Heat Recovery Systems & CHP Vol. 14, No. 1, pp. 61–66, 1994
Printed in Great Britain

0890-4332/94 $6.00 + .00

VALVE CONTROL FOR OPTIMUM PERFORMANCE IN COMPRESSION REFRIGERATION CYCLES

GIOVANNI CERRI and LORENZO BATTISTI
Dept of MS&PA, Engineering Fac, University of Trento, Via Mesiano, 77, 38050 Trento, Italy

(*Received* 28 *January* 1993)

Abstract—A performance optimization system has been studied for refrigeration plants based on vapor compression cycles. The water temperature at the evaporator exit is maintained at the requested value when the refrigeration capacity varies. Minimum power consumption is also achieved. The method is based on the calculation of valve opening set point law. Valve openings are related to the external quantities: water mass flow in the condenser and evaporator, temperature of the water at the condenser inlet, compressor and electric motor serviceability, fouling of heat exchangers, etc. Results of an investigation carried out on a 200 kW refrigeration capacity plant are given and discussed.

1. INTRODUCTION

Vapor Compression Refrigeration plants (VCR) are widely used for air conditioning. A plant scheme and its cycle are shown in Fig. 1. Operations of VCRs require the achievement of two goals:

(i) the water temperature at the evaporator exit ($T15$ or T_{weo}) should remain constant whatever the water temperature at the evaporator inlet is (i.e. refrigeration capacity). Moreover, it should be insensitive to the changes in boundary quantities. Such variables are: the water mass flow for condenser cooling (m_{wc}), its inlet temperature (T_{wci}) and water mass flow at the evaporator (m_{we}). This aspect is particularly important with respect to the conditioning plant equipment design and efficiency;
(ii) power consumption at any cooling capacity (CC) has to be as minimal as possible.

The above aspects are important because of the required capacity variations during the plant operations. The above goals should be achieved at part load operations not only at the design conditions.

This paper refers to plants equipped with reciprocating multi-cylinder compressors although the results are applicable to VCRs with other types of compressors.

The compressor displacement rate has to be adapted to the required refrigerating capacity (RC). Thus reciprocating compressors are equipped with cylinder unloading systems and sometimes with variable speed drives.

The most widely used expansion devices are the traditional thermostatic expansion valves (TEV) with internal or external equalizers. To avoid liquid entry into the compressor cylinders, the control of this kind of valve is based on the vapor superheat at the evaporator exit. This is achieved by controlling the refrigerant mass flow rate entering the evaporator. Of course, in steady state conditions this mass flow is the same throughout all the components [1, 2, 5].

Generally, the use of thermostatic valves leads to large changes in the water temperature at the evaporator exit when the required refrigeration capacity varies. The above occurs with the plant operating at constant velocity and with a constant number of active cylinders. For example for $T_{weo} = 7°C$ (design value), T_{weo} may become 3°C at 80% RC, and −2.5°C at 60% RC.

A microcomputer-based system for the monitoring and control of the vapor compression refrigeration unit has been studied. It establishes the set values for the speed (stepped) of the electric motor, and for the number of unloaded cylinders. Moreover, the system drives a motored expansion valve. The aim is to have a constant temperature of cooled water at the evaporator exit and to operate with minimum power consumption at any refrigeration capacity load.

After a brief description of the system, the method of obtaining the valve opening set points is presented in this paper and some results are discussed.

