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Preview of API 610 12th Edition

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Jones received his B.S. and M.S. degrees in mechanical engineering from Kansas State University and is a registered professional engineer in Texas. He represented Shell on the API Subcommittee on Mechanical Equipment and is a former chairman of the subcommittee. Jones is the task force chairman of API 610 and the International Standards Organization (ISO) convener for the 13709 Working Group. He is the previous chairman of the International Standards Coordinating Committee of the API and head of the U.S. delegation to the various ISO technical committees governing standards for refining and offshore equipment. He is a former member of the International Pump Users Symposium Advisory Committee.

This paper serves as an introduction to API 610 12th Edition, which is expected to be published sometime in 2015. It covers highlights of the proposed changes to the current ISO 13709/API 610 11th Edition and provides insights into the various topics discussed by the API 610 sub-committee.

ABSTRACT

API 610 Eleventh Edition, Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, is being updated to the Twelfth Edition. The Eleventh Edition was identical to the ISO 13709 Second Edition; however, API and ISO have decided to no longer “co-brand” standards and ISO 13709 Second Edition is not being updated.

This tutorial addresses the background process in how the document is updated and indicates the participating companies that contributed to this work. The majority of this paper is focused on addressing the “significant” changes as well as “other” changes that are of particular interest to the reader in understanding changes from the previous ISO/API editions. Included is the background reasoning behind each change. Insight into subject matter for future updates to ISO 13709/API 610 is addressed at the end. The Twelfth Edition draft has been circulating for comment since the second quarter of 2014.

One area of particular interest in the Eleventh Edition was the data sheet program which has been improved and was expected to support electronic data interchange (EDI) for engineering contractors, end users and pump manufacturers.

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Korkowski received his B.S. in industrial engineering from New Jersey Institute of Technology with post-graduate studies in engineering and business administration at Lafayette College and Fairleigh Dickinson University. He is one of the Flowserve representatives on the API 610 Subcommittee task force for the last 18 years and helped produce the API 610 12th Edition document.

EDI was expected to save significant effort in accurately specifying equipment requirements. In the Twelfth Edition, a task force subgroup is again trying to enhance its use.

INTRODUCTION

The American Petroleum Institute (API) publication “Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, 11th Edition, September 2010 is being updated to the 12th Edition. The 12th Edition is expected to be reviewed by the API Subcommittee on Mechanical Equipment (SOME) at the 2014 Fall Refining Meeting in November. The plan is for it to be balloted in the first quarter of 2015 and be published sometime in late 2015.

It is a normal, required part of the update process to compare the previous edition with the current edition of the “standard paragraphs” (i.e., an API document which applies to all API standards). This is accomplished by breaking the document into sections and then having sub-team members read two sections. No two sections are read by the same two-team members. In this edition, the sub-team was tasked with also looking for sections that were illogically organized. These would be reorganized for improved reading in the Twelfth Edition. Several sections such as bearings and bearing housings have been substantially reorganized, so they may appear unfamiliar

to users of the Eleventh Edition even though there is very little change technically.

As one can see from the chronology in Table 1, publication of a standard takes considerable time. If the 610 schedule is met, the standard should be published very close to the target five-year interval between new editions. Historically, the revision process has often been very slow. It is believed that API 610 has never been reaffirmed. (Within the API process, reaffirmation grants a two-year delay before the next publication is due.) Yet, the interval between 610 editions is still average. Table 1 provides some interesting historical data on the various API 610 Editions.

