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Hybrid electric vehicle absorption-compression refrigeration system

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Abstract. The air-conditioning system of electric vehicles shortens the driving mileage of the car. Therefore, a set of absorption-compression hybrid refrigeration system for hybrid electric vehicle are designed. The system consists of power system, absorption refrigeration cycle and compression refrigeration cycle. By using the waste heat of micro gas turbine to drive the absorption refrigeration subsystem, the power consumption of the compressor driven by battery power is reduced, as well. The feasibility of the system is discussed through the thermodynamic cycle calculation. The results show that, a bus air-conditioning with cooling load of 30 kW, the waste heat emitted by a micro gas turbine with a power of 35 kW drives an absorption refrigeration cycle, which can provide about 15 kW of cooling capacity, and the remaining part needs to be compensated by the compression refrigeration cycle, which consumes about 6.71 kW of compressor power. At the same time, compared with the absorption-compression combined refrigeration and electric compression refrigeration, the energy efficiency is improved by about 37.1 %.

1. Introduction
The energy crisis and increasingly serious environmental pollution have long been a global problem. Under the dual pressures of current social energy and the environment, traditional fuel vehicles will inevitably face elimination and replacement because of their high energy consumption and serious exhaust pollution. The new energy vehicles such as electric vehicles will undoubtedly become the direction of future automobile development. Hybrid electric vehicle (HEV) is a vehicle equipped with two or more power sources[1]. It not only has the high efficiency and low emission performance of the pure electric vehicle, but also has the advantages of long mileage and fast refueling performance of traditional internal combustion engine cars. The storage elements and the power generation elements are combined in different ways. Through the optimization, complementation and coordination of different energy sources, the system can achieve low emissions and improve system fuel economy. As an important means to improve the working conditions of the driver, improve the efficiency and the safety of the car and create a healthy and comfortable environment for the crew, the automobile air-conditioning system is essential for the fuel vehicle and the electric vehicle. Compared with ordinary automobile air-conditioning, electric vehicle air-conditioning system is driven independently by the electric motor without engine drive. As the largest auxiliary subsystem of the power consumption of the electric vehicle, the use of air-conditioning system will greatly reduce the driving mileage of the car.

The drive system of hybrid electric vehicle based on micro-turbine generator (MTG) consists of micro gas turbine (MGT), high speed generator, energy storage device (such as storage battery),
motor and control device, etc. As a new generation of power generation equipment, MTG set has few moving parts, simple structure, good reliability, low maintenance cost, and much less emission than traditional motor vehicle engine. It is very suitable for the series hybrid electric vehicle engine. MTG and battery are connected in series to form a compound power, and the two advantages complement each other. Through the control of the controller, MTG and battery work in coordination to make MTG set work in a high thermal efficiency and low emission area. As the auxiliary power source of the automobile, MGT can generate stable flue gas in operation. If the waste heat is fully utilized, it can not only improve the efficiency of MTG but also meet the passengers comfort requirements through the operation of the refrigeration and heat pump system. High temperature flue gas emitted by MGT, and this part of the waste heat can be used to drive absorption refrigeration system. However, the load demand characteristics of the automobile air-conditioning are not consistent with the operating characteristics of the car. The MTG is not always working as the auxiliary power source of the car, so the use of absorption refrigeration cycle alone can not meet the timely demand of vehicle load regulation. At the same time, the air-conditioning system cannot take up too much space due to the limited space of the car. Therefore, it is of practical significance to develop a new, efficient and compact air-conditioning system for hybrid electric vehicles based on the operating characteristics of hybrid electric vehicles, making full use of waste heat sources, and reducing the power consumption of air-conditioning systems.

2. Design requirements for automotive air-conditioning
For the design of vehicle air-conditioning using waste heat, the following requirements must be met:
• Automotive air-conditioning must be miniaturized and light-weighted, and the equipment in the system can be separately installed to facilitate installation on vehicles with limited space.
• The use of water cooling will increase the number of equipment in the system, and the system will become more complicated, which is not conducive to the actual installation and application. Therefore, the air-conditioning system for vehicles should be directly air-cooled.
• Automotive air-conditioning must be able to work normally under conditions such as up and down bumps, vertical and horizontal tilt or jolt, and rapid acceleration and deceleration.
• The working medium used in the automotive air-conditioning is as nontoxic, nonflammable, low working pressure as possible.

According to the above design requirements, the refrigerant working pair and system form of the air-conditioning system are determined.

