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TOPICAL DISCUSSION.

NOTES ON THE EFFICIENCY OF SCREW BOLTS AND THEIR APPLICATION TO STEAM PIPE FLANGES AND OTHER USES.

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The writers, when considering the question of proportioning the area of flange surface, and the number and diameter of bolts for fastening together the segments forming the body of a 12 ft. diameter vacuum pan, were somewhat surprised to find such a wide variation in the proportion given by various makers and authors for the size and number of bolts, and the diameter of flanges for steam and other pipes; and as many of the tables of standard proportions were examined (and would be referred to later) it was thought that a review of the current practice with some notes on the designing of flange joints, would be of interest to the Association.

The writers, therefore, propose to bring this question before you in the form of a topical discussion and exchange of data.

It was not generally considered a difficult matter to make a flange joint steam tight, or indeed a flange joint of any kind, water or air tight; the fact, however, was that, unless the bolts in a flange joint be designed to withstand the stresses caused by the expansion or bending of the pipe, due to heat or other causes, it was only a question of time when it would begin to leak and cause trouble, loss, and annoyance; as no doubt many of the members had found by experience.

The question of making a satisfactory flange joint with either cast iron, wrought iron, or copper pipes, mainly depended on the number and diameter of bolts used, and on the thickness of flange and method of reinforcing the pipes at its junction with the flange. With reference to the use of bolts, it could be safely said that the screw was one of the most mechanical powers, but in the V thread form it had a very low efficiency, its low efficiency however as a mechanical power gave it the important property of not overhauling or slacking back when loaded or screwed up, and it was, therefore, specially suited for bolting together steam pipe flanges and other parts of machinery. The efficiency of any mechanism was measured by the ratio of its useful work to the work consumed, and as the screw was used in practice as a means of performing work and for the purpose of binding and holding together various parts of machines, pipes, and structures. In the former case the thread was made square in section. In the latter case the thread is generally V shaped, and the pitch was determined by recognised standards. Very few experiments had been tried to determine the efficiency of the ordinary screw bolt as usd in practice for screwing up joints and other parts of machinery.

The following paper by James McBride, read before the A.S.M.E., entitled "Experiments on a Screw Bolt," showed that the efficiency of the ordinary screw bolt was much lower than was usually assumed.

The paper was quoted in full as it contained data of considerable value to mechanical engineers.

Mr. McBride says:—

"It goes without saying that the screw is one of the most useful mechanical powers, and among its many qualities is not that of overhauling when loaded. A large part of its efficiency as a mechanical power is sacrificed to obtain this valuable property; but aside from the necessary loss to accomplish this result I doubt if it is generally known how much of the total power applied to turn the nut on an ordinary screw bolt is lost by frcition.

I am quite sure that I did not, and a number from whom I have asked for information, and who, I suppose, ought to know, apparently knew as little about it as myself, and it was to get some practical knowledge on the subject that these experiments were made.

The experiments were made with an ordinary two inch screw bolt, such as can be bought anywhere in the market, and it was not specially prepared for the occasion.

The pitch of the thread was 22/100 in. ($4\frac{1}{2}$ threads to the inch) and of the standard shape. The nut was not faced, and had the flat side to the washer, which was of malleable iron not faced. The surface of the nut and washer in contact, as well as the threads of the nut and bolt, were well lubricated with lard-oil.

The nut was a good fit, and when not loaded was easily run up and down the bolt with the fingers.

The nut was placed 4 ft. 6 in. from the floor, about breast high, and rested on a cross beam supported on wooden trestles. The bolt passed through the floor, and was loaded at its lower end with about 750 lbs. dead weight. To ascertain how much a man could pull on a nut thus placed, a nut was placed 4 ft. 6 in. from the floor on a bolt fixed to the side of a plank, and on one end of a soft and very pliable rope was attached to the handle of a wrench and passed over a pulley 9 in. in diameter, the top of which was level with the nut, and immediately in front of the operator.

To the other end of the rope were attached weights, which rested on the floor. The operator was required to grasp the wrench with one hand on each side of the rope where it was fastened to the wrench, and turn the nut on the bolt, which he could only do by raising the weight from the floor. This apparatus was designed to compel the operator to keep the wrench on a level with the nut or nearly so, thus making the conditions as nearly as possible the same as when he applied his power to turn the nut on the loaded bolt.

It was found after many turns by different men that they could by this device raise weights varying from 182 from the floor by a very powerful man.

The following are the results of the trials with the weights:—

Clark	267 lbs.
Sweeney	227 ,,
Fay	182 ,,
Rogers	208 ,,
Harward	182 ,,

When the wrench was applied to the nut on the loaded bolt, it was found that the men could just move the nut with the following length of lever:—

Clark.. ..	8 in. radium	—a circumference of	50.26 in.
Sweeney ..	13 in.	,, ,, ,,	81.68 in.
Fay	$14\frac{1}{2}$ in.	,, ,, ,,	91.10 in.
Rogers	13 in.	,, ,, ,,	81.68 in.
Harward	$14\frac{1}{4}$ in.	,, ,, ,,	89.53 in.

