

ISO 13709 2nd Edition/ API 610 Eleventh Edition Highlights

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This paper varies in style from other published papers due to the need to incorporate actual or slightly edited wording from the ISO/API standard document. It is composed on the basis that the reader has purchased a copy of the ISO/ API standard and the supporting data sheet program and may use this paper for supplemental information.

ABSTRACT

International standard ISO 13709:2009 (Identical), ANSI/API Standard 610: September 2010 “Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries” also referred to as ISO 13709 Second Edition / API Eleventh Edition has been updated from its previous ISO 13709 1st Edition/ API 610 10th Edition.

This tutorial addresses the background process in how the document was updated along with indicating the participating companies who 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 will be addressed at the end.

One area of particular interest is the data sheet program which has been improved and significantly enhanced to become a guide for engineering contractors, end users and pump manufacturers alike for accurately specifying equipment requirements. A special thank you is extended to the late Anthony (Tony) Semple of Bechtel for his hours of dedication to produce this data sheet program.

INTRODUCTION

The American Petroleum Institute (API) publication “Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, 10th Edition, October 2004” has been updated to the 11th Edition. The 11th Edition is identical to ISO 13709, 2009 (Identical) Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, Second Edition”. The process of publishing identical API and ISO standards is called “co-branding”.

The 9th Edition of 610 and the first Edition of ISO 13709 were also identical. The two documents were submitted separately for ISO Final Draft International Specification and API ballots.

After successfully passing both ballots ISO edited 13709. Even though ISO editors had previously looked at the document, they identified a large number of editorial changes. This resulted in the need to add a lengthy errata to 610 9th Edition. This would have made 610 very inconvenient to use. One would find a requirement in the text and then go to the errata to check for a change. Since the number of editorial changes was so large it was decided to incorporate the changes into the text and re-issue it as the 10th Edition in October of 2004. This is why the 10th Edition came out so soon after the 9th. Technically the 9th and 10th Editions are identical just with only minor word differences. If a pump is purchased to either the 9th or 10th it should be the same pump. The 10th Edition is identical to ISO 13709 1st.

The 9th Edition of 610 and 1st Edition of 13709 were complete technically in 2002. It took 2 years from the time that 610 9th and 13709 1st were technically complete until they were published. By that time the taskforce/working group had reformed and begun work on the 11th Edition of 610/2nd Edition of 13709. This was in mid 2004. Note: API calls groups that work on standards “taskforces” whereas ISO calls them “working groups”. For the rest of this paper, they will be referred to as WG.

One of the difficulties in co-branding standards is that they must go through two parallel voting processes. In API, comments are allowed on all ballots. In ISO, comments are allowed on Committee Draft and Draft International Specification (DIS) Ballots. Technical comments are not allowed on Final Draft International Specification (FDIS) ballots. If documents are allowed to be balloted separately, the WG must resolve comments at least twice as many times. In preparing the 11th Edition of 610, all ballots were simultaneous and it was only necessary to resolve comments once after the DIS. The DIS went out for ballot in November of 2005. On 10 April 2006 the ballot closed and the WG received sixty pages of comments. These were duly resolved and presented to the API Subcommittee on Mechanical Equipment in San Francisco in October of 2006. In light of the large number of comments it

was decided to resubmit the document for a second DIS. The second DIS ballot closed in Jun of 2008. Comments were resolved and ISO 13709 Second Edition FDIS was submitted for ballot in June of 2008. There were no negative ballots and only a few technical comments.

The ballots for the 11th Edition of 610 and the 2nd Edition of 13709 were simultaneous. When API members voted affirmative on 610, they were simultaneously voting to adopt 13709 back as API 610. This allowed both documents to be identical. When a document is balloted as an FDIS, technical comments are not allowed. However editorial comments are allowed. When the WG received the FDIS, a sub-team was selected to proof read the entire document. Members of this team proofread two sections of the document and each of these sections was proofread by two different people. A relatively small number of editorial errors were found. We then proceeded to argue with the ISO editor for over a year until we finally gave up, and the 2nd Edition of 13709 was published on December 15, 2009. Within API we decided to correct the remaining editorial errors and the 11th Edition of 610 was finally published in September of 2010.

As one can see from this chronology, publication of a joint standard may take considerable time. It would be easy to think that the co-branding process slows down the process. It does, yet historically the 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 about average. Table 1 gives some historical data on 610.

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

TASKFORCE FORMATION & OBJECTIVES

Numerous companies have provided experts in their field 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 which contributed personnel to this work are:

Bechtel, Fluor, KBR, Shell, Valero, BP, Marathon Petroleum, Chevron, ConocoPhillips, Petrobras, Aramco, Total, Repsol, Dow, Engridge, Clydeunion, CPC, Ebara, Flowserve, Floway, ITT-Goulds, Ruhrpumpen, Sulzer, Sunstrand, Weir, Samsung, Blackmer, Lawrence, Nikkiso, Nuovo Pignone, MHI, SBEC, European Sealing, Centrifugaltech, ABS pumps, Rexnord, Intelliquip, MHI.

This ISO 13709/ API 610 Working Group (WGTF) was lead by the Chairman Roger Jones of Shell (at the time), 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 this 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

There are about a couple hundred changes overall that were made to the Tenth Edition. It is hard to draw the line to categorize significant as opposed to “others”. However, primarily due to their “newness” to the API 610 document or to the extent of changes made, they are considered significant and will be discussed in greater detail:

1. Addition of Annex K Section K.1 Shaft Stiffness Guidelines for Overhung pumps, which applies to OH2 horizontal centerline mounted overhung pumps and their counterpart OH3 vertical inline pumps
2. Addition of Annex K Section K.2 Bearing “system” life considerations for pump types OH2, OH3, BB1, BB2, and BB3 along with new paragraph •6.10.1.6
3. Introduction of Three Inspection Classes for Casings, Table 14
4. Significant expansion of Torsional Analysis Section 6.9.2
5. Performance testing tolerances and Table 16
6. NACE MR 103 and NACE MR 175
7. Data Sheets, Annex N

Annex K in Tenth Edition

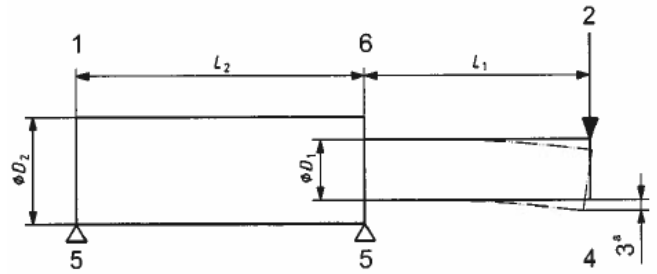
This was formerly “Seal Chamber runout illustrations” which contained Figure K.1 chamber concentricity and Figures K.2 seal face run-out. As Eleventh Edition, they are shown as Figures 27 and 28, respectively, and have been relocated to a more appropriate location in the document text to accompany the seal chamber dimensions. Nothing has changed in the illustrations, however; for Figure 28, the location for face runout measurement has been labeled.

In Eleventh Edition, the New Annex K (informative) now addresses “Shaft Stiffness and Bearing System Life.”

Annex K Section K.1 Shaft Stiffness Guidelines for Overhung pump types OH2 and OH3

This completely new section K.1 addresses a standardized method for calculating the pump shaft flexibility index (SFI) for an overhung pump. Within the standard document text is paragraph •9.1.1.3 which is a bulleted paragraph indicating that “if specified” the shaft flexibility index shall be calculated by the vendor in accordance with this sub-clause and stated on the data sheet.

Shaft flexibility index (SFI) or I_{SF} is commonly referred to L^3/D^4 or often verbalized as “L cubed over D to the fourth”. Figure K.1 shows a simple overhung rotor composition to use for calculating SFI where L_1 is the overhang (centerline of impeller to line bearing) and L_2 is the bearing span.



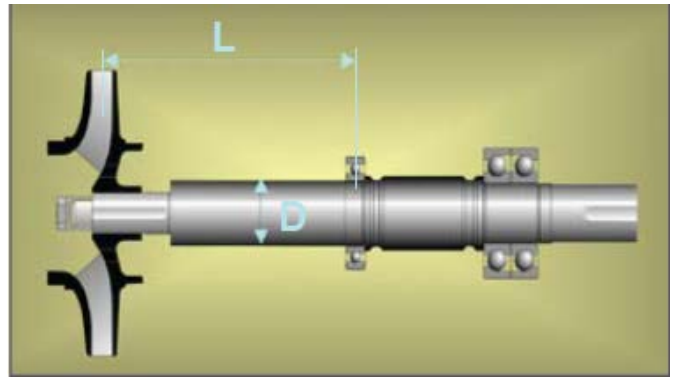
Key

- | | |
|-------------------------------|---------------------|
| 1 simplified shaft | 4 impeller end |
| 2 radial load on the impeller | 5 support (bearing) |
| 3 deflection | 6 overhang pump |

* The deflection is proportional to the load, $\left(\frac{L_1^3}{D_1^4} + \frac{L_1^2 L_2}{D_2^4} \right)$.

Figure K.1 — Simple overhung rotor

Notice that there are two areas of the shaft expressed in the SFI formula $I_{SF} = L_1^3 / D_1^4 + L_1 L_2^3 / D_2^4$. The second portion of this term constitutes only about 20% of the actual total value of the SFI value, so the shortened expression $I_{SF} = L_1^3 / D_1^4$ is generally used.



This relationship of shaft overhang to shaft diameter at the seal sleeve area was used extensively by refiners in the 1970s and 1980s as a means of relating SFI to pump mean time between repair (MTBR). Generally, the “lower” the SFI number for an overhung process pump, the better is the shaft stiffness, thus realizing higher MTBR.

The following two Figures K.2 (SI units) and K.3 (USC units) show the relationship of pump “sizing” factor K_t to SFI. K_t is derived from the formula $K_t = (Q \times H) / N$ where Q is the flow at best efficiency point (BEP) of the maximum impeller diameter; H is the corresponding total head at the BEP; and N is the rotating speed. The legend A through G represents different vintages of overhung pumps manufactured over the years.

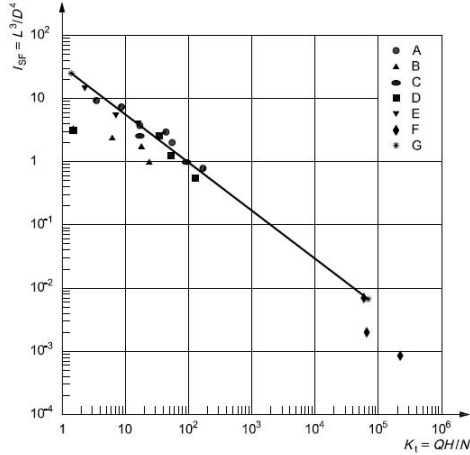


Figure K.2 — Overhung pump shaft flexibility index versus size factor (SI units)

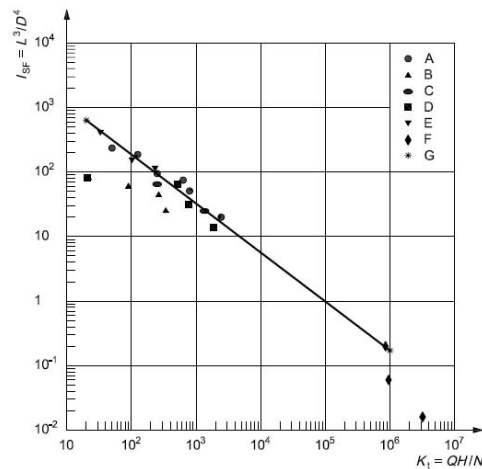


Figure K.3 — Overhung pump shaft flexibility index versus size factor (USC units)

For clarity, units of measure are:

Parameter	Units of Measure	
	SI	USC
D - diameter	mm	in
L - length	mm	in
Q - flow rate	m ³ /h	gpm
H - total head	m	ft
N - rotative speed	r/min	rpm

Basically, small pump sizes with typically single volute casings are represented in the left area of the chart, while medium size dual volute pumps are towards the middle and large pump sizes (large pump-turbines) up to 350000kW (500000 hp) are at the

right end of the chart. This high-side area was derived by equations:

$$(K.4) \quad I_{SF, SI} = 32 \times K_t^{-0.76}$$

$$(K.5) \quad I_{SF, USC} = 6200 \times K_t^{-0.76}$$

The criteria stated in Eleventh Edition are: “ Figures K.2 and K.3 or Equations (K.4) and (K.5) can be used to make a first assessment of the rotor stiffness of a given overhung pump design or a number of similar designs for a given application. An overhung pump whose I_{SF} is more than 1.2 times the chart or equation value is cause to seek justification of the design from the pump vendor”. The WGTf did evaluate a more complicated approach which included the diameter of the shaft between the bearings, but in the end it was decided to go with the SFI method. This was a more scientific way of approaching the subject than the one originally proposed using 100 (USC units) across the board for all pump sizes.

