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ULTRASONIC TORQUE WRENCH

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16. ABSTRACT

Manual and semiautomatic ultrasonic torque wrenches for tightening flared tubing connections for sizes ranging from 0.318 cm (0.125 in.) to 2.54 cm (1 in.) diameter were designed, fabricated, and tested.

During the torquing operation, ultrasonic energy was applied for a predetermined time to the coupling nut of the flared tube connection by means of the wrench head. The vibrational movement of the coupling nut resulted in reduced friction between the mating threads, permitting further tightening without the application of additional torque.

In aluminum, ultrasonic activation of the wrench head during the application of torque produced 27 leaktight assemblies out of 37 as compared to 13 of 32 leaktight assemblies made without the application of ultrasonics. In stainless steel, the application of ultrasonics during torquing produced 54 leaktight connections out of 64 as compared to 41 of 59 torqued without the application of ultrasonics. The application of ultrasonic energy during torquing provided an average of 0.25 rad. (14 deg.) additional rotation of the coupling nut without an increase in measured torque, increased the tensile strains in the coupling nuts, produced higher sealing stresses on the tube flares, and necessitated increased breakaway torque to loosen the assemblies. However, the test showed that the wrench was too bulky and unwieldy to be of practical use.

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Concluded)

LIST OF TABLES

TECHNICAL MEMORANDUM X-53995

ULTRASONIC TORQUE WRENCH

SUMMARY

Manual and semiautomatic ultrasonic torque wrenches for tightening flared tubing connections were designed, fabricated, and tested. The manual wrench was provided with a switch for manual initiation of ultrasonic energy by the operator after a predetermined torque was reached. The semiautomatic wrench was designed for automatic initiation of ultrasonic energy after a predetermined torque was reached. The wrenches were provided with interchangeable 12-point wrench heads to accommodate flared tubing connectors for sizes ranging from 0.318 cm (0.125 in.) to 2.54 cm (1 in.) diameter.

Operation of the wrenches was demonstrated in the tightening of flared tubing connections of 6061-T6 aluminum alloy and CRES 304 stainless steel within the size range noted above. Ultrasonic torquing led to substantial improvement in the leaktightness of the assemblies. When tested at an internal helium pressure of 20×10^6 N/m^2 (3000 lb/in.^{2}), 57 of 101 assemblies which had been torqued using ultrasonics were leaktight as compared to 35 of 91 leaktight assemblies which were torqued without using ultrasonics. Ultrasonic activation effected additional rotation of the nut without increasing the applied torque, increased the tensile strains in the nut, and produced higher sealing stresses on the flared tubes. The breakaway torques required to loosen the nuts were also greater for the assemblies torqued using ultrasonics.

The wrenches were designed so that the wrench head provided ultrasonic vibration in the bell mode — the coupling nut was caused to vibrate like a ringing bell. This ultrasonic vibration resulted in reduced friction between the surfaces of the mating threads, permitting further application of sealing stress without increasing the torque.

The above operations were conducted in a laboratory environment using a bench vise to provide the opposing torque where a standard open-end wrench would be used on the union hex in shop operation. Even under these conditions it was difficult, if not impossible, for the operator to get a "feel" of the wrench. The wrench system as designed proved to be too bulky, heavy, and unwieldy to the point of having no practical application without additional design refinement.

INTRODUCTION

Flared tubing connectors have been a source of continuing leakage problems in fluid systems of missiles and space launch vehicles. Much expense and effort have been expended to solve this problem. Additional effort has been directed toward improving tubing connectors, tubing flares, materials, and processes. Because of the limited success attained in eliminating the leakage problem the search continued for methods and techniques that would reduce or eliminate it.

The concept of using ultrasonic energy to assist in torquing flared tube connections was conceived during a 1964 conference presentation of the effects of ultrasonic energy on aerospace materials. A limited study was made to determine if the use of ultrasonic energy would prove useful in reducing leakage. It was found that the application of ultrasonic energy in research efforts had demonstrated the phenomena of reducing friction between mating surfaces and allowing material deformation without significant change in mechanical properties. It was assumed that the phenomena generated by the application of ultrasonic energy would assist in obtaining leaktight connections in flared tubing connections.

First, the feasibility of ultrasonic wrenching was demonstrated, approaches for ultrasonic coupling to the threaded tube connection were evaluated, effective ultrasonic power levels were determined, and design criteria for practical ultrasonic wrenching tools for flared tubing connections were evolved. The work covered in this report involved the design, fabrication, and evaluation of a prototype manual ultrasonic flared-tubing-connection wrench followed by the design, fabrication, and evaluation of a prototype semiautomatic wrench for the same purpose.

The type of flared tubing connection of primary concern is illustrated in Figure 1. The components of the flared tubing connection are a flared tube, a compression sleeve, a union, and a coupling nut. These were designed and fabricated to specifications as noted on the sketch. The flared tube sizes (outside diameters of the tubes) of immediate interest were within the range of 0.318 cm (0.125 in.) to 2.54 cm (1 in.). The connection is sealed by rotating the nut to compress the tube flare against the union bevel. A large portion of the torque applied for sealing is required to overcome frictional forces acting on the mating thread surfaces between the internal shoulder of the nut and the face of the sleeve and between the sleeve and the tube flare. With standard assembly techniques, the dissipation of applied torque because of friction can vary from one assembly to another, resulting in uncontrollable compression in the seal area and unpredictable sealing. With reduction in the frictional forces,

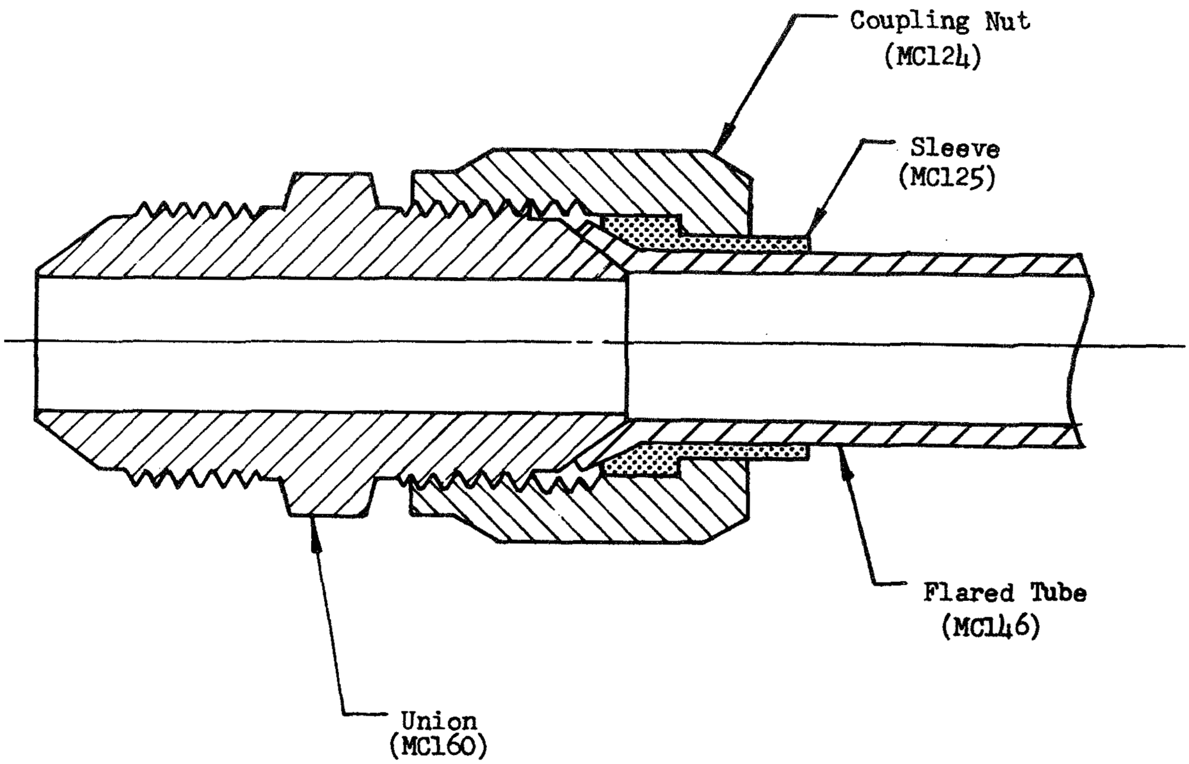


Figure 1. Flared tubing connection.

the applied torque is evidently more effectively and consistently used to compress the tube flare, and the probability of obtaining reproducibly leaktight seals is increased.

Previous laboratory work had shown that friction between metal surfaces can be reduced by appropriate application of ultrasonic energy. In addition, evidence exists that the vibratory energy may transiently improve the deformability of metallic materials.

Since the feasibility of ultrasonic wrenching of flared tubing connections was by no means obvious, laboratory-type ultrasonic transducer-coupling systems and associated wrenching hardware were assembled to activate the threaded connections in each of three vibratory modes (axial, torsional, and bell) at frequencies of 15 kHz (15 kc/sec) and 28 kHz (28 kc/sec), and over a range of ultrasonic power levels up to about 800 electrical watts input to nickel-stack transducers. The ultrasonic systems and parameters were evaluated in the tightening of flared tubing fittings for 0.635-cm (0.250-in.) 6061-T6 aluminum alloy tubing, and fittings for 1.27-cm (0.50-in.) CRES 304 stainless steel tubing, using a range of torque levels.

