

Special-purpose Couplings for Petroleum, Chemical, and Gas Industry Services

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Introduction

Users of this standard should be aware that further or differing requirements may be needed for individual applications. This standard is not intended to inhibit a vendor from offering, or the purchaser from accepting alternative equipment or engineering solutions for the individual application. This may be particularly appropriate where there is innovative or developing technology. Where an alternative is offered, the vendor should identify any variations from this standard and provide details.

This standard requires the purchaser to specify certain details and features.

A bullet (●) at the beginning of a paragraph indicates that either a decision is required or further information is to be provided by the purchaser. This information should be indicated on the datasheet(s), typical examples of which are included as [Annex A](#); otherwise it should be stated in the quotation request or in the order.

Special-purpose Couplings for Petroleum, Chemical, and Gas Industry Services

1 Scope

This standard specifies the requirements for couplings for the transmission of power between the rotating shafts of two machines in special-purpose applications in the petroleum, petrochemical, and natural gas industries. Such applications are typically in large and/or high-speed machines, in services that can be required to operate continuously for extended periods, are often unspared and are critical to the continued operation of the installation. By agreement, it can be used for other applications or services.

Couplings covered by this standard are designed to accommodate parallel (or lateral) offset, angular misalignment, and axial displacement of the shafts without imposing unacceptable mechanical loading on the coupled machines. It is applicable to gear, metallic flexible element, quill-shaft and torsionally resilient type couplings. Torsional damping and resilient type couplings are detailed in [Annex B](#), gear-type couplings are detailed in [Annex C](#), and quill-shaft type couplings are detailed in [Annex D](#).

This standard covers the design, materials of construction, manufacturing quality, inspection, and testing of special-purpose couplings.

This standard does not define criteria for the selection of coupling types for specific applications.

This standard is not applicable to other types of couplings, such as clutch, hydraulic, eddy-current, rigid, radial spline, chain, and bellows types.

2 Normative References

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ANSI/AGMA 9000,¹ *Flexible Couplings—Potential Unbalance Classification*

ANSI/AGMA 9002-C14, *Bores and Keyways for Flexible Couplings (Inch Series)*

ANSI/AGMA 9003-B08, *Flexible Couplings—Keyless Fits (Inch Series)*

ANSI/AGMA 9004-B08, *Flexible Couplings—Mass Elastic Properties and other Characteristics (Inch Series)*

ANSI/AGMA 9103-B08, *Flexible Couplings—Keyless Fits (Metric Series)*

ANSI/AGMA 9104-A06, *Flexible Couplings—Mass Elastic Properties and other Characteristics (Metric Series)*

ANSI/AGMA 9112-B15, *Bores and Keyways for Flexible Couplings (Metric Series)*

ANSI/ASME B1.1,² *Unified inch screw threads, UN and UNR thread form*

ANSI B11.19-2010, *Performance Requirements for Safeguarding*

BS EN ISO 80079-36:2016, *Explosive atmospheres. Non-electrical equipment for explosive atmospheres. Basic method and requirements*

¹ American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314-1560, www.agma.org.

² ASME International, Three Park Avenue, New York, New York 10016-5990, www.asme.org.

ISO 262, *ISO general-purpose metric screw threads—Selected sizes for screws, bolts and nuts*

ISO 286-2, *Geometrical product specifications (GPS)—ISO code system for tolerances on linear sizes—Part 2: Tables of standard tolerance classes and limit deviations for holes and shafts*

3 Terms, Definitions, Acronyms, and Abbreviations

3.1 Terms and Definitions

For the purposes of this document, the following terms and definitions apply.

3.1.1

AFD

adjustable frequency drive

A device that allows speed adjustment accomplished by electrical frequency variation in an AC motor.

3.1.2

angular misalignment

(double-engagement couplings)

Two minor angles between the extension of each machine centerline and the centerline of the structure joining the two flexible elements.

3.1.3

angular misalignment

(single-engagement couplings)

Minor angle between the extensions of two machine-shaft centerlines.

NOTE If the shaft centerlines do not intersect, a single-engagement coupling is not appropriate.

3.1.4

assembly balance

Procedure in which a completely assembled coupling is balanced as a unit.

3.1.5

assembly balance check

Procedure in which an assembled coupling is placed on a balancing machine and the residual unbalance is measured.

NOTE An assembly balance check can be performed on a component balanced coupling, or on an assembly-balanced coupling.

3.1.6

axial displacement

Change in the relative axial position of the adjacent shaft ends of two coupled machines, usually caused by thermal expansion.

3.1.7

component balance

Procedure for achieving coupling balance in which the components or factory assembled subassemblies are balanced separately before assembly of the coupling.

3.1.8

continuous torque rating

Coupling manufacturer's declared maximum steady state torque that the coupling is capable of transmitting continuously for unlimited periods.

3.1.9

coupling rated speed

Highest rotational speed at which the coupling is required to be capable of transmitting the continuous torque rating while simultaneously being subjected to the rated angular misalignment and the coupling rated axial displacement.

3.1.10

crown diameter

Major diameter of the external teeth of a gear-type coupling.

3.1.11

cyclic torque

Cyclic (fluctuating or oscillating) torques that occur from various sources, such as AFD (adjustable frequency drives) or synchronous motors.

3.1.12

cyclic torque rating

The cyclic torque value for a specified number of cycles (see [7.4.3](#) specified by purchaser) for the weakest component in the system.

3.1.13

diaphragm coupling

Coupling that transmits torque radially from the outside diameter of a driving flexible plate or plates to the inside diameter, across the spacer piece, and then from inside to outside diameter of a driven flexible plate or plates. Misalignment is accommodated by the flexing of the plate(s).

3.1.14

disc coupling

Coupling that transmits torque from a driving to a driven bolt tangentially on a common bolt circle. Torque is transmitted between the bolts through a series of thin discs. Misalignment is accommodated by the flexing of the discs.

3.1.15

distance between shaft ends

DBSE

Distance from the extreme end of one shaft (including any threaded end) to the extreme end of the next shaft or, in the case of integral flanges, the distance from the mating faces.

3.1.16

DN

diamètre nominal

Alphanumeric designation of size for components of a pipework system.

3.1.17

double engagement coupling

Coupling with two planes of flexure.

NOTE This arrangement enables couplings of certain types, notably gear and metallic flexible element types, that cannot normally accommodate parallel (or lateral) offset, to do so.

3.1.18

dummy hub

A hub that is used to simulate the moment of mass of a half coupling during rotating equipment testing without the use of the job hub.

3.1.19**factor of safety**

Factor that is used to cover uncertainties in a coupling design (e.g., analytical assumptions such as stress analysis, material properties, manufacturing tolerances).

