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## Modelica Based Dynamic Modeling of Water-Cooled Centrifugal Chillers

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### ABSTRACT

In this paper, the transient model of a water-cooled centrifugal chiller is developed in Modelica with Dymola and the TLK/IfT Library (TIL). The centrifugal compressor is modeled in detail based on the turbomachinery theory. The chiller capacity control is achieved by the combination of variable inlet guide vane (VIGV) and variable speed drive (VSD). The shell-and-tube heat exchangers are discretized based on the finite volume method with convenient and numerically efficient two-phase property evaluations. A thermal expansion valve (TXV) is used to regulate the pressure levels at the condenser and evaporator sides. These models are interconnected to form a centrifugal chiller system. The developed chiller model could be easily integrated to various control applications and it is proposed to validate the controller performance at the simulation phase prior to experimentation.

### 1. INTRODUCTION

Significant of energy consumption for buildings is due to the heating, ventilation and air conditioning (HVAC) systems. Commercial and residential buildings consume over 40% of the primary energy supply in the United States, representing over 72% of all electrical power generation, 55% of natural gas consumption and more than 1/3 of total CO<sub>2</sub> emissions (Holness 2009). Improving the efficiency of building HVAC system is thus critical for energy and environmental sustainability. In central HVAC system of commercial buildings, the chilled-water system supplies chilled-water for the cooling needs of all the building's air-handling units. The system includes a water pump to circulate the chilled water through the evaporator and throughout the building for cooling. Another water loop from the chiller rejects heat to the condenser water, where another pump circulates the heated water to the cooling tower and cooled back. For large system applications, centrifugal chillers have become the most widely used chiller unit for its high efficiency, high reliability, and low maintenance. Among the major devices in a chilled-water system, centrifugal chiller is the most energy-consuming device. Chiller efficiency can be improved via advanced controller design and fault detection, both demanding for high-fidelity dynamic simulation models.

There has been significant amount of work done on steady state modeling for centrifugal chillers (Braun et al. 1987; Beyene et al. 1994; Gordon et al. 1995; Browne and Bansal 1998; Swider 2003; Saththasivam and Ng 2008). Dynamic modelling for centrifugal chillers is critical for developing advanced control technology and fault detection and identification (FDI) scheme. Dynamic models for vapor compression cycle have been previously studied. Dhar and Soedel (1979) modeled a reciprocating compressor with polytropic efficiency, compressor housing and refrigerant mixing with the oil in the sump were included. The heat exchanger was modeled based on moving boundary method, average state was assumed for each phase region due to the assumption of fully mixing condition. The expansion valve was modeled based on isenthalpic process and the dynamics of the sensing bulb were considered. The simulation models were tested with shut-down and start-up and different operations. Popovic and Shapiro (1998) developed a centrifugal compressor model based on control volume approach, i.e., the impeller and diffuser were considered separately as control volumes. The impeller was modeled in detail based on the Euler

equation. The diffuser was modeled as ideal without any losses. Three refrigerant states were considered at the impeller inlet, the diffuser inlet (compressor mid-state) and the diffuser outlet. Two empirical relations were used to predict the mass flow rate and the exit state enthalpy, which were specific to the refrigerant R236ea. The model was compared to a modified Braun's model (Braun et al. 1987) and improvement was reported. Svensson (1999) developed a liquid chiller model based on lumped-parameter modeling of the heat exchangers. Step changes in the condenser-side water flow rates were studied in order to simulate the system performance under disturbances. Wang and Wang (2000) developed a mechanistic, single stage and two-stage centrifugal chiller models. The centrifugal compressor was modeled based on the Euler turbomachinery equation, the energy equation and the equations of impeller velocity component relations. The energy performance of the chiller was simulated by considering compressor polytropic efficiency, hydrodynamic losses and mechanical and electrical losses. However, the condenser and evaporator were modeled based on lumped parameter method and the heat transfer is calculated based on effectiveness model. Grace and Tassou (2000) developed a dynamic liquid chiller model based on the work of MacArthur and Grald (1987) in terms of the heat exchangers. The dynamic model of a semi-hermetic reciprocating compressor was developed based on the first law of thermodynamics applied to a lumped control volume. The expansion valve was modeled based on a simple orifice flow model and a detailed remote phial model. The chiller model was validated with a water-to-water chiller test facility. The model predicted higher start-up condenser and evaporator pressures than the measurements. The modeled dynamics appeared faster than the experiment result for convergence to the steady-state. The evaporator-side model was better than the condenser-side in terms of predicting the transient responses.

