

SAFETY, APPLICATION, AND SERVICE FACTORS AS APPLIED TO SHAFT COUPLINGS

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ABSTRACT

Shaft couplings are rated by their manufacturers as to how much torque they can transmit, how much misalignment they can accommodate, and the maximum speed at which they can operate. Still, manufacturers, buyers, or various organizations often use qualifiers, or factors that significantly reduce the published capabilities of couplings, when the published ratings cannot be used at face value.

It is the authors' intent herein to help coupling users in understanding these factors.

INTRODUCTION

Three correction factors are used, often in combination:

- *safety factors*, which establish the ratio between the breaking point and the maximum catalog ratings,
- *application (experience) factors*, which establish the ratio between the rated torque from the catalog, and the actual torque of a given application, and
- *service factors*, which establish the ratio between the maximum value of a fluctuating torque and the average torque as calculated on the basis of power and speed.

All these factors are subjective, inasmuch as they are determined based on experience and economic parameters. They differ from manufacturer to manufacturer and from user to user. Organizations such as the American Petroleum Institute (API) and American Gear Manufacturers Association (AGMA) have issued standards in which values for the experience and service factors are recommended; in the views of many people, these standards have created as many problems as they were trying to solve. No attempt was made so far to standardize safety factors; these are established by coupling manufacturers. Only a limited amount of data on safety factors was published.

SAFETY FACTORS

A safety factor is the ratio between the breaking point and the rating value of a given device; its magnitude is a compromise

between safety and economics. For equipment in public use (such as elevators) safety factors are mandated by government agencies; in the case of couplings, each manufacturer establishes factors arbitrarily. It can be said that a good safety factor is a "sleeping pill," as it allows both the manufacturer and the user to sleep well in case any device is operated at its catalog rating. Safety comes at a price; the larger the safety margin, the higher the cost of a product.

In the particular case of flexible couplings, safety factors apply to speed, torque, and operating misalignment, as any of these parameters can cause breaking of a coupling component. Couplings with flexible elements (metallic or elastomeric) have an additional parameter to consider: axial displacement. Axial displacement stretches flexible elements, and when resulting stresses become excessive they can also cause failures.

Speed Parameter

Rotating speed creates both radial and tangential stresses in a component; the tangential stresses, also known as "hoop" stresses, are usually larger. Manufacturers publish the maximum speed at which each coupling type and size can operate. The margin of safety of the speed rating is easy to determine through "reverse engineering," as the approximate hoop stress (σ) is easily obtained from the formula:

$$\sigma = \frac{V^2}{K}$$

For σ in pounds/inch²,
V in inches/second, and
material being steel,
K = 1500

where V = rim velocity, and

K = coefficient that depends on material

The strength of materials used in couplings can be found in the literature; most manufacturers publish the materials used in their products, particularly for "special purpose" couplings. Special-purpose couplings are used in high-performance applications, which generally implies high speeds. They are made of heat treated alloy steels, or other high strength metals.

The authors have surveyed the published speed ratings of a large number of coupling manufacturers from many countries. The result of the survey showed that manufacturers use either a constant hoop stress across a line of couplings, or that the stress is a function of rim diameter. Two such cases are shown in Figures 1 and 2.

Outside Diameter	Maximum Speed	Peripheral Velocity	Hoop Stress
inch	rpm	inch/sec	psi
6.56	23,800	8217	44,320
7.84	19,900	8211	44,250
9.09	17,100	8181	43,930
10.47	14,900	8211	44,250
11.94	13,100	8232	44,480
13.06	11,900	8180	43,910
14.31	10,900	8209	44,230
15.75	9,900	8207	44,200
17.00	9,200	8200	44,470

Figure 1. Hoop Stresses in an American Disk-Pack Coupling Line. Notes: The hoop stresses of this coupling line are held constant for all sizes.

Outside Diameter	Maximum Speed	Peripheral Velocity	Hoop Stress
inch	rpm	inch/sec	psi
5.6	33,800	9962	66,160
7.4	24,500	9500	60,170
8.4	21,300	9400	58,900
9.8	18,400	9490	60,040
11.1	16,000	9300	57,700
12.4	13,400	8750	51,040
13.9	11,600	8490	48,050

Figure 2. Hoop Stresses in an European Disk-Pack Coupling Line. Notes: The hoop stresses are not held constant; the safety factor becomes larger as the diameter increases.

