In textbooks and seminar materials mostly only general conclusions about the parallel operation of two or more pumps are given.
A formula is shown in which the flow rates of the individual pumps are simply added for a given constant head.

Parallel Pumping

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Abstract: *At first sight, parallel operation of pumps does not seem to pose any problems. Under real conditions, however, parallel pumping proves to be rather more complex than typically portrayed in the relevant literature. So as to ensure that pumps and systems produce the required flow rates and pressures detailed information is needed about the real curve of both the pumps and the system. The duty point of the individual pumps operated in parallel will shift and this will affect both efficiency and NPSH performance. Only if these changes are taken into account will a parallel pumping system function properly.*

$Q_{tot} = Q_{I} + Q_{II} + Q_{III}$
where $H = const.$

Fig.1 is shown in addition, which displays the information exactly and correctly.

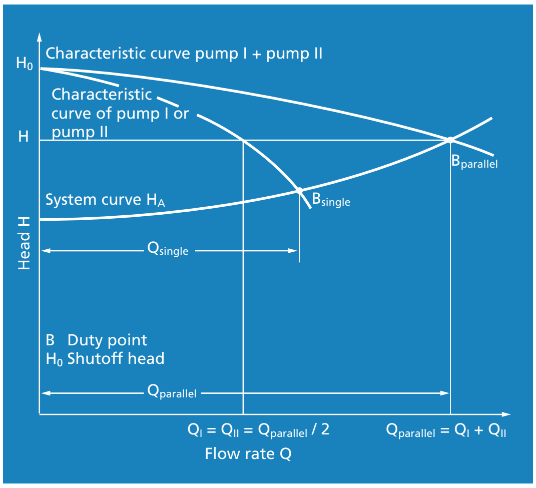


Fig. 1: Parallel operation of two identical centrifugal pumps with stable characteristic curve

Despite this simple formula, the problem continues to occur during pump operation in parallel that the pumps or the pumping system do not achieve the expected flow rate. How such problems can avoided will be explained under the following five headings:

Real versus ideal conditions

The real conditions in pump systems differ from the ideal textbook representation with regard to the system characteristics.

A pump system in which several pumps can run in parallel is generally arranged as shown in Fig. 2.

Each pump takes its flow from an inlet sump via an individual pipe or from a manifold via an individual connection, and discharges into a common manifold.

It is useful to calculate the *system curve* (also called *system characteristic*) using only the static component (for example the difference between the geodetic altitudes of the water levels) and the dynamic component of the losses of the common manifold. The dynamic losses in the individual pipes (individual losses of each pump) are subtracted from their respective pump characteristics. A flat pump characteristic becomes a steeper one taking into account the losses directly attributable to that pump.

By adding the resulting characteristic curves reduced by the individual

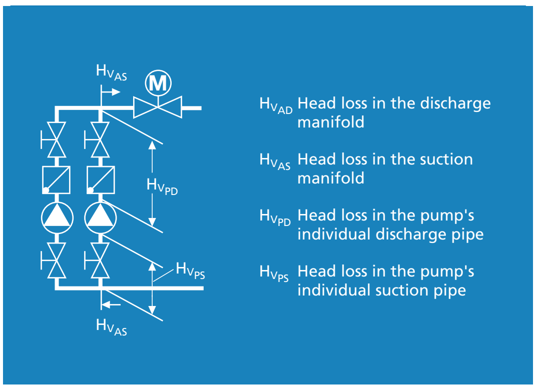


Fig. 2: Pump system schematic (2 pumps operating in parallel)

System curve or characteristic: The system curve (or characteristic) is the graph of the system head H_A required for the installation shown as a function of the flow rate Q. It represents the flow resistance of the system for the pump. It is made up of static and dynamic components. The static components (static head H_{geo} and pressure difference between the inlet and outlet vessel (p_{out}-p_{in})) are independent of the flow rate. The dynamic component consists of the flow resistance in the piping system and grows as the square of the flow velocity ($H_v = f(Q2)$).

losses, the intersection with the system characteristic and therefore the total *duty* or *operating point* and the operating point of each pump can be determined (see Fig. 3).

The form of the system characteristic must be carefully considered when an existing facility is to be expanded, for example by installing further pumps in parallel. As shown in Fig. 4 (presenting the system curve of a cooling circuit), the total flow rate cannot be significantly increased by installing more pumps.

Duty point selection

Each pump's duty point is selected according to the mode of operation, considering the efficiency of each pump and the pump system, as well as the NPSH of the pumps.

