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WASTE HEAT RECOVERY ON DREDGING VESSELS

Limiting global warming requires the maritime sector to transition to a more efficient and sustainable operation. Reducing greenhouse gas (GHG) emissions, such as carbon dioxide and methane is vital to limit the global temperature rise (IPCC, 2021). Several legislative initiatives are in effect or are being discussed, including the IMO GHG strategy and the FuelEU Maritime initiative. This article discusses the potential of waste heat recovery (WHR) technologies to reduce the fuel consumption of dredging vessels. Available WHR technologies are compared based on working principle and operational performance for different types and ratings of internal combustion engines.

Waste heat recovery for marine application

Dredging and offshore vessels are generally powered by diesel or dual-fuel engines. These engines are used as they are robust and have a high power density. However, during the fuel combustion process, these engines also emit greenhouse gases and harmful emissions. Additionally, a significant amount of energy is rejected in the form of heat during their operation and not used for power generation.

Engine manufacturers optimise their engine design to the improve engine efficiency (less waste heat) and reduce emissions. However, the engine efficiency improvement is limited by the Carnot theorem. This limits the efficiency improvement of the in-cylinder processes and reducing the heat transfer to the cooling water is challenging due to the almost adiabatic in-cylinder process (Klimstra, 2006). Internal combustion engines generally waste 50% or more of the valuable fuel energy as heat through the exhaust gas, cooling water loops (both low temperature and high temperature) and radiation.

Effective use of the waste heat flows helps to improve the overall system efficiency and reduces the environmental impact of marine diesel/dual-fuel engines. Generally, this heat is recovered from the exhaust gas and/or cooling water with an economiser and thermal oil system. The recovered heat may be used to heat the vessel's accommodation and enclosed working areas, as heating for the fresh water maker and to heat the fuel in the bunker tanks if needed. Steam as heat transfer medium is often only used in case of larger heat demands. It is used aboard cruise vessels for the heating, ventilation and air-conditioning (HVAC) system, swimming pool, galley, laundry services, or for ships operating in the Arctic region to de-ice the ship. Steam is also used for turbine-driven cargo pumps in case of LNG carriers.

The energy transition results in a change to more sustainable cleaner fuels. These fuels

may not require heating or a much smaller amount than the traditionally used heavy fuel oil (HFO). Therefore, this heat may be used to increase the system efficiency by using a waste heat recovery system. Waste heat can be converted it into electric power, which in turn reduces the vessel's fuel oil consumption, GHG emissions and harmful emissions.
The most suitable WHR technology depends on the waste heat quality (i.e. temperature). The second law of thermodynamics states that energy is not only defined by its available quantity but also by its quality. The quality of energy is expressed using the thermodynamic concept of exergy (Moran, 2010). Singh and Pedersen (2016) consider the exhaust gas as the waste heat flow with the highest energy recovery potential, due to its high temperature and mass flow. Approximately 75% of the total waste heat of a marine diesel engine that can be converted into work is in the exhaust gas.

The literature discussed in this article focuses on maritime application of WHR systems. Other industries, such as the automotive and the chemical industry have performed considerable research on the topic with advanced cycles, which may become relevant in the future. However, these will not be discussed in this article. Singh and Pedersen (2016) provide a comprehensive overview of waste heat recovery system technologies suitable for maritime applications. They have a focus on three options: thermoelectric generators, turbocompounding and bottoming cycles.

Thermoelectric generators are solid-state devices and utilise advanced materials to convert heat directly into electricity through the Seebeck effect (Uyanık et al., 2022). Despite the potential benefits of this solid-state technology, such as its simplicity and reliability, it has not yet been placed on the market due to its low efficiency (about 2%).

Given the impossibility of completely avoiding waste heat, we should make the most of it.

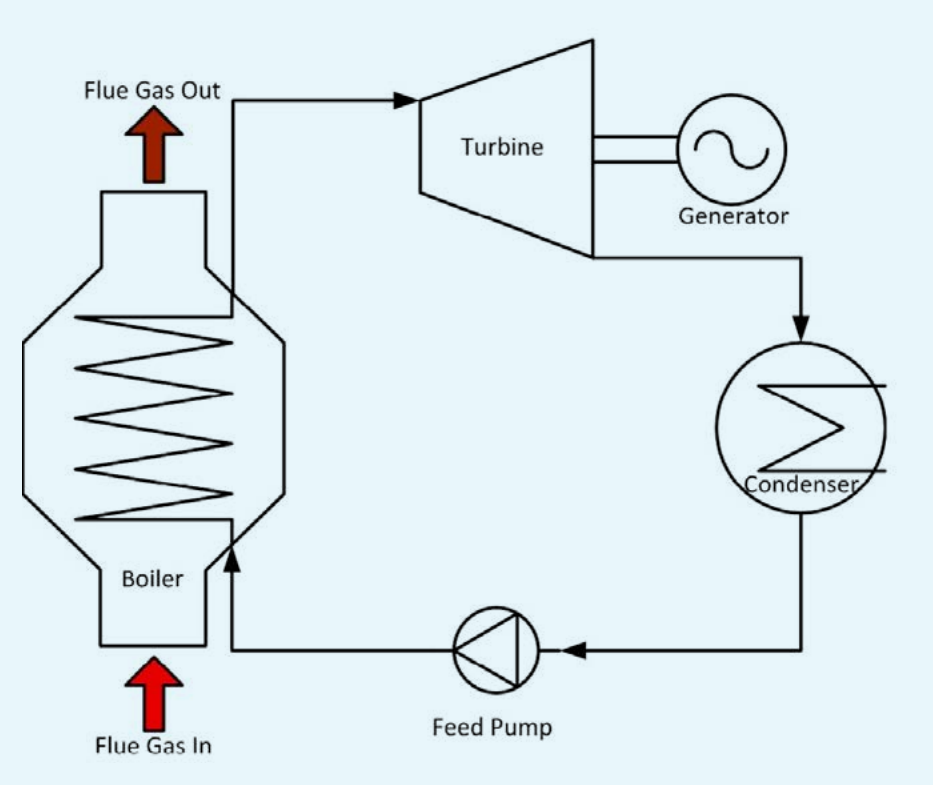


FIGURE 1

Simple Rankine cycle (Singh and Pedersen, 2016).

Turbocompounding is a broad concept related to using a turbine with generator anywhere in the exhaust piping to recover energy of blow-down exhaust gas. Aghaali and Ångström (2015) performed an in-depth review of various turbocompounding configurations including high to low pressure and mechanical to electrical options. Onshore applications have demonstrated fuel improvements of 5-10%. Hybrid electric turbocharging, meaning integrating an electrical machine into the turbocharger is one approach to implementing turbocompounding. This allows the turbocharger to serve as a generator. Westhoeve (2018) showed that a hybrid electric turbocharger for a medium-speed diesel engine can only achieve a fuel reduction of 1.3%. This is because an increase in backpressure can negatively impact the in-cylinder scavenging process, resulting in a poorly operating turbocharger and potentially excessive thermal loading on the engine.

