NOVEL CYCLES FOR POWER AND REFRIGERATION

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ABSTRACT

It is well known that cold can be produced from heat using sorption cooling cycles. It is also well known that low-grade heat can be converted into power also, although the efficiency is low. Considering the fact that both for producing power and cold sorption cycles may be used both conversion processes can be combined and a cogeneration of power and cold from heat can be envisaged.

In this paper, the basic features and some cycle configurations with the potential of being feasible will be discussed, although practical experience is still lacking. The main obstacles for widespread use are the high cost for heat exchange, which defines a large research need. Also control may play an important part.

INTRODUCTION

Low temperature heat sources are abundant. Especially the industrial waste heat is attractive due to the fact that its disposal itself bears expenses because of investment for and operating of the necessary cooling equipment. It is interesting to note that the way of using low temperature heat is quite different, depending on its origin. Typically, for industrial waste heat, upgrading or production of cooling is discussed. In the case of geothermal heat, the conversion to electricity is very prominent today. This paper is supposed to combine these options, and it should demonstrate some development options for the future.

We start off with some considerations about efficiency of individual production of cold and power from low-temperature heat sources. We continue with combining the production. Finally, we discuss some technical solutions.

EFFICIENCY OF SEPARATE PRODUCTION

Heat Driven Chillers

Let us have a look at an absorption chiller in the conventional black-box manner (Figure 1a). It is characterised by the temperatures of the heat sink (environment) T_1 and the respective heat flow, Q_1, the heat source (produced cold Q_0) T_0, and the driving heat temperature T_2 and heat flow, Q_2. The well-known cooling $COP = Q_0/Q_2$ results to:

$$COP = \frac{T_0}{(T_1 - T_0)} \frac{(T_2 - T_1)}{T_2} \quad g \qquad [1]$$

Here, the thermodynamic quality g is the ratio between the real efficiency COP and the thermodynamic limit; it accounts for irreversibilities due to fluid restrictions (staging etc.) or design of the equipment.

Let us now assume that the low temperature heat source exhibits a relatively large glide with temperature of the flow (in) and return (out), T_{2i} and T_{2o} (Figure 1b). We can still use equation [1] if we substitute for T_2 the thermodynamic mean temperature of the heat source, $\breve{T}_2$, which is defined for constant heat capacities by:

$$\breve{T}_2 = \frac{T_{2i} - T_{2o}}{\ln T_{2i} - \ln T_{2o}} \qquad [2]$$

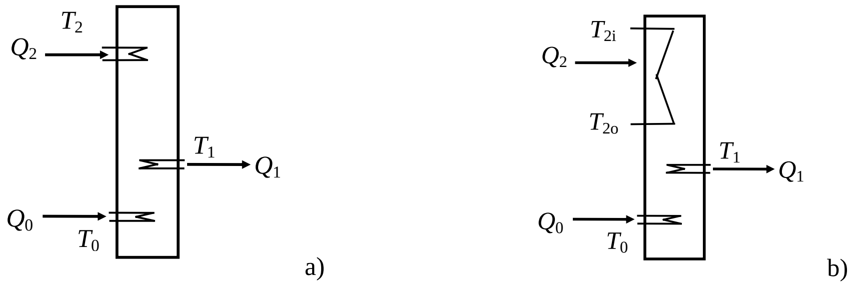

Figure 1: Black-box representation of an absorption chiller; small (a) and large (b) glide in the heat source (supply T_{2i}, return T_{2o});heat sink temperature (cooling water) T_1; production of cold at T_0.

