A Study of Hybrid-Power Gas Engine-Driven Heat Pump Control Strategy Based on Instantaneous Optimization

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Abstract. The control strategy for a coaxial parallel hybrid-power gas engine-driven heat pump system is studied based on instantaneous optimization, aiming at minimizing the equivalent energy consumption. The system includes two main subsystems, drive subsystem and heat pump subsystem. Continuously variable transmission (CVT) is applied to combine the two subsystems for a continuous transmission ratio adjustment. The steady-state models of components of the two subsystems are established, and based on which equivalent energy consumption function is proposed. Meanwhile, a penalty factor is introduced into study of battery to maintain SOC in range of 0.2 to 0.8, where the internal resistance is relatively small. Then computational simulation work is performed on Simulink platform. Result shows that battery SOC runs stable between 0.49~0.66 and equivalent energy consumption is 10.4% lower than that of system with multi-stage gear ratios.

1. Introduction
Gas engine heat pump (GEHP) has advantages of high primary energy efficiency and lower operating cost than electric-driven heat pump[1] and can balance seasonal demand differences of electricity and gas. The government’s advocacy of gas use provides an opportunity for the development of GEHP. However, the engine efficiency of the conventional GEHP is inefficient under partial load condition. To solve this problem, the concept of hybrid-power gas engine-driven heat pump (HPGHP) is proposed by Air Conditioning and Refrigeration Laboratory of Southeast University[2], adding hybrid power technology into GEHP system. Control strategy is the key of hybrid power technology[3], many studies about this have been done. Based on HPGHP system, Wang[4] puts forward an instantaneous optimal control strategy and determines the optimal transmission ratio of different modes; Jiang[5] carries out a logic threshold control strategy based on the optimal engine torque and verifies that HPGHP has relatively higher thermal efficiency and lower exhaust emissions in the whole compressor speed range according to the experimental results; Meng[6] proposes an energy management control strategy based on gas engine economic zone.

All studies above are based on HPGHP with multi-stage gear ratios, which cannot satisfy the matching between gas engine speed and the continuous change of the heat pump loads. To improve this problem, continuously variable transmission (CVT) is applied in the drive subsystem of HPGHP in this paper. Then an instantaneous optimal control strategy is proposed to minimize energy consumption, taking equivalent energy consumption as the objective function.
This paper is organized as follows. The coaxial parallel HPGHP system configuration considered is briefly depicted in section 2. In section 3, the mathematical models of the system are established. Instantaneous optimal control strategy is presented with a modification in section 4. Then the control strategy is experimented to verify the effectiveness and results are revealed in section 5. Finally, conclusions are drawn in section 6. Heating condition is mainly considered.

2. Description of system
As shown in Figure 1, the HPGHP system consists of 3 main subsystems: drive subsystem, heat pump subsystem and heat recovery subsystem. CVT combines the first two subsystem and transports shaft power.

![System of the coaxial parallel-type HPGHP.](image)

Figure 1. System of the coaxial parallel-type HPGHP.

a) **Drive subsystem** mainly includes gas engine, motor and battery pack. The motor works both as generator and electric motor in different modes.

b) **Heat pump subsystem** consists of compressor, indoor and outdoor heat exchanger and expansion valve and can realize heating in winter and cooling in summer by switching four-way valve.

c) **Heat recovery subsystem** is used to collect waste heat from cylinder liner and waste gas.

There are 3 main working modes as described below.

| Mode | Load   | Working status                                      |
|------|--------|-----------------------------------------------------|
| A    | low    | Gas engine drives compressor and charges battery pack |
| B    | moderate | Gas engine drives compressor alone                  |
| C    | high   | Gas engine and motor together output shaft power    |

3. Modeling of the coaxial parallel HPGHP system with CVT
In this section, steady state Mathematical models are set up based on test data. \( \eta, n \) and \( T \) refers to efficiency, speed and torque respectively.

