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A COMPARISON OF REGENERATIVE AND CENTRIFUGAL COMPRESSORS

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SUMMARY

The performance of regenerative and centrifugal compressors are compared and applied to a design problem. It is found for the problem considered that the regenerative compressor is superior in all aspects except thermal efficiency. The case for the regenerative compressor stands on cheaper manufacture, fewer stages, lower stresses, easier transmission, and less maintenance.

NOMENCLATURE

C_2	fluid radial velocity at exit from impeller (m/s)
D_2	rotor tip diameter (m)
g	acceleration due to gravity (m/s^2)
H	head (m)
n	rotational speed (revs. per second)
Q	volume flow rate (m^3/s)
u_2	rotor tip velocity (m/s)
β_2	impeller tip blade angle
ϕ	velocity coefficient, $\frac{C_2}{u_2}$, dimensionless
η	efficiency
Ω	capacity coefficient, $\frac{Q}{D_2^2\, u_2}$, dimensionless
ψ	head coefficient, $\frac{gH}{u_2^2}$, dimensionless
θ	specific speed, $\frac{n\, Q^{\frac{1}{2}}}{(gH)^{3/4}}$, dimensionless

INTRODUCTION

In the last twenty years rotary compressors have tended to supersede reciprocating compressors in many industrial applications. However, the rotary compressors have in general been of the centrifugal type. Regenerative compressors have been used rarely, basically because of lack of knowledge of their performance. It has been generally understood that they give low flow rates at high heads and, as a consequence, regenerative compressors have been limited to use as catalyst pumps in the chemical industry or for lubrication or control.

It is necessary to decide on the significance of the terms low flow rate and high head. In the gas turbine industry 50 kg/s of air would be a low mass flow rate and 15 atm. would be a high pressure rise through the compressor. In the chemical industry 50 kg/s of a gas could be an extremely high mass flow rate and 200 atm. could be a low pressure rise. In the case of gas turbines the compressors are predominantly of the axial type whereas in the chemical industry they are either reciprocating or centrifugal. Specific speed is a better means of classifying rotary compressors. Typically axial compressors have a specific speed of about 0.5, centrifugal compressors 0.1 and regenerative compressors 0.03. These values of specific speed may appear to bear out the comments relating flow rate and head, but it should be remembered that the specific speed also involves rotary speed and so indirectly geometry, stresses, vibrations etc. That is, the specific speed is a more universal design parameter than flow rate and head. In compressible situations the compressor performance is described in terms of a pressure ratio, a non-dimensional mass flow criterion and a non-dimensional rotational speed criterion, whereas for pumps the corresponding performance criteria are head coefficient, capacity coefficient and specific speed.

In the chemical industry compressors are used for pressurising relatively heavy and incompressible gases. Thus when dealing with these gases it is preferable to use pump characteristics though, of course, cavitation limits do not exist. Compressors having similar design features are found to have approximately the same specific speed, head coefficient and capacity coefficient in widely different applications, for example, in hydraulic and pneumatic applications. In deciding if rotary compressors are applicable for a particular application and in choosing between different rotary compressors dimensional design considerations are necessary. It may be that for a particular job specification in order to keep rotor speed down and tip diameter up to acceptable levels the number of stages required for a centrifugal compressor may be excessive. Alternatively in the case of a regenerative compressor its inherent inefficiency may override the practical advantages of fewer stages and lower rotor speed for a given rotor tip diameter. Thus it is necessary to consider compressor geometry, speed, etc. before deciding which is the most practical unit.

CENTRIFUGAL COMPRESSORS

The performance of centrifugal compressors has been discussed extensively in the literature. At present it is sufficient to reproduce optimum performance curves, see Figure 1. These performance curves were obtained from (1), Figures 5.1 and 6.12.

