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Article

Waste Heat Recovery Systems with Isobaric Expansion Technology Using Pure and Mixed Working Fluids

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Abstract: Economic expedience of waste heat recovery systems (WHRS), especially for low temperature difference applications, is often questionable due to high capital investments and long payback periods. By its simple design isobaric expansion (IE) machines could provide a viable pathway to utilize otherwise unprofitable waste heat streams for power generation and particularly for pumping liquids and compression of gases. Different engine configurations are presented and discussed. A new method of modelling and calculation of the IE process and efficiency is used on IE cycles with various pure and mixtures as a working fluid. Some interesting cases are presented. It is shown in this paper, that the simplest non-regenerative IE engines are efficient at low temperature differences between a heat source and heat sink. Efficiency of non-regenerative IE process with pure working fluid can be very high approaching Carnot efficiency at low pressure and heat source/heat sink temperature differences. Regeneration permits to increase efficiency of the IE-cycle to some extent. Application of mixed working fluids in combination with regeneration permits to significantly increase the range of high efficiencies to much larger temperature and pressure differences.

Keywords: Isobaric expansion engines; heat driven pump; compressors; low-grade heat; mixed working fluids

1. Introduction

Today the lion's share of electricity generated globally is consumed by pumps and compressors. For instance pumping systems consume around 20% of world's electricity demand [1]. Compressed air systems alone account for 10% of electricity consumed in the U.S. and European Union industry, 9.4% in China [2].

Most of the electricity production is based on big power stations using heat from the combustion of fossil fuels (such as coal and natural gas) accompanied with CO_2, NOx and particulate emissions. Other methods (solar photovoltaics, wind, tidal, geothermal etc.) constitute a lesser part of electricity market. According to a recent evaluation, 72% of primary high-grade energy is lost after conversion. 63% of this lost energy is waste heat with temperature below 100°C [3]. In fact, the only way to convert low-grade heat to power is based on the Organic Rankine Cycle (ORC). However, the real contribution of ORC produced power to the world's electricity generating capacity is insignificant. For instance, global installed power capacity in 2016 reached 6473 GW of which 61.1% (3954GW) was based on thermal energy sources [4], whereas ORC represented only around 2.7 GW [5].

In other words, the contribution of ORC accounts for 0.04% of the global electricity generating capacity and 0.07% of the generating capacity based on thermal (heat) energy sources. Therefore, it can be said with confidence that low grade and waste heat resources in power generation remain almost untapped. Such an inconsequential contribution of ORC is caused mostly by economic reasons. ORC-based power generation is too expensive in case of ultra-low-grade heat sources (below 100°C) and low power ranges [6],[7].

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For instance, the share of units with power below 500kW does not exceed 2% of total installed ORC capacity [5] i.e. millions of small low-grade and waste heat sources remain unused.

Utilization of waste heat thus provides a possibility to increase efficiency of industrial processes and monetize an otherwise lost energy stream. Waste heat sources are plenty with EU potential estimated at 300TWh/yr. of which one third part in the low temperature range of 100-200°C [8].

Isobaric expansion engines defined in [9] are the oldest type of heat-to-mechanical power converters known. James Watt, Thomas Savery, Denis Papin and Thomas Newcomen, standing at the cradle of this invention [10],[11], set forth the fundamental principles of conversion of heat into work bringing about the industrial revolution. Later on, these machines were displaced by more efficient well-known water steam expansion machines such as piston steam engines as well as steam turbines. Due to simple design by utilization of readily available components, IE machines provide a viable pathway to utilize waste heat steams in the low temperature (difference) range.

However, as it was shown in [9], IE engines can be efficient using some working fluids in a combination with heat regeneration. It will be demonstrated that for the various types of IE-engines configurations attractive results can be obtained for low and ultra-low temperatures waste heat recovery. With a thermodynamic model and the use of the REFPROP 10 database [12], results on some single component and binary mixtures are presented, proving the benefits of mixed working fluids in IE machines on a theoretical basis. We will show that a wide temperature operating range can be achieved with binary mixed working fluids.

As a result, IE technology can be ideally fit for pumping liquids and compression of gases because they permit to use heat for the compression and pumping directly, i.e., without the intermediate step of electricity generation, transmission and further conversion back to mechanical energy typical of today's industry. IE machines can also be attractive for electricity generation in low and medium power ranges.

2. Worthington type isobaric expansion engines

Worthington type heat engines relate to one out of two classes of IE-machines known. Bush-type engines [13], forming the second class, perform a different working cycle. Theoretically, Bush-type engines are more efficient than Worthington ones; however, the high efficiency can only be realized if dead volume of the heat exchangers does not severely influence the engine power and efficiency [14]. On the contrary, Worthington-type engines are less sensitive in this respect and can use any off-the-shelf heat exchanger. In this publication only Worthington-type engines will be considered as they are more market ready.

The Worthington-type IE machines can literally use all types of heat exchangers, including the most economical ones (gasketed and brazed plate, pillow plate etc.). Moreover, the large internal volume can even be useful for dampening pressure pulsations. In addition, this permits Worthington engines to use multiple heat sources and heat sinks with different temperatures. To demonstrate the wide variety of possible Worthington engine configurations some of them are outlined below.

System configurations

Basic principle of one out of many possible modifications of Worthington-type engines is shown in Figure 1.

