

JAN ALBERS, ANNETT KÜHN, STEFAN PETERSEN, FELIX ZIEGLER*

**CONTROL OF ABSORPTION CHILLERS BY INSIGHT:
THE CHARACTERISTIC EQUATION**

**STEROWANIE WEWNĘTRZNE
CHŁODNI ABSORPCYJNYCH:
RÓWNANIE CHARAKTERYSTYCZNE****Abstract**

Usually the control of absorption chillers is done via the temperature of the driving heat. It is set in dependence of the chilled water temperature. The temperature of the cooling water is almost constant and not being used for control. This method exhibits two drawbacks: there is optimization potential with regard to water consumption and fan power which is not addressed, and in solar cooling applications the temperature of the driving heat is not really free to be used for control.

We will present a basic relation between the governing temperatures which can be used to control absorption chillers – especially with solar drive – and which may even be used to optimize water consumption and auxiliary power demand. Moreover, it allows for a diagnosis of deterioration or faulty behavior of the chiller.

After derivation of the so-called characteristic equation experimental examples will be given.

Keywords: absorption chiller, solar cooling, heat transfer, control

Streszczenie

Sterowanie chłodni absorpcyjnych zwykle odbywa się poprzez kontrolę doprowadzanego ciepła, w zależności od temperatury wody chłodzącej. Temperatura wody chłodzącej zwykle jest niezmienna i nieprzydatna do sterowania. Zaprezentowano podstawowe zależności pomiędzy temperaturami sterowania użytecznymi do regulacji chłodni, szczególnie solarnych. Zależności mogą być wykorzystane do optymalizacji zużycia wody i zapotrzebowania na energię zewnętrzną. Co więcej mogą służyć do diagnozowania defektów pracy lub awarii schładzaczy. Rozważania wyników badań pozwoliły na sformułowanie równań charakterystycznych.

Słowa kluczowe: chłodnie absorpcyjne, chłodzenie solarne, transport ciepła, sterowanie

* Dipl.-Ing. Jan Albers, Dipl.-Ing. Annett Kühn, Dipl.-Ing. Stefan Petersen,
Prof. Dr.-Ing. Felix Ziegler, Institut für Energietechnik, Technische Universität Berlin.

1. Introduction

Absorption chillers are being used worldwide for cooling with negligible consumption of electrical energy. In countries such as Japan or China they are energized mainly by firing natural gas, in order to save electricity. From the point of view of primary energy consumption it is much more favourable if the chillers are driven by waste heat, district heat, or solar heat. This is the mainstream application in Europe.

The control of absorption chillers conventionally is done by temperature control of the driving heat input. With using a mixer the temperature of the driving heat is reduced or increased so as to fit to the requirements of the chiller. In order to understand this behavior it is important to realize that an absorption chiller is a device which is totally governed by heat transfer processes. Therefore, it must be possible to derive the part-load characteristics from heat transfer characteristics. This will be exemplified in this paper. Moreover, we will show how this insight can be used for a better control strategy.

The idea was first published in Japan in the 80s [1–3]. Later it was applied to several advanced cycles [4–6] but still is not used extensively.

2. Basics

2.1. Absorption chillers

An absorption chiller consists of four main heat exchangers as shown in Fig. 1. In the evaporator E the cold is produced by evaporating a refrigerant. The vapor is absorbed into a solution in the absorber A, which has to be cooled by cooling water. The diluted weak solution is pumped to the generator G where it is regenerated by heat input. The generated vapor is re-condensed in the condenser C, giving off heat also. The concentrated strong solution flows back to the absorber whereas the condensate flows back to the evaporator. In order to increase the efficiency the solution circuit contains an internal heat recovery, the solution heat exchanger. The efficiency is defined as Coefficient of Performance

$$\text{COP} = \frac{\dot{Q}_E}{\dot{Q}_G} \quad (1)$$

The cooling capacity of a commercial water chiller using water as refrigerant and aqueous LiBr-solution as absorbent is in the range from several 10 kW up to about 10 MW. In Fig. 2 the 10 kW-chiller which is under development at TU Berlin [7] is shown together with the test facility and a cogeneration engine as heat source.

