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Abstract

Peripheral pumps are often applied on refrigeration and circulation mixtures in refrigeration systems.

These turbomachines are identified by the application of two flows on meridional motion and is a superimposed tangential one; this complex toroidal flow is not readily amenable to detailed theoretical analysis because of the considerable flow separation in the impeller blading.

The paper deals with a theoretical and experimental approach to carry out a non-dimensional correlation relating pressure and flow to driving torque, taking into account also the important contribution of the eddy viscosity.

The overall performance data on specific units was used to improve the efficiency and to optimize the impeller configuration.

RECHERCHE THEORIQUE ET CARACTERISTIQUES DE LA PERFORMANCE.

RESUME : Des pompes sont souvent utilisées dans l'industrie du froid et pour les mélanges en circulation dans les systèmes frigorifiques.

Ces turbomachines sont caractérisées par l'application de deux écoulements sur le déplacement méridien et un écoulement tangentiel superposé ; cet écoulement toroïde complexe n'est pas facilement susceptible d'une analyse théorique détaillée en raison de la séparation considérable des écoulements dans les aubes de la turbine.

Ce rapport traite de l'étude théorique et expérimentale pour obtenir une corrélation sans dimensions entre la pression et le débit d'une part et le couple d'entraînement d'autre part, en tenant compte aussi de l'importante contribution de la viscosité tourbillonnaire.

Les résultats généraux de la performance d'appareils particuliers ont été utilisés pour améliorer le rendement et optimiser la configuration de la turbine.

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ABSTRACT

Peripheral pumps are often applied on refrigeration mixtures circulation in refrigeration systems. These turbomachines are identified by the application of two flows: on meridional motion is superimposed a tangential one. This complex toroidal flow is not readily amenable to detailed theoretical analysis because of the considerable flow separation in the impeller blading.

The paper deals with a theoretical approach able to study the phenomena of energy exchange without the introduction of empirical constants and useful to plan an experimental investigation in order to improve the efficency.

NOMENCLATURE

Symbols

- A -Output area on control volume, cross section area
- B -Input area on control volume
- c -Absolute velocity of fluid
- **n** -Perpendicular versor
- M -Torque
- P -Power
- Q -Volume flow rate
- r -Radius
- w -Relative velocity of fluid in impeller
- Y -Specific energy
- α -Angular extension of channel

Γ -Transferred energy parameter
Φ -Stress
η -Efficiency
Π -Power loss
θ -Angular extention of channel
σ -Normal stress
ω -Speed of rotation

Subscripts

1 ,2 -refered respectively at the inlet and outlet sections
c -in the channel
cs -refered to control surface
g -of the impeller
G -medium of channel
h -hydrodinamic
m -meridional
n -normal
o -mechanical
s -of the stage
u -tangential

INTRODUCTION

Periphery machines are classified as turbomachines according to their functional characteristics; they are used in the boundary area betwen volumetric pumps and radial flow pumps and can function at high pressure coefficient with very low specific speeds. Despite the very high head which can be obtained, its application has been limited by a low total efficiency.

This paper deals with the exchange mechanism for energy transfer; this makes it possible to analyse the real possibilites of developement of these machines and the way to achieve appreciable improvements. The periphery pump stage consists of a rotor which may be considered as being similar to a centrifugal impeller with axial intake and outake, and by an anular channel shaped stator with inlet and outlet of the flow rate (Fig.1) The resulting flow is toroidal. half in the impeller end half in the channel. The open channel which closes to a small clearance from the sides and the tip of the rotor, make it possible to obtain the dynamic seal between high and low pressure regions. In view of the working principles described above it is not possible to apply traditional modelling methods used for conventional turbomachines. According to the authors, the fundamental difference of periphery pumps with respect to turbomachines is the presence of two main circulations. The first passes through the impeller in centrifugal direction and through the channel in centripetal direction. The second flows in a tangential direction and determines the flow rate of the pump. The effect of these two circulations, which have different flow rates, makes it possible to study the periphery pump as a turbomachine-ejector system in which the turbomachine operates at a very low reaction number. According to this scheme (see Fig. 2) the working exchange occours in two phases: the meridional flow rate passes through the impeller and enters the channel where it diffuses and exchanges momentum (this phenomenon is illustrated by the ejector in the scheme); the study of this mixing mechanism between the two flow rates is the fundamental operating principle of the machine.

