the suction pressure of the LP compressor is within a certain range, CS2 is activated (S4), and the first slice gets in passive (S5) mode by opening EV10. Being in S5 means that the slice is still delivering cooling but the LP now is controlled by CV12 of CS2. The control of the slices must be very stable as it is operating close to the triple point of CO_2. The OD of the CV12 of CS2 minimizes as the cooling load increases. Opening/closing time of the on/off valves and three-way valve take 90 s resulting in a minimum enabling time of 261 s (about 4.4 mins) until the slice is in standby mode.

The experiments are finished, the detectors are shut-off. The required cooling load and thus, the total mass flow rate decrease at once. Once a minimum required mass flow rate of the LP compressor of CS2 is approached, the suction pressure decreases. Hitting the minimum pressure level of 5.98 bar, the second slice is deactivated and brought back to standby mode. If it stays there for a certain time (500 s), it gets disabled resulting in an one slice operation. CS1 is activated and controls the suction pressure of the system again. A detailed description of the setup of the primary refrigeration system is written in Verlaat (2019) and the logic is explained in detail in the Functional Analysis For Continuous Process Control (CPC) (Hulek, 2021).

3. System modeling

In this section, the development of a dynamic model of System A using Dymola software is performed. The simulation results were obtained for the performance assessment of the demon- stration plant System A.

Dymola is a tool for multi-engineering modelling and simulation of integrated and complex systems based on the Modelica language for systems' design (Tegethoff, 1999). Being component-oriented, one of the major advantages of using Dymola is to graphically drag and drop predefined models to simulate real-time cyber-physical systems by using a wide range of its available libraries. Dymola is also flexible, in the sense that it allows the user to extend the component equations in order to fulfil specific requirements.

TIL suite, the refrigerant and components library connected to the standard library, is obtained from TLK-Thermo GmbH. It is used for steady-state or transient simulations of thermodynamic systems (heat pumps, refrigeration, cooling and heating) and consists of three main sub-libraries. The add-on library TIL3.11.0 includes the fluid property library TILMedia3.11.0 including a wide range of refrigerants and the TIL-FileReader used to import external data such as ambient temperatures and heating loads, and DaVE for good visualization of the thermal systems.

Table 1
Step description of enabling second compressor slice.

S0 - OFF	All actuators are in their **safety state**
S3 - Standby	Operates with **supply gas valve closed** to be ready for start operation in Active step as quick as possible.
S4 - Active	Operates with **active capacity control using the hot gas by-pass** and compressor speed. Slice operating in this step controls the capacity of whole system.
S5 - Passive	Operates with the **LP hotgas bypass valve closed**, delivering maximum of cooling power, operates always in parallel with an active slice.
T01-T12-T23	1. Air cooled gas cooler three-way valve open > CVgc.value 2. Flash gas supply valve open / Discharge gas valve open If MP compressor suction pressure > MPcs.value 3. Signal to HP compressor start If LP compressor suction pressure < MPcs.value 4. Signal to LP compressor start / Signal to Liquid supply valve open For time 1

3.1. Modelica model

3.1.1. Model description

Fig. 5 represents the advanced system model of System A using Dymola. The model consists of three main units: the common equipment, the compressor slices and the cold box, namely the 2PACL. "Vertical Pipeline" is the model describing the height difference from the surface to the cavern, which is zero for System A. The "cold box, 2PACL" represents the module with the input parameters mass flow rate and super-heat (set-point 5 K) of the dummy. Each equipment used in the simulation model is configured with same design parameters as the equipment used in the real system: gas coolers, heat exchangers, compressors with the right sizes and coefficients,…). Furthermore, the ambient air (TT2194) and water temperatures (TT2002 and TT2174) as input parameters are shown, as well as the implemented logic of the system: three orange boxes on the bottom right, namely called "common", "CS2 standby" and "CS2 active". The common logic box controls swapping mechanism of the compressor slices giving the enabling & disabling and activation & shut-down signals. The CS2 standby box controls the components inside CS2 giving signals to each component based on the pre-defined sequence described in 2.2. The CS2 active box gives the active and passive signal to CS2 and CS1 depending on the thermal load of the system, which is a function of the previous described boxes.

