The New API Sucker Rod Joint

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American Petroleum Institute Standard 11B, covering sucker rod dimensional standards, was originally adopted in November 1926. The basic joint design contained in this specification remained the same through 12 editions and numerous supplements up to the 13th edition, issued in January 1960, covering a period of 33 yrs. During this period it accomplished a very worthwhile purpose and, by giving industry a fully interchangeable joint of good design with adequate strength for most well loads, saved the industry many millions of dollars.

During the past few years, however, deeper and deeper wells have been put on the pump. More and more fluid is being raised to obtain the allowable oil production. Sucker rod strings, therefore, are being subjected to higher and higher loads. As a consequence, joint failures are becoming more prevalent and point up the necessity of taking a close look at our joint design to see if it can be improved. This inspection has now been done and the present standard has been changed to incorporate two new items of design as definite improvements. These are the Class 2A - 2B thread and the stress relieved pin. The latter, the stress relieved pin, has also opened up the door for an important third change, the rolled threads.

Class 2A - 2B Threads: During the past 10 to 15 yr great advances have been achieved in screw thread design and investigation. In November 1948, culminating many years of intensive study and research, the United States, together with the United Kingdom and the Dominion of Canada, jointly signed the "Declaration of Accord" which covered a new unified standard for screw threads. The American Standards Association immediately revised their then existing standard and, in 1949, published ASA B1 1-1949 titled "Unified and American Screw Threads". This publication was followed, in March 1951, with the 1950 Supplement to "Screw Thread Standards for Federal Services", which was issued by the National Bureau of Standards of the United States Department of Commerce, and which also conformed to the new "Unified" screw thread standard. Since that time, industry in the United States has generally progressed a long way in the change over to the new thread.

For many years, thread experts had maintained that the class of thread -- that is the fit which specifies tolerances and allowances -- for the API sucker rod connection was much closer than is necessary or desirable. It has been felt by the trade, however, that a close fit requiring precision tools and gages would result in a stronger and more satisfactory joint. This idea was logical; but, in view of the recent advancement in the screw thread art, it behooved the industry to take a second and closer look at the problem. This look taken and, in January 1960, the 13 edition of API Standard 11B was issued specifying the "Unified" Class 2A - 2B fit for sucker rods.

Under the old American standards for screw threads, there were five classes of fits roughly cataloged as follows:

Class 1: Ready interchangeability, with loose fits and maximum tolerances.
Class 2: Good interchangeability, reasonable fits and tolerances. (This class covered the major portion of interchangeable screw thread work in all industries.)
Class 3: Highest grade of interchangeability with close fits and minimum tolerances for a production product.
Class 4: Interchangeability not 100 per cent; may be selective with close fits and little tolerance; a precision product requiring expensive special gages and gaging practice.
Class 5: An interference fit requiring use of wrench for make up. Ordinarily made up only once, such as steel studs in engine blocks.

(Class 4 and 5 have now been abandoned)

The API sucker rod joint did not conform to any of the above enumerated classes, but fell between a Class 3 and Class 4 fit and consequently became a precision product. If the closeness of fit and precise quality of the thread were to be indicative of the strength of the joint and the service to be expected from it, the care and expense required to obtain it might well have been worth the trouble. However, studies proved this not to be true.

A few years ago Professor Buckingham of the Massachusetts Institute of Technology conducted an extensive series of tests to obtain information about the influence of the class of fit on the tensile strength of general purpose bolts and nuts. In this interesting bit of research work, several sizes of bolts and nuts made from several grades of steel were tested in lots: each lot of course, was the same material and consisted of about 25 samples each of Class 1, 2, and 3. Professor Buckingham concluded that "the class of fit is of little or no importance. Any additional time, effort, or money which is expended on the making of these threaded bolts and nuts of ductile materials (steel) with tolerances more exacting than those of Class 1 is an unnecessary waste." He then presented a mass of data which clearly indicated that in tension, the Class 1 thread is just as strong as are the Class 2 or Class 3 thread. By going to more precise fits, nothing is gained strength-wise as far as tensile strength is concerned. Thus, surprising as it may seem, the old sucker rod Class 3 to 4 thread fit added nothing to the strength in tension of the joint over that to be obtained from Class 1 or Class 2 fit. But sucker rod pins fail in fatigue, not tension. How does class of fit affect this type of failure? The closer fit promotes fatigue failure.