3. Absorption–compression hybrid refrigeration cycle
3.1. Working principle of air-conditioning system
Figure 1 is an absorption-compression hybrid refrigeration system. The system consists of three parts: power system, absorption refrigeration cycle and compression refrigeration cycle. The power system consists of a power battery, a controller, a MGT and a generator. The power battery is used as the main driving force, and the MTG is used as the auxiliary power. When the MGT works, the flue gas from the regenerator enters the vapor generator and drives the absorption subsystem to refrigerate. When the MGT stops working, the power battery supplies power to the motor compressor, which drives the compressor to run and realizes compression refrigeration.

The working process of the refrigeration cycle is as follows: after the working solution in the heat recovery vapor generator absorbs the waste heat from the MGT, the refrigerant in the solution is partially vaporized, and the working medium leaves the vapor generator in the form of gas-liquid two phases. The refrigerant steam separated by the gas-liquid separator enters the air-cooled condenser to condense into a liquid, which is then decompressed by the expansion valve and then flows to the evaporator to evaporate to generate refrigeration. The refrigerant steam from the evaporator enters the absorber, then it is absorbed by the weak solution which is from the vapor generator and cooled by the
heat exchanger. The rich solution after absorbing the refrigerant steam is pressurized through the solution pump and heated by the heat exchanger into the generator.

When the cooling capacity of the absorption refrigeration system can not meet the cooling requirement of vehicles, the compression refrigeration system begins to work, and the hybrid refrigeration system meets the cooling demand by jointly operating the absorption and compression refrigeration cycle.

![Figure 1. The working process of the absorption-compression hybrid refrigeration system](image)

### 3.2. Working pair selection

Absorption refrigeration is the use of heat exchange between the heat source and the generator, the working pair is heated in the generator, and the low-boiling point refrigerant vaporizes out of the solution. After the cycle, the cooling capacity is generated. The working pair should be selected according to the following characteristics[2]:

- The difference between the boiling point of the refrigerant and the absorbent is quite large.
- In a certain temperature range, the absorbent and the refrigerant have good intersolubility, and the saturation concentration of the refrigerant in the absorbent varies greatly with the change of temperature.
- The mixed solution has stable chemical properties, no decomposition at high temperature, no poison and no explosion.
- The mixed solution has high heat transfer coefficient, low viscosity, and high thermal diffusivity.

For absorption-compression hybrid refrigeration system, refrigerant must also be used to absorb refrigeration cycle and compression refrigeration cycle simultaneously. At present, relatively mature technology is widely used in industry, mainly for LiBr-H2O and NH3-H2O. LiBr and H2O have large differences in boiling points, are chemically stable, and have strong hygroscopicity, which are widely used as a working pair. However, due to the relatively small per-unit refrigerating capacity of swept volume of water, the corresponding refrigerating apparatus has a large volume, and therefore does not
suitable as the working pair of the absorption of automotive air-conditioning. Due to the toxicity of NH₃, the ammonia solution has a corrosion on the copper tube, and the unfavorable factors such as the explosion of the fire in case of an open flame, the safety hazards in the waste heat refrigeration system for vehicles are high.

Based on the above reasons, and taking into some factors like non-toxic, non-corrosive, non-flammable, moderate working pressure, easy to use air-cooled, and can be used for both absorption and compression refrigeration cycle and so on, choosing HCFCS or HFCS[3] (eg R22, R124, R134a, etc.) as refrigerant and choosing chemical solvent (eg DMAC, DMF, DMEFEGD, etc.) as absorbent. Through comprehensive analysis, it is considered that the selection of R124-DMAC (2-chloro-l,l,l,2-tetrafluoroethane/N',N'-dimethylacetamide) under air-cooled conditions is more suitable. Borde et al.[4] tested the basic properties of R124 and DMAC, and the specific parameters are shown in Table 1.

| Medium | Molecular formula | Molecular weight (g/mol) | Density (kg/m³) | Specific heat (kJ/kg°C) |
|--------|------------------|--------------------------|-----------------|------------------------|
| R124   | C₂H₅F₄Cl        | 136.50                   | 564.79          | 1.13                   |
| DMAC   | C₄H₉NO          | 87.12                    | 942.80          | 1.92                   |

| Medium | Boiling point (℃) | Evaporation latent heat (kJ/kg) | Critical temperature (℃) | Critical pressure (MPa) |
|--------|------------------|--------------------------------|--------------------------|------------------------|
| R124   | -11.0            | 194.0                          | 122.2                    | 3.574                  |
| DMAC   | 165.0            | 519.16                         | 382.4                    | 4.211                  |

