Dynamic modeling and simulation of a centrifugal chiller based on MBSE

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Abstract. Product design by a virtual concept has always been a major subject of many of engineers and manufacturers. Issues should be how close to simulate to the real world by virtual ways. Model Based Systems Engineering (MBSE) in a digital modeling environment provides a methodology to make a virtual mock up easily. In the study, 1D/3D co-simulation was performed on a centrifugal cooling system to check how much accuracy was improved. A specific part of the fields which require to be solved in detail was modeled in 3D for co-simulation while the rest of system remained 1D model. The 1D and 3D solvers run at their respective field domains in parallel and share information. The study considered the multi physics for the components of compressor to refrect the effects of interaction between different physics. An electro magnetics, refrigerant flow & fluid dynamics, heat transfers, power electrics be calculated together to simulate them simultaneously. All 3D models converted to meta model to match time scale with 1D model. This paper shows a case study where a virtual design is successfully applied to the centrifugal chiller system, and dynamic behavior of start up at different cooling loads was simulated and investigated. It can be concluded that the approach by co-simulation as a process of MBSE be able to use to the imposed dynamic behavior on the system. The simulation results under a various load conditions showed that it maintained less than 5% compared with experimental data during from start up to the steady state, moreover, co-simulation showed effective to get a better accuracy.

Keywords. 1D/3D co-simulation, MBSE, virtual product design, multi physics, refrigeration system, centrifugal chiller.
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1. Introduction

A centrifugal chiller is an air conditioning equipment that utilizes a centrifugal compressor in a vapor-compression refrigeration cycle. It is popular because they have relatively few moving parts, easy to maintain and long-lived. However, test facilities at a system level are not to be prepared well because of too high heat capacity to be rejected by facility for measuring capacity of a chiller. Moreover, there has a broad spectrum of industrial needs, it is not easy to check for all of quality issues before factory out. The motivation to make a virtual model would be enough to consider at the point of cost and quality. Althrough the product design by virtual concept has been introduced decades ago, it faced a new era by the help of AI technologies under the big umbrella of Digital Transformation(DX). There has been had not much engineering platforms and tools to make a strong simulation model, which was not an environment to use sufficient interfacing technology to communicate well under the condition of multiple physics, different dimensions and different language codes. Before DX, many of simulation model has handicaps to simulate a real world due to not refined data set or there was less ways to melt it into the simulation model even there has. The research focused on dynamic modeling and simulation of the transient characteristics, in which stop–start operating and capacity modulation through various compressor speed, for control purposes. There can find a lot of studies for dynamic modeling, Bendapudi et al.[1] has shown a comparative literature study of refrigeration system for dynamic behaviors. From the reviews, Dhar and Soedel [2] takes a moving boundary approach in the heat exchangers as a coupled pair of lump representing the liquid and vapor phases. He tried to predict the start-up characteristics of compressor by identifying conditions that could lead to liquid slugging, and also as an aid to the system designer to find an optimized condition. Wang [3] presented that a detailed model of the centrifugal compressor is developed from the first principles. He considered the inlet guide vane or pre-rotation vanes form of capacity control, however, the heat exchangers are treated as single entity lumped effective-NTU. Even there could see a plenty of dynamic models for refrigeration system, most of approach are for a single dimensional model, not included the 3D co-simulation. Moreover, there has not been treated it including the interactions by multi-physics. The study considers 1D/3D co-simulation by the link of different physics at every time interval as time goes on. However, there is no need to make 3D model for all components, in the study, we focused on the modeling for main component of compressor. Heat exchangers are treated as 1D model, it was modeled by a moving boundary method, which is one of single dimensional approach to simulate through the values of inlet/outlet temperatures of chilled and cooling water together with refrigerant properties. The object being modeled for simulation is a commercially available centrifugal chillers, which has two stage compression, magnetic bearings, BLDC motor with variable frequency driver(VFD) and electronic expansion valves(EEVs). Because the 3D models are always heavier than 1D models, co-simulation has to runs at a single dimension basis after taking a reduced order model from 3D full model. The simulation results were compared with the experimental data or the test-based compressor map for validation. The final goal of the study is to make a digital mock up, it can be contributed to development of a scenario based control strategy, performance prediction and quality check by a virtual way.

2. Dynamic modeling approach

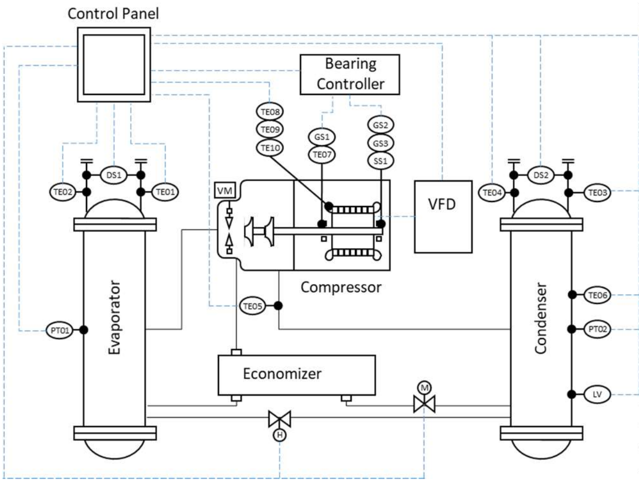
2.1 1D modeling for the system

Simulation study was conducted for the centrifugal chiller, with Modelica by Dassualt company. The model calculates thermodynamic properties according to the variation of input parameters at each phases and states. Fig.1 shows a schematic diagram of the system, included instruments and piping. When we plan a model, measuring data from test and experiment set the same parameters from the output of simulation to be matched them in advance for the purpose of adding a control software in the loop simulation(SILS).

