ASME Section VIII Pressure Vessels Design Considerations for Bolted Flange Connections

The primary purpose of the rules for bolted flange connections in Parts A and B of Appendix II is to insure safety, but there are certain practical matters to be taken into consideration in order to obtain a serviceable design. One of the most important of these is the proportioning of the bolting, i.e., determining the number and size of the bolts.

In the great majority of designs the practice that has been used in the past should be adequate, viz., to follow the design rules in Appendix II and tighten the bolts sufficiently to withstand the test pressure without leakage. The considerations presented in the following discussion will be important only when some unusual feature exists, such as a very large diameter, a high design pressure, a high temperature, severe temperature gradients, and unusual gasket arrangement, and so on.

The maximum allowable stress values for bolting given in the various tables of Subsection C are design values to be used in determining the minimum amount of bolting required under the rules. However, a distinction must be kept carefully in mind between the design value and the bolt stress that might actually exist or that might be needed for conditions other than the design pressure. The initial tightening of the bolts is a prestressing operation, and the amount of bolt stress developed must be within proper limits, to insure, on the one hand, that it is adequate to provide against all conditions that tend to produce a leaky joint, and on the other hand, that is not so excessive that yielding of the bolts and/or flanges can produce relaxation that also can result in leakage.

The first important consideration is the need for the joint to be tight in the hydrostatic test. An initial bolt stress of some magnitude must be provided. If it is not, further bolt strain develops during the test, which tends to part the joint and thereby to decompress the gasket enough to allow leakage. The test pressure is usually One and a half times the design pressure, and on this basis it may be thought that 50 percent extra bolt stress above the design value will be sufficient. However, this is an oversimplification, because, on the one hand, the safety factor against leakage under test conditions in general need not be as great as under operating conditions.

On the other hand, if a stress-strain analysis of the joint is made, it may indicate that an initial bolt stress still higher than 1 ½ times the design value is needed. Such an analysis is one that considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. In any event, it is evident that an initial bolt stress higher than the design value may and, in some cases, must be developed in the tightening operation, and it is the intent of this Division of Section VIII that such a practice is permissible, provided it includes necessary and appropriate provision to insure against excessive flange distortion and gross crushing of the gasket.

It is possible for the bolt stress to decrease after initial tightening, because of slow creep or relaxation of the gasket, particularly in the case of the "softer" gasket materials. This may be the cause of leakage in the hydrostatic test, in which case it may suffice merely to retighten the bolts. A decrease in bolt stress can also occur in service at elevated temperatures, as a result of creep in the bolt and/or flange or gasket material, with consequent relaxation. When this results in leakage under service conditions, it is common practice to retighten the bolts, and sometimes a single such operation, or perhaps several repeated at long intervals, is sufficient to correct the condition. To avoid chronic difficulties of this nature, however, it is advisable when designing a joint for high-temperature service to give attention to the relaxation properties of the materials involved, especially for temperatures where creep is the controlling factor in design. This prestress should not be confused with initial bolt stress, S1, used in the design of Part B flanges.

In the other direction, excessive initial bolt stress can present a problem in the form of yielding in the bolting itself, and may occur in the tightening operation to the extent of damage or even breakage. This is especially likely with bolts of small diameter and with bolt materials having a relatively low yield strength. The yield strength of mild carbon steel, annealed austenitic stainless steel, and certain of the nonferrous bolting materials can easily be exceeded with ordinary wrench effort in the smaller bolt sizes. Even if no damage is evident, any additional load generated when internal pressure is applied can produce further yielding with possible leakage. Such yielding can also occur when there is very little margin between initial bolt stress and yield strength.

An increase in bolt stress, above any that may be due to internal pressure, might occur in service during startup or other transient conditions, or perhaps even under normal operation. This can happen when there is an appreciable differential in temperature between the flanges and the bolts, or when the bolt material has a different coefficient of thermal expansion than the flange material. Any increase in bolt load due to this thermal effect, superposed on the load already existing, can cause yielding of the bolt material, wheras any pronounced decrease due to such effects can result in such a loss of bolt load as to be a direct cause of leakage. In either case, retightening of the bolts may be necessary, but it must not be forgotten that the effects of repeated retightening can be cumulative and may ultimately make the joint unserviceable.

