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Whenever a new edition of a global specification such as the International Organization for Standardization (ISO) or the American Petroleum Institute (API) is released, there is often confusion about the explicit details and rationale of the changes made. This article will address changes that have been incorporated into the new ISO 13709 2nd Edition - ANSI/API Standard 610 11th Edition, Centrifugal pumps for petroleum, petrochemical and natural gas industries. It will specifically discuss significant changes that will impact pump reliability as well as other key changes.

Background. Before reviewing these changes, it is important to understand the background surrounding this new API 610 edition. Developed cooperatively with the ISO 13709 2nd Edition, which was released on December 15, 2009, the API 610 11th Edition was completed in July 2010 and was completed in July 2010 and published in September. The ISO 13709 and API 610 documents are identical in content with the exception of a few minor editorial corrections in the API 610 version.

API documents are routinely updated every five years. The API 610 9th Edition was released in January 2003 and reissued as the 10th edition in October 2004. The API taskforce/ISO Working Group began updating efforts on the 11th edition in 2006. The team addressed the latest developments concerning rotating equipment, including reliability issues, industry issues and proposed changes based upon proven engineering and operating practices. Collaboration with other industry groups such as the Hydraulic Institute, International Electrotechnical Commission (IEC), National Electrical Manufacturers Association (NEMA) and ASTM ensured that the 11th edition reflected those organizations’ latest updates.

Historically, new editions have transitioned into worldwide usage over a period of approximately two years. During this timeframe, engineers typically choose to make purchases that embrace a new edition’s changes, especially those affecting equipment reliability.

Changes addressing reliability. For the new API 610 11th Edition, there are three significant changes with potential impact on pump reliability. These include:

- The addition of Annex K Section K.2 Bearing “system” life considerations for OH2, OH3, BB1, BB2, and BB3 pumps
- The expansion of Torrional analysis, rewritten to explain when each type of analysis is required.

Annex K Section K.1 Shaft Stiffness Guidelines for Overhung Pumps. The API taskforce/ISO Working Group received input from a number of users/contractors who reported evaluating shaft stiffness and discovering wide shaft flexibility among manufacturers. The shaft flexibility index was developed as a straightforward tool to evaluate a true API pump design vs. one that is purportedly labeled API but does not meet the standard’s design requirements.

Shaft stiffness became the differentiator. Fig. 1 shows a simple overhung rotor with \( D \) equaling the shaft diameter under the mechanical seal sleeve and \( L \) equaling the distance from the impeller centerline to the radial bearing. Shaft flexibility index or \( ISF \) in its shortened expression is \( ISF = L^3/D^4 \).

Fig. 2 shows historical data from various overhung pumps. In this figure, \( K_s \), the pump “sizing” factor, is equal to a pump’s BEP flow x TDH/rotating speed. This illustration charts smaller pumps as having higher \( L^3/D^4 \) than larger pumps. The guideline for \( L^3/D^4 \) is that as long as a pump is below the line of shaft stiffness it is following industry practice.

If \( L^3/D^4 \) exceeds the line by 20%, then the customer should seek justification from a pump manufacturer for its design. Fig. 2 represents modern design overhung pumps. Some of the overhung designs reviewed in constructing Fig. 2 actually exceeded the
TABLE 1. Old vs. new OH2 designs

<table>
<thead>
<tr>
<th>Refinery</th>
<th>QTY conversions</th>
<th>QTY repairs before upgrade</th>
<th>Run time, before upgrade</th>
<th>QTY repairs after upgrade</th>
<th>Run time, after upgrade</th>
<th>MTBR, before upgrade</th>
<th>MTBR, after upgrade</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>16</td>
<td>129</td>
<td>2,114</td>
<td>11</td>
<td>511</td>
<td>16</td>
<td>46</td>
</tr>
<tr>
<td>B</td>
<td>9</td>
<td>73</td>
<td>1,458</td>
<td>3</td>
<td>265</td>
<td>20</td>
<td>88</td>
</tr>
<tr>
<td>C</td>
<td>19</td>
<td>149</td>
<td>3,103</td>
<td>10</td>
<td>471</td>
<td>21</td>
<td>47</td>
</tr>
<tr>
<td>Total</td>
<td>44</td>
<td>351</td>
<td>6,674</td>
<td>24</td>
<td>1,248</td>
<td>19</td>
<td>52</td>
</tr>
</tbody>
</table>

