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HIGH VACUUM FLANGE TEST OF A BAKEABLE
ALUMINUM WIRE SEAL

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J. D. Redman

OAK RIDGE NATIONAL LABORATORY

operated by

UNION CARBIDE CORPORATION

for the

U.S. ATOMIC ENERGY COMMISSION

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Reactor Chemistry Division

HIGH VACUUM FLANGE TEST OF A BAKEABLE ALUMINUM
WIRE SEAL

J. D. Redman

MARCH 1964

OAK RIDGE NATIONAL LABORATORY
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HIGH VACUUM FLANGE TEST OF A BAKEABLE ALUMINUMWIRE SEAL

J. D. Redman

ABSTRACT

As a part of the development work associated with the design of vacuum components for the Thermonuclear Project, a program was initiated for the testing of high vacuum bakeable flange and gasket designs for leak-tight sealing requirements. The flange and gasket components, chosen for the initial study because of the simplicity in design and low yield point of the gasket material, consisted of the fully annealed, type 1100, aluminum alloy compressed between plane flange surfaces which had a 32μ in-rms finish. Leak-tight sealing of the 99% aluminum, 10-inch wire was found to be dependent on the yield point and creep properties of the alloy. It was demonstrated, (1) that sealing was accomplished when the bolt loaded flange exceeded the yield point of the strain hardened gasket, and (2) the life of the gasket was dependent on the rate of deformation during thermal cycling between room temperature and 250°C . It was possible to maintain a satisfactory vacuum seal [$<10^{-9}$ torr liter $\cdot$ sec $^{-1}$.(gasket in) $^{-1}$] for more than 18 thermal cycles with four hours per bake at the elevated temperature. Initial sealing of this 0.086-in diameter gasket was achieved with approximately 0.028-in gasket deformation and a force of 1500 lbs (linear in) $^{-1}$ or ($7\frac{1}{2}$ tons $\cdot$ in $^{-2}$).

The maximum loading force required to achieve a seal (i.e., to cause plastic flow) was approximately $8\frac{1}{2}$ tons·in⁻², or 3000 lbs (linear in)⁻¹ of gasket.

INTRODUCTION

An experimental program, in support of the ORNL Thermo-nuclear effort, has been initiated to determine the reliability and reproducibility of flange closure as a function of flange and gasket design for optimum performance in the high vacuum region.* This report describes the facility and procedure for such testing and the results of testing wire gaskets of type 1100 wrought aluminum alloy between plane flange surfaces. In this study quality of the vacuum seal was determined at pressures to 1.5×10^{-8} torr (mm Hg) on repeated closure and with thermal cycles between room temperature and 250°C as functions of torque applied, bolt stress, and gasket strain.

EXPERIMENTAL

Test Facility

The facility, shown in Fig. 1, consisted of a 12-liter test chamber pumped by a 6-inch diffusion pump (NRC No. HS6-1500) fueled with Dow Corning 705 oil. The test chamber was protected from backstreaming pump oil by a liquid nitrogen baffle. The chamber was a spool and hat flange of 9.5-inch

*This program is conducted in cooperation with the Engineering and Design Section of the ORNL Thermonuclear Division.

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Fig. 1. Bakeable Flange Test Facility

diameter stainless steel pipe joined at the test flanges. The chamber was sealed by a 30-mil, 10.75-inch diameter, gold wire to a 3.5-liter water cooled base flange spool assembly. The gold wire, a shear deformation type, was shown to seal on the three occasions tested on bolt loadings to 40 ft-lbs while thermally cycled between room temperature and 150°C.

The test chamber could be baked to 400°C with the gold seal insulated. Heating was accomplished by a removable furnace insulated with 2-inch thickness of marinite and heated by a 3.5 Kw calrod. Temperatures were monitored on (1) the outside wall of the "hat", (2) in a 1-inch deep thermowell in the test flange, and (3) in a similar thermowell in the gold-gasketed base flange.

Pressures within the system were measured by a Bayard-Alpert glass enveloped gauge connected to the "spool" assembly below the heating area. The base pressure of the system (total volume of the test chamber: 15.5 liters) was 1.5×10^{-8} torr.