2PACL is designed to operate at a target pressure for future detectors as low as 8.5 bar (Verlaat, 2019). The pumps of 2PACL need 10 K of sub-cooling bringing the liquid temperatures below−50° (6.8 bar). Thus, the suction pressure of 6 bar is the most important parameter to design carefully having a margin of maximum 0.8 bar. Indeed, a rapid increase in the suction pressure of the LP compressor and hence the evaporation pressure of the primary refrigeration cycle, may affect the 2PACL cycle. When the pressure drops, solid CO_2 may form inside the evaporator limiting the efficiency of the heat transfer. In all cases, the system is designed in a way that solid CO_2 particles cannot be admitted to the LP compressors. A receiver pressure of 55 bar ensures a temperature of 20 °C in the vertical pipelines connecting the surface with the underground caverns. No insulation is applied as the soil temperature is stable at 20° A varying temperature here would mean heat loss inside the pipelines. The trans-critical pressure is calculated using Eq. (1): if $u \leq 38$ as a function of the gas cooler outlet temperature (u in °C), having a pressure at about 76.4 bar. This pressure level is supposed to achieve a good overall efficiency of the primary refrigeration system (Hulek, 2021).

$$y = \left(\left(0.05546u^2\right) - 1.046u + 60.227\right) * 1000 \tag{1}$$

In Table 2 the set-points of the pressures, temperatures and superheats are listed.

3.1.2. Control logic of the model

The control logic of the simulation model in Dymola (Table 3) is similar to the one implemented in System A. The first compressor slice always is in operation. Either it is in ACTIVE (S4) or PASSIVE mode (S5). Being in ACTIVE mode means that its high-pressure bypass valve (CV12) controls the suction pressure of the system while being in PASSIVE mode means that it is completely closed and inactive. The control has the CV12 valve of the activated CS. The second compressor slice can be in shut-

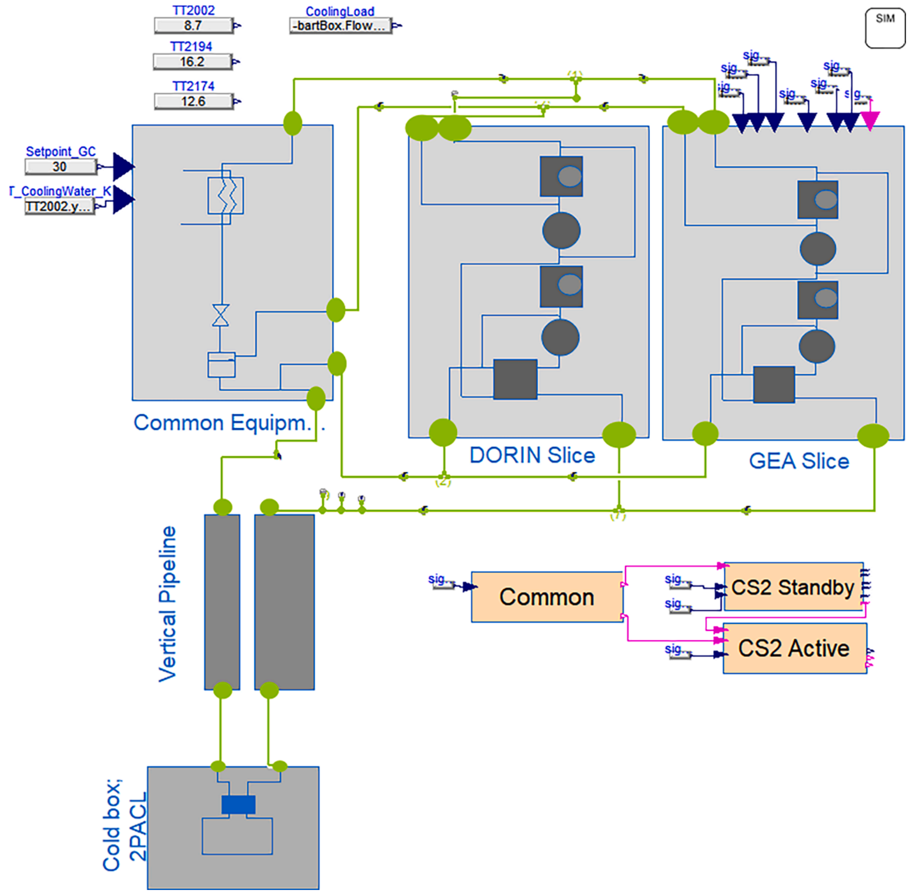

Fig. 5. Model of the primary refrigeration system (System A) in Dymola.

