APPLICATION NOTE
INDUSTRIAL HEAT PUMPS

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SUMMARY

Industrial heat pumps, using waste process heat as the source, deliver heat at a higher temperature for use in industrial process heating, preheating, or space heating. There is debate over their definition, but, in general, they represent a worthwhile method of improving the energy efficiency of industrial processes and reducing primary energy consumption.

Industrial heat pumps (IHPs) offer various opportunities in all types of manufacturing processes and operations. Increased energy efficiency is certainly their most obvious benefit, but few companies have realized the untapped potential of IHPs in solving production and environmental problems. This Application Note demonstrates that IHPs can offer the least-cost option for removing bottlenecks in production processes and allowing greater product throughput and, in fact, may be an industrial facility's best way of significantly and cost-effectively reducing combustion-related emissions.

PHYSICAL PRINCIPLES

A heat pump is essentially a heat engine operating in reverse. The principle is illustrated in Figure 1.

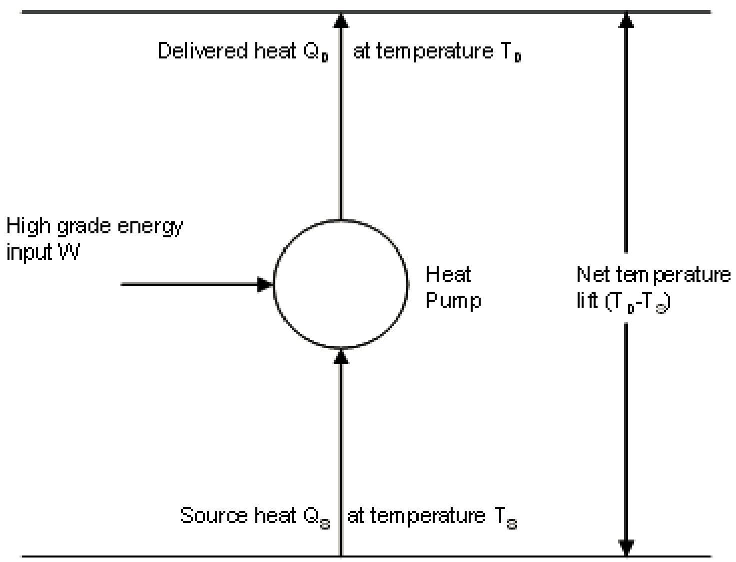

Figure 1 – Heat pump principle.

From the first law of thermodynamics, the amount of heat delivered Q_D at the higher temperature T_D is related to the amount of heat extracted, Q_S, at the low temperature, T_S, and the amount of high grade energy input, W, according to the equation:

$$Q_D = Q_S + W$$

Compared to heat pumps for space heating, which use heat sources such as ground or water, IHPs often have the following advantages:

- High coefficient of performance due to low temperature differences and/or high temperature levels
- High duty factors
- Relatively low investment cost, due to the use of large units and short distances between heat source and heat sink
- The availability of waste heat and the demand for heat occur simultaneously and in close proximity

Despite these advantages, the number of heat pump installations in industry is almost negligible compared to those installed for space heating.

A coefficient of performance (COP) can be defined as:

$$COP = \frac{Q_D}{W}$$

The Carnot coefficient of performance

$$COP_C = \frac{T_D}{T_D - T_S}$$

represents the upper theoretical value obtainable in a heat pump system. In practice, attainable coefficients of performance are significantly less than COP_C. Unfortunately, it is difficult to compare the COPs of different categories of IHP, which differ widely for equivalent economic performance.

When comparing heat pump systems driven by different energy sources it is more appropriate to use the primary energy ratio (PER) defined as:

$$PER = \frac{useful\ heat\ delivered}{primary\ energy\ input}$$

The equation can be related to the coefficient of performance by the equation

$$PER = \boldsymbol{\eta} COP$$

where η is the efficiency with which the primary energy input is converted into work up to the shaft of the compressor.

Heat pump installations

It is possible to use a number of different types of heat pump cycles in industrial applications. These cycles can be categorised in various ways, e.g. as mechanically- or heat-driven, compression or absorption, closed or open cycles. The most important are:

- Closed compression cycle, electric motor-driven
- Closed compression cycle, diesel motor-driven
- Mechanical vapour recompression (MVR)
- Thermal vapour recompression (TVR)
- Absorption cycles (heat pump and heat transformer)

Closed compression cycle

The principle of the simple closed compression cycle is shown in Figure 2.

