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CO_2 as working fluid in a Rankine cycle for electricity production from waste heat sources on fishing boats

Summary report

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TECHNICAL REPORT

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RESULT (summary)

CO_2 based Rankine cycle for electricity production for use in maritime applications has been investigated and compared to conventional Organic Rankine Cycle technology

Performances of such a power cycle equal conventional ORC solutions for high temperature waste heat (exhaust gas). Performances for low temperature waste heat are significantly improved (25%) using CO_2 technology. Energy (fuel) savings up to 10% can be achieved.
It was also showed that CO_2 technology has a potential for size reduction.

The pre-project confirmed that CO_2 based electricity production for maritime application is technically a promising solution. It was also showed that use of such technology could lead to important savings.

KEYWORDS

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1 SUMMARY

CO_2 based Rankine cycle for electricity production for use in maritime applications has been investigated and compared to conventional Organic Rankine Cycle technology.

Performances of such a power cycle equal conventional ORC solutions for high temperature waste heat (exhaust gas). Performances for low temperature waste heat are significantly improved (25%) using CO_2 technology. Energy (fuel) savings up to 10% can be achieved.
It was also showed that CO_2 technology has a potential for size reduction.

The pre-project confirmed that CO_2 based electricity production for maritime application is technically a promising solution. It was also showed that use of such technology could lead to important savings.

2 INTRODUCTION

Only 40% of the energy consumed by a diesel engine is converted to useful work, the rest is lost waste heat. Recent oil price increases, together with new regulations on NOx emissions strongly motivate optimal use of energy in all industries and in particularly for the fishing industry.

Among several actions, waste heat produced on ships could be converted to electricity. This electricity would be used directly on the ship (no storage). The produced electricity from waste heat would in turn reduce the fuel consumption as the generator onboard would have lower requirements.

There is some established technology to produce electricity from heat. For low temperature heat sources like exhaust gas or the engine's cooling water Organic Rankine cycle are generally the preferred solutions (ref 1, ref 2).

Those processes are generally quite bulky and then not ideal for maritime applications. In addition, organic fluids are either toxic, flammable or have high Global Warming Potential or Ozone Depletion Potential ratings (ref 3, ref 4). As a result they are exposed to additional taxes and their use could be restricted in the future.

CO_2 is a natural fluid, it is environmental friendly, non toxic and non flammable. It is also very cheap, made as a by-product of many chemistry processes and the availability world wide is very good. CO_2 based power cycles are receiving increasing attention (ref 15, ref 16) and is a serious working fluid candidate.

This note presents an investigation of a potential CO2 based technology for electricity production on boats.

3 TECHNICAL BACKGROUND

3.1 Low grade energy recovery

The second law of thermodynamics states that all energy sources are not of the same quality: it is possible to convert 1 KW of work into 1kW of heat but it is not possible to produce 1kW of work from 1 kW of heat. In this prospect recovering energy from low temperature sources is challenging. The next section will present important background material to understand inherent limitations of such processes and parameters for optimization.

3.1.1 Maximum thermal efficiency

Thermal efficiency is the ration of work produced divided by the amount of heat absorbed from the heat source. As stated before the efficiency cannot be 100%. Maximum efficiency for a perfect theoretical engine varies with the source temperature. For a constant source (T_{high}) and sink (T_{low}) temperatures (in Kelvin) it writes:

$$\eta_{thermal} = \frac{T_{high} - T_{low}}{T_{high}}$$

Low grade energy sources are generally not constant heat sources, their temperature varies as heat is taken away by the process. Such process with temperature gliding can be seen a many theoretical engines in serial arrangement working with decreasing source temperatures. Integration gives:

$$\eta_{thermal,gliding} = 1 - \frac{T_{low}.\ln\left(\frac{T_{high}}{T_{low}}\right)}{T_{high} - T_{low}}$$

Both theoretical efficiencies are plotted for a sink temperature of 20’C (Figure 1).

