1990

Heat Recovery from Refrigeration Plants Meeting Load and Temperature Requirements

T. B. Herbas
Pontificia Universidade Catolica do Rio de Janeiro

E. A. Dalvi
Pontificia Universidade Catolica do Rio de Janeiro

J. A. R. Parise
Pontificia Universidade Catolica do Rio de Janeiro

HEAT RECOVERY FROM REFRIGERATION PLANTS MEETING LOAD AND TEMPERATURE REQUIREMENTS

T.B. Herbas, Postgraduate Student
E.A. Dalvi, Postgraduate Student
J.A.R. Parise, Associate Professor
Pontifiicia Universidade Católica do Rio de Janeiro
Department of Mechanical Engineering
22453 - Rio de Janeiro - Brazil

ABSTRACT

In refrigeration plants the heat extracted from the refrigerated medium, plus the energy in form of mechanical work from the compressor, is discharged to the atmosphere through the condenser. In the present paper a study is made on the alternatives for recovering this amount of energy to produce hot water, or warm air.

A basic thermodynamic analysis is carried out, showing the most appropriate conditions for heat recovery. A simulation model is also employed for the performance prediction of cases where temperature requirements are present. Data from an experimental unit are also presented and compared with predicted results. Tests were performed over a range of condenser and evaporator temperatures.

NOMENCLATURE

ECR energy conversion ratio, dimensionless
$\dot{Q}_c$ condenser thermal power output (kW)
$\dot{Q}_e$ evaporator thermal power input (kW)
$\dot{Q}_h$ total thermal power output (kW)
$\dot{Q}_i$ intermediate heat exchanger thermal power (kW)
$\dot{Q}_s$ additional heater power (kW)
R heating to cooling load ratio, dimensionless
T_c condensing temperature (^{0}C)
T_e evaporating temperature (^{0}C)
$\dot{W}_k$ single compressor power consumption (kW)
$\dot{W}_1$ low pressure compressor power consumption (kW)
$\dot{W}_2$ high pressure compressor power consumption (kW)

Subscripts

c condenser
e evaporator
h condenser plus supplemental heater
k compressor
s supplemental heater
1 low pressure compressor
2 high pressure compressor

1. INTRODUCTION

The simultaneous requirement for both cooling and heating in many industrial processes has led to the attempt of recovering the condenser heat output in refrigeration plants. A considerable reduction in the overall energy consumption is expected as the condenser output will contribute, in part or totally, to the heat load. Besides, the plant initial investment may be reduced by the supression of the cooling tower.

Several examples of refrigeration heat recovery can be found in the literature. They range from heat recovery in air-conditioning plants [1] and sports centres (ice-rink / swimming pool)

[2] to industrial applications such as milk coolers [3], plastic injection moulders [4], breweries [5], and others. Generally, from the papers investigated, one concludes that waste heat can be recovered advantageously from refrigeration plants. More recently, Ghosh et al [6], studying the economics of heat pump systems for simultaneous heating and cooling, have confirmed that. As for multi-stage cascaded refrigeration-heat pump systems, Gupta [7] presents a paper where such a system is numerically optimized.

There are two basic ways heat can be recovered from refrigeration units: by direct recovery or by heat pump recovery. With the direct heat recovery scheme, illustrated in Figure 1, heat is obtained directly from the condenser. To perfectly match the temperature and power load requirements, in both cooling and heating sides, an additional heating source has to be included. It can be a gas-fired or electrical heater, or even a boiler, in which case the condenser acts as a feed-water pre-heater. Water temperature at the condenser outlet is controlled by adjusting the condensing temperature. Higher temperatures result, of course, in lower cycle efficiencies. One common practice is to install a heat exchanger between the compressor and the condenser to further increase the temperature of the water leaving the condenser. One such example is described in [8]. With the heat exchanger acting as the vapour de-superheater, higher water temperatures can be attained just by taking advantage of the high enthalpy of the compressor discharge gas.