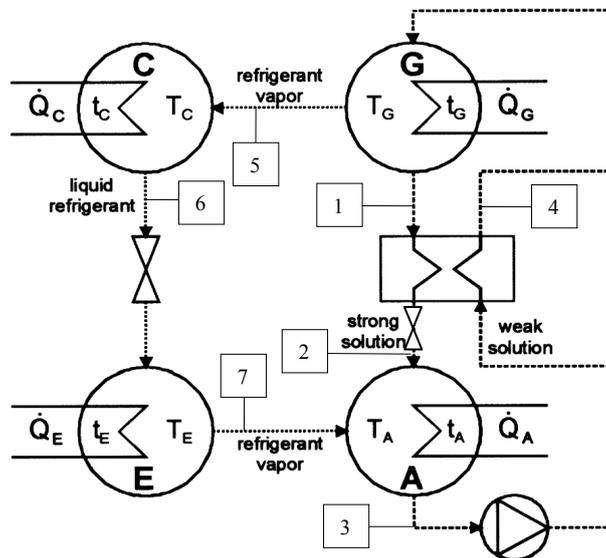


Fig. 1. Flow scheme of a single-effect absorption chiller

Rys. 1. Bilans jednostopniowej chłodni absorpcyjnej



Fig. 2. Test facility at TUB with 10 kW chiller (right) and cogeneration engine (left)

Rys. 2. Stanowisko badawcze w TUB: 10 kW chłodnia (z prawej) i silnik (z lewej)

2.2. Part-load

The part-load characteristics of the 10 kW-chiller are depicted in Figs 3 and 4. We can observe a wide range of variation (2 kW up to 20 kW) and a fairly stable COP. However, the main observation is that the capacity responds not only to the driving heat temperature but also to the temperature of chilled water and cooling water. It seems to be difficult to combine the response to all three external temperatures into one control scheme.

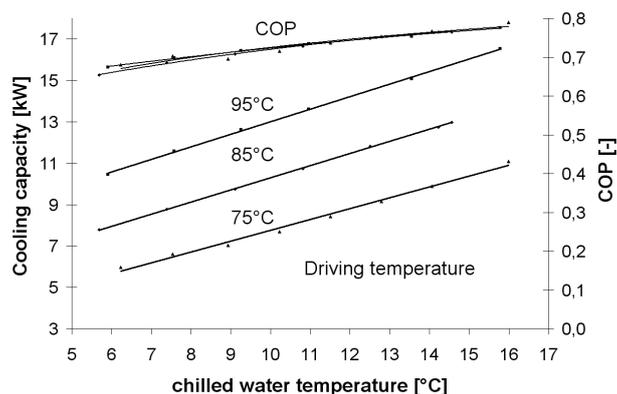


Fig. 3. Dependence of cooling capacity and COP on the temperature of chilled water and driving heat

Rys. 3. Zależność zdolności chłodzenia i COP od temperatury wody chłodzącej i doprowadzanego ciepła

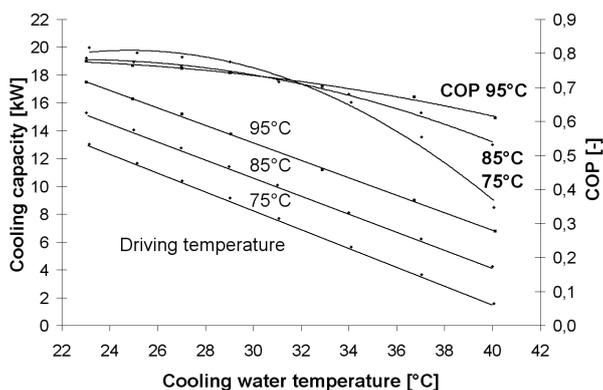


Fig. 4. Dependence of cooling capacity and COP on temperature of cooling water and driving heat

Rys. 4. Zależność zdolności chłodzenia i COP od temperatury wody chłodzącej i doprowadzanego ciepła

3. The characteristic equation

3.1. Balances

We will now derive an equation which allows to describe the system behaviour in dependence on all external temperatures very easily. For each of the four main heat apparatuses (X stands for Generator, Absorber, Condenser, Evaporator) we may establish a heat transfer equation

$$\dot{Q}_X = (UA)_X |T_X - t_X| \quad (2)$$

In this notation T_X denotes the mean internal process temperature and t_X denotes the mean external heat carrier temperature (see Fig. 1). This equation is used predominantly, although it neglects that we have a combined heat and mass transfer problem. Mass transfer resistance is lumped into the heat transfer coefficient U_X as long as no details have to be considered. Please note that the internal temperatures T_X are unknown whereas the external temperatures t_X are known because they are specified or used for control.