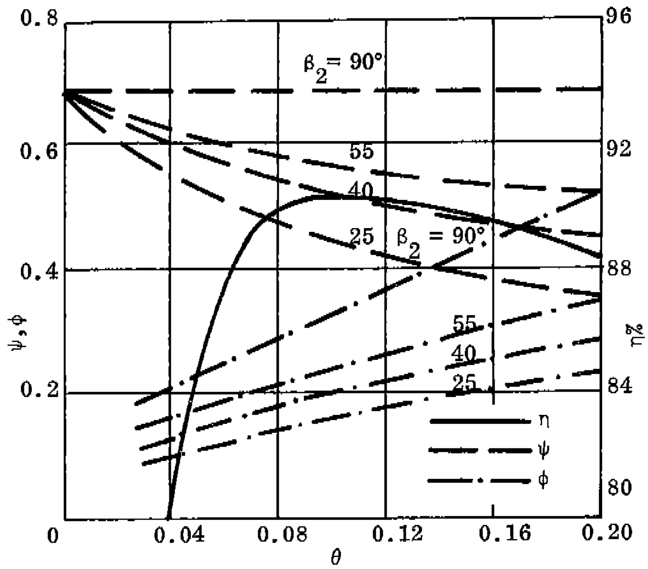

Figure 1 Optimum Impeller Characteristics

REGENERATIVE COMPRESSORS

The literature available on the regenerative machine is limited and deals exclusively with pumps. Also, some controversy has existed about the mechanism of the pumping action. (2, 3) Senoo has put forward a theory on the fluid-dynamic mechanism which is that the motion of a rough surface in a channel containing fluid causes the fluid to be dragged along by the rough surface, the fluid head being increased in the direction of flow. (4)Iversen supported this theory and his analysis based on it gave reasonable predictions of the shapes of performance curves. However, Senoo's theory was discounted by others. (5) Wilson, Santalo and Oelrich suggested an alternative theory which accounts for the observed performance curves equally as well. Their theory is fundamentally different from that of Senoo, they take account of the circulatory motion in the fluid induced by centrifugal forces which causes the fluid to circulate repeatedly through the rotor vanes and the annular channel, see Figure 2. The circulation effectively causes internal multistaging, hence the term 'regenerative flow'.

(6) Senoo in a later paper compared the two theories and concluded that they are compatible in principle.

(7) Shimosaka and Yamazaki suggest that a turbulent flow is stimulated along the annular channel caused by the difference in angular momentum at the tip and root of the rotor vane grooves and hence the pumping action is increased. The fluid in a vane groove flows into the annular channel from the vane tip and returns to the root of the vane. This action occurs at each vane groove simultaneously and the impact mixing and turbulence in the channel is large so raising the pressure of the fluid.

(8) Csanady suggests that the available experimental evidence shows that a secondary circulation is set up as shown in Figure 2. The same fluid particles flow outwards along the rotor vanes several times in one revolution, the effect being similar to the action of several rotors in series. Csanady suggests that the resultant fluid particle path is toroido-helical.

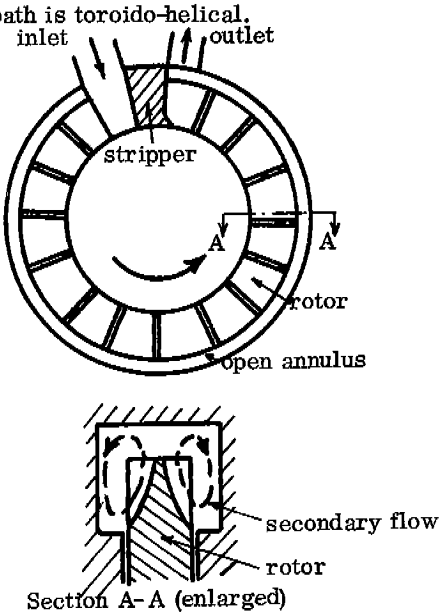

Figure 2 Regenerative Compressor

(5) Wilson, Santalo and Oelrich plotted both experimental and calculated performance curves of a regenerative compressor applicable to air and water. Their results have been used to produce the regenerative compressor performance curves illustrated in Figure 3.