In both cases, the materials are alloyed steels with a tensile strength of approximately 170,000 psi. The safety factor for Case 1 (Figure 1) is determined as:

$$S_F = \frac{170,000}{44,000} = 3.86$$

For the coupling of Case 2 (Figure 2) the safety factor varies according to size, and is best shown in graphical form (Figure 3). In both cases, even the smallest safety factor is large enough to satisfy most users.

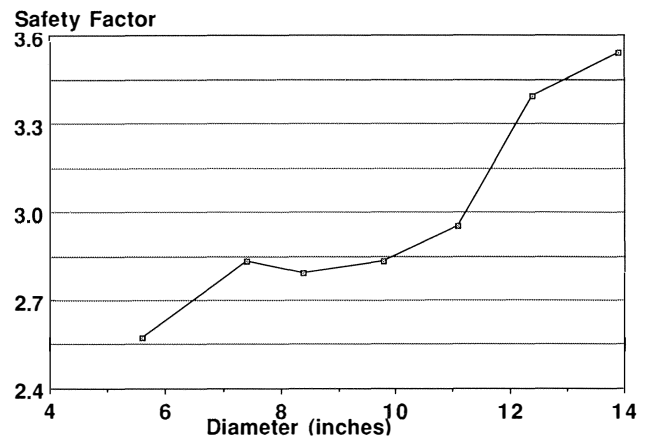


Figure 3. Speed Safety Factor of the Disk-Pack Coupling Line from Figure 2. Notes: Even the smallest safety factor, as related to hoop stresses, is quite satisfactory.

One manufacturer of diaphragm-type couplings relates the maximum speed to the operating axial stretch of its metal-diaphragm coupling (Figure 4). Making speed and axial stretch interdependent is logical; both generate constant stresses in the element, and reducing one allows the increase of the other without exceeding acceptable stresses.

A tabulation of this coupling's maximum rated speed at zero axial stretch, and of the resulting hoop stresses is shown in Figure 5. Operating one of these couplings at the "maximum rated speed" would probably place it in a failure mode.

The resulting safety factor is shown in the graph in Figure 6, based on an alloy steel with a tensile strength of 170,000 psi. If the larger couplings would be operated at the published maximum

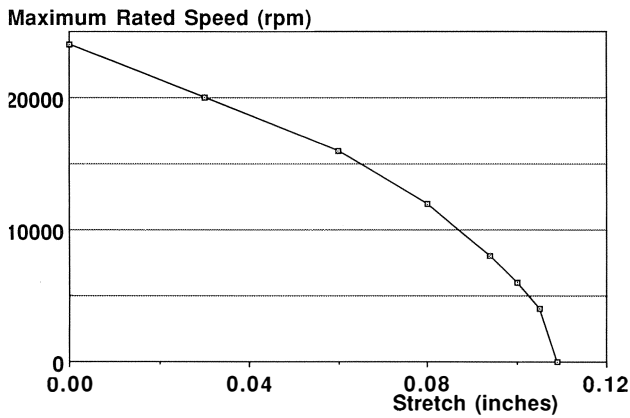


Figure 4. Speed Vs Axial Stretch Graph Used for a Line of American Diaphragm Couplings. Notes: While operation at zero axial stretch is possible and desirable, extending the ratings down to zero speed is superfluous.

Outside Diameter	Maximum Speed	Peripheral Velocity	Hoop Stress
inch	rpm	inch/sec	psi
6.1	40,000	12,840	109,950
7.1	38,000	14,200	134,400
9.2	28,500	13,800	126,960
11.0	24,500	14,200	134,240
13.1	21,500	14,800	146,500
14.9	19,500	15,300	155,900
16.9	18,000	16,000	170,900
18.9	16,500	16,400	179,600
22.6	14,000	16,650	184,900
26.4	12,800	17,780	210,900

Figure 5. Hoop Stresses in the Diaphragm Couplings of Figure 4, at Maximum Rated Speed. Notes: Operation of coupling sizes that fall below the heavy line, at the maximum rated speed, will probably cause failures, as stresses exceed ultimate strength.

