A Dynamic Model of a Gas Engine-Driven Heat Pump in Cooling Mode for Real-Time Simulation

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ABSTRACT: The present study has been conducted to simulate dynamics of a gas engine-driven heat pump (GHP) for the design of control algorithm. The dynamic model of a GHP was based on conservation laws of mass and energy. For the control of refrigerant pressures, actuators such as an engine throttle valve, outdoor fans, coolant three-way valves and liquid injection valves were controlled by P or PI algorithm. The simulation results were found to be realistic enough to be applied for the control algorithm design. The model could be applied to build a virtual real-time GHP system so that it interfaces with a real controller for the purpose of developing control algorithm.

Nomenclature

- c_{pg} : specific heat at constant pressure of refrigerant gas [kJ/kg℃]
- h : enthalpy [kJ/kg]
- $\dot{m}$: mass flow rate of refrigerant gas [kg/s]
- N : engine speed [rpm]
- p : pressure [Pa]
- PD : rate of change of displacement volume of a compressor [m^3/s]
- $\dot{Q}$: volumetric flow rate [m^3/s]
- T : temperature [℃ or K], torque [N-m]
- t : time [sec]
- u : internal energy [kJ/kg]
- UA : overall thermal conductance [W/℃]
- V : volume [m^3]
- v : specific volume [m^3/kg]
- w : specific work [J/kg]

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Greek symbols

- η_v : volumetric efficiency of a compressor
- η_m : mechanical efficiency of a compressor
- θ : rotation angle of an engine crankshaft [radian]
- ρ : density of refrigerant [kg/m^3]
- τ : time constant [sec]

Subscripts

- *amb* : ambient
- *byp* : hot gas bypass valve
- *cd* : condenser
- *comp* : compressor
- *cool* : engine coolant
- *disp* : engine displacement volume
- *eng* : engine
- *ev* : evaporator
- *f* : saturated liquid
- *g* : saturated vapor
- *gas* : refrigerant in gas phase
- *i* : inlet

indoor : indoor
j : any of indoor units
liq : liquid control valve
load : cooling load
o : outlet

1. Introduction

Large commercial buildings have adopted conventionally central HVAC systems which use turbo-chillers and air ducts. However, the central system has the following demerits: need of full-time mechanic staff, high maintenance cost, and inability of separate zone control. In order to overcome such shortcomings, nowadays HVAC systems are in the trend of transition from the central system to several multi-split air conditioners. They have indoor cooling units which exchange heat with an outdoor unit by controlling refrigerant flow rates through them. As the technology associated with the multi-split air conditioner system reaches maturity, its market portion will grow further. The key part of the technology is the control algorithm that makes the system run stably against rapid transitions of cooling load.

Because the multi-split air conditioner system is highly complicated and high investment of time and money is required to develop a control algorithm of such system, it is difficult to catch up with the competitiveness of Japanese products in a short period.(1) So, secure establishment of GHP control algorithm is a prerequisite for the product competitiveness.

In order to develop GHP control algorithm, a GHP and test facilities must be prepared for test runs. However, in the early stage of GHP development, a GHP prototype is not ready to run. Even though a prototype is prepared, operation of thermal environment chambers must be minimized in view of saving cost and development time. An efficient alternative is to use a virtual real-time GHP. A controller under development is interfaced with the virtual GHP and tested to secure integrity of its control algorithm for all kinds of scenarios of loads and disturbances. Controller design through a real time simulator is a common practice in large scale chemical and power plants.(2-4) The same approach can be applied to GHP. However, no literature is found regarding GHP. It is because GHP is commercialized in Japan and Korea only, and details of control algorithm are not published for protection of a company's intellectual asset.

The purpose of the study is to develop a dynamic model of GHP in cooling mode as a first step toward development of a real time GHP simulator for control design.

2. Outline of GHP system for control

Figure 1 shows the schematic of GHP to deal with cooling loads coming from indoor cooling units. Normal cooling load according to weather condition or indoor activity can be processed by engine speed variation. However, extraordinary operation conditions, such as sudden transition of cooling load by powering on or off indoor units, GHP start-up and system operation at very low loads, for example, 'sub-idle cooling load', which refers to the cooling load smaller than that corresponding to idle engine speed, should be assisted by auxiliary actuators, such as a liquid injection valve, a hot gas bypass valve and a coolant three-way

