

First in a 3-part series:

# Centrifugal pump operation at off-design conditions

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**The following is the first in a series of three articles which examine the performance of centrifugal pumps operating at flows either higher or lower than the rate for which they were designed to achieve their highest efficiency. Part 1 examines the effects on radial thrust and temperature rise in the liquid pumped when pumps operate with flows below or exceeding the capacity for best efficiency. Part 2, scheduled for the May issue, will discuss internal recirculation, driver overload in high-speed pumps, and air-binding in pumps operating below capacity with liquids containing high concentrations of dissolved or entrained gases. Part 3 in the June issue will examine the manner in which by-passes must be arranged when process demands fall below minimum flow values recommended by the pump manufacturers.**

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Theoretically, as long as the NPSH (net positive suction head) available is greater than the required NPSH, a centrifugal pump is capable of operating over a very wide range of capacities. The exact operating capacity is determined by the intersection of the pump head-capacity curve with the system-head curve. This operating capacity can be changed by altering either one or both of these curves: varying the pump speed will alter the head-capacity curve, while throttling the pump discharge will alter the system-head curve.

But at any given speed, the performance of a centrifugal pump is at its optimum at only one capacity point, and that is the capacity at which the effi-

ciency curve reaches its maximum. At all other flows, the geometric configuration of the impeller and of the casing no longer provides an ideal flow pattern. Our definition of "off-design" conditions must, therefore, be any condition wherein a pump is required by circumstances to deliver flows either in excess of or below the capacity at best efficiency.

### Operation at high flows

There are two circumstances which might lead to the operation of a pump at flows in excess of its best efficiency or even of its design point. The first occurs when a pump has been oversized by specifying excessive margins in both head and capacity. Under these circumstances, the pump performance and its relation to the system-head curve is shown in Fig. 1. The head-capacity curve intersects the system-head curve at a capacity much in excess of the real required flow. Of course, the pump can be throttled back to the required capacity and the power reduced somewhat. But if, as frequently occurs, the pump runs uncontrolled, it will always run at the excess flow indicated on Fig. 1. Unless sufficient NPSH has been made available, the pump may suffer cavitation damage and, of course, the power con-

sumption will increase for pumps with a specific speed under about 4500 rpm.

The second circumstance occurs when two or more pumps are used in parallel and one of the pumps is taken out of service because the demand has been decreased. Fig. 2 describes the operation of two such pumps. Whenever a single pump is running, its head-capacity curve intersects the system-head curve at flows in excess of the design capacity. This is called the "run-out" point. Here again the available NPSH and the size of the driver must be carefully selected to satisfy the conditions that will prevail at the run-out point.

### Operation at low flows

The most frequent cause of operating a pump at reduced flows is a reduction in demand of the process served by the pump. However, it may be that two pumps operating in parallel are unsuitable for service at reduced flows. One of the pumps on the line may have its outlet check valve closed by the higher pressure developed by the stronger pump.

Operating centrifugal pumps at reduced capacities leads to a number of unfavorable conditions which may take place separately or simultaneously. Some of the consequences which must be anticipated or circumvented are:

1. Operation at less than best efficiency. If the reduced flows are required by the characteristics of the process served by the pump, this effect can be obviated somewhat by using variable speed drives or by using several pumps for the total required capacity and shutting down pumps sequentially as the total demand is reduced.
2. If the pump is of the single volute design, it will be subjected to a higher radial thrust which will increase the load on its radial bearings. A pump

