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SIMULATION OF A TEST BENCH FOR REFRIGERATION SYSTEM COMPONENTS

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***Abstract**. The present work deals with the modeling and simulation of a test bench used in evaluating the performance of components of vapor compression refrigeration systems. The modeling is performed for the system being operated at steady state conditions and it is computationally implemented using the software EES-Engineering Equation Solver as a tool. The resulting model is used for designing of the bench subsystems and also used in its simulation in order to study their range of controllability as well as the input data required for the system is stably operated. The sizing and selection of equipment and components are made based on R134a refrigerant as the working substance, however, the model demonstrated to have numerical stability and to be compatible with the use of refrigerants such as R410a, R22, R407c and other mixtures. The main correlations between the operational variables of the test bench were derived as main result from the steady-state simulation. On the basis of these results, it may be concluded that the layout of the test bench warrants stability and reproducibility of experiments when using this test bench to study vapor compression refrigeration systems.*

***Keywords:** Test bench of cooling systems, operational controllability of refrigeration system.*

1. INTRODUCTION

The test bench, see Fig.1, is to be implemented for a cooling system that presents a good performance; initially the system will work with R134a refrigerant and a single scroll type compressor (CP) with a nominal capacity of 17.6 kW and then the system will be tested with another refrigerants.

The compressor was specified to operate in a limited range between the freezing point of water at the evaporator (EVP) and the maximum allowable water temperature for heat rejection in the condenser (RC) (Sanyo 2003).

Oil is significantly important as a lubricant for moving parts of refrigeration compressors; although it does not take part directly in the refrigeration cycle it may have a direct influence in heat transfer characteristics of both evaporators (EVP) and condensers (RC/PC).

The oil separator has the function of separating, collecting, filtrating solid particles and promoting the return of oil to the compressor thus preventing the oil to flows to both the lowest evaporator point and parts of the system at low temperatures.

The oil separator (OS) efficiency depends on the heat exchangers effectiveness, which is optimum when oil free.

The superheated gas enters the oil separator and is released by the solenoid valve (OSV) which is always opened when the compressor is in operation.

The two-way valve (W2V) has the function of controlling the subcooling temperature of liquid refrigerant in the water inlet line of the rejecting condenser (RC). This valve also contributes to maintain condensing pressure according to the specified parameters for the test.

The existing water-cooled heat rejection system is provided with a cooling tower (CT) of a rated capacity exceeding the possible maximum demand of test bench system. The main function of the rejecting condenser is to provide the balance of the system (that is, it rejects the heat associated to the compressor work).

The process condenser (PC) is intended to release, within the own system an amount of heat equal to the evaporating load in order to give stability and conditions for steady state operation. All parts of the system are insulated to reduce heat losses.

The cool water pump (ECWP) will operate at a constant flow and always will be in operation when the compressor starts (CP).

The hot water recirculation valve (W3VC) has the function of controlling the condenser water outlet temperature (Tc) and the cold water recirculation valve (W3VG) has the function of controlling the evaporator water outlet temperature (Th).

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The thermal Inertial tank (TIT) provides thermal capacitance to the system as well as the required thermal inertia, and it is sized so as to ensure the process gets stabilized within a short period of time and the controlling devices be able to adjust the input parameters during specific tests.

The water outlet temperatures (Ta) and (Tf) at the tank (TIT) are equal because the flows entering to the tank at points (e) and (l) mix themself inside the tank from the inner mixing tree to the exit manifold.

