THE DESIGN OF A HYDRO-PNEUMATIC UNIVERSAL SPAR TESTING MACHINE

A THESIS

SUBMITTED FOR THE DEGREE

OF

MASTER OF SCIENCE IN AERONAUTICAL ENGINEERING

BY

BARCLAY M. LEWIS

Atlanta, Georgia
Daniel Guggenheim School of Aeronautics
Georgia School of Technology
May 24, 1935.

381

Approved:

INDEX

THE DESIGN OF A HYDRO-PNEUMATIC UNIVERSAL SPAR TESTING MACHINE

SUMMARY

The design described herein is that of a Universal Spar Testing Machine developed at the Daniel Guggenheim School of Aeronautics of the Georgia School of Technology, Atlanta, Georgia, by the author as a thesis for the degree of Master of Science in Aeronautical Engineering. The purpose of this development was to provide a means for the further research and investigation of the strength of wing spars.

The machine is composed essentially of five main parts: namely, a hydro-pneumatic loading system, a loading control system, an adjustable or movable support for the loading system, a stationary base or frame for supporting the spar, and a deflectometer for measuring spar deflections.

The machine was designed to accommodate spars up to 20' in length, 12" in depth, and 3" in thickness, and it is capable of producing loads up to 20,000 pounds. In case it is desired to test spars larger than the aforementioned, a scale model of the spar would have to be used.

The loading arrangement consists of a number of hydro-pneumatic units which are to be placed at the rib point supports of a given spar. A medium grade oil is used as a seal in the cells and a maximum of 500 pounds air pressure is

supplied by a two stage compressor.

INTRODUCTION

An important phase of aeronautical research is the experimental determination of the strength of wing spars. In connection with this form of research, the means for securing criteria for spar design have been more or less limited. The ideal solution of this problem would be the development of a machine that would reproduce the conditions experienced by a wing spar in actual flight. Several attempts to solve this problem have been made but up to the present the author has not been able to find published information on a machine capable of furnishing accurate data for any desired loading condition on spars of various sizes, and capable of applying the loads simultaneously.

Certain machines have been built that are capable of testing spars of practically any size but the loading conditions were only approximated by the hanging of weights along the spar (Ref.1,2,3). Other machines have been devised for special loading conditions, such as axial, uniform transverse, and varying transverse, or any combination, but these machines have been limited to a small range in the size of spars to be tested (Ref.4,5). Machines have also been constructed that were adaptable with regard both to the distribution of loads and the method of load application to the spar (Ref. 6, 10). Machines of this type have an adjustable outboard spar support and a loading device composed of a system of levers on which weights are hung. However, the disadvantage of this type is its inability to apply all of the loads to the spar simultaneously. From each of the above cases it is quite evident that the major problem in the design of a spar testing machine is that of the loading system on which the machine depends for its universality.

The investigation described herein was made by the author at the Daniel Guggenheim School of Aeronautics of the Georgia School of Technology, Atlanta, Georgia. The purpose of this investigation was to develop the design of a spar testing machine to be installed and used in the aircraft structural testing laboratory. This investigation was confined to two phases in the development of this machine as follows:

1. The design of a flexible loading unit, several of which could be used for applying loads simultaneously to the spar and for effecting any desired load distribution.

2. The incorporation of these units together with suitable control apparatus in the design of a rigid structure for supporting the test spar and the loading units in any desired manner.

DESCRIPTION OF MACHINE

In the design of this spar testing machine the following requirements were set up as a guide:

1. Flexibility of loading system both as to magnitude and distribution of loads.
2. Simultaneous application of loads.
3. Universal mounting of test spar.
4. Simplicity, compactness, and ease of operation.

After a careful consideration it was decided that a maximum full scale spar of 30'length was the largest for which such a machine would be economically feasible. However, larger spars could be tested in sections or by means of scale models. Assuming a minimum rib spacing of 1' and a maximum total load of 20,000 pounds, it was necessary for each unit to be capable of producing a force of 1000 lbs.

Loading Unit

Several possible methods of load application were considered and it was finally decided that the loading requirements could best be met by utilizing a hydro-pneumatic principle. In the application of this principle the load is applied by a close fitting piston in a cylinder filled with oil into which air pressure is introduced from an external source, a short strut and a shoe being used to transmit the load to the spar. This cylinder is connected to a large air chamber which acts as a pressure reserve and enables the cell to produce practically a constant load, varying only one percent for the full stroke of the piston.