BASIC ANALYSIS

The study of operation principles, may be carried out, according to the scheme proposed on the bases of the following hypotesis

1) Steady flow

2) Constant density

3) Tangential pressure gradient is constant throughout the channel and is indipendent of radius

4) The meridional flow rate is indipendent from overall Δp

5) The radial distribution of the absolute speed in the channel does not vary in tangential direction

The analyses of the working principles for the impeller and the channel may be carried out separately

Impeller flow

A single equation of the dynamic equilibrium may be used to describe the exchange of energy between the fluid and the impeller.

The equation indicated in appendix refers to equilibrium of the moments of the impeller control volume with respect to the rotation axis.

$$M_g = \rho Q_m (r_2 c_{u2} - r_1 c_{u1}) \qquad (1)$$

This equation shows that the torque which operates on the impeller is not caused by the variation of angular momentum of the pump flow rate Q_s but by the meridional flow rate Q_m which on the basis of the above hypoteses, does not vary as the operating conditions of the pump vary

Excluding the shear stress, the power transferred by the impeller to fluid is espressed by

$$P_g = M_g \omega = \rho\, Q_m\, \omega\, (r_2 c_{u2} - r_1 c_{u1}) \qquad (2)$$

It follows that this power increases as c_{u1} decreases as the pump flow rate decreases

Channel flow

As in the case of impeller, the fluid dynamic behaviour in the channel may be described by an overall equation of dynamic equilibrium (see appendix)

The control volume in this case corresponds to the volume inside the channel.

$$\Delta p\, r_G\, A_c = \rho Q_m\, (r_2 c_{u2} - r_1 c_{u1}) \qquad (3)$$

$$Y_s = Q_m / r_G\, A_c\, (r_2 c_{u2} - r_1 c_{u1}) \qquad (4)$$

Equation (4) describes the tipical curve of the machine excluding the loss due to the shear stress on the sides of the channel

The component c_{u1} is stictly correlated to the flow capacity Q_s, in fact, is the result of radial distribution of the tangential component of the absolute speed. The pump flow rate tends to a maximum value when the work of the impeller is nil, that is when $r_2 c_{u2} = r_1 c_{u1}$.

It assumes a value of zero when $c_{u1} = -c_{u2}$ (Fig. 4) and the exchange of energy for diffusion is at maximum. In this case there is a closed tangential circulation in the channel between the cilindrical section which divides the channel radially. On the basis of (4) the power transferred to the fluid may be expressed by:

$$P_h = \rho Q_s Y_s = \rho Q_s Q_m / r_G A_c (r_2 c_{u2} - r_1 c_{u1}) \qquad (5)$$

The ultimate goal of the scheme proposed is to define the parameter $\Gamma = P_h / P_g$ as transferred energy parameter which can be assumed as a measure of the energy transferred to the flow rate inside the channel

$$\Gamma = Q_s / \omega r_G A_c \qquad (6)$$

The function Γ is a right intersecting line on the origin of axis . From (6) it is evident that the maximum of the transfer function would be obtained when the the tangential component in the channel varies linearly with respect to the radius and is equal to the velocity of the blades.

ANALYSIS OF THE PUMP FLOW RATE

The overall efficiency of a periphery machines, results, as in the case of all turbomachines, from the mechanical efficiency produced for the hydraulic efficiency.

Since there are two energy transfer processes, the latter will be supplied from the efficiency of the impeller times the transferred parameter (Γ).

