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Dynamic Simulation of a Thermal Management System Consisting of a CO_2 Heat Pump and a Water-Loop

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Abstract: *A dynamic simulation of a thermal management system which combined a CO_2 heat pump with a water-cooling system have been carried out using Simscape™ according with the change of the compressor rotation speed. Simscape™ built-in models are used for each system component. The simulation results demonstrate that the system's behavior and performance are reasonable in terms of the physical senses, however, it needs to be validated experimentally.*

Keywords: Dynamic simulation; Thermal management system, CO_2 (R-744) heat pump

1. INTRODUCTION

Dynamic simulations of refrigeration systems using CO_2 as a refrigerant in multi-physical domain platform have been carried out some researchers. Because, it is convinent to simulate complicate systems which consist of components in different physical-domains. Pfafferott and Schmitz [1] developed a library for CO_2-refrigeration system in Modelica and compared simulation results with their experimental data and they show a good agreement with the experimental results. But the results showed that the refrigerant state at the expansion valve inlet is deviated from its experimental data. In this case it might cause an effect on the dynamic behavior of the system. Shi et al. [2] developed a dynamic model of a transcritical CO_2 for a supermarket refrigeration system in Dymola to investigate the system performance and validated their model with actual field data. Using their model, they investigated the effects of designe parameters in field such as one- and two-dimensional gas cooler models, discretization number for the liquid pipe, and natural convection on the gas cooler.

In the present study, dynamic simulations of an air-to-air thermal management system which combined a CO_2 heat pump with a water-cooling system have been carried out in the MATLAB/Simulink environment, using Simscape™. The system's dynamic behavior and performance are stated corresponding with a changing of compressor speed in this paper.

2. MODEL

Simscape™ provides some physical models, called component block, in the multi-physical domain. Using these blocks, it is possible to build up models of the

Table 1. System component in Simscape™ physical domain.

Actual		Simscape™	
System	Component	Block model	Physical domain
Heat pump	Compressor	Controlled Mass Flow Rate Source (2P)	Two-Phase Fluid
	IHX	Pipe (2P)	
	Evaporator	Customized	
	Expansion device	Variable Local Restriction (2P)	
Water-cooling system	Pump	Controlled Mass Flow Rate Source (TL)	Thermal Liquid
	Heat exchanger	Pipe (TL)	
Thermal reservoirs		Controlled Temperature Source	Thermal
Air-side convective heat transfer		Convective Heat Transfer	

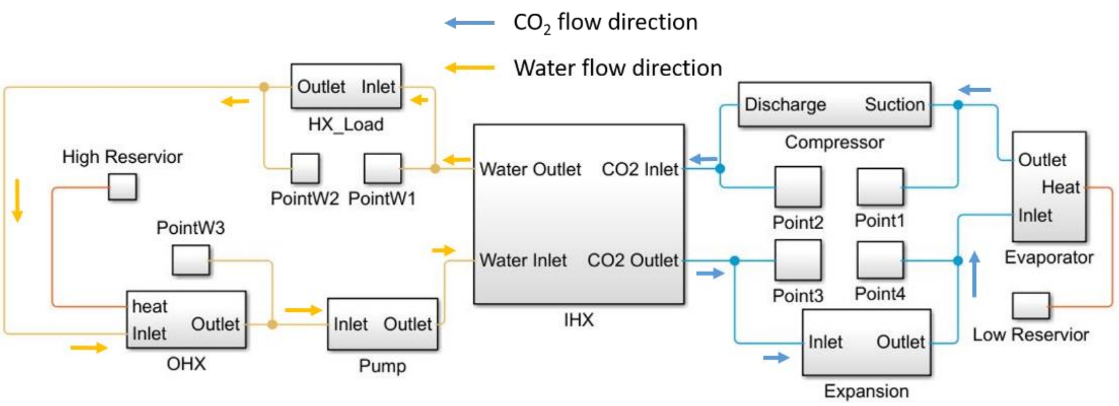

Fig. 1. Schematic of the thermal management system.

physical system and to create customized component blocks or modifying the original built-in blocks by Simscape™ language.