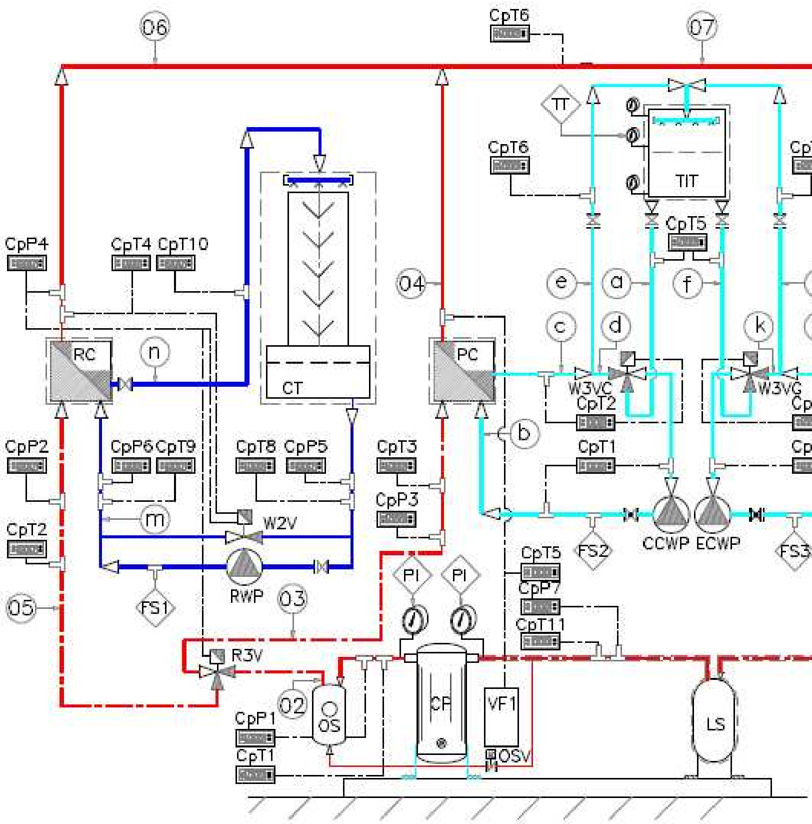

Figure 1. Schematic Diagram of the Test Bench

The subcooled liquid refrigerant lines leaving the condensers at the states 4 and 6 join together at the state 7, so becoming a single line, then flowing through an electrical heater to adjust the subcooling degree to the prescribed setting.

The board of valves have many options of expansion devices which will be actuated by solenoid valves. The test bench will initially operate with an electronic expansion valve (EEV) with adjustable superheat control and suction equalizer (Danfoss 2004).

The liquid separator (LS) is responsible for retaining liquids and condensates in the compressor protecting it from damage.

2. SIMULATION

A design-oriented simulation is conducted to sizing the basic components of the test bench and then to built a parameterization of the operation point so as to reach the best controlling condition and to determine the limiting values of main variables involved in system performance.

2.1 Mathematical Model

The mathematical model is built by direct applications of the mass and energy conservation principles to each component of the test bench as well as of particular relations that allow describing its operational behavior such as thermal effectiveness, temperature at heat exchanger inlets and outlets and volumetric and isentropic compressor efficiencies.

The following assumptions are taken in the model formulation:

- Steady state operation,

- The heat transfer rates in the condensers and the evaporator is controlled by their corresponding water heat transfer coefficients,

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- The thermal tank has a very high thermal capacity and an evenly distributed temperature throughout its volume and
- Negligible heat losses from heat exchangers, valves and piping of both water and refrigerant.

The relation $U = U^*\left(\frac{\dot{m}}{\dot{m}^*}\frac{A^*}{A}\right)^n$ is assumed in order to size the flat-plate-type heat exchangers proposed for the condensation and evaporation processes. In the late expression, the exponent (n) depends on the angle of Chevron of the plates, whilst those values of (U) and (m) labeled by (*) are reference values for the global heat transfer coefficient and mass flow rate respectively. A value of (n) equal to 0.3 is adopted to estimate the global heat transfer coefficient. A relation of the form $U = U^*\left(\frac{\dot{m}}{\dot{m}^*}\right)^{0.8}$ is used for the coil-in-coil rejecting condenser.

In all heat exchangers (PC, RC, EVP), the heat transfer is modeled by using the $\wp - NTU - R$ method with NTU and ($\wp$) defined on the water side. The exit temperature of the single phase fluid may be expressed in terms of the temperature effectiveness ($\wp$) of the heat exchanger, the inlet temperature of the single phase fluid (T_{inlet}) and the change of phase temperature (T_{ph-c}) according to the equation (1) (Shah and Sekulić 2007).

$$T_{Outlet} = (1-\wp)T_{inlet} + \wp T_{ph-c} \tag{1}$$

where $\wp = \frac{T_{Outlet} - T_{inlet}}{T_{ph-c} - T_{inlet}} = \wp_{(NTU,\ R,\ Flow\ Arrangement)}$

The ratio of thermal capacity rates (R) is zero for any heat exchanger with change of phase and regardless the flows arrangement the temperature effectiveness may be calculated by equation (2).