It is universally acknowledged that sucker rod pin failures due to fatigue (99 per cent of such failures) is due to lack of, or loss of, proper initial tightening, which will generate high face pressures at the shoulders and a consequent low range of stress, under load, in the pin (1). A close fit will militate against proper face and shoulder pressure on several counts.

A close fit adds friction to make up and absorbs torque which would otherwise have been used in generating proper face pressure. In the Spring of 1955, the author worked with others on a test string in a well in West Texas. It was a mixed string of tight fits and free running fits alternated from top to bottom. In seven weeks five pin failures were experienced in the tight fit pins, with none in the free running fit pins. The test was then discontinued.

To avoid high local stresses on one side of the pin, a close fit requires precise parallelism of shoulder faces while a looser fit will mechanically accommodate a small amount of misalignment without high local stresses. The Federal Service H28 Handbook covering

Screw Thread Standards reports: "A close fitting thread assembly under some conditions may fail whereas the cause of failure may be eliminated by providing a looser fit. A cap screw that seats only one side of the bearing surface under the head may break off when the screw is tightened. When a screw has a large bearing surface under the head or when the head must be square with a projecting pin, sufficient pitch diameter clearance must be provided to allow for any out-of-squareness of the screw axis with the bearing surface under the head. Thus, as large a pitch diameter tolerance as possible, together with providing proper tolerances on squareness of face with the thread axis where seating is required may avoid the necessity for specifying a heat treated bolt" (2:201).

A close fit will provide little if any clearance for foreign matter and must be kept meticulously clean if high friction during make up and its attendant troubles are to be avoided. The conditions under which sucker rods are run and re-run in particular are far from conducive to this requirement. A looser fit will tolerate more dirt and grit and will avoid many broken pins on this count alone.

A close fit, liberally greased, can and does build up substantial hydraulic resistance to tightening and will thus absorb torque at the expense of proper shoulder and face pressure. This situation has been dramatically demonstrated in the author's plant; however, a looser fill will prevent this undesirable condition.

A close fit requires, in thread form and lead, a precision approaching that required for gages which, in a production operation as in the sucker rod thread, is difficult and uneconomical to obtain. A looser fit will overcome this difficulty with no sacrifice in quality or performance.

When the tightness of fit is excessive, the couplings are hammered by field operators and many are broken or damaged by this hammering.

In a letter written by Mr. C. E. Haven of the National Bureau of Standards, U. S. Department of Commerce, and also a member of the API Correlating Committee on Gages he states: "I am in complete accord with the belief that sucker rod threads should be made to Class 2A - 2B tolerances and that most of the failures in joints are due to insufficient tightening of the threaded connections."

It should be stressed that improvement in the performances of the sucker rod joint should be and is the sole aim. The increased knowledge and experience of the past few years now clearly demonstrates that a Class 2A - 2B fit in accordance with the new "Unified Standards" is not only as strong but is much preferred, from a performance standpoint, to the antiquated Class 3-1/2 sucker rod thread. And it should also be stressed that its adoption does not open the gates to sloppy work. The new fit is just as closely controlled with just as precise gages and gaging practice as was the old fit.

Also, the old and the new have 100 per cent interchangeability. The only effect is a median compromise with one member of the joint the old standard and one the new; hence, such a condition is more desirable than the old.

Undercut Pins: The undercut or stress relieved pin will be specified in the 14 edition of API Standard 11B to be issued this spring.

Undercut pins, or stress relieved bolts, have been in general industry use for over 20 yr, particularly by the aviation and automotive industries. A book, titled Prevention of the Failure of Metals Under Repeated Stress, published by the Battelle Memorial Institute over 20 yr ago, shows several examples of fatigued threads having no undercut section, with the book comments, "design error" or "inadequate design, no stress relieving undercut" Early in the 30's, the Gulf Research Development Company did some work in applying a modified undercut to the sucker rod pin. In 1949 this matter was again discussed in the API Standardization Committee Meeting, after which one manufacturer designed a stress relieved pin and made up several strings with this contour, three of which were run in Seminole in 1950 and the remaining subsequently run in Drumright and Oklahoma City fields.