Table 2
Setpoints of temperatures in °C, super-heat in K and pressure in bar.

Outlet temperature of air-cooled gas cooler	35
Outlet temperature of water-cooled gas cooler	30
Ambient temperature	8.7*
Water temperature in common slice	16.2*
Water temperature in compressor slice	12.6*
Super-heat for HP compressor	20
Super-heat of evaporator	5
Suction pressure LP compressor	6
Suction pressure MP compressor	22.91
Receiver Pressure	55
Trans-critical pressure	Eq. (1)

* Averaged trend of extraction day.

Table 3
Control logic for each compressor slice.

Unit	Initial position	RUN
Compressor Slice 1 (CS1)	ACTIVE (S4)	PASSIVE (S5) if CS2 in Standby CV12 closed Suction Pressure LP compressor > PT5064 SP1 ACTIVE (S4) if CS2 in Standby CV12 OD > 0.001*OD max **OR** Suction Pressure LP compressor < PT5064 SP2
Compressor Slice 2 (CS2)	OFF (S0)	Start to STANDBY (S3) if CS1 in Active CV12 OD < OD max for OD t1 For time 1 **OR if** CS1 in Active CV12 OD < OD min ACTIVE (S4) if CS2 in Standby CV12 OD closed Suction Pressure LP compressor > PT5064 SP For time 1 OFF (S0) if Slice 2 in Standby CV12 > OD min for OD t2

* CV12 OD stands for the OD of the LP bypass valve, OD max and OD min for the valve open status threshold high and low, respectively, and OD t1 and OD t2 for the activation time thresholds.

OFF (S0), STAND-BY (S3) or ACTIVE (S4) mode, depending on the required cooling load. Stand-by means that the CS is pressurized and all components inside the CS are ready to operate. The EVs, except for EV10 (suction gas on/off valve), are opened and CV12 has no control yet.

If a small cooling capacity is needed, CS1 is running with a maximum opening degree of CV12. Hitting the upper limit (which is an adjustable input parameter), CS2 gets the Enabling signal. Then, a series of events happen inside the slice, such as the start-up of the compressors and heat exchangers. If all requirements are full-filled, CS2 is activated and CS1 goes to passive mode. Once the cooling demand drops, the mass flow rate quickly gets back to a minimum. CS2 gets a disabling signal stopping it from running and CS1 is activated controlling the suction pressure.

3.1.3. General parameters and assumptions

At the very beginning of modeling an already existing refrigeration plant, the systems refrigerants need to be chosen in the SIM (System Information Manager) of every new model. For System

A, it is R744A as the main refrigerant, while moist air gas is taken as coolant for the air-cooled gas cooler (17–18) and water for the water-cooled gas cooler (13–14 & 19–1). The following assumptions are made: the oil management system is neglected and the evaporator in the cold box (2PACL) is not modelled. Experimental data from the demonstration plant about the refrigerant mass flow rate and level of super-heat are considered as input parameters instead. A zero pressure drop in components and pipelines is assumed.

To evaluate the logic of the model and to properly compare and validate the simulation results with the experimental results from System A (test plant at CERN), the components used in DEMO need to be replicated. The following components are used: Reciprocating compressor types for the LP (SP11–12) and HP (SP16–17) compressors based on polynomials provided by the manufacturers (DORIN, GEA Bock), plate counter-current flow pre-cooler (10–11), water- cooled gas cooler (19–1), interstage-cooler (13–14) and post heat-exchanger (4–5), orifice valves based on effective flow areas for each valve (CVxx & EVxx) used in the system and a linear directional control valve used for the three way valve.