The mechanical efficiency η_0 depends on the "mechanical" quality and on the braking action of the fluid on the impeller disk, it is very close to one and does not vary as the flow rate Q_s varies. The impeller efficiency on the other hand depends on the flow of the fluid in the impeller and can be expressed as $\eta_g = P_g / (P_g + \Pi_g)$ where the term Π_g consider the power dissipated on the boundary layer flow on the sides of the blades and on the sides of impeller disk opposite the channel in addition to blade entrance losses

The losses resulting for friction do not vary as Q_s varies since the friction is proportional to Q_m^2 which has been considered as constant, the blade entrance losses are proportional to the variation of relative speed at the blade inlet and increase in importance when the machine operates in proximity of the limit $Q_s = 0$ and $Q_s = Q_{MAX}$ of the performance characteristic The inlet speed triangle in fact deforms very considerably as Q_s varies since, as it increases, the tangential circulation increases until the c_{u1} must necessarily change versus, thereby forcing the incidence angle of speed to increase quite considerably (Fig.4)

The efficiency of the impeller assumes an approximately parabolic pattern with a maximum correspondig to a particular c_{u1} which is equal to zero when the impeller has radial blades as in the case which has been illustrated The maximum condition of the overall efficiency is obtained by combining the patterns of the curves $\eta_g(Q_s)$ and $\Gamma(Q_s)$ (See Fig 5)

If the actions of the shear stress are considered, the function derivative of $\Gamma(Q_s)$ decreases as the flow rate increases because of the power dissipation on the sides of the channel that is proportional to Q_s^2. The pattern of the overall efficiency is strongly affected by the pattern of the impeller efficiency which depends mainly on the value of blade entrance losses, above all for identifying the value of the flow rate corresponding to the maximum total efficiency

CONCLUDING REMARKS

The optimization of the flow in the impeller is not sufficient to increase the efficiency of the machine with respect to low flow rates because, at this conditions, the overall operating efficiency is very strongly affected by the diffusion efficiency which become very low for low values of Q_s.

An increase of the overall efficiency values is however possible when the inlet angle is optimized for very high values of Q_s.

APPENDIX

Impeller

With reference to the Fig. 3, when a control volume has been selected for the impeller, the equation of moment of momentum for the rotation axis may be written as follow:

$$\int_{cs} \mathbf{r} \times \mathbf{\Phi}\ \mathbf{n} ds + \int_{cs} \rho \mathbf{r} \times \mathbf{c}\ \mathbf{c}\ \mathbf{n} ds + \mathbf{M}_g = 0 \qquad (7)$$

As a result of the axial symmetry of the rotor, taking into account that the normal stress have not moment and negletting the moment due at shear stress, the equation reduces to.

$$\mathbf{M}_g = -\int_A \rho \mathbf{r} \times \mathbf{c}\ \mathbf{c}\ \mathbf{n} dA - \int_B \rho \mathbf{r} \times \mathbf{c}\ \mathbf{c} \mathbf{n} dB \qquad (8)$$

The integration surfaces A and B are those involving the inlet and outlet momentum flux of the impeller respectively, so that $\rho c_n dA = \rho\ c_n dB = \rho\ dQ_g$ where Q_g is the the impeller rate corresponding to the meridional flow rate Q_m By replacing the medium effective value $\int_{cs} \mathbf{r} \times \mathbf{c}\ ds$ with $r\ c_u$, we can write in scalar form:

$$M_g = \rho Q_m (r_2 c_{u2} - r_1 c_{u1}) \qquad (9)$$

When the medium effective values of the product $r\ c_u$ are the real values, the equation represents the condition of dynamic equilibrium for the impeller excluding the action resulting from the shear stresses on the impeller and on the surface of the interface with the channel which may be negletted considering the geometry examined.