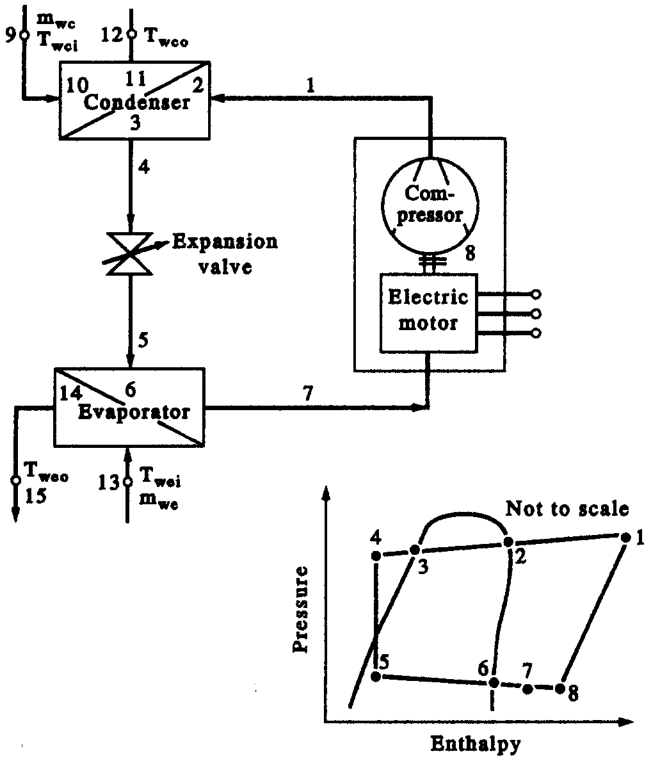

Fig. 1. Schemes of the vapor compression refrigeration plant and cycle.

2. SYSTEM DESCRIPTION

The method is based on a monitoring system by which the level of serviceability (i.e. state) of the VCR components can be established (see Fig. 2).

The actual plant component behavior is described by the new component performance functions and parameters and by deterioration scaling factors. Thus the actual electric motor efficiency curve and compressor volumetric and compression efficiency curves are established taking into account the fouling factors of the heat exchangers. Also coefficients for the calculation of pressure drops and thermal energy losses are updated by measured data or by empirical relationships previously established empirically.

Using the monitoring system, output data as the input to the inverse solution of the plant model, it is possible to obtain the above plant component state parameters and/or functions.

For example, the actual volumetric efficiency (η_{va}) calculated by the above inverse plant model solution is expressed as:

$$\eta_{va} = \eta_{vi} F_{vn} f_{vs}. \tag{1}$$

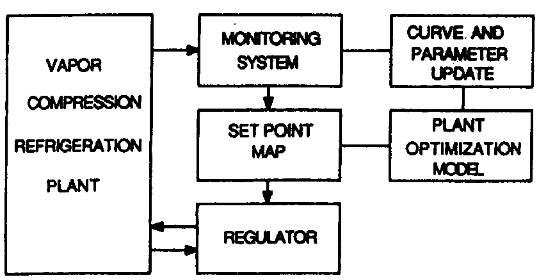

Fig. 2. Block diagram of the Performance Optimization System.

η_{vi} is the ideal volumetric efficiency based on the compressor clearance. F_{vn} is a function accounting for the non ideal phenomena when the compressor is new. Such phenomena are back-flows throughout the valves, leakages, vapor heating due to dissipative actions (friction, cylinder unloading, etc.), and heat transferred essentially from the discharge areas. Experiments have been carried out to investigate F_{vn} for the compressor family considered in the present work. Results show that F_{vn} depends on the motor synchronous speed, on the number of active cylinders, and on the pressure ratio. f_{vs} is the scaling factor and represents the state of the compressor (i.e. it summarizes the history of the machine) and is calculated by the monitoring output data.

A similar approach is adopted for the other apparatus performance curves and parameters. Periodically the actual apparatus performance curves and parameters are automatically calculated. If the above quantities are found to be far from those previously calculated, the new curves and parameters are input as data in the Optimization Plant Performance Model (OPPM). Then the new expansion valve set point map is calculated.

During plant operations, it is possible to find the optimum set value of the electric motor synchronous velocity, number of unloaded cylinders, and valve opening. This is done by entering into the map the external variables; such variables are: condenser water mass flow (m_{wc}) and its inlet temperature ($T_{wci} = T9$), evaporator cooled water mass flow (m_{we}) and inlet and outlet temperatures ($T_{wei} = T13$ and $T_{weo} = T15$). Corrective actions have to be initiated by the regulator if some quantities exceed their set point or threshold values, e.g. the valve opens when T_{weo} rises.

