Waste Heat Recovery for Reciprocating Compressors

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Abstract. In many important industries (oil and gas, process gases, chemical process engineering) multi-stage reciprocating compressors are used, especially for high pressure ratios. The gas is compressed in multiple consecutive stages and cooled after each stage in order to reduce the maximum process temperatures. The rejected heat is often dissipated to the environment and the usable part – the exergy – is lost.

The aim of this paper is to show how the waste heat potential of reciprocating compressors can be used. For this purpose, the waste heat available per stage is quantified for different compression scenarios. Based on this, the processes for the waste heat recovery suitable for the temperature range of the discharge gas as heat source are presented – in particular the structure, working principle and characteristics of the waste heat recovery system. It is shown that the waste heat can be used flexibly for different purposes (heating, power generation, cold supply). The potential of the possible methods of waste heat recovery can be estimated with the aid of the given efficiencies of the respective energy conversion processes.

1. Introduction
In addition to the general imperative to develop resource-saving technologies, legal requirements – such as the Ecodesign Directive of the European Union (Directive 2009/125/EC [1]) – increasingly demand the optimization of the energy efficiency of every product. To meet the high requirements on energy efficiency in the field of reciprocating compressors, which will continue to increase in the future, it is vital to make the best possible use of existing potential for improvement. As in many other technical areas, the topic of waste heat recovery (WHR) plays an important role, since in reciprocating compressors, in particular for multi-stage reciprocating compressor systems in the kilowatt or even megawatt range, the compressed gas is typically cooled against ambient after each compression stage. This represents a considerable amount of energy and remains unused in almost all applications.

The paper is intended to provide a straightforward method for the quantification of the available waste heat and potential uses. Therefore, it introduces and analyses different methods to utilize the recoverable waste heat. It is intended to promote an increased use of WHR systems in the field of reciprocating compressors.
2. Thermodynamics of compression and waste heat quantification

The compression of gases in reciprocating compressors represents an energy supply from the piston to the gas, which leads to an increase of its enthalpy. This energy supply is expressed by an increase in pressure and by an increase in temperature of the compressed gas in de facto all real compression scenarios. The temperature typically rises with increasing pressure ratio, which is why multi-stage compression is often used at high pressure ratios. In these cases, the gas is compressed step by step in several consecutive compression stages (see figure 1). After each compression stage, the gas is cooled by means of an aftercooler, which serves to reduce the thermal load on the compressor components and increases the energy efficiency, as it enables a more isothermal compression process [2]. The enormous potential results from two facts: First, the driving power of multi-stage reciprocating compressor systems is typically in the kW or even in the MW range. Second, in many compressor applications the heat from the aftercooler is simply released to the environment, even though it still contains a usable part of energy – the exergy.

Figure 1. P&ID flow chart of a double-stage piston compressor system.

Figure 2. $p,v$ diagram of a single-stage and double-stage compression with isobaric intercooling

2.1. Quantification of recoverable waste heat

First, an easy way is given to quantify the potential heat under simplifying assumptions for general accessible data of the compression task (different gases, pressure level and suction conditions). The prerequisites for this are:

1. Neglect of kinetic and potential energy.
2. Ideal gas behaviour, i.e. a constant heat capacity is assumed, and the specific enthalpy of the gas is only a function of temperature.
3. Complete cooling of the gas in the aftercooler after each stage, i.e. the intake temperature is the same for each compressor stage ($\theta_{1,in} = \theta_{2,in} = \theta_{in} = \text{const.}$).

For each compressor stage in figure 1, the energy supplied to the gas by the displacer results from the change in its enthalpy flow $\Delta H = H_{out} - H_{in}$. By division by the gas mass flow $m_g$ this can be converted into the specific enthalpy increase $\Delta h$:

$$\frac{\Delta H}{m_g} = \Delta h = h_{out} - h_{in}. \quad (1)$$

Considering assumption 2, the specific enthalpy can be expressed by the associated temperature $\theta$ and the constant specific heat capacity $c_p$:

$$h = h(\theta) = c_p \theta. \quad (2)$$
If the gas is cooled by each aftercooler to the same intake temperature \( \vartheta_{in} = \vartheta_{1,in} = \vartheta_{2,in} \) (assumption 3), it follows that the specific enthalpy difference of the gas between inlet and outlet of the compressor stage \( \Delta h \) corresponds to the specific heat quantity \( q_{ac} \) rejected in the aftercooler:

\[
\Delta h = c_p \vartheta_{out} - c_p \vartheta_{in} = c_p \Delta \vartheta \quad (3)
\]

\[
q_{ac} = c_p \Delta h_{ac} = |c_p \vartheta_{in} - c_p \vartheta_{out}| = c_p \Delta \vartheta \quad (4)
\]

\[
\Delta h = q_{ac}. \quad (5)
\]

