



# TECHNICAL INFORMATION BULLETIN

NUMBER SEVEN

## How to Use System-Head Curves

*You can avoid some pumping pitfalls by using system-head curves in conjunction with pump performance curves. In this article a pump expert shows graphically how to analyze a number of different situations.*

A system-head curve is a graphical representation of the relationship between flow and hydraulic losses in a given piping system. Since hydraulic losses are functions of rate of flow, size and length of pipe, and size, number and type of fittings, each system has its own characteristic curve and specific values.

In virtually all pump applications at least one point on the system curve is given to the pump manufacturer in order to help him select the pump properly. In many cases, however, it is highly desirable to graphically superimpose the entire system curve over the head-capacity curve of the pump. The intersection of the pump curve with the system-head curve defines the operating point of the pump.

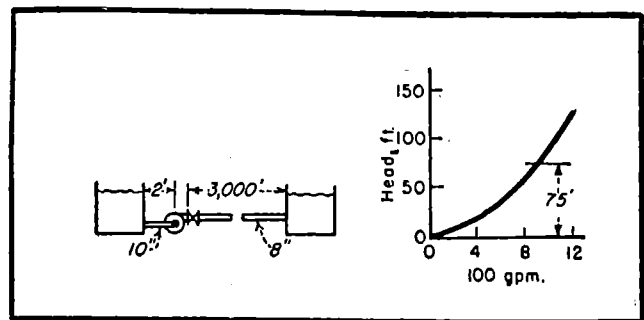
This article reviews, by typical examples, methods of calculating system heads for some common piping layouts and discusses some of the factors which are brought to light by superimposing system curves over pump performance curves.

Hydraulic losses in piping systems are composed of the following: pipe friction losses; fitting losses (valves, elbows, etc.); entrance and exit losses (these normally occur at the beginning and end of a pipeline, respectively); losses due to change in pipe size by sudden or gradual enlargement or reduction in diameter.

In the application of low-head, high-capacity pumps, such as mixed-flow or axial-flow pumps, it is imperative that, when applicable, each of these losses be accurately accounted for, because they usually represent a significant fraction of the total system head. However, in applications involving relatively

high static lift and/or relatively high pipe friction losses (long lines) it is usually unnecessary to account for losses other than pipe friction, since these other losses are an insignificant portion of the total head.

Except for the first two examples to follow, we shall take into account only losses resulting from pipe friction. However, in actual practice each application should be checked to see what is the order of magnitude of the various hydraulic losses. A decision can then be made as to what losses should be accounted for and what ones can be neglected.



### NO LIFT—ALL FRICTION HEAD

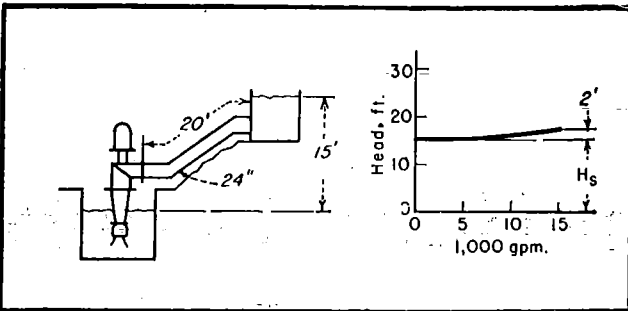
The system-head curve, in the absence of static lift, starts at zero flow and zero head. Since friction losses vary as the square of flow rate, this system curve is parabolic in shape and is a "steep" system curve.

For the case illustrated, here's how the point on the curve is calculated for a flow of 900 gpm. of water:

Entrance loss from tank into 10-in. suction pipe =  $0.5V^2/2g$  = 0.10 ft.  
 Friction loss in 2 ft. of 10-in. suction pipe = 0.02  
 Loss in 10-in. 90-deg. elbow connected to pump (equivalent to 25 ft. of 10-in. pipe) = 0.2  
 Friction loss in 3,000 ft. of 8-in. discharge pipe = 74.5  
 Loss in 8-in. gate valve fully open (equivalent to 5 ft. of 8-in. pipe) = 0.12  
 Exit loss from 8-in. pipe into tank =  $V^2/2g$  = 0.52

Total friction loss  $H_f$  = 75.46 ft.

Friction losses at other flow rates can be computed and plotted to get the system-head curve shown. If all losses were ignored except friction through the discharge pipe the total system head would not change significantly.