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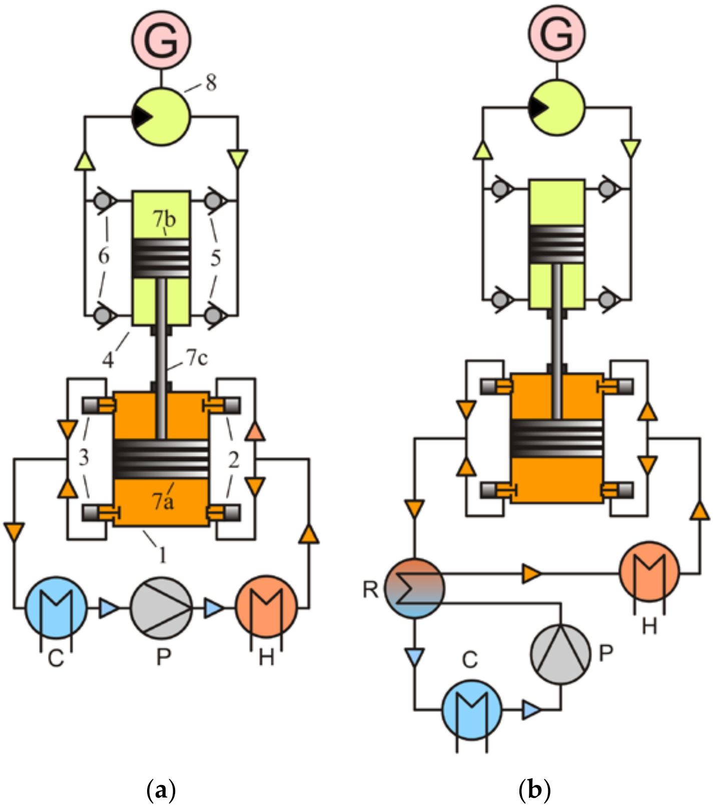

Figure 1. (a) Simplest non-regenerative Worthington-type engine, **(b)** Regenerative Worthington-type engine.

The engine consists of a power or driving cylinder (1) with a pair of admission valves (2) and a pair of exhaust valves (3), as well as pumping cylinder (4) with a pair of suction valves (5) and a pair of discharge valves (6). The driving and pumping cylinders are equipped with driving and pumping pistons 7a and 7b respectively, rigidly connected by a rod 7c. As a result, both pistons and the rod form a so-called differential piston moving together as one unit. The engine scheme also includes a feed pump (P), heater/evaporator (H) and cooler/condenser (C). If the engine is used as a shaft power or electric generator it can be equipped with a hydraulic motor (8) and electric generator (G). In operation the feed pump (P) delivers a liquid driving fluid to the heater (H) where it is heated and turned into vapor. The hot pressurized vapor of the driving fluid is supplied to the upper and lower parts of the driving cylinder alternately, providing a reciprocating motion of the differential piston (7a-7b-7c). The spent working fluid exhausts through the exhaust valves (3). The driving fluid does not expand in the cylinder providing a constant force for the pumping in the pumping cylinder; therefore, each admission and each exhaust valve in the opposite chamber of the driving cylinder are open during every half of the cycle. The spent working fluid exhausted from the driving cylinder goes to the cooler/condenser (C) and then returns to the feed pump (P).

The pumping cylinder operates as a typical double-acting pump with self-acting valves. At certain conditions the efficiency of the engine can be increased several times by means of heat regeneration. The regenerative Worthington-type engine is shown in Figure 1 (right). In this case the heat of the hot fluid exhausted from the driving cylinder is used for preheating of the cold driving fluid in the regenerator (R) installed upstream the heater (H). This increases efficiency by decreasing the consumption of the heat used for the working fluid heating in the heater (H). The regenerator also decreases the heat load on the cooler/condenser resulting in a smaller heat exchange area and size of the condenser.

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In case of high temperature operation, excluding conventional positive seals, a seal-less design can be used. Such a single-acting machine, Figure 2, consists of a cylinder (1) with admission and exhaust valves (2) and (3), a free piston (4) and a diaphragm unit. The diaphragm unit includes two semispherical covers (5) and (6) with a diaphragm (7) clamped in between. The upper cover (6) is equipped with suction and discharge valves (8) and (9). The piston (4) here plays only a role of a heat barrier dividing the hot vapors of working fluid (shown in orange) in the lower part of the cylinder (1) and cold liquid working fluid (shown in blue) in the upper part. In fact, in this design the cold liquid working fluid acts as a driving piston. It transmits the pressure of hot working fluid through the diaphragm (7) to the liquid to be pumped (shown in yellow-green).

Apparently, if the liquid to be pumped can be used also as the driving working fluid, the diaphragm unit can be eliminated. In this case the suction and discharge valves (8) and (9) can be installed directly in the upper part of the cylinder (1). Such a pump can be used, for instance, for pumping of liquefied gases, light hydrocarbons on refineries, in chemical industry etc.

In case of pumping warm or hot liquid the cylinder with the thermal barrier piston can be eliminated at all and the diaphragm unit plays the role of the driving and pumping cylinders at the same time. Here the admission and exhaust valves related to the driving part are also installed directly in the lower cover (5) the diaphragm unit. The diaphragm IE-pump is similar to that of the piston IE-pump; however, the initial and final pressures both in the driving and pumping cylinders are equal. The main advantage of the diaphragm-based design is an almost frictionless and leak-free operation.

