

Power generation from residual industrial heat

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ABSTRACT

Industrial plants continuously reject large amounts of thermal energy through warm liquid or gaseous effluents during normal operation. These energy losses contribute to an inflation of production costs and also threaten the environment. This paper investigates methods of recovering the residual low grade thermal energy and converting it into higher quality mechanical energy using the thermodynamic Rankine cycle principle. For the temperature range of the available thermal energy, water was shown to be a poor working fluid for the conversion system, thus several potential working fluids, including ammonia, synthetic refrigerants, and organic compounds have been considered as alternatives. A comparative analysis led to the identification of different performance evaluation criteria. For example, the water-based Rankine cycle and, to a lesser extent, the ammonia-based Rankine cycle proved to be interesting when the power generation potential per unit working fluid mass flow rate was considered. On the other hand, Rankine-like cycles using dry hydrocarbon working fluids proved much more interesting in terms of energy conversion efficiency, as well as in terms of the net mechanical power generation potential for a given heat source. All performance indicators were low at low temperatures, and improved as the primary heat source was available at higher temperatures. This paper also discusses the influence of various external and internal operating parameters, such as heat source and heat sink temperatures, turbine and pump isentropic efficiencies and the addition of an internal heat exchanger on the overall performance of the energy recovery and conversion system.

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1. Introduction

Cooling towers and smokestacks are among the most common characteristic components of the modern industrial environment. These components are used in order to recycle used liquids and gases by discharging out the residual heat they carry. Nguyen et al. [1] cited the example of the power generation industry and recalled that the average 37% conversion efficiency of fossil fuel thermal energy into mechanical energy implies that 63% of the energy input is going to waste at the generation site. The temperature of the combustion products at the exit of the boiler for a Rankine power cycle and the temperature of the low pressure gas at the exit of a Brayton turbine often exceed 500 °C [2]. It has also been estimated that a typical petroleum refinery in the USA rejects about 256 MW of thermal energy to the environment through warm gases at temperatures of 150–300 °C [3]. The chemical industry, as well as metallurgy and incineration plants represent other examples of industrial processes which draw huge amounts of high quality energy and reject a significant percentage of this energy through warm gaseous or liquid effluents. This energy depredation that occurs during normal operation contributes to an inflation of

production costs, and may also be seen as an avoidable misuse of a precious resource. On the other hand, both the discharge of the residual heat and the production of the corresponding avoidable primary energy threatens the environment, the addressing of which was part of the objective of the Kyoto Protocol [4–6]. A common practice for reducing these energy losses in industry consists of the integration of specially designed and arranged heat exchangers which collect thermal energy from the exit of high temperature heat demanding stages and pass it over to initial and typically low temperature heat demanding stages of the processes. The energy content of the exiting streams is also used occasionally for residential heating, or to power sorption-based refrigeration systems [7,8]. However, Verschoor and Brouwer [9] pointed out that converting heat at a certain temperature into heat at a lower temperature intrinsically causes a degradation of the thermal energy, *i.e.*, an exergy loss. Furthermore, the thermal energy demand strictly depends on site and weather conditions. It is also in a lesser extent than the available waste heat, and delivering thermal energy to distant users is not cost effective [1,3,10]. Finding ways to efficiently recover the waste thermal energy and convert it into a higher quality and more useful energy, shaft power or electricity, has therefore drawn a strong interest from the engineering and scientific community. A simple modification of the design of a Rankine-like power cycle has been reported that resulted in a

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maximization of the working fluid flow rate, a better utilization of the available thermal energy, and eventually to a larger net power generation [10]. Regenerative Rankine cycles have been proposed where the high pressure liquid en route to the evaporator is pre-heated by a vapour bleed operated during the expansion [5] or by the low pressure vapour at the exit of the turbine [11] as a means of increasing the process thermal efficiency. Marrero et al. [12] have proposed a three-step Brayton, steam Rankine and ammonia Rankine combined cycle, while new designs of the combined cycle technology have been introduced in which an initial power cycle using a low temperature thermal energy drives an even lower temperature bottoming turbine [13,14]. From a rather different perspective, Labrecque and Goni Boulama [15] have presented a general procedure of estimating the power generation potential of warm fluid flows of different temperatures and natures. This potential was shown to be high at high temperatures, and notably decreases as the temperature at which the heat is available decreases. It has also been shown that the conversion of low temperature heat into mechanical energy using the conventional steam Rankine cycle is not feasible [3,11,16,17]. Very few papers in the literature have attempted to define a new conversion cycle, because the ease of operation will ultimately constitute a decision parameter probably as important as the efficiency and the economics of the proposed conversion system [9]. Instead, the bulk of the research has concentrated on explaining the reasons for the shortcomings of the steam Rankine cycle [3,6,18,19], establishing a comprehensive list of selection criteria for better suited working fluids [16,18,20], validating thermodynamic property prediction tools for potential alternative working fluids [11,18,21,22] and optimizing the performance of power conversion systems using a given heat source [10,12–14,23].

The selection of an adequate working fluid for the conversion of low temperature heat into electrical energy will be reviewed in this paper with the main objective parameters being high performance and good material property. The superior performance of cycles using dry hydrocarbons will be demonstrated. In the meantime, we will also show that different performance indicators exist for such a problem, and cycles using different working fluids perform differently depending on the considered performance indicator. In the second part of the paper, an extensive sensitivity analysis will be conducted on the effects of the heat source and heat sink conditions, as well as many other internal operating parameters on the overall performance of the conversion cycle.