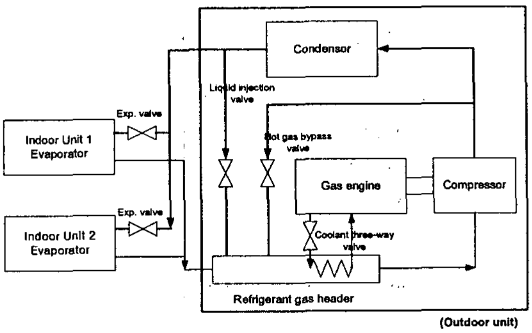


Fig. 1 Schematic of GHP for control.

valve, in addition to engine speed. The liquid injection valve acts as an emergency device to lower degree of overheat at the compressor inlet by injecting pressurized liquid refrigerant to use evaporation heat, so it cools down the refrigerant entering the compressor. The hot gas bypass valve acts as an emergency device to regulate refrigerant pressures beyond the limits of normal pressure control by passing pressurized hot gas back into the compressor inlet, so it is possible to lower excessive high pressure at the compressor exit and to increase excessive low pressure at the compressor inlet whenever needed.

The operable speed range of a gas engine is between 800 rpm and 2,400 rpm so its turndown ratio is about 3 : 1. It is quite narrow compared to the least range of cooling load that is 8 : 1 in terms of number of indoor units attached to a GHP system. To increase the turndown ratio, loading / unloading of compressor is applied. However, when it comes to the sub-idle cooling load, the cooling capacity at idle speed is yet too big to deal with the load. It will lead to frequent starts and stops of the system. To avoid the frequent switching, artificial cooling load is generated. It is made by injecting liquid refrigerant into the gas header and supplying necessary vaporization heat by increased coolant flow rate through a coolant three-way valve to the gas header where the refrigerant exchanges heat with the coolant in a compact plate-type heat exchanger.

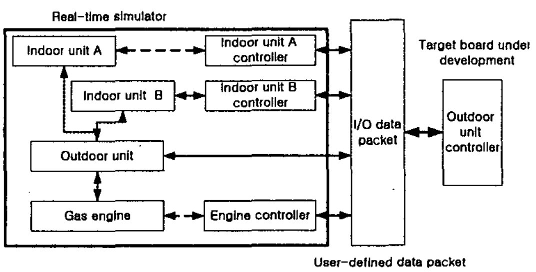


Fig. 2 Controller prototyping environment using a real-time simulator.

On the other hand, when GHP pressures deviate significantly from its set values, which is mainly caused by switching indoor units frequently, emergency control mode is activated where liquid refrigerant is injected and hot refrigerant gas is bypassed back to the compressor inlet.

Figure 2 shows the concept of prototyping a controller interfaced with a virtual real-time GHP plant that is programmed using a dynamic model to be developed in the study. GHP is a system that is heavily coupled among many inputs and outputs and requires a huge and expensive calorimeter to test its control performance because of various cooling loads associated with its many indoor units. To develop GHP control algorithm and its prototype controller in short time and less cost, it is necessary to develop a virtual real-time GHP plant to replace a GHP and a calorimeter. If a virtual real-time GHP plant can be realized by modeling GHP dynamics and making it run in real-time exchanging its inputs and outputs with a target controller via data communication, for example, through serial port, it will save a lot of time and cost for the development of a GHP system.

3. Dynamic model

3.1 Compressor

A P-V diagram of a reciprocating compressor is shown in Fig. 3. Its volumetric efficiency and refrigerant flow rate are defined respectively in Eqs. (1) and (2).[9]

$$\eta_v = \frac{\dot{m}_{comp} v_3}{PD} \tag{1}$$

$$\dot{m}_{comp} = \left[1 + C - C\left(\frac{p_c}{p_b}\right)^{1/n}\right]\frac{PD}{v_b} \tag{2}$$

Here, polytropic exponent n should be determined by experiment, but it is a common

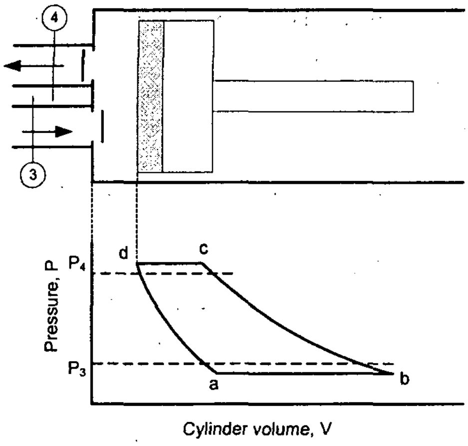


Fig. 3 P-V diagram of the compressor.

