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Modeling and optimization of a hybrid-power gas engine-driven heat pump

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Abstract

Based on the coaxial HPGHP system experiments, the mathematical model of the various components of the coaxial HPGHP system is established to research the matching relations between the drive system and the load demand of the compressor. This paper establishes the thermal efficiency of the engine model based on load rate, to maintain the thermal efficiency always be above 0.25, combined with the thermal efficiency map of the engine, the best economic zone of engine operation is determined. Finally, Torque optimization model of the coaxial HPGHP system was proposed, the transmission ratio of the HPGHP system was optimized and mode control strategy of the coaxial HPGHP system was determined. Simulation results show the thermal efficiency of the HPGHP system can be always been guaranteed above 0.25 and the transmission ratios are respectively 2.9, 1.8 and 1.4 in three different load demand range.

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Keywords: Energy management, HPGHP, SOC, transmission ratio, torque limits, control strategy

1. Introduction

Since the beginning of the 21st century, energy and the environment has become a national strategic issues remaining to be resolved. Natural gas, which is considered as a clean energy, will occupy an important part in the future energy structure. Development of distributed energy systems on natural gas will be a major trend in the future

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of energy development. So development of natural gas as a new energy driving air conditioning equipment will be particularly important. The GHP system is a kind of system which can realize the different heating and cooling load demands through changing the operating conditions of the engine [1-2]. However, the engine will depart from the economic zone in the low- load operation, which leads to a decline in the thermal efficiency. In this paper, air Conditioning and Refrigeration Laboratory of Southeast University proposes a kind of the HPGHP system, which combines hybrid power technology with gas engine heat pump [3-7]. Through the rational allocation of torque, speed and power of engine and the motor, the engine can be remained in the economic zone. Li Yinglin proved that the thermal efficiency is 27%~37% higher than the conventional gas heat pump system [8]. Wang Yanwei put forward a kind of energy control strategy which was based on the minimum gas consumption rate [9]. Wang Jieyue carried out the simulation and optimization of the HPGHP system [10].

However, in order to ensure the HPGHP system runs in the economic area all the time and minimize the gas consumption rate and emission levels, in this paper, through the establishment of the comprehensive charging/discharging efficiency model of the HPGHP system, Finally, the transmission ratio of the HPGHP system is optimized and control strategy mode of the coaxial HPGHP system was determined.

Nomenclature

η_f the thermal efficiency of the gas engine
T_f the torque of the gas engine[Nm]
ω_f the speed of the gas engine[rpm]
T_d the torque of the motor[Nm]
ω_d the speed of the motor[rpm]
ηd the thermal efficiency of the motor
η_{dc} the motor charging efficiency
η_{df} the motor discharging efficiency
η_b transmission efficiency[%]
η_n inverter efficiency[%]
$\overline{\eta}$ average comprehensive efficiency[%]
Q_h the heating capacity[KW]
Qc the cooling capacity[KW]
ω the speed of the compressor [rpm]

Subscripts
η efficiency [%]
T the torque [Nm]
ω the speed[rpm]

Abbreviations
GHP gas engine heat pump
HPGHP hybrid-power gas engine heat pump
SOC the state of charge

2. Principle of the coaxial parallel-type HPGHP system

As is showed in Fig.1,a coaxial parallel-type HPGHP system has two power sources:the engine and the motor.Both the engine and the motor can be run separately or together to drive the heat pump system.The motor which can be used as a generator or a electric motor can work together with the engine throttle opening,which can adjust the engine real-time speed and torque as well as control the engine in its economic zone.As a result, efficiency of the HPGHP system is improved and emissions are reduced.

The HPGHP system working mode is divided into five forms: the engine drive alone(mode D); the motor drives alone(mode M);the engine and motor drive together and the engine is running in the economic zone(mode L);the engine drives compressor and through generator to generate electricity at the same time(mode C);the engine drives the generator to generate electricity(mode S).The motor works as an auxiliary power source.The engine and the motor share a shaft connection and have the same speed when running at the same time.In this paper,in order to match the load demand and engine operation condition,the HPGHP system uses multi-stage transmission ratio,which is designed according to three different load demand range.

