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Model-based Analysis of Novel Heat Engines for Low-Temperature Heat Conversion

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To overcome the lack of economic technologies for conversion of low-grade heat into power, Dutch company Encontech B.V. has recently suggested a novel-type external combustion engine. This Encontech Engine utilizing dense working fluids promises effective energy conversion and large power density at low specific cost. In the study presented in this paper, a simplified thermodynamic model has been developed and applied to identify optimal working conditions and suitable working fluids for the Encontech Engine. Although the utilization of dense working fluids promises several advantages over gas-phase heat engines (e.g., Stirling engine), efficient regeneration is a difficult task. For this reason, approaches to combine the benefits of dense working fluids with enhanced regeneration have been investigated.

1. Introduction

A recent study by Forman et. al. (2016) has shown that around 73 % of the globally converted energy is lost during the transition into the final energy service. Roughly two-thirds of these losses is waste heat. Pinch analysis methods have found wide application in industries and contributed to remarkable improvements in process efficiency by smart linkage between heat sources and sinks within individual processes, plants or entire production sites (Semkov et. al., 2014). However, large portions of energy still remain unutilized. Approximately 42 % of the global industrial waste heat arises at temperatures below 100 °C (Forman et. al., 2016), for which only minor demand exists in many industrial production processes. If utilized at all, low-temperature heat is used for heating of facilities, heat upgrading (heat pumps) or production of cold (absorption refrigerators). Conversion of low-grade waste heat into power is usually not considered due to lacking economic conversion technologies. Organic Rankine Cycles (ORC) constitute the most proven and industrially applied technology to convert low-temperature heat into power. They are considered to be the most efficient and economic technology in temperature ranges of 200-400 °C (De Pascale and Bianchi, 2011), but they have high specific costs of 2,000-4,000 €/kW at lower temperatures (Heberle and Brüggemann, 2015). The Kalina cycle, a variation of the classical ORC utilizing water-ammonia mixtures, is a promising alternative for very low temperature sources. However, studies indicate that the promised benefits of Kalina cycles appear over-estimated, while this process is much more complex and maintenance-demanding than classical ORC processes (Goswami et. al., 2010) Alternative technologies, such as piezo-electric (Hendricks and Choate, 2006) or thermo-electric conversion concepts (Snyder, 2009), are still in the early stages of development and highly cost-intensive.
Encontech BV, a spin-off company of Twente University, has recently suggested a novel-type externally heated engine (Glushenkov and Kronberg, 2012). This device, combining the advantages of the technically interesting but mainly forgotten concepts of the ‘thermocompressor’ (Bush, 1939) and the ‘tidal regenerator engine’ (Hagen et. al., 1976) with additional features (e.g., a self-driven displacer, a hydraulic load and application of several dense working fluids), promises high power density and energy efficiency at low specific cost. The Encontech Engine is hence particularly attractive for conversion of low-temperature heat into power (Kronberg et. al., 2016).

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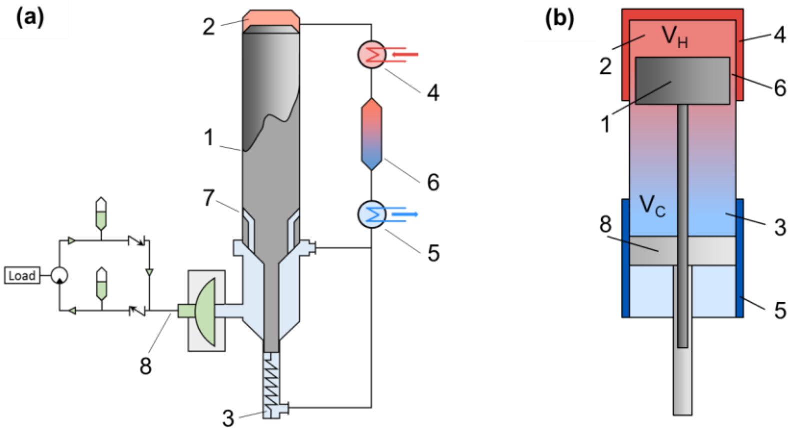

Figure 1: Schematic representation of the Encontech Engine (a) and simplified physical model of externally heated engines (b).