“acceptable” speed, failures would most likely occur. Although making speed and axial stretch interdependent is logical, a safe limit for maximum speeds should always be incorporated.

It is interesting to note that an older catalog of the same manufacturer, as well as the one of their European competitor, list speeds that result in hoop stresses of only 60,000 psi, equivalent to a safety factor of 2.8.

Torque Parameter

Catalog torque often determines the useful life of a coupling rather than the breaking point of the torque-transmitting elements of a coupling. Therefore, safety factors cannot be easily applied to torque. Because of this, many manufacturers make the rated torque, the maximum speed, and misalignment inter-related. For instance, one of the most popular types of special purpose gear couplings has a rated torque that is a function of the operating speed. This coupling’s torque derating factor as a function of speed is shown in Figure 7.

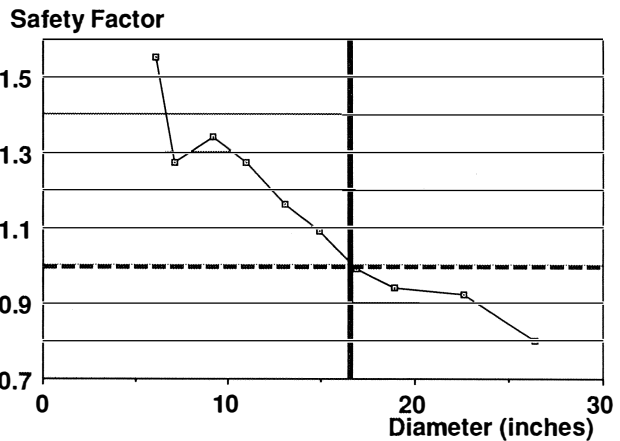


Figure 6. Safety Factors, at Maximum Rated Speeds, in the Diaphragm Coupling Line of Figure 4. Notes: Operation of coupling sizes that fall to the right of the heavy line, at the maximum rated speed, will probably result in failures, as the safety factors are smaller than one.

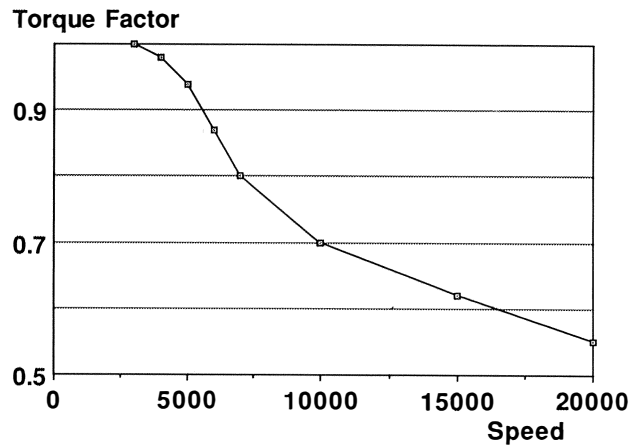


Figure 7. Torque Vs. Speed Curve of a Special-Purpose Gear Coupling Line. Notes: The manufacturer reduces the rated torque at speeds above 3,000 rpm, in order to maintain a satisfactory service life at all speeds.

However, not all coupling types are designed the same way; in all coupling types some components will eventually break if torque reaches a certain level. For instance, in gear-type couplings, flange bolts will break long before the teeth will break, while in spoke-type diaphragm couplings, the spokes will break first.

Peak torque is defined as a momentary or instantaneous load that occurs infrequently, such as startup, surge, or other occasional events. It subjects couplings to very high stresses, which are close to the yield strength of the components. Usually, manufacturers list a rated peak torque that is two to three times larger than the rated torque. One notable exception will be discussed later on.

Alignment Parameter

In couplings with flexible metal membranes, the safety factor is related to the Goodman diagram, which establishes a zone of infinite life, as a function of the alternating and constant stresses in the components. Diagrams published by two different manufacturers are shown in Figures 8 and 9. They provide a curve for the limit conditions (at which failures start occurring), and a “rated”

curve, which represents operations at a given safety margin. While the rated curve from Figure 8 provides a safety factor of two at worst conditions, for both the alternating and the constant stresses, the ratings from Figure 9 provide a safety factor of two for the alternating stresses, but a much smaller and unusually low safety factor of 1.3 for constant stresses at point "A."