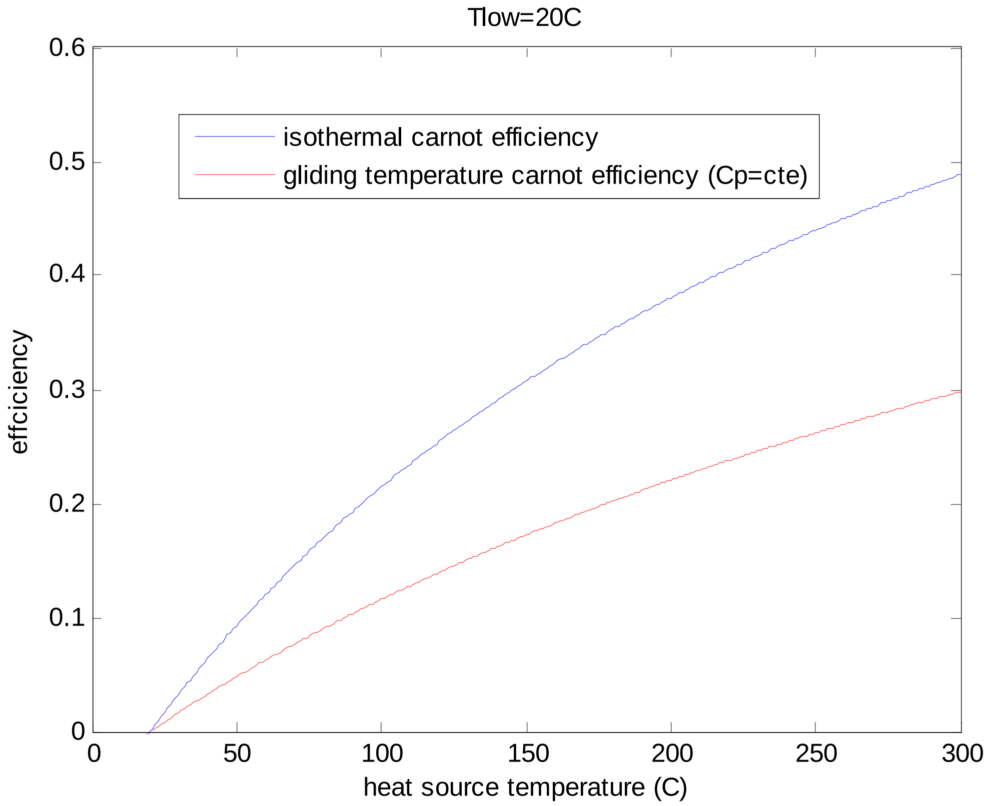

Figure 1: Maximum theoretical efficiency for constant and gliding heat source temperatures.

Maximum efficiency decreases with source temperature, for a 100’C source the maximum efficiency of 10% can be achieved in case of temperature gliding, meaning that the mean temperature of the source being cooled will be lower.

3.1.2 Resource utilization

Thermal efficiency is not fully characterizing a process, in some cases it can even be directly misleading. On can imagine a process where only a very small proportion of the source heat is used to generate useful work with very high thermal efficiency. Despite the high thermal efficiency, only a small amount of work is produced. For practical purpose, this is obviously a bad utilization of the heat source.

Resource utilization can be characterized by the ratio of the produced work (W) divided by the source available energy:

$$\eta_{use} = \frac{W}{\dot{m}Cp\left(T_{high} - T_{low}\right)}$$

$\dot{m}$ is the mass flow rate of the heat source and Cp the heat capacity of the source (assumed constant for the temperature interval considered).

3.2 Existing technology, Rankine cycles

3.2.1 Principle

Rankine cycles makes use of phase transition to improve the performance of a pure gas cycle. An ideal cycle would comprise:

- A. Liquid compression (constant entropy)
- B. the liquid is heated up, possibly with a phase transition,, and possibly super heated in contact with the heat source (constant pressure)
- C. The warm gas is expanded in an expansion machine, e.g. a turbine, producing useful work (constant entropy)
- D. The gas is cooled down and condensed in contact with the heat sink

Liquid compression in a pump requires much less work (small specific volume) than gas compression. In addition heat transfer for boiling and condensation of a fluid is very efficient, and relatively compact heat exchangers can be built.

In some cases the gas at the outlet of the expansion machine (a turbine for example) has a temperature higher than the saturation temperature at this pressure. As a result, this gas must be cooled prior to condensation. This has an energy cost, as this heat to be dissipated could have been used to provide addidtional work.