In the present work, a simplified thermodynamic model for externally heated engines has been developed and applied to demonstrate the potential strengths of the Encontech Engine and to identify promising working conditions and suitable working fluids.

2. The Encontech Engine

The Encontech Engine represents a technically simple and economical alternative to existing heat engines. This regenerative external combustion engine can use any arbitrary heat source of any grade and operates with dense working fluids. Figure 1a gives a schematic representation of the engine. A self-driven piston (1) periodically displaces working fluid between a hot (2) and a cold (3) domain. Heat is supplied to the hot domain by a heater (4), the cold part is cooled (5). A regenerator (6) is applied to recover heat during the cyclic fluid flow. When fluid is displaced from the cold to the hot domain, the engine pressure rises accordingly. Once a certain pressure level is established, a valve (7) opens and working fluid is communicated to a connected hydraulic load circle (8) until the pressure in the engine equals the pressure in the load circle. As a result of the subsequent displacement of hot fluid to the cold domain, the engine pressure decreases until the valve opens again at the lower limiting pressure, and a second pressure equalization step takes place (Glushenkov and Kronberg, 2012). Useful work is generated as the fluid in the hydraulic load circle passes through a hydraulic motor. The engine is flexible with respect to both heat source and load and can therefore be used in various applications, including power generation, refrigeration, compression and pumping. As the engine has been demonstrated to work even at very low temperatures (around 70 °C) of the heat source, it appears to be particularly interesting for conversion of low-grade waste heat (Kronberg et. al., 2016).

3. Modelling approach

Externally heated regenerative engines – including both the Stirling Engine and the Encontech Engine – can be described by the physical model depicted in Figure 1b. The engine model consists of a cylinder filled with the working fluid. The upper end of the cylinder is heated (4); the lower part is cooled (5). In addition, the model employs two pistons whose motions result in changes of pressure p and volumes V inside the engine. The displacer piston (1) divides the cylinder volume into a hot domain (2, volume V_H) and a cold domain (3, volume V_C) and serves to move working fluid from one domain to the other. The volume of the regenerative zone (6) is denoted as V_R and remains constant throughout the cycle. When working fluid is displaced from the cold to the hot domain, it is heated externally and/or partially by regeneration, and vice versa. Useful work is produced, when the power piston (8) moves, i.e., the total engine volume V_{tot} changes. The interdependence of both piston motions defines the thermodynamic cycle. In our approach, we assume frictionless piston movement, uniform pressure in the entire engine and temperatures in the hot (T_H) and cold part (T_C) kept constant by external heating or cooling. The energy conversion efficiency is defined as the ratio of the useful work generated per cycle to the heat supplied to the hot part:

$$\eta=\frac{|W|}{Q_H} \tag{1}$$

The useful work per cycle W is defined as:

$$W = \oint p\, dV_{tot} = \oint p\, dV_H + \oint p\, dV_C \quad (2)$$

The interdependence of pressure and volumes inside the engine can be obtained from the sum of masses contained in each individual part of the engine, i.e. the total mass of working fluid:

$$m_{tot} = m_H + m_C + m_R = \rho(p,T_H)\, V_H + \rho(p,T_C)\, V_C + \rho(p,T_R)\, V_R \quad (3)$$

The heat supply to the hot part Q_H is determined from an energy balance of the hot domain, neglecting changes of kinetic and potential energies. We consider the changes of internal energy U_H, of pressure-volume work and of the enthalpy flows to (superscript: in) or out of (superscript: out) the hot part:

$$Q_H = \oint dU_H + \oint p\, dV_H + \oint h^{out}\, dm_H^{out} - \oint h^{in}\, dm_H^{in} = \oint dU_H + \oint p\, dV_H + \Delta Q_H \quad (4)$$