| Edition | Date | Title (*) | # Pages | Price | Price per Page | Years Between Editions |
|---------|--------|-----------------------|---------|--------|----------------|------------------------|
| 1 | 1954 | (1) | 20 | \$1.50 | \$0.08 | -- |
| 1 | Jan-55 | Sup.1 Mech. Seals | 1 | \$0.25 | \$0.25 | -- |
| 1 | Jan-55 | Sup. 2 Vertical Pumps | 3 | \$0.25 | \$0.08 | -- |
| 2 | Jan-57 | (1) | 22 | \$1.50 | \$0.07 | 2 |
| 3 | Jan-60 | (2) | 26 | \$1.50 | \$0.06 | 3 |
| 4 | Jul-65 | (2) | 26 | \$1.50 | \$0.06 | 5 |
| 5 | Mar-71 | (2) | 35 | -- | -- | 6 |
| 6 | Jan-81 | (2) | 62 | \$20 | \$0.32 | 10 |
| 7 | Feb-89 | (2) | 139 | \$75 | \$0.54 | 8 |
| 8 | Aug-95 | (3) | 193 | \$100 | \$0.52 | 6 |
| 9 | Dec-02 | (4) | 192 | \$175 | \$0.91 | 7 |
| 10 | Oct-04 | (4) | 184 | \$175 | \$0.95 | 9** |
| 11 | Sep-10 | (4) | 184 | \$249 | \$1.35 | 8 |

* Titles Legend:
 (1) = API Specifications for Pumps for General Refinery Services
 (2) = Centrifugal Pumps for General Refinery Services
 (3) = Centrifugal Pumps for Petroleum, Heavy Duty Chemical and Gas Industry Services
 (4) = Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Services
 ** based upon publication data of the 8th Edition

Table 1: API 610 documents historical data

Task Force Formation & Objectives

Numerous companies have provided experts in their fields to produce this updated document. Engineering contractors, end users and pump manufacturers alike comprised an international team to explore, discuss and debate a variety of topics. The following companies and a number of private consultants contributed to this work: Bechtel, Fluor, KBR, Shell, Petrobras, Aramco, Dow, Union, CPC, Flowserve, Floway, IIT-Goulds, Ruhrpumpen, Sulzer, Sunstrand, Weir, DuPont, Hydro, Nuovo Pignone, ABS Pumps, European Sealing and Intelliquip.

This API 610 Task force is led by the chairman, Roger Jones of Aramco Services/KBR, and secretary, Charles Heald of Flowserve.

The Update Process

API standards are on a five-year review cycle. This means that perhaps three years after a standard has been published, a task

force is reformed to review the current standard and determine:

- If the standard requires updating to conform with current technology and market practices; or
- If the standard can be reaffirmed

Presuming the decision is to revise the standard, the task force proceeds to determine how best to do the updating work, makes committee assignments, and recommends proposed changes. All changes must meet with task force approval before they are included in the first draft of the revised standard.

In the review process, the task force must consider all standard paragraphs that are pertinent to the standard and either:

1. Change the standard to agree with the standard paragraph; or
2. Modify the standard paragraph to better suit the standard being reviewed; or
3. Justify that the standard paragraph does not apply to the equipment for which the standard being reviewed applies and remove it

When the task force is satisfied that the revised standard is ready to be presented to the Subcommittee on Mechanical Equipment (SOME), the presentation is scheduled. In the presentation, all changes to the standard must be presented, explained and/or justified. The Subcommittee may request changes be made to the draft or that the task force revisit certain subjects and present them again.

When the SOME is satisfied with the revised standard, it may be submitted to API Headquarters for balloting by all voting members of the American Petroleum Institute. All negative ballots must be resolved before the revised standard can be published. This process usually takes between two and three years, depending on the magnitude of the changes.

SIGNIFICANT ADDITIONS & CHANGES

The API 610 Subcommittee started the process of reviewing about 30 items based on inputs from the SOME, industry leaders, updates from referenced specifications (such as Hydraulic Institute) and subcommittee members. Though eventually there is expected to be more than 100 changes which include minor edits, the key additional or modified items recommended for the Twelfth Edition inclusion are as follows:

1. Addition of shaft guards for all pumps
2. New Informative Annex addressing high-energy pumps
3. Material Columns reduction and improvements to material designations, including non-metallics
4. Updated Annexes for Material class selection guidelines and Material columns
5. Energy density limits for pipeline pumps
6. Performance test points modification
7. Clarification of several definitions and images
8. Re-arranging of certain sections
9. Addition of "data list"; data sheet update
10. Pressure rating for OH, BB1 and BB2 pumps

11. VFD considerations
12. Vertical pumps: TIR on vertical motor mounting flange; can requirements; dynamics
13. Update all paragraph numbers, tables

Shaft Guards

The current API 610 11th edition addresses only coupling guards. Inputs from multiple refineries indicated that safety organizations were pointing out that the area between the pump casing cover and the bearing housing has an exposed shaft area that should be covered. More specifically, this is the shaft area where the mechanical seal gland is located. Furthermore, the drive collar adjacent to the cartridge seal has set screws, which could be a concern if someone placed their hand in that area during pump operation. The sub-committee decided to mandate a shaft guard. A simple decision to make became complicated as we started to define the guard requirements. Basically, the same requirements which apply to coupling guards pertain to shaft guards, with some differences. Unlike the coupling guard, woven wire is an acceptable approach, since this guard does not have the need to be sufficiently stiff (rigid) to withstand a 200 lbf (900N) static point load. However, the shaft guard does require to be sufficiently vented to prevent accumulation of seal emissions, liquid or vapor and have an opening 0.50 inch (1.27cm) in diameter to allow for a portable VOC emission probe 0.25 inch (0.64cm) in diameter to measure emissions within 0.39 inch (1 cm) of the shaft-seal interface area. Further information was provided for pipeline pumps.



Figure 1: Unguarded shaft area vs. guarded

High-energy “Special Purpose” Pumps

In the 11th Edition, high energy was defined as pumps with heads per stage greater than 650 ft (200 m) and power per stage greater than 300 hp (225 kW). Only a stipulation for percentage of radial clearances between the diffuser vane or casing cut-water and the impeller blade in relation to their radii was addressed in the 11th Edition.

The sub-committee realized two things: first, that high energy meant different things to different people, as evidenced by customers who have already written into their specifications what they consider high energy; and second, that irrespective of “the definition” of high energy, the prescription of what exactly should be addressed for any high-energy pump was the more important issue. The decision was made to:

- Re-label these pumps as “Special Purpose”
- Add a “new” annex specifically dedicated to these pumps
- Annex to be “informative” instead of “formative”

The annex contains sections for definition; selection criteria for pressure boundary and rotor; design considerations for pressure boundary components, impellers, diffusers or volutes, shaft seals, bearings and bearing housings; materials; manufacturing; and testing guidelines.

Examples of special purpose pumps are: single-stage 5490 rpm high-speed hydrogen and oxygen F-1 turbopumps used for the Saturn V booster rocket engines; 7500 psi (500 bar) high-pressure, 6000 rpm high-speed, 1600 ft. (500 m) per stage water injection pumps; high-pressure ethylene pipeline pumps; high-pressure boiler feed water pumps; and even possibly unspared 3 to 4 MW refinery charge pumps. It is recognized that special purpose pumps constitute only about 1% of the entire pump population; however, they represent some of the greatest challenges for pump designers and thus the need for special design considerations. Figure 2 represents one approach in defining pump energy level in terms of stage pressure rise.

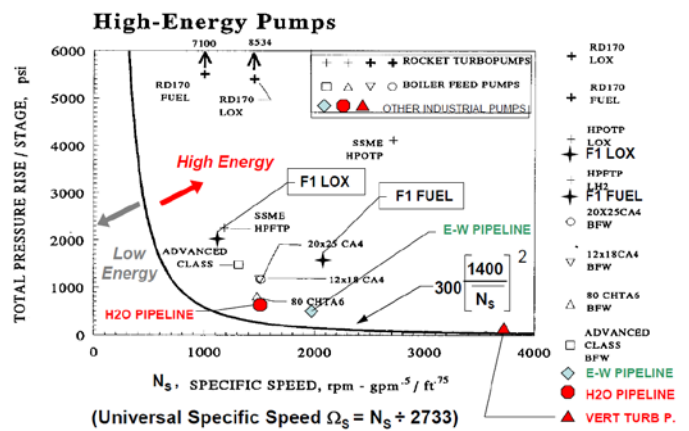


Figure 2: Example of high-energy pumps based on specific speed vs. total pressure rise per stage