### MOSTLY LIFT—LITTLE FRICTION HEAD

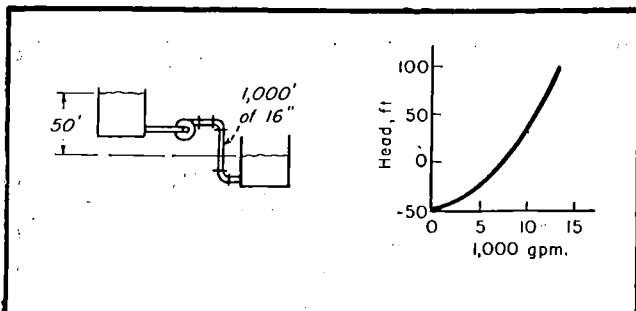
The system-head curve here starts at the static lift  $H_s$  and zero flow. Since, by assumption, friction loss  $H_f$  is relatively small compared to  $H_s$ , we get a "flat" system curve. For the case illustrated, here is the calculation of  $H_f$  for a flow of 15,000 gpm.:

Friction loss in 20 ft. of 24-in. pipe = 0.4 ft.  
 Exit loss from 24-in. pipe into tank =  $V^2/2g$  = 1.6

Total friction loss  $H_f$  = 2.0 ft.

In this example almost 90 percent of the total head of 17 ft. at 15,000 gpm. consists of static lift. However, it should be noted that neglecting the friction and exit losses would result in an appreciable error.

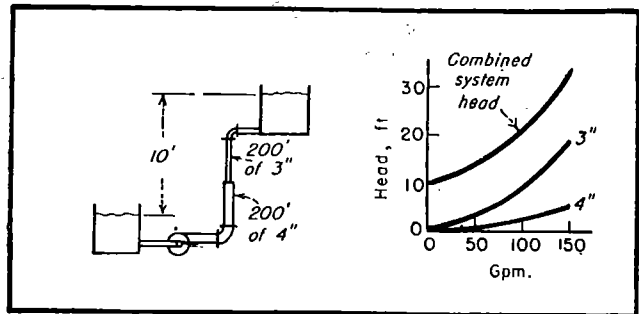
A combination of these two cases is encountered quite often, i.e., where both friction head and static lift are appreciable. The system-head curve is similar to that of the first case with the addition of the static lift.



### NEGATIVE LIFT (GRAVITY HEAD)

In this particular installation flows up to 7,200 gpm. will occur by gravity head alone. To obtain flow rates beyond this, however, requires a pump to overcome the pipe friction losses in excess of 50 ft. The following data define three points on the system-head curve:

At 5,000 gpm., friction loss in 1,000 ft. of 16-in. pipe = 25 ft.  
 At 7,200 gpm., friction loss = available gravity head = 50  
 At 13,000 gpm., friction loss = 150

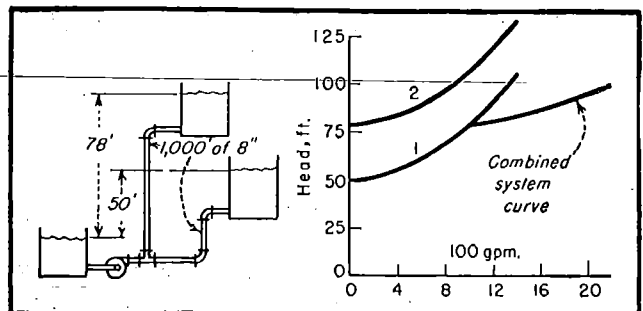


### TWO DIFFERENT PIPE SIZES

Friction losses vs. flow rate are plotted independently for the two pipe sizes. At any given flow rate the total friction loss for the system is the combined loss for the two lines. Thus the combined system curve represents the sum of static lift and the friction losses for all portions of the line.

At 150 gpm., friction loss  $H_{f1}$  in 200 ft. of 4-in. pipe = 5 ft.  
 At 150 gpm., friction loss  $H_{f2}$  in 200 ft. of 3-in. pipe = 19  
 Static lift  $H_s$  = 10

Total head  $H$  at 150 gpm. = 34 ft.

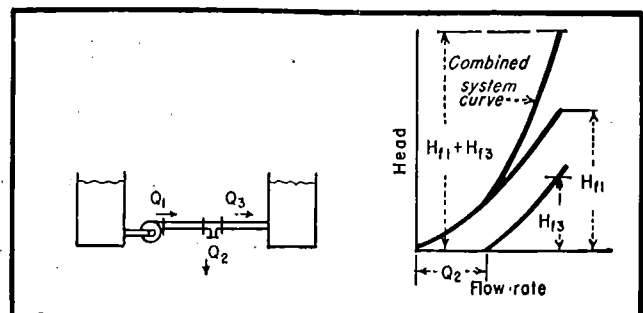


### TWO DIFFERENT STATIC LIFTS

System curves are plotted independently for the two lines. The total system curve is then obtained by adding flow rates for the two lines at the same head.