The last term in Eq(4), $\Delta Q_H = \oint h^{out}\, dm_H^{out} - \oint h^{in}\, dm_H^{in}$, describes the performance of the regenerator, which represents a key feature of externally heated engines. The regenerator stores the heat of the fluid leaving the hot domain while it is cooled down to the cold temperature and returns the stored heat to the fluid on its way from the cold to the hot space. The regenerator outlet temperature $T_{H,R}$ equals the hot space temperature T_H only in the theoretical case of perfect regeneration. In reality, the difference has to be compensated by additional heating ΔQ_H. The approach, together with an equation of state, defining density as a function of pressure and temperature ($\rho(p,T)$), and an energy equation ($h(p,T)$ or $u(p,T)$), can be applied to arbitrary thermodynamic cycles and working fluids. The Encontech Engine can adequately be described by the ideal rectangular cycle, consisting of two isochoric and two isobaric stages, as illustrated in Figure 2.

4. Demonstration of potential strengths of the Encontech Engine

Utilizing dense working fluids, i.e., fluids that are in liquid state at cold temperature T_C and in vapour or supercritical state at hot temperature T_H, is an option to circumvent the most severe problems of externally heated engines using gaseous working fluids. Several studies and many years of development have shown that gas-phase engines suffer from inevitable dead volumes (e.g., for heating and cooling purposes) which decrease the engines efficiency and power density (Wongwises and Kongtragool, 2006). The gaseous working fluid impedes sealing and lubrication of moving parts (i.e., pistons). Moreover, gas-phase engines generally offer poor power density (work per cycle and unit volume) which requires high frequencies (cycles per unit time) to generate a desired power output. This, however, sets further barriers, including balancing of pistons, mechanical losses and wear (Hargreaves, 1991).
The governing equations were implemented in MATLAB and solved numerically to determine specific work per cycle and energy conversion efficiency for given operating parameters. In order to demonstrate the potential strengths of the Encontech Engine, two indicative operation options were compared in a simulation study: an example with dense working fluid R134a and a gas-phase case utilizing nitrogen for reference. Temperatures of T_C = 20 °C and T_H = 80 °C, typical values for low-grade waste heat, were chosen. The lower pressure for R134a was p_l = 5.8 bar, representing the minimum pressure, as the fluid would evaporate spontaneously at lower pressures. For nitrogen, p_l was arbitrarily set to 10 bar. Upper pressures were varied in reasonable boundaries for both cases.

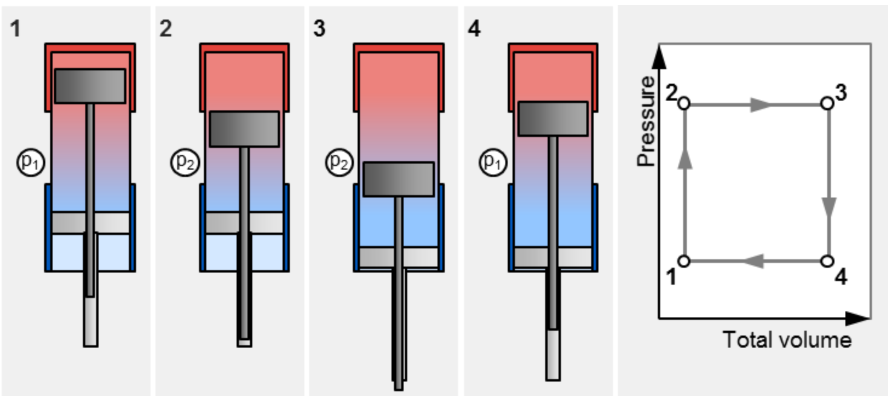

Figure 2: State points (1 to 4), corresponding piston positions and pressure-volume diagram (right) of the ideal rectangular cycle corresponding to our simplified physical model shown in Figure 1b.