In order to quantify \( q_{ac} \), the use of the isentropic efficiency of the compression \( \eta_s \), among other possibilities, is probably the simplest option. This parameter reflects the ratio of specific isentropic enthalpy increase \( \Delta h_s \) and specific real enthalpy increase \( \Delta h \) (with losses) [3]:

\[
\eta_s = \frac{\Delta h_s}{\Delta h} \quad \text{or} \quad \Delta h = \frac{\Delta h_s}{\eta_s}. \quad (6)
\]

Considering the fluid as an ideal gas, \( \Delta h_s \) can be calculated using the following approach [3]:

\[
\Delta h_s = \frac{\kappa}{\kappa - 1} R_s \vartheta_s \left( \frac{\pi}{\kappa - 1} \right). \quad (7)
\]

Here \( \kappa \) is the isentropic exponent, \( R_s \) the specific gas constant, \( \vartheta_s \) the suction temperature (in Kelvin) and \( \pi = p_d/p_s \) the pressure ratio of the compressor stage with \( p_d \) as high pressure and \( p_s \) as suction pressure. If the equations (5), (6) and (7) are combined, it is possible to quantify the specific heat \( q_{ac} \) dissipated in the aftercooler for different gases, pressure levels and suction conditions of a specific compressor stage:

\[
q_{ac} = \Delta h_1 = \frac{1}{\eta_s} \frac{\kappa}{\kappa - 1} R_s \vartheta_s \left( \frac{\pi}{\kappa - 1} \right). \quad (8)
\]

As an example, the results of the specific heat dissipation in the aftercooler for an air compressor for different pressure ratios and degrees of compression quality are shown in figure 3. In the case of a conventional compressed air compressor which takes in 1 000 \( \text{m}^3 \text{h}^{-1} \) of dry ambient air at 20 \( ^\circ \text{C} \) (approx. 1.2 \( \text{kg m}^{-3} \)), discharges it at 5 bar\(_c\) and works with a typical isentropic efficiency \( \eta_s = 0.7 \), approx. \( q_{ac} = 246 \text{ kJ kg}^{-1} \) of specific heat must be dissipated in the aftercooler. With a delivered mass flow \( \dot{m}_g = 1 \text{ 200 kg h}^{-1} (1 \text{ 000 m}^3 \text{ h}^{-1}) \), waste heat of \( Q_{ac} = 82 \text{ kW} \) is therefore available. This is a considerable waste heat potential even for a single-stage compression scenario.

These simplified relationships result from the assumptions specified above and can be used as a first approximation for all compressor scenarios. If the pressures and temperatures at the inlet and outlet of the aftercooler of a specific compressor system are available by measurements, the existing waste heat potential can of course be quantified more precisely by corresponding gas properties.
2.2. Temperature level of waste heat

The maximum temperature level of the waste heat source $\theta_{\text{WHR, max}}$, i.e. aftercooler gas inlet temperature $\theta_{\text{ac,in}}$, plays an almost even more important role for WHR than the amount of heat available from two different perspectives. On the one hand, according to the known Carnot factor for a thermal power process $\eta_c = (\theta_{\text{WHR, max}} - \theta_{\text{amb}})/\theta_{\text{WHR, max}}$, it specifies how much of the waste heat can ideally be recovered (i.e. with no entropy production) at an ambient temperature $\theta_{\text{amb}}$, whereby the usable part increases with increasing temperature of the waste heat. On the other hand, the WHR process and the resulting type of energy as well as the ambient conditions must match the temperature level of the heat source. Due to the temperature gradients that occur during heat transfer, attention must be paid to the pinch point, i.e. the minimum temperature difference between the hot and cold medium within the aftercooler.

For reciprocating compressors, the maximum discharge temperature is typically 200 °C and in some cases slightly above [4]. For many applications this temperature is defined by standards: for oil-lubricated, multi-stage air compressors above 10 bar, it is 160 °C according to [5]. API 618 [6], the world’s leading standard in the oil and gas industry, recommends a general maximum temperature of 150 °C and 135 °C for high hydrogen content. A general limitation to 135 °C within the oil and gas industry is given in [7]. Sometimes higher discharge temperatures are possible under special precautions. For most compression scenarios, minimum outlet temperatures of at least 80 °C occur. In summary, 80 °C to 200 °C can be assumed for most reciprocating compressor applications.