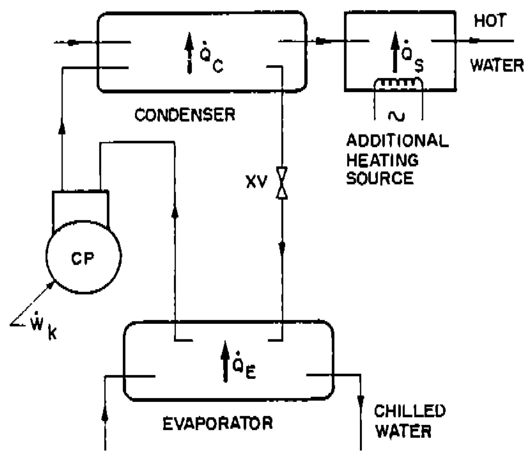

Figure 1 - Direct heat recovery scheme.

The second method, shown diagramatically in Figure 2, is the heat pump recovery scheme, where a heat pump utilizes the refrigeration condenser output as its heat source. The result is a "cascade" two-vapour compression system. The two cycles are interconnected by a refrigerant-to-refrigerant heat exchanger, as in Figure 2, or by separate low pressure condenser and high pressure evaporator, with an intermediate fluid (usually water). The supplemental heater is also included in Figure 2.

The present paper gives an insight on the thermodynamics of refrigeration plant heat recovery systems. The concept of an energy conversion ratio, ECR, the heating load divided by the total energy consumption, is introduced to measure the efficiency of the conversion of electrical energy into heat. In the first part of the paper a simple energy analysis of both recovery schemes is carried out, showing that ECR is strongly dependent on the proportion between the existing cooling and heating load requirements. In section 3 experimental results from an existing refrigeration unit, with direct heat recovery, are shown. Finally, in section 4, a study is made taking into account temperature requirements at both evaporator and condenser/heater outlets. For this study a numerical simulation model was employed.

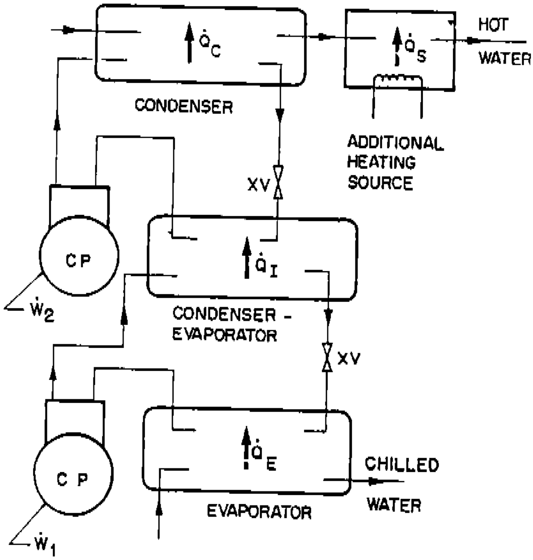

Figura 2 - Heat pump cascade recovery scheme.

2. IDEAL CYCLE ANALYSIS

To give a first assessment of the plant performance, a simplified energy analysis is made for both recovery schemes. For that, the following assumptions are made:

i) vapour compression is carried out isentropically;

ii) there is neither heat loss nor pressure drop across any of the heat exchangers;

iii) the heating process, in the additional heater, is one hundred percent efficient;

iv) the heating load is always greater than the evaporator power input.

Two parameters, ECR and R, are defined. ECR is the total power output, $\dot{Q}_h$, divided by the total power input, as consumed by the compressor and heater.

