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The application of process integration to the optimisation of cruise ship energy systems: a case study

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Abstract:
In recent years, the shipping industry has faced an increasing number of challenges in terms of fluctuating fuel prices, stricter environmental regulations, and concerns about global warming. In this situation, passenger volumes on cruise ships have increased from around 4 million to 13 million from 1990 to 2008 and keep growing today. A small cruise ship can emit about 85 tons of CO_2 per day, and require around 27 tons of fuel per day. To keep up with market demand, while reducing their impact on the environment, cruise ships will need to improve their energy efficiency.
Most previous research in marine technology relates to energy efficiency focused on propulsion, which for most ship types constitutes the largest energy demand. On cruise ships, however, auxiliary heat and electric power also have a significant importance. For this reason, we focus in this paper on the heat demand and its integration with available sources of waste heat on board.
In this study, the principles of process integration are applied to the energy system of a cruise ship operating in the Baltic Sea. The heat sources (waste heat from the main and auxiliary engines in form of exhaust gas, cylinder cooling, charge air cooling, and lubricating oil cooling) and sinks (HVAC, hot water, fuel heating) are evaluated based on one year of operational data and used to generate four operating conditions that best represent ship operations.
Applying the pinch analysis to the system revealed that the theoretical potential for heat integration on board could potentially allow the reduction of the external heat demand by between 35% and 85% depending on the investigated case. A technoeconomic optimisation allowed the identification of the most economically viable heat exchanger network designs: two in the “retrofit” scenario and one in the “design” scenario, with a reduction of 13-33%, 15-27% and 46-56% of the external heat demand, respectively. Given the high amount of heat being available after the process integration, we also analysed the potential for the installation of a steam turbine for the recovery of the energy available in the exhaust gas, which resulted in up to 900 kW of power being available for on board electric power demand.

Keywords:
Low carbon shipping, energy efficiency, process integration, heat integration, pinch analysis

1. Introduction

In a context of demand for increased energy efficiency in shipping, this article proposes an example of the application of process integration to the energy systems of a cruise ship.

1.1. General introduction to shipping

Although total shipping contribution to global anthropogenic CO_2 emissions is today relatively low (2.7% of the total in 2012 [1]) this share is expected to grow in coming years due to the increased transport demand [1]. This tendency goes in contrast with the targets that the shipping industry will be expected to achieve if global temperature increase is to be kept below 2°C compared to pre-industrial levels, as these targets will require shipping to decrease its carbon emissions by more than 80% compared to 2010 levels [2]. Although shipping has been excluded from the latest Paris agreement on climate change, both the IMO and individual actors (such as the European Union [3]) are acting with regulations and policies aimed at limiting CO_2 emissions from shipping.

1.2. Energy efficiency in shipping

Many developments were introduced in the latest years for improving the energy efficiency of the shipping sector. Operational measures include the reduction of auxiliary power consumption, improvements in voyage execution, engine monitoring, trim/draft optimization, weather routing, hull/propeller polishing, and slow-steaming. Design measures include the use of more efficient engines and propellers to improved hull design, pump frequency converters, air cavity lubrication, fuel cells, wind propulsion, waste heat recovery, cold ironing [4].

So far there is not much scientific work published which focuses mainly on the thermal component of the energy system. Extensive work has been published with reference to the application of heat-to-power waste heat recovery (WHR) solutions, specifically steam cycles and organic Rankine cycles [5]. More in general, Shu et al. [6] presented a review of possible means for the recovery of waste heat from two-stroke marine Diesel engines, including heat-to-power and heat-to-cooling technologies.

WHR in the form of heat-to-heat recovery is installed on the majority of ships today. Exhaust gas boilers (HRSG) for the generation of medium-pressure steam to fulfil on-board energy demand are the most common, while the use of low-pressure evaporators for recovering waste heat from the cooling system to generate freshwater on board are also rather widespread [7].

For the majority of the global fleet there is abundance of waste heat in comparison to the on-board heat demand, which makes it unnecessary to optimise this part of the system [8]. On some specific ship types, however, the specifics of the system and of the ship operations make heat demand to become a significantly more relevant energy demand. This is the case of, for instance, cruise ships, where the needs for transport are combined with requirements related to passenger services and comfort, such as air conditioning, cooking and cooling in restaurants, and passenger entertainment facilities [9,10]. The complexity of these energy systems makes them of particular interest for the application of more advanced methods for the optimisation of on board energy systems.

1.3. Process integration

Industrial processes may present significant potentials for fuel and energy savings, as well as reductions of CO_2-emissions. Process integration techniques are powerful tools that aim at minimising the use of external energy utilities by maximising internal heat recovery within the system of interest. The most well-known method is named pinch analysis: it was developed by Linnhoff [11] in the 80's for designing heat exchanger networks in chemical processes and was also applied to industrial sites such as refineries, as discussed by Klemes et al. [12,13]. This technique was at first developed for continuous processes but was extended to batch processes in the last decade, considering heat storage opportunities. The application of these tools to systems specific to the marine industry is more uncommon, possibly because of the challenges to tackle: ships operate in different conditions and their heating demands vary therefore over time. The most relevant works deal with the implementation of waste heat recovery cycles. Frameworks were presented in the literature for the generation of heat exchanger networks operating over ranges of variations in flow rates and temperatures (multiperiod problem). They may deal without the integration of utility systems or without.

1.4. Aim

This paper deals with the application of process integration tools to ship energy systems, taking a cruise ship as case study and with a particular focus on the optimisation of the steam distribution network. As the ship operates in different conditions (port stays, sailing) and environmental conditions (cold winters, warm summers), the process integration cannot be performed on one design point but on a number of representative operational conditions.

2. Problem description

2.1. Ship description

The ship under study is a passenger vessel operating on daily cruises in the Baltic Sea between mainland Sweden (Stockholm) and the island of Åland (Mariehamn), located roughly halfway between Sweden and Finland. The ship is 177 m long and has a beam of 28.6 m and has a capacity of 1800 passengers. The vessel was designed for a cruising speed of 21 knots and was built in Aker Finnyards, Raumo (Finland) in 2004. On the ship, the passengers are entertained in several restaurants, night clubs and bars, as well as saunas and pool, resulting in a heat and electricity demand on board that is more complex than in the case of a regular cargo vessel of the same size.

The typical daily operational pattern of the ship is represented in Fig. 1. This pattern can change slightly during normal operations, but the starting time in Stockholm and Mariehamn is fixed. It should be noted that the ship often stops and drifts in open sea for both saving fuel and for the comfort of the passengers during night hours before mooring at its destination in the morning, if allowed by weather conditions.