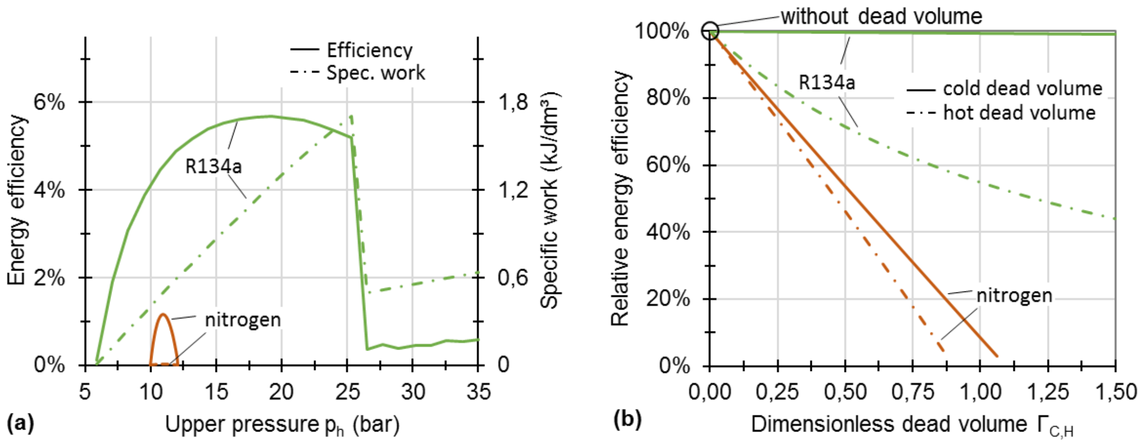

Figure 3: Simulation results: (a) energy efficiency and specific work per cycle in dependence of the chosen upper pressure; (b) influence of dead volumes on the energy efficiency at fixed upper pressure.

4.1 Specific work and energy efficiency

In a first step, in addition to the previously listed assumptions, neither regeneration nor dead volumes were considered. Figure 3a illustrates the simulated results for specific work per cycle and energy efficiency for both fluids as a function of the chosen upper pressure p_h. While the efficiency for the nitrogen example has a parabolic shape, a sudden drop is observed after exceeding a certain pressure (here: approx. 25 bar) for R134a. At this pressure (saturation pressure at T_H), the fluid in the hot part is liquefied. Operation beyond this pressure is possible, yet very inefficient. Moreover, Figure 3a clearly shows that both energy efficiency and specific work are remarkably higher for the dense working fluid. The specific work for the nitrogen example is about two orders of magnitude lower compared to the R134a case and even hardly visible in Figure 3a. Cycles with dense working fluids can hence be operated at significantly lower amplitudes, potentially reducing the difficulties of high-frequency operation mentioned above.

4.2 Influence of dead volumes

Dead volumes are unavoidable in externally heated engines, as they are required for heat transfer and fluid ducting or simply appear as volumes not swept by one of the cylinders. In this study, dead volumes are related to the swept volume V_0 of the engine ($\Gamma_{C,H,R}=\frac{V_{C,H,R}}{V_0}$) to allow a general interpretation of the results.

Figure 3b indicates how existing dead volumes decrease the energy efficiency of the engine. Plotted is the efficiency related to the efficiency determined without dead volumes, the upper pressures were set to the value gaining the highest efficiency for each example (see Figure 3a). The negative influence of dead volumes is significantly lower for the R134a operation, whereas the efficiency of gas-phase operation drops to zero when dead volumes in the hot or cold space are as high as the swept volume of the displacer. Dead volumes in the cold domain in an engine with dense working fluid are hardly an important issue, as the fluid is nearly incompressible there. Consequently, utilizing dense working fluids potentially eliminates one of the most significant drawbacks of conventional externally heated engines.

