7 Changing Pump Performance

In practice, it is often necessary to change the head pressure of a rotary pump whose measurement is already given. At the same time, the revolution speed is not changed. In general, an increase in the pump head pressure is not possible, or it is possible only to a small extent. This can be conducted by a one-side sharpening of the ends of vanes in order to increase the angle of emergence. It will be pointed out that this measure also means that the power consumption increases. If the vane has a sufficient radial extension, the head pressure can be reduced by shortening or hollowing-out the rotor. A change in the head pressure will at the same time lead to a change in the transmission flow.

If the pump works with guide vanes or with a guide ring (ring diffuser), vanes will be hollowed-out, whereas the rotor walls stop operating with the view to the unhindered water flow to the stator. In the case of spiral casing pumps it is better to machine-out the rotor walls too (Fig. no. 12).

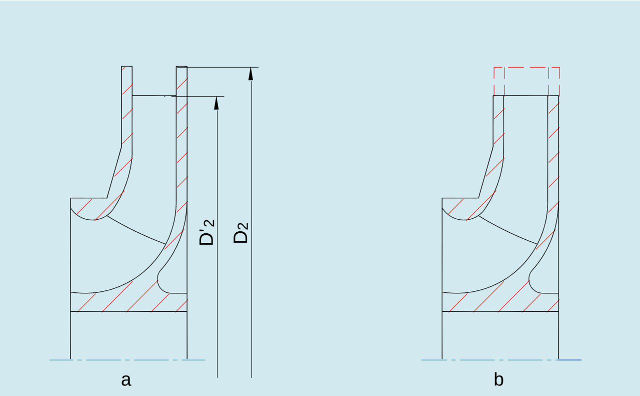


Fig. 12 Shortened and hollowed-out rotor

When determining the rotor's diameter that is adjusted to the new transmission conditions, the following can be assumed:

The delivery height H decreased exponentially with the diameter of the rotor D_2. The transmission flow Q changes in a linear way with the exhaust area of the rotor, for a constant rotor width also in linear way as the diameter D_2, and also linearly with the circumferential speed of the rotor exit that is proportionate to the diameter of the rotor. This leads to the conclusion that the delivery flow also changes exponentially when the diameter of the rotor decreases.

This results in the following equation:

$$\frac{H'}{H} = \frac{Q'}{Q} = \frac{D'^2_2}{D^2_2} \qquad (18)$$

H', Q' and D'_2 stand for the changed parameters. This equation (18) results in:

$$D'_2 = D_2 \cdot \sqrt{\frac{H'}{H}} \quad \text{or} \quad D'_2 = D_2 \cdot \sqrt{\frac{Q'}{Q}} \qquad (19)$$

The new diameter D'_2 can thus be derived as follows:

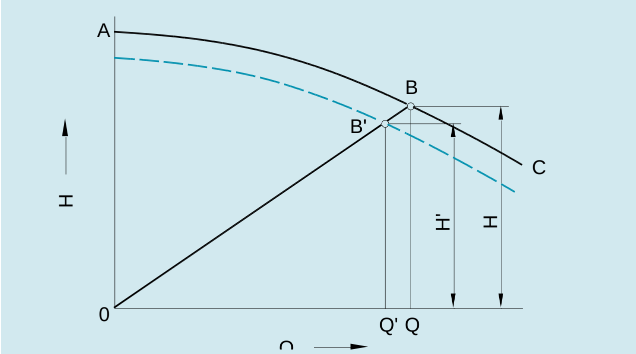


Fig. 13 Determination of the changed rotor diameter

In Fig. 13, AC is the original Q-characteristic of the pump that is regarded as a single-stage pump here; B' is the wanted operation point upon the change of the rotor diameter. We draw the right line OB' whose extension over B' meets the Q-characteristic AC in the point B. The points B' and B provide for the pump head pressures H' and H as well as for the transmission flows Q' and Q, from which the diameter D'_2 can be calculated in accordance with the equation no. 19.