More recently, Bendapudi et al. (2005) developed a dynamic centrifugal liquid chiller model with flooded-type shell-and-tube heat exchangers. The compressor model was with constant speed and the capacity control was achieved by variable inlet guide vanes (IGV). The polytropic efficiency was determined based on steady-state data and was regressed as a second-order polynomial of the volumetric flow rate and polytropic work. Another regression model was applied to determine the maximum capacity condition, i.e., with wide-open IGV, and the actual mass flow rate was then computed based on a linear relationship of the maximum mass flow rate and the IGV position. However, the polytropic efficiency map is only effective after the startup process (up to 90 seconds), a constant polytropic efficiency was assumed at the start-up process and then switched to the regression model afterwards, which makes the start-up prediction less accurate. The heat exchangers were modeled based on the finite volume method with counter-flow assumption. To match the sub-cooling measurement, the initial numerical charge prediction was artificially tuned based on several simulations. The final best initial charge was found to be 75kg and the overall system charge seemed to be maintained about 50kg after the numerical transients.

To the author's best knowledge, for chiller system simulation, due to complex nature of the refrigeration cycle, it seems that all the existing simulation models developed for centrifugal chillers include some simplification for some component(s). For the centrifugal compressor, the dynamic performance was often approximated based on the compressor characteristic map rather than parameterized dynamic models (Browne and Bansal 2002; Bendapudi et al. 2005). Such approximation is sufficient for simulating the steady-state performance of the chiller, whereas may lead to difficulty in simulating transient processes such as startup, load change and shutdown. Some other works presented dynamic models for the centrifugal compressors (Wang and Wang 2000), but modeled the heat exchangers with lumped formulations. The objective of this study is to improve the existing work towards comprehensive dynamic model for the whole chiller to achieve quality simulation performance for all components of centrifugal chiller. The significance of the proposed research includes the following: 1) The dynamic characteristics of the centrifugal compressor was modeled based on the geometry and detailed loss calculations 2) The shell-and-tube heat exchangers were modeled based on distributed parameter approach. 3) The model development time was greatly reduced by using the Modelica language and the Dymola simulation platform.

## 2. DYNAMIC CHILLER MODELING

Figure 1 is a schematic of the water-cooled centrifugal chiller system which consists of four major components: a centrifugal compressor, a condenser, an expansion valve, and an evaporator. In general, the overall chilled-water system consists of one circuiting refrigerant loop and two water loops. The first water loop is circuiting between the condenser and the cooling tower. The second water loop is circuiting between the evaporator and the air handling units (AHU) and produces chilled-water for the cooling coil.