For high-energy pumps, every aspect of the design requires careful review, including rotor stiffness, distribution of residual stresses in metal-to-metal sealing surfaces, determining deflection at critical fits and establishing proper running clearances. Performing structural analysis of impellers and diffusers (or volutes) is essential as is determining the proper NPSH margin based upon incipient NPSH (NPSHi), not just the generic 3% NPSH3. Especially for new designs, FEA of the bearing housing should be done to carefully determine the types of bearings to use. Lastly, the ability to easily assemble and disassemble impellers must be taken into consideration. As for manufacturing requirements, patterns and rigging should provide sound castings while non-destructive testing of highly stressed areas should be performed.

Materials

Changes to the 11th edition annexes for Materials Selection Guidance and Materials & Material Specifications for Pump Parts are proposed. The key changes are:

- Drop the cast iron material columns I-1 and I-2, since API pump manufacturers no longer pour cast iron casings; most likely ANSI pump will do

- Re-defining boiling water and process water in terms of temperature limits while replacing I-1 and I-2 with C-6 materials
- For S-6 materials, use 12% chrome shafts
- Drop columns S-1 and S-3, as there is little usage for cast iron and ni-resist internals
- Remove pressure differential per wear part for non-metallic wear parts
- Remove CA15 for impellers; use CA6NM (as was already required for pump casings in 11th edition) for improved castability, weldability, and being a tougher material more resistant to cracking

Under auxiliary connections, for C-6 materials, 316L piping and fittings are to be used up to 500°F (260°C), and Inconel 625 material for higher temperatures.

Bearing Selection Criteria

Currently in the 11th edition, hydrodynamic radial and thrust bearings are mandated when the energy density (i.e., pump rated power times the rated speed) is 5.4×10^6 hp/min (4.0×10^6 kW/min) or greater. For the 12th edition, this requirement remains for all services except for pipelines, where higher energy density levels of 14.3×10^6 hp/min (10.7×10^6 kW/min) are proposed. Justification for this higher level is basis various successful field installations and considering that pipeline services are characteristic of pumping products with lower product temperatures compared to medium to hot temperature liquids found in refinery services.

Bearing Oil & Housing Temperatures

For non-pressurized bearing systems, such as ring-oiled or splash systems, oil and housing temperature limits have been properly stated as a function of temperature rise, since ambient temperature is an essential part of the criteria.

Performance Test Points

Slight changes from the 11th Edition are proposed. Additional test points (highlighted in blue in Table 2) are now required to help better verify pump performance in the region between shut-off and minimum continuous stable flow (MCSF). The new stipulation is that no two points in the allowable operating range are to be apart by 35% in flow. This is particularly important on medium and higher energy pumps where it is recommended to obtain a vibration signature at the low flow end without damaging the pump (11th Edition currently requires taking a performance reading at shut-off; however, no vibration data is required).

| 11 th Edition | Proposed 12 th Edition |
|-------------------------------------|--------------------------------------|
| Shut-off (no vibration) | Same |
| MCSF (beginning of allowable range) | Same |
| — | Possible 35% filler point |
| — | Min. preferred operating range point |
| — | Possible 35% filler point |
| 95% to 99% of rated flow | Same |
| Rated flow to 105% rated | Same |

| | |
|--|--|
| Approx. BEP (if rated flow is not within 5% BEP) | Same |
| End of allowable operating range | End of curve (end of preferred and operating) region |

Table 2: Performance test points comparison between 11th Edition and proposed 12th Edition

