TORQUE VS. LOAD STUDY
FLARED TUBE FITTINGS

Presented at

Society of Automotive Engineers
Spring Meeting – Miami, Florida
26 APRIL 1960

CONVAIR (ASTRONAUTICS) DIVISION
GENERAL DYNAMICS CORPORATION

TORQUE VS. LOAD STUDY

FLARED TUBE FITTINGS

By

C. M. Richards
Senior Design Engineer

CONVAIR (ASTRONAUTICS) DIVISION
GENERAL DYNAMICS CORPORATION
San Diego, California

6 April 1960

At the meeting of this committee last spring in Cleveland, the subject of the torque to be applied in the assembly of tube fittings was brought up for discussion. At that time it was reported that Convair-Astronautics had started detailed research into this subject and would make its findings available when completed to this Committee.

This study is now finished and we wish to thank this Committee for the opportunity to present our findings for your consideration.

At the outset of our program it was decided to approach torque requirements from the standpoint of the required end result, that is, an adequate loading at the sealing interface. Certain assumptions were necessary in establishing the value of this adequate load. In previous tests leading to the development of the copper seal, it was learned that material must be caused to yield at the seal point in order to achieve the degree of mating which would contain helium gas.

Some of you who were with us in St. Louis last Fall may remember my brief report from the floor concerning this development. At the risk of repetition, I would like to recall for you the results of those tests. It was found that of 160 test samples assembled at standard maximum torques only four were able to contain helium gas at 3,000 psi and most failed below 500. When a material with a low yield point, in this case soft copper, was placed at the sealing interface, the same fittings contained pressures of 3,000 to 4,000 psi with only 9 exceptions -- most of which were traced to gross tube defects. In the case of the material combination used in these tests, the fitting had a yield point of 30,000 psi and the tube material of 75,000 psi. Our assumption now is that the minimum torque required to seal helium with a corrosion resistant steel flared assembly is that torque which will develop a stress of 30,000 psi at the sealing interface.

In order to relate the forces required to develop this level of stress to torque at the fitting nut, tests were run on simulated AN-type fitting specimens.

The test articles consisted of a solid bar of corrosion resistant steel machined on one end to the dimensions of the male flared fitting, another solid bar machined to correspond to a standard fitting sleeve and a standard AN 818 nut. These parts were assembled in a manner to simulate a fitting on a tube and installed in a tension test machine. Measured torques were applied with a standard torque wrench and resultant loads were read directly as tension on the machine dial. In this manner all undetermined variables were eliminated and a true measure of the effect of torque was obtained.

Included as a side study in this test, or fringe benefit if you will, was an investigation of a promising new dry lubricant. Previous galling and friction tests had been conducted on a number of materials applied by a high velocity air jet in a fluid vehicle. Such materials as silver, colloidal mica, lead and graphite were tested. It was found that a combination of graphite in a silicon suspension, which could be air dried at normal room temperatures, gave the best combination of properties for the lubrication of tube fittings. The graphite, which was insensitive with liquid oxygen, was milled to a maximum particle size of two microns which left no problems of system contamination should any material become dislodged. This material is imbedded into the pores of the steel by the very high velocity air stream. The resultant surface shows no measurable change in dimension and essentially no removal of material in use. The male thread and the simulated sleeve shoulder of the torque test samples were treated with this graphite and silicon material. There is a promising field here for investigation of pre-lubricated fittings.

With the figures giving the force available as the result of nut torque, it was a simple set of calculations to relate these figures to terms of stress at critical points in the fitting assembly.

A cross section of a standard flared fitting assembly is shown in Fig. 1.

The loads measured in the tensile test machine represent the net force existing to draw the three elements of the fitting together. The nut applies this force to the shoulder of the sleeve, which in turn transmits the force to the back of the tube flare, and brings the inside of the flare up to the nose of the fitting. This force is resolved into a force acting normal to the sealing surface, which is the force which must develop the required 30,000 psi at the seal.

This normal force may be broken down into two components. One axial and one radial. It will be seen that the radial component is resisted by and, therefore, is the result of hoop stresses set up in the sleeve. From this it may be seen that the limiting factor on the force normal to the seal is the hoop strength of the sleeve. We are all familiar with sleeves which have been overstressed and which have become jammed in the nut. For years we have attributed this phenomena to over-torquing. Unfortunately this is not the case. Reference to the figures obtained in this test show that at standard maximum torques in one size, the stress in the sleeve is in the order of 30,500 psi or above the yield strength of the sleeve material and, in the others, is coming dangerously close.

This means that at standard torques without any internal pressure the sleeve may be caused to fail. When an internal pressure is applied the axial component of this pressure acts in the same direction as the force resultant of torque.

Take the case of a 1-in. diameter fitting torqued to 110 ft. -lb. The calculated hoop stress in the sleeve, resultant from the measured axial load of 3,850 lb., is 19,000 psi. When a proof pressure of 6,000 psi is applied, an additional axial load of 4,000 lb. is added to the 3,850 lb. existing and a resultant hoop stress is developed in the sleeve of 38,600 psi. It is obvious that the limited cross section of the nut in this area, which is already under severe bending loads, is carrying the hoop loads beyond the capacity of the sleeve. A similar condition exists in the 5/8- and 3/8-in. fittings.

This analysis indicated that present maximum torque levels are too high, when considered as applied to a pressurized fitting, and the nut is assuming a load for which it was not designed.

Up to this point we have been discussing the stress levels present at normal torques. What do these stress levels mean in terms of the stress at the seal and how do these stresses compare with those required to seal?

The set of curves shown in Fig. 2 demonstrate what is going on inside of a 1/4-in. standard flared fitting. Using the values of net force obtained from the torque tests converted into sealing stress, we find two curves being a maximum and minimum stress level. The minimum torque required to effect a seal is that torque which will produce 30,000 psi at the seal and taken from the maximum stress measured. The maximum torque required to seal is that torque which produces 30,000 psi on the extrapolated minimum curve. The maximum allowable torque is that torque which produces 30,000-psi hoop stress in the cross section of the sleeve.

You will note that in the case of this 1/4-in. fitting the maximum allowable torque is below the maximum required seal. The standard torque as established by AND 10064 ranges from below the minimum required to slightly above. This is a condition peculiar to the 1/4- and 3/8-in. fittings and accounts for the relatively high reliability of these sizes for helium tight joints.