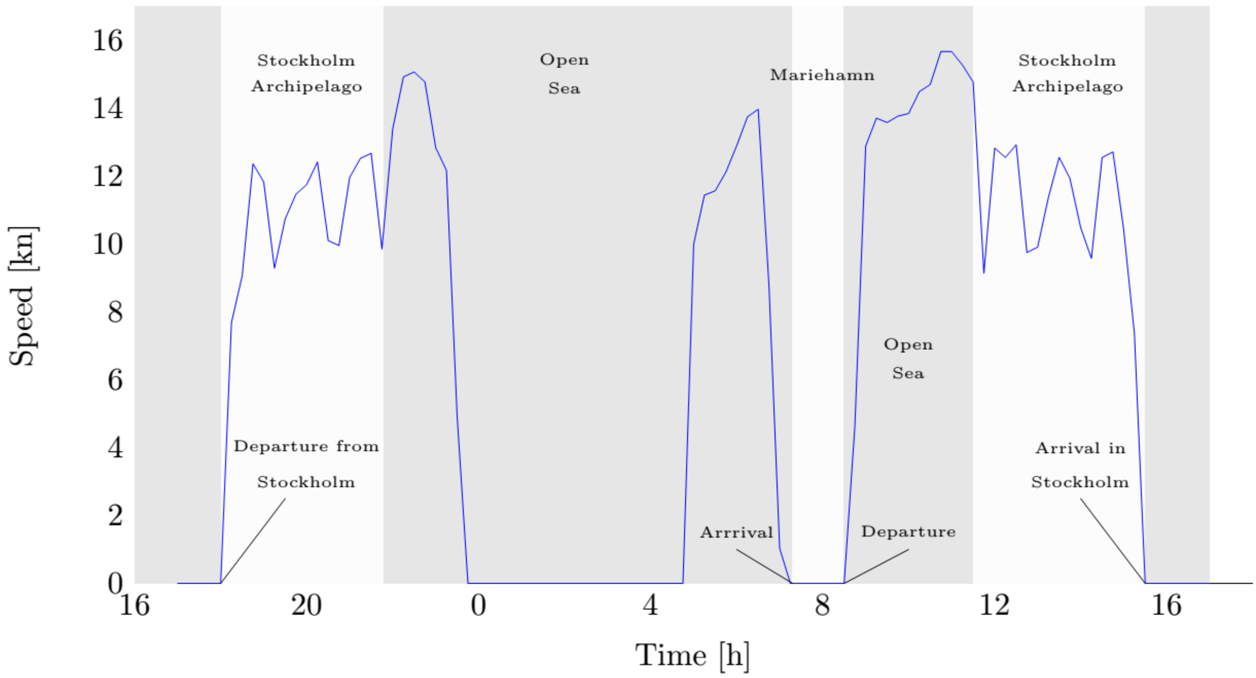


Fig. 1: Typical daily operations of the case study ship

The propulsion system consists of of two equal propulsion lines, each composed of two engines, a gearbox, and a propeller. Four Wärtsilä 4-stroke Diesel engines (ME) rated 5850 kW each are in charge of delivering propulsion power. All engines are equipped with selective catalytic reactors for nitrogen oxides emissions abatement. Although propulsion power is needed whenever the ship is sailing, full speed speed is rarely achieved during normal operations. In the large majority of operational conditions, only one engine per propulsion line is running. It is worth noting that the majority of the time it is from a power propulsion perspective only necessary to run one engine, but two are running because of r safety requirements.

Auxiliary power is needed on board for a number of alternative functions, from pumps in the engine room to lights, restaurants, ventilation and entertainment for the passengers and is provided by four Wärtsilä 4-stroke auxiliary engines (AE) rated 2760 kW each.

Auxiliary heat is needed for passenger and crew accommodation, as well as for the heating of the highly viscous heavy fuel oil used for engines and boilers. These needs are fulfilled by three different systems: the HRSG located on two of the four MEs and on all four AEs; the heat recovery system on the high temperature (HT) cooling water (HRHT), which can harvest heat at low temperature from all eight engines; and the two oil-fired auxiliary boilers (ab), mainly used when the ship is in port, or during winter.

2.2. Heat recovery and integration

As described in the previous section, the ship is already equipped with an integrated heat recovery system which allows avoiding the use of the boilers for a significant amount of time. Based on previous research by the authors on the analysis of the ship's energy systems [10], it was observed that roughly 70% of the overall heat demand on board is satisfied without any additional fuel consumption. The use of the auxiliary boilers can be needed when none of the following happens. First, if the heat demand rises, which happens in some specific moments of the day or when the outer temperature decreases. Secondly, when the available heat decreases, which happens when the ship is in port, or it is sailing at low speed. The current heat recovery system is based on two main media: 6 bar steam generated in the boilers and pressurised water heated up by the HT cooling systems (HTCS) and, when needed, by steam itself.

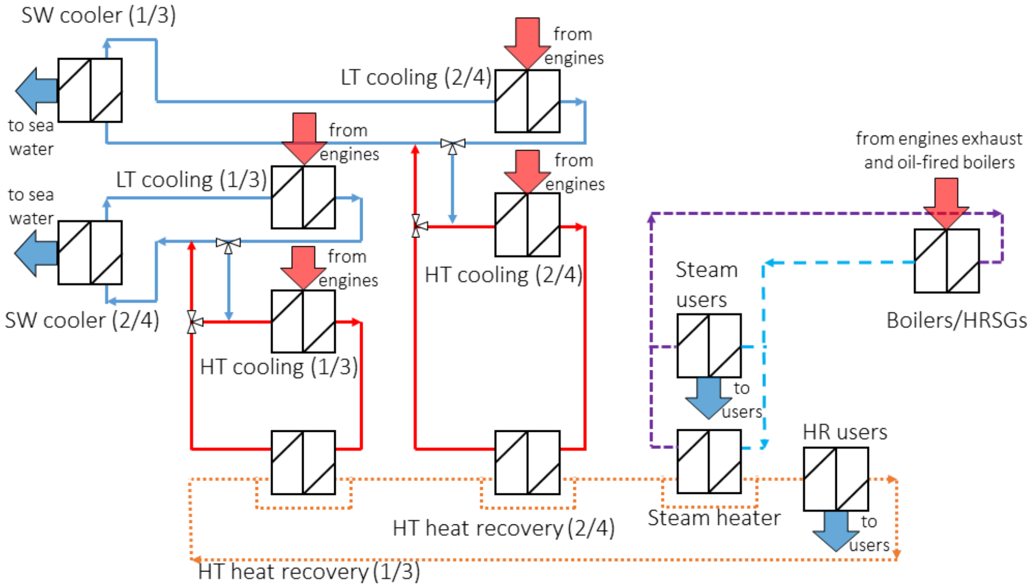


Fig. 2: Overview of the heat exchanger network on board

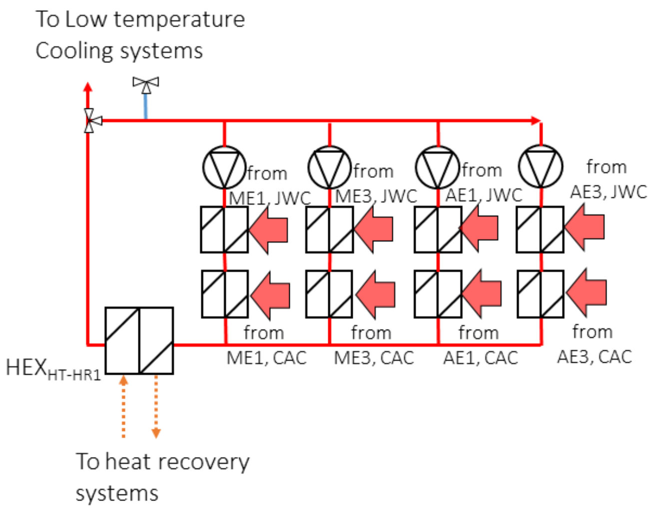


Fig. 3: Detail of the HTCS

A general overview of the existing HEX network on the case study ship is shown in Fig. 2. There are several systems on-board that require cooling. In particular, the MEs and AEs are the main sources of waste heat that needs to be extracted to ensure safe engine operations. On board of the case study vessel, these cooling systems are structured in two different temperature levels: high temperature and low temperature (LTCS) cooling systems. In order to increase redundancy, the cooling systems are divided in two independent systems. Each system is responsible for the cooling of two main and two auxiliary engines (1/3 and 2/4 in Fig. 2).