Channel

The equation of moment of momentum applied to the control volume which is illustrated, is written as follows:

$$\int_{cs} r \times \Phi \ \mathbf{n} ds + k \int_{cs} \rho r \times c \ c \ \mathbf{n} ds = 0 \qquad (10)$$

By negletting, in addition, the shear stresses, we obtain:

$$\int_{Ac} r \times \sigma \ \mathbf{n} dA_c = - \int_A \rho r \times c \ c \ \mathbf{n} dA - \int_B \rho r \times c \qquad (11)$$

As in the previous case with the real medium effective value $r c_u$, we can write the following equation in scalar form :

$$\Delta p \ r_G \ A_c = \rho Q_m (r_2 c_{u2} - r_1 c_{u1}) \qquad (12)$$

The distribution of pressure in a tangential direction is linear to the angle α measured from the inlet section of the channel. In fact, by selecting a section of the channel subtended by the angle α as the control volume, the condition of equilibrium for the moments is written as follows:

$$\Delta p \ r_G \ A_c = \rho Q_m \alpha / \theta (r_2 c_{u2} - r_1 c_{u1}) \qquad (13)$$

θ is the angle at the centre which subtends the channel

The above considerations are in aggrement with experimental findings

Determination of the tangential components of absolute speed

In all the equations considered, the difference $r_2 c_{u2} - r_1 c_{u1}$ appears, and since the radial dimensions of the machine which are rather limited if compared with the average radius of the reference toroid, it is possible to consider the distribution of the tangential component of absolute speed in the respective areas of action A and B as constants and to calculate the products with the average radius of the outlet and inlet surface of the impeller

The law of continuity must be applied to the inlet and outlet areas A and B of the impeller and the speed inversion radius must ensure that $\int_A c_m \alpha r \, dr = \int_B c_m \alpha r \, dr$ with $c_m (r_1) = -c_m (r_2)$

This condition is satisfied by a parabolic distribution of $c_m (r)$

After the meridional flow rate Q_m has been fixed, the inversion radius should be calculated and then the areas of the impeller-channel interface flow surfaces.

According to the outlet triangle it is possible to calculate a medium effective value of c_{u2} according to the relative flow angle and by a slip factor which considers the finite number of blades .

The medium effective value of c_{u1} may be calculated considering the following three conditions.

a) the value of c_{u1} is zero when $c_{u2} A_2 = c_c A_c = Q_s$

b) Q_s is zero when $c_{u2} A_2 = -c_{u1} A_1$

c) Q_s is maximum when $c_c = c_{u2}$

d) then $c_{u1}(Q_s) = (Q_s - c_{u2} A_2) / (A_c - A_2)$

As a first approssimation, the values of c_{u2} and c_{u1} are those calculated for the solution of the integrals in moment of momentum equations.

To establish the value of the meridional flow rate, it is necessary to obtain direct experimental results, or by means of the value of specific energy measured with flow rate as zero. In fact, by using the above condition d) in (4), the specific energy of flow rate becomes:

$$Y_s = \frac{Q_m}{r_G A_c \left(r_2 c_{u2} - r_1 \dfrac{Q_s - c_{u2} A_2}{A_c - A_2} \right)} \qquad (14)$$

which, for $Q_s=0$, makes it possible to calculate Q_m

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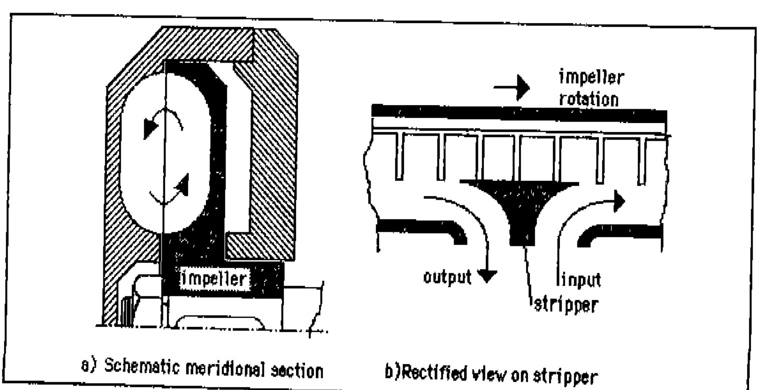

