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ENERGY AND ECONOMIC PERFORMANCE COMPARISON OF GAS ENGINE AND ELECTRIC DRIVEN AIR-TO-WATER HEAT PUMP

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ABSTRACT

A research and development (R&D) project of air-to-water type natural gas engine driven heat pump (GEHP) is proposed to cater for the increasing supply of natural gas in China. As part of R&D, an in-depth technical comparison is performed to evaluate whether GEHP system is competitive with the large existing electric heat pump (EHP) in energy and economic aspect. In order to be as accurate as possible, an experimental engine waste heat mode and semi empirical heat pump system model, which combined to be a GEHP model, are set up to calculate and compare the performance of GEHP and EHP under the various outdoor temperature. The energy consumption and operating cost are calculated by BIN method based on the weather data and energy price of East China.

1. INTRODUCTION

With the increasing supply of natural gas in East China, the gas engine driven heat pump was proposed to use in HVAC field for its merits of reducing the electric peak and saving energy. Gas engine driven heat pump (GEHP) is a heating and cooling system, which employs a natural gas engine to drive the compressor. Compared with conventional electric driven heat pump (EHP) system, GEHP has following advantages [1]:

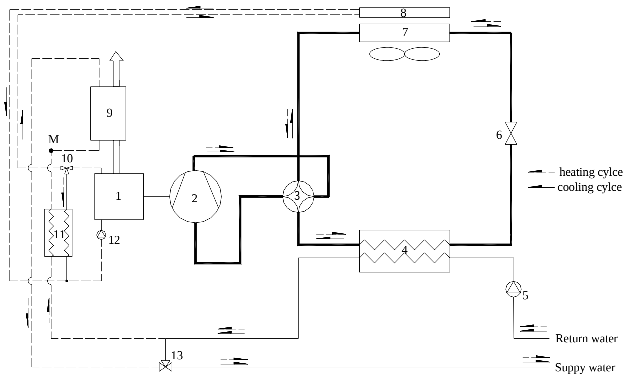
- Higher heating capacity and primary efficiency ratio (PER) because of the extra-recovered heat released by engine cylinder jacket and exhaust gas.
- Excellent part load efficiency and superior comfort due to the convenient speed modulation.
- Equilibrating the consumption of natural gas and electric power in cooling season.
- Reducing the pollution because it is fueled with clean energy resource.

An air-to-water type gas engine driven heat pump is advised to research and develop (R&D) because of its convenience to substitute existing air-to-water electric driven heat pump (EHP) without change domestic air conditioning system. As part of R&D, an in-depth technical comparison is performed to evaluate whether GEHP system is competitive with the large existing electric heat pump on energy and economic aspect. In order to be as accurate as possible, a steady-state model is set up to compare the performance of GEHP with variable speed EHP, especially under the part load condition.

2. SYSTEM DESCRIPTION

The system discussed in this article is an air-to-water heat pump, which can provide cold and hot water for air conditioning purpose in summer and winter respectively. The schematic diagram is shown in Fig.1.The whole system can be divided into two parts: heat pump part and gas engine part, which are connected directly by a shaft. The heat pump part has no difference with the ordinary EHP, including an open compressor, a condenser, an evaporator and an expansion valve. In gas engine part, besides the engine, three additional heat exchangers are used to recover or dissipate the waste heat of gas engine in heating mode or cooling mode.

In heating mode, return water firstly enters the condenser to absorb the condenser heat, then goes through the water-to-water heat exchanger and gas-to-water heat exchanger to recover the engine waste heat, then the hot water again supplies the air conditioning equipments. In cooling mode, the extra heat of engine cylinder is dissipated by the



1. engine
2. compressor
3. four-way valve
4. plate heat exchanger
5. supply water pump
6. expansion valve
7. fin-tube heat exchanger
8. engine waste heat radiation
9. gas to water heat exchanger
10. three-way valve of engine cooling water
11. water to water heat exchanger
12. engine cooling water pump
13. three-way valve of supply water

Fig.1 Scheme of GEHP

radiation to ensure the engine operate reliably. The cold water bypasses through the valve and supplies the air conditioning equipments directly.