The system model should be a virtual model whose dynamic behavior is very similar to the real centrifugal chiller in order to deploy for a failure mode analysis. In the model development, the system is formulated in equations to represent physical system. The basic concept of modeling comes from Thermosys, Alleyne Research Group[4, 5], they suggest a simplified one-dimensional approach to the process of heat transfer and a flow path of heat exchangers, it is capable to simulate a change of an occupied volume according to a phase change of refrigerant. The Thermosys is worth understanding the concept of dynamic approaches of refrigeration system to common users, however, it has to be modified adequately as to apply it for individual's purpose. Shin[6] improved the stability of Thermosys by solving the possibility of diverging caused by initial values. The governing equations be derived from the conservation of mass and energy with assuming one-dimensional refrigerant flow. Terms containing partial derivatives of normalized length can be changed to normal derivatives through the rule of Leibnitz. Since temperature, pressure and the occupied length of phases have the terms of time derivatives, the governing equation represents as a state vector “x” as follows.

$$\mathrm{Z}(x,u)\dot{x} = \mathrm{f}(\dot{x},u) \tag{1}$$

The function “f” at the right hand side of equation (1) represents a force vector, it denotes an energy term entering and leaving through a control volume at the equation of energy conservation, at the same time, it can be considered as a variation of mass at the equation of mass conservation. The equation represents dynamic states because derivative term of the state vector x is not be zero. The heat exchangers including economizer, where contained the most of refrigerant during operation, effect a lot on dynamic behaviors than any other components of chiller system. The dynamic behaviors of compressor and expansion valve mainly come from variation of pressure by dynamics of refrigerant besides heat exchangers came from heat transfer related by heat



Symbol	Description
TE01	Chilled Water Inlet Temp. Sensor
TE02	Chilled Water Outlet Temp. Sensor
TE03	Cooling Water Inlet Temp. Sensor
TE04	Cooling Water Outlet Temp. Sensor
TE05	Comp. Discharge Temp. Sensor
TE06	Sub-Cooler Temp. Sensor
TE07	Comp. Motor Bearing Temp. Sensor
TE08	Motor Wiring Temp. Sensor (R)
TE09	Motor Wiring Temp. Sensor (S)
TE10	Motor Wiring Temp. Sensor (T)
PT01	Eva. Pressure Transmitter
PT02	Cond. Pressure Transmitter
SS1	Comp. Motor Speed Sensor
SG1	Bearing Gap Sensor (I)
SG2	Bearing Gap Sensor (M)
SG3	Bearing Gap Sensor (Thrust)
VM	Vane Motor
DS1	Diff. Pressure Switch (Chilled Water)
DS2	Diff. Pressure Switch (Cooling Water)
LV	Ref. Level Sensor

Fig. 1 - Schematics of water cooled centrifugal chiller

capacity. The meaning is that there has huge gaps of response time between components. Moreover, it is not easy to make a dynamic model for compressor because of its complexity [7]. Since the compressor model is treated as an abnormal state from the viewpoint of time lapse, 3D modeling of the main components such as motor, inverter driver and impeller was performed for dynamic analysis, followed by co-simulation with the 1D system model.

2.2 1D modeling for heat exchangers

There are two different approaches to make a model for a heat exchanger in refrigeration cycle, which are finite volume method(FVM) and moving boundary method(MBM). In the view of time required at the based on the same calculation results, MBM takes shorter than FVM [8].

Tab. 1 – Nomenclature.

Symbols		**Subscript**	
P	pressure [Pa]	c	condenser
T	temperature [K]	t	total
x	quality	r	refrigerant
h	enthalpy [J/kg^{-1}]	w	water
u	internal energy [Jkg^{-1}]	in	inlet
Cp	specific heat [Jkg^{-1}K^{-1}]	out	outlet
Z	height [m]	f	saturated liquid
ρ	density [kgm^{-3}]	g	saturated vapor
m	mass [kg]	sat	saturated
ṁ	mass flow rate [kgs^{-1}]	sub	subcooled
A	area [m^2]	u	upside
S	surface [m^2]	d, dn	downside
V	volume [m^3]		
U	overall heat transfer coefficient [Wm^{-2}K^{-1}]		