Note also that the maximum torque required to seal is far beyond the standard range and beyond the maximum allowable torque; which is that torque which causes a stress beyond 30,000 psi in the sleeve.

As the size increases this condition becomes worse. For 1/2 in. and up, the maximum allowable torque falls below the minimum required.

The family of curves pertaining to the 1/2-in. size is interesting, particularly, in that it forms the first verification of the stress required to seal. (See Fig. 5-8.)

In the copper seal tests only three specimens sealed helium at standard maximum torque. With the seal in place all 40 specimens held 3,000 psi. You will note that the minimum seal stress curve is in the area of 10,000 psi at standard torques. In the 5/8- and 3/4-in. sizes standard torque produces a seal stress in excess of 10,000 psi.

In the 1-in. specimens it was found necessary to increase torque to 135 ft.-lb. to effect a seal with the copper gasket. Reference to the 1-in. curves shows that this is the point at which the minimum seal stress curve is in the range of 10,000 psi.

The disparity between the allowables and the requirements becomes greater as the fitting size increases, indicating that any seal obtained in the -10, -12 or -16 is a matter of chance rather than designed-in capability. This is, of course, a matter of common knowledge to those faced with sealing high pressure helium. These curves only serve to tell why and indicate corrective measures which may be taken.

Another curve on these slides is labeled maximum nut stress - modified. This curve extends the allowable torque range very considerably. Enough to make sealing within allowable torque values a reality.

Figure 9 shows a cross section through a flared fitting, a tube, a modified sleeve and a nut. This is a standard MS 21921.

From the description of the analysis of the standard assembly certain advantages of this design will be apparent. First, the seal is the same and the dependence of the seal is still upon the allowable hoop stress in the sleeve and nut. Here the similarity ends. The sleeve has been reduced to a vestigial structural element and the nut has become a significant mass. The action of the nut is now to not only contain the sleeve but act to reduce its diameter.

The first advantage of this design is to bring the large cross section of the 21921 nut into action as a hoop stressed member for which it was designed. The effect of this is to reduce the hoop stresses to the 4,000/6,000-psi range at the loads measured in this test. This reduction acts to extend the maximum allowable torque and bring the required torques within the maximum allowable range.

The second advantage may not be so apparent but may be of even greater significance. In the original cross section discussed (see Fig. 1), the action of the sleeve was to expand under hoop tension. This expansion caused the sleeve, already a loose fit on the tube, to move still further away at the heel of the flare. The characteristic failure of flared tubes under vibration occurs, 99 failures out of 100, at the root of the flare. The expansion of the sleeve leaves the tube unsupported at the critical area. Note in the modified design (Fig. 9) the nut exerts a radial force inwardly, which swages the sleeve tightly onto the tube. If this effect appears familiar to you think of the flareless assembly. The geometry and the action are the same.

Test samples of this design have been subjected to vibration while under 3,000-psi helium pressure. There have been no tube failures, at over 200,000 cycles of reverse bending at 75% of the minimum yield stress of the tubing material.

To summarize these test findings, refer to Fig. 10. In this figure the required torque range is plotted with the AND standard range, the maximum allowable range with the standard design and the maximum allowable with the modified design.

In all cases the required range is far beyond both the standard range and the allowable with the standard design, while the modified design permits torques far in excess of requirements.

From this it can be seen that the standard design can not be made to seal helium gas without dangerous over-torquing. Some design modification must be made available for this service.

The design modification proposed here incorporates the following qualifications for meeting the requirements:

A. It produces the required seal stresses at reasonable internal fitting stress levels.

B. It does not affect weight of the assembly.

C. It is entirely compatible and interchangeable with existing installations.

D. It yields measurable improvement in vibration performance.

E. Bursting strengths are increased in the order of 20%.

See Figs. 11 and 12 for tabulated test data from which the foregoing figure charts were prepared.