$$\wp = 1 - Exp(-NTU) \tag{2}$$

with, $NTU = \frac{UA}{\dot{C}}$, and $\dot{C} = \dot{m}\,Cp$

2.2 Mathematical equations

2.2.1 Flat Plate Evaporator Model-EVP (EVP)

The evaporator rated cooling capacity may be calculated by equations (3) and (4):
- Water side

$$\dot{Q}_{evp} = \dot{m}_{w,evp}Cp_{w,evp}(T_h - T_g) \tag{3}$$

- Refrigerant side

$$\dot{Q}_{evp} = \dot{m}_r(h_1 - h_9) \tag{4}$$

According to Equation (1), the water exit temperature may be expressed in terms of the evaporation temperature, the water inlet temperature to the evaporator and the temperature effectiveness as in the equation (5).

$$T_h = (1-\wp_{w,evp})T_g + \wp_{w,evp}T_{evp} \tag{5}$$

The evaporator temperature effectiveness is defined along the water path by an expression in the form of the equation (6)

$$\wp_{w,evp} = 1\text{-}Exp(-NTU_{evp}) \tag{6}$$

The thermodynamic properties at the defined states are determined according to the set of equations represented by equations from (7) to (12).

$$h_9 = h_8 \tag{7}$$

$$h_8 = h_{(T_8,P_8)} \tag{8}$$

$$h_1 = h_{(T_1,P_1)} \tag{9}$$

$$T_1 = T_{(P_{evp},x=1)} + \Delta T_{sh} \tag{10}$$

$$P_1 = P_{evp} \tag{11}$$

$$x_9 = \frac{h_9 - h_{f(P_9,x=0)}}{h_{g(P_9,x=1)} - h_{f(P_9,x=0)}} \tag{12}$$

2.2.2 Flat Plate Condenser Model (PC)

The condenser thermal capacity may be calculated by equations (13) and (14)
- Water side

$$\dot{Q}_{cnd} = \dot{m}_{w,cnd} Cp_{w,cnd} \left(T_c - T_b\right) \tag{13}$$

- Refrigerant side

$$\dot{Q}_{cnd} = \dot{m}_{r,cnd} \left(h_3 - h_4\right) \tag{14}$$

The water exit temperature is calculated by Equation (15)

$$T_c = \left(1 - \wp_{w,cnd}\right) T_b + \wp_{w,cnd}\, T_{cnd} \tag{15}$$

with the effectiveness defined by the water side according to the equation (16).

$$\wp_{w,cnd} = 1\text{-}Exp\left(-NTU_{cnd}\right) \tag{16}$$

The set of equations from (17) to (21) allows determining the thermodynamics properties at the states in connection with the process condenser.

$$h_3 = h_{(T_3,P_3)} \tag{17}$$

$$h_4 = h_{(T_4,P_4)} \tag{18}$$

$$T_4 = T_{(P_{cnd},x=0)} - \Delta T_{sc} \tag{19}$$

$$P_3 = P_{cnd} \tag{20}$$

$$P_4 = P_3 - \Delta P_{cnd} \tag{21}$$

2.2.3 Compressor Model (CP)

The compressor power demand is calculated by equation (22)

$$W_{cp} = \dot{m}_r \eta_s \left(h_{2s} - h_1\right) \tag{22}$$

Isentropic Efficiency (Sanaye and Chahartaghi 2010)

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$$\eta_s = B_o + B_1 \frac{P_2}{P_1} + B_2 \left(\frac{P_2}{P_1} \right)^2 \tag{23}$$

Volumetric Efficiency (Sanaye and Chahartaghi 2010)

$$\eta_s = a + b \frac{P_2}{P_1} \tag{24}$$

Refrigerant Mass Flow Rate (Jabardo, Mamani et al. 2002)

$$\dot{m}_r = \eta_v \rho_1 \frac{V_C N}{60} \tag{25}$$

Determination of the properties of thermodynamic states

$$h_2 = h_{(T_2, P_2)} \tag{27}$$

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_s} \tag{28}$$

$$h_{2s} = h_{(P_2, s_1)} \tag{29}$$

$$\rho_1 = \rho_{(T_1, P_1)} \tag{30}$$

2.2.4 Coil-in-Coil Condenser Model (Rejecting Condenser) (RC)

Under the assumption that the heat rejected by the rejecting condenser is numerically equivalent to the compressor power the next relation applies,

$$\dot{Q}_{rc} = \dot{W}_{cp} \tag{31}$$

The thermal capacity on the water side is calculated using the following equation,

$$\dot{Q}_{rc} = \dot{m}_{w,rc} Cp_{w,rc} (T_n - T_m) \tag{32}$$

The thermal capacity on the refrigerant side is,

$$\dot{Q}_{rc} = \dot{m}_{r,rc} (h_5 - h_6) \tag{33}$$

The water exit temperature is calculated by equation (34)

$$T_n = (1 - \wp_{w,rc}) T_m + \wp_{w,rc} T_{cnd} \tag{34}$$

with the effectiveness defined by the water side according to the equation (35).

