WORLD PUMPS February 2014

Oil & gas

Refinery specs: the user's view

Users of centrifugal pumps for refineries often tailor their specifications with additions to the well-known API 610 standard. However, D. Chaware and A. Joshi have found that some additions can prove deleterious. They discuss the shortcomings of some of the most frequently used addenda and offer resolutions.

Pump users and consultants often formulate their own specifications for the purchase of centrifugal pumps for refinery and petrochemical projects. These specifications are normally prepared with supplements or addenda to the widely used ANSI/API 610 standard[1] published by the American Petroleum Institute (API). The primary reasons for users to make such additions to this standard are:

a. To enhance the safety and reliability of the pump.

b. To indicate the user's choice against clauses in ANSI/API 610[1] marked with a bullet (•).

c. To spell out special requirements about the pump vendor qualifications and required product performance track records.

d. To clarify the intent of various clauses to avoid unnecessary debates about their interpretation between users and manufacturers.

e. To cover special plant and applicable statutory requirements.

These addenda are based on the users' experience and understanding of the equipment, recent developments reported in the literature (e.g. Refs 2 and 3)* but not yet captured in the API 610 standard[1], and sometimes on 'myths' about the increase in reliability due to certain additional factors.

*The full list of references is available from the authors on request.

It is observed that such additions can restrict the selection choice, resulting in the purchase of a pump with a higher life cycle cost (LCC) and changes in plant piping layout. The rejected manufacturers become offended as their offers are rejected for no sensible technical reasons. The drawbacks of some of the frequently used additions are discussed below, with suggested resolutions.

Suction-specific speed limits

Clause •6.1.9 of the API standard[1] refers to the suction-specific speed (S) of the pump and offers an option to users to impose a limit on the S value. Most of the addenda impose a limit on S of 11,000 expressed in US units (flowrate in USgpm, net positive suction head required (NPSHR) in ft, speed in rpm) for direct acceptance. Where S is >11,000, the pumps may be rejected, or some of the addenda recommend technical deliberations on pump reliability with manufacturers to determine acceptance.

Most probably this benchmark of 11,000 is based on Hallam's analysis of failure data from a refinery[4], which indicated that the failure rates are higher for pumps with S > 11,000. This higher failure rate is attributed to the presence of suction recirculation at relatively higher flow rates. Hallam's paper mentions only one design attribute: pump specific speed (flowrate in USgpm, head in ft and speed in rpm) < 3,000 rpm. The paper is silent about design attributes such as the shaft stiffness as referred to in Annex K of ANSI/API 610[1], the material composition of the pump components, bearing details, volute casing type, operating parameters and range, pumpage properties and the design year or decade. The importance of this paper, however, cannot be underestimated as it has highlighted the serious lack of product reliability[5]. In short, the limiting value of 11,000 – though unsupported by any systematic and scientific laboratory tests conducted under control conditions – is often mentioned in the users' specifications, though with an option for further deliberation.

Based on this benchmark, selection engineers find it easy to reject pumps with S greater than 11,000, conveniently forgetting that such pumps also work satisfactorily. The recommendation for further deliberation with the manufacturer receives little consideration, due either to lack of expertise to evaluate the design offered or to a demanding project schedule. Thus, in the process, better pumps are rejected. In some cases this benchmark is lowered to 8,500: a value most probably based on its appearance in the Hydraulic Institute's standard for centrifugal pumps[6] and its citation in the literature[7,8].