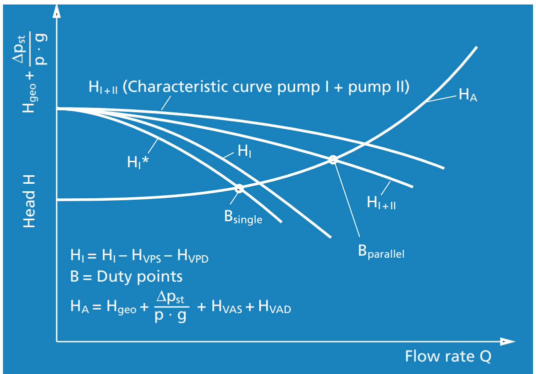
If the efficiency is included in the characteristic curves, it becomes evident that when changing from single to multiple pump operation, the operating point of each individual pump increasingly shifts towards lower load conditions.

The choice of pump size for pumps working in parallel is therefore largely dependent on the expected operating time. If it is planned that most often three pumps will be working in parallel, then the best efficiency point (BEP) of the pumps should be chosen for this condition ($Q_{tot} = 3 \bullet Q_{opt}$, individual pumps). However, it must be ensured that if a single pump is operated on its own, the increased flow of that pump still falls within the permissible range, in particular with respect to *NPSH performance* (see Fig. 5).

In the opposite case, when parallel operation is only occasionally required, for example to meet peak demands, the pump size should be chosen so that single pump operation is near the BEP. During parallel operation the duty points of the individual pumps shift towards low flows with lower efficiency. There is usually no risk of insufficient NPSH.

Duty or operating point: Intersection of the system curve with the pump characteristic curve (for the actual speed of rotation). The only possible working condition of the pump in a non-controlled system.

NPSH performance: When cavitation (formation of vapour bubbles) develops in the impeller, the performance of a pump (head, efficiency, noise emission, etc.) is affected. The pump parameter used to quantify this behaviour is the NPSH value.



System curve: Δp_{st} Difference of the static pressures in the suction- and discharge side vessel; pump curve after substraction of individual losses (see Fig. 2)

Fig. 3: Pump characteristic curve taking into account the losses

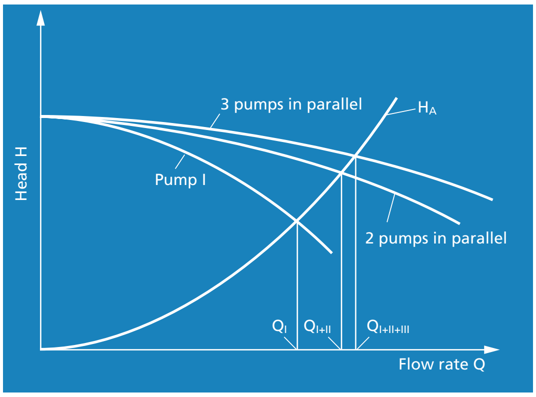
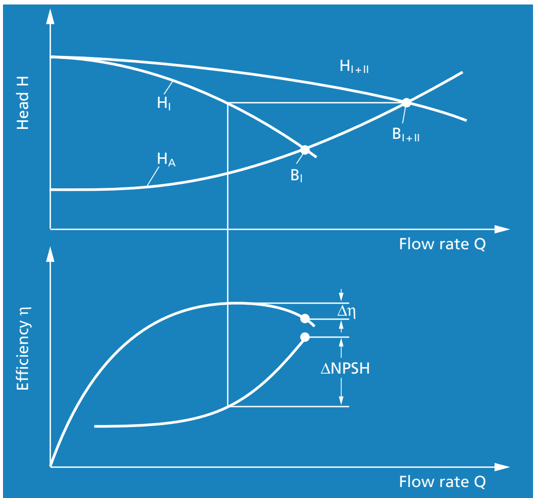


Fig. 4: Pumps operating in parallel: low increase in flow rate with steep system curve



Continuous parallel operation of pumps: $Q_p = Q_{opt}$ is targeted. If one pump stops, the efficiency and the NPSH value deteriorate (Δη and ΔNPSH).

Fig. 5: Shift of operating point when pump working in parallel is stopped

Pump characteristic curves

The pump characteristic curves and the system's mode of operation are of paramount importance. Various types of pumps, particularly when the impeller is cut down substantially below its largest diameter, tend to hydraulic instability, i.e., the developed head at shutoff (Q = 0) is lower than at some other operating point.

If such pumps are connected in parallel, the second pump cannot open its check valve against the high operating pressure produced by the first pump and runs with Q = 0 (Fig. 6). The energy input by the motor is almost completely converted to heat within the pump, and a substantial increase of the temperature of the enclosed fluid can result.