It should be noted that ISO 13709/ API 610 contains various requirements that affect pump shaft size and stiffness. These requirements include the following: a) shaft deflection at the seal faces is limited to 0.05mm (0.002 in) under the worst conditions, b) one and two stage pumps are required to have a dry bending first critical above 120% of maximum continuous speed, c) seal chambers are standardized, d) bearings must be designed for a minimum life of 25,000 hrs and e) comply with the ISO 13709/ API 610 vibration criteria.

These requirements except for seal chamber dimensions collectively drive down the SFI values to a stiffer shaft design while the seal chamber dimensions set the minimum overhang the pump can have and limit how low to push the SFI value.

The following data from a pump company exemplifies the improvement in MTBR after old original pump designs with skinny shafts and high SFI values were upgraded to today’s more robust overhung rotor designs with low SFI values.

Refinery	QTY Conversions	QTY Repairs Before Upgrade	Run Time (months) Before Upgrade	QTY Repairs After Upgrade	Run time (months) After Upgrade	MTBR Before Upgrade	MTBR After Upgrade
A	16	129	2114	11	511	16	46
B	9	73	1458	3	265	20	88
C	19	149	3103	10	471	21	47
Total	44	351	6674	24	1248	19	52

The improvement in MTBR is the bottom line result when providing pump designs with low SFI values. This whole subject of SFI is intended to bring attention to its importance and to serve as a differentiator in evaluating pumps that are not in line with the most current designs. Some of the very old overhung rotor designs had SFI of 300 or even over 1000, which represent very skinny shaft designs. A pump having a SFI of 100 vs. another having SFI of 50 is really not the issue. It is when there is a quantum difference in SFI values that raises the concern and the need to have a discussion with a

pump manufacturer.

Annex K Section K.2 Bearing system life considerations for OH2, OH3, BB1, BB2 and BB3 pumps, and new paragraph • 6.10.1.6

Similar to Annex K, Section K.1, Section K.2 is tied to the new bulleted paragraph • 6.10.1.6 stating that “if specified” the bearing system life calculations shall be furnished. See Annex K, Section K.2 for a discussion on bearing life system.

The API bearing life requirement for at least 25000 hrs with continuous operation at rated conditions and at least 16000 hrs at maximum radial and axial loads and rated speed was stated in Tenth Edition, Table 9, and has not changed. What has been clarified and required is that the criteria must apply to the bearing “system” and not to the individual bearings alone, which was a general practice in the industry. 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.

In both paragraph. • 6.10.1.6 and Annex K, Section K.2 formulas are given to calculate the bearing system life:

$$L_{10h,system} = \left[\left(\frac{1}{L_{10hA}} \right)^{3/2} + \left(\frac{1}{L_{10hB}} \right)^{3/2} + \dots + \left(\frac{1}{L_{10hN}} \right)^{3/2} \right]^{-2/3}$$

where

L_{10hA} is the basic rating life, L_{10h} , per ISO 281 for bearing A;
 L_{10hB} is the basic rating life, L_{10h} , per ISO 281 for bearing B;
 L_{10hN} is the basic rating life, L_{10h} , per ISO 281 for bearing N;
 N is the number of bearings

This produces a “system” life which 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 would calculate to only 25,000 hours. Similarly, if one bearing had a life of 10000 hours, the second bearing would need to be capable of 25,700 hours for a resultant 25,000 hr system life.

Generally, pump manufacturers have several applicable methods to increase the bearing system life. While one approach could be to increase the bearing size, while the counter effects of increased bearing temperature and possible ball skidding must be evaluated. Other modifications such as modified impeller construction perhaps with or without balance holes and eliminating back wear rings are possible solutions depending on the actual conditions of service. Also, it is important to note that proper bearing lubrication by far is more of a concern than achieving the bearing life number.

Most pump manufacturers who have improved their designs over the years will either comply or be close to complying to the 25000 hour / 16000 hours API/ ISO criteria. If a manufacturer does not comply, discussion is needed to evaluate the true advantage of changing the standard pump design to simply achieve higher bearing life.

Inspection 8.2 (7.2)

This section now includes new paragraphs and Table 14 which introduces three additional “material inspection classes” for pressure casings and nozzle welds (where applicable), auxiliary casing connection welds and auxiliary process piping welding. Class I is the basic minimal level requiring visual inspection; Class II is a higher level with MT or PT inspection based upon maximum allowable working pressure (MAWP) and temperature; and Class III is the highest level with MT, PT, UT and/ or RT, based upon specific gravity (primarily lower s.g.) and higher temperature services or the severity of a service (such as HF acid).

8.2.2.1 new – Unless otherwise specified, pressure-casing materials shall be inspected in accordance with the requirements of Table 14.

Table 14 — Pressure casing material inspection requirements

Type of component	Requirements by inspection class ^a		
	I	II	III
—	Minimum	> 80 % MAWP and > 200 °C (392 °F)	< 0.5 SG or > 200 °C (392 °F) and < 0.7 SG, or > 260 °C (500 °F) Extremely hazardous services ^e
Casing ^b : cast	VI	VI, plus MT or PT of critical areas	VI, plus MT or PT of critical areas, plus RT or UT of critical areas
Casing ^b : wrought ^c	VI	VI, plus MT or PT of critical areas	VI, plus MT or PT (critical areas), plus UT (critical areas)
Nozzle weld: casing	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT plus RT (100 %)
Auxiliary connection welds ^d	VI	VI, plus MT or PT	VI, plus MT or PT (100 %)
Internals	VI	VI	VI
Auxiliary process piping: socket-welded	VI	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT
Auxiliary process piping: butt-welded	VI, plus 5 % RT	VI, plus 100 % MT or PT and 5 % RT	VI, plus 100 % MT or PT and 10 % RT

^a Definition of abbreviations:
 VI: Visual inspection
 MT: Magnetic particle inspection
 PT: Liquid penetrant inspection
 RT: Radiographic inspection
 UT: Ultrasonic examination

^b “Casing” includes all items of the pressure boundary of the finished pump casing (e.g. the casing itself and other parts, such as nozzles, flanges, etc. attached to the casing). “Critical areas” are inlet nozzle locations, outlet nozzle locations and casing wall thickness changes. The manufacturer shall submit details of the critical areas proposed to receive MT/PT/RT/UT inspection for purchaser’s approval.

^c “Wrought” materials include forgings, plate and tubular products.

^d Due to complex geometry and thickness variations, it is not practical to RT butt-welded auxiliary casing connections.

^e Extremely hazardous services, as specified by the purchaser.

NOTE: Although the pump is designed to meet specific pressure and temperature requirements and the casing is hydrostatically tested in accordance with the requirements of this International Standard, this alone does not guarantee that the material is of a suitable quality for the service. Casting quality can be affected by considerable variations in material processing. Material standards, such as ASTM, provide minimum requirements for the material itself, but castings can be subject to areas of shrinkage, gas porosity, hot tears, sand inclusions, improper weld repairs, etc. In addition, some materials are prone to grain boundary tears or cracks that can propagate under in-service stresses caused by temperature, pressure, vibration and pipe strain.
Note: These three basic inspection classes represent what many in the industry have written in their specific company specifications over the years. Definition for critical areas is included. These requirements are mandatory “unless otherwise specified”.

8.2.2.2 new – For double-casing pumps, the outer casing pressure/temperature should be used to determine the inspection class of the outer casting (see 8.2.2.1). The inner casing should be inspected to Class I (Table 14).

8.2.2.3 new – The timing of the inspections required by Table 14 shall be as follows:

a) VI/MT/PT shall be performed after final heat treatment in the proof (rough) machined condition. In the proof (rough) machined condition, an additional amount of material remains on areas where machining to critical dimensions and tolerances is required. The additional amount of material removed shall not exceed 1 mm (0,040 in) material stock or 5% of minimum allowable wall thickness, whichever is less.

b) RT/UT of castings shall be performed after final heat treatment.

c) RT of welds and UT of wrought material and welds shall be performed after final heat treatment. UT of wrought material shall be performed prior to any machining operations that can interfere with the UT examination.

8.2.2.4 new – Where the configuration of a casting makes radiography impossible, radiographic examination may be replaced by ultrasonic testing.

Torsional Analysis, Section 6.9.2 (Tenth Edition 5.9.2)

This section basically has been completely re-written and greatly enhanced by incorporating a flow chart which provides a decision tree process in directing the user to determine which of the three types of torsional analyses is required. New paragraph 6.9.2.1 defines three types of torsional analyses:

- undamped natural frequency analysis
- steady-state damped response analysis
- transient torsional analysis

6.9.2.1 new – There are three general types of torsional analyses that are normally performed on pumps:

a) undamped natural frequency analysis: determination of the unit's torsional natural frequencies and associated mode shapes and generation of a Campbell diagram to determine potential resonance points;

b) steady-state damped response analysis: evaluation of the resonance points uncovered in the undamped analysis via a forced response analysis that utilizes representative values for excitation magnitudes and damping; the results are cyclic torques and stresses in all shaft elements in the model, which can then be used to evaluate the structural adequacy of the machine;

c) transient torsional analysis: similar to a steady-state damped response analysis, except that it is done for the transient condition and its results are cyclic torques and stresses as functions of time; by far, the most common application for this analysis type is the start-up of a synchronous motor.

d) A flow chart for the torsional analysis is shown in Figure 29.

6.9.2.2 (tenth ed. 5.9.2.1) changed wording from generic torsional to an undamped natural frequency analysis.

New paragraph 6.9.2.2 f) added – the requirement for undamped natural frequency analysis is needed for vertical pumps with a driver rated 750 kW (1000hp) or higher. New addendum (*in italic*) was added: “The analysis shall be for the train as a whole unless the train includes a device that has weak dynamic coupling, for example, a hydraulic coupling or torque converter. *In all cases, the vendor having unit responsibility shall be responsible for directing any modifications necessary to meet the requirements of 6.9.2.3 through 6.9.2.9*”

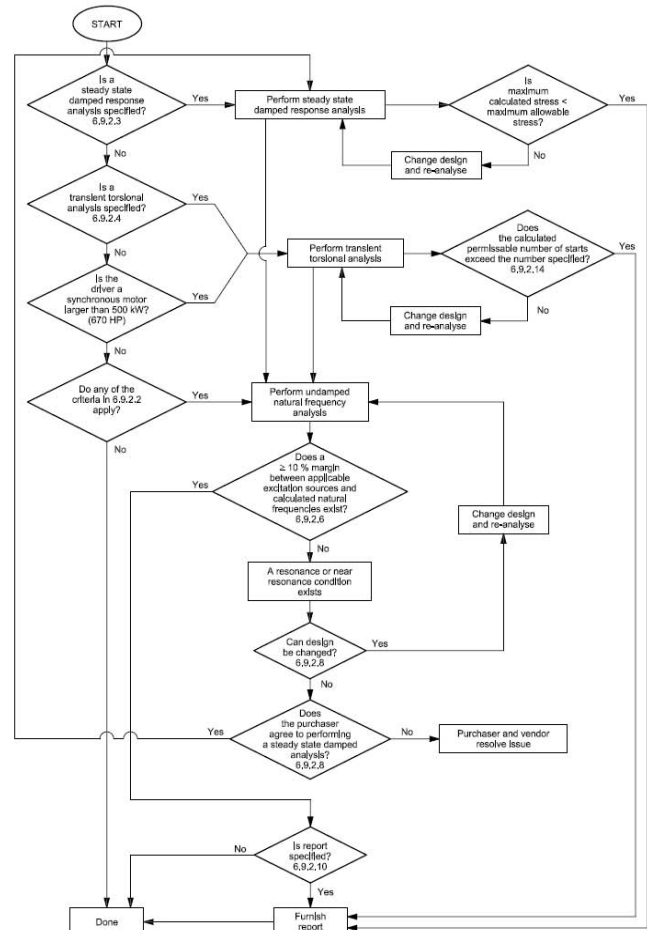


Figure 29 — Torsional analysis flow chart

New paragraph • 6.9.2.3 states:

If specified, for variable-frequency ASDs, a steady-state, damped-response analysis shall be performed. The analysis shall consider all resonant frequencies through 12 times line frequency. Most modern variable-frequency ASDs, when performing properly, produce insignificant torsional vibration and shaft stress. Variable-frequency ASD malfunctions can produce significant excitation. Certain designs still exist that produce significant torsional pulsations.