Alternative Stresses

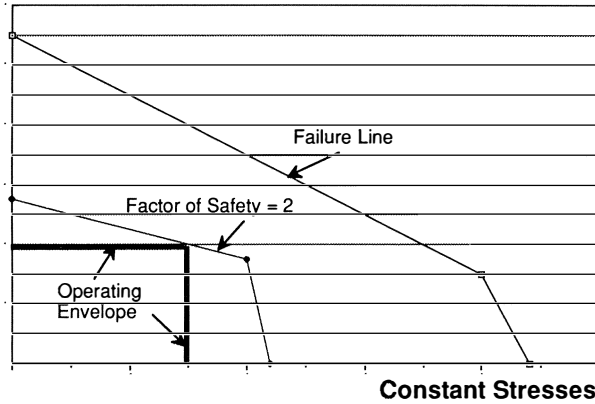


Figure 8. Rating Method of a Special-Purpose Disk-Pack Coupling Line. Notes: This figure illustrates a very conservative rating method, as even the worst conditions still have a safety factor of two for alternating stresses, and even larger for constant ones. On the other hand, the manufacturer mandates a minimum application factor of 1.5.

Alternative Stresses (psi)

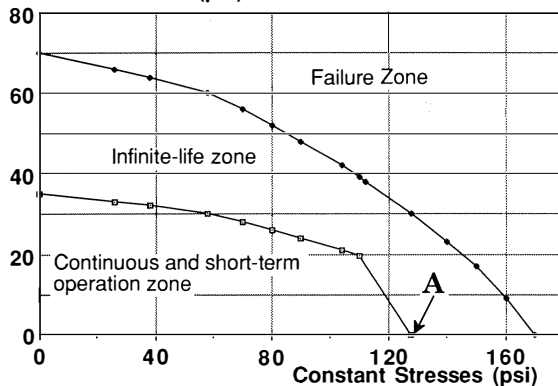


Figure 9. Rating Method of a Special-Purpose Diaphragm Coupling Line. Notes: The manufacturer uses a less-conservative rating method, as operation at point A result in a safety factor of only 1.3, in respect to constant stresses.

Constant stresses are generated by the combination of torque, centrifugal acceleration, and axial stretch. The latter represents the largest stress of the three. Therefore, any unpredictable increase in axial shaft separation, particularly when combined with a momentary peak torque, could cause element failures. The combination of large axial stretch and large torques are often encountered during the startup of a machine.

Some manufacturers publish speed derating factors related to operating misalignment. A European coupling manufacturer reduces the maximum speed as a function of misalignment, as shown in Figure 10.

Speed Factor

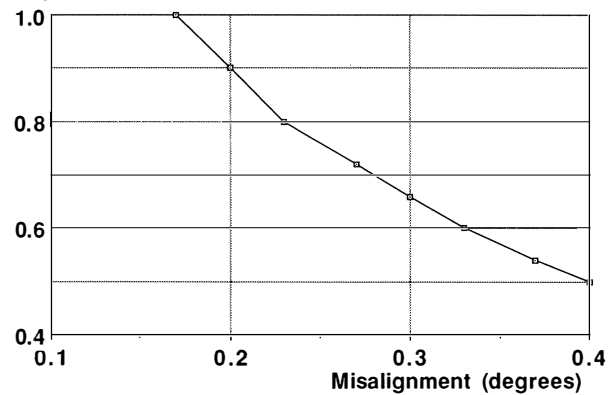


Figure 10. Speed Correction Factor of an European Gear-Coupling Line. Notes: The manufacturer uses a different method (as compared with the one of Figure 7) to maintain a satisfactory service life, at any acceptable misalignment.

Manufacturers of high-misalignment couplings publish torque derating factors related to misalignment. The maximum torque as a function of misalignment, as published by an American coupling manufacturer, is shown in Figure 11.