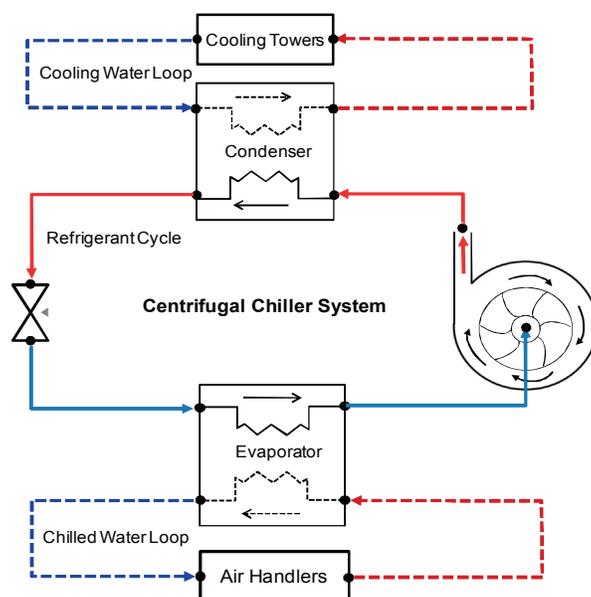


Figure 1: Schematic of water-cooled centrifugal chiller system

In this study, the transient model of a water-cooled centrifugal chiller is developed in Modelica with Dymola and the TLK/IfT Library (TIL). Modelica is an open-source object-oriented and equation based modeling language that features non-causal and flexible model-reuse (Åström et al. 1998; Mattsson et al. 1998; Fritsson 2004) Dymola (Dynamic Modeling Library) is an integrated multi-engineering modelling and simulation platform that supports Modelica. TIL is a Modelica library designed for steady-state and transient simulation of thermo-fluid systems such as heat pump, air conditioning, refrigeration or cooling systems (Richter 2008). In the model development, the chiller system is formulated in DAE, as a natural mathematical form to represent a physical system.

## 2.1 Centrifugal Compressor

The centrifugal compressor is the quickest responding device in the chiller plant. The system balance of centrifugal compressor is achieved through capacity control. In this study, the capacity control is manipulated by adjusting the IGV position and input torque. Based on the classic work on axial-flow compressor by Greitzer (1976), Hansen et al. (1981) introduced the initial model for centrifugal compressors. Fink et al. (1992) extended the Greitzer model and included centrifugal compressors with variable rotor speed. Recently, Gravdahl and Egeland (1999) provided a comprehensive summary of compressor modeling and control including Greitzer and Fink et al.'s original work. In addition, the effects of incidence and fluid friction losses mentioned in (Ferguson 1963; Watson and Janota 1982) were also included. In our work, the compressor model was developed based on these methods summarized in Gravdahl and Egeland (1999). Figure 2 schematically depicts the centrifugal model with boundary conditions and capacity control. It is important to note that the model described in Gravdahl and Egeland (1999) was intended for the typical air compression system, which is an open-loop operation with dry air as the working medium. However, for the chiller system, the working medium is generally refrigerant that works under multi-phase conditions, which makes it more complex and difficult for the underlying modeling framework and consistent numerical initialization.

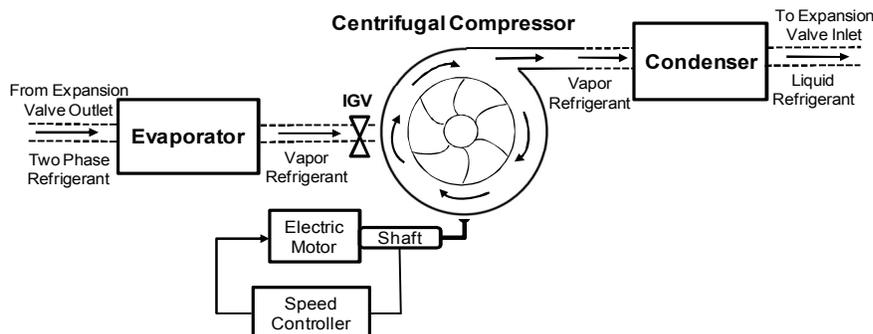


Figure 2: Schematic drawing of centrifugal compression system in chiller

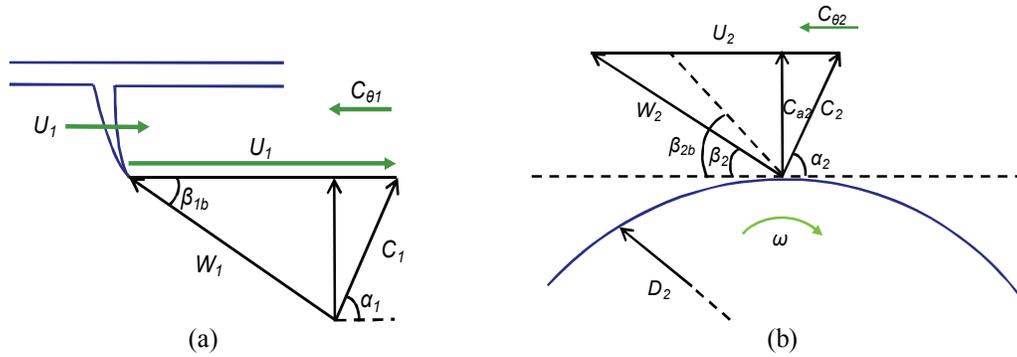


Figure 3: Refrigerant vapor flow angle at where (a) the inducer and (b) the impeller tip

