

# Computer Simulation of a Lithium Bromide-Water Absorption Heat Pump for Temperature Boosting

G. Grossman      K.W. Childs  
ASHRAE Member

## ABSTRACT

A computer simulation model has been developed to predict the performance of an absorption heat pump for temperature boosting of low-grade heat. The model simulated a single-stage, lithium bromide-water system currently being constructed. The effects of waste heat temperature, cooling water temperature, and solution circulation rate were investigated. The temperature boost and delivered capacity increased almost linearly with an increase in the waste heat temperature or a decrease in the cooling water temperature. The system's coefficient of performance (COP) increases slightly under either of these conditions.

## INTRODUCTION

Absorption heat pumps for temperature boosting, also known as heat transformers, have attracted growing interest in recent years due to their potential for waste heat utilization.<sup>1</sup> These systems, operate in a cycle that is the reverse of the conventional absorption chillers. This makes it possible to boost the temperature of low-grade heat typically by 36 to 72°F (20 to 40°C), thereby rendering it useful for industrial and other applications. The absorption heat transformer has an advantage over other types of heat pumps, e.g., vapor compression, by using the waste heat itself as a source of power and not rely on primary energy, except for small auxiliary equipment.

An earlier paper described the conceptual design of an absorption heat pump for temperature boosting, which is the first step toward the construction of a prototype working system.<sup>2</sup> The potential of the absorption heat transformer was explored for recovering low temperature waste heat (140°F or 60°C).<sup>3</sup> An investigation revealed several earlier projects in which absorption heat transformers were built and experimentally investigated.<sup>4-6</sup> Different design alternatives were studied, including staging schemes and various working materials. The performance of individual components, as well as the overall system, was calculated for different operating conditions using a computer model. This model assumed constant heat exchange effectiveness in all the subunits of the system and the attainment of thermodynamic equilibrium at the end of the absorption and desorption processes. These calculations yielded the COP and temperature boost for each set of conditions. The COP is limited to 0.5 and generally comes close to this value with reasonable heat exchange effectivenesses and circulation ratios. The temperature boost in a single-stage system was found to be of the order of the difference between the heat source and heat sink temperatures. A related study investigated the possibility of open cycle desorption, using ambient air instead of a condenser as a sink for the vapor in the reconcentration of the weak absorbent. This system is called for in cases in which the heat source and heat sink temperatures are close to each other. Still another study investigated the potential for improvement in the heat transformer cycle by adding adiabatic absorption and desorption steps to the absorber and desorber of the system, respectively. These adiabatic processes make it possible to obtain an increased temperature boost at a better COP than in the conventional cycle.

Gershon Grossman, Staff Scientist, Energy Division, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

Kenneth W. Childs, Engineer, Computer Sciences Division, Union Carbide Corporation, Oak Ridge, Tennessee.

Research sponsored by the Office of Coal Utilization, U.S. Department of Energy, under contract W-7405-eng-26 with Union Carbide Corporation.

The results obtained in the above studies paved the way for the design and construction of a laboratory prototype system, beginning in February 1982. The system, built by an industrial manufacturer of absorption equipment to ORNL's specifications, is single-stage and uses lithium bromide-water as the working fluid. It is designed to deliver 150,000 Btu/hr (44 kW) of temperature-boosted heat at an input low-grade heat temperature of 140°F (60°C) and cooling water temperature of 59°F (15°C). The system includes the innovative feature of an adiabatic process in the absorber and has two small mechanical pumps for the absorbent solution and the refrigerant.

This study describes a computer simulation of the absorption heat pump, performed in conjunction with the construction of the prototype unit. The objective of the study was to predict the performance of the working system. The results will be compared with the experimental data when it is available. This comparison will help point out phenomena associated with the operation of the machine which were not taken into consideration in the design, as well as shortcomings of the assumptions made in the model and the design. Such computer simulations have proven in the past to be a valuable tool for design optimization once validated by experimental data.<sup>9,10</sup> The present computer simulation is somewhat different from the one performed earlier in the conceptual design. Here, the constraints of a built system, such as fixed heat transfer areas, have to be taken into account.