NOTE Under given design conditions, the factor of safety is the material yield strength divided by the calculated stress, where the stress is a function of torque, speed, misalignment, and axial displacement.

3.1.20**fatigue factor of safety**

Factor of safety at the published continuous rated conditions of torque, speed, misalignment, and axial displacement, used by coupling manufacturers to establish the coupling rating (see [7.2](#)).

NOTE The fatigue factor of safety is further explained and defined in [Annex E](#).

3.1.21**flex-hub coupling**

Gear-type coupling with the external teeth on the hubs and the internal teeth in the sleeves.

3.1.22**gear coupling**

Coupling of the mechanical contact type that transmits torque and accommodates angular misalignment, parallel offset, and axial displacement by relative rocking and sliding motion between mating, profiled gear teeth.

3.1.23**half coupling**

Composite of all of the components of the coupling attached to, and supported from, one shaft, including an appropriate portion of the spacer assembly in the case of a double-engagement coupling or of the flexing elements of a single-engagement coupling.

3.1.24**idling adapter (solo plate)**

Device designed to rigidly hold in alignment the floating parts of certain types of couplings to allow uncoupled operation of the driving machine without dismounting the coupling hub.

3.1.25**lateral offset**

Lateral distance between the centerlines of two shafts that are not parallel, measured perpendicularly to the centerline and in the plane of the shaft end of the driving machine [See [Annex F](#)].

3.1.26**manufacturer**

Agency responsible for the design and fabrication of the coupling

NOTE The manufacturer is not necessarily the vendor.

3.1.27**maximum allowable temperature**

Maximum continuous temperature for which the manufacturer has designed the coupling at the continuous load condition.

3.1.28**maximum continuous angular misalignment**

Maximum angular misalignment at each plane of flexure that the coupling is able to tolerate for unlimited periods.

NOTE Maximum continuous angular misalignment can be expressed as either:

- a) a single value if transmitting the coupling continuous torque rating at the coupling rated speed, and simultaneously subjected to the coupling maximum continuous axial displacement, or
- b) a range of values expressed as an interrelated function of speed, torque, and axial displacement.

3.1.29

maximum continuous axial displacement

Maximum axial displacement the coupling is able to tolerate for unlimited periods.

NOTE Maximum continuous axial displacement can be expressed as either:

- a) a single value if transmitting the coupling continuous torque rating at the coupling rated speed and simultaneously subjected to the coupling maximum continuous angular misalignment, or
- b) a range of values expressed as an interrelated function of speed, torque, and angular misalignment.

3.1.30

maximum continuous speed

Highest rotational speed at which the machine, as made and tested, is capable of continuous operation.

3.1.31

maximum driver power

The maximum power the driver can provide (without hardware modifications) at rated (100 %) speed.

- a) For an electric motor, it is the motor's nameplate rating plus any motor service factor.
- b) For a gas turbine, it is when the gas turbine is operated at maximum allowable firing temperature, rated speed, ambient site conditions, or other conditions as defined by the manufacturer.
- c) For a steam turbine, it is when the steam turbine is operated at rated operating conditions, rated speed, or under other conditions as defined by the manufacturer.

3.1.32

metallic flexible-element coupling

Coupling type that obtains its flexibility from the flexing of thin metallic discs, diaphragms, or links.

3.1.33

minimum allowable speed

Lowest speed (revolutions per minute) at which the machine, as made and tested, is capable of continuous operation.

3.1.34

moment simulator

Auxiliary device intended to simulate the moment of the mass of a half coupling.

3.1.35

momentary torque rating

Torque that corresponds to a factor of safety of 1.0 with respect to the most highly stressed component's material yield strength, allowing for a combination of speed, angular misalignment, and axial displacement.

3.1.36

NPS

nominal pipe size

Value approximately equal to a diameter in inches (e.g., NPS 3/4).

NOTE 1 Refer to ASME B 31.3

NOTE 2 The letters NPS are followed by a value that is related to an approximate diameter of the bore, in inches, for piping up to and including 12 in. diameter. For piping over 12 in. (NPS 12), the NPS value is the nominal OD.

3.1.37

normal operating point

Point at which (normal) operation is expected with optimum efficiency. This is usually the operating point at which the vendor certifies that performance is within the tolerances stated in this standard.

3.1.38

owner

Final recipient of the equipment, who may delegate another agent as the purchaser of the equipment.

3.1.39

parallel offset

Distance between the centerlines of two coupled shafts that are parallel but not in the same straight line [See [Annex F](#)].

3.1.40

peak torque rating

Torque that corresponds to a factor of safety of 1.15 with respect to the most highly stressed component's yield strength, allowing for a combination of speed, angular misalignment, and axial displacement, considering all transient and momentary events.

3.1.41

pilot

rabbet

register

Surface that positions a coupling component, subassembly, or assembly radially with respect to another coupling component.

3.1.42

potential unbalance

Probable net unbalance of a complete coupling.

NOTE 1 Potential unbalance results from a combination of the residual unbalance of individual components and subassemblies and possible eccentricity of the components and subassemblies due to run-out and tolerances of the various surfaces and registers. Since it can be assumed that the actual values of the various contributory unbalances are random in both magnitude and direction, the numerical value of the potential unbalance is the square root of the sum of the squares of all the contributory unbalances. Typical contributory unbalances are:

- a) the residual unbalance of each component or subassembly;
- b) errors in the balance of each component or subassembly resulting from eccentricity in the fixture used to mount the component or subassembly in the balance machine;
- c) the unbalance of each component or subassembly due to eccentricity resulting from clearance or run-out of the relevant registers or fits.

NOTE 2 The concept of potential unbalance is explained more fully, and a worked example is provided in [Annex G](#).

3.1.43

purchaser

Agency that issues the order and purchase specification to the vendor.

NOTE The purchaser can be the owner of the plant in which the equipment is to be installed, the owner's appointed agent, or, frequently, the manufacturer of the driven equipment.

3.1.44**quill-shaft coupling**

Coupling that is both laterally and torsionally flexible, with angular misalignment, parallel offset, and torsional fluctuations being accommodated by elastic deformation of a relatively long, slender shaft.

NOTE Quill-shaft couplings, unless combined with another type, cannot accommodate axial displacement.

3.1.45**residual unbalance**

Level of unbalance remaining in a component or assembly after it has been balanced, either to the limit of the capability of the balancing machine or in accordance with this standard (see [Annex K](#)).

3.1.46**roughness magnitude****R_a**

Arithmetic average of the absolute value of the profile height deviations recorded within the evaluation length and measured from the mean line.

NOTE 1 Adapted from ASME B 46.1–2009, paragraphs 1–4.1.1.

NOTE 2 It is the average height of the entire surface, within the sampling length, from the mean line.

