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بسم الله الرحمن الرحيم

Performance of Steam Regenerative-Turbine Compressors*

أداء ضواغط البخار التربينية الاسترجاعية

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خلاصة في هذا البحث تم دراسة أداء ضواغط البخار من النوع التربيني الإسترجاعي نظرياً. بالرغم من بساطة تركيب هذا النوع من الضواغط إلا أنه قليل الانتشار. تم وصف سريان البخار خلال الضاغط عن طريق كل من معادلة ا لسريان ومعادلة كمية الحركة بالاضافة الى معادلة الطاقة في صورتهم الكلية. تم ذلك بتصميم نموذج نظري حل- عددياً- بعمل برنامج للحاسب الآلي. في هذا البرنامج أخذ في الاعتبار سلوك البخار كمادة نقية عند حساب خواصه خلال سريانه في الضاغط. تم في هذا البحث دراسة تأثير ظروف التشغيل المختلفة على أداء هذا النوع من الضواغط. كذلك تم دراسة تأثير أبعاد الضاغط على أدائه.

Abstract In this work, the performance of steam regenerative-turbine compressor is studied. In spite of the simple construction of this compressor, it is rarely used in practical applications. The flow of steam inside the compressor is described through the continuity, momentum and energy equations in integral form. These governing equations are solved, numerically, by design a computer program. This program is designed , such that the steam properties are calculated taking into account the behavior of steam as a pure substance. The effect of operating conditions, such as speed of compressor impeller and suction pressure, on the performance of the compressor is examined. Moreover, the effect of compressor dimensions, represented by the pocket radius to impeller radius ratio, is studied .

1. Introduction

The regenerative-turbine compressor, as shown schematically in figure (1), consists of impeller and collecting passage. The impeller has pockets, which are formed by flat radial vanes. The steam enters the impeller pockets through a suction port, formed in the stationary part of compressor, where its velocity is increased and hence it is directed to the collecting passage, where it is collected and then leaves the compressor through delivery port . In collecting passage, some of steam kinetic energy is transferred to enthalpy at higher pressure. As it is clear, the construction of the impeller is similar to the pump runner of hydraulic coupling, while the turbine runner here is replaced by the collecting passage [1-2]. In spite of the similarity between this compressor and hydraulic coupling, there are a rare knowledge about it in the literature. The performance of regenerative-turbine pump is examined in reference [3].

Because of the simple construction of this type of steam compressor, it is applied in small vapor-compression desalination units despite of higher expected loses. In such applications, a moderate or small pressure ratio is required and moreover the power lost, from turbomachines design point of view, is considered as an added heating source in such units [4]. In present work, an effort is made to make a simple theoretical model to examine the performance of such compressor when steam is used as working medium. Effect of operating conditions and compressor dimensions is studied using the present proposed model. Seeking for simpilicity, the friction and other losses are not considered in the present analysis.

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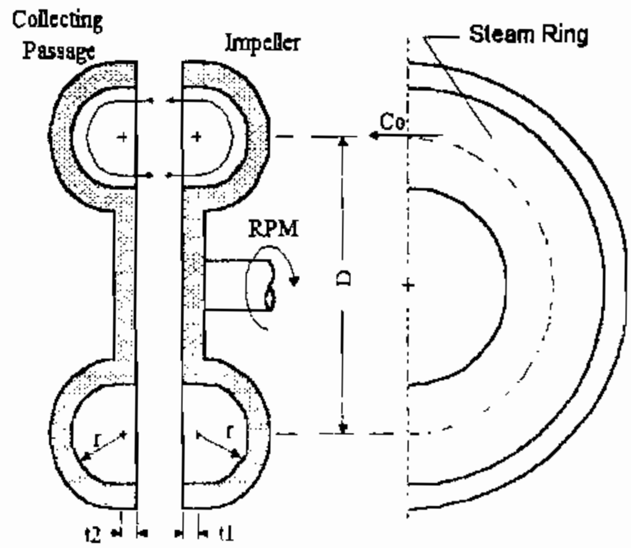


Figure (1) Schematic drawing of compressor showing the steam velocity direction.