In the development and design of this cell it was necessary to determine the manner in which binding and friction would effect its operation and to ascertain the way in which these effects could be kept within a one percent limit. A testing device (Fig. 1) was constructed which consisted of a structural steel frame supporting a load beam mounted on ball bearings. A movable table was used for holding the test cell and a screw jack was employed for raising and lowering this table.

The first cell (Fig. 2) was designed for a working air pressure of 60 pounds which was available in the laboratory. It consisted of a steel cylinder of a 5" inside diameter, a cast iron piston having a length-diameter ratio of 1, a rod, acting as a strut, screwed into the piston, and a shoe resting on a 1/2" hardened ball in a spherical seat at the top of the rod.

This cell was tested at an air pressure of 60 pounds. The results showed that the friction of the oil in the small 1/8" copper tube, which was used as a pressure line, was sufficient to cause a sluggish action of the cell. It was possible, of course, to increase the size of this line but this would have limited the flexibility of the system and would not have entirely eliminated the friction of the oil in the tube. Obviously it was necessary for the pressure line to be free of oil. Consequently, an oil reservoir, large enough to hold a volume of oil somewhat greater than the displacement of the piston, was attached to the cylinder by a short 1/4" nipple. The pressure line, now free of oil, was attached to the top of the reservoir. This addition made it possible to use a smaller pressure line, resulting in a more flexible system.

Before the investigation of the low pressure system was completed, a 500 pound pressure supply was made available in the test laboratory and it was to great advantage both economically and structurally to adopt a small loading unit and use higher pressures.

A smaller cell of similar type was then designed (Fig. 3). In this cell the cylinder and oil reservoir were made in one cast iron block, the cylinder having a diameter of 2" and the reservoir having a diameter of 1-1/2" with a 3/8" hole between the two for the oil passage. The piston was made of steel and was ground with a clearance of .0003". The cylinder was 7-3/16" tall giving the piston a 3" stroke. The arrangement of the strut and shoe was the same as that used in the low pressure cell. A test cell of these dimensions was constructed having a steel cylinder and cast iron piston.

When 200 pounds pressure was put on this cell, it was found that the friction was much too high. Part of the friction evidently was caused by the ball and socket joint in the shoe and the remainder was due to the binding effect resulting from the moment produced on the piston by the side component of the load. To reduce this friction a second ball and socket joint was introduced at the lower end of the strut and the strut was case hardened. This arrangement reduced the binding effect somewhat, but there was still a tendency for the piston to bind due to the side load applied by the strut at the top of the piston. Also, when the piston was at the top of its stroke the strut became unstable and fell over. A small triangular wire spring was fixed in an undercut circular groove in the cap of the cylinder to hold the strut in a central position throughout the travel of the piston. To relieve still further the moment on the piston the ball and socket joint was lowered to the center. This change brought the friction within a two percent limit of the load and eliminated the instability of the strut. It was found that by removing the shoe and allowing the upper ball to bear on a flat steel surface that the friction was reduced to as low as 1/2 of one percent. Such a result would be expected since a ball and socket is subjected to sliding friction while a ball in contact with a flat plate experiences only rolling friction. As a result of this conclusion the hemispherical sockets at each end of the strut were cut with a 1/16" larger radius than that of the ball in order to give a rolling action at the joint. This modification brought the friction within a one percent limit.

Tests on the cell with all the above modifications showed excessive air leakage around the top of the piston at pressures of 100 and 200 pounds. On investigation, it was found that air was coming from the reservoir into the cylinder through the wall of the casting about halfway up. Later it was found that oil leaked through the cast iron piston of the test cell when the ball joint was lowered within an inch of the bottom. These tests revealed the fact that it was very difficult to use cast iron for the cylinder or piston at these pressures. On the basis of this conclusion a new piston was made for the test cell. This piston was made of steel case hardened, ground, and lapped into the cylinder. Tests on this cell at both 200 and 300 pounds pressure proved satisfactory, the friction of the cell being within the one percent limit.

The cylinder and piston of the final cell (Fig.4 and 5) were made of hard manganese steel, the other features being essentially the same. The cylinder was made from solid stock and had an inside diameter of 2.523" giving it a cross section area of 5 sq.in. The piston was case hardened and ground. The reservoir was made of 3-1/2" seamless steel tubing with a 1/2" wall thickness, having 1/2" plugs seated in each end and a 1/4" nipple connection to the cylinder. Two small steel ribs were soldered between the cylinder and reservoir to add to the rigidity of the unit. (Fig.6 shows the sequence in the development of the loading unit.)