3. GEHP SYSTEM MODEL

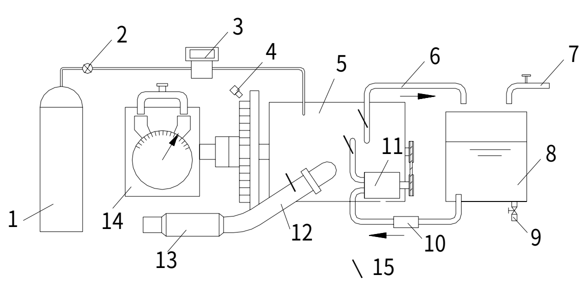
3.1 Engine System

3.1.1 Experimental I.C. gas engine waste heat model

The thermodynamic model of engine can be got by two ways: mathematic or experimental way. Because of the complexity of thermodynamic process in the engine, mathematic modeling is a quite complicated task. Instead, taking experiment is a simple but also accurate way to acquire the thermodynamic characteristics needed for GEHP system modeling [2].

The steady working condition of the engine is the function of load and speed [3]. If the engine load and speed are fixed, all the thermodynamic parameters are kept certainly. In GEHP system calculation, the necessary parameters are engine cylinder jacket heat flux, cooling water flow rate, exhaust gas temperature, exhaust gas flow rate and fuel flow rate.

To get the relations between engine thermodynamic parameters with a wide range of load and speed, an experiment on engine waste heat is performed on a natural gas engine, which is modified from a gasoline engine. The facilities for experiment are shown in Fig.2.



1. natural gas storage
2.pressure relief valve
3.natural gas meter
4.tacho-generator
5.engine
6.hot water pipe
7.supplymental water
8. water tank
9. draw off valve
10.water flow meter
11.water pump
12.exhaust pipe
13.silencer
14.hydrodynamometer
15.thermometer

Fig.2 Diagram of engine waste heat experiment

Among the five parameters needed for GEHP system modeling, fuel flow rate, cooling water flow rate and exhaust gas temperature are measured directly, the other two variables are calculated by the following ways.

Engine cylinder waste heat flux is calculated by

$$Q_{CYL} = c_{p,w} \cdot m_{ecw} \cdot \left(t_{ecw,out} - t_{ecw,in}\right) \qquad (1)$$

Exhaust gas flow rate is calculated by [3]

$$m_{exh} = m_{fuel} \cdot (1 + AF) \tag{2}$$

AF is the air to fuel ratio, which can be calculated by

$$AF = \gamma \cdot (AF)_0 \tag{3}$$

$(AF)_0$ is theoretical air to fuel ratio of natural gas.

Second-order bivariate regression polynomial form is employed here to express the relations between required parameters and speed and load [2].

$$y = c_1 + c_2 n + c_3 n^2 + c_4 T_r + c_5 T_r^2 + c_6 n T_r + c_7 n T_r^2 + c_8 n^2 T_r + c_9 n^2 T_r^2 \tag{4}$$

Here, output y represents the above five required parameters.

There are altogether 66 valid data points with the speed ranging from 940 R.P.M. to 3400 R.P.M., and torque ranging from 50 Nm to 110 Nm. The Constants of Polynomials are shown in Table 1. All of the polynomials have the credence of 90%, so they can perfectly describe the characteristics of engine waste heat.

Table1 Constant of polynomials

	Cylinder heat flux	Cooling water flow rate*	Exhaust gas temperature	Exhaust gas flow rate	Fuel flow rate
c_1	-1.5638e+002	1.4847e-002	2.7681e+002	1.5339e-003	1.3840e-004
c_2	2.2980e-002	1.5424e-004	4.5368e-001	6.4712e-006	3.1668e-007
c_3	1.1600e-005	0	-1.1947e-004	1.2038e-009	-5.7714e-011
c_4	4.8484e+000	0	3.1762e-001	7.3855e-005	-1.37834e-006
c_5	-3.3069e-002	0	1.1069e-003	3.6009e-007	2.0045e-008
c_6	-1.5148e-003	0	-7.0686e-003	8.4211e-008	-9.7800e-010
c_7	-1.9910e-008	0	2.3667e-006	4.1301e-011	2.2233e-012
c_8	1.3667e-005	0	4.6388e-005	8.1626e-010	2.7029e-011
c_9	-9.0345e-010	0	-1.4840e-008	2.2859e-013	-1.0988e-014

*Cooling water pump is driven by engine, so at the given pressure difference, cooling water flow rate is only the linear function of engine speed.