Flange Assembly

Gaskets were tested between plane flanges, Fig. 2, finished to surface of 32 μ in-rms which terminate the spool and hat sections of the test chamber. These flanges were of 7/8-inch thickness and 12-7/8-inch diameter and were drilled to accommodate eight, 1/2-inch diameter bolts. Standard 304 stainless steel, 1/2-13NC hexheaded bolts and 304 stainless

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Fig. 2. Detail of Flange-Gasket Assembly

steel nuts were used without washers or lubrication. Four of the bolts spaced alternately around the flange, were machined to provide a 10° cone at the end and smooth level heads to permit micrometer determination of elongation under stress. Some tests simulated use of 16 bolts by addition of 8 "C" clamps stressed to an equilibrium tension with the bolts.

The gaskets were fabricated from O-temper, type 1100 aluminum alloy wire of 86-mil diameter. The wire gaskets (specified minimum 99% Al and maximum 1% of Fe and Si) were formed and resistance welded into 10-1/8 inch diameter rings.

Procedure

The test procedure consisted in determining the lowest bolt loading which would produce a seal for successive joint closures at room temperature and during thermal cycling to 250°C . The bolts were tightened alternately in pairs 180° apart with calibrated torque wrenches which were periodically checked for precision.

Quality of the seal at room temperature was determined by enclosing the gasket area in a plastic bag, flooding it with helium, and measuring the leakage into the system with a Veeco Model MS9 leak detector. This leak rate measuring device as installed had a sensitivity of 5×10^{-9} torr liters/sec (or 7×10^{-9} atm cc/sec). Quality of the seal at elevated temperature was similarly monitored after flooding the furnace volume with helium.

Measurements of bolt elongation were made by use of a micrometer on the four special bolts; such measurements agreed to ± 0.0002 inch for the four bolts under conditions of average bolt loading. Measurements of gasket deformation (to ± 0.002 inch) were consistently made at 4 points 90° apart on the bolt circle by centering the micrometer on $1/16$ inch ball bearings partially imbedded in the flange at those points.

The furnace was capable of raising the assembly temperature by only $3^\circ\text{C}/\text{minute}$. Even with this heating rate, however, the bolt temperature frequently exceeded the gasket temperature by 50° during the early portion of the baking cycle. This thermal differential did not apparently contribute to bolt relaxation or to failure of an established seal. Thermal differential during the cooling portion of the thermal cycle did not exceed 15°C , with a cooling rate no greater than $1^\circ/\text{minute}$. A thermal differential between the bolts and the gasket would tend to shorten the life of the gasket.

TEST RESULTS

In the curves shown are the results from 8 gaskets tested for quality of closure as a function of bolt loading and gasket deformation for a variety of operating conditions. Also a number of gaskets were prepared for hardness and compressive strain tests performed by the Y-12 Metallurgical Services Section and are included in Figs. 4 and 5. Since the test flange assembly was designed for $1/2$ -inch diameter bolts most

of the tests were limited to a bolt torque of 75 ft-lbs which produced a total bolt stress less than 35,000 psi. In test AC 8, a torque of 85 ft-lbs did not exceed the bolt proof strength, since there was no permanent elongation. Although a leak-tight seal could be maintained more than 18 thermal cycles, the bolting assembly was incapable of supplying sufficient stress to reseal a gasket for multiple closures. The failure of the seals could not be correlated with gasket imperfections, or flange deformation at bolting positions.

It was found, Fig. 3, that when the flange coupling was loaded sufficiently to seal, there was a gasket deformation of 27.5 ± 3.5 mils. Deformation values cited are transient (instantaneous) and not permanent. Measurements of the gasket deformation made at the conclusion of each test were generally 3 or 4 mils less than the final instantaneous value measured under load. In order to obtain the force necessary to deform the gasket sufficiently, a bolt loading of 75 ft-lbs for eight bolts was adequate in some cases. In others a seal was not obtained until the gasket was annealed during the baking cycle. Between gaskets there was a variation of ± 3 to 6 mils from the average deformation as a function of torque and a variation of torque for the same deformation of $\pm 20\%$. Torque values are nominal only since a variation of $\pm 30\%$ has been observed by others for the stressing of commercial nuts and bolts.¹ To obtain a deformation of 31 mils, the maximum necessary for leak tight sealing, a bolt torque

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Fig. 3. Gasket Deformation as a Function of Bolt Loading and Helium Leak Rate as a Function of Deformation

of 115 ft-lbs may be required for eight bolts, assuming the deformation-torque relation to remain essentially linear. This would require a bolt stress 115% greater than that available using the SS 304, 1/2 - 13NC bolt.