2. Governing Equations

The flow inside the compressor is described by the integral forms of continuity, momentum and energy equations. Referring to Figure (1), the steam enters the collecting passage with tangential component of velocity C_o. The change of velocity in both axial and tangential directions is negligible compared with that of tangential component. Figure (2) shows a part of collecting passage, which is considered as a differential control volume of length δx. Applying the conservation laws [5-6]on this element yields to.

$$m + \delta m = m + m_o \delta x \quad , \tag{1}$$

$$A\,\delta p = (m_o \delta x)c_o - [(m + \delta m)(c + \delta c) - mc] \quad , \tag{2}$$

$$(m + \delta m)[(h + \delta h) + \frac{1}{2}(c + \delta c)^2] = m(h + \frac{1}{2}c^2) + m_o \delta x(h_o + \frac{1}{2}c_o^2) \quad , \tag{3}$$

where equations (1-3) are continuity, momentum and energy equations; respectively. As it is clear, these eqnations satisfy the steady state condition. m is the mass flow rate at the inlet cross-section of the control volume. While m_o is the mass flow rate per unit length of the collecting passage, which enters the control volume coming from the compressor impeller. The cross-sectional area of the collecting passage is denoted as A. p is the pressure at inlet of the control volume. c and c_o are the steam velocity at inlet of the control volume and that of steam coming from the impeller; respectively. h and h_o are the enthalpy of steam at inlet and of that coming from the impeller. In addition of the foregoing equations, the equation of state is required to determine the density of steam at various conditions. This eqnation of state can be written in representative form as;

$$h = f(\rho, p) \qquad . \tag{4}$$

As it is well known, there is no single simple relation between steam properties, which satisfies all water phases. According to this fact, it is suitable to use steam tables to determine the steam properties, whenever they are required.

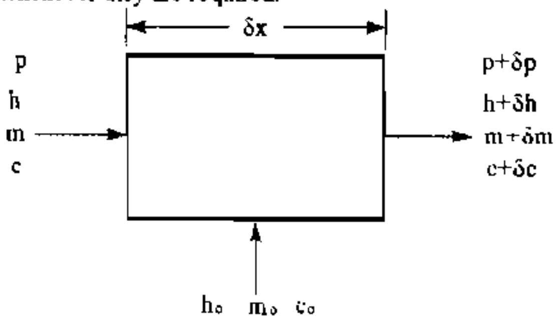


Figure (2) General control volume of length δx isolated from the total length of the collecting passage

3. The Numerical Procedure

In the following sections, the numerical techniqne and the carried out steps to predict the compressor performance at different conditions are presented. Seeking for simplicity, one makes some assumptions. The tangential velocity acquired due to the flowing of steam in the impeller is calculated based on the mean diameter of the impeller D as shown in figure (1). This velocity is calculated according to the following relation as:

$$c_o = \frac{\pi N}{60} D \qquad , \tag{5}$$

where N is the speed of impeller in rpm. The change of the other two components of the velocities is negleeted. The effect of mechanieal friction and the viscous force is also neglected. The maximum possible mass flow rate per unit length of collecting passage $m_{o,max}$ is given according to the relation:

$$m_{o,max} = \frac{\rho_o V_s N}{60 \pi D} \qquad , \tag{6}$$

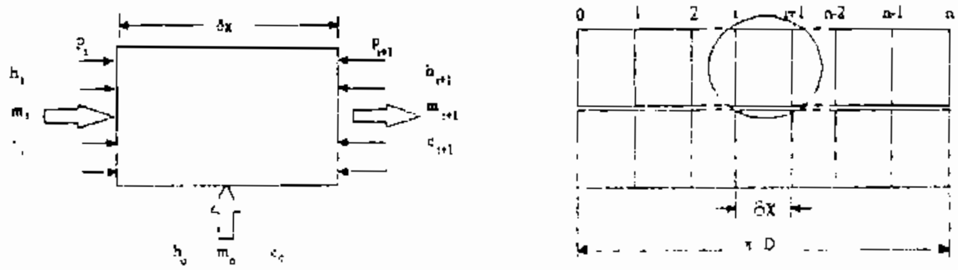
where ρ_o and V_s are the density of steam at the suction of compressor and the volume of impeller pockets; respectively. The volume of the pockets is given according to the following relation as;

$$V_s = V \times \left(1 - \frac{k t}{\pi D} \right) \qquad , \tag{7}$$

where k and t are the total number of pockets and the mean thickness of the vane (mean thickness of pocket wall); respectively. V is the volume of the ring shown in figure(1), which is evaluated through the following relation [7];

$$V = \pi^2 r^2 D/2 + 2 \pi r D t_1 \quad , \tag{8}$$

referring to figure (1) r, D & t_1 are the radius of the pocket cross section, mean diameter of the ring and the length of the pocket tip; respectively.