Baseplates

Wording improvements to the 11th Edition are proposed to more accurately describe the baseplate types. “Drain rim and drain pan” are removed and replaced with:

- Flat deck type with a sloped gutter drain (Figure 3)
- Sloped deck plate mounted between the side rails and extending beneath the pump and driver (Figure 4)
- Sloped deck plate mounted between the side rails and extending only under the pump and coupling (Figure 5)

Also, the following new baseplate types are added:

- Open deck of the above three basic designs with no deck/top plate (Figure 6)
- Non-grouted baseplate of the above designs with attention to the pedestal supports and tying into the side rails
- Non-grouted baseplate with: a gimbal mount; three-point mount; anti-vibration mount (AVM) spring mount; or other for minimizing deflections for nozzle loads or driver torque

The current 11th Edition requires that the purchaser specify which type of baseplate is required.



Figure 3: Flat deck type with a mounted sloped gutter drain



Figure 4: Sloped deck plate between the side rails and extending beneath the pump and driver



Figure 5: Sloped deck plate mounted between the side rails and extending only under pump and coupling



Figure 6: Open deck of the three basic designs with no deck/top plate (grouted or non-grouted)

Details on jackscrew requirements were added. They shall be a minimum of M12 (1/2-13 UNC), whether removable or permanently mounted.

A new requirement for preventing blocking of the area adjacent to the pump bearing housing, mechanical seal and coupling has been included. This is particularly important for OH2 process pumps with auxiliaries for Plan 52, 53 and gas panels, along with seal flush plans with coolers ([Plan 21, 23] see Figure 7).

To facilitate this, non-standard dimensioned baseplates shall be used instead of the standard 0.5 to 12 sizes, and the auxiliaries can then be positioned on the baseplate in front of the pump suction nozzle area. This approach gives access to both sides of the pump back end with room to check the mechanical seal, bearing housing and coupling while the pump is operating, or to easily remove the back-pull-out element for servicing. A similar approach is to be taken for between bearings pumps (see Figure 8).

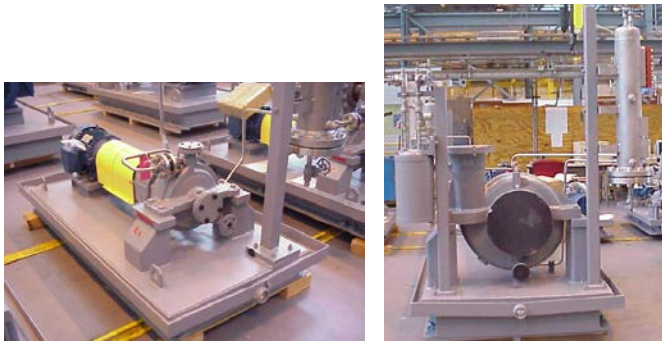


Figure 7: Small and large OH2 pumps with auxiliaries mounted in front of the suction nozzle area

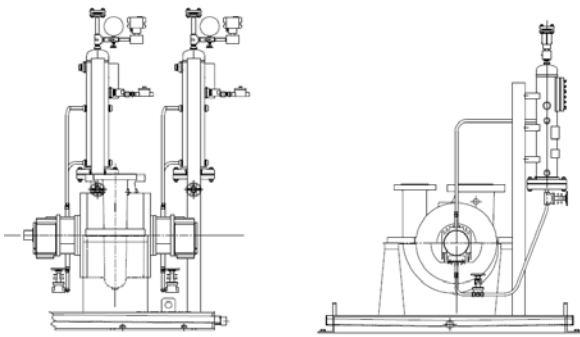


Figure 8: Between bearings pumps with auxiliaries mounted preferably on one side, for easy access

Definitions and Images

As part of the review process for producing the 12th Edition, “Standard Paragraphs” which apply to all rotating equipment were reviewed and compared to the 11th Edition to determine where possible changes in definitions would be required. The definitions needing attention were: maximum allowable working pressure (MAWP), maximum discharge pressure, and properly defining what is meant by “normal”.