• 6.9.2.4 (Tenth ed. 5.9.2.2) has been completely re-written and is now bulleted – If specified, or if the driver is a synchronous motor rated 500 kW (670 hp) or higher, a transient torsional analysis shall be performed. If performed, the time-transient analysis shall meet the requirements of 6.9.2.11 through 6.9.2.14.

New paragraph 6.9.2.7 states:

Torsional natural frequencies at two or more times running speeds shall preferably be avoided in systems in which corresponding excitation frequencies occur. If the natural frequency cannot be moved, it shall be shown to have no adverse effect.

Paragraph 6.9.2.8 (Tenth ed. 5.9.2.4) has the following added:

NOTE: Typically, steady-state, damped torsional analyses of pumps driven by pulse-width-modulated variable-frequency drives have shown acceptably low stresses at the resonant conditions; these have no adverse effects on the machinery train.

New paragraph 6.9.2.9 states:

Unless otherwise specified, if only a steady-state, undamped torsional analysis is performed, a Campbell diagram with a tabulation of the mass elastic data and brief explanation of the calculation method may be furnished to the purchaser in lieu of a report.

New paragraph 6.9.2.11 states:

In addition to the parameters used to perform the steady-state, undamped torsional analysis specified in 6.9.2.2, the following shall be included in the transient torsional analysis:

- a) motor average torque, as well as pulsating torque (direct and quadrature axis) vs. speed characteristics;
- b) load torque vs. speed characteristics
- c) electrical-system characteristics affecting the motor terminal voltage or the assumptions made concerning the terminal voltage, including the method of starting, such as across the line, or some method of reduced voltage starting.

New paragraph 6.9.2.12 states:

The analysis shall generate the maximum torque as well as a torque vs. time history for each of the shafts in the train. The maximum torques shall be used to evaluate the peak torque capability of coupling components, gearing and interference fits of components, such as coupling hubs. The torque vs. time history shall be used to develop a cumulative damage-fatigue analysis of shafting, keys and coupling components.

New paragraph 6.9.2.13 states:

Appropriate fatigue properties and stress concentrations shall be used.

New paragraph 6.9.2.14 states:

An appropriate cumulative fatigue algorithm shall be used to develop a value for the safe number of starts. The purchaser and vendor shall mutually agree as to the safe number of starts. NOTE: Values used depend on the analytical model used and the vendor's experience. Values of 1 000 to 1 500 starts are common. ANSI/API Std 541 requires 5 000 starts. This is a reasonable assumption for a motor since it does not add significant cost to the design. The driven equipment, however, would be over-designed to meet this requirement. EXAMPLE: a 20-year life with 1 start per week equals 1 040 starts. Equipment of this type normally starts once every few

years rather than once per week. It is necessary, therefore, to specify a reasonable number of starts.

Performance Testing 8.3.3 (Tenth Edition 7.3.3)

•8.3.3.2 (Tenth ed. 7.3.3.1) though new requirement i) has been added to the list, this requirement was shown in tenth edition 7.3.3.1 with 65° C (150° F). Eleventh edition now states: unless otherwise agreed, performance tests shall be performed using water at a temperature not exceeding 55° C (130° F)

8.3.3.2 c) (Tenth ed. 7.3.3.2 c)) This paragraph describes criteria for testing liquid seals on water. A note has been added stating: Notably, pressurized dual seals with high barrier-fluid pressures [greater than 4000 kPa (40 bar; 600 psi)] should be reviewed.

• 8.3.3.3 (Tenth ed. • 7.3.3.3)

There are now six test points to be recorded instead of five as previously required and the actual location of three test points have slightly changed (shown in *italic*):

- 1) shutoff (no vibration data required)
- 2) minimum continuous stable flow (*beginning of allowable operating region*)
- 3) *between 95% and 99% rated flow*
- 4) *between rated flow and 105% rated flow*
- 5) *approximately the best efficiency flow (if rated flow is not within 5% of best efficiency flow rate)*
- 6) end of allowable operating region

The following chart shows the comparison between Eleventh and Tenth edition testing point requirements:

Eleventh Edition	Tenth Edition
1. Shutoff	1. Shutoff
2. Minimum continuous stable flow	2. Minimum continuous stable flow
3. Between 95% and 99% of rated flow	3. Midway between minimum and rated flow
4. Between rated and 105% rated flow	4. Rated flow
5. Approximate BEP flow (if rated is not within 5% of BEP flow)	
6. End of "allowable" operating region	5. Maximum allowable flow (120% BEP as a minimum)

Note: "allowable" operating range is defined as the point not exceeding the vibration levels of API 610 or thermal rise or other limitation specified by the manufacturer.

• 8.3.3.3b (Tenth ed. • 7.3.3.3b)

Previously in tenth edition, this sub-paragraph b) simply stated: The test point for rated flow shall be within a tolerance band of +/- 5% of rated flow. The eleventh edition now states: The test data shall be fit to a spline or appropriate polynomial (typically third or fourth order) for head and for power using a least squares method.

The rated/guaranteed flow shall be inserted into the resulting equation and a value for head and power calculated. These

values shall be corrected for speed, viscosity and density (specific gravity). The corrected values of head and power shall be within the tolerance bands allowed in Table 16.

Table 16 (Tenth ed. Table 14)

Condition	Rated point %	Shutoff %
Rated differential head:		
0 m to 75 m (0 ft to 250 ft)	± 3	± 10 ^a
> 75 m to 300 m (> 250 ft to 1 000 ft)	± 3	± 8 ^a
> 300 m (1 000 ft)	± 3	± 5 ^a
Rated power	4 ^b	—
Efficiency	c	
Rated NPSH	0	—

^a If a rising head flow curve is specified (see 6.1.11), the negative tolerance specified here shall be allowed only if the test curve still shows a rising characteristic.

^b With test results corrected to rated conditions [see 8.3.3.3 b)] for flow, speed, density (specific gravity) and viscosity, it is necessary that the power not exceed 104 % of the rated value, from all causes (cumulative tolerances are not acceptable).

^c The uncertainty of test efficiency by the test code specified is ± 2.5 %; therefore, efficiency is not included in the pump's rated performance. In those applications where efficiency is of prime importance to the purchaser, a specific value and related tolerance should be negotiated at the time of the order (see 8.3.3.4).

The following changes have occurred:
 Test tolerances have changed to + / - 3% across the board for differential heads of 0m to > 300m (1000 ft) which represents a slight shift in the mid- to high-head regions. There has been a tightening in the low-head region which is now defined as 0 m to 75m (250 ft) vs. the tenth edition 0 to 150m (500 ft).

Beginning of footnote b) is new and states: With test results corrected to rated conditions [see 8.3.3.3b)] for flow, speed, density (specific gravity) and viscosity, it is necessary that the power not exceed 104% of the rated value, from all causes. The ending: (cumulative tolerances are not acceptable) is the same from Tenth edition.

Footnote c) is new and states: The uncertainty of test efficiency by the test code specified is +/- 2.5%; therefore, efficiency is not included in the pump's rated performance. In those applications where efficiency is of prime importance to the purchaser, a specific value and related tolerance should be negotiated at the time of the order (see 8.3.3.4). For ease of comparison, Table 10 from Tenth Edition is included.

Table 14 from Tenth Edition has been included only to show the comparison to Table 16 Eleventh Edition.

Condition	Rated point %	Shutoff %
Rated differential head:		
— 0 m to 150 m (0 ft to 500 ft)	-2 +5	+ 10 - 10 ^a
— 151 m to 300 m (501 ft to 1 000 ft)	-2 +3	+ 8 - 8 ^a
— > 300 m (1 000 ft)	-2 +2	+ 5 - 5 ^a
Rated power	+ 4 ^b	—
Rated NPSH	0	—

NOTE: Efficiency is not a rating value.

^a If a rising head flow curve is specified (see 5.1.13), the negative tolerance specified here shall be allowed only if the test curve still shows a rising characteristic.

^b Under any combination of the above (cumulative tolerances are not acceptable)

New paragraph • 8.3.3.4 states:
 For higher-power pumps (> 1 MW), performance tolerances other than those in Table 16 can be appropriate. If specified,

pump efficiency at rated flow shall be quoted to the tolerance given by the purchaser and shall be included in the pump's rated performance. If a tolerance is specified for rated efficiency, an additional test point as close to rated flow as practical shall be taken. The rated efficiency and tolerance shall be consistent with the test code being used, with particular attention to the uncertainty of efficiency determined by test to that code. The purchaser should expect that adding efficiency with a specific tolerance to the pump's rated performance usually affects pump cost and delivery.

• 8.3.3.5 (Tenth ed. • 7.3.3.4 sub-paragraph b) is new:
 For ring and splash-oil systems, oil temperatures shall be recorded at the beginning and the end of the test. For pressurized systems, bearing metal temperatures shall be recorded at the beginning and the end of the test. The duration of the test shall be indicated on the test report.

New paragraph • 8.3.3.6 states:
 If specified, the performance test shall be conducted with test stand NPSHA controlled to no more than 110 % of the NPSHA specified on the data sheet. NOTE: It is the purpose of this test to evaluate pump performance with the specified NPSHA at pump suction.

NACE MR 103 and NACE MR 175

Distinction has been made to help understand when each National Association of Corrosion Engineers (NACE) document applies. Basically, *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. As such it has become part of the ISO 13709/ API 610 specification.

Note: The entire section of • 6.12.1.12 (Tenth ed. • 5.12.1.12) has been re-written to reflect the distinctions.

• 6.12.1.12.3 (Tenth ed. • 5.12.1.12) new – If specified, reduced-hardness materials shall be supplied in accordance with ISO 15156-1.

NOTE 1: For the purposes of this provision, ANSI/NACE MR0175 is equivalent to ISO 15156-1.

NOTE 2: ISO 15156 (all parts) which is the equivalent of ANSI/NACE MR0175, applies to material potentially subject to sulfide and chloride stress-corrosion cracking in oil and gas production facilities and natural gas sweetening plants.

6.12.1.12.4 (Tenth ed. 5.12.1.12) If reduced-hardness materials are specified, ferrous material not covered by **NACE MR0103 or ISO 15156-1 (ANSI/NACE MR0175)** shall have

a yield strength not exceeding 620 N/mm² (90 000 psi) and a hardness not exceeding HRC 22. Components that are fabricated by welding shall be postweld heat-treated, if required, so that both the welds and heat-affected zones meet the yield strength and hardness requirements.

Note: Now applies to both NACE documents.

- 6.12.1.12.6 (Tenth ed. • 5.12.1.12b) Renewable impeller wear rings that are necessary to through-harden above HRC 22 for proper pump operation shall not be used if reduced-hardness materials are specified. **Impellers may be provided with either hard-coated or surface-hardened integral wear surfaces or renewable wear rings.** If approved by the purchaser, in lieu of furnishing renewable wear rings, wear surfaces may be surface-hardened or hardened by the application of a suitable coating.

Note: Added section in bold.