In addition, we can write an energy balance for each of the four components

$$\dot{Q}_E = \dot{m}_R (h_7 - h_6)$$

$$\dot{Q}_C = \dot{m}_R (h_5 - h_6)$$

$$\dot{Q}_G = \dot{m}_R h_5 + \dot{m}_{S,s} h_1 - \dot{m}_{S,w} h_4 \quad (3)$$

$$\dot{Q}_A = \dot{m}_R h_7 + \dot{m}_{S,s} h_2 - \dot{m}_{S,w} h_3$$

From a mass balance around absorber or generator we find

$$\dot{m}_{S,s} = \dot{m}_{S,w} - \dot{m}_R \quad (4)$$

With (4) we can rewrite (3) into a more convenient form

$$\dot{Q}_E = \dot{m}_R (h_7 - h_6)$$

$$\dot{Q}_C = \dot{m}_R (h_5 - h_6)$$

$$\dot{Q}_G = \dot{m}_R (h_5 - h_1) + \dot{m}_{S,w} (h_1 - h_4) \quad (5)$$

$$\dot{Q}_A = \dot{m}_R (h_7 - h_2) + \dot{m}_{S,w} (h_2 - h_3)$$

We immediately see that in all four components the heat flow is dominated by the mass flow of refrigerant and an enthalpy difference between vapour and solution which is in the order of magnitude of the latent heat of evaporation. In generator and absorber a second term is added which is due to the pumped solution flow and a relatively small enthalpy difference between the two solution streams.

3.2. Derivation of part-load characteristics

Now we want to look at variations around a nominal point of operation. As long as the solution flow and the concentrations are not changing very much the enthalpy differences will only change marginally, also. We can conclude that the whole variation in heat flow is due to the change in refrigerant flow and write for all components

$$\begin{aligned}\dot{Q}_x &= \dot{m}_R \Delta h_x + \dot{m}_{s,w} \Delta h_{sx} \\ \text{with } \Delta h_{SE} &= \Delta h_{SC} = 0\end{aligned}\quad (6)$$

We now combine (2) and (6)

$$(UA)_x |T_x - t_x| = \dot{m}_R \Delta h_x + \dot{m}_{s,w} \Delta h_{sx} \quad (7)$$

Finally, we re-substitute the cooling capacity from (5) into (7) by eliminating the refrigerant flow

$$(UA)_x |T_x - t_x| = \dot{Q}_E \frac{\Delta h_x}{h_7 - h_6} + \dot{m}_{s,w} \Delta h_{sx} \quad (8)$$

In a first order approximation all coefficients in Equation (7) are constant except for the driving temperature differences and the cooling capacity. This is the numerical expression for the fact that the chiller is totally dominated by heat transfer.

We now solve (8) for the unknown internal temperatures T_x

$$T_x = t_x \pm \left(\frac{\dot{Q}_E \frac{\Delta h_x}{h_7 - h_6} + \dot{m}_{s,w} \Delta h_{sx}}{(UA)_x} \right) = t_x \pm (\dot{Q}_E K_x + K_{sx}) \quad (9)$$

We use the + for absorber and condenser and the – for evaporator and generator. K_x and K_{sx} are well enough constants. K_{SE} and K_{SC} are zero.

In a single stage absorption chiller cycle the internal temperatures of the four heat exchangers can be combined by using Dühring's rule for the solution field of aqueous lithium bromide

$$(T_G - T_A) - (T_C - T_E)B = 0 \quad (10)$$

The parameter B accounts for the different slopes of the boiling lines in the Dühring plot. The value is under normal operational conditions in a range of 1,1 to 1,2 and does not change strongly. Substituting the internal temperatures from (9) into (10) we find

$$(t_G - t_A) - (t_C - t_E)B = (\dot{Q}_E K_G + K_{SG}) + (\dot{Q}_E K_A + K_{SA}) + (\dot{Q}_E K_C + \dot{Q}_E K_E)B \quad (11)$$

We use the definition of a characteristic double temperature difference

$$\Delta \Delta t = (t_G - t_A) - (t_C - t_E)B \quad (12)$$

and find from (11) a linear relationship between the cooling capacity and the characteristic temperature function

$$\Delta\Delta t = \dot{Q}_E (K_G + K_A + (K_C + K_E)B) + (K_{SG} + K_{SD}) \quad (13)$$

This relationship is depicted in Fig. 5 for the data which are shown in Figs 3 and 4. We see that the data can be correlated quite well with Eq. (13) despite the amount of assumptions and the wide range of load variations.