The length of the wrench was measured from the centre of the nut to a point between the hands, and was taken as the shortest length of wrench with which the nut could be moved, and the operator was allowed to jerk or surge as he liked with both feet on the floor, and keeping the wrench as nearly horizontal as possible.

The theoretical value of a 2 in. screw bolt with a .22 in. pitch, and under the foregoing conditions, is as follows—

Clark	.22 in. : 50.26 :: 267 : 60,997 ..	12.290/o
Sweeney ..	.22 in. : 81.68 :: 227 : 84,279 ..	8.990/o
Fay	.22 in. : 91.01 :: 182 : 75,304 ..	9.950/o
Rogers	.22 in. : 81.68 :: 208 : 77,224 ..	9.710/o
Harward ..	.22 in. : 89.53 :: 182 : 74,064 ..	10.120/o
		5)50.960/o
		Mean, 10.190/o

It will be seen from these figures that the practical value was only a mean of 10.190/o of the theoretical, and that 89.810/o of the power applied to the wrench was absorbed by the friction of the nut on the washer, and on the threads of the bolt.

Since beginning this paper two cases have come under my observation which go to show how little is really known of the strains that are put upon screw bolts under various conditions.

One was that of some 7-8 in. bolts, 11/100 pitch (9 threads per inch), the nuts of which were being screwed up with a wrench 28 in. long, operated by two men; leaving out the element of friction, and assuming the combined force applied by the men to be only 10 lbs. (they being in a sitting position, and not able to exert their full force), they should have put a strain of 159,927 lbs. on the bolt, which is about seven times more than its tensile strength.

It is very evident that no such tensile strain was put directly upon the bolt, although a number of them were broken. The friction of the threads alone being sufficient to lock the two parts together so that the bolt was twisted off just below the nut.

The other was that of a 3 1-8 in. diameter, .25 in. pitch (4 threads per inch), the nut and its bearings were fell faced and oiled, the wrench was 42 in. long, having an eye at one end, to which was attached a double and triple pulley block and tackle, operated by four men. Four men with such a tackle can just raise 1,700 lbs., which applied to the end of the 42 in. wrench to turn the nut on the bolt should raise 194,452 lbs., which is about five times the tensile strength of the bolt.

As in the former case, no such strain ever reached the bolt, but assuming that 10 per cent. of it did, it will be seen that this bolt was strained so as to only leave a factor of safety of 2.1 to 1. in.

The following are the theoretical deductions on the efficiency of the screw:—

Trautwine says:—

"In practice the friction of the screw (which under heavy loads becomes very great) makes the theoretical calculations of but little value."

Weisbach says:—

"The efficiency is from 19 to 30 per cent. These figures can obviously only apply to square threaded screws, such as that in a screw-jack."

Lewis gives the following approximate formula for ordinary screws:—

Bolts with V threads:—

$$\text{Efficiency} = \frac{\text{Pitch.}}{\text{Pitch} + \text{Diar.}}$$

For a bolt in 1in diameter, having 8 threads per inch:—

$$\frac{.125}{.125 + 1} = 11\%$$

For a ¾in. bolt having 10 threads per inch:—

$$\frac{1}{.1 - .75} = 11{\cdot}76\%$$

For a 2in. bolt having 4½ threads per inch:—

$$\frac{.22}{.22 + 2} = 10\%$$

Professor Unwin gives three examples of the efficiencies of a V thread screw bolt:—

1st.—Bolt 1in. diameter, 8 threads per inch, 11.5 per cent. efficiency.

2nd.—Bolt 2in. diameter, 4½ threads per inch, 10.3 per cent. efficiency.

3rd.—Bolt 4in. diameter, 3 threads per inch, 8 per cent. efficiency.

It will be seen by a comparison of these figures that Lewis' and Unwin's formulae agree closely with the experimental data obtained by Mr. McBride.

A correspondent of the "American Machinist" describes an experiment with a differential screw-punch, which gave a very low efficiency.

The outer screw was 2in. diameter, 3 threads per inch.

The inner screw 1⅝in. diameter, 3½ threads per inch.

The pitch of the outer screw being 1/3in., and that of the inner screw 2/7in., the punch would advance in one revolution 1/3in. − 2/7in. = 1/21in.

Experiments were made to determine the force required to punch a 11/16in. hole in a plate ¼in. thick, the force being applied at the end of a lever 47¾in. long.

The leverage on the punch would be 47¾in. x 2 x 3.1416 x 21 = 6300 to 1.