Pump Data sheets, Annex N

Annex N (informative) Pump datasheets and electronic data exchange has a new fresh look and is a quantum improvement. It now has: a) drop-down options, containing automatic calculated values in certain cells based on input data, b) pop-ups which identify cross referenced paragraphs along with some text, and c) drop-down lists for uniform selection. Figure N.1 data sheets apply to SI units, Figure N.2 apply to US Customary units. Figure N.3 is a document which shows all the drop-down choices behind specific cells. New fields have also been added to address additional scope items and technical information. All of this information is contained in an excel based program that is an optional purchase item from API.

The screenshot shows a complex data entry form for a centrifugal pump. Key sections include:

- CONSTRUCTION:** API PUMP TYPE, CASING MOUNTING, CASING TYPE, CASE PRESSURE RATING, HYDROTEST, ROTOR, COUPLING, and SPECIAL FITTING.
- NOZZLE CONNECTIONS:** A table with columns for No., Size, Type, Facing, and Position.
- SUCTN PRESS. REGIONS:** Designated for MAMP.
- ROTOR:** Includes fields for SHAFT FLEXIBILITY INDEX (SFI), First Critical Speed, COMPONENT BALANCE, and SHRINK-FIT LIMITED MOVEMENT IMPELLERS.
- COUPLING:** Includes COUPLING and SPECIAL FITTING.

 Pop-up boxes provide detailed technical notes, such as 'Radially split casings shall have metal-to-metal fits...' and 'If specified, for pumps in pipeline service with a maximum operating temperature of 55 °C (130 °F) or less, auxiliary connections may be threaded.'

The above chart is an example of the new API data sheet program which contains drop-down lists and pop-ups for providing easy and consistent information.

Figure N.3—CENTRIFUGAL PUMP DATA SHEET (continued)

Note	CONSTRUCTION				Rev	
1						
2						
3						
4	0.5.5	Facing	Rating	Position		
5		FF	150	END		
6		RF	300	TOP		
7		RTJ	600	SIDE		
8			900	IN-LINE		
9			1500	BOTTOM		
10				BELL	YES, NO	
11	6.4.3.2	Type	Facing	Rating	Position	
12		SW	FF	125	TOP	YES, NO
13		SWF	RF	250	SIDE	YES, NOT, BOTH DIRECTIONS
14			RTJ	300	IN-LINE	YES, NO
15				600		
16				900		
17				1500		
18				SUPPLIER, PURCHASER	9.1.1.3	YES, NO, N/A
19					YES, NO	YES, NO, N/A
20				SUPPLIER, PURCHASER	9.2.2.3	YES, NO, N/A

This third and very useful set of data sheets, Figure N.3, shows only all the drop-down choices.

Section N.2 Electronic data exchange (EDE) is for those companies who are interested in adopting (EDE) which would entail utilizing the standard BSR/HI 50.7 for the digital transfer of centrifugal pump data.

OTHER ADDITIONS & CHANGES OF INTEREST

The following items chronologically appear in the ISO 13709/ API 610 Eleven Edition document. They are mostly new items unless otherwise indicated. Where appropriate () indicates corresponding Tenth Edition paragraph number.

Note: The use of terms “hazardous and flammable” which was used extensively in previous API 610 editions, has been removed due to controversy surrounding their meaning. The National Fire Protection Association (NFPA) and other agencies define these terms differently, and , in reality the purchaser decides what is flammable and/or hazardous.

1 Scope

Relevant industry operating experience suggests pumps produced to this International Standard are cost effective when pumping liquids at conditions exceeding any one of the following:

- discharge pressure (gauge) 1 900 kPa (275 psi; 19,0 bar)
- suction pressure (gauge) 500 kPa (75 psi; 5,0 bar)
- pumping temperature 150° C (300° F)
- rotative speed 3600 r/min
- rated total head 120 m (400 ft)
- impeller diameter, overhung pumps 330 mm (13 in)

This criteria was exactly what was in API 610 8th Edition and actually should have been included in subsequent editions, but was not., hence reinstated in 11th Edition. The only difference is that in 8th Edition the criteria only applied to flammable or hazardous liquids, whereas in 11th Edition, the criteria applies to all fluids.

2 Normative references

The following references are new:

ISO 3117, Tangential keys and keyways
 ISO 9606 (all parts), Approval testing of welders - Fusion welding
 ISO 10721-2, Steel structures - Part 2: Fabrication and erection
 ISO 14120, Safety of machinery – Guards - General requirements for the design and construction of fixed and movable guards
 ISO 15156-1, Petroleum and natural gas industries -Materials for use in H₂S-containing environments in oil and gas production - Part 1: General principles for selection of cracking-resistant materials
 ISO 15609 (all parts), Specification and qualification of welding procedures for metallic materials — Welding procedure specification
 ISO/TR 17766, Centrifugal pumps handling viscous liquids — Performance corrections
 IEC 60034-2-1, Rotating electrical machines — Part 2-1: Standard methods for determining losses and efficiency from tests (excluding machines for traction vehicles)
 EN 953, Safety of machinery — Guards — General requirements for the design and construction of fixed and movable guards
 EN 13463-1, Non-electrical equipment for use in potentially explosive atmospheres — Part 1: Basic method and requirements
 ANSI/ASME B18.18.2M, Inspection and Quality Assurance for High-Volume Machine Assembly Fasteners
 ANSI/ASME B31.3, Process Piping
 API Std 547, General Purpose Form-Wound Squirrel Cage Induction Motors — 250 Horsepower and Larger
 NACE MR 0103 Materials Resistant to Sulfide Stress Cracking in Corrosive Petroleum Refining Environments

The following referenced documents have been removed:

ISO 10438, ASME B17.1, MR0175 materials for sulfide stress corrosion cracking in oilfield equipment, AWS D.1.1, EN 287, EN 288, API 5L, API RP 500, API 686, HI 1.3

3 Terms and Definitions

The following are new terms unless otherwise noted:

3.5 (3.4) best efficiency point BEP – flow rate at which a pump achieves its higher efficiency **at rated impeller diameter**.

NOTE: The best efficiency point flow rate at maximum impeller diameter is used to determine pump specific speed and suction specific speed. The best efficiency point flowrate at reduced impeller diameters is similarly reduced from the value at the maximum impeller diameter.

Note: section in bold including the note was added.

This wording is same as that shown in the latest edition of Hydraulic Institute and reflects industry practice.

3.8 Classically stiff - characterized by the first dry critical speed being above the pump’s maximum continuous speed by the following:

- 20% for rotors designed for wet running only
- 30% for rotors designed to be able to run dry

Note: This definition is used to identify when a torsional and lateral analysis is required

3.24 maximum operating temperature - highest temperature of the pumped liquid, including upset conditions, to which the pump is exposed

3.35 net positive suction head required NPSH₃ – NPSH that results in a 3% loss of head (first-stage head in a multistage pump) determined by the vendor by testing with water

Note: NPSH₃ replaces what was commonly referred to as NPSHR

3.51 similar pump – pump that is accepted, by agreement between purchaser and manufacturer as sufficiently similar to not require a lateral analysis, taking into account the factors listed for an identical pump (3.18)

3.53 stage – one impeller and associated diffuser or volute and return channel, if required)

3.63 wet critical speed – rotor critical speed calculated considering the additional support and damping produced by the action of the pumped liquid within internal running clearances at the operating conditions and allowing for stiffness and damping within the bearings.

3.61 (3.57) vertical in-line pump – vertical-axis, **single stage** overhung pump whose suction and discharge connections have a common centerline that intersects the shaft axis.

Note: Wording in bold was added

4 General

4.1 Unless otherwise specified, the pump vendor shall have unit responsibility. The pump vendor shall ensure that all sub-vendors comply with the requirements of this International Standard and all reference documents.

5 Requirements

5.2 Now labeled as “Statutory requirements” states that the purchaser and the vendor shall mutually determine the measures necessary to comply with any governmental codes, regulations, ordinances, or rules that are applicable to the equipment, its packaging and preservation.

5.3.1 In case of conflict between this International Standard and the inquiry, the inquiry shall govern. At the time of the order, the order shall govern.

6 Basic Design

6.1.1 (5.1.1) The equipment (including auxiliaries) covered by this International Standard shall be designed and constructed for a minimum service life of 20 years (excluding normal-wear parts as identified in Table 20) and at least 3 years of uninterrupted operation. **Shutting down the equipment to perform vendor-specified maintenance or inspection does not meet the continuous uninterrupted operation requirement.** It is recognized that these requirements are design criteria and that service or duty severity, mis-operation or improper maintenance can result in a machine failing to meet these criteria.

Note: Section in bold was added.

6.1.7 (5.1.9) Provision for sealing against atmospheric pressure in vacuum service is especially important when handling liquids near their vapor pressure (such as liquefied petroleum gases). **During operation, the seal chamber pressure shall be at least a gauge pressure of 35 kPa (0,35 bar; 5 psi); see ISO 21049 (API 682).**

Note: Section in bold was added.

6.1.15 (5.1.17) Pumps with heads greater than 200 m (650 ft) per stage and with more than 225 kW (300 hp) per stage shall be deemed **high-energy pumps** and can require special provisions to reduce vane passing-frequency vibration and low-frequency vibration at reduced flow rates.

Note: Section in bold was added; definition for high-energy pump is established.

6.3.5 (5.3.5) Added Note 2 – This sub clause provides minimum requirements consistent with designs existing at the time of publication of this International Standard. For the next edition, all OH, BB1, and BB2 pumps with ISO 7005-1 PN50 flanges will be required to have a casing MAWP equivalent to their flanges.

Added paragraph: The pump-seal chamber and seal gland shall have a pressure-temperature rating at least equal to the maximum allowable working pressure and temperature of the pump casing to which it is attached, in accordance with ISO 21049:2004 (API 682), 3.41.

•6.3.10 (5.3.10) Radially split casings shall have metal-to-metal fits, with confined controlled-compression gaskets, such as an O-Ring or a spiral-wound type. **Gaskets other than spiral-wound may be proposed and furnished if proven suitable for service and approved by the purchaser. Radially split pressure casing joints and bolting shall be designed to seat a spiral-wound gasket (see 9.3.2.3 for VS type pumps).**

Note: Wording in bold was added; the bullet (•) part of this paragraph is “O-rings shall be supplied if approved by the purchaser”.

Added: NOTE Table H.1 shows only spiral-wound gaskets for casing joints. Spiral-wound gaskets are generally preferred because they are perceived by users to have had better availability, are more conducive to material identification, have a broader chemical compatibility and temperature range, contact a wider sealing surface (are less susceptible to leakage because of sealing surface irregularities) and are easier to handle and store than O-rings. ISO 21049 and ANSI/API Std 682/ISO 21049, specifically require O-Ring gaskets on low-temperature [$< 175^{\circ}\text{C}$ (350°F)] pressure-seal gland plates.

Note: This is the justification for Spiral Wound preference.

6.3.16 new – If the manufacture of cast pressure-casing parts requires the use of openings for core support, core removal or waterway inspection and cleaning, these openings shall be designed so they can be closed by welding, using an appropriate, qualified weld procedure, during the completion of

casting manufacture.

Note: Some pump styles may use a bolt-on cover at the cross-over, in which case the ability to be welded closed is required.

6.4.2.6 new – To minimize nozzle loading and facilitate installation of piping, machined faces of pump flanges shall be parallel to the plane as shown on the general arrangement drawing within $0,5^{\circ}$. Bolt holes or studs shall straddle centerlines parallel to the main axes of the pump.

6.4.3 (5.4.3) Auxiliary Connections

•6.4.3.2 new – If specified, for pumps in pipeline service with a maximum operating temperature of 55°C (130°F) or less, auxiliary connections may be threaded.

Note: Basis for this allowance is that it is difficult to get hot work permits to perform welding in remote areas characteristic of pipelines.

• 6.4.3.3 new – If specified, special threaded fittings for transitioning from the casing to tubing for seal flush piping may be used provided that a secondary sealing feature, such as O-rings, are used and that the joint does not depend on the thread contact alone to seal the fluid. The connection boss shall have a machined face suitable for sealing contact.

Note: This is intended to be used on extremely critical services where highest integrity possible for connection is required.

• 6.4.3.6 new – Auxiliary connections to cast iron pressure casings may be threaded.

• 6.4.3.10 new – If specified, piping shall be gusseted in two orthogonal planes to increase the rigidity of the piped connection, in accordance with the following stipulations.