**Gas engine** The fitting functions of thermal efficiency, external characteristic torque and optimal torque of gas engine are obtained by methods of multiple linear regression theory and least square method respectively\([7-8]\). And curves are shown in Figure 2.

\[
\eta_e = \begin{bmatrix} 1 & n_e & T_e & n_e^2 & n_e T_e & T_e^2 \end{bmatrix} \cdot A_b
\]

\[
A_b = \begin{bmatrix} -1.4115 & 1.8403E-4 & 0.0931 & -2.968E-8 & -2.2888E-7 & -0.0015 \end{bmatrix}^T
\]
Motor The efficiency map of motor is established as Figure 3 and torque maximum and minimum of motor is depicted as equation 2 and equation 3 when charging and discharging.

\[
T_{ch-min} = \begin{cases} 
-71 & (n_m \leq 1600) \\
-2.7155 \times 10^{-6} n_m^2 + 0.0313 n_m - 113.17 & (n_m > 1600)
\end{cases}
\]

(2)

\[
T_{f-max} = \begin{cases} 
71 & (n_m \leq 1600) \\
3.1688 \times 10^{-6} n_m^2 - 0.0343 n_m + 117.46 & (n_m > 1600)
\end{cases}
\]

(3)

In Figure 2, the area where \( \eta_c \geq 0.26 \) is considered as the economic zone of gas engine and the curve where \( \eta_c = 0.26 \) is the boundary condition in this study.

Compressor The relationships of compressor shaft power, heating capacity and compressor speed are needed in this study. By experimental data we get Figure 4,5. It shows how shaft power and heating capacity increase with speed. Equation 4 is the fitting function of shaft power with speed.

\[
P_c = -1.5553 \times 10^{-7} n_c^2 + 0.0060432 n_c - 0.52821 \quad (750 \leq n_c \leq 2600)
\]

(4)

4. Instantaneous optimal control strategy

In this section, instantaneous optimal control strategy is studied, aiming at minimizing equal energy consumption (\( J_{min} \)) of the drive subsystem by optimizing the distribution of power from engine and battery.
The idea of instantaneous optimization is to make the energy consumption minimum in each time step ($\Delta t$), hence to get a total minimum consumption. So we need to discuss about consumption in a time step. Introduce a variable $u(t)$ into the definition of $T_e$ and $T_m$, which indicates the ratio of the power from engine to the compressor demand power at time $t$, $u(t) = T(t)_{e} \cdot i_{cv}(t) \cdot \eta_{cv} / T(t)_{e}$. Then, $T_e$ and $T_m$ are functions of $i_{cv}(t)$ and $u(t)$. The objective function can be expressed like

$$\min Q[u(t), i_{cv}(t)] = \Delta Q_e[u(t), i_{cv}(t)] + R(t) \cdot \Delta Q_d[u(t), i_{cv}(t)]$$

$\Delta Q_e[u(t), i_{cv}(t)] = H_e \cdot G_e[u(t), i_{cv}(t)] \cdot \Delta t = \frac{P[u(t), i_{cv}(t)]}{\eta_e[u(t), i_{cv}(t)]} \cdot \Delta t$

$\Delta Q_d[u(t), i_{cv}(t)] = \frac{I_d[u(t), i_{cv}(t)] - U_d[u(t), i_{cv}(t)]}{1000}$

Boundary conditions

$$T_{e_{min}}(n_e) \leq T_e \leq T_{e_{max}}(n_p)$$
$$T_{m_{min}}(n_p) \leq T_m \leq T_{m_{max}}(n_p)$$
$$n_{e_{min}} \leq n_e \leq n_{e_{max}}$$

$H_e$ is the low calorific value and equals 46200kJ/kg. $R(t)$ is equivalent factor. In mode A, $R_{ch}(t) = 1/\eta_{ch}(t)\eta_{e\text{chong}}(t)\eta_{e\text{chg}}$; in mode B, $R(t) = 1$; in mode C, $R_{dis}(t) = \eta_{dis}/\eta_{e}(t)\eta_{e\text{chong}}(t)$. The variable $\eta_{e\text{chong}}(t)$ can be obtained from test data chart. $\eta_{ch}$ and $\eta_{dis}$ are battery charging and discharging efficiency and equals 0.85 and 0.93 respectively.

To make sure that the battery works in range of 0.2 to 0.8, where the internal resistance of lead-acid battery is relatively small, a penalty function proposed by Paganelli[9] is used to modify electrical energy. $X(SOC) = 1 - D_{soc}^{min} + 0.2D_{soc}^{max}$.

$$D_{soc} = \begin{cases} 1 & \text{SOC > SOC}_h \\ \frac{SOC - (SOC_h + SOC_l)/2}{(SOC_h - SOC_l)/2} & \text{SOC}_l \leq SOC \leq SOC_h \\ -1 & \text{SOC < SOC}_l \end{cases}$$

$SOC_l$ is the lower limit and set to be 0.2 here; $SOC_h$ is the upper limit and set to be 0.8. In summary, the objective function is expressed as below after modification.