In 1952 a paper which included an explanation of the advantages of this stress relieving contour was presented to the Mid-Continent District Spring meeting of API at Wichita, Kansas. (3.)

All of the above field tests demonstrated the superiority of this design but the degree of superiority was apparently not enough to convince the operators that at that time, it should be adopted. The matter was carried on the API Sucker Rod Committee's agenda until 1956 when it was dropped for lack of interest on the part of the users.

About 1959 interest was revived, primarily because of increasing operation under higher stresses required for deeper wells and the resulting increasing incidence of pin breakage.

The undercut pin provides three important benefits:

First, and most important, the stress flow through the critical neck of the pin is smoothly steamlined in such a manner that stress concentration at the vulnerable point is greatly reduced.

Second, the long, slender neck has a spring effect, or greater stretch factor, under a given torque and tends to reduce the tendency of the pin to loosen in operation. Basically, elongation is equal to stress times length divided by the modulus of elasticity. In the old 7/8 in. API pin, for instance, that portion of the pin from the shoulder face to the first perfect thread, has an average diameter of 1.148 in. and a length of 0.550 in. Assuming a stress at the root of the thread of 40,000 psi, this pin would stretch 0.00066 in. when properly tightened. The undercut pin has an average diameter of 1.035 in. and a length of 0.717 in. and would stretch 0.00104 in., or 1.6 times as far as the old API pin. This additional stretch tends to absorb or offset relaxation which occurs under operation and will help greatly in keeping the joint tight.

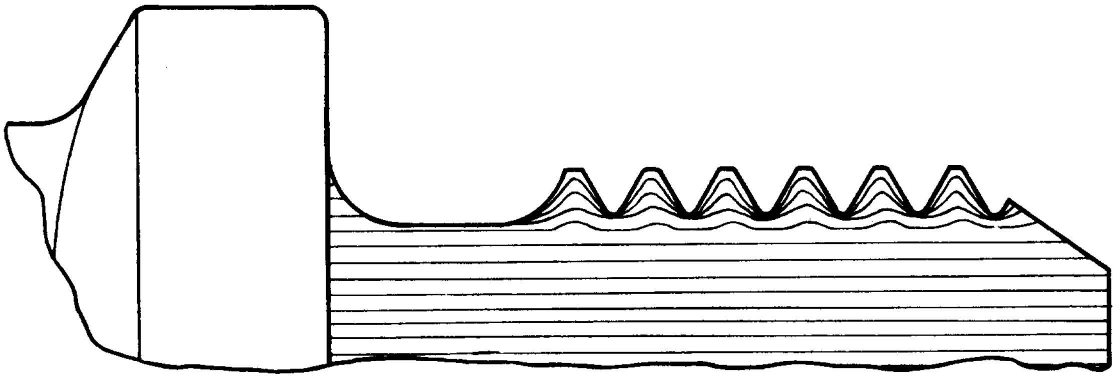
Third, the longer and more slender section of the undercut pin has greater flexibility than does the shorter, thicker counterpart of the present API pin. Therefore, when bent under the same magnitude of deflection, tensile stresses on the extreme fiber on the undercut section of the pin will be appreciable lower than will those on the API pin, thus the tendency for fatigue will be correspondingly reduced. Of course, bending does not occur if the faces do not separate, but they too often do so.

Rolled Threads: API Standard 11B is a dimensional standard; but it does not set forth methods required to obtain those dimensions. Therefore, thread rolling, being one of the several thread forming methods, is not specified in the API Standard. With the inclusion of the undercut thread in the standard however thread rolling becomes the logical and the much-to-be-preferred method of forming the pin threads.