According to Gravdahl (2002), based on the isentropic analysis of ideal gases, the characteristic of the centrifugal compressor could be expressed as

$$\psi_c(\omega, \dot{m}) = \left[ 1 + \frac{\mu r_2^2 \omega^2 - \frac{r_1^2}{2} (\omega - \alpha \dot{m})^2 - k_f \dot{m}^2}{c_p T_1} \right]^{\frac{\kappa}{\kappa-1}} \quad 1(a)$$

$$\alpha = \frac{\cot \beta_{1b}}{\rho_1 A_1 r_1} \quad 1(b)$$

where  $\psi_c(\omega, \dot{m})$  is the pressure rise ratio,  $\omega$  is the rotation speed of the drive motor,  $m$  is the mass flow rate. Unlike the air compression systems (Gravdahl 2002),  $\alpha$  is not a constant in chiller system, rather, it is a function of the inlet refrigerant density, which depends on the operation of chiller control at the evaporator side.  $\beta_{1b}$  is the refrigerant outflow angle at inducer,  $A_1$  is the cross section area,  $r_1$  is the average inducer radius. Also,

$$\mu = \sigma \left( 1 - \frac{\cot \beta_{2b} \dot{m}}{\rho_1 A_1 r_1 \omega} \right) \quad 1(c)$$

where  $\beta_{2b}$  is blade angle at impeller tip,  $r_2$  is the radius at impeller tip,  $k_f$  is the friction coefficient,  $T_1$  is the inlet stagnation temperature,  $c_p$  is the constant-pressure specific heat capacity,  $\kappa$  is the heat capacity ratio and  $\sigma$  is the slip factor.

For the cyclic operation of chiller, the inlet refrigerant of the compressor is often saturated vapor. The vapor flow angle is schematically shown in Figure 3. The mass flow rate of vapor entering the impeller eye (inducer) can be expressed as

$$\dot{m}_r = \rho_r A_1 C_1 \quad 1(d)$$

$$U_1 = \frac{D_1}{2} \omega \quad 1(e)$$

$$D_1^2 = \frac{1}{2} (D_{i1}^2 + D_{h1}^2) \quad 1(f)$$

where  $\dot{m}_r$  is the refrigerant vapor flow rate,  $A_1$  is the cross section area,  $C_1$  is the absolute velocity.  $D_{i1}$  and  $D_{h1}$  are the diameters of the inducer tip and hub casing, respectively. The refrigerant vapor leaves the impeller from the tip with velocity  $C_2$ . At impeller tip, the diameter is  $D_2$  and the tangential velocity is  $U_2$ . The compressor torque can be determined as

$$\tau_c = |\dot{m}_r| r_2 \sigma U_2 \quad 1(g)$$

where

$$U_2 = \omega r_2 = 2\pi N r_2 \quad 1(h)$$

from which the momentum balance equation is formulated.

$$J \frac{d\omega}{dt} = \tau_t - \tau_c \quad 1(i)$$

where  $\tau_t$  and  $\tau_c$  are the motor drive torque and compressor torque, respectively.  $J$  is the momentum of inertia of the spool and rotor. The calculation of isentropic efficiency is based on detailed compressor geometry parameters with consideration of incidence losses and friction losses at impeller and diffuser, respectively and various other losses (Gravdahl and Egeland 1999).

## 2.2 Condenser and Evaporator

Condenser and evaporator modeling requires quality heat exchanger models. For the shell-and-tube heat exchangers in a centrifugal chiller system, the differences between the finite volume (FV) and moving boundary (MB) methods have been extensively studied by Bendapudi et al. (Bendapudi, Braun et al. 2008). It was found that the FV method was more robust for start-up and load change scenarios for both individual components and the complete chiller system. The MB method was also found to be less accurate in terms of the charge prediction. The FV method has about 20% increase in computation time, however, such trade-off is worthwhile considering the improvement in accuracy for the obtained dynamic models. Therefore, the FV method has been adopted in this study.