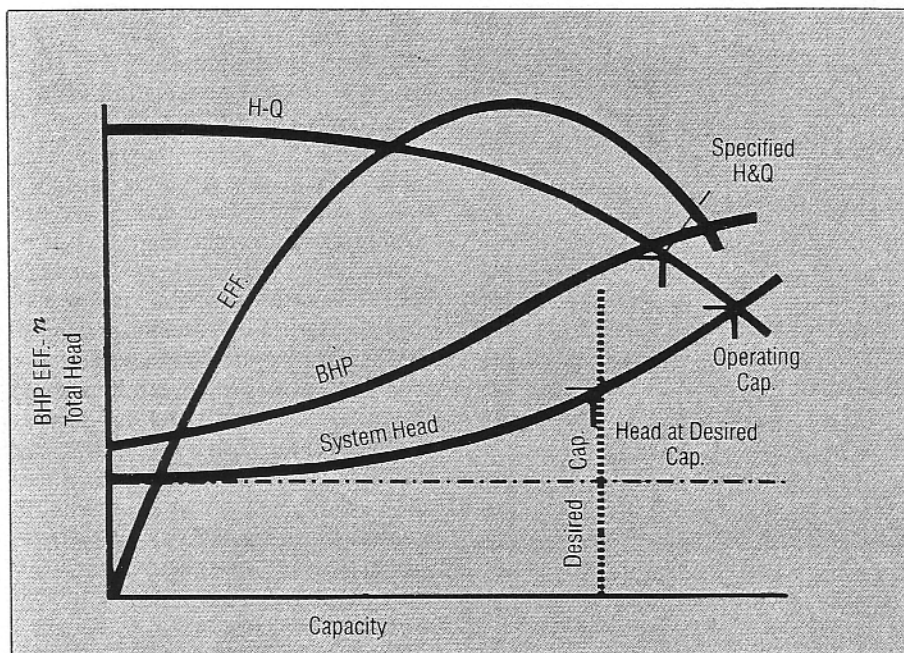


Fig. 1: Effect of oversizing pump

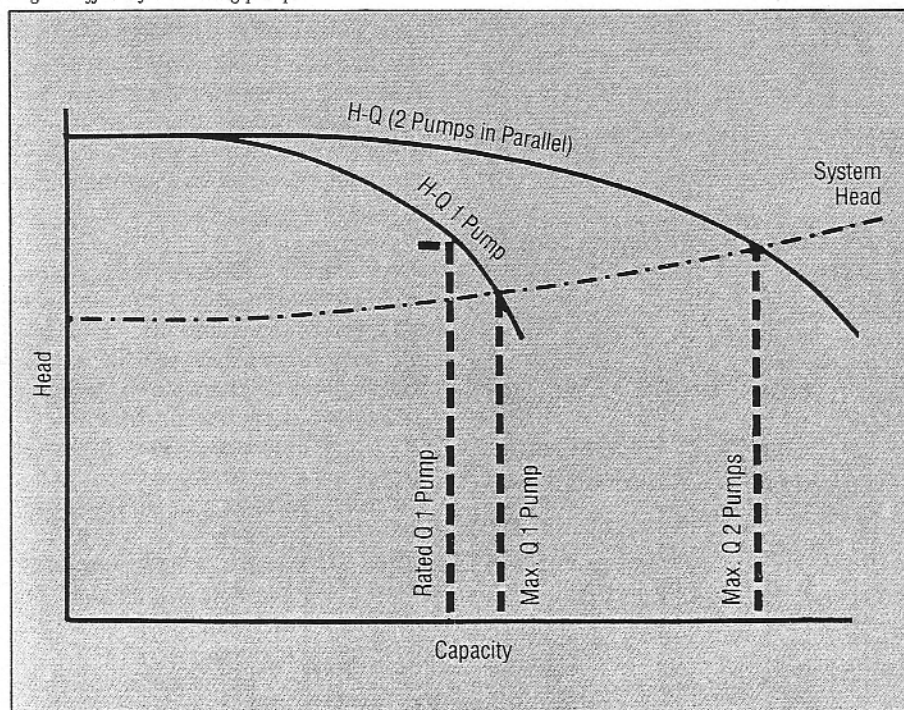


Fig. 2: Two- and one-pump operation

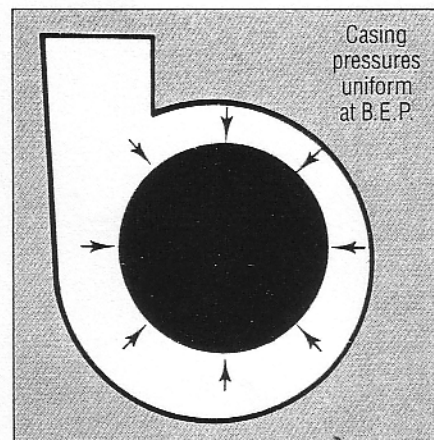


Fig. 3: Radial forces in single-volute pumps at best efficiency point

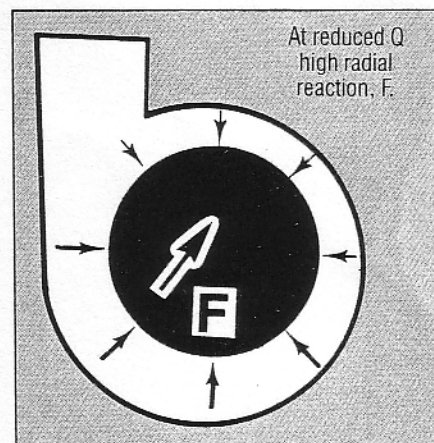


Fig. 4: Radial forces in single-volute pumps operating at reduced flow

that is expected to operate at lower than rated flows must be able to handle a higher bearing load.

3. As the capacity is reduced, the temperature rise of the pumped liquid increases. To avoid exceeding permissible limits, a minimum flow by-pass must be provided. The by-pass, which can be made automatic, will also protect the pump against the accidental closing of its check valve while the pump is still running.



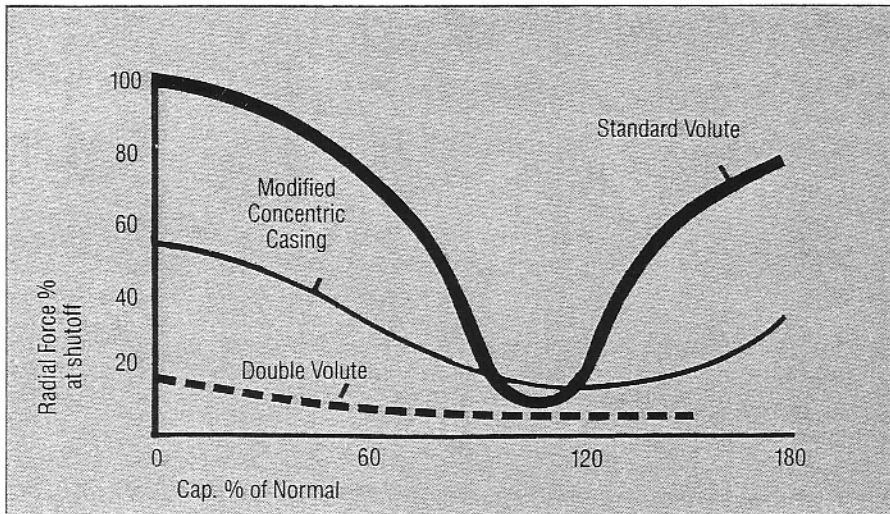


Fig. 5: Radial thrust in volute pumps

- At certain flows below that at best efficiency, all centrifugal pumps are subject to internal recirculation, both in the suction and discharge areas of the impeller. This can cause hydraulic surging and damage to the impeller material similar to that caused by classic cavitation, although taking place in a different area of the impeller.
- High specific speed pumps have power curves that rise with reduced capacities. Unless the driver size has been selected with this fact in consideration, it may be overloaded when operating capacities are reduced.
- If the liquid contains an appreciable amount of entrained air or gas, and if the pump capacity is reduced below a certain minimum, the pump can become air-bound.

Each of the effects may dictate a different recommended minimum operating capacity. Obviously, the final decision must be based on the greatest of the individual minimums. We shall now examine the effects in greater detail except for the first effect which is self-evident.

### Radial thrust

In a single-volute pump casing design (Fig. 3), uniform or nearly uniform pressures act on the impeller when the pump is operated at its best efficiency capacity. At reduced capacities, the pressures around the impeller are not uniform (Fig. 4) and there is a resultant radial reaction (F). Fig. 5 shows that this unbalanced force is greatest at shut-off.

