بسم الله الرحمن الرحيم

Transient Operation of Mechanical Vapour Compression Desalination-System

التشغيل غير المستقر لنظام إعذاب الماء بطريقة الانضغاط الميكانيكي للبخار

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خلاصة: نتيجة لزيادة الطلب على الماء العذب في الحاضر والمستقبل [illegible]

[illegible] لإمكانية الوصول بها لحالة التشغيل المستقرة

Abstract Due to continuous increase of pure water demands for present and future time and due to the limited natural resources of pure water, the used techniques of producing pure water from sea and brackish water have a great interest and expose to a great improvement day by day. The mechanical vapour compression system is one of the most important used methods. This system has many advantages in comparison with other systems. Beside its simplicity in use and control point of view, it has minimum running cost (mainly, the cost of chemical additives).

As a result of running a unit of 10 ton/day nominal capacity, the self-starting of the units based on this technique is a point of great importance and needs deep understanding of the involved processes, especially, in case of transient operation of such units (i.e. the starting condition reaching to the normal operating condition). In this work, a simple qualitative analysis of the starting condition and the possibility of break down of the system (the departure from the stable operating condition) is presented. As a result of this analysis and the results obtained, experimentally, some sort of arrangements must be added to meet the stable starting condition (starting condition which leads to a stable running condition).

Nomenclature

M mass of steam (*kg*)

m mass flow rate of steam (*kg/s*)

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p	pressure (bar)
t	time (s)
V	volume (m^3)
v	specific volume (m^3/kg)

Subscripts

c	compressed steam
f	feed water
g	generated steam in main compartment
p	produced water
1	main compartment of evaporator
2	the high pressure compartment
3	compartment, where the produced water is collected

Introduction

Among the water desalination techniques, the vapour compression (VC) system is expected to be widely used in Egypt because of its advantages. Running and maintenance of such system need no specially qualified labours, and in addition it requires reasonable running cost compared with other methods [1 and 2]. In [3] a survey of water treatment techniques for industrial purposes is presented.

The present study is carried out for 10 ton/day VC unit. A schematic drawing of this unit is shown in figure (1). The evaporator of this unit is physically divided into three compartments. Main compartment (No. 1) is the *tubes compartment*, where the steam is generated. Second compartment (No. 2) is the *high pressure compartment*, where the compressed steam is accumulated. The third one is the *product compartment*, where the produced pure water is collected. The troubles associated with the starting of such units is analysed in the following sections of this work.

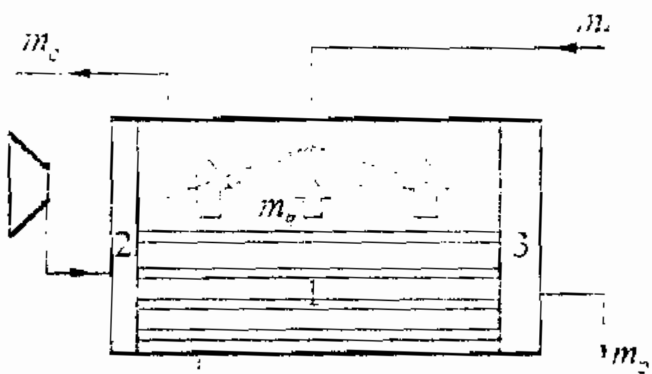

Figure (1) Schematic layout of the mechanical vapour compression unit

Experimental Measurements

To examine the behaviour of the considered VC unit, both pressure and temperature at the suction and delivery of the compressor are measured. Neglecting the heat losses and pressure drop in the connecting pipes, these measured values can be taken as those of the associated evaporator components. The measurements of both temperature and pressure are carried out by commercial bourdon-tube dial gauge. Figures (2 and 3) show a sample of the obtained results.

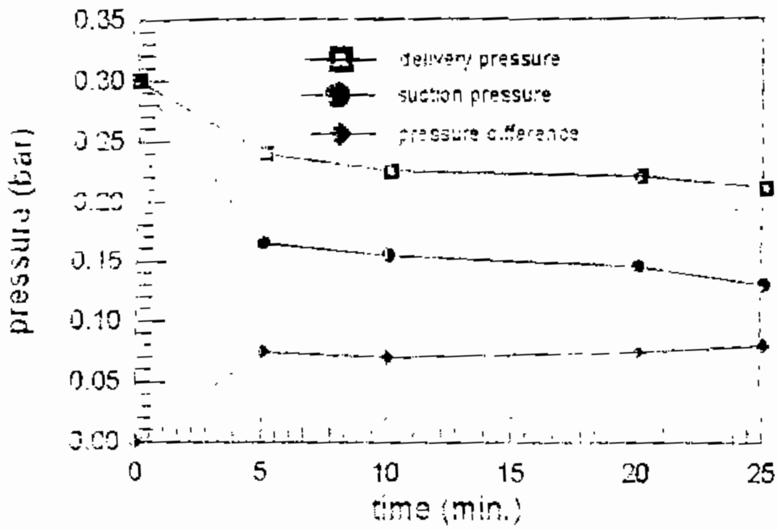

Figure (2) Pressure distribution at the delivery and suction cross-sections of compressor