3.1.47**service factor**

Factor applied to allow for such conditions as off-design conditions, cyclic and other variations, and equipment variations.

NOTE Service factor is not the same as the factor of safety (3.1.19) or the fatigue factor of safety (3.1.20).

3.1.48**single-engagement coupling**

Coupling with only one plane of flexure.

NOTE This type of coupling can accommodate angular misalignment and axial displacement. Single-engagement couplings of some types, notably gear and metallic flexible element types, do not normally accommodate parallel (or lateral) offset. Certain types of single-engagement couplings (not covered by this standard) can accommodate offset misalignment to a limited extent.

3.1.49**spacer**

Part of a coupling that is removable to give access for maintenance and/or removal of the coupling hubs.

NOTE The spacer can be a single component or an assembly.

3.1.50**spacer gap length**

Distance between coupling hubs or sleeves in which the coupling spacer is installed.

NOTE Spacer gap length is not necessarily equal to the distance between the shaft ends.

3.1.51**spark resistant**

Material that is not prone to generating impact sparks under conditions of use.

3.1.52**torsional damping**

Absorption or dissipation of cyclic (oscillatory) rotary energy.

NOTE Torsional damping is necessary in some cases to limit the buildup of steady-state torsional resonant cycles (oscillations) in a system.

3.1.53

torsional natural frequency

Frequency of the undamped, free-rotational vibration of a system composed of revolving mass inertias acting in combination with the restraining torsional rigidities of the connected shafts and couplings.

3.1.54

torsional stiffness

Ratio of the applied torque to the resulting torsional displacement of either a complete coupling or part of the coupling, such as a spacer.

NOTE With some types of couplings, the torsional stiffness is not constant, but is a function of the magnitude of the torque, axial positioning, and, with cyclic torques, also the frequency.

3.1.55

torsionally resilient coupling

Coupling with increased flexibility in a rotational direction, increased capability to recover from flexing, and hysteresis capability.

NOTE Resilience is the ability to recover from deformation under repeated flexing, taking account of energy storage and hysteresis effects. Some types of torsionally resilient couplings can also be designed to accommodate misalignment and/or axial displacement.

3.1.56

total indicator reading

total indicated runout

TIR

Difference between the maximum and minimum readings of a dial indicator or similar device, monitoring a face or cylindrical surface during one complete revolution of the monitored surface

NOTE For a cylindrical surface, the indicated run-out implies an eccentricity equal to half the reading. For a flat face, the indicated run-out implies an out-of-squareness equal to the reading. If the diameter in question is not cylindrical or flat, the interpretation of the meaning of TIR is more complex and may represent ovality or surface irregularities.

3.1.57

trip

Automated shutdown to ensure personnel safety (safety critical).

3.1.58

unit responsibility

Obligation for coordinating the documentation, delivery, and technical aspects of all the equipment and all auxiliary systems included in the scope of the order.

NOTE The technical aspects to be considered include, but are not limited to, such factors as the power requirements, speed, rotation, general arrangement, dynamics, noise, lubrication, sealing system, material test reports, instrumentation, piping, conformance to specifications, and testing of components.

3.1.59

vendor

Manufacturer or manufacturer's agent that supplies the equipment.

3.2 Acronyms and Abbreviations

For the purposes of this document, the following acronyms and abbreviations apply.

ANF	axial natural frequency
-----	-------------------------

ATEX	atmosphere explosibles
CMM	coordinate measuring machine
CNC	coordinate measuring controlled
DBSE	distance between shaft ends
F_s	service factor
HRC	Rockwell hardness measured on the C scale
N_c	lateral natural frequency, rpm
N_x	speed as indicated by x, rpm
U	residual unbalance, g-mm (oz-in.)

4 Requirements

4.1 • Units of Measure

The purchaser's use of an SI data sheet (see [Annex A.1](#)) indicates that the SI system of measurements shall be used for all data, drawings, and maintenance dimensions. The purchaser's use of a USC data sheet (see [Annex A.2](#)) indicates the USC system of measurements shall be used for all data, drawings, and maintenance dimensions.

4.2 Statutory Requirements

The purchaser and the vendor shall determine the measures to be taken to comply with any governmental codes, regulations, ordinances, directives, or rules that are applicable to the equipment, its packaging, and any preservations used.

5 Coupling Selection

5.1 • The purchaser shall specify the type of coupling required. Unless otherwise specified, the coupling shall be a metallic flexible-element coupling. For torsional damping and resilient couplings, refer to [Annex B](#); for gear couplings, refer to [Annex C](#); and for quill-shaft couplings, refer to [Annex D](#). [Figure 1](#) provides guidance for the typical selection process for a coupling.