Default parameter are used for the control logic, which is more detailed explained in Section 3.1.2. The parameter (Table 4) were adjusted aiming the most possible stable pressures, resulting in a smooth transition from one to two operating compressors, and vice versa, and reliable systems operation.

One cycle of operation for each scenario will be evaluated in this paper. Once cycle consists of a zero load one slice operation, and increased cooling demand with the transition to a two slice operation to its full capacity operation, and vice versa.

The first 500 s of each simulation phase is not considered in this paper as the start-up phase of the refrigeration system is not analysed.

3.2. Model development

First, a simplified version of the complex refrigeration plant consisting of a common unit, a dummy load system and one compressor slice were developed. Predefined positive displacement compressor models (called the ”PartialCompressor”) based on Polynomials, provided by the manufacturer Bitzer, of the TIL library and a static trans-critical pressure of 78 bar as a pa- rameter for the high-pressure valve were applied. A full list of components is noted in S. Blust (2019). Realistic pressure, temperature and liquid level parameter, geometry and heat transfer properties as well as setting parameter for the PI controllers of the valves were implemented as a start of replicating the desired refrigeration model.

In this first phase of model development, static tests were run at the beginning followed by dynamic simulations. For the static tests, the minimum possible load (0.6 kW)(equivalent to a minimum mass flow rate) was assumed simulating the non-operating state of the detectors as well as the maximum possible load (50 kW) limited by the selected compressors and defined by the input mass flow rate in the 2PACL box. The maximum achievable load (maximum mass flow rate) is simulating the operational phase of the detectors. Later, one more step was taken having a static test at half load (25 kW). For the dynamic analysis, the minimum and maximum load were kept the same with its load

Table 4
Input parameter.

Actuator	Value	Description
OD max	45 %	Maximum OD of CV12 to enable slice 2 to standby
OD min	10 %	Minimum OD of CV12 to disable slice 2 from standby
OD t1	20 s	Activation time threshold to enable slice 2 when OD of
OD t2	300 s	CV12 is reached
CVgc. value	90 %	Activation time threshold to disable slice 2 when OD of CV12 is reached
MPcs. value	22.11 bar	Minimum OD of three-way valve to open flash gas supply and discharge gas valve
SP9 SP1	6.05 bar	Specific requirement for compressors to start from 30 Hz
SP9 SP2	5.98 bar	running Pressure setpoint to reach Active mode
time 1	5 s	Pressure setpoint to reach Passive mode Waiting time in
t load	500 s	Standby before going to active step
t load	840 s	Time of rise from zero to full load (dataset 2) Time of rise from zero to full load (dataset 1)

constantly rising.

In the second phase, a vertical pipe connecting the underground and surface units was imple- mented. For System A, the height was set to 1 m, as the demonstration plant is located on ground. On System A applied modelling parameter i.e. geometry and heat transfer properties were implemented, and initial values (pressure, temperature levels and enthalpies) were fine- tuned to achieve a smooth start of the simulations. Then, an exact same compressor slice was implemented, both operating in parallel. Once the dynamic tests ran smoothly, the control logic developed jointly by NTNU and CERN was inserted. Like this, one compressor slice can operate while the second one is in stand-by mode. Furthermore, the pressure equation, which is a func- tion of the water-cooled gas cooler outlet temperature, for the trans-critical pressure valve, was integrated and the compressor manufacturers exchanged. In the first compressor slice low and high pressure compressors from the DORIN company and in the second slice compressors from GEA BOCK are operating with a maximum cooling capacity of 50 kW each, having a maximum mass flow rate of 0.20 kg/s for the LP and 0.34 kg/s for the HP compressors.

In test phase three, two mass flow rate profiles of the dummy has been applied using experimental test campaigns of the demonstration plant at CERN, consisting of one cycle including enabling and disabling a second compressor slice. First, the average values of the low and high load were taken to evaluate the performance of the control logic before the real trend was consulted. The results are written in 4.1 and 4.2. To achieve a stable suction pressure under all loading condition, various control parameter were adjusted accordingly. Finally, the actual profile (data set 2) was taken to be validated on the demonstration plant (4.3).