A schematic of the system is shown in Fig. 1. The thermal management system consists of a water-cooling system without phase change and a CO_2 heat pump system. The water-cooling system consists of a pump and three heat exchangers. One is an outdoor heat exchanger (OHX) to reject heat to the high-temperature reservoir, the second is to cool down the 'Load', and the last is an IHX to exchange heat with the heat pump system. The heat pump consists of a compressor, an expansion device, and an evaporator. Table 1 describes component blocks which used to model the present system and what they represent. The detail mathematical models or equations that make up each component block are stated on the Simscape™ Reference [3].

2.1 Compressor

A compressor is set up using 'Controlled Mass Flow Rate Source (2P)' block with the following assumptions:

1. Compression is in isentropic.
2. Heat transfer from or to the surroundings is negligible.

This block gives the specific value of set refrigerant mass flow rate.

2.2 Expansion device

An orifice is set up as an expansion device using 'Variable Local Restriction (2P)' block, under the isenthalpic expansion assumption. By changing the cross-section area of the orifice, the orifice controls pressure loss and refrigerant mass flow rate. The cross-section area of the orifice is controlled by the refrigerant temperature and superheating at the suction.

2.3 Heat exchanger (HX)

2.3.1 Refrigerant-side

Heat transfer and pressure drop of refrigerant flow are calculated based on the mass, momentum, and energy conservation equations using 'Pipe (2P)' block. This block represents a control volume and functions as a unit-segment for calculation. The block works on the following assumptions:

1. One dimensional flow
2. Fully developed flow
3. The gravitational effect is negligible.
4. Change in kinetic and potential energy is negligible.
5. Heat transfer is calculated with respect to the temperature of the fluid bulk.
6. The temperature gradient along the pipe is negligible.

Three conservation equations are as follows:

$$\frac{d}{dt}(\rho V) = \sum_{in}^{out} \dot{m}\,, \tag{1}$$

$$\left(-\frac{dP}{dz}\right) = \left(\frac{G^2 dv}{dz}\right) + \left(\frac{f_D G^2 v}{2D_h}\right), \tag{2}$$

$$\frac{d}{dt}(m\hat{u}) + \sum_{in}^{out} \dot{m}\hat{h} = \dot{Q}\,, \tag{3}$$

where, t, ρ, V, $\dot{m}$, G, v, f_D, D_h, z, m, $\hat{u}$, $\hat{h}$, and $\dot{Q}$ represent the time, the refrigerant specific density, the refrigerant volume in the pipe, the refrigerant mass flow rate, the refrigerant mass flux, the refrigerant specific volume, the Darcy friction factor, the hydraulic diameter, the segment length, the refrigerant mass, the specific internal energy, the specific enthalpy, and the heat transfer rate, respectively. This block has only equipped with a condensation correlation for phase change heat transfer coefficient. So, a customized block is newly made for calculation of boiling heat transfer coefficient using Fang [4] model. This customized block is used for the evaporator modeling.

2.3.2 Water-side

Heat transfer and pressure drop of water flow are calculated based on the mass, momentum, and energy conservation equations using 'Pipe (TL)' block. Including 6 assumptions mentioned in the section 2.3.1, this block takes following additional assumptions:

1. Incompressible flow
2. Frictional pressure drop takes into consideration, only.
3. Convective heat transfer is dominant for inner-pipe flows.

2.3.3 Air-side

Convective heat transfer takes into consideration in 'Thermal' domain using 'Convective Heat Transfer' block. This block functions a simple convective heat transfer by a heat transfer area and coefficient as input parameters. Heat transfer coefficient of air-side is manually calculated by a user-defined script file in a fixed temperature using Park and Jacobi [5] model for louver-fin HXs. The dry air properties are obtained from Lemmon et al. [6].

