the flange faces, fitted with ¾in. diameter bolts, should be reduced in the ratio of $\frac{3000}{10,000} = \frac{3}{10}$

For example, the 5in. diameter pipe has a nett pressure of 300lbs. per square inch between the flanges, then 300 x 3/10 — 90lbs. per square inch, whcih seems insufficient to keep the joint steam tight, even should no other stresses than those due to the steam pressure of 250lbs. per square inch come on the bolts.

The 12in. diameter pipe has 18 bolts, ⅞in. diameter in the flanges; the pressure between flanges is given as 236lbs. per square inch; but as the safe working stress of a ⅞in. bolt should be taken at 3400 instead of 10,000 lbs. per square inch,

Then $\frac{236 \times 34}{100}$ = 80.2lbs. per square inch.; and for the 16in. pipe, the pressure between the flanges would only be $\frac{168 \times 34}{100}$ = 57lbs. per square inch.

Mr. Nagle says that the pressures on the jointing material given in column 5 is small, and it is not surprising that, when slight transverse stress is produced by the expanding pipes, the joint is opened on one side and the flange leaks.

It should be noted that Mr. Nagle's remarks apply to the pressure on the flanges when the bolts are screwed up to a stress of 10,000lbs. per square inch, and would, therefore, apply with much greater force when the bolts are screwed up to the lower stress.

It should be noted that it is not the intensity of the pressure between the face of the flanges that will keep the joints tight when the pipes are subjected to transverse stresses, caused when heating them up or due to elongation. The diameter and number of bolts in the flanges should be such as will cause the metal in the body of the pipe to elongate without unduly stretching the bolts; or, in other words, the sectional area of the bolts should be sufficient to keep ample pressure between the flanges, and at the same time cause the body of the pipe to elongate under the transverse stresses.

With a view to obtaining greater pressures on the jointing materials a tongue and corresponding groove have been adopted in steam pipe flanges by the Chapman and Crane Companies, and while this device is an excellent one, and has the merit of locking the jointing material into a recess, and is therefore a safeguard against the joint being blown out, it will not keep the joint steam tight unless the bolt area is large enough to withstand the transverse stresses, which may come on them, without unduly stretching. Should the jointing material be of an elastic nature, the slight stretching of the bolts in the flanges, caused by transverse stresses, may not cause the joint to leak, as the material is securely locked in the recess, and its elastic nature will keep it in position, but should the joints be made of any kind of material that sets hard and is inelastic, the joints will certainly leak if the bolts be unduly stretched by transverse stress in the pipes.

Columns 7, 10, and 17 show the effective pressure per square inch on the narrow tongues formed in the flanges; assuming the bolts to be strained to 10,000lbs. per square inch, but as was explained when examining the pressure given in column 5, this pressure should be reduced in the ratio stated.

So far as making the joints steam tight, the pressure between the faces of the tongue in the flanges, given in column 17, should make that certain; as the pressure per square inch on the tongues is about $4\frac{1}{2}$ times the steam pressure in the pipes. There is not, so far as the writers are aware, any experimental data on the relation between the pressure per square inch, which should be put on the flange faces, and the steam pressure in the pipe to ensure the flanges being steam tight. It may, however, be taken as a safe empirical rule that the pressure on the jointing material due to screwing up the bolts should be at least equal in intensity per square inch of joint surface to the steam pressure under which the joint is required to be steam tight. The number and diameter of bolts in pipe flanges should, therefore, be such as will (when screwed up to the safe working stress in table 1) press the flange faces together with a pressure per square inch of surface equal to the steam pressure in the pipe, but also have a sufficient margin of strength to withstand any transverse strain that may come upon them from any extraneous causes.

Table No. 3 has been compiled to show the relation between the length of spanner, diameter of bolt, and the stresses to which they are subjected to when being screwed up. The number of threads and area at bottom of threads of the various sizes of bolts are Whitworth's standard.

