DISCUSSION.

Mr. W. D. Cruickshank said, in opening the discussion, it was only after careful study the wide variation and the material discrepancies which had crept into what was recognised as the best standard practice were uppermost, but which by a few simple calculations had been shown, in a number of instances, to border on the absurd. The wonder was, nobody thought of it before.

The object of the paper was not only to show the defects of the present system, but to collect reliable data, so that internal diameter and thickness of pipes should be proportional, that all flanges should be rigid enough and the distribution of bolt section strong enongh to not only ensure tightness under steam, but also to have sufficient margin for resisting the angular and transverse stresses to which, in a greater or less degree, all steam pipes were subjected, not forgetting at the same time to make due allowance and ample provision for expansion. What was wanted was to be able to make all joints in such a manner that, no matter what the stresses might be, the body of the pipe must move, and not the flanges or bolts, and if the tables were revised we should have something commanding confidence, and which would be recognised and used as the standard of the future. This might be done by the authors at their convenience, and would be duly appreciated by the profession.

After reading the paper, various points suggested themselves as appropriately bearing on the subject, and possibly the following remarks would be of interest.

Referring generally to recognised authorities, also to the standard formulae given in text books, it was evident that in the present instance the proportions given in connection with the flanged joint were unreliable, consequently liable to mislead and result in trouble, unless our technical training was in happy combination with actual workshop and everyday working experience.

In New South Wales, the system and opportunities for acquiring technical knowledge ranged high, and he was sure all appreciated the great and good work done by our technical schools; but in mechanical, as in other branches of engineering, there was an instinctive, intuitive personal education, which could not be procured from books, and could only be obtained by having to do the actual work. In support of this, take No. 3 table, where the "length of spanner," "its effective length," "its effective circumference," "theoretical pull," "effective pull," etc., etc., were worked out elaborately, many people would say—too much so, yet we knew that pipe and all other joints were made, and would continue to be made, not by such practically impossible scientific refinements as shown in the tables, but by the instinctive, intuitive knowledge which mechanics possessed, a knowledge that could not be taught, could scarcely be explained, and could only be acquired by personal experience in having to do the work themselves.

What he wished to point out was, that this special practical skill was not by any means confined to steam pipe jointing, as many of the "rules" and "formula" relating to mechanical engineering had to be, and were, heavily discounted by the men who were responsible for carrying out work. But it required years of experience and practical knowledge to discriminate with confidence how far or how much they could depart from theory without impairing efficiency. This was particularly mentioned because, so far as could be seen, his impression was that the present tendency was to credit theoretical deductions and empirical formula with a percentage of importance which was not proportional to the supreme necessity and value of a thorough practical training, and it was in such a subject as that now under discussion that it was most likely to bring out the great importance of having the two principles happily blended. Many instances and illustrations might be given in the application of the above to actual work, showing how necessary it was to exercise care and caution under certain conditions, but the field was vast and the time limited, therefore he could only refer briefly to a few.

Respecting the variation in the tables, one example might be cited. In No. 6 table, which presumably represented the standard practice of mechanical engineers in America, in a 3in. steam pipe the flanges were $7\frac{1}{2}$in. diameter and $\frac{3}{4}$in. thick, secured by four $\frac{5}{8}$in. bolts, for a workng pressure of 200lbs. per square inch. Compare this with the same sized pipe, flange, and pressure in "Denny's" Table (No. 4), representing the standard Clyde practice, and we found their flanges $\frac{7}{8}$in. thick and secured by eight $\frac{3}{4}$in. bolts. The collective sectional area of bolts at thread bottom in the American joint was only .8 of a square inch, while in Denny's joint $3\frac{1}{2}$ square inches to do the same work. Consider the two joints from another point: In the American the plate section in flanges was over 80 per cent., but t he bolt section was only about $6\frac{1}{2}$ per cent., while in the other the plate and bolt sections were 56 per cent. and 30 per cent. respectively.

