

Steady-State Modeling of the Refrigeration Unit for a Marine Refrigerated Container

Jiajun Ren

School of Marine Engineering
Jimei University
Xiamen, China
e-mail: Jiajun_ren@126.com

Wu Chen

School of Marine Engineering, Jimei University
Fujian Provincial Key Laboratory of Naval Architecture
and Ocean Engineering
Xiamen, China
e-mail: chenwu73@jmu.edu.cn

Chaoyu Zheng

School of Marine Engineering
Jimei University
Xiamen, China
e-mail: cyzheng@jmu.edu.cn

Abstract—The mathematical models of compressor, evaporator and condenser in a refrigerated container's refrigeration unit were established, which were used to form the whole system model for steady-state simulation according to the coupling relationship among them. The full-load performance of the refrigeration system was firstly deployed using the system model programmed in the MATLAB software. And the experimental performance tests were secondly carried out using a marine refrigerated container test rig. The errors between simulation results and experimental results are within 10%. Therefore the model has good simulation accuracy and could be used for the future energy-saving analysis of the marine refrigerated container.

Keywords—marine refrigerated container; refrigeration unit; simulation; steady-state model; full-load experiment

I. INTRODUCTION

The inside temperature of marine refrigerated container with refrigeration system can be control at -30 to 20°C when the external temperature being -40 to 50°C. For the refrigeration unit of the marine refrigerated container, scholars have carried out a lot of theoretical research and technical development work and made much valuable research in the fields of mathematical modeling, simulation, optimization and control of the vapor compression refrigeration air-conditioning system [1]. The steady-state simulation of the refrigeration unit can be used to study the effects of the structure parameters on the system components, to inspect the system components' matching, and to simulate the performance of the system under different working conditions, and hence it is widely applied in the field of refrigeration. Because the hot-gas by-pass valve is closed and the suction valve is fully open, the control problems involved is limited during the steady-state operation of refrigeration unit under full-load condition. Therefore to develop a steady-state system model of refrigeration unit is valuable to provide a tool to

study the energy consumption and energy-saving of the marine refrigerated container for the whole voyage.

II. THE REFRIGERATION UNIT SIMULATION MODEL FOR MARINE REFRIGERATED CONTAINER

A. Component model

1) Compressor model :

The effect of compressor in refrigeration system is implemented through the refrigerant migration, so for the system simulation model of compressor, calculating the mass flow through the compressor is the most important, and calculating other parameters which affect the performance of the units, such as the exhaust temperature at the same time. The thermodynamic model of the compressor is as follows [2]:

$$m_{com} = \frac{IV_{th}}{u_s} \quad (1)$$

$$t_d = (t_s + 273.15) \left(\frac{p_c}{p_e} \right)^{\frac{k-1}{k}} - 273.15 \quad (2)$$

$$N_{in} = \frac{I}{h} \cdot V_{th} \cdot I \cdot \frac{p_e \cdot k}{k-1} \left[\left(\frac{p_c}{p_e} \right)^{\frac{k-1}{k}} - 1 \right] \quad (3)$$

m , V , u , N_{in} stand for mass flow rate, volumetric displacement, inlet refrigerant specific volume, input power to compressor; t and p stand for temperature and pressure; k stands for polytropic exponent; I stands for coefficient of capacity; h stands for electrical efficiency; Subscripts th and com stand for theoretical and actual value, s and d stand for suction and discharge condition.

