Dr. Lev Nelik, P.E., APICS

What Does Minimum Flow Have to Do With "L^3/D^4"?

Our repair shop came upon a customer that purposely buys oversized pumps and immediately operates them at nearly 80 percent closed valve. Each year, after the hot days of summer simmer down, they evaluate the pumps' performance by measuring flow. If the flow dropped by more than ~10 percent, they slightly open the valve (say 70 percent closed) to compensate and run the pump at this new valve position the next season.

They felt proud of this "pro-active" maintenance program. They pumped mostly clean water, did not worry much about spills and catastrophic failures, and periodically changed packings. Despite vibration levels being around ~0.7-in/sec (pumps were held to the sole plates by only half the number of bolts required, and some sole plates did not even have holes drilled), they did not feel they had any problems.

Each year, several pumps would reach the point where a valve could not open any further. These units were sent to be overhauled. Once the pump returned from the repair shop, the valve was again set almost closed and the story was repeated.

In this profitable privately-owned company, the same mechanics that took care of the pumps also paved the driveways, fixed roofs, and worked on employee cars in their shop. The pumps were rather noisy, but were installed inside the buildings where anyone rarely went (other than to periodically open the valves a bit more). This noise didn't bother anyone, but for those who did complain, the advice was not to listen, or to use plugs.

Another pump repair shop that serviced these folks was regarded as friendly, reasonable and always willing to pick up a worn out pump at the end of each season on time. The only reason we got involved was by accident. At first, it was rather difficult to discuss the concept of BEP, MCSF, vibration criteria, and the need to actually bolt the pumps down so they didn't just sit there like rocket launches held mostly by the flange connection. We talked about the dangers of running pumps significantly to the left of the BEP, which brought us to the question: "What is MCSF"?

Minimum Continuous Stable Flow (MCSF) is defined as flow below which the pump should not be operated continuously. It can operate there for a short time (such as start-up), but not too long. The main reasons are unstable operation, high radial and axial thrust, vibrations and noise – all of which ultimately deflect the shaft; damage the seal, bearings, and couplings; and reduce reliability significantly. The value

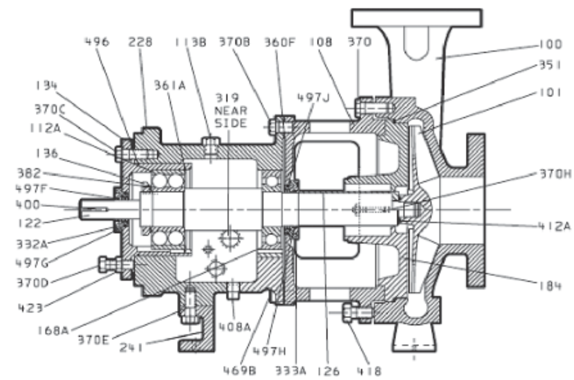

Figure 1. A typical end suction pump design: the shaft is relatively slender, with a high value of L^3/D^4.

MCSF is established by the pump manufacturer and guided by several factors, including pump type, pump energy level, pump Ns and Nss and onset of recirculation, guiding specifications, and experience.

For critical, more sophisticated designs, on-set of suction recirculation often is a main factor.[1] For simpler cases, field feedback and experience present opportunities for simplified methods.[2] To further clarify things, some specifications simply state the allowable *minimum* flow (as well as *maximum* flow, on the other side of the BEP). For example, API-610 spec requires no less than 60 percent as allowable, and no less than 70 percent as preferred.[3]

Regardless of which method is used, the main objective is to prevent the pump shaft from deflecting excessively and causing seals to leak, bearings to overload, and the coupling to overstress. It is *intuitively* (I must be careful here!) known that shorter and thicker shafts resist deflecting force better, thus such shaft designs are preferred. The measure of shaft "robustness" (its resistance to deflection, stiffness) is a so-called ratio, L^3/D^4.

This ratio comes from the deflection formula: $y = WL^3 / 3EI$, or deflection at the end of a cantilevered shaft. Since the moment of inertia for a round bar is $I = \pi \times d^4 / 64$, then substituting we get: $y = (64W / 3E\pi) \times L^3/D^4 = k \times L^3/D^4$. Thus, for the same load, the lower the quantity (L^3/D^4), the lower the deflection. Short, beefy shafts have *lower* L^3/D^4, which is a good thing.

Consider a typical end suction ANSI design below (see Figure 1), a 1.5 x 1-6. The shaft diameter under the sleeve is 1.125-in, and the length of the shaft from the bearing to impeller centerline is approximately 7-in, thus $L^3/D^4 = 7^3/1.125^4 = 214$. Typically, end suction pumps have this "slenderness ratio" between 10 to 300, the lower the better. One of the reasons the design in Figure 1 has high value (214) is because it allows for a double mechanical seal, which requires extra room to fit. Had the design been intended for a single seal only, the shaft could have been shortened, to lower the ratio, but then seal options would be limited.