To show the absurdity, in fact he would say the "danger," of the proportions given in Table No. 6, imagine that in a boiler seam two plates $\frac{3}{4}$ in. thick were being held together by 5-8 in. rivets, pitched 4 in. centres with 200lbs. per square inch pressure. He thought the man who compiled that table would give that boiler a wide berth. Many of the proportions given in the various tables were equally bad, possibly worse, than the above, which was taken at random. However, in any case the useful information contained in the paper was certainly a good object lesson, which showed how necessary it was to exercise care and caution in not taking things for granted merely because they were printed and often find their way into well-known text books.

Another pertinent question suggested iaself, viz., How would you calculate the "force" tending to separate the flanges in the direction of their length? In the paper it was assumed that all the joints were perfectly tight, and in such cases this force was always equal to the area of area, multiplied by the pressure, but in partially tight or leaky joints the force tending to separate must be considerably more, how much more, and what allowance could be made, might be discussed with advantage.

On page No. 12 an empirical calculation was made, showing the result of a temperature difference of 60deg. between the top and bottom of a steam pipe. This and similar calculations were useful in showing how such problems should be solved, provided the assumptions were correct, but practically the figures had a very fractional value. Besides, to him it was by no means clear under what circumstances a temperature difference of 60deg. could possibly exist in a steam pipe.

From experience and observation in marine boilers he had often seen a temperature difference of nearly 300dg. between the top and bottom. The method of proving this was by drawing a bucket of water from the water space below the furnaces and comparing its temperature with that of the sea water, and in many instances finding both temperatures the same, and that had been done repeatedly when the ship was going full speed and over 100lbs. per square inch showing on the steam gauge. This difference in temperature, more or less, was what took place in all boilers in which a considerable quantity of water was below the fire, and where no appliances were available for the circulation of water when getting up steam. Any attempt to calculate the complicated stresses and strains set up under such circumstances (to his mind) could never be satisfactory.

Referring to copper steam pipes generally, it might be d esirable to point out that this material for the existing high pressure was not by any means considered to be so safe and reliable as it should be, although on paper and in accordance with formula the strength margin was at least double, and in many cases more than double, what was allowed for engines, boilers, and other parts of machinery, being as high as 10 and 12 to 1. The principal causes for this were defective brazing, possible burning, and insufficient provision for expanson for the unknown stresses when under steam. The greatest difficulty we had to contend with, in a copper steam pipe where there was a working pressure of, say, 200lbs. to the square inch, was that although the calculated factor of safety was exceptionally high, and although it was tested by hydraulic pressure to 600lbs. per square inch, and although that test was, so far as could be seen, absolutely perfect, yet many instances had occurred when such pipes had given way under ordinary working conditions, and often with disastrous results. At all events, it was well known that defective copper steam pipes had given more trouble, caused more accidents and loss of life than anything else connected with machinery afloat. To show that there was a want of confidence in this material, many of our leading engineers had all main steam pipes, clasped with steel bands, the distance between each being 8in. to 10in., while the British Admiralty had all main pipes continuously wound with stout copper wire, special machines havng been designed for that purpose. For the above and other reasons the introduction and use of a special mild steel for steam pipes was becoming more general, and would eventually displace copper, being stronger, safer, and much more reliable.

With regard to the expansion of copper pipes, when under steam, the usual method of providing for the increase of length due to increased temperature was by easy bends or expansion joints, or both. For the existing high pressure, however, many of the copper pipes were 5/16in. and ⅜in. thick, and the bends in such cases were too rigid to have the requisite amount of give, consequently the unfortunate pipe had to dispose of its expansion as best it could. Even when fitted with expansion joints, and especially when the pipes were far from being straight, the expansion would follow the line or lines of least resistance, and in such cases perhaps less than half the expansion would be developed in the expansion gland, and to prevent such contortions it became necessary to cast lugs on the flanges, to which moveable stays were attached, for the purpose of compelling the movement due to expansion to take place in the expansion gland. Respecting the amount of expansion in copper pipes under steam, speaking approximately, we knew that the difference in temperature between steam at atmospheric pressure (212deg.) and stam at 50lbs. to the square inch was 85deg., whereas the temperature difference between 150 and 200 lbs. per square inch was only 22deg., showing that the difference decreased as the pressure increased, and for practical purposes we might assume that the average temperature difference in copper pipes cold, under steam, for present high pressure was 300deg. F., which could be used as a constant.