$$\wp_{w,rc} = 1\text{-}Exp(- NTU_{rc}) \tag{35}$$

The thermodynamic properties at the defined states are determined according to the equations (36) and (39).

$$h_5 = h_3 = h_2 \tag{36}$$

$$h_6 = h_{(P_6, T_6)} \tag{37}$$

$$T_6 = T_{(P_6, x=0)} - \Delta T_{sc} \tag{38}$$

$$P_6 = P_{cnd} \tag{39}$$

2.2.5 Energy Balance for the Thermal Inertial Tank (TIT)

The following relation represents the energy balance for the thermal inertial tank,

$$(1-y)\dot{m}_{w,evp}Cp_{w,l}T_l + (1-x)\dot{m}_{w,cnd}Cp_{w,e}T_e = \left[(1-y)\dot{m}_{w,evp} + (1-x)\dot{m}_{w,cnd}\right]Cp_{w,mix}T_{mix} \tag{40}$$

$$T_a = T_f = T_{mix} \tag{41}$$

2.2.6 Mass and Energy Balances for the Cold-water Three-way Valve (W3VG)

The mass ratio of the re-circulated water in the evaporator circuit is defined by the equation (42)

$$y = \frac{\dot{m}_{w,evp(k)}}{\dot{m}_{w,evp}} \tag{42}$$

The mass balance for the valve (W3VG) is written as

$$\dot{m}_{w,evp(k)} + \dot{m}_{w,evp(l)} = \dot{m}_{w,evp} \tag{43}$$

$$\dot{m}_{w,evp(l)} = \dot{m}_{w,evp(f)} \tag{44}$$

with the energy balance expressed according to equation (45)

$$yCp_{w,k}T_k + (1-y)Cp_{w,f}T_f = Cp_{w,g}T_g \tag{45}$$

$$T_k = T_h \tag{46}$$

2.2.7 Mass and Energy Balances for the Condensing water Three-way Valve (W3VC)

The mass ratio of the re-circulated water in the condenser circuit is defined by the equation (47)

$$x = \frac{\dot{m}_{w,cnd(d)}}{\dot{m}_{w,cnd}} \tag{47}$$

The mass balance for the valve (W3VC) is written as

$$\dot{m}_{w,cnd(d)} + \dot{m}_{w,cnd(e)} = \dot{m}_{w,cnd} \tag{48}$$

$$\dot{m}_{w,cnd(a)} = \dot{m}_{w,cnd(e)} \tag{49}$$

With the energy balance expressed according to equation (50)

$$xCp_{w,d}T_d + (1-x)Cp_{w,a}T_a = Cp_{w,b}T_b \tag{50}$$

$$T_d = T_c \tag{51}$$

2.2.8 Eletronic Expansion Valve (EEV)

A nonlinear model obtained from manufacturing rating data in (Li, Braun et al. 2004) is used to determine the overall mass flow rate passing through the valve, according to equation (52).

$$\dot{m}_r = c_o\left[2\left(\frac{\Delta T_{sh}}{\Delta T_{sh,\max}}\right) - \left(\frac{\Delta T_{sh}}{\Delta T_{sh,\max}}\right)^2\right]\sqrt{\rho_8(P_8 - P_9)} \tag{52}$$

In the equation (52), the constant Co depends on valve geometry and reserve of capacity. A maximum superheating value is assumed in the valve model.

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2.2.9 Eletrical Heater (EH)

The subcooling degree of refrigerant flows leaving the process condenser and the rejecting condenser differ from each other and an excess of subcooling may eventually occur in the mixing flows at the state 7. An electrical heater is mounted on the system to adjust the subcooling degree of refrigerant entering the expansion devices. The power of electrical heater is calculated by simple energy balance according to equation (53),

$$\dot{W}_{el} = \dot{m}_r \left[h_8 - h_7\right] \tag{53}$$

2.2.10 Refrigerant Three-way Valve (R3V)

Under the assumption that all flows of refrigerant passing through valves are considered to be isenthalpic the enthalpy of flows at states (2), (3) and (5) are equal.