One Japanese gear-coupling manufacturer publishes a composite torque derating graph (Figure 12), in which the catalog rated torque is reduced as a function of both misalignment and speed (actually, the ratio of operating speed to maximum catalog speed). Derating torque as a function of speed is an understandable practice, and has been used by many manufacturers. However, the rated torque in this case can only be used if the coupling operates at zero misalignment. As this is an impractical condition, the validity of this manufacturer rating practice is cast into doubt.

Torque Factor

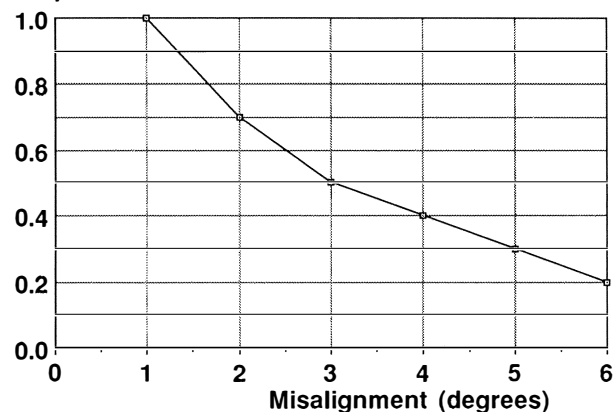


Figure 11. Torque Correction Factor of an American Line of High-Misalignment Gear Couplings. Notes: The manufacturer reduces the rated torque at misalignments larger than 1 degree, in order to maintain a satisfactory service life at any acceptable misalignment.

APPLICATION (EXPERIENCE) FACTORS

Application factors, also named experience factors, establish the ratio between the rated torque from the catalog and the actual torque of a given application.

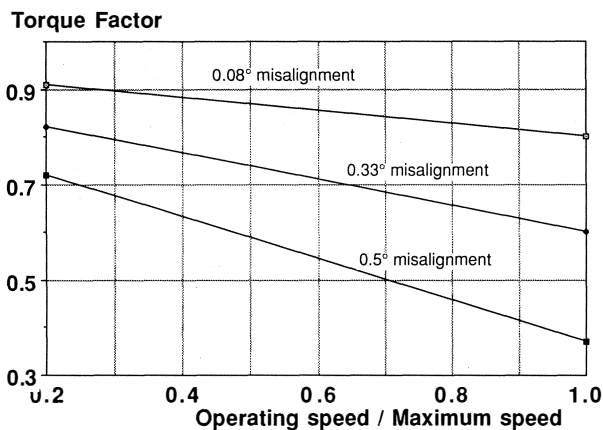


Figure 12. Rating Method of a Japanese Gear-Coupling Manufacturer. Notes: The manufacturer allows operation at rated torques only at zero misalignment! This is quite unacceptable, as at zero misalignment a flexible coupling becomes unnecessary!

The first flexible couplings were of the gear type, and ratings were of no concern: selection was made based on bore capacity, not torque capability. The pitch diameter of early couplings was twice the maximum bore. This geometry resulted in forces on teeth that allowed couplings to operate, for a satisfactory period of time, at whatever torque the shafts could transmit. Although current catalogs provide torque ratings, many types of gear couplings still have an inherent safety built in, if selected, so that the hubs can be installed on the shaft, then they can safely transmit the shaft's torque.

To be competitive, a few gear coupling manufacturers increased the bore capacity of their couplings while maintaining the same pitch diameter for the teeth. Not only was the loading on teeth increased, but the lubricant volume that can be contained within the coupling was drastically decreased. This increase in bore size made torque rating, rather than bore size, the determining parameter in the selection of these types of couplings.

Originally, application factors were established by the coupling buyer. A number of reasons may have led a purchaser to select a larger coupling than the one determined from catalog ratings: a margin was left for future increase in the machine output, the shaft did not fit in the coupling, or the purchaser may have had a lack of trust in the catalog ratings, a fear that was fueled by field failures of "correctly" selected couplings.