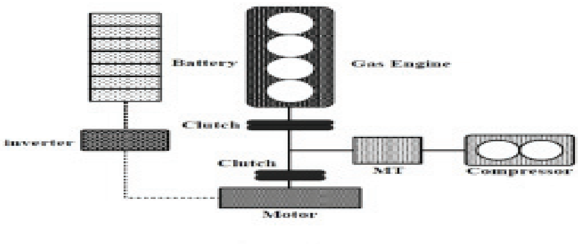

Fig. 1. The coaxial parallel-type HPGHP.

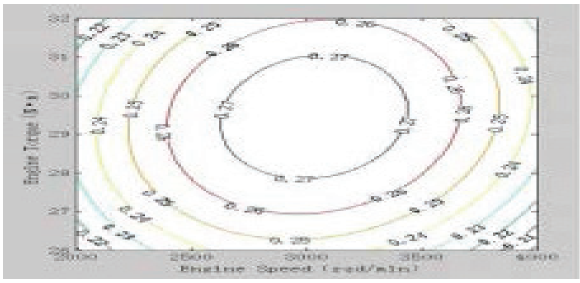

Fig. 2 .Thermal efficiency map of the engine.

3. Mode of the HPGHP system

3.1. Mode of the engine

This paper mainly discusses the steady-state performance of the HPGHP system under different loads of the compressor and focus on the establishment of a steady state model of the engine.By means of experimental modeling methods,the equation of curved surface fits the thermal efficiency of torque and speed,finally,the thermal efficiency of the engine characteristic curve model are established.The maximum output power of the engine is 12.5KW;the maximum torque is 31.9Nm;the highest thermal efficiency is 0.279 and the rated speed is 2900~3100 rpm. According to multiple linear regression theory [11], the establishment of the thermal efficiency characteristic curve model is shown in formula (1) and Fig.2.

$$\begin{bmatrix}\eta_f 1\\ \eta_f 2\\ \vdots\\ \eta_f N\end{bmatrix}=\begin{bmatrix}1 & \omega_{f1} & T_{f1} & \omega_{f1}^2 & \omega_{f1}T_{f1} & T_{f1}^2\cdots\omega_{f1}^l & \omega_{f1}^{l-1}T_{f1} & \cdots & T_{f1}^l\\ 1 & \omega_{f2} & T_{f2} & \omega_{f2}^2 & \omega_{f2}T_{f2} & T_{f2}^2\cdots\omega_{f2}^l & \omega_{f2}^{l-1}T_{f2}\cdots & & T_{f2}^l\\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots\quad\vdots\quad\vdots & & \vdots & \\ 1 & \omega_{fN} & T_{fN} & \omega_{fN}^2 & \omega_{fN}T_{fN} & T_{fN}^2\cdots\omega_{fN}^l & \omega_{fN}^{l-1}T_{fN} & \cdots T_{fN}^l & \end{bmatrix}\times\begin{bmatrix}a_0\\ a_1\\ \vdots\\ a_{k-1}\end{bmatrix}+\begin{bmatrix}e_0\\ e_1\\ \vdots\\ e_N\end{bmatrix}$$

$$A_m=\begin{bmatrix}-2.244 & 1.838E-4 & 0.1524 & -4.169E-8 & 2.362E\text{-}6 & \text{-}2.709E\text{-}3\end{bmatrix}$$

$$\eta_f=\begin{bmatrix}1 & \omega_f & T_f & \omega_f^2 & \omega_f T_f & T_f^2\end{bmatrix}A_m^T \qquad (1)$$

where a_i is model coefficient, e_i is stochastic error, N is number of test points, Am is the coefficients of the regress equation.

