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TIPS ON FATIGUE

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TIPS ON FATIGUE

by

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TIPS ON FATIGUE

PREFACE

Sooner or later, metal structures under repeated load wear out. The problem is to be certain that it is later, rather than sooner. For this reason, the Navy has always encouraged research and development to find ways of making structures last longer.

It has been found, however, that it is not enough for scientists and research engineers to know the secrets of fatigue. If designers, shop men, and inspectors do not recognize the signs of fatigue, then the purpose of research and development has not been realized.

To get this message across, plain simple language and forthright pictures, are used, unhampered by superfluous technical jargon, theory, and detailed data displays.

This approach is one way to ensure that the findings of research become the usable knowledge of the man in the shop and the man on the drawing board. The premise is that research and development are worth every cent they cost—if and only if we make full use of the new ideas they produce.

Lack of communication between those who know and those who need to know is often the prime cause of structural failure. If the knowledge gained through fatigue research over the last 100 years were applied, many fatigue problems would never occur (or recur).

Thousands of documents on classic and applied research of metal and structural fatigue literally bury facts by their weight and profundity. These documents should be left to the experts.

On the other hand, the man at the drawing board is a practical man. He needs practical answers to such fundamental questions as: Is he continually making errors that will result in fatigue problems? The fact that new airplanes are still failing in fatigue indicates this to be true. Examination of such failures indicates that many designers are not even aware that sharp notches are fatigue-prone. Corrections for this one fault could save millions of dollars per year and possibly a few lives.

Hence this book proposes to be nothing more nor less than a simple guide on how to:

1. Recognize potential fatigue problems.
2. Rectify existing problems.
3. Avoid getting into situations that may cause problems.

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Principles mentioned herein have to do with (1) relationships of one structural member to another; and (2) paths of load carried within the individual members. This is not to imply that the more abstruse principles of solid state physics would not also be helpful, but these are far beyond the scope of this work. Before fatigue was considered, airplanes were designed to withstand a given static load. This may be in terms of the number of G's the airplane may be expected to encounter during a maneuver, or (in some cases the largest gust that may be encountered once in a lifetime).

In any event, there was some design number. Knowing the strength of the material, it was possible to figure out how much material was required to carry the load. A perfect design was one wherein the structure would carry 100 percent of the design load and fail at 101 percent. Not only that, no component would be relatively stronger than the next. Just like the "wonderful one-hoss shay that was built in such a logical way that it ran for a hundred years to a day." In his poem, "The Deacon's Masterpiece," Oliver Wendell Holmes (father of Supreme Court Justice) chronicles that no part could fail first because each was constructed of the very best material for the function to be served. Undoubtedly, design also had something to do with it.

While airplanes are still designed to carry a certain static load, fatigue poses the additional problem of estimating how long the airplane will last. Ideally, it should last as long as the designer intended it should.

In designing for static strength, the designer was given a set of rules governing the stress levels to which his materials could be worked. This gave some uniformity in design. The nominal stress levels may have differed with location or purpose. For example, the compression allowable would depend on stringer and bulkhead spacing, while tension allowables might depend on the type of fastener.

In fatigue, designing to a uniform nominal stress would not ensure a uniform fatigue strength. A uniform design for fatigue would involve the product of the nominal stress times the stress concentration. Not knowing the stress concentrations, this would be an impossibility. Acknowledgment is due to all those whose encouragement and assistance have made this book possible. While theirs is the glory for any merit in the work, blame for any fault herein belongs to the author alone.

Assistance came from many sources, all remembered and deeply appreciated, though space limits mention to M. S. Rosenfeld, Naval Air Engineering Center, R. L. Creel and C. P. Baum, Navy Bureau of Weapons for review of the entire book.

The author wishes to especially acknowledge the cartoons of W. Goldsmith and T. Adams; the editorial assistance of R. J. Prichard; the organizational help of Ralph DeSola in the early stages of the work; and the technical assistance of G. G. Green.

For the data and photographs that give this informal work a concrete set of examples, especially in Chapter 5, the author wants to thank the following aircraft manufacturers, operators and organizations:

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North American Aviation, Inc.
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Pan American Airlines
Republic Aviation Corp.
Standard Pressed Steel Co.
Transcanada Airlines
Trans-World Airlines
United Airlines
Western Airlines

This book represents an initial attempt to make all levels of personnel aware of the fatigue problem that exists in aircraft structures. It is anticipated that revision [illegible] will be required in the future; consequently users comments are solicited so that a meaningful revision may be accomplished. Similarly, the photographs represent the best illustrations presently available to depict the problems discussed. These photographs were not specifically taken for this purpose; hence they are not all as clear and uncluttered as would be desired Photographs that illustrate more clearly the problems discussed herein or any other fatigue problems occurring in aircraft structures are desired. Users comments and new photographs suitable for illustration should be forwarded to:

Director (S-3)
Aeronautical Structures Laboratory
Naval Air Engineering Center
Philadelphia, Pa., 19112

San Diego, California
30 Oct 1963

C. R. SMITH

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COOLER

Sunny early today with increasing cloudiness tonight. Cloudy with rain tomorrow. Slightly cooler today and much cooler tomorrow. Detailed report on Page a-11.

The San Diego Union

FINAL ★ ★ ★ ★

ESTABLISHED 1868 | BE 4-7111 | SAN DIEGO, CALIFORNIA, MONDAY MORNING, MARCH 5, 1962 | 38 PAGES | DAILY 10 CENTS

Airliner Crashes With 110 Aboard

British Charter Flight Hits Near Atlantic In Africa

LONDON, March 5 (Monday) (AP)—A chartered British airliner carrying a crew of 10 and about 100 passengers crashed last night in the west African republic of the Cameroon, officials announced early today.

There was no immediate word of casualties.

The four-engine DC7C was reported flying from Mozambique in east Africa to Luxembourg on charter to a London firm, Trans-African Air Coach, Ltd.

NEAR AIRPORT

An Air Ministry spokesman said the crash scene was believed to be near the Douala airport not far from the Atlantic coast. He said the plane carried 101 passengers.

A spokesman for Caledonian Airways, owner of the plane, said the number of passengers was "about 100."

Three stewardesses were reported to be among the crew.

The crash came three days after one of the worst aviation disasters in history, a jet crash near New York's Idlewild airport which took 95 lives.

No details of the circumstances of the Cameroon crash were immediately available here.

DETAILS SOUGHT

Nor was there information on the nationality of passengers. Officials said all efforts were being made to get details from Douala but the process was slow.

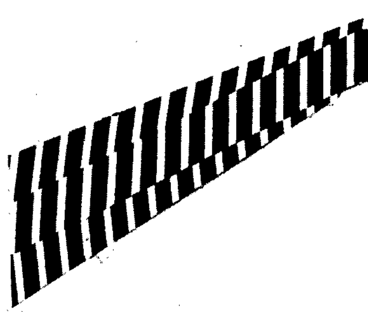
Caledonian Airways, with offices in Prestwick, Scotland, started operating in December after signing contracts with Sabena, the Belgian airline, for the lease of two Douglas DC7Cs to form the nucleus of a fleet for charter service.

The Cameroons, captured from Germany in World War I, were divided between the French in the south and the British in the north.

Last year, the northern part of the territory decided by plebiscite to become part of Nigeria. The southern part voted at the same time to join the republic of Cameroun, which achieved independence from France in 1960.

B I G G E R   H E A D L I N E

I N   T H E   M A K I N G ?



Jet Bounces In Air; 10 Hurt

TOKYO, March 5 (Monday) (AP)—A Pan American World Airways 707 jet airliner carrying 116 passengers bounced in rough air after leaving Honolulu yesterday and 10 passengers required medical treatment at a Wake Island refueling stop.

"We suddenly hit something and the bottom dropped out," Hollywood actor Richard Brooks said. "We went down and the food went up."

Brooks said neither he, nor his actress wife, Jean Simmons, nor their traveling companion, Mrs. Ira Gershwin, was injured.

Dave Jones, Pan American public relations director here, said the plane was not damaged.

He said two of the 10 treated at Wake were given further medical attention here. E. E. Hambree, Memphis, Tenn., suffered a possible fractured wrist while Mrs. Sidney Bergman of Pittsburgh injured her back. The two were not hospitalized.

TIPS ON FATIGUE

1 | THE FATIGUE PROBLEM

1.1 A CENTURY OF STUDY - - - AND FATIGUE STILL FAILS STRUCTURES

The fatigue problem relating to metals and structures has been investigated experimentally for more than a century. In 1849, Jones and Galton investigated cast iron bars in bending. They found that failure occurred in less than 100,000 cycles if loaded to more than one-third of ultimate bending strength. Similar work on wrought iron built-up girders by Fairborn (1860-1861) showed similar results. Wohler's work for the Prussian State Railways goes back to the 1850's when he made an extensive series of tests of various grades of iron and steel subjected to repeated direct tensile and compressive loads, to repeated bending loads, and to repeated torsional loads. Yet we continue to read about and hear about railroad wrecks, automobile smashups, airliner crashes, and other catastrophes directly attributable to fatigue in metallic structures.

1.2 FATIGUE CAN BE BIG, BAD NEWS

"Airliner Crashes with 110 Aboard!" read the black headlines on 5 March 1962. On the same front page, and at the foot of the column describing the loss of the airliner, her crew, and all her passengers, was another report in much smaller type: "Jet Bounces in Air: 10 Hurt." How many such bounces can a structure sustain before it fails and becomes the fact behind an even bigger headline?

Why an even bigger headline? The first plane was a chartered plane on an unscheduled flight. It went down near the coast of West Africa, and whatever happens in remote places never seems as real or critical as what happens closer to home. The second plane carried more passengers and was on a regularly scheduled flight. Its route involved the lives and emotions of hundreds of thousands of people locally. When fatigue failure overcomes the second plane, as it may in the course of time, it is safe to predict that the headline will be bigger, the casualty figures more startling, and the impact on the traveling public even greater.

1.3 TODAY FATIGUE IS A BIGGER PROBLEM THAN EVER

Airplanes in the past were not subjected to loads experienced by present day high speed aircraft. Also, they were built of materials whose tensile strengths were so low that in order to satisfy static strength requirements, stresses for service loading would automatically fall within ranges that would provide an adequate fatigue life.

The fatigue problem has risen at an alarming rate with present day airplanes having high speed and performance. To make matters worse, the materials used in present day airplanes are stronger, yet have no better (and in some cases, poorer) fatigue properties than those used formerly. The result has been an accumulation of service failures, some with fatalities involved. Such situations have involved the designer in unfamiliar areas. Besides, more accurate methods of stress analysis enable us to design structures with greater efficiency and precision.

1.4 THE PANIC FACTOR

Designers are usually shocked by service failures; hence their subsequent designs and modifications of failed parts often include a high panic factor. This panic factor is likely to be far out of proportion to the design improvement needed. The panic factor is born of sudden fright. Sometimes it is compounded with ignorance, and certainly its use is contrary to all the principles of good design.

Unanticipated fatigue failures cause designers to become appalled at the amount of information that seems necessary to estimate the service life of any part or structure. The fact that their knowledge of the ordinary mechanical properties of materials - - ultimate and yield strengths, elongation, modulus of elasticity - - has failed them, leads them to feel fully justified in using the high panic factor. The natural tendency is to "beef-up" the structure that failed, even though this change may not be the solution and the weight penalty extreme. In some cases, a removal of material might solve the problem, whereas a "beef-up" may create a new problem just outside of the "beefed-up" area.

1.5 THE VICIOUS CIRCLE

In aircraft design, every pound of structure added requires additional weight in the form of added power and fuel necessary to carry the added structure. This quickly becomes a vicious circle because more support structure is then needed to sustain the added engine and fuel required to carry the "beefed-up" redesigned structure.

1.6 BREAK THE BIG ONE INTO LITTLE ONES

As long as fatigue is treated as one enormous problem, it never seems to get solved. However, when fatigue is considered as a number of small problems, the solution of each problem becomes apparent. An initial approach to any problem is to list the factors involved, such as:

1. What are the loads?
2. What are the stresses?
3. What are the stress concentrations?
4. How much is the material good for?
5. Could we use a better material?
6. Is that shape necessary?
7. What about careless shop practices?
8. What is the matter with inspection?
9. Why didn't the engineer say so if that is what he wanted?
10. What "bird brain" called out this heat treat?

This list could go on and on. The point is, that in looking over even this short list, the designer, the inspector, and the shop man can each find at least one - - and probably more - - items that he can personally do something about.

1.7 IF EVERYONE DOES HIS OWN JOB WELL ...

Accordingly, if everyone took pains to correct faults in areas where he has influence, a lessening of fatigue failures would surely result.

Take the case of feathered edges. Just because designers did not take the trouble to call for corner radii, and inspectors did not reject parts having sharp corners, repairs were necessary on several recently built airplanes. While the repairs in themselves may not have been costly, the interest at 6 percent on an idle airplane costing $5,000,000 will amount to over $800.00 per day. Add to that the rental value of facilities for repair and wages of an idle crew, and the daily cost is appalling.

A fleet of 300 military airplanes was recently modified to bring them up to desired fatigue life. It cost 3200 man hours per airplane to the tune of more than $11,000,000 for the job. Down time amounted to about 3 months per airplane.

1.8 ALL FOR WANT OF FATIGUE RESISTANCE, THE BATTLE COULD BE LOST

In the case of a military airplane, the cost can be failure to complete a mission, which in a critical situation would be impossible to measure in terms of dollars and cents.

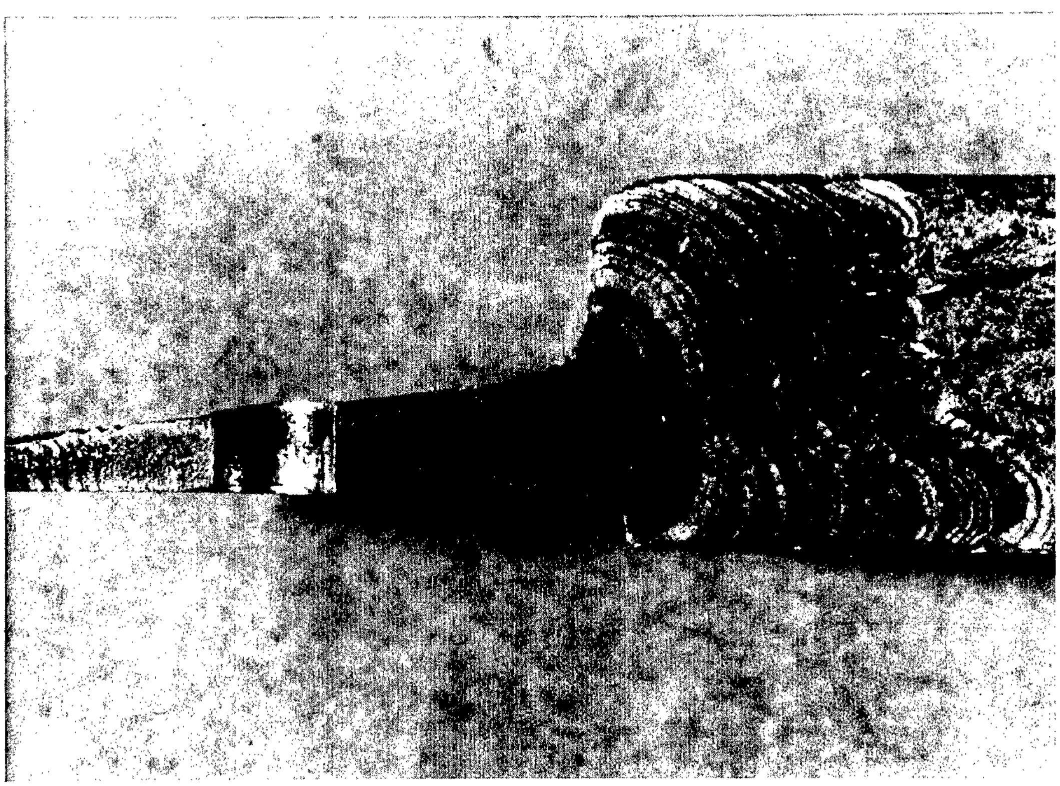


Figure 1.1. "Beach Marks" Identify Progressive Fatigue Failure (See Section 2.8)

1.9 IF YOU CAN GET THEM WHEN THEY'RE LITTLE, FEW PROBLEMS WILL GET BIG

The importance of considering fatigue in design cannot be overemphasized. In many cases, it may seem that the effort necessary for an adequate fatigue design is superfluous and costly. Here is where the fatigue experts should be consulted. In other cases, the optional fatigue design is so simple that little or no cost is involved. These are mainly the items discussed in this book.

2 | BASIC PRINCIPLES

2.1 LET'S KEEP IT SIMPLE

This chapter will present only those principles that practicing engineers, inspectors, and shop personnel can apply. In doing so, some of the more basic fundamentals of crystalline structure, such as slip planes, dislocations, and others will be omitted. These are beyond the scope of this work. While the principles discussed are adequate for the purpose intended, the reader is directed to Appendix C, "Suggested Further Reading," for more detailed technical information.

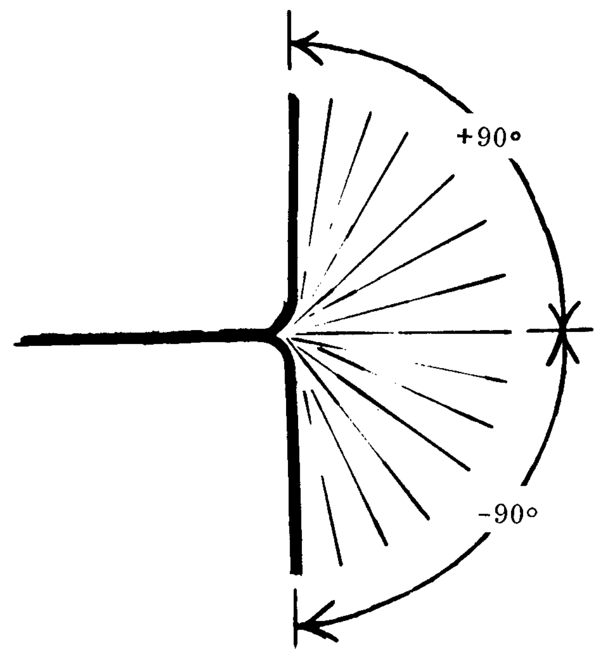
Knowing the behavior of metals under load is vital to the understanding of fatigue. Every day, common occurrences show how metals act when loaded. A thoughtful consideration of these examples is probably the easiest way to summarize some of the more basic principles of metal fatigue.

2.2 IT'S NOT ONLY WHAT YOU DO, IT'S HOW YOU DO IT AND HOW MANY TIMES

2.2.1 UNIDIRECTIONAL VERSUS REVERSE BENDING

Repeated bending is a familiar example of fatigue.

Have you ever noticed that you can break a wire quicker by increasing the bend angle?



5 CYCLES
90-DEGREE REVERSE BENDING

A galvanized 14-gage wire that breaks in four or five cycles of 90-degree reverse bending . . . .

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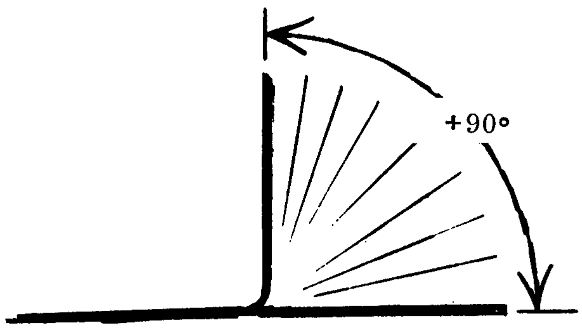
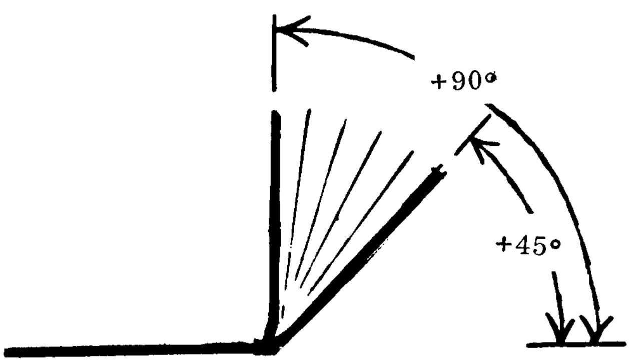


. . . will last for 15 or 16 bends of from 0 to 90 degrees.

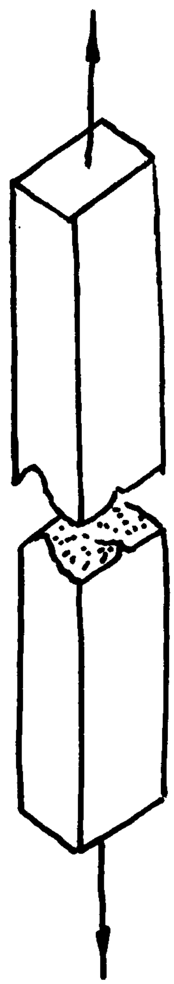
15 CYCLES
0 to 90-DEGREE BENDING



If bent only from 45 to 90 degrees, it may last from 60 to 70 cycles.

70 CYCLES
45-to 90-DEGREE BENDING

83,000 POUNDS



ONE-INCH-SQUARE BAR BREAKS
AT 83,000 POUNDS

Similarly, the life of a structure may depend more on how it is loaded than on the total number of times it is loaded, or on the maximum amount of the loads themselves.

This is best shown by loading an inch-square bar of aluminum alloy that breaks at 83,000 pounds when loaded once.

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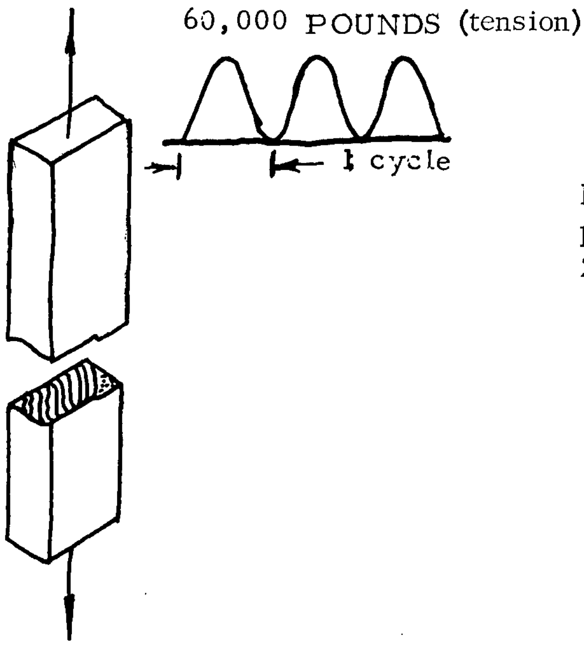
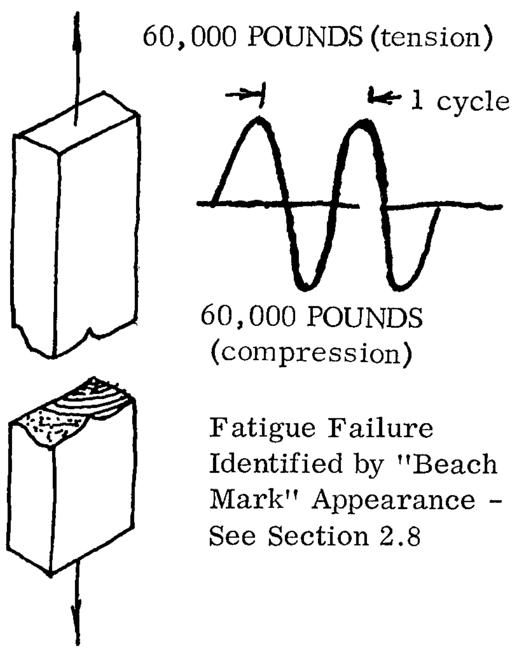


However, if loads of from 0 to 60,000 pounds were applied, it would last about 25,000 cycles.

60,000 POUNDS APPLIED 25,000 TIMES WILL BREAK THE BAR



Loadings from 60,000 pounds tension to 60,000 pounds compression (commonly called "plus to minus 60,000 pounds") would fail the bar in about 4,000 cycles.

REVERSING THE 60,000-POUND LOADS BREAKS THE BAR IN 4,000 CYCLES

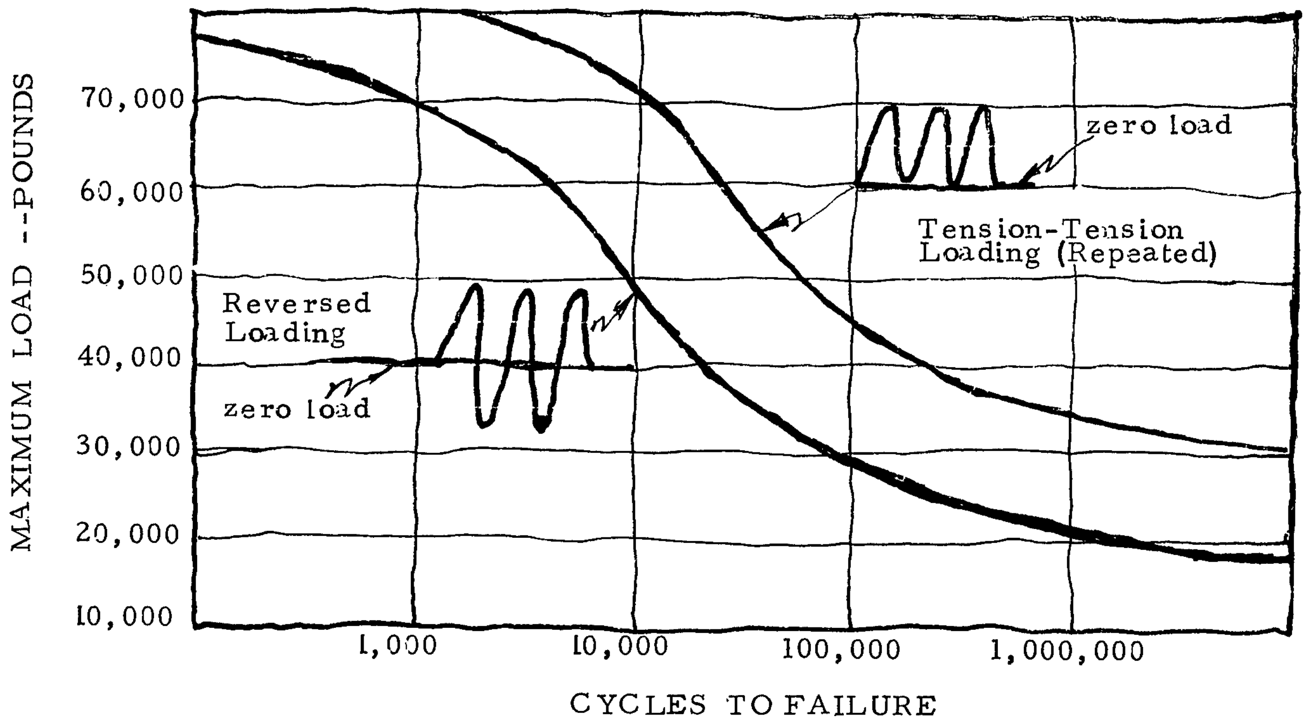


Figure 2.1 Load Versus Fatigue Life for One-Inch-Square Bar of 7075-T6 Aluminum Alloy

Similar relationships between repeated tension and reversed loads are plotted in Figure 2.1. Load is shown on the vertical axis (ordinate) and the number of cycles on the horizontal axis (abscissa). So that the lifetime scale could be condensed to fit on one page, the abscissa has been compressed in what is commonly known as a logarithmic scale. Note that each major division represents 10 times the value of the previous division. Schematic diagrams of load cycles are shown to facilitate reading.

2.3 CONVERTING LOAD TO STRESS

The inch-square bar was used in the preceding example for two reasons. First, a square inch is a standard unit of measure. Second, when fatigue effects are understood in terms of a square-inch cross-section, it is easy to compare the load-carrying ability of our known example with the load-carrying abilities of structures having other dimensions. In other words, load carrying ability is then expressed in terms of pounds per square inch of cross-sectional area.

Commonly expressed in terms of load divided by cross-sectional area, the shorthand or algebraic description for stress is

$$S = \frac{P}{A}$$

where

S = stress in pounds per square inch
P = load in pounds
A = cross-sectional area in square inches

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2.4 FATIGUE SHORTHAND

Since fatigue life is not only dependent on the amount of stress, but also on how the stress is applied, a system has been devised identifying the type of loading, thus

$$R = \frac{S_{min}}{S_{max}}$$

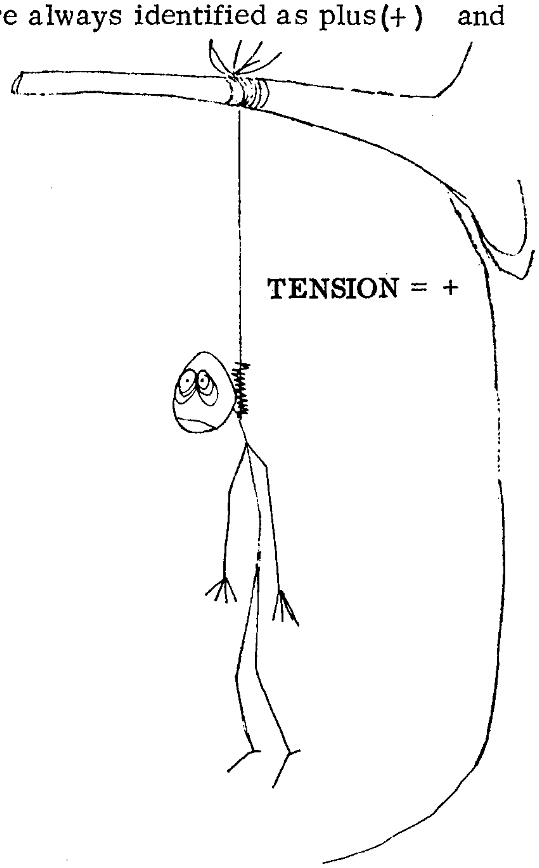
This is simply the ratio of the minimum stress divided by the maximum stress. Using this notation, the curve for repeated tension loading in Figure 2.1 would be identified as $R = 0$, because the minimum load was zero and zero divided by anything is still zero.

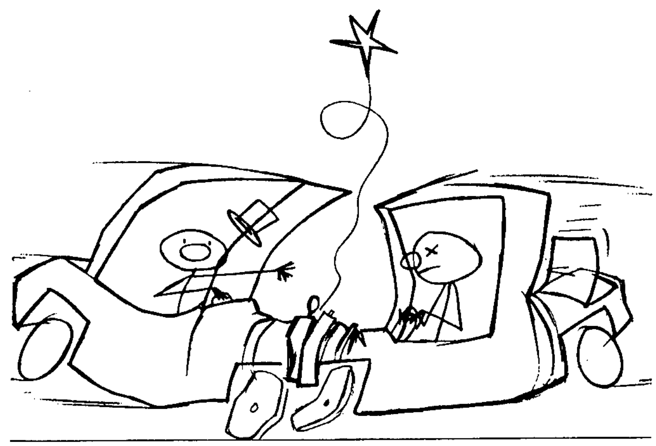
2.5 TENSION AND COMPRESSION LOADING

According to convention, tension stresses are always identified as plus(+) and compression stresses are minus (-). In Figure 2.1 the curve for reversed loading would have a stress ratio of

$$\frac{S_{min}}{S_{max}} = -1$$

since S_{min} is equal to S_{max}, except for the opposite signs.





COMPRESSION = -

2.6 S-N CURVES FOR SMOOTH SPECIMENS

To present fatigue data in brief form, curves of stress versus the number of cycles to failure, (called S-N curves) are used. Since the dimension of the bars in Figure 2.1 was one-inch square, applying $S = P/A$, the curves shown are also S-N Curves. Frequently, a whole family of curves is given in order to show lives for other ratios of stress (R). Figure 2.2 shows a family of curves. Appendix C, "Suggested Further Reading," contains references to S-N data for other materials. Curves for typical airplane structures are given in Appendix A.

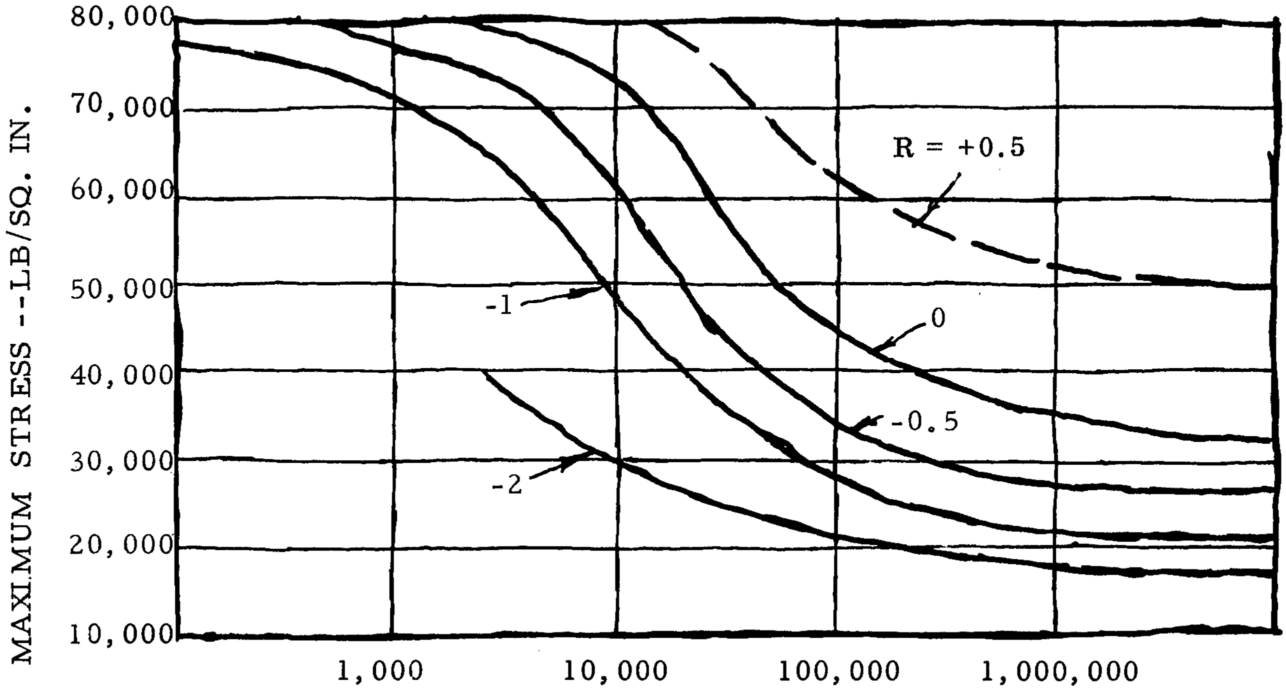


Figure 2.2 A Whole Family of Curves, Showing Lives for Various Ratios of Stress, R.

2.7 STRESS CONCENTRATIONS

In the case of airplane structures, the fatigue behavior will be substantially different from that observed in the square-inch bar, because free flow of stress is interrupted by obstacles such as holes, notches, bumps, and changes of section. Piling up of stress at obstacles such as these is commonly called a concentration of stress, and the obstacles themselves are known as stress raisers.

2.7.1 UNFILLED HOLE -- In the case of the one-inch bar, a small hole through its center would cause the actual stress at the edge of the hole to be about three times that away from the hole.

