ARRANGING DISSIMILAR CENTRIFUGAL PUMS IN SERIES AND PARALLEL

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Abstract: While series or parallel arrangement is acceptable for some applications, arranging dissimilar pumps in series or in parallel may lead to problems, especially if one pump is much larger than the other. Also, to avoid pump damage and loss of combined net head, any individual pump should be shut off and bypassed at flow rates larger than that pump's free delivery, so this curve indicates.

Keywords: pums in serios and parallel, duty point of the system

1. INTRODUCTION

There are many applications where two or more pumps operate in series or in parallel; usually, the pumps are identical, but there are some situations where the pumps are different. Two very dissimilar centrifugal pumps are considered, first in series and than in parallel, and the problems related to widely different performance curves are exposed. Also, the work considers the case of three different centrifugal pumps, first in series and than in parallel, and analyses the performance curves for the two configurations.

2. SOME FUNDAMENTAL PARAMETERS TO ANALYSE THE PERFORMANCE OF A PUMP

Some fundamental parameters are used to analyze the performance of a pump, [1], [3]. The **mass flow rate** of fluid through the pump, $\dot{m}$ is a primary pump performance parameter. In case of an incompressible flow, it is more common to use the **volume flow rate** (or **capacity**), V (mass flow rate divided by fluid density), rather than mass flow rate,

$$\dot{V} = \frac{\dot{m}}{\rho}. \tag{1}$$

Other fundamental parameter of a pump is its **net head**, H, defined as the change in Bernoulli head between the inlet and outlet of the pump,

$$H = \left(\frac{p}{\rho g} + \frac{v^2}{2g} + z\right)_{out} - \left(\frac{p}{\rho g} + \frac{v^2}{2g} + z\right)_{in}. \tag{2}$$

The dimension of net head is length, and it is often listed as an equivalent column height of water (even for a pump that is not pumping water). Consider the special case of incompressible flow through a pump in which the inlet and outlet diameters are identical, $D_{in} = D_{out}$, and there is no change in elevation, $z_{in} = z_{out}$; the equation (2) reduces to

$$H = \frac{p_{out} - p_{in}}{\rho g}. \tag{3}$$

For this simplified case, net head is simply the pressure rise across the pump expressed as a head (column height of the fluid).

The **useful power** actually delivered to the fluid, $\dot{W}_{water}$, is

$$\dot{W}_{water} = \rho g \dot{V} H . \tag{4}$$

All pumps suffer irreversible losses due to friction, internal leakage, flow separation on blade surfaces, turbulent dissipation, etc. Therefore, the mechanical power supplied to the pump must be larger than $\dot{W}_{water}$.

The **supplied power** (the external power supplied to the pump), for the typical case of a rotating shaft, is:

$$\dot{W}_{shaft} = \omega T_{shaft} , \tag{5}$$

where ω is the rotational speed of the shaft (rad/s) and T_{shaft} is the torque supplied power.

One defines **pump efficiency** as the ratio of useful power to supplied power,

$$\eta_{pump} = \frac{\rho g \dot{V} H}{\omega T_{shaft}} . \tag{6}$$

These are the principal pump parameters.

3. PUMP PERFORMANCE CURVES AND MATCHING A PUMP TO A PIPING SYSTEM

It is necessary to present, shortly, the pumps performance curves and the matching a pump to a piping system, [1], [4].