In order to analyze the influence of gas loss on the drive system efficiency and determine the best economic zone of the engine operation,this paper establishes the thermal efficiency of the engine model based on load rate,under the assumption that the thermal efficiency of the engine is a single valued function of load rate.Engine load rate is the ratio of the actual output power to the rated engine power.the thermal efficiency increases first and then decreases with increasing load rate.when the load rate is in the vicinity of 0.75,the thermal efficiency is at its maximum.as is shown in Fig.3.To maintain the thermal efficiency always be above 0.25,namely $\eta_f>0.25$,combined with the thermal efficiency map of the engine,the load rate is controlled in the range of 0.56-0.84 and the output power is controlled in the range of 7-10.5KW.In this power range,the engine is always running on the optimal torque curve.Engine optimal torque curve describes the highest engine thermal efficiency at corresponding points under different output power,and the simulation optimal torque curve is showed in formula(3.2).Fig.4 shows the thermal efficiency of the gas engine on the optimal torque curve.

$$T_f = 4.36\times10^{-4}\omega_f + 28.13 \qquad (2)$$

3.2. Mode of the motor[12]

In the HPGHP system,the motor can be used as the generator in charging mode and electric motor in discharging mode.Motor modeling is similar to engine modeling,By means of experimental modeling methods,the equation of curved surface fits the thermal efficiency of torque and speed,finally,the thermal efficiency of the motor characteristic curve model are established [13].As shown in Fig. 5 is the efficiency map of the motor,whether in the charging or discharging mode,the efficiency of the motor can maintain a high level.The engine and the motor share a shaft connection and have the same speed when running at the same time.The engine speed can be controlled in 2100-3800rpm due to the adoption of multistage transmission ratio,so the motor efficiency can keep almost above 0.9.

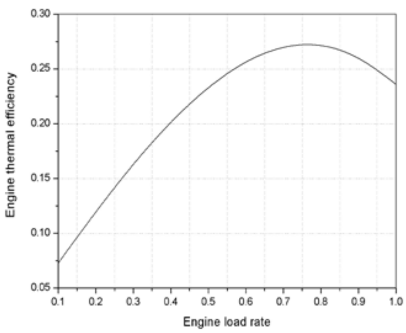

Fig. .3 Engine load rate and thermal efficiency curve.

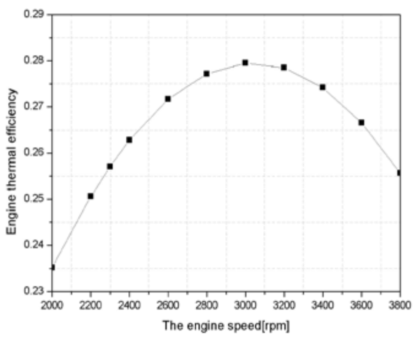

Fig.4. Engine thermal efficiency on the optimal torque curve.

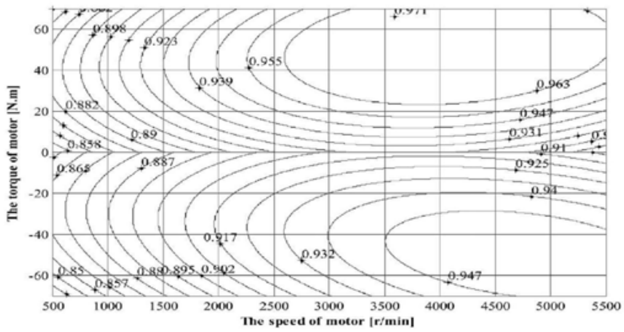

Fig.5. Motor efficiency map

3.3. The influence of compressor speed variation on the HPGHP system [13]

Heating condition applies in winter for heat supply.The indoor heat ex-changer is the condenser and the outdoor heat ex-changer is the evaporator.Environmental parameters of the system are shown in Table 1. Cooling condition occurs mainly in summer.This moment,indoor heat ex-changer is evaporator,and the outdoor heat ex-changer is condenser.The environmental parameters of this operation are shown in Table 2.