Field failures started to occur in alarming numbers about 25 years ago, when a tremendous surge forward occurred in speeds and power of machinery used in the process industries. Along with these advances came misapplications and frequent failures of couplings. One of the authors was involved with nearly thirty coupling failures in one ammonia syngas train (35,000 hp at 10,500 rpm). Similar experiences were encountered on hydrogen compressors. Serious questions about ratings occurred when one coupling manufacturer stated that the life expectancy of its products is about 18 months.

Maintenance of couplings became the bottle neck of the plants' ability to operate machines at increasingly longer periods of times. As coupling manufacturers provided limited help, users resorted to measures of their own, including the development of the Turbomachinery Symposium. Discussions at the Symposium revealed how universal coupling problems are, and the American Petroleum Institute (API) decided to take action.

In 1979, API published Standard 671: "Special-Purpose Couplings for Refinery Services." Among other guidelines for special-purpose couplings was a torque application factor of 175 percent.

The intent of this application factor was not to address the margin of safety on design, rather, "to mandate an adequate experience factor in order to allow for off-design operations which may occur at operating points requiring higher torque than the normal operating point at which the coupling selection is based, as well as equipment variation resulting in higher torque than actual equipment design point."

For awhile, some manufacturers thought that the standard would put them out of business, as they could no longer be competitive. Users accepted exceptions to this rule for some time. When new coupling catalogs were published, they suddenly contained higher ratings in many instances for identical products. An example of such an increase after the issuing of API Standard 671 is shown in Figure 13. Even more striking is a note that appears in the English version of a European coupling catalog: "A service Factor of 1.75 as per API 671 is already incorporated in the (torque) values indicated in the list." However, the ratings listed in this catalog were identical to those published in a pre-API 671 catalog of the same manufacturer.

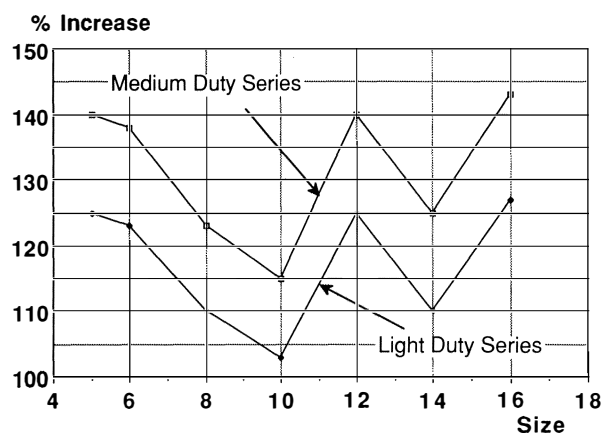


Figure 13. Torque Rating Increases of a Diaphragm-Coupling Line, After API 671 Was Issued. Notes: The manufacturer elected to increase the catalog ratings of two coupling types, without any apparent changes in geometry or material.

Recently, three manufacturers of disk-pack couplings (two American and one European) published catalogs that included their own application factors, as shown in Figure 14. The minimum application factor that must be applied to the "continuous torque rating" is 1.5. Under this condition, the published ratings cannot be used for even the best operating conditions, thus making the ratings meaningless. Furthermore, all three manufacturers list a peak torque which is 1.33 times the rated torque (most coupling

APPLICATION FACTORS

Constant Torque	1.50
API 671	1.75
Moderate Fluctuations	2.00

Figure 14. Application Factor Table, as Published by Three Manufacturers of Similar Disk-Pack Coupling Lines. Notes: The manufacturers of these special-purpose couplings do not allow operation at rated torques, under any conditions. Therefore, the published ratings become meaningless.

catalogs list a peak torque that is a minimum of two times the rated torque). Indeed, multiplying the peak-torque factor of 1.33 by the "minimum" application factor yields exactly 2.0. This suggests that the real coupling ratings are 2/3 of the ones listed.

The question that some engineers might ask is: why use application factors at all? Applying one safety factor on top of the one the manufacturer has already applied to the ratings seems unnecessary and uneconomic.

There are at least two reasons for the use of experience factors:

- A large number of machines are periodically upgraded. One example involves a number of simple-cycle Frame 5 gas turbines with which one of the authors has been involved for over 30 years. Two such machines were originally installed in the early 1960s, and could develop 14,000 hp. Through continuous improvements in power turbine materials, blade geometry, increased compressor section flow, and better governor systems, the same machines now generate in excess of 20,000 hp, an increase of over 45 percent. Because the experience factors originally used in the selection of the load couplings were large, the same couplings are successfully used even today.