Figure (2) shows the pressure at the delivery and suction of the compressor versus the time. As it is clear both pressures, first, decrease rapidly and then slowly. Delivery pressure, then takes almost constant value while, the suction pressure continuously decreases. The pressure difference across the compressor is shown in the same figure. This difference at the start tends to be zero and then, in very short period, it increases sharply. After a period of about five minutes it increases continuously but at a smaller rate.

Figure (3) presents the temperature at the suction and delivery of compressor. As it is shown, the temperature at the suction exhibits a gradual decrease through out the test period, while the temperature at the delivery increases rapidly at the beginning and then slowly. Near the end of the test period it, again, increases rapidly.

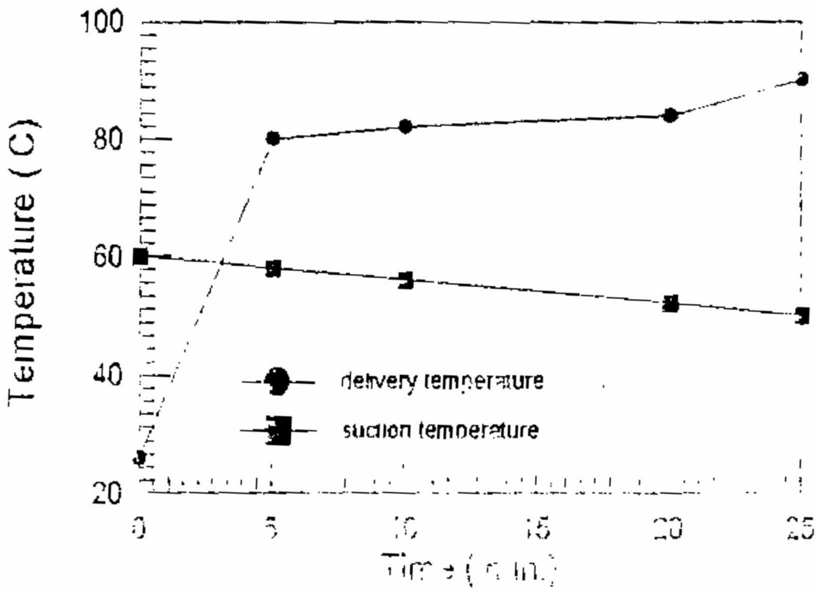

Figure (3) Temperature distribution at the delivery and suction cross-sections of the compressor

From both mentioned figures (2-3), one can see that, the unit fails to achieve a stable operating condition. The effect of feed water temperature is not presented here, but one can conclude according to the carried out tests that, the period after which the unit breaks down is a function of the feed water temperature. It is longer for higher feed water temperature.

Governing Equations

The following analysis is based on the overall mass balance equations [4-5]. Both *tubes compartment* (No.1) and *high pressure compartment* (No.2) are considered. Accordingly, one can write:

Tube compartment:

$$M_1 = \int (m_e - m_c)\,dt + C_1 \qquad (1\text{-a})$$

and thus one can write an expression of specific volume in the main compartment as:

$$v_1 = \frac{V_1}{\int (m_e - m_c)\,dt + C_1} \qquad (1\text{-b})$$

igh pressure compartment:

$$M_2 = \int (m_2 - m_g)dt + C_2 \qquad (2\text{-a})$$

nd in the same manner, one can write an expression of the specific volume in the high ressure compartment as:

$$v_2 = \frac{V_2}{\int (m_c - m_v)dt + C_2} \qquad (2\text{-b})$$

where M_1 *and* M_2 are the total accumulated mass within the tubes- and high pressure compartments, respectively. V_1 *and* V_2 are the total volume occupied by the steam in tubes- and high pressure compartments, respectively. v_1 is the specific volume of steam in tubes compartment while, v_2 is the specific volume in high pressure compartment. One can note that, he pressure in both compartments is inversely proportional to the corresponding specific volume. C_1 *and* C_2 *appear in the equations are constants*, their numerical values play nothing in he following analysis.