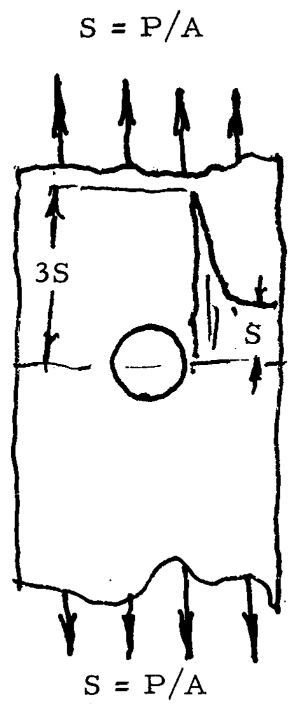


Figure 2.3. Bar with Centrally Drilled Hole

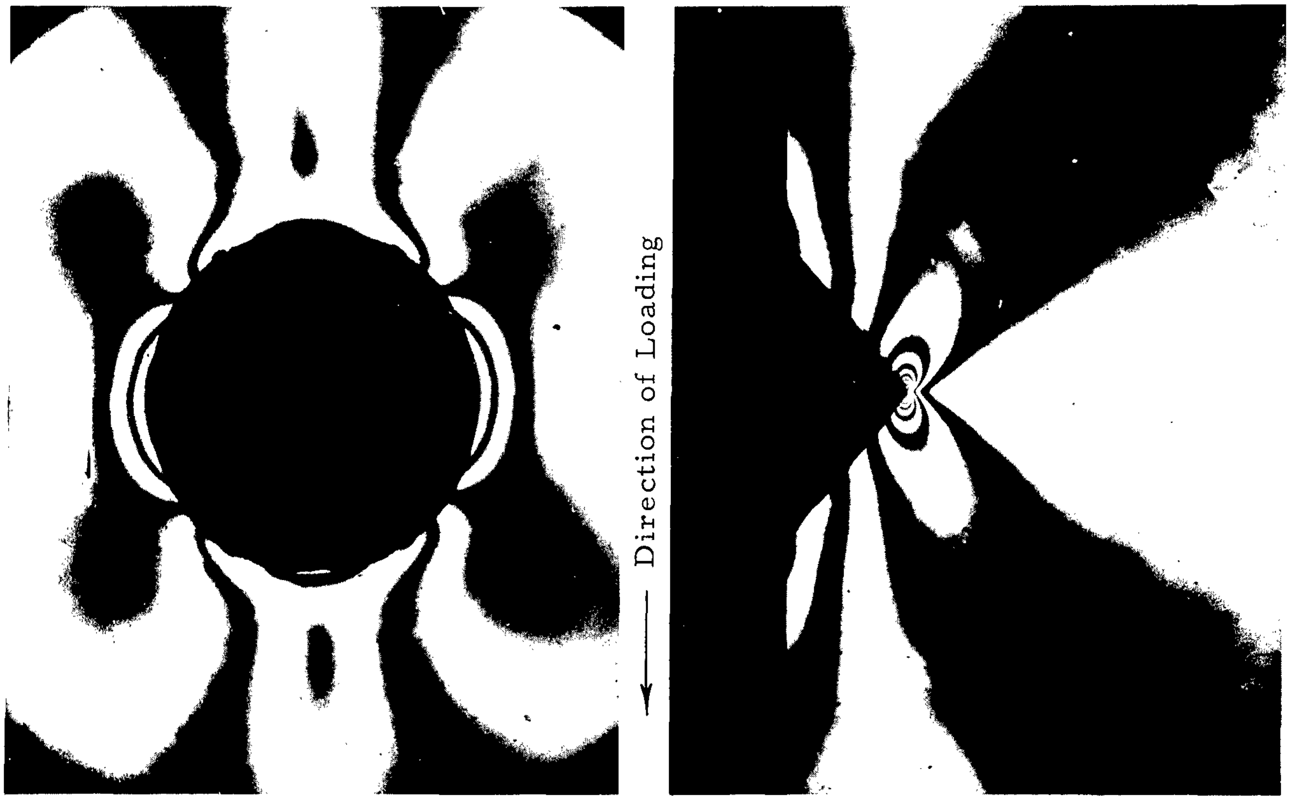


Figure 2.4 Photoelastic Models, Showing Stress at Edge of Hole and at V Notch

The stress distributions around two different concentrations are shown in Figure 2.4. The left picture shows the stress distribution at the edge of an unfilled hole and the right picture shows what happens around a V-notch. Here, an experimental stress analysis technique, called photoelasticity, is used to visually demonstrate locations of highly stressed areas. This technique employs polarized light and clear plastic models in which stressed areas become opaque. This is an especially valuable tool for demonstrating relative merits of design. Amount of stress is directly related to the number of opaque lines and concentration is proportional to the line spacing. As in judging the steepness of the terrain by the contour lines of a topographical map, photoelastic patterns tell the steepness of stress.

2.7.2 WATCH OUT FOR OPEN HOLES -- Most structures have holes. Open holes are usually worse than rivet-filled holes. Stress at an open hole is three times that away from the hole. For this reason, open holes should be avoided in regions of high stress. Where location holes are an absolute necessity, plug them with rivets if possible. Never plug holes with weld, as this creates high residual (locked up) tensile stresses.

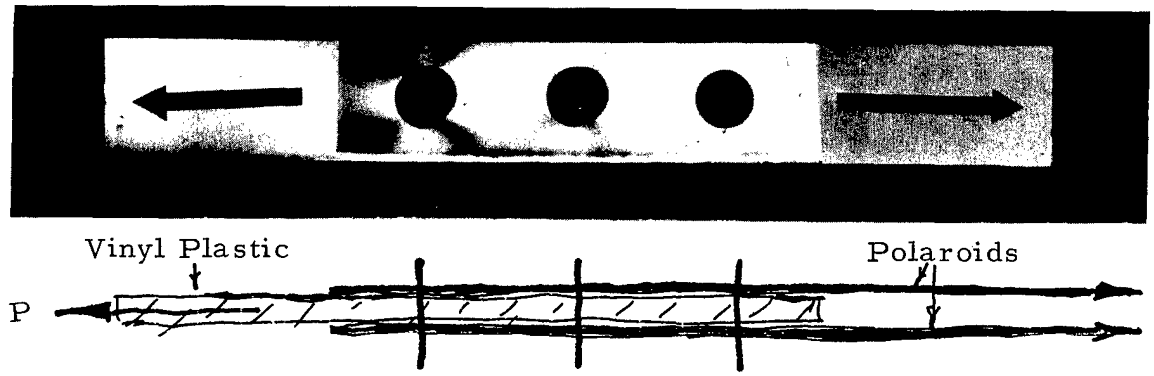


Figure 2.5. Photoelastic Model, Showing Distribution of Load in Fasteners of a Clevis Joint

2.7.3 RIVETS AND BOLTS -- Riveted or bolted joints always constitute problems. One reason is that the load introduced by the rivet or bolt increases the stress at points of concentration.

The second reason is that it is virtually impossible to distribute the load evenly between rivets or bolts, irrespective of workmanship. This is because the second and successive rows of rivets cannot carry their share of load without some stretch in the splicing material between the first two rows. In fact, the stretch in the splicing material should be greater than that of the material being spliced at this point. Note in Figure 2.5 that the fastener nearest the load has the highest stress. A solution to this problem would be to make this fastener incapable of carrying so much of the load. Unfortunately, reducing the size of the fastener is not always a solution. In fact it is an invitation to trouble with the fasteners themselves. Perhaps a wiser choice would be to remove some of the splice material so that it would not be able to overload the first fastener or fasteners. The thinner splice material stretches, thus allowing some of the load to be carried by the second row of fasteners. This is illustrated in Figures 2.6 and 2.7, where edge views of photoelastic models are shown. The model in Figure 2.6 is very similar to the clevis joint shown in Figure 2.5. Being cut from one piece of material, the model in Figure 2.6 clearly shows that a good load distribution cannot be attained by providing a better fit.

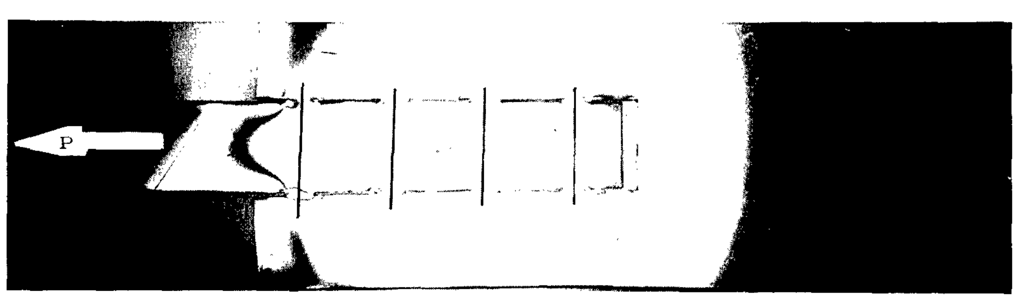


Figure 2.6. Photoelastic Model of Clevis Joint, Showing Edge View of Load Distribution Between Fasteners

Figure 2.7. Improved Load Distribution by Scarfing

2.7.4 SCARFING — Figure 2.7 shows an ideal joint wherein the load at the first fastener is relieved by scarfing the material. Practically, this is hard to do. Other methods of relieving the load at the first fastener will be discussed under joints and joining in Chapter 3. In the meantime, this would be one of the cases where the senior engineer or fatigue specialist should be consulted.

2.7.5 SINGLE VERSUS DOUBLE SHEAR — The photoelastic models of joints so far have been of the clevis or double shear type. Unfortunately, aerodynamic smoothness requires designing many airplane joints in single shear, the worst example being the simple lap joint. The major fault with a lap joint is that the sheets being joined tend to align with each other, causing severe bending stresses at the first fastener. This is shown by the photoelastic model in Figure 2.8. Figure 2.9 is an improved lap achieved through scarfing. Lap joints will be discussed further in Chapter 3.

2.7.6 NOTCHES — Like the open hole, the notch constitutes one of the most harassing problems in fatigue. Figuratively speaking, a notch is comprised of any kind of a discontinuity, the hole being one of the many types. Thus, it is common to speak of the "notch effect" when the notch being considered is in reality a bump.

Accordingly, the term "notch" can be applied to holes, grooves, notches, bumps, etc. Perhaps the most troublesome of all notches is the fillet radius. Machinists seem to enjoy machining neat corners, instead of providing generous radii at the junctions of two surfaces. The importance of providing generous radii is shown in Figure 2.10, where a photoelastic model shows how stress at a fillet can be relieved by providing a more generous radius.

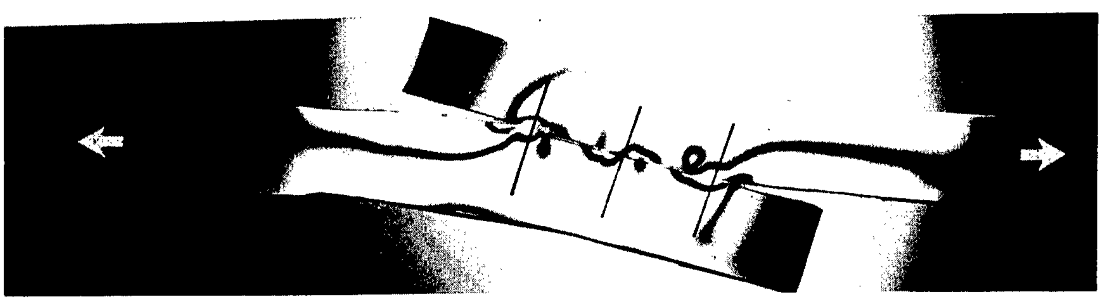


Figure 2.8. Simple Lap Joint -- Note Bending

Figure 2.9. Scarfed Lap Joint

A proper radius is especially critical in machined parts where only a slight change could determine whether the part were satisfactory or not. Most important, however, is to make certain that an additional notch is not created by the machined radii failing to meet the flat surface smoothly as shown in the photoelastic model in Figure 2.11. Further examples of the effects of radii on fatigue are given in Chapter 5.

2.7.7 FRETTING—The erosion of two surfaces rubbing against each other is known as "fretting." The notch effect of the pitted surfaces tends to exaggerate the effect of other notches so that the combination is a superimposed stress concentration. Fretting is easily recognized by

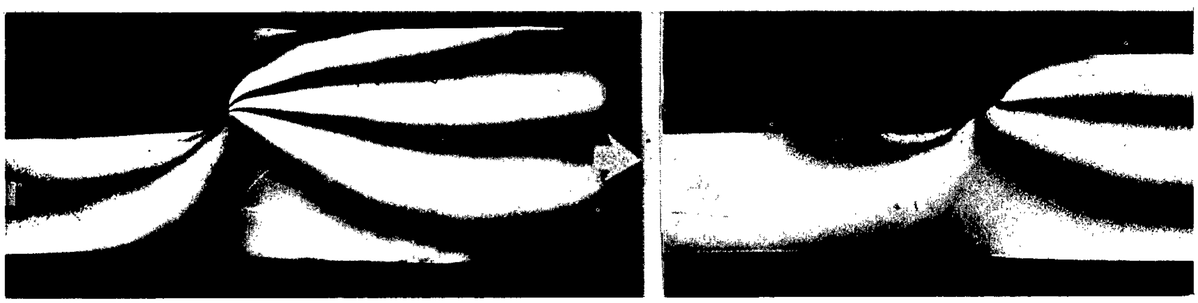


Figure 2. 10 Photoelastic Models Showing Effect of Fillet Radii on Stress

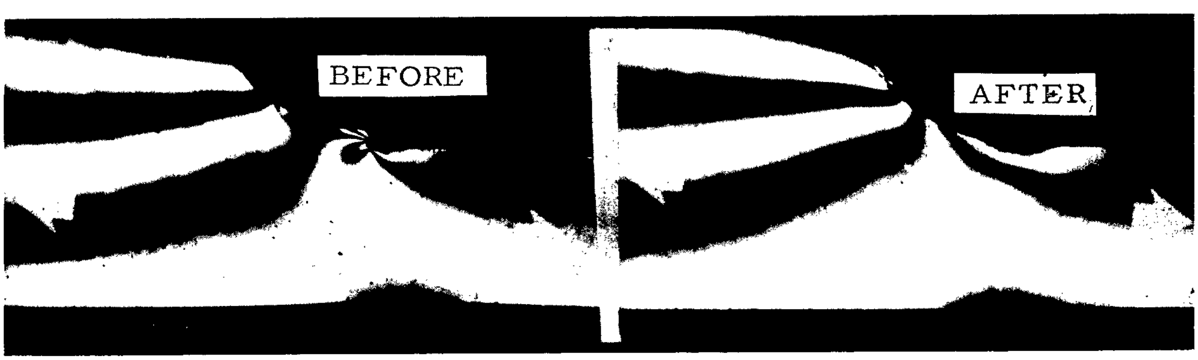


Figure 2. 11 Photoelastic View of Machined Radii Not Meeting -- Stress is Indicated by Number of Dark Lines

powder that sifts out from between the surfaces.

Since fretting is caused by rubbing, anything that will reduce the amount of rubbing will reduce fretting. Equalizing the amount of stretch between material being spliced and splice material, as illustrated in the photoelastic models of scarfed joints, will help. Adhesive bonding (see Section 3.9) also helps. Lubricants can be helpful in special cases. Consult the specialist on this.

2.8 IDENTIFYING FATIGUE FAILURE

Those of us who have had the dubious pleasure of looking at the broken end of a drive shaft or a rear automobile axle, can never forget what it looked like. We may also remember the mechanic saying that the shaft was old and crystallized, and that was why it failed.

This explanation, however, is not necessarily accurate. All metal is crystalline. However, because fatigue cracks propagate through the crystals instead of around them (as in the case of the one-time loading, or static failure in a ductile material) outlines of the crystals come into clear view.

Fatigue cracks propagate at various rates, depending on the material and loading. The stress concentration at the end of a crack, being extremely high, causes the material to fatigue locally so that the crack continues until enough fresh material (not yet fatigued) is engaged to resist loading for another interval. This gives rise to the "beach" mark appearance of fatigue failed parts as shown in Figure 1.1. Sometimes called "tide" marks, they are useful for locating origins of fatigue failure. The beach marks left by high loading are usually spaced farther apart than those caused by low loading. In either case, the origin is usually at a point of stress concentration or nucleus, and the beach marks propagate in circular patterns with the nucleus as the center. Usually, the marks near the origin are obliterated by rubbing of fractured surfaces against each other. Thus in Figure 1.1, the origins were probably at lower corners of the hole; however, beach marks do not appear until some distance away. After fatigue cracking had progressed to the last beach mark at the right, there was insufficient remaining area to carry the load. Static failure finally resulted as indicated by the rough surface at the right.

3 | JOINTS AND JOINING

3.1 TOO MUCH STRESS IN THE WRONG PLACES

There was no fatigue problem in aircraft during the era when they were made of wood. It was only with the advent of all-metal airplanes, and in particular of high-strength metal airplanes, that fatigue became a problem. Why? Too much stress in the wrong places!

The reason we have too much stress in the wrong places is that we have thrown away the simple approach used in gluing wood, and instead we resort to the obvious boiler plate construction. Now, boiler plate construction is fine when used on boilers, but we don't have to build airplanes like that.

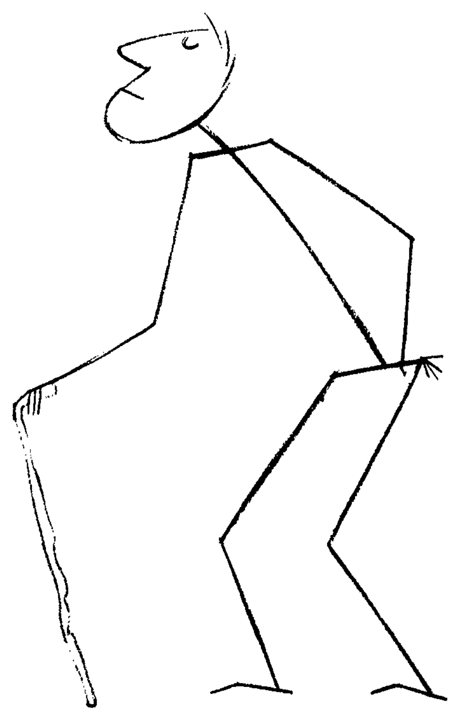
Wooden airplanes were built with cabinetmaker techniques, and the cabinetmaker tried to join his structure in such a manner that the joint was not apparent to the eye -- nor to the stress. When the same technique was used on wooden airplanes, the stress flowed from one piece to another as if they were one. Maybe it was luck, but the result was a continuity of stress flow.

3.2 WHY HAVE JOINTS?

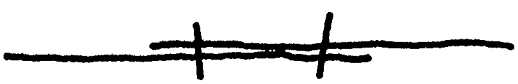
Joining, to begin with, is a technique used only when the structure cannot be built in one piece. Ideally, the load is evenly distributed throughout the structure to afford a continuity of stress flow. Accordingly, the more nearly the joining resembles a single piece in this respect, the better the joint. The cabinetmaker's long-scarf joint very nearly satisfied this condition.

3.3 TYPES OF JOINTS

The most obvious way to join two sheets of material together is to lap the edges of one piece over the other and fasten them



with some device. Historians tell us that man's first attempt of this sort was probably fastening two pieces of animal skin together with a fish bone. The result would be known today as a lap joint.



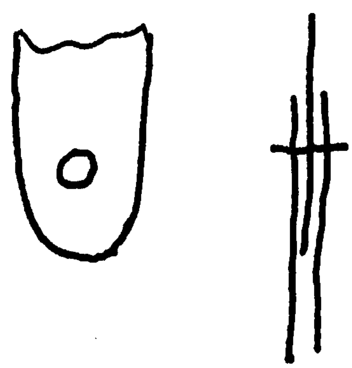
LAP JOINT

A neater way is to butt the two sheets edge-to-edge and fasten each to a third sheet (splice plate) underneath. This gives a smooth surface on one side that is not only pleasing, but also is aerodynamically superior if the sheet happens to be the outside skin of an airplane. The butt joint with a single splice plate or doubler is known as a single shear butt joint.

SINGLE SHEAR BUTT JOINT

DOUBLE SHEAR BUTT JOINT

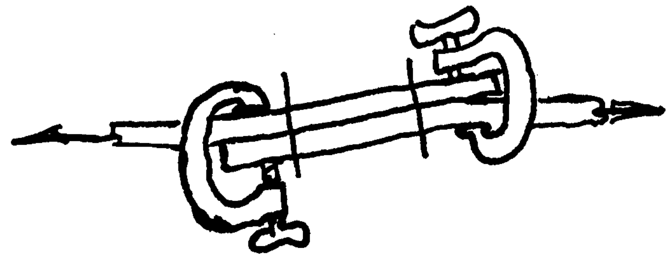
Another type of butt joint holds the two pieces of material with two splice plates. This is called a double shear butt joint. Being symmetrical, it has a fatigue strength superior to that of either the lap joint or single shear butt joint. The double shear butt joint is preferable wherever cost and aerodynamics permit.



LUG

A fourth type of joint is the simple lug. It usually consists of a clevis and a single fitting that is pinned between the clevis by a single bolt or other fastener. This type is generally used for moving parts where bushings or bearings are used for lessening friction.

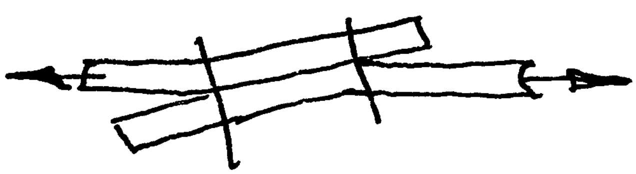
Many variations of the above joints could be mentioned; however, their problems and solutions are similar. Several exceptions, such as hooks and piano hinges, need a specialist's attention.



CLAMPS? -- Not so bad if bending is away from rivets

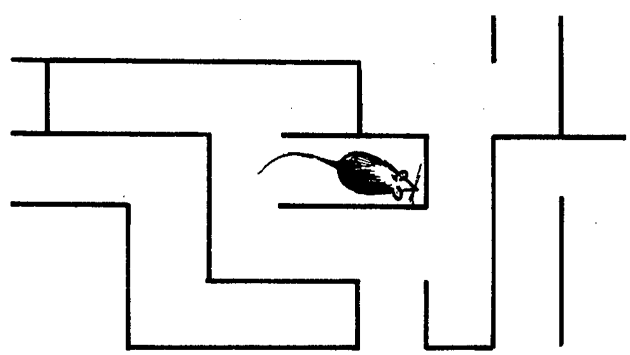
3.4 WHAT'S THE MATTER WITH JOINTS?

3.4.1 LAP JOINTS — While the lap joint is the simplest of all joints, its main problem is that, when the two sheets of material are joined, they tend to align themselves with each other. This causes the sheet to be bent at the first fastener, which is already suffering from too much load (see Figure 2.8). This offset in alignment is commonly called eccentricity.



LAP JOINT -- Sheet bends right where it hurts most

The logical solution would be to let the sheet bend, as long as it didn't bend right where the load was greatest. Ideally, you could clamp the sheet to make it bend at some other point; this would separate the bending stress from the load-carrying shear stress. A trick such as this is frequently called "confusing the stress" or "stress confuser." See Chapter 6 for other stress confusers.

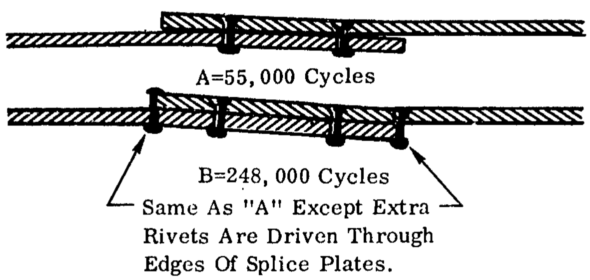


STRFSS CONFUSER

STRESS CONFUSER

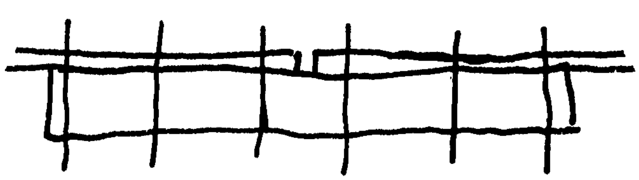
While using a C-clamp would be highly impractical in an airplane, the same results can be obtained by driving extra rivets through the edges of doublers.

In fatigue tests, an ordinary lap joint with two countersunk rivets as in A



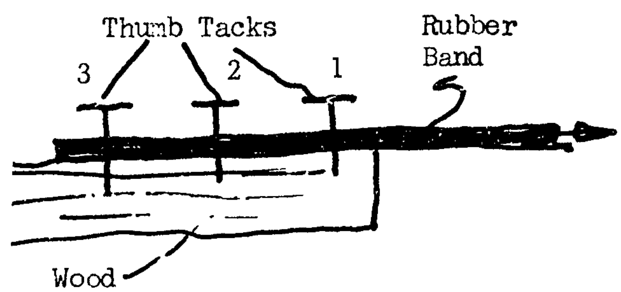
failed after 55,000 cycles of repeated loading. By driving extra rivets through doubler edges as in B, the fatigue life was raised to 248,000 cycles. The second joint lasted longer because the edge-driven rivet could take no load other than that caused by sheet bending, thus passing the shear load on to the next rivet, which was thus relieved of the bending load.

3.4.2 SINGLE SHEAR BUTT JOINTS — The single shear butt joint is really two lap joints facing each other, so it has the same problem as the lap joint. One of its advantages is that the doubler can be made thicker than the material being spliced. This reduces the effects of bending, but it creates an additional problem: the rivet nearest the doubler's edge now carries most of the load just as in the case of the clevis joint shown in Figure 2.5.



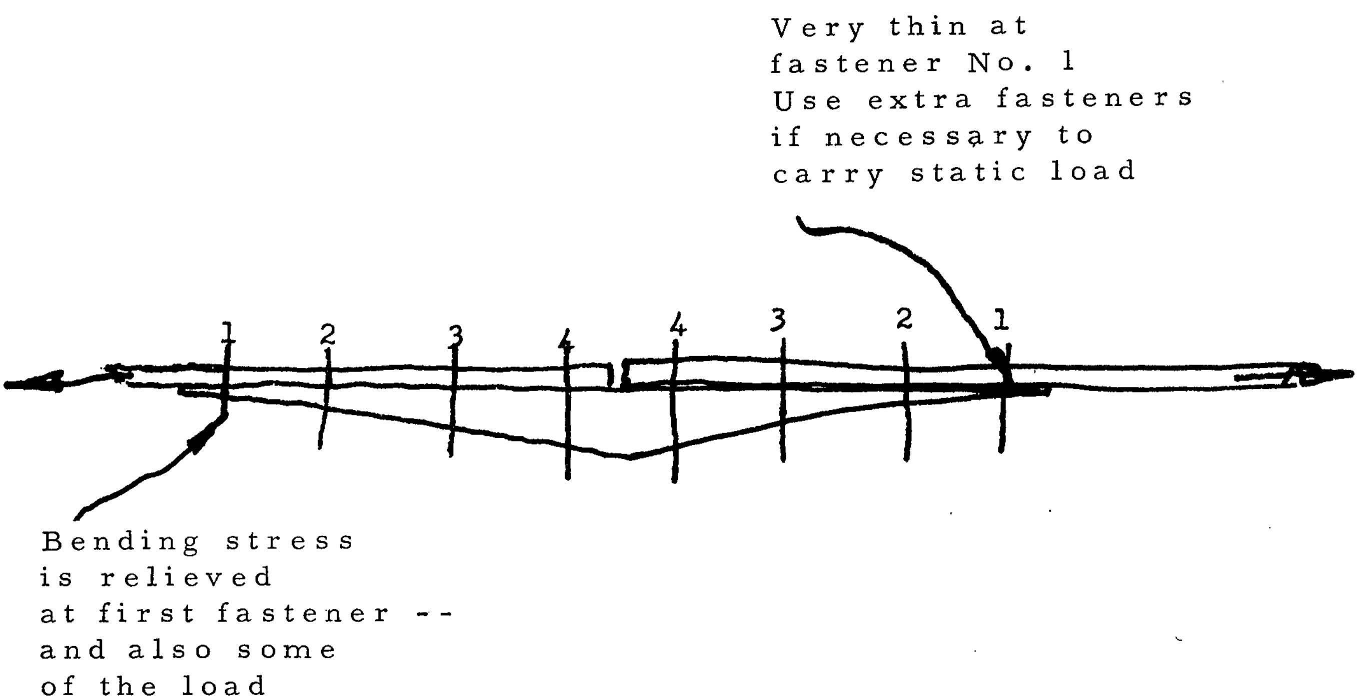
HEAVY SPLICE PLATE -- Reduces bending, but makes rivet carry too much load

To visualize this, stake the end of a wide rubber band to a board with three thumb tacks and pull. Note the amount of deformation required at the No. 1 fastener before the No. 3 starts to carry the load. This would indicate that some provision must be made for the splice plate to stretch if the No. 2 and No. 3 fasteners are to carry their fair share of the load.



EXPERIMENT -- Showing load division between fasteners

Some degree of deformation can be achieved by thinning the doubler material between the first two rows of fasteners so the second row can carry

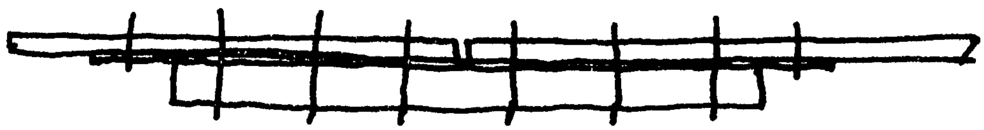


TAPERED SPLICE PLATE

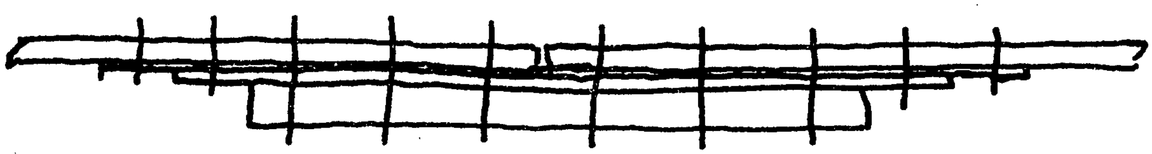
some of the load. Since doubler material must stretch in order to do this, the thickness at the first fastener should be less than half that of the material being spliced.

The value of one-half is arbitrary; the point being that, with this thickness, the No. 1 fastener won't overload the spliced material. High loads would cause doubler material at the No. 1 fastener to yield in bearing -- which is good, if the remaining fasteners can carry the design load. Thus, for static strength it might be a good policy not to rely on the first row of rivets but, instead to provide extra fasteners for the job.

Theoretically, the doubler should taper to almost nothing so that the No. 1 fastener carries an infinitesimal part of the load — something on the order of the cabinetmaker's scarfed wood joint. Generally, the extra machining is impractical; sometimes, however, the weight saving does make it worthwhile. Almost as good as the thick, tapered splice plate is the thin auxiliary doubler next to the material being spliced. The auxiliary doubler should be long enough to engage an extra row of rivets outside the main splice area. Here again, there is a compromise between the practical and theoretical optimum thickness of auxiliary doublers.



AUXILIARY DOUBLER



TWO AUXILIARY DOUBLERS

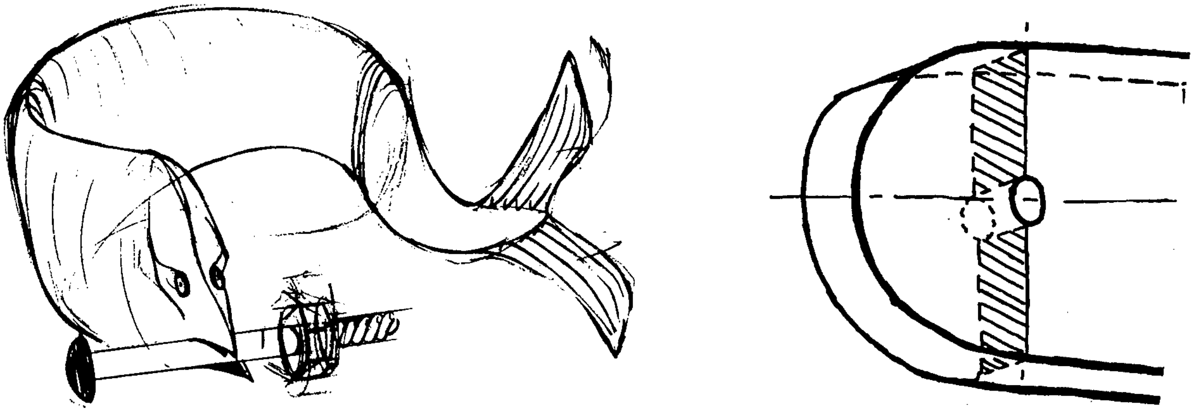
For splices in aluminum alloy, an aluminum alloy auxiliary doubler about one third as thick as the spliced material is about right. Where two auxiliary doublers are used, the one nearest the butt should be about one fifth as thick as the material spliced.

Auxiliary thin doublers when properly used will increase the lifetime more than ten times.

3.4.3 DOUBLE SHEAR BUTT JOINTS — Double shear butt joints are superior to those of the single shear type. This is because the symmetry of the double shear type eliminates the bending effects found in the single shear. However, the double shear type also has the problem of load distribution between fasteners. (See photoelastic models in Figures 2.5 and 2.7). Scarfing, or providing auxiliary doublers as for single shear joints, will improve fatigue life.

3.4.4 LUGS — The lug is a simple form of the double shear joint. Since the joint has but one fastener, the problem of load distribution between fasteners does not arise. Whether or not the fastener (usually a bolt) fits tightly has made a substantial difference in test results. A loose bolt tends to bend more and will sometimes fail in the middle of the tongue, or male fitting. It also introduces an ex-

TIPS ON FATIGUE



WIDE THIN LUG -- easy on bolt but bad on fatigue

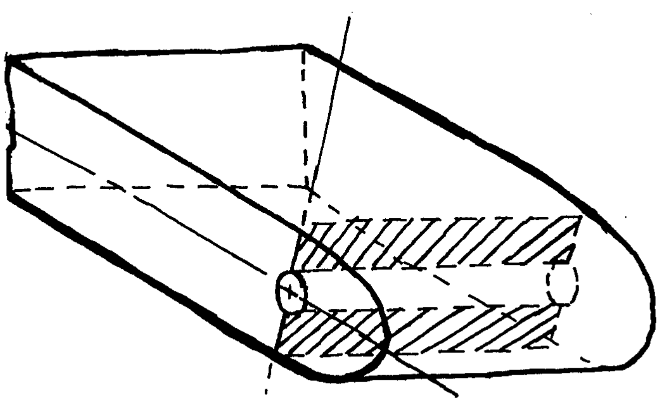
tremely high bending stress on the corners of the clevis, or female fitting. Thick lugs with closely fitted bolts or bushings will have twice the fatigue life of lugs with sloppy fits. A good interference fit will improve the life many times. See Chapter 6 for more on this.

Because of the bending effect, it is a good idea not to stint on bolt size. Indications are that it would be helpful to have the bolt even twice as strong as the lug. This keeps from overloading the corners of the lug.

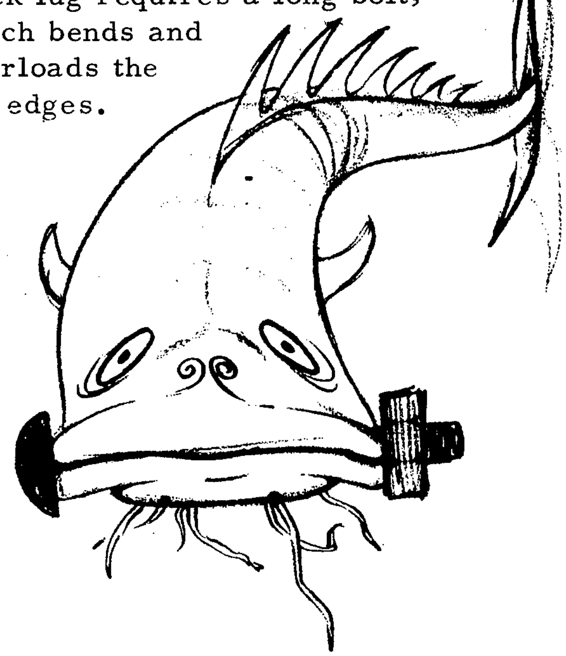
Also, you won't have to worry about the bolt.

Shape of the lug's cross section is very important. A wide thin lug, while relieving bolt bending, causes the stress at the edge of the hole to be many times the average stress away from the hole. (See concentration factors in Appendix B).

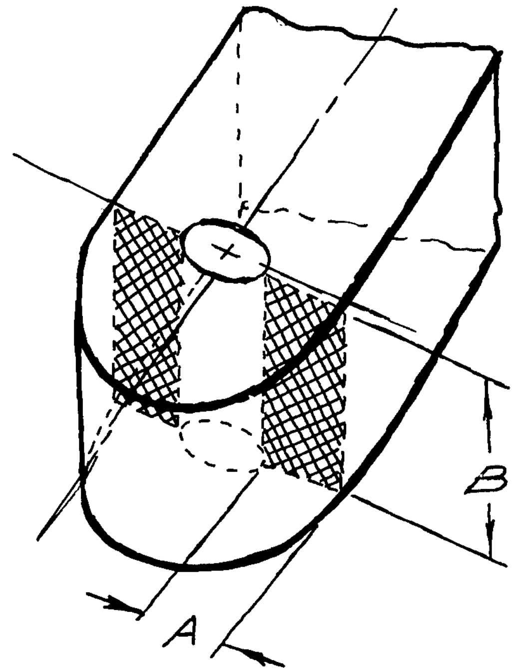
On the other hand, a narrow, thick lug requires a long bolt, which bends and overloads the lug edges.