Table 1.Environmental parameters of test condition

Outdoor dry bulb temperature	Condensing temperature	Evaporating temperature
7℃	46℃	5℃

Table 2 Environmental parameters of test condition

Indoor dry bulb temperature	Condensing temperature	Evaporating temperature
26℃	42℃	5℃

Heating capacity,cooling capacity and input power of compressor can be expressed as a function of compressor speed,as is shown in formula (3)-(5).

$$Q_c = a\omega^m \tag{3}$$

$$Q_h = b\omega^p \tag{4}$$

$$P_i = c\omega^r \tag{5}$$

Where a,b,c,m,p and r are model coefficients and are respectively 0.3427,7.304E-2,3.668E-3,0.5395,0.7937 and 1.055.when the value of m,p,r are larger,the magnitude of cooling capacity and heating capacity as well as input power of compressor will become larger with the increasing speed of compress,As a result,the regulating compressor speed can achieve the variable load operation of the HPGHP system.The relation between heating capacity,cooling capacity,input power of compressor and compressor speed are shown in Fig.6 and Fig.7.

The cooling performance coefficient ICOP,c and the heating performance coefficient ICOP,h are two important technical and economic indexes to evaluate the performance of the HPGHP system,which is shown in formula (6) and formula (7).

$$I_{COP,c} = \frac{Q_c}{P_i} = \frac{a}{c} \cdot \omega^{m-r} \tag{6}$$

$$I_{COP,h} = \frac{Q_h}{P_i} = \frac{b}{c} \cdot \omega^{p-r} \tag{7}$$

The ratio of part load compressor speed to full load compressor speed is equals to the speed ratio of compressor,as is shown in formula (8).

$$\alpha = \frac{\omega}{\omega_m} \tag{8}$$

The ratio of part cooling load coefficient of performance to full cooling load coefficient of performance is described as β cop,c,as is shown in formula(9).Similarly,the ratio of part heating load coefficient of performance to full heating load coefficient of performance is β cop,c,as is shown in formula(10).

$$\beta_{COP,c} = \frac{I_{COP,c}}{I_{COP,c,f}} = \alpha^{m-r} \tag{9}$$

$$\beta_{COP,h} = \frac{I_{COP,h}}{I_{COP,h,f}} = \alpha^{p-r} \tag{10}$$

The ratio of part load cooling capacity to full load cooling capacity is β cop,c,as is shown in formula (11).Similarly,the ratio of part load cooling capacity to full load cooling capacity is β cop,c,as is shown in formula (12).

$$\gamma_c = \frac{Q_c}{Q_{c,f}} = \left(\frac{\omega}{\omega_m}\right)^m \tag{11}$$

$$\gamma_h = \frac{Q_h}{Q_{h,f}} = \left(\frac{\omega}{\omega_m}\right)^p \tag{12}$$

So, $\alpha = \gamma_c^{1/m}$ (13)

$$\alpha = \gamma_h^{1/p} \tag{14}$$

$$\beta_{COP,c} = \gamma_c^{(m-r)/m} \tag{15}$$

$$\beta_{COP,h} = \gamma_h^{(p-r)/p} \tag{16}$$

No matter in the perspective of variation of compressor speed or the variable load demand,when m>r, β COP,c≤1,the value of ICOP,c will increase with the increasing value of α or γ c.Similarly,when p>r, β COP,h≤1,the value of ICOP,h will increase with increasing value of α or γ h.the relation between ICOP,c,ICOP,h and α is shown in Fig.8.

The above discussion indicates that the influence of the ratio of compressor speed and variable load demand on coefficient of performance of the HPGHP system are the same,from which we can conclude that the variable load can be meet by changing the compressor speed;cooling or heating coefficient of performance of the HPGHP system which changed with the variable speed ratio of the compressor depend on the difference of change degree index between the cooling capacity(m) as well as heating capacity(p) and the compressor input power(r).