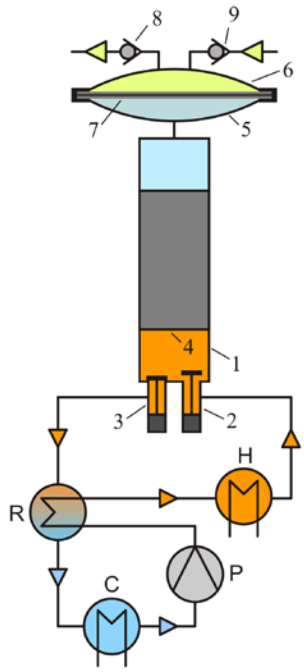

Figure 2. Regenerative Worthington-type engine with thermal barrier piston and diaphragm.

If the IE-machine is used only as a shaft power/electricity generator, the useful work can also be extracted directly from the driving piston by means of some kinematic (crank, swash plate, etc.) mechanism rather than by the hydraulic output. The example with a crank gear is shown in Figure 3 (left). Such a design reminds an old fashion steam engine with a very late cut-off. The admission valve remains open during the whole working stroke and closes at top dead center of the piston. Then the discharge valve opens and remains open during the whole back stroke of the piston. When the piston reaches the bottom dead center, the discharge valve closes and the admission valve opens starting a

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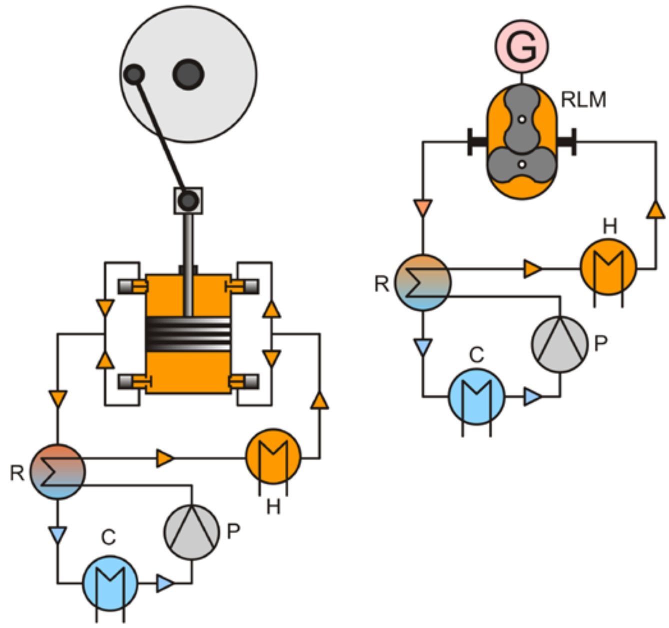
new working cycle. The reciprocating piston with the crank gear mechanism can be replaced with different rotary piston machines, more convenient in some cases.

A straight forward example of such a machine is a so-called rotary lobe or Roots compressor/blower operating in a reverse mode, i.e., as pressure-to-shaft-power converter shown in Figure 3 (right). The set-up also has a pump (P), heater (H), regenerator (R) and cooler (C). To convert pressure of the hot working fluid to shaft power a special positive displacement rotary lobe machine (RLM) is applied. Such an arrangement is similar to a conventional ORC installation. Roots machines were investigated as ORC expanders [15],[16]. However they do not perform expansion because a so-called built-in volume ratio is close to one i.e. in the Roots converter vapor does not expand during the pressure-to-work conversion. In fact, Roots machines operate as typical hydraulic motor rather than a gas expansion machine. As a result, the functionality of the Roots machine does not differ from that of reciprocating Worthington-type ones outlined above. Moreover, the rotary lobes play the roles of pistons and valves at the same time. Such a simplicity in a combination with high volumetric and mechanical efficiencies as well as low costs makes the Root's-based set-ups appealing. The installation does not produce inertial forces. In addition, power capacity of the biggest Roots machines can reach megawatt power range.



(**a**) (**b**)

Figure 3. Worthington-type engine with crank gear ***(a)***. Worthington-type engine with rotary lobe machine ***(b)***.

However, the Roots machines cannot work at high temperatures and high pressure drops. For higher pressures and temperatures other types of volumetric machines are necessary.

3. Cycle Calculations

Heat regeneration processes are rather difficult for modelling, above all in case of cyclic processes. In IE engines using dense working fluids the problem is complicated due to phase changes or operation near the critical point. At such conditions heat capacity is not constant but a strong function of both temperature and pressure. As a result, the second law of thermodynamics poses limitation on the possible heat regeneration degree. This implies that changes of the enthalpies of the fluids/streams participating in the heat exchange process at each temperature level (within the temperature engine cycle temperature interval) matter.

The expansion-compression work performed by the engine per cycle is:

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$$W_E = (P_H - P_L)\Delta V_E \tag{1}$$

where P_H and P_L are the high and low cycle pressures (pressures during the expansion and compression processes), and ΔV_E is the change of vapor volume in the cylinder.

If the cylinder volume at the beginning of the admission process (clearance volume) is negligible:

$$\Delta V_E = \frac{m}{\rho(T_H, P_H)} = mv(T_H, P_H) \tag{2}$$

where m is the total mass of the working fluid involved in the cycle and ϱ and v are density and specific volume of the working fluid.

The pumping work, assuming an adiabatic process with negligible changes in kinetic and potential energies, can be determined as:

$$W_P = m\left(h\left(T_{P,out}, P_H\right) - h(T_L, P_L)\right) \tag{3}$$

where $h(T, P)$ is the enthalpy of the working fluid per unit mass (specific enthalpy), and $T_{P,out}$ is the discharge temperature of the pump.