Theoretical analysis of the problem

The compressor of the present study is of the regenerative-turbine type. The theory of operation and performance of pumps of this type are reported in [6]. According to the nature of this compressor, the approximation of the head-capacity curve of such kind of compressors by a straight line is acceptable. This approximation is carried out because the exact characteristic curve of the compressor is not available and, in the same time, this approximation makes the analysis presented in this work easier. This characteristic curve of he compressor is shown in figure (4).

To explain the behaviour of this VC small unit and the reasons of its unstable starting, the following simple analysis is carried out; with the aid of both the proposed compressor characteristic curve- figure (4)- and the overall mass balance equations (1 and 2). As it is clear, the pressure difference across the compressor is dependent on the different mass flow rates crossing the evaporator compartments. From mass flow rates point of view, there are three possible operating conditions. *The rate of generated steam may be equal to, less than or greater than the compressed steam.* Each of these three operating conditions will be discussed in the following sections.

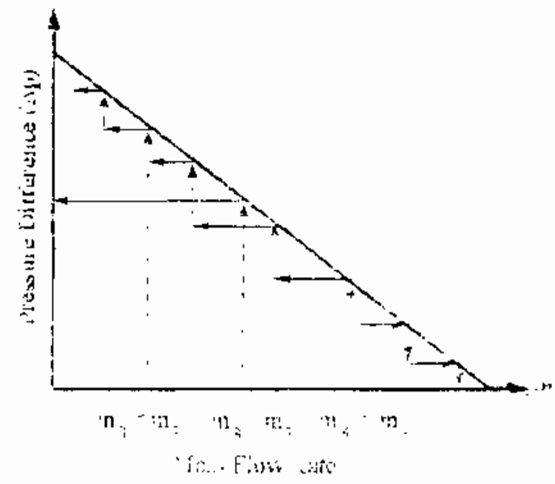

Figure (4) Approximated characteristic curve of regenerative turbine-compressor

a) the generated steam is equal to the compressed steam ($m_g = m_c$) .

In this case, according to equations (1), the mass within the main compartment remains constant and, in turn, the specific volume and hence the pressure are maintained constant. As it is clear from equation (2), if the produced steam (m_c) is equal to the compressed steam and all of m_c is condensed to water, the mass within compartment (2) and also the pressure remain constant. Accordingly the pressure difference across the compressor is maintained constant. Otherwise, the pressure in compartment (2) increases and in turn, the pressure difference across the compressor increases and hence the mass flow rate of compressed steam decreases. These two alternatives are shown in figure (4).

b) the generated steam is greater than the compressed steam ($m_g > m_c$) ;

As it is clear from equations (1), v_1 decreases and thus p_1 increases. From equations (2) and for the case of ($m_c = m_w$), the pressure in the compartment (2) p_2 remains constant and accordingly the pressure difference across the compressor ($\Delta p = p_2 - p_1$) decreases and with the aid of figure (4) the rate of compressed vapour increases till it becomes greater than the generated steam. This condition will be examined in the next possible operating condition (c). On the other hand, if ($m_c > m_w$) the pressure p_2 increases such that the pressure difference Δp decreases. This case leads to a situation looks like to the previously mentioned one (case a). The other alternative is that Δp increases which also leads to the next condition.

c) The generated steam is less than the compressed steam (mg < mc) ;

From equations (2), it is clear that v_2 decreases or remains constant for ($m_g \leq m_c$) and, in turn, p_2 increases or remains constant. While with the aid of equation (1), v_1 increases and hence pressure p_1 decreases. Accordingly, the pressure difference ($\Delta p = p_2 - p_1$) increases and due to the characteristic curve of the compressor figure (4) m_c decreases and, in turn, the generated steam m_g decreases. That is due to the decrease of the convective heat transfer rate because of the decrease of the rate of flowing steam through the evaporator tubes.

In general and according to the foregoing analysis, each of the three above mentioned operating conditions leads to continuous increase in pressure difference across the compressor and finally, the system fails to reach the normal operating condition (break down of the unit).

Conclusions

The proper starting of vapour compression system is that starting which leads to a stable normal operating condition. As it is shown in the foregoing analysis, the proper self starting of vapour compression small units is, practically, very difficult to be achieved. Attention must be made to prevent the rapid decay of pressure in the tube-compartment of the evaporator and in the same time to prevent the instantaneous increase of pressure in the high pressure compartment. One can conclude that, some sort of control must be made to insure a suitable pressure difference across the compressor, especially at starting of such a system. A by-pass between the tube- compartment and low pressure compartment (No.3) is proposed. The test of the examined unit with the suggested by-pass leads, successfully, to stable starting of the unit. It s recommended to make a theoretical model of such a system to study the behaviour of such system, quantitatively, in transient and steady state condition.

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