The heat transferred to the HTCS is further exchanged to the LTCS through a partial mixing, regulated as to keep the maximum temperature in the HTCS below a specific value (generally set to 90^{O}C). Then, the sum of the heat to the HTCS and LTCS is finally transferred to the sea water (SW) through a SW cooler. Seawater is not used directly in the cooling systems to avoid corrosion.

On the case study vessel, there are two parallel systems for heat recovery, one based on the HTCS and one based on the exhaust gas.

Waste heat can be recovered from the HTCS through a heat-recovery system based on pressurized water (HTHR). The existing systems therefore allows recovering the waste heat from jacket water cooling and from the HT stage of the cooling of charge air. The HTHR is then connected to four main users: hot water heater, air conditioning (AC) pre-heater, AC re-heater and the on-board low-pressure evaporator using for the generating technical water. When the heat from the HTCS is not sufficient for fulfilling the demand from the aforementioned consumers, the required extra energy can be provided from the boilers through a steam heater.

The remaining part of the on-board heat demand is fulfilled by the steam distribution system. Steam is generated on-board by using six HRSGs, positioned on all auxiliary engines and on main engines 2 and 3, and two ABs. The two ABs alone, rated approx. 4700 kW each, can satisfy the entire heat demand.

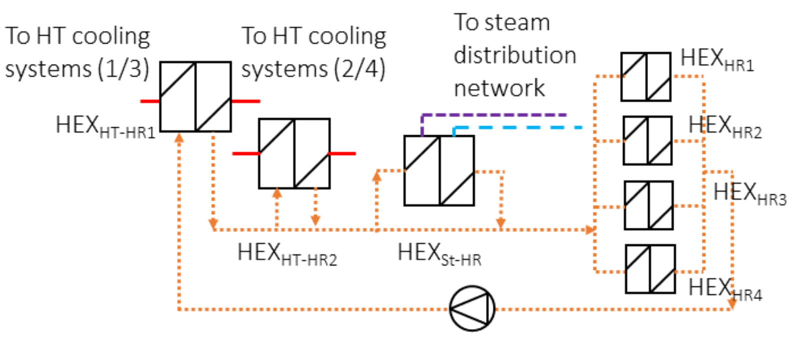


Fig. 4: Detail of the HTHR systems

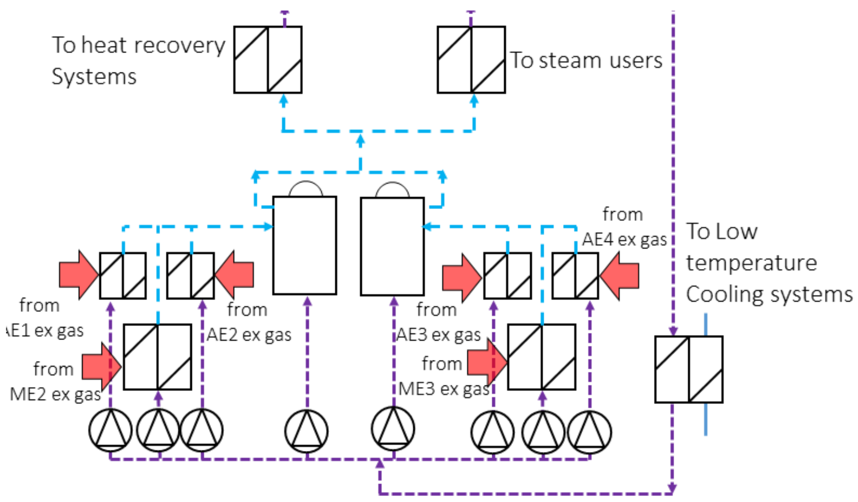


Fig. 5: Detail of the steam generation systems

3. Methodology

In this paper, we refer to the pinch analysis method, as presented by Linnhoff and extended later to industrial sites by Klemes et al. [12]. The problem is derived as a mathematical programming problem. It builds on the heat cascade theory and the original transhipment model proposed by Papoulias and Grossmann [14] and is formulated as a linear optimisation problem as proposed by Maréchal and Kalitventzeff [15], for which the objective is to maximise the internal heat recovery and minimise the operating costs. The total investment costs and part-load behaviour of the heat exchanger network are derived a posteriori, after the optimisation procedure. It is applied to the specific case presented in the previous section, a passenger vessel sailing in the Baltic Sea.

The methodology applied in this work can be summarised in the following four steps:

(1) The process boundaries are defined and the appropriate data is extracted, the heating and cooling demands are therefore identified, in terms of temperatures and heat flows. In the present study, this step was partly performed in a previous work of the authors [10], and the corresponding results are used to generate reference conditions for the analysis, as described in details in Section 3.1.
(2) The minimum temperature difference which can be accepted in the heat exchangers is defined, considering that different types of streams have different film coefficients, which has an impact on the component design and cost. The maximum internal heat recovery and minimum utility demands are evaluated through a pinch analysis.
(3) Different options for improving the current system can be proposed, from a partial redesign of the heat exchanger network, to the modification of operating conditions and integration of new technologies.
(4) These possibilities can be optimised with regards to economic or thermodynamic aspects, by retrofitting the existing heat exchanger network, paying attention to engineering constraints.

The analysis is performed in these steps to investigate (i) whether a retrofit of the current utility system could present significant benefits in terms of energy demands while being economically interesting, and (ii) to which extent additional energy savings can be reached by further system integration.

In the description of the selected case study (Section 2.2) we mentioned the existence of a heat recovery system on board. In this work, however, we perform the heat integration based on the base hot and cold streams without any intermediate heat exchanger installed. This allows comparing the optimal solution obtained according to the application of process integration to the system installed by the shipyard.

3.1. Identification of heating and cooling demands

Based on previous work by the authors [10] the heating and cooling demands for the presented case study could be evaluated, and are summarised in the following sections.

3.1.1. Investigated cases

Given the variable operational conditions of the specific application, both in terms of hot and cold streams, four main cases were investigated based on whether the ship is in port or sailing, and on the season. The main features of the different cases are presented in Table 1 and were used as a basis to determine both hot and cold streams.

3.1.2. Hot streams

The largest part of the available heat on board of most ships is connected to the operations of the Diesel engines on board. This, in turn, refers to four main waste heat sources: exhaust gas, inlet air (also referred to as charge air) cooling, lubricating oil cooling and cylinder wall (also referred to as jacket water) cooling. Of these, only the last three can be associated to cooling demands, while exhaust gas, although representing a source of waste heat, can simply be released as they are to the environment.

Table 1: Description of the investigated cases

	Case 1	**Case 2**	**Case 3**	**Case 4**
Season	Winter	Winter	Spring	Spring
Op. mode	Port	Sailing	Port	Sailing
ME running	0	2	0	2
ME load	0%	40%	0%	40%
AE running	2	2	2	2
AE load	60%	35%	60%	35%
Passengers	1050	1050	1350	1050
Outer air temperature	278	278	285	285

The application of process integration to the hot streams as presented in Table ***2*** is hereafter referred to as the “design” case, where it is assumed that the current utility systems (i.e. the HTCS and LTCS) can be replaced and that the heating demand of a given process stream can be (partly) satisfied by directly recovering heat from another one.