For any percentage of capacity, the

radial reaction is a function of the total head and of the width and diameter of the impeller. Thus, a high-head pump with a large diameter impeller will have a much greater radial reaction force at partial capacities than a low-head pump with a small diameter impeller.

For a given shaft and bearing size, the reactions lead to increased deflection and bearing loading at reduced capacities. Since it may be uneconomical to supply oversize shafts and bearings in many cases, pump manufacturers generally indicate the minimum recommended flow for single volute pumps, frequently at 50% of the best-efficiency-capacity. On the other hand, because this limitation may in itself be uneconomical, it became desirable to design standard units capable of accommodating the effect of radial thrust.

One solution is to use a so-called double-volute design (Fig. 6). Basically this consists of providing two volutes, located at 180° from each other and joined into a common discharge. Such a design comes very close to balancing the radial reactions over the entire range of capacities. Small size pumps may not be readily adaptable to the double-volute design because of foundry problems. They are frequently built with modified concentric casings which reduce the unbalanced radial thrust appreciably when compared to single-volute pumps, as shown in Fig. 5.

### Temperature rise effect

The thermodynamic problem that arises from the operation of a centrifugal pump

at extremely reduced flows is caused by the heating up of the liquid handled by the pump. The difference between the brake horsepower consumed and the water horsepower developed represents the power losses within the pump itself, except for a very small amount lost in the pump bearings. These power losses are converted into heat and transferred to the liquid passing through the pump.

If the pump is operating against a completely closed valve, the power losses are equal to the shutoff brake horsepower and, since no flow takes place through the pump, all this power goes into heating the small quantity of liquid contained within the pump casing. As this process occurs, the pump casing itself heats up and a certain amount of heat is dissipated by radiation and convection to the surrounding atmosphere. If the amount of heat added to the liquid is small, it can be transmitted through the casing with a low differential in temperature between the liquid in the casing and the outside air. However, if the power loss is very high, the liquid temperature might have to reach an exceedingly high value, far in excess of the boiling temperature at suction pressure, before the amount of dissipated heat equals that generated in the pump proper. Operation of the pump under such conditions would have disastrous effects.

The rate of heating the liquid in the pump casing for a given power loss depends both upon the volume of water contained in the casing and upon the surface of casing that can dissipate heat. For practical reasons, dissipation of heat by radiation can be ignored, and the temperature rise can be determined from the formula:

$$T_r/\text{Min} = \frac{42.4 \text{ BHP}_o}{W_p C_p + W_w C_w}$$

in which:

$T_r/\text{Min}$  = Temperature rise, in degrees Fahrenheit per minute

$\text{BHP}_o$  = Brake horsepower at shutoff  
42.4 = Conversion from brake horsepower into BTUs per minute

$W_p$  = Net weight of pump, in pounds

$C_p$  = Specific heat of pump metal (approximately 0.13)

$W_w$  = Net weight of liquid in pump, in pounds

$C_w$  = Specific heat of liquid (1.0 if liquid is water)