3. Suitable processes of WHR

From the given conditions in section 2 the question arises for what purposes the waste heat can be used. In principle, four different forms of use can be differentiated, whose RI flowcharts are shown in the figures 4 to 7.

3.1. Heating purpose

The simplest application is the direct use of the heat transferred for heating purposes (room heating, hot water). For this purpose, the waste heat is typically fed to the consumer by means of a heat transfer medium (shown in purple in figure 4). Advantageous compared to the other WHR systems are little losses with good thermal insulation and low complexity of the system. In addition to the requirement that there is a heat demand at all, a major challenge is that the location and temperature level must match the discharged gas as the heat source. If, for example, the discharge temperature of the compressor is 90 °C and hot water of 80 °C is required at 1 km distance, the WHR is practically meaningless.

Figure 3. Specific heat rejection $q_{ac}$ in the aftercooler for the compression of air ($\kappa = 1.4; R_g = 287.1$ J kg$^{-1}$) at a suction temperature of $\theta_s = 293.15$ K (20 °C) for different pressure ratios $\pi$ and isentropic compression efficiencies $\eta_s$. 
3.2. Thermoelectric generator

A thermoelectric generator (TEG) is based on the Seebeck effect, which describes the relationship between temperature and electric current. It states that a voltage occurs at the free ends of two electrical conductors in contact if the temperature at the contact point is different from that at the free ends. In this way, electrical power can be obtained from a heat source resulting from temperature differences. TEGs are therefore characterized by a comparatively low complexity and have no susceptible moving components. They also advantageously offer electric current as the most flexible form of energy. Nevertheless, there is a serious disadvantage, as the achievable efficiencies are still quite low. According to [8] approx. 20% of the Carnot efficiency can be realised, thus, efficiencies for the conversion of heat into electrical power of 3 to 7% can be achieved for heat source temperatures of 353 to 473 K (approx. 80 to 200 °C) at a heat sink temperature of 300 K (approx. 27 °C) for conventional (and hence affordable) material pairings. Newer material developments based on nanotechnology indicate an increase of 1 to 4% of the overall efficiency. Goldsmid [9] suggests using TEGs if the temperature difference between heat source and heat sink is below 100 K, especially since there are no suitable alternatives for using the waste heat in this temperature range. This corresponds to the gas temperatures at the outlet of many compressor systems with rather low-pressure differences.

3.3. Organic Rankine Cycle (ORC)

In an Organic Rankine cycle, a thermal power process, a working fluid (see figure 6 shown in green) passes through a closed circuit. In the liquid state (no or almost no subcooling) the high-pressure liquid absorbs the waste heat from the aftercooler in a heat exchanger (E). The transferred thermal energy is then partly converted into mechanical energy in the expansion machine (X) and typically further converted into electrical energy by means of an electric generator (G). If the temperature at the expander outlet is still above the actual condensation temperature, part of the energy remaining in the working fluid can serve for preheating the high-pressure liquid by the aid of an internal heat exchanger (IHX). The circuit is closed by a condenser (C) and the pump (P).
Figure 8. \(\vartheta_s\) diagrams of ORCs for the WHR at the aftercooler with a gas inlet temperature of 200 °C for standard \(n\)-butane cycle without superheating and without IHX (left), for a transcritical \(n\)-butane cycle with IHX (middle), and a classic steam cycle (water) with superheating (right).

For the temperature gradient of the waste heat from 200 °C to ambient temperature available in the aftercooler, organic working fluids, which have aroused great interest in recent years, are particularly suitable. The maximum temperature of the waste heat of ORC processes is given in review publications at up to 300 °C to 400 °C [10]. This is mainly because many organic fluids have a retrograde boiling curve and the expander outlet is in the area of superheated gas and not in the wet steam area. Liquid fractions, which can have an adverse effect on the expansion machine, therefore do not occur. Furthermore, due to the location of the critical point, a transcritical process is also possible in which the two temperature curves in the heat exchanger can be matched to each other (see middle diagram in figure 8). This reduces exergy losses during heat transfer, and, with a suitable ORC fluid selection and process control, it ensures a favourable utilisation of the gas temperature gradient. ORCs prove to be particularly advantageous under constant operating conditions [11], matching with many compressor applications. A disadvantage is the complexity of the system which leads to higher costs as well as a more complex control system.