The thermodynamic mean is reasonably near to the arithmetic mean (Ziegler, 1998). We find immediately equation [3] which is displayed in Figure 2:

$$COP = \frac{T_0}{(T_1 - T_0)}\left[1 - \frac{T_1}{T_{2i} - T_{2o}} ln\left(\frac{T_{2i}}{T_{2o}}\right)\right] g \approx \frac{T_0}{(T_1 - T_0)}\left[\frac{T_{2i} - T_1}{T_{2i}} - \frac{1}{2}\frac{T_1(T_{2i} - T_{2o})}{T_{2i}^{\;2}}\right] g \quad [3]$$

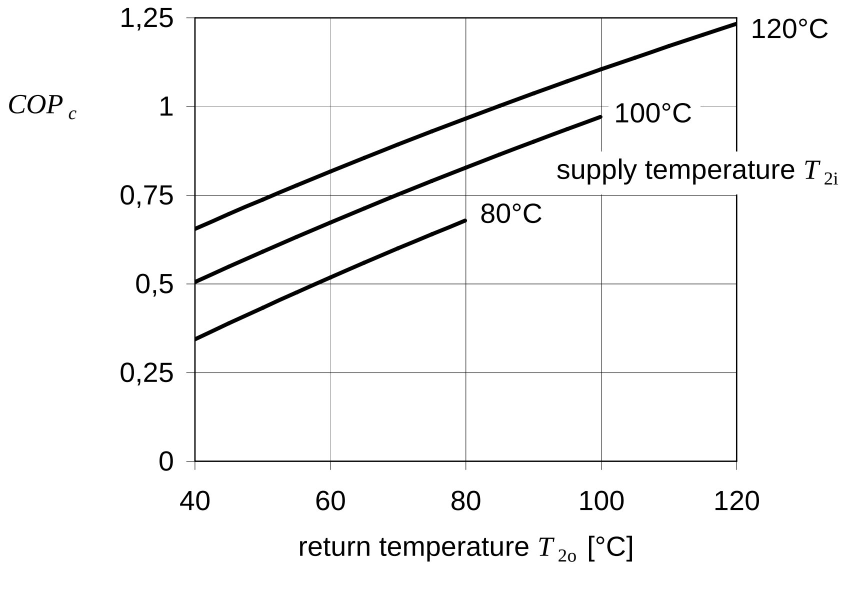

Figure 2. Coefficient of performance COP of a heat driven chiller according to equation [3] in dependence of the return temperature T_{2o} of the driving heat source. Parameter: Supply temperature T_{2i} of the driving heat source. heat sink temperature 31°C, driving temperature difference in the heat exchangers 5K, cold production at T_0=9°C, thermodynamic quality of the process g=0.7.

Please note that in this Figure irreversibilities both for external heat exchange and internal features are contained. It is interesting to see that even with relatively low return temperatures acceptable *COP*s can be attained. The prerequisite, of course, is the availability of adapted machinery. In our equation, we assume that for any external temperature we can find an adequate cycle with only decent irreversibilities. This normally is not the case. Here we are only interested in the potential limits.

Power Plants

Let us now look at thermal power plants. If we use the well-known Carnot-efficiency with the mean input temperature according to [2] we find equation [4], which is displayed in Figure 3. It is obvious that the efficiencies are quite low in comparison with fossil fired plants, and even in comparison with the efficiency of a heat driven chiller.

$$\eta = \left[1 - \frac{T_1}{T_{2i} - T_{2o}} \ln\left(\frac{T_{2i}}{T_{2o}}\right)\right] g \approx \left[\frac{T_{2i} - T_1}{T_{2i}} - \frac{1}{2}\frac{T_1 (T_{2i} - T_{2o})}{T_{2i}^{\ 2}}\right] g \qquad [4]$$