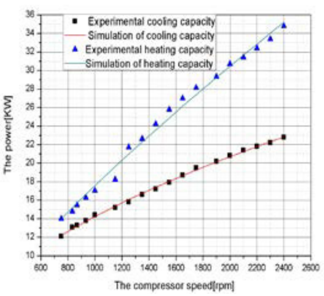

Fig.6.The relation between the heating capacity, The cooling capacity and the speed of compressor.

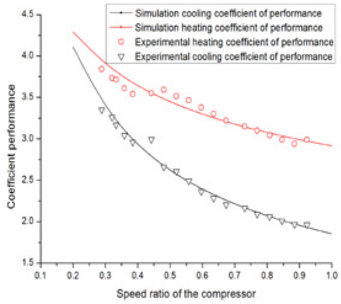

Fig.7.The relation between the Compressor power and the speed of compressor.

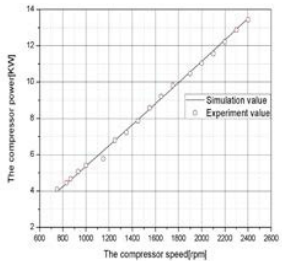

Fig.8.The relation between speed ratio of the compressor and coefficient of performance.

4. Optimization of the transmission ratio and a new energy management control strategy

Designing the transmission ratio of the HPGHP system must consider to meet the speed requirements as well as energy consumption efficiency.The transmission ratio is relation between maximum speed of the engine and compressor,and optimum speed of the engine in the economic zone.therefore,the transmission ratio is different at different load demand range.From formula(2).the output power of the engine is controlled in 7-10.5KW,this moment,the compressor speed,the cooling capacity and the heating capacity are controlled in 1250-1900rpm,16-20KW,21-29KW,respectively.

When the cooling capacity or heating capacity is in 16-20KW or 21-29KW,respectively.the compressor speed is controlled in 1250-1900rpm.the engine speed is controlled in 2300-3600 by the model of optimized torque curve.in this case,r=1.6, $\overline{\eta}$ =0.26554,r=1.7, $\overline{\eta}$ =0.26955,r=1.8, $\overline{\eta}$ =0.27132,r=1.9, $\overline{\eta}$ =0.27085,r=2.0， $\overline{\eta}$ =0.26815,the maximum of comprehensive efficiency of the HPGHP system is 0.27132.therefore,r=1.8.the HPGHP system drive into the mode M.similarly.when the cooling capacity or heating capacity is less than 16KW or less than 21KW,respectively.the compressor speed is controlled in 750-1250rpm and the engine speed is controlled in 2300-3800 by the model of optimized torque curve.in this case.r=2.7, $\overline{\eta}$ =0.2655, r=2.8, $\overline{\eta}$ =0.267125,r=2.9, $\overline{\eta}$ =0.267975,r=3.0, $\overline{\eta}$ =0.26795,r=3.1, $\overline{\eta}$ =0.26705.the maximum of comprehensive efficiency of the HPGHP system is 0.267975.therefore,r=2.9.the HPGHP system drive into the mode C when the value of SOC is less than 0.3.otherwise the HPGHP system drive into the mode M.similarly.when the cooling capacity or heating capacity is greater than 20KW or greater than 29KW,respectively.the compressor speed is controlled in 1900-2400rpm and the engine speed is controlled in 2900-3600 by the model of optimized torque curve.in this case.r=1.3, $\overline{\eta}$ =0.27532,r=1.4, $\overline{\eta}$ =0.27739,r=1.5, $\overline{\eta}$ =0.2757,r=1.6, $\overline{\eta}$ =0.27019,r=1.7, $\overline{\eta}$ =0.26077,the maximum of comprehensive efficiency of the HPGHP system is 0.27739.therefore,r=1.4.the HPGHP system drive into the mode L.the relation between the comprehensive efficiency and compressor speed is shown in Fig.13.