Table 2: Hot streams, design case

Streams	**Fluid**	**Case 1 and 3**				**Case 2 and 4**			
		$\dot{Q}$	$\dot{m}$	T_{in}	T_{out}	$\dot{Q}$	$\dot{m}$	T_{in}	T_{out}
Charge air cooler (ME1)	air	0	0	-	-	318	4.2	393	323
Charge air cooler (ME2)	air	0	0	-	-	318	4.2	393	323
Charge air cooler (AE1)	air	375	3.9	412	323	99.8	2.1	367	323
Charge air cooler (AE2)	air	375	3.9	412	323	99.8	2.1	367	323
Jacket water cooler (ME1)	-	0	-	393	393	252	-	393	393
Jacket water cooler (ME2)	-	0	-	393	393	252	-	393	393
Jacket water cooler (AE1)	-	293	-	393	393	171	-	393	393
Jacket water cooler (AE2)	-	293	-	393	393	171	-	393	393
Lubricating oil cooler (ME1)	lub oil	0	0	-	-	308	28.3	333	338
Lubricating oil cooler (ME2)	lub oil	0	0	-	-	308	28.3	333	338
Lubricating oil cooler (AE1)	lub oil	217	14.2	333	341	126	14.2	333	337
Lubricating oil cooler (AE2)	lub oil	217	14.2	333	341	126	14.2	333	337
Exhaust gas (ME1)	eg	0	0	-	-	1166	5.9	616	433
Exhaust gas (ME2)	eg	0	0	-	-	1166	5.9	616	433
Exhaust gas (AE1)	eg	1028	4	671	433	608	2.2	689	433
Exhaust gas (AE2)	eg	1028	4	671	433	608.3	2.2	689	433
Boilers									

Table 3: Hot streams, retrofit case

Streams	**Fluid**	**Case 1 and 3**				**Case 2 and 4**			
		$\dot{Q}$	$\dot{m}$	T_{in}	T_{out}	$\dot{Q}$	$\dot{m}$	T_{in}	T_{out}
HT cooling systems	water	923	16.7	350	363	1083	100	360	363
LT cooling systems	water	845	16.7	309	321	1466	100	309	313
Exhaust gas (ME1)	eg	0	0	-	-	1166	5.9	616	433
Exhaust gas (ME2)	eg	0	0	-	-	1166	5.9	616	433
Exhaust gas (AE1)	eg	1028	4	671	433	608	2.2	689	433
Exhaust gas (AE2)	eg	1028	4	671	433	608	2.2	689	433
Boilers									

As described in Section 2.2, however, the ship is already equipped with intermediate cooling systems which are used to extract heat from the Diesel engines and reject it to the sea. In the case where the process integration is applied to an the existing vessel, hence, the available sources of waste heat should be selected among the LTCS and HTCS, as shown in Table *3*.

In this case it is assumed that the heat exchangers between the process and utility streams cannot be retrofitted and the only degree of freedom lies in a possible integration of the different utility subsystems (steam, cooling water, tank heating, etc.) with each other.

3.1.3. Cold streams

The cold streams represent the heat demand on board. In order to reduce the complexity of the system and the uncertainty, the heat users are here subdivided in eight main categories, which are hereafter presented. The air conditioning systems require both a preheater and a reheater, which in the existing system are driven by hot water at respectively 50^{O}C and 80^{O}C. Showers and other accommodation-related features require extensive supply of hot water. Several tanks on board need heating for keeping the stored fluids to an appropriate level of viscosity. Additional heat is required by the machinery in the galley, at a rate depending on the time of the day and on the amount of passengers. Finally, a number of other consumers exist on board (e.g. separators, heaters in the engine room) that are all grouped in a constant demand. The cold streams are summarised in ***Table 4***.

Table 4: Cold streams

Streams	**Fluid**	**Case 1 and 2**				**Case 3 and 4**			
		Power	Flow	T_{in}	T_{out}	Power	Flow	T_{in}	T_{out}
AC preheater	water	1754	31	309	323	695	31	318	323
AC reheater	water	1068	21.3	341	353	570	21.3	347	353
Hot water heater	water	73	-	343	343	90	-	343	343
Hot water steam heater	water	22	-	343	343	27	-	343	343
Galley	steam	36	0.02	433	433	45	0.02	433	433
Others	steam	190	0.09	433	433	190	0.09	433	433
Sea water	water								

Integrating the individual heating and cooling demands of all processes present on the cruise ship is challenging, since these energy demands are varying over time and streams belonging to different subsystems may be geographically located far from each other. At present, low- and medium-temperature excess heat (cooling demand) is discharged into the environment through a common cold utility system (cold water) while the heat deficits at high temperature (heating demand) are satisfied by using a common hot utility system (pressurised steam).

The present work builds on the technology and thermodynamic levels of details, and they have been referred earlier as “retrofit” and “design” heat exchanger network (HEN), to denote whether direct modifications of the process-utility heat exchangers were possible.

3.2. Techno-economic modelling

3.2.1. Heat exchanger network

A HEN that achieves the maximum internal recovery is designed for each of the presented cases, following the rules of thumbs of the pinch method, i.e. (i) no external heating should be provided below the pinch point, (ii) no external cooling should be provided above the pinch point, and (iii) no heat should be transferred across it. Several network layouts may be equivalent from an energy perspective, and focus is therefore kept on minimising the number of additional components and reducing the heat transfer area. However, it is essential to develop HENs that can be operated in several load conditions.

The temperature levels of each energy demand are corrected to account for the fact that a minimum temperature difference between the hot and cold sides of the heat exchangers is required to ensure appropriate heat transfer and avoid too large heat exchangers. This minimum temperature difference is generally selected based on the type of heat exchangers (e.g. shell and tube or plate) that can be used, on the type of fluid (e.g. organics or water), and on the type of heat exchange (e.g. exchange of sensible heat or phase change). In this work, they are assumed to be 2, 4 and 8 K for phase-changing, liquid and gaseous flows respectively.Phase-changing streams are characterised by higher film coefficients than liquid and gaseous flows, which explains the smaller values of the minimum temperature differences allowable in the heat exchangers.The corrected temperatures are defined as:

$$T_h^* = T_h - \frac{\Delta T}{2}; \text{and } T_c^* = T_c - \frac{\Delta T}{2}; \text{with } \frac{\Delta T}{2} = \begin{cases} 2K \text{ for phase changing streams} \\ 4K \text{ for liquid streams} \\ 8K \text{ for gaseous streams} \end{cases}, \quad (1)$$

The minimum demands for external heating and cooling can therefore be determined, together with the pinch point of the cruise energy system. The latter illustrates the temperature level across which no heat should be transferred, and divides the whole energy system into a heat sink where only external heating should be provided, and a heat source with only external cooling.

Based on these constraints, the heat exchanger network and utility system are (re-)designed to maximise internal heat recovery and minimise the consumption of external utilities. The heat exchangers are assumed counter-current and ideal, i.e. without pressure drops or heat losses; it is also assumed that the specific heat capacity of each fluid is constant over the operating temperature range.