Because the temperature rise may be so

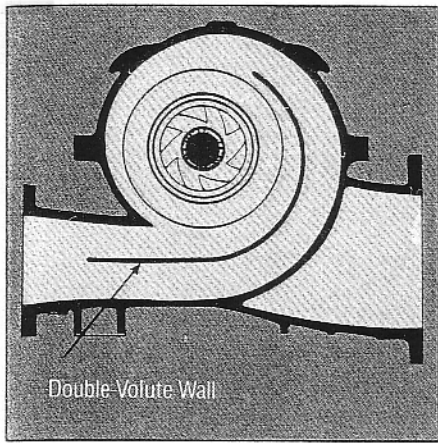
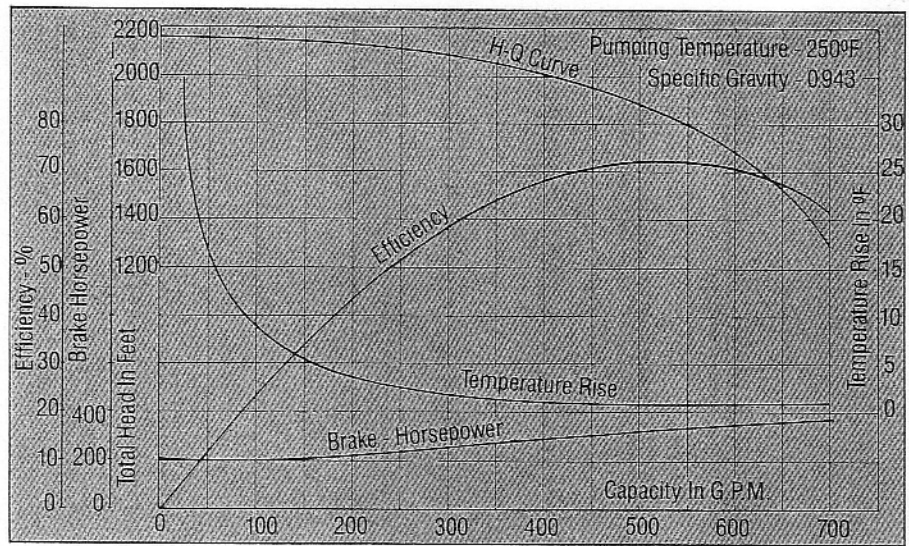


Fig. 6 (above): Cross-section of double-volute casing pump. Fig. 7 (above, right): Centrifugal pump performance curve including temperature rise curve. Fig. 8 (right): Temperature rise in centrifugal pumps related to total head and efficiency



rapid that there is no time to transmit the heat from the liquid to the pump casing, it is generally safer to neglect the casing altogether. The formula is then simplified to:

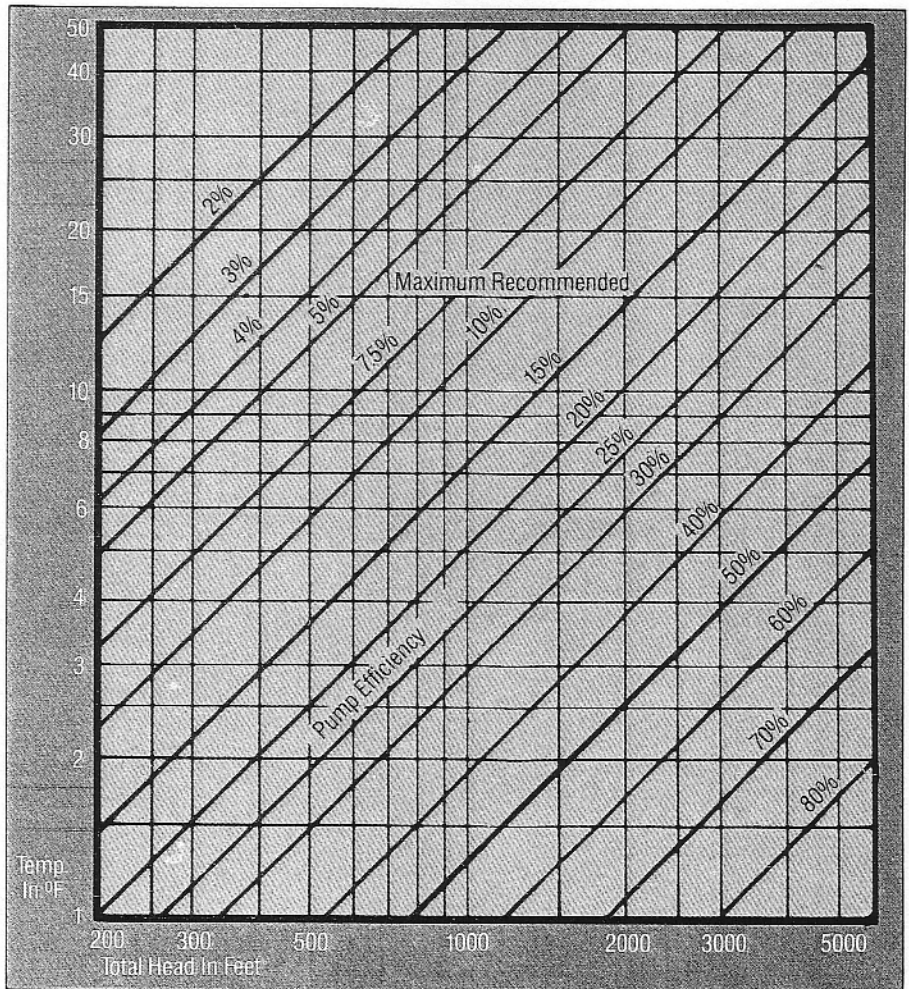
$$T_r/\text{Min.} = \frac{42.4\text{BHP}_s}{W_w C_w}$$