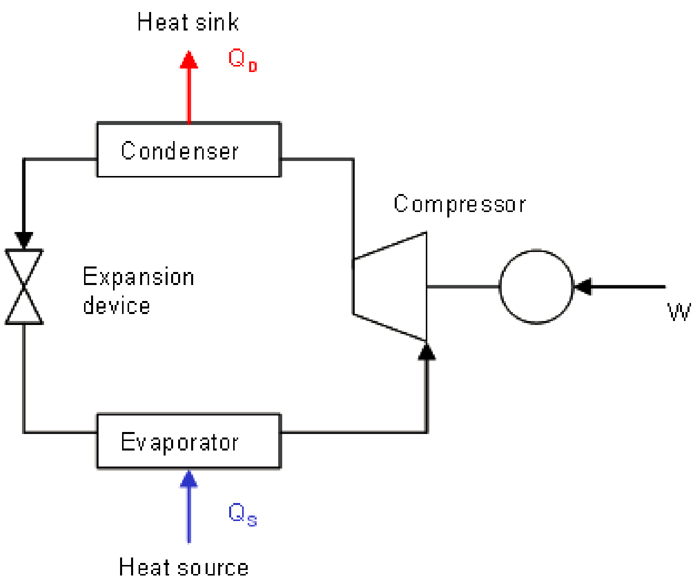

Figure 2 – Closed compression cycle.

The COP is given by

$$COP = \frac{Q_D}{W} \approx \frac{Q_D}{Q_D - Q_S}$$

For a specified design of cycle, the Carnot efficiency can often be regarded as a constant with varying T_D and T_S, provided these variations are moderate.

To increase the COP of this type of cycle, internal modifications of the simple cycle are normally carried out:

- Sub-cooling of the condensate after it has passed through the condenser can be performed by the heat sink. With this arrangement, the heat output from the heat pump increases without any increase in compressor work. As a result, the cycle is often very economical, at least when the sink is in liquid form. Typical improvements of the COP and the capacity are approximately 1% per degree K of sub-cooling.
- Another possible improvement is to divide the expansion of the condensate from the condenser into two stages. The vapour part of the working fluid after the first expansion valve is compressed, without

passing it through the evaporator, so the temperature lift and the need for compression of this vapour part decrease, and the COP as well as the capacity increases. This arrangement, known as an *economiser*, is often used in industrial applications. The cycle assumes a two-stage compressor.

- A rather similar arrangement is provided by the *flash intercooler*, in which the proportion of vapour to be compressed is increased, but the temperature lift is reduced. This is achieved by using the superheat of the vapour coming from the first compressor to evaporate some of the liquid working fluid from the first expansion stage. This type of cycle theoretically gives better performance than the economiser cycle. Its disadvantage is pressure drop in the intercooler, and a risk of entrainment of liquid drops in the second-stage compressor.
- In situations where the temperature lift needed is large, multi-stage (*cascade*) cycles are possible options. They allow different working fluids to be used at each stage, and reasonable pressure ratios to be achieved in each compressor. Heat pumps coupled in series are beneficial when there are large temperature gradients on the heat sink and source.

Three different compressor types are used in closed compression cycle heat pumps: reciprocating, screw and turbo compressors. Reciprocating compressors are used in systems up to approximately 500 kW heat output, screw compressors up to around 5 MW and turbo compressors in large systems (above about 2 MW heat output). The COP can be approximately determined from the Carnot coefficient COP_C. This coefficient varies with the working fluid, but typical values are 0.448 for a reciprocating compressor, 0.55 for a screw compressor and 0.64 for a turbo compressor.

ELECTRIC MOTOR-DRIVEN SYSTEM

The most common type of compressor drive is the electric motor. Its efficiency varies from 70% to 97% depending on its size and loading. In industrial heat exchanger applications, it is normally possible to operate at efficiencies above 90% from full load down to less than half the nominal load.

Figure 3 shows typical COP values versus evaporation temperature, with condensation temperature and type of compressor as parameters, for an electric motor-driven closed-compression economizer cycle.