### THE COMPUTER SIMULATION MODEL

The absorption heat pump analyzed here is described in Fig. 1 in a schematic form. In Fig. 1 solid lines indicate the flow is in the liquid phase and dashed lines indicate vapor. As mentioned earlier, the system is single-stage and operates using lithium bromide-water. The major subunits of the system are the evaporator, absorber, desorber, condenser, and recuperative heat exchanger. The adiabatic portions of the absorber and desorber are represented by the cross-hatched boxes. Numbers in the diagram indicate state points of the working fluids. A detailed description of the operation of the heat pump has been given in Refs 2, 3, and 8.

The computer simulation model is based on heat and mass balances heat transfer equations, and thermodynamic property relations written in terms of the state point properties for each of the system's subunits and matched at the connecting points. The following equations (see the notation in Fig. 1) control the behavior of the absorber:

mass conservation of salt:

$$m_7 C_H = m_4 C_L \quad (1)$$

overall mass balance:

$$m_4 = m_7 + m_8 \quad (2)$$

energy balance, solution side:

$$Q = m_7 h_7 + m_8 h_8 - m_4 h_4 \quad (3)$$

energy balance, water side:

$$Q = m_3 c_p (T_9 - T_3) \quad (4)$$

heat transfer:

$$Q = UA \frac{[(T_{7e} - T_9) - (T_4 - T_3)]}{\ln \frac{T_{7e} - T_9}{T_4 - T_3}} \quad (5)$$

thermodynamic equilibrium:

$$T_4 = f(P, C_L) \quad (6)$$

where

- $C_L$  = solution low concentrations
- $C_H$  = solution high concentration
- $Q$  = heat released from the absorber
- $m$  = mass flow rate at a given state point
- $T$  = temperature at a given state point
- $h$  = enthalpy at a given state point
- $P$  = pressure in the absorber.

The temperature  $T_{2e}$  is determined from a heat and mass balance for the adiabatic absorber. The thermodynamic equilibrium relation, Eq 6, relates the temperature of the solution leaving the absorber to its concentration and to the pressure. The pressure is the same as the vapor pressure of the refrigerant in the evaporator. Equilibrium data for lithium bromide-water was obtained from Ref 11.

A set of equations similar to the above may be written for the desorber. Somewhat simpler sets, not including the mass balances or thermodynamic equilibrium, pertain to the other components of the system. A method incorporating a simultaneous solution of the entire set of nonlinear equations was developed. The method makes use of a subroutine, HYBRDI, developed by J. J. Moré, et al., to solve the system of equations.

The input parameters to the program include (1) waste heat temperature and flow rate in the evaporator, absorber, and desorber, (2) cooling water temperature and flow rate in the condenser, and (3) solution circulation rate as given by the characteristic of the solution pump. Also given are the design parameters of the machine, including heat transfer areas and coefficients in the different subunits. Most of the input parameters are measurable and controllable quantities set by the designer or the operator. The only exception concerns the heat transfer coefficients. In another absorption simulation program written for a chiller, some of these coefficients were calculated. Others were assumed, based on practical experience. Insufficient information existed on calculating absorption heat transfer coefficients and the dependence of the coefficients on the particular geometry and heat transfer surface. It was thus decided in the present model to input the overall heat transfer coefficient at each of the subunits. Estimated values were used in the calculations presented in this paper. They will be replaced by measured values once the machine is tested.

The COP, temperature boost, and other operating parameters are calculated from the temperatures and heat transfer rates. The COP is determined as the ratio of useful heat supplied by the absorber to the waste heat supplied to the evaporator and desorber. The energy use by the pumps is quite small, and is not considered in the calculation of the COP.

## RESULTS AND DISCUSSION

The computer model was used to simulate the system's performance for a variety of operating conditions. As a reference point for evaluation of the effect of different parameters, a nominal condition was selected which corresponds to the design point for the prototype heat pump. The temperatures of the waste heat and cooling water at the design point are 140°F (60°C) and 59°F (15°C), respectively; and the design boosted-temperature heat output is 150,000 Btu/hr (44 kW). The nominal condition is described in Tab. 1, which is a transcribed version of a typical computer output. The table lists first the following input parameters:

1. the input waste heat and cooling water temperatures
2. the product of overall heat transfer coefficient and heat transfer area (UA) for each of the system's subunits
3. the mass flow rate of water in the evaporator, absorber, desorber, and condenser, and of the strong solution leaving the solution pump
4. an assumed deviation from thermodynamic equilibrium, expressed in degrees, at the outlet from the adiabatic absorber and desorber

Next, the following calculated quantities are shown:

1. the temperature, mass flow rate, and concentration at all the state points corresponding to Fig. 1
2. the heat quantities and heat exchange effectivenesses in the evaporator, absorber, desorber, condenser, and recuperative heat exchanger
3. the temperature boost
4. the coefficient of performance

Similar calculations were made for the other selected sets of operating conditions.