Column No. 1 gives sizes of bolts ranging from $\frac{1}{2}$in. to $2\frac{1}{4}$in. diameter, and column No. 2 the number of threads per lineal inch. Columns Nos. 3 and 4 the lengths in inches of spanners which should be used in screwing up the different sizes of bolts. No. 3 giving the actual length, while column No. 4 shows the effective length of spanner, i.e., the actual length, 2 in.; this 2in. being considered the distance from the end of the spanner where the centre of effort in screwing up a bolt would be applied. Column No. 5 gives the effective circumference; from this column and column No. 2 the theoretical leverage on the bolt can be readily calculated: for example, we find that in screwing up a bolt 1in. diameter we get a theoretical leverage of 704 to 1, i.e., effective circumference of centre of effort on spanner = 88in. x 8 the number of threads per inch = 704; then for every pound pull on the spanner there is, theoretically, 704lbs. pull on the fibres of the bolt, but only 10 p.c. (vide Mr. McBride's experiment) of this is effective $\therefore \frac{704}{10} = 70.4$lbs. effective pull. Now, we find by table No. 3 that a bolt 1in. diameter can be safely stressed up to 2150lbs. $\therefore \frac{2150}{70.4} = 31$lbs. pull needed on the spanner as per column No. 10; the torsional stress on the 1in. bolt, as shown on column No. 13, is 14in. x 31 lbs. = 434 inch lbs. This is well under the twisting strength, viz., 826 inch lbs. given in column No. 12.

It will be noticed in column 13 that the actual twisting stress to obtain the required pull on bolt is, in the case of the small diameter bolts, well below the safe limit, and approaches nearer the safe limit for the larger sizes; example, for bolts $\frac{1}{2}$in. diameter the actual twisting stress is 6in. x 5.7lbs. = 34.2 inch lbs. against the safe stress of 14 inch lbs., only $\frac{34 \cdot 2 \times 100}{84} = 40$ p.c. Whereas for the $2\frac{1}{4}$in. bolt the actual twisting stress is 7648 against the safe working stress of 10,000, as much as 76 p.c. The reason to be assigned for this difference can safely be said to be that in actual practice a bolt as small as $\frac{1}{2}$in. diameter is subject to rough treatment in being screwed up, and a pull at the spanner of as much as 50lbs. or more is often given, which would be about 9 times the calculated amount; with a 50lbs. pull the fibre stress would be as much as 26,608lbs. per square inch, which is considerably more than it should be; a pull of 60lbs. would increase the twisting moment to 240 inch lbs., or more than 50 p.c. of its resistance to breaking by twisting.

Table No. 4, columns 1 to 6, give the proportion of flanges and bolts for copper steam pipes from 1in. to 8in. diameter, designed for a working pressure of 160lbs. per square inch. The table has been taken from a specification for copper steam pipes by Mr. W. D. Cruickshanks.

Column 7 gives the area of flange face after deducting the area of the bolt holes.

Column 8 gives the pressure per square inch with which the faces of the flanges are pressed together when the bolts are screwed up to the stress given in Table 1, and column 9 gives the total stress on the bolts binding the flanges together.

Column 10 gives the safe stress on one bolt.

Column 11 gives the pressure in lbs. per square inch with which the flanges are pressed together when the full pressure of 160lbs. per square inch is on the pipe.

Take the 6in. diameter pipe for an example.

Area of pipe = 28.27 square in. x 160 = 4523lbs. of force tending to separate the flanges due to steam pressure.

There are 12 bolts, $\frac{7}{8}$in. diameter, binding the flanges together. Then 1450 x 12 = 17,400lbs. of force binding the flanges together $\therefore$ 17,400 − 4523 = 12,877lbs., then 12,877 ÷ 77·5 = 166lbs. per square inch binding the flanges together when under a steam pressure of 160lbs. per square inch.

There are two of the pipes which have the pressure between the flanges considerably less than 160lbs. per square inch, and three of them are only slightly under it. The others, more especially the larger diameter, are sufficiently in excess to ensure a steam-tight joint.