practice to approximate it with the isentropic exponent k. For the refrigerant R-22, k is 1.16. The clearance coefficient C is defined as follows and its value is about 0.05.

$$C = \frac{V_d}{V_b - V_d} \tag{3}$$

Work per cycle and power consumed are calculated from the following relations.

$$W_{comp} = \frac{n}{(n-1)} p_b v_b \left[\left(\frac{p_c}{p_b} \right)^{(n-1)/n} - 1 \right] \tag{4}$$

$$\dot{W} = \frac{\dot{m}_{comp} w_{comp}}{\eta_m} \tag{5}$$

3.2 Evaporator

Cooling load is processed by vaporizing the liquid refrigerant in the evaporator and the superheat of the refrigerant is regulated by an electronic expansion valve located at the exit of the evaporator. Heat balance equation in the evaporator is described as follows.

$$m_{ev} \frac{du_{ev}}{dt} = \dot{Q}_{load} - \dot{m}_{ev,j} (h_{ev,o} - h_{ev,i}) \tag{6}$$

Cooling load in the evaporator can be expressed by multiplication of the enthalpy difference between the inlet and exit of the evaporator and the vapor throughput. Since superheat of R22 is controlled within 5℃ its sensible heat is less than 1% of latent heat. So contribution of the sensible heat to the cooling load is neglected for simplicity as follows.

$$\dot{Q}_{load} = UA_{indoor} (T_{indoor} - T_{ev,sat}) \tag{7}$$

In addition, Eq. (6) is rearranged to obtain a first-order differential equation with respect to the enthalpy at the evaporator exit by assuming that the internal energy in the evaporator is approximated to the enthalpy at the evaporator exit.

$$m_{ev} \frac{dh_{ev,o}}{dt} + \dot{m}_{ev,j} h_{ev,o} \approx \dot{Q}_{load} + \dot{m}_{ev,j} h_{ev,j} \tag{8}$$

Pressure build-up in the evaporator can be explained from the following consideration. Evaporation of refrigerant by cooling load increases pressure in the evaporator whereas suction of refrigerant gas into the compressor decreases the pressure. At steady state, evaporation amount by cooling load is balanced by the amount of refrigerant gas sucked into a compressor. As cooling load increases, evaporation amount increases and the pressure in the evaporator increases when a compressor runs at fixed speed at the moment. So the compressor speed should be increased to keep the pressure constant.

3.3 Gas header

All of refrigerant gas leaving each indoor unit is collected in a refrigerant gas header and sucked into a compressor. As shown in Fig. 4, the gas in the course of heading toward the refrigerant gas header receives additional heat and, in some cases, additional refrigerant in gas or liquid phase from auxiliary devices, such as a liquid injection valve, a hot gas by-

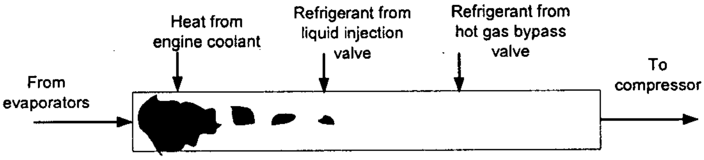


Fig. 4 Heat and mass balance in the gas header for auxiliary control.

pass valve and an engine coolant heat exchanger, to control superheat, refrigerant pressures and temperatures.

The behavior of the pressure of the superheat gas in the gas header can be approximated by the following ideal gas relation.

$$P_{ev} V_{gas} = m_{gas} R T_{gas} \tag{9}$$

The amount of refrigerant evaporated from each indoor unit and the intake amount into the compressor are changed by varying cooling load. The following equation describes the amount change in the gas header, which leads to the pressure change in the evaporator.

$$\frac{dP_{ev}}{dt} = \frac{R}{V_{gas}}\left(T_{gas}\frac{dm_{gas}}{dt} + m_{gas}\frac{dT_{gas}}{dt}\right) \tag{10}$$

At that moment, the rate of change of the mass of the superheat gas accumulated in the gas header is determined by the following mass flow rates from each indoor unit, a liquid injection valve, a hot gas bypass valve, and the intake amount into the compressor.

$$\frac{dm_{gas}}{dt} = \sum_{j=1}^{n} \dot{m}_{ev,j} + \dot{m}_{liq} + \dot{m}_{byp} - \dot{m}_{comp} \tag{11}$$

