2 Characteristic Curves

1. The Characteristic Curves of the Pump

The pump head pressure of the rotary pump changes with the transmission flow and the rotational speed in a wide range. At a constant rotational speed the relationship between pump head pressure and transmission flow provides data from which a curve can be plotted on a right-angle co-ordinate system. This is the pump head pressure characteristic curve, QH-curve or even Q-characteristic. The last designation "Q-characteristic" results from the fact that the dependence of the pump head pressure from the transmission flow is determined on the test station by throttling, i.e. by step-by-step narrowing of the outlet of the pump. At each rotational speed there is a H = f(Q) and a P = f (Q) (input) characteristic curve, which are related to each other. All the pump head pressure and power characteristic curves represent the map of the pump. (Fig. 2)

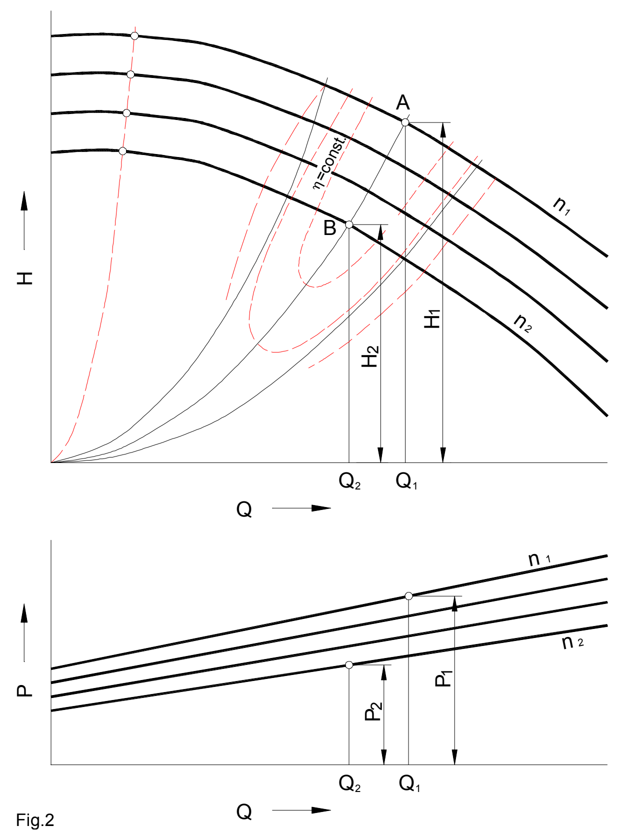


Fig.2

For liquids of low viscosity, the characteristic curves of a rotary pump without cavitation are related to each other by Affinity Laws. If Q designated the transmission flow, H the pump head pressure, P the absorbed power and n the rotational speed, the following relations exist:

$$Q_2 = Q_1 \cdot \frac{n_2}{n_1}\ ,\ H_2 = H_1 \cdot \left(\frac{n_2}{n_1}\right)^2\ ,\ P_2 \approx P_1 \cdot \left(\frac{n_2}{n_1}\right)^3 \qquad (8)$$

The equation applies only to the useful power as the power consumption includes losses that do not change with the 3rd power of the rotational speed. Also when applying the Affinity Laws to the transmission flow and pump head pressure the resulting rotational speed n_2 should not deviate more than approx. 20 to 25% from the beginning rotational speed.

Since the transmission flow changes in a linear, the pump head pressure in a square relation to the rotational speed, corresponding points (i.e. A and B) are located on a parabola with the vertex at the point of origin. All points of such a parabola have geometrically similar curves for different rotational pump speeds. If for **one** point of the parabola non-impact entry of the liquid into the vane channels is considered, then a non-impact entry applies to all other points of the parabola. They are therefore referred to as parabolas of equal impact condition.

Within its field of validity the Affinity Laws offer the possibility to calculate Q-characteristics for other rotational speeds from a measured Q-characteristic for a specific rotational speed. All these Q-characteristics are congruent.

One could conclude from the above description that the efficiency is the same for all maps on such a parabola. This however is not the case. Equal efficiency is found in elliptical curves that are closed within themselves (Fig. 2). The fluctuation is a result of the above-mentioned fact that the power consumption includes losses that do not change exactly with the 3rd power of the rotational speed. These are the mechanical losses in the bearings and the shaft seal as well as the wheel face friction loss. Additionally, the head losses inside the pump do not exactly follow the squared law due to the change of the Reynold figure. In a given transmission flow a high efficiency can only be expected if a specific rotor diameter is used. In smaller and especially greater diameters the highest achievable efficiency is diminished. Pumps with large rotor diameter at small transmission flows are therefore not economical. In that case a multi-stage design with optimised rotor diameter is preferable here.