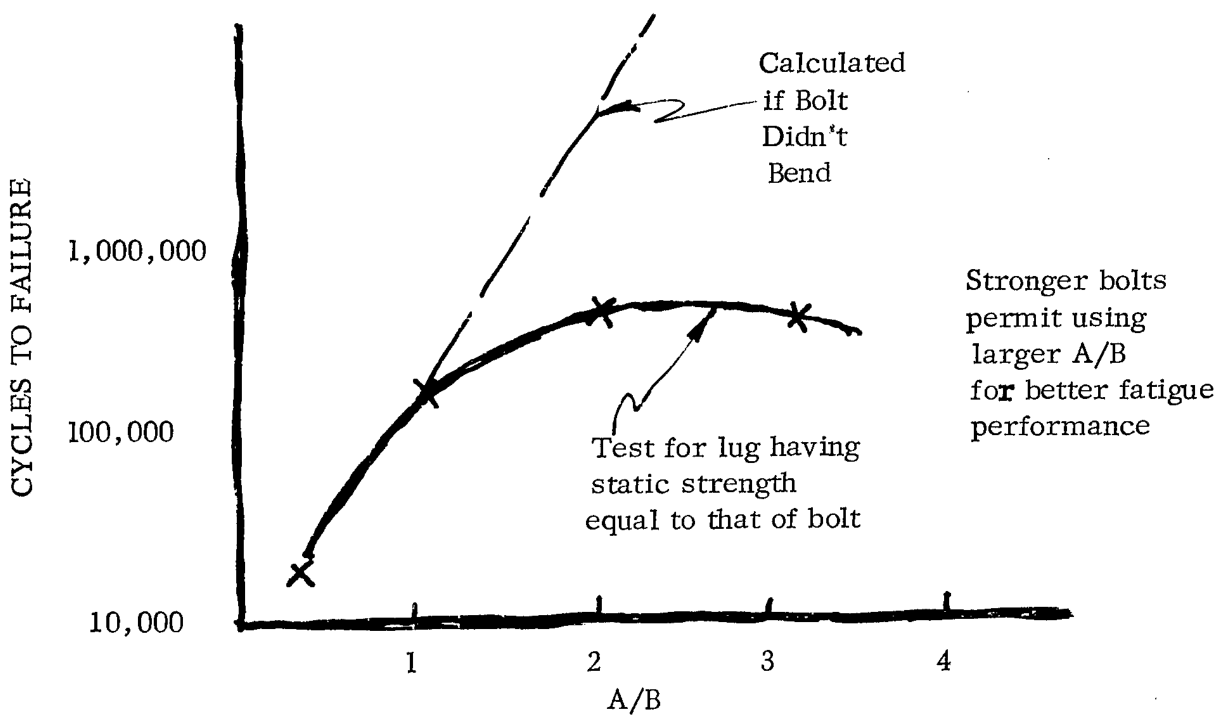


NARROW THICK LUG -- better than wide, thin lug, but bends bolt too much.



Tests have shown that the lug with cross sections with ratios of A/B between 1 and 3 would be best where the bolt shearing strength is equal to the strength of the lug.

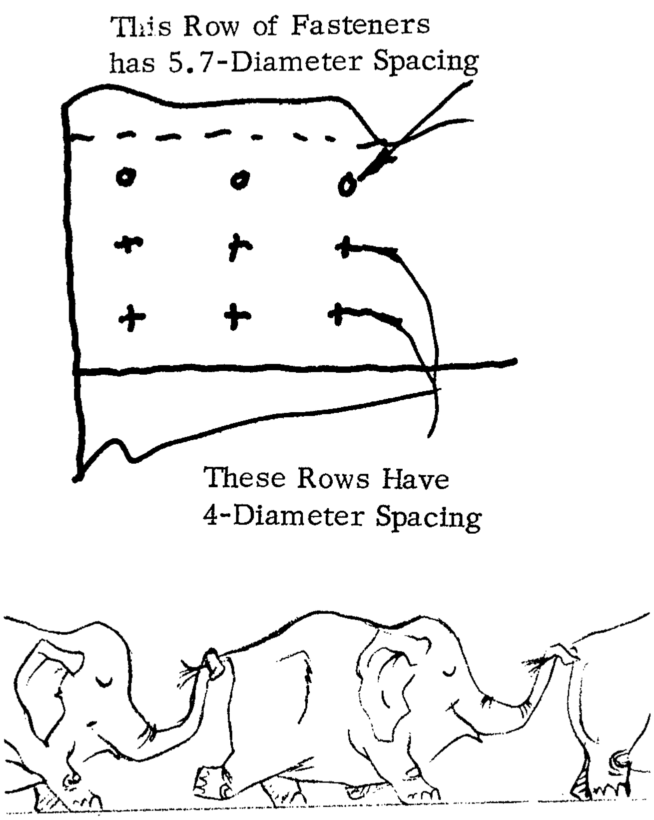




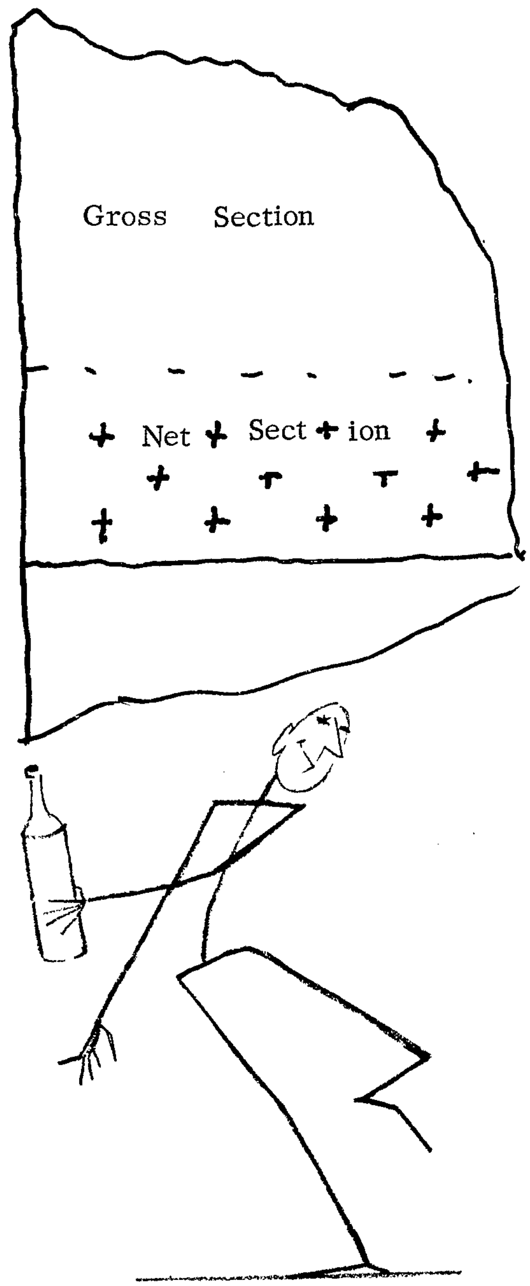
3.5 FASTENER SPACING

Much has been written about spacing fasteners in a joint, most of it from a static strength point of view. Accordingly, handbooks recommend fastener sizes and spacing that leave as much of the area as possible in the spliced material. For static strength, the net cross sectional area is usually no less than 75 percent of the area away from the splice. (Net cross section is the area remaining after removal of material for holes.) This would give a fastener spacing (commonly called pitch) of about four times the fastener diameter. To further enhance the static strength, the first fastener nearest the load is sometimes reduced in size.

Frequently, fasteners are staggered. Joints of this type have never proven to be any better than the tandem pattern. Staggering, however, is desirable for fuel sealing or other uses where joints should not leak.



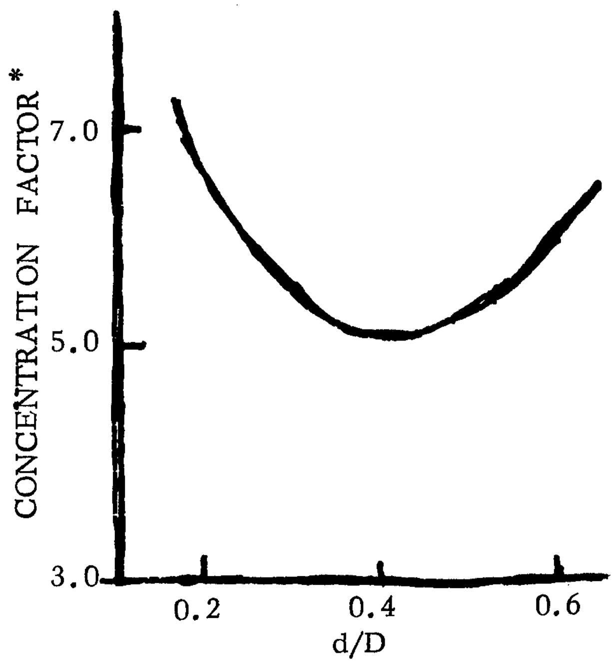
TANDEM PATTERN
Good Static Strength Joint



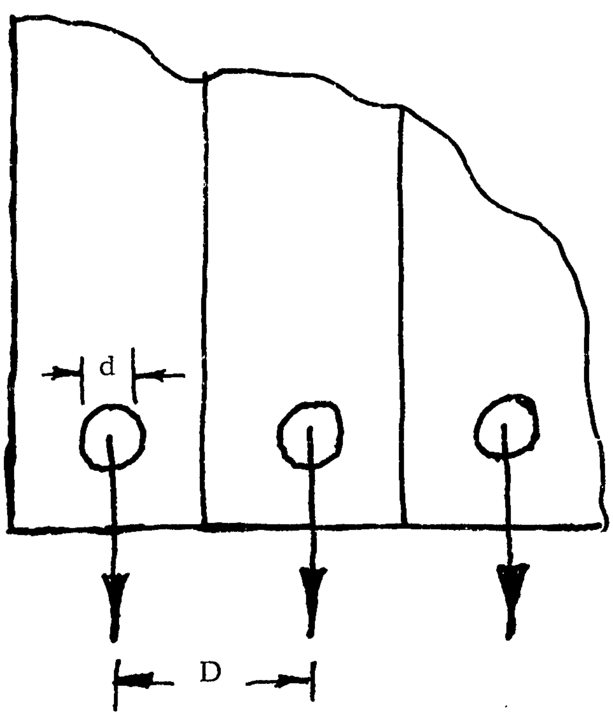
STAGGERED PATTERN
OK for leak prevention

730-755 O-64—3

Design practices used for optimum static strengths seldom apply to fatigue. Again, in fastener spacing, the best practice for static strength falls far short of good fatigue design. As has been shown, the stress at the edge of the first fastener hole is the most critical for fatigue. Accordingly, the best fastener pattern would be that which would lessen the stress here.



In terms of the average stress away from the first row of fasteners, the pattern can be likened to a series of lugs. This would indicate that the optimum spacing for fasteners normal to the direction of loading would be about 2.5 diameters. In terms of static strength, this would amount to a joint whose strength was 60 percent of the structural strength away from the joint (in area clear of holes). Test data indicate that a spacing of 2.5 diameters between rows is also helpful.



While it is unlikely that the spacing indicated will be used deliberately in practice, the idea of reducing the spacing might be helpful if the static strength margin were more than adequate and there were a need to increase fatigue strength.

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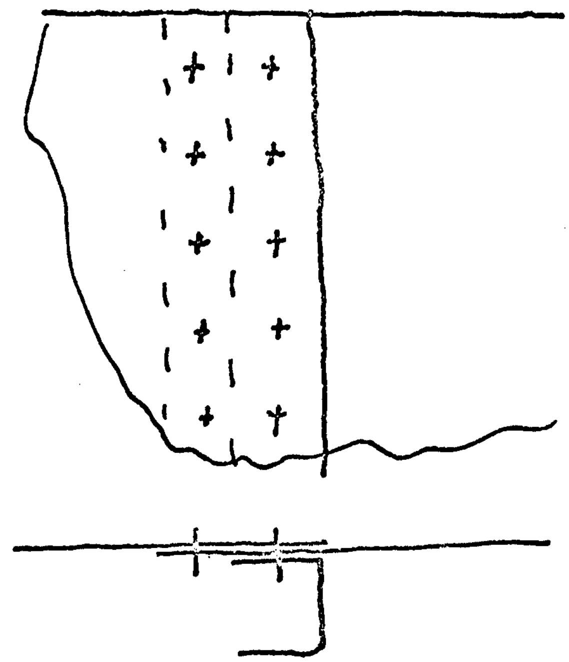
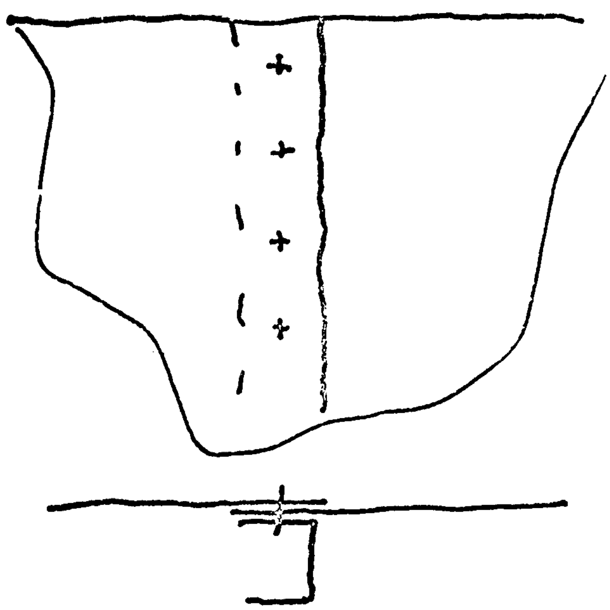
* See Appendix C for other stress concentration factors

Edge distance should always be considered along with fastener spacing. Except where the fastener is used through the edge of a doubler for clamping purposes (see Section 3.4.1), a minimum spacing of two-diameters from the center of the fastener to the edge of the doubler is recommended.

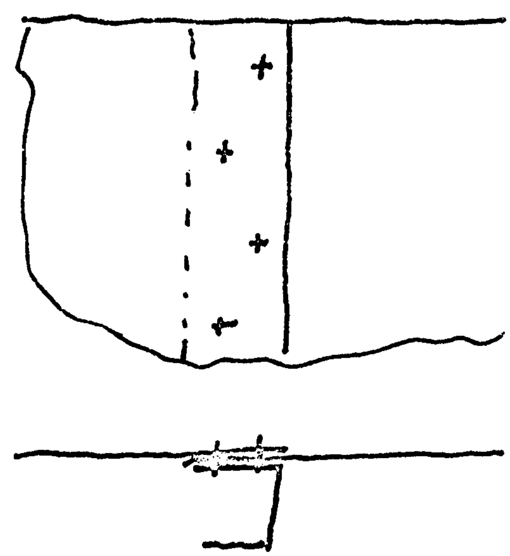
The number of rows of rivets is usually governed by static strength requirements and available space. For fatigue resistance, it is desirable to use two or more rows. Never one! This is particularly true where splices are made over rib flanges. Often, one row is used on the excuse that the flange is too narrow to use two. You can always find room for another row away from the flange, or you can stagger a row.



TWO OR MORE ROWS OF FASTENERS ARE A MUST



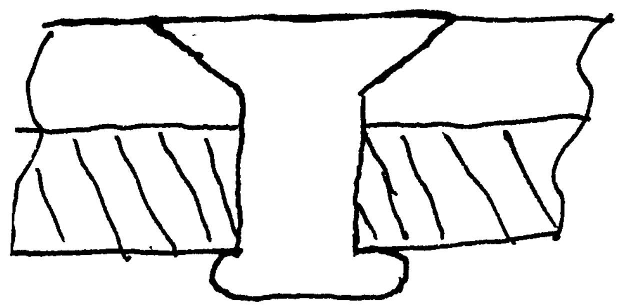
NEVER USE A SINGLE ROW OF FASTENERS



STAGGERED ROW

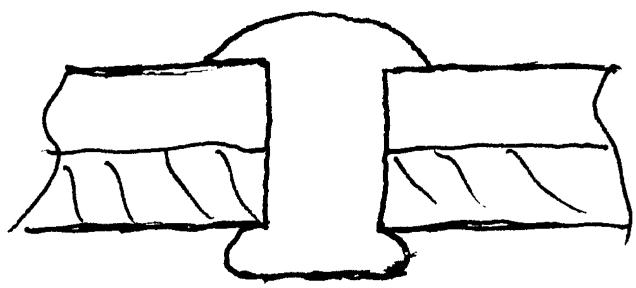
3.6 COUNTERSUNK FASTENERS

Countersunk fasteners are used for aerodynamic or hydrodynamic smoothness. Few other cases will justify the use of a countersunk fastener-- not even the doors in the galley. Where material thickness permits, holes for fastener heads are machined. These are called "machine countersunk fasteners." Where the countersink is provided for by dimpling a recess in the skin, the process is called "dimple countersinking."

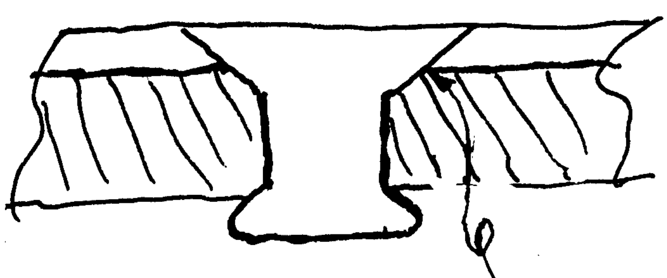


BETTER

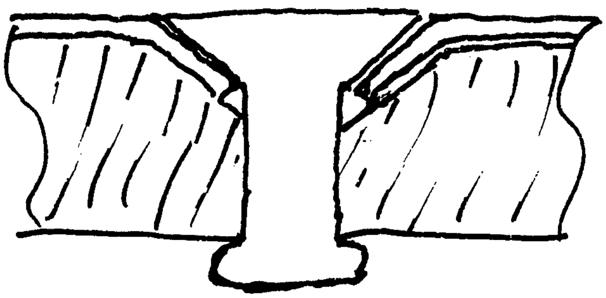
3.6.1 MACHINE COUNTERSINKING — Use of machine countersinking is not recommended where the material thickness is less than 1.5 times the depth of the head. The ragged edge caused by the countersink tool creates a superimposed stress concentration that invariably results in fatigue cracks. (See "Feathered Edges," Chapter 5)



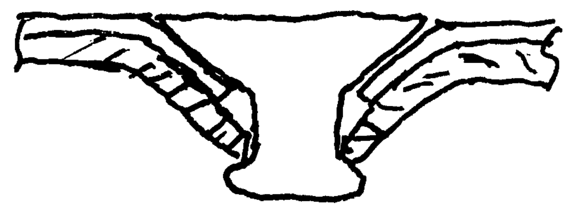
BEST



POOR



DIMPLE COUNTERSINK OVER HEAVY SUBSTRUCTURE



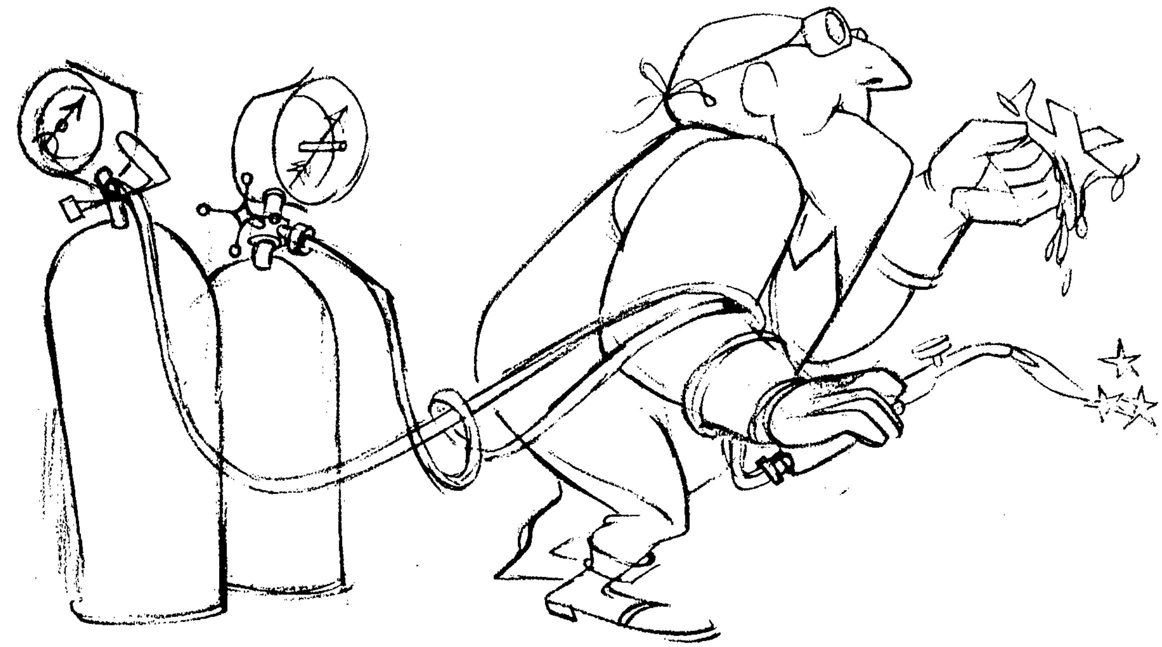
DIMPLE COUNTERSINK OVER LIGHT SUBSTRUCTURE

3.6.2 DIMPLE COUNTERSUNK FASTENERS -- are to be used wherever skin thickness does not permit machine countersinking. When assembled over heavy substructure, part of the substructure is machine countersunk to accept the protrusion caused by dimpling the skin.

When assembled over light substructure, the substructure is dimpled also.

While most dimpled joints have a superior fatigue life to those with machine countersink, extreme care is needed in forming dimples to prevent cracking around the periphery of the dimple. This is usually caused by too sharp a dimple radius or by not supplying the proper amount of heat on forming.

It is a good policy to use a more generous bend radius than the minimum specified in handbooks.



3.7 WELDING

Books have been written on welding and welding processes. No attempt will be made here to go into the many ramifications of welding: the advice of a specialist should be sought when designing welded joints. However, a few pertinent facts regarding effects of welding on fatigue are in order.

To begin with, it is not in the cards for a weld to develop fatigue strength equal to that of the parent material. Many vendors of special equipment and materials will make such claims; however, their claims are based on laboratory test specimens, rather than on full-scale structures. Welds in specimens don't have as many defects as those in the field. Defects such as voids, slag inclusions, and dirt are good places for fatigue to start.

Since fatigue failures always start at stress concentrations, welding should be done in a manner that affords the least concentration. Figure 3.1 shows a comparison of various weld configurations for static strength, fatigue strength, and cost. For purposes of comparison, a butt weld having complete penetration and a flush surface is considered as 100 percent (not necessarily 100 percent of the strength for parent material). Photoelastic models indicate state of stress at the welds.

Comparisons of welds in stainless steel are presented in Figure 3.2. It will be noted that the butt weld has very nearly the same fatigue strength as that of the base metal. As previously stated, caution should be exercised in using data from small specimen tests for estimating strength of fabricated structures. However, the data serve to illustrate possibilities of good welding.

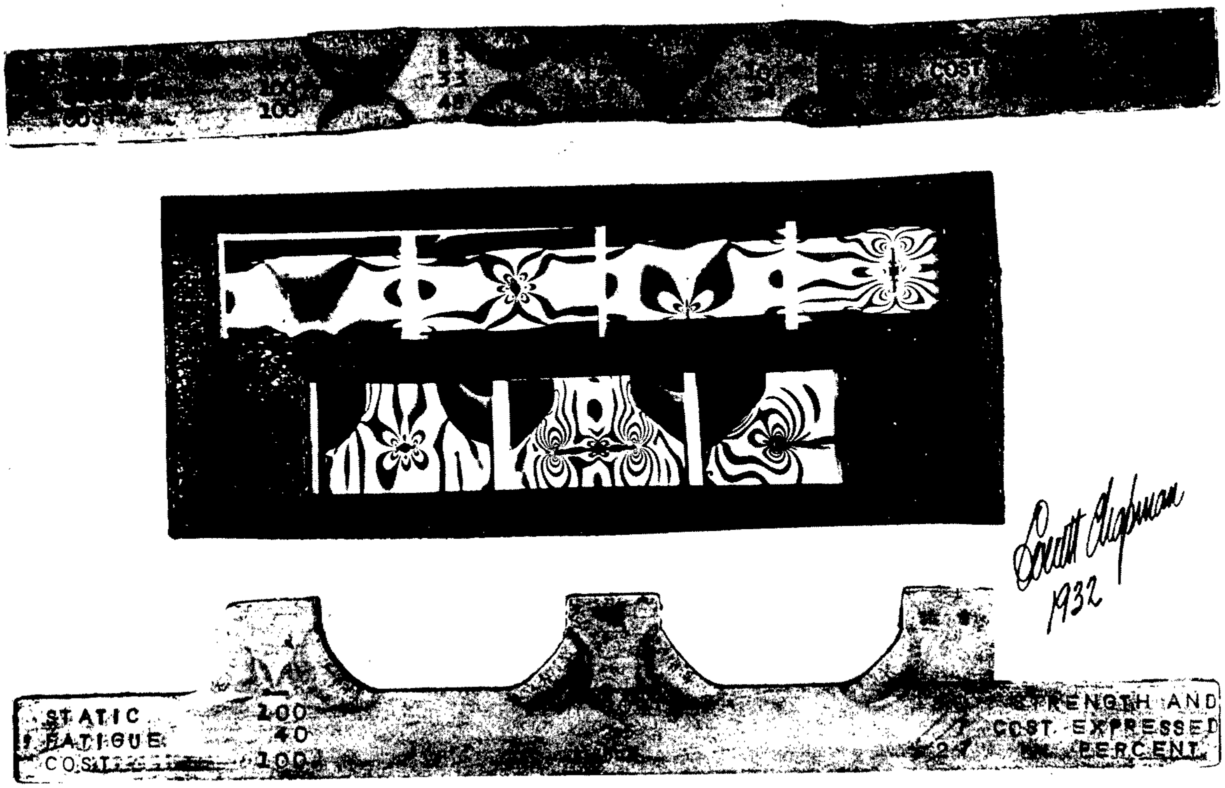


Figure 3.1 Welded Joints Compared for Static Strength, Fatigue Strength and Cost.

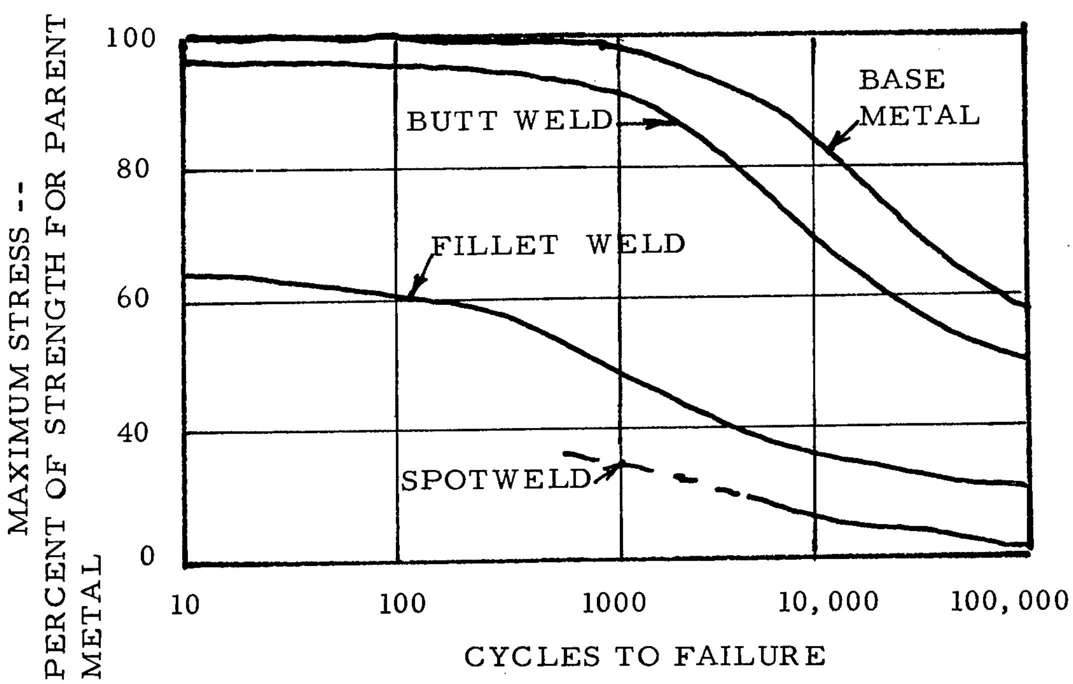
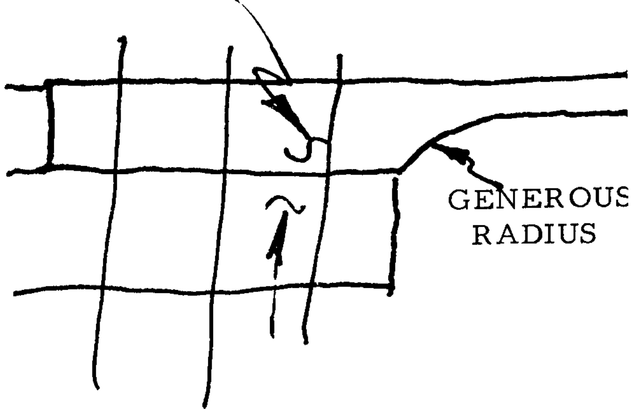


Figure 3.2 Fatigue Strength Comparisons of Welds in Stainless Steel



3.8 SCULPTURED STRUCTURE

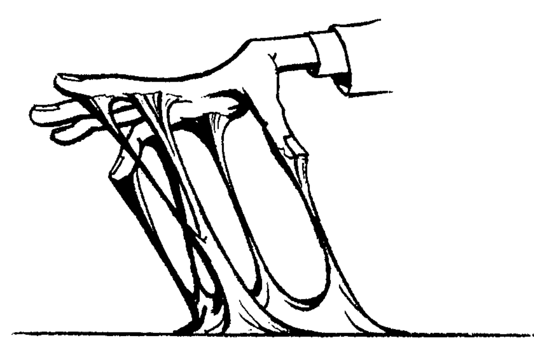
In keeping with the ideal that a perfect structure would be built of one piece, sculptured structures (machined from one piece of material) have come into use.

These sculptured parts are put together in erector-like fashion, extra thick material being provided at attachment points to negate the effect of holes. For static considerations, a 25 percent increase in thickness might be satisfactory, but for fatigue considerations, the amount of buildup (land) depends on the stress raiser. For the simple hole, a buildup of three times the nominal skin thickness might be satisfactory; then again, it might not. The reason is that the thick land builds up a misalignment at the center of the material and the resulting eccentricity accentuates the effect of concentrating the stress. At the time of the present writing, no good approximations are available for the right thickness of land material needed. In all cases, need for extremely generous fillet radii and good workmanship is paramount.

3.9 ADHESIVE BONDING

Finally, getting back to our cabinetmaker's technique, adhesive bonding for metals has become quite common. When used with cabinetmaker technique (scarfed very thin at the edges), bonded joints have superior fatigue strength. One reason it is not more widely used is that it is difficult to inspect for defects. Also, most adhesives deteriorate with time and environmental exposure. As a result, a combination of bonding and mechanical fastening is often used in primary structures, sort of like wearing belt and suspenders. In such cases, however, the bonding is used for fuel sealing, vibration damping, or other special purposes.

Most adhesives become brittle when cold and scorch when hot. Consult the specialist on this.



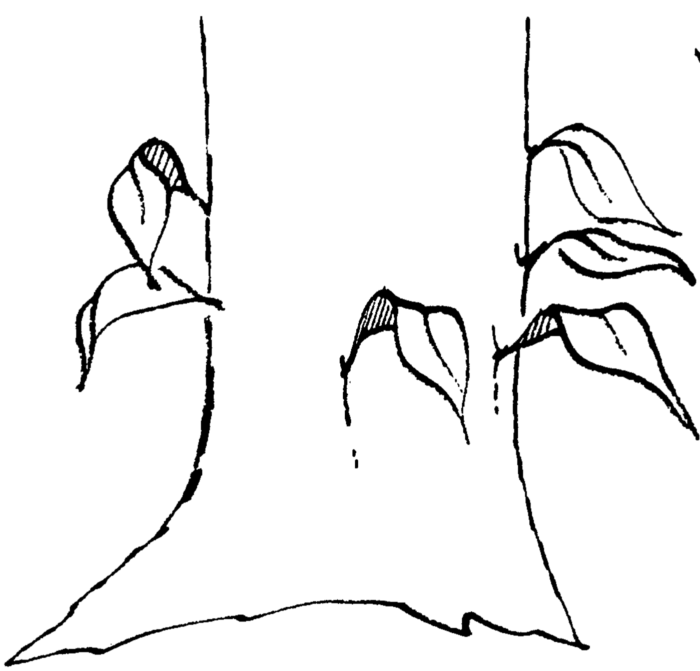
4 DEVELOPING AN INTUITION FOR FATIGUE

4.1 SEAT OF THE PANTS TECHNIQUE

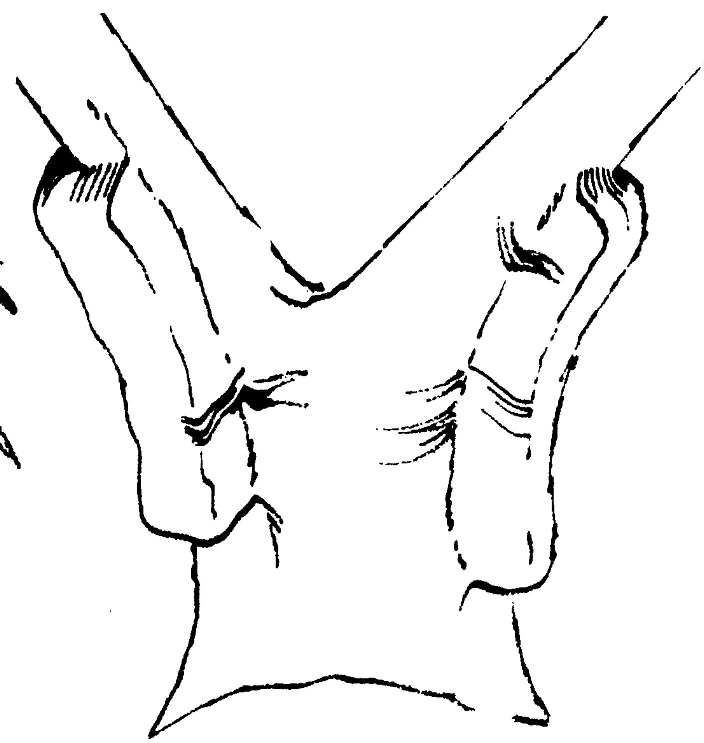
We have frequently heard the term "flying by the seat of the pants." This has more merit than some of the instrument specialists would care to admit. Certainly the number of G's to which the airplane is exposed would largely be a function of the comfort or distress experienced by the pilot. Yet the feel for the amount of trim required for special situations is largely a result of practice and intuition. Similarly, one can develop an intuition for fatigue resistance.

4.2 NATURE'S WAY OF DESIGNING

Boiler plate construction is man-conceived. You never see tree fronds connected directly to the huge trunk of a tree; nor do you see branches attached to the trunk with huge gobs of extra wood. Yet man, in splicing



NATURE DOESN'T DO THIS

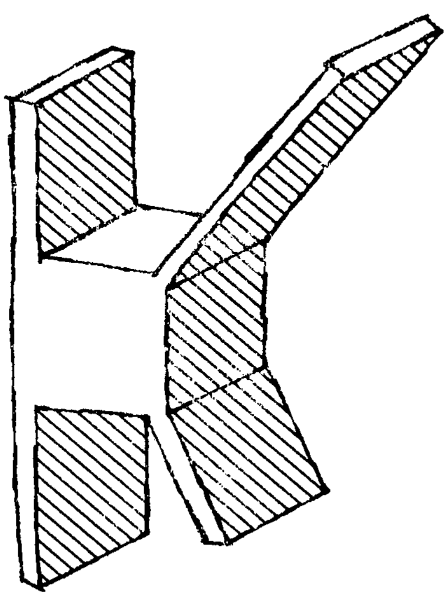


NOR THIS

two pieces of material, often uses a doubler the same thickness or thicker than the material being spliced. Consequently, most of the load is dumped on the first row of fasteners in the splice.

Intuitively, we should know that the second row of fasteners cannot begin to carry load without some "give" at the first row. Naturally, we don't want the "give" to occur in the structure that we are trying to protect. Intuition should tell us that the doubler material between the first two rows of fasteners should be thinned down so it will stretch without overloading the first row of fasteners. See Chapter 3 for more detail.

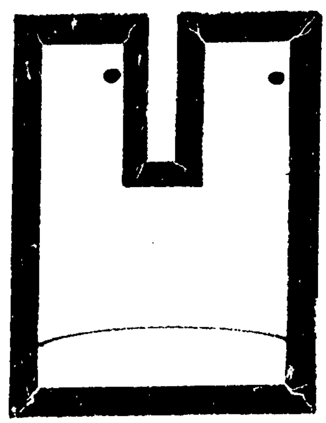
Similar to the heavy splice doubler, the huge gob of material that is machined away to almost nothing in spots should rub our intuition responses



the wrong way. Sometimes the small fingerlike extensions are made for attachment purposes. There is nothing wrong with this. What's wrong is that the fingerlike extension ends abruptly in a gob of material where stress cannot help but concentrate.