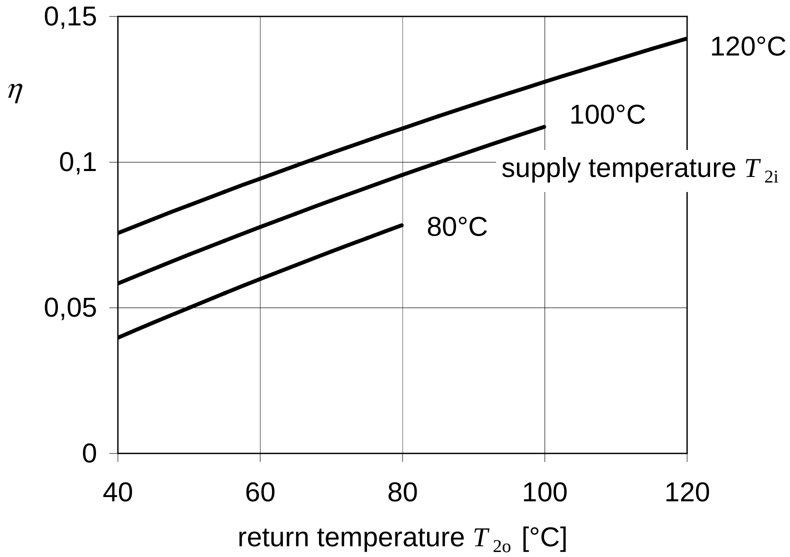

Figure 3: Efficiency η of the power station according to equation [4] in dependence of the return temperature T_{2o} of the driving heat source. Parameter: Supply temperature T_{2i} of the driving heatsource. heat sink temperature 31°C, driving temperature difference in the heat exchangers 5K, thermodynamic quality of the process g=0.7.

COMPARISON

If we compare equation [3] to [4] we see that - even for real machines – in order to calculate the *COP* for cooling under comparable circumstances (temperature of driving heat source and heat sink, quality) the power station efficiency η has to be multiplied by the coefficient of performance $T_0/(T_1 - T_0)$ of a reversible compression chiller working between T_0 and T_1:

$$COP = \frac{T_0}{(T_1 - T_0)} \eta \qquad [5]$$

The ratio $T_0/(T_1 - T_0)$ typically is in the order of 10. The thermal efficiency of producing cold, therefore, is about one order of magnitude higher than the one of power plants. That means that – from a given low temperature heat source - ten times more cooling energy than electrical energy can be produced.

EFFICIENCY OF COMBINED PRODUCTION

Basics

The scheme of a combined cycle producing power and cold from a heat source with gliding temperature is displayed in Figure 4: we just have to add the power output, *P*.

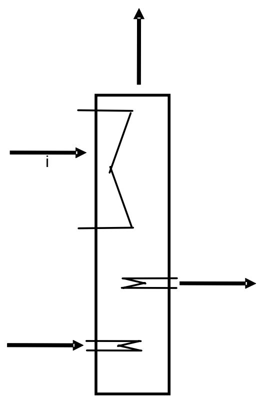

Figure 4: Black-box representation of an absorption chiller according to Figure 1,but producing power P also

The driving heat input, Q_2, is now the energy which is required to drive both the cooling and the power producing process. In order to describe the performance we need two different, independent numbers because we have two different outputs. These may be chosen from the electrical efficiency, P/Q_2, the cooling efficiency, Q_0/Q_2, and the power-to-cold ratio, P/Q_0. We find:

$$\frac{Q_0}{Q_2} = \frac{1 - \frac{T_1}{T_{2i} - T_{2o}} \ln\left(\frac{T_{2i}}{T_{2o}}\right)}{\frac{P}{Q_0} + \frac{T_1 - T_0}{T_0}} \quad g \tag{6}$$

$$\frac{P}{Q_2} = \frac{1 - \frac{T_1}{T_{2i} - T_{2o}} \ln\left(\frac{T_{2i}}{T_{2o}}\right)}{1 + \frac{T_1 - T_0}{T_0} \frac{Q_0}{P}} \quad g \tag{7}$$

With using the equations [3] and [4] we can write also:

$$\frac{Q_0}{Q_2} = \frac{COP}{1 + \frac{T_0}{T_1 - T_0} \frac{P}{Q_0}} \qquad [8]$$

$$\frac{P}{Q_2} = \frac{\eta}{1 + \frac{T_1 - T_0}{T_0} \frac{Q_0}{P}} \qquad [9]$$