2. Working fluids for low temperature Rankine cycles

The layout of the standard steam Rankine power cycle is depicted in Fig. 1. High pressure liquid water is vaporized and super-

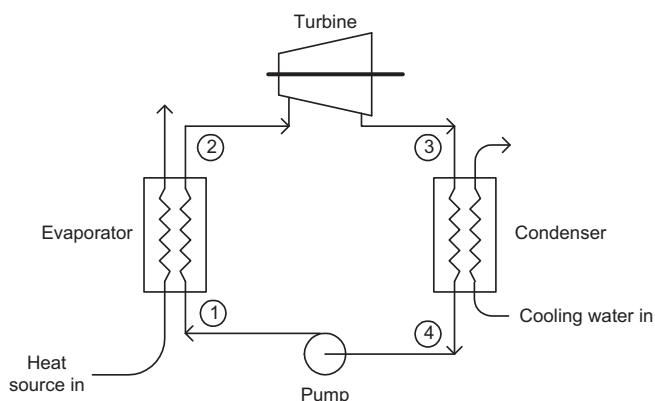


Fig. 1. Schematic of a Rankine power cycle.

heated in a boiler and sent to a turbine for expansion and work generation. At the exit of the turbine, the steam at a minimum 90% quality is condensed by heat transfer to a low temperature reservoir, usually the atmosphere, then re-pressurized and returned to the boiler for a new cycle. The heat required for the vaporization is usually a result of the combustion of some fossil fuel, and the temperature of the steam at the inlet of the turbine can exceed 600 °C for a pressure of about 50 bar. However, for the industrial waste heat considered as the heat source in this paper, the temperatures range only in the 100–250 °C interval and it is thus not possible to produce steam at very high pressures. Water is in fact a low vapour pressure chemical, meaning that for a given pressure it needs to be at a relatively high temperature before it turns into the vapour phase. As an example, water at 3 bar vaporizes at a temperature of 150 °C, a temperature high enough for ammonia at up to 100 bar to be vaporized.

Water is also referred to as a “wet fluid” [5,10,21], which means that it has a tendency to prematurely condense during expansion as a consequence of its bell-shaped temperature–entropy diagram. Premature condensation is highly detrimental to the operation of a turbine, and is generally prevented by a superheat of the steam before it enters the turbine. The superheat represented by the portion 2–2' of Fig. 2a constitutes a major limitation for the steam Rankine cycle should it be used as a means of converting low temperature heat into mechanical energy. Because of the superheat requirement, a medium size 30 bar conventional Rankine turbine admits steam at 400–500 °C, although the saturation temperature at 30 bar is only 233 °C.

Another limitation of the conventional Rankine cycle is the fact that the temperature variation curve of water in the boiler, the acceptance curve, includes a plateau which corrupts the performance of the heat exchanger. The area between the cooling curve of the heat source and the acceptance curve of the working fluid is in fact a measure of the irreversibilities associated with the process. This contention has been raised elsewhere to explain the superior performance of counter-flow heat exchangers when compared to parallel-flow heat exchangers. Fig. 3a illustrates the phase change temperature plateau of a pure substance undergoing evaporation. The same phase change plateau exists during the condensation, but the condensation side of the problem is of less concern because the cooling medium is generally rather affordable. The constant temperature phase change constraint, together with the finite size of the heat exchangers, implies that the highest temperature the working fluid can reach in the conversion cycle is well below the temperature at which the warm effluents are released from the industrial plant, and the lowest temperature reached by the working fluid is greater than the cooling medium temperature. These two temperatures concurrently determine the pressure drop, the enthalpy variation and the work generation potential in the turbine of the conversion cycle.

Apart from the above thermodynamic considerations, the use of water as the working fluid in a low temperature Rankine cycle is likely to require extremely high velocities in the turbine, excessive friction losses, increased lubrication and sealing challenges, increased blade erosion risks, and ultimately an erratic turbine operation [18].

2.1. High vapour pressure working fluids

As opposed to water, ammonia is a substance that changes phase at relatively low temperatures, and it is thus referred to as a high vapour pressure chemical. High pressure ammonia vapour can theoretically be produced using the low temperature heat transported by industrial effluents and introduced into a turbine where, upon expansion, considerable work can be generated. The interest in ammonia is even more accentuated because of its

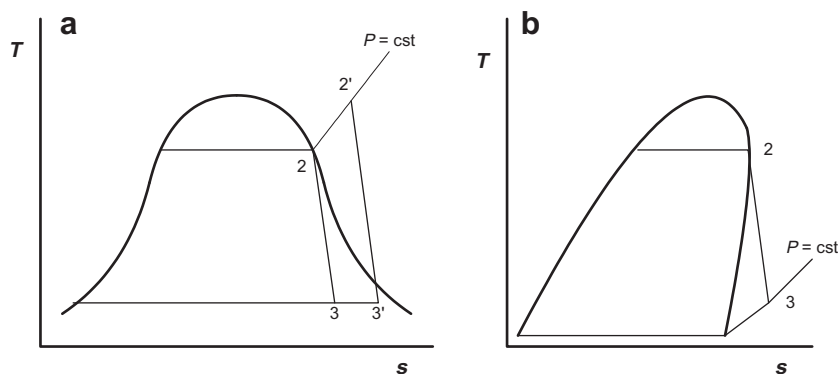


Fig. 2. Temperature–entropy diagrams for (a) wet and (b) dry fluids.

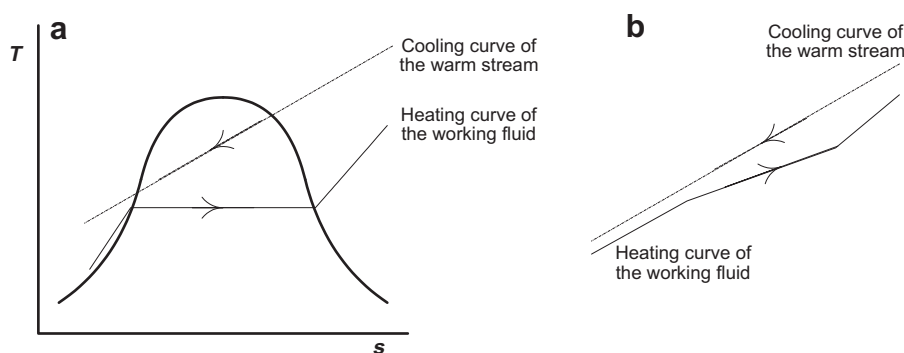


Fig. 3. (a) Phase change temperature plateau for a pure substance and (b) phase change over a finite temperature interval for a fluid mixture during evaporation.

affordability (6th most abundant chemical produced worldwide). Methane, ethane, propane, propadiene, and generally most short carbon chain hydrocarbons present high vapour pressures. Synthetic refrigerants R142b (chlorodifluoromethane) and R11 (trichlorofluoromethane) also present high vapour pressures compared to water, but which are lower than that of ammonia. Because of increasingly tight regulations on chlorofluorinated refrigerants [6], these will not be discussed further in this paper. Light alcohols methanol and ethanol constitute other examples of relatively high vapour pressure fluids with 3.5 bar and 2.2 bar saturation pressures at 100 °C, respectively.