3. SOLUTION

The resulting model is implemented on the environment of the calculating software EES, and the computation model is constituted by a set of equal number of equations and variables simultaneously solved with initial guesses compatible with typical operational data of vapor compression refrigeration systems.

3.1 Entrance Data

The evaporator used in the test bench is a brazed-plate type Heat Exchanger, manufactured by Alfa Laval, Model: AC-70X-50M (G67, H34, B32) with a heat transfer area of 2,448 m^2 for water entering at 12 ºC and leaving at 6 °C, water mass flow rate of 0,6975 kg/s and a heat transfer rate of 17,6 kW. The mass flow rate of refrigerant R134a is 0,1203 kg/s at an evaporating temperature of 1,2 °C and superheating of 3,8 K. The overall coefficient of heat transfer (reference value) is equal to 1234 W/(m^2K).

The process condenser used in the test bench is also a brazed-plate type Heat Exchanger of Alfa Laval, Model: CB30-50H (H21, B21) with a heat transfer area of 1,392 m^2, water entering at 30 °C and leaving at 35 ºC, the mass flow rate of water is 0,8419 kg/s, heat transfer rate of 17,6 kW, and the mass flow rate of refrigerant R134a 0,0931 kg/s at a condensing temperature of 40,2 °C, the refrigerant subcooling is estimated at 3,8 K and the overall coefficient of heat transfer (reference value) is equal to 2280 W/(m^2K).

Table 1. Entrance Data for Heat Exchangers (ALFA-LAVAL 2013)

Parameters	Evaporator	Condenser	Rejecting Condenser
Water mass flow rate (kg/s)	0,6975	0,8419	0,2037
Refrigerant mass flow rate (kg/s)	0,12025	0,0931	0,0251
Exit water temperature (ºC)	6,0	35,0	35,0
Heat transfer coefficient (W/(m^2K))	1234	1706	250
Heat transfer area (m^2)	2,448	1.392	1,229
Subcooling degree (K)	-	3,8	variable
Superheating degree (K)	3,8	-	-
Average temperature TIT (ºC)	20,0		

4. RESULTS

The model was used to simulate operating conditions similar to those expected when operating the real test bench of the refrigeration and air conditioning laboratory at the PUC-RIO, yet to be constructed.

Figure 2 shows that when fixing both superheating and subcooling, the cooling capacity increases linearly but slightly with an increasing condensation temperature, however, this dependence jumps at higher levels when increasing superheating degrees.

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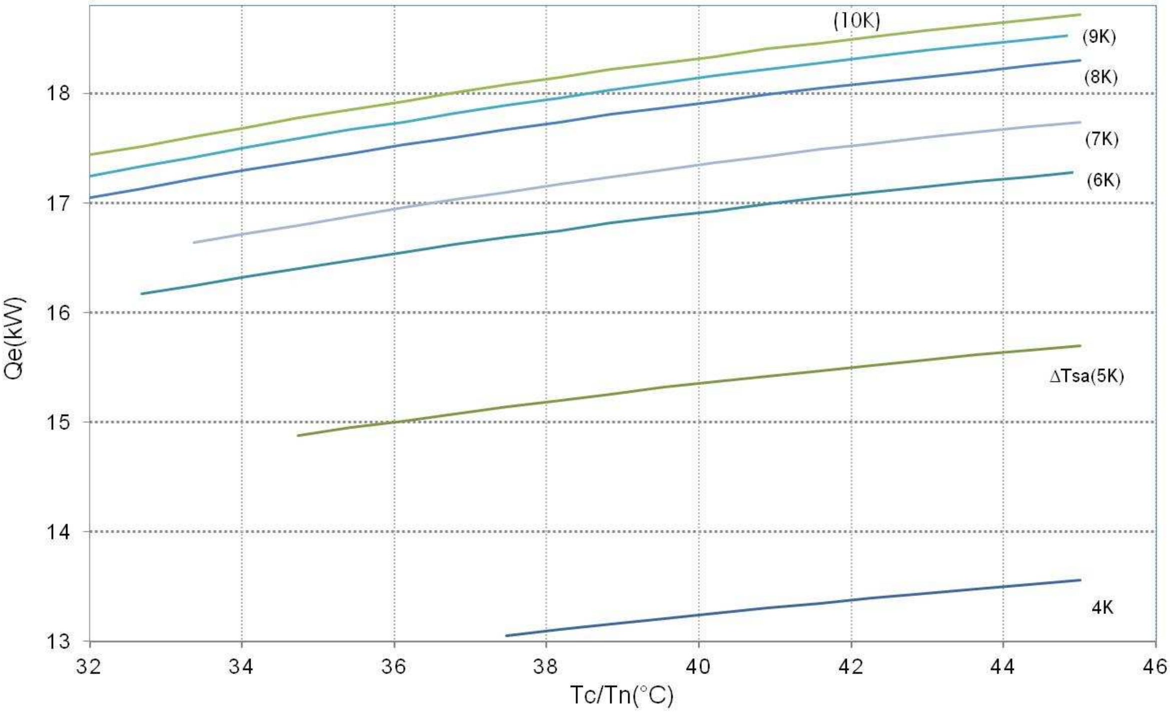