4.3 SHARP CORNERS

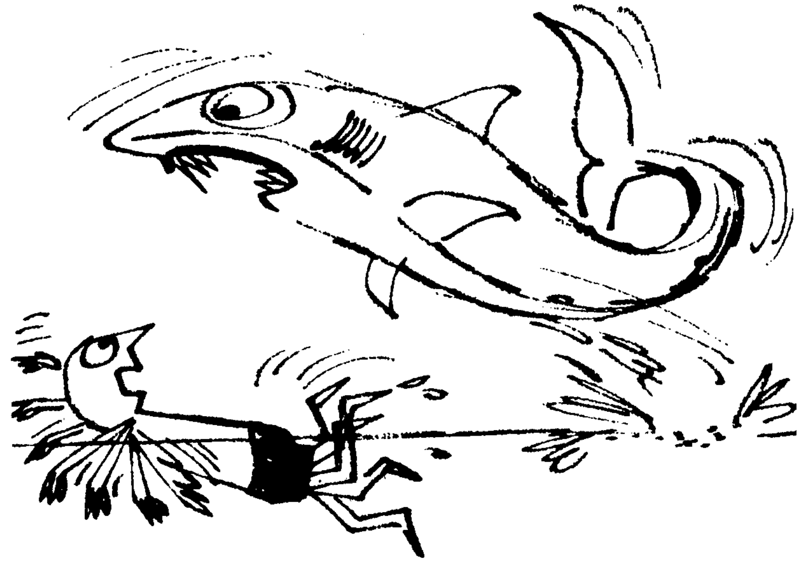
Sharp corners, either internal or external, have produced more fatigue failures in metallic structures than any other one common fault. There is nothing aesthetic

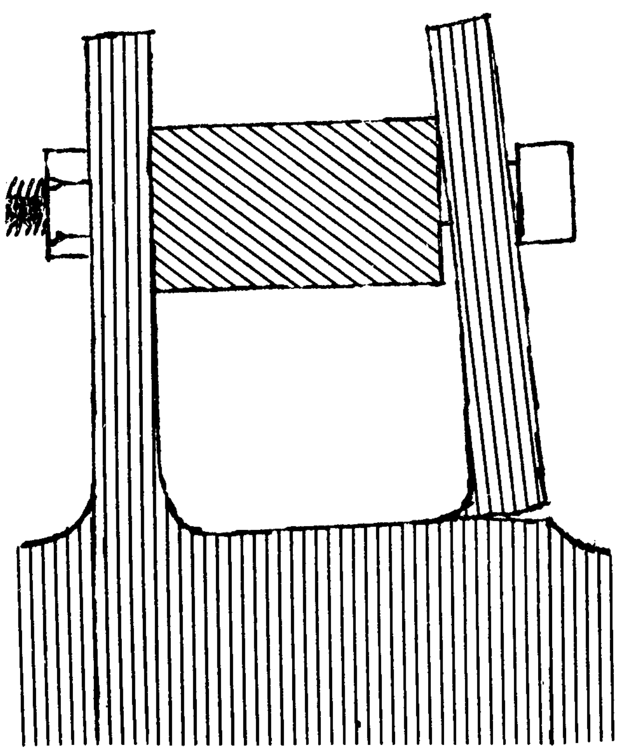


about a sharp corner. In fact, any workman who takes pride in his work takes special pains in smoothing off sharp edges and corners. Examples of what happens when sharp corners are left unfinished are given in the next chapter.

4.4 DON'T FORCE IT

A bigger hammer is rarely the answer. Intuition should tell us that when something has to be forced, everything is not well. An example of this is trying to fit a heavy lug into a clevis that is too small. This fault is usually obvious. Just as bad, but not so obvious, is the case where the clevis has been sprung by bolt tightening because the spacer bushing was too short.





4.5 ROUGH FINISH

Similar to the sharp corners is the rough finish. Like the cheap automobile paint job special, the rough finish reflects sloppy workmanship. Intuition should tell us that an excessively rough machine job is not right. Where our intuition may need further training is in telling us how much roughness is tolerable. While sawtooth finishes are never tolerable anywhere (see Chapter 5), we should sense that long straight machined surfaces would not have to be so smooth as fillet surfaces. A good rule of thumb is to use exceptional care at changes in section sometimes even to the point of hand finishing.

4.6 POWER OF KNOWING

Intuition has been defined as the gift of knowing without recourse to inference or reasoning, but it takes a lot of observation and logic to get that gift.

Previous chapters have presented some of the pure reason of what fatigue is and how to avoid it. The next chapter will present some examples of fatigue.

While it is hoped that the suggestions presented herein will prepare the reader to better cope with whatever situation arises, the reader will have to acquire his own experience.

As the father who tries to prepare his son for the pitfalls in life -- none seem to be like the ones described.

Experience with similar situations, however, should help. The important thing to remember is not the individual problem, but the logic used in its solution. Later, you'll get to the point that you automatically know what's right -- that's intuition.

5 | PAST EXPERIENCE

5.1 STRUCTURAL BLUNDERS AND HOW TO AVOID THEM

As the Parson said, "Everybody feels what's right, but don't always do right." The feeling for what's right can be stimulated by object lessons of what's wrong.

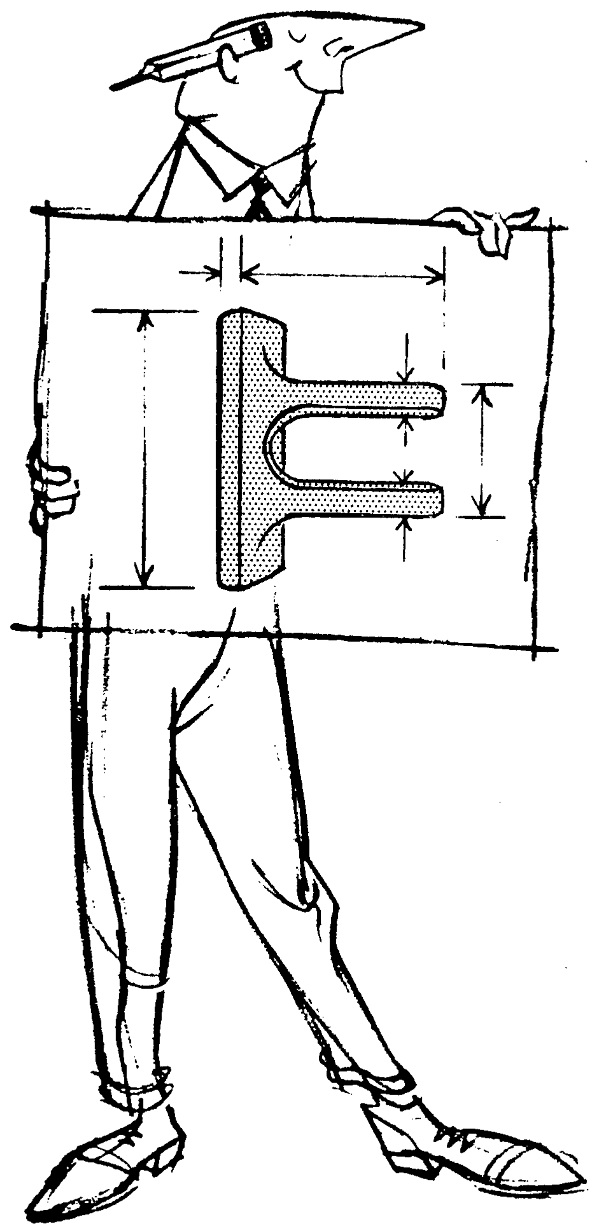
That's why the Parson preached fire and brimstone on Sunday -- and that's why this chapter is devoted to those wrongs that are known by such terms as blunders, butches, botches, boo-boos, goofs, mistakes, and what have you.

When we see the consequences of wrong, we are driven to do what's right. Take blunders, for example. There are design blunders, shop blunders, and that horrible group of atrocities that are conceived as design blunders and perpetuated as shop blunders. The designer who says "I don't need to specify details, the shop man can take care of that" would be horrified to hear the shop man say "If the fool designed it that way, I'll build it that way.'

TIPS ON FATIGUE

The photographs of fatigue failures collected in this chapter have been furnished by various manufacturers and airplane operators who hope that seeing will make believers of designers, inspectors, and shop men.

Examples of design butches start off with notches in various forms, including designs with no radii, short fillet radii, and sharp bend radii. Special cases of notches include square holes and feathered edges. Feathered edges may more logically be blamed on the shop, since most standard shop procedures include rounding off all corners. Nevertheless, where sharp edges are anticipated, special pains should be taken to see that the drawing clearly spells out rounding off all corners. This is especially critical where careless handling might result in nicking sharp exposed edges. Poor load distribution is another design botch, and closely related to this is the case of superimposed stress concentration in its various forms.



TIPS ON FATIGUE

Shop boo-boos involve poor craftmanship, failure to recognize design goofs, and failure to alert engineering when a design or shop mistake is discovered. Especially critical are dings resulting from careless handling practices. While short rivet or bolt edge distances can usually be blamed on the shop, frequently engineering drawings do not make allowance for sufficient space to install the number of rivets or bolts required.

No matter who was at fault, a good craftsman would never leave insufficient edge distance or mismatched machined surfaces, even if the blueprint called for them. Even a poor craftsman would blush if he were shown the failure that resulted because he had installed a bolt of the wrong length.

5.2 NOTCHES

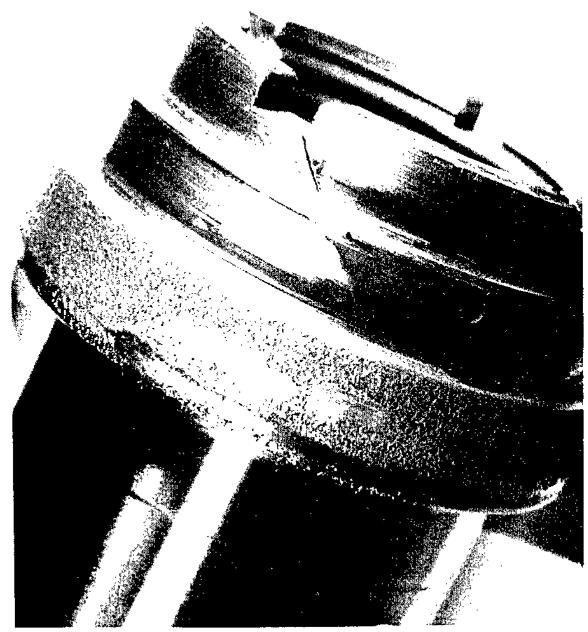
5.2.1 SHARP NOTCH — The very existence of the sharp notch is a result of stupidity, carelessness, and unwise penny-pinching.

The manufacturer may aim at saving a penny, but the user will always pay heavily, and often the manufacturer will pay later in the form of warranties. In any event, machine time saved is peanuts when compared with overall structural integrity costs — and, in many cases, using machine cutters with proper edge radii will avoid the sharp notches at no extra cost.

Thousands upon thousands of fatigue failures similar to those shown on the opposite page could have been avoided by good design and craftsmanship.

If every shop foreman impressed every tool salesman with the need, machine toolmakers would soon design cutters so that such crimes would be impossible to commit. The sharp radius, as well as the means for making it, should be outlawed the way they outlawed gun slingers. Tools for making sharp radii should be kept under lock and key for use on nonstructural parts, such as ash trays.

Similar failures can also occur in well filleted parts where the fillet has tool marks or scratches.



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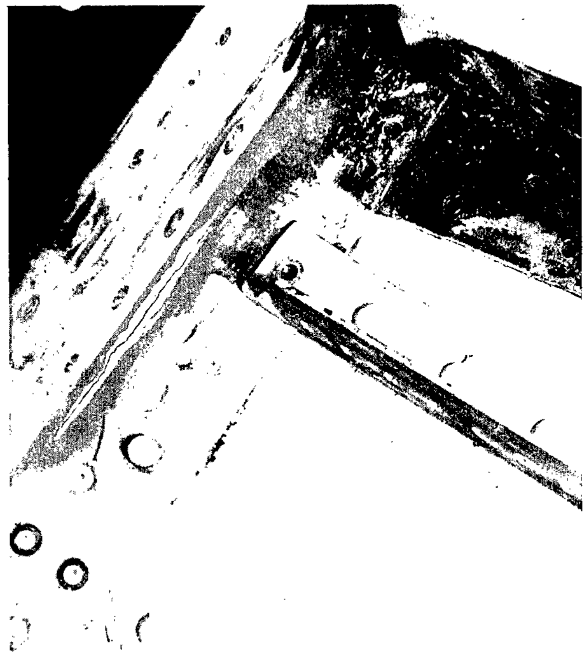
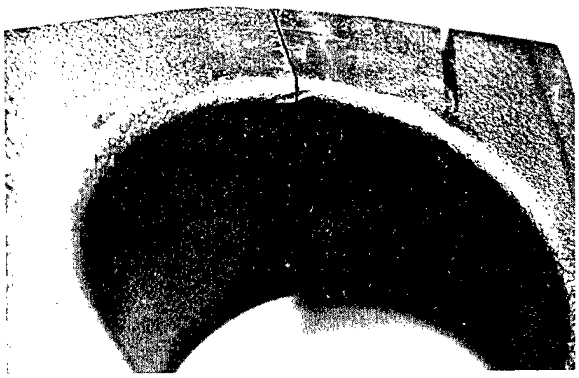
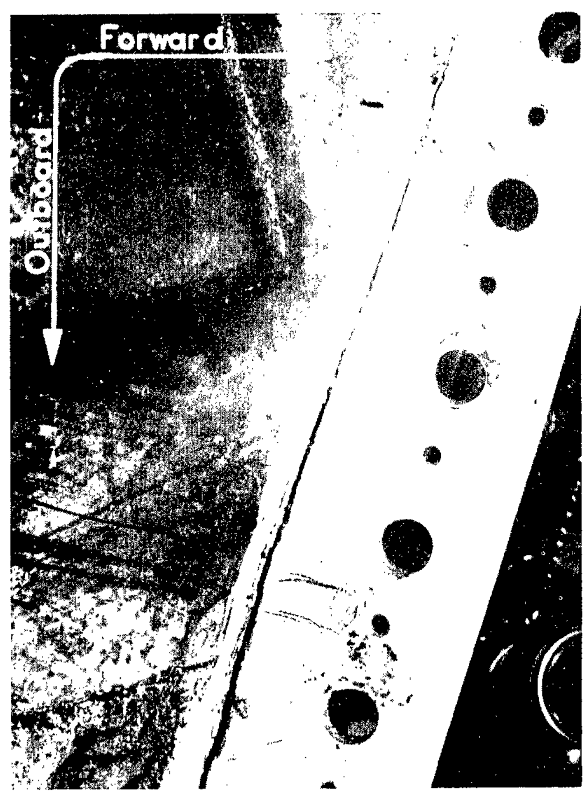
5.2.2 DINGS -- Dings can be intentional or accidental. Upper left shows the fatigue failure resulting from an accidental blow by a blunt object. Lower left shows failure resulting from identification marks, and upper right shows how marks left by a rivet bucking bar cause a wing spar to crack in the radius.

Failure at lower right is similar to upper right, except the tool was an impact screwdriver used for removing the wing fuel cell cover attachment screws.

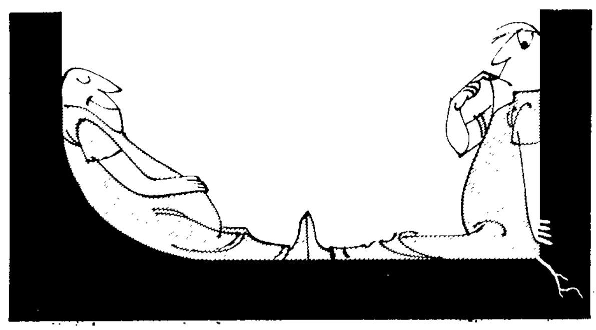
Precautions should be taken in handling parts to avoid denting and nicking. Should dings such as shown be discovered, they should be called to the attention of the foreman and fatigue specialist. These can frequently be corrected.






Forward
Outboard

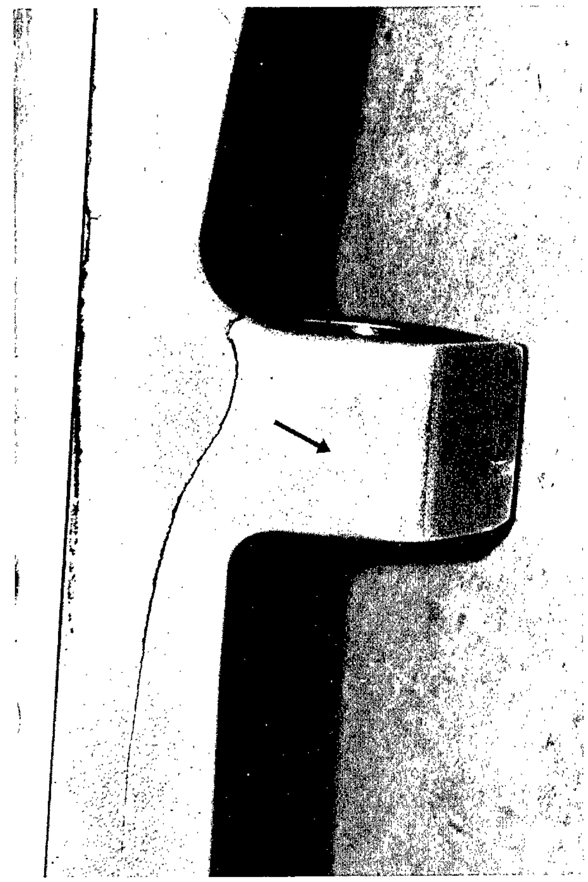
5.2.3 FILLET RADIUS — Think big when it comes to fillet radii. Radius of the lug in the upper left photo would seem ample, but it only had a fifth of its expected life. A similar lug in the upper right photo has a radius that was increased by removing some of the material. This lug with the larger radius had a fatigue life eight times greater.



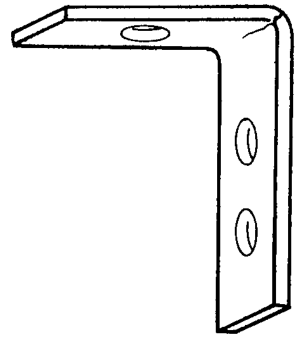
The lower photo shows fatigue failure through a fillet that would have seemed to be OK, but the fact that the part failed in service indicates a larger radius should have been used, especially since there was no problem of clearance in this area.

A good policy is to use as large a radius as space and practicality permit, especially at changes of section thicknesses.

Original Life = N Cycles



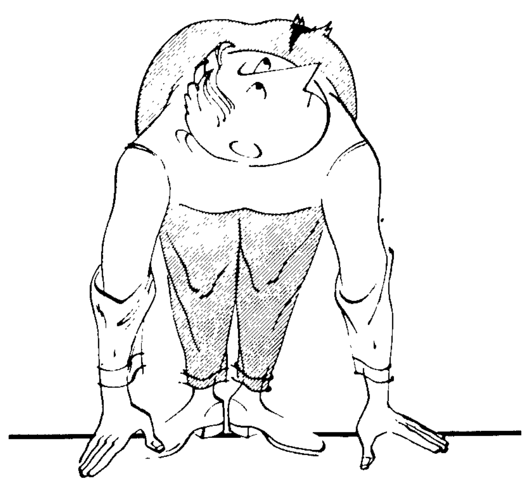
Redesigned Life = 8 N Cycles

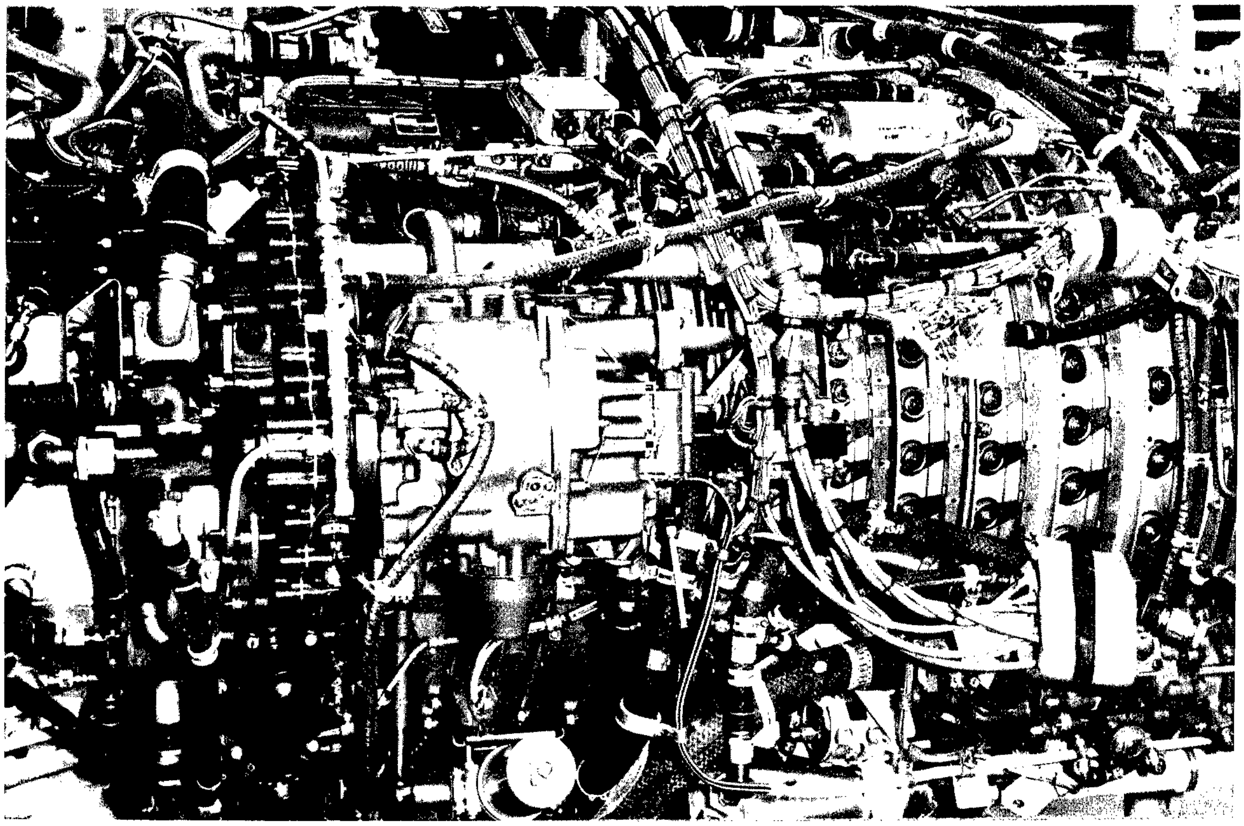


5.2.4 SHARP BENDS OR NUISANCE FAILURES -- The clip in the sketch costs about 10 cents, but, buried in the maze shown in the photograph, it took 32 man-hours to replace. While failures of clips such as this may never constitute more than a nuisance, the replacement cost is appalling.

The problem here is that someone thought the sharp bend in the clip radius looked cleaner and was within the allowable bend radius for the material. Unfortunately, specifications for bend radii were based on the ability to form without cracking at the time of bending. Little thought, if any, was given to fatigue performance.

Use a bend radius that is as generous as possible without destroying functional performance. In some cases, using a stainless steel clip that is one gage thinner than the original aluminum clip would be in order. Consult the specialist on this.

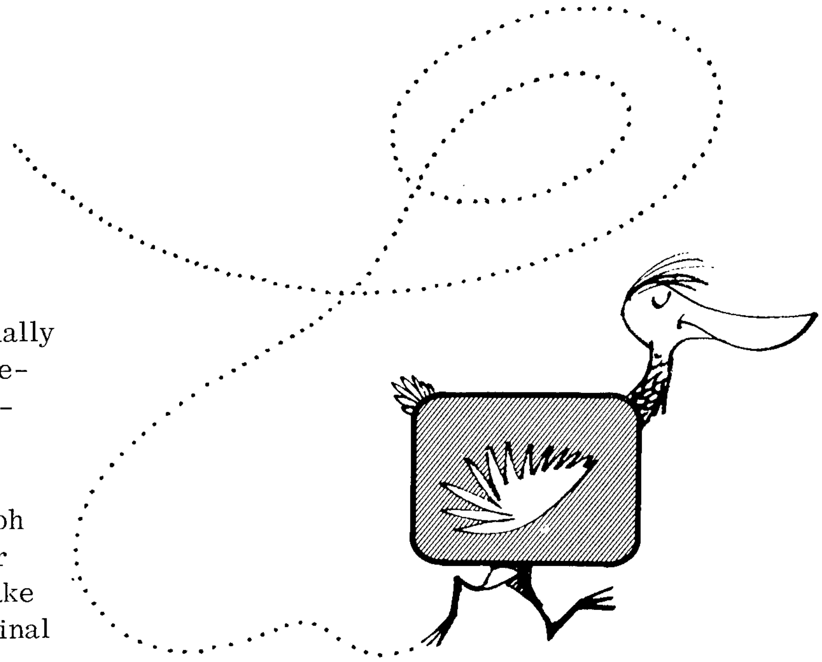




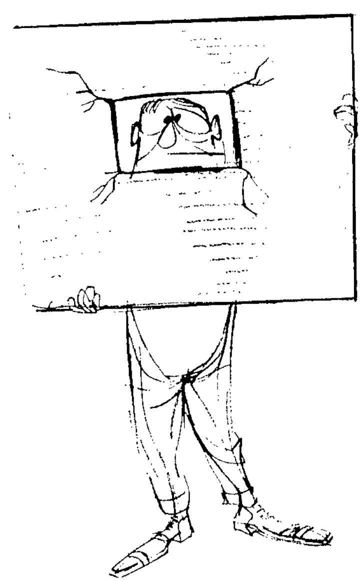
5.2.5 SQUARE HOLES—There is usually a purpose in making square holes. Sometimes holes are made square for no purpose at all.

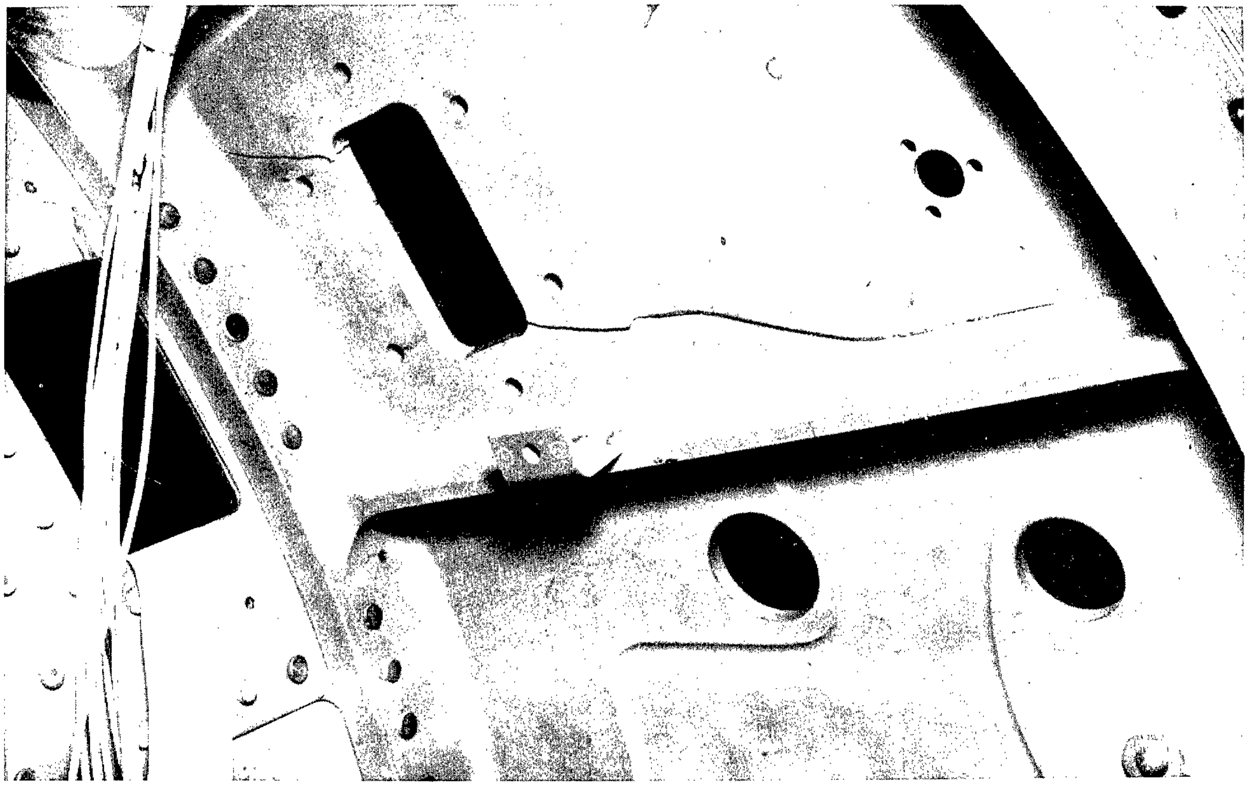
The hole in the upper photograph was made to accommodate a rectangular duct. Solution to the problem was to make an elliptical hole encompassing the original rectangular hole. Fatigue life was five times that with the rectangular hole.

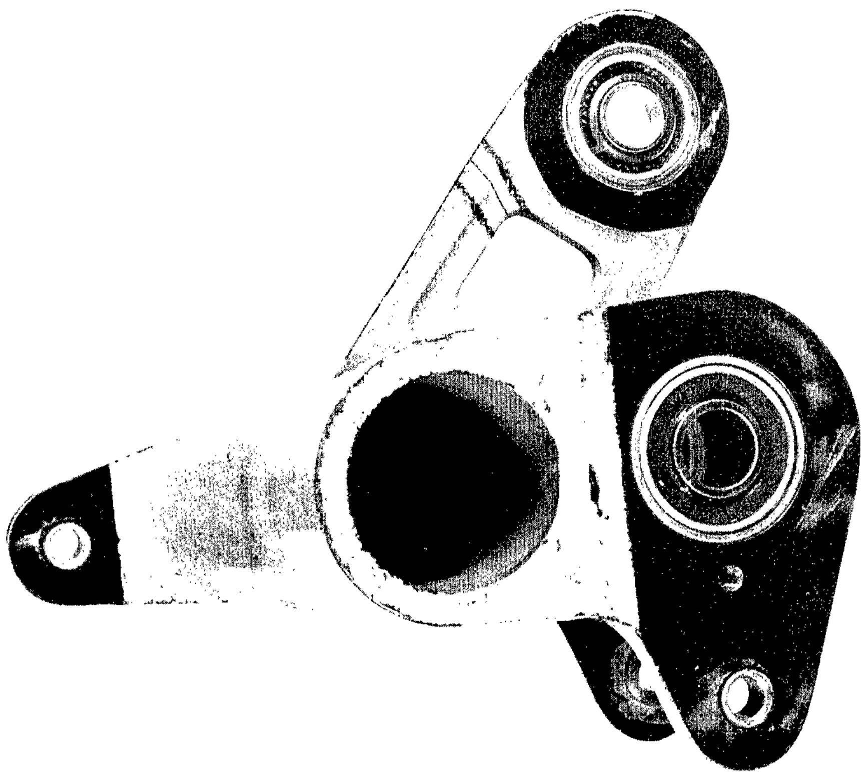
The hole shown in the lower photograph was made for no purpose other than to save less than two ounces of weight. It could have been round, better yet, forgotten altogether. The part failed despite the seemingly generous corner radii.



"Rectangular Duct"





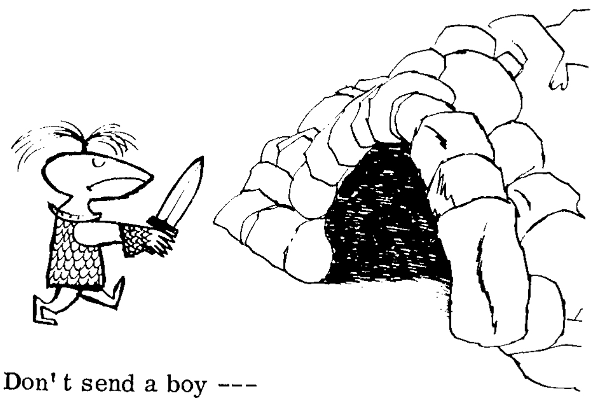


5.3 FEATHERED EDGE AND SHARP CORNERS

No one in his right mind plans to have sharp corners or feathered edges. They usually occur because the designer didn't pay attention to what would happen when his two- or three-drawing views were integrated. He would be surprised to find raw edges of the kind that caused service failure in these examples.

Besides being a stress raiser in itself, the feathered edge or sharp corner is easily nicked. Thus, a structure employing similar parts may fail most frequently at the nicks on feathered edges, and someone might wrongly infer that feathered edges were all right -- it was only the nicks that were wrong.

However, if there were no feathered edges, there would probably be fewer nicks. A nick on top of a feathered edge constitutes a superimposed stress concentration. While every concentration doesn't cause failure, why have concentrations that could be so easily avoided. The time could better be spent reducing the effect of concentrations that can't be avoided.



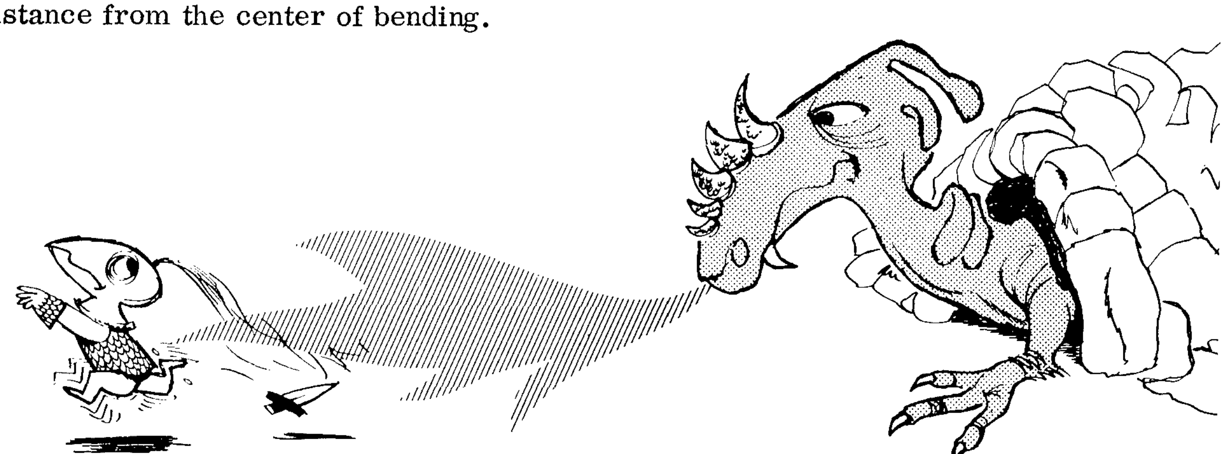
Don't send a boy ---

In the example shown, the thin flange added little to the strength of the forging. This causes the stress at the outermost flange fibers to be higher than it would have been in the plate surface without the flange. Whereas thin flanges attached to heavy main structures can sometimes be condoned on the basis of increased rigidity, this is done at the expense of increased stress and should not be attempted without the approval of the specialist.

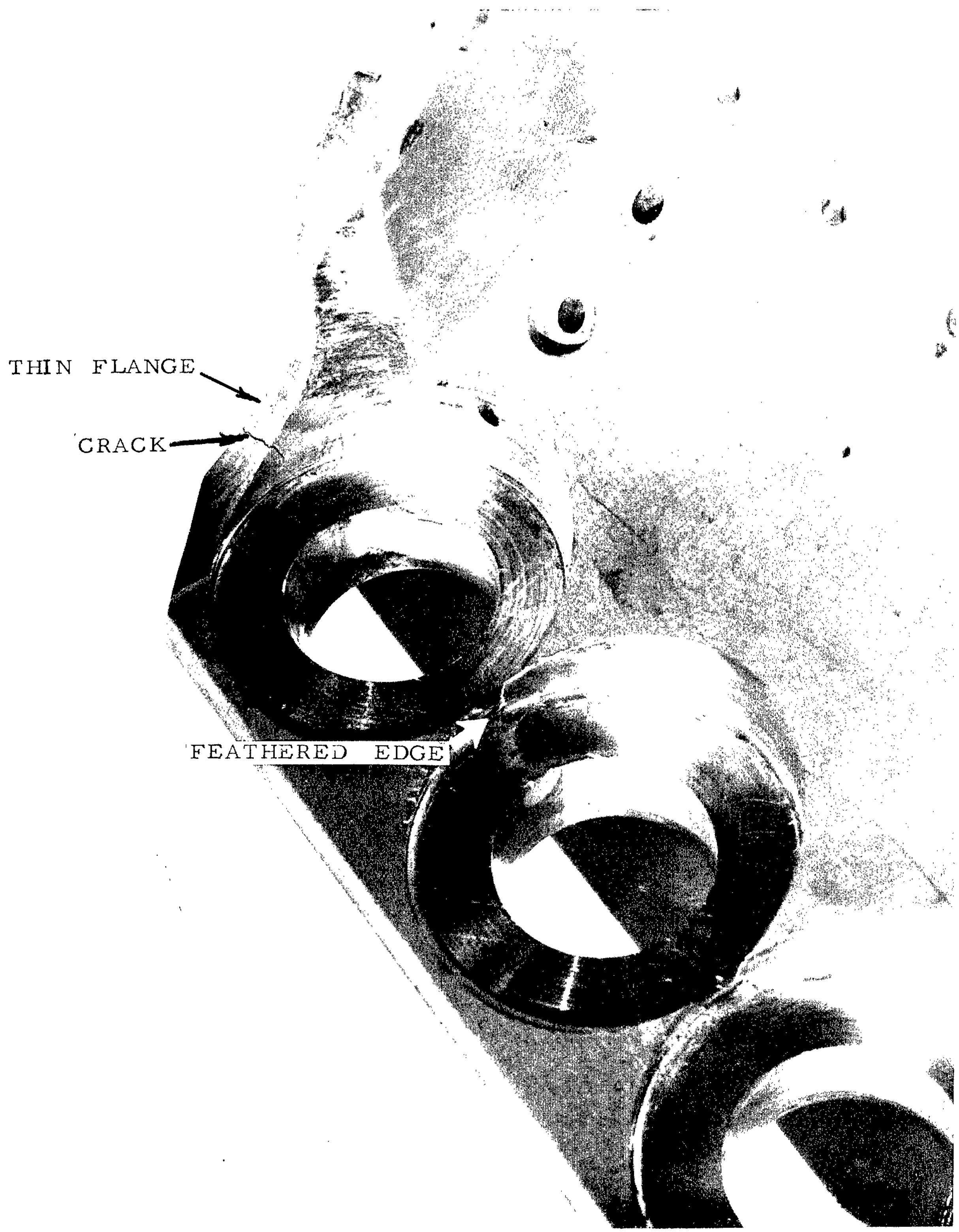
5.3.1 THIN FLANGES — This part has faults shown in the preceding example because the spotfaces resulted in feathered edges. These feathered edges might ultimately have caused a fatigue failure, but an additional fault—that of using thin flanges to carry load in bending—caused the failure even sooner.