Bolt elongation measurements permitted calculation of axial force per unit stress area of the stressed bolts according to Hooke's law and were related to the gasket loading area for the corresponding compressive stress on the deformed gasket. In Fig. 4 are shown stress-strain curves comparing experimental data points from the bolt loaded flange tests, prior to baking, with those obtained from direct thrust compressive strain tests. The direct thrust tests were made on three gaskets using a tensile-compression machine with a 1-inch ram and a strain gage for correlating instantaneous deformation. The data from the tensile-compression machine showed a break in the yield stress curve at 28 mils instantaneous deformation corresponding to a stress of $6.9 \text{ tons} \cdot \text{in}^{-2}$. The yield strength for aluminum 1100 with an H 12 temper determined by the 0.2% offset method is $7.5 \text{ tons} \cdot \text{in}^{-2}$. The gasket material, initially an O-temper with a yield strength of 5,000 psi,² was strain hardened with cold-working to a high temper. In the three graphs shown in Fig. 5, changes in the average stress of the test gaskets were correlated with temper and strain hardening as a function of torque. The data taken from the Metals Handbook,² shown in the middle graph, was for a 33-mil aluminum wire.

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Fig. 4. Comparison of Stress-Strain Curves for Direct Thrust Test and Bolt Loaded Flange Tests

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Fig. 5. Stress on Gasket Near the Yield Point as
Related to Strain Hardening

The hardness tests (performed on gaskets prepared in the test cell by cold working from 0 to 85 ft-lbs torque, by heat treating only, and by heat treating followed by cold working) were superficial hardness tests on the Rockwell T-scale using a 1/16-inch diameter ball with a 15 Kgm load. Due to the softness of the material the hardness numbers were reliable only for comparative purposes, but were sufficient to show that the cold-worked gasket strain hardened under 85 ft-lb bolt loading to a temper corresponding to the H12 temper of the alloy. Aluminum 1100 with an H12 temper has a yield strength of $7.5 \text{ tons} \cdot \text{in}^{-2}$ which was approximately the stress necessary for the 28-mil gasket deformation shown in Fig. 4. Gaskets annealed by the flange test baking cycle had the lower temper and a yield strength of $1-2 \text{ tons} \cdot \text{in}^{-2}$ as determined by the residual axial bolt force after baking. When annealed gaskets were restressed they again became strain hardened. No gasket was strain hardened to a temper greater than that of H12 temper.

Test gasket AC8 was thermally cycled for a total baking time at 250°C of 72 hours, and is described in detail. For this test eight new SS 304, 1/2-13NC bolts were installed. No evidence of thread relaxation or permanent deformation of the bolts could be observed. A graphic history of this gasket, typical in many respects, is shown in Figs. 6 and 7. Prior to baking, the gasket failed to seal because of a leak at the weld, but it subsequently sealed near 135°C during the initial bake.

Fig. 6. Deformation and Leak Rate as a Function of Thermal Cycling Between Room Temperature and 250°C

Fig. 7. Stress-Strain History of Test Gasket AC8

Initially the deformation of the gasket, Fig. 6, was 4 mils per bake or 3.5% per hour and required that the bolts be restressed prior to each bake. After six 4-hour baking periods and about 65% reduction in gasket thickness, the deformation diminished to an average of 2.0% per hour during baking. Eventually, due to the increase in area of the gasket in contact with the flange surfaces, the critical stress of $7.5 \text{ tons}\cdot\text{in}^{-2}$ necessary to seal the strain hardened gasket could not be maintained by 8 bolts loaded to 85 ft-lbs. Eight "C" clamps were installed alternately about the flange with the standard bolts and stress was applied in such a manner that similar axial loading was maintained on both classes of bolts. The 16 bolts were restressed to 85 ft-lbs for each thermal cycle in the primary deformation area, and stress on the gasket was maintained between 7.5 and $8.5 \text{ tons}\cdot\text{in}^{-2}$. In the minimum deformation area there was less strain hardening from the facts that very little bolt restressing was necessary and the yield strength remained below the diminishing stress value. After 72% reduction of the gasket thickness there was no further bolt restressing. Small, almost immeasurable, creep continued which was apparent in a bolt relaxation of about 3.5 mils in 7 cycles, and a slowly increasing leak rate. If the bolts were restressed to maintain a constant stress on the gasket the stress-strain data points should fall along the broken curve in Fig. 7. The tests were concluded after 18 cycles assuming the seal would fail when the stress had

decreased to a value lower than the yield point of the fully annealed gasket.