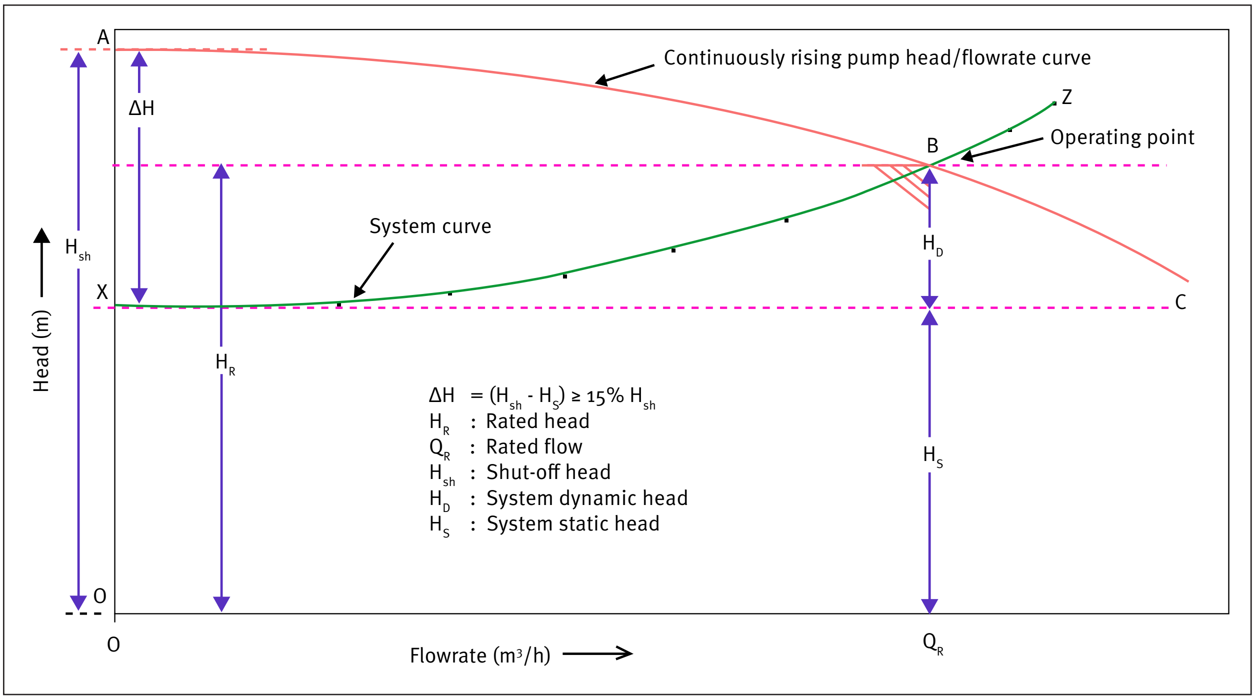

Figure 1. Condition to determine the permissible pump head rise.

It is noted that imposing such simplistic benchmarks can result in a higher LCC. This LCC escalation is due to:

1. Selection choice restrictions. We carried out a survey covering 522 pumps of overhung OH2 type as per ANSI/API[1] from different manufacturers, which indicated that with benchmarks of S <8,500 and <11,000 the respective selection choice would be reduced to 12% and 86% of the total.

2. Myth-based selection. The postulate that pumps with S below 11,000 or 8,500 are reliable is dangerous in view of the fact that pumps with S less than 11,000 or 8,500 have also failed on site.

3. Lower values for S. It is noted that certain manufacturers with capabilities for reliable designs with higher S values have accepted this limitation as a design criterion, thus setting the trend to design pumps with lower S. This forces users to go for costly options such as slow and larger machines or a greater number of machines, or costlier measures to increase available NPSH.

On the other hand, a scientific study by Stoffel and Jaeger[9] clearly indicated that it is possible to design and manufacture reliable pumps with S > 11,000. This fact is supported by the many pumps with S > 11,000 operating satisfactorily, though no published statistics are available. The credibility of such benchmarks is thus doubtful, and the following measures are recommended:

1. Do not impose such simplistic benchmarks related to S in the specification.

2. Minimum continuous stable flow (MCSF) in relation to the recirculation flow value, to decide on the operating range for a given application, should be based on experiments and not on theoretical calculations. Several approaches for experimental determination mentioned in the literature[10–12] may prove useful. We have successfully used the approach mentioned by Fraser[10] for checking suction recirculation in relation to MCSF to rectify a problem related to cooling tower pumps.

3. Compare shaft stiffness factors to determine the soundness of the design. This comparison gives a fairly good idea about the design, as discussed later in this article.

4. Carry out a design review of models available with the prospective vendors before granting approval. Literature dealing with design review guidelines is available[13–16]. Superior metallurgy for the impeller is recommended. The guidelines[17], supported by field performance, from vendors can be used, though they are most likely to be suitable for products developed by a particular vendor.