The flange on an I-beam is an example of supplying material where needed. Just as important is the removal of material where not needed. Thus, removing some of the material provides a generous fillet, (as in Section 5.2.3) and improves fatigue life. Another example of the need for material removal will be shown in Section 5.4

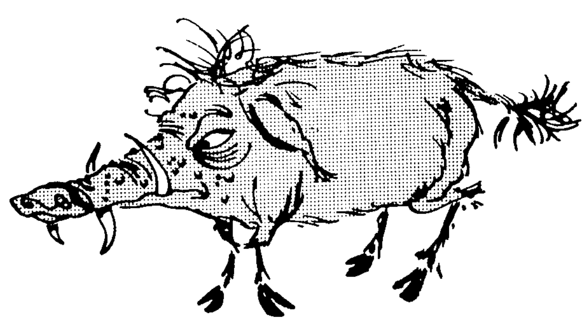
This does not mean that it is always undesirable to have areas a long distance from the center of bending. The opposite is true. Take the case of the I-beam. The flange is remote from the center of bending; but, there is enough beef to more than compensate for its distance from the center of bending.

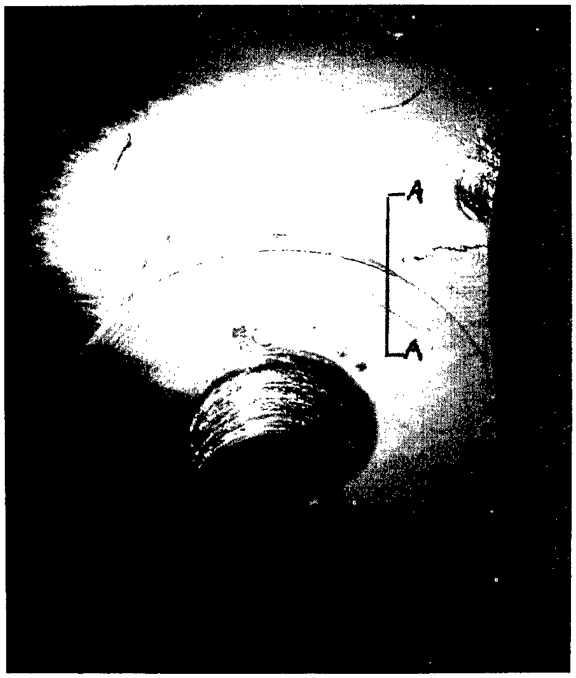


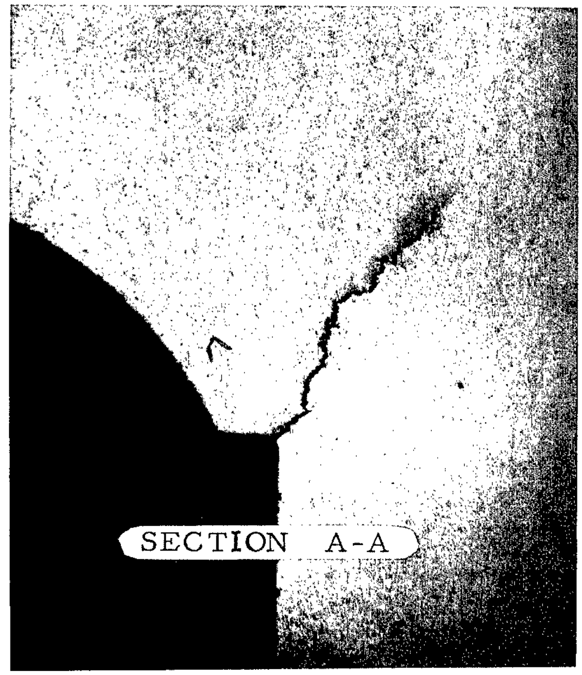
--- to do a man's job!

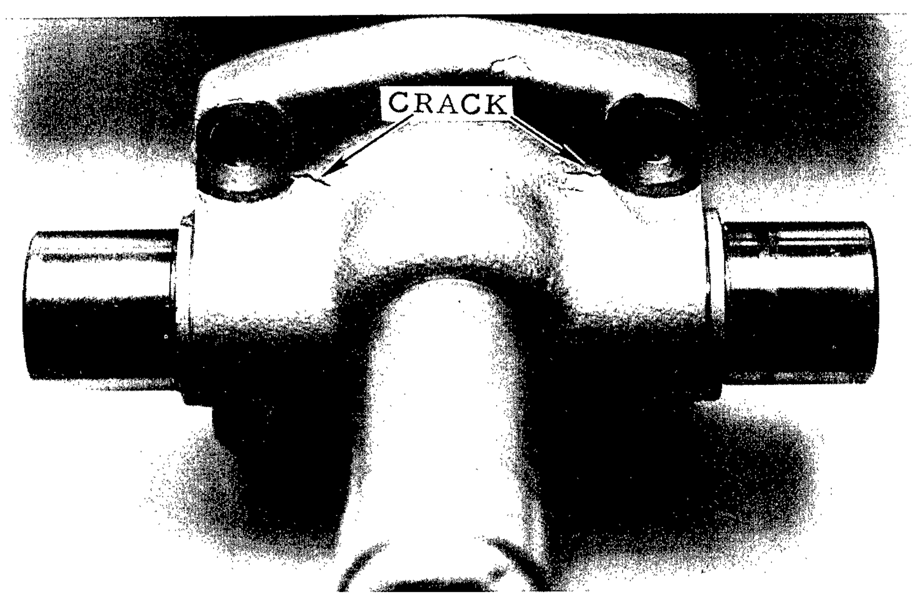

THIN FLANGE
CRACK
FEATHERED EDGE

5.3.2 TROUBLESOME SPOTFACE — Spotfacing is a common cause of fatigue failure. While not quite as bad as the example shown on the previous page — in that no vulnerable outstanding legs were left — the sharp corner combined with a feathered edge is a sure cause of trouble.



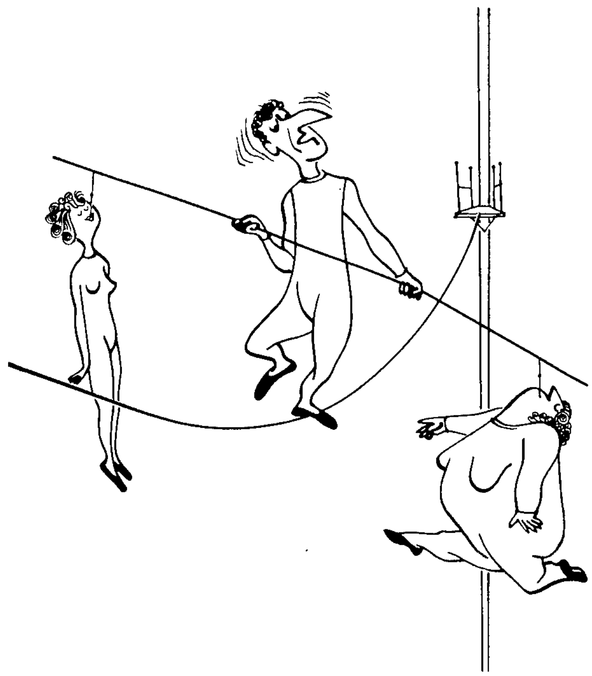

A
A


SECTION A-A


CRACK

5.4 BAD LOAD DISTRIBUTION

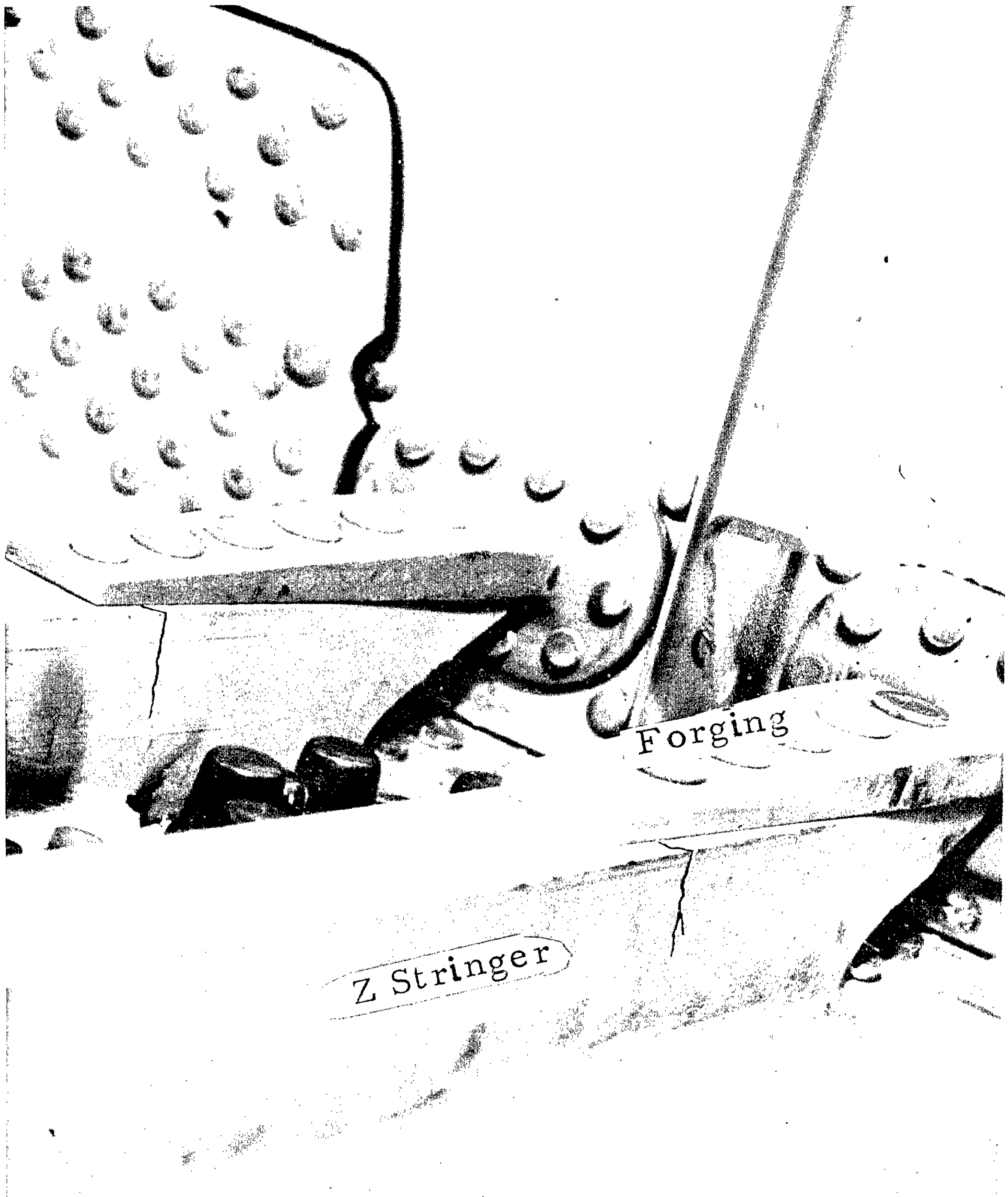
5.4.1 STRINGER FORGING—These stringers failed because there was too much stress at the end fasteners. Since the stress here is caused by a number of items, including axial loading, bending, and fretting, a reduction in any or all would improve life.



An attempt was made to reduce the stress here by using smaller rivets. As you see—it didn't help. While using smaller rivets at the ends of splices is common practice for static strength, their use for fatigue situations has never proven helpful.

One solution is thinning the forging (see Section 3.4.2) to relieve the bending effects of the single shear attachment. This also prevents overloading the first fasteners and at the same time reduces fretting by equalizing the stretch in stringer and forging.

Another solution is to replace the end rivets with interference fit fasteners (see Section 6.3). This is especially helpful where an easy fix is wanted to bring existing structures up to required life. Replacing end rivets with somewhat larger diameter tapered bolts will ensure removing fatigue damaged material (provided no cracks remain visible) besides providing a better-than-new fatigue strength.



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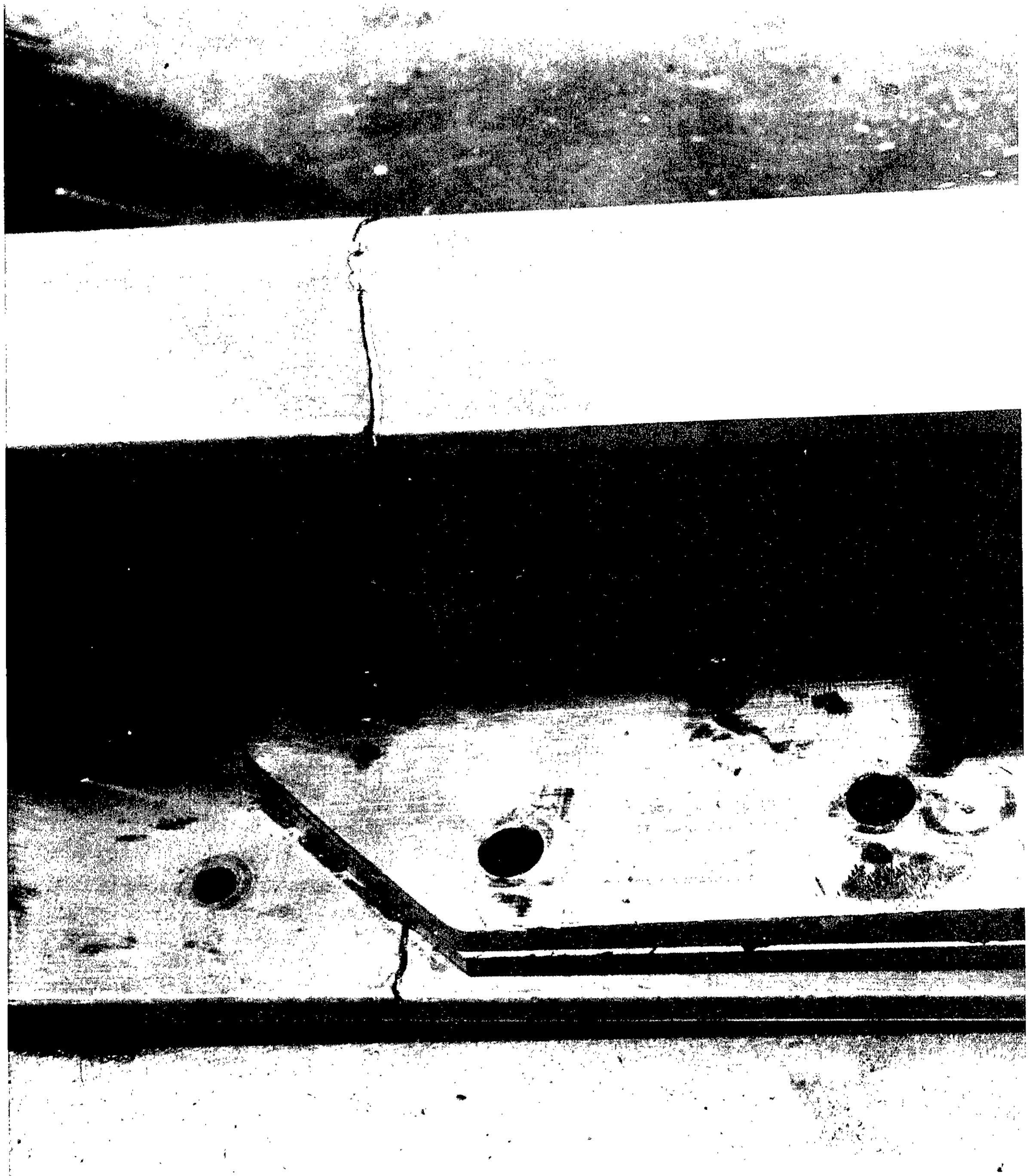
5.4.2 SPLICE — This splice contains the same problem as the dagger fitting on the previous page. Being more complicated, the problem is not so easily diagnosed. Thus, the usually proposed solution is to increase the thickness of the material being spliced. While this may solve the fatigue problem, the extra weight penalty may be extreme.

Solution for this problem is the same as for the stringer attachment forging, except that the loads on all first line rivets must be relieved. A common practice is to cut finger-like extensions to the splice doubler as will be shown on the next page. This, however, is not so effective as tapering the doubler or, better yet, cutting the fingers and also tapering the doubler.

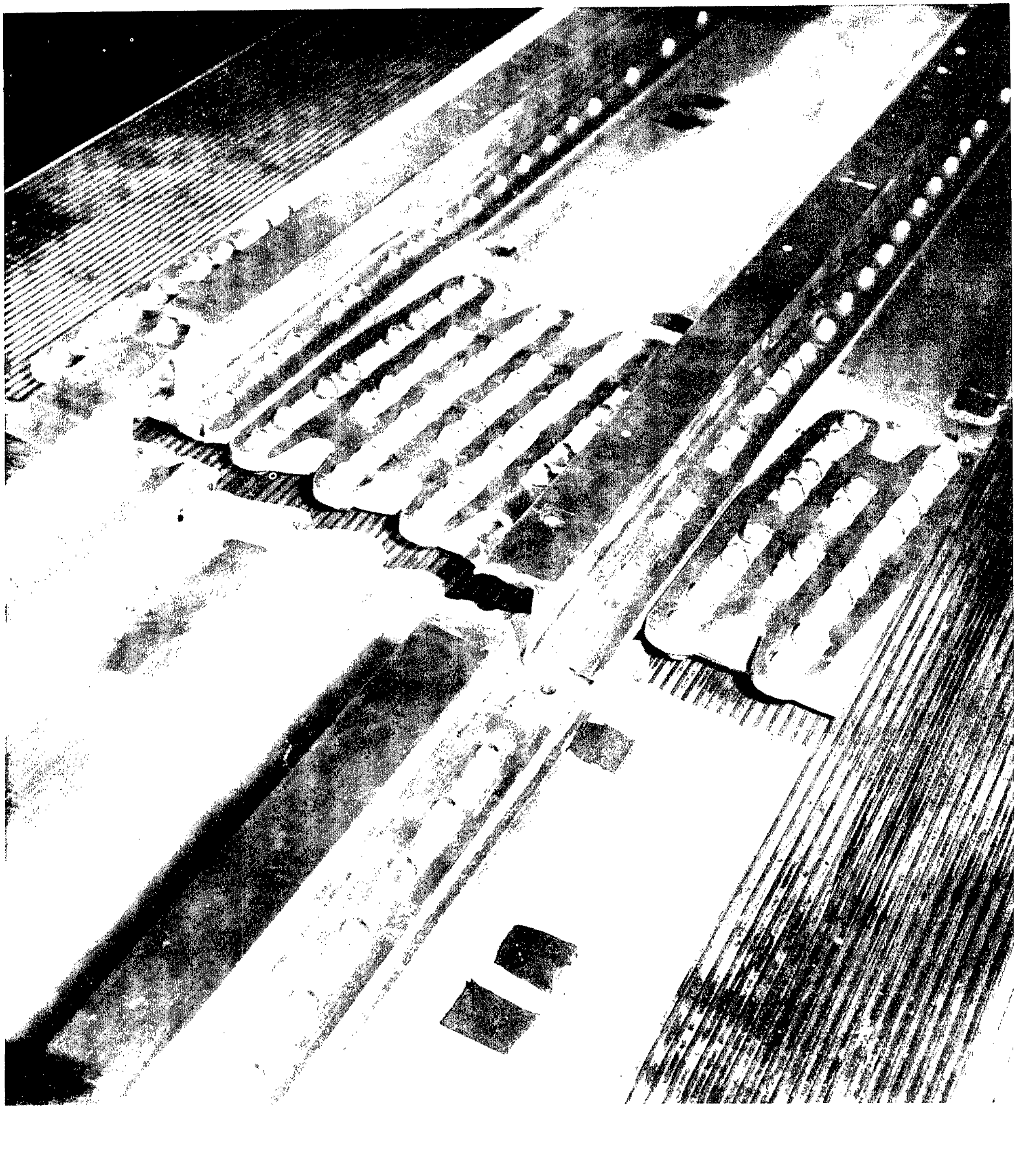


5.4.3 THE CONTINUOUS MEMBER – It would be fine to have all continuous members with no intervening splices. This would end many of our fatigue problems. Since we do have splices, the idea of making only a partial splice at a given location leads to trouble.

This example would appear at first to have been caused by using splice plates that were too thick*, as in the previous example. The primary failure, however, was not at the end rivet as the photograph would indicate. This failure happened after the continuous stringer broke directly over the splice area, dumping its share of the load on the skin.

What happened was that there was too much give in the riveted joint, so that the skin was unable to carry its fair share of the load, causing the stringer to fail. The skin failure was secondary.

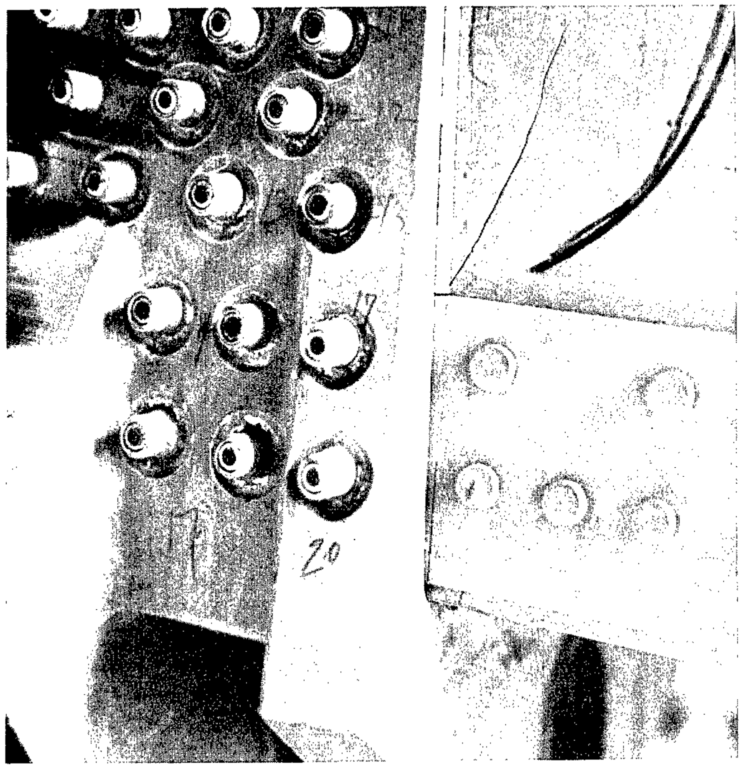
* They really were too thick at the end rivets. This might ultimately have caused the failure as shown had not the initial failure been in the stringer.



5.4.4 HEAVY FORGING -- These are typical situations where heavy pieces are attached to relatively thin under structure, resulting in the failure of the structure.

Wherever situations like these arise, the solution is to feed out the stress through successive layers of thinner material or to provide gradual taper to the first point of attachment.

If this is not possible, perhaps the heavy piece can be split normal to the direction of stress so the under structure can have "Breathing Room." To lower the stress fluctuation resulting from the hole itself, taper bolts at points of attachment will help.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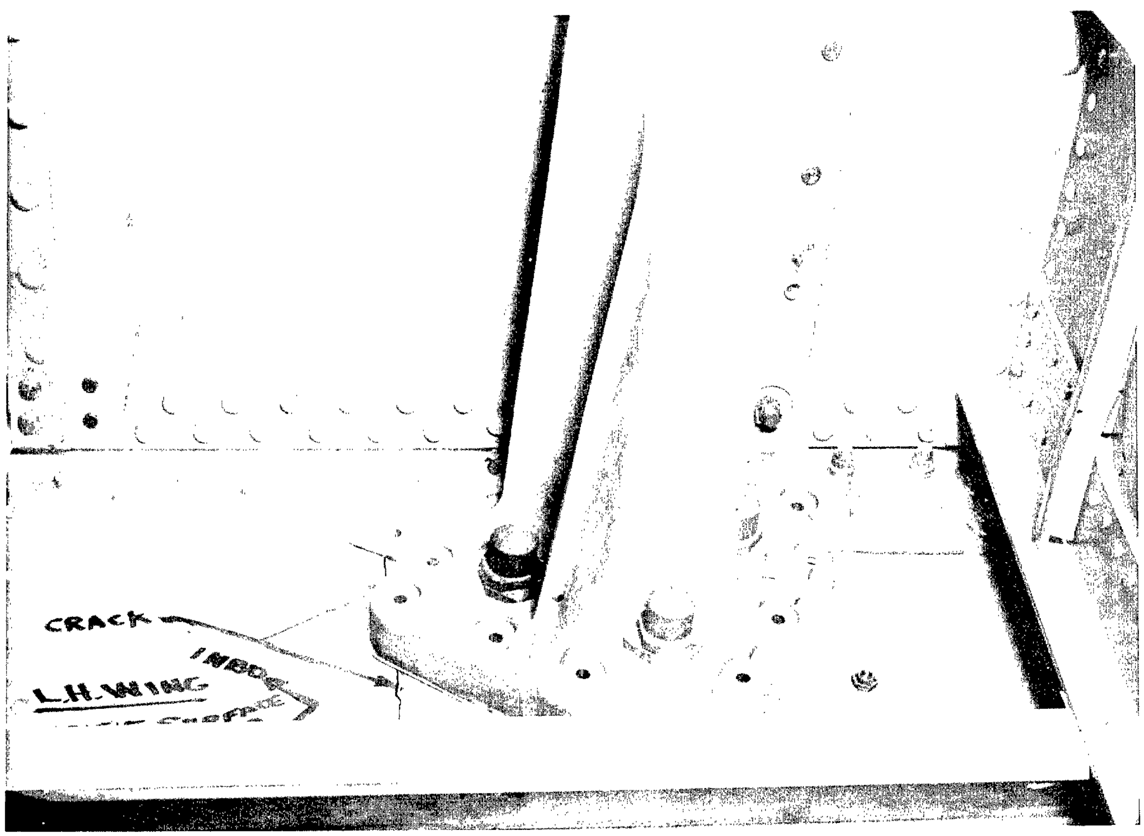
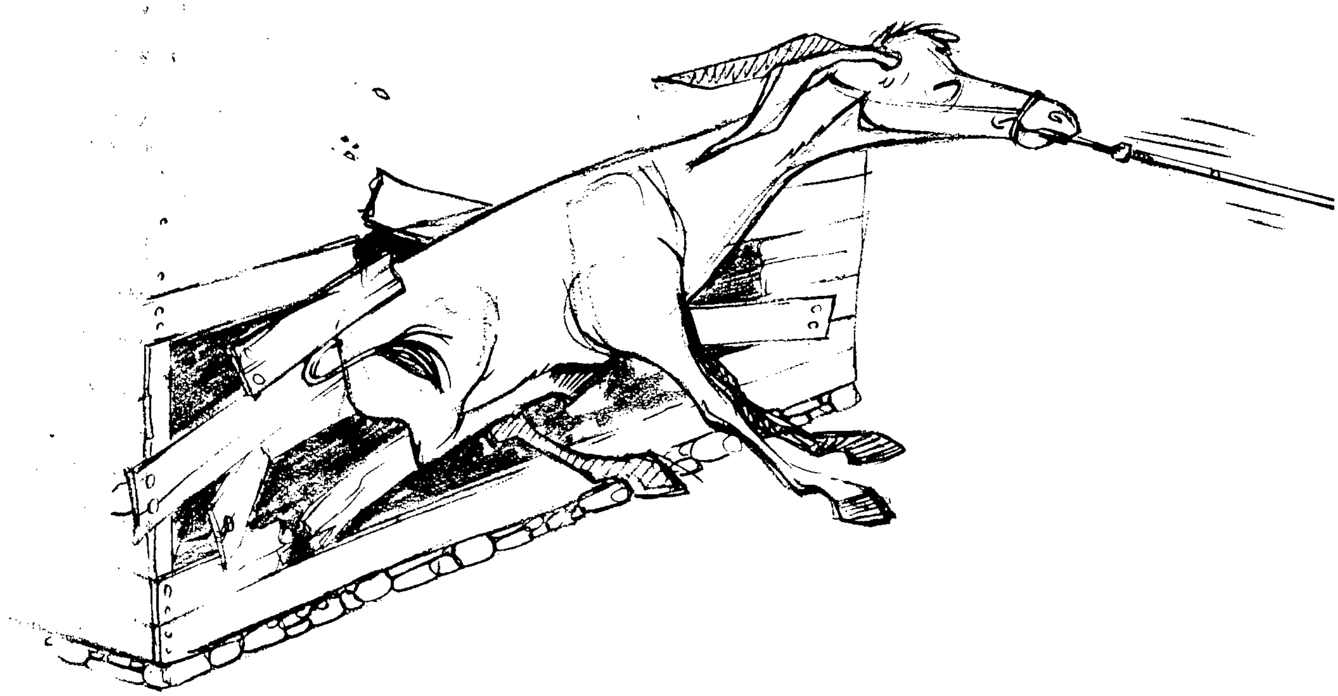
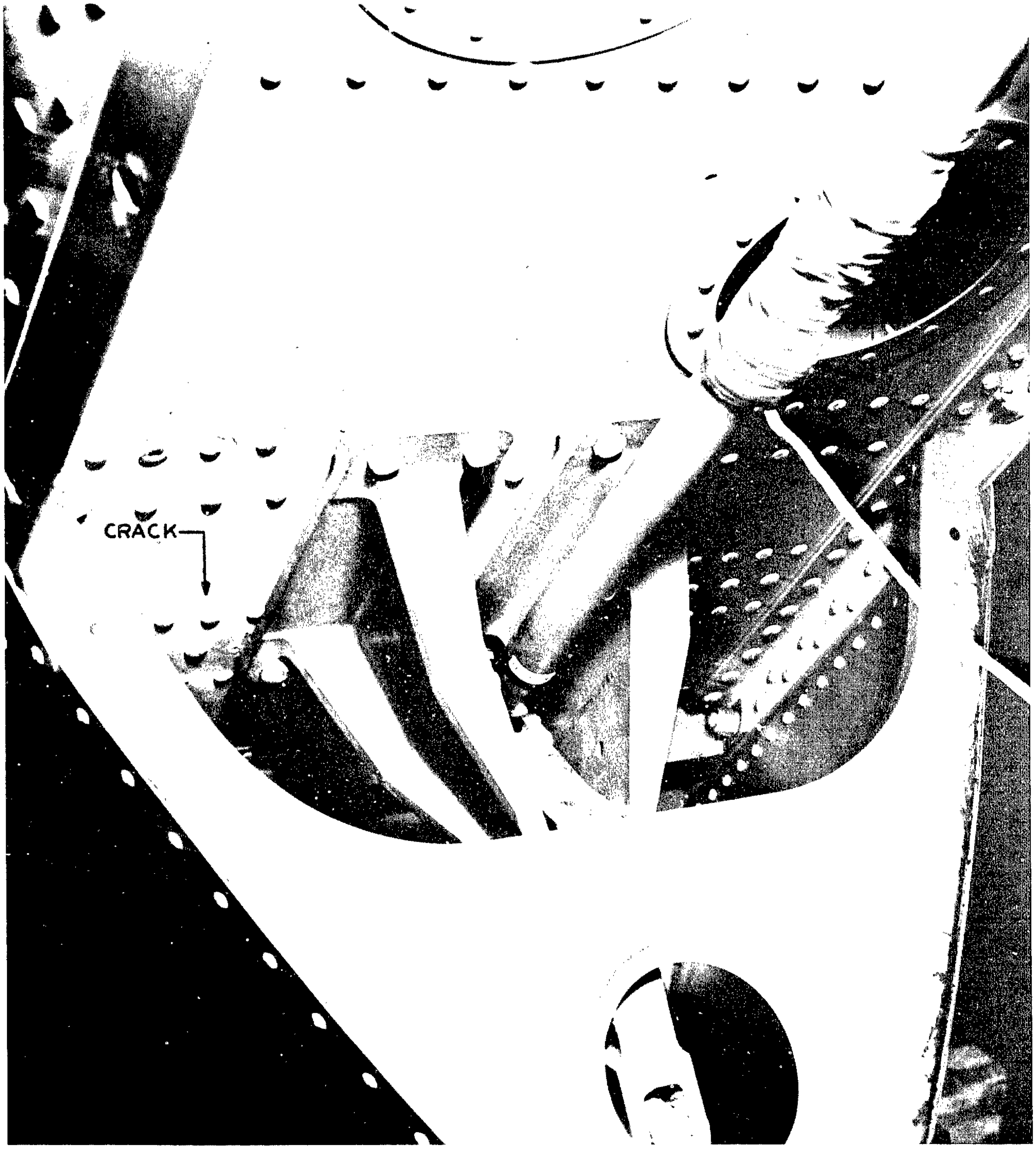

CRACK
L.H. WING

5.4.5 NO BACKUP STRUCTURE — One of the fundamentals of science is that for every action, there has to be a reaction. Here, a husky control bracket was attached to a flimsy spar web structure with results as shown. Solution is to provide support structure (backup structure) having strength equal to that of the fitting. This differs from the examples shown in 5.4.4, which were known to have adequate support structure.



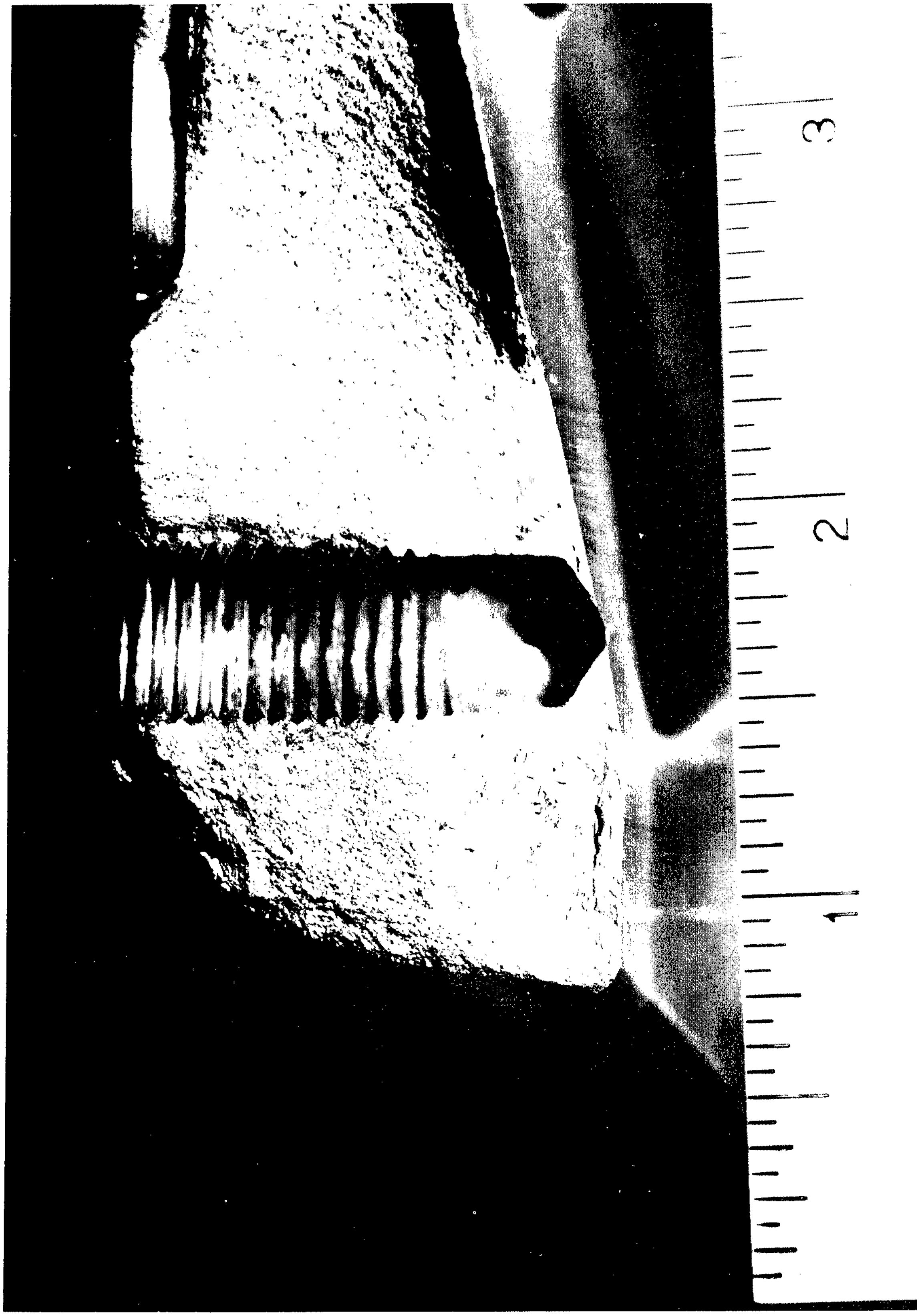
This is a typical problem where one person designs a component while another designs the supporting structure. In this particular case, it is doubtful that much thought was given to supporting structure. Rather, the bracket was attached to a structure that was designed for another purpose.