This work demonstrated that ultrasonic activation of the wrench head permitted additional angular rotation of the fittings without increasing the torque level. Increased tightening was also evident from the fact that non-ultrasonic breakaway torque required to loosen the fittings was higher for the ultrasonically tightened than for the non-ultrasonically tightened assemblies. Helium-leak tests at a pressure of 24.1×10^5 N/m^2 (350 lb/in.^{2}) gage pressure indicated greater reproducibility of leaktight connections. Repeated ultrasonic wrenchings of the same connection at the same power and torque levels resulted in less additional rotation than occurred on initial assembly, suggesting that local yielding associated with ultrasonic metal working had effected a better "fit" between the components. This was confirmed by metallographic examination of sectioned connections, which revealed greater contact area on the tube flare with the ultrasonic assemblies. Hardness measurements revealed a slight work-hardening effect from the vibratory energy application.

Comparison of the results obtained in tightening connections with the various modes of ultrasonic activation showed that the above effects were obtained to a significant degree with both the axial and the bell modes; the torsional mode was less effective. The bell mode was selected for activation of the end-item wrenches, since it is more adaptable to wrenching flared tubing connections because of their ability to be excited radially in flexure.

There appeared to be no effect of ultrasonic frequency within a practical range. Frequencies of 15 kHz (15 kc/sec) and 28 kHz (28 kc/sec) appeared to be equally effective, and other work at frequencies up to 60 kHz (60 kc/sec) had also revealed no significant effect. Operating frequencies for production units may therefore be selected primarily on the basis of practical physical size.

Experimentation over a range of ultrasonic powers indicated that, for a given size connection, there is a maximum power level beyond which the components may be damaged. The range of powers that produced useful tightening effects was about 50 to 150 W for 0.635-cm (0.250-in.) aluminum components, and 200 to 400 W for 1.27-cm (0.50-in.) stainless steel components. Considering the flared surface areas of the two types of tubes, the indicated power density was about 165 W/cm^2 (1065 W/in.^{2}) for the aluminum, and about 159 W/cm^2 (1025 W/in.^{2}) for the steel tubing. These comparable power densities seem to imply that only a minor fraction of the applied power is used to provide material deformation, which depends on the yield strength of the material, and that a major portion of the power acts to overcome friction.

The above power figures are electrical watts input to nickel transducers, which were used in the experimental arrays. Ceramic transducers would be more practical for production-type ultrasonic wrenches, since they are less massive. Moreover, ceramics such as lead zirconate titanate have electromechanical conversion efficiencies roughly twice as high as those of nickel; the transducer-coupling systems can therefore be designed with about half the input power capacity, i.e., to provide power densities of about 85 W/cm^2 (550 $W/in.^2$) of tube flare surface.

On the basis of the experimental effort, design criteria were evolved for a practical ultrasonic wrench for flared tubing connections covering the above size range, as follows:

1. Compatibility with reasonable wrench geometries, particularly with regard to size and access to restricted locales.

2. Mechanically removable wrench heads.

3. Operational frequency within the range of 20 to 60 kHz.

4. Ceramic transducer assemblies having a power capacity of about 500-W input.

5. Ultrasonic coupling system to excite the wrench head in the bell mode.

6. Force-insensitive transducer-coupling mounting system to provide frequency stability and minimum vibratory energy loss to the wrench body and torque indicating system under variable force application.

7. Integral static torque indicator, as on ordinary torque wrenches.

8. If possible, an "additional degrees totalizer," for assessment of additional tightening achieved with ultrasonics.

9. Compact solid-state frequency converter, to maximize portability and minimize weight, with an output of approximately 500 high frequency electrical watts.

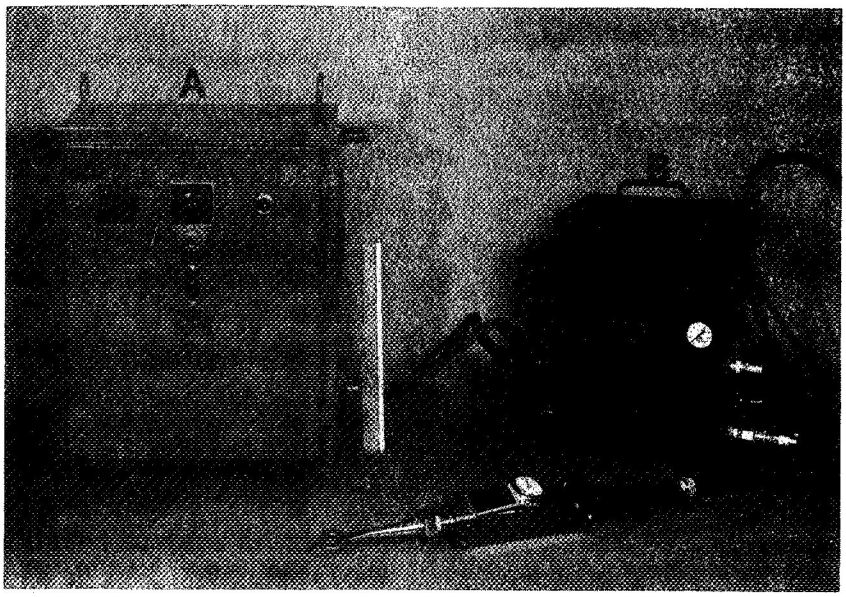
10. Ultrasonic power switch conveniently located, possibly on the wrench handle, and a power-step switch on the frequency converter calibrated in terms of tube coupling size.

11. Cables between the frequency converter and transformer and between the transformer and wrench of convenient length to permit flexibility of operation.

On the basis of the above criteria, a manual ultrasonic wrench was designed, fabricated, and evaluated. Experience with the wrench indicated the desirability of certain design modifications, and a second wrench was designed and fabricated to incorporate such changes and to provide certain semiautomatic features.

DESCRIPTION OF MANUAL ULTRASONIC WRENCH

A prototype ultrasonic flared-tube-connection wrench for manual operation was designed and fabricated based on the design criteria established. The complete wrench, shown in Figure 2, consisted of the wrench assembly, frequency converter, junction box, and interconnecting cables. A calibration adapter incorporating a 26-Ω resistive load was provided to serve as a reference standard for checking the output of the frequency converter.



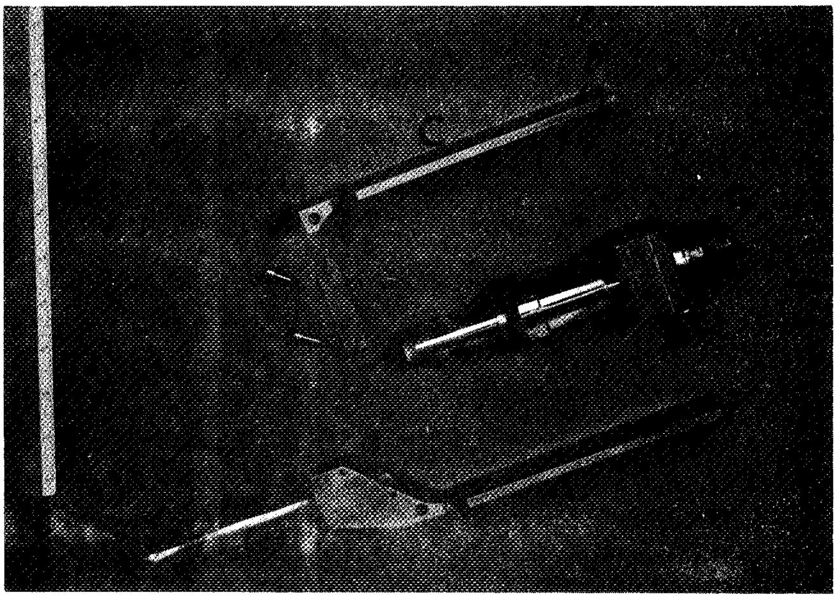
Legend
A. Frequency Converter
B. Junction Box
C. Wrench Assembly

Figure 2. Manual ultrasonic wrench.

Wrench Assembly

The wrench assembly incorporated a 28-kHz (28 kc/sec) ceramic (lead zirconate titanate) transducer assembly having a power rating of 500 W pulse duty (3 sec on, 3 sec off) and 300 W continuous duty. The transducer assembly was supported in the wrench body via a force-insensitive mount, which minimized frequency shift and loss of vibratory energy to the torque wrench beams and indicating mechanism under the variable applied torque loads. Air-cooling channels were provided in the transducer assembly to prevent overheating and depolarization of the ceramic transducer.

The transducer assembly, steel mounting block, and steel side deflection beams, comprising the major components of the wrench body, are shown in Figure 3. A torque indicator meter was mounted on a reference plate extending from the support plate for the transducer and side rails. The dial plunger was spring-loaded against one side rail and thus provided a deflection indication that was proportional to the side rail deflection. The dial was calibrated from 0 to 170 N-m (0 to 1500 in.-lb). A thumb switch was provided on the handle to trigger the ultrasonic activation.



Legend
A. Transducer Assembly
B. Mounting Block
C. Side Deflection Beams

Figure 3. Components of ultrasonic wrench body.