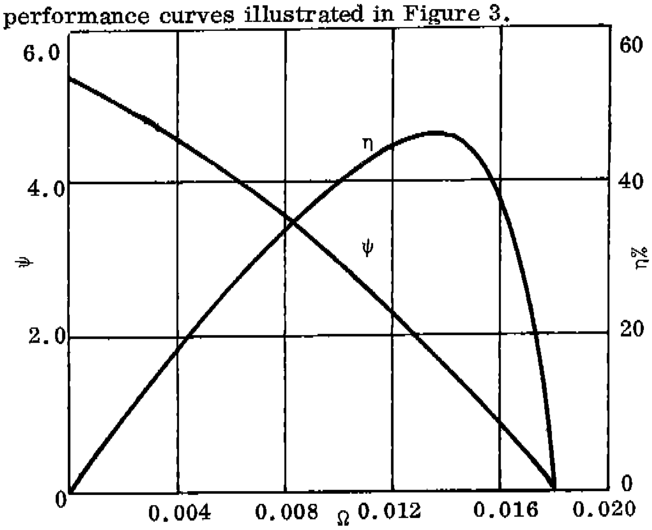

Figure 3 Regenerative Compressor Characteristics

COMPARISON OF CENTRIFUGAL AND REGENERATIVE COMPRESSORS

From Figure 1 the maximum optimum efficiency of a centrifugal impeller is 90.25% and the associated specific speed is 0.685. If the blade angle at the tip of the rotor is 90° the head coefficient is 0.685, whereas, for a blade angle of 30° the head coefficient is 0.47. From a blade stressing standpoint 90° blade angle is better than 30° but from diffuser fluid mechanic considerations the reverse is true. Figures 4 and 5 for 90° and 30° blade angle respectively are obtained from Figure 1 and are dimensional characteristics of centrifugal impellers. As head (m^2/s^2) and volume flow rate (m^3/s) are used these two graphs are independent of fluid properties, that is, they are applicable to all fluids.

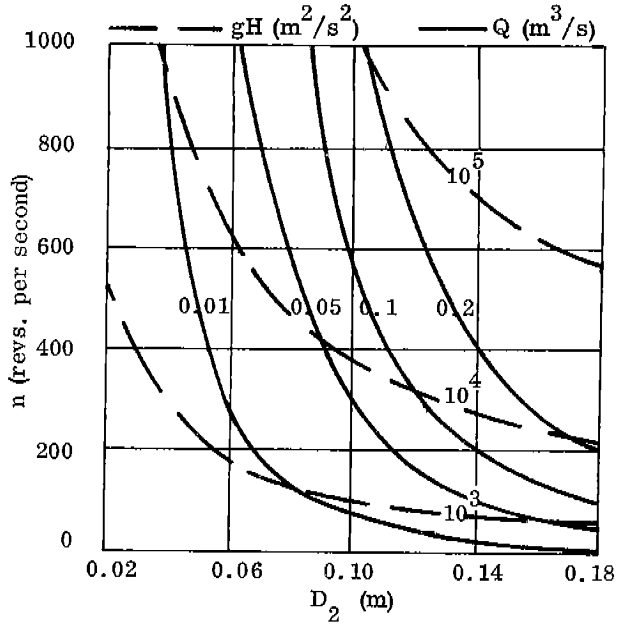

Figure 4 $\beta_2 = 90°$, $\eta = 90.25\%$, $\theta = 0.1$, $\psi = 0.685$

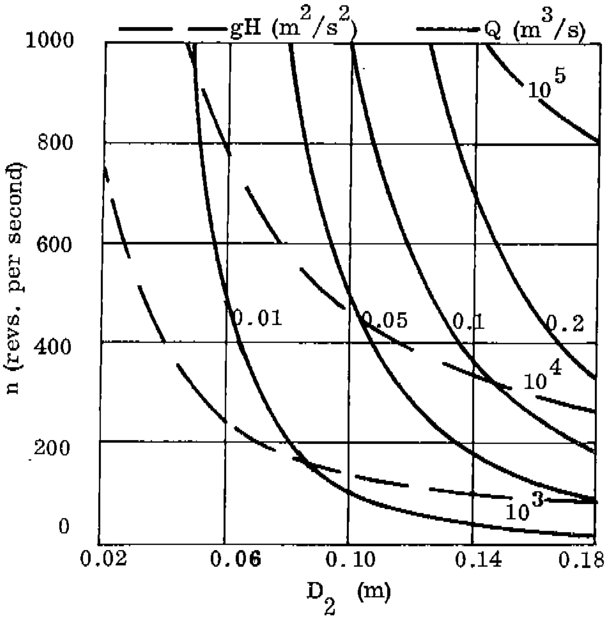

Figure 5 $\beta_2 = 30°$, $\eta = 90.25\%$, $\theta = 0.1$, $\psi = 0.470$

From Figure 3 the maximum efficiency of the regenerative compressor tested by Wilson, Santalo and Oelrich is 45%, the associated head coefficient is 1.5 and the capacity coefficient is 0.014. The corresponding specific speed is 0.0244. As in the case of centrifugal impellers the performance characteristics of the regenerative compressor can be plotted dimensionally, see Figure 6. A physical comparison of centrifugal impellers and regenerative compressors can be obtained from Figures 4, 5 and 6.