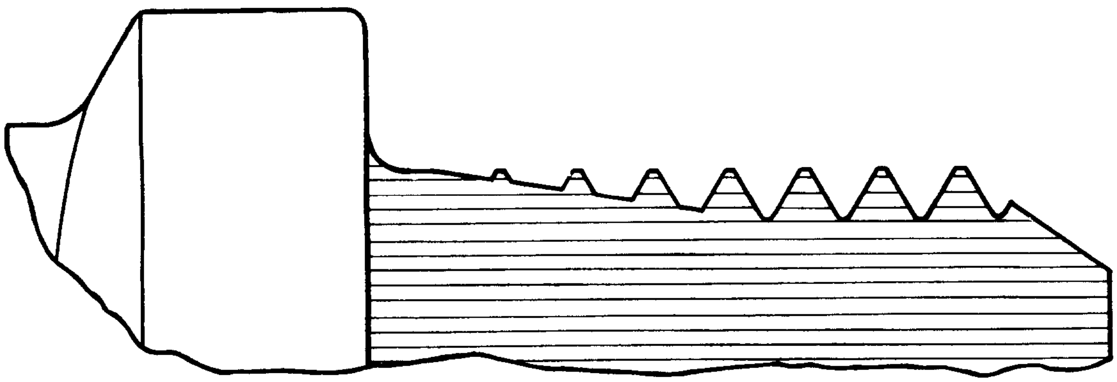
Thread rolling is a simple cold forging process. Three round, hardened steel dies, the faces of which are threaded with the desired thread configuration, are forced over the straight turned cylindrical pin blank, and thus cause the metal to flow into the required thread form.

Thread rolling is not new; this method of forming threads has been used for over 50 yr, primarily by the fastener industry. It is particularly applicable to forming threads on straight cylindrical stock on which the unthreaded portion is undercut sufficiently to clear the die rolls. The old API pin with its 9° vanish cone presented difficulties to this method; consequently, its use was never seriously considered until the undercut pin design was suggested.

Thread rolling does not cut the fibers of the pin as does thread cutting; instead, it compresses the metal at the roots and extrudes it up into the crests (Fig. 1). The residual compression induced at the roots of the threads is highly effective in minimizing fatigue breakage. Literature points out that fatigue life is increased over cut threads by a factor of 50 to 75 per cent (4). Rolling between smooth dies leaves the threads with smooth burnished flanks and with freedom from tears and chatter or cutter marks which can serve as focal points of stress to induce fatigue failures. Also, the smooth flanks of the rolled thread effectively reduce friction on make-up and thus convert more of the force used in torquing the connection to generating high shoulder face pressure. The superiority of the finish is illustrated in Figure 2. Rolling also, by cold working the metal,



(a) Undercut Pin With Rolled Threads



(b) API Pin With Cut Threads

Fig. 1

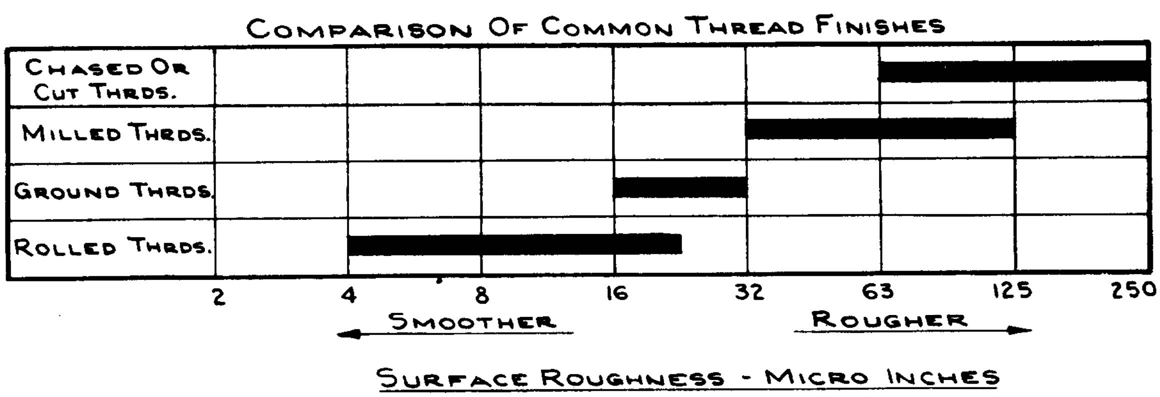


Fig. 2

increase the tensile strength of the pins as well as the shear strength by about 10 per cent. Accuracy, which in the case of rolled threads, depends on the accuracy of the dies, is more closely held than cut threads since thread rolling dies do not wear in the same manner as do thread cutting tools. Wear, rather than being concentrated on sharp cutting edges, is distributed over broad surfaces, and the rolling action is relatively free of friction. Thus the dies are given greater life.