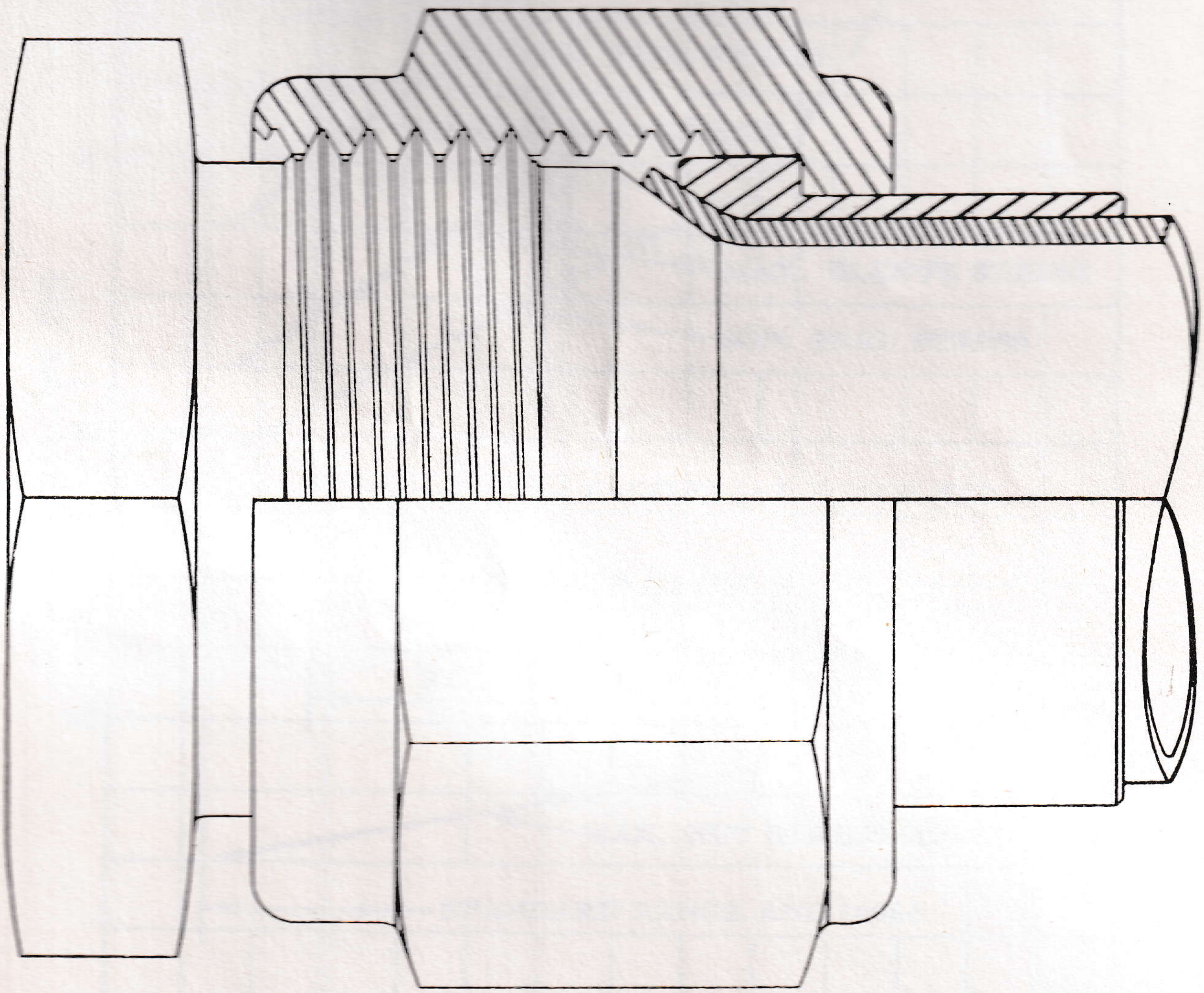
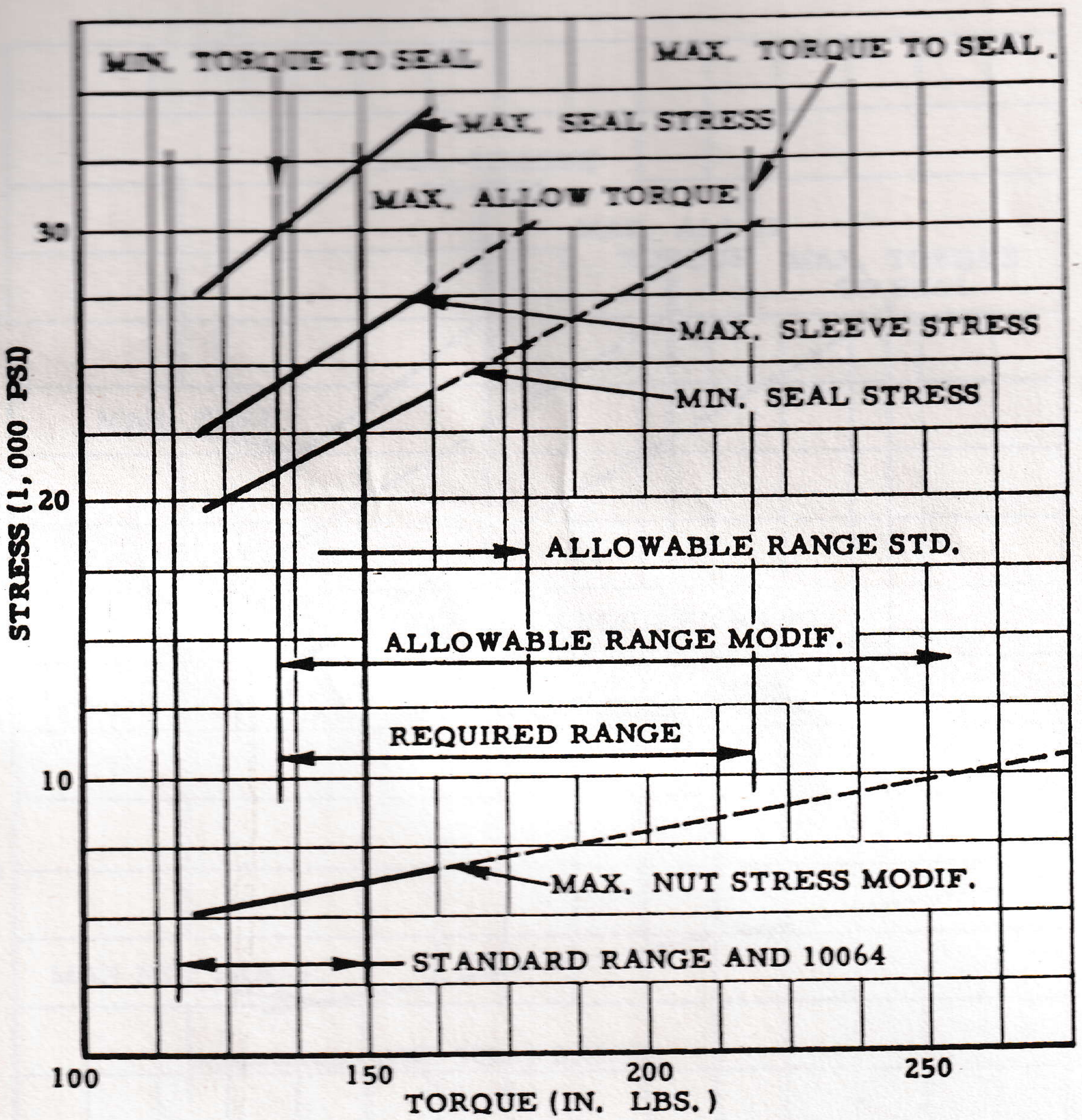
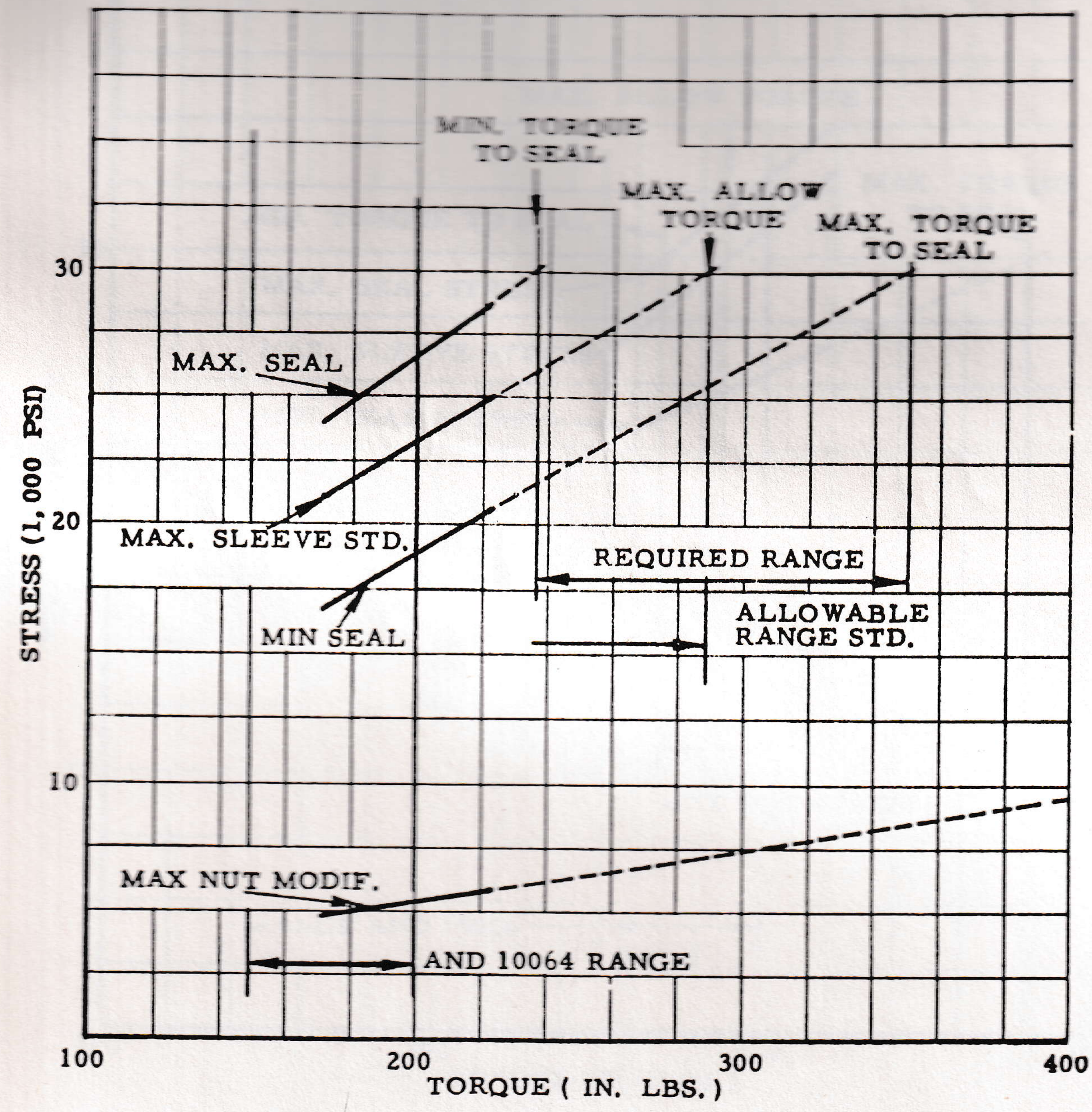