Figures 2 and 3 describe the change in temperature boost and COP, respectively, brought about by changes in the waste heat and cooling water input temperatures. The UAs and flows are maintained constant at the values given in Tab. 1; the deviation from thermodynamic equilibrium is assumed to be zero. The conditions of the design point are indicated on the figures. The machine covers a range of combinations of waste heat and cooling water temperatures deviating from nominal conditions. As expected, the temperature boost and COP increase with the waste heat input temperature and decrease with the cooling water temperature. The increase in temperature boost is almost linear with the waste heat temperature; the change in COP is very small.

Figure 4 shows the changes in high and low concentrations with the input waste heat temperature for three typical cooling water temperatures. The solid lines describe the high concentration and the broken lines are the low concentration. Observe that for a higher waste heat temperature or lower cooling water temperature, both concentrations in the system and the difference between them are larger. The former effect is related to the more efficient desorption associated with a higher source and lower sink temperature. The latter is a result of the higher temperature boost obtained under these conditions. The capacity of the machine is proportional to the temperature boost with a fixed flow rate in the absorber. Therefore, an increased amount of refrigerant must be circulated when the temperature boost is higher. Since the solution flow rate induced by the pump is essentially fixed, this increase is brought about by a larger concentration gap.

Figure 5 shows the effect of a deviation from thermodynamic equilibrium at the end of the adiabatic absorption and desorption steps.<sup>19</sup> Since there is some uncertainty about the ability to achieve equilibrium at these points, the possibility has been considered in the model of some residual subcooling at point 7e and superheating at 5e. Figure 5 shows that deviations of 1.8 and 3.6°F (1 and 2°C) from equilibrium at both points cause a reduction in the temperature boost of about 1.3 and 2.5°F (0.7 and 1.4°C), respectively. There is also a minor reduction in COP, of the order of 0.001.

Figure 6 describes the effect of a change in solution flow rate on performance. The COP, temperature boost, and high and low concentrations are plotted as functions of the solution flow rate, which is normalized with respect to its nominal value. The waste heat and cooling water temperatures are maintained fixed at the design point values 140°F (60°C) and 59°F (15°C), respectively. The UAs and water flow rates in the evaporator, absorber, desorber, and condenser are assumed to remain constant; the heat transfer coefficient in the recuperative heat exchanger varies with the flow to the power 0.8, typical for turbulent flow. When the solution flow rate increases, the COP decreases, as expected, due to higher circulation losses and temperature boost decreases. At the same time, the difference between the high and low concentrations decreases, as shown in Fig. 6(b). Conversely, when the solution flow rate is reduced, the difference in concentrations increases, circulation losses are reduced, and a higher COP is obtained. The temperature boost increases as well, reaches a maximum, and then drops as the flow rate is reduced further. Reducing the solution flow rate, within reasonable limits, increases the number of transfer units (NTU) in the absorber and desorber, and to a lesser extent in the recuperator. This results in a higher heat exchange effectiveness in these units. However, reducing the flow may cause difficulties in properly wetting the heat transfer surface. In the present system there is some margin for reducing the flow of solution to obtaining improved performance.

The influence of variations in the UAs of the different subunits was not investigated in detail in this study. Several computer runs indicate the expected trend of enhanced performance with an increase in the heat transfer areas and coefficients. However, a meaningful study of the effect of these variables must be in the form of a design optimization. This depends upon a knowledge of the actual transfer coefficients and the relative cost of the heat transfer surfaces used in the different parts of the system.