After the transmission ratio is determined,there,,the engine and motor drive together in mode L.the whole system meet the power balance.as is shown in formula (17).Similarly.the engine drives compressor and through the generator to generate electricity in mode C.as is shown in formula (18).The engine drives alone in mode D,as is shown in formula (19).

$$\omega_{\mathrm{y}} \cdot T_y = (\omega_f \cdot T_f + \eta_{\mathrm{em}} \cdot \omega_d \cdot T_d) \cdot \eta_{\mathrm{b}} \tag{17}$$

$$\omega_{\mathrm{y}} \cdot T_y = (\omega_f \cdot T_f + \omega_d \cdot T_d / \eta_{\mathrm{em}}) \cdot \eta_{\mathrm{b}} \tag{18}$$

$$\omega_{\mathrm{y}} \cdot T_y = \omega_f \cdot T_f \cdot \eta_{\mathrm{b}} \tag{19}$$

The engine runs on the optimum torque curve when the HPGHP system is in mode L,this moment,the compressor speed and the discharge motor torque both increase with increasing load demand and are in 8.6-8.8Nm and 2-

3.5KW,respectively.as is shown in Fig.14 and Fig.15.Similarly.When the HPGHP system is in mode C,the compressor speed and the discharge motor torque both increase with increasing load demand and are in -12--11.5NmNm and -4.5--2.5KW,respectively.as is shown in Fig.16 and Fig.17.

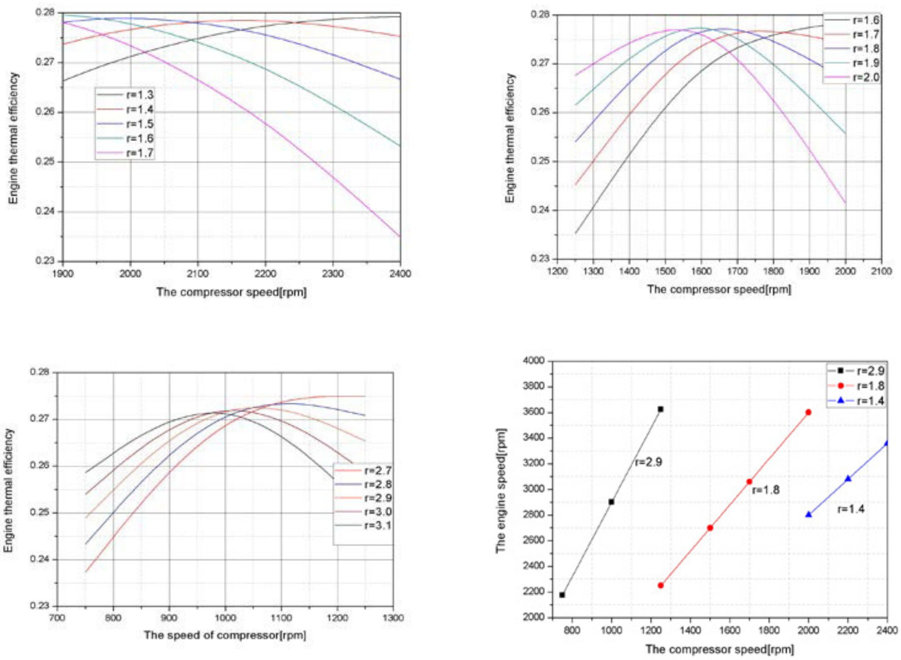

Fig. 9. The relation of the compressor speed,energy thermal efficiency and transmission ratio.

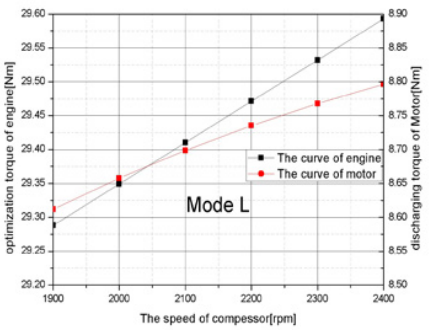

Fig.10.The relation of the compressor speed,Optimization torque of engine and the discharging torque of motor in the Mode L.