Fig. 1 AN STANDARD FLARED FITTING



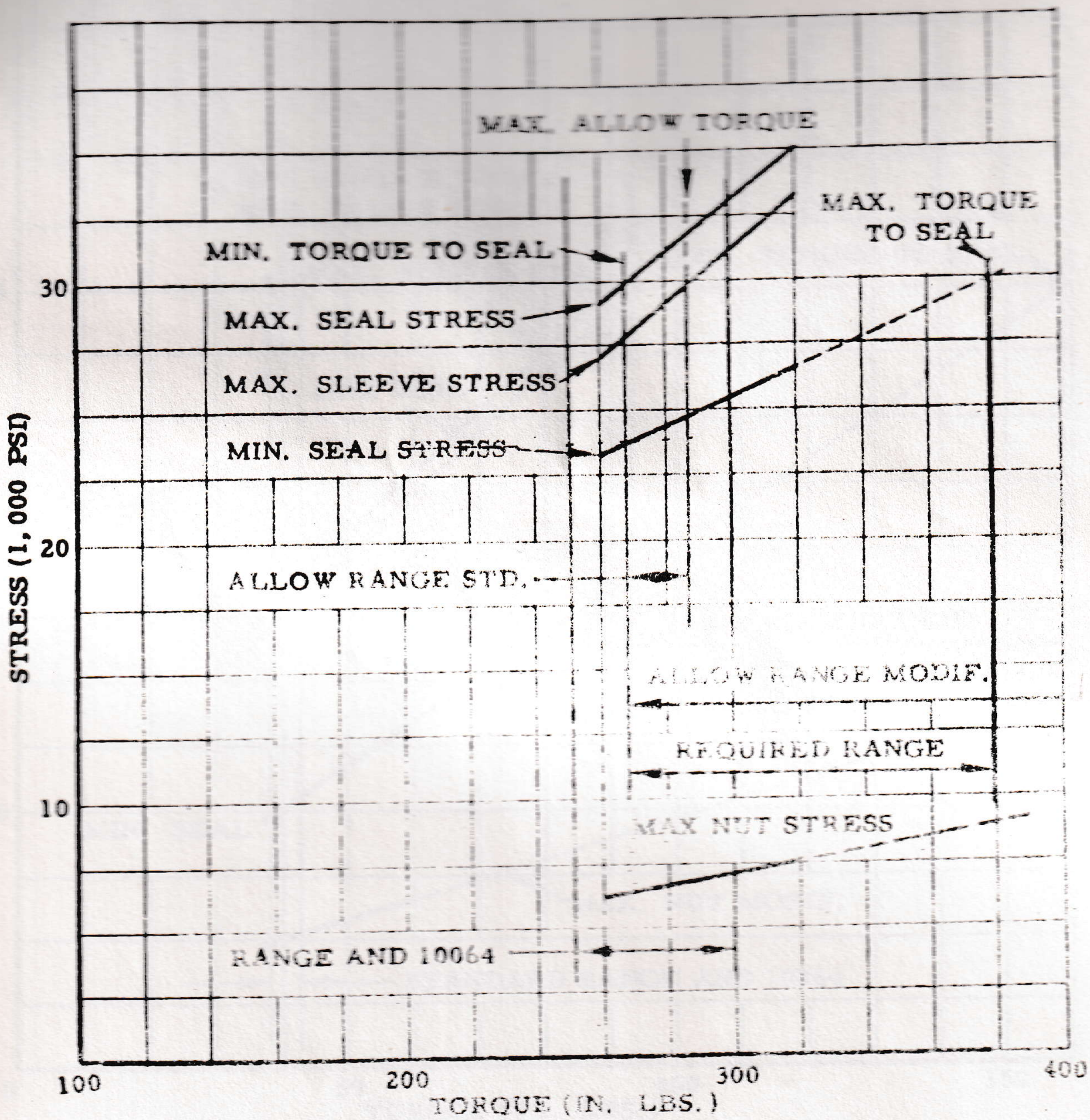
-4 1/4 IN.