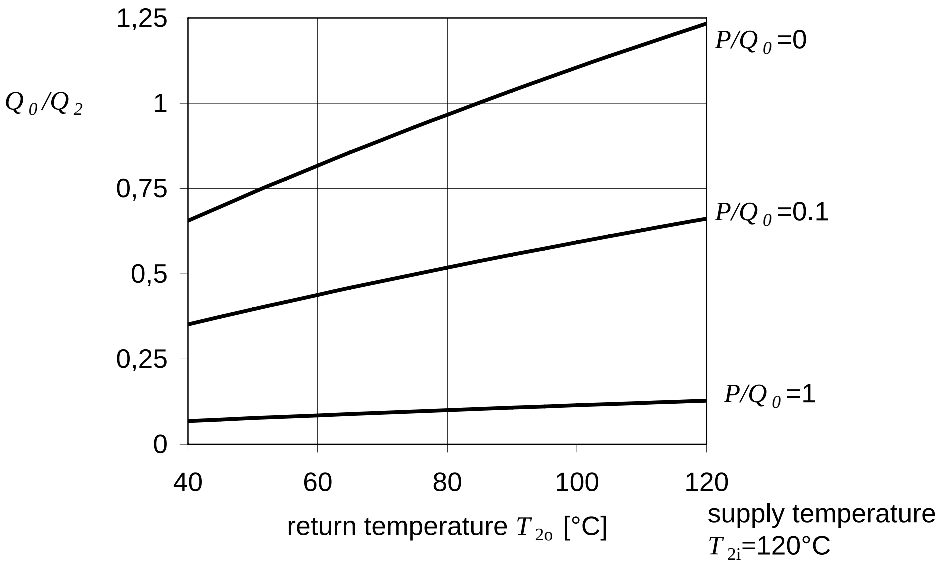

Figure 5: Performance characteristic Q_0/Q_2 according to equation [6] in dependence of the return temperature T_{2o} of the heat source. Parameter: ratio of power to cold, P/Q_0supply temperature T_{2i} = 120°C. Heat sink temperature 31°C, cold production at T_0=9°C, driving temperature difference in the heat exchangers 5K, thermodynamic quality of the process g=0.7.

Both equations are displayed in Figures 5 and 6. The ratio of cold production (heat flow Q_0) to power output P or vice versa is used as a parameter. The uppermost curves in both cases denotes the case where merely cold and no power is produced (Figure 5) or vice versa (Figure 6). It is obvious - and also easily understandable from basic thermodynamics - that the cold output of the system responds much stronger to an increase in power output (Figure 5) than the power output responds to an increase in cold output (Figure 6). As a consequence we can immediately conclude that the co-production of cold in a power station will have only very little effect on the primary energy need of the system, whereas the co-production of power from an absorption chiller will have significant consequences.

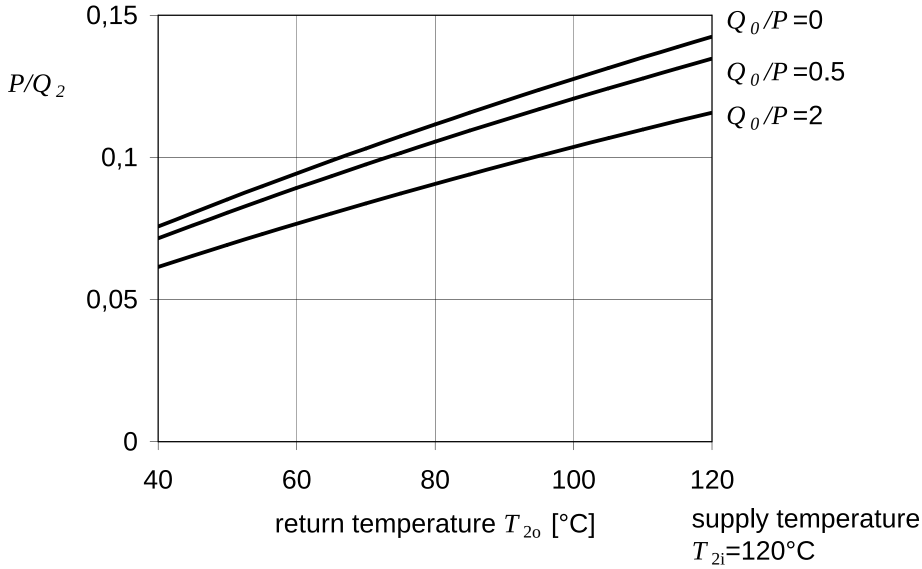

Figure 6: Performance characteristic P/Q_2 according to equation [7] in dependence of the return temperature T_{2o} of the heat source. Parameter: ratio of cold to power, Q_0/P supply temperature T_{2i} = 120°C heat sink temperature 31°C, cold production at T_0=9°C, driving temperature difference in the heat exchangers 5K, thermodynamic quality of the process g=0.7.