Assuming an isentropic pump operation the discharge temperature can be found from the constant entropy equation:

$$S(T_L, P_L) = S\left(T_{P,out}, P_H\right) \tag{4}$$

The net value of the work produced during the cycle is found as:

$$W = W_E - W_P \tag{5}$$

The working fluid is heated up in a heater from the pump discharge temperature (usually slightly above the temperature of the cooler or the low cycle temperature) to the desired inlet temperature of the power cylinder (high cycle temperature T_H).

Without heat regeneration the amount of heat supplied to the working fluid in the heater is

$$Q_H = m\left(h(T_H, P_H) - h\left(T_{P,out}, P_H\right)\right) \tag{6}$$

The thermal efficiency, defined as the useful work produced during a full cycle in relation to the supplied heat, is

$$\eta_{IE} = \frac{W}{Q_H} = \frac{(P_H - P_L)v - \left(h\left(T_{P,out}, P_H\right) - h(T_L, P_L)\right)}{h(T_H, P_H) - h\left(T_{P,out}, P_H\right)} \tag{7}$$

The supplied heat can be reduced by recovering a part of the heat of the working fluid exhausted from the power cylinder in a recuperative heat exchanger, Figure 4. In this case working fluid after the feed pump is heated in the regenerator from $T_{P,out}$ to some higher temperature $T_{R,out}$ at the outlet of the regenerator. Thus the duty of the heater becomes

$$Q_H = m\left(h(T_H, P_H) - h\left(T_{R,out}, P_H\right)\right) \tag{8}$$

$T_{R,out}$ value is determined by the regenerator process. Its scheme is shown in Figure 4.

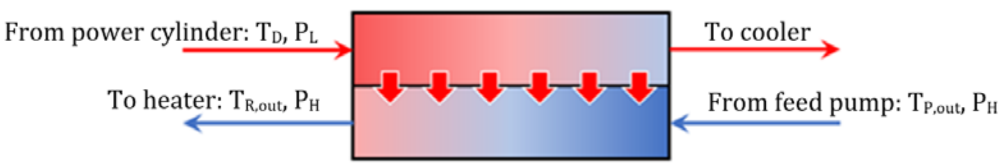

Figure 4. Scheme of the regenerator.

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In the regenerator cold stream from the feed pump at pressure P_H is heated by the hot stream discharged from the power cylinder. The temperature and pressure of the fluid discharging from the power cylinder (fluid delivering heat) depend on the way how the discharge is arranged.
We assume that the pressure of the fluid exhausted from the power cylinder decreases from the high cycle pressure P_H to the low cycle pressure P_L in a throttle located before the regenerator. The exhaust valve of the power cylinder may play the role of the throttle. This pressure reduction process is considered as a Joule-Thompson expansion.

Neglecting the changes in kinetic and potential energy, the temperatures in the power cylinder and at the inlet of the regenerator, T_E and T_D, are related by an equation ensuring no enthalpy change in this process:

$$h(P_E, T_E) = h(P_L, T_D) \tag{9}$$

in which P_E is the pressure of the working fluid in the power cylinder (before the exhaust valve) which change during the discharge process consisting of the pressure reduction stage and displacement of the working fluid from the power cylinder at the low cycle pressure.
Change of P_E and T_E during the discharge process can be obtained from the energy balance assuming the adiabatic process in the power cylinder:

$$\rho(T, P)dh = dP \tag{10}$$

Although the amount of working fluid in the cylinder reduces the energy, the balance equation, Eq. (10) is the same as that for adiabatic expansion of a fluid of constant mass.
From Eq 9, 10 the variable inlet temperature of the fluid delivering heat T_D can be calculated. Then, applying a method presented in [17] enthalpy of the working fluid entering the heater can be obtained.

To simplify the calculations of the regenerator process we assumed that the working fluid coming from the power cylinder to regenerator is well mixed. In other words, instead of variable inlet temperature T_D its average value found from the enthalpy balance is used. This assumption is justified if amount of the working fluid in the regenerator is much larger than m. Otherwise the assumption results in a conservative value of heat exchanged in the regenerator.
The specific enthalpy of the fluid after discharging and mixing is $h(T_H, P_H) - (P_H - P_L)v$. Therefore T_D can be obtained from equation

$$h(T_D, P_L) = h(T_H, P_H) - (P_H - P_L)v \tag{11}$$

If temperatures of the streams entering the regenerator are known the maximum thermodynamically allowed heat transfer between two fluids, or the maximum amount of regenerated heat Q_R is [17]

$$Q_R = h(T_D, P_L) - h\left(T_{P,out}, P_H\right) - Dh \tag{12}$$

where Dh is the maximum enthalpy difference within the temperature interval $T_{P,out} \leq T \leq T_D$:

$$Dh = max\left(h(T, P_L) - h(T, P_H)\right), \quad T_{P,out} \leq T \leq T_D \tag{13}$$

Accordingly, the maximum enthalpy of the working fluid entering the heater is:

$$h\left(T_{R,out}, P_H\right) = h\left(T_{P,out}, P_H\right) + \Delta h_{max} = h(T_D, P_L) - Dh \tag{14}$$