## CONCLUSION

A computer simulation model has been developed to predict the performance of an absorption heat pump under a variety of operating conditions. The model indicated that higher waste heat temperatures or lower cooling water temperatures enable the system to deliver an increased temperature boost and greater capacity, accompanied by a slight increase in COP. Under these conditions, the system's both high and low concentrations increase and so does the difference between them. The calculations also demonstrated the effects of deviation from thermodynamic equilibrium and solution flow rate.

## REFERENCES

1. M. H. Chiogloji, Industrial Energy Conservation (New York: Marcel Dekker, 1979).
2. G. Grossman and H. Perez-Blanco, "Conceptual Design and Performance Analysis of Absorption Heat Pumps for Waste Heat Utilization," ASHRAE Transactions, 88 (1982), 1: paper no. 2691.
3. H. Perez-Blanco and G. Grossman, Cycle Performance Analysis of Absorption Heat Pumps for Waste Heat Utilization, ORNL/TM-7852, Oak Ridge National Laboratory, Oak Ridge, Tenn., 1981.
4. "Heat Transformer Makes Waste Heat Efficient Source," Design News, Oct. 22, 1979.
5. G. Cohen, J. Salvat, and A. Rojey, "A New Absorption-Cycle Process for Upgrading Waste Heat," Proceedings of the 14th Intersociety Energy Conversion Engineering Conference, Vol. 2, (Boston: Aug. 5-10, 1979), pp. 1720-1724.
6. E. C. Clark and O. Morgan, "Chemical Heat Pumps for Industry," Proceedings of the 16th Intersociety Energy Conversion Engineering Conference (Atlanta: Aug. 9-14, 1981).
7. H. Perez-Blanco and G. Grossman, "Open Cycle Absorption Heat Pumps for Low Grade Heat Utilization," ASHRAE Transactions, 88 (1982), 1: paper no. HO-82-5/3.
8. G. Grossman, "Adiabatic Absorption and Desorption for Improvement of Temperature-Boosting Absorption Heat Pumps," ASHRAE Transactions, 88 (1982), 2: paper no. 2723.
9. G. C. Vliet, M. B. Lawson, and R. A. Lithgow, "Water-Lithium Bromide Double Effect Absorption Cooling Cycle Analysis," ASHRAE Transactions, 88 (1982), 1: paper no. HO-82-5/2.
10. G. Grossman et al., "Solar Powered Environment Control Criteria and Realization," Proceedings, The International Solar Energy Society Conference, 1 (Atlanta: May 1979), pp. 720-27.
11. ASHRAE Handbook, 1977 Fundamentals Volume, New York, 1977.
12. J. J. Moré, B. S. Garbow, and K. E. Hillstrom, User Guide for MINPACK-1, Argonne National Laboratory, Argonne, IL, ANL-80-74, 1980.

TABLE 1

## Nominal Condition for Heat Pump Simulation

Input parameters:

waste heat temperature: 60°C (140°F)  
 cooling water temperature: 15°C (59°F)  
 deviation from thermodynamic equilibrium: 0°C

	Heat Transfer Coefficient x Area (UA)		Mass Flow Rate	
	kw/°C	Btu/min·°F	kg/s	lb/min
Evaporator	11.65	368.3	3.04	402.1
Absorber	6.53	206.4	0.45	59.3
Desorber	6.10	192.8	0.90	118.6
Condenser	13.04	412.2	1.94	256.2
Recuperator	1.43	45.1	0.18	23.7

Calculated parameters:

temperature boost = 23.41°C (42.14°F)  
 COP = 0.486

State Points (See Fig. 1)	Temperature		Mass Flow Rate		Concentration
	°C	°F	kg/s	lb/min	% weight LiBr
1 low-grade heat input to evaporator	60.000	140.000	3.040	402.130	—
2 low-grade heat output from evaporator	56.349	133.429	3.040	402.130	—
3 low-grade heat input to absorber	60.000	140.000	0.448	59.310	—
4 weak solution output from absorber	77.208	170.975	0.198	26.173	47.99
5 weak solution input to desorber	58.073	136.532	0.198	26.173	47.99
5e solution output from adiabatic desorber	43.023	109.441	0.195	25.822	48.64
6 strong solution output from desorber	50.342	122.615	0.179	23.720	52.95
7 strong solution input to absorber	72.696	162.852	0.179	23.720	52.95
7e solution output from adiabatic absorber	85.180	185.325	0.181	23.973	52.39
8 vapor from evaporator to absorber	53.936	129.084	0.019	2.453	—