(a) Mass and energy Transfer through a general cell of collecting passage

(b) Cells identifications of both collecting passage (above) and impeller

Figure (3) Devisions of both collecting passage and impeller of compressor used in numerical technique

Governing equations (1-3) are, numerically, integrated by dividing the collecting passage to n cells, as shown in figure (3). Referring to figure (3-a), the application of conservation laws (1-3) on the shown general cell yields to the following recursive relations as;

$$m_{i+1} = m_i + m_o\, \delta x \quad , \tag{9}$$

$$P_{i+1} = P_i + \frac{1}{A} \left[(m_o\, \delta x) c_o - (m_{i+1}\, c_{i+1} - m_i\, c_i) \right] \quad , \tag{10}$$

$$h_{i+1} = \frac{1}{m_{i+1}} \left[m_o \delta x \left(h_o + \frac{1}{2} c_o^2\right) + m_i \left(h_i + \frac{1}{2} c_i^2\right) \right] - \frac{1}{2} c_{i+1}^2 \quad . \tag{11}$$

where i varies from 0 to n (total number of cells). m_i is the flow rate in the collecting passage at the cross-section i ($m_i = \rho_i\, c_i\, A$) . The cross sectional area of the collecting passage is denoted as A. To calculate the pressure gain at different possible flow rates, One defines the degree of filling ϕ as the ratio between the actual mass flow rate of the compressor to the maximum possible mass flow rate ($\phi = m_o / m_{o,max}$, $1 \ge \phi \ge 0$). Through out the calculations, it is assumed that the thermodynamic properties (ρ, p, h,...etc.) remain constant inside the cells and the change of them takes place, only, at the boundaries of each cell.

A computer program is designed to solve the previously described theoretical model. This program is designed such that, one can study the effect of various operating conditions and the characteristic dimensions of the compressor. To determine the values of the steam properties whenever they are required, a subprogram is designed and linked to the main program. This subprogram is designed based on the steam tables.

4. Results And Discussions

In the following, the effect of impeller speed and suction pressures is presented. Also effect of compressor size on its performance is shown. First of all, the thermodynamic properties of steam through out the collecting passage are depicted by figures (4-7). In these figures, a compressor of 0.3 m mean impeller radius R and of 0.05 m pocket radius r is studied [(see figure (1)]. Impeller speed, in this case is taken as 10,000 rpm and suction pressure is 0.2 bar. Figure (4) shows the pressure distribution along the collecting passage at different values of mass flow rate; corresponding to degree of filling ϕ of 0.10, 0.25, 0.50 and 0.75. As it is clear, the pressure increases gradually in the down stream direction till it reaches its max. value at the exit of the passage (corresponding to θ=360°). It is true for smaller mass flow rates (ϕ = 0.10, 0.25). While for higher mass flow rates (ϕ = 0.50, 0.75) the pressure has its maximum value at a position inside the passage such that this position is nearer to the exit of passage as the flow rate is smaller. As it is noted, for high values of flow rates (e.g. ϕ = 0.70) the compressor fails to do its job (p/p_o at exit < 1.0). The density of steam along the collecting passage is shown in figure (5). The trend of the curves, here, is similar to that of pressure as it is clear from figures (4-5). Figure (6) shows the enthalpy of steam along the passage at different values of the degree of filling ϕ. In general, the enthalpy of steam decreases in down stream direction. The rate of decrease of enthalpy is higher as the flow rate increases. Figure (7) presents the velocity of steam along the collecting passage. The value of the velocity increases almost linearly in down stream direction, especially, for case of small and moderate flow rates. At degree of filling ϕ = 0.70, the velocity increases rapidly, especially, near the exit of passage and its value exceeds the value of acquired velocity (c_o).