Heat recovery can also be achieved through the implementation of bottoming thermodynamic cycles. Two types of Rankine systems are commonly used for waste heat recovery, namely the organic Rankine cycle (ORC) that uses an organic fluid and the steam Rankine cycle (SRC) that uses water/steam. These Rankine cycles utilise heat to evaporate (and superheat) a working fluid. This is then expanded through an expander device such as a steam turbine or screw expander, to generate work. The working fluid is then condensed by a cooling medium such as seawater and its pressure is increased by a circulation pump. The most basic layout of such a cycle consists of a boiler, a steam turbine, a condenser and a feed water pump. Figure 1 presents an overview of a Rankine cycle.

The main advantage of bottoming cycles over turbocompounding is the negligible impact on engine performance as it is a separate system. This means that a bottoming cycle system will not affect the engine, while for a turbocompounding it may result in an increased thermal loading of the engine. The impact of the bottoming cycle is limited to its effect on the engine's backpressure. The backpressure of the total exhaust gas system (including silencer, after treatment, etc.) should be within the specified engine limits. The use of an economiser/boiler for the bottoming cycle, a heat exchanger in the exhaust system, results in lower backpressure increase than the use of additional turbines in the exhaust piping as in the case of turbocompounding. (Aghaali and Ångström, 2015).

Mondejar et al. (2018) conducted a thorough review of ORC power systems for maritime use, analysing three application cases: a container ship, a bulk carrier and an oil tanker. However, all vessels were equipped with a two-stroke diesel engine, which is currently not used in new-build dredging vessels. The case studies used the jacket water heat for preheating the working fluid and exhaust gas heat for evaporating the working fluid. Fuel savings of about 6-10% were achieved with the ORC technology for all three vessel types (Mondejar et al., 2018).

Sturm and Banck (2016) analysed a 309 kW ORC system with a Caterpillar MaK 8M46DF engine. This system used a saturated steam system as an intermittent heat transfer loop. Fuel savings of 4-8% were achieved at various engine loads in gas mode and 4-7% in diesel mode due to reduced exergy in the exhaust flow. The maximum power output of the ORC system was achieved with a 65% engine load, but no simulations were conducted for a larger ORC system. Their fuel savings and payback analysis suggested that one ORC unit for four 8M46DF engines would result in a fuel saving of 1.4% and a payback period ranging from 2.1 to 11.3 years depending on fuel price and scenario.

Detailed analyses of waste heat recovery potential for specific cases are reported in literature. There is a relative lack of discussion on simple SRC systems for waste heat recovery aboard ships, as most of the research in this field tends to focus on ORC systems for maritime applications. Moreover, there are no studies that have compared ORC and SRC technologies for various marine engine types. These studies have also not investigated the impact of the varying load conditions present on dredging vessels, but only investigated the more stable load of cargo vessels. Dredging does not continuously require maximum engine loading. Thus, a comparative analysis of the ORC and SRC bottoming power cycles from various marine engine types and loads is relevant.

Comparison of steam Rankine cycle with organic Rankine cycle

The main difference between the steam Rankine cycle (SRC) and organic Rankine cycle (ORC) is the working fluid. ORCs use organic fluids with lower boiling points, such as refrigerants or hydrocarbons, while SRCs use water as the working fluid.

The selection of the working fluid influences the system performance, system layout and operational parameters. The temperature of the waste heat flow determines the working fluid and the operating conditions for the waste heat cycle. One of the key factors is that the temperature of the waste heat should exceed the boiling point of the working fluid but to what extent depends on the type of fluid. The working fluids may be categorised as dry, wet and isentropic fluids based on their saturation vapour curve. This curve defines if the working fluid remains in vapour phase (dry) or condensates during expansion (wet). Water is a wet fluid and requires superheating to prevent condensation during the expansion process in the steam turbine.

Table 1 shows the various parameters in which organic working fluids differ from water. The two dry fluids pentafluoropropane (R245fa) and trans-1-Chloro-3,3,3-trifluoropropene (R1233zd(E)) are often used in ORC systems for maritime applications. These fluids have low boiling temperatures (15-20°C) at atmospheric pressure, compared to water (100°C) and lower critical pressures (pc) and temperatures (Tc) than water. In addition, these fluids have a low decomposition temperatures, which is problematic when direct heating of these fluids with exhaust gases. For this reason, ORC units use an intermediate thermal oil or hot water heating loop between the exhaust funnel and the ORC unit. The thermal oil or hot water loop are already in place in ships with bunker (fuel) heating and is heated to around 150-180°C. This heating loop allows the ORC system to be located in the most suitable place on the vessel.

ORC systems commonly use dry working fluids. Compared to wet fluids such as steam, dry fluids require a small amount of superheating to prevent condensation during the expansion process. The small difference between the specific volumes of the liquid and

TABLE 1

Comparison of R245fa, R1233zd(E) and water (Eyerer et al., 2019).

Parameters	R245fa	R1233zd(E)	Water
Chemical formula	$CF_3CH_3CF_2$	$CF_3CH{=}{=}CHCF_3$	H_2O
Critical pressure (PC) (bar)	36.5	35.7	217.8
Critical temperature (TC) (°C)	154.01	165.6	373.9
Boiling point (°C)	14.81	17.97	100
Decomposition temperature (°C)	~ 300	> 250	3000
Latent heat at boiling point (kJ/kg)	196.23	195.52	2260
Molar mass (g/mol)	134	130.5	18.02
Slope	Dry (positive)	Dry (positive)	Wet (negative)
Ozone Depletion Potential	0	0.00034	-
Global Warming Potential 100yr (CO2-eq)	1030	7	-
Atmospheric life-time	7.7 years	25 days	-
Flammability	Non-flammable	Non-flammable	Non-flammable
ASHRAE Std 34 safety class	B1	A1	-

