

2008

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ABSORPTION RESORPTION CYCLE FOR HEAT RECOVERY OF DIESEL ENGINES EXHAUST AND JACKET HEAT

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ABSTRACT

Diesel engines on board of fishing trawlers produce significant quantities of heat. About 20% of the fuel energy is recoverable from the exhaust gases at temperatures of 320 to 400°C and about 30% of the fuel energy is recoverable from the jacket coolant system at a temperature level of 75 to 100°C. On board of fishing vessels there is a need for refrigeration. Rapid freezing requires low evaporating temperatures in the equipment used, generally plate freezers and air coolers in the related freezing stores. The evaporating temperature can then be as low as -55°C. At this evaporating temperature and heat rejection to the sea water, single stage absorption cycles can only be driven by the hot exhaust gases and have very low COP's. The paper investigates the possibilities of a two stage absorption resorption cycle to make optimal use of the recoverable heat available at the two different temperature levels. The paper gives a comparison between model results for the cooling capacity obtainable under different operating conditions for both single and two stage cycles. The two stage option delivers approximately a 25% larger cooling capacity.

1. INTRODUCTION

In fishery trawlers often the harvested fish is directly frozen on board. For these applications generally very low temperatures are required: around -55°C.

In these ships diesel engines are applied with large power output. Rest heat is generally available at both high temperature (from the exhaust gases) and mid-temperature (from the motor cooling cycle). Part of the rest heat available in the exhaust gases is used for steam production. The case under consideration has about 2.5 MW heat available at temperatures above 180°C. Further about 2.3 MW heat will be available at temperatures around 90°C from the water jacket cooling.

In this paper, the possibility of application of the lower temperature heat in combination with the high temperature heat for driving a refrigeration cycle is investigated. The situation is schematically illustrated in Figure 1.

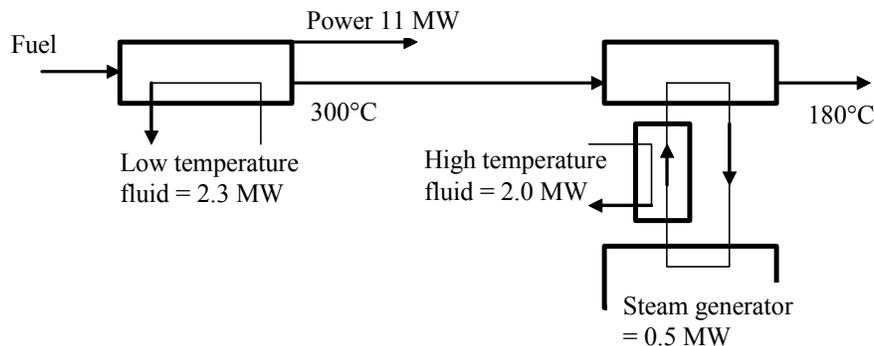


Figure. 1. – Availability of heat at different temperature levels.

The cycle may reject heat to sea water with a temperature of at maximum 30°C. Refrigeration is required at -55°C.

Single stage absorption operation of ammonia water cycles is only possible when the driving temperature is 180°C or higher. This implies that two stage operation must be considered. Several two stage cycles have been considered but they all failed in allowing the use of both temperature levels.

Finally a two stage absorption resorption cycle has been identified that makes it possible. This paper discusses the lay-out of the cycle and compares the performance of the cycle to the performance of a single-stage cycle that makes use of only the high temperature heat. It confirms the potential of such a two stage cycle for optimum use of the rest heat of diesel engines.

2. AVAILABILITY OF HEAT

This paper concerns a specific fishing trawler that combines fishing with the production of frozen fish 5400 hours/year. Under these conditions both main and auxiliary engines will be in operation, delivering the largest power but also the largest quantity of heat. Generally a part of exhaust gases heat is used for the production of steam but still a large amount remains available for other application as heat fired refrigeration cycles. Table 1 gives recoverable heat and its temperature level.

Table 1. – Availability of heat.