Two interchangeable steel wrench heads of open-end, 12-point design were provided for tightening connections associated with tubing sizes of 0.635 cm (0.25 in.) and 1.27 cm (0.50 in.). Subsequently, wrench heads for other size connections, fabricated of titanium alloy, were provided for use with either the manual or the semiautomatic wrench. The wrench heads were designed to provide a 50 percent margin of safety over the specified torque values as stipulated in Specification MC-245. These data for the various tubing sizes are presented in Table 1.

TABLE 1. SPECIFICATION AND DESIGN TORQUE VALUES FOR ULTRASONIC WRENCH HEADS

Tube Diameter, cm (in.)	Specification[a] Torque Ranges Aluminum, N-m (in.-lb)	Steel, N-m (in.-lb)	Design Maximum Torque, N-m (in.-lb)
0.318 (0.125)	3.39 - 4.52 (30-40)	6.78 - 9.04 (60-80)	13.6 (120)
0.635 (0.250)	7.91 - 13.6 (70-120)	15.3 - 17.0 (135-150)	25.4 (225)
0.953 (0.375)	14.7 - 20.3 (130-180)	30.5 - 33.9 (270-300)	50.9 (450)
1.27 (0.50)	33.9 - 45.2 (300-400)	50.9 - 56.5 (450-500)	84.8 (750)
1.90 (0.75)	73.5 - 90.4 (650-800)	102 - 113 (900-1000)	170 (1500)
2.54 (1.0)	102 - 113 (900-1000)	136 - 158 (1200-1400)	170 (1500)

a. Specification MC-245

The wrench heads were readily interchangeable; a locating pin on the wrench body coupling face and a mating hole on the wrench head coupling face permitted precise alignment, and a retaining nut was tightened over the joint with a torque of 88.2 to 101.8 N-m (65 to 75 ft-lb). A 0.076-mm (0.003-in.) annealed copper shim was placed between the body and the head to ensure efficient transmission of vibratory energy across this interface.

The wrench body, from the hose connection on the handle to the coupling face, was 0.3683 m (14.5 in.) long; and each wrench head, from its coupling face to the center of the dodecagon, was 0.13335 m (5.25 in.) long. This provided an overall length of 0.50165 m (19.75 in.). The weight of the wrench assembly was approximately 5 kg (11 lb).

The frequency converter, for converting ordinary 60-Hz (60-cycle) electrical power to high frequency electrical power and delivering it to the transducer, was a solid-state (silicon-controlled rectifier) unit. The output frequency of the unit was continuously variable over the range from 27 to 29.5 kHz (27 to 29.5 kc/sec), so that it could be adjusted to match the resonant frequency of the wrench assembly with different wrench heads attached. The converter was designed to operate from a standard 120-V, 60-cycle, single-phase power line with a current of 10 A. Its maximum output was 500 W. The unit incorporated a power selector on the front panel, calibrated in terms of tubing diameter, and a built-in solid-state timer to provide a power pulse adjustable between 3 and 5 sec.

In the circuit between the frequency converter and the transducer was a junction box that housed the impedance-matching network, transformer, spark gap, and inductance coils, as well as the controls for providing clean, filtered, cooling air to the transducer (air manifold, pressure regulator, and pressure gage). The cooling air requirement was 0.005 standard m^3/sec (1 ft^3/min) at a pressure of 3.5×10^4 to 7×10^4 N/m^2 (5 to 10 lb/in.^{2}).

The cables for transmitting the ultrasonic power and the necessary control signals from the frequency converter to the junction box and from the junction box to the transducer were enclosed in lightweight, rubber-covered metallic tubing. This tubing also carried the cooling air for the transducer. The high voltage cable for coupling the junction box to the transducer was approximately 3 m (10 ft) long. Two low voltage cables were provided for alternate use in coupling the frequency converter to the junction box; one of these was 0.6 m (2 ft) long for convenient laboratory use of the wrench, and the other was 17 m (55 ft) long to permit remote operation.

The wrench was designed for simple manual operation in the tightening of flared tubing connections. With the appropriate wrench head installed on the wrench body, the master switch on the frequency converter is turned on and, after a short warm-up period, the power selector switch is set for the appropriate size tubing. The resonance controls inside the frequency converter cabinet are adjusted to the resonant frequency of the wrench assembly, as indicated by peak deflection in the frequency meter. The wrench is now ready for use. The flared-tube-fitting coupling nut is tightened to the desired torque level without ultrasonic activation. At this point, the thumb switch is depressed triggering the ultrasonic power pulse. This causes an instantaneous drop in indicated torque, and the nut is immediately tightened further until the specified torque is again reached. The ultrasonics automatically switches off at the conclusion of the preset pulse time interval, and the torque can then be relaxed.

PERFORMANCE OF MANUAL ULTRASONIC WRENCH

Evaluation of the performance of the manual wrench involved the torquing of 0.635-cm (0.250-in.) 6061-T6 aluminum alloy and 1.27-cm (0.50-in.) CRES 304 stainless steel tubing connections, which were selected as representative of the size range of interest. The additional rotation obtained with ultrasonic activation was measured, helium leak tests were performed on the assembled connections, and the non-ultrasonic breakaway torque required to loosen the assembled units was measured.

The torquemeter, which was incorporated on the wrench body, and which provided indication of the torque applied during tightening, was calibrated using dead weights. The calibration was checked by placing one end of a short length of hexagonal metal stock into the socket of a commercial torque wrench and the opposite end into the ultrasonic wrench head, and applying torque via the ultrasonic wrench. The reaction torque indicated on the scale of the commercial wrench coincided with the torque indicated on the ultrasonic wrench meter within the accuracy of line widths. The indicated torque values were therefore accurate within a fraction of a newton-meter.

The ultrasonic wrench, without ultrasonic power, was also used for breakaway torque measurements.

Special pressure fixtures were designed and fabricated for helium leak testing, which was carried out at pressures of 0, 68×10^5, and 20×10^6 N/m^2 (0, 1000, and 3000 $lb/in.^2$).

During initial evaluation of the wrench, a combined total of about 140 tubing connections of the two sizes were assembled, either with or without ultrasonic activation. The ultrasonic torquing provided several degrees additional relative rotation between the nut and the union. In helium leak tests, only about 15 percent of the non-ultrasonic control assemblies and about 30 percent of the ultrasonically wrenched assemblies were leaktight. Analysis of the performance of the wrench indicated that it was functioning as intended, and it appeared that the difficulty was associated with the fitting components.

Final evaluation of the wrench was carried out with 0.635-cm (0.250-in.) aluminum MC components and 1.27-cm (0.50-in.) stainless steel MC components. The aluminum assemblies were tightened at torque levels ranging from 7.91 to 15.82 N-m (70 to 140 in.-lb), and the stainless steel assemblies were tightened at torque levels of 50.85 and 56.50 N-m (450 and 500 in.-lb). For the ultrasonically tightened assemblies, the ultrasonic pulse time was 3 sec; ultrasonic power was generally 50 W for aluminum and 150 W for stainless steel.

As in the earlier work, many of the assemblies leaked at elevated pressure; but, at both pressure levels and for both materials, a substantially greater percentage of the ultrasonically wrenched assemblies than control assemblies were leaktight. Among the aluminum assemblies, leaktighteness at the maximum pressure of 20×10^6 N/m^2 (3000 $lb/in.^2$) was achieved with 16 of 36 of the ultrasonically tightened units and only 4 of 60 of the control units. For the stainless steel at the same pressure, 14 of 48 of the ultrasonic and 10 of 42 of the non-ultrasonic units were leaktight. Thus the ultrasonic wrenching produced significantly increased incidence of leaktight assemblies.

The average additional rotation under ultrasonic influence for the leaktight assemblies ranged from 0.05 to 0.26 rad. (3 to 15 deg) for the aluminum and from 0.7 to 0.25 rad. (4 to 14 deg) for the stainless steel connections.

The non-ultrasonic breakaway torque valaues averaged 43 percent higher with the aluminum and 9 percent higher with the stainless steel for the ultrasonically wrenched than for the non-ultrasonically wrenched assemblies, confirming the improved tightening obtained.

Because of the different response to ultrasonic torquing of assemblies metallographic examination of selected assemblies was performed. This work involved nine aluminum assemblies, four with commercial unions and five with MC unions. Microhardness measurements revealed that the MC sleeve and union were harder than the MC flare, but the commercial union was softer than the flare. The effect of this difference in hardness was evident in photomicrographs of sections of each type of assembly. In assemblies containing the commercial union, the union, being the softest component in this assembly, was slightly deformed at the inside tip. No such deformation occurred in assemblies made with the MC unions; in this case, the sleeve penetrated into the back of the softer flare. It was concluded that when the union is softer than the flare, the union yields and permits a greater contact area between the sealing surfaces of the flare and the union opposite the point of penetration of the sleeve, and extended intimate contact between the sealing surfaces does not occur.

Evaluation of the manual ultrasonic wrench verified its performance for the intended purpose and confirmed the effects of ultrasonic application during tightening of flared tubing connections, as evidenced by the increased relative rotation between the nut and the union, the greater percentage of leaktight assemblies under pressure, and the higher breakaway torque required to loosen the assemblies.