2.4 Pump

'Controlled Mass Flow Rate Source (TL)' block is used for modeling a pump with the following assumptions:

1. Friction loss during pumping procedure is negligible.
2. Heat transfer from or to the surroundings is negligible.

Using this block, the water mass flow rate is controlled.

2.5 Thermodynamic properties

REFPROP ver. 10.0 (Lemmon et al. [6]) is linked with MATLAB. A look-up table of working fluid properties is built at the MATLAB workspace. Properties on this table are parameterized by pressure and normalized specific internal energy. Using 'Two-Phase Fluid Properties (2P)' block, the properties table is brought into the Simscape™ physical domain. Thermodynamic properties of water are built in 'Thermal Liquid Properties (TL)' block. The water properties in this block are sourced from Coolprop, an open-source fluids database.

Table 2. Geometric specification of each component.

Component	Value
Compressor displacement (cm^3)	4
Expansion device	
Full open diameter (mm)	5
Min. diameter (mm)	5×10^{-4}
Evaporator	
Hydraulic diameter (mm)	1.5
Length (m)	1.3
Number of tubes	46
Total heat transfer area of air-side (m^2)	2.5
OHX	
Hydraulic diameter (mm)	5
Length (m)	1
Number of tubes	34
Total heat transfer area of air-side (m^2)	5
IHX (Double-pipe HX)	
Hydraulic diameter CO_2-side (mm)	10
Hydraulic diameter water-side (mm)	20
Length (m)	10
HX for the Load (Tubular)	
Hydraulic diameter (mm)	10
Length (m)	5
Number of tubes	1

Table 3. Simulation condition.

Content	Value	Remarks
Heat pump system		
High-pressure side		
Initial pressure (MPa)	7	
Initial temperature (K)	300	
Low-pressure side		
Initial pressure (MPa)	5	
Initial temperature (K)	300	
Water-cooling system		
Initial pressure (MPa)	0.101	
Initial temperature (K)	300	
Load (kW)	5	Constant
Thermal reservoir (Dry air)		
High-temperature side		
Temperature (℃)	30	Constant
Mean air velocity ($m\ s^{-1}$)	17	Constant
Low-temperature side		
Temperature (℃)	20	Constant
Mean air velocity ($m\ s^{-1}$)	3.5	Constant

2.6 Thermal reservoir

High and low-temperature reservoirs are set up using 'Controlled Temperature Source' and using 'Signal Builder' it is possible to control the temperature of the reservoirs with time. Dry air takes into consideration as a heat transfer medium.

3. SIMULATION

3.1 Conditions

Geometric specifications of each system component are summarized in Table 2. The water mass flow rate is fixed in 0.2 kg s^{-1}. Superheating at the suction keeps more than 3 ℃ and the expansion device opening is controlled by this set superheating. Simulation conditions and initial parameter set-up for each component are summarized in Table 3. For calculation, the 'ode23t', one of implicit variable time-step solvers, is selected. With changing the compressor speed as shown in Fig. 2, the system's behavior and performance are investigated.

3.2 Results

Fig. 3 shows the dynamic behavior of CO_2 heat pump and its performance corresponding with the compressor speed change which mentioned in Fig. 2, for 3 hours (10800 seconds).

In Fig. 3 (a) and (b), when the compressor starts up and it runs on 1500 rpm (revolutions per minute), the system temperature and pressure spread out and reach each steady-state position. After 1000 seconds later the compressor speed decreases to 1000 rpm, the pressure and temperature in the high-pressure side decrease and the pressure and temperature in the low-pressure side increase. After 4000 seconds, the compressor speed changes up and down and the trend of the system pressure and temperature follows up its speed. The system's dynamic behavior is shown in Fig.4 based on the pressure-enthalpy diagram.