While the above cited fluids are interesting when their pressures at the turbine inlet are concerned, this advantage turns into a shortcoming when their minimal allowed pressures at the turbine exit are considered. In fact, in order for ammonia for example to be condensed under atmospheric temperature conditions, its pressure has to be maintained above 10 bar. This significantly limits the pressure drop, the enthalpy variation and the work generation potential at the turbine when a Rankine-type power cycle using one of these fluids is used for the recovery and the conversion of low temperature waste heat into mechanical energy. The use of high vapour pressure chemicals as working fluids for power cycles would also result in a cost penalty (high pressure piping and equipment). In the meantime, their use in vapour compression refrigeration systems is perfectly justified [6].

Another disadvantage of high vapour pressure working fluids is that they are also in most cases wet fluids, *i.e.*, as water, they tend to prematurely condense during the expansion, and consequently need to be superheated before entering the turbine. Lastly, these fluids also go through a constant temperature and pressure plateau during the evaporation process, which negatively affects the quality of the heat transfer at the evaporator.

2.2. Organic working fluids

The use of organic fluids in Rankine-like power cycles using low temperature heat sources was introduced in an early paper by Iqbal et al. [20]. These authors identified a list of desired criteria which the perfect working fluid should meet. At the top of the list, a good working fluid should not condense during expansion to save for the extra superheating thermal energy consumption. It should present a large enthalpy variation over a short temperature range. Its pressure at the turbine inlet should not be too high in order to minimize stress on the equipment and prevent excessive power requirement for pressurization. Also the pressure at the turbine exit should not be too small in order to avoid high vacuum operation complications. At the same time, the ideal working fluid should present a low density in the vapour phase and a low viscosity in the liquid phase in order to reduce equipment size and head losses. While it is clear that the ideal working fluid will most probably never be developed, most of the desired properties are satisfied by a large number of organic fluids [5,6,11]. In particular, a large family of hydrocarbons tends to become dryer and dryer during expansion because of the overhanging shape of their temperature–entropy diagrams (see illustration on Fig. 2b). Along with other authors, Saleh et al. [21] have compared Rankine-like power cycles using different working fluids including high vapour pressure refrigerants, wet and dry fluids, and they concluded the superiority of cycles using dry fluids for the considered heat source temperature range.

After a closer examination at hydrocarbons, it was discovered in this study that short carbon chain chemicals have extremely high vapour pressures. At the other end of the spectrum, long carbon chain chemicals (eight or more carbons) present extremely low vapour pressures. For example, the saturation pressure for octane at

120 °C is only 0.47 bar, and the pressure has to be lower than 0.03 bar in order for it to condense under standard atmospheric operation conditions. As a result, a Rankine-like power cycle using octane would operate entirely under a high vacuum. The high potential hydrocarbons are then limited to those with 4 to 7 carbons, which is still a large pool considering alkanes, alkynes, alkenes and, for the same molecule, different structural arrangements or isomers. Alkanes are readily available, stable and feebly reactive. Pure alkenes are used in polymerization processes and are quite rare in nature. Alkynes tend to have high vapour pressures and are often subject to polymerization with detonations. The higher molecular masses of these hydrocarbons are also likely to mitigate the technological challenges experienced by low temperature steam Rankine power cycles (high turbine velocities, large friction losses, sealing difficulties, etc.) [18].

Table 1 gives some quantitative information about the vapour pressures and critical conditions for several chemicals, many of which we have initially considered as possible working fluids. Most hydrocarbon thermodynamic properties are taken from a reference handbook by Ferris [24].

It is worth noting that hydrocarbon flammability risks are relatively well handled in daily life, especially in an industrial plant. Cavallini [6] and Jung et al. [25] have also indicated that hydrocarbons are gaining wide acceptance for use in refrigeration, while Badr et al. [18] highlighted the little value of available quantitative stability data. In addition, the use of hydrocarbons as working fluids does not involve any fossil fuel cost concerns (hydrocarbons are actually less expensive than refrigerants [16]), as the working fluid is continuously recycled in the energy conversion unit. For the interested reader, a case-study including a summary of capital investment and O&M costs for a pilot 1.5 kW electricity generation plant using *n*-pentane as the working fluid has been published by Nguyen et al. [1].

Table 1
Vapour pressures and critical conditions for potential working fluids.

Usual name	Molar mass (g/mol)	Vap. pressure at 120 °C (bar)	Vap. pressure at 30 °C (bar)	Critical temperature (°C)	Critical pressure (bar)
Water	18.02	1.98	0.0427	374.2	221.18
Ammonia	17.03	91.0	11.6	132	112.78
Methane	16.04	–	–	–82.52	46.00
Ethane	30.07	–	46.5	32.28	48.84
R-141b	116.95	10.4	0.942	204.15	44
Propane	44.10	–	10.8	96.67	42.50
Cyclopropane	42.08	50.5	8.26	124.76	54.95
Propylene	42.08	–	13.1	91.61	46.12
i-Butane	58.12	28.3	4.08	134.98	36.48
n-Butane	58.12	22.2	2.84	152.01	38.00
i-Butene	56.11	26.5	3.57	144.75	40.00
Cyclobutane	56.11	16.7	1.85	186.78	49.80
1-Butyne	54.09	21.0	2.23	170.05	49.50
1,3-Butadiene	54.09	25.2	3.27	152.22	43.27
i-Pentane	72.15	10.9	1.09	187.24	33.81
n-Pentane	72.15	9.10	0.823	196.51	33.69
Neopentane	72.15	16.3	2.01	160.61	31.99
Cis-2-Pentene	70.13	8.96	0.794	202.78	36.78
Cyclopentane	70.13	6.50	0.513	238.61	45.08
1-Pentene	70.13	10.4	1.02	191.63	35.26
n-Hexane	86.18	3.90	0.251	234.15	29.69
Benzene	78.11	2.99	0.158	288.90	48.95
Cyclohexane	84.16	2.87	0.164	280.39	40.73
1-Hexyne	82.14	3.85	0.220	243.05	36.2
2-Hexyne	82.14	2.64	0.135	275.85	35.3
n-Heptane	100.20	1.82	0.0776	266.99	27.41
1-Heptene	98.19	2.07	0.0952	264.14	28.30
Cycloheptane	98.19	1.04	0.0373	331.15	38.40
Toluene	92.14	1.31	0.0489	318.64	41.08
n-Octane	114.23	0.471	0.0246	295.54	24.87

Organic fluids therefore present two major advantages when compared to water. On one hand, they have higher vapour pressures, *i.e.*, they potentially allow for higher pressures and enthalpies at the inlet of turbines. On the other hand, most organics do not have a tendency to condense during expansion, which precludes any requirements for superheat before the turbine inlet or high pressure exit conditions from turbines, in addition to also being valuable for turbine operation.