The main cause comes from how many segments to be considered for calculation. MBM switches set of equations according to the change of phase boundary, and it is relatively simple to calculate heat exchager at transient conditions, besides FVM should be divided it many of small nodes along the flow direction. In case of need to taking the same or similar calculation speed of a model as to match a real product, MBM is better. However, when we takes MBM, there are big issues to find an exact boundary of phase change by comparing the exit enthalpy with the enthalpy at saturation line for good result [9]. At the beginning of caculation, there should have information of length of phase to initiate calculation. In the study, the volume of refrigerant is calculated by the sensing data from a refrigerant level sensor equipped at the heat exchangers. There need five models to represent different phases in heat exchangers, where three models for condenser and two for evaporator. In case of condenser, superheat region can be neglected because the length of superheat region is not appeared significantly,and the capacity of heat at the region is very small. For the evaporator of flooded type, a satuarted refrigerant, where be treated as a superheatd gas with 1.0 of quality, flows into the inlet of compressor because evaporator be designed as the role of accumulator at the same time. Finally, the condenser model can be described as governing equations for two zones, which are two-phase and subcooled region, it represented as equation (2) and (3). Moreover, only one governing equation be needed to describe two-phase zone for evaporator. It is shown as equation (4).

$$\begin{bmatrix} (\bar{\rho}_{r,sub}-\bar{\rho}_{r,sat})\dfrac{\mathrm{d}V_{sub}}{\mathrm{d}z} & \dfrac{\partial\bar{\rho}_{r,sat}}{\partial P_c}V_{sat} & 0 & 0 & 0 & 0 \\ (\bar{\rho}_{r,sub}h_{r,f}-\bar{\rho}_{r,sat}\bar{u}_{r,sat})\dfrac{\mathrm{d}V_{sub}}{\mathrm{d}z} & \dfrac{\partial\bar{\rho}_{r,sat}}{\partial P_c}V_{sat}\bar{u}_{r,sat}+m_{r,sat}\left(\bar{x}\dfrac{\mathrm{d}u_{r,g}}{\mathrm{d}P_c}+(1-\bar{x})\dfrac{\mathrm{d}u_{r,f}}{\mathrm{d}P_c}\right) & 0 & 0 & 0 & 0 \\ \bar{\rho}_{r,sub}\dfrac{\mathrm{d}V_{sub}}{\mathrm{d}z}(\bar{u}_{r,sub}-h_{r,f}) & 0 & m_{r,sub} & 0 & 0 & 0 \\ 0 & 0 & 0 & m_{w,sub}C_{p,w} & 0 & 0 \\ 0 & 0 & 0 & 0 & m_{w,sat,u}C_{p,w} & 0 \\ 0 & 0 & 0 & 0 & 0 & m_{w,sat,d}C_{p,w} \end{bmatrix} \begin{bmatrix} \dot{z} \\ \dot{P}_c \\ \dot{\bar{u}}_{r,sub} \\ \dot{\bar{T}}_{w,sub} \\ \dot{\bar{T}}_{w,sat,u} \\ \dot{\bar{T}}_{w,sat,d} \end{bmatrix} = X_1 \quad (2)$$

Where,

$$X_1 = \begin{bmatrix} \dot{m}_{r,in} - \dot{m}_{r,out} \\ \dot{m}_{r,in}h_{r,in} - \dot{m}_{r,out}h_{r,f} - U_{sat}\left(A_{t,up} - A_{sub}(z)\right)\left(\bar{T}_{r,sat} - \bar{T}_{w,sat,u}\right) - U_{sat}A_{t,dn}\left(\bar{T}_{r,sat} - \bar{T}_{w,sat,d}\right) \\ 2\dot{m}_{r,out}\left(h_{r,f} - h_{r,sub}\right) - U_{sub}A_{sub}(z)\left(\bar{T}_{r,sub} - \bar{T}_{w,sub}\right) \\ 2\dfrac{S_{t,sub}}{S_{t,dn}}\dot{m}_{w,sat,d}C_{p,w}\left(T_{w,in} - \bar{T}_{w,sub}\right) + U_{sub}A_{sub}(z)\left(\bar{T}_{r,sub} - \bar{T}_{w,sub}\right) \\ 2(1 - \dfrac{S_{t,sub}}{S_{t,dn}})\dot{m}_{w,sat,d}C_{p,w}\left(T_{w,in} - \bar{T}_{w,sat,u}\right) + U_{sat}(A_{t,up}\ A_{sub}(z)\left(\bar{T}_{r,sat} - \bar{T}_{w,sat,u}\right) \\ 2\dot{m}_{w,sat,d}C_{p,w}\left\{2\dfrac{S_{t,sub}}{S_{t,dn}}\bar{T}_{w,sub} + 2\left(1 - \dfrac{S_{t,sub}}{S_{t,dn}}\right)\bar{T}_{w,sat,u} - (\bar{T}_{w,sat,d} + T_{w,in})\right\} + U_{sat}A_{t,dn}\left(\bar{T}_{r,sat} - \bar{T}_{w,sat,d}\right) \end{bmatrix}$$