Table 5, column 1 to 6, gives the proportion of flanges and bolts for copper steam pipes designed for a steam pressure of 250lbs. per square inch. The table is taken from Low and Bevis's Treatise on machine design (page 183).

The flange is of special design, and has been patented by Mr. R. B. Pope, of Dumbarton. The illustration shows the details of the flange and the arrangement adopted to reduce the area of joint face. By this design the area of joint face has been reduced to less than $\frac{1}{4}$ of that of an ordinary flange for the same size of pipe.

The average pressure per square inch between the faces of the flanges given in the table is about 905lbs. per square inch. There should, therefore, be no difficulty with this type of flange in making steam-tight joints with steam pressures ranging from 250 to 300 lbs. per square inch.

Take the 7in. pipe for an example:

7in. diameter = 38·48 sq. in. x 250 = 9620lbs.
Stress on bolts = 3000 x 9 = 27,000lbs.
Then 27,000 — 9620 = 17,380.
And 17,380 ÷ 17·1 sq. in. = 1016lbs. per sq. in.

Table 6, column 1 to 6, gives the proportions of pipe flanges and bolts recommended by a committee of the American Society of Mechanical Engineers for steam pressures from 75lbs. to 200lbs. per square inch.

Column 7 gives the area of the flange for joints.

Column 8 gives the pressure per square inch with which the flange faces are pressed together when the bolts are screwed up to the stress given in Table No. 1.

Take the 6in. diameter pipe for an example:

6in. diameter = 28·27 x 200 = 5654lbs. force tending to separate the flanges due to steam pressure.
Safe stress on bolts = 900lbs. x 8 = 7200 total.
Then 7200 — 5654 = 1546lbs.
And 1546 ÷ 63·2 = 24·5lbs. per square in.

So that these flanges could not be expected to remain long steam-tight.

The steam pressure at which the 6in. pipe flange would be steam-tight, assuming the pressure between the flange faces to be at least equal to the steam pressure in the pipe, is about 77lbs., as shown by the following example:—

6in. diameter = 28.27 sq. in. x 77 = 2177lbs. force tending to separate the flanges due to steam pressure.

Then 7200 – 2177 = 5023lbs. and 5023 ÷ 63·2 sq. in. = 77lbs. per square inch, forcing the flanges together; which is equal to that assumed as the pressure in the pipe, and even at 77lbs. per square inch there is no margin in the bolts to withstand any transverse stress that may come upon the pipe. The flanges of the 8in., 9in., 10in., 12in., 14in., and 15in. diameter pipe have all a minus value, that is to say, the flange faces instead of being pressed together when the steam pressure is on the pipes are being drawn apart with a force of 2853, 1920, 1692, 5200, 4980, and 940 lbs. respectively. Or, dividing these figures by the number of bolts in the flanges, it will be seen that the bolts are subject to a stress of 356, 160, 140, 433, 415, and 60 lbs. respectively beyond the stress allowed in Table 1.

The ratio of flange area and bolts section in the 10in. pipe would not permit of more than 90lbs. pressure being carried with safety, and the 3½in. diameter pipe should not carry more than 35lbs. per square inch. Under the action of these pressures the pressure on the flange faces per square inch would be equal to the steam pressure in the pipes.

The remarks of Mr. Pearson in the paper on pipe flanges apply very strongly to the pipe flanges and bolt proportions in this table.

Table 7, columns 1 to 6, gives the proportions recently adopted for steam pipe joints in the practice of the C.S.R. Co. for steam pressures of 100lbs. per square inch.

The table is self-explanatory, and shows that for 100 lbs. per square inch steam pressure, the flange joints, if properly made, should remain steam-tight, provided no transverse stress come on the pipe.

Take the 6in. diameter pipe for example:

6in. = 28·27 sq in. x 100 = 2827lbs. of force tending to separate flanges due to steam pressure.

Safe stress on bolts = 1450 x 8 = 11,600lbs.