The procedure described above gives only approximate values because some impacts that can hardly be assessed are not considered in this procedure. The inaccuracy increases with the increased deviation between D'_2 and D_2. This is why it is recommended not to fall below the diameter relation $D'_2/D_2 = 0.8$. In case of EDUR guide vane pumps, this restriction causes no difficulties because nozzle rings are graded very precisely.

Caution should be exercised in the case of rotors with small diameters because a relatively small shortening of the rotor disc diameter leads to a considerable reduction of the pump capacity. Due to the existing uncertainty, it is generally recommended to machine-out the rotor disc to a diameter larger than calculated, and subsequently to determine the final diameter after the reduction in performance has been determined.

What is important is the pump efficiency after the rotor has been machined-out. Generally, the increase in the distance between the rotor and the stator will lead to a loss in the degree of efficiency. In case of narrow rotors with large diameters, a modest reduction will lead only to a small reduction in the degree of efficiency, because the rotor frictional path and the friction at the side of the rotor are reduced.

By contrast, in the case of rotors that are designed for larger transmission flows at smaller pump head pressures, machining-out results in an immediate considerable decrease in the degree of efficiency as a result of the deterioration of water flow in the shortened vane channels. Another problem encountered in these rotors is the uneven vane length in the front and back cover disc. In order to avoid a disproportionately high loss in the pump capacity, it is advantageous to have the rotor on the back side with an external diameter smaller than on the front side. In the meridian profile, this will lead to a steep outlet edge, thus completing the transition to a half-axial rotor.

From time to time, it is necessary to increase the pitch of a Q-characteristic, for example in case of a pressure-related pump control in order to maintain the necessary pressure difference between the start-up and turn-off pressure. This change in the pump capacity can be obtained by means of a sharp aperture (Picture no. 14). It is built in the pressure fitting in a way that the flow is directed against the sharp edge of the aperture.

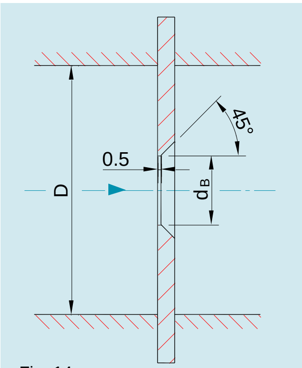


Fig. 14

An insertion of the aperture in the pressure line can lead to a considerable deviation in the head loss if the internal pipe diameter does not correspond with the D value stated in the diagrams. Furthermore, there must be a minimal distance of 2 pipe diameters behind a gauge or a pipe curvature.

The throttling caused by the aperture leads to a head loss H_V that is dependent on the aperture diameter d_B, the pipe diameter D and the flow rate Q.

Relations between the stated diameters are evident in the attached work sheet “Head loss H_v through throttling apertures”.

A Calculation Example

The Q-characteristic of a rotary pump presented in the picture no. 15 should be changed by building in an aperture at the pressure fitting of the pump in such a way that at Q = 15 m³/h, the pump head pressure is reduced from H = 33 m (Point A) to 23 m (Point B), that is to say by H_V = 10 m. The pressure fitting has a diameter of 32 mm.

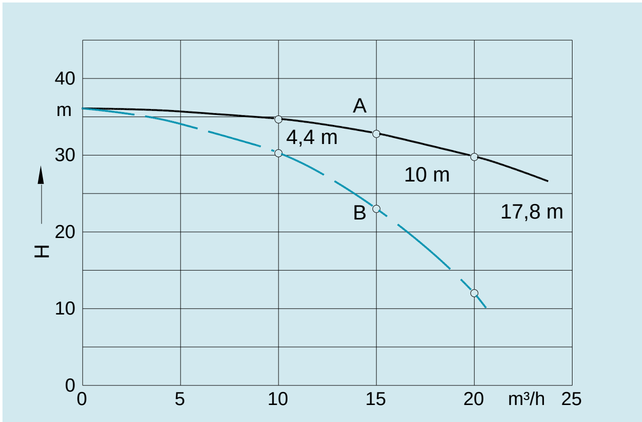


Fig. 15

In the diagram for DN 32 in the work sheet “Head loss H_V through throttling apertures” we can see for H_V = 10 m and Q = 15 m³/h an aperture diameter is 20.4 mm (linearly interpolated).