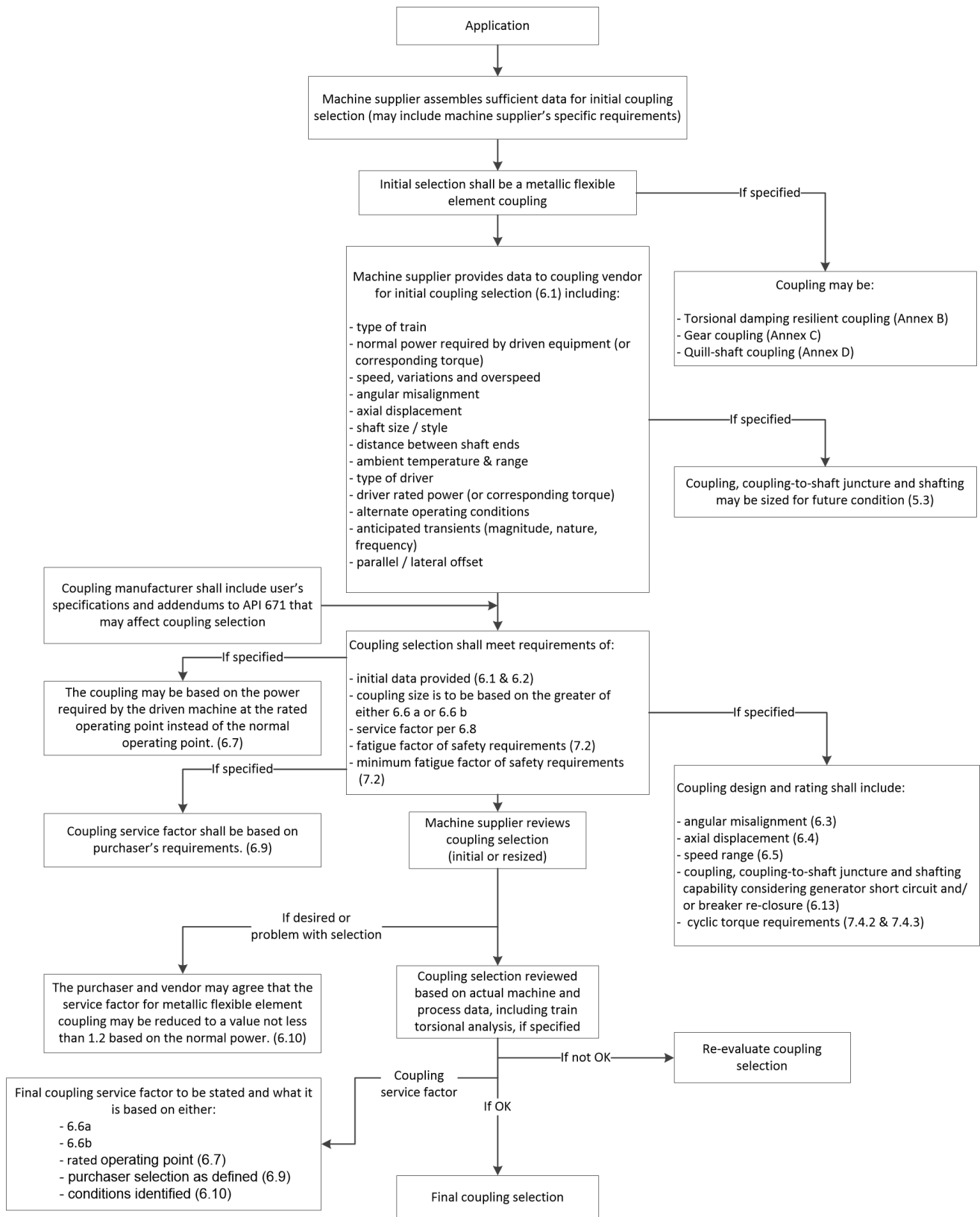


Figure 1—Typical Coupling Selection Process

5.2 The coupling shall be selected based on the equipment loading and shall be capable of simultaneously transmitting the steady-state torques, cyclic torques, and maximum transient torques under all conditions of

angular misalignment, axial displacement, speed, and temperature, to which it will be subjected in service as identified in [Section 6](#).

5.2.1 The purchaser shall specify the period of uninterrupted continuous operation. Shutting down the equipment to perform planned maintenance or inspection during the uninterrupted operation is not acceptable.

NOTE 1 It is recognized that there are some services where this objective is easily attainable and others where it is difficult.

NOTE 2 Auxiliary system design and design of the process in which the equipment is installed are very important in meeting this objective.

5.2.2 The vendor shall advise in the proposal any component or system designed for a finite life and identify any component or maintenance requirement that would result in the need to shut down the equipment within the uninterrupted operational period.

5.3 • If specified, the coupling and coupling-to-shaft juncture may be sized for a future condition.

5.4 • For trains with multiple couplings, the sizing of each coupling shall consider load profiles/sharing for different conditions, as specified.

6 Coupling Design

6.1 The coupling shall be designed based on the equipment loading and shall be capable of simultaneously transmitting the steady-state torque, cyclic torques, and the maximum transient torques under all conditions of angular misalignment, axial displacement, speed, and temperature, to which it will be subjected in service as identified in [6.2](#) through [6.16](#).

6.2 • The purchaser shall specify the following requirements, where applicable:

- a) type of train;
- b) normal power;
- c) normal speed;
- d) rated speed of driver or driven equipment;
- e) minimum operating speed;
- f) maximum continuous speed;
- g) any defined overspeed (including trip);
- h) angular misalignment;
- i) axial displacements;
- j) shaft sizes and styles;
- k) distance between shaft ends;
- l) ambient temperature (maximum and minimum);
- m) type of driver;
- n) maximum driver power;

- o) driver power rating (list and driver service factor);
- p) expected transient and cyclic torque conditions, including magnitude, nature, and number of occurrences of transients to which the coupling will be subjected in service;

NOTE This information is used to establish the required endurance strength or fatigue strength of the coupling components. The coupling design depends on the number of load cycles, allowing for a combination of speed, angular misalignment, and axial displacement considering all transient and momentary events that can lead to high torque-amplitudes, such as startup, short circuit, or auto reclosing, considering the related number of load cycles.

- q) data for the coupling guard to allow windage calculations;
- r) information for the torque measuring system identified in [11.3](#) (if a torque measuring system is provided);
- s) oil properties available for gear coupling applications.

6.3 • The purchaser shall specify the maximum angular misalignments the coupling is expected to experience during startup, normal operation, and shut-down of the coupled machines, normally expressed as parallel (or lateral) offset and/or angular misalignment between the coupled shafts. These values shall allow for all the known effects on the machines from thermal, pressure, and dynamic forces. Unless otherwise specified, the steady-state angular misalignment capability across each flexible element shall be not less than 0.2°.

6.4 • The purchaser shall specify the maximum axial displacements the coupling is expected to experience, expressed as the amount and direction of the relative movement of the shaft ends toward or away from each other as the coupled machines go through their startup, normal operation, and shut-down cycle. These dimensions shall be given from the machine at ambient conditions and while it is in a non-operating position. Unless otherwise specified, the minimum steady-state axial deflection (plus/minus) capability shall be determined by the largest shaft diameter divided by 125.

6.5 • The purchaser shall specify the speed range (minimum speed to maximum continuous speed) and trip speed for the fully assembled coupling, moment simulator, and solo plate for both continuous speed and any defined overspeed.

6.6 The steady-state selection torque, T_s , used to select the coupling shall be the greater of [6.6 a\)](#) or [6.6 b\)](#).

- a) The steady-state selection torque, T_s , expressed in newton-meters (inch-pounds force), shall be determined based on the driven equipment demand using Equation (1). The coupling service factor, F_s , shall be based on the type of the coupling.

$$T_s = T_n * F_s \quad (1)$$

Where:

T_n is the steady-state torque expressed in newton-meters (inch-pounds force), and shall be determined using Equation (2);

F_s is the coupling service factor specified in [6.8](#).

$$T_n = (K_1 * P_{normal}) / N_{normal} \quad (2)$$

Where:

K_1 is a constant, equal to 9550 (63,000);

P_{normal} is the input power required by the driven machine, expressed in kilowatts (horsepower), at the specified normal operating point;

N_{normal} is the speed corresponding to the normal operating point's power, expressed in revolutions per minute.

- b) For metallic flexible-element couplings, the steady-state selection torque, T_s , expressed in newton-meters (inch-pounds force), shall be determined based on the maximum driver torque at maximum driver power using Equation (3). This coupling service factor, F_s , is considered to be a constant for this method.

The same methodology can be used for other types of couplings using an agreed-upon service factor.