CRACK

5.5 SUPERIMPOSED STRESS CONCENTRATION

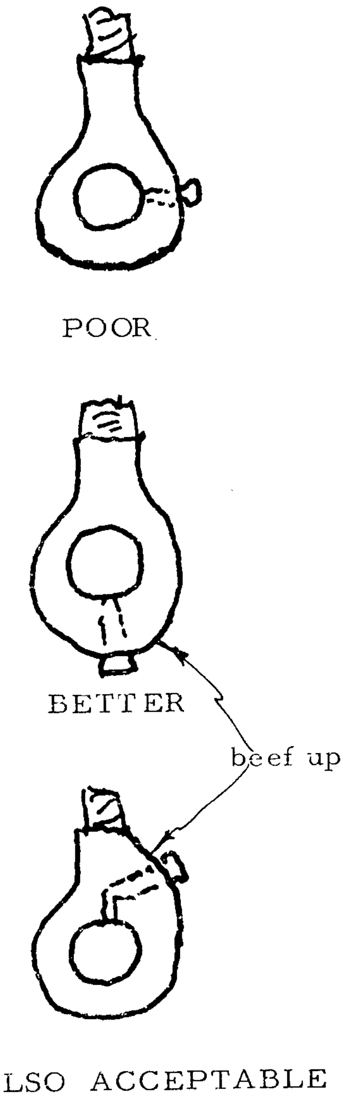
As though notches were not bad enough, they can be made worse by superimposing one upon another. Thus, we have examples of feathered edges that also terminate at heavy sections and fillet radii that don't match the rest of the machined surface.

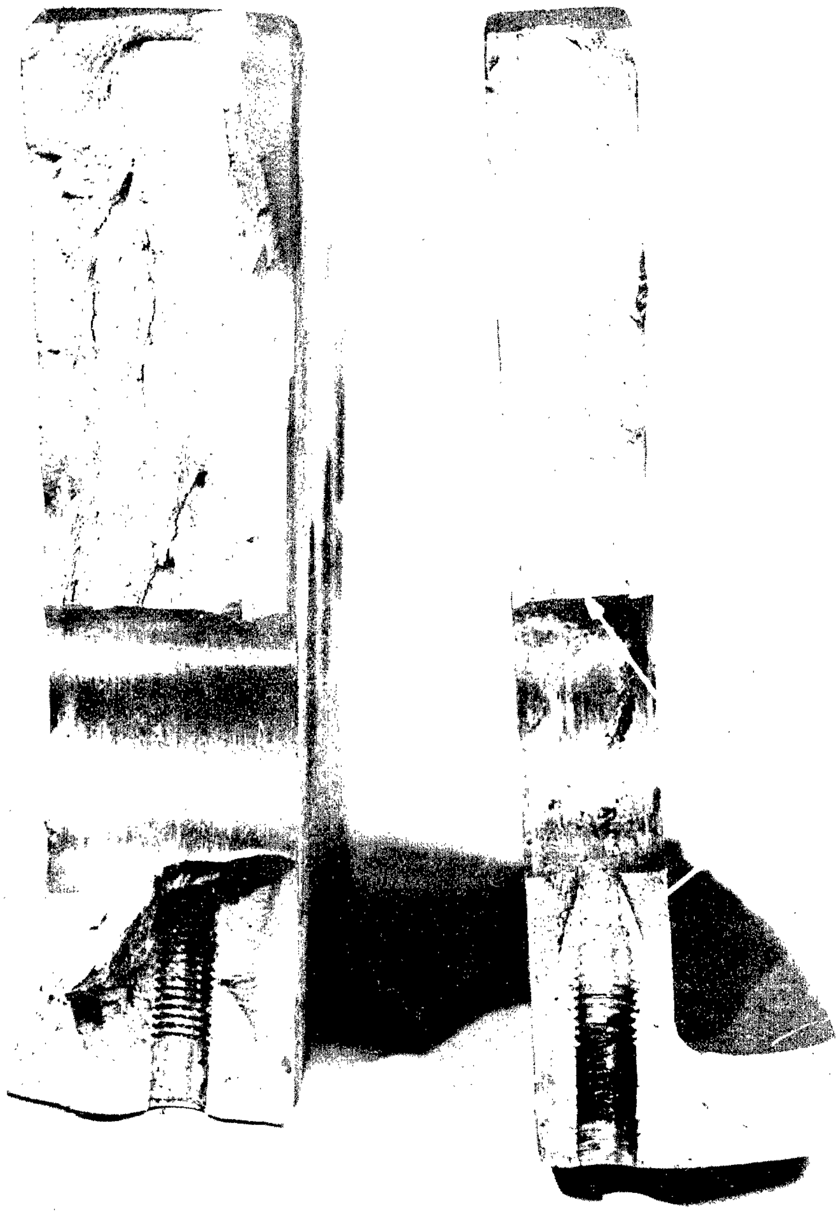
5.5.1 HOLE PARTLY THROUGH — The opposite photo shows how fatigue can start at a hole that was not drilled all the way through. The solution is simple: finish drilling the hole.



5.5.2 INTERSECTING HOLES—Here is a case where a hole was tapped into the highly stressed region of another hole in the tension flange of a wing beam. This is similar to the case of a grease fitting hole for a bearing.

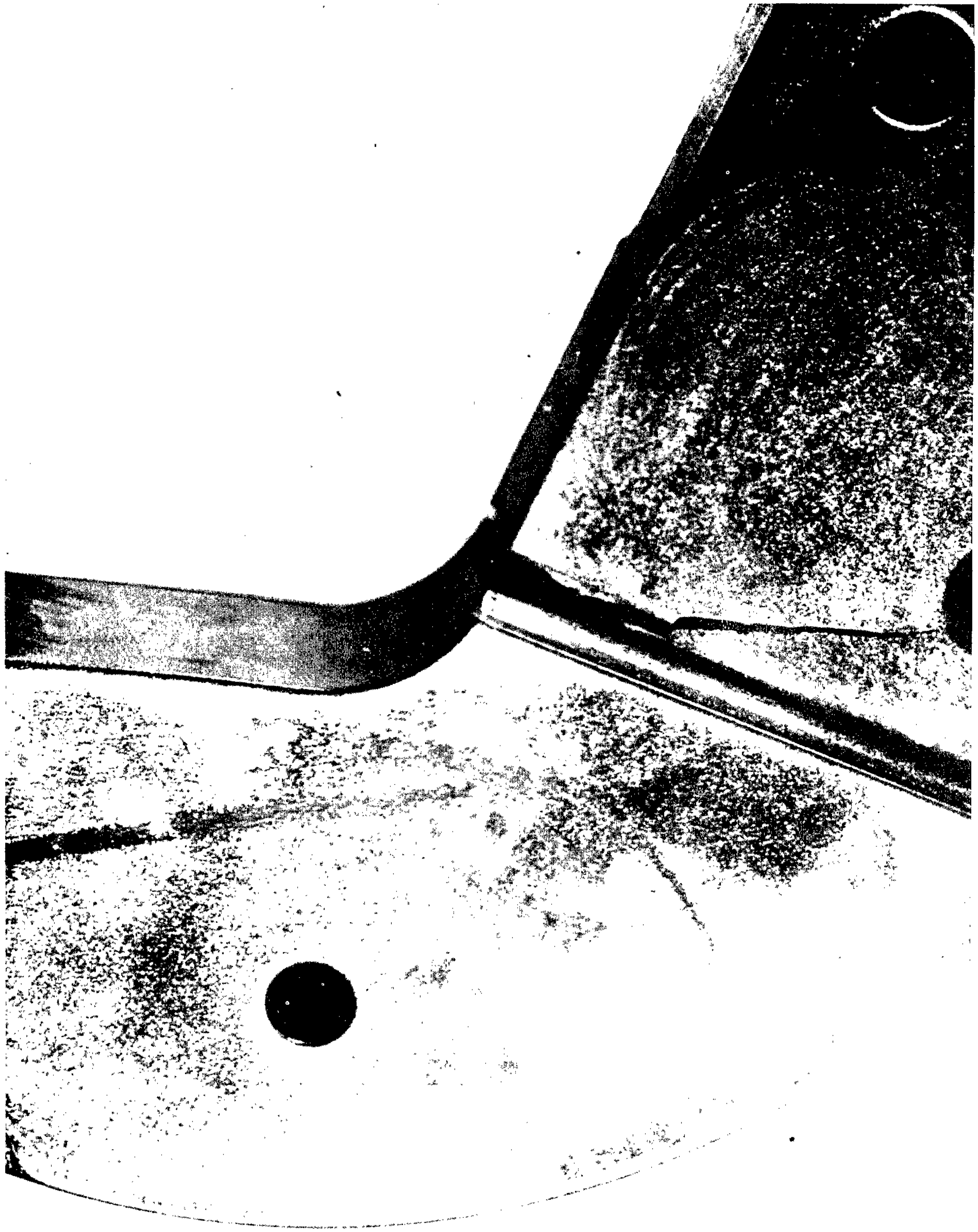
If you must have a hole, the idea is to move it to the least damaging position. A little local reinforcing (beef up) is also helpful.





FORWARD

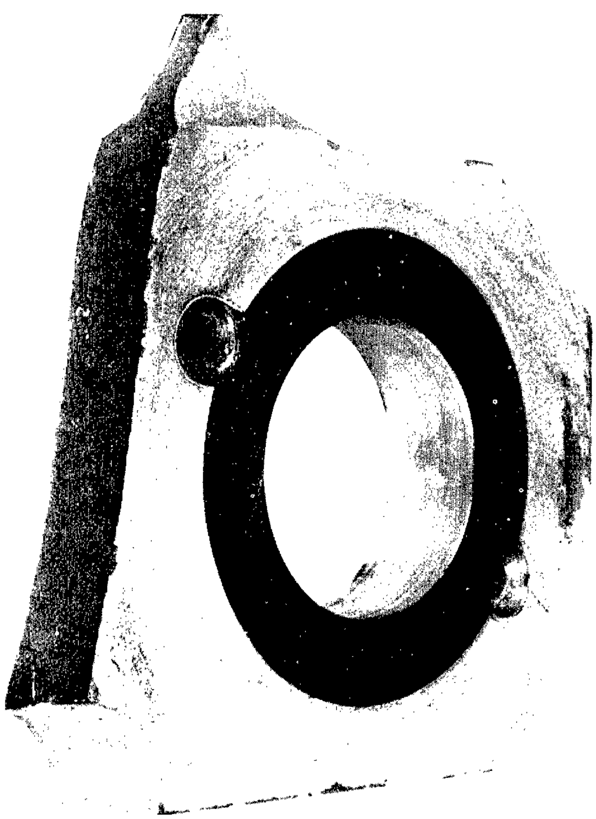
5.5.3 RADIUS AT CHANGE OF SECTION — There always has to be a radius of some sort at any change of section. Likewise, there has to be a radius at a change of direction. However, you don't have to make one radius right on top of the other. The radius for change in section should have been made at another location. If this were impossible, both radii should have been enlarged to permit a more gentle transition.



5.5.4 ROUGH SURFACE FINISH — A finish such as that illustrated here also constitutes a superimposition of stress. Where a stress raiser already exists, as in this case, it is foolish to let a rough surface like this get by, especially with tool marks normal to the direction of loading.

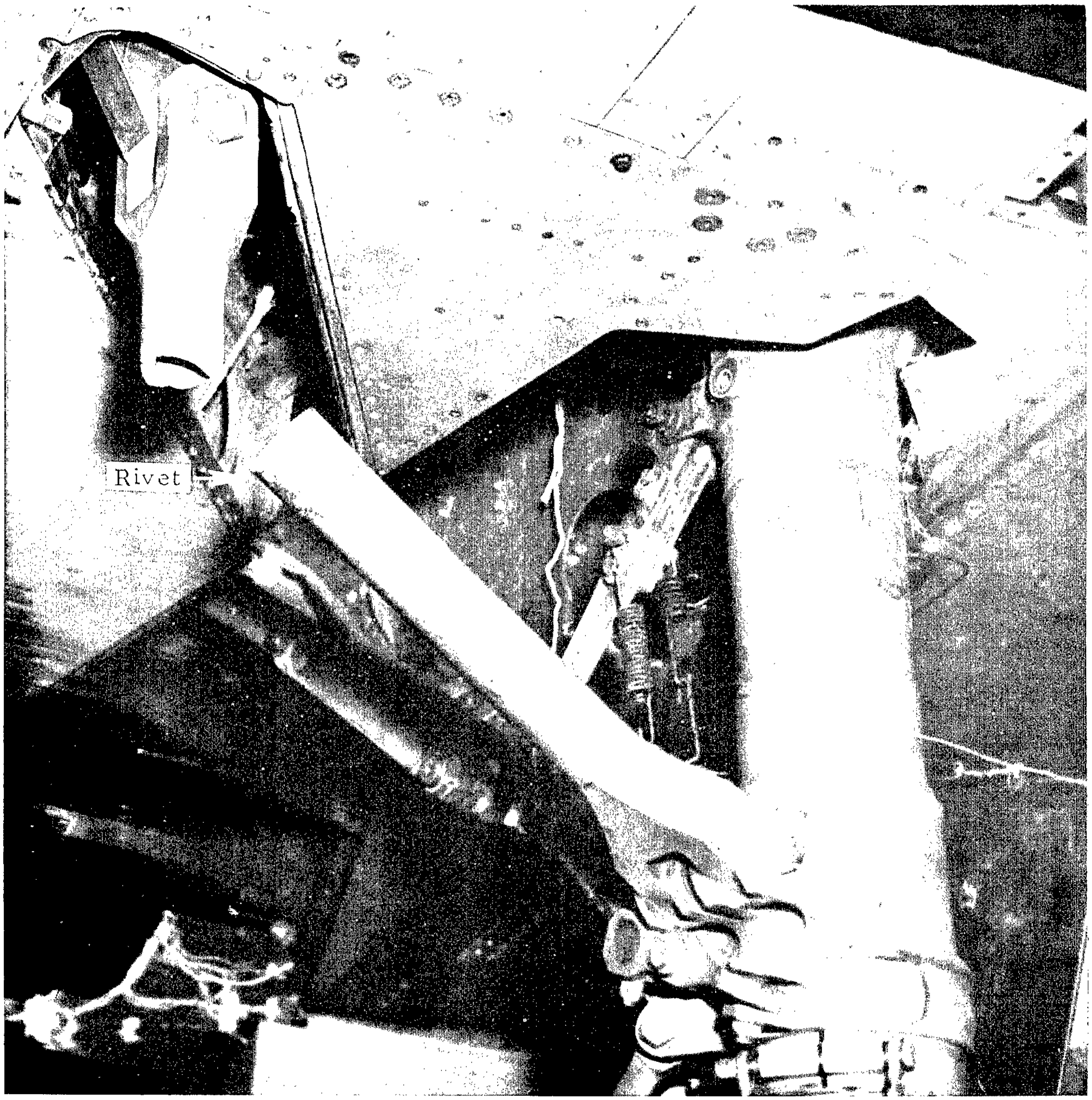
730-755 O-64--6

5.5.5 CROSS GRAIN -- The parts shown failed for two reasons, rough surfaces, and the material's grain structure is normal to the direction of loading. While it might have been possible to avert failure by machining a smooth surface, the wrong direction of grain makes such a solution highly speculative.



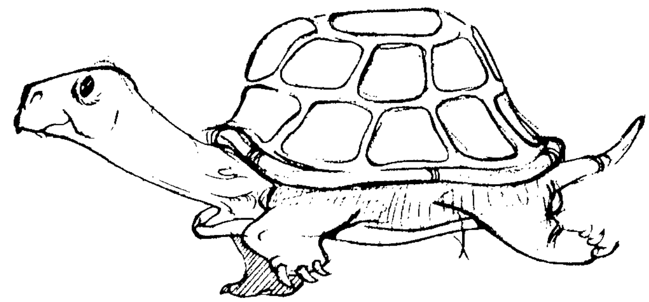
5.6 AUXILIARY ATTACHMENTS—

The part shown was from a landing gear assembly. Failure occurred through a rivet hole used for attaching a schafing shield. Similar things happen to frames having attachment screws to support hydraulic lines, upholstery, or what have you? It would have been better to tie the part on with rope (also try adhesive bonding) than to take chances on fatiguing as shown.

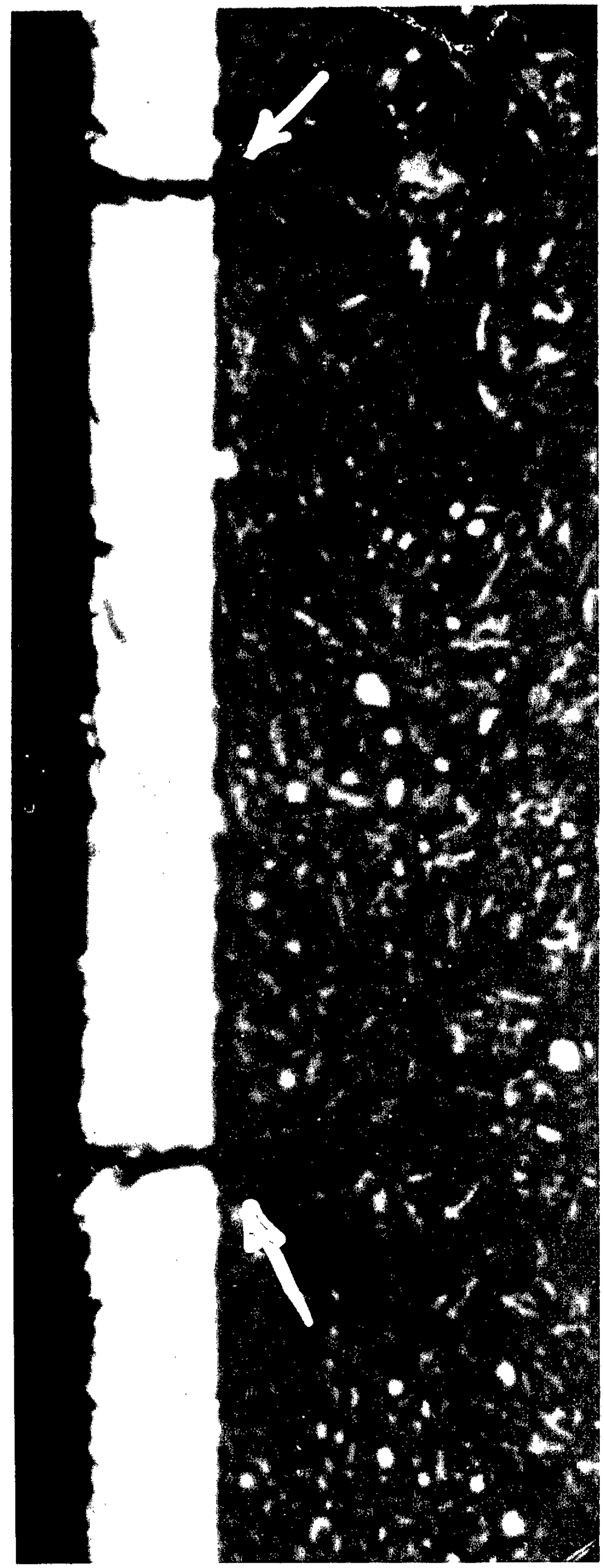

Rivet

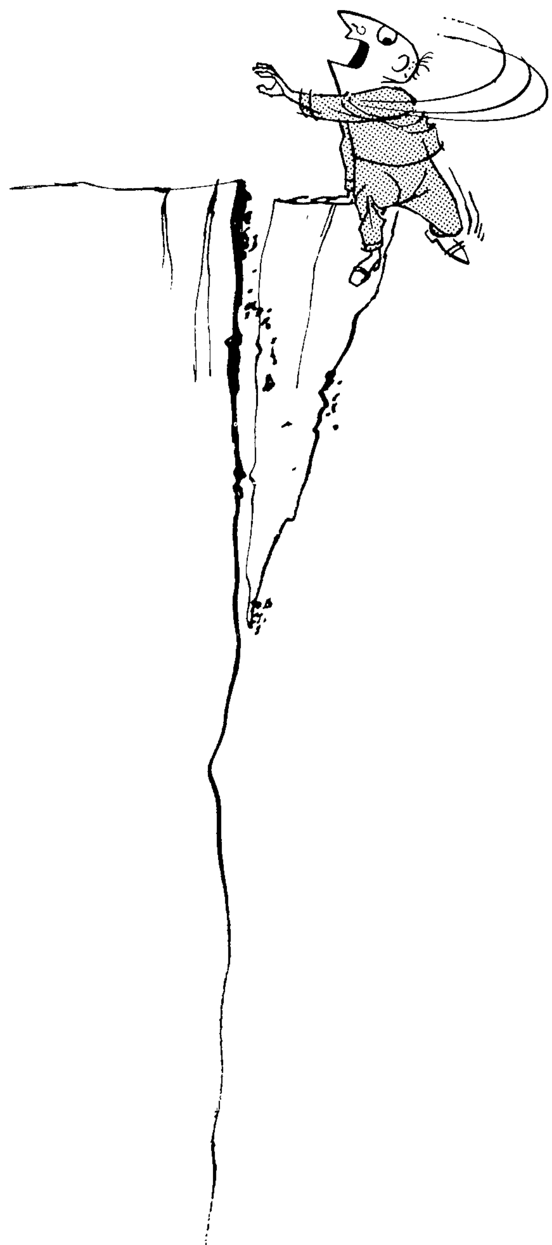
5.7 HARD PLATING

Here is a typical example of where a part was chrome plated to make it more wear resistant. It wasn't more fatigue resistant. The cracks in the plating act as stress raisers that eventually fail the part the plating is supposed to protect.



Shot peening prior to plating is a common inhibitor of fatigue cracking in chrome plated parts. It is unwise to chrome plate parts for dimensional buildup or wear resistance without the help of the specialist.



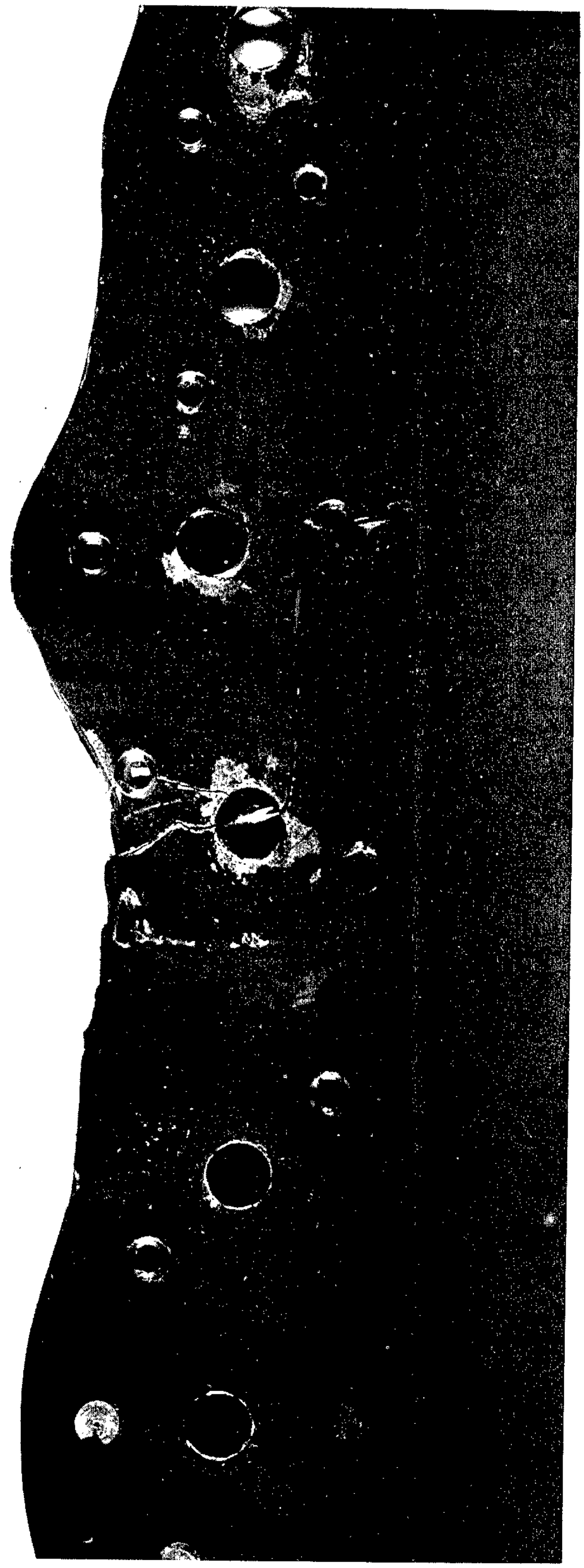


5.8 SHOP BLUNDERS

While most of the examples previously shown can be blamed directly on design, a number could equally well have been caused by shop blunders. Thus, we have the hole that was not tapped all the way through and the rough surface that was not smoothed. The following examples can be blamed almost entirely on shop practices.

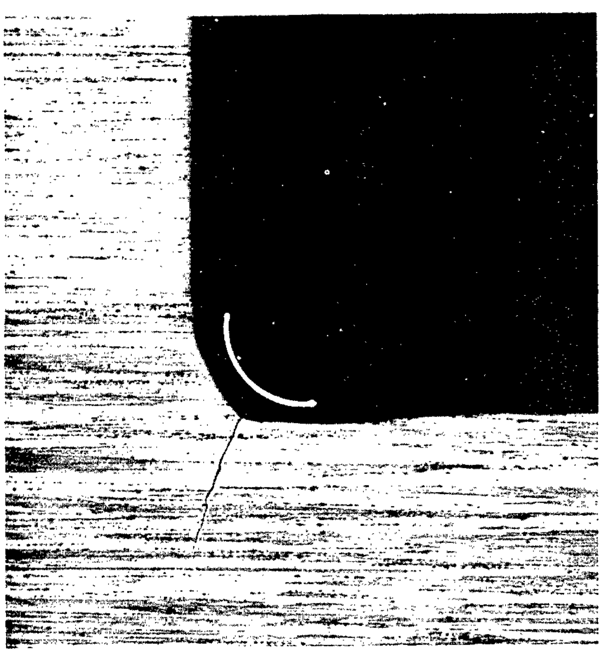
5.8.1 NO EDGE DISTANCE — The

engineering drawing may not have specified the exact locations of holes for nut plates; however, standard shop practices should be such that this would never happen. As shown, there was insufficient room for nut plates to be spaced in such a manner that holes would fall between nut plates. Also, note that the edge surface finish was nothing to brag about.

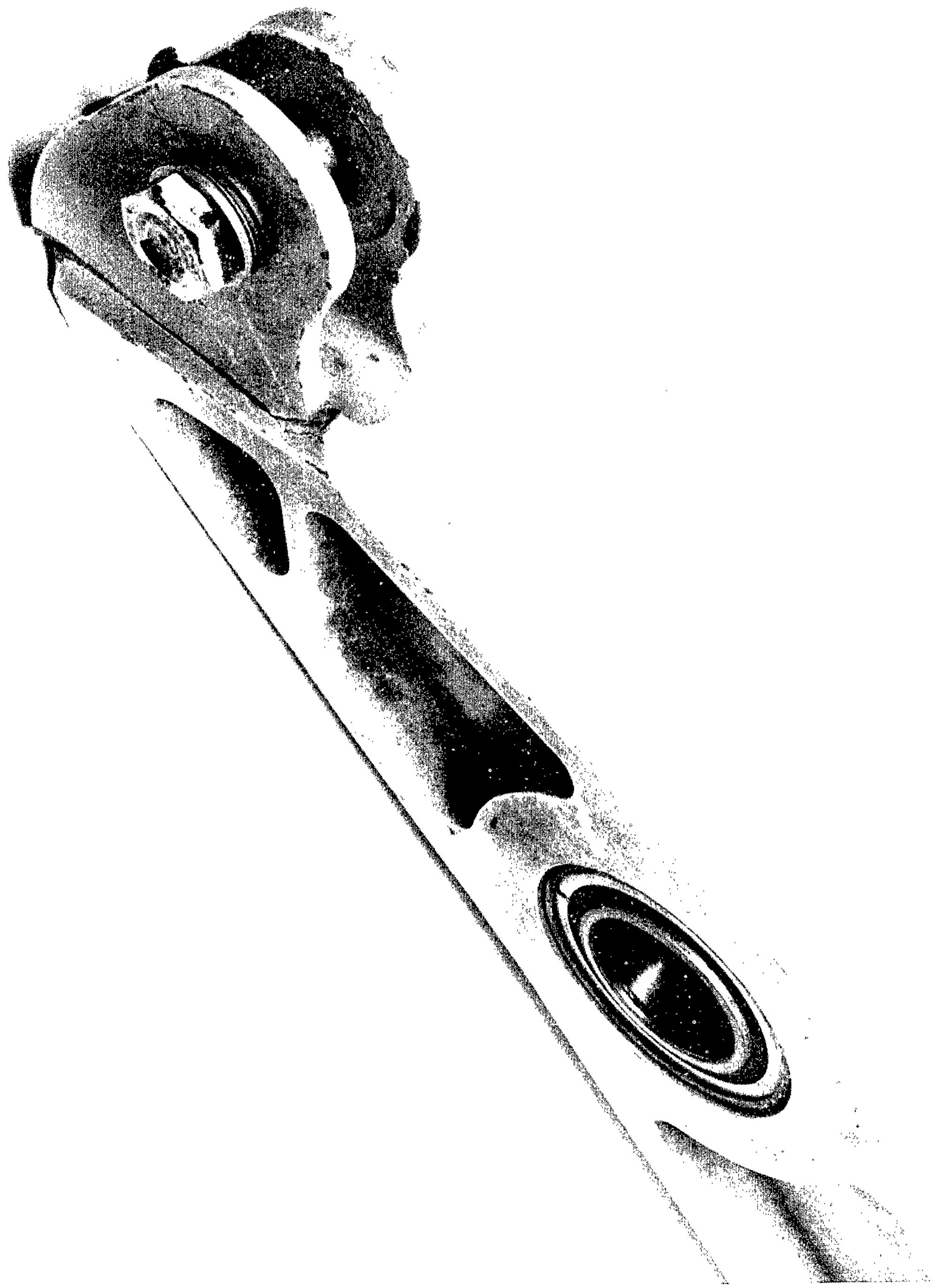


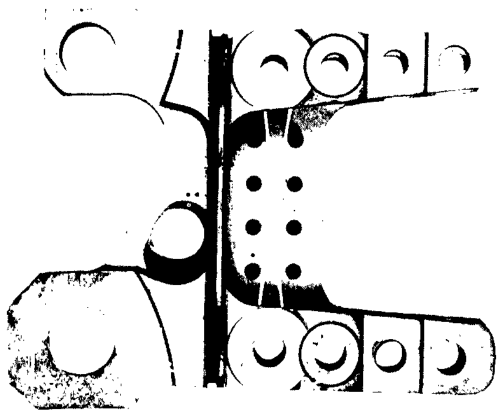
5.8.2 MISMATCH -- Even excellent machinists often machine a curved surface that doesn't meet its straight counterpart, leaving what amounts to a superimposed stress raiser. While it is not so bad where the two surfaces are convex, the concave ones usually result in failure as indicated in the illustration.

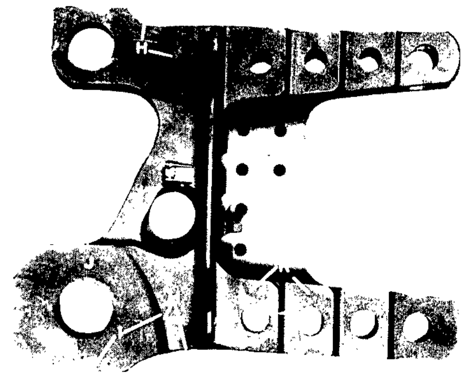
Many drawing room manuals specify the maximum allowable mismatch. It so happens (not in these cases) that fatigue failures have resulted where the mismatch was within tolerance. Care should be exercised in permitting mismatches in critical areas -- even within specified tolerances. While it would be virtually impossible to define the amount of mismatch that can be permitted in every case, a rule of thumb is to use extreme care with concave surfaces.



5.8.3 EXCESSIVE CLAMPING—The bolt on this part was tightened without having the proper spacer bushing. Fatigue failure finally set in, as might be expected. Make sure you have the right length bushing and the right length bolt and THINK TWICE BEFORE TIGHTENING (see 4.4).







ORIGINAL FITTINGS

A. WRONG GRAIN DIRECTION
B. SPOTFACES
C. INADEQUATE FILLET RADIUS
D. SHARP CORNERS
E. FEATHERED EDGES
F. ROUGH SURFACE
G. UTS=240-250 ksi

REDESIGNED FITTING

H. ENLARGED FILLET RADIUS
I. ROUNDED CORNERS
J. SMOOTH SURFACE FINISH
K. NO SPOTFACES
L. UTS=210-220 ksi

5.9 MURPHY'S LAW

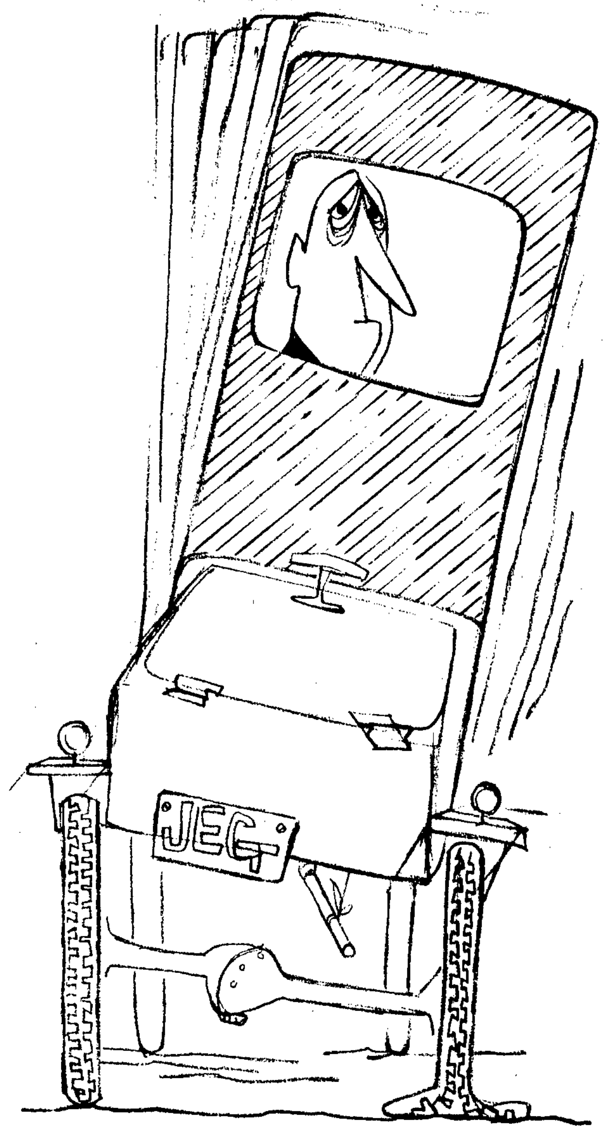
Murphy's Law states that if it were possible to botch up a job, someone will surely find a way to do it. DON'T UNDERESTIMATE GROUP EFFORT! As shown in the upper left photograph, the combined efforts of engineering and shop very nearly succeeded in doing everything wrong.

Botches in this one part include sharp edges, bad spotfaces, small fillet radii, rough surface, cross grain, and others. In addition, there were signs of hydrogen embrittlement due to cyanide-bath cadmium plating. Hydrogen embrittlement is a term used for low ductility caused by absorption of too much hydrogen during processing. These nice big words sound authoritative when used to explain failures for which no real reason (other than poor design or workmanship, which we hate to admit) is apparent. Decarburization (another mouthful that means loss of carbon due to poor processing) was also apparent to a minor extent.

Proper processing was insufficient to bring the part up to required life, so it was necessary to perform a major overhaul. This included providing a better surface finish, removing sharp edges, and providing more generous fillet radii. The reworked part is shown in the upper right photograph. Tests on similar parts revealed a life of approximately four times that sustained by original parts with no increase in weight.

6 MAKING THE MOST OF A BAD SITUATION

6.1 BAD SITUATIONS



6.1.1 THE CASE OF THE FLAT TIRE

-- Have you ever had a flat tire while driving along the countryside and discovered that your spare was also flat? . . . No pump or patching material . . . so you finally decided to drive it flat. Then there was the joker who passed you and yelled, "Don't you know you got a flat tire?" And you felt like sticking a big sign on your rear bumper saying, "I know it's flat - - so what?"

If you have had such an experience, then you can imagine what it's like to have a service failure in an area where there just isn't enough room for replacement with a huskier part. Maybe you are already using material as strong as you dare.