- Gussets shall be of a material compatible with the pressure casing and the piping shall be made of either flat bar with a minimum cross section of 25 mm by 3 mm (1 in by 0.12 in) or round bar with a minimum diameter of 9mm (0.38in).
- Gusset design shall be as shown in Figure 20.

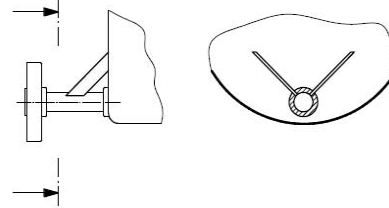


Figure 20 — Typical gusset design

c) Gussets shall be located at or near the connection end of the piping and fitted to the closest convenient location on the casing to provide maximum rigidity. The long width of gussets made with bar shall be perpendicular to the pipe and shall be located to avoid interference with the flange bolting or any maintenance areas on the pump.

d) Gusset welding shall meet the fabrication requirements (see 6.12.3), including PWHT when required, and the inspection requirements (see 8.2.2) of this International Standard.

e) Gussets may also be bolted to the casing if drilling and tapping is done prior to hydrotest.

f) Proposals to use clamped or bolted gusset designs shall be

submitted to the purchaser for approval.

Note: Increasing number of end users are specifying gusseting, so it was added to the ISO 13709/ API 610 specification.

6.4.3.14 (5.4.3.10) All pumps shall be provided with vent and drain connections, except that vent connections may be omitted if the pump is made self-venting by the arrangement of the nozzles. **Pumps that are not self-venting shall be provided with vent connections in the pressure casing, as required (see 6.8.10). If the pump cannot be completely drained for geometrical reasons, this shall be stated in the proposal. The operating manual shall include a drawing indicating the quantity and location(s) of the liquid remaining in the pump.**

Note: Section in bold has been added.

6.6 (5.6) Rotors

6.6.1 (5.6.1) Unless otherwise specified, impellers shall be of the fully enclosed, semi-open or open type.

Note: Previously, only fully enclosed impellers applied with other types with purchaser's approval.

6.6.14 new – If the vendor can demonstrate that electrical or mechanical runout is present, the demonstrated amount of runout can be vectorially subtracted from the measured vibration during the factory test as long as it does not exceed 25 % of the allowed peak-to-peak vibration amplitude or 6,5 μm (0,25 mil), whichever is less.

Note: This addition allows to remove some of the runout amount from some of the vibration readings at test.

6.8 (5.8) Mechanical Shaft Seals

• **6.8.9 new** – Seal chambers shall be designed with space available to provide for an additional flush port to approximately the centre of the chamber and extending vertically upward. If specified, this port shall be drilled and machined for a piping connection. Tapered pipe thread connections are not allowed.

Note: This addition allows to typically used to measure seal chamber pressure or to install SFP Plan 23.

• **6.8.13 new** – The vendor and purchaser shall agree on the maximum static and dynamic sealing pressures that can be anticipated to occur in the seal chamber and the vendor shall state these values on the data sheet [see 6.3.5 c]

6.9 (5.9) Dynamics

6.9.2 (5.9.2) Torsional Analysis

Note: Extensive changes are covered in the above first portion of this paper.

6.9.3 (5.9.3) Vibration

6.9.3.2 (5.9.3.2) new – Figure 33 – Locations for taking vibration readings on a) vertical in-line OH3 and b) high-speed integrally geared (OH6) pumps.

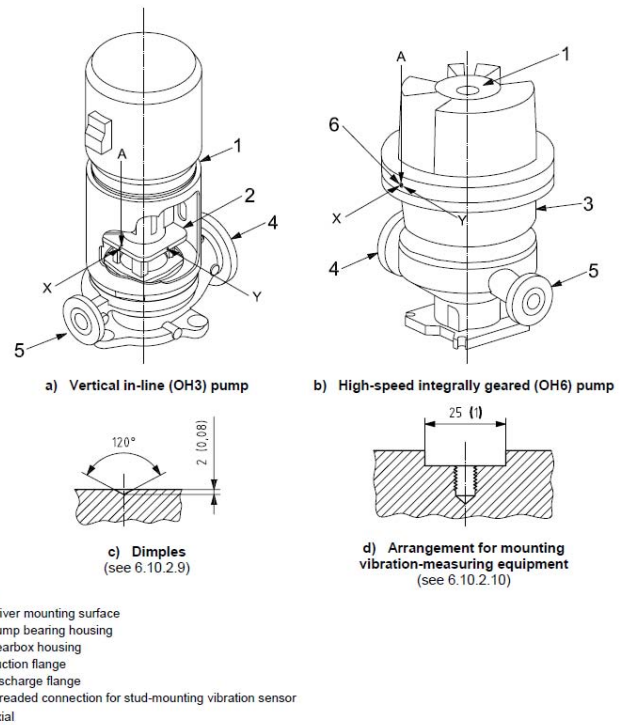


Figure 33 — Locations for taking vibration readings on [a)] vertical in-line (OH3) and [b)] high-speed integrally geared (OH6) pumps

Table 8 (Table 7) – Vibration limits for overhung and between bearing pumps

Discrete frequencies measured on the bearing housing has limit of $v_f < 2,0 \text{ mm/s RMS}$ (0,08 in/s RMS).

Also improved definitions for: v_u is the measured overall velocity; v_f is the discrete frequency velocity, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 400 lines; A_u is the amplitude of measured overall displacement; and A_f is the amplitude of displacement at discrete frequencies, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 400 lines;

Note: Value for discrete frequencies is now the correct value which goes back to API 610 8th Edition and which was intended to be used in API 610 9th and 10th Editions.

Table 9 (Table 8) – Vibration limits for vertically-suspended pumps

Discrete frequencies measured on the bearing housing has limit of $v_f < 3,4 \text{ mm/s RMS}$ (0,13 in/s RMS). Discrete frequencies for hydrodynamic guide bearing, measured on the pump shaft (adjacent to bearing) for $f < n$: $A_f < 0,33A_u$. *Also improved definitions for:* v_u is the measured overall velocity; v_f is the discrete frequency velocity; A_u is the amplitude of measured overall displacement; and A_f is the amplitude of displacement at discrete frequencies, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 400 lines;

Note: Value for discrete frequencies are now the correct values which go back to API 610 8th Edition and which was intended to be used in API 610 9th and 10th Editions.

Figure 34 – Vibration limits for horizontal pumps running above 3600 r/min or absorbing more than 300 kW (400 hp) per stage.

Added – NOTE 2: The vibration limit for discrete frequencies is: $v_f < 0,67 v_u$ allowable from Figure 34.

Note: This applies to all bearing types including hydrodynamic bearings.

6.4 (5.9.4) Balancing

6.9.4.1 (5.9.4.1) added – For single-stage BB1 and BB2 pump rotors with interference fit components, the vendor may choose to balance the assembled rotor (in accordance with 9.2.4.2) instead of balancing major rotating components individually.

6.10.1 (5.10.1) Bearings

6.10.1.3 (5.10.1.4) added – Non-metallic cages shall not be used.

•6.10.1.6 new – Bearing system life

Note: Extensive changes are covered in the above first portion of this paper.

Table 10 (Table 9) – Bearing selection

Changed: limits

a) Rolling-element bearing speed: For all bearing types, the bearing manufacturer's published nominal speed limitations shall not be exceeded. For ball bearings, factor ndm for individual bearings shall not exceed 500 000 for oil lubricated and 350 000 for grease lubricated bearings.

Note: ndm factor for grease has been added.

added – NOTE 1: The bearing temperature limits in 6.10.2.4 can limit ndm factors to even lower values.

added – NOTE 2: Roller and spherical bearings generally have lower speed limitations than ball bearings.

6.10.2.7 (5.10.2.8) reworded: If oil-mist lubrication is specified, the requirements of 6.10.2.7.1 or 6.10.2.7.2 shall apply.

Note: Previous API 610 Tenth edition had some ambiguities in requirements for pure vs. purge oil mist requirements. To simplify understanding of the requirement; one section now addresses pure mist while the second addresses purge mist.

6.10.2.7.1 (5.10.2.8 d) changed – For pure oil mist lubrication, bearings and bearing housings shall meet the following requirements:

a) A threaded 6 mm (NPS 1/4) oil-mist inlet connection shall be provided on the housing or end cover for each of the spaces between the rolling element bearing or bearing set and the bearing housing end seal.

b) Oil-mist fitting connections shall be located so that oil mist can flow through rolling element bearings.

NOTE: Reclassifiers and oil-mist fittings are normally installed in the field.

Note: Removed 10th Edition 5.10.2.8 a) wording: "If bearing housing design is such that short circuiting cannot be avoided, directional oil-mist reclassifiers may be furnished to ensure

positive oil-mist circulation through the bearings".

6.10.2.7.2 new – For purge oil-mist lubrication, bearings and bearing housings shall meet the following requirements in a) to d) below:

a) A threaded 6 mm or 12 mm (NPS 1/4 or 1/2) oil-mist connection shall be located in the top half of the bearing housing to act also as a vent-and-fill connection.

b) Constant-level oilers shall be provided, and a mark indicating the oil level is required on the bearing housing.

Bearing lubrication is by a conventional oil bath, flinger or oil ring system.

c) Constant-level sight feed oilers shall be equipped with overflow control to allow excess coalesced oil from the mist system to drain from the bearing housing so that oil level in the sump is maintained at proper level. The oil shall be contained to prevent it from draining onto the baseplate.

d) Constant-level sight feed oilers shall be piped so that they operate at the internal pressure of the bearing housing, do not vent excess mist at the bearing housing, or allow oil to drip to the baseplate.

6.10.2.7.3 new – For both pure and purge mist applications, a drain connection shall be located on the bottom of the bearing housing to provide complete oil drainage. (See 6.10.2.7.5.)

6.10.2.7.5 (5.10.2.8 a) The oil-mist supply, reclassifier(s) and drain fitting shall be provided by the purchaser. Unless otherwise specified, directional reclassifiers, if required, shall be provided by the machine manufacturer.

Note: Before responsibility was not assigned, whereas now it is clearly by the purchaser.

6.12 (5.12) Materials

•6.12.1.12 • (5.12.1.12) NACE Requirements

Note: Extensive changes are covered in the above first portion of this paper.

6.12.3 (5.12.3) Welding

•6.12.3.1 Table 11 (•5.12.3.1 Table 10)

Table 11 – Welding requirements

Requirement	Applicable code or standard
Welder/operator qualification	ASME BPVC IX or ISO 9606 (all parts)
Welding procedure qualification	Applicable material specification or, where weld procedures are not covered by the material specification, ISO 15609 (all parts), ASME BPVC IX or ANSI/ASME B31.3
Non-pressure-retaining structural welding, such as baseplates or supports	ISO 10721-2
Magnetic-particle or liquid-penetrant examination of the plate edges	ASME BPVC VIII, Division 1, UG-93(d)(34)
Post-weld heat-treatment	Applicable material specification, EN 13445-4, ASME BPVC VIII, Division 1, UW 40, or ANSI/ASME B31.3
Post-weld heat-treatment of casing fabrication welds	Applicable material specification, EN 13445-4, or ASME BPVC VIII, Division 1

Table 11 now contains all latest applicable codes.

6.12.3.3 d) new - Plate edges shall be inspected by magnetic-particle or liquid penetrant examination as required by internationally recognized standards, such as ASME BPVC Section VIII, Division 1, UG-93 (d) (3)

Note: This is primarily applicable to OH2 casing covers out of plate as well as to the vertical pumps fabricated heads, bowls, etc.

6.12.4 (5.12.4) Low Temperature Service

6.12.4.2 (5.12.4.2) added – The vendor should, therefore, exercise caution in the selection of materials, fabrication methods and welding procedures for parts intended for services below 40°C (100°F)

Note: Previous Tenth Edition stated: services between -30°C (-20°F) and 40°C (100°F)

6.13.5 new – Nameplates and rotation arrows (if attached) shall be of austenitic stainless steel or of nickel-copper alloy [equivalent to Monel]. Attachment pins shall be of the same material as the nameplate or rotation arrow. Welding is not permitted as an attachment method.
Footnote added: Monel is an example of a suitable product available commercially.