$$ECR = \frac{\dot{Q}_h}{\dot{W}_k + \dot{Q}_s} \tag{1}$$

Also important is the heating to cooling load ratio, R, defined as,

$$R = \frac{\dot{Q}_h}{\dot{Q}_e} \tag{2}$$

2.1 Direct Recovery

Referring to Figure 1, the total power output, $\dot{Q}_h$, is given by,

$$\dot{Q}_h = \dot{Q}_s + \dot{Q}_c \tag{3}$$

Also,an overall energy balance over the system gives,

$$\dot{W}_k + \dot{Q}_e = \dot{Q}_c \tag{4}$$

Substituting equations (3) and (4) into equation (1),

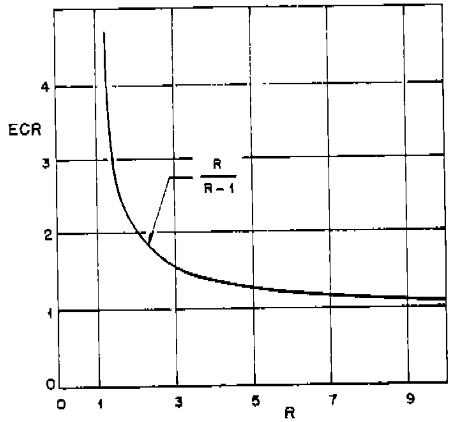

Figure 3 - Ideal energy conversion ratio as a function of the heating to cooling load ratio.

$$ECR = \frac{\dot{Q}_h}{\dot{Q}_h - \dot{Q}_e} \tag{5}$$

Rewritten in terms of R, equation (5) becomes,

$$ECR = \frac{R}{R-1} \tag{6}$$

Equation (6) shows that the ideal energy conversion ratio depends on the heating to cooling ratio solely. This implies that, ideally, there would be no difference in making the condenser operate at a higher temperature, so as to reduce the additional heat source load. In a real system the influence of other parameters may prevail, as demonstrated later in this paper.

2.2 Heat Pump Recovery

For this case there is the work input from both compressors plus the energy consumed in the additional heater. Thus,

$$ECR = \frac{\dot{Q}_c + \dot{Q}_s}{\dot{Q}_s + \dot{W}_1 + \dot{W}_2} \tag{7}$$

An energy balance over each of the refrigerant circuits, Figure 2, gives.

$$\dot{W}_1 = \dot{Q}_i - \dot{Q}_e \tag{8}$$

$$\dot{W}_2 = \dot{Q}_c - \dot{Q}_i \tag{9}$$

Substituting equations (8) and (9) in equation (7).

$$ECR = \frac{\dot{Q}_c + \dot{Q}_s}{\dot{Q}_s + \dot{Q}_c - \dot{Q}_e} \tag{10}$$

From equation (2),

$$\dot{Q}_h = \dot{Q}_s + \dot{Q}_c = R\,\dot{Q}_e \tag{11}$$

With the substitution of equation (11) into (10), equation (6) is again obtained,

$$ECR = \frac{R}{R-1} \tag{6}$$

2.3 Comments on the Ideal ECR

It results that equation (6), plotted in Figure 3, applies for both direct and heat pump recovery systems. It is seen that ECR tends to unity as R increases. This means that the greatest benefit from refrigeration heat recovery (higher ECR's) is obtained when cooling and heating loads are in same order of magnitude. On the other hand, large payback periods (i.e., low ECR's) are expected when $\dot{Q}_h$ is much greater than $\dot{Q}_e$. This was expected as, for example, the contribution from a small size refrigeration unit to pre-heat feed-water to a large capacity boiler would be negligible, making any investment on heat recovery not attractive. In fact, for the majority of cases described in the literature, R values did not exceed 3.

3. TESTING OF A DIRECT RECOVERY SYSTEM

Experimental results from an existing test rig [9] were employed to analyse, at various condensing and evaporating pressures, the performance of a water chiller unit with direct heat recovery. The plant lay-out is shown in Figure 4. The condenser was of the shell-and-coil type and chilled water was obtained by means of a series arrangement of three evaporators, each with its own thermostatic expansion valve. The electrical heater, placed downstream the condenser, is not indicated in Figure 4.