4. Results and discussion

4.1. Simulation model using averaged input parameter

In Fig. 6, the three bottom plots represent the key components pressure profile for a first, compared to a second averaged mass flow rate (evaporator) ranging from 0.05 kg/s to 0.18 kg/s (Fig. 6, top plot) corresponding to an average cooling load in the range of 14 kW to 50 kW respectively. This first test campaign takes 14 min (dotted line) for the cooling load to reach its maximum, while the second one takes about 8 min (solid line). The pressure profile for the suction (blue line), receiver (red line) and trans-critical pressure (green line) can be seen.

The trans-critical pressure profiles of both data sets show stable values around 78.7 bar ± 1 bar, while the receiver pressure is constantly kept around 55 bar ± 0.2 bar. The suction pressure shows stable behavior at about 6 bar for both data sets with a maximum fluctuation of $+0.3$ bar and -0.02 bar. Negligible pressure fluctuations in the suction line are seen for data set 1 during one cycle ($+0.005$ bar and -0.003 bar).

Based on the results seen in Fig. 6, it can be assumed to have a stable and reliable operation for both data sets, 1 and 2. The steeper the mass flow rate profile (the faster the transition from a zero to full load operation), the greater the effect on the suction pressure. The next simulation step (Section4.2) is done taking data set 2 as input for the mass flow rate of the evaporator.

4.2. Simulation model using experimental input parameter

In the following section, the second data set is studied considering the same parameter as in Section 4.1. The reasons to take the second data set is the faster rise of mass flow rate from zero to full load (in 8 min) and the comparatively short test campaign phase (short cycle), which is about 4.5 h resulting in a fast configuration/simulation time. The ambient temperature is assumed to be 16.2 °C for all test campaigns, which is the average value of the extraction day of data set 1. Depending on this value, the speed of fan of the air-cooled gas cooler is adjusted. The water temperature of the water-cooled gas cooler in the common equipment unit is fixed to 8.7 °C and to 12.6 °C in the compressor slices,

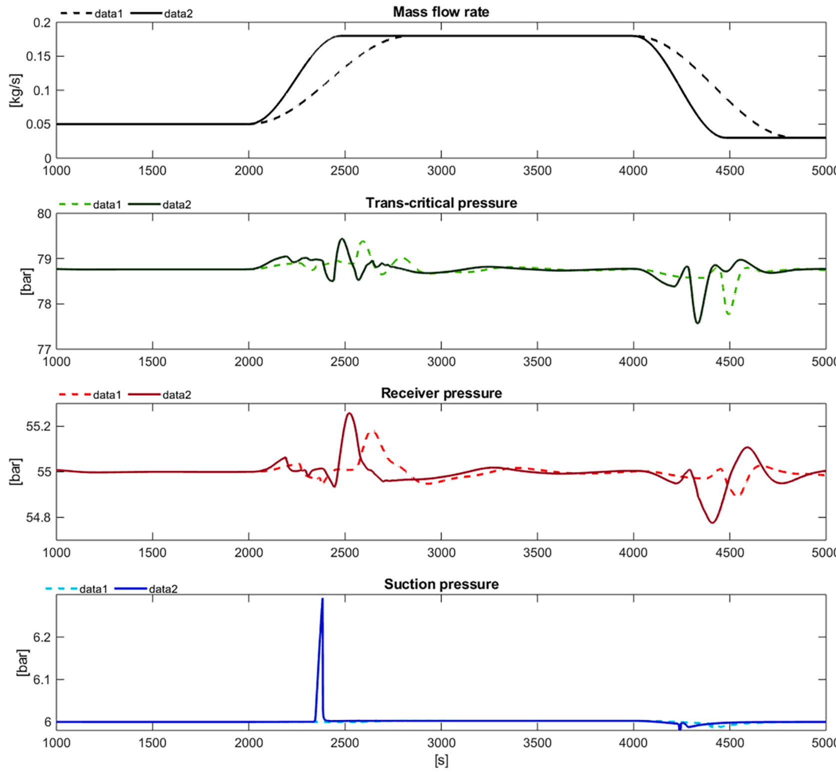

Fig. 6. Averaged mass flow rate of 0.05 to 0.180 kg/s in 14 min (data set 1 dotted) and in 8 min (data set 2 solid)(top plot). The resulting pressure profiles: suction (blue), liquid receiver (red) and trans-critical pressure (green) of the mass flow rate profiles of data set 1 (dotted) and data set 2 (solid) respectively.