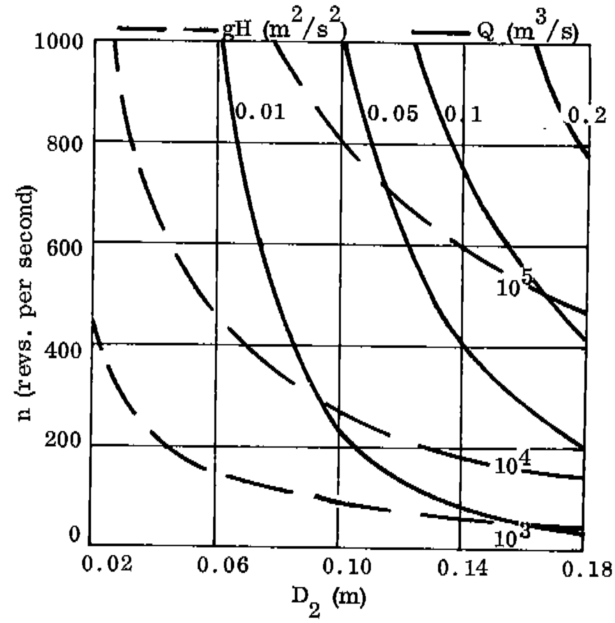

Figure 6 $\eta = 45\%$, $\Omega = 0.014$, $\psi = 1.5$

At the present time blade stresses in centrifugal compressors are not a major worry and, therefore, it is reasonable to design for good fluid mechanics, low hub stresses and ease of manufacture. This is best done with a small tip blade angle, say 30°. At small blade angles the radial velocity component of the fluid at exit from the impeller is small making diffuser design easier. Comparing Figures 4 and 5 it can be seen that for a given head and flow rate the rotational speed is the same for 90° and 30° blade angle but the tip diameter is substantially greater for the 30° case. This means that the hub diameter will be larger for the 30° case and so the compressor will be more able to transmit the power required. Also when considering the chemical industry because of relatively low volume flow rates and high heads, rotary compressors have small tip diameters. Thus for manufacturing reasons it is better to design with small blade angles. It is proposed therefore to compare a 30° tip blade angle impeller with the regenerative compressor.

In the chemical industry relatively incompressible gases with high density, of the order of water, and high pressure, perhaps about 1 000 bar, are used frequently. Also, mass flow rates about 50 kg/s would be common. Consider a flow rate of 0.1 m^3/s and a head of 10^5 m^2/s^2. The corresponding mass flow rate for a gas of density 500 kg/m^3 is 50 kg/s and the pressure rise is 500 bar.

Extrapolating the 0.1 m^3/s and 10^5 m^2/s^2 curves in Figure 5 indicates that for a centrifugal compressor the impeller rotational speed is 1 600 revs. per second and the tip diameter is 0.9 m. From Figure 6 the corresponding rotor speed and tip diameter of the regenerative compressor are 500 revs. per second and 0.17 m. Clearly the regenerative compressor is slower running and diametrically larger and therefore stronger than the centrifugal compressor. For the case considered the power used for compression in either compressor is 5 MW and therefore, the power required to drive the centrifugal impeller is 5.54 MW and for the regenerative compressor 11.11 MW as the respective efficiencies are 90.25% and 45%. The tip diameter of the centrifugal impeller is 0.09 m and so it is improbable that the compressor shaft would be greater than 0.03 m giving a maximum stress in excess of 200 MN/m^2, a substantial stress. The shaft diameter of the regenerative compressor could be two or three times that of the centrifugal compressor and, even though the rotational speed is lower and the power required higher, the maximum stress would be considerably less than 200 MN/m^2.