FIG. 2 Overhung pump shaft flexibility vs. size (USC unit s.) (Source: ISO 13709 2nd Ed./API 610 11th Ed. Appendix K Figure K.3)

$L^3/D^4$ shaft stiffness line by a factor of 10. It is important to realize that providing the customer with the $L^3/D^4$ ratio is an “if specified” bulleted item in API.

Table 1 shows the differences between previous OH2 designs with characteristic long slender shafts and extremely high $L^3/D^4$ values and today’s robust designs with shorter shaft spans, larger diameters and very low $L^3/D^4$ values. In this sampling, the mean time between repair (MTBR) improved from < two years to > four years. (Values derived by dividing run time by number of repairs.) This table illustrates the importance of shaft stiffness evaluations and the positive impact that properly designed pumps can have on MTBR performance.

Annex K Section K.2 Bearing ‘system.’ Life considerations for OH2, OH3, BB1, BB2 and BB3 pumps are discussed here. For decades API 610 has required each individual bearing to be designed for a life of 25,000 hours (i.e., three years) continuous operation (at rated flow) and 16,000 hours at maximum radial and axial loads, typically minimum continuous stable flow. Recent work on other API standards raised the issue of this being inconsistent with the API requirement for pump design to be suitable for a three-year uninterrupted run.

The identified problem is that “system” life is shorter than the shortest life of the individual bearings in the system. For years, all manufacturers have treated this requirement as applicable to “each” bearing instead of the bearing “system.” The 11th edition has added a new formula to this section that calculates system life, noting that the combination of both radial and thrust bearings (system) should comply with the ISO/API bearing life requirements. Results from this formula show that system life is shorter than the shortest life of the individual bearings in a system. For example, if each bearing by itself had a life of 37,500 hours, the bearing system life, when calculated by the new formula, would be only 25,000 hours.

Bearing loads, and ultimately bearing life, are functions of pump type, type of impeller (single suction vs. double suction), impeller configuration (balance holes, no balance holes), wear ring diameters, suction pressure and bearing types. Traditionally, API 610 has had various requirements that build a “safety factor” into selection of the bearings. It is also a fact that these requirements be derived from the largest set of hydraulics for each bearing housing size. This means that, for all or most other hydraulics, API 610 pump designs should automatically exceed the bearing life requirements. This positively impacts API 610 pump manufacturers who should not have to change their designs. However, in cases where the bearing system life number does not comply, discussion is needed.

Pump manufacturers have a number of tools that they can use in these more extreme applications. Changing bearing type or size, or perhaps using unbalanced construction in a pump, are some of the typical ways to increase bearing life. Applying bearings larger than those in current service may come at the price of increased bearing temperature or may even produce bearing skidding. So, in these cases, discussion of the available options is imperative. The requirement to provide bearing life figures is also an “if specified” bulleted item in API.

**Torsional analysis.** The subcommittee rewrote this section in response to persistent questions from customers regarding the types of torsional analysis and when each type is required. A flow chart was composed to provide a simple “yes/no” decision-tree method for practical guidance. The three types of analysis are:

a) Undamped natural frequency
b) Steady state damped response analysis
c) Transient torsional

**Other key changes.** Key changes to address include:

**Basic nomenclature.** Net positive suction head required (NPSHR) has been replaced with NPSH3—net positive suction head required, in meters (feet). This more accurately reflects that NPSH testing is based upon a 3% head drop. Therefore, the “R” was deleted and the “3” was added to the NPSH term (NPSH3).