2.3. Ammonia–water mixture working fluid

Mixtures of pure substances change phase over a finite temperature interval, *i.e.*, not at a constant temperature as pure substances do. As a result, the acceptance curve of the working fluid and the cooling curve of the warm exhaust stream are closer to one another, which improves the heat exchanger performance. This was illustrated in Fig. 3, together with a comparison with the heat exchanger presenting a phase change temperature plateau. An important review article by Radermacher [19] discussed the use of mixtures as working fluids in power cycles and heat pumps. The ammonia–water couple already well established in absorption cooling captured most of the interest of the research community. In addition to being familiar, the ammonia–water mixture presents the advantage of a higher vapour pressure than water, and that vapour pressure can be adjusted by changing the mixture composition to better fit the temperature of the heat source. Moreover, because water and ammonia have close molar masses, the change in composition will not dramatically affect the design of the equipment and piping system. This has been cited as one of the advantages the ammonia–water mixture has over the lithium bromide–water mixture in absorption refrigeration.

The ammonia–water Rankine cycles introduced in the literature include different degrees of vaporization of the working fluid in the boiler accompanied by the use of separators and mixers. The ammonia rich high pressure vapour is directed towards the turbine inlet while the lean liquid solution is directed towards the turbine exit and mixed with the low pressure vapour before the condensation [3,14,26]. In the most recently reported application, the use of the ammonia–water mixture working fluid benefited from the availability of an extremely low temperature (–160 °C) heat sink which allowed expansion down to an exceptionally low pressure in the turbine [14].

The use of mixtures adds one more degree of freedom to the conversion unit, and represents an unquestionable advantage when compared to pure substance working fluid cycles. However, the fine-tuning and accurate control of the evaporation ratio in the boiler, together with the use of separators and mixers represent too big a challenge for plant operators. As a result, despite an impressive number of invention patents and 25 years of development, the Kalina technology [26] has yet to make a footprint in the commercial market. It is finally worth mentioning that the ammonia–water mixture, as each of its individual components, presents the same worrisome tendency to prematurely condense during expansion.

3. Analysis

With reference to the simplified Rankine cycle layout of Fig. 1, the heat source consists of 300 kg/s warm industrial gases comparable to standard air. The temperature at which the heat source is available is specified at the inlet of the evaporator and constitutes one of the parameters that will be varied in the course of the analysis. The working fluid in the cycle is specified. The evaporator is assigned the constraint that the exit temperature of the working fluid is 20 °C below the inlet temperature of the heat source. The

condenser is assigned the constraint of a working fluid exit at the saturated liquid state. The reference condensation temperature is 30 °C; this value will be varied in the course of the analysis. For the specified heat source and heat sink conditions, the higher and lower pressure levels in the conversion cycle are determined iteratively so as to satisfy a complete vaporization of the working fluid at the higher pressure and a condensation at the heat sink temperature. The selection of the pressure levels and the flow rates of the working fluid in the cycle is based on a maximum power output objective with the additional constraint that the moisture content of the vapour at the exit of the turbine should never exceed 5%. The two heat exchangers are adiabatic meaning that heat is only transferred from the warmer to the cooler fluid. Line pressure losses are neglected. The turbine and the pump are adiabatic. Their respective isentropic efficiencies are defined as follows:

$$\eta_{\text{isentropic}}(\text{turbine}) = \frac{h_2 - h_3}{h_2 - h_{3s}} \quad (1)$$

$$\eta_{\text{isentropic}}(\text{pump}) = \frac{h_4 - h_{1s}}{h_4 - h_1} \quad (2)$$

where h is the specific enthalpy. The numerical subscripts refer to locations on the cycle layout of Fig. 1 and the subscript s refers to an isentropic operation of the upstream component. These two isentropic efficiencies will be varied in the analysis, the reference value being 85% for both the turbine and the pump.

The net power output of the conversion cycle is the sum of the mechanical shaft power generated at the turbine and the pressurization power consumption:

$$\dot{W}_{\text{net}} = \dot{W}_{\text{turbine}} + \dot{W}_{\text{pump}} \quad (3)$$

where the mechanical power at the turbine and the pump are given respectively by:

$$\dot{W}_{\text{turbine}} = \dot{m}(h_2 - h_3) \quad (4)$$

$$\dot{W}_{\text{pump}} = \dot{m}(h_4 - h_1) \quad (5)$$

where \dot{m} is the working fluid mass flow rate in the cycle.

The heat transfer rate from the heat source to the working fluid at the evaporator and the heat transfer rate from the working fluid to the heat sink at the condenser are, respectively:

$$\dot{Q}_{\text{evaporator}} = \dot{m}(h_2 - h_1) \quad (6)$$

$$\dot{Q}_{\text{condenser}} = \dot{m}(h_4 - h_3) \quad (7)$$

Finally, the overall conversion cycle thermal efficiency is defined as:

$$\eta_{\text{cycle}} = \frac{(h_2 - h_3) + (h_4 - h_1)}{(h_2 - h_1)} \quad (8)$$

where the numerator and the denominator, respectively represent the net mechanical energy generation rate by the cycle and the rate of the thermal energy extraction from the heat source.