The radial hydraulic thrust is almost non-existent when a pump operates near its BEP, but increases rapidly towards the shut-off by the approximate formula[2]: $R = k \times H \times D \times b / 2.31$, where H is pump head in feet, D and b are impeller OD and width at the OD (in inches), and k an empirical factor. For single volute pumps, $k = 0.36 \times [1 - (Q/Q_{BEP})^2]$ (same ref. [2], although even higher k-values were reported as well). For the 1.5 x 1-6 design, $D_{max} = 6.06$", b ~ 1.0". Consider the pump performance in Figure 2.

We will not use a rated point (which is application specific) at reduced diameter, but instead consider a full diameter scenario. At the BEP (100-gpm), the pump head is 130-ft, but radial thrust is zero, since k = 0. At the shut-off (155-ft), radial thrust is maximum (k = 0.36): $R = (0.36 \times 155 \times 6.06 \times 1.0) / 2.31 = 146.4$ lbs, and shaft deflection at the impeller is: $y = 147 \times 7^3 / [3 \times (30 \times E{+}6) \times 3.14 \times 1.125^4) / 64] = 0.007$ in.

Deflection at the seal is somewhat less and can be calculated by the similar formula, which produces a value roughly half of that at the impeller centerline, i.e. 0.0035-in. As most seal manufacturers recommend, less than 0.002-in deflection is allowed at the seal faces for their proper, non-leaky, operation. Obviously in our example, near shut-off operation produces deflection almost double the value allowed by seal manufacturers. It is also possible to back-calculate the flow at which deflection at the seal would be exactly 0.002-in, by making the following tabulation (see Table 1).

Valve at	flow-ratio	Head, ft	k	radial load, lbs	y, in @imp	y, in @seal
shut-off	0%	155	0.36	146.43	0.0071	0.0036
	50%	150	0.27	106.28	0.0052	0.0026
	60%	145	0.23	87.67	0.0043	0.0021
	75%	140	0.16	57.86	0.0028	0.0014
fully open (BEP)	100%	130	0.00	0.00	0	0

Table 1

Table 1 shows that when the valve is closed to choke the flow to about 60 percent of the BEP, deflection at the seal reaches limiting value. Thus the pump should not be allowed to operate below this flow. The OEM performance curve shows MCSF is allowed to be as low as 10-gpm (less then 10 percent of the BEP), where point shaft deflection at the seal exceeds the limit; albeit for small, low energy pumps, this rule appears to be relaxed. (In defense of the OEM, no known studies show a relationship between the *deflection* at the seal faces and *seal life*, and a 0.002-in rule of thumb is the only known make-it-or-break-it criteria.[4])

For these reasons, larger pumps typically utilize a *double volute*, for which radial thrust is more balanced and shaft deflections reduced significantly. A concentric volute is another example of reducing shaft deflections, but with some sacrifice of pump efficiency.

A modification of the ANSI end suction design recently entered the market, where rotor design allows a much greater variation of the operating flow, significantly below a BEP point. However, it requires a case-by-case evaluation of the system parameters to ensure the dimensional envelope allows exact retrofit of a problematic pump, which the new design replaces.

By the way, whatever happened to that customer? Nothing so far. But I will let you know in a few years.

P & S

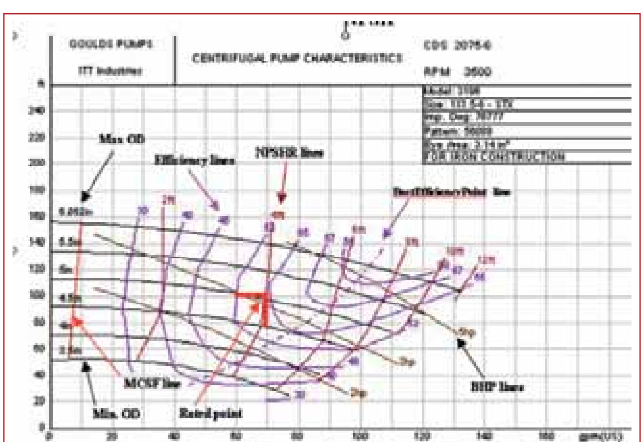

Figure 2. Typical performance characteristics of the end suction ANSI pump, size 1.5 x 1-6. *Source: 2004 Goulds Pumps Manual, ITT Industries*

References

1. Frazer, H., "Flow Recirculation in Centrifugal Pumps," presented at ASME meeting, 1981.
2. Stepanoff, A., *Centrifugal and Axial Flow Pumps*, 2nd edition, 1957
3. API-610, *Centrifugal Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services*, Washington, DC.
4. Hydraulic Institute, *Mechanical Seals for Pumps: Application Guidelines*, Parsippany, NJ, 2006.

Dr. Nelik (aka "Dr. Pump") is president of Pumping Machinery, LLC, an Atlanta-based firm specializing in pump consulting, training, equipment troubleshooting, and pump repairs. Dr. Nelik has 30 years experience in pumps and pumping equipment. He has published over 50 documents on pump operations, the engineering aspects of centrifugal and positive displacement pumps, and maintenance methods to improve reliability, increase energy savings, and optimize pump-to-system operations. With questions, comments, or to attend his Pump School, he can be contacted at www.PumpingMachinery.com.