- Experience factors should be used as a correction factor in those cases when manufacturers use less-than-desirable margins when they rate couplings. An indiscriminate, across-the-board experience factor would lead to more uneconomic selections than wise ones. This statement is particularly valid for cases where ratings are artificially inflated.

SERVICE FACTORS

Service factors were introduced by manufacturers of gear couplings with curved faced teeth. At low misalignment, these couplings can transmit less torque than the ones with straight teeth, because of larger contact pressures. Manufacturers of couplings with curved teeth often publish the same ratings as the ones of couplings with straight teeth, but apply a "service factor," a practice that was not used with couplings having straight teeth.

Service factors are a significant selection factor for applications where torque fluctuates cyclically. *Couplings must be selected for the maximum torque that occurs during one cycle.* As this value is seldom known, coupling manufacturers used previous experience to establish a ratio between the maximum torque and the average torque (as determined through calculations, using power and speed). The actual torque (curve B) and the calculated torque (curve A) is shown in Figure 15 for a cyclic torque application.

The areas under the two curves from Figure 15 are identical, as they represent the power that flows through the coupling. Because torque, not power, defines the required rating of a coupling, the selection of a coupling's size must be made for the maximum torque that occurs during one cycle, and not for the power transmitted.

The maximum torque during each cycle should not be confused with the peak torque, which occurs only occasionally.

Original equipment manufacturers know the torque curves of their machines, and can make judicious coupling selections. Without a torque curve, the maximum torque must be estimated as a percentage of the average torque. This percentage factor is the service factor.

As each coupling manufacturer has its own list of service factors, attempts were made to make them uniform. A standard titled "Load Classification and Service Factors" was published in 1968 by AGMA (No. 514.01). With the advent of elastomer couplings, it was found that various materials react differently to torque fluctuations, and that service factors must, therefore, be made a function of coupling material, along with application. AGMA's standard has since been withdrawn; only manufacturers' data should be used.

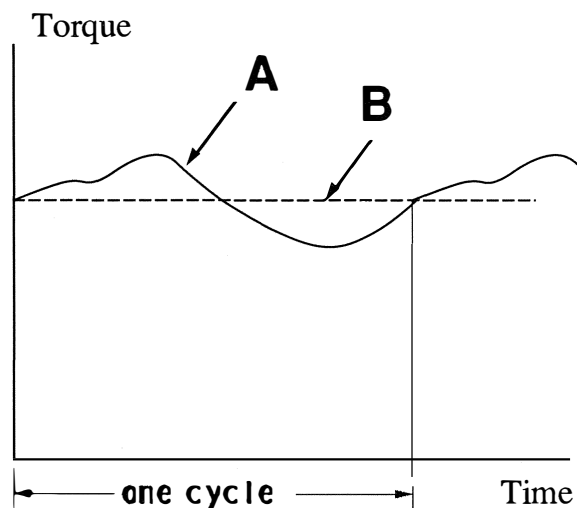


Figure 15. Torque Fluctuation through a Coupling. Notes: The areas under curve A (actual torque), and curve B (calculated torque) are identical, and represent transmitted power. Couplings must be selected to accommodate the maximum torque within one cycle, rather than the power transmitted.

To help with coupling selection, manufacturers publish long lists of service factors.

Example

A reciprocating compressor rated at 38 hp is driven by an electric motor rated at 40 hp at 1800 rpm. Assuming that the selected coupling type has a service factor of two, the selected coupling size should have a rating of:

$$\text{Rating (HP/100 rpm)} = \frac{38 \times 2}{1800/100} = 4.22$$

Note that in selecting the coupling size, the power consumed, and not the one of the driving machine, should be used in the calculation.