3.1.2 ENGINE WASTE HEAT RECOVERY SYSTEM

As Fig.1 shows, in the heating mode, the cylinder waste heat is recovered by a water-to-water heat exchanger and exhaust gas waste heat is recovered by a gas-to-water heat exchanger.

The heat that cool water gains in water-to-water heat exchanger can be calculated by

$$Q_{WWHE} = c_{p,w} m_{rw} \left(t_M - t_{PHE,out}\right) \tag{5}$$

If heat loss is ignored

$$Q_{WWHE} = Q_{CLY} \tag{6}$$

Heat balance in gas-to-water heat exchanger satisfies

$$Q_{EWHE} = c_{p,w} m_{sw} \left(t_{sw} - t_M\right) = c_{p,exh} \left(t_{exh,in} - t_{exh,out}\right) \tag{7}$$

The heat transfer process can be expressed by the logarithmic mean temperature difference method

$$\frac{Q_{EWHE}}{(UA)_{EWHE}} = \frac{\left(t_{exh,in} - t_{sw}\right) - \left(t_{exh,out} - t_{hrw}\right)}{\ln\left[\left(t_{exh,in} - t_{sw}\right) / \left(t_{exh,out} - t_{hrw}\right)\right]} \tag{8}$$

Here, $(UA)_{EWHE}$ is the product of overall heat transfer coefficient and heat transfer area of gas to water heat exchanger. In this paper, the variation of $(UA)_{EWHE}$ is negligible because it will not affect calculation results of energy consumption much. So $(UA)_{EWHE}$ is regarded as constant at nominal condition provided by heat exchanger manufacture.

3.2 HEAT PUMP MODEL

3.2.1 COMPRESSOR MODEL

Compressor is also an important component of GEHP system and its performance affects the system much. To acquire a more practical compressor model, several semi-empirical equations for variable speed compressor are employed here [4].

The mass follow rate can be calculated by

$$m_{ref} = nV\rho_{in}\left(1 + C_k - D_k\left(\frac{P_{out}}{P_{in}}\right)^{\frac{1}{\kappa}}\right) \tag{9}$$

An isentropic efficiency is used to relate inlet and outlet enthalpies

$$\eta_k = \frac{h_{out,isentropic} - h_{in}}{h_{out} - h_{in}} \tag{10}$$

It can be calculated by

$$\eta_k = A_k\left(\frac{P_{out}}{P_{in}}\right) + B_k \tag{11}$$

A_k, B_k, C_k and D_k are empirical constant coefficients, which can be fitted from compressor manufacture's test data.

The input work of compressor is calculated by

$$Ne = m_{ref} \cdot \left(h_{out} - h_{in}\right) \tag{12}$$

The relation between work and torque is given by

$$Ne = T_r \cdot n / 9550 \tag{13}$$

3.2.2 HEAT EXCHANGER MODEL

3.2.2.1 PLATE HEAT EXCHANGER

A plate heat exchanger is used here as the water cooling and heating facility for its virtues of compact structure and high efficiency. It works as evaporator in summer and condenser in winter respectively. To simplify the calculation, a one-dimension and lumped parameter model is used here for plate heat exchanger.

In cooling mode, heat gained in refrigerant side and heat lost in water side can be calculated as

$$Q_{PHE,C} = m_{ref}\left(h_{ref,PHE,out} - h_{ref,PHE,in}\right) = m_{sw}c_{p,w}\left(t_{rw} - t_{sw}\right) \tag{14}$$

The logarithmic mean temperature difference method is used here to express the heat transfer process in plate heat exchanger.

$$\frac{Q_{PHE,C}}{(UA)_{PHE,C}} = \frac{\left(t_{rw} - t_{ref,PHE,in}\right) - \left(t_{sw} - t_{ref,PHE,out}\right)}{\ln\left[\left(t_{rw} - t_{ref,PHE,in}\right) / \left(t_{sw} - t_{ref,PHE,out}\right)\right]} \tag{15}$$