NOTE 1 Adjustable frequency drive (AFD) motor drivers provide various torque curves; therefore, the maximum torque likely will not occur at the rated speed. This is also true with some gas turbine drivers with a flat torque curve.

NOTE 2 A coupling selection method is considered as some users install drivers much higher in capable power than the train requires, such as common drivers. This selection method results in couplings that are oversized for the application. This selection method proportionally splits the oversizing of the couplings for each of the driven machine(s).

$$T_s = T_d * F_s \quad (3)$$

Where:

T_d is the torque at maximum driver power, expressed in newton-meters (inch-pounds force), and shall be determined using Equation (4);

F_s is the coupling service factor, equal to 1.2.

$$T_d = (K_1 * P_{driver}) / N_{driver} \quad (4)$$

For adjustable frequency drivers, T_d is to be the maximum torque capability of the driver.

Where:

K_1 is a constant, equal to 9550 (63,000);

P_{driver} for a coupling connected directly to a driver (or gearbox and driver), P_{driver} is the maximum power of the driver as described in 3.1.31, expressed in kilowatts (horsepower); for couplings between multiple items of driven equipment, P_{driver} shall be the maximum power of the driver as described in 3.1.31, expressed in kilowatts (horsepower), minus the power of the driven equipment based on T_n (the additional power above each machine's T_n is to be proportionally split for each driven machine);

N_{driver} is the speed corresponding to the 100 % rotational speed, expressed in revolutions per minute.

6.7 • If specified, the steady-state selection torque, T_s , expressed in newton-meters (inch-pounds force) shall be determined based on the torque required by the driven machine at the rated operating point and the corresponding speed rather than at the normal operating point (per 6.6a) using Equation (5). This can be appropriate if the power required by the driven machine at the rated operating point is significantly higher than that at the normal operating point.

$$T_s = T_r * F_s \quad (5)$$

Where:

T_r is the torque required by the driven machine at the rated operating point, expressed in newton-meters (inch-pounds force), and shall be determined using Equation (6);

F_s is the coupling service factor specified in 6.8.

$$T_r = (K_1 * P_{rated}) / N_{rated} \quad (6)$$

Where:

K_1 is a constant, equal to 9550 (63,000);

- P_{rated} is the power required by the driven machine, expressed in kilowatts (horsepower), at the rated operating point;
- N_{rated} is the speed corresponding to the rated operating point's power, expressed in revolutions per minute.

NOTE 1 This situation can occur, for example, in the case of a centrifugal compressor with several different specified operating duties.

NOTE 2 Basing the coupling continuous torque rating on the driven-machine rated operating point rather than the normal operating point can result in a coupling that is undesirably large or heavy. This can be significant if one or both of the connected machines is sensitive to overhung mass.

6.8 Unless otherwise specified in [6.9](#) or [6.10](#), the minimum coupling service factor shall be 1.5 for a metallic flexible-element coupling; 3.0 for torsional damping and resilient couplings (see [Annex B](#)); 1.75 for gear couplings (see [Annex C](#)); and 1.5 for quill-shaft couplings (see [Annex D](#)).

NOTE At the proposal stage for new equipment, the final drive characteristics are often unknown. These factors are normally satisfactory to cover unknowns so that coupling sizes do not increase during engineering.

6.9 • If specified, the purchaser may select a service factor value and state the basis of the service factor.

6.10 The coupling service factor, F_s , may be reduced by agreement between the purchaser and the vendor if:

- the characteristics of the driver and driven machines and the operating process are well understood, or
- all reasonable attempts to achieve the specified steady-state torque service factor fail to result in a coupling mass and subsequent overhung moment commensurate with the requirement for rotor dynamics of the connected machines.

In no case shall the service factor, F_s , for metallic flexible-element couplings be reduced to a value lower than 1.2 based on the input power required by the driven machine at the specified normal operating point.

6.11 If the machine train is driven by an induction motor, the coupling, coupling-to-shaft juncture, and machinery shafting shall be capable of transmitting 115 % of the expected transient torque encountered during a startup without damage.

6.12 If the machine train is driven by a synchronous motor, the coupling and coupling-to-shaft juncture shall be capable of withstanding the cyclic torque associated with startup for the expected number of starts specified by the purchaser. A limited life fatigue stress analysis shall be performed to verify this.

NOTE For initial coupling selection, a large cyclic torque requirement is typically assumed until all conditions are known so that the torsional response analysis can be completed.

6.13 • If specified, the coupling and coupling-to-shaft juncture shall be capable of transmitting a specified transient torque associated with a generator short circuit and/or a motor breaker re-closure without damage.

NOTE During these situations, large torques can occur, which require that the total system be subsequently examined for possible damage.

6.14 The coupling-to-shaft juncture shall be capable of operating at a steady-state torque determined in accordance with 6.6a, substituting a service factor of 1.75. Torsional damped applications shall have a coupling-to-shaft juncture service factor of 3.0.

6.15 The torque capacity of a coupling-to-shaft juncture that is hydraulically fitted and keyless shall meet the following unless otherwise agreed:

- methods and equations in ANSI/AGMA 9003-B08 or ANSI/AGMA 9103-B08. Other methods of calculation may be used with the approval of the purchaser;

- b) coefficient of friction of 0.15;
- c) assumed length of hub engagement shall not include O-ring and oil-distribution grooves in the hub or the shaft.

6.16 Couplings may be designed to transmit torque through flange face friction. Unless otherwise agreed, a value of 0.15 shall be assumed for the coefficient of friction. However, the fasteners for coupling flanges, including integral flanges, shall not be less than that required to transmit the specified torques, as identified in 6.11 through 6.13 in shear.

6.17 Coupling shall provide drain passages in components, which could potentially trap and accumulate liquids. A minimum of two drain passages shall be provided in each affected component.

6.18 Spare parts shall meet all the criteria of this standard.

7 Coupling Ratings

7.1 The vendor shall state the continuous torque rating at the coupling rated speed while simultaneously subjected to the maximum continuous angular misalignment and the maximum continuous axial displacement.

7.2 The fatigue factor of safety at the continuous torque rating and any published combination of speed, angular misalignment, and axial displacement shall be determined using the proportional increase method with either the modified Goodman diagram or constant-life curves (see [Annex E](#)), together with the mean and cyclic stresses induced under the evaluated conditions. If the modified Goodman diagram is used, the fatigue factor of safety shall not be less than 1.25 for the flexible element and 1.5 for all other coupling components. If the constant-life curve is used, the fatigue factor of safety shall be not less than 1.35 for the flexible element and 1.5 for all other coupling components. Regardless of the method used, data for material strength shall be drawn from published industry standards or test data.