7.1.4 (6.1.3) changed – Motors shall have **nameplate** power ratings, excluding the service factor (if any) at least equal to the percentages of power at pump rated conditions given in Table 12. However, the power at rated conditions shall not exceed the motor nameplate rating. **The smallest acceptable motor power rating to be supplied is 4 kW (5 hp)**. If it appears that this procedure leads to unnecessary oversizing of the motor, an alternative proposal shall be submitted for the purchaser's approval.

7.2 (6.2) Couplings and guards

7.2.2 b) (6.2.2 b)) added – NOTE 1: The use of bolt heads or flexible element fasteners alone to retain the spacer if a flexible membrane ruptures might not provide adequate support because they are subject to wear if and when failure occurs.

7.2.2 d) (6.2.2 d)) changed – d) **The distance between the pump and driver shaft ends (distance between shaft ends, or DBSE) shall be greater than the seal cartridge length for all pumps other than OH type or at least 125 mm (5 in)** and shall permit removal of the coupling, bearing housing, bearings, seal and rotor, as applicable, without disturbing the driver, driver coupling hub, **pump coupling hub** or the suction and discharge piping. **For Types BB and VS pumps, this dimension, DBSE, shall always be greater than the total seal length, l, listed in Table 7, and shall be included on the pump data sheet (Annex N).**

NOTE 2: The DBSE dimension usually corresponds to the nominal coupling spacer length

Note: In bold, beginning part of paragraph has changed. New wording added to address BB and VS pump types. NOTE 2 is also new.

7.2.6 (6.2.6) changed – Flexible couplings shall be keyed to the shaft. Keys, keyways and fits shall conform to AGMA 9002, Commercial Class. **Shaft coupling keyways shall be cut to accommodate a rectangular cross section key. Sled-runner type keys and keyways shall not be provided. Keys shall be fabricated and fitted to minimize unbalance.**

Note: New portion in bold was added. Sled-runner key and keyways are not allowed.

•7.2.15 new – If specified for coupling guards with potentially explosive atmospheres, an “ignition hazard assessment” (risk analysis) in accordance with EN 13463-1 shall be conducted and a suitable report

7.3 (6.3) Baseplates

•7.3.1 (6.3.1) changed – The purchaser shall specify the rim or pan type as follows: a) drain rim surrounding the entire baseplate; b) drain pan surrounding the entire baseplate; c) partial drain pan that covers the entire width of the baseplate.

Note: Now it is the responsibility of the purchaser to specify exactly what style of baseplate is required.

•7.3.6 (6.3.4) added – If specified, in addition to shim packs, a stainless steel spacer plate of not less than 5 mm (0,200 in) thickness, machined on both sides, and of the same length and width as the specific mounting feet, shall be furnished and installed under all equipment feet, including the pump, driver, and any speed increaser or reducer.

Note: This is a new bulleted portion to the paragraph

•7.3.14 new – If specified, the baseplate shall be supplied without a deck plate, i.e. open deck design.

7.3.17 (3.14) added – To prevent distortion, machining of mounting pads shall be deferred until welding on the baseplate in close proximity to the mounting pads has been completed.

7.5 (6.5) Piping and appurtenances

•7.5.2.4 (6.5.2.5) added – Orifice openings shall not be less than 3 mm (0,12 in) in diameter. **Orifice hole size shall be stamped on the orifice plate. The purchaser shall specify orifice tagging or labeling requirements.**

Note: Added portion in bold.

7.5.2.5 new – Drain valves and a drain manifold shall be supplied for pumps that require more than one drain connection. The drain manifold shall be inside the drain pan limits.

Drain valves are not required for pumps that can be drained with one drain connection. The vendor shall provide space on the baseplate for a purchaser-supplied drain valve inside the pump drain pan or drain rim.

•7.5.2.8 (• 6.5.2.8) changed – The purchaser shall specify where flanges are required in place of socket-welded unions. With purchaser approval, socket-welded unions may be used in place of flanges at the first connection from the seal gland.
Note: Removed wording “flammable and hazardous fluids” since now the requirement for socket-welding connections apply to all fluids (except for those allowed when using cast-iron I-1 and I-2 materials)

added – NOTE: Threaded connections are allowed on gland connections (see 6.4.3.11). If the remainder of the piping arrangement is flanged, repeated assembly and disassembly

can overstress this threaded connection, since adjustment is possible only in 90° increments. Socket-welded unions supplied in stainless steel tend to leak after repeated assembly and disassembly.

7.5.2.9 new – Threaded piping joints may be used only on seal glands, instrumentation connections and for pumps of cast iron construction (Class I-1 or I-2 in Table H.1).

7.5.2.10 new – Transmitters and pressure gauges shall have block-and-bleed valves.

Inspection 8.2 (7.2)

Note: Extensive details on the additions of the three classes of casing inspection requirements are covered in the above first portion of this paper.

•8.2.2.4 (•7.2.2.1) changed – **Unless otherwise specified, inspection methods and acceptance criteria shall be in accordance with those in Table 15** as required by the material specification. If additional radiographic, ultrasonic, magnetic-particle or liquid-penetrant examination of the welds or materials is specified by the purchaser, the methods and acceptance criteria shall also be in accordance with the standards shown in Table 15. Alternative standards may be proposed by the vendor or specified by the purchaser. The welding and material inspection data sheet in Annex N may be used for this purpose.

Note: Words added in bold reinforces that Table 15 compliance is required. The rest of the paragraph is virtually the same as shown in Tenth Edition.

Table 15 (Table 13) changed – *updated to latest specification references and added visual inspection details.*

Table 15 — Materials inspection standards

Type of inspection	Methods	Acceptance criteria	
		For fabrications	For castings
Radiography	ASME BPVC, Section V, Articles 2 and 22	ASME BPVC, Section VIII, Division 1, UW-51 (for 100 % radiography) and UW-52 (for spot radiography)	ASME BPVC, Section VIII, Division 1, Appendix 7
Ultrasonic inspection	ASME BPVC, Section V, Articles 5 and 23	ASME BPVC, Section VIII, Division 1, Appendix 12	ASME BPVC, Section VIII, Division 1, Appendix 7
Liquid-penetrant inspection	ASME BPVC, Section V, Articles 6 and 24	ASME BPVC, Section VIII, Division 1, Appendix 8	ASME BPVC, Section VIII, Division 1, Appendix 7
Magnetic-particle inspection	ASME BPVC, Section V, Articles 7 and 25	ASME BPVC, Section VIII, Division 1, Appendix 6	ASME BPVC, Section VIII, Division 1, Appendix 7
Visual Inspection (all surfaces)	ASME BPVC, Section V, Article 9	In accordance with the material specification and the manufacturer's documented procedures	MSS SP-55

•8.2.2.8 new – If specified, pressure boundary parts of alloy materials shall be subject to positive material identification (PMI) using recognized testing methods, instrumentation and standards. The purchaser and vendor shall agree on the specific parts tested, procedures used and acceptance criteria. Only techniques providing quantitative results shall be used. Mill test reports, material composition certificates, visual stamps or markings shall not be considered substitutes for PMI testing.

NOTE: PMI is not available to differentiate between grades of carbon steels.

Note: This is the first time PMI is addressed in ISO13709/ API 610 compared to previous editions and reflects the current

industry practice for using it.

8.3 (7.3) Testing

8.3.2 ((7.3.2) Hydrostatic Test

8.3.2.1 new – The intent of a hydrostatic test of a centrifugal pump casing is to ensure that the design and construction of the pump pressure containing components and joints are leak-free from ambient conditions to the maximum operation conditions defined on the data sheet.

8.3.2.3 new – The test set-up and/or apparatus shall not provide stiffening that improves the integrity of any joint.

8.3.2.12 (7.3.2.1 a)) changed – Double-casing pumps, horizontal multistage pumps, integral-gear pumps (as described in 6.3.6), and other special-design pumps as approved by the purchaser may be segmentally tested. **Seepage past internal closures required for testing of segmented cases and operation of a test pump to maintain pressure is acceptable.**

Note: Added wording in bold. It is possible to isolate internal parts to just check for leakage, not pressure drop.

8.3.2.13 (7.3.2.4) added – NOTE: It is not necessary to hydrostatically test piping systems assembled with tubing or threaded connections after assembly.

8.3.2.14 new – Unless otherwise specified, single-stage overhung-pump casing components with a radial joint (mean gasket diameter) 610 mm (24 in) in diameter or less may be hydrostatically tested as components or subassemblies provided that joint design integrity has been proven by qualification testing for the specific size of pump and pressure rating in question.

8.3.3 (7.3.3) Performance Testing

Note: Extensive changes are covered in the above first portion of this paper.

8.3.4 (7.3.4) Optional tests

•8.3.4.3.2 (7.3.4.2.2) changed – A 3% drop in head (**first stage head on pumps with two or more stages**) shall be interpreted as indicating performance impairment, thus the terminology NPSH3. The first-stage head of pumps with two or more stages shall be measured using a separate connection to the first-stage discharge if possible. If this is not feasible, testing of the first stage only should be considered. **With purchaser approval, first-stage head may be determined by dividing total developed head by the number of stages.**

Note: Added wording in bold; vendor to advise if optional approach is needed and purchaser to approve.

• 8.3.4.3.3 new – The NPSH required test shall determine the actual NPSH required at a 3% head drop. Unless otherwise specified or agreed, curves shall be developed at constant flow by reducing the NPSHA to a point where the head curves break away from that developed with sufficient NPSHA (8.3.4.3.2) by at least 3%. The NPSH required test shall start with at least the same NPSHA as the performance test and at least twice the NPSH3 shown on the proposal curve. The first two test points

shall not differ by more than the uncertainty of the head measurement. If the second test point at the same flowrate shows a decrease in differential head then the NPSHA shall be increased to a value sufficient to establish two consecutive points of equal head. The first two points shall be separated by a minimum of 1 m (3 ft) of NPSHA. These NPSH3 curves shall be developed and submitted in accordance with Hydraulic Institute Standards (ANSI/HI 1.6) or ISO 9906, as specified.

The test shall not proceed beyond a 20% head breakdown (20% of first-stage head for multistage pumps).

NOTE: If 8.3.3.6 is specified, it is possible that the head has already been affected by insufficient NPSHA, so starting at a higher NPSHA is desirable.

8.3.4.4 (7.3.4.3) Complete unit test

• **8.3.4.4.2 new** - The acceptable vibration limits of each component of the train shall be as per its applicable standards and specifications, except for reciprocating engines. (In this case, limits shall be mutually agreed upon by purchaser, pump vendor and engine supplier.)

Note: This addresses complete unit test option.

8.4 (7.4) Preparation for shipment

8.4.8 new - Horizontal pumps, and all furnished drivers and auxiliaries, shall be shipped fully assembled on their baseplates, except as noted below. Coupling spacers with bolts and other items, such as minimum flow orifices that are not part of the assembled pumping unit, shall be separately boxed, tagged and securely attached to the baseplate.

8.4.9 new - Drivers for vertical pumps and horizontal drivers with a mass over 200 kg (450 lb) may be removed after shop mounting and alignment and shipped separately but alongside pump. Vertical pumps with suction cans shall be shipped with the suction cans (barrels) removed.

8.4.10 new - If it is necessary to ship other major components separately, prior purchaser approval is required.

8.4.11 new - Metal filter elements and screens shall be cleaned and reinstalled prior to shipment. Non-metallic filter elements shall be shipped and installed in an unused condition.

8.4.12 new - Suitable rust preventatives shall be oil-soluble and compatible with all pumped liquids.

9 (8) Specific pump types

9.1.1 (8.1.1) Horizontal (type OH2) pumps

9.1.1.2 new – The distance between the pump and driver shaft ends (distance between shaft ends, or DBSE) shall permit removal of the coupling spacer and back pullout assembly without disturbing the driver, coupling hubs or casing.

Note: This is also mentioned in 7.2.2 b)

• **9.1.1.3 new** – If specified, the shaft flexibility index shall be calculated by the vendor in accordance with K.1 and stated on the data sheet.