The thermodynamic properties of the working fluid and those of the heat source are calculated using a commercial software. Validation tests were performed which confirmed that the program can reproduce the saturation curves of many of the considered candidate working fluids. Sample cycle calculations were also carried out with water as the working fluid, and the results were successfully compared with hand calculations using property tables available in a reference thermodynamic textbook [2].

4. Results and discussion

The remainder of this paper is dedicated to the performance analysis of power generation systems that recover heat from low temperature heat sources and use organic chemicals as working

fluids in Rankine-like cycles in order to produce mechanical energy. It is recognized that there are several possible ways of using the thermal energy conveyed out by the industrial warm used fluids; however it is not the objective of this paper to compare different recovery techniques. Such a comparison would in fact require a comprehensive knowledge of the specific energy needs of the industrial plant, the needs of its immediate neighbourhood, as well as the local energy costs.

4.1. Selection of the working fluid based on the temperature of the heat source

Different fluids have varying abilities for extracting heat from a warm body, and in generating mechanical energy when expanded in a turbine. Furthermore, power cycles using different working fluids have different heat-to-work conversion efficiencies. The thermodynamic properties associated with these abilities (heat capacity, enthalpy and exergy) depend on the temperature and pressure variation intervals, and these intervals are in turn functions of the temperatures at which the high and the low temperature media are available.

A selection of chemicals from Table 1 was tested as working fluids for Rankine-like power cycles generating work from a low temperature industrial gas stream as sketched in Fig. 1. For each potential working fluid, the maximal and minimal pressures in the conversion cycle and, where applicable, the vapour superheat at the turbine inlet were determined and the mechanical power generation potentials were calculated. The procedure has been detailed in Section 3 above. Fig. 4 below compares the performances of cycles using different working fluids for different heat source temperatures.

Fig. 4 indicates that the work generation potential is generally low when the heat source temperature is low, and it monotonically improves as the temperature at which the warm industrial waste fluids are available increases. The figure also shows large deviations between recovery cycles using the six considered working fluids. By far, the cycle using water displayed the highest performance in terms of power generation potential for a unit mass flow rate of circulated working fluid. For example it overperformed by 80% compared to the second best cycle, using ammonia, over the entire heat source temperature range considered.

The degree of utilization of the available heat conveyed by the warm industrial effluents is not apparent in Fig. 4. Another way of assessing the performances of power cycles using different working fluids would therefore be to compare the thermodynamic efficiencies of the conversion of the recovered thermal energy into mechanical energy (see Fig. 5).

Fig. 5 corroborates the observed trend in Fig. 4 of low performances at low temperatures, which improve as the temperature at which the heat is available increases. However, in contrast to Fig. 4, the conversion cycle using water as the working fluid is now the least performing, and the cycle using benzene becomes the most efficient for the entire considered heat source temperature range. The relatively 'good performance' of the water Rankine cycle observed in Fig. 4 thus failed to disclose an extraction and use, i.e., a utilization, of much more thermal energy from the warm gases compared to the power cycles using the other working fluids. Incidentally, the capacity to extract heat from a given heat source is one of the beneficial properties that explains the success of water as a working fluid for high temperature conventional Rankine cycles.

It is also important to mention that the three dry fluids considered in Fig. 5 (*n*-heptane, benzene, *i*-pentane) proved to be superior compared to the three wet fluids (propane, ammonia, water). This is consistent with previous results and predictions [16,20,21].

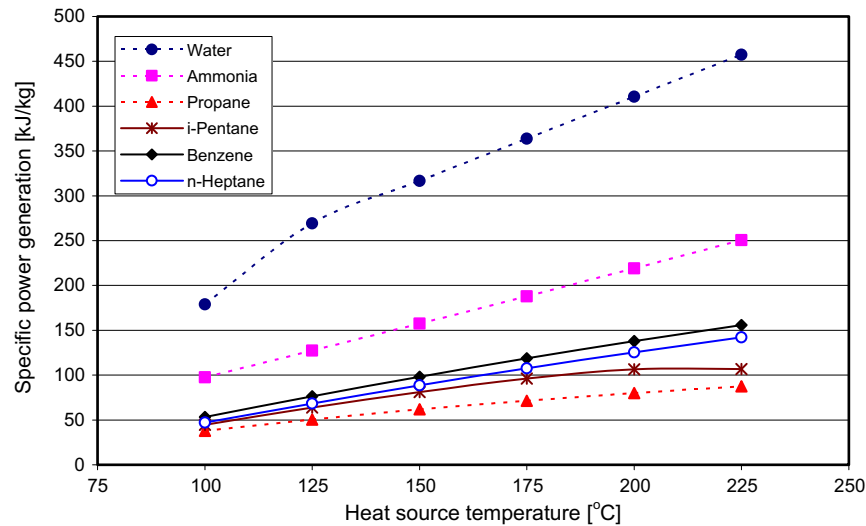


Fig. 4. Power generation potential per unit working fluid mass flow rate by Rankine-like power cycles as a function of the heat source temperature.

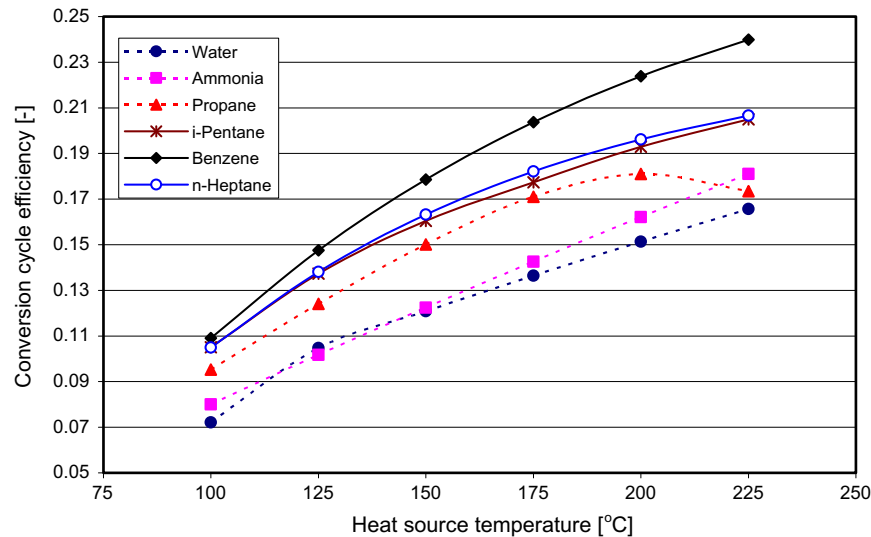


Fig. 5. Energy conversion efficiency by Rankine-like cycles using different working fluids as a function of the heat source temperature.