In addition to the difficulties created by yielding of the bolts as described above, the possibility of similar difficulties arising from yielding of the flange or gasket material, under like circumstances or from other causes, should also be considered.

Excessive bolt stress, whatever the reason, may cause the flange to yield, even though the bolts may not yield. Any resulting excessive deflection of the flange, accompanied by permanent set, can produce a leaking joint when other effects are superposed. It can also damage the flange by making it more difficult to effect a tight joint thereafter. For example, irregular permanent distortion of the flange due to uneven bolt load around the circumference of the joint can warp the flange face and its gasket contact surface out of a true plane.

The gasket, too, can be overloaded, even without excessive bolt stress. The full initial bolt load is imposed entirely on the gasket, unless the gasket has a stop ring or the flange face detail is arranged to provide the equivalent. Without such means of controlling the compression of the gasket, consideration must be given to the selection of gasket type, size and material that will prevent gross crushing of the gasket.

From the foregoing, it is apparent that the bolt stress can vary over a considerable range above the design stress value. The design stress values for bolting in Subsection C have been set at a conservative value to provide a factor against yielding. At elevated temperatures, the design stress values are governed by the creep rate and stess-rupture strength. Any higher bolt stress existing before creep occurs in operation will have already served its purpose of seating the gasket and holding the hydrostatic test pressure, all at the design pressure and temperature.

Theoretically, the margin against flange yielding is not as great. The design values for flange materials may be as high as five-eights or two-thirds of the yield strength. However, the highest stress in a flange is usually the bending stress in the hub or shell, and is more or less localized. It is too conservative to assume that local yielding is followed immediately by overall yielding of the entire flange. Even if a "plastic hinge" should develop, the ring portion of the flange takes up the portion of the load the hub and shell refuse to carry. Yielding is far more significant if it occurs first in the ring, but the limitation in the rules on the combined hub and ring stresses provides a safeguard. In this connection, it should be noted that a dual set of stresses is given for some of the materials. In the ASME Boiler & Pressure Vessel Code, Section VIII: Division 1, Table UHA-32, the lower values should be used in order to avoid yielding in the flanges.

Another very important item in bolting design is the question whether the necessary bolt stress is actually realized, and what special means of tightening, if any, must be employed. Most joints are tightened manually by ordinary wrenching, and it is advantageous to have designs that require no more than this. Some pitfalls must be avoided, however. The probable bolt stress developed manually, when using standard wrenches, is:

Where S is the bolt stress and d is the nominal diameter of the bolt. It can be seen that smaller bolts will have excessive stress unless judgement is exercised in pulling up on them. On the other hand, it will be impossible to develop the desired stress in very large bolts by ordinary hand wrenching. Impact wrenches may prove serviceable, but if not, resort may be had to such methods as preheating the bolt, or using hydraulically powered bolt tensioners. With some of these methods, control of the bolt stress is possible by means inherent in the procedure, especially if effective thread lubricants are employed, but in all cases the bolt stress can be regulated within reasonable tolerances by measureing the bolt elongation with suitable extensometer equipment. Ordinarily, simple wrenching without verification of the actual bolt stress meets all practical needs, and measured control of the stress is employed only when there is some special or important reason for doing so.

Allowable Bolt Stress

The ASME Boiler & Pressure Vessel Code, Section VIII: Division 1, Appendix S in particular deals with the bolt stress. For example, a flange designer should determine the necessary tightening at the given operating temperature specifically in accordance with the allowable stresses for the bolt material at the operating temperature. These allowable stresses are based on the particular material; and their strength at operating temperature.