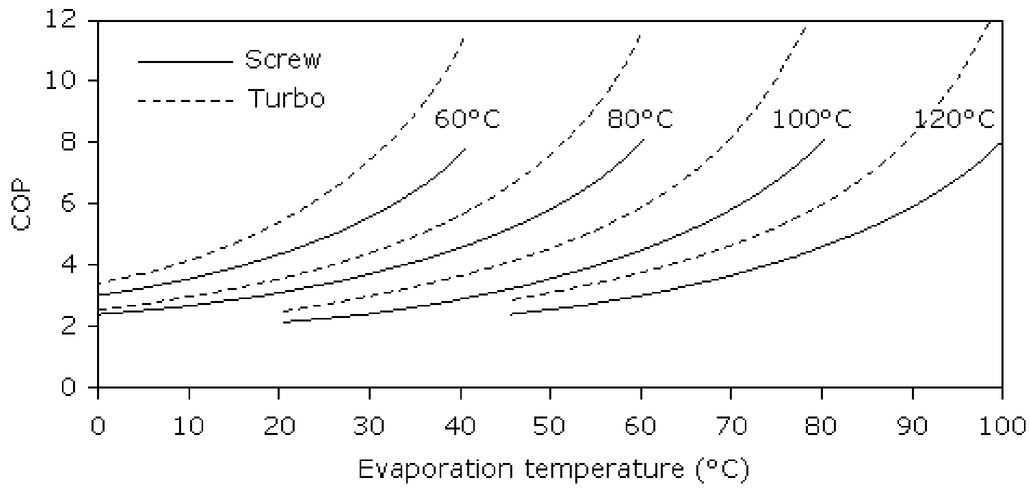

Figure 3 – COP versus heat pump characteristics.

It can be seen that the turbo compressor gives a higher COP than the screw compressor, especially at small temperature lifts.

DIESEL ENGINE-DRIVEN SYSTEM

When a diesel engine is used to drive the compressor, the waste heat from the engine can also be utilised to heat the sink. Heat that is of use in industrial applications is available from the engine in the exhaust gases and also, for applications at a temperature around 100°C, in the cooling water. One scheme for utilizing this heat is shown in Figure 4.

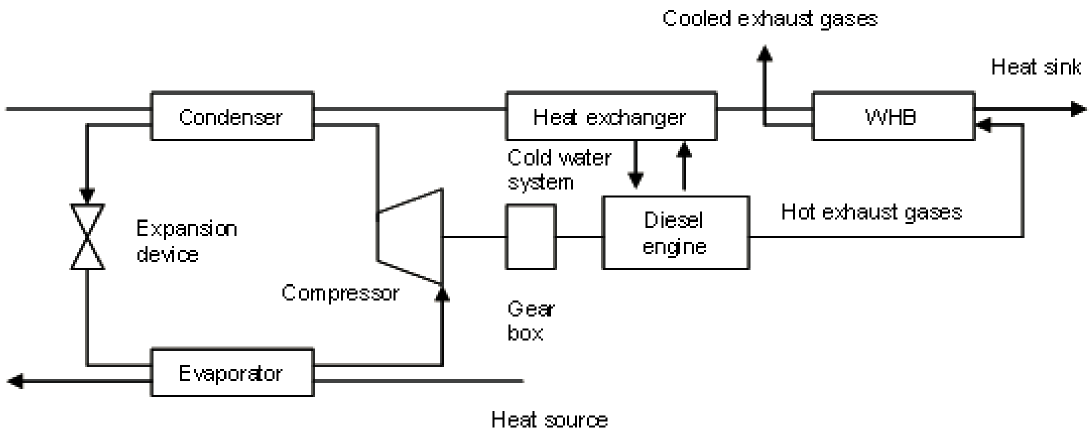

Figure 4 – Diesel engine - driven heat pumps.

The mechanical efficiency for modern diesel engines suitable for heat pump drives is above 0.4 (values of up to 0.45 are found).

The COP of a diesel engine-driven heat pump can be calculated from the COP of the heat pump cycle itself, with the following equation:

$$COP = \eta_m \times COP_{HP} + (\eta_{tot} - \eta_m)$$

where:

COP_{HP} is the COP of the heat pump itself, compensating for the electrical efficiency

η_m is the mechanical efficiency of the diesel engine

η_{tot} is the total efficiency of the diesel engine

MECHANICAL VAPOUR RECOMPRESSION (MVR)

Mechanical vapour recompression is the technique of increasing the pressure, and thus also the temperature, of waste gases thereby allowing their heat to be reused. The most common type of vapour compressed by MVR is steam, to which Figures 5 and 6 refer.

There are several possible system configurations. The most common is a semi-open type, in which the vapour is compressed directly (also referred to as a direct system). After compression, the vapour condenses in a heat exchanger where heat is delivered to the heat sink. This type of MVR system is very common in evaporation applications.