DESIGN AND FABRICATION OF SEMIAUTOMATIC WRENCH

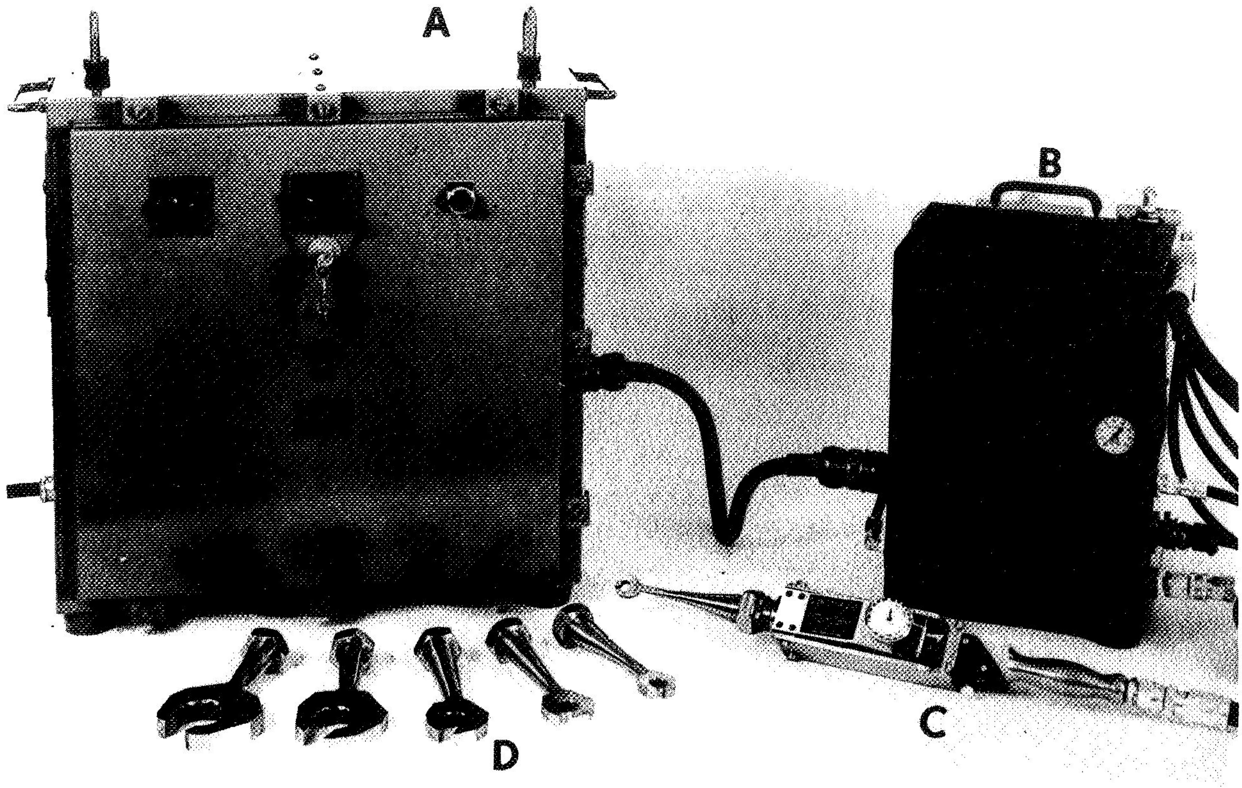
In preliminary use, the operators found the manual wrench to "feel" heavier than standard torque wrenches and to be rather unwieldy to manipulate. It appeared desirable to reduce the weight of the assembly and to redistribute the mass so that the center of gravity would be located closer to the handle. In addition, it was desired that the wrench incorporate automatic triggering of the ultrasonic pulse, so that the operator would provide only torquing effort.

The second wrench, therefore, was designed and fabricated with consideration for these modifications, at the same time fulfilling the design criteria. Modifications were made to the junction box and the frequency converter as well as to the wrench assembly. The complete equipment assembly is shown in Figure 4.

Two avenues were available for reducing the weight of the wrench assembly; redesign of the wrench to make it smaller and to operate at a higher frequency, and fabrication of the wrench components of lighter-weight materials.

The advisability of fabricating a smaller, high-frequency wrench for the smaller size connections was considered. As the operating frequency is increased, the physical size of the transducer and coupling members is reduced because of shorter wavelength. A design was projected for a 40-kHz (40-kc/sec) unit, which would weigh less than about 0.9 kg (2 lb), could be used with connections for tubing sizes up to 0.953 cm (0.375 in.), and could accommodate a maximum torque of 28.3 N-m (250 in.-lb). A wrench of this size, however, would be impractical for the larger size fittings, and action in this direction was discontinued.

With respect to weight reduction on the existing wrench design, only limited modifications could be made because the major components (transducer assembly, side beams, and torque indicator support plates) for the wrench



Legend

A. Frequency Converter
B. Junction Box
C. Wrench Assembly
D. Interchangeable Wrench Heads

Figure 4. Ultrasonic semiautomatic wrench assembly.

body had been fabricated at the same time and of the same material (steel) as those of the first wrench. The wrench heads had not yet been fabricated, and it was decided to make these components of 6Al-4V titanium alloy, which would reduce their weight by about half and at the same time shift the center of gravity somewhat closer to the handle.

Three sets of titanium alloy wrench heads were required; two sets comprising six heads of the open-end, 12-point design covering the size range of fittings for 0.318-cm (0.125-in.) to 2.54-cm (1-in.) tubing, and one set of five heads of closed-end, 12-point design. After fabrication, the wrench heads were checked for frequency response and were found to fall within the desired range of 27.5 to 29 kHz (27.5 to 29 kc/sec).

A further modification to the ultrasonic wrench was the incorporation of a device for automatically triggering the ultrasonic pulse at a preset torque level. This device involved an on-off switching arrangement in which the switch was activated by deflection of the side rail under torque. A knurled contact screw was threaded through the side rail. The screw thread of the micrometer-like device was selected so that four revolutions would advance the screw a distance equivalent to the lateral deflection of the side rail under the maximum expected operating torque. The face of the micrometer contact screw cap and barrel was calibrated in inch-pounds of torque. In operation, the control signal microswitch was triggered by a taper on the micrometer screw extension. As the side rails deflected, the tapered section of the rod was brought into coincidence with the microswitch arm, forcing switch closure.

Figure 5 shows the wrench assembly with the micrometer screw and with a set of open-end wrench heads.

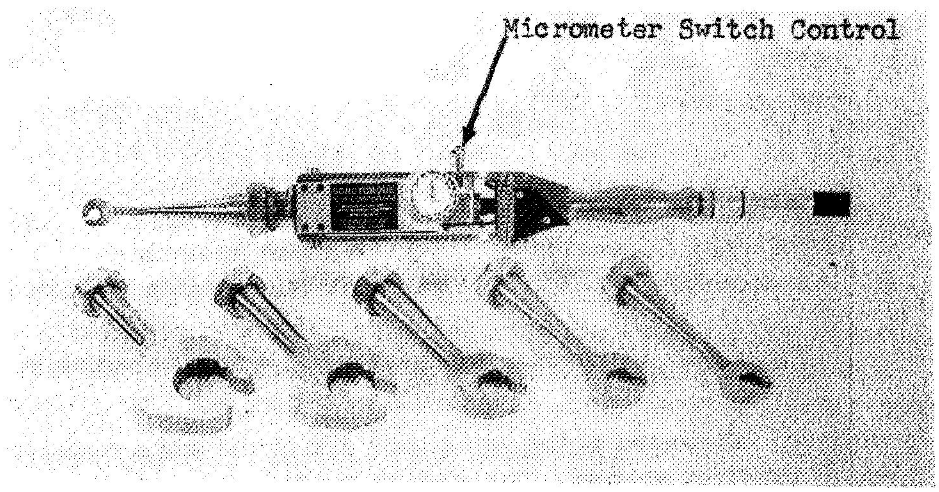


Figure 5. Ultrasonic wrench assembly with wrench heads.

The frequency converter for the second wrench was the same in external appearance and basic design as that provided with the manual wrench, but operating characteristics were improved to provide a more flexible drive unit. Automatic frequency control was incorporated to eliminate the necessity for operator adjustment to resonance each time a different-size wrench head was installed. A signal from the power transmission line to the transducer synchronized the frequency of a free-running, constant-amplitude, solid-state multivibrator circuit so that the converter would operate at the frequency of maximum power delivery from the wrench head into the tube fittings. The timer provided for ultrasonic power pulse durations variable from 3 to 5 sec.

During initial checkout of the modified frequency converter, it was found that the inrush of current with initiation of the ultrasonic pulse tended to trip the protective circuit breakers, and a controlled ramp-type power buildup was incorporated. The oscillator control-components were also altered to reduce the frequency range, which was broader than necessary.

Minor modifications were made to the junction box that housed the electrical components and cooling air controls. Air and oil filters were provided on the rear of the box, and the housing was modified to provide mounting holes for the filters. The outlet connection for the cable to the wrench was modified to provide horizontal rather than vertical orientation of the outlet fitting.

EVALUATION OF SEMIAUTOMATIC ULTRASONIC WRENCH

Performance of the semiautomatic wrench was evaluated by tightening six sizes of 6061-T6 aluminum alloy and CRES 304 stainless steel flared tubing connections of the type previously used.