Another important result observed in Fig. 5 is the intersecting of the efficiency variation curves of the Rankine cycles using different working fluids as the temperature of the heat source changes. More specifically, the efficiency of the water-based cycle and that of the ammonia-based cycle intersect with each other several times over the considered heat source temperature interval. Furthermore, propane appears to be a better working fluid than water and ammonia for a large part of the heat source temperature range, but the efficiency variation curve associated with propane hits a maximum at approximately 200 °C, then inverts its slope. Eventually, propane ceased to be a better working fluid than ammonia when the considered heat source temperature reached 225 °C. On the other hand, while the efficiency of the water-based cycle continues its ascending progression to reach 37% when the temperature of the heat source is 700 °C, the efficiency of the propane-based cycle continues to decrease down to about 10% for the same heat source temperature. The general profile of the propane-based conversion cycle efficiency is not unique, and is related to the critical conditions of the chemical (see Table 1).

It is recalled that other authors have adopted a pure theoretical second law approach and compared the conversion efficiencies of the Rankine-like cycles using different working fluids over a comparable heat source temperature range [5]. As a result, these authors were able to verify the improved power cycle efficiencies as the heat source temperature increased, as well as the intersecting of different efficiency curves. It should ultimately be noted that the working fluid selection for Rankine-like power cycles strongly depends on the temperature of the heat carrying medium.

A third performance indicator can be defined by assessing the net work generation potential by a Rankine-like cycle designed for upgrading the thermal energy rejected by a given industrial plant. To illustrate this third performance indicator, an industrial process that continuously discharges 300 kg/s of warm air at a given temperature between 100 °C and 225 °C was considered. A Rankine-like conversion cycle adapted to this industrial plant recovers the maximum possible thermal energy from the exhausts and converts it into the largest possible amount of mechanical energy. The equipment and other design specifications presented in Section 3 remain unchanged.

For all considered working fluids, the maximal and minimal pressures were first determined, and then the working fluid flow rates were maximized in order to extract the largest possible part of the heat conveyed out by the warm industrial fluids. Mechanical work data have been calculated for different heat source temperatures, and Fig. 6 below has been generated.

As already suggested by Figs. 4 and 5, Fig. 6 indicates that the net work generation potential by a Rankine-like cycle is low when the heat source temperature is low and regularly increases as the heat is available at higher and higher temperatures. Fig. 6 also indicates a better performance of organic fluids as compared to water and ammonia. As an example, the use of benzene as the working fluid instead of water would result in a 45% to 53% higher power generation over the entire considered heat source temperature range. As in previous paragraphs, several intersecting of different curves are observed, confirming that the actual temperature at which the heat is available is a key design criterion for the recovery–conversion system. It has for example been observed in a previous study that at higher temperatures, benzene performs poorly compared to toluene and p-xylene [16].

In Fig. 6, it can also be seen that the slope of the net power generation curve associated with propane significantly decreases past about the 200 °C heat source temperature. However, this is not a precursor of an upcoming decrease in the net power generation potential. The power generation potential will indeed continue to increase, and a simulation with a 700 °C heat source temperature indicated that the potentials for all considered working fluids significantly increased, but water has become the most attractive working fluid, followed by ammonia and benzene. This is an expected result that confirms the superiority of water as the working fluid for conventional fuel burning power plants.

Finally, we believe that the results observed in Fig. 6 (net work generation potentials) would most probably be of greater interest for industry managers than those in Fig. 5 (thermodynamic energy conversion efficiencies) and those in Fig. 4 (work generation potentials per unit circulated working fluid mass flow rate). That said, to obtain the results seen in Fig. 6, and consistently with results in Fig. 4, the circulated working fluid flow rates are not the same for the six considered recovery–conversion cycles. The equipment and piping are thus not the same, and the associated costs may vary.

In the remainder of this paper, the effects of some selected external and internal operating parameters on the overall performance of the recovery–conversion power cycles are discussed.

4.2. Influence of cooling medium temperature

While a high heat source temperature allows for the production and introduction of a high pressure working fluid vapour into a turbine, the availability of a very low temperature heat sink allows for an expansion down to a lower and lower pressure inside the turbine. In either case, the result is an important pressure drop, a large enthalpy variation and an important work generation potential.

Table 2 summarizes results obtained by varying the temperature of the heat sink used by the low temperature Rankine-like conversion cycle. The temperature of the warm industrial gas exhausts is maintained at 150 °C. The heat source flow rate and other constraints on the equipment are described in Section 3. The working fluid considered for this simulation is benzene. The temperature of the cooling medium has been gradually raised, forcing condensation to occur at increasing temperatures from 5 °C to 35 °C. Accordingly, the condensation pressure gradually increased, and the pressure drop within the turbine shrank. The mass flow rate of the working fluid was then maximized in order to maximize the utilization of the available free thermal energy.

The table shows a progressive alteration of all performance indicators: the power generation per unit working fluid mass flow rate has dropped by 29%, the efficiency of the very thermodynamic conversion of heat into work has dropped by 22%, and the net power generation potential by the cycle has dropped by about 38% for the considered condensation temperature interval. The somewhat less important decrease in the thermodynamic conversion efficiency as compared to the two other indicators is explained by the fact that the power generation potential decreases faster than the thermal energy utilization (see Eq. (8)).

Table 2
Effects of cooling medium temperature on power cycle performance (benzene).