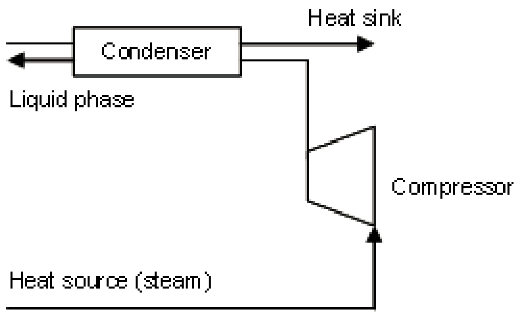

Figure 5 – Mechanical vapour recompression.

The other type of semi-open system lacks the condenser, but is equipped with an evaporator. This less usual configuration can be used to vaporise a process flow that is required at a higher temperature, with the aid of mechanical work and a heat source of lower temperature.

Figure 6 shows the COP versus temperature lift for a typical MVR system using a screw compressor. It can be deduced that MVR systems give very high COP and that the COP is very dependent on the temperature lift.

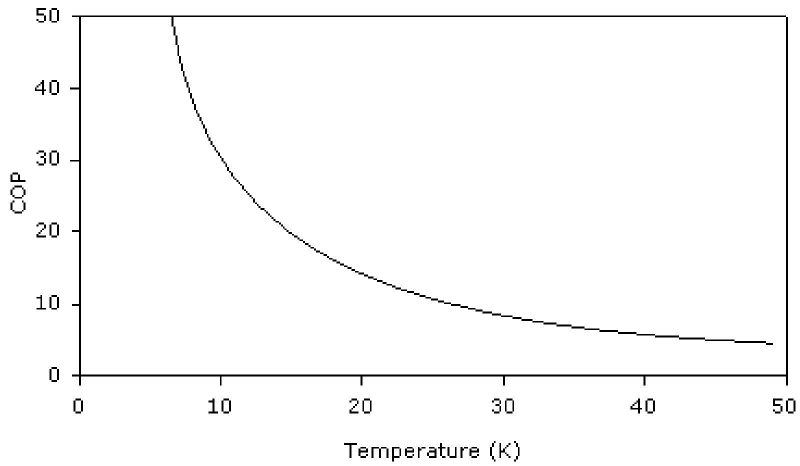

Figure 6 – COP versus temperature lift for MVR using a screw compressor.

By far the most important design issue in MVR systems is the choice of the compressor type. Compressors for MRV systems are of two main types: turbo and positive-displacement compressors.

THERMAL VAPOUR RECOMPRESSION (TVR)

With the TVR type of system, heat pumping is achieved with the aid of an ejector and high-pressure vapour. It is therefore often simply called an ejector. The principle is shown in Figure 7. Unlike the MVR system, a TVR heat pump is driven by heat, not by mechanical energy. Thus, compared to an MVR system, it opens up new application areas, especially in situations where there is a large difference between fuel and electricity prices.

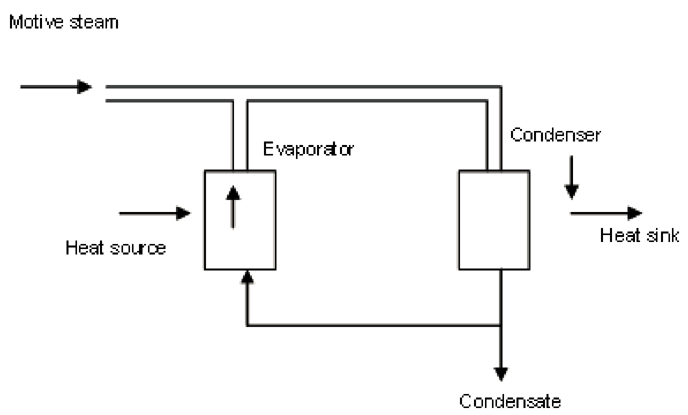

Figure 7 – Thermal vapour recompression.

The TVR type is available in all industrial sizes. A common application area is evaporation units.

The COP is defined as the relation between the heat of condensation of the vapour leaving the TVR and heat input with the motive vapour. Figure 8 shows COP versus temperature lift, defined as the temperature difference between condensation and evaporation temperatures. As can be seen from the Figure 8, the COP is modest.