In heating mode, heat lost in refrigerant side and heat gained in water side can be calculated as

$$Q_{PHE,H} = m_{ref}\left(h_{ref,PHE,in} - h_{ref,PHE,out}\right) = m_{sw}c_{p,w}\left(t_{hrw} - t_{rw}\right) \tag{16}$$

The flow in the plate heat exchanger become count-flow in heating mode, and heat transfer process then is represented by

$$\frac{Q_{PHE,H}}{(UA)_{PHE,H}} = \frac{\left(t_{ref,PHE,out} - t_{rs}\right) - \left(t_{ref,PHE,in} - t_{hrw1}\right)}{\ln\left[\left(t_{ref,PHE,out} - t_{rs}\right) / \left(t_{ref,PHE,in} - t_{hrw1}\right)\right]} \tag{17}$$

3.2.2.2 FIN-TUBE HEAT EXCHANGER

A fin-tube heat exchanger is used here to achieve the heat transferred between air and refrigerant. It works as a condenser and evaporator in summer and winter respectively. Also a one-dimension and lumped parameter model is used here for fin-tube heat exchanger.

In cooling mode, heat lost in refrigerant side and heat gained in air side can be calculated as

$$Q_{FTHE,C} = m_{ref}\left(h_{ref,FTHEin} - h_{ref,FTHE,out}\right) = m_a c_{p,a}\left(t_{a,ouu} - t_{a,in}\right) \tag{18}$$

The heat transfer process in tube-fin heat exchanger can be represented by

$$\frac{Q_{FTHE,C}}{(UA)_{FTHE,C}} = \frac{\left(t_{ref,FTHE,in} - t_{a,in}\right) - \left(t_{ref,FTHE,out} - t_{a,out}\right)}{\ln\left[\left(t_{ref,FTHE,in} - t_{a,in}\right) / \left(t_{ref,FTHE,out} - t_{a,out}\right)\right]} \tag{19}$$

In heating mode, heat gained in refrigerant side and heat lost in air side can be calculated as

$$Q_{FTHE,H} = m_{ref}\left(h_{ref,FTHE,out} - h_{ref,FTHE,in}\right) = m_a c_{p,a}\left(t_{a,in} - t_{a,out}\right) \tag{20}$$

Heat transfer process is expressed as

$$\frac{Q_{FTHE,H}}{(UA)_{FTHE,H}} = \frac{\left(t_{a,in} - t_{ref,FTHE,out}\right) - \left(t_{a,out} - t_{ref,FTHE,in}\right)}{\ln\left[\left(t_{a,in} - t_{ref,FTHE,out}\right) / \left(t_{a,out} - t_{ref,FTHE,in}\right)\right]} \tag{21}$$

In above heat transfer equations, (UA) is the product of overall heat transfer coefficient and heat transfer area of plate heat exchanger and fin-tube heat exchanger. Because the secondary fluids (air and water) flow rate are constant, the variation of (UA) cased by refrigerate flow rate change is not so enormous that the calculation result of energy consumption will be affected remarkably. To simplify the calculation they are assumed as constant equal to the number in nominal condition, which is provided by heat exchanger manufacture.

3.2.3 Expansion valve

Refrigerant flow through expansion valve can be regarded as isentropic process. Therefore, energy equation can be simplified as

$$h_{EV,in} = h_{EV,out} \tag{22}$$

In steady state, the mass flow rate in expansion valve equals to mass flow rate in compressor.

Eqs. (1)~(22) constitute the whole GEHP steady-state model. If the engine system model is omitted, the EHP model is obtained and its performance can be calculated.

4. CALCULATION CONDITION

4.1 CLIMATE CONDITION

Climate condition is a vital factor affecting the performance of air-sourced heat pump. In addition, it affects the building load remarkably. In this paper, BIN method [5] is employed to calculate the energy consumption, therefore, the temperature frequency is adopted here to represent the climate condition. In this paper the temperature frequency of Shanghai area [6] is adopted here to represent the climatic character of East China.

4.2 BUILDING LOAD

To simply the calculation, the building load is assumed as straightly linear to outside dry temperature. Therefore the cooling and heating load at given outside temperature can be calculated by

$$Q_x = Q_d \frac{t_i - t_o}{t_i - t_{o,d}} \tag{23}$$

In fig.3 and 4, the solid black line shows the cooling and heating load variation with the change of outside temperature.