5. Check the pump's track record for identical applications and the reputation of the vendor[18,19].

Implementing the above steps may sound cumbersome initially but such exercises are one-time requirements and can benefit the user in the long run.

Continuous head rise

Clause •6.1.11 concerns the preference for 'stable' head/flowrate curves (a continuous head rise to shut-off) in general and as a requirement for parallel operation. The word 'stable' appears to be a misnomer since all head/flowrate curves have unstable zones. For parallel operation, a minimum rise of 10% from rated flow (not normal flow!) to shut-off or zero flow is specified. It is noted that many addenda specify a rise of 5% for single pump operation and a minimum of 10% for pumps in parallel. Many promising offers have been rejected just because the head rise was a little short of the required 5% or 10%. On the other hand, we have observed pumps with a 1–2% head rise that are running in parallel without any problem. Communication with the API revealed that the guideline of a 10% head rise was a conservative purchase requirement. The API further mentioned that system characteristics are not a purchase issue and hence no mention of it is made.

This conservative approach of a minimum head rise of 5% or 10% without considering

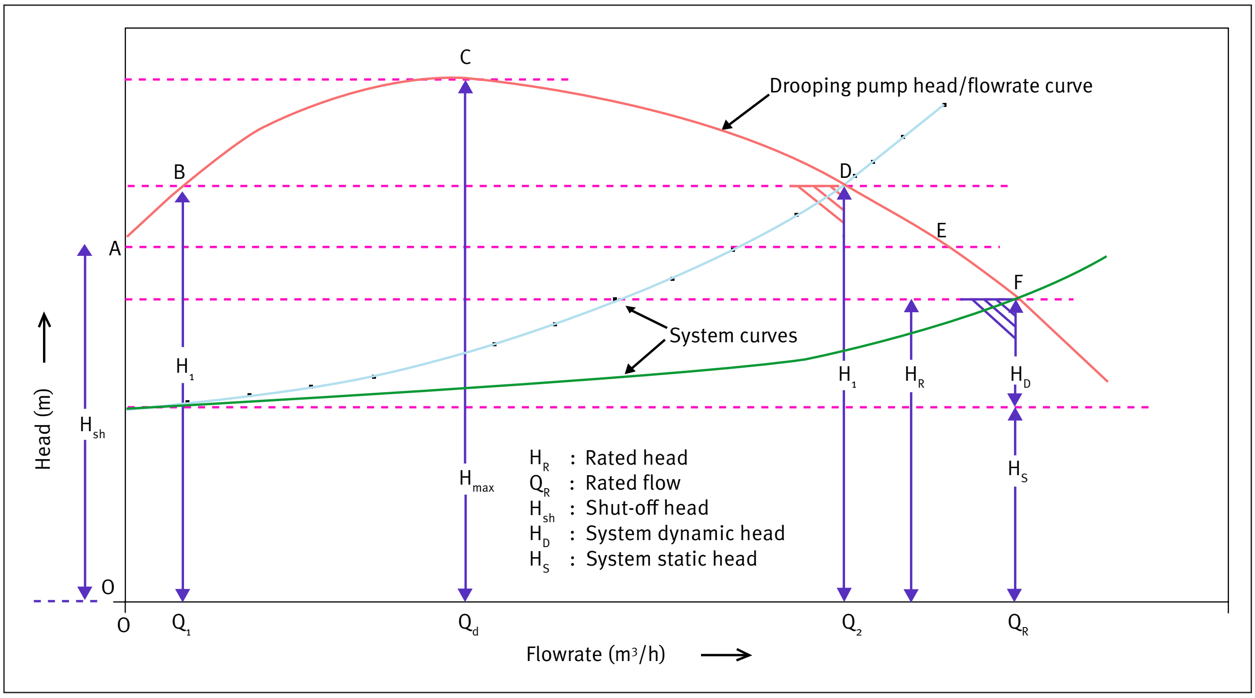

Figure 2. Condition to determine acceptance of a pump with a drooping head/flow rate curve.

the system characteristics may not be to the advantage of users as it restricts the selection. It is important to note that in refinery and petrochemical plants most of the pumps usually operate continuously at flowrates lower than the rated flow. In such operations pumps with relatively flat head/flowrate curves can prove energy-efficient in comparison with those with steep curves[20].