Fig. 2 UNIT STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY CRES



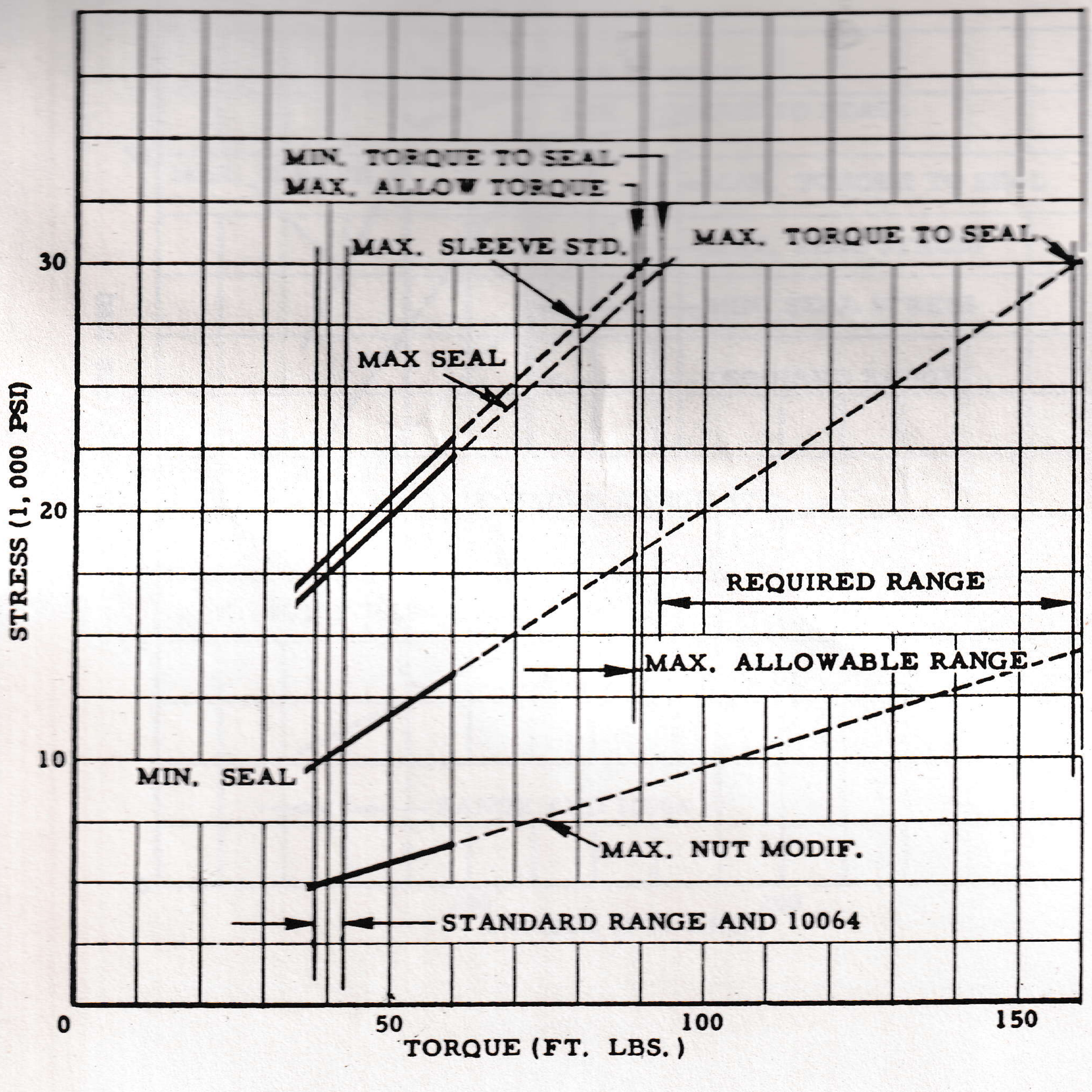
-5 5/16 IN.