Parameter	Turbine			Conversion cycle	
	Condensation temperature (°C)	Inlet pressure (bar)	Exit pressure (bar)	Generated power (kJ/kg)	Overall efficiency (%)
	5	3.79	0.0481	130	21.9
	10	3.79	0.0625	123	21.1
	15	3.79	0.0803	117	20.3
	20	3.79	0.102	110	19.5
	25	3.79	0.129	104	18.7
	30	3.79	0.161	98.2	17.9
	35	3.79	0.199	92.2	17.0

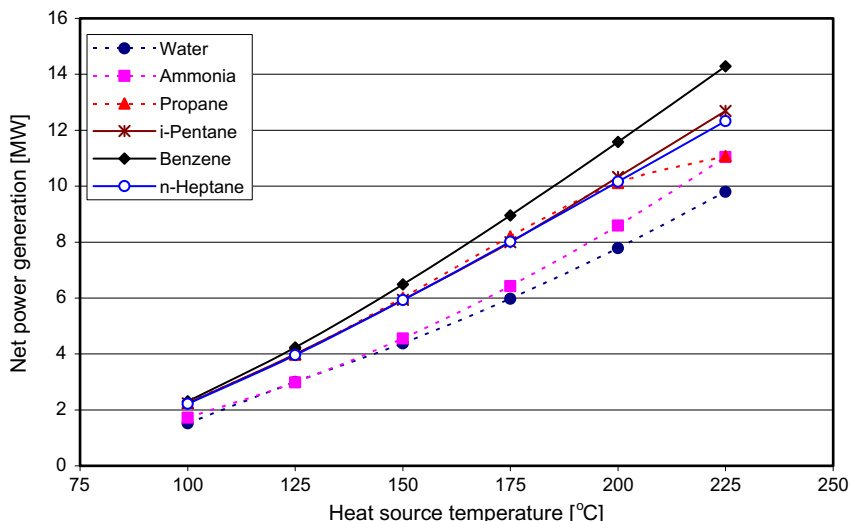


Fig. 6. Net power generation by Rankine-like cycles using different working fluids as a function of the heat source temperature.

For comparison purposes, for a heat source at 150 °C and two different heat sink temperatures of 5 °C and 35 °C, an ideal Carnot power cycle would return the conversion efficiencies of 34% and 27%, respectively. The relative deviation between the actual cycle and the Carnot cycle therefore does not display any significant heat sink temperature effect. Finally, the qualitative trends in Table 2 apply to conversion cycles using other working fluids and other pressure and temperature conditions.

4.3. Influence of working fluid flow rate

The working fluid flow rate is an extremely important parameter in the analysis of power generation cycles using residual industrial thermal energy. Among other things, it explains the quite impressive difference between Figs. 4 and 6. Water first proved to be the fluid that, on a unit mass flow rate basis, allows for the most important mechanical power generation. However, this superior performance was realized at the expense of a more important utilization of the available thermal energy. In fact, when the mass flow rates for different working fluids were maximized in order to make use of the largest possible part of the available thermal energy, the trends inverted, and water became the least interesting working fluid. The net power generation potential is in fact directly proportional to the circulated working fluid flow rate (Eqs. (3)–(5)). The latter is constrained by the temperature and the flow rate of the warm industrial fluid stream, as well as the design specifications of the heat exchangers. The working fluid flow rate has however no effect on the first law efficiency of the thermodynamic conversion (Eq. (8)).

4.4. Influence of turbine and pump second law efficiencies

The performances of individual components affect the overall performance of the proposed energy recovery–conversion cycle. The size, the design and an efficient maintenance routine of the primary heat exchanger together directly determine the quality of the heat transfer between the waste warm gases and the working fluid. On the other hand, depending on the design of the turbine (blade geometry and material, thermal insulation, etc.), the actual operating conditions (viscous dissipation, nature of fluid, vapour quality, etc.) and the maintenance routine, a turbine may operate very close or very far from its ideal isentropic conditions. In order to simulate these conditions, the turbine second law efficiency has been arbitrarily varied between 70% and 95%. Most important calculated data are shown in Table 3. The considered working fluid is *n*-heptane with maximum and minimum pressures of 6.96 bar and 0.0761 bar, respectively corresponding to a vaporization temperature of 180 °C and a condensation temperature of 30 °C. All other design specifications are described in Section 3.

For the same amount of thermal energy extracted from the heat source (unchanged evaporator inlet and exit conditions), a quasi-linear increase in the mechanical energy generation is observed

when the second law efficiency of the turbine is increased. In the meantime, the temperature of the working fluid at the exit of the turbine is decreasing, indicating an increase of the enthalpy variation across the turbine. This is consistent with the boost in power generation at the turbine on one hand, and consistent with the decline of the heat transfer rate to the cooling medium at the condenser on the other hand. The heat recovered at the evaporator is not affected and the pressurization work is unchanged, so that the increase in work generation readily translates into an improvement of the energy conversion cycle efficiency. In quantitative terms, the 36% increase of the turbine second law efficiency has resulted in a 36% increase of the work generation potential, and an increase in the same order of magnitude of the overall first law cycle efficiency. It is again remembered that from a technological point of view, a high turbine isentropic efficiency is synonymous to higher equipment and O&M costs.

The same analysis has been conducted on the influence of the pump second law efficiency on the overall conversion cycle performance. Isopentane was selected as the working fluid for this simulation with an 8.13 bar saturated vapour at 105 °C at the exit of the evaporator, and a 1.08 bar pressure at the exit of the turbine corresponding to a saturation condensation of 30 °C. The turbine second law efficiency was held constant at 85% while pump efficiency was gradually varied from 70% to 95%.

Results in Table 4 show a decrease in the energy demand by the pump as its efficiency is increased, together with a reduction in the preheating of the working fluid during the pressurization. Consequently, the thermal energy required at the evaporator for the heating and vaporization of the working fluid slightly increases. The net mechanical power output has slightly increased as well, and the energy recovery–conversion cycle's overall efficiency only experiences a very slim improvement.

4.5. Influence of the addition of a preheater

In most of the simulated cases presented in this paper, the working fluid is a dry hydrocarbon, that is, it still carries a non-negligible sensible heat at the exit of the turbine, and that sensible heat is simply disposed of at the condenser. However, in actual industrial thermal power plants, it is a common practice to insert another heat exchanger called a preheater (also often referred to as a regenerator or an internal heat exchanger in the literature) that uses part of the thermal energy of the low pressure vapour exiting the turbine to preheat the high pressure liquid working fluid flowing towards the evaporator [2,11]. This is generally accepted to result in an increase of the conversion plant efficiency.