3.3. Advantages of absorption-compression hybrid refrigeration air-conditioning

The absorption refrigeration system undertakes the partial cooling load of the automobile, reduces the refrigerating capacity required for the electric compression air conditioner, cuts down the power consumption of the automobile air-conditioning system, and solves the problem of shortening the driving range of the vehicle due to the power consumption of the air-conditioning system operation. The waste heat from the MGT is used to drive the absorption chiller and heating air, effectively improving the efficiency of waste heat utilization, realizing the full utilization of waste heat, and saving energy to improve the overall energy efficiency of the system.

The air-cooled condenser is adopted, compared with the conventional absorption chiller, since there is no cooling tower, it reduces the occupying volume of the air-conditioning and realizes the compact arrangement of the air-conditioning in the limited space of the automobile.

4. Thermodynamic cycle calculation

4.1. Absorption refrigeration subsystem

4.1.1. Design parameters of absorption refrigeration cycle. The vehicle is selected as a bus, the air-conditioning cooling load was 30 kW, and the MTG was used as an auxiliary power source with a power of 35 kW. The absorption refrigeration sub-cycle design parameters are listed in Table 2.

| Item                              | Value |
|-----------------------------------|-------|
| Inlet temperature of cooling air (℃) | 35    |
| Outlet temperature of cooling air (℃) | 45    |
| Evaporation temperature (℃)       | 3     |
| Condensation temperature (℃)      | 55    |
| Solution inlet temperature of absorber (℃) | 55    |
| Solution outlet temperature of absorber (℃) | 50    |
4.1.2. Thermodynamic calculation process of absorption refrigeration

- **Generator**

\[ q_g = h_1^v + f (h_{11}^l - h_{10}^l) - h_{11}^l \]  

(1)

Among them, \( f \) is the circulation ratio of the solution, kg/kg; \( h_1^v \) is the specific enthalpy of the refrigerant vapor at the outlet of the vapor generator, kJ/kg; \( h_{11}^l \) is the specific enthalpy of the rich solution into the vapor generator; \( h_{11}^l \) is the specific enthalpy of the weak solution at the outlet of the vapor generator.

- **Absorber**

\[ q_a = h_6^v + f (h_7^l - h_8^l) - h_7^l \]  

(2)

Wherein, \( h_6^v \) is the specific enthalpy of the refrigerant vapor at the inlet of the absorber; \( h_7^l \) is the specific enthalpy of the weak solution into the absorber; \( h_8^l \) is the specific enthalpy of the rich solution out of the absorber.

- **Condenser**

\[ q_{e, abs} = h_1^v - h_{41}^l \]  

(3)

Here \( h_{41}^l \) is the specific enthalpy of the refrigerant liquid at the outlet of the condenser.

- **Evaporator**

\[ q_{e, abs} = h_6^v - h_{46}^l \]  

(4)

In the formula, \( h_6^v \) is the specific enthalpy of the refrigerant vapor at the outlet of the evaporator.

- **Solution pump**

\[ w_{pump} = \frac{f \Delta p}{1000 \rho \eta_{pump}} \]  

(5)

Where \( w_{pump} \) is the power consumption of the unit refrigerant solution pump, kJ/kg; \( \Delta p \) is the pressure difference between the inlet and outlet of the solution pump, pa; \( \rho \) is the density of the rich solution through the solution pump, kg/m; \( \eta_{pump} \) is the efficiency of the solution pump.

- **Coefficient of performance**

\[ cop_{abs} = \frac{q_{e, abs}}{q_g + w_{pump}} \]  

(6)

4.1.3. Calculation results of absorption refrigeration cycle

Under the design condition, the gas-emission scope of the absorption refrigeration subsystem is 0.09. The cooling air flows through the absorber and then passes through the condenser, and the total temperature rises to 10 °C. In the design calculation, the flow resistance of the refrigerant steam between the condenser, the generator and the compressor outlet is ignored, and the pressure of the generator is equal to that of the condenser; the working pressure of the absorber is 10 kPa lower than
the evaporation pressure, and the heat loss of the hybrid refrigeration system is neglected. The calculation results are shown in Table 3 below:

### Table 3. Thermodynamic calculation results of absorption refrigeration cycle under design conditions

| Equipment         | Status point | Temperature (℃) | Pressure (MPa) | Mass fraction | Mass flow rate (kg/s) | Load (kW) |
|-------------------|--------------|-----------------|----------------|---------------|-----------------------|-----------|
| Absorber          | 8            | 50              | 0.1665         | 0.555         | 0.8803                | 42.45     |
|                   | 7            | 55              | 0.1665         | 0.465         | 0.7322                |           |
|                   | 6            | 8               | 0.1765         | 1             | 0.2897                |           |
| Vapor generator   | 10           | 130             | 0.8641         | 0.555         | 0.8803                |           |
|                   | 11           | 147             | 0.8641         | 0.465         | 0.7322                | 58.68     |
|                   | 1            | 147             | 0.8641         | 1             | 0.2897                |           |
| Heat exchanger    | 11           | 147             | 0.8641         | 0.465         | 0.7322                | 153.35    |
|                   | 12           | 55              | 0.8641         | 0.555         | 0.8803                |           |
|                   | 9            | 50              | 0.8641         | 0.555         | 0.8803                |           |
|                   | 10           | 130             | 0.8641         | 0.555         | 0.8803                |           |
| Condenser         | 2            | 147             | 0.8641         | 1             | 0.2897                | 46.23     |
|                   | 4            | 55              | 0.8641         | 1             | 0.2897                |           |
| Evaporator        | 5            | 3               | 0.1765         | 1             | 0.2897                | 30        |
|                   | 6            | 8               | 0.1765         | 1             | 0.2897                |           |
| Solution pump     | 8            | 50              | 0.1665         | 0.555         | 0.8803                | 0.92      |
|                   | 9            | 50              | 0.8641         | 0.555         | 0.8803                |           |

As can be seen from Table 3 above, the $\text{cop}_{\text{abs}}$ of absorption refrigeration is 0.511 under design conditions. According to the following formula, it is possible to calculate the 35 kW output power of the MTG which can provide about 15 kW cooling capacity, and the remaining cooling load needs to be compensated by the compressor.

$$Q_0 = Q_g \cdot \text{cop}_{\text{abs}}$$  \hspace{1cm} (7)

### 4.2. Compression refrigeration subsystem

#### 4.2.1. Design parameters of compression refrigeration cycle

The compression refrigeration sub-cycle design parameters are listed in Table 4.

### Table 4. Compressed refrigeration sub-cycle design parameters

| Item                        | Value |
|-----------------------------|-------|
| Cooling load(kW)            | 15    |
| Inlet temperature of cooling air(℃) | 35    |
| Outlet temperature of cooling air(℃) | 45    |
| Outlet temperature of cooling air(℃) | 3     |
| Condensation temperature(℃) | 55    |
| Superheat degree of vapour(℃) | 5     |
| Supercooling degree of liquid(℃) | 3     |
| Isentropic efficiency(%)    | 70    |

#### 4.2.2. Thermodynamic calculation process of compression refrigeration

- **Condenser**

$$q_{e,\text{comp}} = h_3^v - h_4^l$$  \hspace{1cm} (8)
Where \( h \) is the specific enthalpy of refrigerant vapor at the compressor outlet, kJ/kg

- Evaporator
  
  \[ q_{e,comp} = h_6^v - h_4^l \]  
  \[ (9) \]

- Compressor specific work
  
  \[ w_{comp} = h_5^v - h_6^v = \frac{h_6^v - h_6^l}{\eta_{comp}} \]  
  \[ (10) \]

Here \( \eta_{comp} \) is the Isentropic Efficiency of the refrigeration compressor.

- Coefficient of performance
  
  \[ cop_{comp} = \frac{q_{e,comp}}{w_{comp}} \]  
  \[ (11) \]

### 4.2.3. Calculation results of compression refrigeration cycle

According to the design parameters in Table 4 above, the design and calculation results can be obtained as follows:

Table 5. Thermodynamic calculation results of compression refrigeration cycle under design conditions

| Equipment | Status point | Parameter | Temperature (℃) | Pressure (MPa) | specific enthalpy (kJ/kg) | Mass flow rate (kg/s) | Load (kW) |
|-----------|--------------|-----------|-----------------|----------------|--------------------------|----------------------|-----------|
| Condenser | 2 | 68 | 0.8641 | 402.43 | 0.1412 | 20.20 |
| | 4 | 52 | 0.8641 | 259.38 | 0.1412 |
| Evaporator | 5 | 3 | 0.1765 | 259.38 | 0.1412 | 15 |
| | 6 | 8 | 0.1765 | 365.64 | 0.1412 |
| Compressor | 3 | 68 | 0.8641 | 402.43 | 0.1412 | 5.19 |
| | 3s | 56 | 0.8641 | 391.85 | 0.1412 |

\[ cop_{com} = 2.89 \]