A rotational speed of 1 600 revs. per second for a centrifugal compressor is excessive, creating transmission and many other problems. If the centrifugal compressor runs at the same speed as the regenerative compressor, that is, 500 revs. per second, it can be seen from Figure 5 that four stages would be required to achieve a head of 10^5 m^2/s^2 with a flow rate of 0.1 m^3/s. The impeller tip diameter would increase from 0.09 m to 0.125 m improving slightly on the shaft stressing problem. It may be decided that 500 revs. per second is too high a speed and then one stage would be inadequate for the regenerative compressor. For example, 300 revs. per second would mean a two stage regenerative compressor or an eleven stage centrifugal compressor. Thus from manufacturing, running and maintenance considerations the regenerative compressor has much to offer. The number of stages required in a centrifugal compressor could be reduced by designing an optimum impeller which was not the maximum optimum. Figure 7 is generated from Figure 1 for an optimum impeller with specific speed 0.05, head coefficient 0.55 and efficiency 85%. For the case being considered and a speed of 300 revs. per second the number of stages required is five. The same thing is possible with the regenerative compressor but the cost is an even lower efficiency, see Figure 8.

CONCLUSIONS

The regenerative compressor has found little use in industry and yet its performance makes it attractive for a number of applications. Its basic failing is a low efficiency. However, the efficiency of the compressor tested by Wilson, Santalo and Oelrich could be improved on by profiling the rotor vanes and perhaps by modifications to the geometry of the annular channel and diffuser. Efficiencies in the region of 90% may not be possible but a considerable improvement on 45% should be attainable. In the case of centrifugal compressors even though the impeller efficiency is about 90% the overall efficiency of a multistaged unit would be substantially lower.

The main advantage of the centrifugal compressor is that it requires a smaller prime mover and less fuel to run. In every other respect the regenerative compressor for application of the type discussed in the article is superior. Clearly, there is a case for much more research into regenerative compressors.

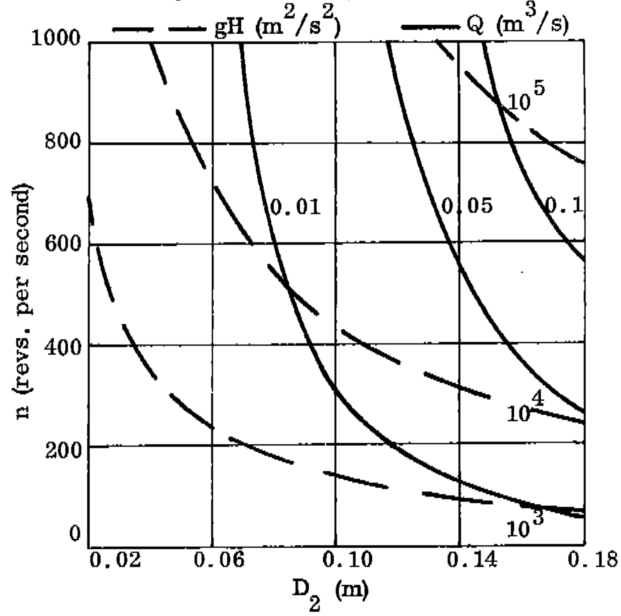

Figure 7 $\beta_2 = 30°$, $\eta = 85\%$, $\theta = 0.05$, $\psi = 0.550$

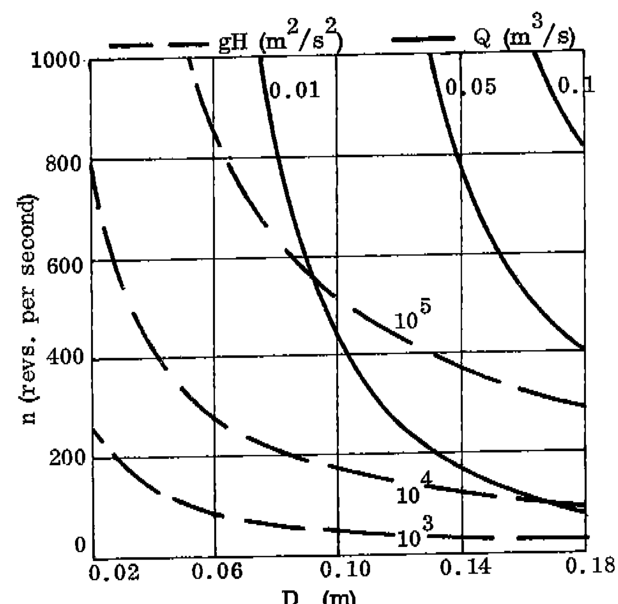

Figure 8 $\eta = 30°$, $\Omega = 0.007$, $\psi = 3.85$

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