By building in an aperture, a pump obtains a new Q-characteristic. It can be calculated as follows:

If the initial values are Q_1 and H_{V1}, the sought variables Q_2 and H_{V2}, the result is:

$$H_{V2} = H_{V1} \cdot \left(\frac{Q_2}{Q_1}\right)^2 \qquad (20)$$

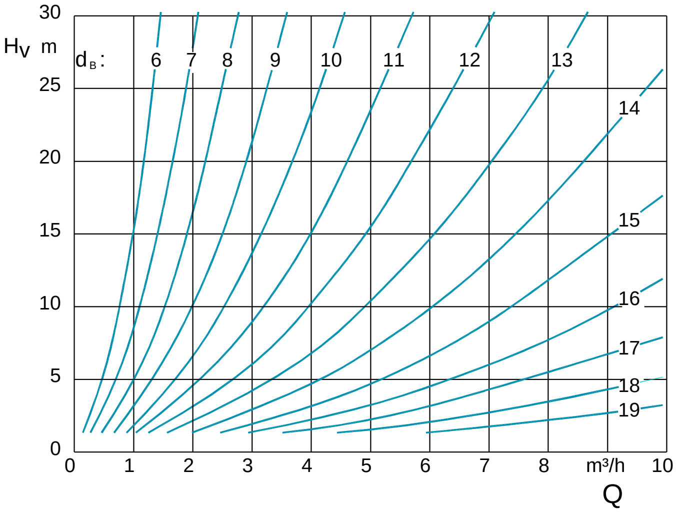
The loss at Q_2 = 10 m³/h amounts to:

$$H_{V2} = 10 \cdot \left(\frac{10}{15}\right)^2 = 4{,}4 \quad \text{m}$$

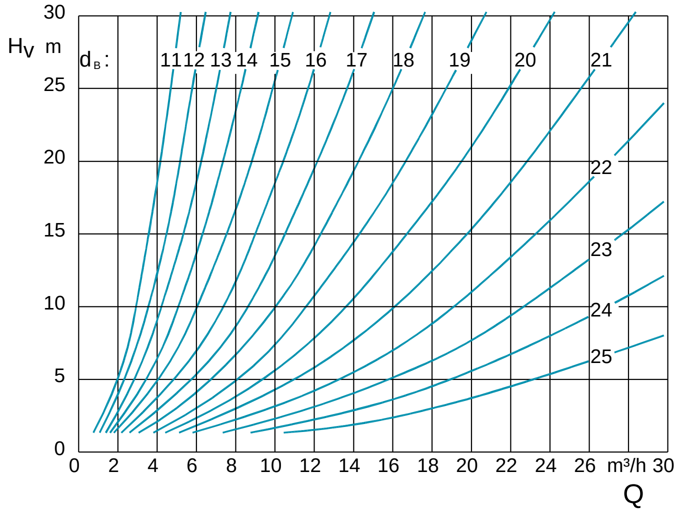
At Q = 20 m³/h:

$$H_{V2} = 10 \cdot \left(\frac{20}{15}\right)^2 = 17{,}8 \quad \text{m}$$

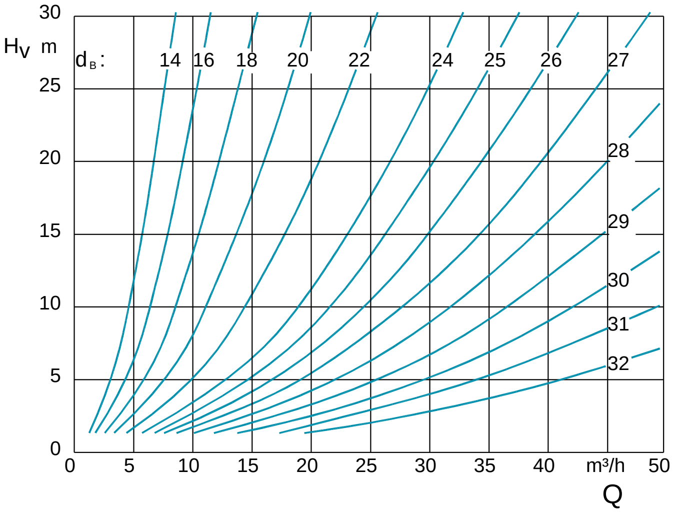
Pressure fitting DN 25 (D=25mm)



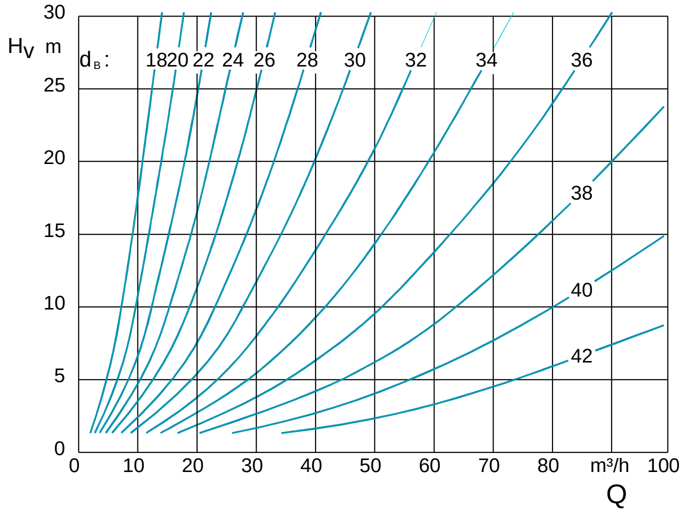
Pressure fitting DN 32 (D=32mm)



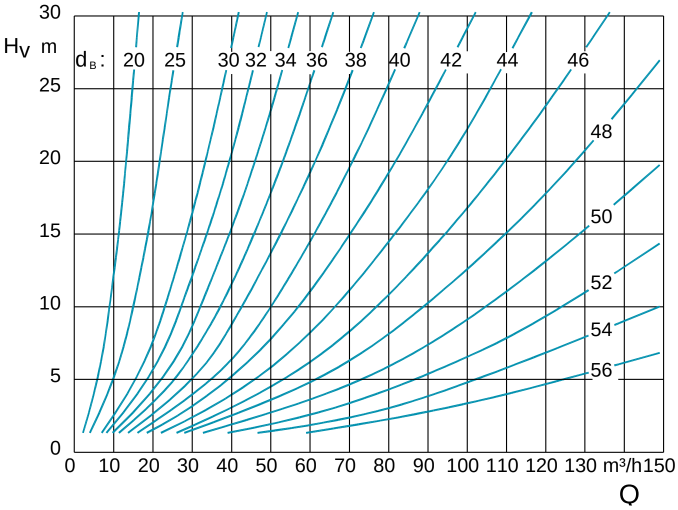
Pressure fitting DN 40 (D=40mm)