Loss of efficiency due to super heating can be partially recovered using an internal heat exchanger or “recuperator”. This additional heat exchanger uses some of the superheat to warm up the liquid out of the pump (ref 11). As a result heat absorption is performed at higher temperatures (the heating at the lowest temperature part for the working fluid is performed in the recuperator which does not count in the thermal efficiency calculation).

High temperature applications traditionally make use of water as the working fluid. Its good heat conductivity, large heat capacity and small pump work makes it a very good working fluid.
Unfortunately the condensation pressure at low temperature is very low, ruling out its use for these applications (typically below 350C, ref 4).

3.2.2 ORC

Various fluids have been used to improve the performance of the Rankine cycle at lower source temperatures. Among them organic compound are generally preferred, ref 12, ref 14). They combine reasonable condensation pressure, good heat transfer and low pump work.

This technology is to a certain extent established for land based applications. Those applications range from use of geothermal or solar energy to biomass fired plants. Industrial players are established (Ormat, Turboden, UTRC) and provide solutions from a few hundred of kW to a few MW output power.

Maritime application gives particular safety restriction. The working fluid should then preferably be non-flammable and non-toxic. Use of alkanes, aromatics or alcohol compounds is then to be considered cautiously.

Despite development of new chemicals is still ongoing (ref 13), more attention has been paid to environmental impact of the working fluids. Organic compounds used for refrigeration and electricity production (ORC) can have a large impact on Ozone depletion and global warming. Because of the bad score in Global Warming Potential and Global Warming Potential, some compounds have already been forbidden (all chlorine based compounds, well known as CFC). More restrictions are expected to come in the future (ref 3).

3.2.3 CO2 Transcritical cycle

CO_2 has a low critical temperature 31.1C. Its use in a Rankine cycle differs from most other compounds. The basic steps are the same, but there is no boiling or phase change taking place during heat absorption since as the CO_2 has a supercritical pressure in the heat exchanger for heat absorption from the heat source (Figure 2).

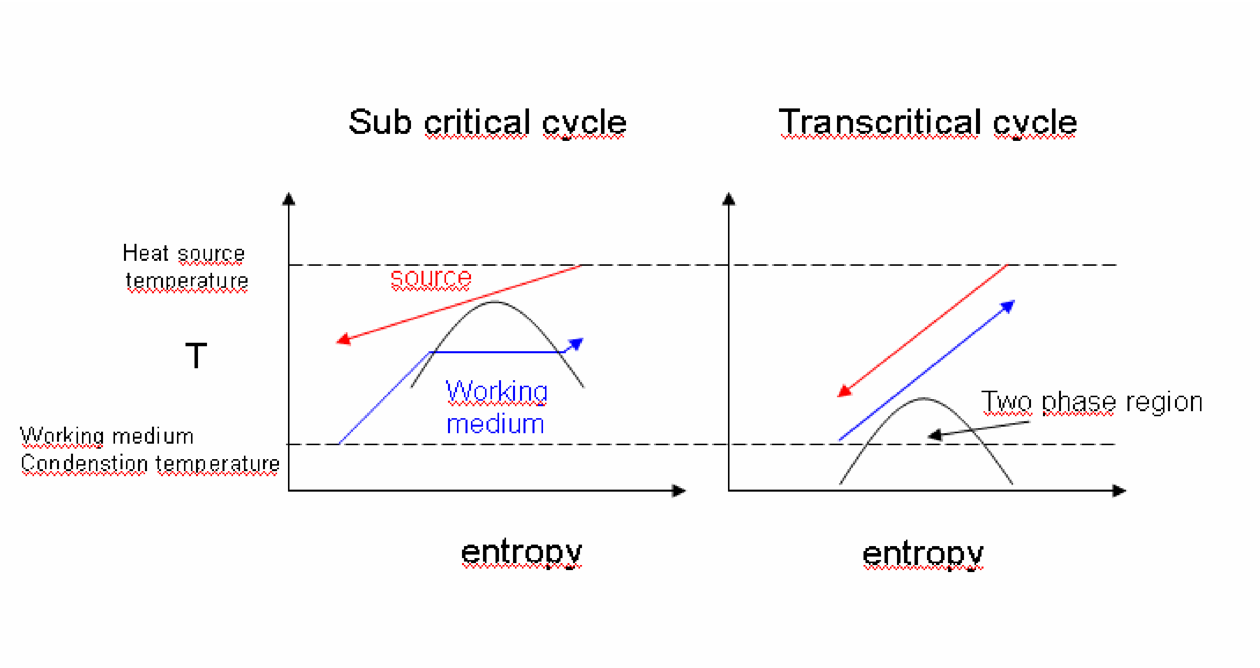

Figure 2: Schematic of the subcritical (R134a) and transcritical (CO2) cycles.