The refrigerant mass flow rate is shown in Fig. 3 (c). There are over-shootings in the mass flow rate value when the compressor speed changes abruptly, but the mass flow rate keeps constant during the simulation. Each heat exchanger capacity changes with a change of the compressor speed as shown in Fig. 3 (d). The amount of increase on the IHX capacity is bigger than with that of the evaporator when the compressor speed increases. Because, the increase of the compressor speed results in its power consumption increase as shown in Fig. 3 (e) and the increased power which be input to the heat pump system should be rejected to the water-cooling system by heat transfer at the IHX. Furthermore, because the amount of increase in input power is bigger than the amount of increase in evaporator's cooling capacity, the system performance decrease when the compressor speed increases as shown in Fig. 3 (f).

The simulation results of the water system are shown in Fig. 5. The water temperature increases having a similar trend with that of the CO_2 heat pump system. After the water flows out from the Load, the water temperature increases because it absorbs the heat from the 'Load'. Then this water rejects its heat to the high-temperature thermal reservoir.

4. Limitations

The current air-to-air thermal management system which integrated a CO_2 heat pump with water cooling system demonstrate the dynamic behavior and performance varied with the CO_2 compressor rotation speed. The simulation shows reasonable results on the refrigerant pressure, temperature, heat exchangers capacity and COP corresponding with the change of compressor speed. However, it is not yet validated. Therefore, it is necessary for the simulation results to compare with experimental

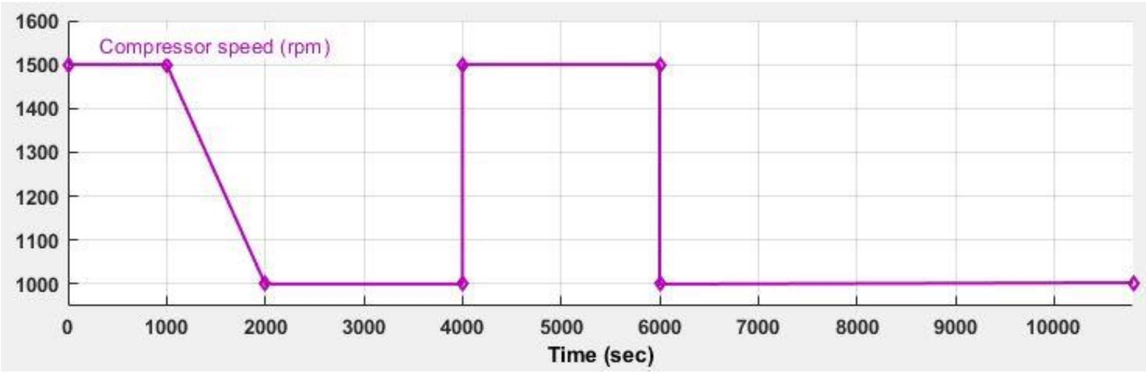

Fig. 2. Variation of the compressor speed.