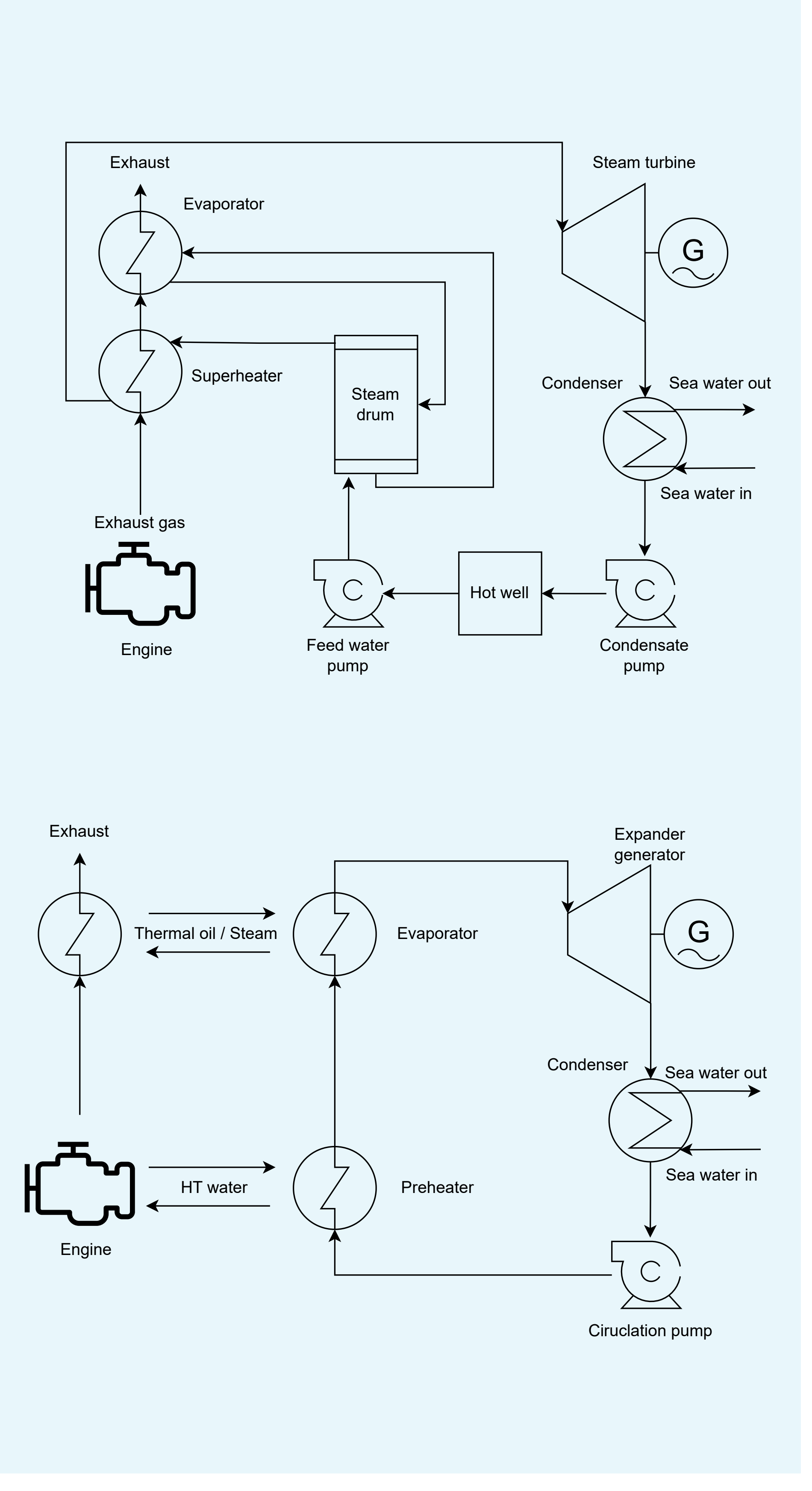


FIGURE 2

System overview of steam Rankine cycle (above) and organic Rankine cycle (below).

vapour phases allows for heating, evaporation, and superheating to take place in a single heat exchanger. This results in a relatively simple ORC configuration compared to the SRC (Mondejar et al., 2018).

In an SRC, water from the steam drum is not evaporated and superheated in one step in the evaporator. A mixture of saturated liquid and saturated steam is obtained by transferring heat from the exhaust gases to the water as it flows through the evaporator. The mixture then flows to the steam drum where it is separated into saturated water (about 75%) and saturated steam (about 25%). The saturated steam passes through the superheater and thereafter to the steam turbine. Superheating is necessary to prevent the condensation of water droplets on the high-speed steam turbine blades as this may cause damage. The steam drum also provides a source of saturated steam for other heating applications on the vessel. After being expanded in the steam turbine, the steam undergoes condensation with seawater in the condenser and is then pumped into the hot well. From there, it is pumped back into the steam drum.

Power generation with an expander, such as a turbine or screw expander, requires a working fluid mass flow and an enthalpy difference over the turbine. This is provided with expansion from a higher temperature and pressure towards a lower temperature and pressure. To achieve this, the working fluids are pressurised, increasing the saturation temperature to match the waste heat temperature. In an ORC unit, the circulation pump pressurises the working fluid to over 20 bar (g), which requires 10-15% of the expander power output. A maritime SRC uses a steam drum pressure of 7 bar (g) for which the feed water pump consumes less than 1% of the steam turbine power output. Most energy is extracted from the working fluid expansion if it condenses just above the temperature of the cooling medium (usually seawater). For the SRC with water a vacuum condenser is required to lower the condensation temperature from 100°C to around 35°C, while a condenser pressure of approximately 1 bar (g) is required for the ORC system to raise the condensation temperature from 15°C to around 35°C.

The Carnot cycle efficiency is often used in studies to represent the theoretical efficiency of a heat engine. However, this calculation is based on assumptions of an infinitely large hot source and cold sink, which are not realistic in practice. The exhaust gases cool down 100-150°C throughout the superheater and evaporator, indicating that the assumption of an infinite large hot source is invalid (DiPippo, 2007; Sturm and Banck, 2016).

The maximum theoretical thermal efficiency is determined with the triangular cycle efficiency of Equation 1:

$$\eta_{th} = \frac{T_{hot} - T_{cold}}{T_{hot} + T_{cold}} \quad [1]$$

where T_{hot} is the temperature of the hot source and T_{cold} the temperature of the cold sink in Kelvin. The maximum thermal efficiency is determined for both the SRC and ORC with an assumed pinch point of 10°C between the hot and cold sides of a heat exchanger. The SRC achieves a maximum thermal efficiency of 31% with an exhaust gas temperature of 285°C and a cooling water temperature of 20°C. While for the ORC, a maximum thermal efficiency of 21% is found with a working fluid temperature of 140°C and a cooling water temperature of 20°C. The SRC has a greater potential for heat recovery and higher thermal efficiency, because the ORC requires an intermediate loop (as shown in Figure 2) to prevent the working fluid from decomposing.