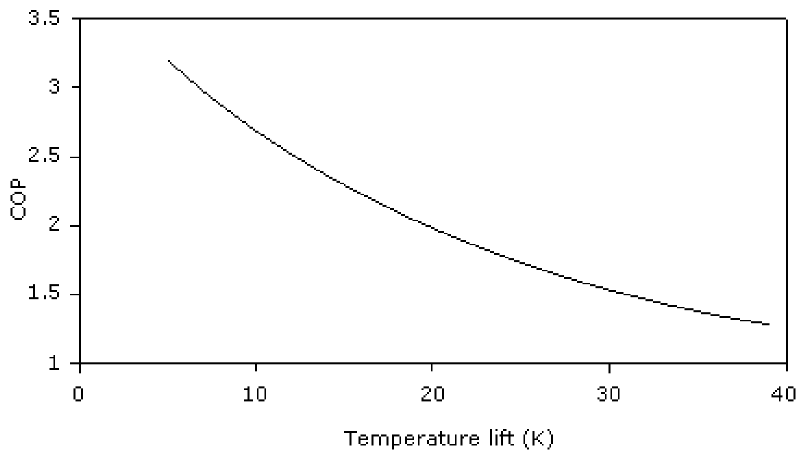

Figure 8 – COP versus temperature lift for TVR.

ABSORPTION HEAT PUMPS

Absorption heat pump cycles are based on the fact that the boiling point for a mixture is higher than the corresponding boiling point of a pure, volatile working fluid. Thus the working fluid must be a mixture consisting of a volatile component and a non-volatile one. The most common mixture in industrial applications is a lithium bromide solution in water ($LiBr/H_2O$).

The fundamental absorption cycle has two possible configurations: absorption heat pump (AHP, Type I) and heat transformer (AHP, Type II), which are suitable for different purposes. The difference between the cycles is the pressure level in the four main heat exchangers (*evaporator, absorber, desorber and condenser*), which influence the temperature levels of the heat flows.

ABSORPTION HEAT PUMP, TYPE I

In the absorption heat pump cycle, heat is lifted from a low temperature level to a medium temperature level. This is achieved by supplying heat at high temperature level.

There are three parameters of interest. The COP determines the amount of heat that can be delivered in relation to the heat supplied. The second parameter is the possible temperature lift that can be achieved at various magnitudes of the three temperature levels and the third is the maximum possible temperature level at which heat can be delivered. The limiting factor is the risk of crystallization. These parameters are shown for two working fluids in Table 1.

Working pair	COP	Max. temp. lift (°C)	Max. delivery temp. (°C)
LiBr/H2O	1.6 – 1.7	45 – 50	100
Alkitrate	1.6 – 1.7	50	200

Table 1 – Type 1 absorption heat pump characteristics.

From Table 1 it is clear that the absorption heat pump with LiBr/H_2O as the working pair can only be used for heat demands below 100°C, due to the limit of the delivery temperature. This significantly limits the use of the cycle in industrial applications. COP is further decreased to 1.3 or 1.4, when boiler efficiency is taken into account.

Absorption heat pump, Type II (heat transformer)

In the heat cycle, heat is supplied at a medium level. Part of this heat is transformed to a high level and the remainder is discharged at a low level. The heat transformer is useful for recovering industrial waste heat at a medium temperature level and replacing primary heat.

For industrial applications, LiBr/H_2O is the only working pair in use.

As in the case of the absorption heat pump, there are three parameters of major interest. The COP, which determines the relation between delivered heat at high temperature and recovered heat at medium temperature, is 0.445-0.49. Thus, nearly half the waste heat can be transformed to high and useful temperature level. The maximum operating temperature is 150°C which, for instance, implies that low-pressure steam can be produced.

CHARACTERISTICS OF HEAT PUMPS

TECHNO-ECONOMIC COMPARISON OF IHP TYPES

Each of the heat pump types considered is applicable to a specific operating temperature range. Some of these ranges overlap, which makes it possible to choose between the types in design situations. Approximate technical operating limits of the various types and typical costs for three sizes are summarised in the Table 2, from which it is possible to select those IHP types that are feasible in a specific situation.