This has very important implications for system performance. The main advantage is enhanced heat recovery in the main heat exchanger. For a classical boiling working fluid, the minimum temperature difference in the heat exchanger (called the pinch) takes place somewhere inside the heat exchanger. This limits the temperature of the working fluid out the heat exchanger, limiting the thermal efficiency of the cycle. This also limits the lowest obtainable heat source temperature at the exit of the heat exchanger, limiting the energy extracted from the source and then again limiting resource utilization (ref 16).

Ideally, the temperature of the working medium in the heat exchanger should be everywhere equal to the heat source at this location minus the minimal temperature difference acceptable (typically 10C). This is called temperature matching.

CO_2 critical pressure is rather high: 73.7 bar, so is the condensing pressure at room temperature (60bar at 20C). This could be seen as an inconvenience at first sight, but it actually is an advantage.

Heat transfer with this dense gas is very good and volumetric efficiency high, thus avoiding large heat exchanger volumes characterizing usual gas processes. In the vicinity of the critical region the heat transfer is actually better than for most boiling fluids. The density of the gas at the exit if the turbine (at condensation pressure) is rather large, allowing for the development of very compact equipment.

CO_2 is cheap, generally recovered as by-product, it does not deplete the ozone layer and is carbon neutral, and is easily available all over the world.

4 CO2 BASED POWER CYCLE FOR MARITIME APPLICATION

Assessment of a CO_2 based power technology for maritime applications has been performed. Cycle simulations were performed with the engineering tools ProII. Performances were compared to an R134a based cycle. R134a as working fluid was recommended by UTRC in a project initiated by Fiskarlaget (ref 6).

Potential electricity production and fuel saving are estimated in this report for several types of ships common in Norwegian waters. Last, a brief status on component availability and need for development is given.

4.1 Performances, comparison with R134a

Use of conventional ORC technology for maritime application is currently examined in a project initiated by Fiskarlag and managed by SINTEF Fiskeri og Havbruk (ref 5, ref 6). The technology provider, UTRC recommended two working fluids: R245fa and R134a. For R245fa we have a lack of reported results and it is not available in our simulation tools. R134a was therefore chosen for comparison.

Many parameters can influence the result of a simulation, it is therefore very important to list the conditions for those simulations, and to make sure that they are identical for the two working fluids. There are two heat sources on a fishing boat: exhaust gas and engine’s cooling water. Those sources are very different and will be investigated separately.

Turbine efficiency was set to 0.8 which is a common value reported in the literature. Pump efficiency was set to 0.7 which is also a common value reported in the literature. Additional losses due to pressure drop in the heat exchangers have not been taken into account.

It is considered that the heat sink has a constant temperature. Mean temperature of Norwegian waters is around 9’C. The condensing temperature is then chosen to 20’C (11 K temperature difference in the condenser, which is considered to be relatively high). Extra pump work for forcing working fluid condensation is not included in the calculations.

4.1.1 Recovery from exhaust gas

Exhaust gas is reported to be released at 300-350’C. We chose a heat source temperature equal to 320’C. The maximum efficiency, power out to heat in, with such a gliding temperature source is close to 30%, which is promising. Unfortunately, it is reported deposition of solids if the exhaust gas is cooled down below 180’C. This would limit the amount of energy that can be recovered by the working fluid. Recent investigation showed that this problem could be handled using proper design for the heat exchanger (ref 17). More research is needed to close this issue. We chose to use 180’C as the lowest exhaust temperature out of the heat absorption heat exchanger in our simulations.

In the simulations, the temperature of the exhaust gas is cooled to 180’C to ensure maximum source utilization. Thermal efficiency is then fully characterizing the process. The heat exchanger’s minimum temperature difference between exhaust gas and working fluid in the main heat exchanger is checked to ensure that such heat exchanger could be realistic.

In an R134a process with recuperator, thermal efficiency up to 16% were obtained while the CO2 process showed a thermal efficiency up to 14% for a high pressure side equal to 100 bar. For a high pressure side equal to 160 bar the thermal efficiency is improved to 20%.