If thermal cycling of AC8 were to continue the gasket would continue to deform either with a continuation of the final creep rate or, at an accelerated creep rate, to fracture. In case of the former, the creep would be expected to continue until the closure failed because the increasing gasket area would be so large that there would be insufficient bolt loading capability. If the curve in Fig. 7 is projected to the yield strength of the fully annealed alloy, 5000 psi, the curve intersects this value at 87.5% reduction or 10.7 mils. Stress calculation for this area with the axial loading force of the bolts using the average elongation values also yields 5000 psi, assuming there would be no bolt relaxation. Actually, as the gasket creeps there is bolt relaxation, requiring restressing.

SUMMARY AND DISCUSSION

In the tests performed, which were limited to only one baking time and temperature variable, it is apparent that the sealing characteristics of the nominal 10-inch diameter gasket were related to the mechanical properties of the aluminum material. For a leak tight seal of the 86 mil wire between flat flange surfaces, procedure and bolt loading requirements such as bolt size, grade, and number must satisfy two conditions: (1) The bolt force available must be sufficient

to seal prior to thermal cycling and also be sufficient to overcome strain hardening corresponding to at least an H12 temper. (2) The force must be adequate to maintain a seal during thermal cycling while the gasket is continually deforming. The first condition was satisfied by loading the flange until the yield strength of the strain hardened gasket, about $7.5 \text{ tons}\cdot\text{in}^{-2}$, was reached. The yield strength for the O-temper alloy, $2.5 \text{ tons}\cdot\text{in}^{-2}$, was reached with about 5% reduction of the gasket. This reduction was insufficient for sealing. Under additional stress, the O-temper alloy strain-hardened to H12 temper, yield strength $7.5 \text{ tons}\cdot\text{in}^{-2}$. Direct thrust compression tests showed a break in the yield stress curve of the cold worked material at 28 mils deformation for a stress of $6.9 \text{ tons}\cdot\text{in}^{-2}$. A stress-strain curve for the average of the experimental points indicated a 28-mil deformation for a stress of $7.5 \text{ tons}\cdot\text{in}^{-2}$. Stress values obtained from the bolt tension as measured by elongation per unit area of instantly deformed gasket were in good agreement with the direct compression test values and the reported yield strength for the H12 temper alloy.

The maintenance of a seal during thermal cycling was complicated by the bake cycle dependent deformation of the gasket. In the primary or steeper region of the deformation curve, bolt relaxation following baking required restressing the bolts prior to each bake to maintain a constant stress of 7.5 to $8.5 \text{ tons}\cdot\text{in}^{-2}$. Greater bolt loading was required than prior

to baking to overcome the increasing effective area of the gasket until reaching the minimum creep region where there was less strain hardening and successively lower yield points.

A projection of the data indicates that a tight seal ($<10^{-9}$ torr liter/sec/inch) can be maintained until the applied stress drops below the yield point of the fully annealed gasket. Assuming a continuance of the average minimum deformation of 0.6 mils per cycle for a constant bolt loading stress, corresponding to the average maximum bolt elongation of .0027-inch, a 5000 psi gasket stress would be reached after 77 mils deformation (90% reduction) or a total of about 38 four-hour bake cycles. The experiment has shown that at least 18 four-hour 250°C bakes can be performed using a bolting assembly capable of producing a maximum stress of 7.5 to 8.5 tons $\cdot\text{in}^{-2}$ for a 60% (51.5 mils) gasket reduction. Assuming a bolt strain of 0.133% (the average in these tests) it is possible to determine the stress area of a bolt with the necessary axial stress potential from Hooke's law for the deformed cross sectional area of the gasket.

$$A_{\text{bolt}} = \frac{S \cdot A_G \cdot L_B}{Y \cdot \Delta l_B}$$

S = Stress on gasket - 17,000 lbs $\cdot\text{in}^{-2}$

A_G = Cross sectional area of the gasket
per bolt for a 10-in diameter
wire

$L_B / \Delta l_B$ = 1/bolt strain

Y = Modulus of elasticity for SS 304,
2.9 x 10⁷ lbs $\cdot\text{in}^{-2}$

An approximation of torque for a given bolt tension can be obtained from the equation^(1,4)

$$T(\text{in-lbs}) = K D S$$

where K = the torsional constant, which is about 0.2 for the lowest coefficient of friction (0.15 for dry driven semi-finished bolts); D = bolt diameter; and S = bolt tension or clamping force for SS 304 bolts with a bolt loading of 63,700 psi. There is variation in the coefficient K depending on scale on the bolt and erosion of the flange surface by the nut-bearing surface (see Appendix 1). Bolts having the necessary stress area for 8- and 16-bolt assemblies are listed in Table 1.