$$\begin{bmatrix} \dfrac{\partial\bar{\rho}_{r,sat}}{\partial\bar{x}}V_{sat} & \dfrac{\partial\bar{\rho}_{r,sat}}{\partial P_c}V_{sat} & 0 & 0 \\ V_{sat}\left\{\bar{u}_{r,sat}\dfrac{\partial\bar{\rho}_{r,sat}}{\partial\bar{x}} + \bar{\rho}_{r,sat}(u_{r,g} - u_{r,f})\right\} & \mathrm{V}_{\mathrm{sat}}\left\{\bar{u}_{r,sat}\dfrac{\partial\bar{\rho}_{r,sat}}{\partial P_c} + \bar{\rho}_{r,sat}\left(\bar{x}\dfrac{\mathrm{d}u_{r,g}}{\mathrm{d}P_c} + (1 - \bar{x})\dfrac{\mathrm{d}u_{r,f}}{\mathrm{d}P_c}\right)\right\} & 0 & 0 \\ 0 & 0 & m_{w,sat,u}C_{p,w} & 0 \\ 0 & 0 & 0 & m_{w,sat,d}C_{p,w} \end{bmatrix} \begin{bmatrix} \dot{\bar{x}} \\ \dot{P}_c \\ \dot{\bar{T}}_{w,sat,u} \\ \dot{\bar{T}}_{w,sat,d} \end{bmatrix} = X_2 \quad (3)$$

Where,

$$X_2 = \begin{bmatrix} \dot{m}_{r,in} - \dot{m}_{r,out} \\ \dot{m}_{r,in}h_{r,in} - \dot{m}_{r,out}h_{r,f} - U_{sat}A_{t,up}(\bar{T}_{r,sat} - \bar{T}_{w,sat,u}) - U_{sat}A_{t,dn}(\bar{T}_{r,sat} - \bar{T}_{w,sat,d}) \\ 2\dot{m}_{w,sat,d}C_{p,w}(T_{w,in} - \bar{T}_{w,sat,u}) + U_{sat}A_{t,up}(\bar{T}_{r,sat} - \bar{T}_{w,sat,u}) \\ 2\dot{m}_{w,sat,d}C_{p,w}\{2\bar{T}_{w,sat,u} - (\bar{T}_{w,sat,d} - T_{w,in})\} + U_{sat}A_{t,dn}(\bar{T}_{r,sat} - \bar{T}_{w,sat,d}) \end{bmatrix}$$

$$\begin{bmatrix} (\rho_{r,f} - \rho_{r,g})\frac{\mathrm{d}V_{sub}(z)}{\mathrm{d}z} & (V_{tot} - V_{sub}(z))\frac{\mathrm{d}\rho_{r,g}}{\mathrm{d}P} + V_{sub}(z)\frac{\mathrm{d}\rho_{r,f}}{\mathrm{d}P} & 0 \\ (\rho_{r,f}u_{r,f} - \rho_{r,g}u_{r,g})\frac{\mathrm{d}V_{sub}(z)}{\mathrm{d}z} & \frac{\mathrm{d}}{\mathrm{d}P}(\rho_{r,f}u_{r,f})V_{r,f} + \frac{\mathrm{d}}{\mathrm{d}P}(\rho_{r,g}u_{r,g})(V_{tot} - V_{r,f}) & 0 \\ 0 & 0 & m_w C_{p,w} \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{P}_e \\ \dot{\bar{T}}_{w,sat,u} \end{bmatrix} = X_3 \tag{4}$$

Where,

$$X_3 = \begin{bmatrix} \dot{m}_{r,in} - \dot{m}_{r,out} \\ \dot{m}_{r,in}h_{r,in} - \dot{m}_{r,out}h_{r,out} - U_{evap}A_{tot}(\bar{T}_{r,sat} - \bar{T}_w) \\ \dot{m}_w C_{p,w}(T_{w,in} - T_{w,out}) + U_{evap}A_{tot}(\bar{T}_{r,sat} - \bar{T}_w) \end{bmatrix}$$

2.3 3D modeling for compressor

Three dimensional models have been built for the centrifugal compressor, which include an electric motor, inverter driver, two impellers and flow paths of refrigerant such as diffusers, return channel and volute, to reflect interaction of components. Compression system of refrigerant is shown in Fig. 2. The model consists of inlet tube, the 1st impeller, the 1st diffuser, return channel, multiple inlets from economizer, the 2nd impeller, the 2nd diffuser and volute. ANSYS CFX has been used to simulate the compressible turbulence flow and the mesh size is about 33 million of cells. The method of moving reference frame (MRF) is used for rotating impellers, also share stress transport (SST) model is used. The boundary conditions have been set to be able to simulate almost the same range of a real product, where 0.3~228Hz for rotating speed, 300~643kPa for compressor inlet pressure and 2.7~12.7kg/s for mass flow rate of compressor outlet. There has additional mass flow inlet from the economizer before the entrance of the 2nd impeller. The variable frequency driver (VFD) drives a motor by the method of pulse width modulation (PWM). There is a ripple in motor current wave during the PWM process to make the current be sinusoidal and this ripple is transformed as a component of torque ripple whose causes a additional performance degradation due to raise a vibration and noise on motor and add heat up to motor in rotor magnet or copper winding. The study considers motor and VFD together to find out how much influenced it to the system. The normal reduced order model(ROM) for motor is made by Maxwell from ANSYS after considering temperatures at the magnet of motor and stator winding firstly, then the model is connected for calculating total loss of motor to 1D VFD model where after adding a calculation result of PWM current. Twinbuilder provides a role of platform to integrate information of all 1D models. Finally, motor ROM can be obtained after combining normal ROM of motor and 1D VFD model. The calculation range of model for motor set to the same as the other models, whose are component models of compressor for co-simulation. Fig.3 shows a model for VFD and motor.