Example:—

Assume a straight copper steam pipe 32 feet long, working at any pressure between 150 and 200 lbs. Temperature difference, cold and hot, 300deg. Co-efficient of expansion (copper), .00000958.

Length before expansion, 32 x 12 x 382 inches.

Then .00000958 x 384 = .00367872.

And .00367872 x 300deg = 1 .1 inch.

So that practically we could take it as being correct enough to say that copper steam pipes, working at modern high pressure, would expand 1 inch for every 32 feet; or, to put it another way, forming an approximate rule and easily remembered, the average expansion was one thirty second of an inch per foot, t he expansion being directly proportional to the length.

Many examples might be given where well known formula could only be used within certain limits, outside of which they became ridiculous and of no account. Take, for instance, round bars subjected to torsion, it was laid down as an axiom that their strength varies as the cube of the diameter. This was true in so far as the experiments with, say, inch round bars were concerned, but in bars or shafts of greater dimensions it might be, and often was, misleading, and in many instances actual experience had to step in and materially increase the theoretical dimensions as much as 100 per cent. under certain conditions. The factor of safety in crank and other shafting (by calculation) was more than double what was usually allowed in similar parts of engines, and yet they frequently broke. This, however, was a large question, and might have special reference at some future time.

Again, the "standard rules" for ascertaining the strength of pipes and cylinders subjected to internal stress was very simple, being always equal to the strength in pounds, multiplied by the thickness and divided by the diameter, the result being the bursting pressure.

$\frac{\text{Strength in lbs. } \square'' \text{ x thickness.}}{\text{Diameter}}$=B.P. But although the rule was simple it often happened that we must discriminate in applying it. To illustrate:—Take a steel cylinder, 100" internal diameter and 1" thick, the strength being 60,000lbs. per sq. inch. Then $\frac{60{,}000 \times 2''}{100''} = 1200$lbs., representing destruction. Allow a strength margin of 6 and the W.P. was 200lbs. per sq. inch, and we would be perfectly justified and quite safe working at that pressure. But take a cast iron cylinder of exactly the same dimensions, and where strength was 15000lbs. sq. inch, its destruction pressure would be $\frac{150000 \times 2''}{100''} = 300$lbs. per sq. inch. Allowing the same strength margin, 6, we would have $\frac{300}{6} = 50$lbs. as the W.P. But the practical working and knowledge of the two metals came in, and we quite realised the necessity of largely increasing the strength margin in the case of cast iron cylinders, and principally for two reasons—1st, because cast iron was not by any means so reliable as steel, and, 2nd, because we could never depend absolutely upon the sectional strength being sound or of uniform thickness; hence, to satisfy practical experience, th estrength margin was often doubled. Another point in illustration was that the thickness formed a very small fraction of the diameter—in fact, small enough to justify us in assuming that the resistance of the material in internal tension was equally distributed throughout the entire sectional thickness, and this meant that every particle of the material did its fair share of the work. It, however, becomes a very different question when the diameter and thickness approached each other, as in hydraulic rams and in pipes of small diameter under exceptionally heavy pressure, as it often happened that adding to the thickness did not increase its strength. It was evident that in such cases the particles forming the inside skin carried the brunt of the intenral stress, while the particles near to and forming the outside diameter would be doing very little work, possibly none at all. So far as he knew, this matter relating to the consideration of the proportional thickness to diameter, where it began and where it ended, had never been definitely demonstrated and could scarcely be calculated, but everyday experience recognised the difficulty and made provision accordingly.