The design and operation requirements for overhung pump

rotors are detailed in several areas of this standard. K.1 lists these requirements and establishes a standardized process of calculating a shaft flexibility index that may be used to evaluate these latter parameters and to establish a baseline for the comparison of shaft flexibility.

Note: Extensive changes are covered in the above first portion of this paper

•9.2 (8.2) Between-bearings pumps (types BB1, BB2, BB3 and BB5)

• **9.2.1.5 new** - For pumps with machined and studded suction and discharge nozzles, the vendor shall provide the minimum acceptable length for break-out spool pieces to facilitate maintenance activity. Spool pieces should be provided by the purchaser.

9.2.4.2 (8.2.4.2) Rotor balancing-Table 19 (Table 17) changed – Added column for rotor balance procedure basis ISO 11342 and updated the corresponding notes.

Table 19 — Rotor balance requirements

Component fit on shaft	Maximum continuous speed r/min	Flexibility factor, L^4/D^2 mm ² (in ²)	Rotor balance procedure(s) ^a	Rotor balance grade
Clearance	≤ 3 800 ^a	No limit	C	c
	≤ 3 800	No limit	C + B or D	G2.5 (B#7) ^d
Interference	> 3 800	≤ 1.9 · 10 ⁹ (3.0 · 10 ⁶)	C + B or D	G1 (4#7) ^{d,e}
NOTE See Table 17 for shaft and rotor runout requirements.				
^a To allow for 5 % speed increase.				
^b See ISO 11342.				
^c Balance correction during assembly is not feasible because clearance fit does not maintain corrected balance.				
^d Approximately equal to the midpoint of the corresponding ISO balance quality grade.				
^e If rotors of higher flexibility are used at speeds above 3 800 r/min, achieving and maintaining this balance level requires special attention to design, manufacture and maintenance.				

9.2.5 (8.2.5) Bearings and bearing housings

9.2.5.2.2 (8.2.5.2.2) changed – Thrust collars **shall be replaceable and shall be mounted to the shaft with an interference fit** to prevent fretting and positively locked to prevent axial movement.

Note: Added wording in bold.

9.3 (8.3) Vertically suspended pumps (types VS1 through VS7)

9.3.2 (8.3.2) Pressure Casings

9.3.2.3 new - Assemblies designed for O-ring seals only do not require flanges and bolting designed to seat a spiral-wound gasket (see • 6.3.10).

Note: • 6.3.10 does require customer approval when o-ring sealing is used instead of spiral wound gaskets.

9.3.3 (8.3.3) Rotors

9.3.3.1 (8.3.3.2) added – For pumps with a shaft length over 4 500 mm (177 in), the vendor may propose an alternative total indicated runout (over 80 μm (0,003 in)) limit to the purchaser for approval.

9.3.8.2 (8.3.8.2) Mounting plates

9.3.8.4 new - The outside corners of the sole plate or mounting plate imbedded in the grout shall have at least 50 mm (2 in) radii in the plan view. (See Figure D.1.)

Note: Figure D.1 is in Annex D, standard baseplates

9.3.10.1 (8.3.10.1) added – The components that constitute the pressure casing are the casing (bowls), column and discharge head. **It is not necessary that bowls on VSI pumps in S-6 materials be 12 % chrome; they can be carbon steel.**

Note: Words in bold have been added.

10.3(9.3) Contract Data

10.3.2.3 new - If specified, the vendor shall furnish an outline of the procedures to be used for each of the special or optional tests that have been specified by the purchaser or proposed by the vendor.

Annex A Specific speed and suction-specific speed

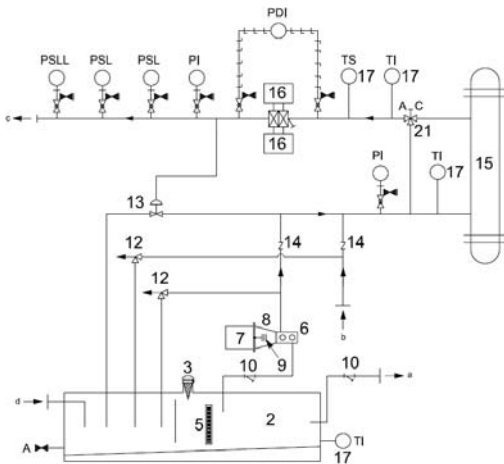
added – Note 2: For simplicity, industry omits the gravitational constant from the dimensionless equations for specific speed and suction-specific speed.

Annex B Cooling water and lubrication system schematics
deleted – cooling plans D

Note: This was Figure B.3 in Tenth Edition; with this deletion, figure numbers have changed.

Figure B.8 and Table B.1 – Key items for Figure B.8 with additional requirements

Note: This replaces Tenth Edition Figure B.10 and Table B.1 – Lubricating-oil system schematic and is quite different and more clearly understood.



See Table B.1 for description of key items and console class.

NOTE 1 In the title, "BP0" indicates that the base plate is part of equipment, "BP1" indicates a stand-alone console, reservoir-mounted. The P&IDs are similar.

NOTE 2 Figure modified from ISO 10438-2:2007, Figure B.1. For the purposes of this provision, API 614-08 is the equivalent of ISO 10438-2:2007.

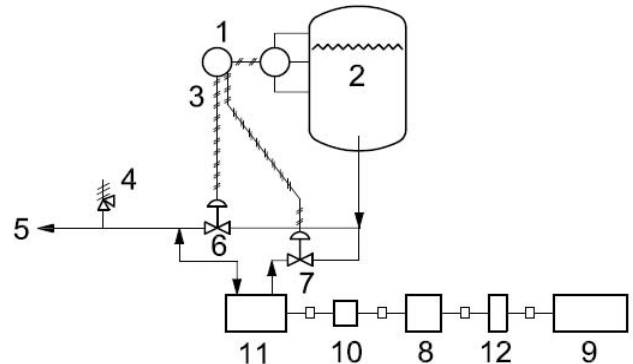
Figure B.8 — Class II-P0-R1-H0-BP0-C1F2-C0-PV1-TV1-BB0 or Class II-P0-R1-H0-BP1-C1F2-C0-PV1-TV1-BB0

Table B.1 — Key items for Figure B.8 with additional requirements

Key item	Identification/subclause	Note/option	Comments
	Basic design, 4.1	Specify	Console class & code Class II – P0-R1-H0-BP0-C1F2-C0-PV1-TV1-BB0
1	Baseplate		
2	Oil reservoir, 4.4	Specify	4.4.2 Bottom sloped to drain
3	Filter/breather		
4	Drain	Specify	4.4.3 Drain connection (with valve and blind flange) at least 5 cm diameter
5	Level gauge	Specify	4.4.5 d) Oil level glass
6	Lube oil pump		
7	Lube oil pump motor		
8	Pump/motor bracket		
9	Coupling		
10	Strainer		
11	Foot valve/strainer		
12	Pressure-limiting valve		
13	Pressure-control valve		
14	Check valve		
15	Cooler, 4.6		
16	Filter		
17	Thermowell		
18	Vent		
19	Drain		
20	Reservoir heater	Option	4.4.7 a) Electric immersion heater is optional
21	Temperature control valve	Option	4.6.13 Thermostatically operated three-way temperature control valve (TV1) is optional
	Oil piping, 5.2		
PSLL, PSL, PI, PDI	Instrumentation, Clause 6 Pressure indicators/switches Differential pressure		See ISO 10438-2:2007, Figure B.25. See ISO 10438-2:2007, Figure B.32.
a, b, c, d	To shift-driven pump From shift-driven pump To equipment bearing housings From equipment bearing housings	Change Add	Oil drain piping shall have a minimum slope of 1:50 (20 mm/m [0,25 in/ft]) 6.2 Table 3: a) PSLL for shutdown on low-low oil pressure b) TS for high oil temperature at cooler outlet c) TI in the oil drain line from each bearing or lubricated coupling

Annex C Hydraulic power recovery turbines

Figure C.1 – Typical HPRT added – New arrangement b) Pump drive at greater than motor speed; relabeled a) Pump Drive at motor speed. Key legend now shows the addition of a gear.



b) Pump drive at greater than motor speed

Note: b) is new diagram added to Figure C.1

Annex E Inspector’s checklist

changed – reference paragraph numbers to correspond with latest Eleventh Edition changes.

added – Oil and bearing temperature (certified)

added – Mechanical run test

Annex F Criteria for piping design

new – NOTE: In order to evaluate the actual machine distortion (at ambient conditions), the piping alignment checks required in API RP 686, Chapter 6, should be performed. API RP 686

allows only a small fraction of the permitted distortion resulting from use of the values from this annex.

Note: This applies to nozzle load checks which the customer will do in the field.

changed – Section F.2 Vertical in-line pumps now specifically applies to OH3 and OH6 pumps only.

Annex G Materials class selection guidance

Table G.1 footnote h added – Materials selected for low-temperature services shall meet the requirements of 6.12.1.6 and 6.12.4. Casting alloy grades LCB, LC2 and LC3 are shown only for reference. **Grades LCB, LC2 and LC3 refer to ISO 4991. C23-45BL, C43E2aL and C43L are equivalent to ASTM A352/A352M, grades LCB, LC2 and LC3.** Use equivalent materials for wrought alloys.
Note: Words in bold have been added.

Annex H Materials and material specifications for pump parts

Note: Table H.4 in Tenth Edition became H.3; Table H.5 became H.4; Table H.3 “miscellaneous material specifications” was removed as it contained guidelines for materials such as babbitt, bronze, hard-facing, ni-resist, sheet gasket, precipitation-hardening nickel alloy, precipitation-hardening stainless steel, low-carbon nickel-moly-chrome alloy and nickel-copper alloy which basically are no longer used.

Table H.1 Material classes for pump parts

changed – Case and gland studs for material class I-1 and I-2 changed from carbon steel to 4140 alloy steel; added – label “Trim Material” to part description.

changed – Footnote j “Super Duplex” Pitting Resistance Equivalency (PRE) has been changed to:

PRE = wCr + 3,3wMo + 16wN where **w** is the percentage mass fraction of the element indicated by the subscript

Note: Wording in bold has been added.

New – Footnote m) For applications where large differences of thermal expansion can result if austenitic stainless steel fasteners are used, alternative fastener materials, such as 12% or 17% Cr martensitic steel, with appropriate corrosion resistance, may be used.

Table H.1 Material classes for pump parts

changed – Extensive changes made to reflect latest material standards: CA 15 no longer allowed as pressure casting for 12% Cr class; added ASTM A276 Gr. 316L for bar stock (material class: austenitic stainless steel); added – the ASTM A955 Gr 1B73A/4A for bar stock (material class: Duplex); deleted – A 351 Gr CD4MCu (material class: Duplex); deleted A351 Gr CD3MWCN (material class: Duplex)