Figure 2. Influence of Condensation Water Temperatures on the Evaporation Capacity at constant Refrigerant Superheating.

The Fig. 3 shows that under the conditions specified above, the compressor rotation speed basically depends on the refrigerant superheating which means the test bench has a significant stability of operation.

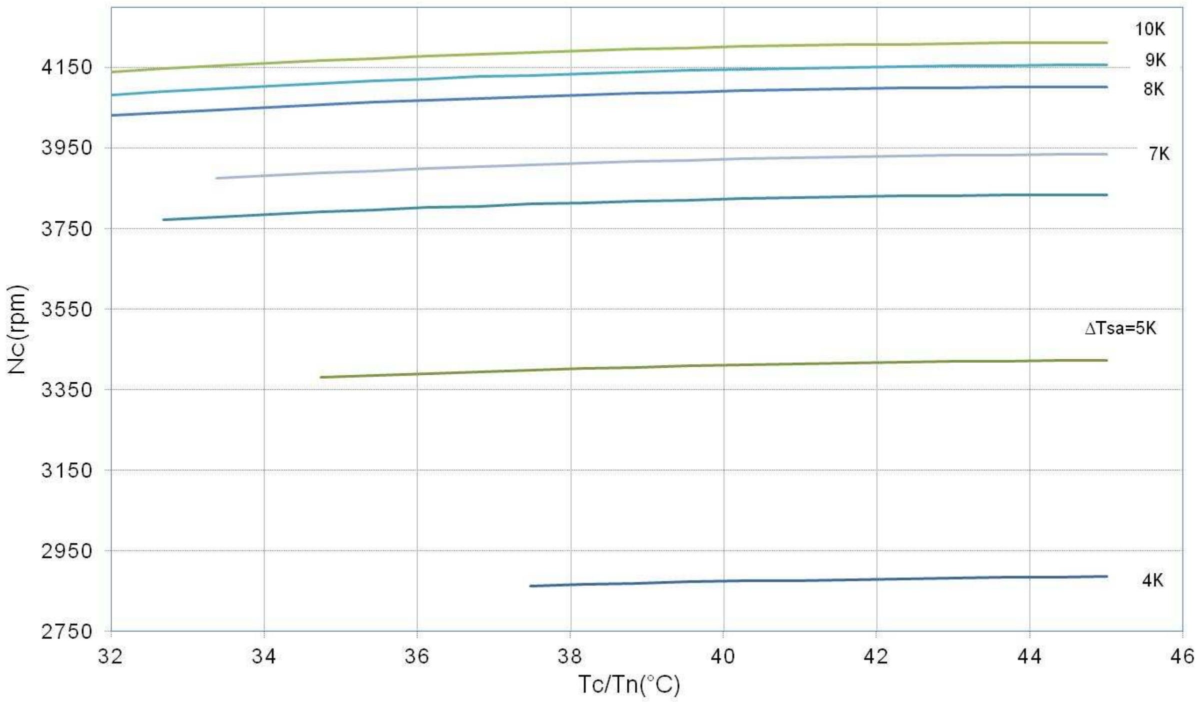

Figure 3. Influence of Condensation Water Temperatures on the Compressor Rotation Speed at constant Refrigerant Superheating.