Larsen et al. (2013) state that dry hydrocarbons, such as cyclohexane, toluene and benzene gives the highest efficiency compared to other ORC working fluids as these can be heated directly with exhaust gas heat. However, the selection of the working fluid must also meet environmental and maritime safety requirements, especially in terms of flashpoint, auto-ignition temperature, toxicity and compliance with SOLAS Ch II-2 regulations. Although hydrocarbons have excellent performance characteristics, they have a flashpoint below the prescribed lower limit of 60°C by the SOLAS regulations for fluids inside the engine room. This makes them unsuitable for use in machinery spaces of Category A unless double-walled piping are used and risk assessments are performed. Due to the fire hazard issue of hydrocarbons, Larsen et al. (2013) recommend R245fa as the optimal working fluid candidate despite its high global warming potential of 1030 over 100 years (GWP100yr). It has recently been suggested that some working fluids will be banned due to their high GWP. R1233zd(E) with a low GWP100yr of 7 is proposed as a drop-in replacement for R245fa. In the maritime market, all standalone ORC systems use non-hydrocarbon working fluids such as the two hydrofluorocarbons presented in Table 1. This allows for the installation of these modular ORC units in the engine room without compromising safety (Mondejar et al., 2018).

Steam has other safety risks such as flashing: the pressurised water turns into steam when expanding due to a leakage or when tapped off to a low pressure pipe. If a saturated water flow suddenly expands from 7 bar (g) to atmospheric pressure, the condensate immediately changes state to steam and expands 1,500 times. In condensate lines, there is the risk of flashing, giving potential shocks due to the large volumetric change of the condensate to steam. This is typically mitigated by cooling down the pressurised condensate below the boiling point corresponding to the lower pressure level, although this is not possible with unforeseen leakages.

A modelling approach is required to effectively handle the interacting components and thermodynamics on the system's performance.

Methods

A simulation study is used to evaluate the operational performance of the ORC and SRC for dredging vessels. For this study, two steady-state models were developed in Matlab Simulink. The specific working fluid conditions for each node in the ORC cycle were obtained by using a thermodynamic library "Coolprop version 6.4.1", while for the SRC, steam tables were accessed by means of "X Steam", a thermodynamic properties library of water and steam based on IAPWS IF-97 formulation.

The developed SRC model is calibrated with performance data provided by the supplier of the SCR technology for a built working vessel. The ORC model is calibrated to represent a standardised module with data obtained from an ORC unit supplier. Verification is performed through parameter sweep studies. Validation of the models is limited to comparing the performance predictions with those provided by the supplier for other conditions than the data used for calibration.

The SRC and ORC models comprise of different types of components. Several components of both systems depicted in Figure 2 can be modelled similarly for the SRC and the ORC. The SRC contains more components, such as an evaporator, steam drum and superheater, which increases the complexity of this model.

The net efficiency (η_{net}) of both cycles is defined as the net electrical power output of the system (P_{net}) divided by the total heat input (Q_{in}) (Equation 2). For the ORC, this is the hot water heat transferred into the preheater (Q_{HT}) and the thermal oil heat (Q_{TO}) into the evaporator. For the SRC, this is the exhaust heat transferred into the evaporator (Q_{evap}) and the superheater ($Q_{superheater}$). The power demand for the Balance of Plant (P_{BoP}) must be subtracted from the generated power ($P_{generator}$) and consists of the circulation pump and sea water pumps for the WHR systems.

$$\eta_{net} = \frac{P_{net}}{Q_{in}} = \frac{P_{generator} - P_{BoP}}{Q_{HT} + Q_{TO}}\ for\ ORC, \quad = \frac{P_{generator} - P_{BoP}}{Q_{evap} + Q_{superheater}}\ for\ SRC \quad [2]$$

Both cycles start with a pump model with the assumptions that the pumps pressurise an incompressible substance and that the pump has a constant pressure, which (neglects kinetic and potential energy changes. The required pump power (P_{pump}) includes the electrical generator losses (η_{gen}) and mechanical losses (η_{mech}) for which constant efficiency rates are assumed for the smaller consumers, the circulation pumps. The power required to operate the pump (P_{pump}) is calculated using Equation 3:

$$P_{pump} = \frac{\dot{m}_{cond} \cdot (h_{out} - h_{in})}{\eta_{is} \cdot \eta_{mech} \cdot \eta_{gen}} \quad [3]$$

The enthalpy of the fluid leaving the pump is determined assuming constant entropy and only pressure changes. The isentropic efficiency (η_{is}) is used as ratio between isentropic power and actual pump power. The pump model includes a head-speed fit on the performance curve to include variable speed operation and the backbone characteristic of a centrifugal pump/ compressor is used to define the efficiency at lower pump speeds.

The steam turbine/screw expander power output (P_{turb}) is calculated with Equation 4 and 5 using an enthalpy balance, which includes the superheated working fluid mass flow $(\dot{m}_{sh})$ and the expander efficiency (η_{tur}). The mechanical efficiency (η_{mech}) is assumed to be constant, while the isentropic efficiency (η_{is}) is determined by fitting performance data from the SRC supplier. A constant isentropic efficiency is assumed for the ORC, which is valid for a screw expander (Mondejar, 2018).

$$P_{turb} = \dot{m}_{sh} \cdot \eta_{tur} \cdot (h_{in} - h_{out}) \quad [4]$$

$$\eta_{tur} = \eta_{is} \cdot \eta_{mech} \quad [5]$$

Many heat exchanger elements are present in both models: the evaporator and condenser of both the SRC and ORC system, as well as the preheater (ORC), thermal oil boiler (ORC) and superheater (SRC). These heat exchangers are modelled as logarithmic mean temperature difference (LMTD) elements. The amount of heat (Equation 6) is proportional to the medium mass flow $(\dot{m})$, the temperature difference and the specific heat of the medium (c_p).

$$Q = \dot{m}_{hot} \cdot c_{p,hot} \cdot (T_{hot,in} - T_{hot,out}) = \dot{m}_{cold} \cdot c_{p,cold} \cdot (T_{cold,out} - T_{cold,in}) \quad [6]$$

The steam drum has two-phases (liquid and vapour) and is modelled as a two-phase energy balance with saturated steam and saturated water in equilibrium. Changing the pressure in the steam drum results in a very rapid release or adsorption of energy to/from the steam and water. The fast dynamics enables the use of low order models to represent the steam drum processes (Astrom and Bell, 2000). The drum pressure response is captured for a varying input of heat, feed water flow rate, the enthalpy balance of the feed water and steam drum water, and steam flow rate with the condensation enthalpy.

The available waste heat on dredging vessels varies during operation due to the engine load variations. Proportional integral (PI) feedback controllers are used to regulate the circulation pumps of the working fluids in both the SRC and ORC cycles. The evaporation temperature is kept constant by adjusting the working fluid mass flow rate of the circulation pump.