Fig. 3 UNIT STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY



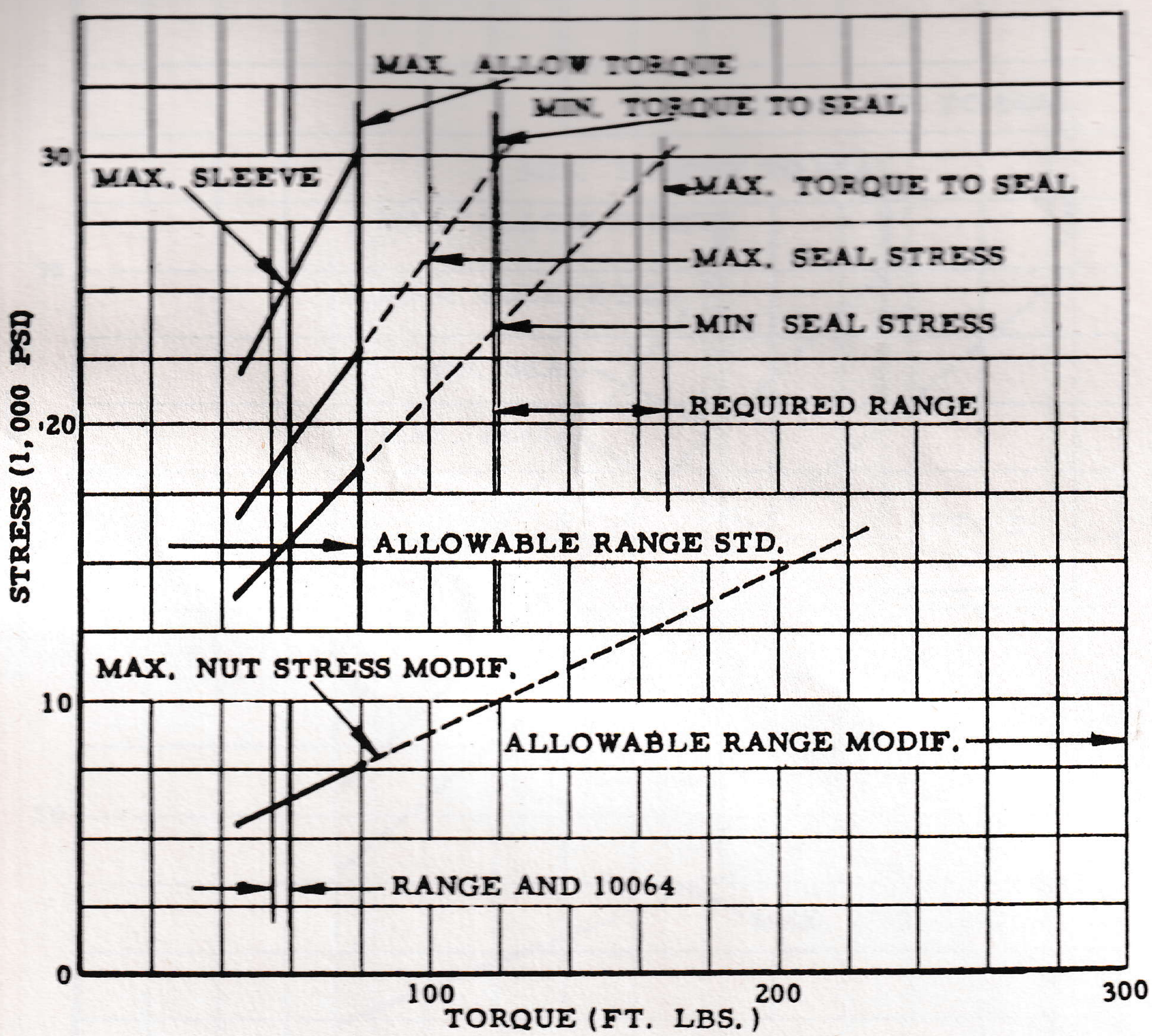
-6 3/8 IN.

Fig. 4 UNIT STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY - CRES



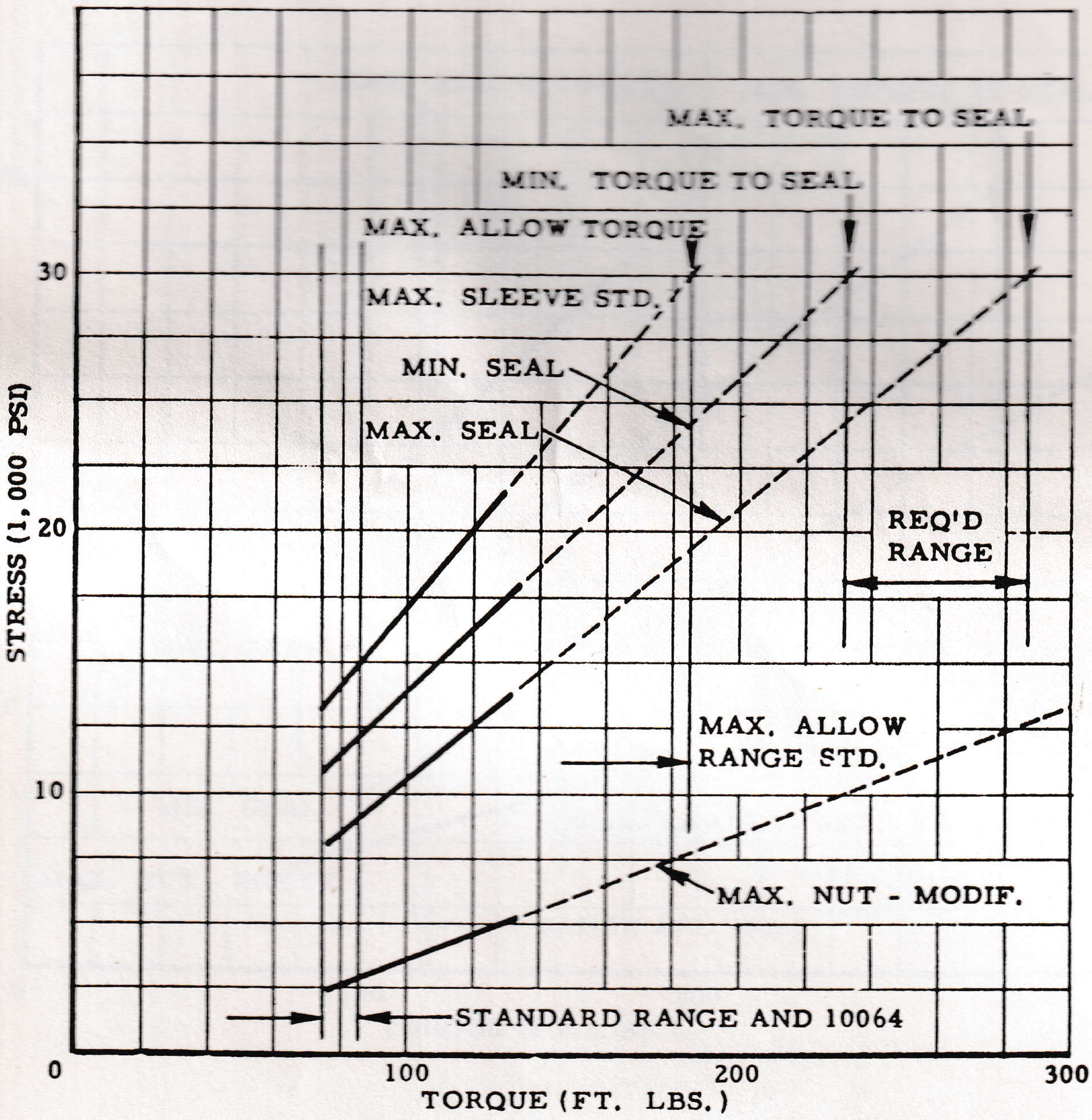
-8 1/2 IN.