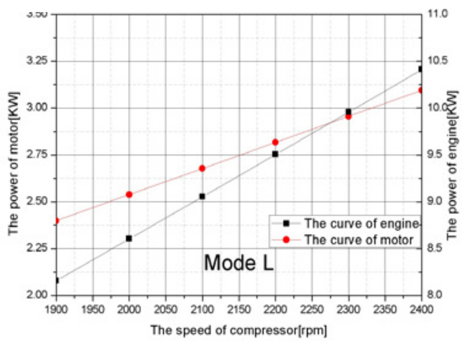

Fig.11.The relation of the compressor speed and the power of engine/motor in the Mode L.

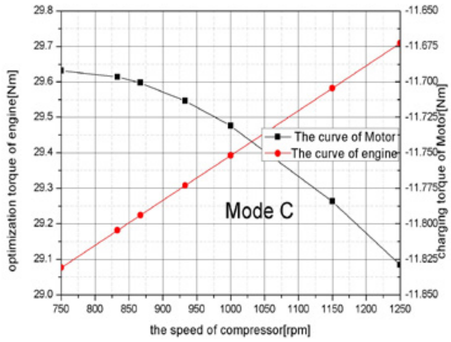

Fig.12.The relation of the compressor speed,Optimization torque of engine and the discharging torque of motor in the Mode C.

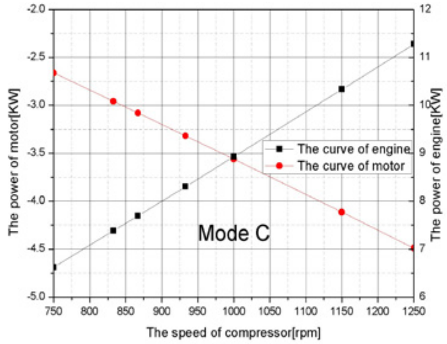

Fig.13.The relation of the compressor speed and the power of engine/motor in the Mode L.

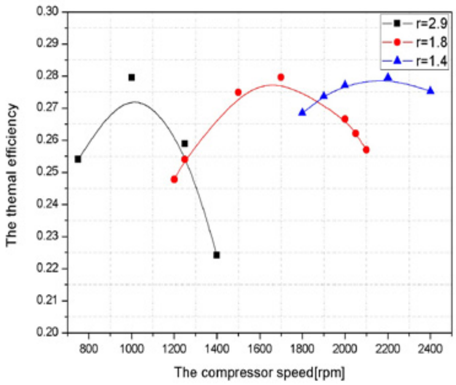

Fig.14.The relation between the thermal efficiency and the speed of compressor.

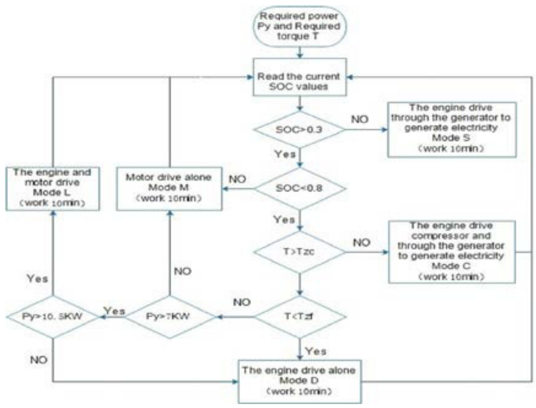

Fig.15.The principle of the optimal curve control strategy.

The control strategy of the HPGHP system is shown as Fig.15 and the thermal efficiency of engine is shown as Fig.14.the control strategy is as follow:

A new control strategy is put forward as follow: For a momentary power demand, the value of SOC should be judged firstly.If the value of SOC is less than 0.3, the HPGHP system enters into mode S.If the value of SOC is greater than 0.8, the HPGHP system enters into mode M.If the value of SOC is between 0.3 and 0.8,demand torque should be calculated firstly.if the demand torque is between the charging limits and discharging torque limits,the HPGHP system operates in mode D;if the demand torque is less than the charging torque limits,the HPGHP system operates in mode C;if the demand torque demand is higher than the discharging torque limits,and then judging the load demand.If the load demand is less than 7KW, HPGHP system operates in mode M, If the load demand is greater than 10.5KW, HPGHP system operates in mode L,otherwise HPGHP system operates in mode D.After 10mins running time in the corresponding mode,the value of SOC should be re-read, and the above steps should be repeated to select the operating mode.