At 550 gpm., friction loss in 1,000 ft. of 8-in. pipe = 10 ft.  
 At 1,150 gpm., friction loss = 38  
 At 1,150 gpm., friction + lift in Line 1 = 38 + 50 = 88  
 At 550 gpm., friction + lift in Line 2 = 10 + 78 = 88

Therefore, the flow rate for the total or combined system at a head of 88 ft. is 550 + 1,150 = 1,700 gpm. Thus, to produce a flow of 1,700 gpm. through this system a pump capable of generating 88 ft. head would be required.



### PART OF FLOW DIVERTED ELSEWHERE

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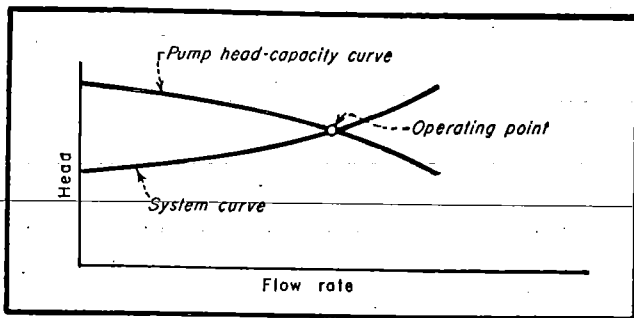
(See graph on preceding page)

In this case we assume that the quantity of liquid being tapped off at the intermediate point is constant.

Friction loss vs. flow rate is plotted in the regular manner for Line 1. The curve for Line 3 is displaced to the right at zero head by an amount equal to  $Q_2$ , since  $Q_2$  represents the quantity passing through Lines 1 and 2 but not through Line 3. The combined system curve is obtained by adding, at a given flow rate, the head losses for Lines 1 and 3.

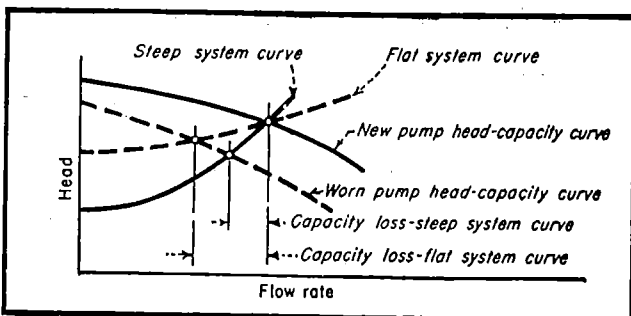
Assume  $Q_2$  is 300 gpm., Line 1 is composed of 500 ft. of 10-in. pipe and Line 3 is composed of 50 ft. of 6-in. pipe:

At 1,500 gpm. through Line 1, friction loss for Line 1	= 11 ft.
Friction loss for Line 3 (1,200 gpm.)	= 8
Total friction loss at 1,500 gpm. pump delivery	= 19 ft.



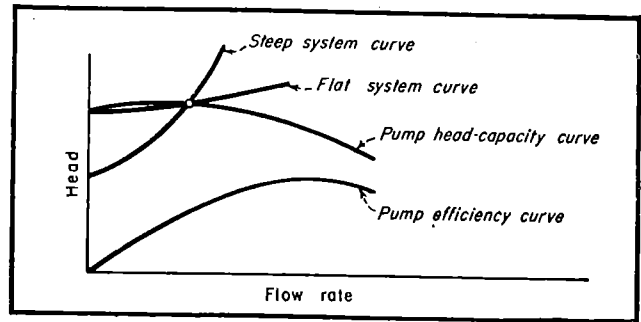
## DEFINING THE OPERATING POINT

Superimposing the system curve on the pump performance curve shows at what point the pump will operate. The shape or slope of both system curve and pump curve are significant in certain applications, as shown in several examples following.



## EFFECT OF PUMP WEAR ON CAPACITY

We are all aware that when a pump wears there is certain to be a loss in performance. The amount of loss for a given amount of wear, however, will depend to a large extent on the characteristics of the system. As shown in the graph, the loss in capacity is greater for a given amount of wear if the system curve is flat as compared with a steep system curve.