2. The Characteristic Curve of the System

The pump head pressure of a plant generally consists of a static pump head pressure H_{stat} and the loss of head H_v in the connected pipe lines and aggregates. The **static pump head pressure H_{stat}** is determined by the height difference of suction and pressure water level, i.e. by the geodesic pump head pressure H_{geo} (Fig. 1) and the pressure difference $p_{II} - p_I$ between suction and pressure container.

$$H_{stat} = H_{geo} + \frac{10{,}2 \cdot (p_{II} - p_I)}{\rho} \qquad (9)$$

In this equation p_I and p_{II} are in bar and the density ρ in kg/dm³.

The **loss of head H_v** changes almost square to the transmission flow. This relation of friction losses and transmission flow can therefore be represented with a parabola-like curve.

The total of H_{stat} and H_V results in the characteristic curve of the plant (Fig. 3). It shows the resistance head that must be overcome by the pump for each transmission flow. The point of intersection B of the plant characteristic curve with the Q-characteristic is the actual working point of the pump.

When concerning a plant that is solely dedicated to transit liquids and which only consists of pipe lines with standard shaped pieces and gauges, the H_V values necessary for the drawing of the curve can be calculated with help of the diagrams from the attached EDUR work sheet "Pipe Friction Losses".

Often, however, systems will contain aggregates or machines to be cooled for which the head loss is only known at a specific transmission flow. In that case the pump head pressure of the plant can only be calculated for this point.

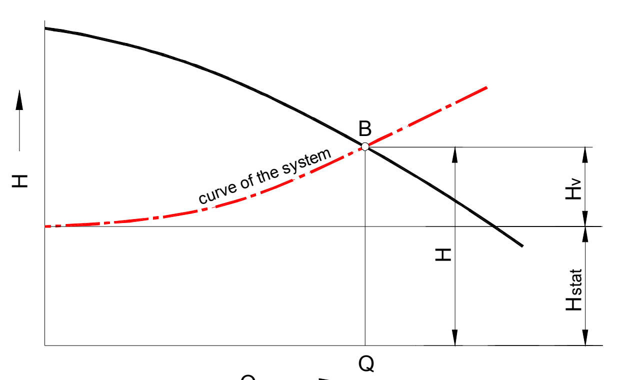


Fig. 3

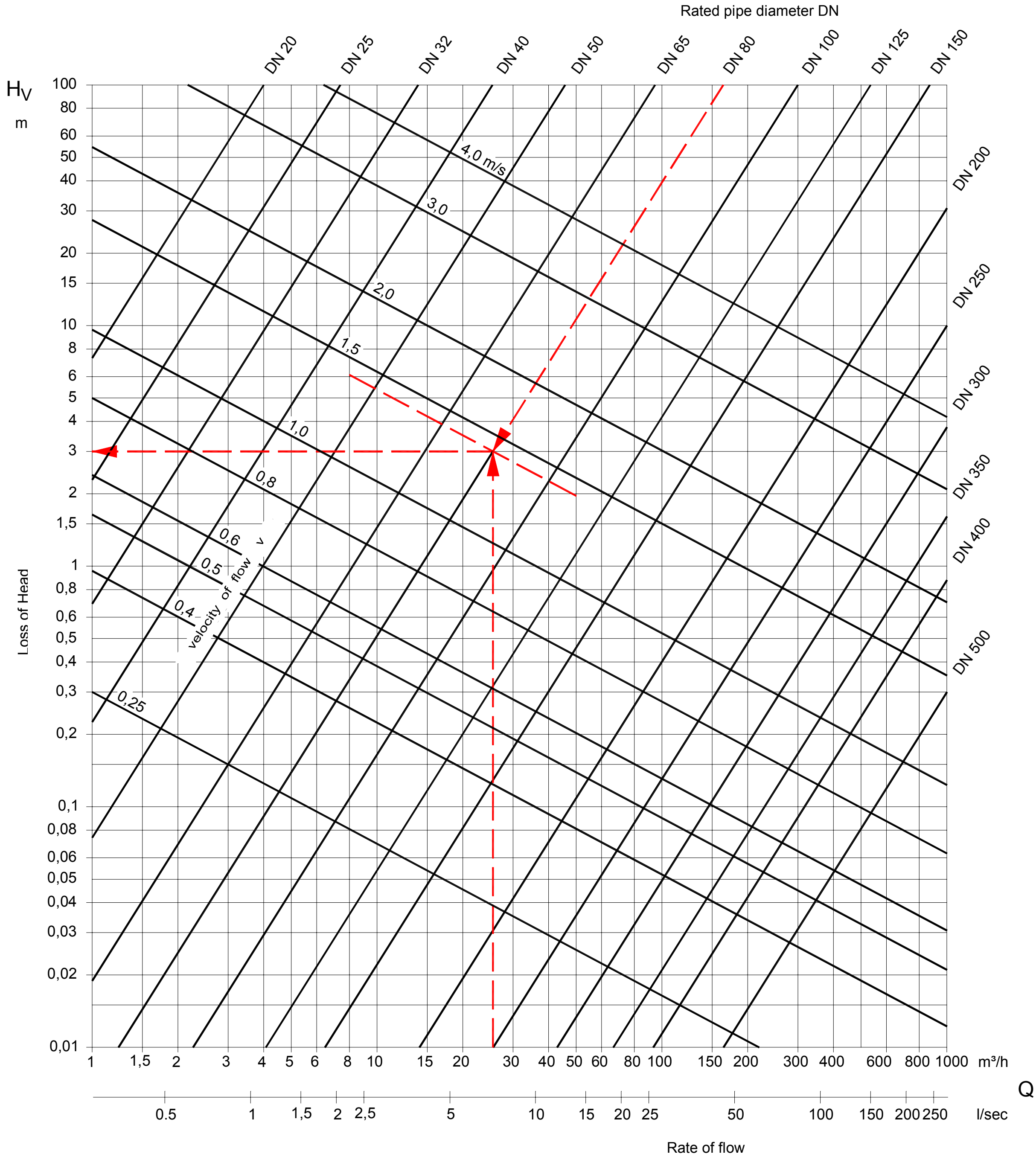
Nevertheless, the progression of the characteristic curve can be determined with sufficient accuracy when utilizing the relation