To prevent wet soot deposition in the exhaust, the exhaust gas temperature leaving the heat exchanger is maintained above 180°C. However, reducing the mass flow on the cold side of the heat exchanger to achieve this could lead to excessively high temperatures. An exhaust gas bypass valve similar to those found in thermal oil boilers is required. The working fluid mass flow can be varied with the circulation pump such that evaporation conditions can be reached.

Further derivation of the equations and model development can be found in Westhoeve et al. (2022). This paper provides more details on the mathematical models and their assumptions as used in this study.

Results

This section shows the performance predictions obtained with the ORC and SRC models. The results of this study are presented in a generic way, based on four engine load points similar to the product guides provided by engine manufacturers. This allows for a general overview of the results and for designers to use the results for early-stage design estimates of dredging vessels. The operational profile of the vessel is important for the viability of the ORC/SRC system, but a specific operational profile has not been selected for this study as it varies for each project.

The net power output, thermal efficiency and fuel savings are compared for one large medium-speed engine MAN 6L48/60CR rated at 7200 kW. The engine operate as diesel-electric by driving a generator. Another engine scenario with a medium-speed two-stage turbocharged Wärtsilä 31 can be found in Westhoeve et al. (2022).

Operational performance of ORC and SRC

The SRC provides a 30-40% higher net power output for the MAN 6L48/60CR engine compared to the ORC, depending on the engine load (Figure 3). The maximum net power output for the SRC is 364 kW while for the ORC it is 247 kW. However, for dredging vessels with a wide range of engine loads during a dredging cycle, the recovered power decreases towards mid-load operation. At mid-load, the net power output for the SRC is reduced by 40% from 364 kW to 216 kW, and for the ORC by 46% from 247 kW to 133 kW. The difference in power recovered by the ORC and SRC can be attributed to the temperature of their superheated working fluids. To transfer heat to the ORC, an intermediate thermal oil loop is used, which utilises a temperature of 170°C to evaporate the R245fa working fluid at 140°C in the evaporator. This results in a considerable amount of exergy loss due to the use of a working fluid with a relatively low evaporation temperature and the heat transfer that occurs in the intermediate loop. In contrast, the SRC uses steam that is evaporated at approximately 170°C (at 7 bar (g)) and then further superheated to approximately 285°C by the superheater.

Thermal efficiency and fuel saving of ORC and SRC

The average thermal efficiency of the ORC and SRC are 8% and 16% respectively as shown in Figure 4. These values represent the percentage of heat that enters the cycle and is converted into net electrical power output.

The superior power output with steam compared to organic fluids is attributed to its higher superheated temperature.

FIGURE 3

Net power output of SRC and ORC for MAN 6L48/60CR.

FIGURE 4

Thermal efficiency and fuel saving of ORC and SRC with MAN 6L48/60CR.

The fuel saving is in the order of 5-7% with the SRC and 3-4% with the ORC. The energy that is recovered is directed towards the board net in the form of electricity. By assuming that the recovered energy is not supplied by the diesel generator, the amount of fuel saved can be determined. It is not expected that there will be significant change in the specific fuel oil consumption of the engine due to the implementation of WHR because the generated power represent about 5% of the maximum continuous rating of the engine.

Economic feasibility of ORC and SRC

The business case is assessed for the ORC and SRC assuming the dredging vessel has 6,500 operational hours a year. Three fuel price levels are considered: €500/tonne, €750/tonne and €1,500/tonne representing cheap fuel, business-as-usual and expensive (bio)fuel. In assessing the business case, one large steam system is evaluated that is matched with the maximum continuous rating of the MAN 6L48/60CR engine. For the ORC, multiple modules are placed in parallel to match with the engine. Again, to avoid any debates regarding the operational profile, the financial outcomes are presented as a function of engine loading for either two or three primary engines online. No maintenance costs or any additional crew costs due to maintenance are included.

The steam system is relative expensive and may not yield a reasonable payback period solely through fuel oil savings depending on the fuel price and any future emission taxing. For expensive fuel of €1,500/tonne, the payback period ranges between 3 and 5 years depending on average engine load. For moderate fuel price levels of €750/tonne, in the range of the current low sulphur marine gasoil (LSMGO) bunker prices, the payback period increased to 6-11 years. For very cheap fuels of €500/tonne, the payback period increases to an unacceptable 10-16 years. For the ORC, the payback periods are relatively short compared to the SRC, ranging from 1-3 years for expensive fuels, 3-5 years for moderate fuel prices and 4-7 years for very cheap fuel.

The Net Present Value (NPV) may be used to determine the profitability of an ORC or SRC unit. This method involves estimating the difference between the present value of cash inflows and the present value of cash outflows over time. The Internal Rate of Return (IRR) is used as a tool to evaluate the profitability of this investment based on NPV = 0 and

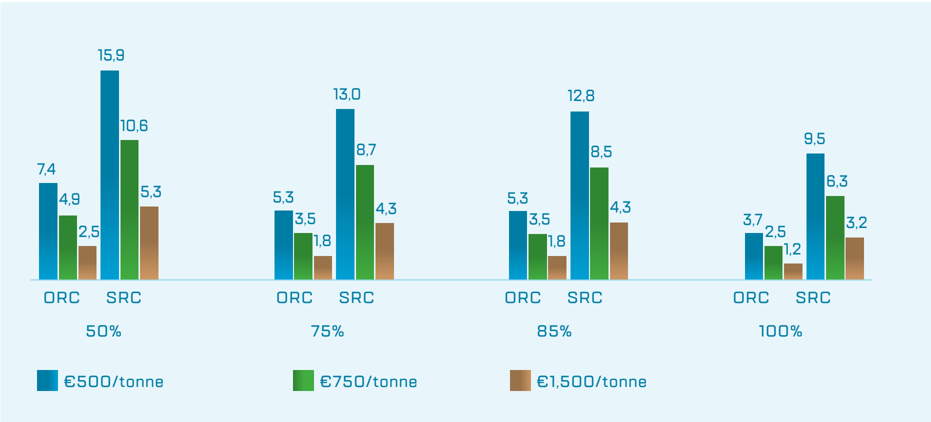


FIGURE 5

Payback period of ORC and SRC with MAN 6L48/60CR.