Load Table

The load table (Fig.8) consisting of an 8" H beam, 30' long acts as a support for the loading units. This beam was designed for a maximum deflection of approximately one inch for the worst loading condition. (Appendix A for details) A guide made of two 5" Zs and slidable on the vertical Z assembly is secured to the end of this H beam. The lateral spar supports made of 2" steel straps are also secured to this beam. This entire assembly is supported by two 7 ton Black Hawk hydraulic jacks (Appendix A for position of jacks for minimum deflection of beam).

Frame

The frame (Fig.8) consists of a horizontal assembly made of two 4" Z beams which are braced by a light 7" channel and secured to the concrete floor, and a vertical assembly of similar cross section secured to the brick wall by two 5" Zs. The purpose of the horizontal assembly is to provide a support for the two hydraulic jacks and to act as a base for the outboard spar support. The vertical assembly furnishes a support for the inboard spar fitting and acts as a guide for the adjustable load table. The inboard spar fitting consists of two 5"x 3" angles secured to the vertical assembly. This fitting may be adjusted for both thickness and depth of test spar.

The outboard spar support (Fig.8) consists of a lower slidable member, a pair of tie rods attached to this member, a singletree attached to the top of the tie rods, and straps running from the singletree to the strut point of the spar. The lower member may be clamped by means of two large bolts at any point along the horizontal assembly thus giving the linkage any desired slope. The lower member is fitted at each end with eyes which are pinned to small clevises into which the tie rods are secured. This arrangement allows motion in two planes thus equalizing the tension in the tie rods. The upper end of each tie rod is secured to the singletree by a ball and socket adjustment. There is a clevis, to which the straps are connected, pinned to the center of the singletree. A series of holes are drilled in the straps to allow for complete adjustment of the linkage. It would have been desirable to have designed this linkage as a single wire system but due to the presence of the load table under the spar it was necessary to adopt a two wire system as noted.

This linkage is to be made of Chrome-Molybdenum steel (S.A.E.No.4130) having an Elastic Limit of 88,600 pounds per sq.in. in the annealed state.

Deflectometer

The device for measuring spar deflections consists of a reinforced wooden beam and a set of height gauges. This beam is secured to the horizontal assembly by four 3/8" x 1-1/2" steel straps. The top is finished to provide a smooth surface

The manifold, 2" in diameter and 36" long, is made of a seamless steel tube plugged at the ends. It has a row of holes on one side for the cell and air chamber valves and holes on the other for the loading, release, and gauge valves, all of which are fastened under the control table.

A table of suitable dimensions is used as a mounting for the gauges, compressor switch, and pressure valves of the loading system which are arranged in convenient order(Fig.7). High pressure needle valves are used and their stems extent through the top of the table and are bushed in a copper plate secured to the top of the table.

METHOD OF OPERATION

The spar to be tested is secured to the inboard spar fitting and the outboard support is attached and adjusted according to specifications. The load table is then raised until the top of the cells are about four inches from the spar. Wooden buffer blocks of suitable thickness are screwed to the lateral spar supports and these supports are adjusted against the sides of the spar and clamped in place. The loading cells are placed at the rib points of the spar and each cell is pre-loaded as determined by a specified load distribution curve. The height gauges are placed on the wooden beam and zero readings are taken at intervals along the spar. The load table is now raised by means of the jacks and the load is transferred to the spar. The deflections at the reference points are now obtained. The table is lowered, more pressure is applied to the cells, and the increased load is applied to the spar by raising the load table as before. Deflections are taken for each increase in load. Loads are increased until the spar fails or until a specified maximum load is reached.

CONCLUSION

It is not to be inferred by the foregoing discussion that this machine is a final solution of the problem but it is the nearest approach as compared with other similar devices known to the author. Recommendations for the further development of this type of testing machine are as follows:

1. The addition of special devices for involving a time factor in the application of loads; i.e., sudden removal and application of loads.
2. The use of a hydraulic device for measuring spar deflections and reproducing the curvature of the spar graphically.
3. The application of this principle to rib tests and tests of wings.