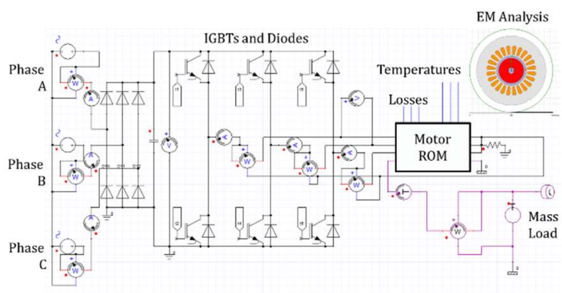


Fig. 3 – The model for VFD and motor.

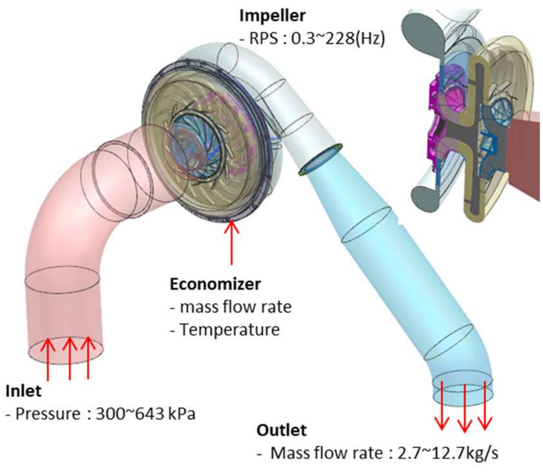


Fig. 2 – The model for a compression system.

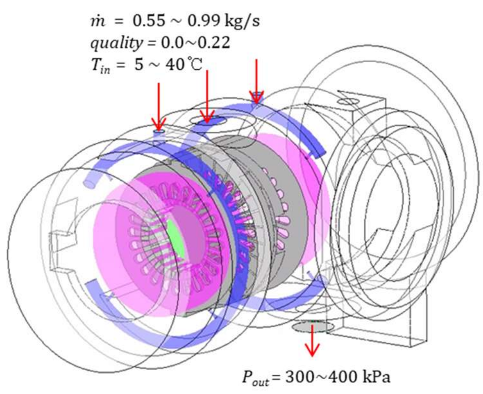


Fig. 4 – The cooling model for motor.

It is essential to cool down at all time for many parts of compressor inside, in the case, motor and inverter circuit are main component for cooling at the model. There is no need to consider for the model to bearings because the system has oil free magnetic bearings. Therefore, the CFD analysis for cooling by refrigerant has done only to the motor and inverter circuit as shown in Fig.4 and Fig5.

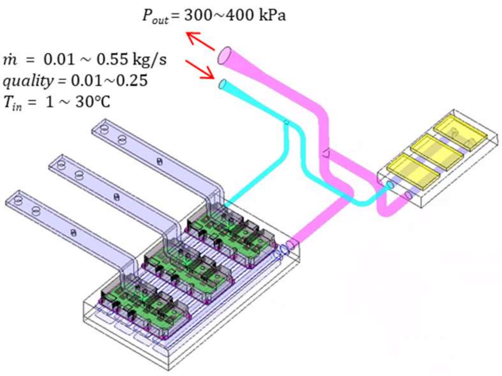


Fig. 5 – The cooling model for IGBT inside of VFD.

The multiple reference frame(MRF) method was introduced for calculation of heat transfer at cooling parts of motor to consider rotational motion of refrigerant flow. To calculate heat transfer between refrigerant and motor surface, a permanent magnet, insulation sheet for bonding, vanish and stator has considered into the CAE model through the 47million grids. After the 3D CAE model of motor converted to a 1D ROM equipped with input and output parameters, 1D ROM exchanges their data with 1D condenser model because refrigerant for cooling comes from condenser. Moreover, for the calculation of heat transfer at inverter driver by refrigerant, IGBTs and Diodes were considered as a main heat source of circuit board.

There have to keep in mind to get a reasonable result by calculation, a model considers a heat transfer coefficient for IGBT dummy and contact surface of circuit parts together with thermal resistance of most of chips. As the same way as motor, 3D inverter model also converted to 1D model for co-simulation. The 12million grids was used for model.