CO_2 heat exchangers with pressure rating up to 150 bar has been developed for refrigeration purposes. This pressure rating should also be realistic for RC plants.

4.1.2 Recovery from engine’s cooling water

Cooling water exits from the engine at 90’C, due to the large heat capacity of water there is actually a lot of energy to recover. The maximum efficiency with such a gliding temperature source is close to 10%. Recovering such low grade energy is a real challenge.

In the simulations, the minimum temperature of the working fluid in the boiler/gasheater was set to 10’C. This time the temperature of the heat source at the exit of the boiler is not fixed, therefore both thermal efficiency and resource utilization have to be investigated to characterize the process

In an R134a process, resource utilization up to 3% could be achieved with a thermal efficiency equal to 6.6%. A source utilization of 3% can appear very bad. But it has to be compared with the maximum source utilization achievable in theory with this gliding temperature source: 9% (all the available energy of the source used with highest possible thermal efficiency).

For the CO_2 process resource utilization up to 4% could be achieved with a thermal efficiency equal to 5.6%. This is typical for a CO_2 trans-critical process: the thermal efficiency is slightly lower than a usual process but as the process is able to cool down the source at lower temperature, the resource utilization significantly improved.

4.2 Results for fishing boats

Energy data for the Norwegian fleet have been gathered by SINTEF Fiskeri og Havbruk and by COWI (ref 7, ref 9). Those data where used to investigated how much electricity could be generated by a CO_2 based power cycle.

Several groups of boats have been investigated, from small fishing vessels to large factory ship for cod fishing and on board processing.

Energy consumption have been divided in different categories and examined for several phases of the operation. We estimated the electricity production for each phase of operation, and used operational profiles to calculate an average electricity production. For small boats, we did not have the operational profiles, minimum and maximum electricity production are then reported.

Available heat

It is commonly reported that for modern diesel engines, only 40% of the consumed fuel is actually used to produce useful work. The rest is lost as heat 30% in the exhaust gas, 25% in the cooling water and 5% in the lubrication oil.

It was then possible to estimate available heat from energy consumption data. This method will be referred as method A.

For exhaust gas, we also have an experimental curve related available heat (exhaust gas cooled from 320C to 180C, ref 7, fig 5.4) to the work delivered by the engine. This method will be referred as method B.

The two methods were used to estimate the electricity production, differences are quite large but acceptable taking into account the large variation between different boats of the same group.

There are generally two engines on a boat, a main engine used to drive the propeller and a help engine which generate the electricity needed on board. If exhaust gas from the two engines exit the boat in the same chimney, all the heat can be used. If several chimneys are installed, it is probably not possible to use exhaust gas from the help engine. The two cases have been investigated.

Electricity production calculation

Thermal efficiencies of 14% for exhaust gas and 5.6% for the cooling water were used for the calculations.

For the method A, it has to be taken into account that not all the available heat is actually used by the power cycle (this issue was referred as the resource utilization in the previous chapters). If the exhaust gas is cooled from 320 to 180 instead of 20C, the effective efficiency is then (320-180)/(320-20)*thermal efficiency=6.5%. Only 6.5% of the available heat is converted to electricity.

The cooling water is cooled from 90C to 30C instead of 20C, only 4.8% of the available heat is converted to electricity.

Last, fuel consumption was compensated such that the energy needs of the boats were satisfied by the sum of the energy from the engines and the produced energy from waste heat.

Results

	exhaust gas methode A			
	both engines		**main engine**	
	mean production (kW)	*% of consumption*	*mean production (kW)*	*% of consumption*
kystbåt	6 to 25	4.7	4 to 20	3 to 3.8
kolmuletrål	132	4.7	100	3.4
ringnåt	87	4.7	63	3.2
torkstråler	176	4.7	132	3.4

	exhaust gas methode B			
	both engines		**main engine**	
	mean production (kW)	*% of consumption*	*mean production (kW)*	*% of consumption*
kystbåt	6.7 to 23	5	4.5 to 19	3.5
kolmuletrål	161	6	131	4.5
ringnåt	116	7	83	3.5
torkstråler	184	5.6	163	4.3

	cooling water methode A			
	both engines		**main engine**	
	mean production (kW)	*% of consumption*	*mean production (kW)*	*% of consumption*
kystbåt	3.7 to 15	2.9	2.4 to 12	1.8 to 2.4
kolmuletrål	82	2.9	62	2.1
ringnåt	54	2.9	39	2
torkstråler	110	2.9	82	2.1

Table 1: Potential electricity production and energy saving for several group of boat. Contribution from cooling water (thermal efficiency equal to 5.6%) and exhaust gas (thermal efficiency equal to 14%) are presented apart.