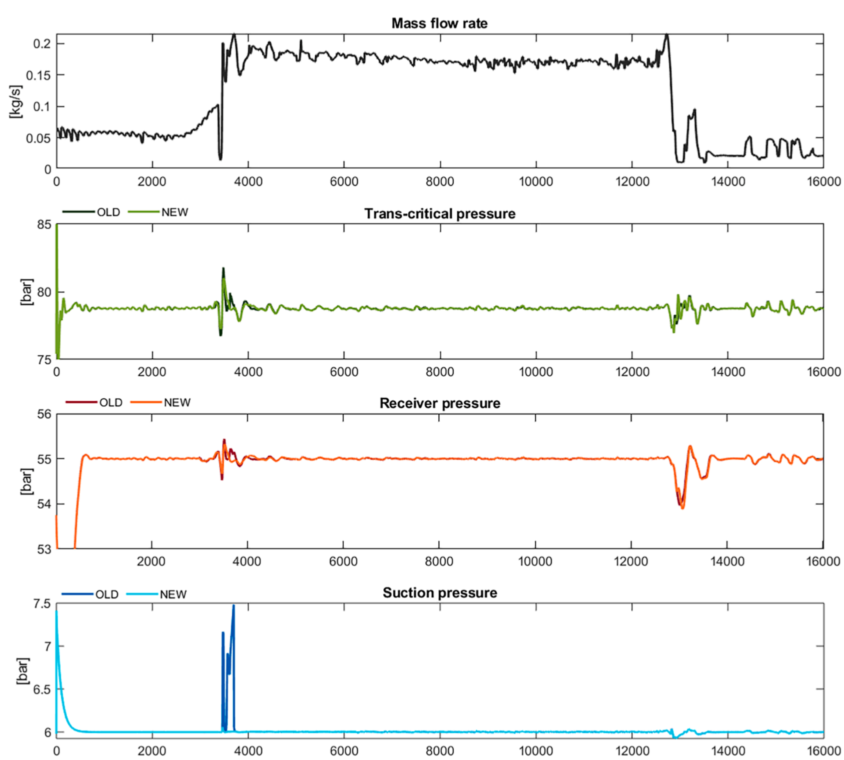

Fig. 7. Experimental mass flow profile of data set 2 extracted on 27.10.2021 from the demonstration plant at CERN (top plot, black line) and the resulting pressure profiles: trans-critical (2nd top plot, green), liquid receiver (2nd bottom plot, orange / red) and suction pressure (bottom plot, blue) of the mass flow rate profiles of data set 2 operating with the initial control logic (OLD) and modified control logic (NEW).

which are adjusted by the water-cooled gas cooler respectively. The mass flow rate starts at 0.05 kg/s, rises to 0.178 kg/s in about 8 min and falls again to about 0.03 kg/s in 200 s. The experimental mass flow rate profile can be seen in Fig. 7, top plot. In Fig. 7, the three bottom plots show the trans-critical (green), the receiver (red / orange) and the suction pressure (blue) for the initial control logic (OLD) and modified control logic (NEW).

The trans-critical and receiver pressure profiles remain stable during one cycle: The trans-critical pressure profile fluctuates at 78.7 bar ±2 bar and the receiver pressure is found to be stable at about 55 bar ±1 The suction pressure, with a set-point of 6 bar, fluctuates between +1.5 bar & -0.06 bar.

To improve the stability of the profiles, another actuator is included saying that the enabling process is only canceled if the OD of CV12 (hot gas by-pass valve of LP compressor) stays above 45 % for more than 100 s.