The evaluation techniques were essentially the same as those used with the manual wrench. Assemblies of each size were tightened to a torque level within the specified range without ultrasonics and with ultrasonic excitation at the preselected power level (within the range of 50 to 400 W) for an interval of 3 or 5 sec, and the additional relative rotation between the nut and the union obtained under ultrasonic influence was noted. Helium leak tests, to a sensitivity of 10^{-8} cm^3 of helium per sec, were carried out at internal pressures of 0, 68×10^5, and 20×10^6 N/m^2 (0, 1000, and 3000 lb/in.^{2}). After leak-testing, each flared tubing connection was disassembled using the wrench without ultrasonic activation, and the required breakaway torque was recorded. Some of the connections were disassembled and reassembled up to 15 times to ascertain the effect of repetitive torquing on leaktightness.

The coupling nuts used in the assemblies were either unlubricated or dry-film lubricated. Each type was used approximately equally with ultrasonic and non-ultrasonic assemblies. There appeared to be no significant difference except that the additional relative rotation with ultrasonic torquing was greater for assemblies using the dry-film lubricated nuts.

The values of additional relative rotation between the nut and the union obtained with ultrasonic activation showed wide scatter, ranging from 0.03 to 0.70 rad. (2 to 40 deg) for the aluminum assemblies and from 0 to 1.35 rad. (0 to 77 deg) for the stainless steel. The values were generally higher the first time a connection was assembled than for subsequent torquings. The average additional rotation for all aluminum assemblies on the first torquing was 0.24 rad. (13.7 deg); on the fifth torquing, 0.25 rad. (14.0 deg); on the tenth and fifteenth, each 0.175 rad. (10.0 deg). For the stainless steel assemblies, the average on the first torquing was 0.20 rad. (11.5 deg); on the fifth, 0.12 rad. (7.1 deg); on the tenth, 0.11 rad. (6.2 deg); and on the fifteenth, 0.10 rad. (6.0 deg). These data are quite similar to those reported earlier and suggest that the initial ultrasonic application somewhat deformed the tube flare to provide a better fit, while in subsequent torquings the major ultrasonic phenomenon was friction reduction.

Aluminum assemblies were mated for leak testing with and without the use of ultrasonics. On the first tightening, 26 non-ultrasonic and 28 ultrasonic assemblies were tested. For the ultrasonically assembled units, 19 of 28 were leaktight without internal pressure and 12 of 28 at both levels of internal pressure. For the non-ultrasonic assemblies, 13 of 26 were leaktight at zero pressure and 7 of 26 were leaktight at both internal test pressures. Any of these assemblies that did not leak at a pressure of 68×10^5 N/m^2 (1000 lb/in.^{2}) likewise did not leak at the higher pressure. These data indicate a significantly increased probability of leaktightness with ultrasonic torquing the first time a connection is assembled and equivalent data for all tightenings indicate still greater improvement with ultrasonic torquing. These data may be conservative since the assemblies selected for repeat torquings were in those sizes for which initial tightening did not yield a large percentage of leaktight connections. Time did not permit repeat torquings for all tube sizes. Nevertheless, the data confirm significant improvement in the leaktightness of the aluminum assemblies with ultrasonic application.

Stainless steel assemblies were prepared for leak testing with and without the use of ultrasonics. For the first-time torqued steel assemblies, 22 were made without and 24 with ultrasonic power application. For the non-ultrasonic assemblies, the numbers that were leaktight were 12 of 22, 11 of 22, and 9 of 22, respectively, for each of the three pressures. The corresponding figures for the ultrasonic assemblies were 15 of 24, 13 of 24, and 10 of 24, respectively. The increase in the numbers of leaktight connections because of ultrasonics was less impressive than with the aluminum assemblies.

Non-ultrasonic breakaway torque was measured on both the aluminum and stainless steel assemblies that had been assembled with and without the use of ultrasonics. In each case, the ratio of breakaway torque to tightening torque was determined, and the overall improvement with ultrasonic application was noted.

For the non-ultrasonic aluminum assemblies, the ratios of breakaway torque to tightening torque ranged from 53 percent for the 0.318-cm (0.125-in.) tubing size to 104 percent for the 0.953-cm (0.375-in.) size, with values for the other sizes falling between these extremes. Similar data for the ultrasonic assemblies ranged from a low of 57 percent to a high of 158 percent. Examination of the initial data indicated that repeat torquing did not alter the breakaway torque range for a given size tubing and thus the breakaway-to-tightening torque ratio was not significantly altered. The overall breakaway-to-tightening torque ratio for the non-ultrasonic assemblies was 82 percent, and that for the ultrasonic assemblies was 95 percent, representing an overall improvement of 16 percent.

The data for the steel assemblies followed a similar trend. The breakaway-to-tightening torque ratio for the non-ultrasonic connections ranged from 43 to 101 percent, with an overall average of 70 percent. Similar data for the ultrasonic assemblies ranged from 61 to 114 percent, with an average of 86 percent, or an overall improvement of 23 percent.

Thus, ultrasonic application during torquing yields an assembly that required a higher breakaway torque than assemblies made without ultrasonic power. Such assemblies should be less likely to vibrate loose, and, initially, leaktight connections should have a higher probability of remaining leaktight over their expected service life.

In the tightening of flared tubing connections, the additional rotation of the nut with respect to the union achieved with ultrasonic excitation provides an increment of additional compressive stress on the tube flare. It has been postulated that this additional rotation also leads to a higher tensile load on the nut. To confirm this measurements were made of the tensile strains in the nut. The magnitude of the additional stress depends to some extent on the thread pitch, which for the flared tubing assemblies ranges from 24 threads/in. for the smallest size tubing to 12 threads/in. for the largest size. Thus, a 10-degree relative rotation between nut and union will yield 0.00295 cm (0.00116 in.) of relative travel for the small tubing and 0.00589 cm (0.00232 in.) for the large size. However, not all of this relative linear translation is manifest as additional compressive stress on the flare. Also involved are radial expansion of the nut and compression of the union. For a 1.0-rad. (60-deg) V-thread approximately 40 percent of the load is transposed radially by the threads. In some instances the radial expansion of the sleeve was sufficient to permanently deform the sleeve and essentially lock it to the nut (Fig. 6). Thus, the increment of compressive load resulting from the additional rotation depends partially on the cross-sectional area of nut and union and partially on radial deformation of the sleeve.

On the basis of available equations, calculations were made of the theoretical sealing stresses on the flare under given tightening torque levels for specific geometries of flared tubing connections. In addition, strain measurements were made on the nut under typical ultrasonic and non-ultrasonic torquing conditions to provide an indication of the additional axial tensile stress resulting from ultrasonic torquing. Precise clamping or locking loads resulting from bolt-nut fastener techniques are difficult to determine because of frictional forces at the threads and the radial expansion of the nut as noted above. The technique generally used for computing such clamping loads is via the nut-bolt torque equation:

$$T = K D P_t$$

where

T = torque, N-m (in.-lb),

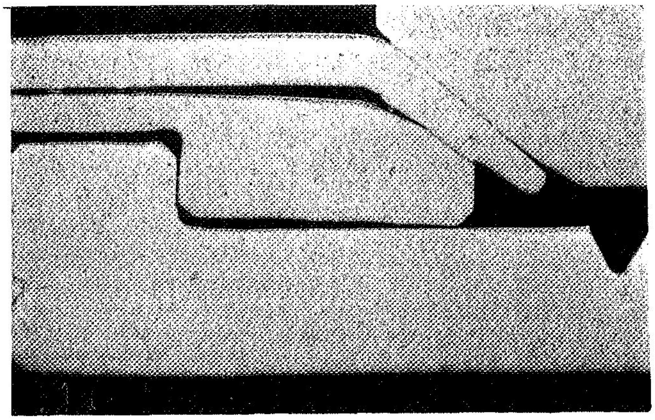
K = torque coefficient, incorporating a constant coefficient for friction, thread pitch, etc.,

D = nominal thread diameter, cm (in.),

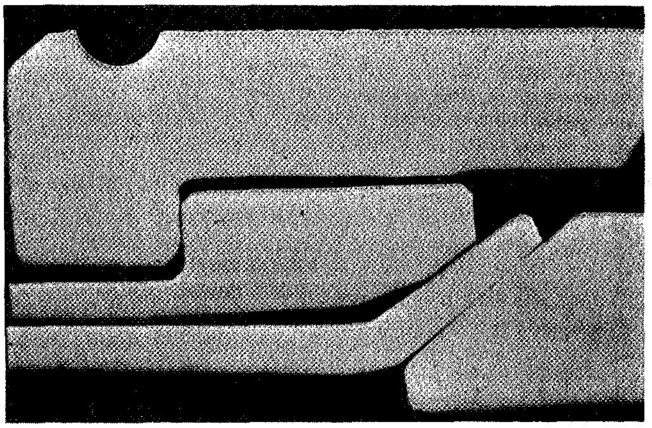
P_t = axial clamping force, N (lb.).

This equation appears to neglect a number of pertinent factors, such as longitudinal or axial "relief" because of hoop strain in the nut, quality of threads, type of lubricant, etc.; however, these factors are accommodated with reasonable accuracy by adjustment of the K factor. Letting $K = 0.25$, the clamping force

$$P_t = \frac{T}{0.25D} = \frac{4T}{D}$$



A. Sleeve Deformed to Contact Nut



B. Minimum Deformation of Sleeve

Figure 6. Deformation of sleeve in vicinity of flare and locking between nut and sleeve shoulder.