The expansion valve adopted for this study is a multi-hole shell type with a cylindrical sleeve. A stepping motor provides the sleeve movement to produce the valve opening variation (see Fig. 3).

3. STEADY STATE PLANT MODEL

The steady state model takes into account the plant apparatus: evaporator, electric motor and reciprocating compressor group, condenser, and expansion valve.

Heat transfer processes have been formulated adopting the NTU-effectiveness approach. Both the condenser and evaporator have been modelled as multi-zone heat exchangers. Each zone is

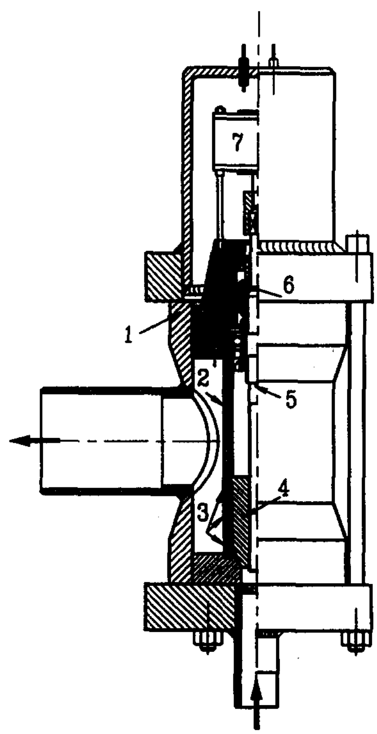

Fig. 3. Proposed multi-hole expansion valve with stepping motor: 1, valve housing; 2, cylindrical shell; 3, holes distributed along a spiral; 4, sleeve; 5, screw driving the sleeve; 6, sealing system; 7, stepping motor.

characterized by the refrigerant state: desuperheating, condensing, and subcooling (for the condenser), evaporating, and superheating (for the evaporator).

The compressor is semi-hermetic—the superheated refrigerant enters the electric motor then enters the compressor suction area. After the compression process the refrigerant is discharged into the condenser.

All the electrical power enters the refrigerant being compressed, one fraction as heat due to dissipative actions and the complementary fraction as compression work. Some thermal power is exchanged between the compressor discharge areas (mufflers and tubing) and the external environment.

The expansion valve is modelled by the relationship between the refrigerant mass flow, valve opening (Ω_v) and pressure drop. This equation applies to the liquid phase because the refrigerant flashing does not occur inside the fluid passage throughout the orifices; the liquid flow is essentially at measurable conditions.

The model is completed by the refrigerant and water thermodynamic and transport physical property equations.

The model form is a set of non-linear equations that are solved by the simultaneous solution method proposed by Cerri (1991) [4]. To obtain such a solution an unknown variable vector, X, is chosen. The vector components are all the refrigerant, water, and tube wall temperatures, the fractions representing the extensions of the heat exchanger zones and the rotational speed of the compressor.

The model has been used to solve the following problems:

(1) The unknown variable vector is completed with the valve opening Ω as a further unknown quantity, then the solution vector $\bar{X}$ is determined solving the following minimization problem:

$$\text{minimize} \left\{ \frac{1}{\text{COP}} \middle| m_{we} = m^*_{we},\ T13 = T^*_{wei},\ T15 = T^*_{weo},\ m_{wc} = m^*_{wci},\ T9 = T^*_{wci} \right\} \tag{2}$$

(* shows the actual values of the degree of freedom have been fixed) constraints in (2) are the externally dependent quantities entering the plant. In this way it is possible to calculate the valve openings for constant cool water temperature at the evaporator exit and for minimum power consumption.