≠ Pumps = Shutoff heads

For reasons of better utilization of the pumps, a small pump (for example OMEGA 300/560) is installed next to several larger pumps (for example RDLO 500/790) in some pump systems to achieve a more complete coverage of the flow range with good efficiency (see Fig. 7). By properly choosing the speed and diameter both pump sizes can be designed to have the same *shutoff head*. It should be kept in mind, however, that depending on the motor, the actual operating speed can cause significant head differences. These can lead to a major *shift of the operating point*, particularly for the smaller pump.

For the smaller pump, which usually has a higher operating speed and correspondingly higher *inlet and discharge velocity*, the so-called individual losses must be carefully calculated.

≠ Pumps
≠ Shutoff heads

In some rare cases pumps can be run in parallel when they do not have the same shutoff head. As shown in Fig. 8, satisfactory operation can only be achieved when the shutoff head of the smaller pump is not exceeded. For smaller flow rates the same problem occurs as for a performance curve instability, i.e. the smaller pump is running but cannot open its check valve, and the motor energy heats up the fluid in the pump.

A similar mode of operation and risk are present when a second pump is started up with a variable speed drive in parallel to a pump already running.

Throughflow in the second pump will only start when its shutoff head exceeds the operating pressure of the first pump (intersection with the system curve). Except for this problem, a variable speed pump is an ideal partner to a fixed speed pump for parallel operation. Sufficient instrumentation with flow meters and pressure gauges is helpful in this case.

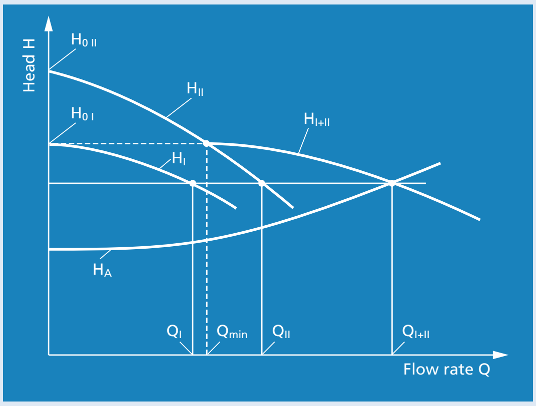


Fig. 8: Parallel operation of two dissimilar pumps

Shutoff head: Head developed by the pump for zero flow rate (operating against a closed valve).

Shift of operating point: When several pumps are operating in parallel against the same system curve, the total flow rate is divided among them according to their individual flow rate for the head imposed by the system. If the actual characteristic curve of a pump deviates from that of the expected characteristic, for example due to design tolerances or variation of the speed, the actual operating point of the pump can vary substantially from the expected one.

Inlet / discharge velocity: The average flow velocity (Q/A) at the pump inlet or discharge

$$\mathbf{A = cross\text{-}section = \frac{1}{4} \cdot \pi \cdot D^2_{S,D}}$$

can be calculated using the actual inlet or outlet diameter.

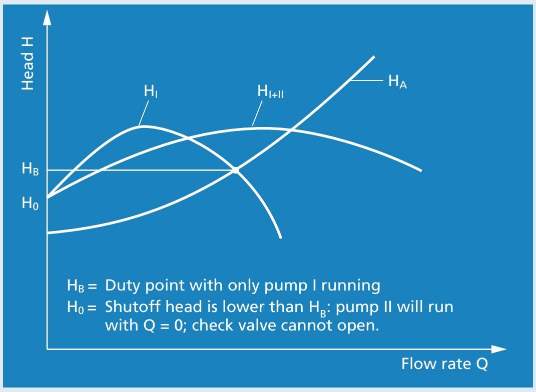


Fig. 6: Parallel operation of two pumps with unstable characteristic curve

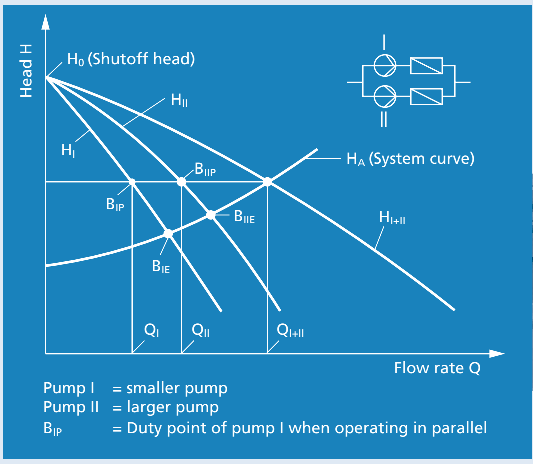


Fig. 7: Parallel operation of two pumps with equal shutoff heads