5 RESULTS AND DISCUSSION

5.1 PERFORMANCE OF GEHP

The important parameter to evaluate the performance of GEHP is the primary energy ratio (PER), which is defined as

$$PER = \frac{Cooling/heating \quad ability}{priamry \quad energy} \tag{24}$$

As the EHP, PER can be calculated by

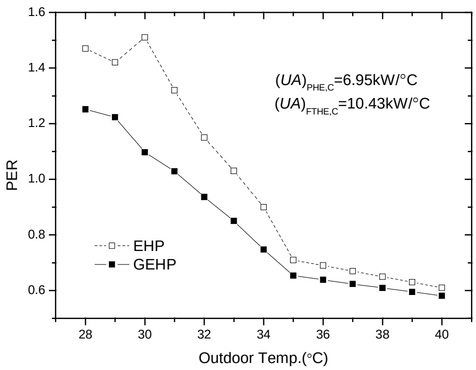
$$PER = COP\eta_1\eta_2 \tag{25}$$

Here, *COP* is the coefficient of performance of heat pump and η_1,η_2 are the efficiency of power plant and power transmission.

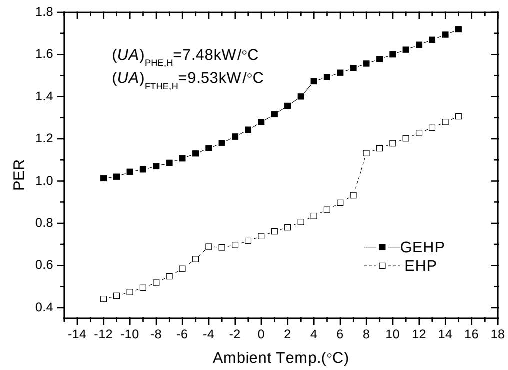
5.2 PERFORMANCE OF GEHP AND EHP IN VARIABLE SPEED MODE

In both cooling and heating mode, the capacity of GEHP can be conveniently modulated by speed modulating system. The analysis under variable speed mode is more meaningful than in constant speed mode. One important thing should be noted is that the compressor speed could not be modulated randomly. Too low engine speed will lead to unstable engine output torque while too high speed will lead to compressor failure. In both GEHP and EHP, the accepted speed range is set from 1000R.P.M. to 3000R.P.M. If the speed requirement is out of this range, the compressor speed is limited to its maximum or minimum speed.

Fig.3 compare the performance variation of GEHP and EHP in the variable speed mode. In cooling mode, the PER of GEHP is lower than that of EHP, especially under lower outdoor temperature. This is because that the engine thermal efficiency is lower than $\eta_1\eta_2$. Under lower outside temperature, the decreasing of cooling load lead to reduction of engine speed, consequently the thermal efficiency degrades much. In heating mode, *PER* of GEHP is much higher than EHP. The increasing of *PER* in GEHP is the result of the waste heat recovery in engine. In calculation, the additional electric heating is considered in lower outdoor temperature in EHP system.