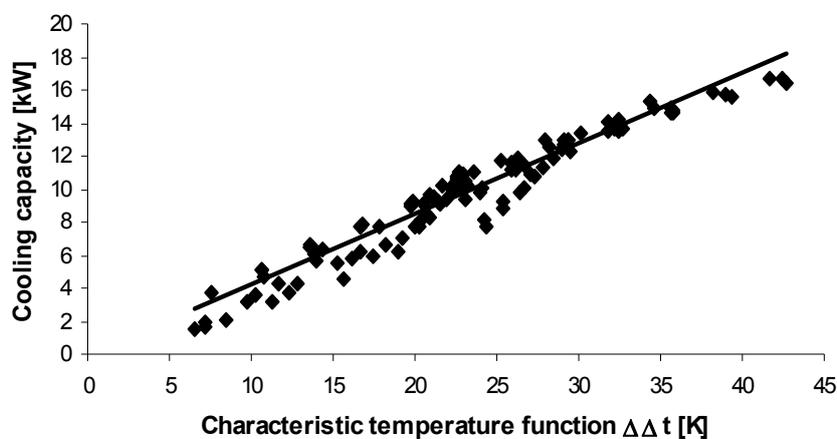


Fig. 5. Data from Figs 3 and 4 in the characteristic plot

Rys. 5. Wyniki z rysunków 3 i 4 w zależności od charakterystycznej funkcji temperatury

3.3. Adjustment

Of course, the method can be refined. After all, this is necessary because the assumption that all the relevant parameters are constant is not really valid. However, we still keep the information about a joint effect of the three external temperatures (absorber and condenser are combined together) and find

$$\dot{Q}_E = 0,42 \cdot t_G - 1,05 \cdot t_{AC} + 0,75 \cdot t_E + 1,3 \quad (14)$$

We can define a new double temperature difference

$$\Delta\Delta t' = t_G - 2,5 \cdot t_{AC} + 1,8 \cdot t_E \quad (15)$$

Then we find for the cooling capacity, as shown in Fig. 6

$$\dot{Q}_E = 0,42 \cdot \Delta\Delta t' + 0,9 \quad (16)$$

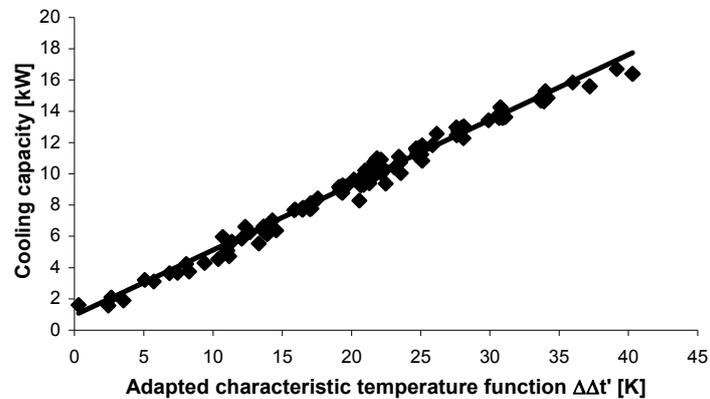


Fig. 6. Characteristic plot with adjusted parameters

Rys. 6. Wyrównane parametry wykresu charakterystycznego

A more physical approach to the adjustment is ongoing work.

3.4. Application for control and diagnosis

The characteristic temperature difference has an important meaning: the cooling capacity is affected only when the characteristic temperature difference changes. Consequently, a change in one external temperature can always be counteracted by a change in one of the other temperatures, and the characteristic function gives the rationale to do the right change. This gives rise to a very simple but versatile control algorithm.

The information which is coded in the characteristic temperature function allows for replacing a change in driving temperature against a change in cooling water temperature. This gives rise to the possibility to optimise the electricity consumption of the cooling tower fans and pumps against the temperature of the driving heat which will in turn influence the collector efficiency. Also the consumption of water may be incorporated. Moreover, a deviation of the characteristic behavior allows to detecting malfunctions in the system, such as fouling at the heat exchangers or bad vacuum. We will show a very special case in the following.

In European solar cooling installations the Yazaki WFC-10 chiller (35 kW nominal capacity) has been implemented about 50 times. This chiller has a quite compact, highly individual design incorporating a generator which at the same time serves as thermosyphon solution pump, and an evaporator without sump. Therefore its part-load behavior under conventional control is not very satisfactory as depicted in Fig. 7 [8].

The solid line shows the characteristic curve as it was to be expected from basic considerations. The real behavior deviates dramatically due to the evaporator running partly dry or overflowing, depending on the generator. From the characteristic plot, we can find the operational conditions for which the chiller hits the characteristic curve: this gives the

optimal points of operation. It necessitates the combined control of generator and cooling water, which is not done normally.

The new control algorithm has been implemented recently in two installations. Preliminary results show a considerable increase in efficiency, as was to be expected. Quantitative results will be presented in further papers.