Performance chart

We can also combine equations [6] and [7] to give us the driving input heat flow Q_2 which is required for a given output of cold, Q_0, and power, P:

$$Q_2 = \frac{P + \frac{T_1 - T_0}{T_0} Q_0}{\left[1 - \frac{T_1}{T_{2i} - T_{2o}} ln\left(\frac{T_{2i}}{T_{2o}}\right)\right] g} = \frac{P + \frac{T_1 - T_0}{T_0} Q_0}{\eta} \qquad [10]$$

Normalising, and using *COP* from [3] and η from [4] as parameters we arrive at the simple relationship:

$$\frac{P/Q_2}{\eta} + \frac{Q_0/Q_2}{COP} = 1 \qquad [11]$$

This relationship is displayed in Figure 7. It may be used as a performance chart of the cogeneration process because it gives the interdependence of taking both products from one heat source in a very simple way. As a parameter the return temperature of the driving heat source was chosen.

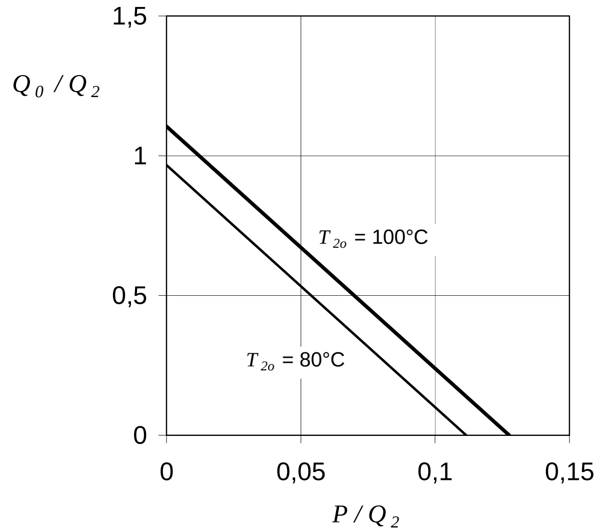

Figure 7: Performance chart: Q_0/Q_2 depending on P/Q_2 according to equation [11]. Parameter: return temperature T_{2o} of the heat source. supply temperature T_{2i} = 120°C heat sink temperature 31°C, cold production at T_0=9°C, driving temperature difference in the heat exchangers 5K, thermodynamic quality of the process g=0.7.

The coefficient of power reduction with cold production, ß', gives us the change in power output, when the output of cold is increased at given driving heat input:

$$ß' = \left(\frac{\partial P}{\partial Q_0}\right)_{Q_2} = -\frac{T_1 - T_0}{T_0} \qquad [12]$$

The change in cold output, when the output of power is increased at given driving heat input is the reverse:

$$\left(\frac{\partial Q_0}{\partial P}\right)_{Q_2} = -\frac{T_1}{T_0 - T_0} \qquad [13]$$

Cycle configurations

By looking at the technology which is available today we note that in thermal cooling as well as low-temperature power plants sorption processes play a major role. There are some consequences: the know-how in absorption cooling technology has to be spread to the power plant engineers; the absorption cooling community should look over the fence and consider power plants. Finally, and this is our goal here, taking into account that absorption refrigerators and, e.g., Kalina cycles (Leibowitz et al., 1999, Köhler, 2005) make use of the same fluids, we can design new cycles combining the production of cold and power (Figure 8).