What now? Shop is still turning out parts (like those that broke) by the barrel-full, and you're faced with the need for a quick decision. You have three possible decisions: (1) you can do nothing and hope that the rest of the parts won't be so bad; (2) you can try to fix it up and hope that the fix is OK; or, (3) you can fix a few samples and test the parts to see if the fix is any good.

6.2 DECISIONS

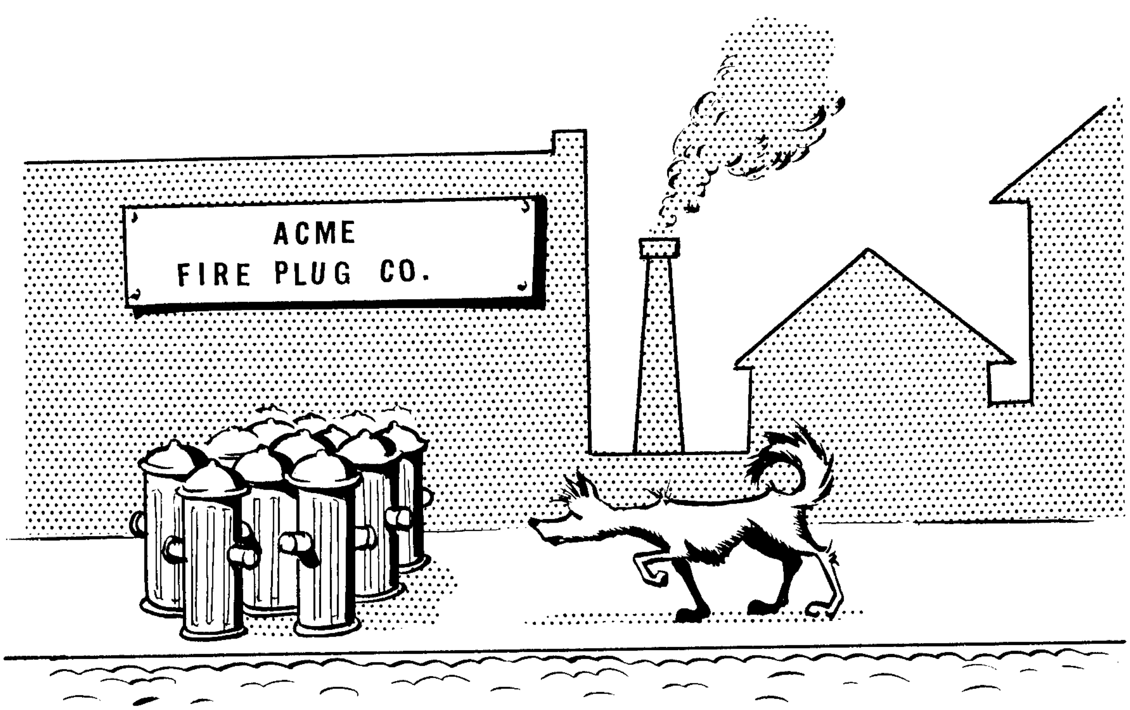
Our decision on a structural fix should be based on the facts we've learned so far. The purpose of this chapter is to arrange these facts in such a manner that our decisions can be easier. What are these facts?

Some of the basic principles were presented in Chapter 2. Also shown were methods for reducing stress at fillets by providing a more generous radius.

The subject of joints was introduced in Chapter 3. Of particular importance was the fact that a small change in basic design could result in a vast improvement in fatigue life. In Chapter 5 we found what happens when we violate principles of good design and fabrication. As in violation of principles of good health, the corrective measures may be slow, painful, and bad tasting.

6.3 SIZING UP THE SITUATION

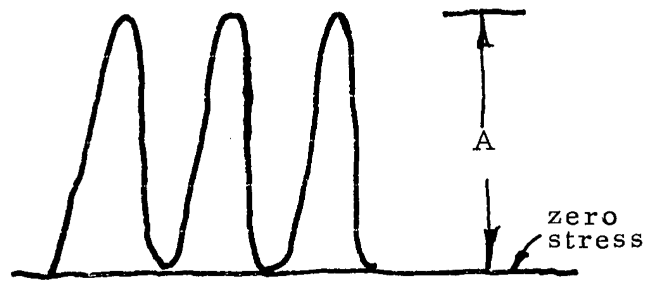
As far as fatigue is concerned, remember that a structure will never fail except at a stress concentration. Accordingly, let's worry about stress at the concentration and never mind about what happens elsewhere, at least not for the time being. This simplifies our problem. The next thing is to visualize what can be done to this particular stress to make the most of the situation.



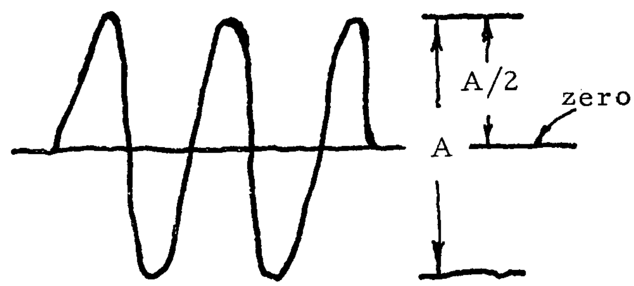
DECISIONS -- ALWAYS DECISIONS

TIPS ON FATIGUE

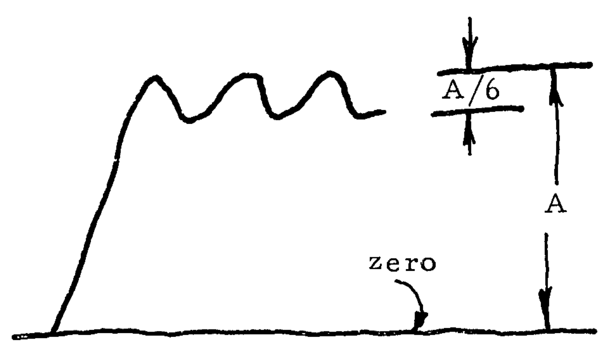
We start out with a stress like this . . . .



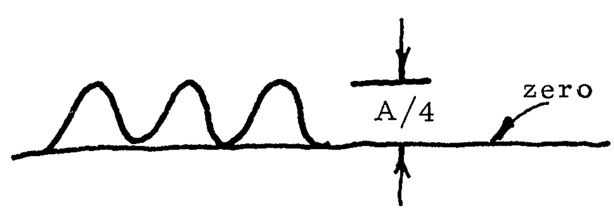
which could possibly be shifted to this . . . .



or again it might be changed to act like this . . . .



or something might be done to make it behave like this . . . .



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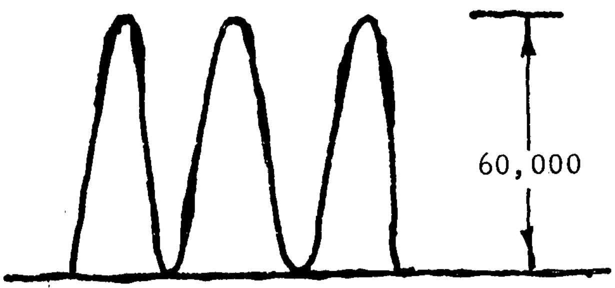
While there may be others these cases will suffice to start. Suppose a part were loaded so that stress at the concentration fluctuates from 0 to 60,000 psi (R = 0). Such a part could be expected to

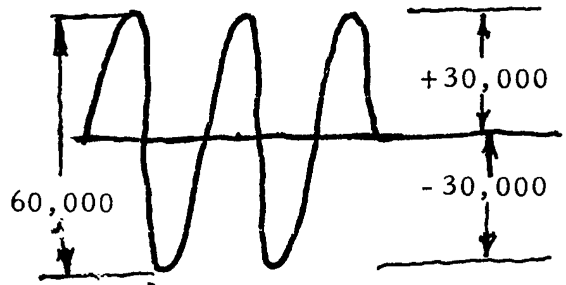


30,000 Cycles, Stress
Range = 60,000 psi, R = 0

last for about 30,000 cycles, according to the S-N curve for R = 0 shown in Figure 2.2. Where the stress range is defined as the difference between the maximum and minimum stress, total stress range would be 60,000 psi.

6.3.1 MOVE THE WHOLE STRESS DOWN

If it were possible to do something to the structure locally so that the stress at the concentration would be 30,000 psi in compression (-30,000 psi) when the part was unloaded, the original loading should cause the localized stress to fluctuate between -30,000 psi and +30,000 psi (±30,000 psi). This would correspond to the same stress range as before, but the life now would be

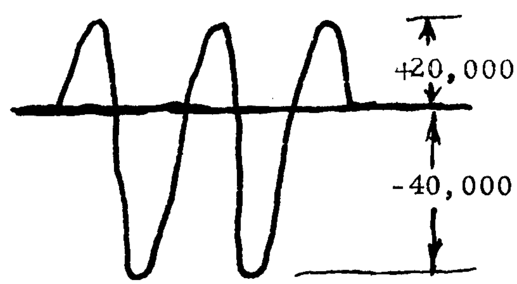


90,000 Cycles, Stress
Range 1 60,000 psi, R = -1

90,000 cycles (curve for R = -1) and maximum stress = 30,000 psi, Figure 2.2.

The introduction of a compressive stress at the concentration sounds like a nice trick if you can do it. That is, the compressive stress should be a permanent affair - - locked up so it can't get away - - and should be in just exactly the right spot. Such stresses are commonly known as residual stresses. Residual stresses can either be in tension or compression.

Similarly, if the stress could be made to behave as though it were cycling



200,000 Cycles, Stress
Range = 60,000 psi, R = -2

from minus 40,000 psi to plus 20,000 psi, a lifetime of 200,000 cycles would result (curve for R = -2 and maximum stress = 20,000 psi, Figure 2.2).

When a part having a stress raiser is loaded in tension so as to cause the material at the concentration to yield locally, the permanently deformed material must go into compression when the load is removed and the part springs back. Such items as hooks lend themselves to this type of correction. Generally, the amount of overload is critical and should be specified by the specialist.

A practical method called "shot peening" is used to introduce residual compressive stress for a longer life. In the first instance, a compressive layer at the notch, amounting to 30,000 psi, will do the trick. In the second, a layer of 40,000 psi would be required. Both are easily achieved, it being common practice to introduce residual compressive stresses as high as two thirds of the material's compressive yield strength. Other methods of introducing protective compressive stress layers include controlled mechanical peening, vapor blasting, surface rolling, and a process called "coining." Surface rolling is especially suitable for cylindrical objects such as bolts. Figure 6.1 shows how thread rolling was used to improve the fatigue life of bolts.

Fatigue life of a part can be improved by providing a better finish. This is particularly true when the original part failed because of machine mismatch. (A mismatch occurs where the two machine surfaces do not meet--see section 5.8.2).

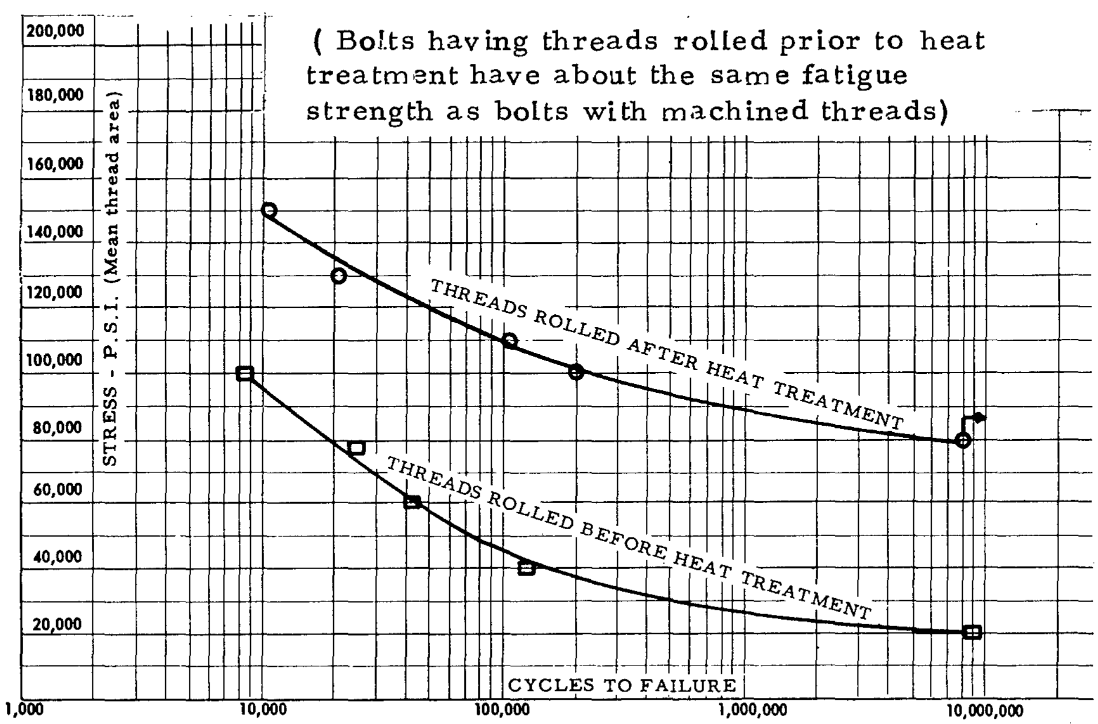
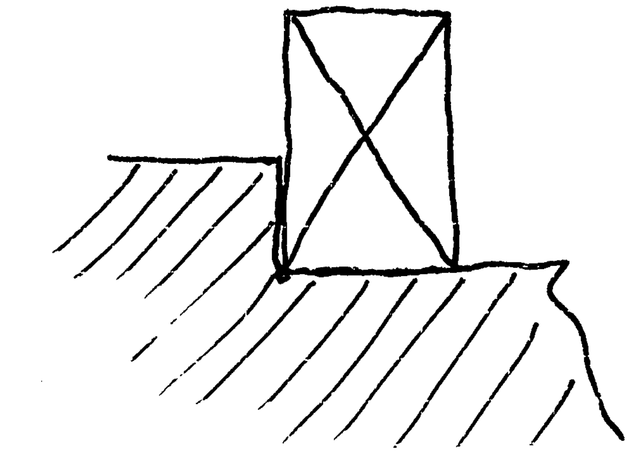
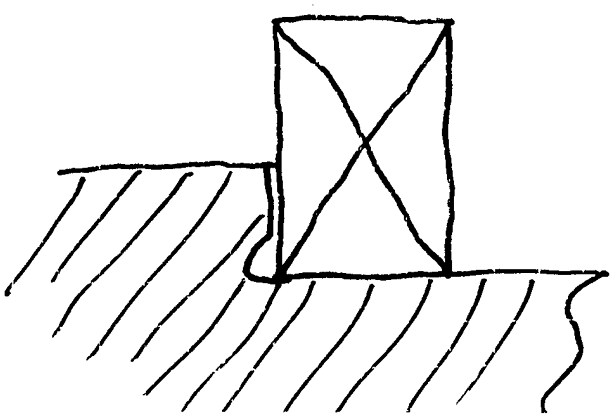


Figure 6.1 Thread Rolling

Figure 2.11 shows how stresses can be very severe as a result of this. Simply smoothing out the radius would be a solution to such a mismatch. Other cases may not be so simple. We may find a part made with a sharp notch so that a bearing could sit close



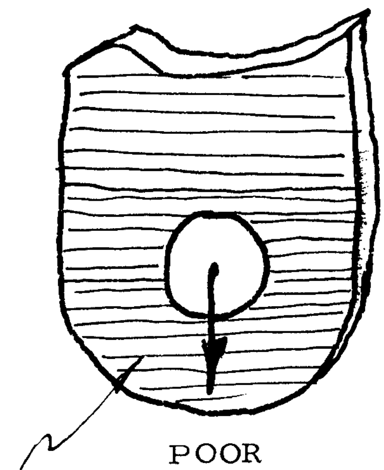
POOR



BETTER

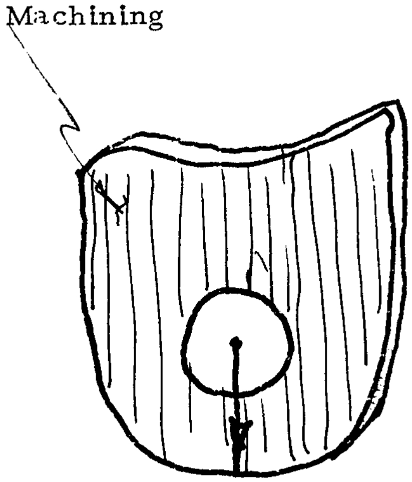
to the shoulder of a shaft. Even so, it is sometimes possible to improve the distribution of stress by removing some material to eliminate the notch and still have a close fit, as shown in the lower sketch.

Let's take the case of a lug. Assume that the machining direction were normal to the direction of the load. As we have already seen, this would constitute a superimposed stress concentration. However, if the machining had been parallel to the



POOR

Direction of Machining



BETTER

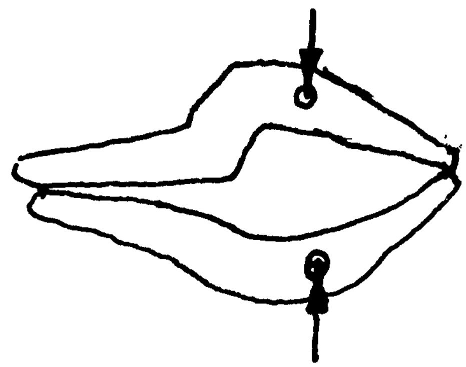
direction of the loading, a substantial increase in fatigue life would result. This not only applies to rough surfaces but would be equally applicable to any surface finishing.

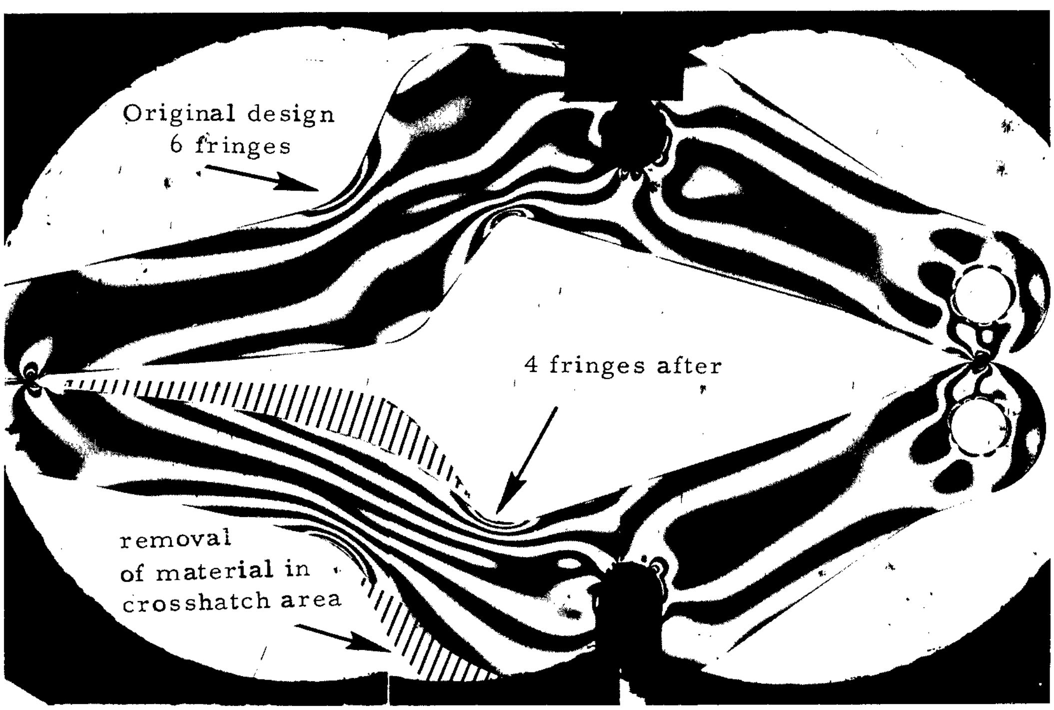
6.3.2 CUT OFF THE TOP OF THE CURVE -- For ordinary rough surfaces, sometimes smoothing will do the trick; but the manner in which the smoothing is done will have a lot to do with how satisfactory the part will be. For example, the part that failed in Section 5.4 might have been satisfactory if the direction of the machining had been with the radius.

The importance of removing material to improve fatigue life cannot be over emphasized. This not only results in a lighter structure, but one that is trouble free.

Our examples so far have been for removing material in fillets. Equally important, perhaps more so, is removing undesirable external humps.

Photoelastic models in Figure 6.2 show how a beam can be reworked to lower the stress at the concentration. Except for removal of material shown in the crosshatched area, the lower model is identical to the upper. Being identically loaded (one against the other) the revised model had the stress lowered to two thirds of the original value. This would increase fatigue life more than ten times. Removing





NOTE: Stress is directly proportional to number of fringes

Figure 6.2 Removing Material Lowers Stress

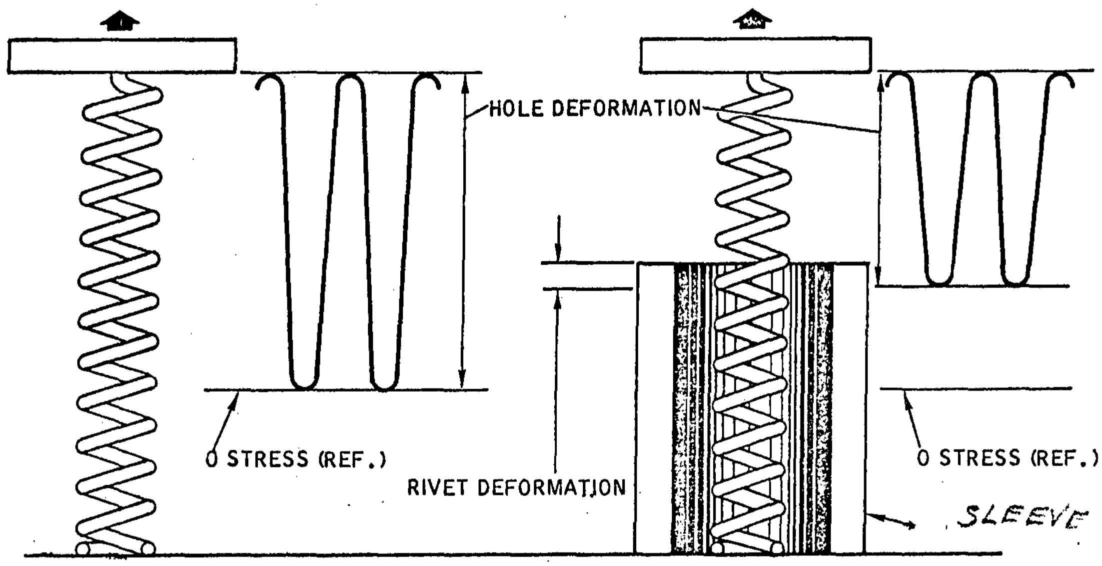


Figure 6.3 Analogy of Stress Cycle as Influenced by Interference Fit

a small amount of material makes the difference between satisfactory and unsatisfactory performance.

6.3.3 CUT OFF THE BOTTOM OF THE CURVE—Another method of shifting the stress cycle is by means of an interference fit fastener or a pressed-in bushing. Here's how it works. Turn back to Figure 2.2. Let's imagine that our part failed because the actual stress was cycling at R = 0 with a maximum stress of 50,000 psi. If we could make the stress behave as though loading were at R = +0.5, we see that the life would then be about 10,000,000 cycles. In other words, if we could block the hole and keep the stress from returning to zero, the load could be reduced to zero, yet the stress would remain at 25,000 psi in the example we have selected. Practically, this is easy to do. Figure 6.3 illustrates the principle involved in the use of an interference fit. Where the deflection of the spring is likened to a stress cycle, it can be seen that an introduction of a hose segment to prevent the return to zero would lower the excursion without adding to the maximum deflection.

Figure 6.4 shows the effect of various amounts of pin interference on the fatigue life of small lug specimens. While fatigue life does not seem to be lowered appreciably by excessive amounts of interference, look out for possible stress corrosion. With tapered bolts, now commercially available, the amount of interference can be closely controlled. Amounts of interference should be those recommended by the tapered bolt manufacturer.

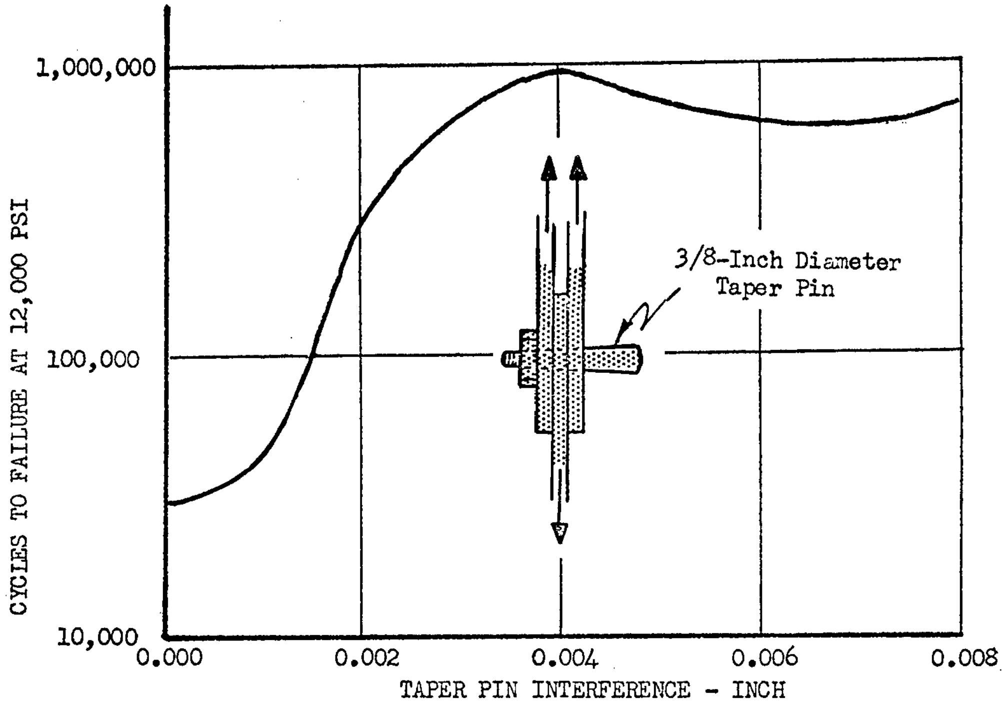
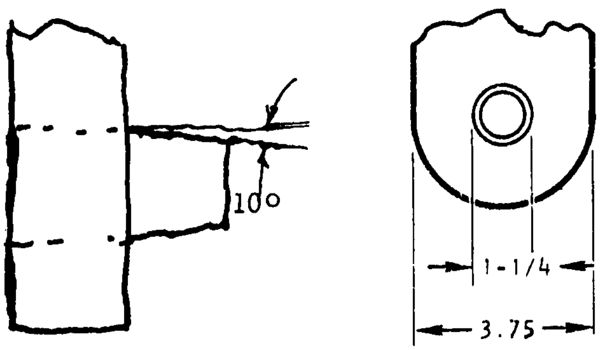


Figure 6.4 Effect of Interference Fit on Fatigue Life

Figure 6.5 shows a graph for large lugs with interference-fit bushings. In order to achieve the interference shown, the bushing must be longer than the lug thickness, with excess length chamfered 10 degrees. Excess length is ground off after pressing.



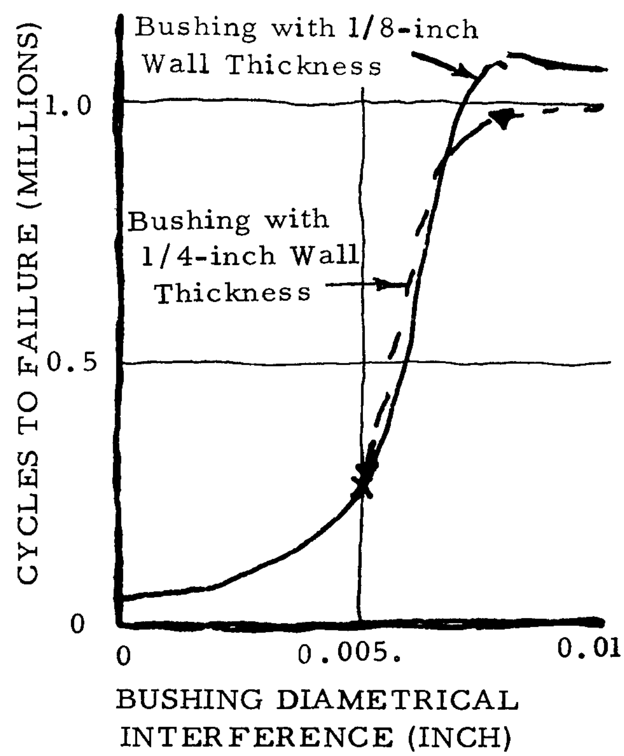


Figure 6.5 Press-Fit Bushings

When more than an 0.002 inch interference is used with an ordinary bushing, the bushing tends to broach the holes.

Examples in Chapter 3 show how fatigue life could be improved by adding edge-driven rivets to relieve the effects of bending in a riveted joint. The problem area in any riveted joint will usually be at the row of fasteners near the load.

Figure 6.6 shows how this can be achieved by providing a larger hole in the doubler (at the first loaded fastener) while using an interference-fit bolt in the material being protected. This acts very much like the edge-driven rivet described in Chapter 3:

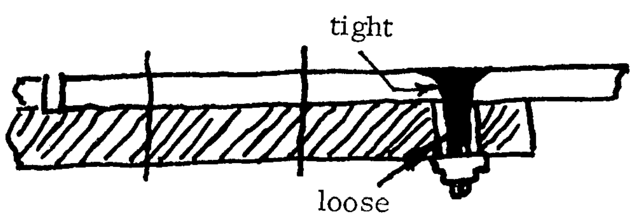


Figure 6.6 Interference Bolt Installation

axial and bending loads are separated, carrying bending loads only at the first row and passing shear loads to the second row. Both devices are examples of what are commonly called stress confusers, but the interference-fit bolt has the advantage that it can still carry shear load in an emergency, which the edge-driven rivet cannot do.

Fatigue tests of joints similar to that shown in Figure 6.7 show a fatigue life increase of from 117,000 cycles to 1,371,000 cycles over similar joints with 1/4-inch diameter rivets. Using an interference fit fastener without the oversize hole in the doubler resulted in a fatigue life of 323,000 cycles. Figures given are averages for five tests each.

7 CHECK LIST FOR FATIGUE RESISTANCE

7.1 A GREMLIN'S MENAGERIE OF COMMON OVERSIGHTS

In the light of what we have learned about fatigue, let's review some of the most common oversights that cause fatigue failure.

When things go wrong in an airplane, it is common practice to blame it on a gremlin -- one of those mythical foot-high, ill-humored imps that have been disrupting the works since man first took to the air. In the case of fatigue, however, it would seem that even a gremlin would need some help. It's as though the gremlin has a menagerie of common oversights, each "animal" with its own structural characteristics that only a gremlin could love.

Since no one loves these critters, it is common practice to say they belong to someone else -- that therefore they are someone else's responsibility -- and that if they are ignored, maybe they will go away. The critters never do go away. In fact, if let alone, they tend to multiply.

There are ways to breed this menagerie of oversights, and there are ways to reduce it. When the designer feels he is being pestered by a seemingly stupid question from a shop man who has found what the shop thought was an oversight, it is recommended that the lofty engineer listen with interest, respect, and appreciation.

Maybe this time the shop man hasn't found an oversight, but it's a cinch he won't keep looking for one if his efforts aren't properly appreciated -- if he comes in ten times and only the tenth is an oversight, it may be the one that saves the designer's reputation and 100 lives.

TIPS ON FATIGUE

On behalf of the shop man, there undoubtedly will be cases where a poor design has been allowed to get by because of cost or other excuses. In areas of very low stress, this might not be serious, but always remember that NEW AIRPLANES STILL FAIL IN FATIGUE because oversights have been allowed to get by.

Only a few of the critters will be shown, but enough to relay the idea. The thing to remember is that whenever one is found, the time to make the change is now. You may sometimes change a part that might have lasted its required lifetime, but it's better to be safe than sorry.

The gremlin's menagerie is intended to jog your memory by associating related ideas. Numbers in parenthesis at the right refer to appropriate sections in the previous text.

While there may be many other items that need checking, the few mentioned here should suffice for stimulating the alertness required to catch mistakes before they become serious.

A good plan would be to make a game of listing all possible design and shop errors. Try to categorize them. You will find many times the few listed herein. Also, by this time you will know what needs to be done to fix it. Good luck!