Recoverable heat	Temperature range	Power
Exhaust heat	450-200°C	1.99 MW
Jacket cooling water	95-75°C	2.34 MW

A single stage absorption cycle can only operate when driven by the high temperature exhaust gases. This means that only 1.99 MW is available to drive the cycle. The COP of a cycle operating with an evaporating temperature of -55°C, a heat rejection temperature of 30°C and an end generation temperature of 180°C is 0.23. The maximum cooling capacity delivered by the single stage cycle is 458 kW. The required freezing capacity of the ship at -55°C is 2.7 MW. Details of the here reported study can be found in Gerritsen (2007).

In the past several attempts have been undertaken to utilize both recoverable heat sources. Longo et al. (2005) used the energy recovered from the exhaust gases of an internal combustion engine to drive a double effect water-lithium bromide cycle and the heat recovered from the cooling jacket to drive a single effect water-lithium bromide cycle. The two absorption cycles were integrated having the evaporator and absorber in common. The evaporating temperatures required for the present application prevent the use of water-lithium bromide. Under the external conditions imposed also such an ammonia-water refrigeration plant cannot operate.

After investigation of the extended list of options discussed by Alefeld and Rademacher (1994) a new cycle has been identified that allows for the delivery of heat at two different temperature levels while the external conditions can be maintained. The proposed cycle is illustrated in Figure 2 and, so far as known to the authors, has not been previously proposed.

As illustrated in the figure, the system operates at three pressure levels. The resorption cycle (generator 3 and absorber 2) operates only at the low and high pressures, while the absorption cycle operates at the three pressure levels. The high pressure results from the operating conditions of condenser 2. This is a sea water cooled condenser and it is expected that at maximum it assumes condensation temperatures of 32°C. The intermediate pressure results from the operating conditions of condenser 1. This condenser is cooled by the generator of the resorption cycle (generator 3) so that the intermediate pressure depends on the operating conditions in the resorption cycle. The low pressure is imposed by the evaporating temperature: -55°C. Both the vapor flow produced by the evaporator and by the generator of the resorption cycle (generator 3) are absorbed in absorber 1. The pump at the outlet of the low pressure absorber (absorber 1) circulates the strong solution to both the mid and high pressure generators. The internal heat exchangers between strong and weak solution flow shown in Figure 2 play an important role in obtaining acceptable performance. The distribution of the main flow to generator 1 and 2 is related to the heat input to each of the generators. The purity of the ammonia vapor leaving the generators is improved by rectification. The vapor from the mid pressure generator flows directly to condenser 1. Part of the vapor from generator 2 (high pressure generator) is absorbed by the absorber of the resorption cycle (absorber 2) while the rest of the vapor is condensed in condenser 2. The condensate of both high pressure and mid pressure condensers is expanded into the evaporator. The weak solution that leaves the low pressure generator is pumped to the high pressure absorber through an internal heat exchanger (not shown in Figure 2).