The calculation of the coefficient of performance of the compression refrigeration in Table 5 only considers the isentropic efficiency of the compressor. When the actual automobile air conditioner compressor is converted to consume engine power, the mechanical efficiency of the compressor and the motor efficiency must also be considered. Therefore, the coefficient of performance value of the actual automotive air-conditioning system is lower than the calculated value of Table 5. If the mechanical efficiency \( \eta_m \) is taken as 0.9 and the motor efficiency \( \eta_{mot} \) is 0.86, according to the formula (12), the input power of the compressor \( W_{input} \) is 6.71 kW, which means that when the compressor is used to compensate the absorption refrigeration, an additional power of 6.71 kW is required.

\[ W_{input} = \frac{W_{comp}}{\eta_m \eta_{mot}} \]  
  \[ (12) \]

### 4.3. Energy efficiency improvement analysis

#### 4.3.1. Calculation of electric compression refrigeration

If the cold load of 30 kW is cooled fully by compression refrigeration, and the value of \( cop_{com} \) is 2.89, then the power of the compressor is calculated to be about 10.38 kW according to the formula (13). The mechanical efficiency \( \eta_m \) is taken as 0.9 in section 4.2.3 and the motor efficiency \( \eta_{mot} \) is 0.86, according to formula (12), the input power required for the compressor is 13.41 kW.
4.3.2. Calculation of absorption-compression combined refrigeration cycle. If the absorption refrigeration with exhaust waste heat recovery is used to take a part of the cooling load, according to 4.1.3 section, the micro gas turbine waste heat driven absorption chiller can provide about 15 kW cooling load, and the remaining 15 kW cooling load should be supplied by the compressor, the power consumption of the compressor is approximately 6.71 kW. The power consumption of the solution pump can be calculated according to formula (14), in the equation, \( q_m \) is the mass flow rate of the solution; \( f \) is the solution circulation ratio, \( \Delta p \) is the pressure difference between the inlet and outlet of the solution pump; for the density of \( \rho \) the specific value can be obtained in table 3. \( \eta_{pump} \) is the efficiency of the solution pump, the general value is 0.8. It can be calculated that the power needed to supply the solution pump is approximately 0.92 kW.

\[
W_{pump} = \frac{q_m \cdot f \cdot \Delta p}{1000 \cdot \rho \cdot \eta_{pump}}
\]  
(14)

The heat dissipation of the absorber is large, and the power of the exhaust fan is 800 W. Therefore, for absorption refrigeration, the total power consumption can be calculated by the formula (15), where \( W \) is the power consumption of the compressor in the absorption-compression combined refrigeration cycle. After calculation, the absorption-compression composite refrigeration system consumes a total of 8.43 kW of power.

\[
W_{total} = W_{comp,p} + W_{pump} + W_{fan}
\]  
(15)

Compared with the compression refrigeration and the absorption-compression combined refrigeration, the energy efficiency improvement can be calculated by the formula (16), and the energy efficiency can be improved by about 37.1 %.

\[
\xi = \frac{W_{input} - W_{total}}{W_{input}}
\]  
(16)

5. Conclusion
A set of absorption-compression hybrid refrigeration air-conditioner is designed for hybrid electric vehicles, using the waste heat of the micro gas turbine exhaust gases to drive the absorption refrigeration system, reduce the cooling load that the battery power needs to drive the compression. Recycle and utilization of waste heat reduces electricity consumption of the automobile air-conditioning system and prolongs driving range of vehicle.

For a bus air-conditioning with cooling load of 30 kW, the waste heat emitted by a micro gas turbine with a power of 35 kW drives an absorption refrigeration cycle that cannot meet the requirements of the automotive air-conditioning. It needs to take an air-conditioning cooling load of about 15 kW by compression refrigeration cycle, using compression refrigeration to compensate absorption refrigeration, and an additional power of 6.71 kW should be provided.

Compared with absorption-compression combined refrigeration and electric compression refrigeration, energy is saved and the energy efficiency is improved by about 37.1 %.

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