The maximum volume flow rate through a pump occurs when its net head is zero, $H = 0$; this flow rate is called the pump's **free delivery.** The free delivery condition is achieved when there is no flow restriction at the pump inlet or outlet – in other words when there is no **load** on the pump. At this operating point, $\dot{V}$ is large, but H is zero; the pump's efficiency is zero because the pump is doing no useful work, as is clear from eq. (6). At the other extreme, the **shutoff head** is the net head that occurs when the volume flow rate is zero, $\dot{V} = 0$, and is achieved when the outlet port of the pump is blocked off. Under these conditions, H is large but $\dot{V}$ is zero; the pumps efficiency (6) is again zero, because the pump is doing no useful work. Between these extremes, from shutoff to free delivery, the pump's net head may increase from its shutoff value somewhat as the flow rate increases, but H must eventually decrease to zero as the volume flow rate increases to its free delivery value. The pump efficiency reaches its maximum value somewhere between the shutoff condition and the free delivery condition; this operating point of maximum efficiency is appropriately called the **best efficiency point** (BEP), and is notated by an asterisk (H^*, V^*, W_{shaft}^*). Curves of H, η_{pump}, and W_{shaft} as functions of $\dot{V}$ are called pump performance curves (or characteristic curves); typical curves at one rotational speed for a centrifugal with backward-inclined blades are plotted in Fig.1. The pump performance curves change with rotational speed.

For steady conditions, a pump can operate only along its performance curve. Thus the operating point of a piping system is determined by matching system requirements (required net head) to pump performance (available net head). In a typical application, $H_{required}$ and $H_{available}$ match at one unique value of flow rate – this is the **operating point** or **duty point of the system**, where

$$H_{required} = H_{available} . \tag{7}$$

The operating point of a piping system is established as the volume flow rate where the system curve and the pump performance curve intersect (Fig. 2).

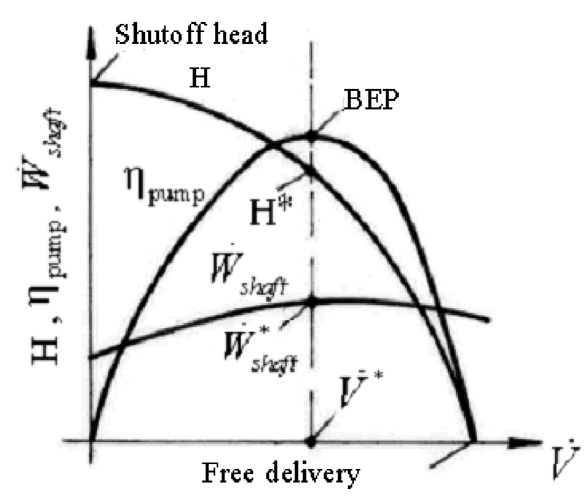

Fig. 1. Typical pump performance curves for a centrifugal pump

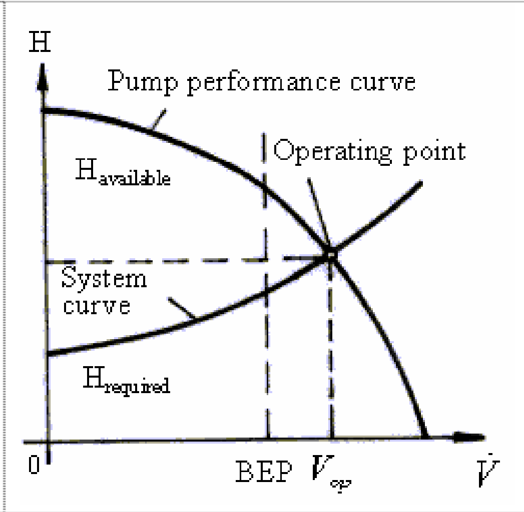

Fig. 2. Operating point of a piping system

4. ARRANGING TWO VERY DISSIMILAR PUMPS IN SERIOS AND IN PARALLEL

When faced with the need to increase volume flow or pressure rise by a small amount, one might consider adding an additional smaller pump in series or in parallel arrangement with the first pump. While series or parallel arrangement is acceptable for some applications, arranging dissimilar pumps in series or in parallel may lead problems, especially if one pump is much larger than the other, [1], [2].

A better course of action is to increase the original pumps speed and/or input power (larger electric motor), replace the impeller with a larger one, or replace the entire pump with a larger one. The logic for this decision can be seen from the pump performance curves, realizing that pressure rate and volume flow rate are related.

Arranging dissimilar pumps in series may create problems because the volume flow rate through each pump must be the same, but the overall pressure rise is equal to the pressure rise of one pump plus that of the other. If the pumps have widely different performances curves, the smaller pump may be forced to operate beyond its free delivery flow rate, whereupon it acts like a head loss, reducing the total volume flow rate.