The efficiency of a regenerative IE-cycle can now be written as:

$$\eta_{IE+R} = \frac{W}{Q_{H+R}} = \frac{(P_H - P_L)v - \left(h\left(T_{P,out}, P_H\right) - h(T_L, P_L)\right)}{h(T_H, P_H) - h(T_D, P_L) + Dh} \tag{15}$$

To evaluate how an IE machine performs compared to other machines, the calculated efficiency is compared with the Carnot efficiency η_C showing the theoretical maximum, the Novikov η_N [18] efficiency which is a realistic measure for practically achievable efficiencies. This is shown in literature [19] for a variety of power plants i.e. nuclear, geothermal, steam, etc. The relation is also known as the Curzon-Ahlborn efficiency and is a result of the assumption that all engine parts are ideal, yet the heat transfer from the reservoirs is irreversible. The Novikov efficiency relates only to the theoretical maximum power point of the (heat)engine or cycle, but is widely used for comparing power generating processes. Also a newly established empirical correlation based on a survey of 34 commercial ORC installations η_{ORC} [20] is used for comparison. This relation is derived from operational ORC systems and the efficiencies that were reported. Definitions are as follows:

Carnot

$$\eta_C = 1 - \frac{T_L}{T_H} \qquad (16)$$

Novikov

$$\eta_N = 1 - \sqrt{\frac{T_L}{T_H}} \qquad (17)$$

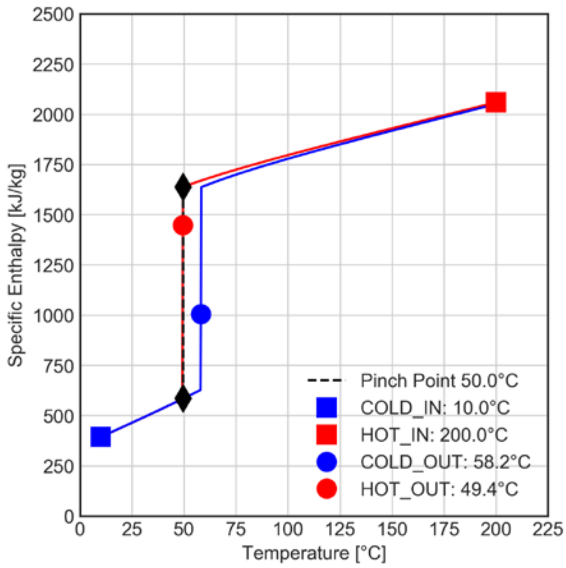
Gangar et. al. [20]

$$\eta_{ORC} = -0.93\eta_N^2 + 0.87\eta_N \qquad (18)$$

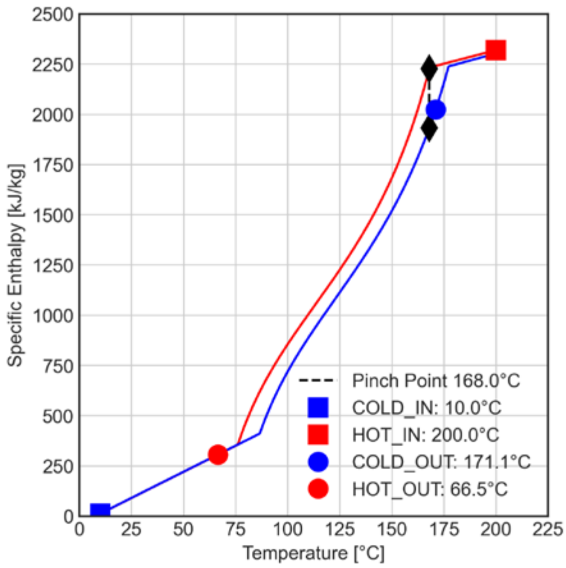
3. Results

All presented results are from running the calculation model against the physical properties of the working fluids referenced from REFPROP version 10 [12]. For single component working fluids we show that with low temperature and pressure difference the efficiency of the IE-cycle will approach the Carnot efficiency. With mixtures the calculation becomes more involved, while multiple mixtures are supported by REFPROP we chose to present the mixtures that have a firm backdrop in practice and literature: ammonia/water (NH_3 / H_2O) [21],[22] mixtures and carbon dioxide / acetone (CO_2/C_3H_6O) [23],[24]. These mixtures show a tremendous increase in efficiency when regeneration is applied to the IE-cycle.

By following the calculation method as described in eq. 1-15 and in [17], the cumulative enthalpy curves are determined for pure ammonia as a working fluid compared with ammonia water 65/35 %wt. at equal conditions. By employing a mixture, the theoretical maximum heat transfer with regeneration between the heat delivering and heat receiving fluid is increased drastically. This is graphically shown in Figure 5.



(a)



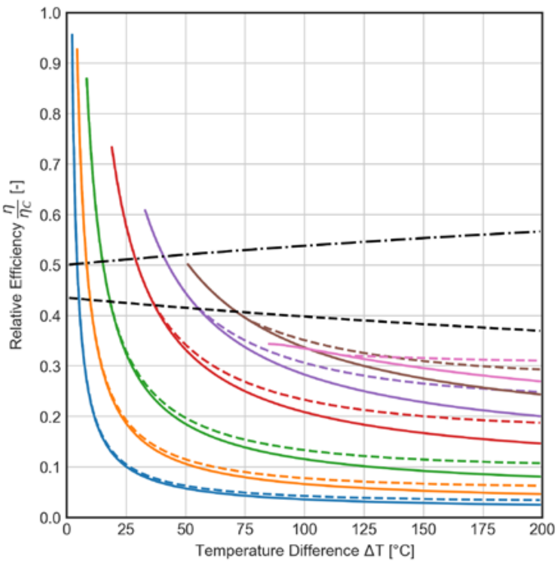
(b)

Figure 5. Calculation of the maximum possible heat transfer with regeneration for pure ammonia (a) and an ammonia/water mixture 65/35 wt.% (b) at P_L = 20Bar ΔP=5bar.