For example, if the pump handles water ( $C_w = 1.0$ ) and contains 100 lb of liquid, and if the brake horsepower at shutoff is 100, the water temperature will increase at the rate of 42.4°F per minute. Operation at shutoff under these conditions is very dangerous. But with a low-head, high-capacity pump that contains 5,000 lb of water and that takes the same amount of power at shutoff, the rate of temperature increase will be only 0.85°F per minute—hardly serious if the operation against a closed discharge valve is not prolonged.

If liquid is flowing through the pump, conditions become stabilized and the amount by which the temperature at the discharge will exceed the suction temperature can be calculated for any given flow. The heat transmitted to the liquid in the pump is equivalent to the product of the difference between the brake horsepower and the water horsepower by 2,545 (Btu equivalent of 1 hp-hr). The temperature rise can therefore be calculated from the formula:

$$^{\circ}\text{F rise} = \frac{(\text{BHP} - \text{WHP}) \times 2545}{(\text{Flow in lbs/hr}) \times C_w}$$

A more convenient formula relates the temperature rise to the total head and to



the pump efficiency:

$$^{\circ}\text{F rise} = \frac{\text{Total Head in Feet}}{778C_w} \left( \frac{1}{e} - 1 \right)$$

This formula can be used to plot a temperature-rise curve directly superimposed on the performance curve of a centrifugal pump (Fig. 7) which repre-

sents the characteristics of a boiler feed pump designed to handle 550 gpm of 250°F feedwater against a total head of 1,800 ft. As shown, the temperature rise increases very rapidly with a reduction in flow. This is caused by the fact that the losses at low deliveries are greater when the flow of liquid that must absorb the



heat developed in the pump is low. For example, Fig. 7 shows that for a capacity of 50 gpm, the temperature rise is 17°F, whereas, at the full capacity of 550 gpm, it is less than 1°F.

The chart in Fig. 8 gives a graphical solution of this formula for water and allows determination of the minimum permissible operating capacity, once the maximum permissible temperature rise has been selected. When liquids other than water are handled by the pump, it becomes necessary to correct the resulting answer for the difference in the specific heats of the liquids.

If the pump is fitted with a balancing device, a certain portion of the suction capacity is returned either to the pump suction or to the suction supply vessel. Thus, the discharge capacity does not represent the true flow through the pump. In addition, a temperature rise takes place in the balancing device leakage caused by the breakdown in pressure through the device. This additional temperature rise can be estimated as:

$$\frac{dT \text{ } ^\circ\text{F in balancing device} = \text{Head drop through the device}}{778}$$

778

The formula for the temperature rise and the chart in Fig. 8 can still be used, provided a correction is made to take care of the increase in pump flow representing the balancing device leakoff and temperature rise in the balancing device.

The maximum permissible temperature rise varies over a wide range, depending on the type of installation. The fundamental objective is to avoid destructive flashing within the pump.

With homogeneous liquids such as water, flashing must be avoided altogether. With multifraction liquids, such as most hydrocarbons, some flashing can be tolerated. For hot water pumps, such as boiler feed service, it was generally recommended to limit the temperature rise to 15°F, and this established the minimum permissible flow. With our expanded knowledge, other factors than temperature rise establish minimum recommended flows for these pumps, as will be seen later. When pumps handle cold water, the temperature rise may be permitted to reach 50 or even 100°F in some cases. Again, other restrictions may be used in establishing minimum flows. ■

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