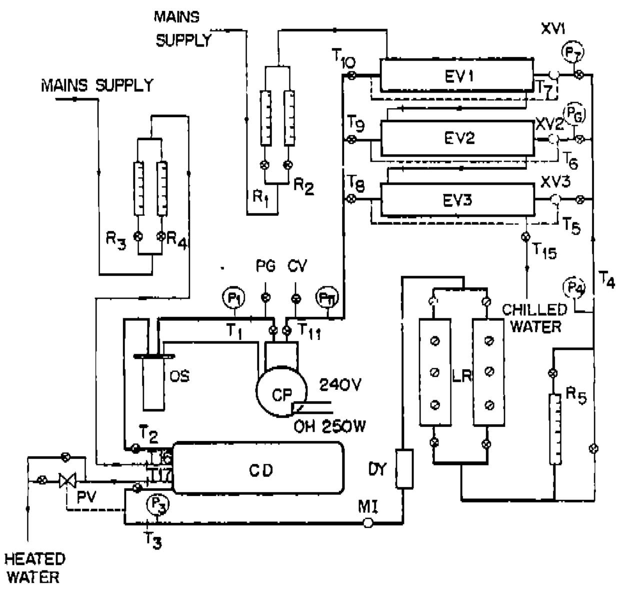

Figure 4 - Lay-out of the water chiller unit. Legend:
CD - condenser; CP - compressor; CV - charging valve; DY - drier;
EV1 - top evaporator; EV2 - middle evaporator; EV3 - botton evaporator;
LR - liquid receivers; M1 - moisturer indicator; OH - oil reater;
OS - oil separator; PG - purge valve;
PV - pressure regulating valve; R - rotameter;
XV - expansion valve.

Data from a total number of 26 runs have been collected. Compressor speed was kept at 400 rpm while the condensing temperature varied from 30°C to 50°C, and the evaporating temperature, from 0°C to 15°C.

Figure 5 illustrates the compressor power requirement, and the cooling and heating capacities of the unit. It can be seen that large heating coefficients of performance, COP_h, reaching a maximum of 7, were obtained. A decrease in the condenser heating capacity is observed with decreasing evaporating temperatures. Although the condenser power output could be increased by as much as 50% when the condensing temperature, T_c, was brought down from 50^0C to 30^0C., the compressor power requirement showed little variation with T_c. Figure 5 also reveals that, for any operating condition, evaporator and condenser capacities are practically the same. This is due, in part, to the large values of COP_h and, mainly, to the heat loss from the condenser shell which has not been insulated from ambient

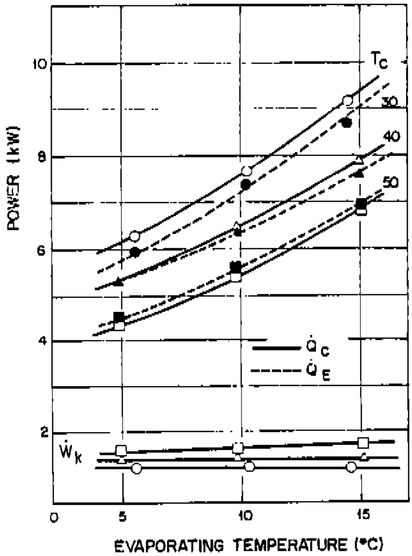

Figure 5 - Cooling and heating capacities and power requirement of a refrigeration unit.

On Figure 6 values of ECR were plotted against R, for condensing temperatures of 30^0C and 50^0C. The same trend of the ideal ECR was observed. Better results were obtained for the condensing temperature of 30^0C. It means that, for better heat recovery, the condenser should be allowed to operate at its lowest possible temperature. At first, this result would seen to be in contradiction as better results are obtained with the lowest condenser outlet temperatures. However, with lower a condensing pressure there is less compressor power consumption. Besides, higher condensing temperatures led to higher heat losses to the environment (usually at 15-25^0C). Compared with the ideal ECR curve, it can be seen that the range of R values for which heat recovery is still favourable becomes even smaller.