2) Condenser model :

In the stable operation of the refrigeration unit, the refrigerant with superheated-gas state enters the condenser; it is gradually cooled to gas-liquid two-phase state and leaves the condenser as super-cooled liquid state by further cooled at last. The heat transfer of the condenser is composed of two parts: the air side flow heat transfer and the refrigerant side flow heat transfer. The capacities of the heat transfer in the two sectors are equal under steady condition. In the establishment of the condenser model, the condenser coil is divided into three parts, they are desuperheating region, two-phase region and subcooling region. The log mean temperature difference method is applied to each zone to determine the heat transfer rate [3]. The thermodynamic model of the condenser is as follows:

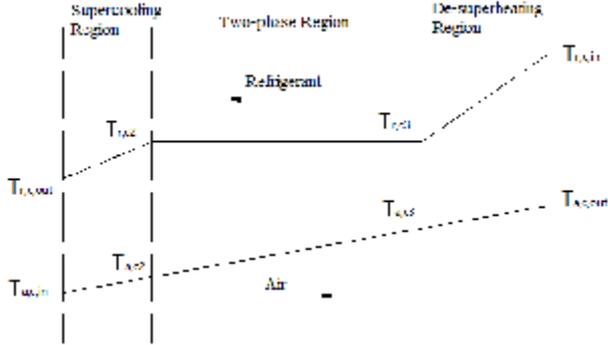


Figure 1. Schematics of temperature change of condenser heat exchange process.

The air side heat transfer:

$$Q_{a,c} = m_{a,c} (h_{a,c,out} - h_{a,c,in}) \quad (4)$$

To calculate the refrigerant heat transfer according to the temperature difference of heat transfer:

$$Q_c = Q_{c,sh} + Q_{c,tp} + Q_{c,sc} \quad (5)$$

$$Q_{c,sh} = K_{c,sh} A_{c,sh} \Delta T_{c,sh} \quad (6)$$

$$Q_{c,tp} = K_{c,tp} A_{c,tp} \Delta T_{c,tp} \quad (7)$$

$$Q_{c,sc} = K_{c,sc} A_{c,sc} \Delta T_{c,sc} \quad (8)$$

To calculate the refrigerant heat transfer according to the flow enthalpy:

$$Q_{r,c} = m_r (h_{r,c,in} - h_{r,c,out}) \quad (9)$$

The convective heat transfer coefficient in single-phase region of the tube is obtained by Dittus-Boelter correlations [4], and condensation heat transfer coefficient in two-phase region is obtained by the Shah correlations [5]. And the air side heat transfer coefficient is obtained by the McQuiston correlations [6].

And because of:

$$Q_c = Q_{a,c} = Q_{r,c} \quad (10)$$

We can solve the temperature, enthalpy and degree of subcooling at outlet of the condenser from Equations (4) - (10).

$Q, \Delta T, A, h, K, T_c$ stand for heat transfer rate, the average temperature difference of logarithms, heat

transfer area, enthalpy, overall heat transfer coefficient and condensing temperature; Subscripts a, r, c, in, out, sh, tp and sc stand for air, refrigerant, condenser, inlet, outlet, desuperheating, two-phased and subcooling.

3) Evaporator model:

In the stable operation of the system, the gas-liquid two-phase refrigerant with low quality enters the evaporator and gradually vaporizes by absorbing heat from the inner space of container, and finally leaves the evaporator with the state of superheated gas. In the establishment of the evaporator steady lumped parameter model, both air outside the tube and refrigerant inside are regarded as a countercurrent heat transfer, and the evaporator coil is divided into two-phase region and superheat region for the modeling. Calculate the heat flow of each part and establish the heat transfer model of both refrigerant in tube and the air outside respectively according to the energy conservation law. The thermodynamic model of the evaporator is as follows [7]:

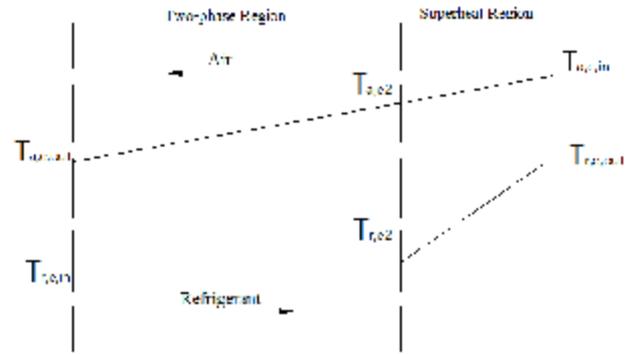


Figure 2. Schematics of temperature change of evaporator heat exchange process.