The influence of inserting a preheater into the low temperature recovery–conversion power cycle is discussed in this paragraph by considering a heat source at 100 °C. The working fluid is isobutane with a 13.7 bar pressure at the inlet of the turbine and 4.1 bar at the exit of the turbine. At this point, the working fluid is still superheated (temperature of 41 °C whereas the saturation temperature

Table 3
Effects of turbine efficiency on power cycle performance (*n*-heptane).

Parameter	Turbine			Condenser	
	Inlet pressure (bar)	Exit pressure (bar)	Power generated (kJ/kg)	Rejected heat (kJ/kg)	Cycle Overall efficiency (%)
70	6.96	0.0761	103.3	530.9	16.1
75	6.96	0.0761	110.7	523.6	17.3
80	6.96	0.0761	118.1	516.2	18.5
85	6.96	0.0761	125.4	508.8	19.6
90	6.96	0.0761	132.8	501.4	20.8
95	6.96	0.0761	140.2	494.4	22.0

Table 4
Effects of pump efficiency on power cycle performance (i-pentane).

Parameter	Pump	Evaporator	Turbine		Cycle
			Inlet pressure (bar)	Exit pressure (bar)	
Pump efficiency (%)	Power input (kJ/kg)	Heat recovered (kJ/kg)			Overall efficiency (%)
70	1.648	454.2	8.13	1.08	13.68
75	1.539	454.3	8.13	1.08	13.70
80	1.443	454.4	8.13	1.08	13.71
85	1.358	454.5	8.13	1.08	13.73
90	1.283	454.6	8.13	1.08	13.75
95	1.215	454.7	8.13	1.08	13.76

Table 5
Effects of the addition of a preheater on power cycle performance (i-butane).

Parameter Preheater savings (kJ/kg)	Evaporator Heat recovered (kJ/kg)	Turbine Power output (kJ/kg)	Condenser Heat rejected (kJ/kg)	Cycle Overall efficiency (%)
0.000	380.9	40.86	342.1	10.18
3.809	377.1	40.86	338.3	10.29
7.618	373.3	40.86	334.5	10.39
11.43	369.5	40.86	330.7	10.50
15.24	365.7	40.86	326.9	10.61
19.05	361.9	40.86	323.1	10.72

is 30 °C). It enters the preheater where different fractions of its sensible heat are transferred to the high pressure liquid exiting the pump. The first simulation in Table 5 corresponds to the base case, i.e., without the preheater, the second simulation corresponds to the case where the preheater supplies 1% of the thermal energy needed for the vaporization, the third simulation corresponds to the case where the preheater covers 2% of the total thermal energy demand, and so on.

It was observed that, for the same working fluid flow rate, both the thermal energy extracted from the warm gases at the evaporator and the heat rejection to the environment through the condenser progressively decrease, and the overall cycle efficiency is improved. Results on the very last line for example indicate that the integration of a preheater that contributes 5% to the energy demand for the vaporization improves the conversion cycle efficiency by a little more than 5% for the same net power generation.

Alternatively, one could have made use of the same external thermal energy and use the energy savings the preheater creates to circulate a larger working fluid flow rate in order to generate more mechanical work at the turbine. The outcome would be the same in terms of the efficiency of the thermodynamic heat-to-work conversion. Many other combinations of extra heat exchangers have been discussed in the literature [5,10,26] and are currently in use on industrial Rankine-like thermal power plants.

4.6. Use of organic fluid mixtures

As discussed in Section 2, the use of mixtures as working fluids would allow for an evaporation process over a finite temperature range, and thus a better quality for the heat transfer between the heat source and the working fluid at the evaporator. Some researchers in the literature [9,22,25] have therefore suggested the use of organic fluid mixtures in order to fully benefit from all the advantages these fluids have to offer: higher vapour pressures than water, no need for superheat, and a better heat transfer performance at the evaporator. While our preliminary investigations have not confirmed these predictions, this is one of the development directions of the present study.

5. Conclusions

The recovery of the low temperature thermal energy rejected by industrial installations during normal operation and its conversion into a higher quality mechanical energy has been discussed in this paper. First, the limitations of the conventional Rankine thermodynamic cycle have been highlighted: the low vapour pressure of water, the low quality of the heat transfer at the evaporator because of the phase change temperature plateau, and the superheat requirement as a consequence of the intrinsic tendency of water to prematurely condense during expansion. These limitations guided the research towards more appropriate working fluids to use in Rankine-like recovery–conversion power cycles. The advantages

and disadvantages of several alternative working fluids have been presented followed by a parametrical study on one of the most promising solutions, which is the use of dry organic fluids. The latter fluids in fact have higher vapour pressures than water, and most importantly, they do not require superheating before they are admitted into the turbine.

This study illustrates the existence of different performance indicators reflecting the fluids' different abilities in extracting energy from the heat source (net specific power output) on one hand, and their ability to convert the extracted thermal energy into mechanical energy (cycle thermodynamic efficiency and net power output) on the other hand. The recovery cycle using water as the working fluid performed the best in terms of the ability to extract energy from the heat source, and was the one which produced the largest amount of work per unit mass flow rate of circulated working fluid. However, the water-based power cycle displayed a poor conversion efficiency of the extracted heat into work. Finally for a given heat source at a temperature between 100 °C and 225 °C, the cycle using water performed the poorest in terms of the net power generation potential when compared to ammonia, propane, isopentane, benzene and *n*-heptane.

The parametrical study showed that all three performance indicators were monotonically and significantly improved as the temperature at which the free thermal energy was available increased. All performance indicators were also improved when the temperature of the cooling medium was lowered which, as high heat source temperatures, led to larger pressure drops and enthalpy variations within the turbine. It was also possible to observe that the net power production increased linearly with the mass flow rate of the working fluid, though the latter is constrained by the availability of the heat source and the design of the heat exchanger. The effects of other individual component performances on the overall cycle performance were also thoroughly investigated and important results were obtained and discussed.

Finally, a modification of the standard Rankine cycle layout by the addition of an internal heat exchanger (the preheater) that recovers part of the sensible heat of the working fluid flowing out of the turbine and passes it on to the high pressure liquid before it enters the evaporator has been discussed, and has proven to be an effective means of improving the overall cycle efficiency.

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