The practical approach to making a decision about head rise is illustrated in Figure 1, where the system resistance curve (XBZ) is plotted and superimposed on the pump head/flowrate curve (ABC). A pump – with or without <10% continuous rise in head – can be accepted as long as the difference, ΔH, between pump shut-off head, H_{sh}, and system static head, H_S, is more than 15%. This conservative margin of 15% is based on the maximum permissible tolerance of 10% given in Table 16 of the API 610 standard[1]. The user can reduce it further in view of selecting an energy-efficient pump with a flat curve[20] or one readily available from the vendor, as long as the user is sure that the minimum pump shut-off head is higher than the maximum system static head.

This 'preferred' requirement of a continuously rising head/flowrate curve is often treated as mandatory, leading to the rejection of pumps with drooping head/flowrate characteristics: for example, curve ACDF in Figure 2 with peak head (H_{max}) at a capacity (Q_d) other than zero flow. We have observed many cases where manufactured pumps were rejected by users because a minor droop was noticed during testing. It is feared that pumps with drooping head/flowrate curves are prone to operate at two flows (Q_1,Q_2) at the same head, $H_1 > H_{sh}$ (Figure 2). This operation at two flows (Q_1,Q_2) occurring at the same head (H_1), which is also referred to as 'hunting', is simply not possible[21], as there can only be one operating point where the system resistance curve intersects the pump head/flowrate curve, as long as the system static head is less than the pump shut-off head. Also, the instability phenomenon in compressors due to fluid compressibility is not possible in pumps. Site experience has proved that pumps with a drooping curve can work without any problem. For single pump operation, the approach mentioned above requiring a difference of 15% or less between shut-off head and system static head can be used. In the case of parallel operation, in addition to the above approach, it is necessary to take into account the severity of the droop and to enlist the help of the vendor. It should be noted that pumps with a drooping curve are usually more efficient, simple in construction, compact and cheaper[21,22].

Clause •6.1.11 further mentions the use of an orifice plate to achieve a continuous rise to shut-off. It is to be noted that an orifice plate only changes the slope of the head/flowrate curve; it cannot remove the droop. In view of the above approach considering system resistance along with the head/flowrate curve, the use of an orifice plate, with its associated energy loss, may not be required.

Dynamic balancing grades

Clause •6.9.4.4[1] gives users the option to specify ISO 1940-1 grade G1, instead of grade G2.5 as per clause 6.9.4.1[1], for the dynamic balancing of components such as the impeller, balancing drums and similar major rotating components. Based on this option, users recommend G1 for components such as impellers and coupling halves, with a view that this will reduce vibration level and improve reliability. Such costly precision grades may not be required, as discussed below.

Dynamic balancing only addresses the mechanical unbalance, whereas the rotodynamic behaviour of the pump is determined by hydraulic and mechanical forces. Based on the literature[23], it is clear that in the case of impellers the hydraulic unbalances are dominant and are about five times the magnitude of the mechanical unbalance according to G2.5, considering precision-cast components. For sand-cast components, which are normally used for pumps, the ratio can be much higher. Reducing the residual mechanical unbalance by imposing precision grades such as G1, which are not repeatable, is therefore not going to result in the smooth running of the pump. It is interesting to note that clause •7.2.3[1] gives an option of G6.3 for the coupling. As per ISO 1940, the grade recommended for the rotor is G6.3. We have not noticed any reliable correlation between the precision grade and vibration readings during performance tests.

In view of the above, it is clear that imposing a costly precision grade such as G1 is not justified in most cases, unless recommended by the manufacturer. The following guidelines are suggested as long as the vibration levels are as per Table 8 in the API 610 standard[1]:

1. Grade G2.5 for pump components and G6.3 for rotors up to a speed of 3,000 rpm.

2. Grade G6.3 is recommended for couplings. Balancing individual components of the coupling to G1 is practically difficult and hence such a requirement should not be imposed.