5. Conclusion

1.The influence of the ratio of compressor speed and variable load demand on coefficient of performance of the HPGHP system are the same,from which we can conclude that the variable load can be meet by changing the compressor speed;cooling or heating coefficient of performance of the HPGHP system which changed with the variable speed ratio of the compressor depend on the difference of change degree index between the cooling capacity m as well as heating capacity p and the compressor input power r.

2.The switching principle of different operation modes has been established based on the mathematical model of the charging/discharging comprehensive efficiency.The simulation results show that when the compressor speed is less than 1250 rpm, the operation mode of the HPGHP system enters Mode C; when the compressor speed is between 1250 rpm and 1900 rpm,the operation mode of the HPGHP system enters Mode D;when the compressor speed is more than 1900 rpm,the operation mode enters Mode L.

The transmission ratio of the HPGHP system is optimized and mode control strategy of the coaxial HPGHP was determined.the thermal efficiency of HPGHP can be always been guaranteed above0.25 and the transmission ratios are respectively 2.9, 1.8 and 1.4 in three different load demand range.

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References

[1] Elgendy E, Schmidt J. Experimental study of gas engine driven air to water heat pump in cooling mode. Energy 2010;35:2461-2467.
[2] Huang Xueming,Analysis of Economic Operation and Thermodynamic of The Gas Heat Pump System[D].Beijing:Beijingjianzhugongchengdaxue,2011.
[3] Tyrus JM, Long RM, Kramskaya M. Hybrid electric sport utility vehicles. IEEE Trans Veh Technol 2004;53:1607-1622.
[4] Huang JY, Gonzalez LF. On parallel hybrid-electric propulsion system for unmanned aerial vehicles. Prog Aerosp Sci 2012;51:1-17.
[5] Huang KD, Tzeng SC. A new parallel-type hybrid electric-vehicle. J Appl Energy 2004;79(1):51-64.
[6] Juang YS, Lin SS, Kao HP. A knowledge management system for series-parallel availability optimization and design. J Expert Syst Appl 2008;34(1):181-193.
[7] Borba Bruno Soares MC, Szklo Alexandre, Schaeffer Roberto. Plug-in hybrid electric vehicles as a way to maximize the integration of variable renewable energy in power systems: the case of wind generation in northeastern Brazil.Energy 2012;37:469-481.
[8] Li, Y.L., Zhang, X.S., Cai, L, 2007. A novel parallel-type hybridpower gas engine-driven heat pump system. Int. J.Refrigeration 30, 1134-1142.
[9] Wang, Y., Cai, L., Yu, Y., et al., 2013. Performance study of parallel-type hybrid-power gas engine-driven heat pump system. Energy Build. 62, 37-44.
[10] Wang, J.Y., Cai, L., Wang, Y.W., et al., 2013. Modeling and optimization matching on drive system of a coaxial parallel-type hybrid-power gas engine heat pump. Energy 55,1196-1204.
[11] Guang-meng Zhou.Universal characteristics curve plotting method based on MATLAB. I. C. E &Powerplant; 2009;2:34-36.
[12]Wenxiu Jiang, Liang Cai.,et al.,Simulation and validation of a hybrid-power gas engine heat pump, International Journal of Refrigeration,50, 114-126.
[13] Qiu Lin,Wu Yuan,Study on Variation of Performance Coefficient of Gas Engine Heat Pump with Compressor Rotation Speed[J],GAS&HEAT 2006;Vol.26 NO.5:42-44.