4. EFFECT OF TEMPERATURE REQUIREMENTS

On the ideal cycle study only energy (heat and work) quantities were involved whilst on the experimental analysis, in the previous section, the only concern with temperature regarded that of condensing and evaporating temperature levels. In practice, however, there is a definite requirement regarding the temperatures of water (or air, in smaller units) achieved at both evaporator and condenser/heater outlets. To evaluate the effects of these temperatures on the performance of the plant as a total energy system, two simulation models were employed.

For the direct recovery scheme, which is basically a vapour compression cycle, a simulation model from Parise [10] was employed. It is based on the application of simple mathematical models for each of the main components of the cycle. They resulted in a set of

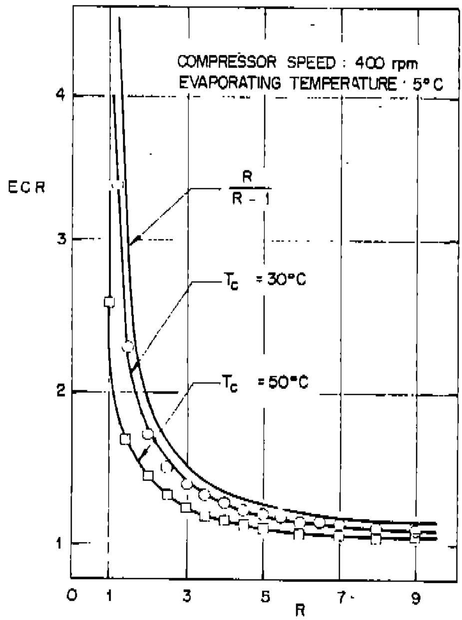

Figure 6 - ECR versus R for an experimental refrigeration unit.

non-linear equations, which was solved by an appropriate algorithm. For modelling the compressor it was assumed that compression followed a polytropic process of known index. This provided the state of the refrigerant at the compressor discharge. Refrigerant mass flow rate was evaluated from the theoretical volumetric efficiency multiplied by an empirical coefficient, to account for valves and piston rings losses. Both condenser and evaporator were treated as having a known overall heat transfer coefficient as well as the heat transfer area. The expansion valve, of the thermostatic type, was simulated by assuming that the evaporator outlet superheat was known. Among other parameters, the model was able to predict both condensing and evaporating temperatures, cold and hot water outlet states and compressor power consuption.

The cascade recovery scheme (a heat pump with the refrigeration condenser output as its heat source) was simulated with a computational code derived from the basic vapour compression cycle model [10]. For simulation purposes two separate heat exchangers (low pressure condenser and high pressure evaporator), with a intermediate coupling liquid (water) were employed. The basic vapour-compression algorithm was applied for both high ared low pressure circuits until convergence in the coupling liquid temperatures was achieved.

The two models were applied to simulate typical refrigerant plants, with characteristics close to the experimental rig [9].

Results of the simulation analysis are shown in Figures 7 to 10 . With the simulation model a few parameters were made to vary. The main objective was to assess the influence of the temperature levels on the predicted performance of both systems. Thus, for the direct recovery scheme, the varying parametrs were : condenser water inlet temperature (30, 40, 50 and 60°C), required hot water outlet temperature (70, 80 and 90°C) and evaporator water inlet temperature (10, 20, 30 and 40°C). With the heat pump recovery scheme similar parametrs were chosen: the high pressure condenser water inlet temperature (30, 40, 50 and 60°C), low pressure evaporator water inlet temperature (10, 20, 30, 40 and 50°C) and hot water required temperature (70, 80 and 90°C).