The mean force applied at the end of the lever was 95lbs.; if there was no friction the total force would be 6300 x 95 = 598,500lbs. The force required to punch the iron, assuming a shearing resistance of 50,000lbs. per square inch, would be 50,000 x 11/16in. x 3.1416 x $\frac{1}{4}$ = 27,000lbs., and the efficiency of the punch would be $\frac{27,000}{598,500}$ = only 4.5 p.c., that is to say there was 95.5 p.c. of the power applied lost, with the larger screw only used as a means of forcing the punch through the plate. The mean force at the end of the lever was only 82lbs., the leverage in this case was 47$\frac{3}{4}$ x 2 x 3.1416 x 3 = 900.

The total force on the punch, including friction, is 900 x 82 = 73,800lbs., and the efficiency $\frac{27,000}{73,800}$ = 36·7%.

The screws were of tool steel, well fitted and lubricated with lard-oil and plumbago. This experiment shows that there is considerable loss in friction when the ratio of the purchase is too high as in the above example.

Having briefly referred to some of the experimental data, and also to some of the theoretical data on the efficiency of the screw as given by some of the standard authors on applied mechanics, reference will now be made to the application of the screw bolt to flange steam pipes and other classes of flange joints as found in current practice.

The first question to be discussed in this connection is the safe stress to which screw bolts of various diameters should be subjected to when being screwed up tightly as in making a flange joint or binding together the caps and brasses of main bearing or connecting rod end brasses.

The figures given by the different authorities which have been consulted are as under:—

Safe Working Stress of Screw Bolt.

D. K. Clark	3360lbs per sq in.	
Professor Unwin	6000lbs per sq. in.	for faced joints
Professor Unwin	3000lbs. per sq. in.	for rough joints and gaskets
Professor Jones (American authority).	6000lbs. per sq. in.	

Mr. Seaton points out that a bolt when being screwed up is subjected to two distinct stresses, viz., tension and torsion, and that the tensile or axial stress cannot be set up without at the same time producing the twisting stress.

Professor Unwin has demonstrated that tightening the bolt against a flange or load of any kind produces an increase of stress equivalent to 17 p.c. increase in the tension, and points out that there is therefore good reason in the practice of not using bolts of less than $\frac{3}{4}$in. diameter in joints that have to be screwed up hard

Table No. 1, on working stress in bolts by Seaton, is based on the relation.

Working stress per sq. in. = (are at bottom of thread) $\frac{5}{12}$ x C where C = 5000 for iron or mild steel and 1000 for Muntz or gun-metal.

For iron or steel bolts above 2in. diameter, and gun-metal or bronze ones above $3\frac{1}{2}$in. diameter, the moment of the twisting stress is so small proportionately that it may be neglected.

When comparing the various tables of proportions of bolts in steam pipes and other flanges the figures given in Seaton's table will be used.

The next question to be discussed is the length of spanner for the different sizes of bolts.

Let "D" = Diameter of Bolt.

Unwin	gives	length	=	12 to 15 D.
Seaton	,,	,,	,,	12 D.
Jones	,,	,,	,,	16 D.
Low	,,	,,	,,	13 D.

The length of spanner in this discussion will be taken at 16 D. (effective length = 16 D. – 2in.), and the pull on the spanner by one man as equal to 50lbs. The length of spanner for other sizes of bolts may also be taken at 16 D. – 2in , and the power applied to the end of the spanner should be fixed so that when screwed up the stress should be approximately that given in Seaton's table; the average pull of a man when using both hands on a spanner may be taken at 100lbs., and when using one hand about 50lbs.

Before examining the various tables of bolt proportions, attention may be directed to the discussion of a paper by Mr. Pike on steam piping in the transactions of the A.S.M.E., volume (xv.) page 544, where the following comparson of the number and size of bolts and dimensions of flanges occur.

Mr. Pearson, in discussing the paper, advocates the use of cast iron pipes as against wrought iron, and said he visited a large number of power plant stations to enquire into the current practice in the manufacture and erection of steam piping, and found all the pipes leaking more or less at the joints, and attributed this to the metal not being thick enough in the body of the pipe, and also to insufficient thickness of the flanges; and the number of the bolts in them being too few for securing the flange faces together with sufficient pressure, he gives the two following examples of a weak flange joint and of a joint that should give no trouble through leaking. In the latter example the faces of the joint are ground metal to metal, the pipes are of cast iron, the metal in the body of the pipe and in the flange of the heavier design seems unnecessarily thick.