So we now have three important improvements in the design of our sucker rod joints. However, lest the enthusiasm with which these improvements have been presented lull the operator into an undue feeling that care is not needed in make-up, we hasten to add and to underline, <u>that joints still must be clean and well greased</u>, and <u>reflect a free running fit to shoulder contact</u>. The Class 2A- 2B thread fit will accommodate more foreign matter than will the old fit, but the threads must still be reasonably clean. If the coupling cannot be spun up to shoulder contact, freely by hand, corrective measures must be taken or trouble is in the making.

It is now generally recognized that in a free running threaded connection, 90 per cent of the applied torque is absorbed in overcoming friction (50 per cent at the shoulder faces and 40 per cent in the threads) and leaving only 10 per cent of this applied torque available for generating shoulder face pressure. Any undue friction in the threads, therefore, will quickly absorb this 10 per cent at the expense of the shoulder loading, so essential in preventing pin and coupling breakage, and leaving the joint loose, in effect, for our purposes.

Torque Make-Up: It has been pointed out in many articles and field experience has amply demonstrated that joint make up with the proper torque is absolutely necessary on well loaded strings if the occurrence of broken pins and couplings is to be avoided (5). The new design features in the API pin promote more consistent results from any given torque effort, allow more leeway in the actual torque required, but definitely do not eliminate the necessity for properly tightening each joint. The same old torque recommendations for the various sizes and grades of rods are still valid.

Consistently obtaining the proper torque with snap wrenches or even with power tongs is a problem. To illustrate this point in the case of hand tools, the recommended torque for 1 in., high-tensile rods is 866 ft lb which is equivalent to a pull of 86.6 lb at the end of a 10 ft lever arm or cheater: attempting to obtain this tightness consistently with a 22 in. rod wrench, even a snap wrench, takes a husky man several blows with plenty of beef behind them, on every joint. It should