NOTE This subsection defines (for the coupling manufacturer) the minimum fatigue factors of safety and the methods for applying them to recognized material properties. This definition standardizes the basis for continuous coupling ratings. Details of the design, such as equations and analysis used to derive the stresses, are often considered proprietary and are not a point of documentation.

7.3 The vendor shall state the momentary and peak torque ratings of the coupling.

NOTE Typically, if the momentary or peak torque rating is exceeded, damage may occur to various components of the coupling, such as flex elements, fasteners, and shaft juncture.

7.4 For applications where cyclic torques can occur (such as from synchronous motor, generator, or reciprocating compressor), the coupling design shall be determined by completing a fatigue analysis and plotted on a fatigue diagram. (see [Annex E](#)).

7.4.1 The fatigue analysis shall take into account the normal operating conditions in conjunction with the transient conditions (see [6.1](#)). Depending on whether the cyclic loads are considered for a limited number of occurrences or considered to be infinite, either a low-cycle fatigue analysis or high-cycle fatigue analysis shall be performed for the various torque-transmitting components in the coupling. For cyclic fatigue analysis, the resultant means and cyclic stresses plotted on a fatigue diagram shall fall under the applicable fatigue diagram based on a specific number of cycles, such as 10^4 , 10^5 , 10^6 , or 10^7 cycles.

7.4.2 • The fatigue diagram used shall be based on the purchaser specifying the number of occurrences of transients to which the coupling will be subjected in service. For high-cycle fatigue analysis, the analysis and safety factors shall be as specified in [7.2](#). Transient applications do not require the application of a service factor. The fatigue diagram should show the steady state (non-cyclic) stress plotted against the alternating stress generated at the coupling rated speed while simultaneously subjected to the maximum continuous angular misalignment and the maximum continuous axial displacement.

7.4.3 • If specified, the cyclic torque rating for the number of cycles specified (as a default 10^7 cycles), based on the combination of normal operating torque, maximum angular misalignments, maximum axial displacement, and maximum continuous speed, shall be provided on the coupling drawing.

7.4.4 In applications where potentially damaging, infrequent, maximum momentary or fault torques occur, the provision of an overload device may be considered to protect the coupling and equipment.

7.4.4.1 The overload device shall be designed to prevent the overload device from prematurely tripping due to low- or high-cycle fatigue of any sacrificial elements.

7.4.4.2 The overload device shall be designed to ensure that the device remains relatively intact after a trip until the connected equipment is brought to a stop.

8 Coupling Requirements

8.1 Metallic Flexible-element Couplings

8.1.1 The requirements of [8.1](#) apply only to metallic flexible-element couplings. For torsionally resilient couplings, gear couplings, and quill-shaft couplings, the requirements of [Annex B](#), [Annex C](#), or [Annex D](#), respectively, shall apply.

8.1.2 Flexible elements shall be metallic.

8.1.3 If the flexible elements of a coupling are combined in a factory-assembled pack, the coupling spacer shall be removable without disturbance to the factory assembly of the elements.

8.1.4 If a tapered bore or integral flange is specified for one or both ends of the coupling, the vendor shall supply spacer shims to adjust the spacer gap. The shims shall provide a range of adjustment of ± 1.6 mm ($\frac{1}{16}$ in.), i.e., 3.2 mm ($\frac{1}{8}$ in.) total, for shafts with a nominal diameter less than 102 mm (4 in.). For shafts with a nominal diameter 102 mm (4 in.) or greater, the shims shall provide a range of ± 3.2 mm ($\frac{1}{8}$ in.), i.e., 6.4 mm ($\frac{1}{4}$ in.) total. For reduced moment style couplings for shafts with a nominal diameter less than 102 mm (4 in.), shims shall provide a range of adjustment of ± 0.8 mm ($\frac{1}{32}$ in.), i.e., 1.6 mm ($\frac{1}{16}$ in.) total.

8.1.5 The design of metallic flexible-element couplings shall be such that in the event of complete failure of the flexible element or elements in one plane of flexure, the spacer assembly is retained in approximately its normal position.

NOTE Once the coupling's flexible element(s) has failed, it is expected that the vibration will increase, and continual operation of the coupling is not possible.

8.1.6 • If specified, couplings shall incorporate a feature that allows the transmission of load for a limited period in the event of a complete flexible element failure to allow safe shutdown of equipment.

NOTE For example, this feature can be considered for turbine main-drive couplings to provide an additional safety factor to prevent turbine overspeed in the event of coupling failure.

8.1.6.1 The purchaser should provide the torque load requirements, the time period for operation, and any other features or requirements.

8.1.6.2 The vendor shall provide the following information on this device:

- a) type of feature;
- b) how the feature functions;
- c) risks associated with the operation of this feature;

- d) details of the system requirements if an oil spray system is required to provide cooling to accommodate operating with the backup system engaged.

8.2 Machining

All coupling parts, other than fasteners and flexible discs, shall be machined all over to minimize inherent unbalance. All exposed surfaces shall be finished to a surface roughness not exceeding $3.2\text{ }\mu\text{m}$ (125 microinches) arithmetic average (R_a). Chamfers shall be provided for interference fits. Fillet radius shall be applied to avoid high stress concentration.

8.3 Spacer

All couplings shall be of the spacer type. The spacer shall be of sufficient length to allow removal of coupling hubs and to allow for maintenance such as adjacent bearings and seals without removal of the shaft or disturbance of the equipment alignment. The minimum spacer length shall correspond to a distance between shaft ends of 460 mm (18 in.).

NOTE The spacer gap length is not necessarily the same as the spacer length.

8.4 Hub Type

- The purchaser shall specify whether the coupling shall be designed to fit to integral flange(s) or be provided with removable hub(s).

8.5 Integral Flanges

8.5.1 If the coupling is to fit to integrally flanged shaft end(s), the coupling vendor and the manufacturer with unit responsibility shall agree on the flange geometry. The holes shall be machined with computer numerical-controlled (CNC) equipment. A drill fixture (or template) shall be used only if CNC equipment is not available. In either case, inspection shall be performed to assure proper location of the mating coupling flanges. The coupling vendor shall supply the drill fixture (or template), if required.

8.5.2 The coupling shall mate directly to the integral flange without an adapter. If an adapter is necessary, it shall require purchaser approval.

NOTE Addition of an adapter between the coupling and the integral flange increases the overhung moment and adds another fit whose runout and clearance is necessary to control to maintain balance.

8.6 Hubs

8.6.1 Removable Hubs

8.6.1.1 Removable coupling hubs shall be fitted to the shaft using an interference fit to positively locate and center the hub, and therefore the rest of the coupling, on the shaft.

8.6.1.2 • Hubs may have a parallel or tapered bore, with or without keyways, or may be splined with a major diameter fit, or splined with a side fit and fore and aft locating pilots, as specified by the purchaser.