ISO 13709:2009(E) added metric Was EN 10213-2 was FC 300

was class 2

was C 35e

was blank

was 2604-2-F31

Note: CA15 was removed for pressure casting

was EN 10213-2

New

New

why are these not the same

Table H.2 – Material specifications for pump parts

Material class	Applications	International ISO	USA ASTM	UNS ^a	EN ^b	Europe Grade	Material No.	Japan JIS
Cast iron	Pressure castings	1895/Gr 250	A276 (C352) Class 30	F12401	EN 1561	EN-GJL-250	JL 1040	G 5501, FC 230
	General castings	1895/Gr 300	A48/A48M Class 300/300-03	F11701 ^c F12101	EN 1561	EN-GJL-250 EN-GJL-300	JL 1040 JL 1050	G 5501, FC 250/300
	Pressure castings	4991 G23-45-AH	A219/A219M Gr W52	J03002	EN 10213	GP 240 GH	1 0619	G 5151, CI 5CPH 2
Carbon steel	Wrought forgings	683-18-C25	A299 Class 2	K03506	EN 10222-2	P 280 GH	1 0426	G 3202, CI 5FVC 2A, 5NB7
	Bar stock pressure	683-18-C25	A696 Gr B40	G10200	EN 10273	P 295 GH	1 0481	G 4051, CI 525C
	Bar stock general	683-18-C45e	A576 Gr 1045	G10450	EN 10083-2	C 45	1 0503	G 4051, CI 545C
	Balls and studs	2604-2-F31	A193/A193M Gr B7	G41400	EN 10269	42 Cr Mo 4	1 7225	G 4107, Class 2, 5NB7
	Nuts	683-18-C45	A194/A194M Gr 2H	K04002	EN 10269	C 45 E	1 1181	G 4051, CI 545C
	Plate	9128-A P 355 TN/ P4 355 TN	A106/A106M Gr B	K03506	EN 10208-1	L 245 GA	1 0459	G 345E, Gr. STPT 370H10
	Pipe	9321-2 PH05	A106/A106M Gr B	K03506	EN 10208-1	L 245 GA	1 0459	G 4851, CI 525C G 3202, CI 5FVC 2A, 5FVC2B
	Fittings	—	A105/A105M	K03504	—	—	—	—
	Bar stock	—	A434 Class BB A434 Class CC	G41400F	EN 10083-1	42 Cr Mo 4	1 7225	G 4105, CI 5CM 440
	Balls and studs	2604-2-F31	A193/A193M Gr B7	G41400	EN 10269	42 Cr Mo 4	1 7225	G 4057, Class 2, 5NB7
Pressure castings	—	A48/A48M Gr CA8MM	J01540	EN 10213	GX 4 Cr Ni 15-4	1 4317	G 5131, CI 5C3 6, 5C5 6X	
General castings	—	A743/A743M Gr CA 15	J01150	EN 10203	GX 12 Cr 12	1 4011	G 5121, CI 5C5 1, 5C5 1X1	
Pressure castings	—	A743/A743M Gr CA8MM	J01540	EN 10203	GX 4 Cr Ni 15-4	1 4317	G 5121, CI 5C3 6, 5C5 1X1	
Wrought forgings pressure	683-13-3	A192/A192M Gr F50 Cl 1	S41000	EN 10250-4 EN 10222-5	X12 Cr13	1 4006	G 3214, Gr. SUS 410A G 3214, CI 5U3	
Wrought forgings general	683-13-2	A473 Type 410	S41000	EN 10088-5	X 12 Cr 13	1 4006	G 3214, Gr. SUS 410 A	
Bar stock pressure	683-13-3	A473/A473M Type 410	S41000	EN 10272	X12 Cr 13	1 4006	G 4303 Gr. SUS 410 or 403	
Bar stock general	683-13-3	A276 Type 410	S41400	EN 10088-5	X 12 Cr 13	1 4006	Gr. SUS 410 or 403	

Note: This is an example of changes made to the first page of Table H.2

Table H.3 (H.4) – Non-metallic wear part materials added – PFA/CF reinforced composite material

Table H.3 – Non-metallic wear part materials

Material	Temperature limits °C (°F)		Limiting pressure differential per wear part linear measure of 25 mm (1.0 in) kPa (bar, psi)	Application
	min.	max.		
Polyether ether ketone (PEEK)	-30 (-20)	135 (275)	2 000 (20; 300)	Stationary parts
Chopped-carbon-fibre filled				
Polyether ether ketone (PEEK)	-30 (-20)	230 (450)	3 500 (35; 500), or 14 000 (140; 2 000) if suitably supported	Stationary or rotating
Continuous-carbon-fibre wound				
PFA/CF reinforced composite	-46 (-50)	230 (450)	2 000 (20; 300)	Stationary parts
Carbon graphite				Stationary parts
Resin-impregnated	-50 (-55)	285 (550)	2 000 (20; 300)	
Babbitt-impregnated	-100 (-150)	150 (300)	2 750 (27.5; 400)	
Nickel-impregnated	-195 (-320)	400 (750)	3 500 (35; 500)	
Copper-impregnated	-100 (-450)			

Non-metallic wear part materials that are proven compatible with the specified process liquid may be proposed within the above limits. See 6.7.4 c).
Such materials may be selected as wear components for mating against a suitably selected metallic component such as hardened 12 % Cr steel or hard-faced austenitic stainless steel. Materials may be used beyond these limits if proven application experience can be provided, and if approved by the purchaser.

Table H.4 (H.5) – Piping materials

changed – Flange bolting in Tenth Edition shown as low alloy steel (ASTM A193 Gr B7; ASTM A194 Gr 2H) is replaced with 4010 alloy steel. Removed the Tenth Edition footnote that “ASTM standards listed are examples of acceptable materials for each type.

changed – Under auxiliary process liquid column, the two categories changed to “Material classes I-1 and I-2” and “All weldable materials”.

Auxiliary process liquid		Auxiliary process fluid	
Category		Category	
Material classes I-1 and I-2	All weldable materials	Non-flammable/ Non-hazardous	Flammable/ Hazardous

Note: This is in line with previous references whereby the use of wording flammable/ hazardous has been removed from the API/ISO document. It is now clear that welded piping connections are required for all API material classes except for I-1 and I-2 which have cast iron casings and represent the only time that threaded piping connections are allowed (as is the mechanical seal gland connection).

Note: Annex G (though informative) shows I-1 and I-2 to be used for water, BFW and sodium carbonate services.

Table H.4 — Piping materials

Component	Fluid					
	Auxiliary process liquid		Steam		Cooling water	
	Category		Gauge pressure kPa (bar, psi)		Nominal size	
	Material classes I-1 and I-2	All weldable materials	< 500 (5, 75)	> 500 (5, 75)	Standard ≤ DN 25 (1 NPS)	Optional ≥ DN 40 (1 1/2 NPS)
Pipe	Seamless ^a	Seamless ^a	Seamless ^a	Seamless ^a	—	Carbon steel (galvanized to ISO 10684 or ASTM A153/ A153M)
Tubing ^b	Stainless steel (Seamless type 316)	Stainless steel (Seamless type 316)	Stainless steel (Seamless type 316)	Stainless steel (Seamless type 316)	Stainless steel (Seamless type 316)	—
All valves	Class 800	Class 800	Class 800	Class 800	Class 200 Bronze	Class 200 Bronze
Gate and globe valve	Bolted bonnet and gland	Bolted bonnet and gland	Bolted bonnet and gland	Bolted bonnet and gland	—	—
Pipe fittings and unions	Forged class 3 000	Forged class 3 000	Forged class 3 000	Forged class 3 000	Malleable iron (galvanized to ISO 10684 or ASTM A153/ A153M)	Malleable iron (galvanized to ISO 10684 or ASTM A153/ A153M)
Tube fittings	Manufacturer's standard	Manufacturer's standard	Manufacturer's standard	Manufacturer's standard	Manufacturer's standard	—
Fabricated joints ≤ DN 25 (1 NPS)	Threaded	Socket-welded	Threaded	Socket-welded	Threaded	—
Fabricated joints ≥ DN 40 (1 1/2 NPS)	—	—	—	—	—	Purchaser to specify
Gaskets	—	Austenitic stainless steel spiral-wound	—	Austenitic stainless steel spiral-wound	—	—
Flange bolting	—	4140 alloy steel	—	4140 alloy steel	—	—

^a Schedule 80 shall be used for pipe sizes from DN 15 to DN 40 (NPS 1/2 to NPS 1 1/2); schedule 40 shall be used for sizes DN 50 (2 NPS) and larger.

^b Acceptable tubing sizes (in accordance with ISO 4200) are the following:
 12,7 mm dia. - 1,05 mm wall (1/2 in dia. - 0,065 in wall);
 19 mm dia. - 2,6 mm wall (3/4 in dia. - 0,096 in wall);
 25 mm dia. - 2,9 mm wall (1 in dia. - 0,109 in wall).

Annex I (Annex I) Lateral analysis

I.1.1 (I.1.1) changed – If a lateral analysis is required (see 9.2.4.1), the method and assessment of results shall be as specified in I.1.2 through I.1.5. Table I.1 illustrates the analysis process. The method and assessment specified are peculiar to **horizontal-axis** liquid-handling turbomachines.

Note: Wording changes in bold. The impact of this change is that this requirement does not apply to vertical pumps.

Table I.1 (Table I-1) Added – “with proven operating record”

Table I.1 — Rotor lateral analysis logic diagram

Step	If...	then...
1	the pump and conditions of service are identical or similar to an existing pump with a proven operating record	analysis is not needed.
2	the rotor is classically stiff (6.9.1.2),	analysis is not needed.
3	neither 1 nor 2 is true,	analysis is required.

I.1.2 a) (I.1.2 a)) changed – the rotor's first, second and third "dry" **bending natural frequencies** (see 9.1.2) .

Note: Words in bold changes from Tenth Edition “critical speeds”

I.1.2 (I.1.2) added – NOTE 2: Usual design practice is to investigate overhung modes, coupling and thrust collar, and set their first bending natural frequency at separation margin at least 20% above the highest potential excitation frequency (based on maximum continuous speed) before carrying out lateral analysis of the rotor.

I.1.2 b) (I.1.2 b)) added – At the end of b) paragraph NOTE: Though the higher order damped natural frequencies can be close to impeller vane passing frequency, there is no experience in liquid-handling turbomachines pointing to rotor-dynamics problems caused by such proximity. This is deemed a consequence of the complex mode shape(s), relatively low excitation energy and sufficient damping at the higher frequencies involved.

I.1.3 (I.1.3) changed – Critical Dampening conditions:

	11th ed.	10th ed.
$\xi \geq$	0,15	0.2
$\delta \geq$	0,95	1.2
$Fa \leq$	3.33	2.5

Note: Dampening factors changes as indicated.

added – NOTE 1: The values given for critically damped conditions in liquid-handling turbomachines differ from those in API standards for gas- or vapor-handling turbomachines. The difference reflects successful operating experience with liquid-handling turbomachines designed using the values in this annex.

added – NOTE 2: Damping of $\xi \geq 0,08$ over the range f_{n1} / f_{run} from 0,8 to 0,4 is supported by design and operating experience with liquid-handling turbomachines, which demonstrates that designs satisfying this requirement have not suffered problems with subsynchronous rotor vibration.

Annex K Shaft stiffness and bearing system life

Note: Extensive additions are covered in the above first portion of this paper

Annex N Pump datasheets and electronic data exchange

Note: Extensive changes are covered in the above first portion of this paper

UPCOMING ISSUES FOR NEXT EDITION

The Taskforce Chair / Working Group Convenor is currently preparing a survey to identify issues that need to be addressed. If there are a sufficient number of issues, a “new work item proposal” will be submitted to API and ISO asking for

authorization to re-activate the Working Group. At this point, the following issues have been identified to address: Seal Guarding, High Energy Pumps, Annex H: Composite Wear Parts/ Material Codes and Baseplates/ Nozzle Loads.

CONCLUSIONS

This paper has presented the seven significant changes/additions and a series of other changes that are of interest to the reader in a comparative narrative of ISO 13709 / API 610 11th Edition to the 10th Edition. To truly understand all the requirements of ISO 13709/ API 610 11th Edition, the authors of this paper recommend readers purchase the latest document along with the data sheet program from API.

All users of this document are encouraged to submit technical inquiries, suggestions and even corrections to the Standards Department, API,
1220 L Street, NW,
Washington, DC 20005
or send email to standards@api.org.

The API-610 Taskforce/ ISO 13709 Working Group will be meeting on an annual basis to consider all received technical inquiries. Depending on the nature of the changes, agreed upon modifications to the standard may then be issued as an addendum.

On the regular review cycle for the standard, these addenda will be considered and compiled along with any other changes deemed necessary to update the standard, and these changes will comprise the next standard publication draft. This draft will then be submitted to the balloting process for approval to publish the next edition of the standard. By following this annual process, the standard will be able to consider improvements more responsively, and the review process which now occurs approximately every 5 years will be supplemented with an annual event. It is envisioned that this process will simplify and expedite the lengthy review process currently required to update and issue new editions of the standard.

API documents require the effort of several individuals with a serious commitment to further refine this standard. Those with this level of commitment are welcomed to provide their expertise to the Taskforce/Working Group.

REFERENCES

- American Petroleum Institute, ANSI/API Standard 610 Eleven Edition, September 2010 (ISO 13709:2009 (Identical), Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries
Korkowski, F., Jones, R.L. and Sanders, J.D. December 2010 *What is new in API 610 11th Ed. (ISO 13709 2nd Ed.)?* Hydrocarbon Processing

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