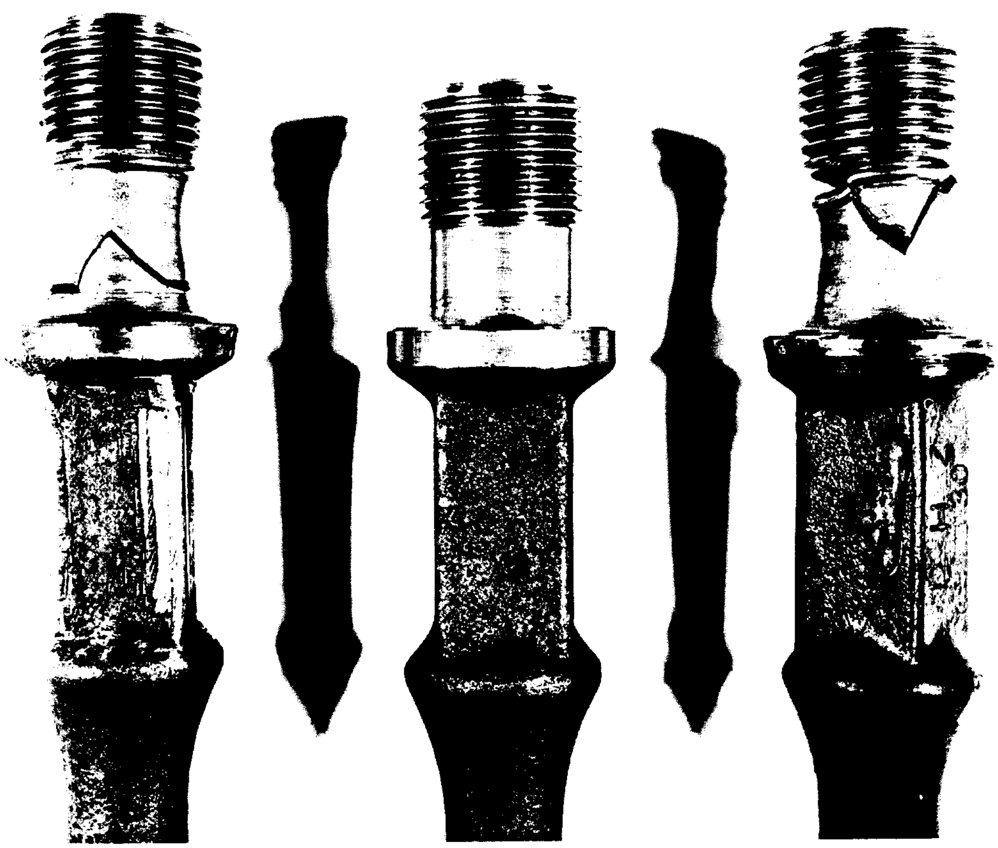


Fig. 3

also be remembered that, in the morning, a rod crew running a tapered string in a deep well starts with the small rods and, in the evening after a long hard day, finishes with the large rods, requiring the highest torque. Human nature being what it is, such practice particularly on heavily loaded and deep strings, is often disasterous.

Obviously, when running 1-1/8 in., 1 in., 7/8 in. and even well loaded 3/4 in. rods, power tongs are the only positive means of obtaining the required torque. But power tongs can and often do give erroneous results. The writer has seen used air tongs that, because of worn and dirty gears which absorbed the power input in internal friction, were putting out far less torque than indicated by the gage. On tongs he has seen gages that were broken or obviously wrong, and other conditions which made it difficult or impossible to check actual torque out-put. The pin on the right, in Figure 3, was pulled off with used hydraulic tongs supposedly set for the required torque of 512 ft lb. The one on the left resulted from the application of 1,260 ft lb carefully measured in the author's plant: The center pin is normal for comparison purposes. Obviously in this instance, the tongs were producing almost 2-1/2 times the torque they were supposed to be producing.

It then becomes plainly apparent that the use of power tongs, in itself, is not the answer to the proper tightening problem. They must be carefully maintained and checked; the gages should be regularly tested; and, most important, the power out-put of the tongs should be measured on every job. One simple and convenient means of doing this is shown in Figures 4 and 5. Here, the back-up fork underneath the tong is dropped out of the way, and substituted is a regular rod wrench which has a chain or wire line attached to the end. A Dillon weight gage is inserted in this chain and the chain is run at right angles to the wrench and run to the derrick leg and anchored. When the tong is engaged with the upper wrench square on the joint and power is applied, this special rig, applied to the lower wrench square, acts as "back-up" The reading of the Dillon gage is pounds times the distance from the joint center to the chain attachment near the end of the wrench in feet. It is the torque being applied to the joint in foot pounds. However, this figure is true and accurate only when there is very little weight on the elevator holding the lower rod. But this rig can be used and is successfully used to check tongs at the beginning of a job. Several other simple and similar means to accurately check tongs at the well