Taking into account the direction of the force vectors (angle of flare) shown in Figure 7, the sealing force

$$P_s = 1.836\ P_t = 7.34\ \frac{T}{D}$$

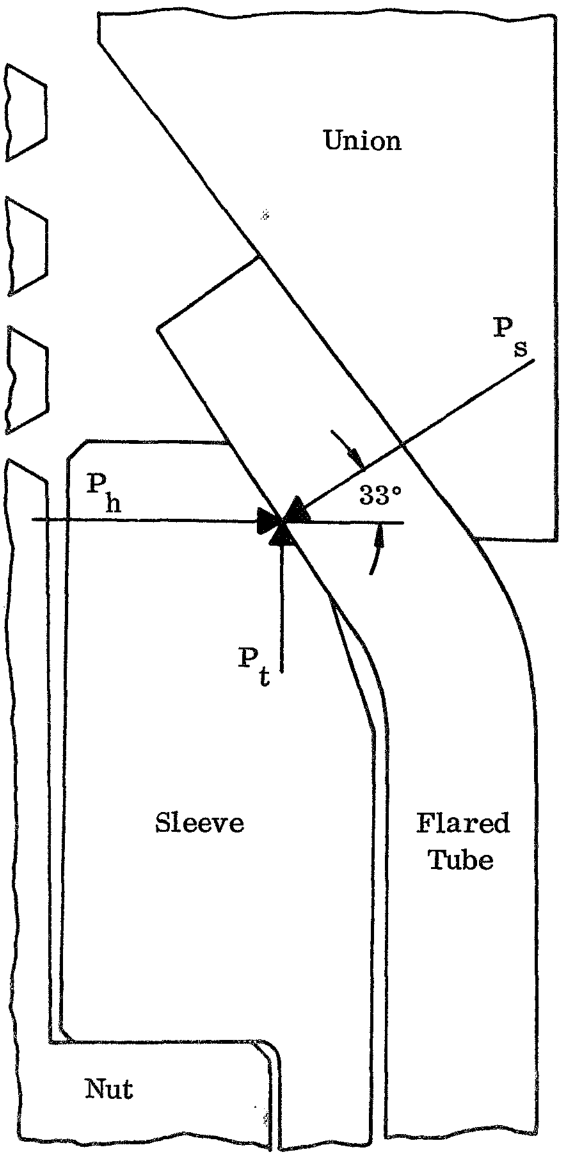


Figure 7. Forces acting at the seat of a flared tubing connection.

Sealing stresses calculated from this equation for aluminum and stainless steel assemblies within the tubing size range from 0.318 cm (0.125 in.) to 2.54 cm (1 in.) tightened to minimum and maximum specification torque levels are provided in Tables 2 and 3. Preliminary calculations of actual sealing stresses from strains measured in the unthreaded portion of the nut showed fair agreement with some of these values. It thus appeared that K = 0.25 was representative for this geometry.

For measurement of the strain in the nut during assembly, miniature type strain gages were attached to one face of the nut as shown in Figure 8. The top gage was located just below the shoulder of the nut, and the center gage was in the nut thread area but above the zone of contact with the threads of the union.

Strain measurements were made on two sizes of aluminum tubing assemblies and four sizes of steel assemblies. With the strain gages in place, the nut was tightened to the specified torque without ultrasonic power applied, providing a strain value identified as peak strain during torquing. When the tightening torque was released, the strain in the nut

TABLE 2. CALCULATED SEALING STRESSES FOR ALUMINUM FLARED TUBING CONNECTIONS

Tube Diameter cm (in.)	Tube Wall Thickness, cm (in.)	Sleeve-to-Flare Contact Area, cm^2 ($in.^2$)	Flare-to Fitting Sealing Area, cm^2 ($in.^2$)	Axial Force: Per N-m Torque, kg	Axial Force: (Per in.-lb Torque, lb)	Minimum Torque, N-m (in.-lb)	Sealing Stress, 10^6 N/m^2 (kpsi)	Maximum Torque, N-m (in.-lb)	Sealing Stress, 10^6 N/m^2 (kpsi)
0. 318 (0. 125)	0. 050 (0. 020)	0. 1832 (0. 0284)	0. 1258 (0. 0195)	5. 13	(12. 8)	3. 39 (30)	249. 56 (36. 2)	4. 52 (40)	332. 29 (48. 2)
	0. 050 (0. 020)		0. 1477 (0. 0229)				212. 33 (30. 8)		282. 65 (41. 0)
0. 478 (0. 188)	0. 050 (0. 020)	0. 2380 (0. 0369)	0. 2167 (0. 0336)	4. 25	(10. 6)	3. 39 (30)	119. 95 (17. 4)	7. 91 (70)	279. 20 (40. 5)
0. 635 (0. 250)	0. 050 (0. 020)	0. 1922 (0. 0298)	0. 1754 (0. 0272)	3. 72	(9. 30)	7. 91 (70)	302. 64 (43. 9)	13. 56 (120)	517. 05 (75. 0)
	0. 071 (0. 028)		0. 1909 (0. 0296)				278. 51 (40. 4)		475. 68 (69. 0)
	0. 088 (0. 035)		0. 1806 (0. 0280)				294. 37 (42. 7)		503. 26 (73. 0)
0. 795 (0. 313)	0. 071 (0. 028)	0. 1929 (0. 0299)	0. 1522 (0. 0236)	3. 20	(8. 00)	7. 91 (70)	248. 18 (36. 0)	13. 56 (120)	424. 67 (61. 6)
	0. 106 (0. 042)		0. 1774 (0. 0275)				257. 14 (37. 3)		441. 21 (64. 0)
0. 953 (0. 375)	0. 071 (0. 028)	0. 2742 (0. 0425)	0. 2225 (0. 0345)	2. 88	(7. 20)	14. 6 (130)	336. 42 (48. 8)	20. 34 (180)	474. 99 (68. 9)
	0. 088 (0. 035)		0. 2193 (0. 0340)				348. 14 (50. 5)		481. 89 (69. 9)
	0. 124 (0. 049)		0. 1780 (0. 0276)				428. 80 (62. 2)		592. 88 (86. 0)
1. 27 (0. 50)	0. 088 (0. 035)	0. 5413 (0. 0839)	0. 4948 (0. 0767)	2. 24	(5. 60)	33. 90 (300)	280. 58 (40. 7)	45. 20 (400)	372. 27 (54. 0)
	0. 124 (0. 049)		0. 4851 (0. 0752)				282. 65 (41. 0)		377. 10 (54. 7)
	0. 165 (0. 065)		0. 3787 (0. 0587)				361. 93 (52. 2)		482. 58 (70. 0)
1. 59 (0. 625)	0. 106 (0. 042)	0. 5335 (0. 0827)	0. 4638 (0. 0719)	1. 84	(4. 60)	48. 59 (430)	348. 14 (50. 5)	62. 15 (550)	445. 35 (64. 6)
	0. 147 (0. 058)		0. 4426 (0. 0686)				364. 69 (52. 9)		466. 72 (67. 7)
	0. 210 (0. 083)		0. 1600 (0. 0248)				1009. 97 (146. 5)		1289. 17 (187. 0)
1. 90 (0. 75)	0. 124 (0. 049)	0. 1780 (0. 0276)	1. 1942 (0. 1851)	1. 50	(3. 75)	73. 45 (650)	166. 83 (24. 2)	90. 40 (800)	204. 75 (29. 7)
	0. 198 (0. 078)		0. 5806 (0. 0900)				342. 63 (49. 7)		420. 53 (61. 0)
	0. 241 (0. 095)		0. 2496 (0. 0387)				796. 94 (115. 6)		978. 94 (142. 0)
2. 54 (1. 0)	0. 147 (0. 058)	1. 6233 (0. 2516)	1. 5226 (0. 2360)	1. 22	(3. 06)	101. 70 (900)	148. 22 (21. 5)	124. 30 (1100)	179. 24 (26. 0)
	0. 165 (0. 065)		1. 5033 (0. 2330)				149. 59 (21. 7)		183. 38 (26. 6)
	0. 241 (0. 095)		0. 7774 (0. 1205)				289. 54 (42. 0)		351. 59 (51. 0)
	0. 317 (0. 125)		0. 0567 (0. 0088)				3962. 67 (574. 8)		4843. 03 (702. 5)