This modification results in a more stable operation of the entire system. The results indicate a minor improvement of the trans-critical (light green) and receiver (orange) and suction (light blue) pressure. As another enhancement, time 1 (Input parameter 4) is reduced to 0.1 s, which basically is no waiting time in standby mode before entering the active step of slice 2. This change was found to be beneficial (less fluctuation) for the suction pressure profile. An OD min = 5 % instead of 10 % was found to have a negligible effect on the fluctuation, and no effect on the maximum pressure difference of the suction pressure. Reducing OD t1 to 20 s or increasing OD t2 to 500 s, on the other hand, was found to have no effect on the suction pressure profile. The reason is, once only one slice is in operation, it takes control of the suction pressure of the primary refrigeration system (see 2.2). Thus, the threshold of a second slice to go to OFF from stand-by does not have an effect on the pressure.

The results of the modifications (an additional logical operator and shortening of time 1 (4)) are seen in Fig. 7 (NEW lines). A minor improvement of the trans-critical and receiver pressure is found, while the suction pressure now fluctuates between ±0.06 bar for the enabling and disabling process respectively, which is 25 times less than for the initial control logic. This means, in theory, the trends can be improved and stabilized by fine-tuning the parameter of the control logic. The main objective of this analysis was to straighten the suction pressure, which was successful as the rise during enabling is reduced by 4 %.

The analyzed mass flow rate profile is representative for all operational mass flow rates having an enabling and disabling process with a transition from zero to full load slower than 500 s. The longer the transition from zero to full load, the smoother the suction pressure profile. One operational cycle is fully dependent on the detectors. The long vertical pipelines of the whole refrigeration complex make the system slow response, having an effect on the mass flow rate in the evaporator. As long as the maximum mass flow rate is smaller than the maximum possible capacity of the LP compressors (< 201 g/s), the actual value is not important. The maximum mass flow rate, and thus, the maximum capacity of the system, is limited by the LP compressors while the minimum rate is limited by the evaporator, or the dummy in case of the demonstration plant.

4.3. Experimental validation of the numerical model

Fig. 8 shows a comparison between the numerical and experimental results of data set 2. It can be seen that generally the experimental pressure profiles for the receiver, suction and trans-critical pressure are less stable (more fluctuation) compared to the numerical results. The fluctuation for all three pressure profiles are heavier at the beginning and end of the test campaign. The average of the experimental, trans-critical pressure shows a fluctuation of 10 bar at the beginning and end, and 6 bar in between. The experimental receiver pressure shows a constant 0.7 bar deviation from the corresponding numerical values. The experimental suction pressure shows a total deviation of 2 bar.

At about 4000 s and 13 000 s, the second compressor slice was either

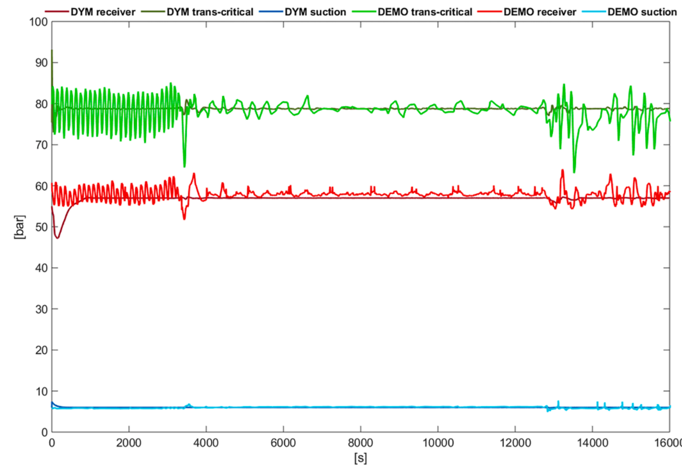

Fig. 8. Comparison of simulation (DYM) and experimental (DEMO) results of System A and data set 2. Trans-critical pressure (green), receiver pressure (red) and suction pressure (blue).

started or shut down. Thus, one full cycle of operation is seen in Fig. 8. It seems like a two slice operation of the demonstration plant is more stable than a one slice operation having less fluctuation in all pressure profiles. The reason of more fluctuating experimental pressure profiles are, based on the engineers at CERN among others, the following: During the period of data extraction, the commissioning of the refrigeration plant was done. During commissioning phase many components were tested/adjusted resulting in less stable pressure profiles. Furthermore, both oil tanks are successively emptied every 5–10 min and a hardly controllable incoming water temperature/mass flow of the water tank due to network issues (fluctuation of 6 K) have a great influence on the receiver pressure causing fluctuations on the pressure profiles. The parameter in the simulation model (such as PID controllers) easily can be adjusted and fine-tuned more accurately, and actuator can be added or cancelled resulting in more stable pressure trends.