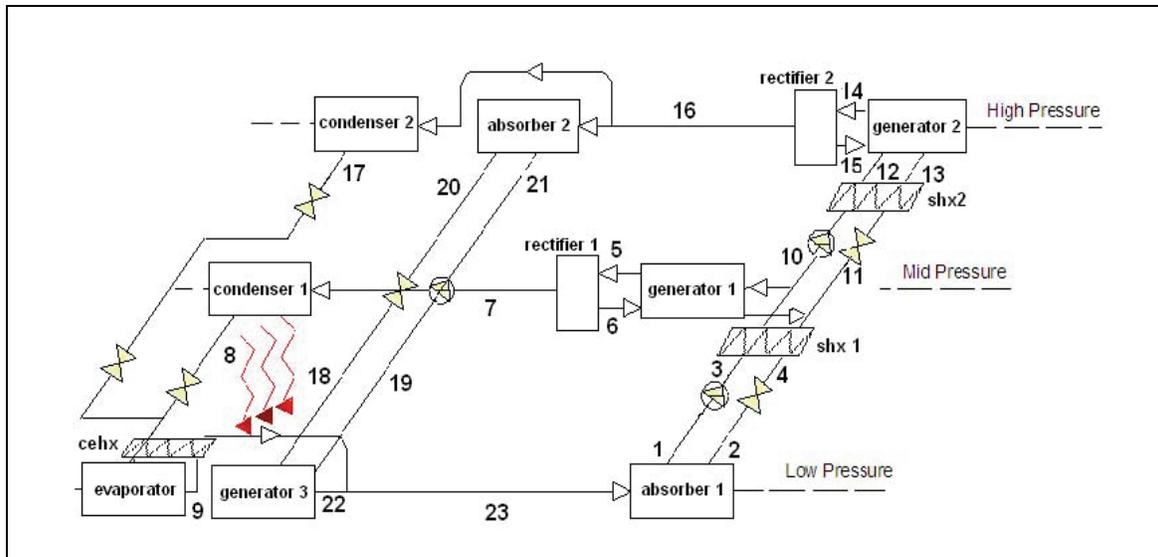


Figure 2. – Two stage absorption-resorption cycle with desorbers driven by high temperature exhaust gases (generator 2) and by mid temperature motor cooling fluid (generator 1).

3. COMPONENT MODELING

In order to model a complete absorption cycle, at first the components need to be modeled. This will be done in the following subsections. A basic absorption cycle consists of seven different components: absorber, generator, condenser, rectifier, evaporator, pump and the valve. The modeling approach will follow the modularity principle, making re-use possible of the components for a different desired cycle. The equations needed for the modeling approach are listed below for each component of the absorption cycle.

2.1 Absorber model

Figure 3 is a model diagram of the absorber. The relevant variables are given as known inputs or outputs of the absorber component. Refrigerant vapor with concentration (x_2), mass flow (m_2), enthalpy (h_2), temperature (T_2) and pressure (p_2) enters the absorber where it is absorbed by the weak solution

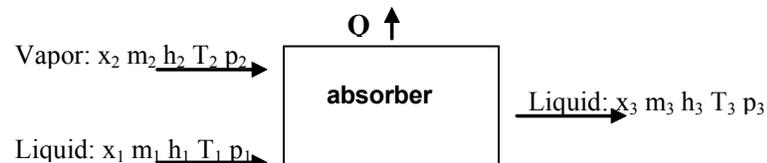


Figure 3. – Causality diagram for the absorber

flow. Absorption heat (Q_{abs}) is rejected to a sea water flow. The strong solution that leaves the absorber is assumed to have a fixed subcooling (5K). All flows enter and leave the absorber at the same pressure ($p_1=p_2=p_3$). The model has 8 inputs ($x_1, m_1, h_1, T_1, x_2, m_2, h_2, T_2$) and 5 outputs ($m_3, x_3, h_3, T_3, Q_{abs}$). The mass and energy balances give:

$$m_1 + m_2 = m_3 \quad (1)$$

$$m_1 x_1 + m_2 x_2 = m_3 x_3 \quad (2)$$

$$Q_{abs} = m_1 h_1 + m_2 h_2 - m_3 h_3 \quad (3)$$

The enthalpy and concentration of the ammonia can be determined from FluidProp (Colonna and van der Stelt, 2004) when other state variables are known. Since the absorber outlet is often subcooled, which affects the performance of the absorption process negatively, a subcooling of 5K has been considered. The outlet temperature T_{3sub} is assumed to be 5K lower than the saturation temperature:

$$h_3 = f(T_{3sub}, q = 0, x_3) \quad (4)$$

$$T_{3sub} = T_3 - 5 \quad (5)$$

$$x_3 = \text{fluidprop}(T_3, P_3, x_3) \quad (6)$$

The absorber is modeled in Simulink. In the model a temperature difference of 6K is assumed between the cooling water inlet temperature and the strong solution outlet temperature. The pressure drop between evaporator and absorber is assumed negligible so that the absorber pressure is 30 kPa (0.3 bar). The enthalpy at the absorber outlet is calculated with Eq. (4), while the concentration NH_3 in the outlet stream is calculated with Eq. (6).