The air side heat transfer:

$$Q_{a,e} = m_{a,e} (h_{a,e,in} - h_{a,e,out}) \quad (11)$$

To calculate the refrigerant heat transfer according to the temperature difference of heat transfer:

$$Q_e = Q_{e,sh} + Q_{e,tp} \quad (12)$$

$$Q_{e,sh} = K_{e,sh} A_{e,sh} \Delta T_{e,sh} \quad (13)$$

$$Q_{e,tp} = K_{e,tp} A_{e,tp} \Delta T_{e,tp} \quad (14)$$

To calculate the refrigerant heat transfer according to the flow enthalpy:

$$Q_{r,e} = m_r (h_{r,e,out} - h_{r,e,in}) \quad (15)$$

The convective heat transfer coefficient in single-phase region of the tube is obtained by Dittus-Boelter correlations, and boiling heat transfer coefficient in two-phase region is obtained by the Kandlikar correlations [8]. And the air side heat transfer coefficient is obtained by the McQuiston correlations.

And because of:

$$Q_e = Q_{a,e} = Q_{r,e} \quad (16)$$

We can solve the temperature, enthalpy and the degree of superheat at outlet of the evaporator from Equations (11) - (16).

The subscript e stands for evaporator.

B. Simulation procedure

This paper builds a complete model of the refrigeration unit which is coupled by subsystem models according to the conservation of mass and the conservation energy law. The steady-state simulation of refrigeration system flowchart is shown in figure 3. The role of expansion valve in the refrigeration system is to regulate the degree of superheat at the evaporator outlet and make it equal to a constant value. So in the whole system, the degree of superheat will be set to a constant value, which is a judgmental condition by adjusting the evaporating temperature, therefore the expansion valve model is omitted. In the modeling process, first of all, the compressor model is established to calculate the mass flow under the different return air temperature in container and different ambient temperature by using initial assumptions of condensing temperature and evaporating temperature, and then using the mass flow as the input of the condenser model to calculate the degree of subcooling in the condenser. A received unit is set at the condenser outlet; the gas-liquid mixture liquid exists with the saturated liquid form during the stable operation of the system, so the degree of subcooling at the condenser outlet is 0. Set up the degree of subcooling on the calculation of the condenser model equal to 0 for the first condition, if the degree of subcooling satisfies the condition, the model will let the calculation continue. Otherwise the condensing temperature is the new guess value for the next iteration. Using the outputs from the compressor model and condenser model, calculate the degree of superheat at evaporator outlet and set up the degree of superheat equal to 5 for the second condition. If it satisfies the condition, the evaporating temperature is correct. Otherwise the evaporating temperature is the new guess value for the next iteration until the degree of superheat satisfied conditions. The model will iterate until both two conditions are satisfied.