Fig. 1 Geometry of peripheral pump

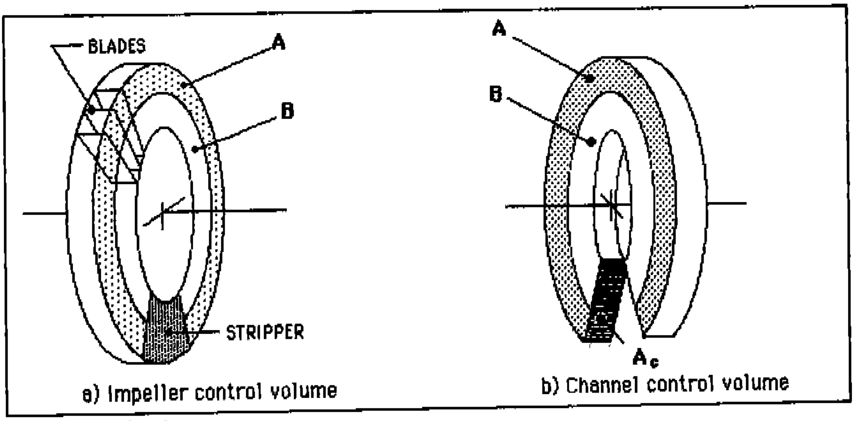

Fig.3: Control volumes

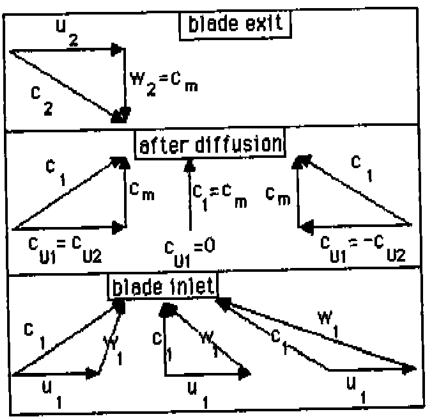

Fig. 4 Velocity triangles

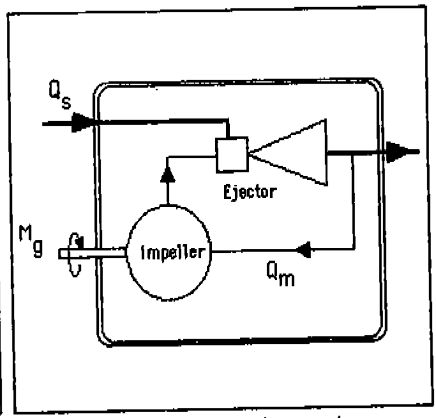

Fig.2 Turbomachine-ejector system

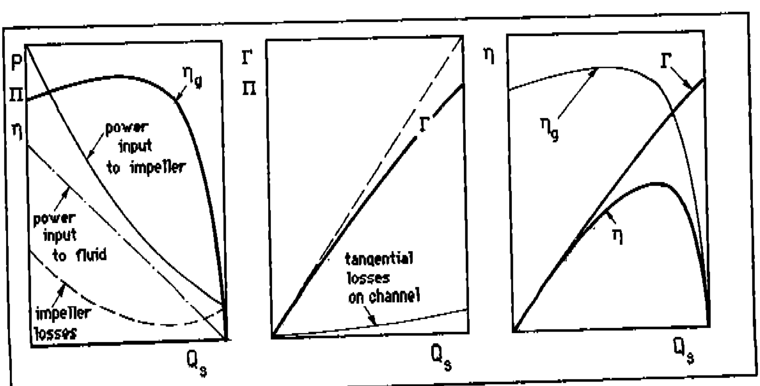

Fig. 5: Theoretical powers, losses and efficiencies trends versus Q_s