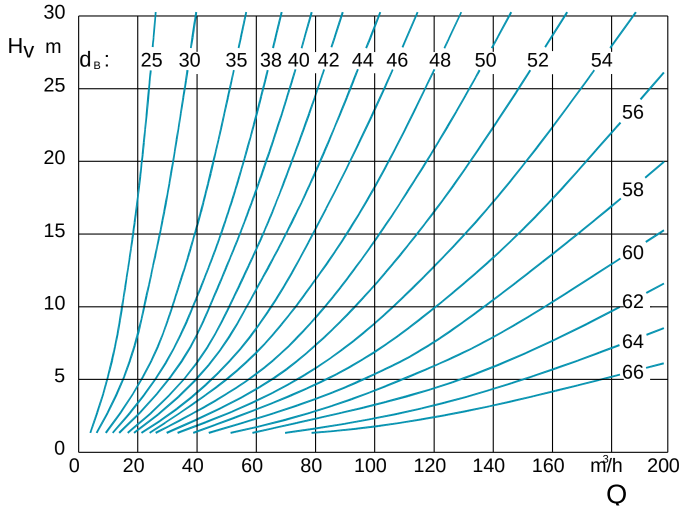
Pressure fitting DN 50 (D=50mm)

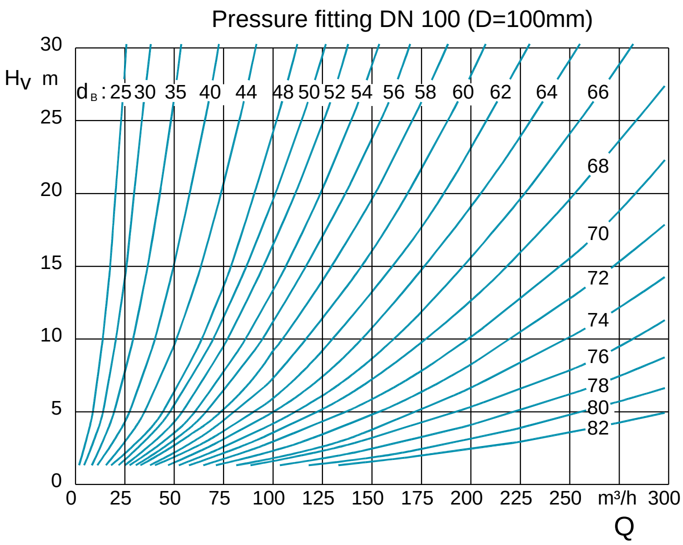


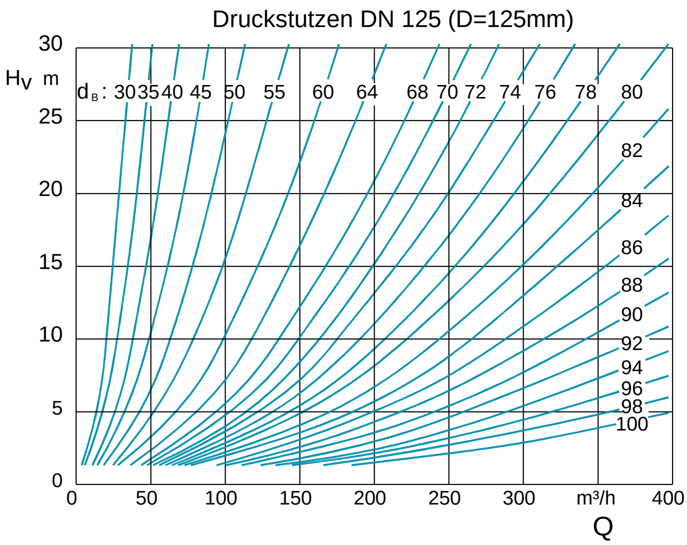
Pressure fitting DN 65 (D=70mm)



Pressure fitting DN 80 (D=82mm)




Pressure fitting DN 100 (D=100mm)
Hv m
dB: 25 30 35 40 44 48 50 52 54 56 58 60 62 64 66
68
70
72
74
76
78
80
82
m³/h
Q


Druckstutzen DN 125 (D=125mm)
Hv m
dB: 30 35 40 45 50 55 60 64 68 70 72 74 76 78 80
82
84
86
88
90
92
94
96
98
100
m³/h
Q