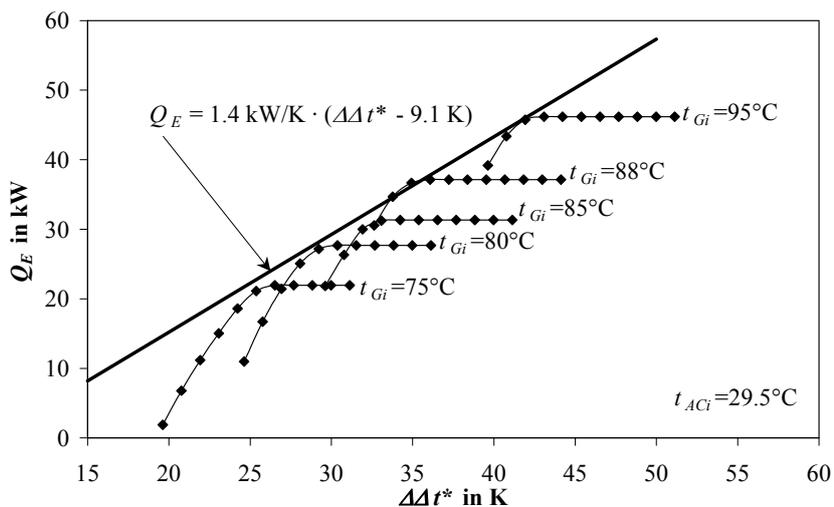


Fig. 7. Characteristic equation and manufactures data for a H₂O/LiBr-absorption chiller with thermally driven solution pump

Rys. 7. Równanie charakterystyczne i wyrównane wyniki dla chłodni absorpcyjnej H₂O/LiBr z termicznie sterowaną pompą

4. Conclusion

The heat transfer mechanisms in absorption chillers dominate their behavior. This gives rise to the possibility to reduce the complex response to all external heat carrier temperatures to a linear relationship between heat flow and a linear function of these external temperatures: the characteristic temperature function. The theoretical background has been presented and experimental data have been discussed. Finally, it has been outlined that the method of the characteristic temperatures can be used as a tool for improving the part-load features of a commercial chiller.

Symbols

A	– heat exchange surface	[m ²]
B	– Düring-coefficient	
COP	– coefficient of performance	

h	– specific enthalpy	[kJ/kg K]
Δh	– difference of specific enthalpy	[kJ/kg K]
K	– constant	
\dot{m}	– mass flow rate	[kg/s]
\dot{Q}	– heat flow rate	[kW]
t	– external temperature	[°C]
T	– internal temperature	[°C]
$\Delta\Delta t$	– characteristic temperature function	[°C]
$\Delta\Delta t'$	– adjusted characteristic temperature function	[°C]
$\Delta\Delta t^*$	– adjusted characteristic temperature function	[°C]

Indices

1 to 7	– state points	i	– inlet
A	– absorber	X	– component
C	– condenser	R	– refrigerant
E	– evaporator	S	– solution
G	– generator	W	– weak solution

References

- [1] Takada S.: *Absorption Freezer*, Japan Freezer Association, Japanese, 1982.
- [2] Furukawa T. et al.: *Study on Characteristic Temperatures of Absorption Heat Pumps*, Proc. 20th Japan Heat Transfer Conf., June 1983, 508-510, Japanese.
- [3] Furukawa T., Sonoda T.: *Characteristics of H₂O/LiBr Absorption Heat Pumps for the Temperature Change of External Fluids*, XVII Internationaler Kongress für Kältetechnik, Wien, Austria, 24–29.08.1987.
- [4] Riesch P., Scharfe J., Ziegler F., Völkl J., Alefeld G.: *Part-Load Behavior of an Absorption Heat Transformer*, Proceedings of the 3rd Int. Symposium on the Large Scale Applications of Heat Pumps, Oxford, England, 155-160, BHRA, The Fluid Engineering Centre, Cranfield, Bedford, 1987.
- [5] Hellmann H.-M., Schweigler C., Ziegler F.: *A simple method for modeling the operating characteristics of absorption chillers*, Proc. of Eurotherm Seminar No. 59, Nancy-Ville, France, 6–7 July 1987, 219-226.
- [6] Schweigler C., Ziegler F.: *Single-effect/double-lift chiller: operational experience and prospect*, Proc. of the 20th International Congress of Refrigeration, Sydney, 19–24 September 1999.
- [7] Kühn A., Petersen S., Ziegler F., Kohlenbach P., Harm M., Schweigler C.: *Operational results of a 10 kW absorption chiller for low-grade driving heat*, International Sorption Heat Pump Conference, Denver, USA, 2005.
- [8] Albers J., Ziegler F., Asdrubali F.: *Investigation into the influence of the cooling water temperature on the operating conditions of thermosyphon generators*, International Sorption Heat Pump Conference, Denver, USA, 2005.