3. For speeds above 3,000 rpm, grade G1 may be adopted for pump components and rotors with interference-fit components after consulting the manufacturer.

Shaft stiffness factor

Clause• 9.1.1.3[1] refers to Annex K, relating to the shaft stiffness factor (SFF) for overhung single-stage pumps. SFF requirements for other types with between-bearings and vertical suspension are as yet not addressed. Using the SFF to evaluate the design can prove very beneficial[24] and is strongly recommended. Clause 6.9.1.3[1] mentions limiting shaft deflection to achieve seal reliability by shaft stiffness. It is interesting to note that the use of SFF for design reviews was suggested more than two decades ago[25] but is rarely mentioned in users' standards. A higher SFF value indicates a slender shaft design, which is more prone to failures at flows away from BEP.

We carried out a comparison of SFF values for a standard series of overhung pumps from four different manufacturers, which clearly showed that one of the vendors was offering pumps with a relatively higher SFF. Further investigation proved that this pump series was originally designed according to the 6th edition of API 610 but was offered to beat the competition. It is therefore strongly recommended to compare the SFF values of pumps offered by different vendors, as per API guidelines[1]. Expressing SFF as a function of QH/n (flow×head/rotational speed) at BEP rather than at the rated point in API 610[1] is very insightful.

Sound measurement in tests

Clause• 8.3.4.5[1] concerns the situation where the user insists on sound level measurements during the performance test. It is practically impossible to carry out a sound level measurement test properly on a pump on a test bed designed for hydraulic performance. This is because noise from sources such as the driver of the pump, valves and piping, and the noise originating from other machines located on the shop floor are practically impossible to isolate from the measurement. A practical approach would be to ask the manufacturers to submit sound level data pertaining to the pump based on tests carried out under controlled conditions under the guidance of acoustic experts for evaluation purposes.

Stringent test tolerances

Table 16 of Clause •8.3.3.3[1] specifies performance test tolerance levels for acceptance at the pump manufacturer's shop, with an option in clause •8.3.3.4 to change these levels for pumps with a power rating >1 MW. It is noted that users recommend no negative tolerance for rated and shut-off head, no positive tolerance for power, etc. This stringent rescheduling of the tolerances by users is done in the hope of increasing the accuracy and reliability of the test.

Such measures, especially allowing no negative tolerance on the rated and shut-off head, have proved futile as they do not take into account the inherent variations in performance due to innate variations in the as-cast dimensions of the casing and impeller, dimensional variations affecting the close running clearances and the inherent limitations of random and systematic errors of the measuring equipment. With such constraints, additional efforts to fulfil stringent tolerance requirements entail rework and repeated testing, adding to the pump cost and delivery time. Sometimes the failure of these efforts demands raising the performance deviations and adds further costs. It is interesting to note that the data from which the rated head and flowrate are calculated may be relatively inaccurate. Also, most of the processes may not really require pumps with stringent tolerances. In view of all these facts, a practical approach is to follow the requirements of Table 16[1] for rated and shut-off heads.

Conclusion

The typical cases discussed here indicate that some of the additions to the basic API 610 standard[1] can ultimately prove costly to the user. A pragmatic approach, taking the requirements into consideration, is needed. ■

Bibliography

[1] American Petroleum Institute, *Centrifugal pumps for petroleum, petrochemical and natural gas industries*, ANSI/API Standard 610, 11th Edition, (September 2010).

Contact

Deepak Chaware, general manager – mechanical
Essar Oil Ltd
Refinery Site, 39 KM, Jamnagar-Okha Highway
Vadinar 361305, Gujarat, India
Tel: +91 9909992483
Email: Deepak.Chaware@essar.com
www.essar.com

Avinash Joshi, consultant: centrifugal pumps
Kirloskar Ebara Pumps Ltd
Pune, India 411016
Mobile: +91 9890100758
Email: as.joshi@kirloskarebara.co.in

International Rotating Equipment Conference

This paper was first presented at the 2nd Pump Users International Forum held in Düsseldorf, Germany, September 2012, and is reproduced with permission from VDMA eV.