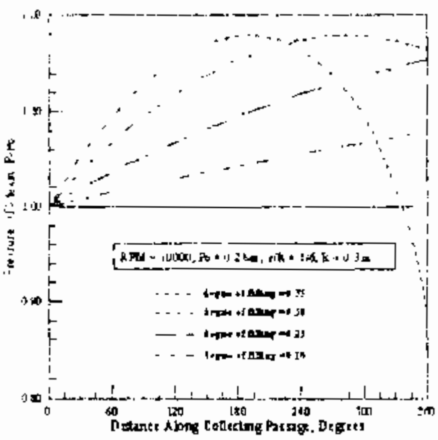


Figure (4) The pressure distribution inside collecting passage

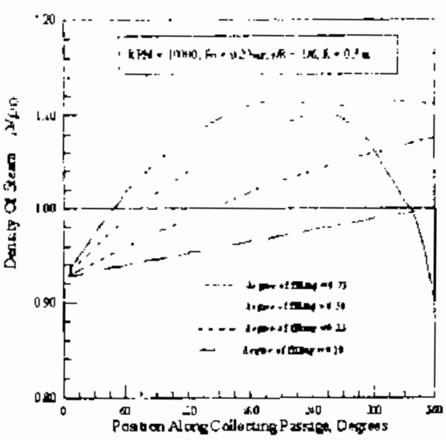


Figure (5) Density of steam inside Collecting passage

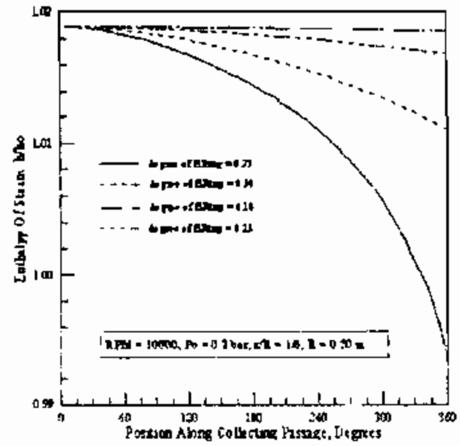


Figure (6) Steam velocity inside collecting passage

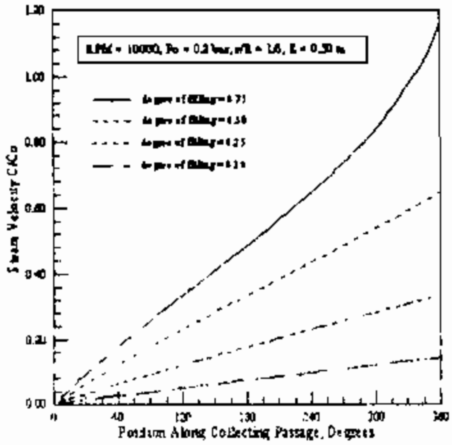


Figure (7) Steam velocity inside collecting passage

Through figures (8-9), the effect of impeller speed on the compressor performance is presented. Impeller of 0.30 m mean radius and 0.05 m pocket diameter is examined. In this figures delivery pressure versus mass flow rate is depicted. In figure(9) mass flow rate is expressed, implicitly, by the degree of filling ϕ. As it is shown, the delivery pressure is increased as the impeller speed increases. For every speed, the pressure increases as the flow rate increases till it reaches a maximum value at ϕ is equal to about 0.40. After this maximal value, the pressure decreases sharply for higher speeds. As it is well known, the left branches of these curves represents the unstable operation of the compressor (pressure increases as mass flow rate increases).

Figures (10-11) shows the effect of suction pressure on the performance of compressor. The same previously mentioned impeller is examined. As the suction pressure decreases the delivery pressure ratio p/p_o increases. This increase of pressure ratio increases with the increase of degree of filling ϕ until $\phi \cong 0.4$ and then decreases with increasing ϕ.