Hydrostatic testing, which in the majority of cases is necessary to verify the system, is done at one and a half times the operational pressure. Consequently, a flanged joint designed in accordance with the ASME Code, which should be hydrostatic tested with a pressure higher than the design pressure, will require a higher initial stress on the stud to successfully pass the test.

Appendix S of the ASME Boiler & Pressure Vessel Code, Section VIII: Division 1 speaks in great length establishes that in order to pass the hydrostatic test, the bolts must be stressed to whatever level is required to satisfactorily pass the test. This introduces additional problems. In cases where low yield bolt material is being used, the stresses required in bolts sufficient to satisfactorily pass the test may exceed the yield point of the bolt material causing the bolts to fracture.

Bolt Load Formulas

The ASME Unfired Pressure Vessel Code, Section VIII, Division 1 defines the initial bolt load required to seat a gasket sufficiently as:

$$W_{m2} = \pi b G y$$

The required operating bolt load must be at least sufficient, under the most severe operating conditions, to contain the hydrostatic end force and, in addition, to maintain a residual compression load on the gasket that is sufficient to assure a tight joint ASME defines this bolt load as:

$$W_{m1} = \left(\frac{\pi}{4}\right) G^2 P + 2b\,\pi\,GmP$$

After W_{m1} and W_{m2} are calculated, then the minimum required bolt area A_m is determined:

$$A_{m_1} = \frac{W_{m1}}{S_a}$$

$$A_{m2} = \frac{W_{m2}}{S_a}$$

if $A_{m_1} \geq A_{m2} \quad A_m = A_{m_1}$

if $A_{m2} \geq A_{m_1} \quad A_m = A_{m2}$

Bolts are then selected so that the actual bolt area A_b is equal to or greater that A_m

A_b = (Number of Bolts) x (Minimum Cross-Sectional Area of Bolt in Square Inches)

$A_b \geq A_m$

The maximum unit load $Sg_{(max)}$ on the gasket bearing surface is equal to the total maximum bolt load in pounds divided by the actual sealing area of the gasket in square inches.

$$Sg_{(max)} = \frac{A_b S_a}{\frac{\pi}{4}\left[(OD - 0.125)^2 - (ID)^2\right]} \quad \left\{ \text{Spiral Wound Gaskets} \right.$$

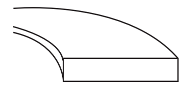
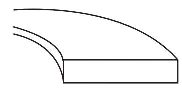
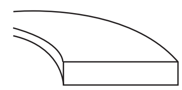
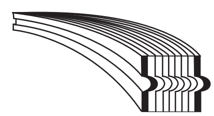
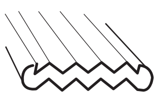
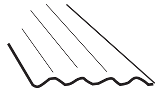
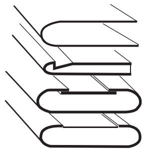
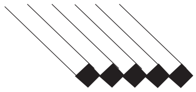
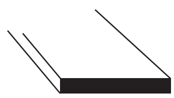
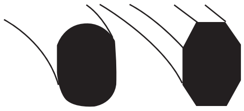
$$Sg_{(max)} = \frac{A_b S_a}{\frac{\pi}{4}\left[(OD)^2 - (ID)^2\right]} \quad \left\{ \text{All Other Types of Gaskets} \right.$$

Notations Symbols and Definitions

Except as noted, the symbols and definitions below are those given in ASME Boiler and Pressure Vessel Code.