IHP Type	Max. sink temp. (°C)	Max. temp. lift (°C)	Installation cost 0.5 $MW_{heat\ output}$ (€/$kW_{heat\ output}$)	Installation cost 1 $MW_{heat\ output}$ (€/$kW_{heat\ output}$)	Installation cost 4 $MW_{heat\ output}$ (€/$kW_{heat\ output}$)
Electric motor CCC	120	80	450 – 700	320 – 550	240 – 420
Diesel motor CCC	130	90	520 – 770	390 – 620	300 – 490
MVR	190	90	Not available	380 – 450	135 – 220
TVR	150	40	Not available	210 – 270	100 – 120
Absorption, Type I (LiBr/H_2O)	100	50	340 – 390	300 – 350	250 – 290
Heat transformer (LiBr/H_2O)	150	60	800 – 900	720 – 830	590 – 680

Table 2 – Techno-economic comparison of IHP types.

There are many possible methods of evaluating investment options and performing economic comparisons. In industry, the payback period method is well known, and is often used for a quick comparison of alternatives. It is assumed that the useful heat generated in the IHP replaces heat from an existing boiler (efficiency = η_b). The payback period for any IHP project can be expressed using the equation:

$$PBP = \frac{I}{\left(\frac{B_{fuel}}{\eta_b} - \frac{B_{drive}}{COP}\right) \times 8760 - m_{IHP}}$$

where:

PBP is the payback period [operating years]

I is the project investment cost [currency units per $kW_{heat\ output}$]

B_{fuel} is the fuel price [currency, unit price per kWh]

B_{drive} is the price of drive energy to the heat pump [unit price per kWh]

ηb is the efficiency of existing heating equipment e.g. boiler

COP is the COP of the IHP

m_{IHP} is the annual maintenance cost of the heat pump [cost per $kW_{heat\ output}$].

Acceptable payback periods vary between different countries and industry sectors, and also depend on the type of installation. However, acceptable payback periods are normally between two and three years.

From the comparisons made, some general conclusions can be drawn:

- Provided the operating temperatures are such that both types can operate, the MVR and the TVR have the shortest payback periods of all heat pump types, and are strong competitors. The payback periods obtained are economically attractive except at low fuel prices.
- The payback period of the electric motor driven IHP is strongly influenced by the electricity price and the COP (the temperature lift). A COP below 4 is normally not acceptable, but at a COP of 6 there is a good possibility of economically favourable installation.
- From the definition of the payback period, it is clear that it is important to investigate possibilities for decreasing the total investment cost for a heat pump installation, i.e. the heat pump itself, its installation and other associated costs, not just the cost of the equipment.

CRITERIA FOR POSSIBLE HEAT PUMP APPLICATIONS

The first step in any possible IHP application is to identify technically feasible installation alternatives and possibilities for their economic installation. In simple operations, where the process in which the IHP will be used consists of only a few streams with obvious sink and source, a thorough assessment is normally not necessary. In these cases, only the characteristics of the sink and source are of importance for the feasibility and selection of the IHP. The obvious parameters are:

- Heat sink and source temperature
- Size (in terms of heat load) of the sink and source
- Physical parameters of the sink and source, such as phase and location.

The sink and source temperatures determine which IHP types can be used in a specific application. Approximate operating limits of the various types were already summarised in Table 2. In Table 3 the heat ratio between the heat sink and source (q-value) is presented for each of the IHP types.

IHP Type	**Q = heat sink / heat source**
Electric motor CCC	1.1 – 1.5
Diesel motor CCC	1.3 – 3
MVR	1.1 – 1.4
TVR	1.7 – 10
Absorption, Type I ($LiBr/H_2O$)	2.5
Heat transformer ($LiBr/H_2O$)	0.5

Table 3 – q-value for industrial heat pumps.

From Tables 2 and 3 it is possible to identify the IHP types that are feasible. In a simple application, the possible types can then be evaluated using practical and economic considerations.

Industrial applications of heat pumps

Heat pumps in drying operations

A variety of drying processes are employed in petrochemical plants when a condensable substance such as water vapour is to be removed from a “non-condensable” substance, such as air. Heat pumps have already found wide application in such cases in the timber and paper industry.

With this equipment, the solvent is no longer withdrawn with the exhaust air in the vapour phase as in conventional dryer applications but is condensed on the cold evaporator surface and withdrawn as a liquid. In this way, the heat of condensation can be recovered. A reduction of the primary energy requirement is possible. Studies show that there are optimum working conditions for heat pumps in air dryer service. Several flow schemes have been compared and the primary energy reduction has been evaluated by replacing the conventional ventilation dryers with heat pumps.

If the evaporator of a heat pump is installed in the exhaust channel and the condenser at the fresh air inlet, it becomes possible to recover the sensible heat of the exhaust air and the latent heat of condensation of the water vapour it contains. This arrangement is called a “recuperator heat pump”.