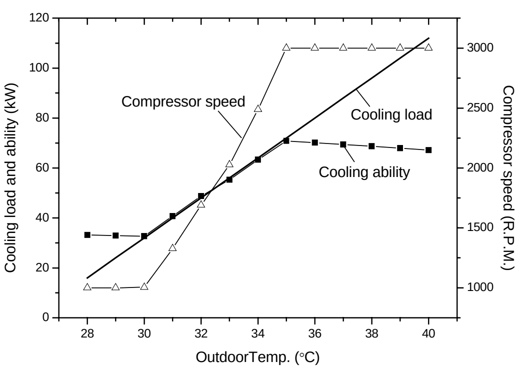


(a) Cooling mode

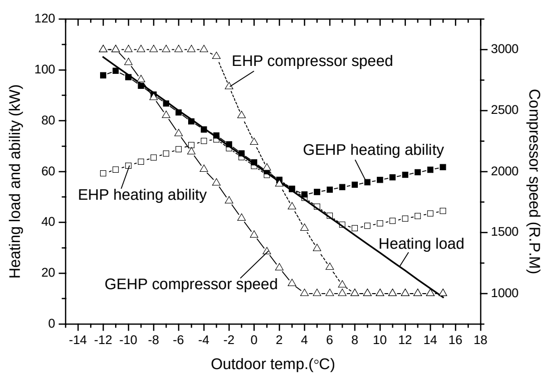


(a) Heating mode

Fig.3 PER comparison of GEHP and EHP in cooling and heating mode



(a) Cooling mode



(a) Heating mode

Fig.4 Heating(cooling) load, ability and speed variation of GEHP and EHP

Further analysis on cooling/heating ability and speed variation of GEHP and EHP with variation of outdoor temperature is shown in Fig.4. Fig.4(a) shows that GEHP has same cooling ability with EHP. However, in heating mode, the situation is different. Because the waste heat is recovered for heating, GEHP has much higher heating ability than EHP. In the low temperature, the heating quantity of GEHP can match the heating requirement well unless the outdoor temperature decrease to –9°C, while the EHP is only limited to –3°C. So GEHP has better heating performance than EHP, especially in lower outside temperature.

In heating mode, with the increasing of outdoor temperature, both GEHP and EHP works in part load situation. Speed variation curve shows that GEHP arrives its constant speed mode when the outdoor temperature is 4°C. If the outdoor temperature is higher than 4°C, the heating ability will be higher than heating requirement, and then some energy will be wasted. To reduce the energy consumption, certain control strategy is necessary for GEHP under the part load condition.

When GEHP works in its lowest constant speed mode, the engine performance degrades much. Not only the thermal efficiency of engine decrease, but also the available waste heat is much lower. Therefore, the characteristics of engine affect the performance of GEHP much.

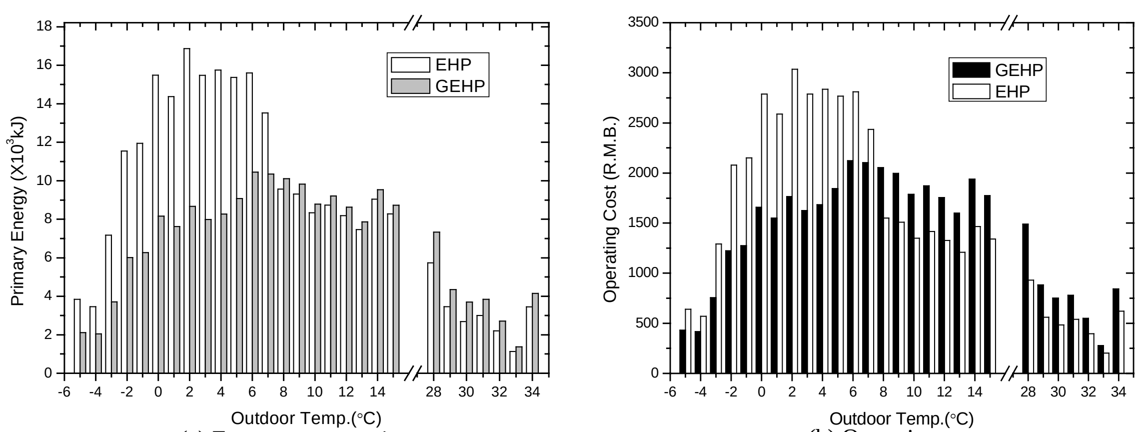
5.3 ENERGY CONSUMPTION AND OPERATING COST ANALYSIS

5.3.1 ENERGY CONSUMPTION

The energy saving is the one of the most important topics in the world. It is necessary to compare energy consumption of GEHP and EHP. As mentioned above, BIN method is employed here to calculate the energy consumption based on outdoor temperature frequency of Shanghai area. The BIN method of energy estimating has the advantage of allowing off-design calculation [5]. In the calculation, the lowest temperature of each temperature interval represents the whole temperature interval. The -6°C represents all the temperature lower than -5°C and 34°C represent all the temperature higher than 34°C. Fig.6 shows the comparing result of the primary energy consumption of GEHP and EHP in the climate of Shanghai area.