Table 1 presents the results fro four type of boats. “Ringnot” and “kolmuletrål” are actually the same boats but using different fishing techniques.

The percentage of energy (or fuel) that can be saved with a CO2 based power cycle does not depend crucially on the type of boat, up to almost 10% can be saved if both cooling water and exhaust gas are exploited.

The electricity production and therefore the dimensions of the power generating system vary a lot between the groups of boat. Electricity production ranges from 6kW fro small boats to 200kW for the biggest vessels. Obviously, practical solutions are not the same for these different sizes. Dimensioning of the components for those power cycles would require further work.

4.3 Components for a trans-critical CO2 cycle

Maritime applications impose additional restrictions to a power cycle. If weight is not a necessary limitation on the large ships, volume is definitely an issue. Critical components for the final dimensions of this process are the heat exchangers and the expansion turbine.

4.3.1 Heat exchangers

CO_2 has critical point at 73.8 bars and 31.05°C, meaning that in the two systems would be comparing supercritical CO_2 heat transfer and pressure drop against phase-changing heat transfer and pressure drop for R134a.

Trying to compare heat transfer coefficient and pressure drop, the heat flux and pressure drop gradient is integrated along a tube through a typical operating range. For CO_2 all calculations are carried out in the supercritical region because the pressure is above the critical pressure.

When comparing the two working fluids using equal geometry (7mm ID), heat load (15000 W/m² and mass flow 500 kg/m²s) the calculated heat transfer coefficient, pressure drop gradient and corresponding temperature profile along a tube is shown in Figure 3. The values are calculated from 30°C in the ”cold end”. The saturation temperature for R134a is 65 °C and the CO_2 pressure is 100 bars.

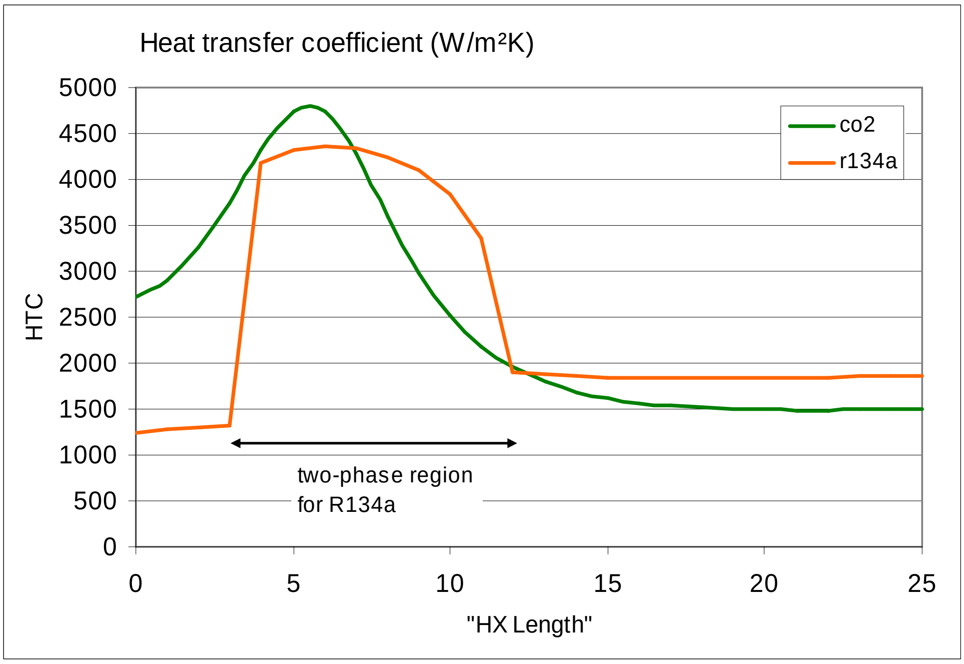

Figure 3: Heat transfer coefficient for CO_2 compared to R134a.