The results of the simulation model are proven on System A: Experimental results (mass flow rate and superheat) are taken from System A at CERN and implemented in the Dymola model. The models parameter and logic were adapted according to the parameter of the refrigeration plant. The average of each pressure line (experiments) correspond to the numerical pressure lines. Experimental results mostly are more fluctuating due to the mentioned reasons. The most important parameter (suction pressure) correspond very well and for the receiver pressure the fluctuation is ±3 bar, which is 6 °C in total. The trans-critical pressure needs to be revised.

5. Conclusion

The R744 System is designed as a trans-critical concept fulfilling the specific needs of detector cooling having strict requirements on temperature stability at very large load differences. The detector needs to stay cold and stable at all time. Thus, the low temperature must be guaranteed with and without the detector electronics powered. The modular concept of 50 kW (or 70 kW on the final system) slices allows for switching on or off a slice depending on the demanded cooling power. One first demonstration plants is built, system A, to mainly evaluate the systems control logic. In parallel, the R744 refrigeration plant including its control logic is modelled in Dymola tool to validate the experimental results.

The simulation results of System A indicate that the initial control logic can be modified to fulfill the systems requirements. For example, a new actuator, saying that the enabling process is only canceled if the opening degree of the by-pass valve of the low-pressure compressor of the active slice stays above a certain value for a specific time, could be included. Furthermore, some parameter could be adjusted to fine-tune the switching on or off of another slice. A series of simulations are done having a mass flow rate trend of system A extracted and used as an input for the model. After adjusting specific parameter to achieve appropriate pressure trends, namely trans-critical, receiver and suction pressure, the simulation results are compared with the experimental ones. It can be seen that the pressure profiles of the simulation model are more stable during one cycle. The reason is a complex series of events happening in experiments such as fluid leakage, badly controllable components, fluctuating ambient / water temperature or influence of the existing of oil in such a system, which is not considered in a simulation model. The experimental pressure trends were not yet stable, having fluctuations, on the extraction day. Controller need to be fine-tuned, parameter adjusted and the control logic revised. However, with the support of a well-functioning simulation model, a stable system performance is achieved before the refrigeration system will be built on side.

Next step is to fine-tune the control logic of the demonstration plant based on the simulation model and evaluate the behavior of the system on-site to guarantee a stable and reliable operation all year long.

Points forts

- Système de refroidissement à deux fluides R744 et R744 : modules d'unité de compression de vapeur d'appoint trans critique à boucle contrôlée par accumulateur biphasé à pompage de liquide sans huile et trans critique
- Concept de refroidissement modulaire pour les mises à niveau des détecteurs ATLAS et CMS du Grand collisionneur de hadrons du CERN pour une demande de refroidissement dynamique (0–550 kW) inférieure à -50 °C
- Développement d'un modèle numérique à l'aide d'un logiciel dynamique appelé Dymola/Modelica & TIL
- Résultats de simulation validés expérimentalement sur une installation de démonstration du CERN (Système A)

Data extraction

All extracted data are uploaded in dataverseNO (https://doi.org/10.18710/YVECMK).

CRediT authorship contribution statement

Stefanie Blust: Conceptualization, Data curation, Formal analysis, Visualization, Writing – original draft, Writing – review & editing. **Pierre A.C. Barroca:** Writing – review & editing, Resources. **Yosr Allouche:** Writing – review & editing. **Armin Hafner:** Funding acquisition, Methodology, Project administration, Resources, Software, Supervision, Writing – review & editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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