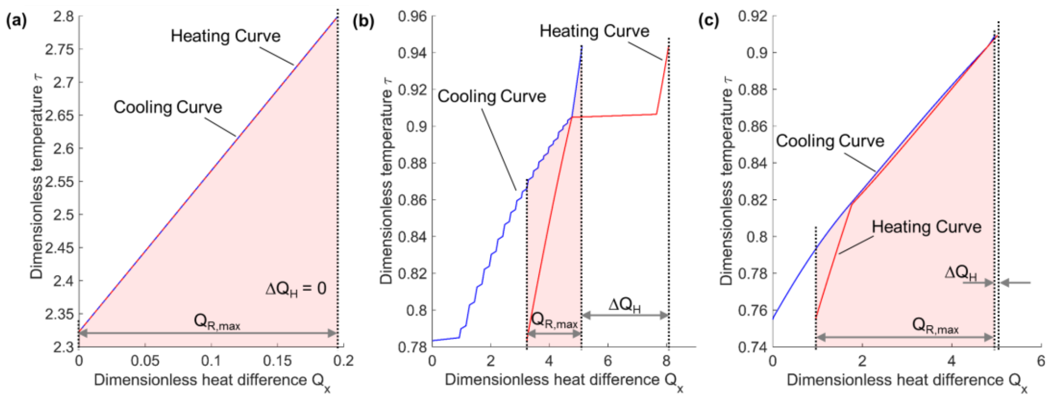

Figure 4: Regeneration potentials for nitrogen (a), r134a (b) and a blend of propane, butane, pentane and ethane with 25 wt.-% of each component (c), $Q_x = Q/(p_{crit}\, v_{crit})$, $\tau = T/T_{crit}$.

4.3 Regeneration

When heat is regenerated, energy efficiency can increase remarkably. To estimate the maximum regeneration efficiency, we applied an approach similar to the Composite Curve methodology in Pinch Analysis (Kemp, 2007). The Composite Curves indicate the accumulated heat demand (during heating from T_C to T_H) and availability (during cooling from T_H to T_C) per temperature level. Obviously, a heat demand for heating can only be met by regeneration, if enough heat is available at a higher temperature. In terms of Composite Curves, this restriction is satisfied when the heating curve never exceeds the cooling curve. In the limiting case, when both curves contact (ΔT_{min} = 0), the amount of heat exchanged in the regenerator reaches its maximum. In this case, the regeneration efficiency, defined in the following equation, is also at a maximum:

$$\eta_{R,H(max)} = \frac{Q_{R(max)}}{Q_{R(max)} + \Delta Q_H} \qquad (5)$$

In reality, heat transfer resistances necessitate a positive minimum temperature difference ($\Delta T_{min} > 0$). Regeneration efficiency consequently decreases with increasing ΔT_{min}. Figure 4 illustrates how differently the composite curves may be shaped. With an ideal gas, to which nitrogen is close, perfect regeneration is theoretically possible, so that no additional heating is required. For isobaric phase transition of R134a, in contrast, the necessary heat can be provided by regeneration only partially, because heating demands for more heat at saturation temperature at high pressure than is available from the cooling phase.

Hence, even if the (poor) regeneration potential is exploited entirely, energy efficiency can only be increased marginally. For nitrogen, instead, energy efficiency could theoretically reach Carnot's efficiency, if the assumptions (minimum temperature difference of 0 K, no dead volume for regeneration) held. Apparently, rather poor regeneration is a feature of dense working fluids. However, even without regeneration, energy efficiency is still better compared to the gas-phase operation, unless the latter achieves regeneration efficiencies of 80% or more.