TABLE 3. CALCULATED SEALING STRESSES FOR STAINLESS STEEL FLARED TUBING CONNECTIONS

Tube Diameter, cm (in.)	Tube Wall Thickness, cm (in.)	Sleeve-to-Flare Contact Area, cm^2 ($in.^2$)	Flare-to-Fitting Sealing Area, cm^2 ($in.^2$)	Axial Force		Minimum Torque, N-m (in.-lb)	Sealing Stress, 10^6 N/m^2 (kpsi)	Maximum Torque, N-m (in.-lb)	Sealing Stress, 10^6 N/m^2 (kpsi)
				Per N-m Torque, kg	(Per in.-lb Torque, lb)				
0.318 (0.125)	0.050 (0.020)	0.1832 (0.0284)	0.1258 (0.0195)	5.13	(12.8)	6.78 (60)	496.36 (72.0)	9.04 (80)	664.58 (96.4)
	0.050 (0.020)		0.1477 (0.0229)				423.98 (61.5)		565.99 (82.1)
0.478 (0.188)	0.050 (0.020)	0.2380 (0.0369)	0.2167 (0.0336)	4.25	(10.6)	10.17 (90)	358.48 (52.0)	11.30 (100)	399.16 (57.9)
0.635 (0.250)	0.050 (0.020)	0.1922 (0.0298)	0.1754 (0.0272)	3.72	(9.30)	15.25 (135)	583.92 (84.7)	16.95 (150)	648.72 (94.1)
	0.071 (0.028)		0.1909 (0.0296)				536.35 (77.8)		596.33 (86.5)
	0.088 (0.035)		0.1806 (0.0280)				565.30 (82.0)		630.11 (91.4)
0.795 (0.313)	0.071 (0.028)	0.1929 (0.0299)	0.1522 (0.0236)	3.20	(8.00)	20.34 (180)	634.24 (92.0)	22.60 (200)	708.01 (102.7)
	0.106 (0.042)		0.1774 (0.0275)				661.82 (96.0)		736.27 (106.8)
0.953 (0.375)	0.071 (0.028)	0.2742 (0.0425)	0.2225 (0.0345)	2.88	(7.20)	30.51 (270)	710.08 (103.0)	33.90 (300)	791.43 (114.8)
	0.088 (0.035)		0.2193 (0.0340)				723.18 (104.9)		803.83 (116.6)
	0.124 (0.049)		0.1780 (0.0276)				889.32 (129.0)		989.97 (143.6)
1.27 (0.50)	0.088 (0.035)	0.5413 (0.0839)	0.4948 (0.0767)	2.24	(5.60)	50.85 (450)	420.53 (61.0)	56.50 (500)	468.10 (67.9)
	0.124 (0.049)		0.4851 (0.0752)				423.98 (61.50)		471.54 (68.4)
	0.165 (0.065)		0.3787 (0.0587)				543.24 (78.80)		603.91 (87.6)
1.59 (0.625)	0.106 (0.042)	0.5335 (0.0827)	0.4638 (0.0719)	1.84	(4.60)	73.45 (650)	523.94 (76.0)	79.10 (700)	567.37 (82.3)
	0.147 (0.058)		0.4426 (0.0686)				551.52 (80.0)		594.26 (86.2)
	0.210 (0.083)		0.1600 (0.0248)				1523.57 (221.0)		1644.21 (238.5)
1.90 (0.75)	0.124 (0.049)	0.1780 (0.0276)	1.1942 (0.1851)	1.50	(3.75)	101.70 (900)	277.50 (33.0)	113.0 (1000)	256.45 (37.2)
	0.198 (0.078)		0.5806 (0.0900)				473.61 (68.7)		525.70 (76.4)
	0.241 (0.095)		0.2496 (0.0387)				1102.35 (159.9)		1225.61 (177.8)
2.54 (1.0)	0.147 (0.058)	1.6233 (0.2516)	1.5226 (0.2360)	1.22	(3.06)	135.60 (1200)	197.16 (28.6)	158.20 (1400)	230.25 (33.4)
	0.165 (0.065)		1.5033 (0.2330)				199.23 (28.9)		233.01 (33.8)
	0.241 (0.095)		0.7774 (0.1205)				386.06 (56.0)		450.86 (65.4)
	0.317 (0.125)		0.0567 (0.0088)				5280.80 (766.0)		6163.92 (894.1)

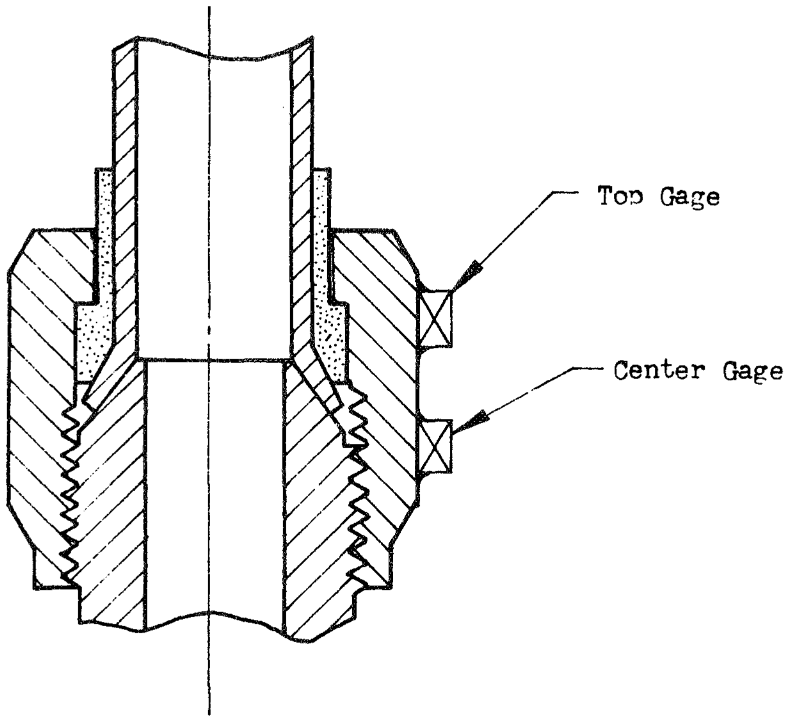


Figure 8. Strain gages mounted on flared tubing assemblies.

dropped slightly, yielding a residual strain value. Subsequent loosening of the nut effected a slight increase in strain, at breakaway torque, followed by a drop to zero. The nut was again tightened to the specified torque level, and ultrasonic power was applied for 5 sec. The additional rotation of the nut relative to the union effected with ultrasonic activation yielded higher strain values.

Table 4 presents the average peak and residual strains both with and without ultrasonic activation obtained during torquing of each size fastener. Also shown are the percentages of increase in strain for the ultrasonic assemblies, as well as the ratio of residual strain to peak strain.

The data show a substantial increase in both peak and residual strain for the ultrasonically torqued aluminum assemblies and smaller increases for the stainless steel assemblies. However, the ratios of residual to peak strain are approximately the same for the ultrasonic and non-ultrasonic assemblies of a given size. It is interesting that in the case of the stainless steel, this ratio decreases with increasing fastener size.

TABLE 4. SUMMARY OF AVERAGE STRAIN DATA

Material	Tube Diameter cm (in.)	Average Strain Without Ultrasonics, 10^{-6} m/m		Average Strain With Ultrasonics, 10^{-6} m/m		Increase Because of Ultrasonics, percent		Ratio of Residual to Peak Strain	
		Peak	Residual	Peak	Residual	Peak	Residual	Non-Ultrasonic	Ultrasonic
Aluminum	0.635 (0.250)	400	310	544	419	36	35	0.775	0.768
	1.27 (0.50)	196	154	378	273	93	77	0.786	0.722
Steel	0.635 (0.250)	988	898	1031	963	4	7	0.909	0.934
	1.27 (0.50)	406	—	425	—	5	—	0.84[a]	0.84[a]
	1.90 (0.75)	149	106	153	114	3	8	0.711	0.745
	2.54 (1.0)	150	86	176	99	17	15	0.573	0.563
	Overall Average							0.751	0.746

a. Interpolated; not included in average.

It is apparent that if the flared tubing assemblies are to be helium leaktight with high internal pressures, the sealing stress during torquing must be sufficient to plastically deform the flare in the flare seal area in order to effect near perfect matching of the tube flare to the union bevel surface. The yield strength of the 6061-T6 aluminum alloy is approximately 241.3×10^6 N/m^2 (35 000 lb/in.^{2}) and that of 304-1/8 hard stainless steel is approximately 517×10^6 N/m^2 (75 000 lb/in.^{2}). It would be expected that the peak stress acting on the flare during torquing would exceed the yield strength and that the residual sealing stress after release of torque would remain near the yield value.

As noted in Table 5, the theoretically derived sealing stresses fall substantially above and below the above yield values. If the mating surfaces were of high quality in finish, contour, flare angle, etc., differential elastic deformations would probably yield leaktight integrity at the lower indicated sealing stress values. In the cases of stresses substantially higher than yield, either there are controlling factors (such as radial deformation of the sleeve) that are not included in the equation, or a very high degree of flare deformation occurs.

The sealing stress values calculated from peak strain measurements on the nut were generally near or above the yield values, and those calculated from the residual strain measurements were generally below yield values. Measured values for the 0.635-cm (0.250-in.) steel assemblies were substantially above yield and also higher than the theoretically computed stress. Values for the 2.54-cm (1-in.) assemblies were below the yield value but were close to the theoretical stress.

Table 5 summarizes the ratios of measured stresses to theoretical sealing stresses and shows the percent increase in these values because of ultrasonics wrenching. For the aluminum assemblies, ultrasonic activation effected increases of 24 to 76 percent in peak stress, and 17 and 77 percent in residual stress. With the stainless steel assemblies, the improvements ranged from 3 to 20 percent. These data further confirm the increased likelihood of obtaining leaktight assemblies with the ultrasonic wrench.