Efficiency of single component working fluids

Efficiencies of several single component working fluids in Worthington cycle with and without regeneration were investigated before [9]. However, in order to understand some critical features of the Worthington cycle we investigated several tens of such fluids calculating a relative (to Carnot) efficiencies at different temperature and pressure differences. As it turned out, all these working fluids demonstrated a similar behaviour shown in Figure 6 for ammonia, water, CO_2 and acetone as typical examples. It can be seen that the highest relative efficiency is reached at the point of phase transition at high cycle pressure P_H. At low temperature and pressure differences (below 5-10°C and 0.5-2 bar) these efficiencies approach to the Carnot efficiency. Any further increase of temperature and pressure difference leads to a decrease of the efficiency. This decline is notably pronounced for the highest efficiencies: the higher the efficiency the sharper the drop. High efficiencies (above 0,7 of the Carnot efficiency) are possible within the temperature range about 20°C. The efficiencies which are of practical, industrial interest (mostly above 0,5 of the Carnot efficiency) can be reached at the temperature difference below 50°C.

It can be also seen that regeneration can slightly extend the temperature and pressure differences. However, the benefit of a regenerative cycle is small and increases towards higher temperature differences. When a pure working fluid becomes supercritical a gradual increase in efficiency is seen, so regeneration becomes more important under supercritical conditions.



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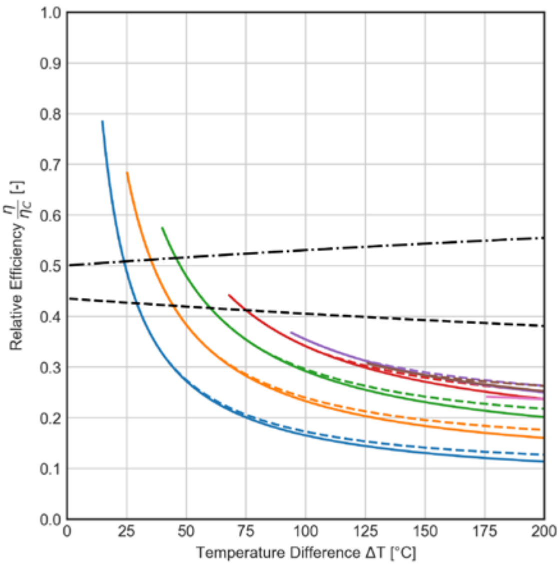

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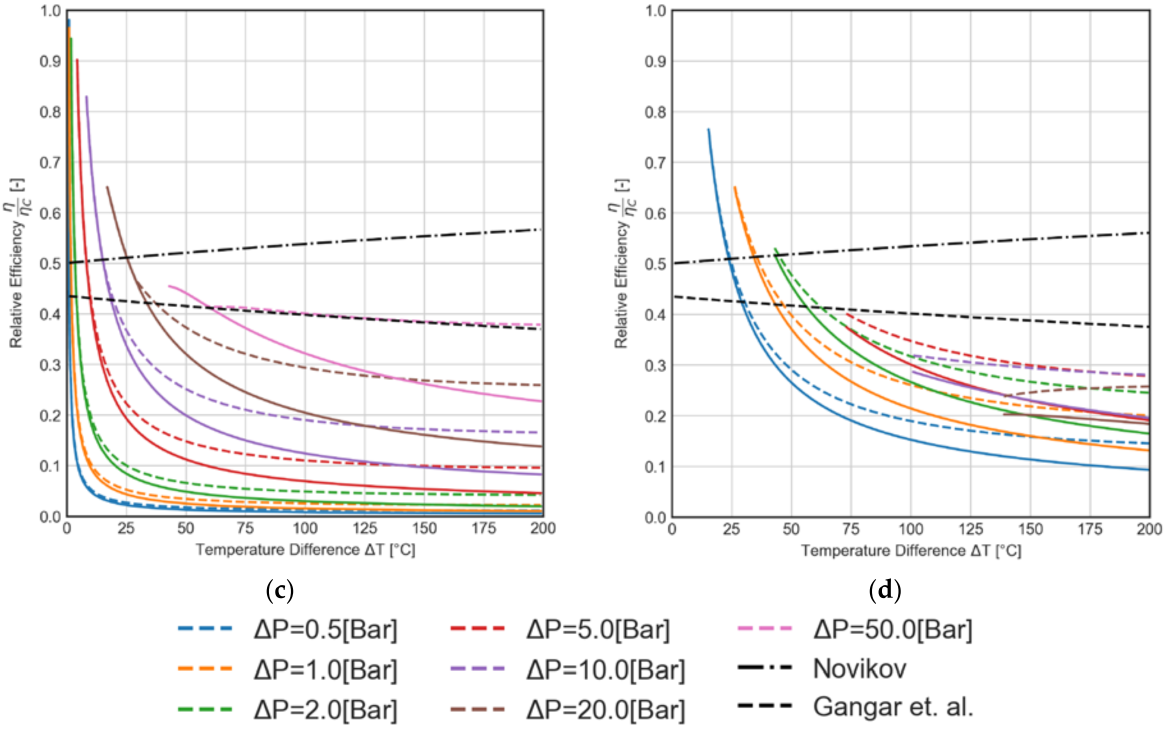

Figure 6. Specific efficiency of pure working fluids: ammonia (TL = 10°C, PL=6.15 Bar) (a), water (TL = 90°C, PL=0.70 Bar) (b), CO2 (TL = 10°C, PL= 11.07 Bar) (c), acetone (TL = 46°C, PL=0.71 Bar) (d). Solid line = without heat regeneration, Dashed line = with heat regeneration.