For the test bench operating at both constant superheating and compressor rotation speed, the refrigeration capacity is nearly constant with respect to the condensing water temperature as illustrated in Fig. 4, but it inhibits a significant dependence with respect to the compressor rotation speed and superheating as expected. This figure also shows a linear relation between the temperatures of cold water leaving the evaporator and the temperature of condensation water entering the condenser. For constant superheating and rotation speed, the cold water temperature increases with an increase in the condensation temperature. At a constant condensation water temperature the temperature of water leaving the evaporator depends strongly on superheating and rotations speed as seen in the figure.

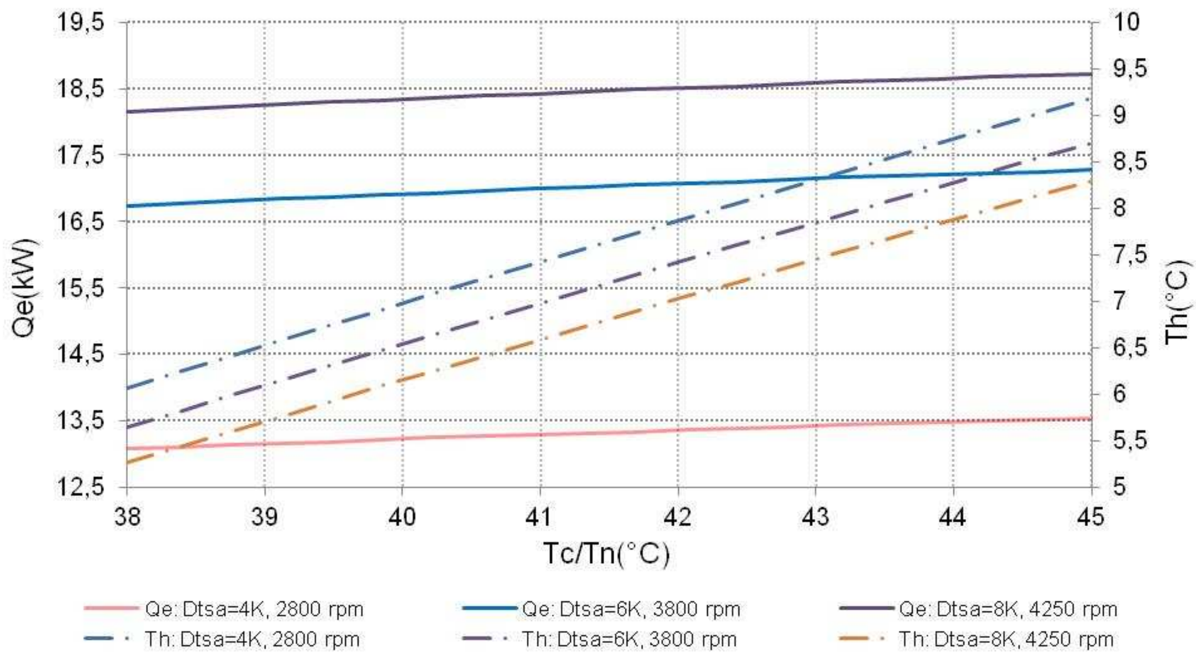

Figure 4. Influence of Condensation Water Temperatures on the Evaporation Capacity and the Cold Water Temperature at constant superheating and compressor rotation speed.

According to Fig. 5 the condensation temperature and refrigeration capacity increase linearly with respect to rotation speed, meanwhile the evaporation temperature decreases.