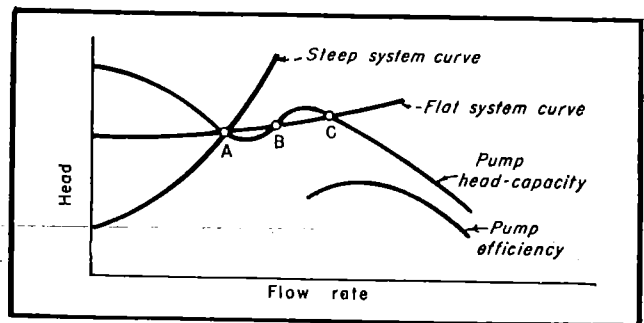


## LOW-SPECIFIC-SPEED PUMP NEAR SHUT-OFF

A characteristic of low-specific-speed pumps is a relatively flat head-capacity curve at capacities near shut-off. Occasionally it is necessary to pick such a pump because of a high-head, low-flow requirement. (Ordinarily pumps are physically capable of operating continuously near shut-off.) If this is the case the system curve should certainly be examined before specifying the pump.

It is evident from the curves that a shallow or flat system curve in conjunction with a flat pump curve can lead to performance trouble if the pump is slightly off in head-capacity or if the system curve is calculated slightly too low.

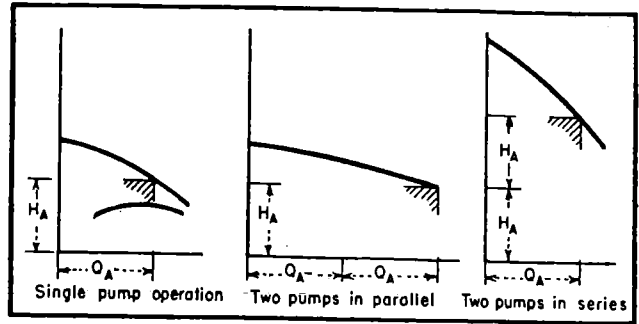
Obviously, with a steep system curve no hydraulic performance trouble would be expected.



## HIGH-SPECIFIC-SPEED PUMP NEAR THE DIP

A characteristic of some high-specific-speed pumps is that the head-capacity curve dips at capacities to the left of peak efficiency. Occasionally the need arises to operate pumps at or near the dip. In order to see whether performance under these conditions would be satisfactory let us look at the system curve.

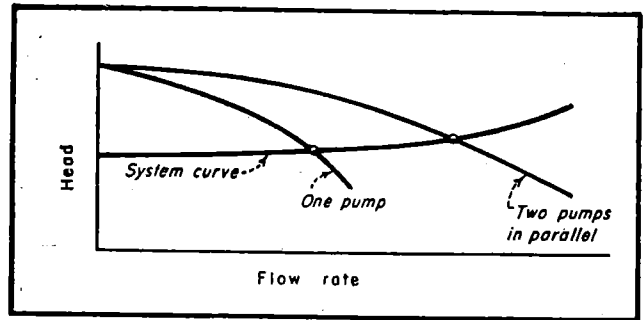
Suppose A is the desired operating point. For some transient high head condition, with a steep system curve no trouble should be encountered. A shallow system curve, however, which intersects the pump curve at three points, might result in unsatisfactory operation if the pump hunted between the three capacities.



**PARALLEL OR SERIES OPERATION**

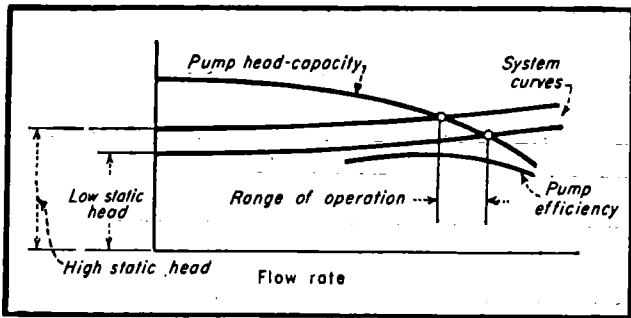
Frequently where there is a wide range in demand two or more pumps may be operated in parallel or series to satisfy the high demand, with just one of the pumps used for low demands. For proper specification of the pumps and evaluation of their performance under various conditions, the system curve should be used in conjunction with the composite pump performance curves.

For pumps in parallel, performance is obtained by adding the capacities at the same head. For pumps in series, performance is obtained by adding the heads at the same capacity.



**PUMP PERFORMANCE IN PARALLEL**

Superimposing the system curve on the pump performance curves clearly indicates what flow rates can be expected and at what heads each of the pumps will be operating.



**VARIATIONS IN THE SYSTEM HEAD**

System demand can vary because of variations in suction or discharge surface levels or greater pipe friction because of increased pipe surface roughness. The conditions shown in the graph, where static head is varied, represent a favorable operating range as far as the efficiency of this particular pump is concerned. The curves also graphically illustrate the magnitude of drop in system flow rate when high head condition occurs.



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