Fig. 5 UNIT STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY



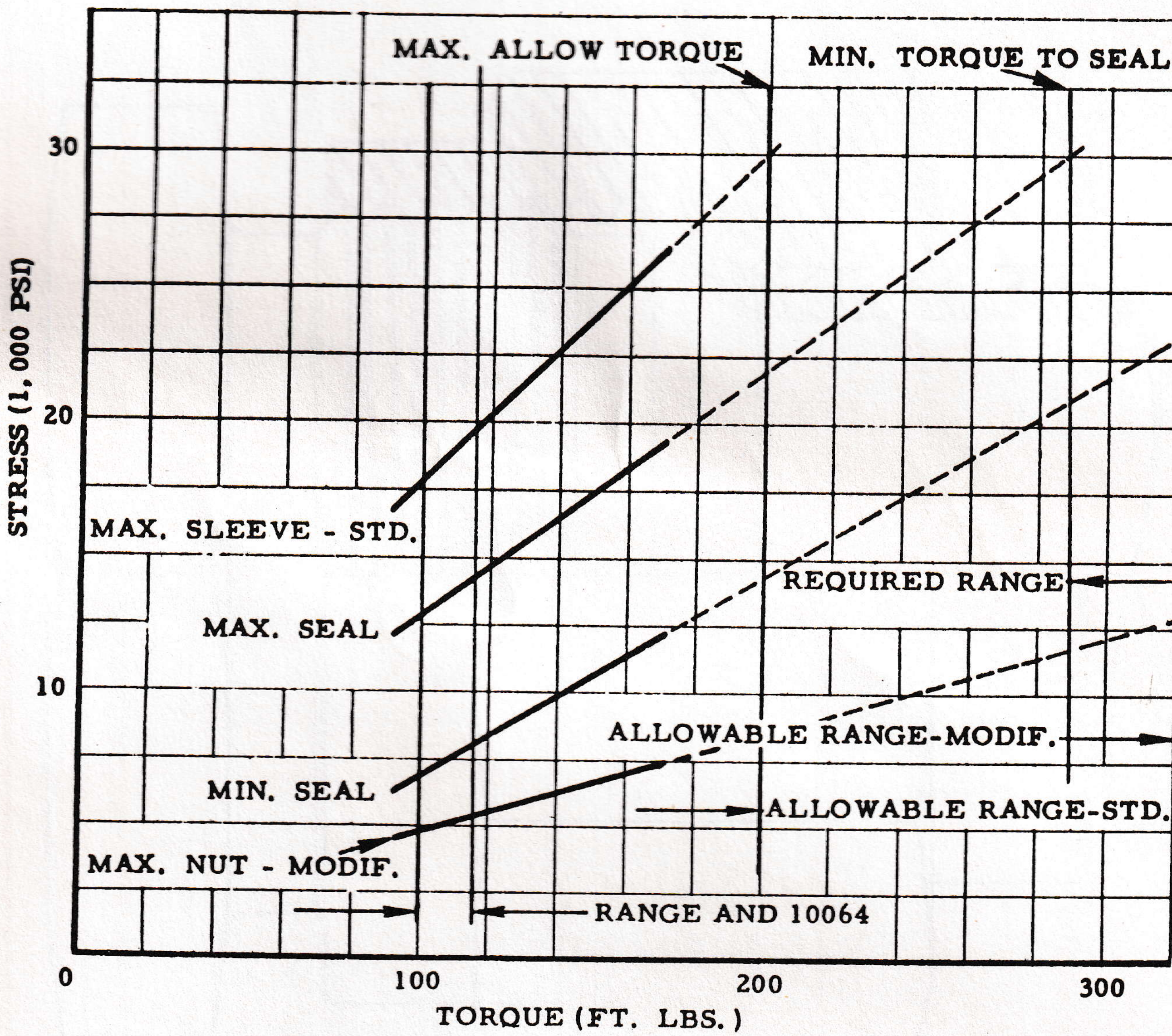
-10 5/8 IN.

Fig. 6 STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY - CRES



-12 3/4 IN.

Fig. 7 UNIT STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY



-16 1 IN.