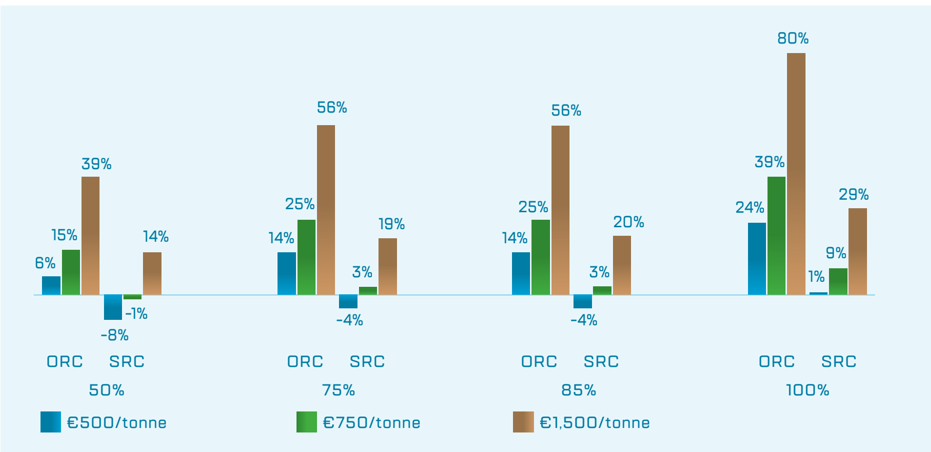


FIGURE 6

Internal Rate of Return for 10 years with ORC and SRC with MAN 6L48/60CR.

determining the discount rate over a timeframe of 10 years. Ultimately, the decision on whether to invest in the ORC or SRC unit will depend on the individual circumstances and goals of the investor. Figure 6 shows no profitable scenario for the SRC system and cheap fuel. For moderate fuel price levels, the discount rate ranges between -1 and 9% depending on engine load. Expensive fuels results in a profitable case for the SCR with a discount rate of 14-29%.

The ORC has a significant higher profitability level than the SRC. A discount rate of 6-24% is found for cheap fuels, 15-39% for moderate fuel price levels and 39-80% for expensive fuels.

Comparison of fuel savings with an ORC and different engine platforms

The waste heat recovery (WHR) potential of the ORC has been evaluated for six different engines operating at a constant speed application for generator sets loaded at 75%. This load condition is chosen because generator sets rarely operate at 100% load. The maximum net power output of each ORC unit is 100 kW and the number of units required depends on the engine size and type of fuel used, as shown in Table 2.

The fuel saving potential is mainly influenced by the exhaust gas temperature, with engines having higher exhaust gas temperatures showing higher fuel savings of 4-5%.

TABLE 2

Engine specification and peak ORC recovery potential for 75% engine loading.

Engine load: 75%	Caterpillar	Himsen	Wärtsilä			MAN	MTU
Type	3512C	6H25/33HR	12V31	9L46DF		6L48/60CR	12V4000
Fuel	MGO	MGO	MGO	MGO	LNG	MGO	Gas
Max power rating (kW)	1784	1890	7080	10305		7200	1492
Nominal speed (rpm)	1800	900	720	600		500	1600
Actual power rating (kW)	1338	1418	5310	7729		5400	1119
Exh. mass flow (kg/s)	1.9	2.6	9.8	17.2	12.5	12.1	1.6
Exh. temperature (°C)	445	345	288	304	407	295	453
Number of ORC units	1	1	2	2**	3	3	1
Gross power of all ORC units excluding BoP (kW)*	75	64	168	271	397	215	65
Net total power ORC (ekW)	60	50	136	228	334	170	51
Net thermal efficiency (%)	8.0	7.9	8.1	8.8	8.8	8.0	7.9
Fuel saving (%)	4.5	3.5	2.6	3.0	4.3	3.2	4.6

* BoP: Balance of Plant.

** Third ORC unit switched-off to reduce BoP power demand.

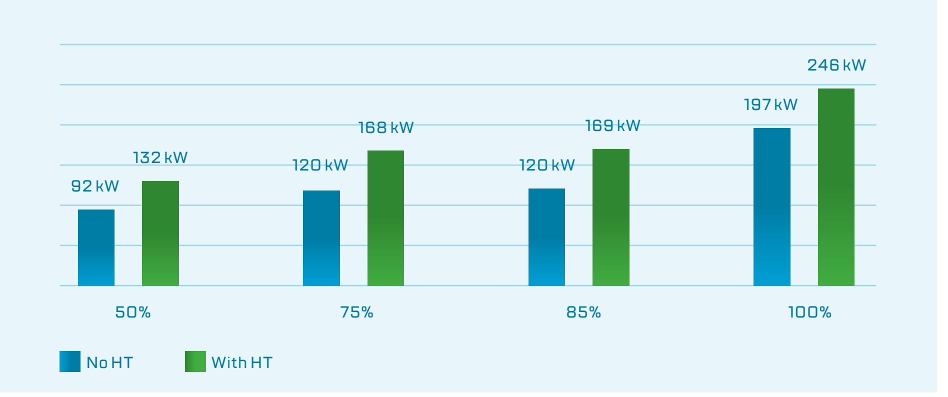


FIGURE 7

Net power of the ORC units with/without preheater for the MAN 6L48/60CR.

FIGURE 8

Thermal efficiency and fuel oil saving of the ORC with/without preheater for MAN 6L48/60CR.

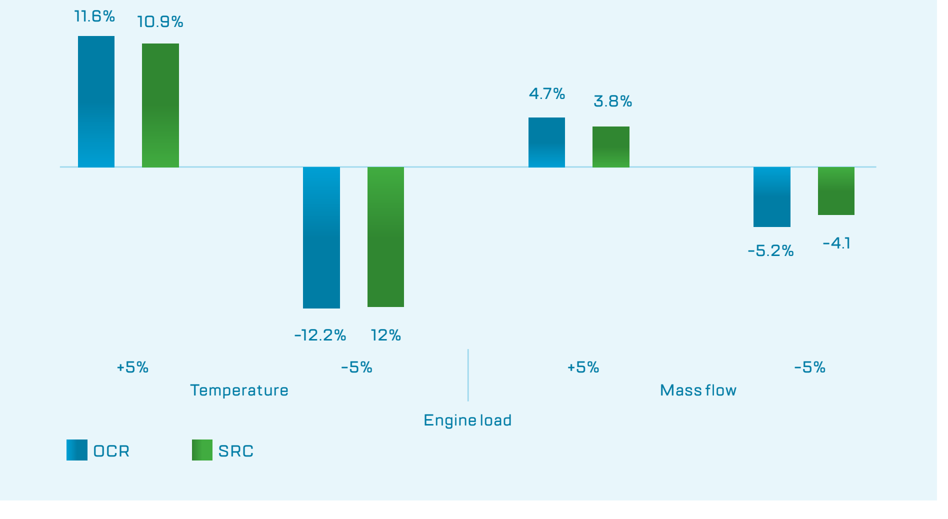


FIGURE 9

Sensitivity of MAN 6L48/60CR exhaust gas temperature and mass flow on WHR power output.

Consideration of WHR systems becomes particularly relevant for vessels equipped with high-speed or gas engines.