Again, consider cylinders, where exposed to collapsing stresses, such as large pipes, tubes, flues, and furnaces. The recognised formula was that their strength varied inversely as the length, inversely as the diameter, and as the square of the thickness. Under certain conditions this rule was all right, but only under certain circumstances and within certain limits, after which it became absurd in exceptionally thick, long, and also in very short tubes. Bearing out the above, a peculiar incident happened some time ago in one of the other colonies. In an ordinary designed return tubular marine boiler, one of the boiler tubes leaked suddenly and too freely, resulting, unfortunately, in loss of life. The Government appointed three engineering experts to enquire into the cause, and report. They ordered quite a number of the tubes to be drawn, measured the minimum thickness, and calculated the bursting and safe working of each tube. The results were actually compiled in a tabulated form and published in all the principal newspapers of Australia. The unfortunate mistake was that the tubes in question were not exposed to tensional or bursting pressure at all.—in fact, they seemed to have ignored or forgotten that all their calculations should have been for tubes subjected to collapsing pressure, not bursting. (Tubes were 3½in. inside and 7ft. long.) But the point to which particular attention was directed is the following:—After reading the report in full, he asked himself this question: Assuming you had to give evidence in this case, how should the strength of such boiler tubes be calculated? At the first blush many engineers would, and in fact did, say, "Oh, by the rules which regulate the collapsing and working pressures on furnaces." But the absurdity of this is at once apparent, as a boiler tube 10 feet long would only be entitled to one-half the working pressure of one 5 feet long—that was, if the strength varied inversely as the length. It was very doubtful if this question could be answered with definite and distinct correctness; but we might say this: We know very well that in very small tubes, where the diameter and thickness approach each other, and where the diameter was a very small fraction of the length, in such cases the length might be eliminated and the calculations simplified by multiplying the strength constant by the thickness and dividing by the diameter — $\frac{\text{Constant x Thickness}}{\text{Diameter}}$ = B.P. or W.P., as the case may be. But at what particular proportion of diameter to thickness or length to diameter would justify the inclusion or exclusion of the length factor was practically an unknown quantity.

Some of the calculated collapsing tables started at 6in. Wilson started at 9in., and assumed that all tubes from 6in. or 9in. upwards should be calculated by the formula previously mentioned. Its application, however, to 6in., 9in., or even 12in. tubes was extremely doubtful, only in such cases it was satisfactory to know the variation was on the safe side.

The only experimental information that he knew of, having special reference to the actual strength of small tubes when exposed to tensional and collapsing stresses, would be found in D. K. Clarke's "Steam Enigne," 2nd volume, page 649, and was deduced from Russell's experiments on solid drawn iron tubes $\frac{1}{8}$in. thick, and ranging from $3\frac{1}{4}$in. to $1\frac{3}{4}$in. external diameter. The average bursting pressure per square inch of surface was 5300 lbs., and the average bursting pressure per square inch of section was about $22\frac{1}{2}$ tons. The average collapsing pressure per square inch of surface was 3425lbs. per square inch, while the average collapsing pressure per square inch of section was 18 tons. No details were given as to how the experiments were carried out, but the results showed that very small tubes were enormously strong, although in what ratio they weakened as the diameter increased, or what the proportional dimensions should be when the length must form a factor in the calculation, was not definitely known.

In Smiles's well-known book, "Self Help," its author never framed a better or more suggestive sentence than when he said that "marriage, like good government, was a series of compromises," and this was equally applicable to good engineering, and specially so as regarded mechanical engineering.

The principal object of his remarks had been a brief attempt to emphasise the fact that, to know when, why, and to what extent one could and must compromise with theoretical deductions, scientific formulae and empirical rules formed one of the most important factors in the education of an engineer.

Mr. James Shirra said every practical engineer had had various experience of the troubles of leaky steam pipe joints, and usually attained almost instinctively to some notion of how to meet the difficulties, but not many, he feared, kept a record of their experience in such apparently commonplace matters in their note-books, or could say right off what should be done in the various circumstances of the case. For the circumstances were very various, and it was necessary to go to the root of the matter in considering it.

The peculiar stresses in a flanged joint first demanded our attention. That in the pipe could be taken as wholly a tensile one. If it was due only to the pressure of the contained steam, it was, as in any other hollow cylindrical structure, a circumferential one, which could be easily calculated from the internal pressure, and a longitudinal one, equal to half the other, the wide flanges easily withstood the first, but not so advantageously the latter. This longitudinal stress was much increased by the bending stress which might come on the pipe, which increased the tension on one side, while diminshing or even putting in compression the other. When the normal longitudinal stress on one side of a pipe under pressure was doubled by a bending stress, and was, therefore, just equal to the circumferential one, and so quite within the resisting power of the material, there was no longitudinal stress on the opposite side at all, and as their metallic structures usually failed by the buckling of their comphession members, a pipe under pressure was very rigid, as anyone who had noticed a canvas hose