2.4 The cycle modeling for co-simulation

The modeling approach of the study is a 1D dynamic model that can calculate the system, which is to convert 3D models of the compressor's main parts into 1D, respectively, and combine it with the heat exchangers and expansion model already made in 1D. The reason to make a model for the main components of the compressor in 3D firstly, is to observe those components in more detail because the compressor is an important component. Also, the reason to make ROMs for all 3D models is for running co-simulation with different physics at one time for every time intervals. The study had checked soundness of all 1D models after reducing their order because ROM could has a loss of accuracy as compared to the full 3D model in case of inadequate conversion. The normal approach to making a 1D refrigeration cycle model with considering multi-physics is a one-way coupled model, not much for two way coupled model. It comes from a difference of reaction-time between physics, for example a thermal characteristics reacts relatively much slower than an electrical characteristics when operation condition changes according to time, so simulation could be stop. Fortunately, a couple of the software solutions opens an opportunity to simulate a system by two-way coupled situation under the circumstance of multi-physics. Fig.6 shows the parameters to use as input-output between 1D models, it is schematic diagram of co-simulation of refrigeration system.

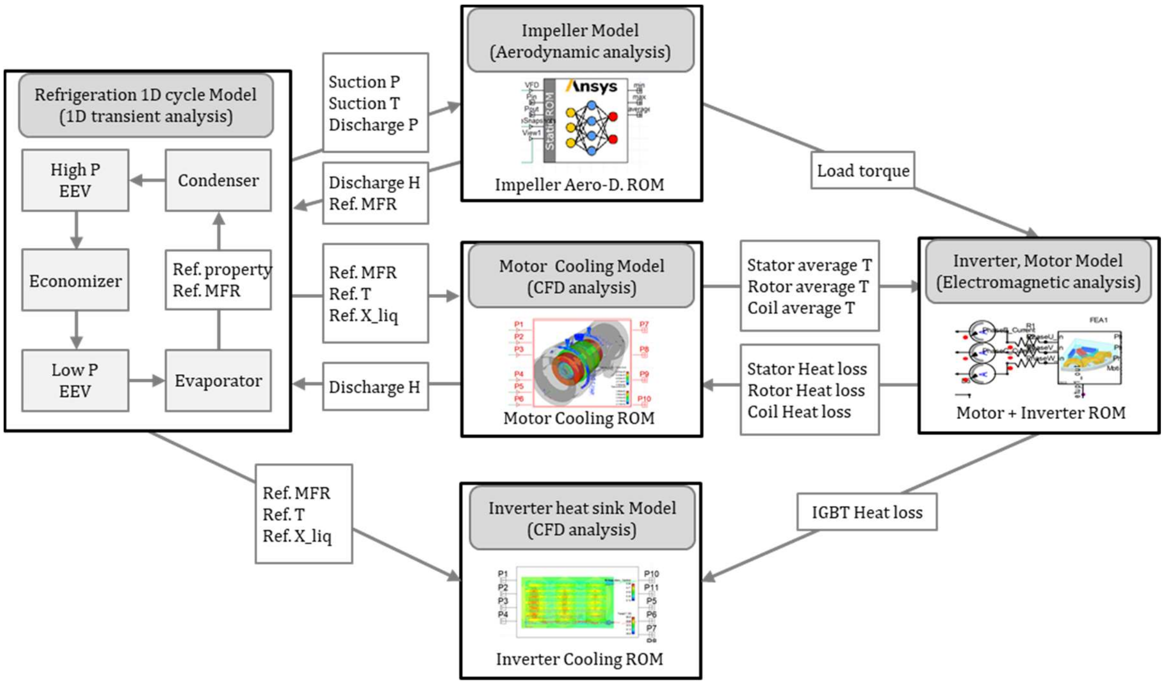


Fig. 6 – The relationship of the single dimensional models for refrigeration cycle.

As you can see at Fig.6, the simulation model for refrigeration cycle is consist of five 1D models, they calculate not only their own properties but also exchange data each other during calculation.

3. Simulation results and discussion

3.1 Model validation under various load conditions

The measured data for validation is an experimental data from the centrifugal chiller whose are manufactured by LG Electronics, it was obtained from the test facility of LGE under start-up and load-change conditions according to AHRI550/590. Simulation has been done from start-up until it reaches a steady state as the same way as doing experiment. The Fig.7~Fig.10 represents that the comparison of the results of the simulation and experiment at various load conditions, figures show a variation of pressure and heat capacity with time.

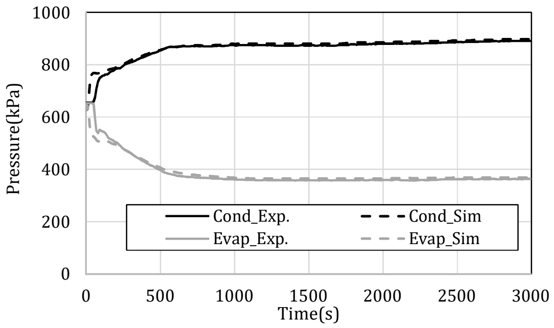


Fig. 7 – The result of simulation to the pressure under 100% load.