Table 1. Comparison of Bolt Requirements for Flange Closure and Recommended Bolts

	Stress Area (in ²)	Max. Torque (ft-lbs)
For 8-bolt assembly	.3000	115 ^a
For 16-bolt assembly	.1500	85 ^a
3/4-10 NC bolt	.3340	265
9/16-12 NC bolt	.1820	110

^aBased on experimental evidence

For convenience in determining instantaneous cross sectional area of the gasket during testing, the effect of slip-plane friction between the gasket and flange surfaces was neglected. Under compression that part of the gasket

touching the flange surfaces is prevented from expanding laterally due to friction, and a bowing in the middle occurs. The degree of bowing and consequently the stress is proportional to the coefficient of friction. In a paper³ on friction in plane-strain compression tests, stress-compression strain in the region of 7-13 tons·in⁻² was reported as a function of interfacial friction of fully annealed aluminum wire. The coefficient of friction with no lubrication was $0.8 \pm .07$, depending on the surface conditions. Frictional effects of air oxidation and freshly polished surfaces were reported observed. Also there was a marked increase of friction with increased die pressure, and the accumulation of metallic debris with no lubrication, which resulted when lubricated, in lower friction. This work suggests the possible significance of frictional conditions in high vacuum flange closure studies.

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APPENDIX 1

The equation, $\text{Torque} = KDS$, is more convenient for use in the field than is the equation derived from Hooke's law. However, it is subject to large variations. When a bolt is wrenched two stresses are applied: (1) torsional, and (2) axial or tensile. The K constant, or torsional coefficient, is difficult to determine because of its dependence on the coefficient of friction which may vary by 60%. The K factor for dry-driven, semi-finished, steel nuts and bolts is usually given as 0.2,^(1,4) and is determined by the equation:⁴

$$K = \mu_B \frac{R_B}{D} + \mu_T \frac{R_T}{D} \sec B + \frac{R_T}{D} \tan C$$

$$K = K_1 + K_2 + K_3$$

where μ_B = coefficient of friction at the bearing face of the nut; D = nominal bolt diameter; μ_T = coefficient of friction at the thread contact surfaces; R_B = effective radius of action of frictional forces on the bearing face of the nut; R_T = effective radius of action of frictional forces on the thread surfaces; B = thread half-angle (30° for the 60° thread); and C = helix angle of the thread. The coefficients, K_1 and K_2 , represent the torque required to overcome friction at the bearing face of the nut, and at the contact surfaces of the threads. The frictional coefficient generally used (which was the lowest value found in the literature) was $\mu = 0.15$. Coefficient K_3 is related to the mechanical advantage of the bolt and accounts for about 10% of the torque.

Table 2 shows the wide variation in the literature of values for the coefficient of sliding kinetic friction of steel on steel with dry surfaces. Nearly all the values reported appeared to be for mild or case hardened steel. In one case¹⁰ was reported a value of 1-1.5 for austenitic (300 series) stainless steels, and a $\mu = 0.58$ for the static friction of mild steel. Since static friction is somewhat higher than kinetic friction the μ for kinetic friction would be near the values reported by others.^(7,8) In Table 3 is shown the wide variation in bolt tension for the same torque for various values of μ given in Table 2, using a dry 1/2 - 13 bolt.

In addition to the uncertainty of μ for known surface conditions of stressed new bolts is the uncertainty of μ for constantly changing surface conditions during thermal cycling and bearing surface lapping. Experimental evidence of changing surface conditions is shown in Table 4. Bolt tension determined by the torque equation, using the conventional value of 0.2 for K, varied by as much as 50 per cent from the more nearly correct values determined by the equation based on Hooke's law. The variation is correlated with changes in the frictional condition of the bearing surfaces. The lowest μ encountered was 0.20 for well-lapped threads free of scale and nut bearing surfaces when washers were used. On the basis of these results the K coefficient during operation varied between 0.273 and 0.400.