The heat balance equation to simulate temperature of the refrigerant vapor in the gas header is as follows:

$$m_{gas}\frac{du_{gas}}{dt} = \sum_{j=1}^{n} (\dot{m}h)_{ev,o,j} + \dot{Q}_{cool} + (\dot{m}h)_{liq} + (\dot{m}h)_{byp} - (\dot{m}h)_{comp,i} \tag{12}$$

The following approximation also holds:

$$h_{byp} \approx h_{comp,i} + w_{comp} \tag{12a}$$

$$h_{liq} \approx h_{comp,i} - h_{fg} \tag{12b}$$

Since the internal energy u_{gas} can be approximated to the enthalpy of the refrigerant at the compressor inlet $h_{comp,i}$, combining Eqs. (11) ~(12b), the following differential equation with respect to $h_{comp,i}$ can be derived.

$$m_{gas}\frac{dh_{comp,i}}{dt} \approx \sum_{j=1}^{n} (\dot{m}h)_{ev,o,j} + \dot{Q}_{cool} + \dot{m}_{byp} w_{comp} - \dot{m}_{liq} h_{fg} - \left(\sum_{j=1}^{n} \dot{m}_{ev,o,j} - \frac{dm_{gas}}{dt}\right) h_{comp,i} \tag{13}$$

The superheat at the compressor inlet can be estimated by solving Eq. (13).

3.4 Condenser

Application of the above argument to the compressor leads to the following relation:

$$\frac{dm_{cd,g}}{dt} = \dot{m}_{comp} - \dot{m}_{byp} - \dot{m}_{cd,f} \approx 0 \tag{14}$$

$$M_{cd}\, c_{p,steel}\frac{dT_{cd,sat}}{dt} = -\dot{Q}_{cd} + (h_{comp,out} - h_{cd,f})\dot{m}_{cd,f} \approx 0 \tag{15}$$

$$\dot{Q}_{cd} \cong UA_{cd}(T_{cd,sat} - T_{amb}) \tag{16}$$

$$P_{cd} \approx P_{sat} = fn(T_{cd,sat}) \tag{17}$$

In Eqs. (14) and (15), transient terms are neglected for ease of calculation. The condensation temperature at the condenser, $T_{cd,sat}$, is solved from Eqs. (14)~(16). Finally, the condenser pressure is obtained from the Eq. (17).

3.5 Expansion valve

The opening of an electronic expansion valve is determined by number of pulse steps applied to the valve stepper. An experimental result shows that the flow rate through the valve has a following linear relation with respect to number of pulse steps to a stepper.[6]

$$\dot{m}_{ev,j} \propto steps \times \sqrt{\rho \Delta p} \tag{18}$$

3.6 Engine dynamics

Engine power is regulated by the opening angle of a throttle valve through which a mixture of air and CNG (compressed natural gas) flows. Since mixture flow rate is non-linear with respect to the opening angle of the valve due to geometric characteristics of its butterfly valve type, the following relation is assumed for simplicity.

$$\frac{\dot{m}_{mix}}{\dot{m}_{mix,\max}} = \frac{bmep}{bmep_{\max}} = \sin\phi \tag{19}$$

In Eq. (19), bmep stands for brake mean effective pressure and is defined as the mean pressure applied to the piston top surface by the gas in the combustion chamber during one engine cycle.[7] Its magnitude is proportional to the amount of the mixture trapped in the combustion chamber and its maximum is about 850 kPa for naturally aspirated spark-ignition engines of modern passenger cars.[3]

$$2\pi T_{eng} = \frac{bmep\ V_{disp}}{2} \tag{20}$$

$$\dot{W}_{eng} = T_{eng}\dot{\theta} = \frac{2\pi T_{eng} N}{60} \tag{21}$$

The dynamic relation with respect to engine speed N with a compressor connected to the engine is as follows:

$$J_{eng}\ddot{\theta} = T_{eng} - T_{comp} - B\dot{\theta} \tag{22}$$

Here, J is the moment of inertia of the engine connected to a compressor. Laplace transformation leads to the following transfer function:

$$\frac{N(s)}{T(s)} = \frac{30/\pi}{Js+B} = \left(\frac{30}{\pi B}\right)\frac{1}{\tau_{eng}s+1} \tag{23}$$

Here, T is net torque obtained by subtracting compressor-absorbed torque from engine-generated torque. The compressor-absorbed torque is determined by circulating flow rate of the refrigerant and pressure difference across the compressor and so forth.