Energy consumption comparison shows that, GEHP saves energy only in the range of outside temperature lower than 7°C. When the temperature is higher than 7°C, GEHP consume more primary energy than EHP. In heating mode, when outdoor temperature is higher than 7°C, both of GEHP and EHP are limited to their max speed, but more heating volume is wasted by GEHP if heat pump is continually operated. In cooling mode, GEHP consume more primary energy is due to the low thermal efficiency of gas engine.

In the whole, the GEHP can save about 23% primary energy than EHP annually based on the climate of Shanghai area.



(a) Energy consumption (b) Operating cost

Fig.7 Energy consumption and operating cost comparison of GEHP and EHP

5.3.2 OPERATING COST

Operating cost is an import issue to be discussed for the reason that it affects the payback period ultimately, for which the owner mostly concerns. Fig.8 shows the operating cost comparison of GEHP and EHP with the variation of outside temperature based on energy price of Shanghai area. The energy price is ￥2.10/m^3 (RMB) for natural gas and ￥0.61/kWh (RMB) for electric. Comparing result also shows that GEHP has excellent economic performance in the low temperature due to the excellent performance of GEHP. While in higher outside temperature, the economic of EHP is better. Integrally, GEHP can save 11.3% operating cost annually.

6. CONCLUSIONS

GEHP has better heating performance than EHP due to utilization of recovered waste heat of engine. However, GEHP does not always work better than EHP in whole heating and cooling season. The weakness of cooling performance of GEHP is due to the waste of engine waste heat and lower engine thermal efficiency. To improve the cooling performance of GEHP, the thermal efficiency of engine should be improved at least to 30%, or waste heat should be utilized.

Compared with EHP, GEHP consumes more energy and operating cost in part load of heating mode, which is due to the speed limitation of engine. Therefore, certain control strategy (such as on/off control) is still quite necessary for GEHP.

Integrally, GEHP can save about 23% primary energy consumption and 11.2% operating cost based on the climate and energy price of East China.

NOMENCLATURE

A	heat transfer area (m^2)
AF	air to fuel ratio
A_k	coefficient of compressor model
B_k	coefficient of compressor model
$c1\sim c9$	polynomial constant
C_k	coefficient of compressor model
COP	coefficient of performance
c_p	heat capacity at constant pressure (kJ kg^{-1} K^{-1})
D_k	coefficient of compressor model
H	enthalpy of refrigerant (kJ kg^{-1})
m	mass flow rate (kg s^{-1})
Ne	input work of compressor (kW)
N	compressor or engine speed (R.P.M.)
P	pressure (Pa)
Q	heat transfer rate (kW)
Q_x	heating or cooling load (kW)
T_r	torque (N m)
T	temperature (°C)
V	displacement volume of compressor (m^3)
Y	output value of engine waste heat mode
Greeks	
Λ	coefficiency of extra air
P	density of refrigerant (kg m^{-3})
K	polytropic exponent
η_k	isentropic efficiency

Subscripts

a	air
C	cooling
CYL	cylinder
d	design condition
EV	expansion valve
EWHE	exhaust to water heat exchanger
ecw	engine cooling water
exh	exhaust gas
FTHE	fin-tube heat exchanger
fuel	natural gas fuel
H	heating
i	indoor
in	inlet
M	the point of output of WWHE
o	Outdoor
out	Outlet
PHE	plate heat exchanger
rw	return water
ref	Refrigerant
sw	supply water
WWHE	water to water heat exchanger
w	Water

REFERENCES

[1] D.D. Colosimo, 1987, Introduction to engine-driven heat pumps – concepts, approach, and economics, ASHRAE Trans. vol.93, no.2: p.987~996.
[2] R.P. Rusk, J.H. Van Gerpen, 1990, Development and use of a mathematical model of an engine-driven heat pump, ASHRAE Transactions, vol.96, no.2: p.282-290.
[3] D.M. Jiang, 1988, Theory of internal combustion engine, Mechanical industry press, Beijin.
[4] McQuiston, F. Parker, J., 1994, Heating ventilation and air-conditioning, design and analysis, 4th edition, Wiley.
[5] H.J. Sauer, R.H. Howell, 1983, Heat pump system, John Wiley & Sons, Inc., New York.
[6] P.L. Chen, 1990, Manual of air conditioning and refrigeration technology, Tongji University Press, Shanghai, 1990