The energy efficiency of the ORC process is typically quantified by means of the overall thermal efficiency \(\eta_{th}\), which represents the relationship between the heat flow transferred at the aftercooler \(\dot{Q}_{ac}\) (corresponding to the change in the enthalpy flow of the compressed gas in the aftercooler \(\Delta H_{g,ac}\)) and the electrical power \(P_{el}\) obtained at the expander:

\[
P_{el} = \eta_{th}\dot{Q}_{ac} = \eta_{th}\Delta H_{g,ac} = \eta_{th}m_gc_{pg}\Delta\vartheta_{g,ac}.
\]

For instance, \(\eta_{th}\) is between 8 % and 20 % for an aftercooler inlet temperature \(\vartheta_{ac,in}\) of 200 °C [11]. In the ORC market, which has been growing in recent decades, ORC systems are available in the output range from approx. 1 kWel to 70 MWel [10, 14], which completely covers the power range of WHR for reciprocating compressors.

3.4. Absorption refrigerator

The absorption refrigeration circuit is a well-known and increasingly used technology in the field of refrigeration technology, which offers the possibility of transforming waste heat into cooling capacity. In the underlying absorption process, a mixture of two different soluble substances is used. During the mixing process, the liquid component, the so-called absorbent, absorbs another fluid being gaseous, called the refrigerant. This exothermal process needs to be cooled to keep it running. To separate the two substances again in a downstream part of the cycle the mixture has to be heated until the refrigerant reaches its boiling temperature at a higher-pressure level. This part of the process can be realised e.g.
with the help of waste heat in a temperature range of approx. 80 °C to 170 °C [13]. These substances, in which NH\textsubscript{3} is the refrigerant dissolved in H\textsubscript{2}O, have gained particular technical relevance [14]. Based on the suitable temperature range, this mixture is selected for further consideration (Another possibility results from a H\textsubscript{2}O/LiBr solution, which is not considered further here but must not be excluded in principle for further investigations).

The process of desorption of NH\textsubscript{3} takes place in the generator (G) shown in figure 7, in which the waste heat \( Q_{ac} \) is transferred from the aftercooler to the refrigerant-absorbent-mixture (shown in blue). At high temperatures and pressures, the NH\textsubscript{3} releases the condensation heat \( Q_{amb,c} \) into the environment while being liquefied in the condenser (C) and is throttled to the low-pressure level and to lower temperatures in an expansion device (X). This enables the heat absorption \( Q_{0} \) in the evaporator (E) at temperatures below the ambient temperature even down to sub-zero temperatures. By this, the process provides cooling capacity by supplying heat above the ambient temperature. An absorption refrigeration system also includes the absorber (A), in which the NH\textsubscript{3} is dissolved again in the solvent (water), a pump (P) for increasing the pressure of the mixture and a throttle (T) for returning the refrigerant-weak solution from the generator to the absorber. An internal heat exchanger in the solution cycle is used in almost all cases to increase the efficiency of the system. Further information on the special features and challenges of an absorption refrigeration system can be found, for example, in [13, 15].

Although the technical expenditure for the plant is typically even higher than for the ORC, they offer the advantage that they have been in use for decades, have been sufficiently tested and are reliable in operation. Absorption systems are available in the range from 5 kW\textsubscript{in} to 3 MW\textsubscript{in} [14] and thus cover the relevant range of reciprocating compressors. The efficiency of absorption systems is usually specified with the so-called heat ratio \( \zeta_A \), which specifies the ratio of the cooling capacity \( Q_0 \) provided to the supplied heat \( Q_{ac} \):

\[
Q_0 = \zeta_A Q_{ac} = \zeta_A \Delta H_{g,ac} = \zeta_A m_g c_p \Delta T_{g,ac}.
\]

The heat ratio is around 0.7 for existing systems [16]. The input power of the pump is often neglected due to its comparatively small value. This means that the available cooling capacity can be estimated roughly but conveniently simplified if the available heat supply to the absorption system is known.

4. Possible uses of provided cooling

As with the direct use of waste heat for heating purposes, it is also possible to use the cooling capacity for other processes in the vicinity of the compressor, e.g. air-conditioning. However, since it depends on the individual on-site conditions, this cannot be further analysed and quantified at this point.

In addition, the absorption refrigeration circuit makes it possible to cool compressor components. There are three possibilities for this application:

1. Cooling of cylinder components to increase the isentropic efficiency of the compressor.
2. Cooling of sealing elements (piston rings, packing rings).
3. Cooling of the suction gas to increase the suction gas density and, by that, the delivered mass flow.