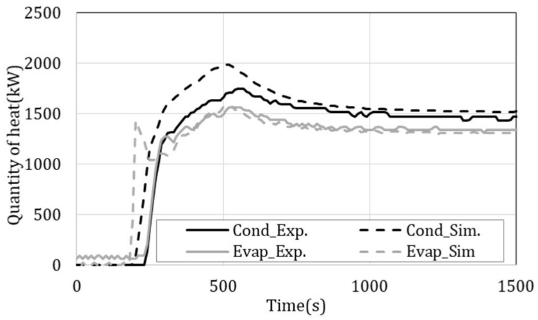


Fig. 8 – The result of simulation to the capacity under 100% load.

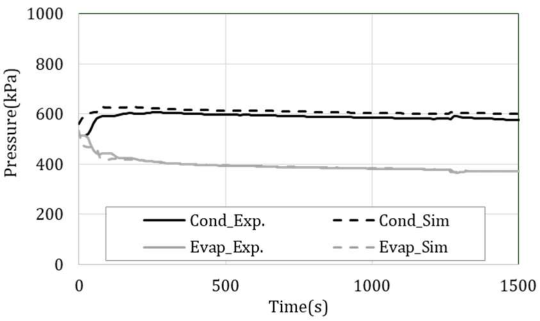


Fig. 9 – The result of simulation to the pressure under 25% load.

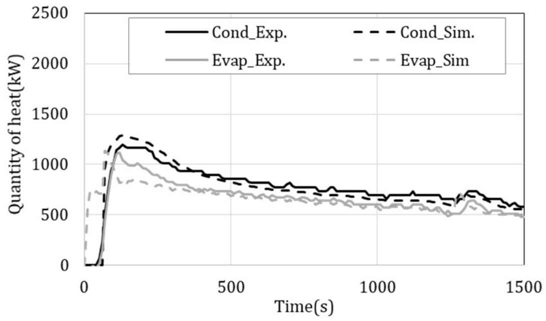


Fig. 10 – The result of simulation to the capacity under 25% load.

There find a reasonable accuracy between simulation and experiment data at all load conditions, however, there has a gap between two values especially for a few minutes after start up. There are two main reasons, first one is that the initial point of calculation could not be set the same as experiment because of their own control strategy of chiller. When an operator pushes the start-up button to turn on a chiller, the controller immediately commands the chiller to run 136Hz, which is the minimum frequency at nothing to do with loads, however, control strategy checks stability of system at every Hz. The result of that, there can be happened a time delay of about 30~45 seconds depending on the communication status. It will be improved by adding the "software in the Loop simulations"(SiLs) as a further study. One other reason is that the simulation model can covers only the same range of operation as experiment, which is from 136Hz to 228Hz. There was not enough data from 0 to 135Hz, so extrapolation method be used to cover the calculation range. It also need to be improved by further study. It would be a common issue when a ROM be introduced for co-simulation in case of less consideration of the model coverage. The accuracy of ROM depends on whether calculation domain is inside of a pre-defined territory. Therefore, diverging could be happen when there be found a case of over-territory during calculation of refrigeration cycle for energy balance under transient condition. It is highly recommended that the separate pre-processing always be required in the 1D/3D interface connection process to ensure that the calculation territory of ROM and the input/output value range of 1D cycle set to be not deviate from each other.

3.2 The effect of co-simulation

The study introduced a concept of the co-simulation not only for improving the prediction accuracy of performance at design stage but also that can be used for manufacturing and quality control. An interaction between physics are normally influenced to a cycle system, co-simulation deserves to be considered. Moreover, 3D CAE model, which will be converted to 1D model, is better to describe a physical phenomenon because it included exact information for the structure and physical dimension. The comparison between two different approaches has shown in Fig.11.

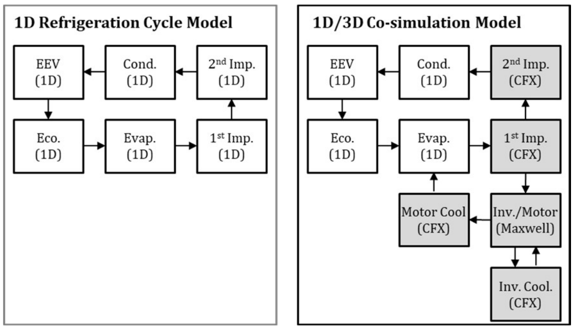


Fig. 11 – The concept of two different modelling approach.

In the co-simulation approach, the impeller aerodynamic characteristics, electromagnetic characteristics of motor, and heat loss, which were difficult to make model at the 1D analysis, were implemented by adding a 3D analysis model. There be shown how much improved accuracy in terms of performance by co-simulation in Fig. 12. The results from "1 D only" simulation both for the capacity of condensing and for evaporating predicted a higher value than experiment while the co-simulation has shown much close value compare to experiment under all load conditions. In the co-simulation, the performance of analysis improved by considering the loss models that could not be reflected in the 1D only model. Moreover, with the help of a detailed 3D analysis for aerodynamics of impellers, the accuracy under low load condition at 25% has also improved.