Efficiency of binary mixture working fluids

It was suggested in [9] that application of mixed working fluids could lead to some efficiency improvements due to better regeneration. In [25] some hydrocarbon mixtures were investigated for Bush-type engines and a slight rise in efficiency was found.

In contrast to pure working fluids, mixtures of working fluids can also provide significant benefits, because the temperature of the reservoirs i.e. heat source and sink temperatures often are not constant [26]. In this case a mixture with a suitable temperature glide can be used to match evaporator and condenser profiles more precisely.

We investigated numerically several tens of binary mixtures with different composition in the temperature range up to 200°C. Among them are binary mixtures of different hydrocarbons, oxygenates (alcohols, ethers etc.), fluorinated refrigerants, hydrofluoroethers (Novec™), inorganic substances etc. The main goal of this investigation was to evaluate whether mixed working fluids are able to considerably improve efficiency of Worthington-type IE-engines due to improved heat regeneration. As it turned out, many mixtures demonstrated a serious rise in efficiency as compared to single component working fluids.

Compared to pure working fluids: at high temperature differences binary mixtures in combination with regeneration can increase efficiency several times. In Figure 7 results are presented for a range of pressures of a 65/35 wt.% mixture of ammonia/water and a 60/40 wt.% CO_2/acetone system. For lower pressure differences the relative efficiencies are the highest >50%. Benefits of regeneration are even more pronounced with the CO_2/acetone mixture, showing efficiencies above the Novikov efficiency for large heat source temperature difference: up to 100°C ΔT for ammonia/water and up to 200°C ΔT for CO_2/acetone.

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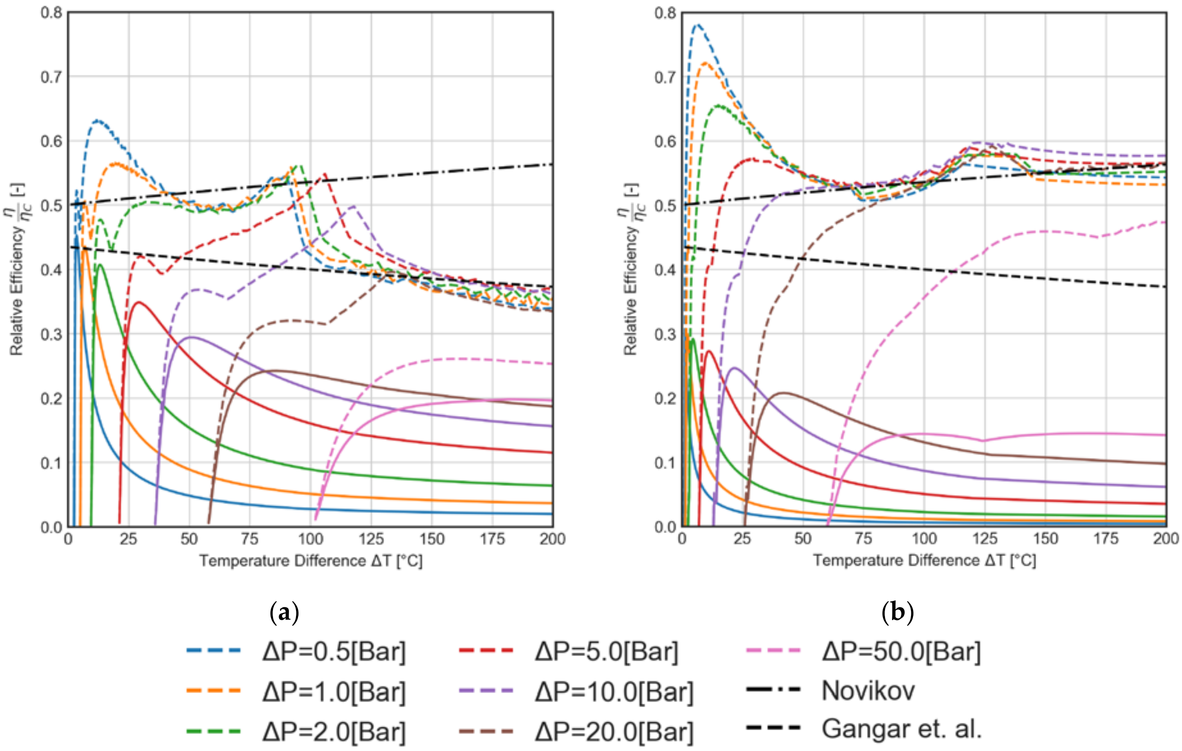

Figure 7. Specific efficiency calculation for 65/35 %wt. ammonia/water (T_L=30°C, P_L= 6.0 Bar) (a) and 60/40 %wt. CO_2/acetone system (TL=30°C, P_L= 35.62 Bar) (b).