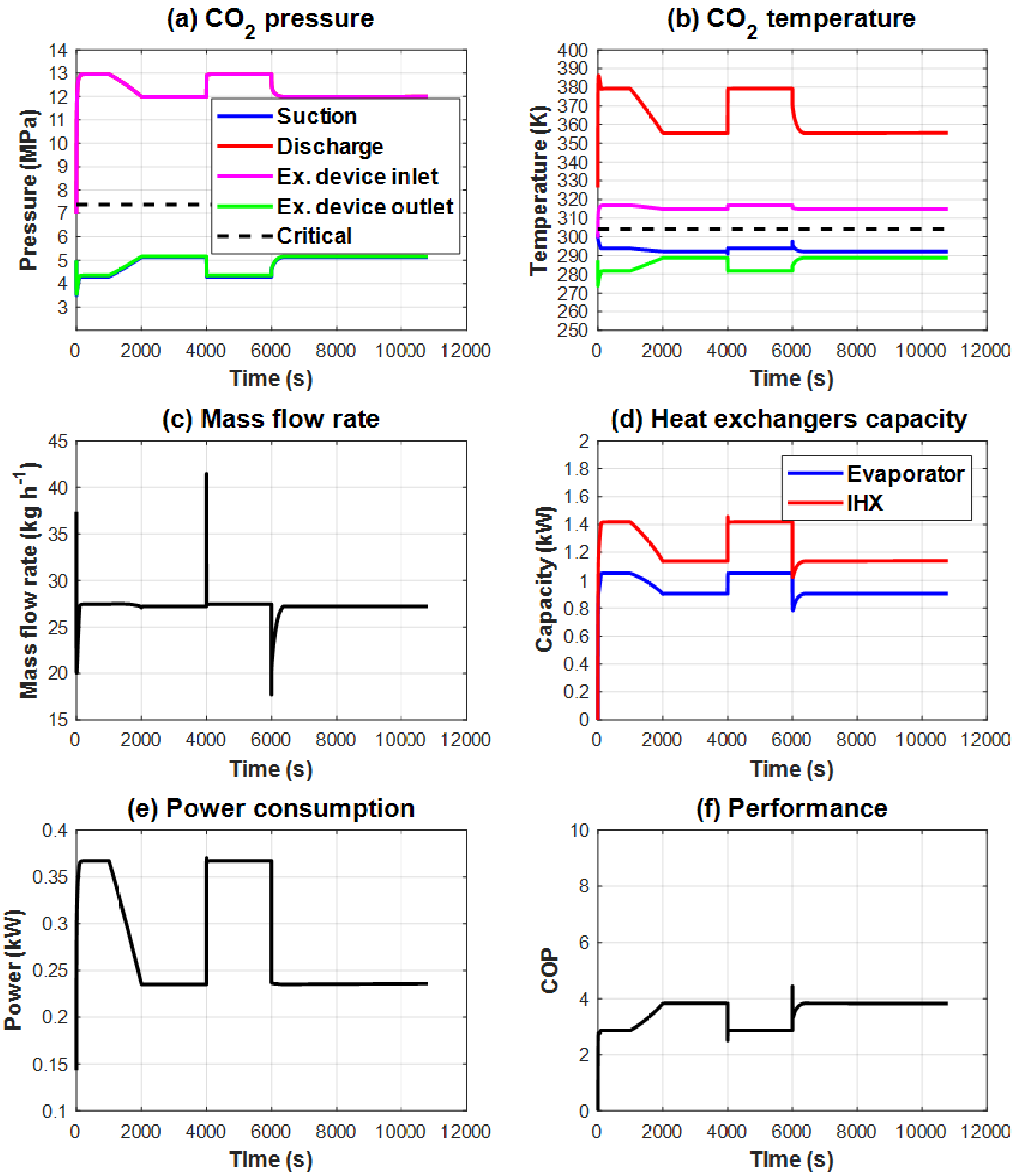

Fig. 3. Simulation results of the CO_2 heat pump system; (a) system pressure, (b) system temperature, (c) refrigerant mass flow rate, (d) heat exchanger capacity, (e) compressor power consumption, (f) system performance.