2.2 Pump model

Figure 4 is a model diagram of a pump. The relevant variables are given as inputs or outputs of the diagram. In the pump the ammonia solution is pumped to a higher pressure (p_2). Also the fluid temperature rises to a higher value. The enthalpy after the pump is equal to the inlet enthalpy plus the specific work

input in the pump $W_{\text{pump_spec}}$. The solution mass flow and concentration remain constant as the fluid passes the pump. The model has 4 inputs (x_1, m_1, h_1, T_1) and 5 outputs ($x_2, m_2, h_2, T_2, W_{\text{pump}}$). An efficiency, η_p , is assumed

to calculate the specific work input to the pump from the ideal specific work. The mass and energy balances give:

$$m_1 = m_2 \quad (7)$$

$$x_1 = x_2 \quad (8)$$

$$W_{\text{pump}} = m_2 h_2 - m_1 h_1 \Rightarrow m_1 = m_2 : \frac{W_{\text{pump}}}{m_1} = h_2 - h_1 \quad (9)$$

$$W_{\text{pump}} = \frac{m_1 v_1 (p_2 - p_1)}{\eta_p} \quad (10)$$

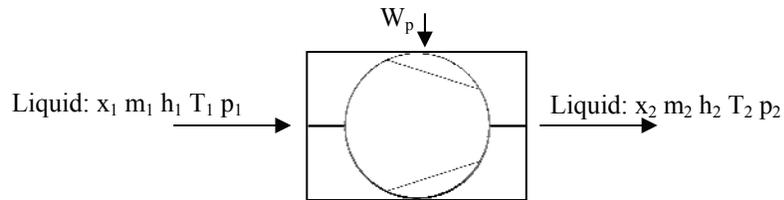


Figure 4. – Causality diagram for a pump

2.3 Internal heat exchanger model

Figure 5 shows the model diagram of the internal heat exchanger (shx). In this model the temperature difference between the two fluids at the hot end of the heat exchanger is assumed. In the heat exchanger the cold strong solution is heated by the hot weak solution.

The pressure drop is assumed negligible and both flows are at the same pressure: $p_1=p_2=p_3=p_4$. The purpose of the heat exchanger is to increase the enthalpy of the strong solution from h_1 to h_3 as much as possible (ideally to the saturation state).

The model has 10 inputs ($x_1, m_1, h_1, T_1, p_1, x_4, m_4, h_4, T_4, p_4$) and 10 outputs ($x_3, m_3, h_3, T_3, p_3, x_2, m_2, h_2, T_2, p_2$). The temperature difference between the two fluids at the hot end of the heat exchanger is a parameter.

The mass and energy balances give:

$$m_2 = m_4 \quad (11)$$

$$m_1 = m_3 \quad (12)$$

$$m_2 (h_4 - h_2) = m_1 (h_3 - h_1) \quad (13)$$

The enthalpy of the weak solution as it leaves the heat exchanger is obtained from Eq. (14):

$$h_2 = \text{fluidprop}(T, p_{\text{cond}}, x_4) \quad (14)$$

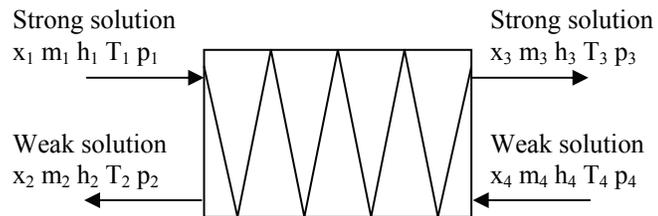


Figure 5. – Causality diagram for the internal heat exchanger

2.4 Model of the generator and rectifier

A model diagram of the generator is given in Figure 6. In this figure, the relevant variables are given as inputs or outputs of the generator / rectifier.

Strong solution enters the generator where it is heated to generate vapor consisting predominantly of the refrigerant component. The weak solution flows out of the generator. The desorption process takes place by adding thermal energy Q_{gen} to the generator. It is assumed that

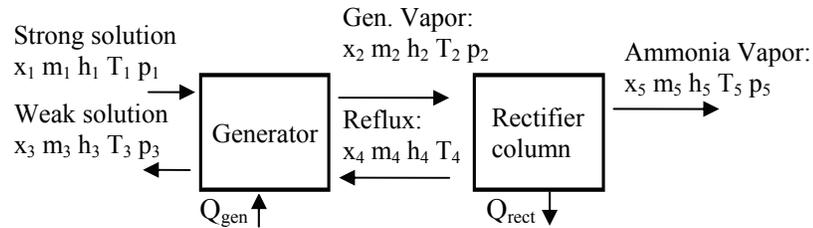


Figure 6. – Causality diagram for the generator / rectifier

the weak solution that leaves the generator is at saturated conditions. Pressures of the streams are equal: $p_1=p_2=p_3=p_4$. The pressure is fixed by the condenser pressure.