Mr. Pearson, speaking in 1893, said:—"A year ago the pipes in use for 120 to 160 lbs. per square inch had the following dimensions:—Bore of pipe, 14in.; diameter of flange, 22½in.; thickness of metal in body of pipe, ¾in. to 1in.; number of bolts, 12; diameter of bolts, 1in. The dimensions he had adopted in fitting out the powerhouse steam pipes at Sehenectady are as follows:—Bore, 14in. diameter; diameter of flange, 22½in.; thickness of flange, 2½in.; thickness of pipe, 1¼in.; number of bolts, 20; diameter of bolts, 1¼in."

In making a comparison between the two types of flanges a steam pressure of 140lbs. per square inch is taken, being the mean between 120 and 160 lbs. per square inch. 14in. diameter = 154 square inch area x 140lbs. per square inch = 21,560lbs. stress separating the flanges due to steam pressure.

Area of flange, 22½in. diameter = 397.6 square inch, deducting the area of the bore of the pipe = 154 square inches, and the holes in the flange = 11.9 square inch, there is left the area of the joint face, ∴ 154 square inches + 11·9 = 165.9 square inches.

Then 397.6 square in. – 165.9 sq. in. = 231.7 square inch joint area of flange face, assuming the bolts to be stressed up to 3900lbs. per square inch.

Then the area at the bottom of the thread of a 1-inch diameter bolt equals .554 square inch x number of bolts in flange equals .554 x 12 equals 6.68 square inch area.

Then 6.65 square inch x 3900lbs. (safe stress of a 1in. bolt) = 25,935lbs. as the safe stress on the bolts.

The stress on the bolts caused by the steam pressure acting to separate the flanges is 154 square in. x 140 = 21,560lbs.

Then 25,935lbs. – 21,560 = 4375lbs. available for forcing the flange faces together when under steam pressure.

Therefore $\frac{4375}{231.7}$ = 18.8lbs. per square inch as the pressure on the joint face, which is about 13.5 per cent. of the steam pressure in the pipe, and the joint would not, therefore, be likely to remain tight very long.

In the pipes adopted by Mr. Pearson, the bolts in the flanges were $1\frac{1}{4}$in. diameter, requiring a $1\frac{3}{8}$in. clearing hole, and $1\frac{3}{8}$in. diameter = 1.485 square inch area x 20 = 29·7 square inches area ∴ 154 + 29·7 = 183·7 square inch and 397.6 – 185.7 square inch = 214 square inch as the area of the joint face. Sectional area of bolts at bottom of threads = 16.98 square inch x 4700lbs. (Table 1) = 79,806lbs. total stress available for forcing the flanges together when there is no steam pressure on them.

The stress on bolts caused by the steam pressure acting to separate the flanges is 21,560lbs.

Then 79,806lbs. – 21,560lbs. = 58,246lbs. available for forcing the flange faces together when under steam pressure ∴ $\frac{58,246\text{lbs.}}{214\text{ sq. in}}$ = 272lbs. per square inch as the pressure on the face of joint. It will be seen that in the latter example the pressure on the flang faces is nearly twice the working steam pressure, so that the joint would be perfectly steam tight, provided no excessive strain is put on the bolts, due to unequal expansion in the pipe.

It may be further noted that the second example of flange joint is pressed with $\frac{722}{18\cdot 8}$ 14.5 times more force than the first example of flange joint.

The question of steam pipe flanges has been further discussed in a paper recently read before the American Society of Mechanical Engineers, on "Pipe Flanges and their Bolts," by Mr. A. F. Nagle, who lays great stress on the effect of heat in causing elongations and transverse stress or flexure, and proposes a form of steam pipe that is designed to relieve the flange joints of all transverse stresse sdue to heat.

A report of this paper appeared in "Engineering," dated July 21st, 1900, and may be referred to for full particulars of the proposed design. Mr. Nagle, in the paper referred to, discusses the current practice of the leading American makers, and gives particulars of steam pipes up to 24in. diameter with their flanges and bolts. The pipes designed to carry a steam pressure of 250lbs. per square inch. In comparing the various pipe flanges, he assumes a fibre stress in the bolts of 10,000lbs. per square inch, although it is stated in the paper that considering the torsional stress in screwing up and other causes, that 7000lbs. per square inch is probably as great a stress as it is safe to put on the bolts. The latter figure agrees with that given in Seaton's table, which is based on the relation existing between the torsional and tensile strength of bolts of different diameters.

Table No. 2 contains full particulars of the sizes of pipes, diameter of flanges, size and number of bolts referred to in Mr. Nagle's paper.

Columns 1, 2, 3, 4, and 5 of the table are those used by the Champan Vale Company, with ordinary flanges, in which the jointing material covers the whole face of the flanges.

Column 5 shows the pressure per square inch on the face of the flanges, assuming the fibre stress in the bolts to be 10,000lbs. per square inch. If, however, the working fibre stress be taken from Seaton's table, viz., 3000lbs. per square inch for a $\frac{3}{4}$in. bolt, then the nett pressure on the gasket or jointing material between