9	temperature-boosted heat output from absorber	83.411	182.141	0.448	59.310	-
20	low-grade heat input to desorber	60.000	140.000	0.897	118.620	-
21	low-grade heat output from desorber	48.295	118.931	0.897	118.620	-
22	condensate from condenser to evaporator	22.163	71.894	0.019	2.453	-
23	cooling water input to condenser	15.000	59.000	1.937	256.230	-
24	cooling water output from condenser	20.729	69.312	1.937	256.230	-
27	vapor from desorber to condenser	50.342	122.615	0.019	2.453	-

	Heat Quantity		Heat Exchange Effectiveness
	kW	Btu/min	
Evaporator	46.43	2642	0.60
Absorber	43.93	2500	0.93
Desorber	43.93	2500	0.69
Condenser	46.43	2642	0.80
Recuperator	8.52	485	0.83

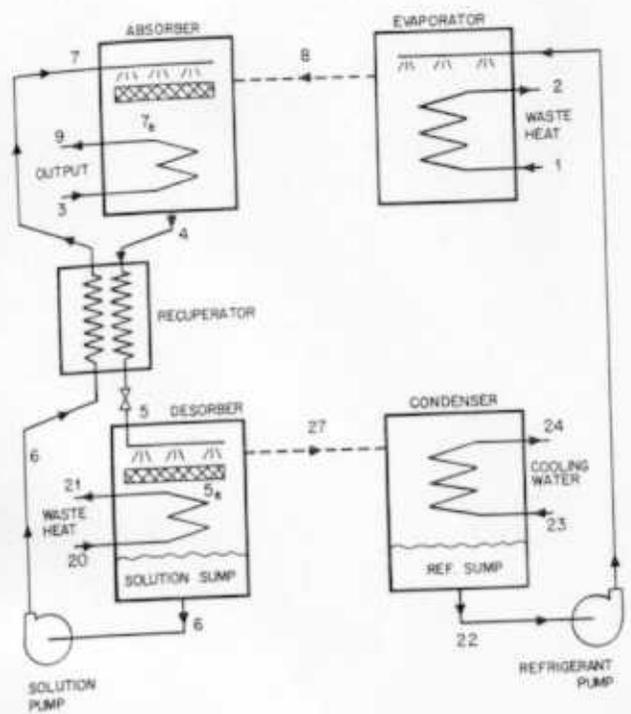


Figure 1. Schematic description of single-stage absorption heat pump

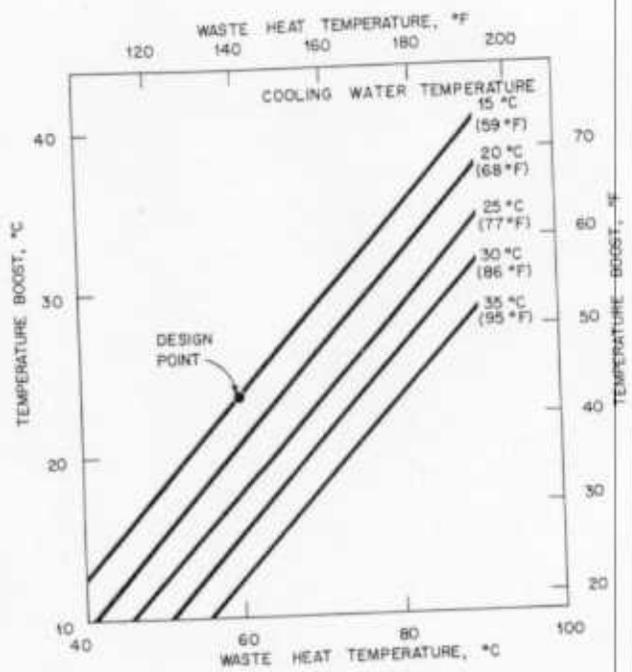


Figure 2. Temperature boost as a function of waste heat temperature for different cooling water temperatures