Fig. 8 UNIT STRESS vs. NUT TORQUE FLARED FITTING ASSEMBLY

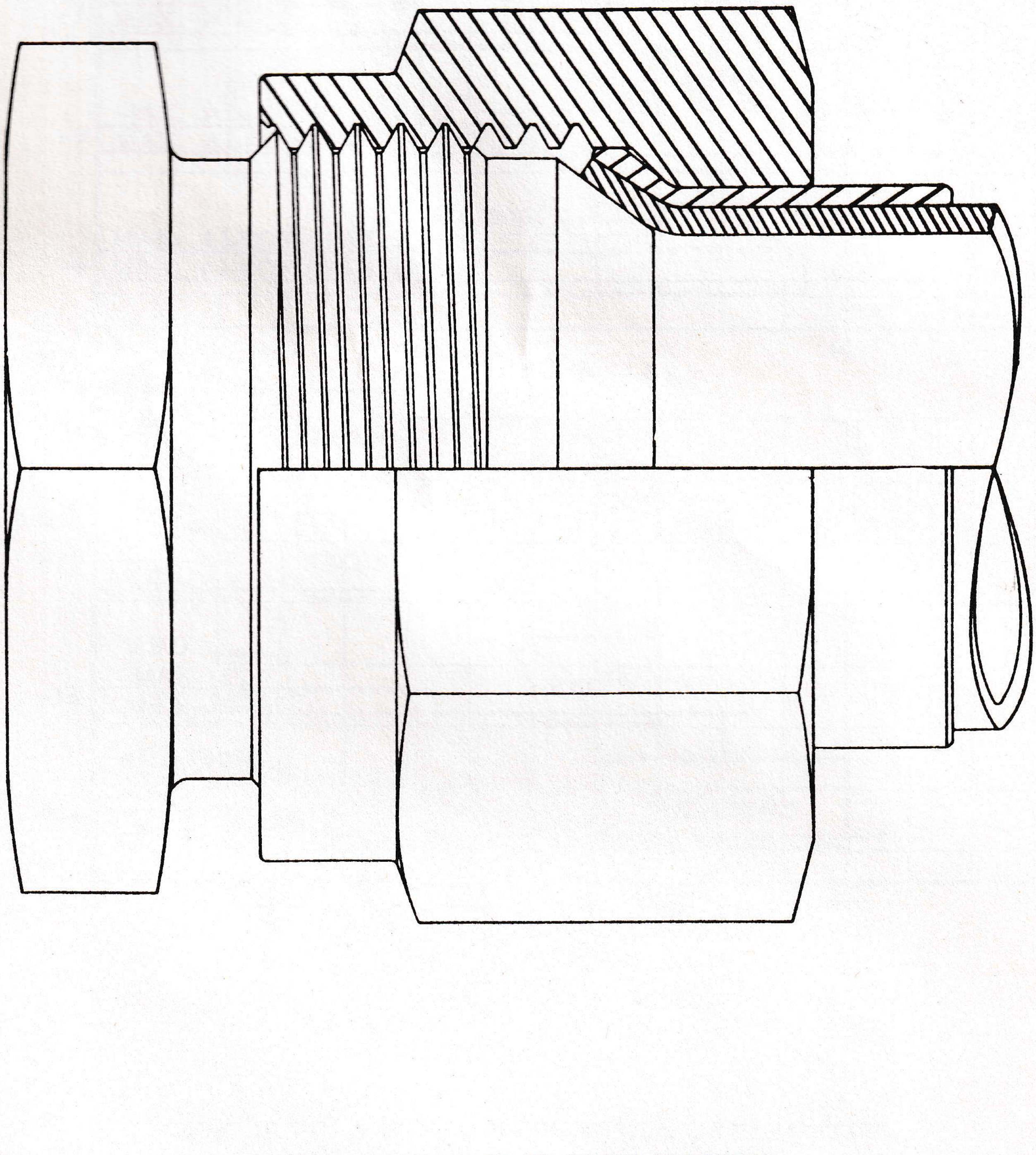


Fig. 9 MODIFIED FLARED FITTING

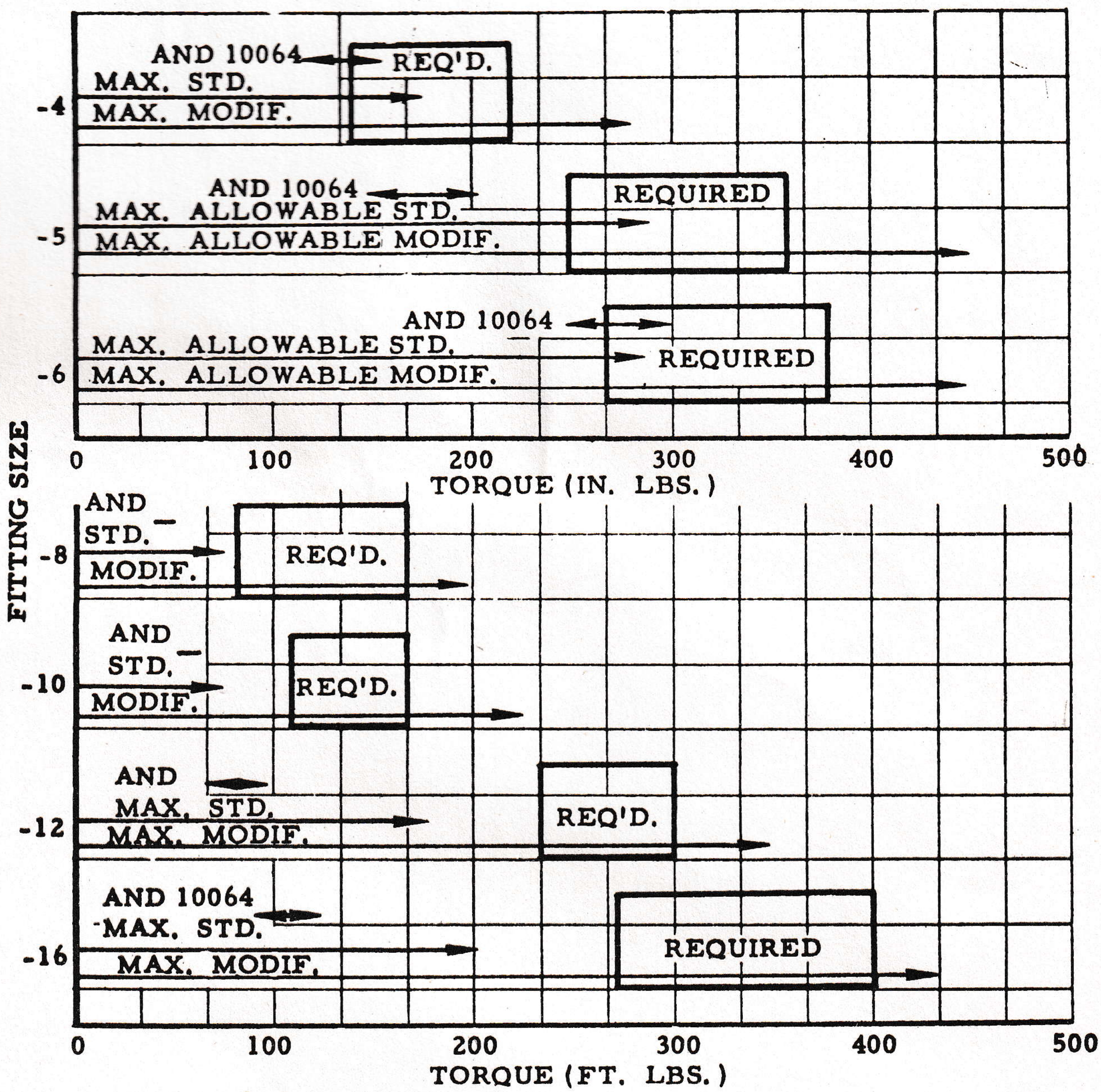


Fig. 10 TORQUE REQUIREMENTS & CAPABILITIES