8.6.1.3 • The degree of interference shall be specified by the purchaser and is subject to approval by the vendor.

8.6.1.4 The following guidelines are recommended for hub-to-shaft fits:

- a) The interference fit for straight-bore, keyed hubs should be from 0.00050 mm/mm (in./in.) to 0.00075 mm/mm (in./in.) of bore diameter.
- b) The interference fit for tapered-bore, keyed hubs should be at least 0.001 mm/mm (in./in.) of bore diameter.

- c) The interference rate for taper-bore, hydraulically fitted hubs should be based upon the torque that is required to be transmitted. As a maximum, the interference rate should not exceed 0.003 mm/mm (in./in.) of bore diameter to avoid potential problems.

8.6.1.5 • If specified for straight-bore, keyed hubs, the shaft sizes and coupling bores shall be in accordance with ANSI/AGMA 9002-C14, ANSI/AGMA 9112-B15, or ISO 286-2.

8.6.1.6 For tapered-bore, keyed hubs, the inspection procedures shall be in accordance with ANSI/AGMA 9002-C14 or ANSI/AGMA 9112-B15.

8.6.1.7 For tapered-bore, hydraulically fitted hubs, the inspection procedure shall be in accordance with ANSI/AGMA 9003-B08 or ANSI/AGMA 9103-B08.

8.6.1.8 The surface roughness, arithmetic average roughness (R_a), of hub bores shall not exceed:

- a) 3.2 μm (125 microinches) for straight-bore, keyed hubs,
- b) 1.6 μm (63 microinches) for tapered-bore, keyed hubs,
- c) 0.8 μm (32 microinches) for keyless hubs.

8.6.1.9 The out-of-roundness of the hub bore, whether straight or tapered, shall not exceed 5.1 μm (0.0002 in.) total indicator reading (TIR) for bores less than or equal to 102 mm (4 in.) in diameter, and shall not exceed 12.7 μm (0.0005 in.) TIR for hub bores greater than 102 mm (4 in.) in diameter. Roundness measurements shall be made before any keyways are cut.

8.6.2 Tapered-bore Hubs

8.6.2.1 • If a tapered-bore coupling is specified, the purchaser shall specify the taper (see [Annex I](#)).

8.6.2.2 Unless otherwise specified, keyless, hydraulically fitted hubs shall have a taper of 1:24 ($1/2$ in./ft of diameter).

8.6.2.3 Unless otherwise specified, non-hydraulically fitted hubs shall have a taper of 1:16 ($3/4$ in./ft of diameter).

8.6.2.4 • The taper of tapered-bore hubs shall be verified. The method of verification can either be using matched ring and plug gauges or by the coordinate measuring machine (CMM) technique. The verification method and acceptance criteria for the coupling hub-to-shaft end fit shall be determined by the purchaser.

8.6.2.4.1 The acceptance criteria for matched ring and plug gauge checking, using the matched ring and plug gauges specified in [11.2.5](#), is:

- a) Tapered bores for keyless, hydraulically fitted hubs shall have at least an 85 % blued fit (surface contact) to the taper gauge. See API RP 687 for bluing check information.
- b) Tapered bores for keyed hubs shall have at least a 70 % blued fit (surface contact) to the taper gauge before cutting keyways.

NOTE 1 ANSI/AGMA 9003-B08 or ANSI/AGMA 9103-B08 provides guidance for checking the contact.

NOTE 2 ANSI/AGMA 9002-C14 or ANSI/ASME 9112-B15 specifies the bore check before cutting keyways.

8.6.2.4.2 The acceptance criteria for the CMM checking is identified in [8.6.2.4.2.1](#) and [8.6.2.4.2.2](#). [Annex J](#) provides information for performing the checks.

NOTE This process is significantly easier with a programmable direct drive CMM and should only be performed on a CMM with the necessary level of accuracy and repeatability.

8.6.2.4.2.1 If the profile of the bore's surface is inspected using a CMM technique, it shall be performed using a minimum of 32 points at a minimum of four different locations (minimum eight points per location) through the length of the bore. Two of the locations should be as close as possible to the large end bore and the small end bore.

8.6.2.4.2.2 If a scanning method is used, a minimum of 1440 degrees (four sweeps) from the large end bore to the small end bore through the length of the hub replaces the 32 points in [8.6.2.4.2.1](#). Unless otherwise specified for keyless hubs, the taper shall not exceed ± 0.0001 mm/mm (0.0012 in./ft) taper on diameter, or an average included angular deviation of 20 seconds and the profile of the surface or cone shall not exceed 0.02 mm (0.0008 in.). Unless otherwise specified for tapered hubs with keyways, the taper shall not exceed ± 0.00012 mm/mm (0.0014 in./ft) taper on diameter, or an average included angular deviation of 24 seconds and the profile of the surface or cone shall not exceed 0.024 mm (0.00096 in.) before cutting keyways.

8.6.2.5 The design of the tapered-bore hub shall provide for a shaft-end retaining nut or plate, including the necessary wrench clearance. The direction of the thread on the retaining nut shall be such that rotation of the coupling hub relative to the shaft (attributable to slippage under load) forces the hub more tightly on the taper.

8.6.3 Additional Requirements for Keyed Hubs

8.6.3.1 • The purchaser shall specify the number and configuration of the keyways.

8.6.3.2 Keys, keyways, and inspection methods shall conform to ANSI/AGMA 9002-C14 or ANSI/AGMA 9112-B15.

8.6.3.3 The bottom corners of all keyways shall be radiused with a smooth transition from the keyway walls. Corner radii shall conform to ANSI/AGMA 9002-C14 or ANSI/AGMA 9112-B15.

8.6.3.4 Keyed coupling hubs shall have a sufficient number of threaded puller holes so that the hub can be removed if fitted with the proper interference or advancement. Puller holes for bores that are less than 64 mm ($2\frac{1}{2}$ in.) shall have a minimum nominal diameter of 6 mm ($\frac{1}{4}$ in.). Puller holes for bores that are greater than or equal to 64 mm ($2\frac{1}{2}$ in.) shall have a minimum nominal diameter of 10 mm ($\frac{3}{8}$ in.). Puller holes shall have standard coarse threads.

8.6.3.5 The key(s) in the shaft for the keyed hub should completely fill the keyways of both the shaft and the coupling hub to avoid balancing errors or effects. The key(s) should not overhang the end of the shaft. Stepped, fully crowned keys should be used if the key extends past the end of the hub. Alternative key(s) should be approved by the purchaser.

NOTE Typically, service keys are supplied by the equipment vendor and not the coupling vendor.