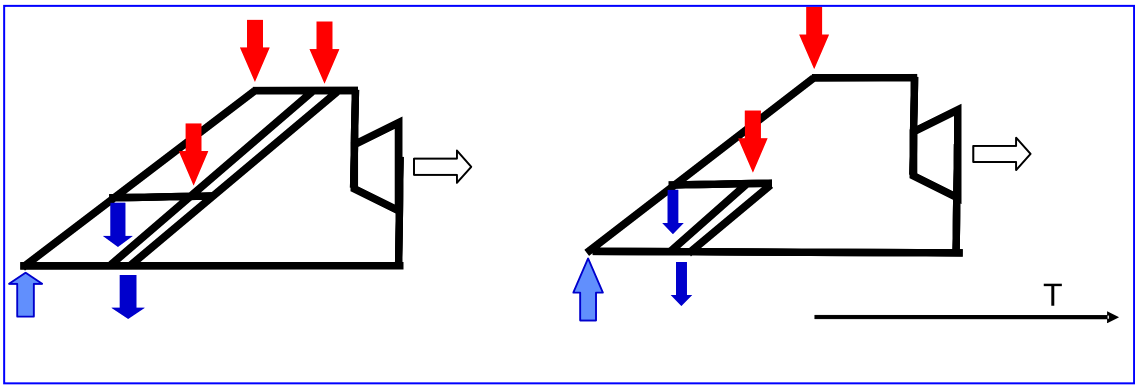

Figure 8: Graph of combined cooling-power plants in the pressure temperature plot. Heat input or output is indicated by vertical arrows;work output is indicated by horizontal arrows.

The right hand cycle can be looked at as a Kalina-cycle to which a single-effect chiller has been added. The left hand cycle is more complicated. It is a superposition of a Kalina-cycle and a double-effect absorption chiller. This cycle is especially valid if the temperature glide in the heat source is large. It needs, of course, more components and control. The right hand cycle is a quite simple one.

Another somewhat simpler configuration of the left cycle of Figure 8 is displayed in Figure 9 in the form of a flow scheme.