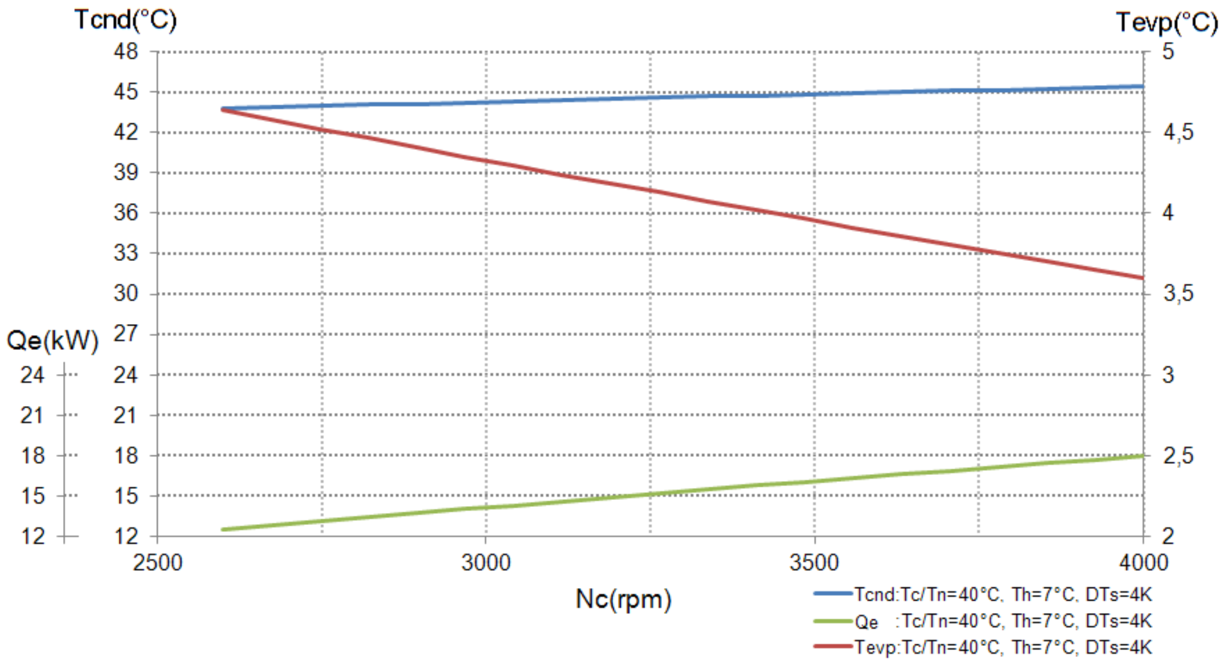

Figure 5. Influence of the Compressor Rotation Speed on the Evaporation Capacity and Condensation and evaporation Temperature at constant conditions for Tc/Tn, Th and ΔTs.

5. CONCLUSIONS

The simulation allows defining the input variables and predicting the output responses, thus imposing operational limits of the test bench. The model demonstrated numerical stability and allowed to capture the correlations between the different variables of operation. The simulation also shows that the test bench layout ensures stability in operation and during testing.

6. ACKNOWLEDGEMENTS

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8. RESPONSIBILITY NOTICE

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NOMENCLATURE:

Símbolos gerais

A	Area, [m2]
a	Coefficient for the function in equation (24), [-]
B	Coefficient for the function in equation (23), [-]
b	Coefficient for the function in equation (24), [-]
$\dot{C}$	Thermal capacity rate, [kW/K]
C_o	Constant in equation (52), [-]
COP	Coefficient of performance, [-]
c_p	Specific heat at constant pressure, [kJ/kg-K]
ΔE	Energy differential, [kJ]
ΔT_s	Subcooling degree of the system, [K]
ΔT_{sa}	Superheating degree, [K]
ΔT_{scr}	Subcooling degree at the RC exit, [K]
ΔT_{sr}	Subcooling degree at the CP exit, [K]
dt	Time differential, [s]
h	Specific enthalpy, [kJ/kg]
$\dot{m}$	Mass flow rate, [kg/s]
NTU	Heat transfer units, [-]
Nc	Compressor speed, [rpm]
n	Exponent for the heat exchanger model, [-]
P	Pressure, [kPa]
$\dot{Q}$	Heat transfer rate, [kW]
R	Ratio of thermal capacity rates, [-]
T	Temperature, [˚C]
U	Global Heat Transfer Coefficient, [$kW \cdot m^{-2} \cdot K^{-1}$]
x	Recirculation Mass flow ratio of water in the condenser, [kg/kg]
y	Recirculation Mass flow ratio of water in the evaporator, [kg/kg]

Greeks:

Δ	Variation, [-]
η	Efficiency, [-]
ρ	Density, [kg/m3]
$\wp$	Temperature effectiveness of heat exchanger, [-]
$\sum$	Summation, [-]

Subscripts

cnd	Process Condenser
cr	Rejecting Condenser
e, evp	Evaporator
cp	Compressor
el	Electric
max	Maximum
r	Refrigerant
s	System
w	Water

Acronyms

CP	Scroll compressor type
CT	Cooling tower water
ECWP	Water cool pump
CCWP	Water condensation pump