The vapor desorbed in the generator enters the rectifier, where it is cooled to liquefy mainly the water constituent of the vapor leaving almost pure ammonia vapor ($x_2=1$) that leaves the column. The reflux which is sent back to the generator has (ideally) the same composition as the solution entering the generator, the same pressure and is at its saturation temperature. The model has 4 inputs (x_1, h_1, T_1, Q_{gen}) and 9 outputs ($m_1, x_2, m_2, h_2, T_2, x_3, m_3, h_3, T_3$).

The mass and energy balances give:

$$m_1 + m_4 - m_2 - m_3 = 0 \quad (16)$$

$$m_1 x_1 + m_4 x_4 - m_2 x_2 - m_3 x_3 = 0 \quad (17)$$

$$m_1 - m_5 - m_3 = 0 \quad (18)$$

$$m_2 x_2 - m_4 x_4 - m_5 x_5 = 0 \quad (19)$$

$$m_2 - m_4 - m_5 = 0 \quad (20)$$

$$m_1 h_1 + m_4 h_4 - m_2 h_2 - m_3 h_3 = Q_{gen} \quad (21)$$

These five equations are sufficient to calculate all the mass flows. The compositions and enthalpies needed for the equations are determined with FluidProp (Colonna and van der Stelt, 2004):

$$x_1 \Rightarrow \text{output}(\text{absorber}) \quad (22)$$

$$x_2 = \text{fluidprop}(P_2, T_2, q_{q=1}) \quad (23)$$

$$x_3 = \text{fluidprop}(P_2, T_3, q_{q=0}) \quad (24)$$

$$x_4 = x_1 \quad (25)$$

$$x_5 = 1 \quad (26)$$

$$h_1 \Rightarrow \text{output}(\text{shx}) \quad (27)$$

$$h_2 = \text{fluidprop}(P_2, q_{q=1}, x_2) \quad (28)$$

$$h_3 = \text{fluidprop}(P_3, q_{q=0}, x_3) \quad (29)$$

$$h_4 = \text{fluidprop}(P_4, q_{q=0}, x_4) \quad (30)$$

$$h_5 = \text{fluidprop}(P_5, q_{q=1}) \quad (31)$$

The temperature of the weak solution that leaves the generator is determined by the temperature of the heating medium. It is considered here that thermal oil is heated by the exhaust gases of the engine, entering the generator at 200°C and leaving it at 190°C. The weak solution leaves the generator with a temperature of 180 °C. This temperature determines the ammonia concentration of the weak solution. The composition of the generated vapour (x_2) is determined by the generator pressure (P_2), (saturation) temperature and composition of the liquid entering the generator (x_1), as is shown in Eq. (23). The assumption is made that the generated vapour is in equilibrium with the entering liquid.

2.5 Condenser model

In the condenser the vapour from the rectifier column is liquefied. The latent heat of the vapour is transferred to the sea water flow. A model diagram of the condenser is shown in Figure 7.

Superheated vapor enters the condenser and saturated liquid leaves the condenser. Condensation takes place at constant pressure depending on the amount of ammonia vapor that enters the condenser. The model has 5 inputs (x_1, m_1, h_1, T_1, p_1) and 6 outputs ($x_2, m_2, h_2, T_2, p_2, Q_{cond}$). The overall heat transfer coefficient multiplied by the heat transfer area form a parameter in this model.