$$\min Q[u(t), i_{cv}(t)] = \Delta Q_e[u(t), i_{cv}(t)] + X(SOC) \cdot R(t) \cdot \Delta Q_d[u(t), i_{cv}(t)]$$

$$\max \left( \frac{T_{e_{min}} \cdot \eta_{cv}}{T_e}, 1 - \frac{1}{T_{e_{max}} \cdot \eta_{cv}} \right) \leq u \leq \min \left( \frac{T_{e_{min}} \cdot \eta_{cv}}{T_e}, 1 - \frac{1}{T_{e_{max}} \cdot \eta_{cv}} \right)$$

$$n_{e_{min}} \leq n_e \leq n_{e_{max}}$$

5. Simulation results and discussion

Computational simulation work was carried out on platform of Matlab/Simulink. Compressor speed increased linearly from 750 to 2600rpm in simulation duration, which is set to 3600s and time step is 1s. The initial value of SOC is 0.6. The simulation results are shown below.

According to the value of $u$, we can get the working range of compressor speed for each mode. As shown in Figure 6, $u$ decreases to 1 as compressor accelerates to 1390rpm, almost remains steady at 1 between compressor speed of 1390 to 1810rpm then declines again with the acceleration of speed. So between 750 to 1390rpm, mode A it is; 1390 to 1810rpm, mode B; 1810 to 2600rpm, mode C.
From Figure 7, the optimal CVT transmission ratio $i_{cvt}$ decreases from 4.1 to 1.15 continuously as compressor speed increases, while decreasing rate is getting smaller and smaller. Figure 8 displays that the optimal torque of motor and engine. We can see motor torque changes shapely in both mode A and C, while engine torque remains relatively stable at about 29.5 to 30Nm. This shows how HPGHP system works, using a sharply changing motor torque to keep gas engine working in high efficiency zone.

As we can see in Figure 9, due to the effect of the penalty function, SOC remains in the setting range of 0.2–0.8 throughout the simulation duration. It increases firstly to 0.66 then keeps still when compressor speed runs from 1390rpm to 1810rpm, and decreases to 0.49 at last.

To prove the effectiveness of the proposed control strategy, the equivalent energy consumption in this study is compared with that of the HPGHP studied by Wang[4], which is a HPGHP system with multi-stage gear ratios (Mode A: $i=3.2$; Mode B: $i=2$; Mode C: $i=1.5$). In Figure 10, S1 refers to the previous study and S2 refers to the HPGHP with CVT. In the same simulation condition, S2 has a lower equivalent energy consumption of about 10.4% than S1.
6. Conclusion

1) CVT is introduced into hybrid-power gas engine-driven heat pump system and an instantaneous optimal control strategy is proposed based on that. In the study of control strategy, the penalty function is introduced to maintain SOC in range of 0.2~0.8 and result shows it’s effective.

2) Based on the instantaneous optimal control strategy, switching law is established. The results show that the HPGHP system works in Mode A when compressor speed is less than 1390rpm; system works in Mode B with compressor speed between 1390 to 1810rpm; system works in Mode C when compressor speed is beyond 1810 rpm.

3) Comparing with the HPGHP with multi-stage gear ratios, the HPGHP with CVT has a better energy saving performance under instantaneous optimal control strategy. In the same simulation condition above, the HPGHP system with CVT has a lower equivalent energy consumption of about 10.4% than system with multi-stage gear ratios.

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List of symbols:

- GEHP --- gas engine heat pump
- HPGHP --- hybrid-power gas engine-driven heat pump
- CVT --- continuously variable transmission
- SOC --- state of charge of battery pack
- $T_{f,\text{max}}$ --- motor maximum discharging torque(Nm)
- $T_{ch,\text{max}}$ --- motor maximum charging torque(Nm)
- $G_e$ --- gas consumption rate(kgs-1)
- $P_c$ --- compressor shaft power(kW)
- $U_b$ --- battery voltage(V)
- $H_l$ --- low calorific value(kJkg-1)
- $I_b$ --- battery current(A)
- $\eta_g$ --- gas engine thermal efficiency
- $\eta_{ch}$ --- battery charging efficiency
- $\eta_{dis}$ --- battery discharging efficiency
- $\eta_{cvt}$ --- transmission efficiency of CVT
- $\eta_{cv}$ --- transmission efficiency of CVT
- $Q$ --- equivalent energy consumption (kJ)

Subscripts:
e --- engine  m --- motor  c --- compressor
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