Figure 4 shows the schematic of the condenser model in Dymola. The evaporator model is quite similar. These heat exchangers were modeled based on the counter-flow assumptions. The water flows are inside the tubes for both condenser and evaporator. With the concentric heat exchanger assumed, the shell-side heat transfer area is computed from the outer surface area of the water tubes and the surface enhancement factor.

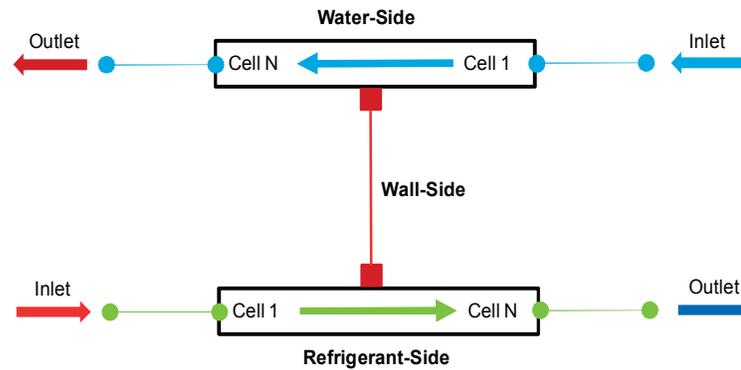


Figure 4: Dymola layout of the condenser model

For the refrigerant side, the basic element is the RefrigerantCell. In each RefrigerantCell, the mass and energy balance can be established as (Richter 2008)

$$\frac{d\rho}{dt} = \left( \frac{\partial \rho}{\partial h} \right)_p \frac{dh}{dt} + \left( \frac{\partial \rho}{\partial p} \right)_h \frac{dp}{dt} \quad 2(a)$$

$$V \frac{d\rho}{dt} = \dot{m}_{in} + \dot{m}_{out} \quad 2(b)$$

$$\frac{dh}{dt} = \frac{1}{M} \left( \dot{m}_{in} (h_{in} - h) + \dot{m}_{out} (h_{out} - h) + \dot{Q} + V \frac{dp}{dt} \right) \quad 2(c)$$

where  $\rho$  is the mean density,  $V$  is the volume,  $h$  is the mean specific enthalpy,  $p$  is the pressure, and  $\dot{m}_{in}$  and  $\dot{m}_{out}$  are inlet and outlet refrigerant mass flow rate, respectively.  $M$  is the refrigerant mass,  $h_{in}$  and  $h_{out}$  are the inlet and out specific enthalpies, respectively, and  $\dot{Q}$  is the heat transferred to the cell. For the water-side, the basic element is the LiquidCell, the basic cell equations are similar to those of the RefrigerantCell except that the dynamic mass and energy equations are transformed with temperature, rather than the pressure and enthalpy pair, as the state variable.

## 2.3 Thermal Expansion Valve

The thermal expansion valve (TXV) is a key component for the refrigeration cycle, which is used to control the superheat at the outlet of the evaporator by adjusting the system pressures levels. The TXV is controlled by

adjusting the valve flow area via a thermal sensing bulb. The dynamics of the TXV is dominated by the response of the sensing bulb. The dynamic response of the bulb is usually first-order with a time constant. The change of the bulb temperature will result in a change of the refrigerant pressure inside the bulb and causes the flow area to change accordingly. A detailed computation of the flow area will depend on the geometry of the valve. In the current chiller model development, a simple expansion valve model is adopted by assuming a quadratic relation of the pressure difference across the valve and the corresponding mass flow rate. The effective flow area could be computed internally based on certain geometry parameters, or it could be provided as an external input.

Assuming one-dimensional flow, for the control volume drawn in Fig. 4, it is assumed that there is no transient mass storage across the TXV. And the throttle process is isenthalpic, i.e.,  $h_{in} = h_{out}$ . The mass flow rate from the TXV is

$$\dot{m}_{in} = \dot{m}_{out} = A_{eff} \sqrt{2\rho_{in}(p_{in} - p_{out})} \quad (3)$$

where  $\dot{m}_{in}$  and  $\dot{m}_{out}$  are the inlet and outlet mass flow rates, respectively,  $h_a$  and  $h_b$  are the inlet and outlet specific enthalpies, respectively,  $p_{in}$  and  $p_{out}$  are the inlet and outlet pressures, respectively,  $\rho_{in}$  is the inlet refrigerant density, and  $A_{eff}$  is the effective flow area of the valve.