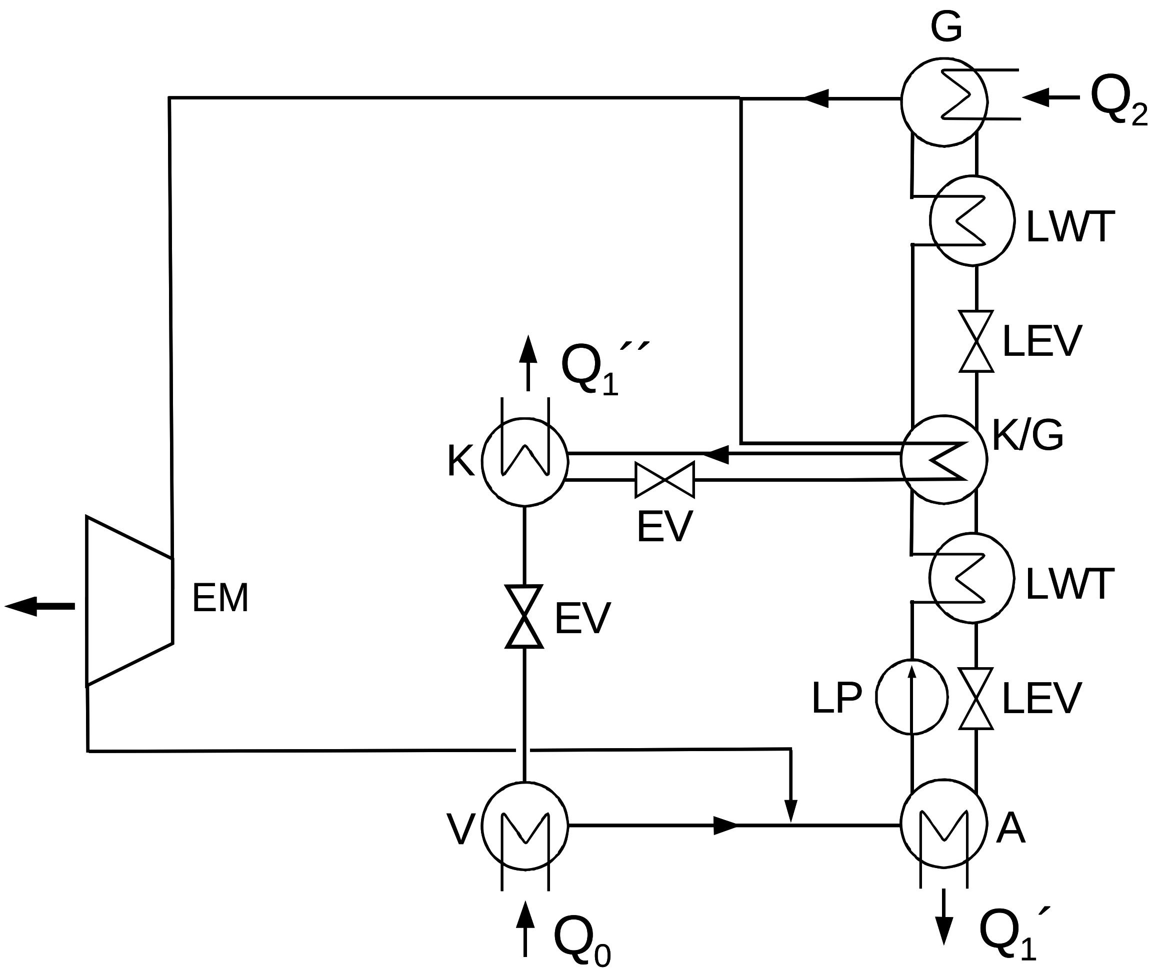

Figure 9: Flow scheme of a double-effect absorption cooling-power plant, producing cold Q_0 and power P from waste heat Q_2 A: Absorber; EM: Exp. machine; EV: Exp. valve; G: Generator; K: Condenser; LEV: Solution valve; LWT: Solution hx; V: Evaporator

This cycle, again, features a double-effect absorption chiller in which a turbine has been inserted in parallel to the throttling devices, i.e. between high and low pressure, or Generator G and Absorber A, respectively. The cycle is almost the same as in Figure 8 left, with only the heat exchange being different. However, the amount of power which can be produced is less in this cycle configuration. The performance chart of the cycle – analogously to Figure 7 – is plotted in Figure 10. For a heat input of 1 MW the production of Cold can be almost 1.2 MW (pure chiller), whereas the power will be less than 100 kW (pure sorption power cycle). Most important is the freedom to adjust the ratio between cold production and power production.

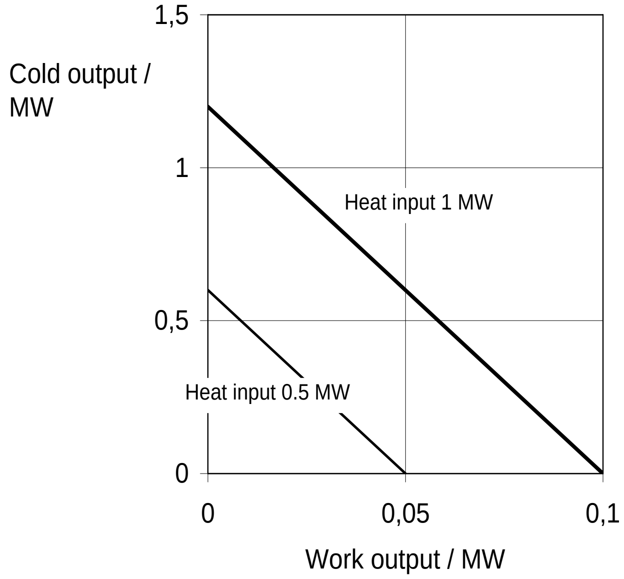

Figure 10: Performance chart of the cycle from Figure 9

CONCLUSION

We have discussed the separate and combined production of cold and mechanical power from low-temperature heat sources. Before these combined cycles become mature technologies many technological obstacles mainly originating from the expanders have to be overcome. Research in this field has just started.

We should note, moreover, that all low-grade heat technologies – chillers and heat transformers as well as some power cycles (sorption power cycles, Kalina) – may make use of sorption technology. So, disregarding which kind of energy service we want to procure, sorption cycles will play a major role. This should be enough reason for enhancing basic research in this field. As the largest fraction of cost is born by the heat exchangers, especially the area of heat transfer has to be promoted.

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