Arranging dissimilar pumps in parallel may create problems because the overall pressure rise must be the same, but the net volume flow rate is the sum of that through each branch. If the pumps are not sized properly, the smaller pump may not be able to handle the large head imposed on it, and the flow in its branch could actually be reserved; this would inadvertently reduce the overall pressure rise.

5. THREE DISSIMILAR PUMPS IN SERIOS

When two or more identical or dissimilar pumps are operated in series, the combined net head is simply the sum of the net heads of each pump, [1], [2], at a given volume flow rate,

$$H_{combined} = \sum_{i=1}^{n} H_i \ . \tag{8}$$

The equation (8) is illustrated in Fig. 3 for three dissimilar centrifugal pumps in series. In this example, pump 1 is the weakest and pump 3 is the strongest.

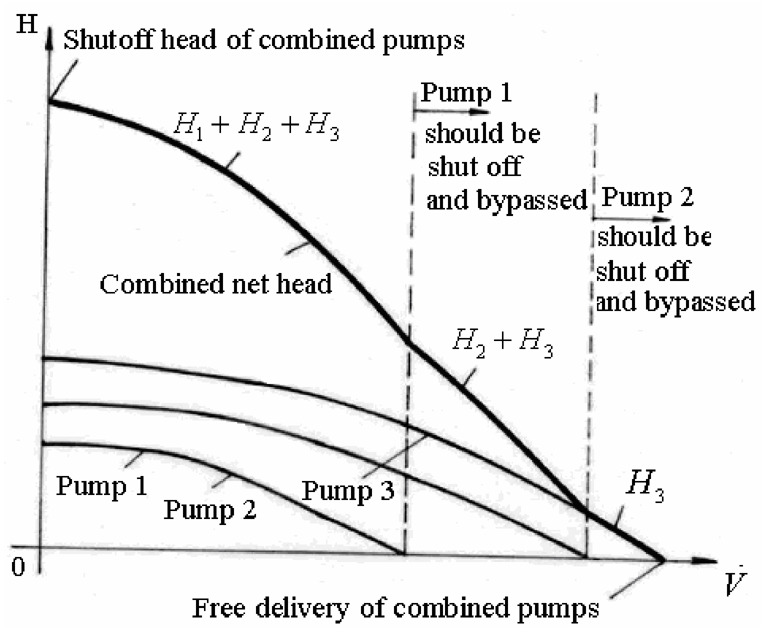

Fig. 3. Pump performance curve for three dissimilar pumps in series

The shutoff head of the three pumps combined in series is equal to the sum of the shutoff head of each individual pump. For low values of volume flow rate, the net head of the three pumps in series is equal to

$$H_{combined} = H_1 + H_2 + H_3 \,. \tag{9}$$

Beyond the free delivery of pump 1 (to the right of the first vertical dashed line in Fig. 3), pump 1 should be shut off and bypassed. Otherwise it would be running beyond its maximum designed operating point, and the pump or its motor could be damaged. Furthermore, the net head across this pump would be negative as previously discussed, contributing to a net loss in the system. With pump 1 bypassed, the combined net head becomes

$$H_{combined} = H_1 + H_2 \,. \tag{10}$$

Similarly, beyond the free delivery of pump 2, that pump should also be shut off and bypassed, and the combined net head is then equal to H_3 alone,

$$H_{combined} = H_3 \,, \tag{11}$$

as indicated to the right of the second vertical dashed line in Fig. 3. In this case, the combined free delivery is the same as that of pump 3 alone, assuming that the other two pumps are bypassed.

If the three pumps were identical, is would not be necessary to turn off any of the pumps, since the free delivery of each pump would occur at the same volume flow rate.

6. THEREE DISSIMILAR PUMPS IN PARALLEL

When two or more identical or dissimilar pumps are operated in parallel, their individual volume flow rates are summed,

$$\dot{V}_{combined} = \sum_{i=1}^{n} \dot{V}_i \,. \tag{12}$$

As an example, one considers the same three pumps, but arranged in parallel. The combined pump performance curve is shown in Fig. 4.