For engines with lower exhaust gas temperatures, the maximum fuel savings percentage is lower. The Wärtsilä 12V31 engine has lower exhaust gas temperatures due to two-stage turbocharging and therefore only achieves a fuel saving of 2.6% at 75% load. While the Wärtsilä 9L46DF engine has a higher fuel saving potential of 4.3% in dual-fuel mode running on gas compared to 3% in diesel mode due to the temperature difference of the exhaust gas flow even with a reduced exhaust mass flow.

ORC performance with/without hot water preheater

Some maritime ORC systems can utilise the main engine cooling water loop for preheating of the working fluid, while others solely rely on steam or thermal oil (160-180°C) to heat and evaporate the working fluid.

Systems without a preheater result in a 20-30% reduction in net ORC power output (Figure 7) and fuel saving potential from 3.7-4.0% to 2.1-3.0% (Figure 8). The thermal efficiency increase with preheating is low due to the low exergy (potential work) of the high-temperature (80-90°C) cooling water flow of the marine diesel engine, which is less efficient than the temperature level of the thermal oil flow (180°C).

Sensitivity of WHR performance with varying exhaust gas conditions

A sensitivity analysis was conducted to assess the impact of varying in operating conditions. The largest impact was found for the exhaust gas temperature and mass flow on the performance of the WHR system. The MAN 6L48/60CR medium-speed engine at 85% load is used as case study. Results show that exhaust gas temperature has the greatest impact, with a 5% increase or decrease resulting in a 10-12% change in power output. The exhaust mass flow has a lower sensitivity with only a 4-5% impact.

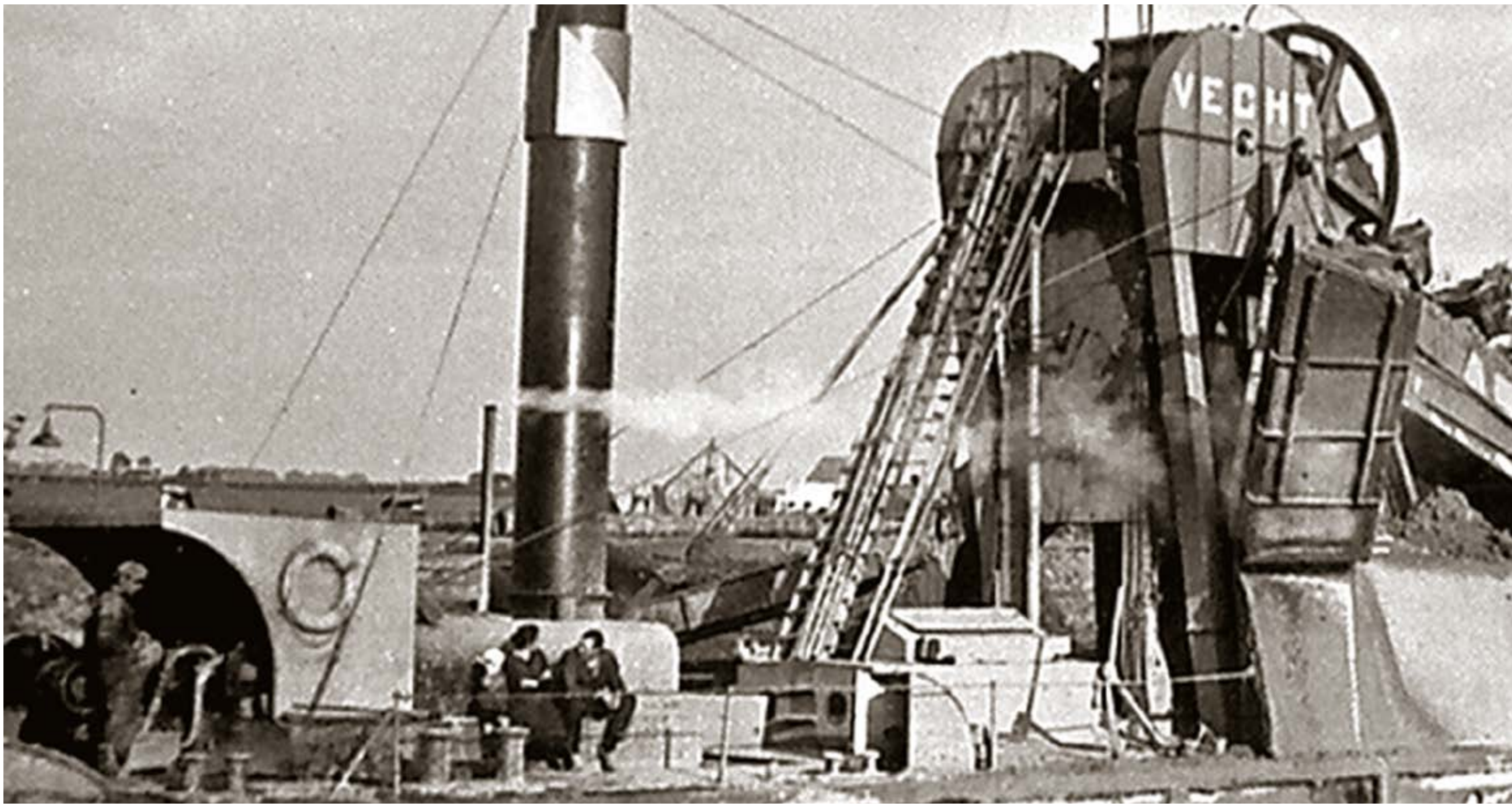


FIGURE 10

Dredging into a new era: 1861 marks the first steam powered bucket dredger in the Netherlands. Photo © Baars B.V.

Discussion

This study has found fuel savings and thermal efficiency for the ORC of 2-4% and 8% respectively. For the SRC, fuel savings of 4-6% and thermal efficiency of 16-17% are found. The SRC system has higher fuel savings and thermal efficiency, this is similar to what is found in literature. Although SRC related performance results as waste heat recovery are not widely available in literature compared to the considerable amount of ORC research for maritime application.

> **The operational profile of the dredging vessel is key for assessing the feasibility of waste heat recovery systems.**

For the ORC, Sturm and Banck (2016) found a higher fuel savings of 4-7% in diesel mode and 4-8% in gas mode with a thermal efficiency of 14%. The higher fuel savings and thermal efficiency of Sturm and Banck (2016) may be explained by several factors. While Sturm and Banck (2016) also used R245fa as their working fluid, they used a higher evaporation pressure of 33 bar (a) and temperature of 150°C versus the 21 bar (a) and 140°C used in this study. Sturm and Banck (2016) also made use of a turbine with an assumed constant efficiency of 82%. A constant efficiency for a turbine is not valid for the off-design operation. The ORC results of this article used a screw expander with a relative low, but constant efficiency of 70% (also for off-design conditions).