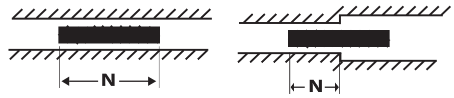
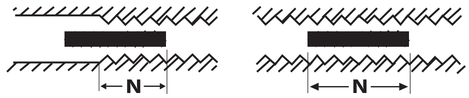
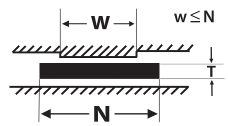
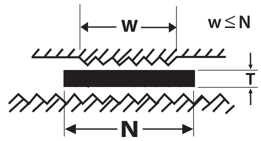
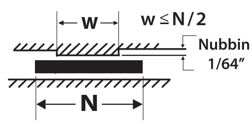
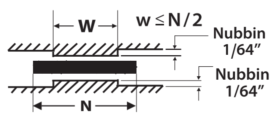
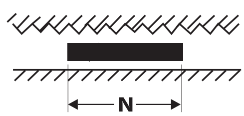
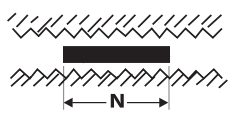
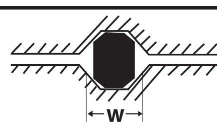
- A_b = Actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, square inches
- A_m = Total required cross-sectional area of bolts, taken as the greater of A_{m1} or A_{m2}, square inches
- A_{m1} = Total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions
- A_{m2} = Total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket sealing
- b = Effective gasket or joint-contact-surface seating width, inches (Table 2)
- b_0 = Basic gasket sealing width, inches (Table 2)
- G = Diameter at location of gasket load reaction (Table 2)
- m = Gasket factor (Table 1)
- N = Width, in inches, used to determine the basic gasket seating width b_0, based upon the possible contact width of the gasket (Table 2)
- P = Design pressure, pounds per square inch
- S_a = Allowable bolt stress at ambient temperature, pounds per square inch
- S_b = Allowable bolt stress at ambient temperature, pounds per square inch
- S_g = Actual unit load at the gasket bearing surface, pounds per square inch
- W_{m1} = Required bolt load for operating conditions, pounds
- W_{m2} = Minimum required bolt load for gasket seating, pounds
- y = Gasket or joint-contact-surface unit seating load, minimum design seating stress, PSI (Table 1) pounds per square inch

Table 1 - Gasket Materials and Contact Facings

Gasket Factors (m) for Operating Conditions and Minimum Design Seating Stress (y) **NOTE:** This table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table 2-5.1. The design values and other details given in this table are suggested only and are not mandatory.					**Refer to Table 2-5.1**	
Gasket Material		**Gasket Factor m**	**Min. design seating stress y (psi)**	**Sketches and Notes**	**Use Facing Sketch**	**Use Column**
Self-Energizing types O Rings, Metallic, Elastomer other gasket types considered as self-seating		0	0	---	---	---
Elastomers without fabric. Below 75 Shore Durometer 75 or higher Shore Durometer		0.50 1.00	0 200		1 (a, b, c, d) 4, 5	II
Elastomers with cotton fabric insertion		1.25	400			
Vegetable fiber		1.75	1100			
Spiral-wound metal, with nonmetallic filler	Carbon Stainless or Monel	3.00	10000		1 (a,b)	
Corrugated metal, double jacketed with nonmetallic filler	Soft Aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.50 2.75 3.00 3.25 3.50	2900 3700 4500 5500 6500		1 (a,b)	
Corrugated metal	Soft Aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.75 3.00 3.25 3.50 3.75	3700 4500 5500 6500 7600		1 (a,b,c,d)	
Flat metal jacketed with nonmetallic filler	Soft Aluminum Soft copper or brass Iron or soft steel Monel 4-6% chrome Stainless steels	3.25 3.50 3.75 3.50 3.75 3.75	5500 6500 7600 8000 9000 9000		1a, 1b, 1c*, 1d*, 2*	
Grooved metal	Soft Aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	3.25 3.50 3.75 3.75 4.25	5500 6500 7600 9000 10100		1 (a,b, c, d) 2, 3	
Solid flat metal	Soft Aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	4.00 4.75 5.50 6.00 6.50	8800 13000 18000 21800 26000		1 (a, b, c, d) 2,3,4,5	I
Ring joint	Iron or soft steel Monel or 4-6% chrome Stainless steels	5.5 6.00 6.50	18000 21800 26000		6	

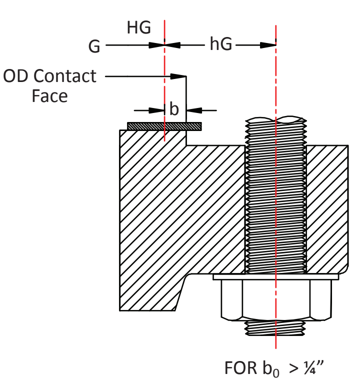
*The surface of a gasket having a lap should be against the smooth surface of the facing and not against the nubbin.