__________ SHARP NOTCHES

__________ GOUGES

__________ NICKS

__________ ALLOW STRETCH BETWEEN FIRST TWO ROWS OF FASTENERS

__________ SHARP EDGES

__________ RAPID CHANGE IN SECTION

__________ CHEWED UP SURFACES __________

__________ FAT GOBS OF EXCESS MATERIAL __________

__________ LONG SPANS OF UNSUPPORTED STRUCTURE __________

__________ DANGLING MEMBERS __________

__________ HIDDEN MISTAKES

__________ HARD PLATING

__________ TROUBLESOME SPOTFACE

__________ EXTRA HUMPS

____

____

____

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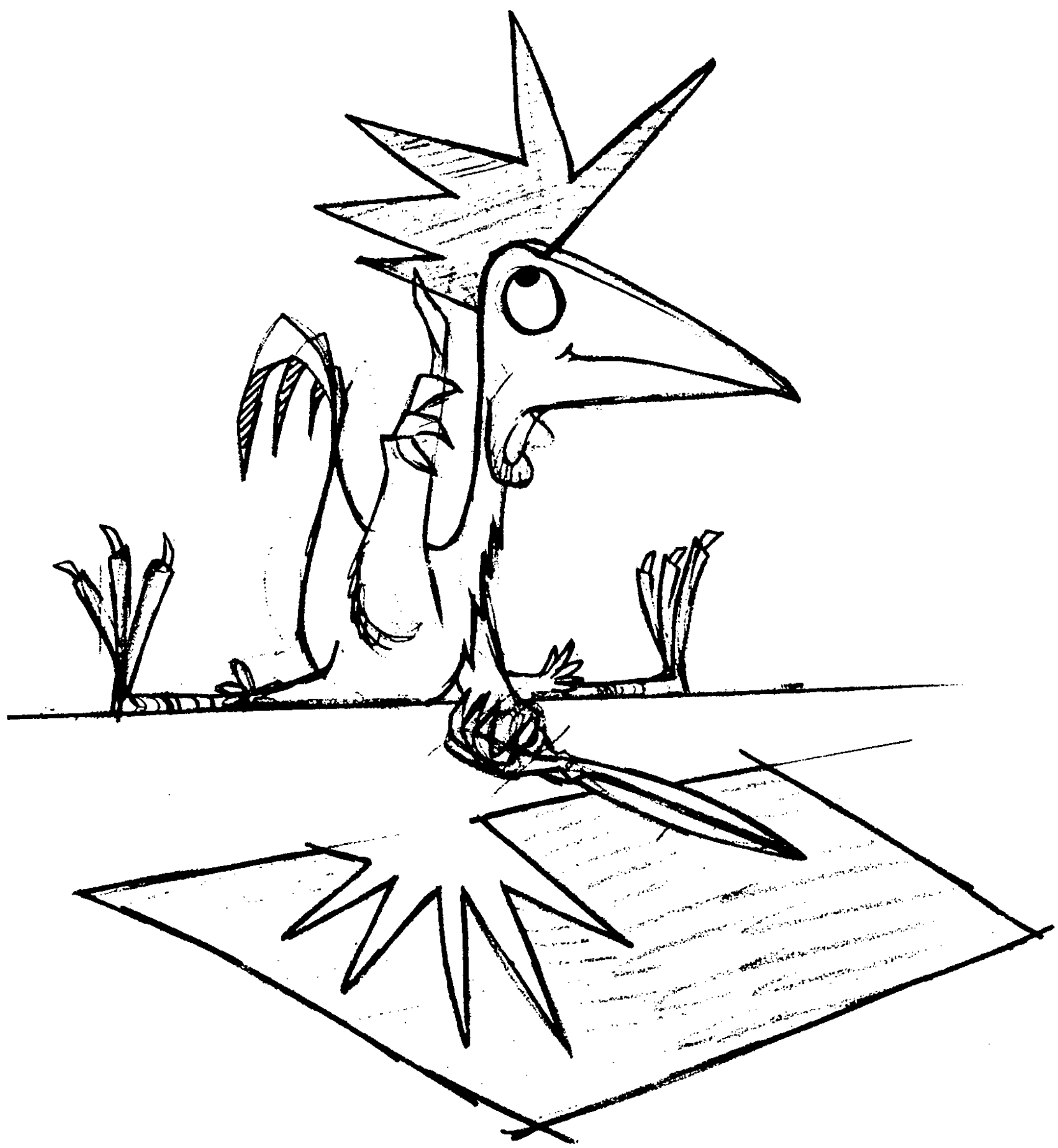
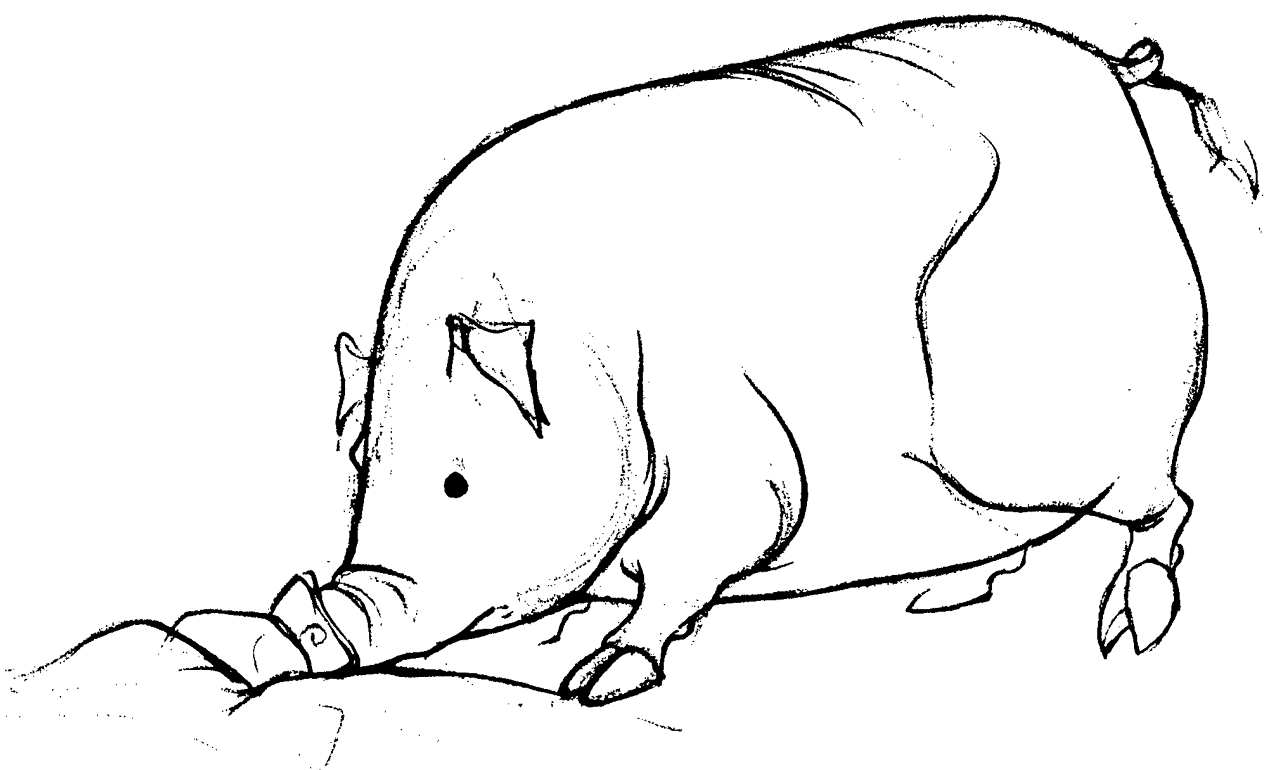
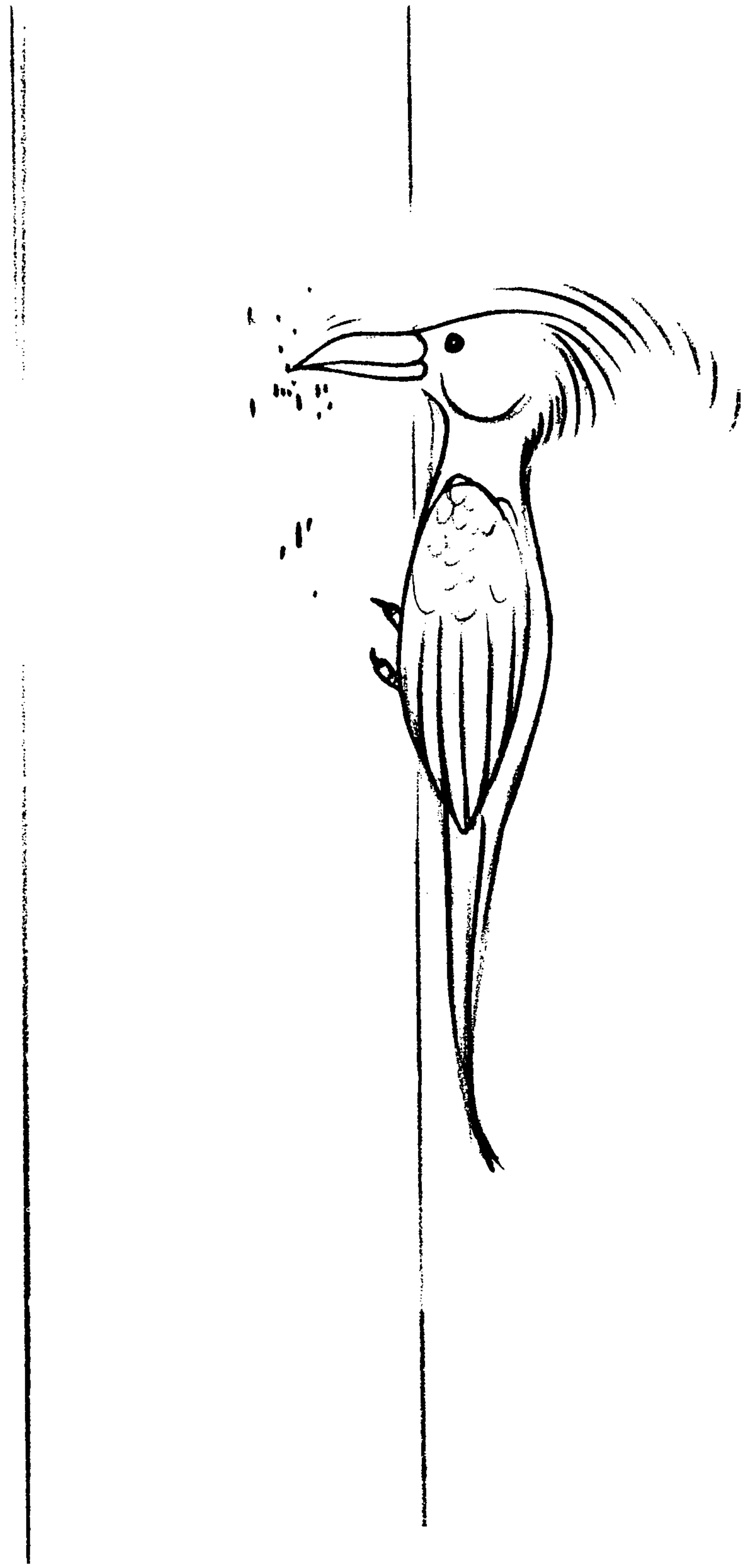


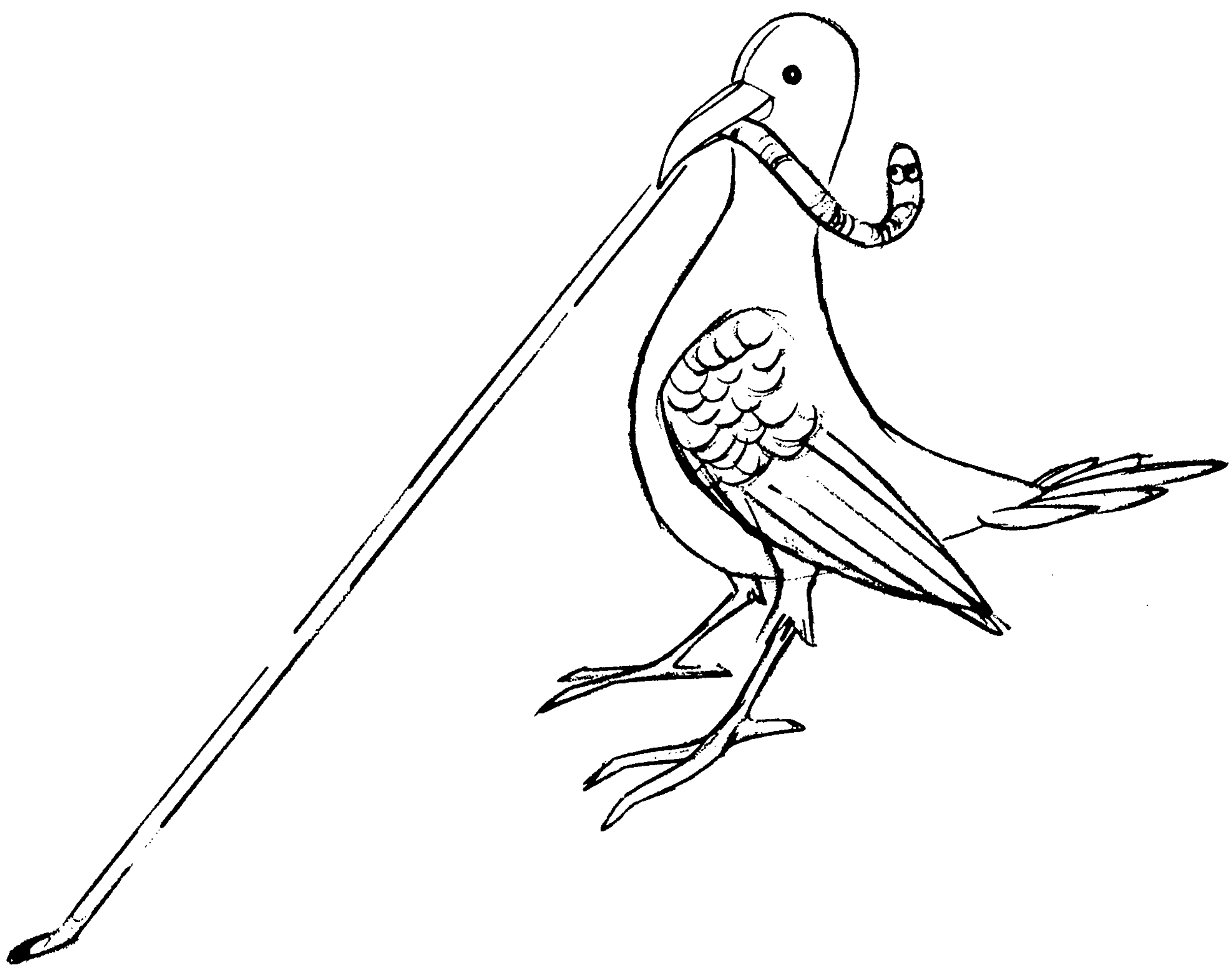
SHARP NOTCHES (5.2.1)



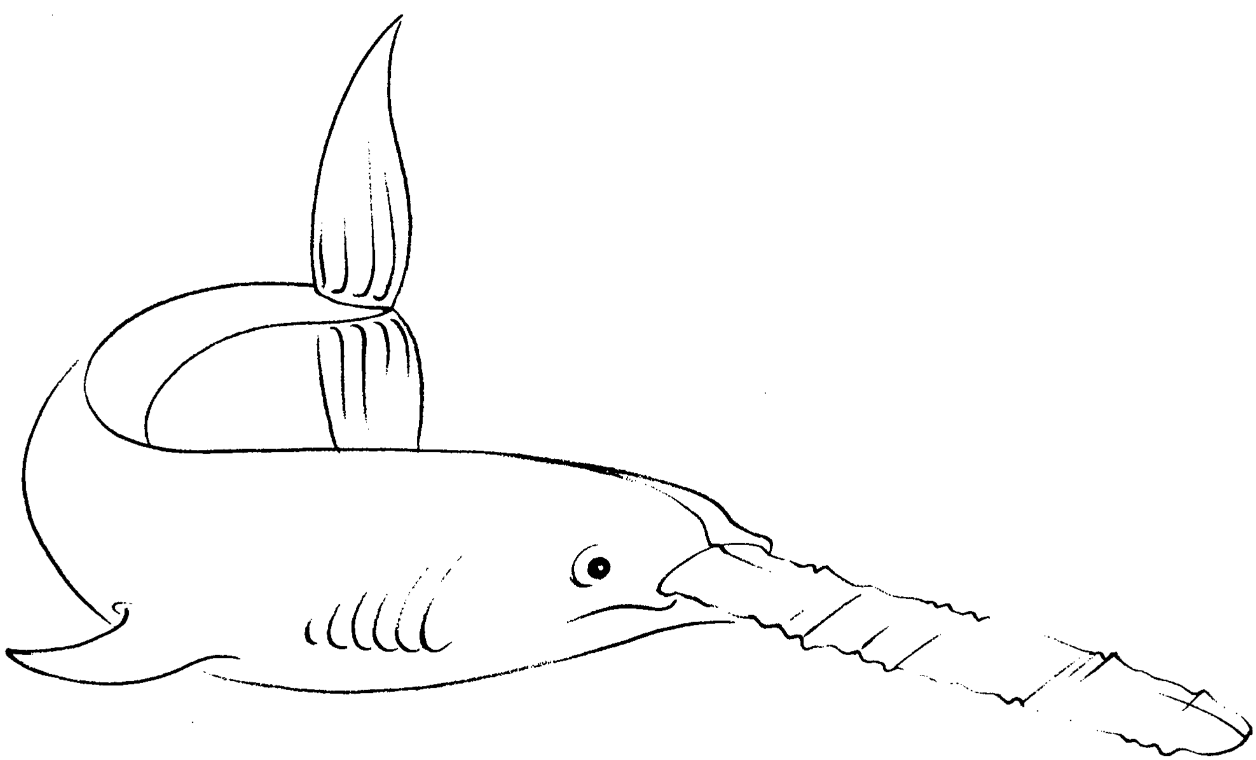
GOUGES (5. 5. 4)



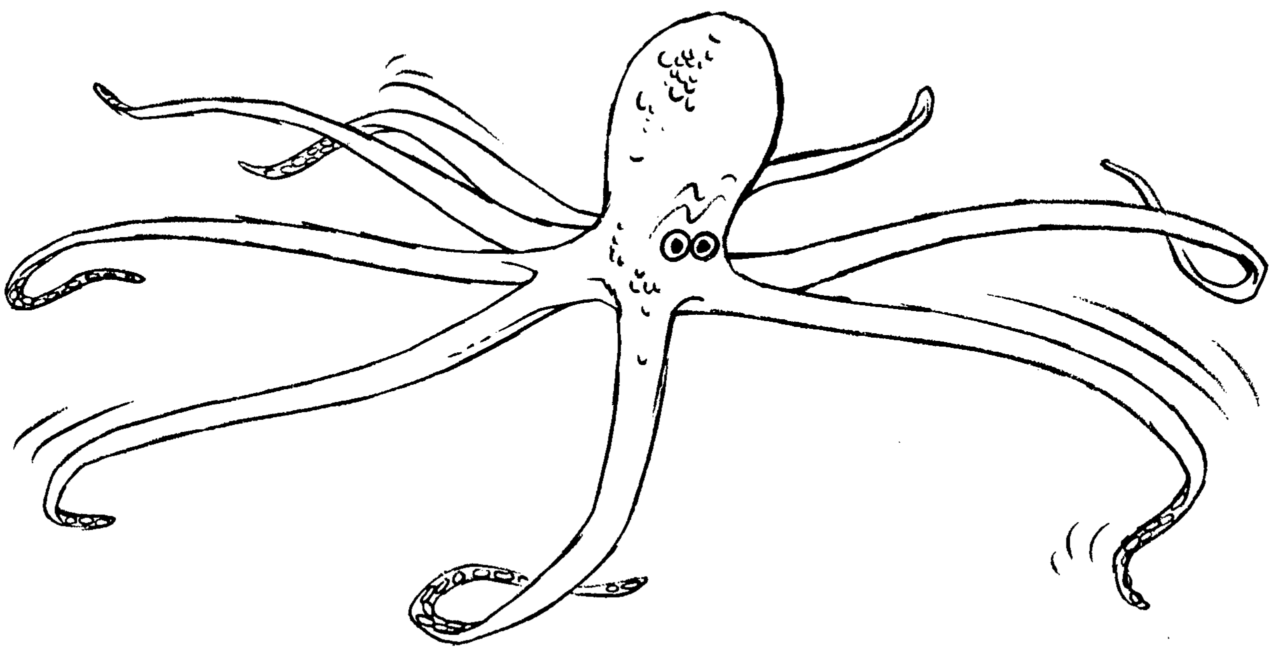
NICKS (5. 2. 2)



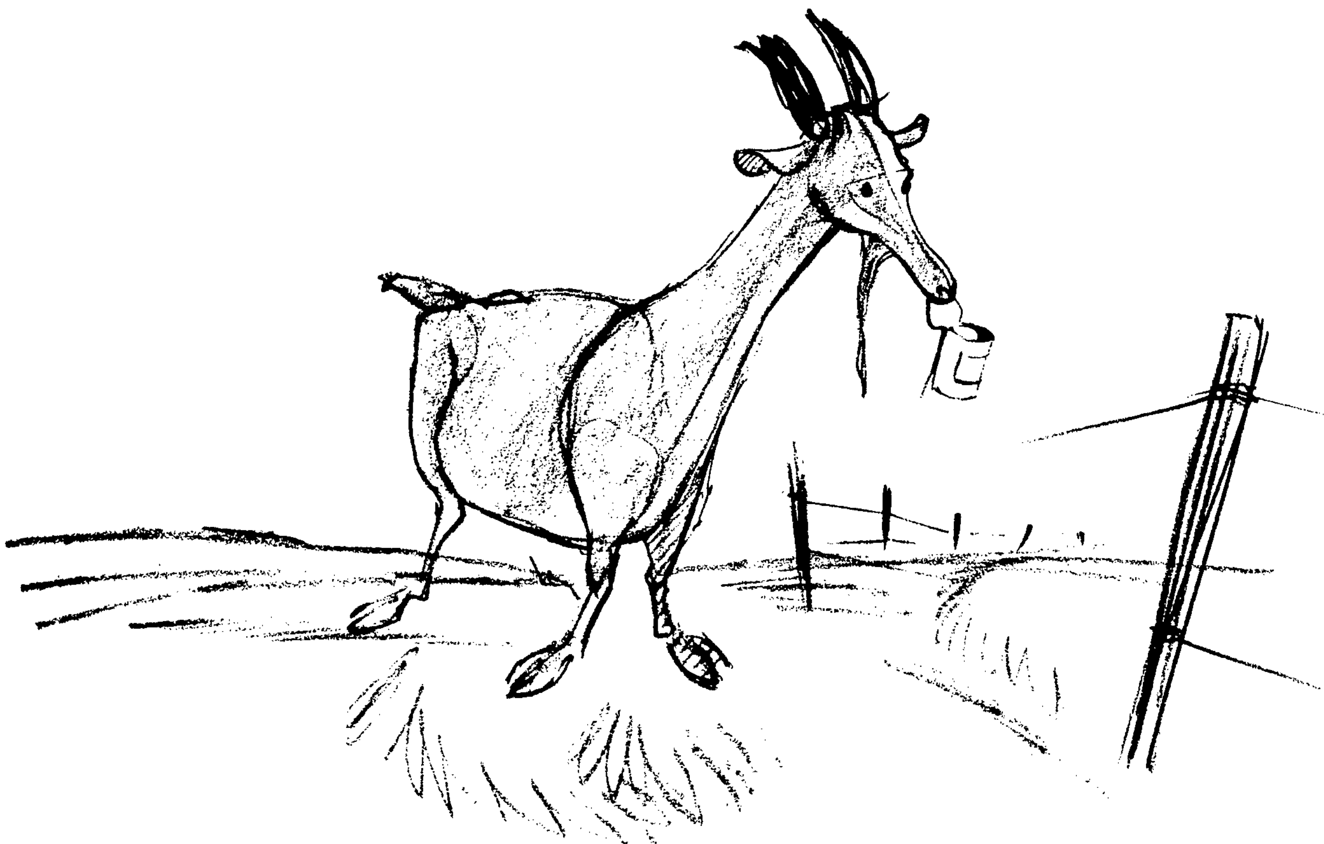
ALLOW STRETCH BETWEEN FIRST
TWO ROWS OF FASTENERS (2.7.3)



SHARP EDGES (5.3)



RAPID CHANGE IN SECTION (5. 5. 3)



CHEWED UP SURFACES (5.8.1)

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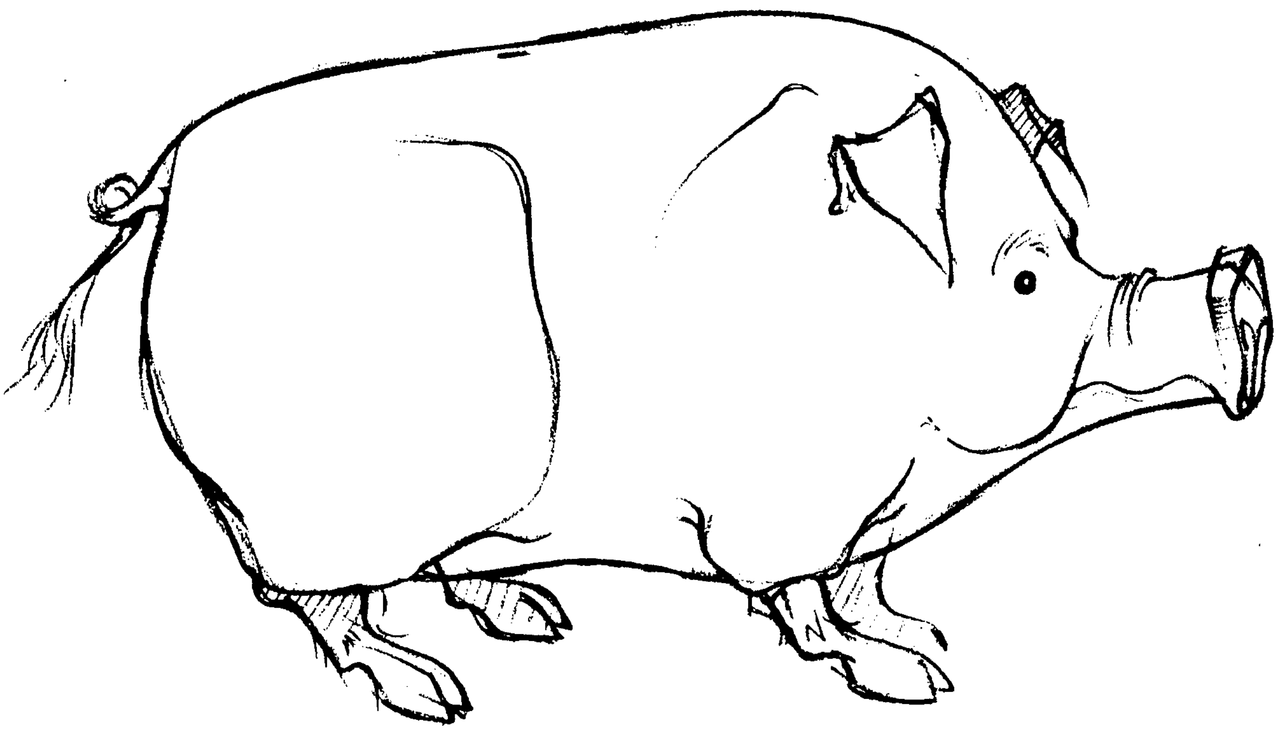
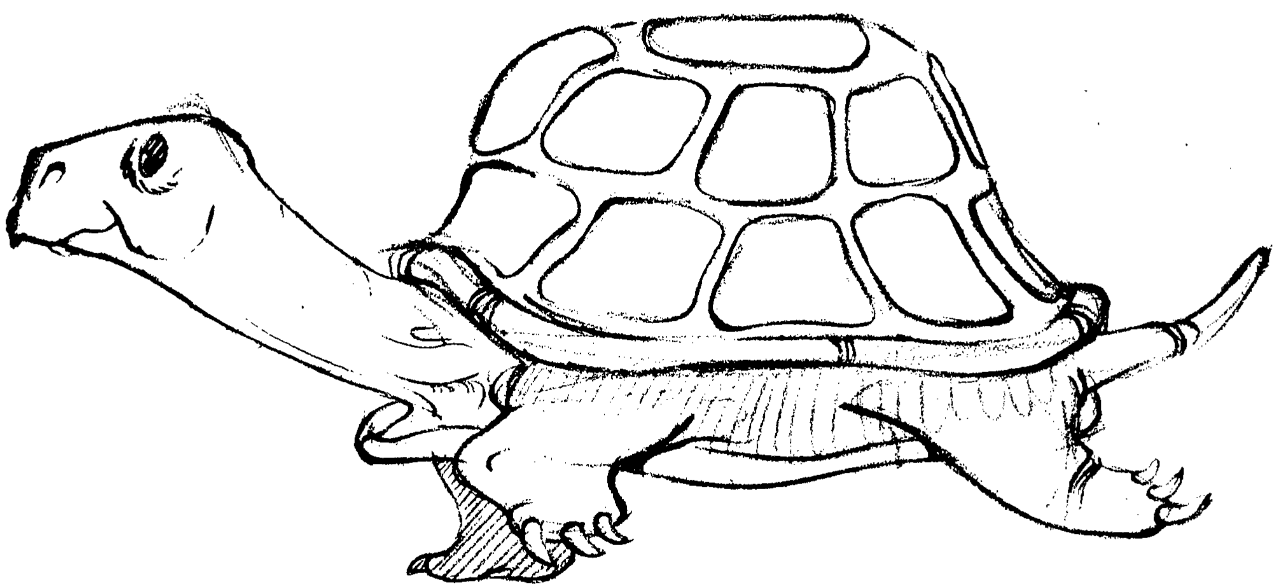


FAT GOBS OF EXCESS MATERIAL (6.3.2)

LONG SPANS OF UNSUPPORTED STRUCTURE

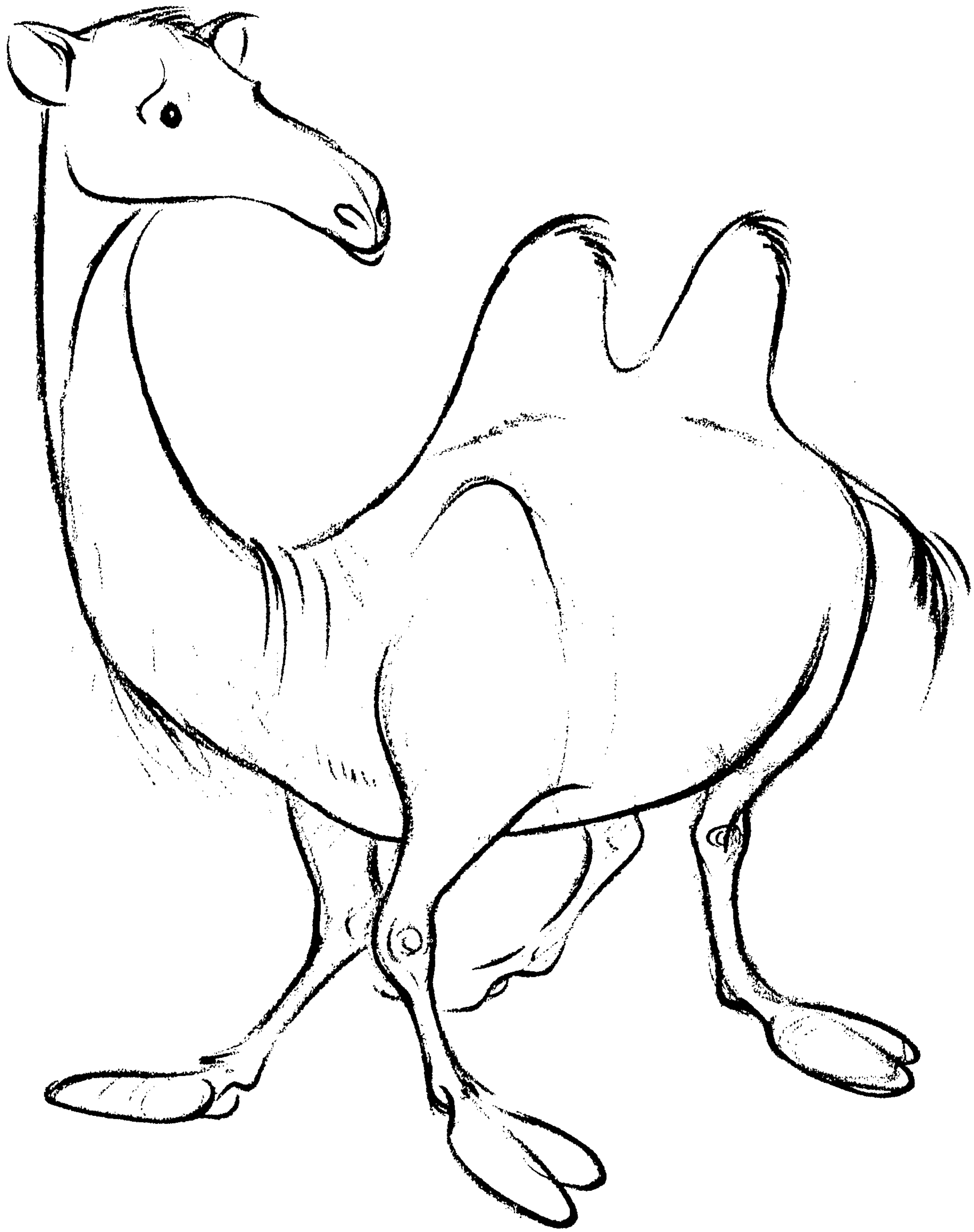
DANGLING MEMBERS

HIDDEN MISTAKES



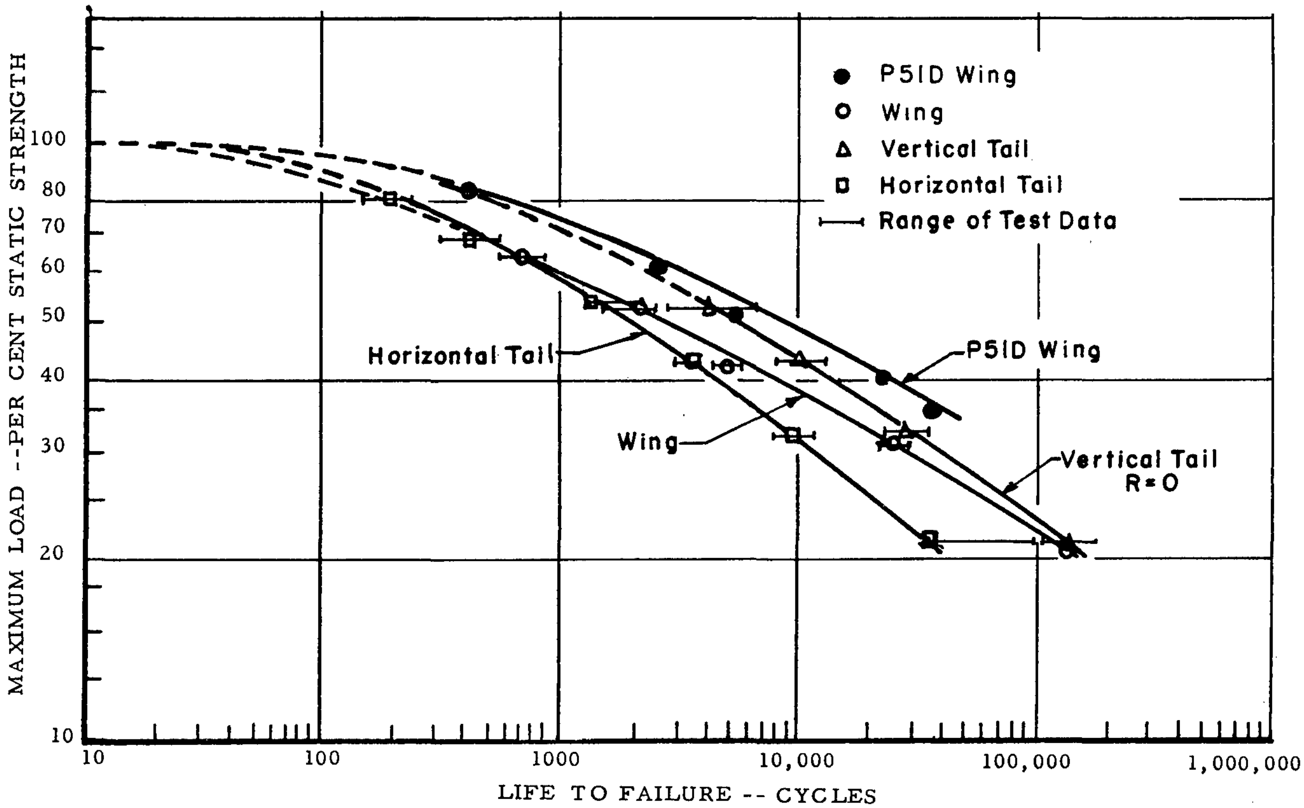
HARD PLATING (5.7)

TROUBLESOME SPOTFACE (5.3.2)



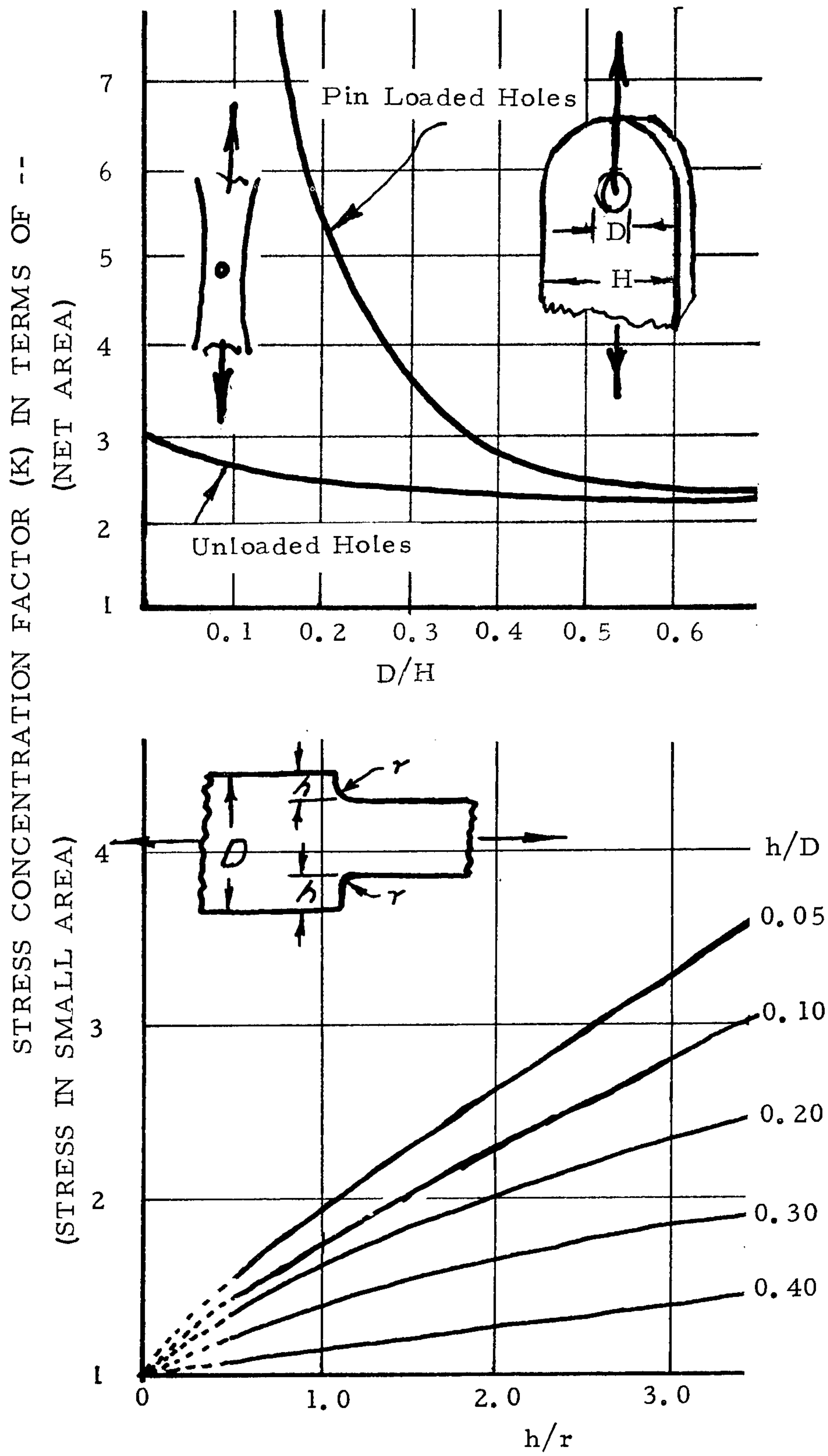
EXTRA HUMPS (6.3.2)

APPENDIX A--FATIGUE LIFE OF TYPICAL STRUCTURES



NOTE: Load values are given in per cent of static failing loads. Note that the vertical and horizontal scales are both compressed. Except for the vertical tail, where R = 0, cycling was from a load representing 1 G to the maximum value indicated.

APPENDIX B -- STRESS CONCENTRATIONS



APPENDIX C—SUGGESTED READING

Following is a list of references on fatigue for those who wish to pursue the subject further. While there are many other valuable references, those listed represent a cross section of the thinking during the last decade.

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- STP 91, Manual on Fatigue Testing
- STP 203, Fatigue on Aircraft Structures, 1957
- STP 237, Symposium on Basic Mechanism of Fatigue, 1959
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Forrest, P. G. Fatigue of Metals, Pergamon Press, New York, 1962

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