Table 2. Coefficient of Sliding Kinetic Friction of
Steel on Steel

Coefficient of Friction (μ)	Lubrication	Authority
0.15 - 0.25	Dry	(5)
0.15	Dry	(1), (10)
0.42	Dry	(6), (7)
1 - 1.5 ^a	Dry	(9)
0.11 - 0.19	Heavy machine oil	(8)
0.08	Grease, Lard	(6)
0.03 - 0.15	Heavy machine oil and graphite	(8)
0.10	Turbine oil and graphite	(6)

^aFor austenitic stainless steels

Table 3. The Effect of Frictional Variation on
Calculated Bolt Tension

Torque (Ft-Lbs)	R_B/D	R_T/D	$\mu_B = \mu_T$	K	1/2 - 13NC Bolt Tension (lbs)
85	.0700	.0440	0.15	0.204	10,200
85			0.25	0.324	6,300
85			0.42	0.53	3,850
85			1.0	1.23	1,650

Table 4. Variation Between Torque and Bolt Strain in Determining Bolt Tension Due to Changes in μ

Test No.	Bolt Tension (lb·bolt ⁻¹) as determined by		Percent Deviation Between Calculations	Percent Average Deviation	μ for Exp'l. Agreement	Condition of Bearing Surfaces
	Torque	Bolt Strain				
AC 1	2500	4200	40	39		New bolts, unlapped, bright surfaces
	3200	5400	41			
	3600	6600	45			
	5900	8400	30			
AC 3	3900	5400	28	27	0.20	Used, well-lapped bolts, bright thread surfaces, some erosion of nut bearing mating surfaces
	5500	7200	24			
	6400	9000	29			
AC 3	1600	2400	33	38		After 2 bakes some scale on thread surfaces
	5200	9000	42			
AC 6	2700	4800	44	46		After 6 bakes scale buildup on threads and further flange surface erosion
	3400	6000	43			
	4500	9000	50			
AC 8	3900	6000	35	35		New bolts, unlapped surfaces, eroded flange surfaces at nut bearing faces
	5200	7800	33			
	5700	9000	36			
AC 8	5200	9600	46	43		After 2 bakes
	5500	9000	39			
AC 8	5500	9600	43	43		After 6 bakes
AC 11	2400	4800	50	49	0.31	Same bolts as used in AC 8, after 18 bake cycles and 1-3 mil erosion in flange surfaces
	3460	6600	48			
AC 11	3000	4800	37	38	0.31 between threads and 0.21 bearing surfaces	Same conditions as in AC 11 above but with smooth, flat washers on the nut bearing faces
	3700	6000	38			
	3900	6600	41			
	4100	6600	38			

APPENDIX 2

A comparison of the tensile strengths for a few of the different types of steel bolts is given in Table 5. Carbon steel bolts (medium-carbon steel, 0.55 per cent C maximum above grade 2, and low-carbon, 0.28 per cent C, for grade 2) are recommended for use only below 700°F.² They are graded according to tensile strength developed by different heat treatments and are identified by head markings. Also shown are some of the stainless steels with tensile strengths corresponding to the carbon steels. Some stainless steels can be used at elevated temperatures to 1000°F. There is no bolt head marking system for the stainless steels. A complete list of the steels pretreatment, and tensile strength is given in The Metals Handbook, Vol. 1, 8th Ed., American Society for Metals, 1961.

Table 5. Comparison of Tensile Strength for Different Types of Steel^{2,8}

S A E CARBON STEEL			STAINLESS STEEL		
Grade	Bolt Head Identification	Tensile Strength psi min.	Grade	Tensile Strength psi min.	Type
2	No Mark	69,000	405	70,000	Ferritic
3		110,000	502	70,000	Martensitic
5		120,000	430	75,000	Ferritic
6		140,000	301	110,000	Austenitic
8		150,000	304	85,000	Austenitic
			316	85,000	Austenitic
			431	125,000	Martensitic
			301 $\frac{1}{2}$ -hard temper	150,000	Austenitic
			201 full hard	185,000	Austenitic

Ferritic: 11-27% Cr; no nickel; magnetic; little corrosion resistance; can be cold worked.

Martensitic: 4-18% Cr; 0-2.5% Ni; magnetic; can be heat treated and work hardened.

Austenitic: 16-19% Cr; 3.5-13% Ni; non-magnetic; can be cold worked; best degree of corrosion resistance.

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