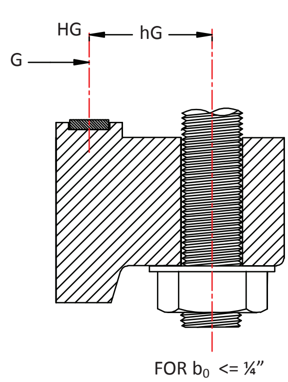
Table 2 - Effective Gasket Seating Width

Sketch #	Facing Sketch	Basic Gasket Seating Width, b_0	
		Column I	Column II
1(a)		$\frac{N}{2}$	$\frac{N}{2}$
1(b) See Note (1)			
1(c)		$\frac{w+T}{2}; \left(\frac{w+N}{4}\ \text{max.}\right)$	$\frac{w+T}{2}; \left(\frac{w+N}{4}\ \text{max.}\right)$
1(d) See Note (1)			
2		$\frac{w+N}{4}$	$\frac{w+3N}{8}$
3		$\frac{N}{4}$	$\frac{3N}{8}$
4 See Note (1)		$\frac{3N}{8}$	$\frac{7N}{16}$
5 See Note (1)		$\frac{N}{4}$	$\frac{3N}{8}$
6		$\frac{w}{8}$	...

Effective Gasket Seating Width, b

$b = b_0$, when $b_0 \leq$ ¼ in.; $b = 0.5\sqrt{b_0}$, when $b_0 >$ ¼ in.





NOTES:

(1) Where serrations do not exceed 1/64" depth and 1/32" width spacing, sketches (1b) and (1d) shall be used.

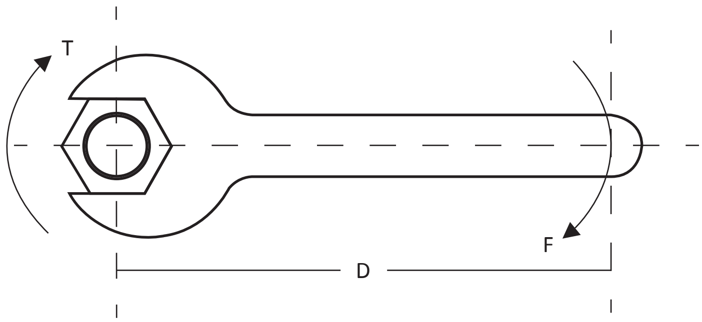
(2) The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

Technical & Design

Mechanics of Loading and Stressing a Bolt

When the nut is turned in a tightening direction by the applied torque (T), the flange, nut and bolt head surfaces are eventually put in contact along planes. This contact prevents further forward advance of the nut. Continued tightening is accomplished at the expense of metal deformations. While there are combination of metal deformations involved, normally only the tensile deformation in the bolt body is used to evaluate the resulting load.

When further torque is applied to the nut, the nut threads act on the bolt threads so as to pull the bolt up through the nut. This pull results in a lateral and axial deformation of the bolt body. The tensile force causing this deformation also causes an equally compressive load to be transferred from the nut and bolt head to the flanges.



A step by step analysis of how a loading device for threaded fasteners works is as follows:

F x D = T

Where: F = Force in inch-lbs or (N)

D = Distance in inches, feet or (m)

T = Torque in inch-lbs, ft-lbs, (Nm)

For instance, if you were to pull on the end of a 12" wrench, distance D, with a force of 50 lbs, force F, the resulting torque would be:

F x D = T

50 lbs x 12" = 600 inch-lbs (222 N x 0.3048 m = 68 Nm)

or

50 lbs x 1' = 50 ft-lbs

An important step, often difficult to understand, is how we go from torque to bolt tension force. For a given size bolt and nut, a scale can be attached and measure the force F, shown above, and with a fixed distance D, the torque can be calculated. During this operation the bolt elongates and the amount of elongation could be measured. Another bolt of the same size is mounted in a tensile testing machine. This machine measures bolt tension force versus elongation. In the former case torque versus elongation was measured. In the latter case bolt tension force versus elongation was measured. Thus, torque versus bolt tension force can be correlated.