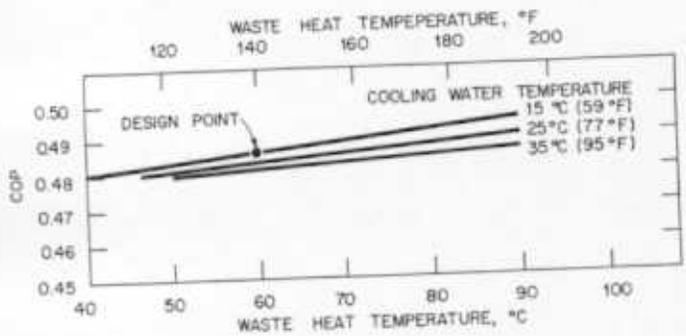


Figure 4. Solution concentrations (in % LiBr by weight) as functions of waste heat temperature for different cooling water temperatures. Solid lines describe the high concentrations; broken lines describe the low concentrations

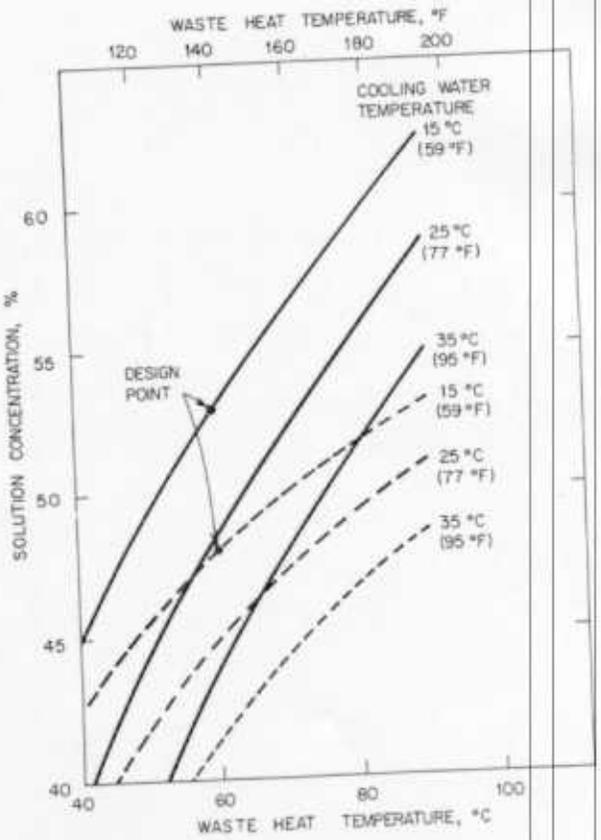


Figure 3. COP as a function of waste heat temperature for different cooling water temperatures