Then 11,600 – 2827 = 8773lbs.

And 8773 ÷ 79·7 = 109lbs. per square inch pressing the flanges together when the steam pressure is on the pipe.

Table No. 8 gives the proportion of flanges and bolts for C.I. pipes to carry 100lbs. per square inch used by Alley and Maclellan.

The table is self-explanatory, and from an inspection of the figures in column it will be seen that the bolts are too few in number and too small in diameter to ensure a steam-tight joint.

Table No. 9, by the same maker, gives the proportions of flanges and bolts for C.I. pipes to carry a pressure of 200lbs. per square inch. The figures in column 9 show that there are not sufficient bolts in the flanges, or they are not large enough in diameter to ensure a steam-tight joint at a pressure of 200lbs. per square inch. In the 12in. and 16in. diameter pipes the figures show that there would be a risk of jointing material being blown out through insufficient pressure between the faces of the flanges.

An examination of the various tables giving the proportions of flanges and bolts for C.I. pipes shows that there is a very wide variation between them. In Mr. Nagle's table the pressure per square inch on the gaskets is ample to ensure a steam-tight joint, but in some of the other tables of proportions for C.I. pipes it is doubtful if the flanges would remain long steam-tight unless under favourable conditions.

The proportions of flanges and bolts given in W. D. Cruickshanks' specification for copper steam pipes, and also in Pope's design of flange, give sufficient pressure between the flange faces to ensure a steam-tight joint at the steam pressure for which they are designed.

It is evident from the remarks of Mr. Pearson and Mr. Nagle that the proportion of flanges and bolts should be designed to withstand any transverse stress to which the pipe may be subjected. It is, however, a difficult matter to determine the extent of any transverse stress that may come on a steam pipe, due to heat, or to the elongation of one pipe at right angles to another, causing a bending moment on the flanges.

It is not intended by the writers in this discussion to attempt to give a table of flange and bolt proportion for the various steam pressures, but rather to indicate the principle which seems to them should be kept in view in designing and arranging the number of bolts to be used in the various kinds of steam pipe joints, so that transverse strains may be provided for.

Much of the transverse stress on pipes may be obviated by the judicious use of properly arranged expansion joints, but the transverse stresses due to heating up the pipe cannot be obviated. They may, however, be much reduced in long ranges of steam pipes by having an ample number of drain cocks, and also by heating the pupes up slowly when raising steam.

When getting up steam, the difference in temperature between the top part of the pipe and the bottom part may often be as much as 50° or 60° Fah., and as that increase of temperature would cause a compressive stress of nearly two tons per square inch on the top part of a cast iron pipe, if it be prevented from expanding, and as the bottom part of the pipe acts as a tie to prevent the top part expanding, due to the increase of temperature, the bottom part of the pipe is put in tension, and as the compression and extension of C.I. for equal stress is nearly equal the compressio n stress in the top part of the pipe would cause a tensile stress in the lower part of the pipe about 1 ton per square inch, and it is the tensile stress put on the lower bolts in pipe flanges by the elongation of the pipe that cause them to stretch and the joints to leak.

The elongation of C.I. under a tensile stress of 1 ton per square inch is about 1/5555 part of its length, and as probably the compression of the top part of the pipe and the extension of the lower part acting in opposite directions make the actual strain on the lower part, with a difference of temperature of 50° or 60° Fah., only about 1 ton per square inch. If, therefore, two 9 feet lengths of C.I. pipe be connected by a flanged joint, and subjected to stresses caused by the unequal heating of the pipe to the extent of, say, 50° or 60° Fah., then the expansion of the top part of the pipe may cause the bottom part to extend about 1/5550 part of its length, which is approximately 1/22 part of an inch. The length of the bolts in the flange between head and nut may be taken at 2.5in., and as W.I. elongates about 1/10,000 part of its length, with a stress of 1 ton per square inch of section. The 1/10,000 part of 2.5in. is 1/4000 of an inch. So that the extension or elongation of the bottom part of the pipe is 36 times greater than the bolt, when the bolt is under a stress of 5 tons per square inch. If the sectional area of the bolts in the flange is insufficient to cause a tensile stress in the lower part of the pipe of about 1 ton per square inch of effective section, then the elongation must take place in the bolts, with the result that the joints will leak. In submitting these figures as the probable difference of temperature, and the consequent strain that may come on a C.I. steam pipe, it is hardly necessary to mention that they are only approximate, and that under certain conditions the stresses and strains on pipe flange joints may be much greater.