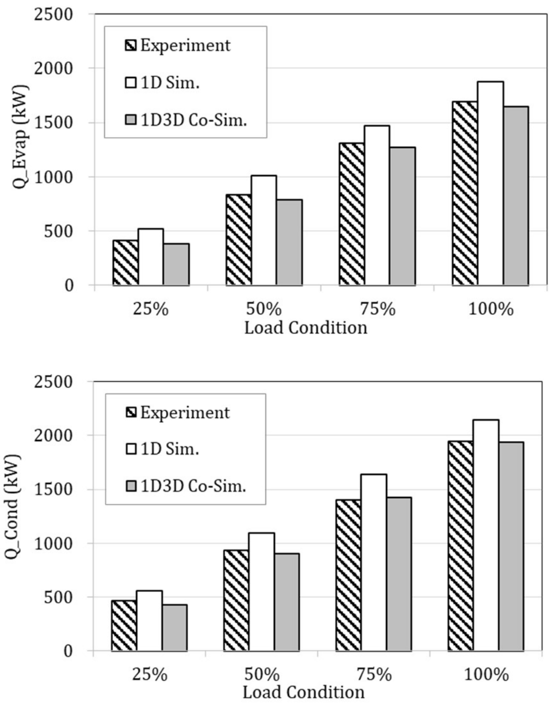


Fig. 12 – The effect of co-simulation on the accuracy of heat capacity.

3.3 Model validation under random change of load

It was observed whether the simulation of the dynamic behavior tracked well under the condition of load scenario. The load scenario was created by selecting 3 pieces of random operation data stored in the real product first, and then attaching them each other to get a series data as long as 50,000 seconds. It becomes a virtual load for model validation. The points with discontinuous data caused by connecting are around 16,000 seconds and 45,000 seconds. Fig. 13 is a dynamic behavior comparing simulation results and scenario data, (a) is a pressure and (b) is an electric current. A capacity of heat can be calculated from the change of electric current because capacity of heat could not directly stored as a data in the control device of chiller. This conversion is one of benefits from doing co-simulation of 1D/3D. Fig.13 shows a pressure and an electric current, the difference between the simulation result and experiment is well tracked within 10%. In case of scenario based analysis, it proceeded from an operating state instead of setting the initial condition as a stop state as shown in Fig. 7~Fig.10. From the results of that, the dynamic behavior of the model is well simulated independent of initial value setting.

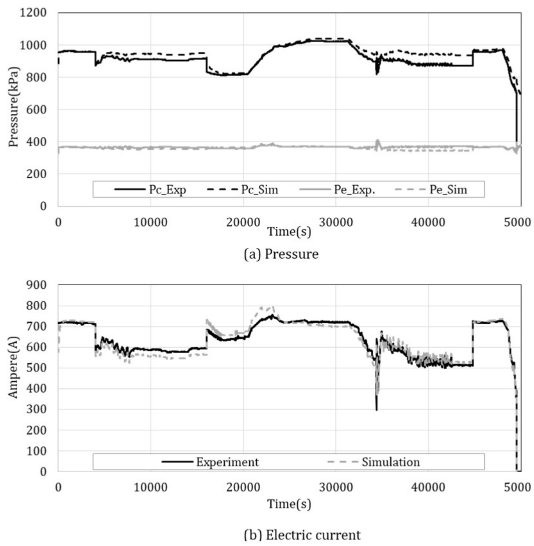


Fig. 13 – The result of simulation on pressure and electric current under the condition of a virtual load.

4. Conclusion

This study presents a detailed modeling approach to a centrifugal chiller as a vapor compression refrigeration system by introduced MBSE methodology of object-oriented structure. The model shows an inaccuracy of less than 2% under the rated condition and 10% under the partial load condition when the simulation result be compared with the experiment. The main contributions brought by the modeling methodology is that 1) the 1D-3D co-simulation significantly improved the accuracy of prediction compared to the 1D analysis only, 2) considering a number of interrelated physics at one time delivered more accurate results if their interaction placed in a state of strong coupled, 3) the

3D models can be converted to 1D model by use of ROM without time lag. At the same time, the accuracy of 1D model maintained as decrease of less than 2% compare to 3D model. The followings are a suggestion as a future study to make a digital mock up by dynamic modeling. 1) At the stage of 1D architecting during MBSE approach, the main component of refrigeration system like a compressor and heat exchangers needs to be refined up to a smaller level, and then make it as an independent library. It is an essential stage for the model to keep expendability even it takes time. 2) In case of adding a 3D modeling for heat exchangers gives a better result than the current model, which did not include 3D model other than compressor. Finally, data domain need to be wide enough so that calculation works well when you making a ROM. Although this study was conducted on a centrifugal chiller system with a centrifugal compressor and a shell & tube heat exchangers, the basic principle of the refrigeration system is the same. Therefore, the result of the study can be applied very similarly to the residential air conditioning system whose have a positive displacement compressor and the fin & tube heat exchangers.

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The datasets generated during and/or analysed during the current study are not publicly available because it is property of LGE but are/will be available depending on purpose. Please contact to yj.hwang@lge.com·