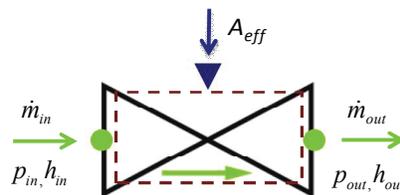


Figure 5: Control volume of the TXV model

### 3. SIMULATION STUDY

When the four components were connected together, simulation failed to start due to initialization problem. To obtain consistent initial conditions of systems of differential-algebraic equations (DAE) for centrifugal chiller simulation, a direct initialization method was proposed along with a three-step preprocessing scheme to resolve the consistent initialization problem (Li, Li et al. 2010).

Simulation study was conducted to investigate the dynamic behavior and steady-state performance of the centrifugal chiller model with R134a as the refrigerant. Two inputs were applied to the compressor model for capacity control, i.e., the IGV position and the input torque. The inlet water flow rates of the condenser and evaporator were 16.7kg/s and 13.2kg/s, respectively. The inlet water temperatures of the condenser and evaporator were 22°C and 16°C, respectively. The initial IGV position and the input torque were set to be 70% and 600N·m, respectively. Figure 6 shows the Dymola layout of the centrifugal chiller model. To test the dynamic responses of the chiller model, ramp inputs were applied to the compressor inputs after the initial numerical transients as shown in Figure 7(a).

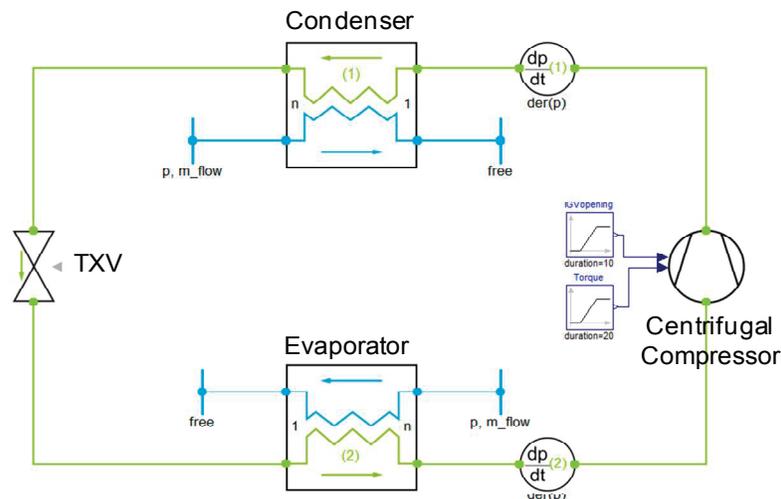


Figure 6: Dymola layout of the centrifugal chiller model

Figures 7(b) to 7(d) present the transient and steady-state performance of the chiller model in terms of refrigerant vapor flow rate at compressor inlet, compressor isentropic efficiency, superheat and sub-cooling temperature, chilled-water temperature, and system pressures.