4. Discussion

It can be seen that the simple non-regenerative cycle can provide a high thermal efficiency (close to the maximum possible Carnot efficiency), using a low temperature and pressure difference. However, the Worthington-type machines can provide such high efficiencies only within a narrow temperature-pressure range. At the higher temperature and pressure differences, the relative efficiency inevitably drops down. Regeneration is able to extend this high-efficiency temperature difference to some extent. Nevertheless, such a simple cycle can be of interest for a number of applications. For instance, in utilisation of low-temperature geothermal energy which can be economical if heat, rejected by IE-engine is used for district heating or in industry (drying etc.). Other examples are heat utilization of low temperature proton exchange membrane fuel cells (LTPEMFC), generally operating between 55°C and 80°C, waste heat recovery of marine diesel engines circulating water etc. The impressive efficiency at low temperature/pressure difference permits to use relatively inexpensive off-the-shelf Roots machines in an IE-cycle.

Another way to extend the temperature difference covered by the simple non-regenerative cycles is a cascade. Such a two-stage cascade is shown in Figure 8 as an example. The cascade is a combination of two IE-installations shown previously in Figure 1; each IE engine utilizes a certain temperature difference. The condenser / cooler (CH) of the first stage in this case plays the role of heater/evaporator for the second stage of the cascade.

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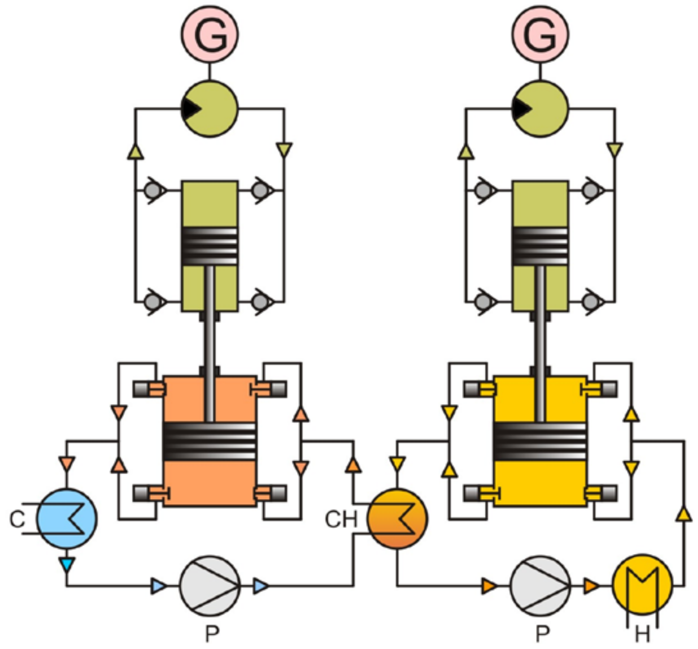

Figure 8. Heat-to-electricity converter based on Worthington-type engine working in cascade configuration.

Therefore, every engine can convert heat with a small reservoir temperature difference with a relatively high efficiency. Every engine in the cascade, shown in Figure 8, serves its own pump/motor. However, the cascade set-ups can be designed differently, when all engines drive only one pump and/or hydraulic motor etc. Generally, the cascade can utilise more broad temperature differences as compared single stage IE-engines. Cascade setups are thus suitable to use heat from different temperature waste streams. In this case every stage can use not only the heat from the upstream stage but heat from any other suitable source. This is of interest for the recovery of heat from for example diesel engines by coupling exhaust- and cooling water waste heat i.e., multiple heat streams with two different temperatures. Refineries, chemical and food industry are other typical examples of multiple temperature level heat sources for IE cascade application.

Mixed working fluids make it possible to operate in much broader temperature and pressure difference ranges typical of medium and high-pressure reciprocating piston IE engines. IE-engines with mixed working fluid can be used for power generation and as heat-driven pumps and compressors. The examples are pumps for irrigation, for transfer liquefied gases and refrigerants, for district heating and water supply, air compressors, refrigeration and air conditioning compressors, compressors of heat pumps etc. Air compressors are of interest also in view of the last efforts on development of compressed air energy storage systems. The second particular case today is a high-pressure compression of hydrogen for storage and hydrogen refuelling stations etc.

5. Conclusions

A high quantity of recoverable energy is still lost in most industrial processes. Possible ways of (ultra) low-grade heat utilization by different types of isobaric expansion (IE) machines are presented and discussed. It is shown that efficiency of non-regenerative machines with single-component working fluids can be high, approaching the Carnot efficiency at a low heat source/heat sink temperature difference. Regeneration permits to increase this efficiency only to some extent. For an efficient utilization of higher temperature differences there are two ways. The first one is a so-called cascade i.e., a combination two or more IE machines; each of them utilizes only some part of the available temperature difference. The second is an application of binary mixtures as working fluid in a combination with regeneration. At high temperature differences such a combination can increase efficiency several times when compared to single component working fluids.

The IE engine can become a suitable means for recovery of heat energy by its simple design, and economical application. Ongoing research and experiments will further develop IE technology and improve system configurations for the plethora of applications.

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Data Availability Statement: In this section, please provide details regarding where data supporting reported results can be found, including links to publicly archived datasets analyzed or generated during the study. Please refer to suggested Data Availability Statements in section "MDPI Research Data Policies" at https://www.mdpi.com/ethics. If the study did not report any data, you might add "Not applicable" here.

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