Integration of an ammonia-water absorption refrigeration system with a marine Diesel engine: A thermodynamic study

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Abstract
This paper examines through a thermodynamic analysis the feasibility of using waste heat from marine Diesel engines to drive an ammonia-water absorption refrigeration system. An energy balance of a diesel engine shows that sufficient waste heat is provided. The results illustrate that higher performance of the system is obtained at high generator and evaporator temperatures and also at low condenser and absorber temperatures.

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1. Introduction
As fuel prices soar, marine diesel engines manufacturers are beginning to scout for more energy-efficiency solutions. This can be achieved by increasing the engine efficiency and developing technologies with low pollutant emissions. The increase of the diesel engine efficiency can be achieved by valorizing the waste thermal energy. Onboard ships, a considerable amount of primary energy may be saved by valuing the waste heat rejected during the operation of the main propulsion systems.

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Indeed, a large amount of waste heat energy is released into the environment. Therefore, an increased attention is being paid for the utilization of this waste energy. This valorization can be intended using absorption refrigeration systems in which the surplus waste heat from the main engine is used to separate a refrigerant vapor from a binary solution.

Traditionally, vapour compression systems are the common onboard ships refrigeration systems. These systems are powered using electric energy produced through burning of increasingly more expensive fuel. The challenges of environment protection and energy economy have led to a growing interest in non-conventional refrigeration systems such as absorption refrigeration. In fact, in an absorption refrigeration system, the refrigeration effect is produced through the use of two fluids and a heat source rather than electrical input as in the more familiar vapor compression system. Therefore, energy recovered from waste heat streams can provide a considerable part of refrigeration and air conditioning needs at no additional cost. As a result, the use of surplus heat rejected by the main propulsion engine provides substantial fuel savings, involving the reduction of pollutants. The performance of this system affects the attainable amount of energy savings. Improving their performance can be only performed using thermodynamic analysis.

While several investigations have been devoted to absorption refrigeration systems [1-18], only few past studies have addressed the concept of driving absorption refrigeration systems by waste heat from internal combustion engines [19-27].

It is clear from the literature survey that absorption refrigeration systems have received a considerable interest among the research society during the last years and a substantial amount of work has been published on the subject. Nevertheless, in the above mentioned work, there seems to be only few attempts to integrate absorption refrigeration systems with marine diesel engines.

Hence, following the above mentioned studies, this paper presents a thermodynamic analysis of an absorption refrigeration system driven by waste heat from a Diesel engine. The calculation and analysis of an absorption refrigeration system require the availability of simple and reliable mathematical model for the determination of thermodynamic properties of the working fluids. Combined to fundamental thermodynamic relations, it can generate all the properties required to carry out a thermodynamic analysis of the cycle.
2. Marine Diesel engine energy balance

Although the great technological development of modern marine Diesel engines, only a part of the energy contained in the fuel is converted to power output. The maximum efficiency remains lower than 45%. The main losses are dissipated as heat in the exhaust gases, coolants, and transferred to the environment.

An energy balance of a marine Diesel engine indicates how the energy contained in the fuel is used or lost. Indeed, the injection of a mass of fuel in the hot air in the combustion chamber produces a large amount of heat. However, the mechanical work requires only a fraction of the energy produced. The residual energy is, in fact, discharged at various places during his stay in the cylinder. For a marine Diesel engine, a first-law analysis yields

\[
\dot{Q}_s = \dot{W} + \dot{Q}_c + \dot{Q}_r + \dot{Q}_e
\]

where, \( \dot{Q}_s \) is the heat supplied to the Diesel engine, \( \dot{W} \) the mechanical energy output, \( \dot{Q}_c \) the heat transferred to the cooling systems, \( \dot{Q}_r \) the heat rejected in exhaust gases, \( \dot{Q}_e \) the heat transferred to the environment by radiation.

Figure 1 shows an example of an energy balance of a modern marine Diesel engine. The heat transferred to coolants includes charge air cooling (17.8%), jacket water cooling (4.8%) and lubricating oil cooling (3.2%). In addition, 25.1% of the total energy is lost, released into the atmosphere during the exhaust outlet. Finally, a small part of the energy (0.6%) is lost by radiation. At first glance, the analysis of heat flows shows that there are three engine waste heat streams, at different temperature levels, that have potential to be recovered: exhaust gas (300-600°C); charge air (200°C); jacket water (80-100°C).