8.6.4 Alternate Hub Design

Alternative coupling-to-shaft juncture methods (providing an interference fit) other than keyed or hydraulically fitted hubs may be proposed as an alternative, as determined by the purchaser, provided such methods comply with all the requirements of this document, including but not limited to torque transmission and balance requirements. Any exceptions shall be approved by the purchaser.

8.7 • Idling Adapter (Solo Plate)

If uncoupled operation is specified and the coupling has floating components when the coupling spacer is removed, the vendor shall supply an idling adapter for drive end coupling. The idling adapter shall center and maintain the balance of the coupling.

NOTE An idling adapter (solo plate) does not fulfill the function of a moment simulator. An idling adapter (solo plate) can be designed to also serve as a moment simulator.

8.8 Moment Simulator

8.8.1 • If specified, a moment simulator shall be supplied. The purchaser shall provide the vendor with the measurement of the distance from the end of the shaft to the centerline of the adjacent bearing. This moment simulator shall not differ from the contract coupling overhung moment by more than $\pm 10\%$.

NOTE A moment simulator can be designed to also serve as a solo plate.

8.8.2 If a moment simulator is specified in [8.8.1](#), with the purchaser's approval a "dummy hub" may be used to function as a moment simulator instead of using the job hub along with an additional component.

8.9 Component Fit Tolerances and Potential Unbalance Calculations

8.9.1 Except for nonmetallic insulating parts (see [8.11](#)), components of couplings operating at speeds greater than 1800 rev/min shall be centered by means of piloted or rabbeted fits.

8.9.1.1 The eccentricity of these fits shall not exceed 0.00008 mm/mm (0.00008 in./in.) of diameter TIR or 0.013 mm (0.0005 in.) TIR, whichever is greater.

8.9.1.2 Fits that tighten under centrifugal loading are preferred.

8.9.1.3 The fit shall range from a loose fit of 0.025 mm (0.001 in.) to an interference fit, with the actual fit determined by balancing requirements.

8.9.1.4 For couplings operating at 1800 rev/min or less, fits are only required if needed to meet the balance tolerance.

8.9.2 The face runout of mating faces (except for flexible elements and nonmetallic insulating parts) shall not exceed 0.00008 mm/mm (0.00008 in./in.) of diameter TIR or 0.025 mm (0.001 in.) TIR, whichever is greater. For couplings operating at 1800 rev/min or less, face runout tolerances are only required if needed to meet the balance tolerance.

8.9.3 • If specified, the coupling vendor shall perform calculations to verify the potential unbalance of the complete coupling, in the plane of the center of mass of each half coupling. The calculation shall be for the coupling specified.

NOTE An example of a potential unbalance calculation is given in [Annex G](#).

8.9.3.1 The potential unbalance shall not exceed ANSI/AGMA 9000-C90, class 9, potential mass center displacement of 0.05 mm (0.002 in.), for couplings operating at speeds 1800 rev/min or less.

8.9.3.2 For couplings operating at speeds greater than 1800 rev/min up to 5000 rev/min, the potential unbalance shall not exceed ANSI/AGMA 9000-C90, class 10, potential mass center displacement of 0.027 mm (0.001 in.).

8.9.3.3 For couplings operating at speeds greater than 5000 rev/min, the potential unbalance shall not exceed ANSI/AGMA 9000-C90, class 11, potential mass center displacement of 0.013 mm (0.0005 in.).

8.9.3.4 The center of mass is the location at which the mass of the half coupling can be considered to be concentrated, and is normally referenced from the equipment shaft end, with a positive location being beyond the shaft end and a negative location being within the shaft.

8.9.4 If a potential unbalance calculation has been specified and performed in accordance with [8.9.3](#), the registers and the fits that locate components with respect to each other can require greater accuracy than the requirements of [8.9.1](#) or [8.9.2](#).

8.10 Fasteners (Including Studs)

8.10.1 Fasteners for all piloted flanges, including integral flanges, and customer-interface flanges shall have a diametrical clearance of not greater than 0.13 mm (0.005 in.) in the holes of one flange. Where the bolts are threaded into one flange, this requirement does not apply. The hole location shall allow the balance requirements to be met.

8.10.2 For couplings that operate at 1800 rev/min or less, the location and the clearance between the bolt and the hole shall be such that the balance requirements can be satisfied.

8.10.3 Metallic, deformed-thread, self-locking fasteners shall be used. Castellated lock nuts are not acceptable. Lock washers shall not be used.

8.10.4 The coupling vendor shall recommend the interval of the replacement of fasteners.

8.10.5 The threads of the fasteners shall comply with ISO 262, class 1, or ANSI/ASME B1.1. The quality of the nuts shall be at least equal to that of the bolts or studs.

8.10.6 The coupling vendor shall specify the required bolt torque or tensile load and shall state whether this torque is for dry or lubricated torquing.

8.10.7 Fasteners shall be held within tolerances, on both dimensions and mass, sufficient to permit interchange within the same set of fastener or substitution of a spare set of fasteners without affecting the coupling integrity or resulting in the balance being outside the prescribed limits. See [9.3.4](#).

8.10.8 A minimum of 10 %, a minimum quantity of two spare coupling fasteners for installation shall be supplied with each coupling and each spare set of coupling fasteners.

NOTE 1 Couplings are weight sensitive and replacement of individual fasteners versus a matched set can cause a coupling balance issue.

NOTE 2 A complete set of spare nuts to maintain the required prevailing torque requirement can be considered.

NOTE 3 A complete set of matched fasteners to facilitate repeated assembly and disassembly can be considered.

8.11 • Electrical Insulation

If specified, the coupling shall be electrically insulated to prevent the flow of electrical current from one shaft to the other through the coupling. Any nonmetallic insulating parts are exempt from the requirements of [8.9.1](#) and [8.9.2](#).

NOTE Electrical insulation is normally achieved by the insertion of insulating material between the flanges and around the flange bolts, including bolt head or nut faces.

8.12 Dynamics

8.12.1 The axial natural frequency (ANF) of a single element convoluted diaphragm coupling shall not occur within 10 % of the specified operating speed range. The vendor shall identify the ANF in the proposal.

NOTE Multi-disc, multi-diaphragm, and non-convoluted single-element flexible couplings do not exhibit a detrimental axial vibration response.

8.12.2 The lateral natural frequency (N_c) of that portion of the coupling between and including the flex elements, assuming infinitely stiff supports, shall be at least two times the highest specified operating speed for the uniform-tube-equation methodology as specified in ANSI/AGMA 9004-B08 or ANSI/AGMA 9104-A06, or at least 1.5 times using a more rigorous analysis based on actual geometry (for example, finite-element analysis). The purchaser and vendor shall agree who shall perform these calculations and the assumptions used.