As seen, the heat transfer coefficient for CO_2 in the supercritical region is comparable to that of evaporating R134a in the two-phase region. In the superheated region, the heat transfer coefficient for R134a is higher for R134a. The corresponding frictional pressure drop is shown in Figure 4.

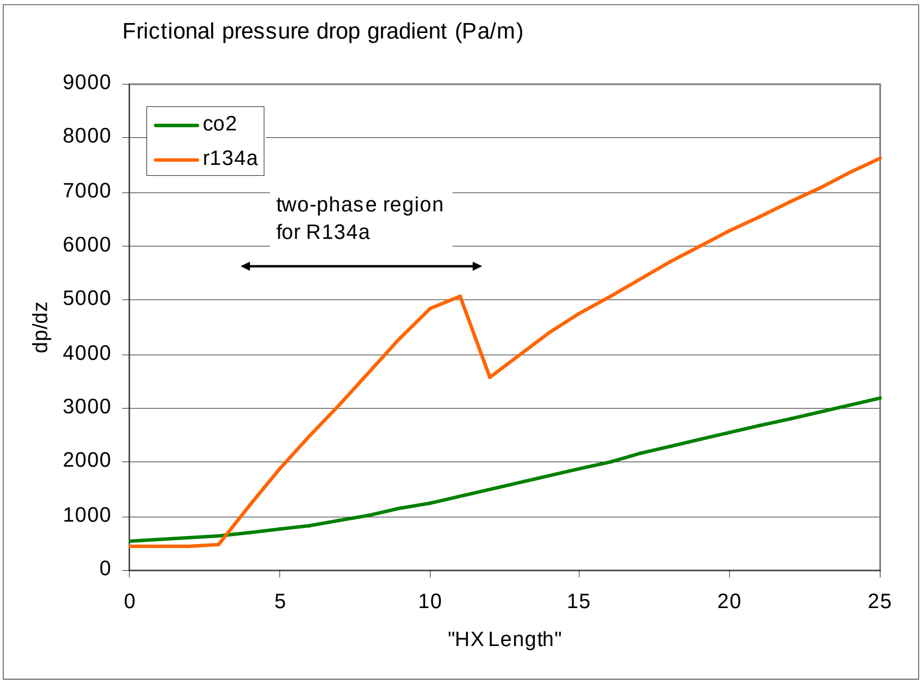

Figure 4: Frictional pressure drop for CO_2 and R134a.

When the frictional pressure drop between CO_2 and R134a is compared under equal conditions and geometry is clear the CO_2 will result in a much lower pressure drop than R134a. This means that it should be possible to design a more compact heat exchanger when using CO_2 as working fluid.

It must be emphasized that the comparison above is much simplified, but it illustrates the potential for making compact and efficient heat exchangers for CO_2 as working fluid.

4.3.2 Expander

The physical dimensions of the expansion device are dominated by the gas density at its outlet. The CO2 based power cycle has a high condensation pressure, as a result the density of the gas after expansion is larger than for usual ORC solutions.

Milora ad al. developed a correlation for a relative measure of the total turbine exhaust area (the turbine factor). Table 2 shows the values of this turbine size factor for several working fluids. CO2 has a very low size factor, which means that very compact expanders can be built.

fluid	water	R11	R12	CO2
turbine size factor	17	5	0.8	0.08

Table 2: Turbine size factor for several working fluids.

CO_2 expander is not an off the shelf technology, but some companies have developed expansion machinery for cooling applications. It is belived that the same concepts and machines can be adapted for application in work recovery applications.

5 CONCLUSION

CO_2 based electricity production for maritime applications have been investigated.

Performance of such a power cycle equals conventional ORC solutions for high temperature waste heat (exhaust gas). Performances for low temperature waste heat are significantly improved (25%) with a CO2 technology. Energy (fuel) savings up to 10% can be achieved.

It was also showed that CO2 technology has a potential for size reduction.

The pre-project confirmed that CO2 based electricity production for maritime application is technically a promising solution. It was also showed that use of such technology would lead to important savings.

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E. Næss. ENOVA seminar: energi effektivisering og energigjenvinning i aluminiumverk, med hovedvekt på kraftproduksjon fra spillevarme. 2007/06/13

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