(2) A further equation is added to the model, it is the regulator equation which relates the valve opening to the other quantities. Here the problem that has to be solved presents the following constraints:

$$\{m_{we} = m^*_{we},\ \dot{Q}_{cool} = \dot{Q}^*_{cool},\ m_{wc} = m^*_{wc},\ T9 = T^*_{wci}\}. \tag{3}$$

The adopted method is based on minimization of a composite objective function. One component is a performance function, like the inverse of the COP, the other is the plant imbalance. The method is based on a recursive quadratic programming technique. Temperatures have been chosen as unknown variables because they allow the direct calculation of the various physical quantities.

For the condenser and evaporator there is the opportunity to choose whether condensation and evaporation occur inside or outside the tubes.

4. RESULTS AND COMMENTS

A 200 kW refrigeration capacity plant having a water cooled condenser and equipped with a 4 cylinder reciprocating compressor has been considered for the present application.

Calculations have been performed assuming R22 as the refrigerant fluid. Other quantities are: $m_{we} = 8\ \text{kg s}^{-1}$, maximum evaporator water temperature difference, $T_{wei} - T_{weo} = 6°\text{C}$, set value for $T_{weo} = 7°\text{C}$, $m_{wc} = 11.5\ \text{kg s}^{-1}$, $T_{wci} = 22°\text{C}$, $T5_{min} = -15°\text{C}$, $T2_{max} = 60°\text{C}$.

The thermostatic valve which has been taken into consideration is one usually adopted by the manufacturer, and has an external equalizer and a phial cross charge. The simplified empirical relationship between valve opening, evaporator saturation temperature ($T6$), superheated vapor

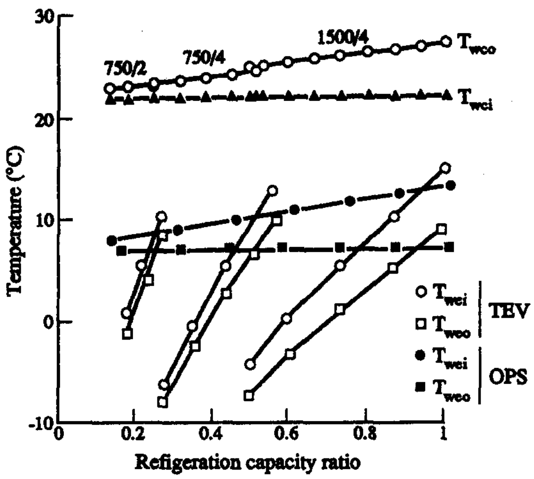

Fig. 4. Condensing water and cooled water temperatures with the thermostatic valve and with the optimum performance system.

temperature ($T7$), and (SS) turns of the superheating setting screw from the factory setting [3, 6] is

$$\Omega_v = A_{rr}\ (T7 - T6 - a_1\ T5^{e1} - SS\ a_2\ T5^{e2})\ a_3\ T5^{e3}. \tag{4}$$

The opening action arising from the unbalanced pressure across the valve port has been neglected in this equation. The constant values are: $A_{rr} = 7.33$; $a_1 = 1.68\ 10^{-17}$; $e1 = 7.1$; $a_2 = 6.80\ 10^{15}$; $e2 = -6.3$; $a_3 = 2.62\ 10^{-31}$; $e3 = 12.47$.

Calculations have been carried out with the above valve, and with the optimum performance system proposed in the paper.

Comparisons between the two methods are shown in Figs 4 and 5. Figure 4 shows the water temperatures at both the entrance and exit of the evaporator and of the condenser. The figure shows that during operations with the TEV, T_{weo} does not remain constant. When there is a variation of refrigeration capacity rate of the compressor, whether by velocity reduction at about 58%; or by cylinder unloading, at about 28%, T_{weo} rises again at the set value. Then T_{weo} decreases again with the required refrigeration capacity ratio (RCR). This means that when operating with the TEV, oscillations of the cooled water temperature at the evaporator exit occur as a consequence of the load variations. With OPS, T_{weo} remains constant across the whole field of RCR variation. Water condenser temperatures behave in the same way with both the TEV and OPS.