The results of strain measurements were also verified by conducting torque tension tests. Simulated tube specimens using machined steel bars to simulate flared tubes were used. The simulated specimens were mounted in a tension test machine, and the specimens were torqued to varying percentages of maximum specified torque with and without the application of ultrasonic energy. The use of ultrasonics produced an average increase of 8.8 percent

TABLE 5. SUMMARY OF SEALING STRESSES AT FLARE/UNION INTERFACE COMPUTED FROM STRAIN DATA

Material	Tube Diameter, cm (in.)	Tube Wall Thickness, cm (in.)	Ultra-sonic Power, watts	Computed Sealing Stress: Peak During Torquing, 10^6 N/m^2 (kpsi)	Computed Sealing Stress: Residual, 10^6 N/m^2 (kpsi)	Theoretical Sealing Stress, 10^6 N/m^2 (kpsi)	Residual Stress as Percent of Theoretical	Increase Because of Ultrasonics, percent: Peak	Increase Because of Ultrasonics, percent: Residual
Aluminum	0.635 (0.250)	0.0889 (0.035)	0	275.7 (39.99)	227.4 (32.98)	504.0 (73.16)	45		
			50	343.0 (49.75)	266.8 (38.70)	504.0 (73.16)	53	24	17
	1.90 (0.75)	0.2413 (0.095)	0	388.9 (56.41)	269.7 (39.12)	861.8 (125)	31		
			300	683.2 (99.10)	476.5 (69.12)	861.8 (125)	55	76	77
Steel	0.635 (0.250)	0.0889 (0.035)	0	993.2 (144.07)	928.0 (134.60)	586.0 (85.0)	158		
			50	1043.3 (151.30)	993.2 (144.07)	586.0 (85.0)	169	5	7
	1.27 (0.50)	0.0889 (0.035)	0[a]	500.1 (72.55)	420.1 (60.94)[c]	468.1 (67.9)	90		
			150[a]	523.1 (75.88)	439.4 (63.74)[c]	468.1 (67.9)	94	5	5
			0[b]	512.9 (74.40)	430.9 (62.50)[c]	468.1 (67.9)	92		
			150[b]	615.5 (89.28)	517.1 (75.00)[c]	468.1 (67.9)	110	20	20
	1.90 (0.75)	0.125 (0.049)	0	544.0 (78.90)	392.3 (56.90)	230.3 (33.4)	170		
			300	562.8 (81.64)	425.8 (61.80)	230.3 (33.4)	185	3	9
	2.54 (1.0)	0.165 (0.065)	0	347.0 (50.30)	201.3 (29.20)	233.0 (33.8)	86		
			400	404.7 (58.7)	231.6 (33.60)	233.0 (33.8)	99	17	14

a. Stress in nut in area of shoulder on first torquing.
b. Stress in threaded portion of nut on fifteenth torquing.
c. Interpolated values.

in tension during torquing. The average increase was 10.6 percent for aluminum and 6.16 percent for stainless steel. This increase in tension was converted to increase in sealing stress and showed an average increase of 14.9×10^6 N/m^2 (2185 lb/in.^{2}) in sealing pressure.

ENGINEERING CONSIDERATIONS

The ultrasonic wrenches that were designed, fabricated, and evaluated represent first-generation tools for a technology that is new. The efficacy of these first units in achieving a higher percentage of leaktight assemblies than conventional wrenches has been demonstrated.

Although their use to date has been limited to laboratory use only, personnel using the wrenches have provided suggestions that indicate directions for further evolution. Operators have commented that the ultrasonic wrench looks big and feels heavy in comparison with standard torque wrenches, that the wrench heads are larger, but that "it works." The wrench can now be modified from the "human engineering" standpoint, as well as for manufacturing and cost reduction, without compromising performance.

One proposed alteration involves a change in the materials of construction. In these first units the wrench body was made of steel and the transducer body of beryllium-copper; by fabricating both of these components of 6Al-4V titanium alloy, the overall weight of the assembly can be reduced from about 5 kg (11 lb) to about 2.7 kg (6 lb). The wrench heads can be made of high strength tool steel, which will reduce the size of the heads and eliminate the difficult machining associated with the titanium alloy.

Another proposed change involves elimination of the integral handle/side-beam design of the wrench body for a tubular geometry. The deflection beams could be formed either by machining out the central portion of the tube body or by welding deflection plates onto the two tubular end sections. The torque readout device can be either a dial indicator for high precision or a pointer extension over a calibrated arc.

Experience has further indicated that an ultrasonic wrench of reduced size is desirable for tightening connections for flared tubing of 0.954 cm (0.375 in.) and smaller. Such size reduction can be obtained with a wrench designed to operate at a higher frequency of, for example, 40 kHz (40 kc/sec). The size of the transducer is decreased as the frequency is increased, and the wrench body can be made correspondingly smaller.

The frequency converter supplied with the semiautomatic wrench is of modular construction. The plug-in modules for oscillator control and for power output can be replaced with modules for the higher frequency, and the two size wrenches can thus operate from the same basic frequency converter.

CONCLUSIONS

Based on the tests conducted and use of the wrench, the following advantages were concluded.

1. Evaluation of manual and semiautomatic ultrasonic wrenches in the tightening of flared tubing connections confirmed the validity of previously established wrench design specifications.

2. The ultrasonic wrenches facilitated tightening of connections of 6061-T6 aluminum alloy and of CRES 304 stainless steel flared tubing within the size range of 0.318-cm (0.125-in.) to 2.54-cm (1-in.) diameter.

3. Use of the ultrasonic wrench increases the probability of obtaining leaktight assemblies of flared tubing connections. The incidence of leaktight aluminum assemblies (Table 6) at zero internal pressure increased from 13 of 32 to 27 of 37, and those at an internal pressure of 20×10^6 N/m^2 (300 lb/m^2) from 7 of 32 to 14 of 37. For the stainless steel assemblies (Table 7), leaktightness at zero internal pressure improved from 41 of 59 to 54 of 64 and at the elevated pressure from 28 of 59 to 43 of 64.

4. The non-ultrasonic breakaway torque required to loosen the ultrasonically tightened assemblies increased by an average of 16 percent for the aluminum connections and by an average of 23 percent for the stainless steel connections.

5. For leaktight assemblies, the average additional rotation of the nut with respect to the union was 0.32 rad. (18.6 deg) for the aluminum and 0.20 rad. (11.3 deg) for the stainless steel connections. This additional rotation increased the tensile strain in the nut, as measured by strain gages, by 36 to 93 percent for the aluminum and by 3 to 17 percent for the stainless steel.

6. Sealing stresses on the assemblies, as calculated from the measured strains, were consistently higher with ultrasonic tightening.

TABLE 6. SUMMARY OF LEAKTIGHTNESS DATA ON ALL WRENCHED ALUMINUM FLARED TUBING ASSEMBLIES

Tube Diameter, cm (in.)	Total No. of Assem-blies Tested	No. of Leaktight Assemblies at Test Pressure of		
		$0\ N/m^2$ (0 psi)	$68 \times 10^5\ N/m^2$ (1000 psi)	$20 \times 10^6\ N/m^2$ (3000 psi)
Non-Ultrasonic				
0.318 (0.125)	4	4	3	3
0.635 (0.250)	12	4	2	2
0.953 (0.375)	5	0	0	0
1.27 (0.50)	2	2	1	1
1.90 (0.75)	4	2	1	1
2.54 (1.0)	5	1	0	0
Totals	32	13	7	7
Ultrasonic				
0.318 (0.125)	8	8	7	7
0.635 (0.250)	14	4	3	3
0.953 (0.375)	6	6	0	0
1.27 (0.50)	3	3	2	2
1.90 (0.75)	2	2	1	1
2.54 (1.0)	4	4	1	1
Totals	37	27	14	14

TABLE 7. SUMMARY OF LEAKTIGHTNESS DATA ON ALL WRENCHED STAINLESS STEEL FLARED TUBING ASSEMBLIES

Tube Diameter, cm (in.)	Total No. of Assemblies Tested	No. of Leaktight Assemblies at Test Pressure of		
		0 N/m^2 (0 psi)	68×10^5 N/m^2 (1000 psi)	20×10^6 N/m^2 (3000 psi)
Non-Ultrasonic				
0.635 (0.250)	16	9	7	6
0.953 (0.375)	4	4	2	2
1.25 (0.50)	28	23	20	19
1.90 (0.75)	6	0	0	0
2.54 (1.0)	5	4	1	1
Totals	59	41	30	28
Ultrasonic				
0.635 (0.250)	30	24	19	14
0.953 (0.375)	8	7	7	7
1.25 (0.50)	19	19	19	19
1.90 (0.75)	-[a]	-	-	-
2.54 (1.0)	7	4	3	3
Totals	64	54	48	43

a. The 1.90-centimeter (0.75-inch) assemblies could not be sealed and were excluded from averages.

7. Torque tension tests showed that application of ultrasonics results in significant increase in tension load at the same torque level. The amount of increase, however, decreases with each successive retightening of the nut. An overall average of 8.8 percent increase in tension load was effected by application of ultrasonics. This increase is equivalent to an increase of 14.9×10^6 N/m^2 (2185 $lb/in.^2$) in sealing pressure.

The above advantages of the use of the ultrasonic torque wrench are offset by the following disadvantages:

1. The wrench requires excessive clearance for access to the coupling nut and rotation of the handle.

2. The wrench system requires that the frequency generator, junction box, connecting cables, and the wrench must be transported to the work site. The wrench also requires electrical power and air for operation.

3. The wrench is large, bulky, and unwieldy. None of the operators who used the wrench could get a "feel" for the wrench.

From the above advantages and disadvantages, it is concluded that the wrench operates and produces the results intended when used under laboratory conditions. Because of the disadvantages, it is evident that additional design and development are required before a useable wrench will be available.

APPROVAL

TM X-53995

ULTRASONIC TORQUE WRENCH

By E. J. Minter

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This document has also been reviewed and approved for technical accuracy.

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