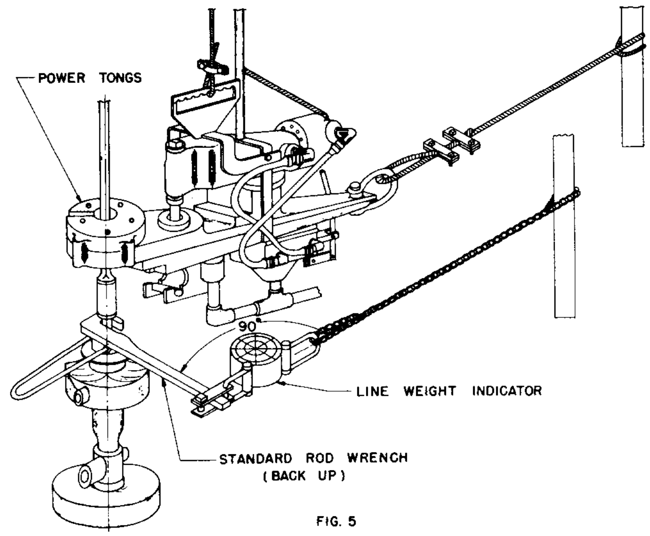


FIG. 5

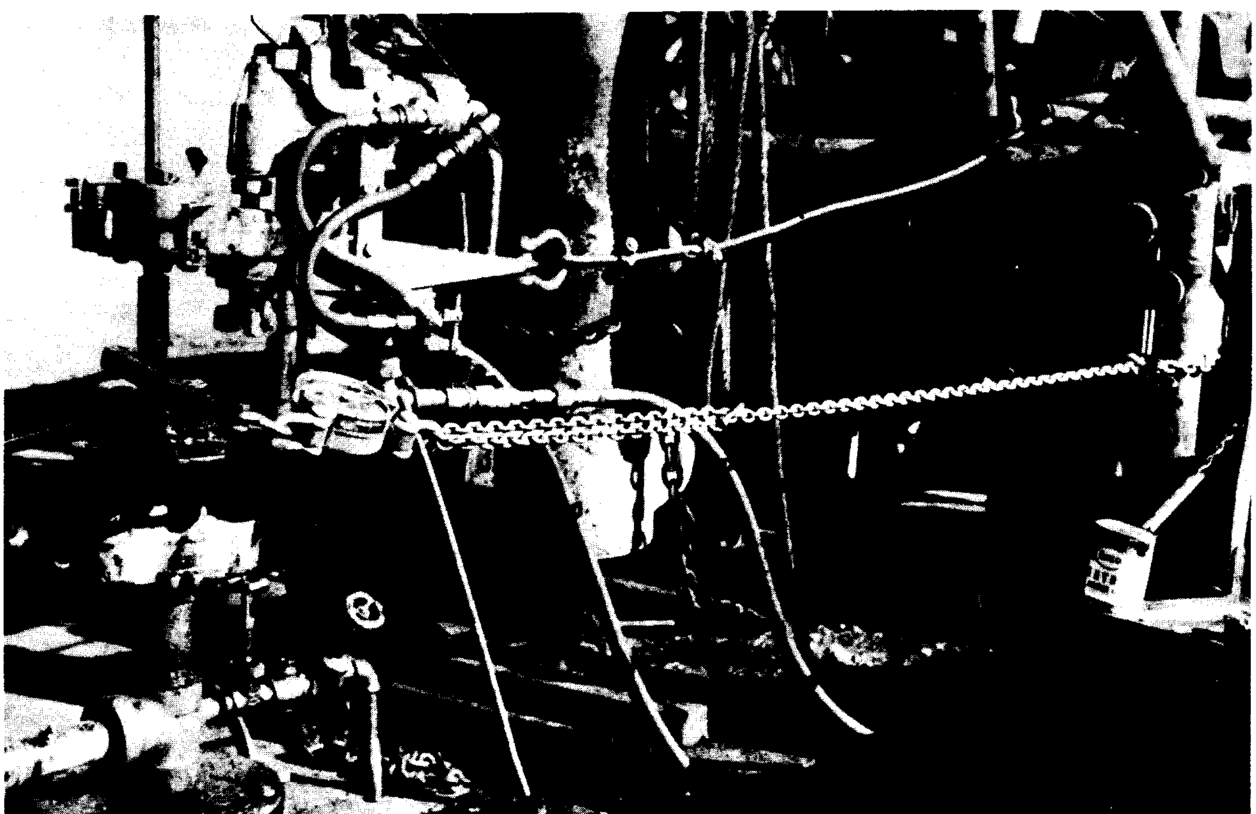


Fig. 4

or in the shop can be rigged up, and they should be devised and regularly used.

The change in class of fit, the change in configuration and the application of rolled threads as outlined above, constitute a major advancement in sucker rod pin design. The benefits inherent in these improvements can only be realized by attention to the following practices:

A. Joints should be kept clean and well lubricated so they will reflect a free running fit to shoulder contact.

b. Pin should be kept shoulder face and coupling face free from bumps, gouges and other imperfections which prevent a metal to metal seal.

c. Power tongs are used in making up joints.

d. Power tongs must be carefully maintained and regularly calibrated; torque out-put should be checked on every job.

e. All rod tools must be kept in good order.

If these practices are rigorously followed, most joint trouble will be at an end.

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