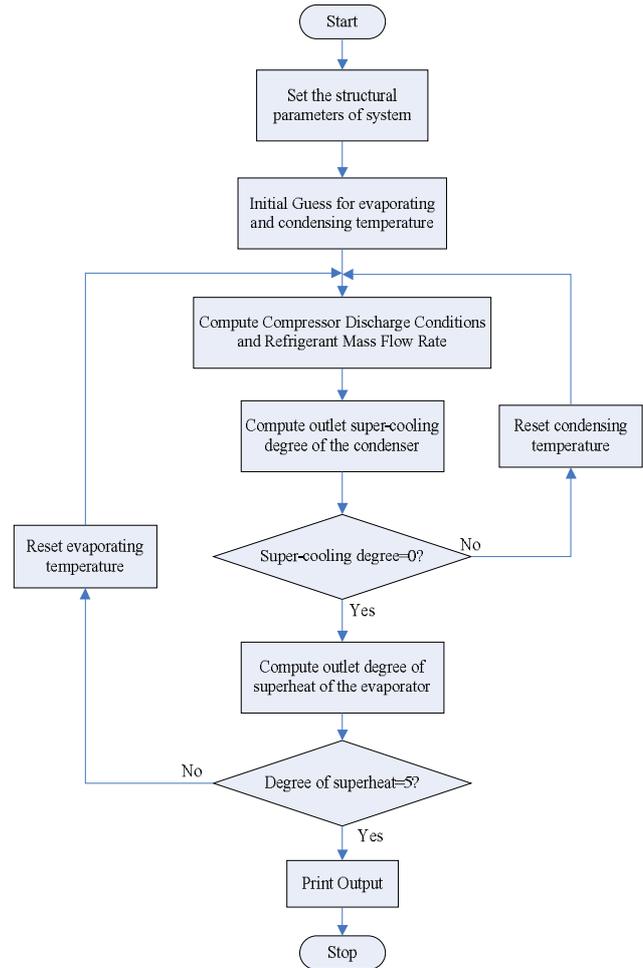


Figure 3. Flowchart of simulation program.

III. PERFORMANCE EXPERIMENT

In order to verify the accuracy of the simulation model, a full-load steady-state experiment is carried out by using a standard 20 feet reefer container experiment table and the refrigerant is R134a. The refrigeration unit of reefer container is mainly composed of the constant speed scroll compressor, condenser fan, evaporator fan, suction regulating valve, liquid accumulator, flow meter and the corresponding control system. It can achieve the requirements of different goods refrigerated or frozen. The refrigeration unit of refrigerated container experiment adopts the hot gas bypass and suction regulation as the capacity regulating mode of the refrigeration unit under the refrigeration model.

This experiment is carried out under the environmental temperature is 18.8°C ($\pm 0.5^{\circ}\text{C}$) and the condition of relative humidity is 68%. Two 2kW/220V heaters and a 6kW/380V heater are placed in the reefer containers. Simulate the thermal load in the container by assembling and adjusting the heater power. The running heat load of evaporator fan is 600W and the leakage heat rate is 26W/k [9] when calculating the leakage heat. The total thermal load for the reefer container is the sum of the simulation container thermal load, fan thermal load and the leakage heat of the container.

The experimental procedure is: Set the temperature in the container to low temperature and then start the heater,

regulate the simulated heat load of container for 5kW after reaching the set temperature. The return air temperature in the container gradually increases, in this process, the BSV (hot-gas by-pass electromagnetic valve) is closed, and the SMV (suction regulating valve) is opened to 100%. Confirm the refrigeration unit has run to steady state after the return air temperature is stable, at this time the total heat load of the reefer container is equal to the refrigerating capacity of the refrigeration unit, because the refrigeration unit have not carried on the suction throttling and hot gas bypass, the refrigerating capacity of this state is the full-load refrigerating capacity. After collecting the data of the refrigeration unit, the simulated heat load will be added to 6kW. When the unit runs to steady state with higher return air temperature, collect the system data again. According to the above method, collect the system data of stable operation respectively under conditions of the simulated thermal load is 7 kW, 8 kW and 9 kW in the reefer container.

IV. RESULTS AND DISCUSSION

The full load refrigerating capacities provided by the refrigeration unit under the stable operation of different environment temperature and different container temperature inside are different. This paper calculated the full-load refrigerating capacity and the power draw of compressor by the steady-state model of refrigerating unit under the same environment temperature and different set temperature. The BSV (hot-gas by-pass electromagnetic valve) need to be closed and the SMV (suction regulating valve) need to be opened to 100% to keep the refrigerating capacity equals to the full-load refrigerating capacity in this condition [10].