The performance of a recuperator heat pump can be improved if a combustion engine is used to drive the compressor. In this case, the waste heat of the engine can be used for additional heating of the inlet air stream (see Figure 9). The efficiency of the process can be further improved by heat exchange between the incoming and outgoing air.

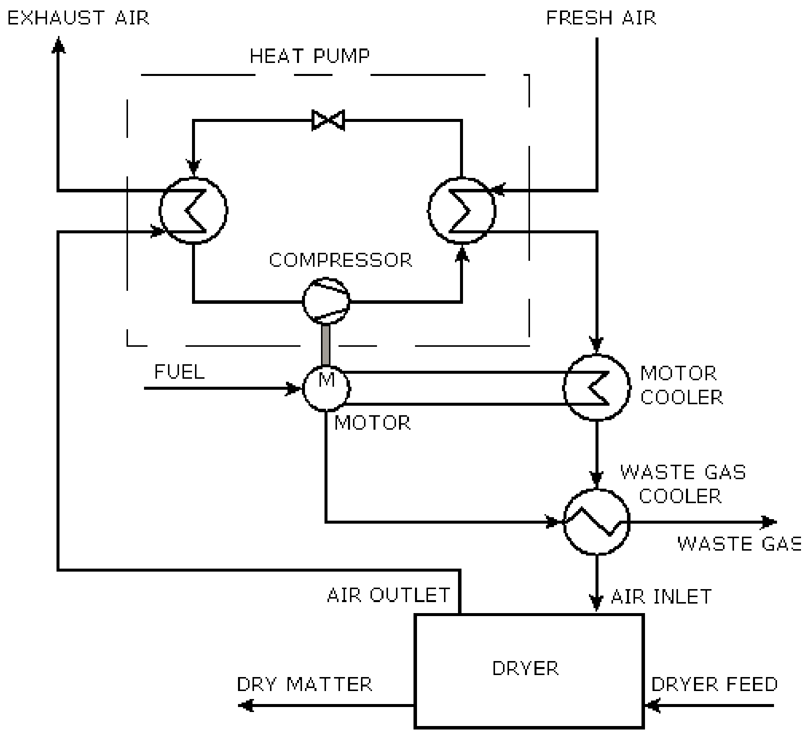

Figure 9 – Recuperator heat pump and utilisation of the motor waste heat.

Heat pumps in distillation

The high energy requirements of distillation can often be reduced by using a heat pump system to “pump” heat from the condenser to the reboiler. This is accomplished by using compression to raise the temperature level of the available heat from that of the condenser to that of the reboiler. Studies have been carried out to develop guidelines for conditions under which heat pumps can be economical in distillation process design. The flow scheme considered consists of a column, reboiler and condenser. The heating medium for the reboiler was condensing steam. The pressure and condensing temperatures were determined by the reboiler temperature, while the cooling medium chosen depended on the overhead condensing temperature.

The simplest alternative to the conventional design involving the use of a heat pump is to replace the steam heating of the reboiler by a condensing refrigerant at a relatively high pressure and to replace the coolant by an evaporating refrigerant at relatively low pressure. Thus, the reboiler becomes the condenser and the column condenser becomes the evaporator of a heat pump system. This type of heat pump system is shown in Figure 10. The column itself is not changed from the conventional system, but the heat exchangers will be quite different. The ratio of the heat pumped to the reboiler Q_R to the compression work W required can be approximated by:

$$COP = \frac{Q_R}{W} = \eta_{tot}\frac{T_3}{T_3 - T_1}$$

Where η_{tot} is the combined compressor efficiency and fluid cycle efficiency.

If the refrigerant is properly chosen T_2 and T_3 in Figure 10 of the flow diagram of a conventional distillation plant will be almost the same, and the equation becomes:

$$\frac{Q_R}{W} = \frac{\eta_{tot}(T_R + \Delta T_R)}{T_R - T_C + (\Delta T_R + \Delta T_C)}$$

The T_R in the numerator of the equation has little effect, but the total ($\Delta T=\Delta T_R+\Delta T_C$) in the denominator is a critical performance parameter. To minimize the compression work required to pump the heat, both Q_R and ΔT should be minimized. However, lowering ΔT means increasing the heat exchanger area and consequently the capital cost. For many heat pump systems the range of economically feasible values for ΔT is limited.