Casisi et al. (2020) performed simulations for an ORC system operating at higher temperatures with toluene as a working fluid resulting in higher fuel savings and a higher thermal efficiency than the ORC operating at lower temperatures. The study used an intermediate thermal oil loop (between the exhaust gas and the ORC system) with a temperature of 350°C. Toluene was used as the working fluid with an evaporation temperature of 250°C. They found fuel savings of 5-6% and thermal efficiency of 22-23% for a simple ORC coupled to a Wärtsilä 6L50DF operating in gas mode. This study also investigated the preheated cycle, resulting in a fuel savings of 6-8% and a thermal efficiency of 22-24% depending on the cycle layout. An ORC system with a regeneration cycle resulted in a thermal efficiency of 26-27% and 6-7% fuel savings. These results are supported by the Carnot theorem, which allows for higher efficiencies when the temperature difference between the heat and cold source increases. Onshore SRC systems may achieve 33-47% thermal efficiency by adding complexity to the system with reheaters, regenerators and supercritical cycles (Vatopoulos et al., 2012). However, an important factor to consider on board vessels is the safety issues with using hydrocarbons at high pressure and temperature as working fluid and the space such a system would use on board.

This study used quasi-steady state models to evaluate the WHR cycles in which operation over time is represented by a succession of

Steam cycle based systems will return on (new-build) dredging vessels as waste heat recovery systems to reduce the vessel's impact on the environment.

steady states. This method neglects any time dependent dynamics or transients, such as energy and mass accumulation. Mondejar et al. (2016) use this approach to simulate the ORC performance for the round trip of a passenger ferry on a minute-by-minute basis. They also state that this same approach is used by others studies in the subject. In case dynamic WHR system performance is required, Grimmelius et al. (2010) shows how to build such a model using resistance and volume elements, finite volumes for the heat exchangers and pseudo-steady state models for the pump and turbine.

Conclusions

In this study, two waste heat recovery technologies are compared by means of a steady-state models representing an ORC and an SRC. The application of waste heat recovery systems on dredging vessels resulted in fuel savings of 2-4% for organic Rankine cycle systems and 4-6% for steam Rankine cycle systems for the cases evaluated in this study. This difference between both systems is a result of the net thermal efficiency, which is 8% for the ORC and 16-17% for the SRC. The reduced efficiency of the ORC system can be attributed to the lower maximum temperature of the chosen working fluid, which is selected for safety reasons on board vessels, among other factors.

This study found that the fuel savings, which may be obtained by the application of a WHR system on-board dredging vessels depend on many factors, such as the engine design and size, the fuel properties, the environmental conditions and the operational profile of the vessel. Compared to medium-speed engines, high-speed engines, with inherently lower engine efficiency, have greater potential for recovering waste heat due to the higher temperature of their exhaust gas flow. It was observed that the WHR performance is significantly affected by changes in exhaust gas temperature, while variations in exhaust gas mass flow have a relatively proportional effect. Because of all the dependencies discussed, the design of WHR systems should be considered during the early design process to achieve an economically and technically feasible system with high fuel savings.

The operational profile of the vessel has a large impact on the fuel saving potential and the economic feasibility of a waste heat recovery system. The WHR system size should be based on the operational profile to get the highest fuel savings, emission reduction and/or economic feasibility, not on the maximum power of the vessel. Operating the WHR in off-design loads will reduce the performance of the system in part-load due to reduced efficiency of the systems. This is particularly important for steam-based systems, in which the turbine efficiency reduces significantly at lower loads. Therefore, a smaller optimised WHR system may result in higher overall fuel savings than a larger system when part-load operation is considered.

The economic feasibility of the WHR systems depend on among others, the investment costs and the fuel expenses saved during the operation. This study found payback periods of 1-7 years for the ORC systems and 3-16 years for SRC systems depending on the operational profile and used fuel price scenario. In the future, emission taxes may provide additional economic benefits (i.e. lower emission costs) for vessels with WHR systems due to the lower fuel consumption and emissions.

Summary

The maritime energy transition will have a profound effect on the design of dredging vessels in the coming years. Dredging vessels are currently powered by internal combustion engines (ICEs), which have a significant energy loss through the heat in the exhaust gas. Waste heat recovery (WHR) systems may reduce the fuel consumption and improve the energy efficiency of vessels by converting this heat to additional electrical power.

This article investigates the effect of applying state of the art WHR systems on the fuel consumption and the total system efficiency of dredging vessels. The study compares the steam Rankine cycle (SRC) and the organic Rankine cycle (ORC). To achieve this, simplified steady-state models have been developed for both systems.

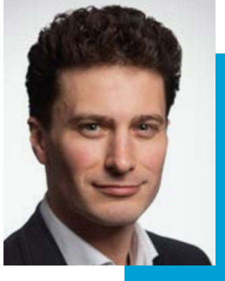
The fuel savings and energy consumption reduction have been determined for medium-speed and high-speed diesel engines and for dual-fuel (DF) and spark-ignited (SI) gas engines. The results show fuel savings of 2% to 6% depending on several factors. The economic feasibility depends on fuel price and engine loading, but lays between 1-7 years for the ORC and 3-16 years for the SRC. The exhaust gas temperature has a large influence on the recovery potential and a higher temperature results in a higher fuel savings by the WHR system.

Jan Westhoeve

Jan is a sustainability engineer at Van Oord, where he develops and implements sustainable energy solutions within the company's fleet. Before joining the company in 2023, he worked as a research and development engineer at Royal IHC, focusing on the design of marine energy systems. In 2018, Jan gained a Master's in mechanical engineering, with a specialisation in marine engineering, from Delft University of Technology in the Netherlands. With his expertise in sustainability and marine energy systems, Jan is dedicated to contributing to sustainable work vessels for the maritime industry.

Benny Mestemaker

Benny is a senior specialist at Royal IHC. He has been with the company since 2009 and his focus is on developing knowledge regarding the use of alternative fuels and the application of new drive systems. He is part of the CEDA Commission on Decarbonisation and the chair of the CIMAC NMA NL. In 2011, Benny obtained his Master's in mechanical engineering, with a specialisation in diesel engines, from the Delft University of Technology in the Netherlands. He is dedicated to contribute to making the maritime industry more sustainable and preparing work vessels for the maritime energy transition.



Lindert van Biert

Lindert is an assistant professor at the Maritime and Transport Technology department of Delft University of Technology. His research focusses on characterisation, modelling, simulation and application of marine power and propulsion systems, and the adoption, storage and bunkering of renewable fuels. Lindert holds a Master's in mechanical engineering and obtained his PhD on his dissertation titled "Solid oxide fuel cells for ships: system integration concepts with reforming and thermal cycles".

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