3. Thermodynamic analysis

In order to carry out a thermodynamic analysis, the conservation laws of mass and energy have been applied to each component of the system. Every component has been considered as a control volume exchanging heat, work and inflow and outflow streams with its surrounding. To simplify the theoretical model, the following assumptions have been considered:

- The analysis is carried out under steady state conditions and assuming thermodynamic equilibrium at all points of the cycle;
- Ammonia at the generator and evaporator outlets is assumed as saturated vapour;
Ammonia at the condenser outlet is saturated liquid;
- Pressure losses in the pipes and all heat exchangers are negligible;
- Heat exchange release to surroundings does not occur.

The absorption system considered uses ammonia as the refrigerant and water as the absorber to create a continuous cycle for the ammonia. Figure 2 shows a schematic of a basic ammonia-water absorption refrigeration system. It consists mainly of a generator, a condenser, an evaporator, an absorber, a solution heat exchanger and a circulation pump. The generator uses waste heat from the marine Diesel engine to separate ammonia vapour from the concentrated ammonia solution. In the condenser, high pressure ammonia vapours are cooled and condensed to liquid state. Liquid ammonia leaves the condenser and flows to the evaporator through an expansion valve. The refrigerant then enters the evaporator, where it receives heat from the cold source. Then, ammonia vapour enters the absorber, where a weak solution of water and low concentration ammonia absorbs the refrigerant and, at the same time, releases heat to the neighbourhood. The ammonia-water solution flows back to the generator through a circulation pump to undergo a new cycle.

Based on the above assumptions, the governing equations for mass and energy conservation are given by the following expressions:

\[ \sum \dot{m}_i - \sum \dot{m}_o = 0 \]  
\[ \sum \dot{m}_i x_i - \sum \dot{m}_o x_o = 0 \]  
\[ \dot{Q} - W = \sum \dot{m}_i h_i - \sum \dot{m}_o h_o \]  
where \( x_i \) and \( x_o \) correspond to the inlet and outlet ammonia mass fractions in the solution.

The application of the mass and energy balance equations to the generator yields

\[ \dot{m}_s = \dot{m}_i + \dot{m}_h \]  
\[ \dot{m}_{s,x} = \dot{m}_i + \dot{m}_h x \]  
\[ \dot{Q} = \dot{m}_i h_i + \dot{m}_h h + \dot{m}_s h_s \]  
From equations (5) and (6), mass flow rates of strong and weak solutions can be calculated:

\[ \dot{m}_s = \frac{1 - x_s}{x_s - x_i} \dot{m}_i \]  
\[ \dot{m}_{s,x} = \frac{1 - x_{s,x}}{x_s - x_i} \dot{m}_i \]  
Equation (9) allows to define the circulation ratio of the system:

\[ f = \frac{\dot{m}_{s,x}}{\dot{m}_i} \]
The energy balance of the solution heat exchanger yields

\[ T_i = εT_s + (1 - ε)T_s \]  

\[ h_i = h_s + \frac{\dot{m}_i}{\dot{m}_s} (h_s - h_s) \]

where, \( ε \) is the solution heat exchanger effectiveness.

The energy required to pump the solution from the low pressure (absorber) to the high pressure (generator) is:

\[ w' = (p_s - p_i)\nu \]

where \( \nu \) is the specific volume of the strong solution.

Then, enthalpy at point 6 can be calculated:

\[ h_i = h_i + (p_s - p_i)\nu \]

Finally, the mass and energy balance equations applied to the absorber, the condenser and the evaporator give the following expressions:

\[ Q_a = \dot{m}_a h_a + \dot{m}_w h_w - \dot{m}_h h_h \]  

\[ Q_c = \dot{m}_c (h_c - h_i) \]  

\[ Q_e = \dot{m}_e (h_i - h_e) \]

The performance of refrigeration systems are usually measured using the coefficient of performance (COP). This parameter is defined as the ratio of the useful effect produced to the energy input of the system:

\[ COP = \frac{\dot{Q}}{\dot{Q} + W'} \]

4. Results and discussions

The analysis involves the determination of effects of generator, absorber and evaporator temperatures on the performance of the system. The performance parameters considered are coefficient of performance and the circulation ratio.

A computer program has been prepared in order to calculate the performance of the ammonia-water absorption refrigeration cycle accordingly to the model presented in the above section. The program uses subroutines allowing the calculation of thermodynamic properties of the binary solution as function of temperature and concentration [5]. Fixed data used in the simulation are summarized in Table 1. The operating condenser and absorber temperatures are 20 to 40°C and depend on the cooling water conditions. The solution heat exchanger effectiveness is defined as the ratio of the temperature drop of strong solution to the temperature difference of the strong and weak solutions entering the heat exchanger.