It is also important to note that friction and variability of the lever arm length (where the wrench is gripped) are all variables that contribute to the inaccuracies of using hand wrenches. Accurately translating this torque number into compressive stress requires a good understanding of the condition of the mating surfaces and friction factors involved throughout the bolted connection.

Common Metals Description

304 Stainless Steel: An 18-8 (Chromium 18-20%, Nickel 8-10%) Stainless with a maximum recommended working temperature of 1400°F (760°C). At least 80% of applications for non-corrosive services can use Type 304 Stainless in the temperature range of -320°F to 1000°F (-196°C to 538°C). Excellent corrosion resistance to a wide variety of chemicals. Subject to stress corrosion cracking and to intergranular corrosion at temperature range of 800°F to 1500°F (427 to 815°C) in presence of certain media for prolonged periods of time.

304L Stainless Steel: Carbon content maintained at a maximum of .03%. Recommended maximum working temperature of 1400°F (760°C) with same excellent corrosion resistance as type 304. The low carbon content tends to reduce the precipitation of carbides along grain boundaries. Less subject to intergranular corrosion than type 304.

316 Stainless Steel: 18-12 Chromium-Nickel steel with approximately 2% of Molybdenum added to the straight 18-8 alloy, which increases its strength at elevated temperatures and results in somewhat improved corrosion resistance. Has the highest creep strength at elevated temperatures of any conventional stainless type. Not suitable for extended service within the carbide precipitation range of 800°F to 1650°F (427°C to 899°C) when corrosive conditions are severe. Recommended maximum working temperature of 1400°F (760°C).

316-L Stainless Steel: Continuous maximum temperature range of 1400°F to 1500°F (760°C to 815°C). Carbon content held at a maximum of .03%. Subject to a lesser degree of stress corrosion cracking and also to intergranular corrosions than type 304.

347 Stainless Steel: 18-10 Chromium-Nickel steel with the addition of Columbium. Not as subject to intergranular corrosion as Type 304 is subject to stress corrosion. Recommended working temperature ranges 1400 to 1500°F (760°C to 815°C) and in some instances to 1700°F (927°C).

321 Stainless Steel: 18-10 Chromium-Nickel steel with a Titanium addition. Type 321 stainless has the same characteristics as Type 347. The recommended working temperature is 1400 to 1500°F (760°C to 815°C). and in some instances 1600°F (871°C).

410 Stainless Steel: 12% Chromium steel with a maximum temperature range of 1200°F to 1300°F (649°C to 704°C). Used for applications requiring good resistance to scaling at elevated temperatures. Is not recommended for use where severe corrosion is encountered but is still very useful for some chemical applications. May be used where dampness, alone or coupled with chemical pollution, causes steel to fail quickly.

502/501 Stainless Steel: 4-6% Chromium and 1/2 Molybdenum alloyed for mild corrosive resistance and elevated service. Maximum working temperature is 1200°F (649°C). If severe corrosion is anticipated, a better grade of stainless steel would probably be a better choice. Becomes extremely hard when welded.

Alloy 20: 45% Iron, 24% Nickel, 20% Chromium, and small amounts of Molybdenum and Copper. Maximum temperature range of 1400 to 1500°F (760°C to 815°C). Developed specifically for applications requiring resistance to corrosion by sulfuric acid.

Aluminum: Its excellent corrosion resistance and workability makes it ideal for double jacketed gaskets. Maximum continuous service temperature of 800°F (427°C).