As can be seen from Figure 4, with the increasing of the return air temperature inside, the full-load refrigerating capacity of the refrigeration unit increases, this is a result that the increasing of both evaporation temperature and refrigerant mass flow two factors act together. When the refrigeration unit runs to steady state, the evaporating temperature of the refrigerant in the evaporator is generally 8-10°C higher than the return air temperature and gets higher with growing of return air temperature. By the pressure-enthalpy diagram if the condensing temperature is constant in the compression refrigeration cycle, when the evaporation temperature increases, the refrigerating capacity per weighting of the refrigerating system increases. At the same time the refrigerant mass flow also increases to obtain the larger total refrigerating capacity. This make the total refrigerating capacity of the refrigeration unit has the increasing trend when the return air temperature increases in the container. The trends of simulation refrigerating capacity and the experimental refrigerating capacity are same and the maximum error is 8.1%.

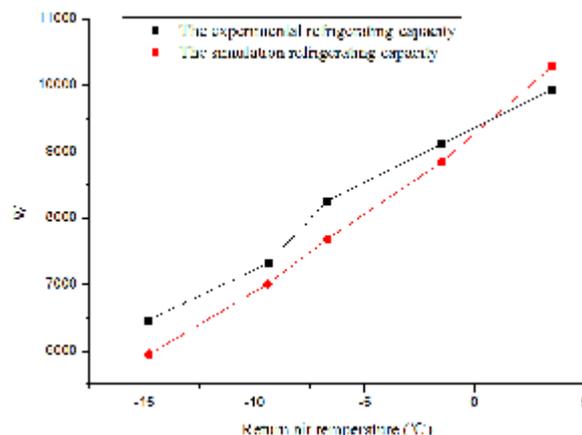


Figure 4. Refrigeration capacity vs. return air temperature.

As can be seen from Figure 5, with the increasing of return air temperature inside the container, the compressor power draw also showed an increasing trend. The constant condensing pressure which on account of unchanged condensing temperature, the increasing evaporation pressure which is due to the evaporating temperature gets higher and the increasing the mass flow led to the power draw of the compressor increases with the return air temperature gets higher. The trend of the simulation compressor power is the same as the trend of experimental compressor and the maximum error is 3.8%.

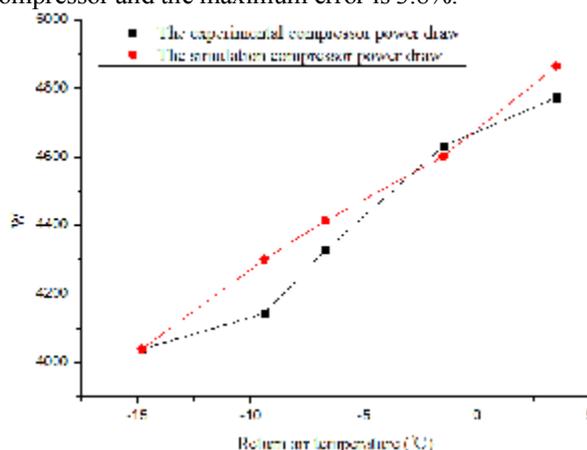


Figure 5. Compressor power draw vs. return air temperature.

V. CONCLUSION

In this paper the sub-models of condenser and evaporator were built based on the steady-state lumped-parameter-method, and formed a system model of refrigeration unit with the compressor sub-model. Using the steady-state simulation model of the refrigeration unit, the system's full-load performance including the refrigeration capacity and the power draw of compressor was calculated when the environmental temperature being set at 18.8°C and the relative humidity being 68%. With the increasing of the temperature of return air inside the container, the evaporation temperature of the refrigeration unit and the refrigerant mass flow all increased, meanwhile the system's refrigeration capacity and the power draw of compressor also trended to increase. It was found that the simulation results were consistent with the experimental results and their error was reasonable. Therefore the system model can be used to judge the performance of the

refrigeration unit for the marine refrigerated container in different external environment temperature and different setting air temperature; it could provide the basis for the subsequent analysis of energy saving of refrigerated containers for the whole voyage.

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