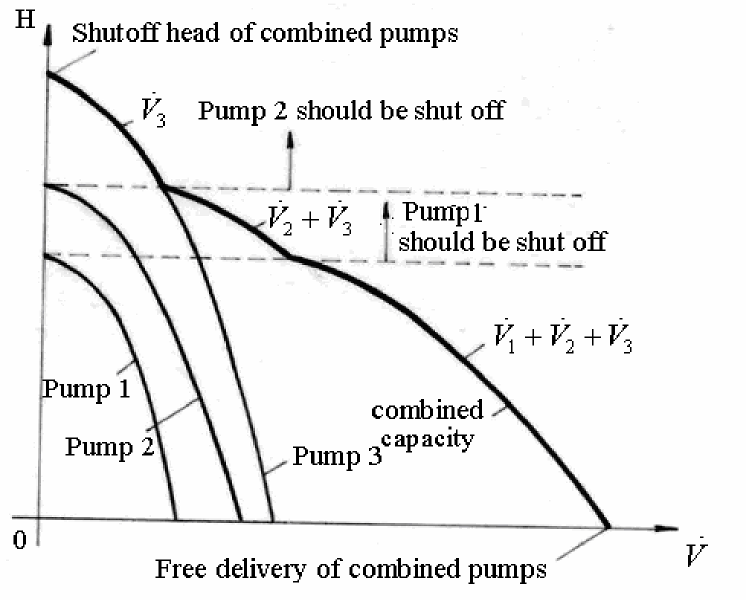

Fig. 4 Pump performance curve for three dissimilar pumps in parallel

The free delivery of the three pumps arranged in parallel is equal to the sum of the free delivery of each individual pump. For low values of net head, the capacity of the three pumps in parallel is equal to

$$\dot{V}_{combined} = \dot{V}_1 + \dot{V}_2 + \dot{V}_3 \tag{13}$$

Above the shutoff head of pump1 (above the first horizontal dashed line in Fig. 4) pump 1 should be shut off and its branch should be blocked (with a valve). Otherwise it would be running beyond its maximum designed operating point, and the pump or its motor could be damaged. Furthermore, the volume flow rate through this pump would be negative as previously discussed, contributing to a net loss in the system. With pump 1 shut off and blocked, the combined capacity becomes

$$\dot{V}_{combined} = \dot{V}_2 + \dot{V}_3 . \tag{14}$$

Similarly, above the shutoff head of pump 2, that pump should also be shut off and blocked. The combined capacity is then equal to V_3 alone,

$$\dot{V}_{combined} = \dot{V}_3 , \tag{15}$$

as indicated above, the second horizontal dashed line in Fig. 4. In this case, the combined shutoff head in the same at that of pump 3 alone, assuming that the other pumps are shut off and their branches are blocked.

In practice, several pumps may be combined in parallel to deliver a large volume flow rate. Ideally all the pumps should be identical. So, if the three pumps where identical, it would not be necessary to torn off any of the pump, since the shutoff head of each pump would occur at the same net head.

6. CONCLUSIONS

While series or parallel arrangement is acceptable for some applications, arranging dissimilar pumps in series or in parallel may lead problems, especially if one pump is much larger than the other.

The pump performance curve for the three pumps in series shows that at low values of volume flow rate, the combined net head is equal to the sum of the net head of each pump by itself. Also, to avoid pump damage and loss of combined net head, any individual pump should be shut off and bypassed at flow rates larger than that pump's free delivery, so this curve indicates.

The pump performance curve for the three pumps arranged in parallel shows that the combined capacity is equal to the sum of the capacity of each pump by itself. However, to avoid pump damage and loss of combined capacity, any individual pump should be shut off at net heads larger than that pump's shutoff head, as this performance curve indicates.

The analysis of the arranging dissimilar centrifugal pumps in series and parallel is useful for the establishment of the exploitation optimal solution.

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