5. Enhancement of regeneration

Enhancement of regeneration potentially leads to remarkable rise in energy conversion efficiency. However, as demonstrated in the previous section, regeneration potential is rather poor for pure dense fluids with isobaric phase transition. The utilization of blends is an opportunity to bypass the main difficulty of large heat demands at a single (boiling) temperature. If selected properly, blends evaporate in a temperature range between the given source and sink temperatures, allowing the Composite Curves to approach each other more closely (see Figure 4c) and hence yielding higher regeneration potential. A set of different pure fluids and blends, summarized in Table 1, was investigated for given temperatures of T_C = 20 °C and T_H = 80 °C. It was assumed that 90 % of the maximum regeneration efficiency (as determined with the Composite Curve approach) can be achieved in each case. The results for energy efficiency and specific work are visualized in Figure 5: The effect of regeneration on the energy efficiency is significant for examples with blends. However, when energy efficiencies with regeneration are compared, utilization of blends instead of pure fluids leads to minor improvements only. Moreover, this positive effect corresponds to reduced specific power, so that expectations regarding the utilization of mixtures are not fully satisfied for the considered examples.

Thus, identification of the most promising blends and their compositions still represents a subject of ongoing research. Another option to enhance regeneration is to employ fluids which are liquid at cold space temperature T_C and in supercritical state at hot space temperature T_H (i.e., $T_C < T_{crit} < T_H$). Ethane with T_{crit} = 32.2 °C is a suitable choice for the considered example, generating both high energy efficiency and specific work per cycle. A potential drawback for real engine operation is the high operational pressure required to keep ethane in liquid state at cold temperature (see Table 1).

Table 1: Investigated working fluids and blends with chosen operational pressures

Working Fluid	Low Pressure (bar)	High Pressure (bar)	Working Fluid (Blends)	Composition (wt.-%)	Low Pressure (bar)	High Pressure (bar)
R134a	5.80	18.68	Propane/Butane	50-50	5.55	16.83
Butane	2.10	9.56	Propane/Pentane	50-50	5.65	11.89
Propane	8.50	23.88				
Nitrogen	10.00	11.00	Propane/Butane/ Pentane/Ethane	25-25-25-25	14.63	22.73
Ethane	38.00	83.96				

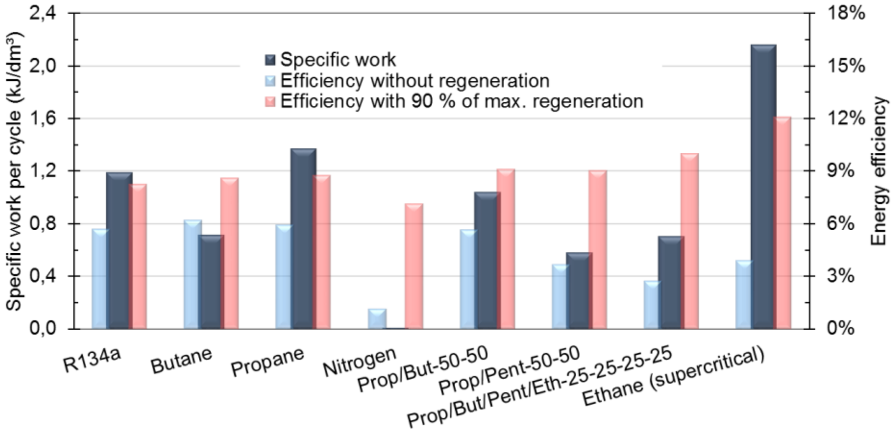

Figure 5: Simulation results on regeneration potential (energy efficiency with and without regeneration) and specific work for selected working fluids and blends.

6. Conclusions

A simplified thermodynamic model was developed and applied to the Encontech Engine, a novel device for conversion of heat (particularly low-grade waste heat) into power. In addition to the analysis of the potential advantages of the Encontech Engine, this model was employed to identify suitable working fluids for selected conditions. It was demonstrated that the utilization of dense working fluids, a key feature of the Encontech Engine, offers significant specific power at relatively high energy conversion efficiency. Moreover, one of the most severe drawbacks of gas-phase heat engines, the strongly negative influence of dead volumes, is alleviated. To address the rather poor regeneration potential of dense working fluids, utilization of blends and supercritical fluids was studied. For the investigated mixtures, a slight improvement of energy efficiency was observed, whereas specific power was reduced. Supercritical fluids were found to offer the most efficient energy conversion together with large specific work, yet they require high operational pressure.

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