‘Flammable’ and ‘hazardous’ terms. These terms have been removed from the entire API 610 11th Edition document. There has been much controversy surrounding the meaning of “flammable” and “hazardous.” National Fire Protection Association...
(NFPA) and other agencies define these terms differently, and, in reality, the purchaser decides what is flammable and/or hazardous. The purchaser can decide to use an API pump in a selected service. Thus, there is no impact if the terms are dropped from the document.

**NACE MR 103 and NACE MR 175.** Distinction has been made to help understand when each National Association of Corrosion Engineers (NACE) document applies. **NACE MR103** becomes the key document applicable to oil refineries, liquefied natural gas (LNG) plants and chemical plants. The traditionally used **NACE MR175** is now specifically noted as applying to sulfide- and chloride-stress-corrosion cracking services in oil and gas production facilities and natural-gas sweetening plants.

For years, **NACE MR175** was the only NACE document that was applied for materials subjected to stress-corrosion cracking covered in **API 610**. Upon further investigation, it was learned that **NACE MR103** was, in fact, more applicable to the majority of equipment purchased to **API 610**, so it has been added.

**Pump performance testing.** The document now emphasizes that performance testing should be conducted based upon the “uncertainty” requirements of **ISO 9906**, which primarily addresses instrumentation controls. Test tolerances have changed to +/- 3% across the board for differential heads of 0 m to > 300 m (1,000 ft), representing a slight shift in the mid- to high-head regions. There has been a tightening in the low-head range, which is now defined as 0 m to 75 m (250 ft) vs. the previous 0 to 150 m (500 ft). Defined recorded test points have also slightly changed, as reflected in Table 2.

**Non-destructive testing (or NDE).** Guidelines for NDE were first introduced in the 10th edition. The subcommittee members agreed that more definitive guidelines for pressure casings, nozzle and connection welds were needed. The 11th edition now expands NDE to include when certain types of NDE are required. New Table 14 defines “three inspection classes”—I, II and III. Class I applies when minimal visual inspection is needed—basically for all services. Class II applies to situations when the casing is >80 percent MAWP and > 200°C (392°F) and requires magnetic particle (MT) and dye penetrant (PT) inspections. Class III is for extremely hazardous services. These include services for fluids with: low specific gravity (< 0.50 sg) with temperatures to 200°C (393°F), low specific gravity (< 0.29 sg) with temperatures > 200°C (392°F) and temperatures > 260°C (500°F). All require additional radiographic (RT) and ultrasonic (UT) inspections.

**Annex N: Pump data sheets and electronic data exchange.** Data sheets have been extensively improved. Previously designed to be completed with a pencil, the new datasheets are now electronic. Rather than words and a circle appearing for each datasheet option, there is simply a blank with a “drop down” list. In addition to their ease of use on a computer, the new datasheets eliminate all the circles and choices, making them shorter.

**Pipe gussetting.** This new item details pipe gusseting (when required). With more customers seeking gusseting of pipe connections to the pump casing, the subcommittee decided to include these details in the ISO/API document. **Note:** However, that gussetting remains an “if specified” by customer item.

**Conclusion.** Even though these new changes and additions to **API 610 11th Edition (ISO 13709 2nd Edition)** are designed to increase MTBR, it is also essential to:

- Choose correctly designed pumps with the right materials for your specific applications

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**TABLE 2. Defined recorded test points, comparison of 10th and 11th editions**

<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Shutoff</td>
<td>1. Shutoff</td>
</tr>
<tr>
<td>3. Between 95% and 99% of rated flow</td>
<td>3. Midway between minimum and rated flow</td>
</tr>
<tr>
<td>4. Between rated and 105% rated flow</td>
<td>4. Rated flow</td>
</tr>
<tr>
<td>5. Approximate BEP flow (if rated is not within 5% of BEP flow)</td>
<td>5. Maximum allowable flow (120% BEP as a minimum)</td>
</tr>
<tr>
<td>6. End of “allowable” operating range</td>
<td></td>
</tr>
</tbody>
</table>

**• Follow best practices for pumping**

**• Assess complete operational systems to identify and eliminate problematic equipment.**

The result will be greater reliability and efficiency and lower operating costs. **HP**

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