Valve opening RCR curves for both the TEV and OPS are given in Fig. 5. These curves show the differences of valve openings between the thermostatic controlled valve and the OPS controlled valve. The COP does not show differences between OPS and the present TEV used at factory setting.

External variables: m_{wc}, T_{wci}, m_{we}, T_{wei}, and the set point for T_{weo} have a strong influence on the valve opening. Figure 6 shows the OPS valve opening vs T_{wci} when synchronous speed, number of active cylinders, and RCR are kept constant: the opening decreases when the water temperature at the condenser entrance rises.

Using the TEV, if the plant manufacturer puts $SS = -2$ turns (anti-clockwise) the vapor superheating is reduced to half that for $SS = 0$, and decreases with RCR more rapidly. The curve of T_{weo} vs RCR (Fig. 4) is shifted toward lower values.

5. CONCLUSIONS

It has been shown that it is possible to implement part load refrigeration plant behavior with the optimum performance concept to obtain minimum power consumption and to maintain cooled water temperature at the evaporator exit constant, i.e. independent of the RCR.

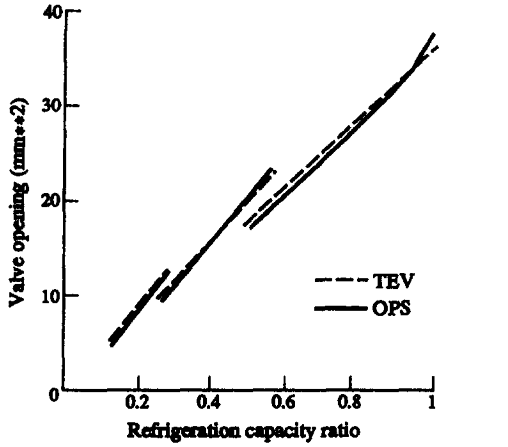

Fig. 5. Valve opening vs capacity with the thermostatic valve and with the optimum performance system.

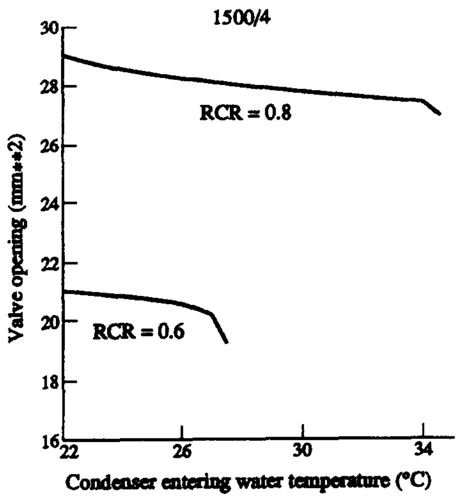

Fig. 6. OPS valve opening vs water temperature at the condenser inlet.

Moreover, it has been shown that with a traditional thermostatic valve there is about the same power consumption OPS while the cooled water temperature decreases with the refrigeration capacity being reduced.

Acknowledgements—This work was partially funded by research grant MURST 40%. The authors thank Mc Quay Italia of the Snyder General Co., Mr. Luca Paolella and Mr. Massimo Di Giosia for their kind collaboration.

REFERENCES

1. W. F. Stoecker and J. W. Jones, *Refrigeration & Air Conditioning*. McGraw-Hill, Singapore (1982).
2. D. Parnitzki, Digital control for heat pumps with minimized power consumption. *Int. J. Energy Res.* **13**, 167–178.
3. Y. T. Wang, D. R. Wilson and D. F. Neale, Heat-pump control. *IEE Proc.* **130**, No. 6, November (1991).
4. G. Cerri, Un metodo generale per la ottimizzazione dei sistemi termodinamici. *L'Energia elettrica* **68**(12) (1991).
5. M. Zaheer-uddin, Multiple setpoint control of a heat recovery system. *Heat Recovery Systems & CHP* **12**, 23–27 (1992).
6. M. R. Conde and P. Suter, A mathematical simulation model for thermostatic expansion valves. *Heat Recovery Systems & CHP* **12**, 271–282 (1992).