The mass and energy balances give:

$$m_1 = m_2 \quad (32)$$

$$x_1 = x_2 \quad (33)$$

$$Q_{cond} = m_1 h_1 - m_2 h_2 \quad (34)$$

Further the following equations apply for the condenser:

$$h_1 = \text{fluidprop}(P_{cond}, q_{q=1}) + c_p \Delta T_{\text{superheating}} \quad (35)$$

$$h_2 = \text{fluidprop}(P_{cond}, q_{q=0}) \quad (36)$$

$$Q_{cond} = UA \Delta T_{\log} \quad (37)$$

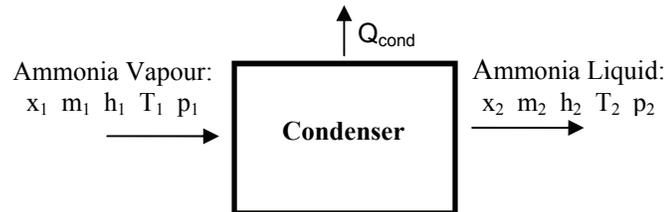


Figure 7. – Causality diagram for the condenser

2.6 Model of the condenser-evaporator heat exchanger

This heat exchanger is modelled in a similar way as the internal solution heat exchanger discussed in Section 2.3.

The effectiveness is given by

$$\varepsilon_{shx} = \frac{T_{11} - T_{10}}{T_8 - T_{10}} \quad (38)$$

And the energy balance by:

$$m_8 h_8 + m_{10} h_{10} = m_9 h_9 + m_{11} h_{11} \quad (39)$$

2.7 Evaporator model

The condensed ammonia flows from the condenser first through the condenser-evaporator heat exchanger and then through the throttling device that decreases the pressure. At this low pressure the remaining ammonia liquid evaporates taking heat from the

application fluid. In practical applications the heat is taken from a

secondary refrigerant flow which is circulated through the cooling system. A model scheme is shown in Figure 8.

The mass and energy balances give:

$$m_1 = m_2 \quad (40)$$

$$x_1 = x_2 \quad (41)$$

$$m_2 h_2 = m_1 h_1 + Q_{evap} \quad (42)$$

With

$$h_2 = \text{fluidprop}(P_{evap}, q_{q=1}) \quad (43)$$

$$T_2 = \text{fluidprop}(P_{evap}, q_{q=1}) \quad (44)$$

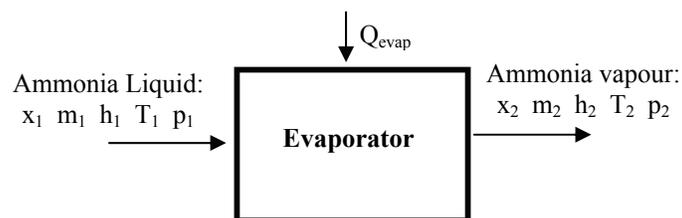


Figure 8. – Causality diagram for the evaporator

$$P_1 = P_2 \quad (45)$$

In the previous sections, the equations needed to model the components of the absorption cycle have been presented. These equations have been implemented in Simulink to simulate each component. In doing this the known variables are the inputs, and the idea is to calculate the unknown outputs. Both a single stage and the two stage absorption resorption (shown in Figure 2) cycles have been implemented in Simulink.

2.8 Example operating conditions

Table 2 shows the results for the two stage cycle when the sea water temperature fed to both absorber and condenser is set to 32 °C. The low temperature generator shows an outlet temperature of 85 °C while the high temperature generator operates at 180 °C.

Table 2. - Pressure, concentration, enthalpy and the mass flow at the different the state points

state points	T [C]	P[kPa]	x[kg/kg]	h[kJ/kg]	m [kg/s]
1	34	30	0,125	61,3	8,00
2	60	30	0,025	233,0	7,18
3	58	80	0,125	181,9	3,51
4	85	80	0,025	336,4	3,15
5	58	80	0,8	2361,9	0,47
6	58	80	0,125	181,9	0,11
7	-38	80	1	1411,1	0,36
8	-38	80	1	23,3	0,36
9	-55	30	1	1382,4	0,49
10	58	80	0,125	181,9	4,49
11	85	80	0,025	336,9	4,03
12	152	1217	0,125	585,5	4,49
13	180	1217	0,025	751,1	4,03
14	152	1217	0,6	2162,9	0,84
15	152	1217	0,125	585,5	0,38
16	31	1217	1	1486,8	0,46
17	31	1217	1	348,6	0,13
18	-46	30	0,7	-246,4	1,11
19	-38	30	0,575	-296,4	0,79
20	33	1217	0,7	157,4	1,11
21	59	1217	0,575	149,4	0,79
22	-55	30	1	1382,4	0,33