ACKNOWLEDGMENTS

It is with pleasure that I acknowledge the assistance of Professor Montgomery Knight who made helpful comments and criticisms, Professor T. Edward Moodie who also aided by his suggestions, and Mr. W. C. Slocum who constructed the calibration jig and the loading units of the spar testing machine.

REFERENCES AND BIBLIOGRAPHY

1. Weaver, E.R.: Static Test of the Engineering Division Experimental Steel Spar. Air Service Information Circulars, Vol.VI, June 15, 1925, No.519.

2. Weaver, E.R.: Report of Static Test of Aeromarine Experimental Duralumin Spar. Air Service Information Circulars, Vol.VI, July 1, 1925, No.524.

3. Newell, J.S.: Comparison of Tests on Experimental 15-inch Metal Spars and 11-ft. Chord Metal Wing Ribs. Pts.1 and 2, Air Service Information Circulars, Vol. VI, March 1, 1926, No.556.

4. Niles, A.S.,Jr.: Tests on 6-1/4 Inch Metal Spars. Air Service Information Circulars, Vol.VII, August 15, 1928, No.622.

5. Brown, C.G.: Test of Armstrong-Whitworth Steel Spars Under Combined Axial and Transverse Loading. Air Service Information Circulars, Vol.VII, December 1, 1930, No.656.

6. Gerard, I.J.: Primary Importance of Mechanical Testing in Aircraft Construction. (in Journal of the Royal Aeronautical Society, Vol.35, July, 1931, Page 579).

7. Marks, Lionel S.: Mechanical Engineers' Handbook. Third Edition. Page 235. Pub.by McGraw-Hill Book Company. New York. 1930.

8. Niles, Alfred S. and Newell, Joseph S.: Airplane Structures. Page 380. Pub.by John Wiley and Sons, Inc.New York, 1929.

9. Seely, Fred B.: Advanced Mechanics of Materials. Page 220. Pub. by John Wiley and Sons, Inc. New York, 1932.

10. Langley, M.: Metal Aircraft Construction. Pages 40-46. Pub. by Sir Isaac Pitman and Sons, Ltd. London, 1934

APPENDIX A

Location of Supports for Minimum Deflection

It is assumed that for a condition of minimum deflection that the deflection at the ends of the beam is the same as that at the middle of the beam. Since both points have the same deflection, the sum of the moments of the areas of the moment diagram for half the beam taken about the end point must be zero.

See Fig. 9-A for Diagram

$$A_1 = \frac{2}{3} \times \frac{wl^2}{8} \times \frac{l}{2} = \frac{wl^3}{24}$$

$$A_2 = \frac{wx^2}{2} \times \frac{l}{2} = \frac{wx^2 l}{4}$$

$$A_3 = \frac{1}{3} \times \frac{wx^2}{2} \times x = \frac{wx^3}{6}$$

$$\frac{wl^3}{24}\left(\frac{5}{16}l + x\right) - \frac{wx^2 l}{4}\left(\frac{l}{4} + x\right) - \frac{wx^3}{6}\left(\frac{3}{4}x\right) = 0$$

$$\frac{5}{384}wl^4 + \frac{wl^3 x}{24} - \frac{wx^2 l^2}{16} - \frac{wx^3 l}{4} - \frac{wx^4}{8} = 0$$

$$\frac{5}{48}l^4 + \frac{l^3 x}{3} - \frac{x^2 l^2}{2} - 2x^3 l - x^4 = 0$$

On solution of this equation we get:

$$x = \frac{4}{10}l$$

$$\partial = \frac{5}{384} \times \frac{wl^4}{EI} - \frac{wx^2 l^2}{16EI} \qquad (a)$$

Design of H-Beam (Load Table)

See Fig. 9-B for Diagram

For supports at ends:

$$\partial = \frac{5}{384} \times \frac{wl^4}{EI}$$

$$\partial = \frac{5 \times 1 \times (20 \times 12)^4}{384 \times 29 \times 10^6 \times 112.8}$$

$$\partial = \frac{5 \times 1 \times 3,317,760,000}{384 \times 29 \times 10^6 \times 112.8} = .0132'' \ (w = 1 \text{ lb/in})$$

Assuming a maximum loading condition of 1000 lbs./ft.

Weight of beam = 32.6 lbs./ft.

Weight of each cell = (approx.) 30 lbs.

Total load = 1,063 lbs./ft. = 88.8 lbs./in.