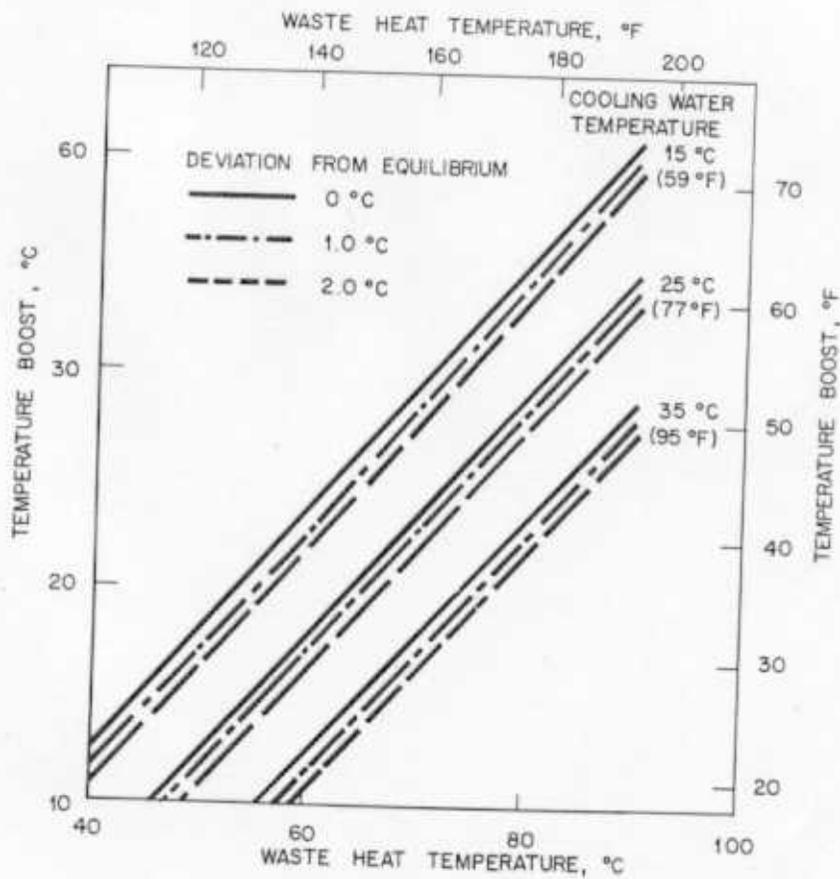


Figure 5. Effect of deviation from thermodynamic equilibrium on the temperature boost

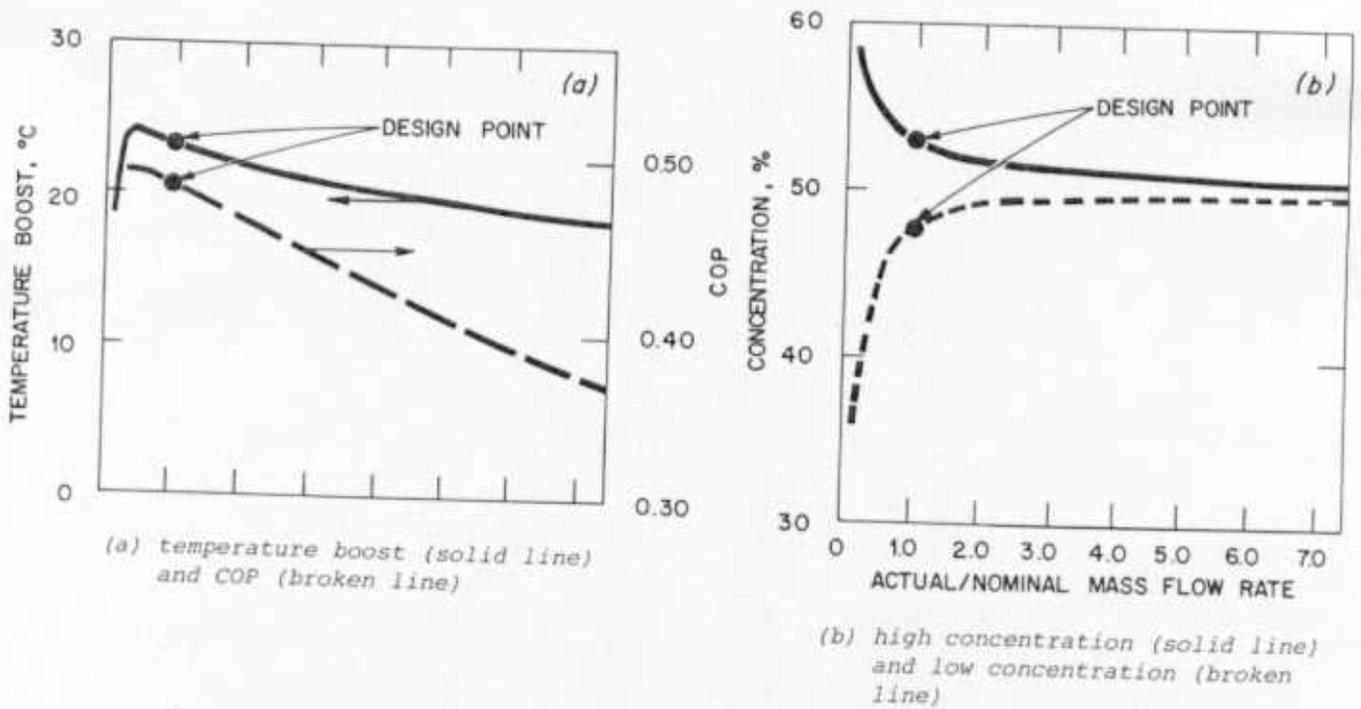


Figure 6. Effect of change in solution flow rate with fixed waste heat and cooling water temperatures