Surge

Surge and auto-oscillation (see next section) are system instabilities that involve not just the characteristics of the pump but those of the rest of the pumping system. They result in pressure and flow rate oscillations that can not only generate excessive vibration and reduce performance but also threaten the structural integrity of the turbomachine or other components of the system. In section (Gb) we provide more detail on general analytical approaches to this class of system instabilities. But for present purposes, it is useful to provide a brief outline of some of the characteristics of these system instabilities. To do so, consider first figure 1(a) in which the steady-state characteristic of the pump (head *rise* versus mass flow rate) is plotted together with the steady-state characteristic of the rest of the system to which the pump is connected (head *drop* versus mass flow rate). In steady-state operation the head rise across the pump must equal the head drop for the rest of the system, and the flow rates must be the same so that the combination will operate at the intersection, O. Consider, now, the response to a small decrease in the flow to a value just below this equilibrium point, O. Pump A will then produce more head than the head drop in the rest of the system, and this discrepancy will cause the flow rate to increase, causing a return to the equilibrium point. Therefore, because the slope of the characteristic of Pump A is less than the slope of the characteristic of the rest of the system, the point O represents a quasistatically stable operating point. On the other hand, the system with Pump B is quasistatically unstable. Perhaps the best known example of this kind of instability occurs in multistage compressors in which the characteristics generally take the shape shown in figure 1(b). It follows that the operating point A is stable, point B is neutrally stable, and point C is unstable. The result of the instability at points such as C is the oscillation in the pressure and flow rate known as "compressor surge."

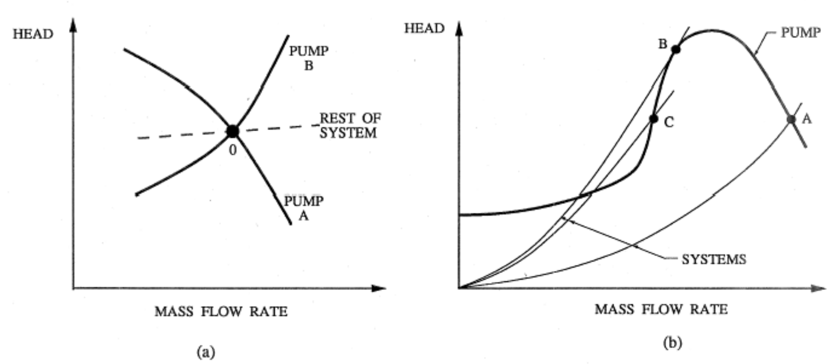

Figure 1: Quasistatically stable and unstable operation of pumping systems.

While the above description of quasistatic stability may help in visualizing the phenomenon, it constitutes a rather artificial separation of the total system into a "pump" and "the rest of the system." A more general analytical perspective is obtained by defining a resistance, R_i^*, for each of the series components of the system (one of which would be the pump) distinguished by the subscript, i:

$$R_i^* = \frac{d(\Delta H)}{dm} \tag{Mbff1}$$

where ΔH is the quasistatic head drop across that component (inlet head minus discharge head) and is a function of the mass flow rate, m. By this definition, the slope of the pump characteristic in figure 1(a) is $-R^*_{PUMP}$, and the slope of the characteristic of the rest of the system is R^*_{SYSTEM}. It follows that the earlier established criterion for stability is equivalent to

$$\sum_i R_i^* > 0 \tag{Mbff2}$$

In other words, the system is quasistatically stable if the total system resistance is positive.

Perhaps the most satisfactory interpretation of the above formulation is in terms of the energy balance of the total system. The net flux of energy out of each of the elements of the system is $m(\Delta H)_i$. Consequently, the net energy flux out of the system is

$$m \sum_i (\Delta H)_i = 0 \tag{Mbff3}$$

which is zero at a steady state operating point.

Suppose the stability of the system is now explored by inserting somewhere in the system a hypothetical perturbing device which causes an increase in the flow rate by dm. Then the new net energy flux out of the system, E^*, is given by

$$E^* = dm \left[\sum_i (\Delta H_i) + m \frac{d \sum_i (\Delta H_i)}{dm} \right] \tag{Mbff4}$$

$$E^* = m\ dm \sum_i R_i^* \tag{Mbff5}$$

where the relations (Mbff1) and (Mbff3) have been used. The quantity E^* could be interpreted as the energy flux that would have to be supplied to the system through the hypothetical device in order to reestablish equilibrium. Clearly, then, if the required energy flux, E^*, is positive, the original system is stable. Therefore the criterion (Mbff2) is the correct condition for stability.

All of the above is predicated on the changes to the system being sufficiently slow for the pump and the system to follow the steady state operating curves. Thus the analysis is only applicable to those instabilities whose frequencies are low enough to lie within some quasistatic range. At higher frequency, it is necessary to include the inertia and compressibility of the various components of the flow. Greitzer (1976) (see also 1981) has developed such models for the prediction of both surge and rotating stall in axial flow compressors.

It is important to observe that, while quasisteady instabilities will certainly occur when $\sum_i R_i^* < 0$, there may be other dynamic instabilities that occur even when the system is quasistatically stable. One way to view this possibility is to recognize that the resistance of any flow is frequently a complex function of frequency once a certain quasisteady frequency has been exceeded. Consequently, the resistances, R_i^*, may be different at frequencies above the quasistatic limit. It follows that there may be operating points at which the total dynamic resistance over some range of frequencies is negative. Then the system would be dynamically unstable even though it may still be quasistatically stable. Such a description of dynamic instability is instructive but overly simplistic and a more systematic approach to this issue requires the methodologies of sections (Gbc).