Then:

$\partial = .0132 \times 88.8 = 1.18''$ (Maximum deflection)

Position of jacks for minimum deflection:

$$1 + 2x = 20$$

$$x = \frac{4}{10} 1$$

$$1 + \frac{4}{5} 1 = 20$$

$$1 = 11.1'$$

$$x = 4.45'$$

On substituting these values in Equation (a):

$\partial = .00038''$(for a w of 1 lb./in.)

Maximum loading condition = 88.8 lbs./in.

Hence: $\partial = 88.8 \times .00038 = .0337''$ (Minimum deflection)

APPENDIX B

Design of Outboard Support Members

The material to be used is Chrome-Molybdenum steel with an elastic limit of 88,600 lbs./sq.in. in the annealed state.

See Fig. 10-A for Diagram

$$R_1 \times 15 - (20 \times 1) \times 10 = 0$$

$$R_1 = \frac{200}{15} = 13.33 \text{ Kips.}$$

$$R_2 = 20 - 13.33 = 6.67 \text{ Kips.}$$

Tension in wires:

$$\frac{T}{13.33} = \frac{(109)^{\frac{1}{2}}}{3}$$

$$T = \frac{13.33 \times 10.44}{3} = 46.5 \text{ Kips}$$

or 23.25 Kips in each wire

<u>Lower support</u>

See Fig. 10-B for Section

$$I_{AA} = \left(\frac{1}{12} \times 1\tfrac{1}{4} \times 5^3 - 2\left[\frac{1}{12} \times 1\tfrac{1}{4} \times 1^3 + 1 \times 1\tfrac{1}{4} \times 1\tfrac{1}{4}^2\right]\right)$$

$$I_{AA} = 8.89 \text{in.}^4$$

$$I_{BB} = \left(\frac{1}{12} \times 5 \times 1\tfrac{1}{4}^3 - 2\left[\frac{1}{12} \times 1 \times 1\tfrac{1}{4}^3\right]\right)$$

$$I_{BB} = \left(\frac{625}{768} - \frac{125}{384}\right) = (.814 - .359)$$

$$I_{BB} = .455 \text{in.}^4$$

$$\frac{T_H}{\frac{13.33}{2}} = \frac{10}{3}$$

$$T_H = \frac{133.3}{6} = 22.2 \text{ Kips}$$

$M_A = T_H \times 4.75 = 22,200 \times 4.75 = 105,500$ lb. in.

$T_V = 6.67$ Kips

Hence:

$M_B = T_V \times 4.75$

$= 31,600$ lb. in.

$$S_A = \frac{M_A c_A}{I_{AA}}$$

$$S_A = \frac{105,500 \times 2.5}{8.89}$$

$S_A = 29,700$ lbs./sq. in.

$$S_B = \frac{M_B c_B}{I_{BB}}$$

$$S_B = \frac{31,600 \times \frac{5}{8}}{.455}$$

$= 43,400$ lbs./sq. in.

$S_{max} = 43,400 + 29,700 = 73,100$ lbs/sq. in.

$\frac{88,600}{73,100} = 1.21 =$ Safety factor (based on Elastic Limit)

<u>Friction of lower support</u>

Coefficient of friction of metal on metal

$= .15$ (lowest) (Ref. 7)

T_H (Total) $= 22,200 \times 2 = 44,400$ lbs.

Using 1" bolts (heat treated) to 100,000 lbs./sq. in.

Area of bolt $= .7854$ sq. in.

Load from bolt $= .7854 \times 100,000 = 78,540$ lbs.

Total load from bolts $= 78,540 \times 2 = 157,080$ lbs.

Total load on upper surface $= 157,080 - 13,333 =$

143,747 lbs.

Frictional force of upper surface =

$.15 \times 143{,}747 = 21{,}562$ lbs.

Total load on lower surface =

$157{,}080 + 13{,}330 = 170{,}410$ lbs.

Frictional force of lower surface =

$.15 \times 170{,}410 = 25{,}600$ lbs.

Total frictional force =

$25{,}600 + 21{,}562 = 47{,}162$ lbs.