Table 1. Fixed data use in the simulation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Generator temperature, °C</td>
<td>60-120</td>
</tr>
<tr>
<td>Condenser temperature, °C</td>
<td>20-45</td>
</tr>
<tr>
<td>Absorber temperature, °C</td>
<td>20-45</td>
</tr>
<tr>
<td>Evaporator temperature, °C</td>
<td>-10-10</td>
</tr>
<tr>
<td>Solution heat exchanger effectiveness, %</td>
<td>70 - 100</td>
</tr>
<tr>
<td>Solution mass flow rate, kg/min</td>
<td>3.5</td>
</tr>
<tr>
<td>Ammonia mass flow rate, kg/min</td>
<td>1</td>
</tr>
</tbody>
</table>
Figure 3 shows the combined influence of generator and condenser temperatures on the COP and the circulation ratio of the cycle for absorber and evaporator temperatures fixed to 25 and -5°C, respectively. The effectiveness of the solution exchanger was set to 80%. Generally, the coefficient of performance increases with increasing the generator temperature as shown in Fig. 3. In addition, the increase in the condenser temperature results in deterioration in the COP. It is noted that for every condensation temperature, there is a lower limit of the generator temperature below which the cycle cannot be achieved. It is interesting to have this limit as low as possible if the waste heat source used is the engine jacket cooling water. The latter has a temperature ranging from 80 to 100°C. The circulation ratio increases considerably for generator temperatures approaching their lower limits (Fig. 3). Higher values of the circulation ratio require unacceptable dimensions of solution pumps. This explains the impossibility of achieving a cycle at very low generator temperatures. Increasing the condenser temperature in turn increases the circulation ratio.

Figure 4 shows the variation of the COP and the circulation ratio according to the generator temperature for various temperatures of the absorber. The fact that temperatures prevailing in the absorber and the condenser are in the same level, the effect of the absorber temperature is similar to that of the condensation temperature on the COP and the circulation ratio.

The evaporator temperature is directly related to the temperature of the product to be cooled. The main advantage of using ammonia-water mixture is to reach negative cooling temperatures. Fig. 5 shows the combined effect of generator and evaporator temperatures on the coefficient of performance and the circulation ratio. These curves have been obtained for fixed condenser and absorber temperatures and
solution heat exchanger effectiveness. For each evaporator temperature, the increase of the generator temperature involves an increase in the COP (Fig. 5.a). It is clear that increasing the evaporation temperature improves significantly the COP. It is also noted here that for each evaporation temperature, there is a lower limit of the generator temperature below which the cycle is impractical.

Similarly, the circulation ratio increases considerably for generator temperatures approaching their lower limits as shown in Fig. 5.b. The decrease in the evaporation temperature in turn increases the circulation ratio for generator temperatures lower than 95°C. Beyond this value, the effect generator and evaporator temperatures on the circulation ratio is negligible.

For generator, condenser, absorber and evaporator temperatures fixed to 90, 25, 25 and -5°C respectively, the cycle performance have been calculated for various effectiveness of the solution heat exchanger (Table 2). The solution heat exchanger effectiveness is defined as the ratio of the temperature drop of strong solution to the temperature difference of the strong and weak solutions entering the heat exchanger. The increase of the effectiveness improves significantly the coefficient of performance. However, it has no effect on the circulation ratio of the cycle.

**Table 2.** Effect of the solution heat exchanger efficiency on the performance of the system

<table>
<thead>
<tr>
<th>$\varepsilon$</th>
<th>COP</th>
<th>$f$</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>0.6057</td>
<td></td>
</tr>
<tr>
<td>80</td>
<td>0.6326</td>
<td></td>
</tr>
<tr>
<td>90</td>
<td>0.6618</td>
<td>3.75</td>
</tr>
<tr>
<td>100</td>
<td>0.6937</td>
<td></td>
</tr>
</tbody>
</table>

5. **Conclusions**

An absorption refrigeration cycle using an ammonia-water solution as the working fluid has been investigated in this work. The system is integrated with a marine diesel engine which provides waste heat at different temperatures levels.

The system has been theoretically investigated by calculating their performance using a thermodynamic model developed according to the first law of thermodynamic applied to each component of the system. The model is coupled to a set of equations of state allowing the reliable calculation of the thermodynamic properties of the binary mixture used. The thermodynamic study of the cycle has been carried out for various operating conditions by varying generator, condenser, absorber and evaporator temperatures. It is found that high performance of the cycle is obtained at high generator and evaporator
temperatures and also at low condenser and absorber temperatures. Furthermore, the increase in the solution heat exchanger effectiveness improves the coefficient of performance with no effect on the circulation ratio.

References