Images for vertically suspended pump types VS6 and VS7 were

improved to show both flat bottom and ellipsoidal cans. An image was added for “near centerline supported” BB1 pump.

Data Sheets and “Data List”

Improvements are being made to the 11th Edition data sheets to cover all new changes within the API 610 document along with the updating of the proper paragraph numbers. In addition, there will be a “data list” which is a neutral data file that can be used to exchange conditions-of-service details for making pump selections. It is a tool that supports electronic data exchange (EDE) effectively and minimizes the possible errors in transposing numbers from contractor to pump manufacturer back to contractor and end user to complete the electronic loop.

Pressure Rating for OH, BB1, BB2 Pumps

The 11th Edition (as well as all previous API 610 revisions) required that OH, BB1 and BB2 pumps be rated for 600 psi (41 bar). The 11th Edition had a special note stating that by the time the 12th Edition is issued, OH, BB1 and BB2 pumps would be required to have a pressure rating equal to that of a 300 lb flange, which is 740 psi (51 bar) at 100°F (38°C). Further discussions revealed that the majority of pump sizes generate heads that are relatively low. This translates to the current 600 psi (41 bar) pressure requirement, to which most pump manufacturers comply. The final decision was made to revert back to the 600 psi (41 bar) rating for these pump types. It should be noted that most manufacturers do have, as an option, higher pressure pump designs, especially for high suction pressure applications which require 600 lb, 900 lb and even 1500 lb flanges and heavier wall thickness casing designs.

Bearing-housing Resonance Test

Additional clarifications are being added to advise what should be done if resonance conditions cannot be detuned. A note has been added regarding VFD applications to explain that it may not be possible to achieve all the applicable frequency separation margin requirements, in which case the purchaser and pump manufacturer may want to take additional readings. With VFDs, certain operating speed ranges can be blocked out and when operating at reduced speeds, the resonance should be lower.

Vertical Suspended Pump Requirements

Three areas have been expanded and modified. The first concerns changing the tolerance required for the driver shaft and base from 0.001 in (25µm) to 0.002 in/ft (0.17 mm/m). This is based on the logic that it is impossible to hold the same tolerance on a small motor flange as a very large motor.

Next are the casing details relative to type VS6 pumps. An explanation is given to outer barrel construction materials relative to having a pipe with weld cap design with butt welds and radiography (RT) vs. a pipe with a flat plate design with fillet welds inspected by either dye penetrant (PT), magnetic particle (MT) or ultrasonic (UT). The key with either design is for the outer barrel to meet the maximum allowable working pressure (MAWP). Suction barrels or cans can have either elliptical or flat bottom heads, again meeting the MAWP requirements and use full-penetration welds. If elliptical bottom heads are specified, they will be either be ellipsoidal or

torispherical. Longitudinal welds of seam-welded pipe for casing walls of pump heads and suction barrels are to be 100% RT inspected.

The third area for improving vertical pump requirements is the dynamic section, which remains a bulleted paragraph. Clarification was added to describe that when a dynamic analysis is required by a customer, it means the complete pump, including the belowground components and the driver structure on either its foundation or support structure. Three new notes have been added to address the extent of detail required for the models, guidelines for verticals per Hydraulic Institute, and how to handle situations when separation margins are not achieved.

Disassembly After Testing

Further explanation is added that for BB3 and BB5 pump types, it may not be possible to drain all the water after testing, and though it is important to do so, the optional approach of disassembling the pump may be invasive to a point of impacting the mechanical integrity of the pump.

Structural/Dynamic Analysis

The API 610 section on torsional analysis, along with the flow chart, is being updated to reflect minor improvements in wording. The definition for steady-state “forced” analysis (it was damped in the 11th Edition) has been more accurately rewritten. Similarly, transient torsional analysis is now defined as transient “forced response” analysis. A clarification for performing an undamped natural frequency analysis when using VFDs and ASDs was added along with a note that certain designs, especially older vintage units may produce high torsional pulsations.