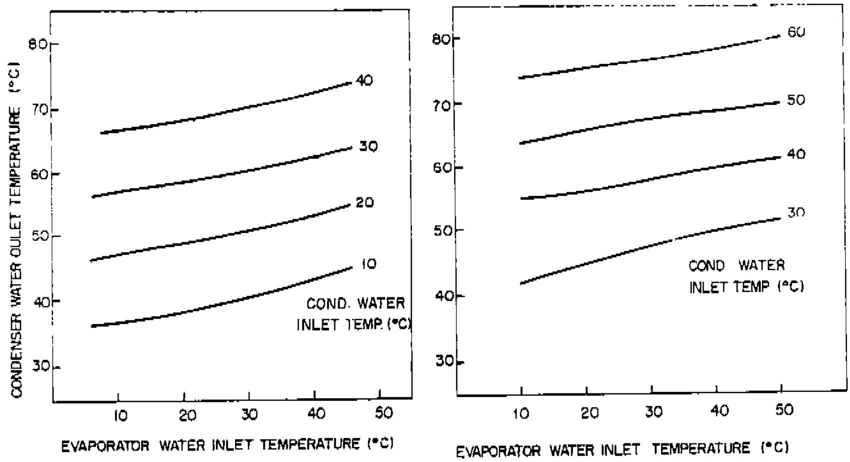

Figure 7 - Condenser water outlet temperature os function of water inlet temperatures at both evaporator and condenser. a) Direct recovery schem(b) Cascade recovery scheme.

Figure 7 outlines, for both schemes, the condenser contribution, as far as temperature levels are concerned, to the water heating process. As expected the condenser water outlet temperature increases with both refrigerated medium and condenser return water temperatures. Note that the cascade system allows for higher temperatures to be obtained at the condenser, as the temperature up-grade is made in two stages instead of one.

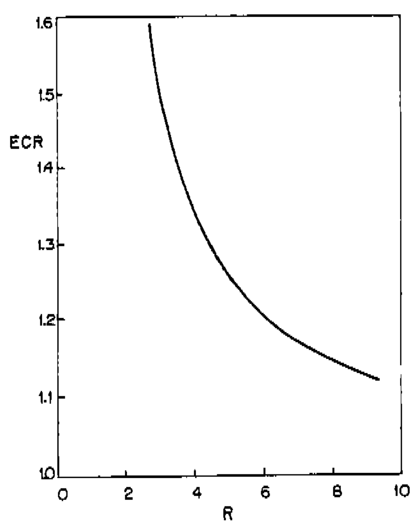

Figure 8 - Predicted ECR against R for varying water temperatures at evaporator inlet (10 - 40^0C) and overall system required outlet (70 - 90^0).

In Figure 8 , a plot was produced for the predicted ECR against R, for all studied

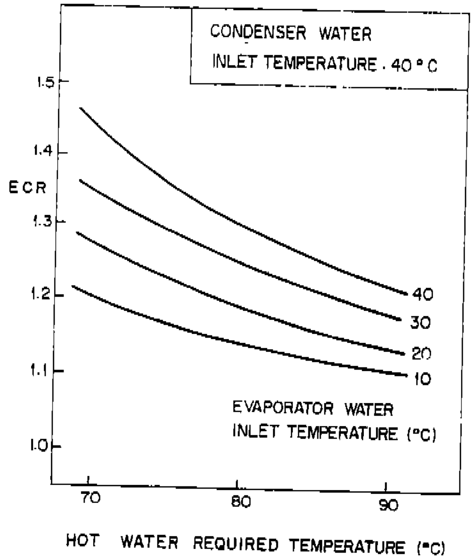

Figure 9 - System energy conversion ratio versus hot water required temperature for the direct recovery scheme.

conditions, including both recovery schemes. It can be noted that it resulted in the same type of curve of Figure 3, derived from the ideal cycle analysis. Values very close to the ideal ECR were obtained, as the simulation model, in its present form, did not take heat losses into account. Besides, the polytropic index of compression was chosen so that the compression process was nearly isoentropic.

Figure 9 illustrates the effect that the hot water required temperature has on the system ECR, for a given set of operating conditions. Clearly the system overall eficiency decreases when higher outlet temperatures are pursued. The same trend was observed for the cascade scheme, as shown in Figure 10. This is because the higher the outlet temperature the more load will be relied upon the less efficient heater.

It is worth mentioning that the ECR values were comparatively low (never exceeding 2). Therefore, although equation (6) would apply for this specific case under study, there is a definite limitation imposed upon the range of possible values of R and, consequently, ECR, under which the system can operate. These restrictions are due to the temperature levels of the cycle. On the other hand the ECR always remained above unity, showing that, at least all energy consumed has been converted into useful heat.