4. GHP control algorithm

The GHP control algorithm in cooling mode is as follows. Each indoor unit deals with its indoor cooling load by controlling superheat of the refrigerant at its exit. The resulting flow rate through indoor units and the volumetric displacement rate (=rotational speed times displacement volume) of the compressor determine the refrigerant pressure at the inlet of the compressor. The outdoor unit controls the pressure within some specified range by regulating the volumetric displacement rate. Thus the resulting rate reflects magnitude of cooling load. A technical challenge of the heat pump with multi-split indoor units is limited variation of cooling capacity compared to the wide range of cooling load variation imposed by many indoor units. The turndown ratio of engine speed is about 3 : 1. To increase the turndown ratio, loading/unloading of individual compressor cylinders is also employed. In spite of such implementation, continuous GHP operation is not possible for very low cooling loads (for example, when a single indoor unit is in service at light load). In that case, the GHP might repeat starts and stops frequently to cope with such low loads and, as a result, adverse effect on system durability and poor control performance is inevitable. Thus, an idea is suggested that the system is supplied with artificial cooling load whose amount makes it run continuously. The load is generated by circulating part of hot engine coolant through the plate-type heat exchanger installed near the inlet of the compressor. The heat exchanger exchanges heat with the refrigerant and the coolant throughput is controlled by the opening angle of the coolant three-way valve. This type of artificial cooling load generation is a widely-accepted technique. Figure 5 shows an example of control algorithm to implement such idea.[8] In this study the same idea was adopted for simulation.

Since thermal dynamics of a GHP system corresponds to linear combination of first-order transfer functions of many mechanical components although they are coupled complicatedly to form the system, a system response to any

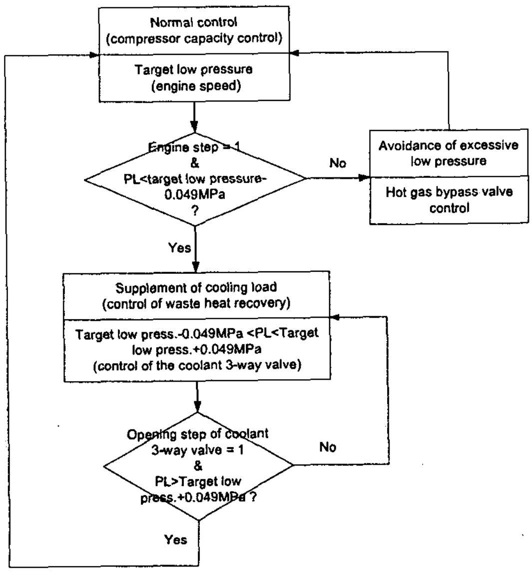
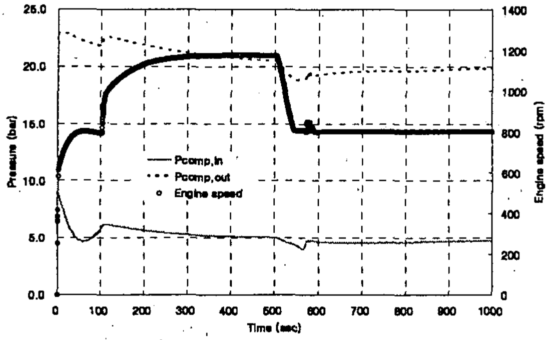


Fig. 5 Control algorithm of Yanmar GHP.

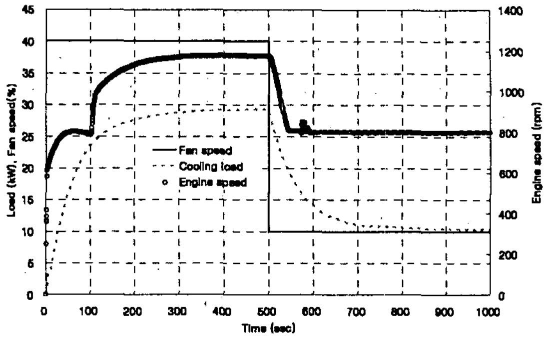
control effort approaches stably to a certain steady-state value without severe excursions. Therefore, it is possible to construct a stable control system by designing a cascade-type control algorithm where fast control loops are placed as inner ones.[5] In this study, cascaded PI loops were applied to all components that need feedback control loops. For example, engine speed and opening angle of the coolant three-way valve were PI-controlled for the refrigerant low pressure. And the inverter speed of the outdoor fan motor was P-controlled for the refrigerant high pressure.