The flow rates and temperatures of each process stream may vary with time, which implies that the heat exchanger network operates under different sets of conditions, and goes into part-load conditions, which has an impact on the overall heat transfer coefficient. The heat transfer rate across a heat exchanger $\dot{Q}$ is defined as a function of the logarithmic mean temperature difference ΔT_{ml} between both sides, the heat transfer area A and the overall thermal coefficient U:

$$\dot{Q} = UA\Delta T_{ml}, \quad (2)$$

Neglecting the fouling resistances and resistances associated with conduction through the heat exchanger materials, the overall heat transfer coefficient is a function of the individual heat transfer coefficients on the hot h_{hs} and cold h_{cs} sides, as well as the heat transfer area on each side. Assuming that the two exchange areas are in the same order of magnitude, the overall heat transfer coefficient can then be expressed as:

$$\frac{1}{U} = \frac{1}{h_{cs}} + \frac{1}{h_{hs}}, \quad (3)$$

In design conditions, the overall heat transfer coefficient for water/water heat exchangers without phase change is taken to 850 W/m^{2}K and 15 W/m^{2}K for air coolers. In off-design conditions, the individual coefficients for each fluid decrease with the flow rate, following a power-law expression in the form of:

$$h_{off-des} = h_{des}\left(\frac{\dot{m}_{off-des}}{\dot{m}_{des}}\right)^{\alpha}; \text{and } \alpha = \begin{cases} 0.6 \text{ on the cold side} \\ 0.8 \text{ on the hot side} \end{cases}, \quad (4)$$

The heat exchanger networks proposed in this work were simulated on Aspen Plus version 8.6.

The thermophysical properties of air, water and lubricating oil are derived from the Peng-Robinson equation of state, which is an equation of the cubic family widely used.

In the case that a new heat exchanger is implemented, the grassroot costs of the heat exchanger network are given as a function of the heat transfer areas, using the capacity-based cost correlations of Turton [16] for shell and tube heat exchangers. The correlations present an uncertainty of +/- 30% and account for the purchased equipment cost, pressure and material factors, and the additional investment costs related to the contingencies and additional fees:

- the purchase cost (C_{pc}) of a heat exchanger as in 2004 is expressed as:

$$\log(C_{pc}) = 3.224 + 0.2419\log(A_{HT}) + 0.09128[\log(A_{HT})]^2, \tag{5}$$

- it is then adjusted with pressure and material factors to calculate the bare module costs, considering copper as heat transfer material, and given by:

$$C_{bm}^0 = C_{pc}(1.8 + 1.5 f_m f_p), \text{ with } \mathrm{f_m} = 1.6 \text{ and } \mathrm{f_p} = 10^{\left(0.06499 + 0.05025\log(p) + 0.01474(\log(p))^2\right)}, \tag{6}$$

- the inflation between the reference year of the cost data and the date of the estimate is estimated by using the Marshall and Swift index, so that:

$$C_{bm} = C_{bm}^0\left(\frac{MSI}{MSI^0}\right), \tag{7}$$

- the total investment costs when installing the equipment items is finally deduced by:

$$C_{gr} = (1+\alpha_1)\sum_i C_{pc,i} + \alpha_2 \sum_i C_{bm,i}; \text{ with } \alpha_1 = 0.18 \text{ and } \alpha_2 = 0.35, \tag{8}$$

In a retrofit case, where only additional heat exchanger area is provided, the cost of the heat exchanger network is estimated by Axelsson et al. [17]**:**

$$C_{rt} = 1.1(30000 + 750A_{HT,\mathrm{sup}})^{0.81}, \tag{9}$$

Where $A_{HT,sup}$ stands for the additional heat transfer area that should be installed.

The expected lifetime for the heat exchanger network is 30 years and an interest rate of 10% is considered, as well as an availability factor of 8000 hours per year.

3.2.2. Steam Rankine cycle

The overall system performance can be improved by implementing cogeneration systems, recovering waste heat from the exhausts of the main and auxiliary engines, when applicable. For example, the main engines are run only while the cruise ship is sailing, while at least one auxiliary engine is run in all conditions. The integration of a steam Rankine cycle may be appropriate, as heat would then be recovered from the engine exhausts and converted partly into power. In the case that an extraction turbine is installed, a fraction of the steam can be recovered at a medium pressure level for direct heat supply. In the other case, if a simpler layout cycle is preferred, the steam cycle can be used only for power generation purposes, and the heat demand can be covered by directly recovering heat from the exhaust gases or by producing steam in a boiler, as it is currently.

The following assumptions are considered when modelling the steam cycle: the isentropic efficiency of the steam turbines is set to 80% and the pump efficiency to 90%. The heat and mechanical losses in the boiler and turbomachinery components are neglected. The minimum temperature for rejection of the exhaust gases is 160°C, and the lower bound for the condensing temperature is set to 30°C. The steam properties are derived from the steam tables of the International Association for the Properties of Water and Steam [18]. The grassroot costs of the steam cycle are estimated following the same costing method, using different coefficients for the cost correlations of Turton et al. [16] and Ulrich [19] for each component (steam turbine and centrifugal pump):

- the purchase costs C_{pc} are expressed as:

$$\log(C_{pc}) = 3.722 + 0.4401\log(\dot{W}_{ST}), \tag{10}$$

for a steam turbine

$$\log(C_{pc}) = 3.5793 + 0.3208\log(\dot{W}_{PMP}) + 0.0285[\log(\dot{W}_{PMP})]^2, \tag{11}$$

for a centrifugal pump.

- it is then adjusted with an installation or material factor, which gives:

$$C_{bm} = 3.5C_{pc}, \tag{12}$$

for the steam turbine, and

$$C_{bm}^{0} = C_{pc}(1.8 + 1.5 f_m f_p), \text{ with } f_m = 1.8 \text{ and } f_p = 10^{\left(0.1682 + 0.3477 \log(p-1) + 0.4841 (\log(p-1))^2\right)} \quad (13)$$

for the pump

The inflation between the reference year of the cost data and the date of the estimate, and the total investment costs when installing the equipment items are estimated as showed in Equations 7 and 8.

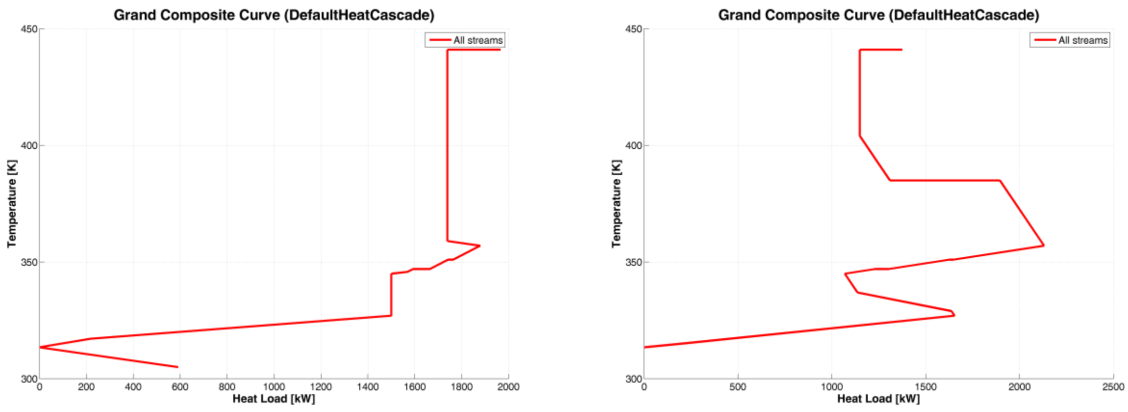
4. Results and discussion

4.1. Opportunities for energy savings

An energy and process integration analysis is performed on the cruise ship case data, comparing the minimum energy requirements with modifying and without modifying the existing heat exchanger network. The total energy requirements for the energy system do not differ from one approach to another, the only difference is lying in the amount of heat that can be exchanged internally, and therefore the demands for external heating and cooling (Table 1).