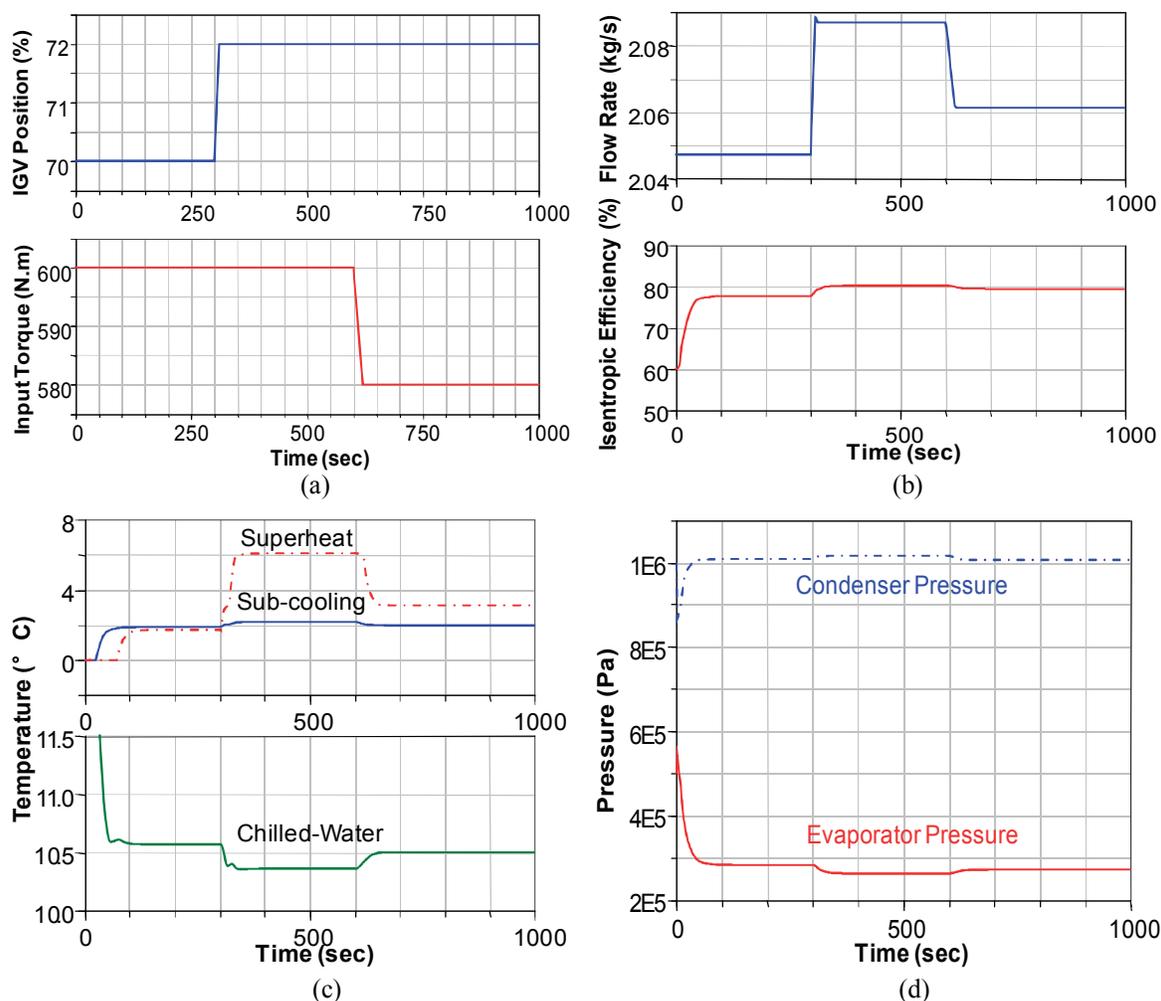


Figure 7: Simulation study of the dynamic centrifugal chiller model where (a) is the ramp changes of IGW position and input torque, (b) is the mass flow rate at the compressor inlet and isentropic efficiency of the centrifugal compressor, (c) is the superheat and sub-cooling temperature at the refrigerant side and chilled water temperature from the evaporator and (d) is refrigerant pressures at condenser and evaporator.

#### 4. CONCLUSIONS

In this paper, a Modelica based transient model is developed for a centrifugal water-cooled chiller using Dymola and TIL. The dynamic model of centrifugal compressor is included by considering the mass, momentum and energy balances with various losses throughout the compressor. The shell-and-tube heat exchanger is developed based on a standard tube-and-tube heat exchanger from TIL with major modifications for the calculations of heat transfer and geometry of the shell-side. A TXV model is used to regulate the refrigerant flow rate with a simple treatment to the effective flow area. Simulation study is conducted for a nominal chiller operation with dynamic performance tests by changing the compressor IGW positions and the input torque in sequence. The simulation results from the chiller model yield reasonable predictions. The developed chiller model will be further improved with experiment study in near future.

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