5. Simulation result

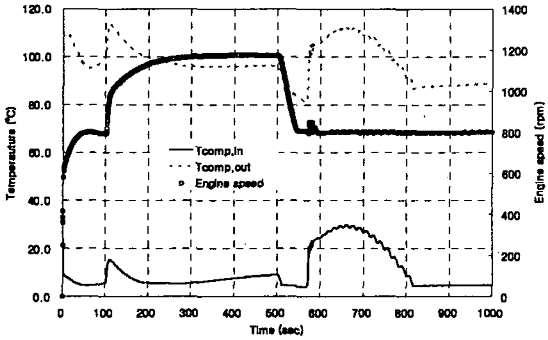
Figure 6 shows an example of simulated GHP operation obtained by programming the modeling result and control algorithm on SIMULINK®.[10] The target compressor was a four-cylinder reciprocating type with the total displacement volume of 554.2 cm^3. Two indoor units in Fig. 1 are exposed to 35℃ of indoor temperature. For the sake of continuous operation, the gas engine is forced to run at its lowest speed of 800 rpm even for a cooling load which is lower than the minimal cooling capacity that the GHP system can produce. One of main concerns in the view of control performance is the behavior of refrigerant at low pressure and COP at such low cooling loads under which the coolant three-way valve and the liquid-injection valve become main actuators to control the pressure and temperature of the refrigerant at the compressor inlet. To monitor the concern, the cooling load was set to be 30 kW for the first 500 sec and 10 kW later on as shown in Fig. 6(b). For the variation of cooling load, Fig. 6(a) shows that simulated refrigerant high and low pressures follow well the target pressures of 20 bar and 5 bar, respectively. Since engine speed can not be reduced below 800 rpm to prevent refrigerant pressure drop, liquid refrigerant has to be injected into the refrigerant gas header



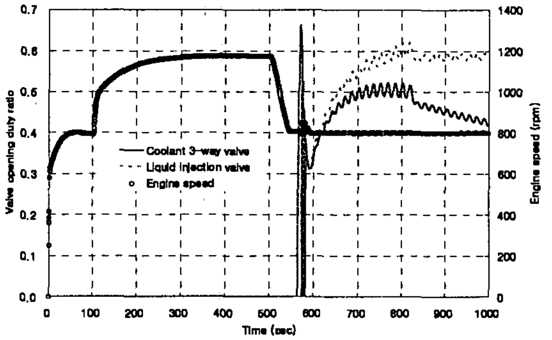
(a) Refrigerant pressures



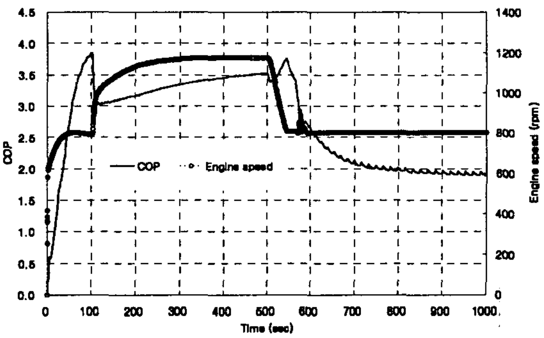
(b) Fan speed and cooling load



(c) Refrigerant temperatures at compressor



(d) Coolant 3-way and liquid injection valves



(e) COP

Fig. 6 Simulation results.

and it should be vaporized by the hot engine coolant via the plate-type coolant heat exchanger, as shown in Fig. 6(d), so that the engine keeps running at 800 rpm. Figure 6(c) represents refrigerant temperatures at the inlet and exit of the compressor.

Figure 6(e) shows COP for the same period. It is shown that COP is about 3.7 for normal load, but it decreases below 2 for very low cooling load. Therefore, the limit of low load for continuous operation must be designed with energy saving target taken into account.

6. Conclusions

The following conclusions were obtained from the study of simulating GHP system dynamics to design control algorithm and evaluate control performance:

(1) Modeling the refrigerant cycle is quite complicated due to two-phase flow heat transfer phenomena. However, through reasoning of simplified physics to derive differential equations, a simple model was obtained which simulated thermal dynamics to serve as a plant for control loop design.

(2) The behavior of simulated temperatures, pressures and COP was found to be similar to those of real systems. Although model refinement is needed through quantitative comparison with experimental data, it may be served at least as a virtual GHP system to test many algorithm candidates. Further development of the model would make it possible to evaluate energy savings and control performance quantitatively with respect to various control strategies.

Acknowledgement

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