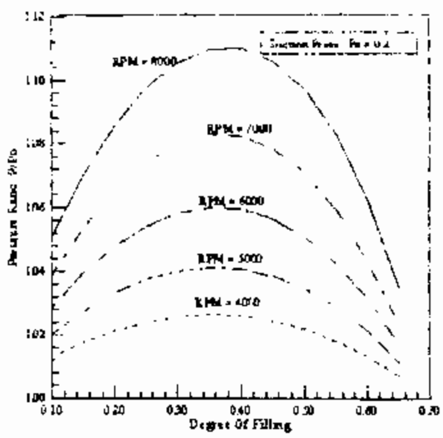


Figure (8) Dimensionless delivery pressure versus degree of filling at different values of RPM

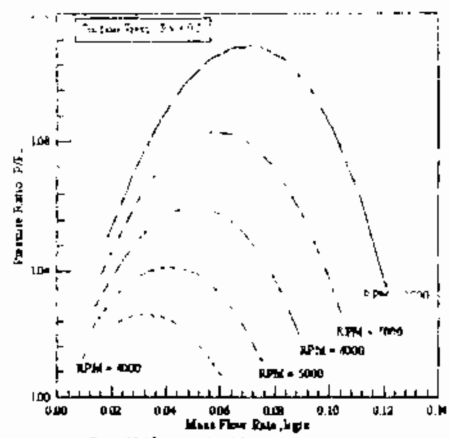


Figure (9) Dimensionless delivery pressure against mass flow rate at different values of RPM

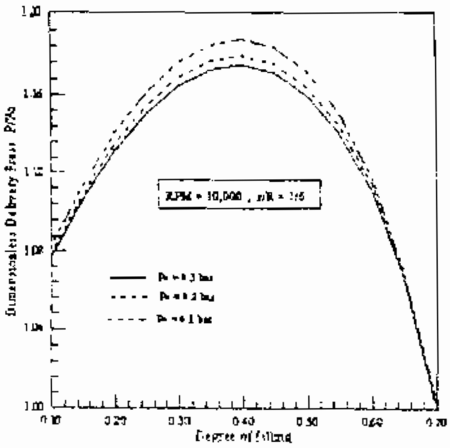


Figure (10) Effect of suction pressure on the pressure at exit of collecting passage

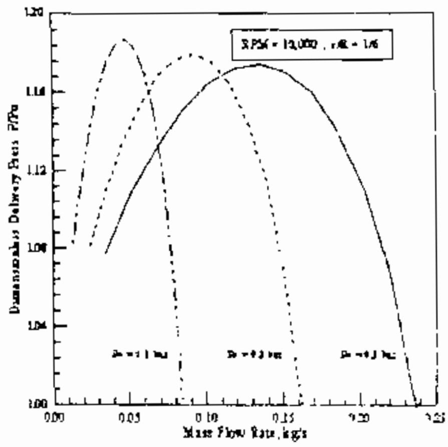


Figure (11) Effect of suction pressure on the pressure at exit of collecting passage

Figures (12-15) show the effect of compressor size on its performance. The pocket size is taken constant (r = 0.05 m) while the mean radius of impeller is varied, corresponding to the pocket to impeller radius ratio of 1/6, 1/5, ¼, 1/3, as 0.30, 0.25, 0.20, 0.15 m; respectively. Figure (12) shows delivery pressure versus degree of filling ϕ. As it is shown in figure, the value of pressure increases for larger impellers at all values of degree of filling but this increase is not significant for higher values of ϕ. Figure (13) shows delivery steam velocity (c/c_2) versus degree of filling ϕ. The effect of impeller diameter seams to be very small, especially, at smaller values of ϕ. Figure (14) shows the pressure against the mass flow rates. As it is clear, the pressure increases as the impeller diameter increases. In figure (15), the relation between the steam delivery velocity versus steam mass flow rate is presented. According to this figure, the velocity increases linearly with increasing mass flow rate for all values of impeller radius R. As it is clear, the delivery velocity has a considered value compared with that acquired in the impeller especially at higher values of flow rate. Using relations similar to that presented in figure (11-15), one can predict the proper size of impeller to satisfy specified operating conditions. In some operating conditions, as it is previously mentioned, the delivery velocity is more than that is required. In this case, an externally attached diffuser may be the proper solution to convert some of steam kinetic energy to enthalpy at higher pressure

5. Conclusions

The theoretical model presented in this work proposes a satisfactory tool to predict the performance of steam compressor of regenerative-turbine type under certain operating conditions (suction pressure, speed...etc.). In addition, one can according to this model estimate the proper size of compressor to satisfy specified operating conditions. It is recommended to extend this model to take in account the effect of different expected losses. An experimental model, perhaps, is recommended to be carried out to examine the validity of this model and to decide whether it requires some modifications or corrections to improve its results. This comparison is impossible to be presented in this work because of the rarity of publications concerned with this type of compressors.