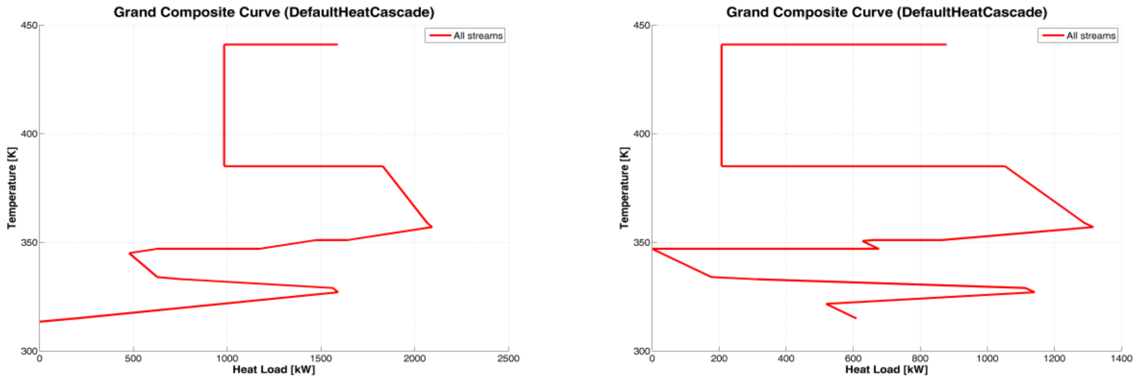
Table 5: Heating and cooling demands, with and without system integration. All values are in kW

	Case 1			Case 2			Case 3			Case 4		
	B	S	C	B	S	C	B	S	C	B	S	C
Heating demand		3140			4140			1620			2820	
Cooling demand		1760			2550			1770			2550	
Heat available from HRSG		2056			3549			2056			3549	
External heating demand	3140	1970	1380	4140	3055	1590	1620	695	235	2820	1735	1150
External cooling demand	1760	590	0	2550	1465	0	1770	845	385	1550	1465	610
Internal heat recovery	0	1170	1760	0	1085	2550	0	925	1385	0	1085	1670
Heat from boilers	1084	0	0	591	0	0	0	0	0	0	0	0
Heat available for power gen.	0	86	676	0	494	1959	436	1361	1821	729	1814	2399



(a) (b)

Fig. 6: Temperature-heat profiles of the cruise energy system for Case 1: a) retrofit, b) complex heat exchanger network.



(a) (b)

Fig. 7: Temperature-heat profiles of the cruise energy system for the design heat exchanger network: a) winter, b) spring.

The energy savings differ significantly from case to case (Fig. 6) – the total heating demand is higher than the cooling demand in three of the four studied cases. This suggests that heat should in any case be covered, at least partly, by heat recovery from the engines and steam production. On the contrary, the cooling demand can be reduced sharply, in particular when the cruise ship is located at the harbour in winter conditions, by proper integration of the heating and cooling flows. It is worth mentioning that reducing the cooling demand may not be any problem of concern on a ship as the sea water is not a shortage.

A comparison between the maximum energy recovery with minimum process integration (retrofit scenario – integration of the utility system) and with the enhanced one (design scenario – integration of the process streams) shows that:

- Energy savings of at least 900 kW can be reached with minimal integration. This represents from 22% to 52% of the system heating demand and from 36% to 53% of the cooling demand.
- Energy savings of at least 1385 kW and up to 1760 kW can be achieved with maximum process integration. This represents from 33% to 85% of the system heating demand and from 54 to 79% of the cooling demand.

A comparison of the heat temperature profiles for each condition (Fig. 7), shows that the location of the pinch point varies significantly from one season to another. This illustrates that the energy system of the cruise ship behaves as a global heat sink in winter conditions, and both as a heat sink and a heat source in spring conditions.

4.2. Design of the heat exchanger network

4.2.1. Re-design of the utility/utility heat exchangers

The pinch method aims to minimise the consumption of external utilities, while avoiding the integration of additional components and excessive heat transfer area. It is still essential that the heat exchanger networks can be operated in different load conditions. The proposed configurations are selected with regards to the number of heat exchangers, which should be equal or close to the minimum value calculated with the Euler's theorem in graph theory.

For example, the maximum energy recovery is 1170 kW with integration of the water cooling system in the first case, when the ship is located at the port in winter time, but this requires two additional heat exchangers. The AC preheater should be connected to the both the HTCS and LTCS, and the remaining heating and cooling demands should be covered by external utilities. However, such a network would be impracticable in other operating conditions: when the ship is sailing, a direct heat exchange between the AC preheater and the LTCS is unfeasible because of the mismatch of temperatures and small heat transfer driving forces.

In the case that the water from the HTCS system can be rerouted into the initial heat exchanger between the AC preheater and LTCS, about 1190 kW of energy can be exchanged internally, which results in a reduction of the heating and cooling demands by 27 and 47%, respectively, when the cruise is sailing in winter conditions. However, this alternative is not practicable in the spring , as the water circulating in the HTCS only needs to be cooled down by 3°C instead of 13°C, and an analysis of the off-design behaviour of the HEX network shows that the water from the AC preheater would be heated beyond the desired temperature of 50°C.

Two main possibilities appear to be feasible in all operating conditions, with a minimum number of design changes:

- A first possibility (hereafter referred to as R1) is to add only a single heat exchanger between the AC preheater and the HTCS, although such a solution does not achieve the maximum possible savings in any case. This means that the use of cooling water and steam to cover the cooling and heating demands of these streams is necessary as a backup solution. However, it presents the advantage that this heat exchanger can be operated in all situations, since the temperature difference between the water streams is of at least 27°C.

This heat exchanger is designed for the largest flow and demand of the high-temperature of the cooling system, which corresponds to the winter sailing case (Case 2), and the energy savings amount to 1080 kW, but decrease to 415, 350 and 920 kW in the cases 1, 3 and 4, respectively, because of the much smaller flowrate of cooling water processed when the ship is located at the harbour.

The cost of this additional heat exchanger is about (85 +/- 30%) kUSD, as no additional heat transfer area is required on any other components.

- A second possibility (R2) is to add a heat exchanger between the HTCS and AC reheating, which results as well in energy savings at all conditions. However, the heating demand of the AC reheater is smaller than the cooling demand of the HTCS in spring conditions, and the opposite case is encountered in winter conditions, which implies that both additional heating and cooling will be required, depending on the season.

This heat exchanger is designed for the largest flow and demand of the high-temperature of the cooling system, which corresponds to the winter sailing case, and the energy savings amount to 1070 kW, but decrease to 460, 330 and 770 kW in the cases 1, 3 and 4, respectively, which corresponds to the same trend as for the first possibility. The cost of this additional heat exchanger is about (50 +/- 30%) kUSD.

When compared to the heat recovery system installed on board, it can be noted that the proposed ones (R1 and R2) are much simpler, as they only require the installation of one additional heat exchanger, compared to the five that are installed today. In addition, the proposed system does not include any intermediate fluid. The results of this study confirm, however, the fact that the recovery from the LTCS is not particularly convenient when all operational conditions are taken into account.