It is almost impossible to determine the tensile and transverse stresses that are liable to come on a range of C.I., W.I., or copper pipes in a boiler and steam plant installation, and therefore no rule can be laid down to provide for them. It may, however, be taken as a safe empirical rule that the number and diameter of bolts in a flange joint should have ample tensile strength to bind the flange faces together with sufficient force to keep them steam-tight, and they should have sufficient tensile strength to make the metal in the body of the pipe stretch without unduly straining the bolts in the flange, and thereby cause the flanges to leak. Where the pipe is not liable to transverse stress it is sufficient if the bolt area be large enough to bind the flange faces together with a pressure per square inch equal to the steam pressure on the pipe.

It may, however, be safely said that all pipes are more or less subject to tensile and transverse stresses, and should therefore have the flanges designed and fitted with sufficient number of bolts to resist them.

In flange joints where no transverse stresses or stresses caused by elongation occur, it becomes a simple question to determine the number of bolts required to keep the joints tight. For example, a vacuum pan 12ft. in diameter, built of segments having vertical flanges 4in. wide, fitted with bolts 1in. diameter spaced 5in. from centre to centre, 12ft. x 12in. = 144in. Vacuum in pan, say equal to 13.5lbs. per sq. in., 144in. x 13.5 lbs. = 1944lbs.; then 1944 ÷ (2 x 4) = 243lbs. per sq. in. as the pressure with which the joint material will be pressed together by the pressure of the atmosphere on the outside of the pan, and as the bolts are spaced 5in. apart each bolt binds an area of 4in. x 5in. = 20 sq. in., so that each bolt should be screwed up to a pressure on the flange faces slightly inexcess 243 x 20 sq. in. = 4860lbs. to ensure the jointing material between the flanges being under a slight initial pressure when the vacuum is on the pan; in this class of joint the pressure acts to keep the joint tight.

Take another example of joint, that of a semi-circular vessel, 68in. diameter, flange 4in. wide, steam pressure 75lbs. per sq. in., 68in. = 3632 sq. in. area x 75 = 272,400lbs.; the area of joint surface is 226in. x 4in. = 904 sq. in., which should be pressed together with a pressure of at least 75lbs. per sq. in., then 904 x 75 = 67,800lbs.; now the bolts for securing the joints should be equal to withstand the whole of stress due to the steam pressure, and screwing up to make t he joint steam-tight.

Therefore, 272,400 + 67,800 = 340,200lbs. These joints were fitted with 93 bolts $\frac{3}{4}$in. diameter, each bolt was, therefore, stressed up to 4200lbs. per sq. in., or about 4 times greater than the safe stress of a $\frac{3}{4}$in. bolt. These joints have always given trouble, the joint not keeping tight more than a few weeks. Larger bolts have been fitted in some of them, but they are not likely to be made a good steam-tight job until they are fitted with bolts about $1\frac{1}{8}$in. diameter.

The review and discussion of the proportion of pipe flanges and bolts in the foregoing remarks have been confined to steam pipe flanges, and had the notes not become so voluminous reference would have been made to the standard proportions given by various authorities for ordinary water pipes, and fo rhydraulic pipes; and also to the proportions of bolts for securing connecting rod end and bearing caps. However, this is a subject some of the other members may introduce by-and-bye. The notes which the writes have submitted for discussion will give any data they may have on the question of troublesome steam flange joints, so that the discussion may be a fairly exhaustive one.