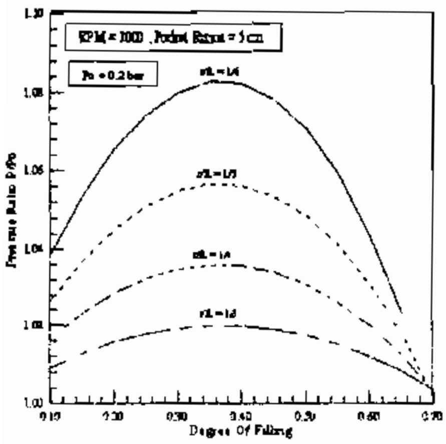


Figure (12) Effect of the ratio between pocket and mean radius on the dimensionless pressure at the exit of collecting passage

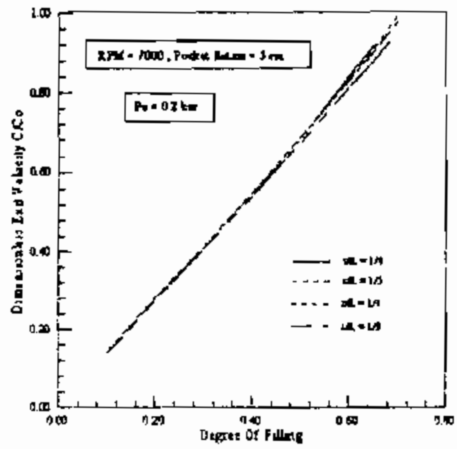


Figure (13) Effect of the ratio between pocket and mean radius on the dimensionless velocity at the exit of collecting passage

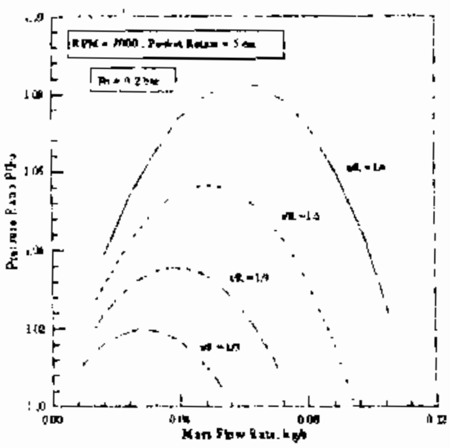


Figure (14) Effect of the ratio between pocket and mean radius on the dimensionless pressure at the exit of collecting passage

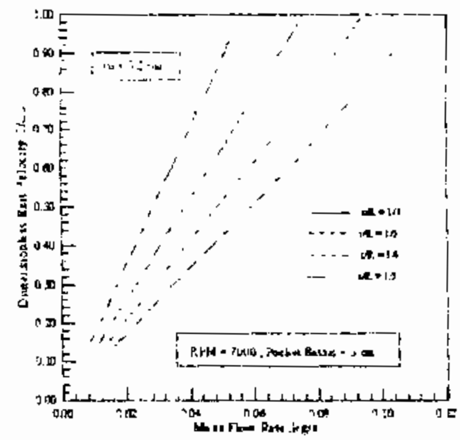


Figure (15) Effect of the ratio between pocket and mean radius on the dimensionless velocity at the exit of collecting passage

Nomenclature

A	cross-sectional area of collecting passage
c	velocity of steam in collecting passage
c_o	acquired velocity of steam due to passing through impeller
D	mean diameter of both impeller and collecting passage
h	enthalpy of steam in collecting passage
h_o	enthalpy of steam at suction port of compressor
i	identifier to denote the position along the passage
k	total number of impeller pockets
m	steam mass flow rate associated with passage
m_o	steam mass flow rate per unit length of collecting passage
N	speed of impeller in rpm
p	steam pressure

p_o suction pressure
R mean radius of impeller
r radius of impeller pockets
t mean thickness of pockets wall
t_1 tip length of impeller pockets
t_2 tip length of collecting passage
V total volume of impeller ring
V_s net volume of impeller ring, which occupied by steam
x length along the collecting passage

Greek Symbols

δ denotes the change of certain variable
ϕ degree of filling
ρ density of steam in collecting passage
ρ_o density at suction of compressor

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