Updated Paragraph Numbers and Tables

Since the 12th Edition will not be co-branded with ISO, throughout the document including all tables and charts, the order of dimensional units has changed from metric first (U.S. customary) to U.S. customary (metric). The decision was made to keep the ISO references since in many cases there are no other equivalent references.

OTHER AREAS OF DISCUSSION AND INTEREST

The following items were discussed and evaluated by the API 610 sub-committee with a decision to either not change the 11th Edition wording or to not include them at all in the 12th Edition. They are included in this paper as a means of representing information that may be beneficial to members of the oil & gas community.

Nozzle Loads

Discussions centered on whether the forces and moments shown in the nozzle load chart are still current or whether they should be changed. This was raised because more engineering contractors are requesting at least two times the API nozzle loads for the pump package (i.e., pump with baseplate). The decision was made to leave the values alone; however, we

added design options under the baseplate section for three-point mount, spring loading (referred to as AVM, anti-vibration mount), which will provide higher nozzle load capability.

NPT Gland Connection

Much discussion and investigation determined whether it was feasible to change from the current default for an NPT connection at the mechanical seal gland to a higher integrity connection joint. The sub-committee presented a number of options to the SOME and addressed the pros and cons of each. The conclusion was to keep the 11th Edition wording for the NPT connection as a default. The SOME provided feedback that this joint has not really been a problem when proper field installation practices are followed; and because the 11th Edition already contains a bulleted paragraph addressing a higher integrity joint for those customers who want it. Another viable approach and solution for those who do not want an NPT connection at the mechanical seal gland is to provide a machined flange or socket welded connection off the casing cover for the primary seal flush line. Only gland auxiliary connections for Plan 52 and 53 handling non-process liquids would be NPT.

Constant Level Oilers

The sub-committee was requested to review whether oilers used on pump bearing housings should be removed. The various pros and cons were discussed. Some argued that operators may overfill the bearing housings because when they do not see oil in the oiler, they just pour more oil into the oilers, which leads to overheating the oil and leakage out of the bearing housing end covers. Use of bull’s-eyes seemed to be a solution; however, these small oil indicators do coke up and it is hard to see them from a distance. The conclusion was reached to continue to require constant level oilers on the bearing housings because they serve as a good indicator for operators to quickly see whether oil is needed from a distance.

Incorporating API 685 Sealless Pumps Into API 610

Currently there are several paragraphs in API 685 that read almost exactly like API 610. However, because there are so many unique design elements characteristic of sealless pumps, it is recommended to keep these two documents separate.

Wear Ring Running Clearances

The question was posed as to whether there was a need to change the 11th Edition wear ring clearances, i.e., increase them or possibly decrease them. Note that the 11th Edition clearances are exactly the same as those from API 610 5th Edition. Also, opening these API clearances by 0.005 inch (125µm) applies to all services with liquid temperature above 500°F (260°C). However, considering today’s technology for improving wear surfaces and the utilization of non-metallic materials, it was a good discussion to have. These improvements were promoted on the basis of improving product reliability and mean time between repair (MTBR), and not necessarily to increase efficiency. On the basis that technically there is not enough field data to verify the impact of closing up metal wear ring clearances, the decision was not to change them for now. Regarding non-metallic rings, closing clearances is possible, especially when efficiency is extremely important on a given

service. However, the parameters of liquid temperature and cleanliness of service should be considered along with the consideration that the clearances will open over time.

CONCLUSIONS

This paper has highlighted most of the upcoming changes that are expected to be approved for the final publishing of the 12th Edition of API 610. It furthermore has provided insights into the various other points of discussion that the API 610 sub-committee addressed and the reasons behind whether changes were actually necessary.

We welcome all comments and suggestions for topics both within and beyond what has been addressed in this paper for additional consideration.

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