4.1. Cooling of cylinder components

The cooling of cylinder components is a well-known process in which a coolant flows through cooling channels usually arranged in the shell of the cylinder. This flow cooling is used to increase the heat dissipation compared to passively cooled compressors (without flow cooling) and has several positive effects on the compressor:

1. Reduction of the discharge temperature.
2. Reduction of the thermal load of the components in the cylinder area.
3. Increasing the efficiency of the compressor.
Active cooling of the cylinder area is not used for all reciprocating compressors but can be a prerequisite for continuous operation in highly stressed applications [17, 18]. In order to investigate the influence of cylinder flow cooling on the energy efficiency of the compressor, various approaches can be found in the literature [19, 20, 21]. However, since the underlying heat transfer depends on the specific geometry and many other boundary conditions the relationship between heat dissipation and the energy efficiency of the compressor cannot be generalized or quantified at this point, but has to be assessed individually for every compressor.

4.2. Cooling of sealing elements
The sealing elements of reciprocating compressors are used to seal gas-filled volumes against ambient (leakage) or to prevent backflow from higher to lower pressure levels within the compressor. Here, the sealing rings at the piston and the sealing rings at the piston rod have to be distinguished. They work as contact seals on an opposing surface. Both sealing surfaces are moved relative to each other, whereby a friction path is covered per revolution, which corresponds to twice the stroke. In this way, for typical average piston speeds \( c_p \approx 2 \text{ to } 8 \text{ m s}^{-1} \) considerable distances of approx. 50 000 to 250 000 km can occur per year in continuous operation. In addition to the distance covered, the contact force in the friction surface is also decisive, since the sealing elements are pressed on by the pressure difference between the two volumes to be sealed. For the piston rod seal, for example, the friction power \( P_{\text{fric,rod}} \) released during sealing can be determined according to [22] by further considering the sealing surface \( A_{\text{Seal,rod}} = \pi d_{\text{rod}} w_{\text{SR}} \) with the piston rod diameter \( d_{\text{rod}} \) and the sealing ring width \( w_{\text{SR}} \) and the coefficient of friction \( \mu_{\text{fric}} \) (approx. 0.2 for oil-free operation, 0.1 with lubrication):

\[
P_{\text{fric,rod}} = 2 s n \mu_{\text{fric}} A_{\text{Seal,rod}} (p_0 + p_1) / 8.
\]

In this way, the frictional heat, which must be dissipated, can easily be quantified for any compressor design and for the associated boundary conditions of the compression task. Based on the considerations made in [22], an approximate expression for the friction power occurring at the piston rings of a double-acting piston with two working chambers that have the same suction and discharge pressure can be established (with the sealing surface again \( A_{\text{Seal,piston}} = \pi d_{\text{piston}} w_{\text{SR}} \)):

\[
P_{\text{fric,piston}} = 2 s n \mu_{\text{fric}} A_{\text{Seal,piston}} \frac{p_2}{2} \frac{\kappa}{\kappa-1} \left( \frac{\kappa-1}{\kappa} - 1 \right),
\]

4.3. Cooling of the suction gas
With conventional systems, the intake gas cannot be cooled any further than close to the ambient temperature, as there is usually no lower temperature level. This can be achieved by providing cooling capacity through an absorption refrigerator. If an approximately isobaric cooling of the suction gas is assumed, this increases the density of the suction gas and thus the flow rate. The heat dissipation for suction gas cooling \( \dot{Q}_{0,s} \) corresponds to the change in the enthalpy flow of the suction gas \( \dot{H}_{g,s} \), whereby assuming ideal gas behaviour and constant specific heat capacity \( c_{p,g,s} \) follows:

\[
\dot{Q}_{0,s} = \dot{H}_{g,s} = \dot{m}_g \Delta h_{g,s} = \dot{m}_g c_{p,g,s} \Delta \theta_{g,s}
\]

5. Summary
Since in reciprocating compressors the waste heat in the aftercooler is usually transferred to the environment after each compression stage and thus remains unused, heat recovery should be sought in the interests of energy-efficient compressor plant design. Since the affected energy quantities are in the order of magnitude of the drive power of the individual compressor, considerable energy saving potentials result from the use of waste heat.

Four different processes were presented for the waste heat recovery, which are suitable for the specified output range and the available temperature level of the gas to be cooled. They are characterised
by the fact that a significant proportion of the waste heat can be recovered. In addition, the processes offer the possibility of flexibly converting the recovered heat into three different forms of energy: Heat for heating purposes, electricity or provision of cooling. Based on the equations given, the waste heat and the benefits that can be achieved using the various processes can be easily quantified. The aim for each compressor must be to best reconcile the supply of waste heat and the local demand. For this purpose, the economic viability of the waste heat recovery systems will also have to be examined in future.

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