Brass: Excellent to good corrosion resistance in most environments, but is not suitable for such materials as acetic acid, acetylene, ammonia, and salt. Maximum recommended temperature limit of 500°F (260°C).

Carbon Steel: Commercial quality sheet steel with an upper temperature limit of approximately 1000°F (538°C), particularly if conditions are oxidizing. Not suitable for handling crude acids or aqueous solutions of salts in the neutral or acid range. A high rate of failure may be expected in hot water service if the material is highly stressed. Concentrated acids and most alkalis have little or no action on iron and steel gaskets which are used regularly for such services.

Copper: Nearly pure copper with trace amounts of silver added to increase its working temperature. Recommended maximum continuous working temperature of 500°F (260°C).

Hastelloy B®: 26-30% Molybdenum, 62% Nickel, and 4-6% Iron. Maximum temperature range of 2000°F (1093°C). Resistant to hot, concentrated hydrochloric acid. Also resists the corrosive effects of wet hydrogen chlorine gas, sulfuric and phosphoric acid and reducing salt solutions. Useful for high temperature strength.

Hastelloy C-276®: 16-18% Molybdenum, 13-17.5% Chromium, 3.7-5.3% Tungsten, 4.5-7% Iron, and the balance is Nickel. Maximum temperature range of 2000°F. Very good in handling corrosives. High resistance to cold nitric acid of varying concentrations as well as boiling nitric acid up to 70% concentration. Good resistance to hydrochloric acid and sulfuric acid. Excellent resistance to stress corrosion cracking.

Inconel 600®: Recommended working temperatures of 2000°F (1093°C) and in some instances 2150°F (1177°C). It is a nickel base alloy containing 77 % Nickel, 15% Chromium, and 7% Iron. Excellent high temperature strength. Frequently used to overcome the problem of stress corrosion. Has excellent mechanical properties at the cryogenic temperature range.

Incoloy 800®: 32.5% Nickel, 46% Iron, 21% Chromium. Resistant to elevated temperatures, oxidation, and carburization. Recommended maximum temperature of 1600°F (871°C).

Monel®: Maximum temperature range of 1500°F (815°C) containing 67% Nickel and 30% Copper. Excellent resistance to most acids and alkalis, except strong oxidizing acids. Subject to stress corrosion cracking when exposed to fluorosilic acid, mercuric chloride and mercury, and should not be used with these media. With PTFE (Polytetrafluoroethylene), it is widely used for hydrofluoric acid service.

Nickel 200: Recommended maximum working temperature is 1400°F (760°C) and even higher under controlled conditions. Corrosion resistance makes it useful in caustic alkalis and where resistance in structural applications to corrosion is a prime consideration. Does not have all the around excellent resistance of Monel®.

Titanium: Maximum temperature range of 2000°F (1093°C). Excellent corrosion resistance even at high temperatures. Known as the "Best solution" to chloride ion attack. Resistant to nitric acid in a wide range of temperatures and concentrations. Most alkaline solutions have little if any effect upon it. Outstanding in oxidizing environments.

Zirconium: Bio-compatible and non-toxic, excellent corrosion resistance to strong alkalies, most organic and inorganic acids and salt water environments where even the best stainless steels are not sufficient.

Duplex 2205: Dual Ferritic-Austenitic steel offers an excellent combination of both strength and corrosion resistance. Higher content of chrome and molybdenum provides superior resistance to general, pitting and crevice corrosion, while providing a higher yield strength over standard austenitic grades. Suitable for environments containing chlorides and hydrogen sulfide, dilute sulfuric acid solutions, organic acids.

AL6XN®: Of the 6 Moly group of materials, readily available in numerous forms. Superaustenitic stainless steel with excellent resistance to chloride pitting and crevice corrosion, and stress-corrosion cracking. Originally developed for seawater applications, offers good resistance to alkaline and salt solutions.

NOTE: Maximum Temperature ratings are based upon hot air constant temperatures. The presence of contaminating fluids and cyclic conditions may drastically affect the maximum temperature range

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