Service factors vary widely between different types of couplings using elastomer flexible elements. When elastomers are subjected to continuous flexing, they absorb part of the energy transmitted through the coupling. The energy absorbed (damping) is transformed into heat, raising the coupling's temperature, which in turn softens and weakens the elastomer. The amount of heat absorbed is a function of the magnitude of torque fluctuation, the operating speed, and the type of elastomer. Generally, rubber has a smaller damping coefficient than urethanes, and it therefore absorbs less energy, under the same operating conditions. Without the cooling provided by the windage (caused by rotation), the elastomer elements would become very hot, and their strength diminishes. The authors have seen cases where elements actually melted, because the coupling guard did not allow for any air circulation.

Because of the larger damping factor, manufacturers of couplings utilizing urethanes recommend a larger service factor than the ones used with rubber couplings. A comparison of service factors of similar couplings, using rubber and urethanes is shown in Figure 16.

The torque generated by the driving machine can also be cyclic, as is the case with reciprocating engines. Therefore, the maximum torque transmitted through a coupling during one cycle is a function of the torque variation of the driving and of the driven

Application	Rubber	Urethane
Agitators	1.0	1.5
Vane Blowers	1.0	1.5
Car pullers	1.5	2.0
Lobe-type compressors	1.5	2.0
Crane drives	1.5	2.0
Cooling tower fans	1.5	2.5
Paper mill chippers	2.5	3.5
Banbury mixers	2.5	3.0

Figure 16. Service Factors Vs Element Material. Notes: Service factors for couplings using urethanes are larger than the ones that use rubber, as urethanes absorb more energy (therefore become hotter) when subjected to flexing, as caused by misalignment or torque fluctuations.

machines. Most coupling manufacturers list two service factors, one for the driver and one for the driven, factors *which are additive*. The sum of the two factors becomes the new service factor. It is important to remember that:

- When the torque consumed is constant in time, the service factor for the driven machine is one.
- When the torque generated is constant in time, the service factor for the driving machine is zero.

Example:

A reciprocating compressor is driven by a four-cylinder gasoline engine. The compressor's service factor is two, while the engine's service factor is 0.5. The selection of the coupling should be made using a correction factor of $2 + 0.5 = 2.5$.

A survey of service factors will reveal that recommendations of coupling manufacturers from many nations have the smallest factor larger than one for even the best possible conditions. This practice indicates that the coupling ratings are effectively meaningless. The service factor tabulation of an European coupling manufacturer is shown Figure 17.

CONCLUSIONS AND RECOMMENDATIONS

Coupling users must be knowledgeable about the process of coupling selection, in order to avoid the pitfalls of meaningless or misleading ratings, and the possibility of ending up with an undersized coupling. To this end, the following recommendations are made:

- Users should question the veracity of *coupling catalogs* that contain ratings which cannot be used even under the best conditions.

Agitator, light liquid	1.25
Agitator, heavy liquid	2
Compressor, centrifugal	1.25
Compressor, reciprocating	1.75
Fan, blower small	1.25
Fan, blower heavy	1.75
Generator	1.5
Propeller	2
Wood and plastic machinery	1.5

Figure 17. Published Service Factors of an European Disk-Pack Coupling Line. Notes: No service factor of one is listed! The published ratings are therefore meaningless, as they cannot be used for any application.

- Coupling manufacturers should establish an "industry standard" that covers safety and service factors. Without such a standard, users' associations should consider mandating minimum safety factors, and request proof that these factors are met.

- *Service factors* for smooth service should always be unity (1), as the only purpose of service factors is to compare rough service to smooth service.

- Manufacturers should avoid cataloguing *application factors*, as such factors are strictly users' tools.

- An indiscriminate, across-the-board value for *experience (application) factors* would not serve either purpose of such factors: to allow for future upgrading, and to numerically show users' confidence level in a coupling type, or in a manufacturer. However, a standardized experience factor (such as the one recommended by API 671) should be used whenever there is no previous experience with a particular type of equipment or coupling.

ACKNOWLEDGEMENTS

The authors used data from published catalogs of a large number of coupling manufacturers in the preparation of the paper. Actual data will be made available on request.

The examples herein were chosen from manufacturers in three continents only to show that the practices described are universal in nature. Even though tables and figures use data from specific catalogs, which may be "American," "Japanese," or "European," the type of data shown can be found in many other catalogs. The examples given are not intended to single out a particular manufacturer or country.