Singletree (Fig.8-3)

For section see Fig.10-C

$$I_{AA} = \frac{1}{12} \times 1 \times 3^3 - \frac{1}{12} \times 1 \times \left[\frac{5}{8}\right]^3)$$

$$I_{AA} = 2.23 \text{in.}^4$$

$$M = 23{,}250 \times 4.25 = 99{,}000 \text{ lb.in.}$$

$$S = \frac{Mc}{I_{AA}}$$

$$S = \frac{99{,}000 \times 1.5}{2.23} = 66{,}500 \text{ lbs./sq.in.}$$

$$S.F. = \frac{88{,}600}{66{,}500} = 1.33 \text{ (based on Elastic Limit)}$$

Chrome-Molybdenum Steel:

(Ref.8) Ultimate tension strength = 200,000 lbs/sq.in.

Bearing strength = 146,000 lbs/sq.in.

Shearing strength = 120,000 lbs/sq.in.

$\frac{5}{8}$" pin in shear

$A = .3068$ sq.in.

$A_{Total} = 2 \times .3068 = .6136$ sq.in.

S_{Total} = .6136 x 120,000 = 73,600 lbs/sq.in.

Total load on bolt = 46,500 lbs.

$$S.F. = \frac{73,600}{46,500} = 1.58$$

<u>Eyes</u> (Fig.8-5)

Shear:

$$A = \frac{3}{8} \times 1 \times 2 = .75 \text{ sq.in.}$$

Load = .75 x 120,000 = 90,000 lbs.

$$S.F. = \frac{90,000}{23,250} = 3.37$$

Tension:

$$A = 2 \times \frac{3}{8} \times 1 = .75 \text{ sq.in.}$$

Load = 88,600 x .75 = 66,500 lbs.

$$S.F. = \frac{66,500}{23,250} = 2.86 \text{ (based on Elastic Limit)}$$

Bearing:

A = 1 x 1 = 1 sq.in.

Load = 1 x 146,000 = 146,000 lbs.(uncorrected)

$$\text{Load} = \frac{146,000}{2.8} = 52,100 \text{ lbs.}$$

(2.8 = Frost and Richard's Constant Ref.9)

$$S.F. = \frac{52,100}{23,250} = 2.24$$

<u>Upper clevis</u> (Fig.8-6)

For section see Fig.10-D

Shear:

$$A = \frac{1}{2} \times 2 \times \frac{1}{2} \times 2 = 1 \text{ sq.in.}$$

Load = 1 x 120,000 lbs.

$$S.F. = \frac{120,000}{46,500} = 2.58$$

Tension:

$$A = \frac{1}{2} \times \frac{1}{2} \times 2 \times 2 = 1 \text{ sq.in.}$$

Load = 1 x 88,600 = 88,600 lbs.

$$S.F. = \frac{88,600}{46,500} = 1.9 \text{ (based on Elastic Limit)}$$

Bending:

$$y_1 = \frac{2(\frac{3}{4})(\frac{1}{2}) + 1(\frac{1}{4})}{2 \times 2} = \frac{10}{16} = \frac{5}{8}"$$

$$I_{AA} = \frac{1}{3}\left(\frac{27}{8} + \frac{1}{8}\right) - \frac{25}{32} = .385 \text{ in.}^4$$

$$c = 1\tfrac{1}{2} - \frac{5}{8} = \frac{7}{8}"$$

$$S = \frac{Mc}{I_{AA}}$$

$$S = \frac{23,500 \times 1.1875 \times .875}{3.85} = 63,400 \text{ lbs/sq.in.}$$

$$S.F. = \frac{88,600}{63,400} = 1.4 \text{ (based on Elastic Limit)}$$

<u>Lower clevis</u> (Fig.8-13)

Shear:

$$A = \frac{3}{8} \times 2 \times \frac{7}{8} = \frac{21}{32} = .656 \text{ sq.in.}$$

$$\text{Load} = .656 \times 120,000 = 78,600 \text{ lbs.}$$

$$S.F. = \frac{78,600}{23,250} = 3.38$$

Tension:

$$A = \frac{7}{8} \times \frac{3}{8} = \frac{21}{64} = .328 \text{ sq.in.}$$

$$\text{Load} = .328 \times 88,600 = 29,000 \text{ lbs.}$$

$$S.F. = \frac{29,000}{23,250} = 1.24 \text{ (based on Elastic Limit)}$$

<u>Straps</u> (Fig.8-9)

Tension:

$$A = \frac{3}{4} \times \frac{3}{16} = \frac{9}{64} = .141 \text{ sq.in.}$$

$$\text{Load} = .141 \times 88,600 = 12,492 \text{ lbs.}$$

If loads over 13,000 lbs. are to be used, larger straps will be needed.