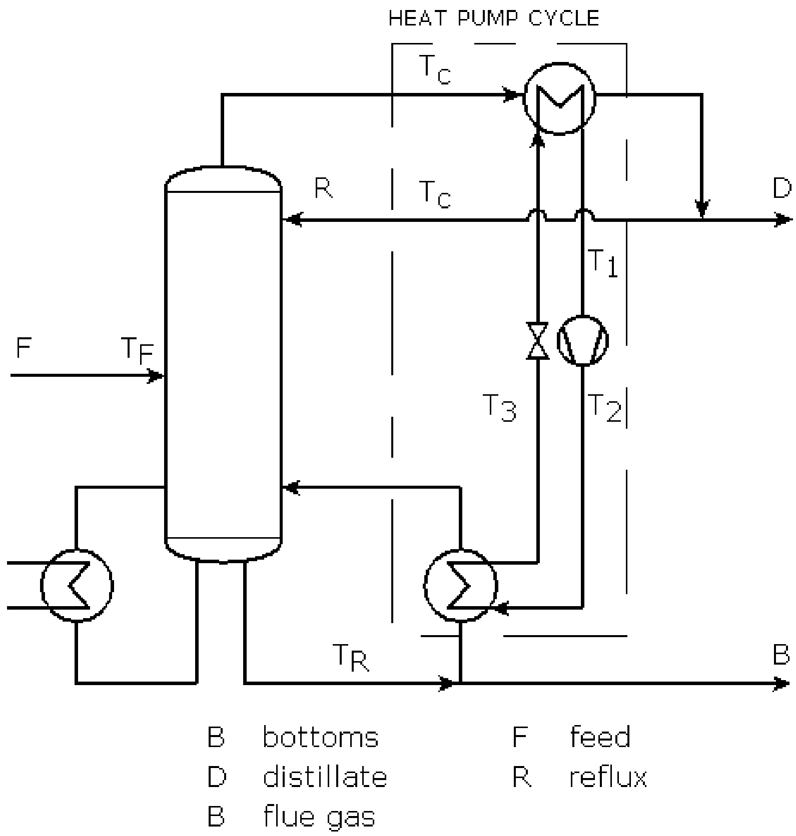

Figure 10 – Distillation unit with closed heat pump cycle.

The results of a comparison of a conventional column and one with an MVR heat pump are given in Table 4.

		Conventional distillation plant	Column with heat pump
Design data	Column diameter	2.5 m	
	Packing height	12 m SULZER Packing Type BX	
	Top pressure/temp.	175 mbar / 124°C	
	Bottom pressure	199 mbar/135.5°C	
	Column pressure (bottom)	24 mbar	
	Boil-up rate	36,000 kg/h	
	Heat of vaporisation (top)	74.5 kcal/kg	
Energy	Energy required for boil-up (kW)	3,120	-
	Turbo-blower duty (kW)	-	310
	Reboiler duty (kW)	-	23
	Total energy amount (kW)	3,120	330
Energy cost	Steam cost (€/h at a rate of €25/tonne)	140	1
	Electrical energy cost (€/h at a rate of €0.06/kWh)	-	18.6
	Total hourly energy cost	-	19.6
	Total annual energy cost	1,125,000	157,500
	Cost of energy consumption (%)	100	14

Table 4 – Case study heat pumps in distillation.

CONCLUSIONS

The economics of an installation depend on how the heat pump is applied in the process. The identification of feasible installation alternatives for the heat pump is therefore of crucial importance taking the heat pump and process characteristics into account. The initial procedure should identify a few possible installation alternatives, so the detailed project calculations can concentrate on a limited number of options.

Each of the commercially available heat pump types has different operating characteristics, and different operating temperature ranges. Thus, for a particular application, several possible heat pump types may be practical. Technical, economic and practical process criteria determine the most suitable type.

For all types, the payback period is directly proportional to installation costs, so it is important to investigate possibilities for decreasing these costs for any heat pump installation.

BIBLIOGRAPHY

[1] Scientific rapport, Ekonomische aspekten van de toepassing van de warmtepomp in de Belgische industrie, KU Leuven, September 1982

[2] Moser, F and Schnitzer, H, Heat Pumps in Industry, Elsevier Science Publishers, New York, 1985

[3] Industrial Heat Pumps, CADDET Energy Efficiency, Sittard, 1997

[4] Handboek Industriële Warmtepompen, Kluwer Techniek, Deventer, 1998