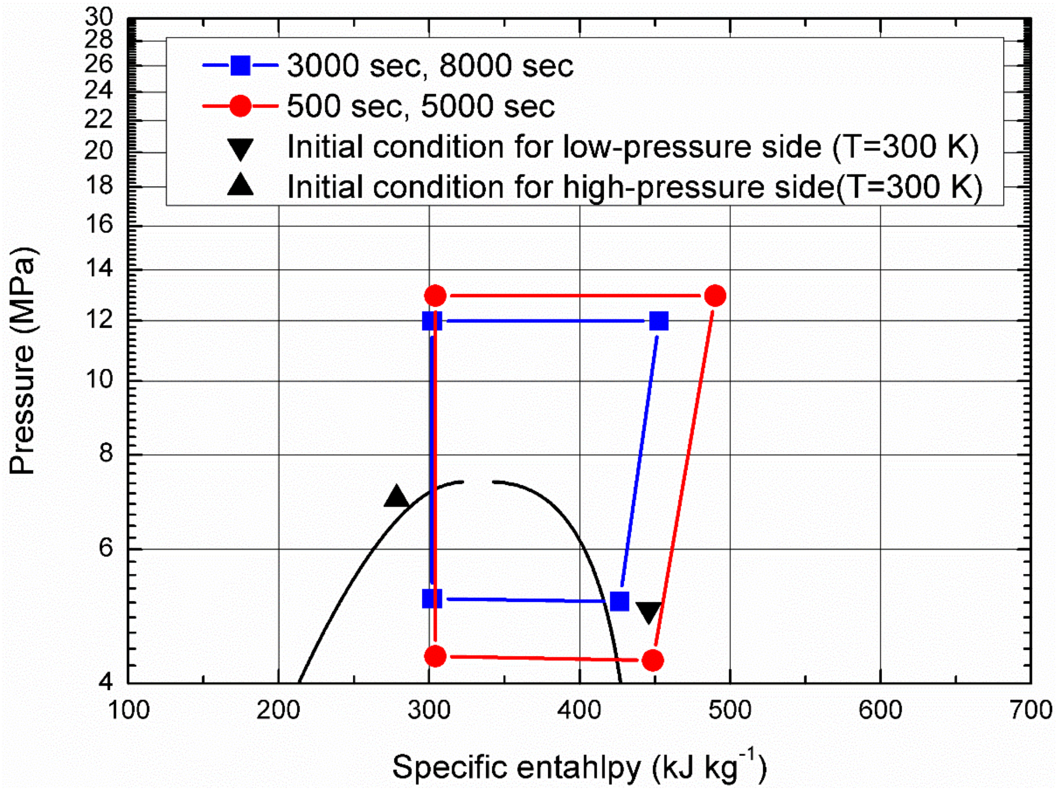

Fig. 4. Pressure-enthalpy diagram of the CO_2 heat pump system.

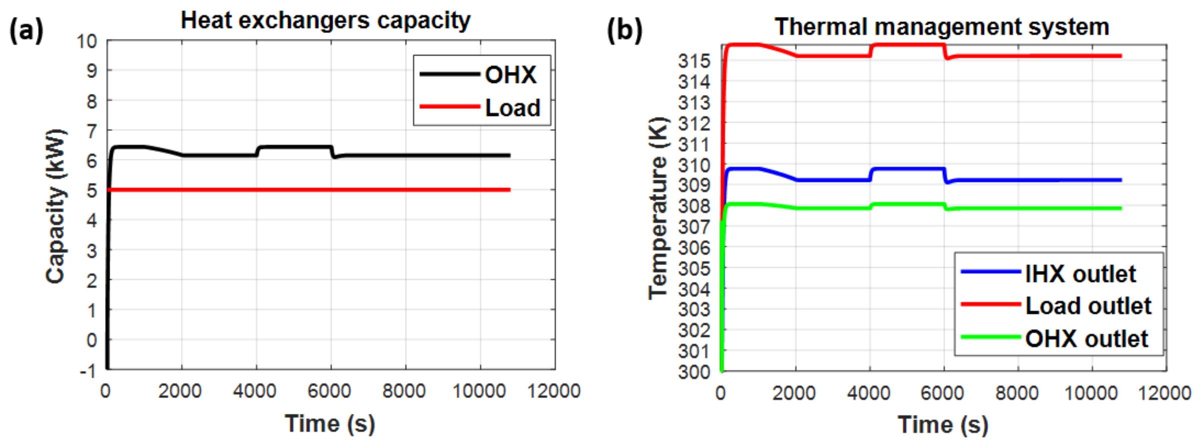

Fig. 5. Simulation results of the water-cooling system; (a) heat exchanger capacity, (b) water temperature.

results of a proper prototype or laboratory scale test-bench.

In the present model, the air-side heat transfer is set up in 'Thermal' domain. So, it is difficult to take into account the flow-dynamic effects caused by air-flow and its distribution. It is necessary to update the air-side of heat exchangers in detail for an air-to-air thermal system. Because the heat transfer rate of air-side has a dominant effect on the total system performance in air-to-air systems.

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