5. CONCLUDING REMARKS

Heat recovery from refrigeration plants has been analysed. It has been shown that considerable reduction on the energy consumption can be obtained, particulary when similar heating and cooling loads are involved. Also, from the experimental results, it has been found that lower condensing temperatures yeld higher ECR's. However, if the required hot water temperature is well above the condensing temperature, most of the heating load will be left to the conventional heater, thus reducing the ECR. In short, a considerable amount of heat can be recovered from refrigeration plants although at low temperature levels. In practice, it will be difficult to find systems capable at operating with heating loads close to the refrigeration load (i.e., R close to 1).

The present analysis has been restricted to R values greater than 1. For lower heating loads, although exceeded by the condenser output, there would be the temperature requirement to be met. Further research is necessary.

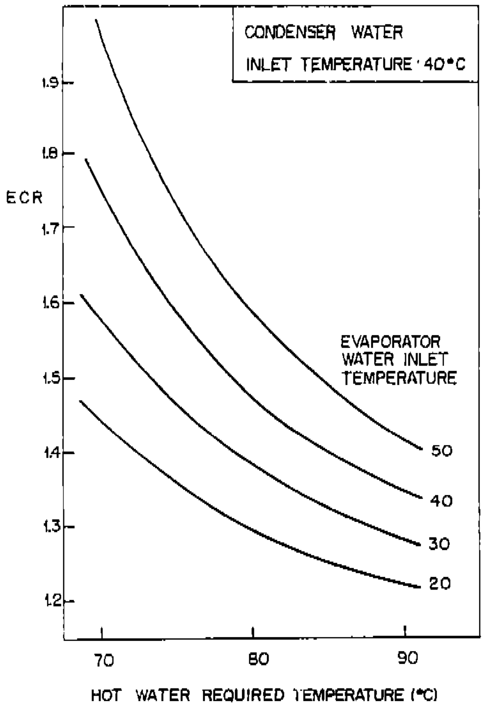

Figure 10 - System energy conversion ratio versus hot water required temperature for the cascade recovery scheme.

REFERENCES

1. H.L. von Cube, Air-conditioning system with de-centralized heat pumps for heat recovery and internal energy transfer, Clima Commerce Int., n 4, (1972).
2. K. Hartmann, Heat Pumps with Turbo-Refrigerastion Machines in Cascade Arrangement, Klimakaltechnik, n 4, pp 88-92, (1975).
3. L.F. Charity et al, Heating Water with Milk Cooler Using Heat Pump Principle, Agric Engng, pp 216-219, April, (1952).
4. Electricity Council Notes, Heat Pump Maintains Plastic Injection MouldingOutput, Case History, EC3822 U.K., (1978).
5. E. Perry, R Kilburn and M.V. Casey, A Heat Pump System to Recover Waste Heat from A Refrigeration Plant in a Brewery, Int Chem Eng Symp Series, n 48, pp 25. 1-10, U.K. (1977).
6. S. Ghosh, S. Devotta and V.S. Patwardhan, The Economics of Heat Pump Systems for Simultaneous Heating and Cooling, Heat Recovery Systems and CHP, vol 7, n 2, pp 159-166, (1987).
7. V.K. Gupta, Optimum Desing of Multi-Stage Cascaded Refrigeration Heat Pump System, Heat Recovery Systems and CHP, vol 6, n 3, pp 235-244, (1986).
8. C.J. Andreen (editor), Recovering Energy from Refrigeration, Energy Technology, n 2, pp 12-13, Sweden (1988).
9. J.A.R. Parise and W.G. Cartwright, Experimental Analysis of a Diesel Engine Driven Water-to-Water Heat Pump, Heat Recovery Systems and CHP, vol 8, n 2, pp 77-85, (1988).
10. J.A.R. Parise, Simulation of Vapour-Compression Heat Pumps, Simulation, 46:2, pp 71-76, February 1986.