4. MODEL RESULTS

Figure 9 shows the predicted cooling capacity for both single-stage and the two-stage absorption cycles as a function of sea water temperature. It is assumed that the engine operates at 85% of full load so that 1500 kW heat are available to heat generator 1. Then 1990 kW is needed to heat generator 2. As it can be seen from the graph, the two stage absorption refrigeration gives better results than the single stage model. The model predicts, for both cases, a linear decrease of the evaporator capacity with sea water temperature. Since the temperature difference between refrigeration cycle and external medium is maintained constant, the area of the heat exchangers changes for each condition of the sea water temperature. Figure 9 does not apply for a fixed design but indicates the best attainable performance depending on the sea water temperature. It makes the advantage of the absorption resorption two-stage cycle clear. The question is of the extra investment in the more complex cycle compensates for the extra delivered cooling capacity. When compared to the single stage absorption cycle an extra investment of 172 k€ is expected for the two-stage cycle, see Gerritsen (2007) for details. If the difference in cooling capacity between the two options (160 kW) would be delivered by a vapor compression cycle with COP=1.43 then the diesel engine would need to deliver yearly 604195 kWh_e extra. The related CO₂ emission savings amount then to 369 ton CO₂ per year (0.61 kg CO₂/kWh). Considering a heavy fuel oil price of 330 \$/ton, the electricity costs 0.062 €/kWh (Gerritsen, 2007) and the yearly savings in comparison with the single stage absorption machine amount 37.5 k€. The extra investment will pay itself back in 5 years.

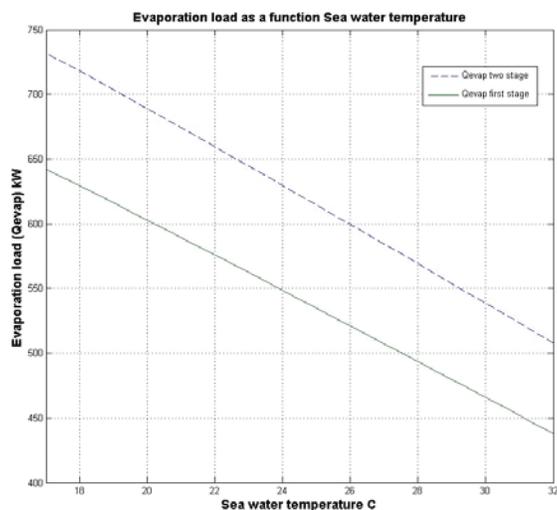


Figure 9. – Cooling capacity as a function of the sea water temperature for single stage and the cycle shown in Figure 2

5. CONCLUSIONS

The paper proposes an absorption resorption two stage cycle that allows for the combined use of high and medium temperature waste heat of diesel engines to operate low temperature refrigeration cycles. It has been concluded that: The cycle allows for a significantly larger cooling capacity than single stage absorption refrigeration driven by only the high temperature waste heat.

It requires an extra investment in comparison to the single stage absorption refrigeration but it prevents the emission of large quantities of CO₂

The payback time of the extra investment is acceptable.

ACKNOWLEDGEMENT

The authors thankfully acknowledge the support provided by SenterNovem (project number 0377-05-03-01-013).

NOMENCLATURE

A	heat transfer area, m ²	U	overall heat transfer coefficient, kW/m ² K
h	enthalpy, kJ/kg	v	specific volume, m ³ /kg
m	mass flux, kg/s	W	shaft power, kW
p	pressure, bar	x	weight concentration of refrigerant, kg/kg
Q	heat flux, kW	ε	heat exchanger effectiveness
T	temperature, °C	η _p	pump efficiency

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