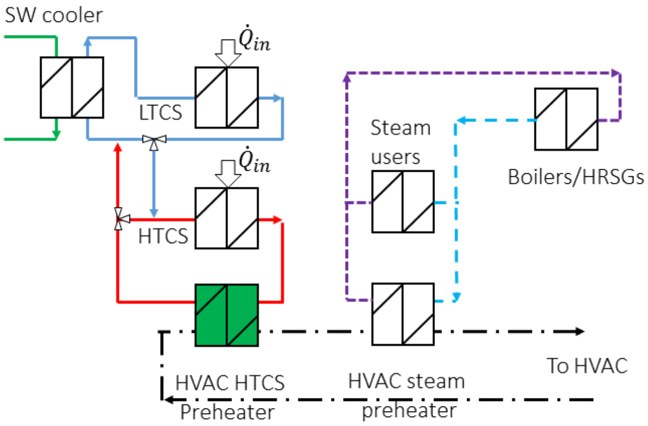


Fig. 8: HEN configuration in the R1 scenario. Note that the R2 case only differs by the additional heat exchanger being placed on the HVAC reheater instead of the preheater

4.2.2. Re-design of the process/utility heat exchangers

As illustrated in Table 1, it is theoretically possible to further increase the energy recovery by a better integration of each individual heat flow within the cruise ship energy system. The findings suggest that enough energy can be recovered so that no external cooling utility is required in the winter season. This is achievable in practice by (D1):

- using the heat from the charge air, jacket water and lubricating oil coolers in the HVAC preheater when the cruise ship is located in the harbour;
- recovering the heat surplus from the charge air and jacket water coolers connected to the main engines, and discharging into the HVAC reheater when the cruise ship is sailing.

The main drawback associated with this layout is the lower temperature of the preheated water by 5°C when the cruise is sailing, because the load of the auxiliary engines is lower. The amount of heat recovered and discharged into the preheating system decreases in consequence from 1750 to only 910 kW, and heat should be provided by the HRSGs. The total energy savings reach, in this case (Case 2), about 1900 kW, which is about 3 to 4 times higher than the energy savings depicted when re-designing only the utility system (R1 and R2).

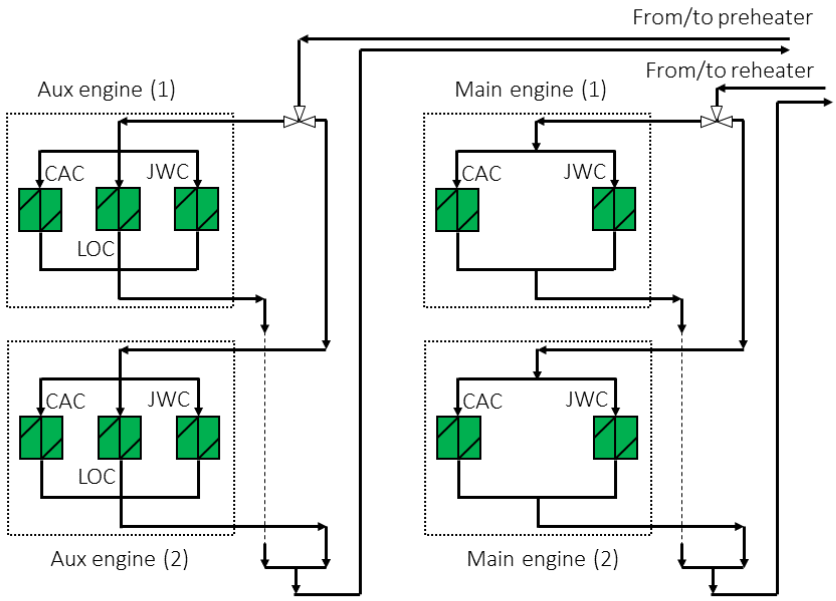


Fig. 9: HEN configuration in the D1 scenario

Table 6: Heating and cooling demands, results from the different proposed HEN solutions. All values are in kW

	Case 1			Case 2			Case 3			Case 4		
	R1	R2	D1	R1	R2	D1	R1	R2	D1	R1	R2	D1
Heating demand		3140			4140			1620			2820	
Cooling demand		1760			2550			1770			2550	
Heat available from HRSG		2056			3549			2056			3549	
External heating demand	2725	2680	1380	3060	3070	2240	1270	1290	805	1900	2050	1430
External cooling demand	1345	1300	0	1470	1480	650	1320	1340	855	1630	1780	1160
Internal heat recovery	415	460	1760	1080	1070	1900	350	330	815	920	770	1390
Heat from boilers	669	624	0	0	0	0	0	0	0	0	0	0
Heat available for power gen.	0	0	676	489	479	1309	786	766	1201	1649	1499	2119

An evaluation of the same heat exchanger network in spring conditions shows the opposite trend: additional cooling is required to bring the hot streams to the desired temperatures. In particular, the air temperature at the outlet of the charge air coolers connected to the auxiliary engines is 2 to 3°C higher than the desired one because of the smaller heating demand of the AC preheater and reheater. The energy savings are thus smaller than in winter conditions, reaching about 1390 kW when the ship is sailing. From a practical perspective, however, all engines should be provided with a backup cooling system which could be used in case of the standard heat recovery systems are not able to operate. Engine cooling on-board is of vital importance for the engines, which cannot be operated without appropriate cooling even for short intervals. This would, in practice, allow achieving the required temperature on the charge air cooling in any operational condition.

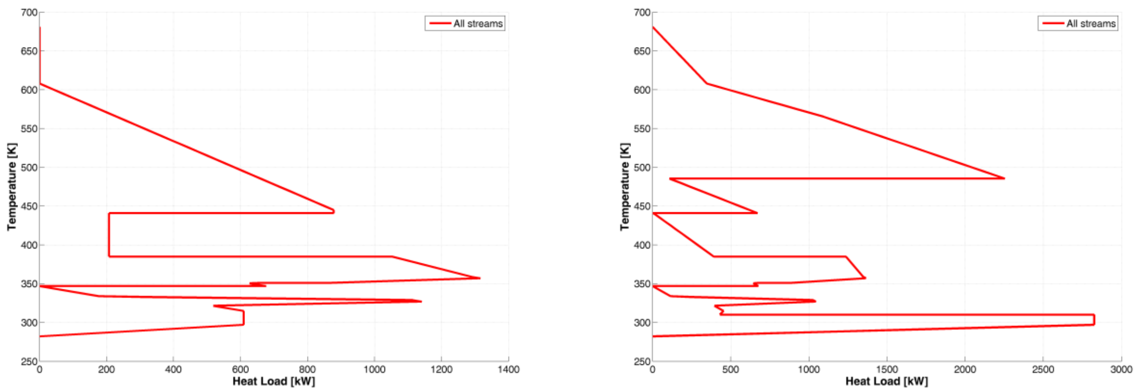
An economic evaluation based on the correlations given in Turton et al. suggests that the cost of the complete heat exchanger network, assuming that the heat exchangers with the cold and hot utilities are already installed, is about M$ (1.4 +/- 30%). These costs are mainly dominated by the price of the air coolers, as a result of a low convective heat transfer coefficient between air and water near atmospheric pressures. Compared to the cases presented in Section 4.2.1, the grassroot costs are significantly higher, which illustrates the trade-off between the capital (heat exchanger investments) and operating (lower heat recovery and greater fuel consumption) costs.

4.3. Integration of the steam cycle

As shown in Table 6, most of the heat demand from the ABs can be avoided even in the R1 and R2 HEN designs. In Cases 2-4, if there is no further use of on-board waste heat, there would be no reason for the installation of more complex HEN, such as the one proposed in D1, as the additional heat recovered from the HTCS would simply mean a lower use of the heat available from the exhaust gas.