$$H_{V2} = H_{V1} \cdot \left(\frac{Q_2}{Q_1}\right)^2 \qquad (10)$$

to calculate the head losses for several transmission flows Q_2 and plotting the points versus the static pump head pressure. The characteristic curve is a very helpful tool in the selection of an appropriate pump for an existing or planned plant. The technical drawing of the system and pump characteristic curves enables a reliable assessment of the actually increase to be expected in transmission flow when applying a larger pump, especially during the extension of existing systems.

EDUR – Work sheet
„Pipe Friction Losses“

Velocity of Flow v and Loss of Head H_V in Straight Pipelines with regard of 100 m Length of the Pipeline



The loss of head Hv as ascertained in the diagram is approximative only with regard to galvanized steel pipes and/or grey cast iron bituminous pipes respectively.

Multiplier for other pipelines:

New rolled steel pipes	app. 0.85
Copper pipes or plastic pipes	app. 0.7
Old cast or steel pipes	app. 1.25
incrustated pipes	app. 1.7 (the data referred to should be taken from the diagram H_V at actual, narrowed cross-section only!)

EDUR – Work sheet
„Pipe Friction Losses“

Equivalent Pipe Lengths in m for Valves and Fittings referred to a Velocity of Flow of 2.0 m/s

Nominal width	20	25	32	40	50	65	80	100	125	150	200	250
Gate valve, completely open	0.2	0.3	0.4	0.5	0.7	0.9	1.2	1.5	1.9	2.3	3.3	4.6
Passage valve	4.0	5.0	7.0	9.0	12.0	16.0	20.0	25.0	31.0	38.0	52.0	66.0
Full-way valve	1.0	1.4	1.6	2.3	3.0	4.0	5.3	6.8	8.4	11.0	15.0	19.5
Full-way check valve	2.4	3.3	4.1	5.8	7.8	10.6	13.8	17.0	21.0	26.0	35.0	44.0
Foot valve with suction strainer	3.0	4.1	5.1	7.3	9.7	13.2	17.2	21.0	26.0	32.0	43.5	55.0
Bend 90°	0.3	0.4	0.5	0.7	1.0	1.3	1.7	2.1	2.7	3.2	4.5	6.0
Elbow 90°	0.9	1.3	1.5	2.2	2.9	4.0	5.2	6.8	8.7	10.6	14.5	19.0

Example:
Search for the loss of head anf for flow velocity in a pipeline DN 80 of 50 m length with 4 pieces of bends 90° and 2 pieces of gate valves.
Flow rate 25 m³/h.

25 m³/h, DN 80
H_V = 3.0 m upon 100 m length of straight pipeline.
Velocity of flow app. 1.4 m/s.

Straight pipeline	50	m
4 bends 90° DN 80 corresp. to	6,8	m
2 gate valves DN 80 corresp. to	2,4	m
calculation of pipeline	59,2	m

$$H_V = \frac{3 \cdot 59{,}2}{100} = \underline{\underline{1{,}78}} \quad \text{m}$$

Flow velocity and loss of head indicate the suitable nominal diameter for pipes. In case of extremely high flow velocity and large loss of head it is advisable to choose a larger nominal diameter also regarding the noise of flow.