Freeing the heat available from the engines exhaust gas becomes profitable if this is used for a heat-to-power recovery cycle, as a consequence of its higher temperature. The integration of a steam cycle is therefore investigated for the case where most heat is available, i.e. when both the main and auxiliary engines are running, so during sailing, and when the heating demand is the lowest, so during the spring. At present, a heat recovery steam generation system is installed, in order to recover heat from the engine gases and to produce part of the steam required on-site, but not a steam turbine for heat-to-power conversion.

A comparison of the temperature-heat profiles of the cruise energy system, including the hot (exhaust gases) and cold (cooling water) streams (Fig. 10), shows that (i) the heat contained in the engine exhausts is sufficient to cover the complete heating demand of the cruise, (ii) the large temperature difference between the exhaust gases and the cold streams suggests that integrating a steam cycle can lead to a better use of the high-temperature waste heat, (iii) the implementation of this heat-to-power technology results in an increase of the cooling water consumption by a factor 3.



(a) (b)

Fig. 10: Temperature-heat profiles of the cruise energy system, including the utilities in the design scenario and Case 4: a) without and b) with the integration of the steam cycle

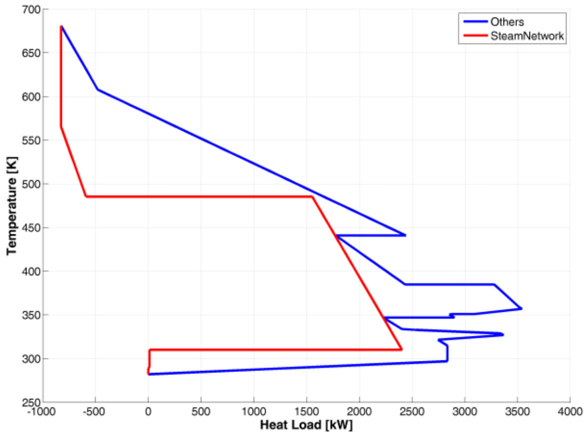


Fig. 11: Temperature-heat profile of the steam Rankine cycle in relation to the cruise energy system

A more detailed analysis of the operating conditions of the steam Rankine cycle (Fig. 11) suggests that the optimum pressure for steam production is around 20 bar, for a superheating of 80°C and a condensing temperature of 30°C. The power production reaches about 900 kW, and the heat recovered from the engine gases is either converted into mechanical power through the steam cycle, or used for heating purposes on-site. A preliminary economic evaluation suggests that the complete cycle (turbine, pump, condenser and boiler) has a cost of about $M (7.5 +/- 30%).

It should be noted, however, that using all of the 900 kW of power from the turbine for on-board electric power demand would substantially reduce power demand from the auxiliary engines, and therefore also redue the amount available waste heat both for on-boar heat demand and for heat-to-power conversion. Although when this aspect is taken into account the power generated by the steam turbine would be reduced, its contribution would still be significant to increase the energy efficiency of the ship's energy systems.

Future work should address the off-design behaviour and control strategy of the steam network in the optimisation procedure. Several solutions that are depicted in the design procedure may be discarded as they would be impracticable under severe partial loads of the ship energy systems.

4.4. Considerations on the application of process integration to cruise ship energy systems

In this article we proposed the application of process integration to ship energy systems, and in particular to the specific case study of a cruise ship. This application, to the best of our knowledge, has never been proposed before, and one of the aims of this article was, consequently, to explore the challenges of the application of process integration to ship energy systems. This allowed identifying two main challenges that future similar applications should take into further account.

The first point relates to the flexibility of the stream flow conditions. In standard methods of process integration, the conditions of hot and cold flows are well defined and unflexible. In ship energy systems, however, many flows could be adapted according to a limited number of constraints. The temperature in the HTCS is one of the examples: considerations related to engine maintenance generally suggest to keep such temperature withing the 70ºC – 90ºC boundaries, but could potentially vary within this range without drawbacks on the operations of the engine. Future applications of process integration to ship energy sytems should therefore put additional focus on the identification of "hard" and "soft" flow constraints and, therefore, on the possibilities of further improving the heat integration by acting on the soft constraints.

The third point relates to the flexibility of the operational conditions. Methods for the application of process integration to variable operational conditions (e.g. batch and time-dependent operations [20]) exist, and in this work this has been taken into account by considering four operational cases in the analysis. The complexity of the different variables that play a role in the definition of both heating (number of passengers on board, outer air temperature, sea water temperature, time of the day) and cooling (ship speed, auxiliary power requirements, outer air temperature) suggests that the results could be improved both in the potential for energy savings and in the accuracy of the results by increasing the level of detail in the analysis. Given the large amount of resulting operational conditions, investigating alternative HEN designs should be applied in a more systematic way. The peculiarity of the daily variations of the case proposed in this study also suggest that heat storage is a retrofit solution that is worth further investigation.

5. Conclusion

In this paper, we investigated the potential for improving the energy efficiency of the energy system of a cruise ship through the application of process integration.

In the first part of the analysis, the application of pinch analysis suggested that there is a large potential for heat integration on board, particularly in the case where the current utility systems can be replaced and that the heating demand of a given process stream can be (partly) satisfied by directly recovering heat from another one. The achievement of the theoretical maximum of heat integration would allow eliminating all needs for external use of oil-fired boilers, while allowing up to 2400 kW of waste heat to be used for auxiliary power generation.

In the second part of the analysis, three heat exchanger networks were proposed for improving the level of heat integration. The R1 and R2 retrofitting solutions allow heat to be transferred from the high temperature cooling systems to the HVAC pre-heater and re-heater, respectively. This allows the complete elimination of all heat demand from the auxiliary boilers in all cases but the winter/port one, where the demand is reduced by 38% and 42% respectively. The D1 case instead, allowing the use of heat directly from the engines, allowed avoiding all use of auxiliary boilers in all studied cases.

Finally, the potential for generating electric power onboard using the extra heat available from the exhaust gas was investigated for the spring/sailing case. This showed the potential of generating up to 900 kW of electric power, which would therefore allow reducing the fuel consumption from the auxiliary engines.

In conclusion, we showed that the application of process integration to ship energy systems, and in particular to cruise ships, allows identifying solutions that generate substantial energy savings by achieving a higher level of heat integration within the system. In particular, increasing the use of low-temperature heat from the cooling systems allows freeing high-temperature heat recovered from the exhaust gas, that can be instead be used for power generation in, e.g., a steam power cycle.

Nomenclature

A	area, m^2
C	cost, USD
h	heat transfer coefficient, $W/(m^2K)$
$\dot{m}$	mass flow, kg/s
p	pressure, bar
T	Temperature, K
U	Global heat transfer coefficient, $W/(m^2K)$

Acronyms

AE	Auxiliary engine
AB	Auxiliary boiler
HEN	Heat exchanger network
HRHT	Heat recovery on high temperature cooling systems
HRSG	Heat recovery steam generator

HTCS High temperature cooling systems
IMO International Maritime Organisation
LTCS Low temperature cooling systems
ME Main engine
MSI Marshall Swift Index
SW Seawater
WHR Waste heat recovery

Subscripts and Superscripts

bm Bare module (cost)
c Cold
des Design
h Hot
HT Heat transfer
off-des Off-design
pc Purchase cost
PMP Pump
ST Steam turbine

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