NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 4324

METHOD FOR DETERMINING THE NEED TO REWORK OR REPLACE COMPRESSOR ROTOR BLADES DAMAGED BY FOREIGN OBJECTS

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Washington
September 1958

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METHOD FOR DETERMINING THE NEED TO REWORK OR REPLACE COMPRESSOR ROTOR BLADES DAMAGED BY FOREIGN OBJECTS

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SUMMARY

A method is presented for deciding whether jet-engine compressor blades that have been nicked in service are safe from fatigue failure with continued engine operation. This method involves determining the vibratory stress distribution of the blades near the leading and trailing edges, and fatigue testing a number of blades with damage inflicted in the laboratory. A curve of limit vibratory stress (highest vibratory stress for safe fatigue operation) against damage location is charted. The maximum-vibratory-stress level of the blade in the engine is also shown on this chart and is compared with the limit vibratory stresses. From this curve the maximum safe vibratory stress, or the stress at which no nick-damaged blade would be in danger of failure by fatigue, can be determined for these blades.

Predicted limit curves for a blade from a specific compressor rotor were checked against the results of fatigue testing many of these blades with nicks on the leading and trailing edges over a wide range of span locations. In general, the fatigue test data were found to be on the conservative side of the predicted limit curves with the worst fatigue data close to these curves. The method was thus judged to be a realistic criterion for inspecting damaged blades.

In addition, some ways of reworking blades with edge nicks were studied. For the particular blades used in this investigation, the increase in the fatigue strength of the damaged blades due to reworking was found to be appreciable.

INTRODUCTION

The primary purpose of this investigation was to evolve a criterion for judging when a jet engine must be torn down for compressor blade replacement because of blade life impairment resulting from foreign-object damage. Indecision arises when the blades in a jet engine have suffered slight damage or when only a few blades in the engine have been damaged. In the absence of factual data, the natural tendency in drafting inspection specifications is to be cautious and reject nearly all damaged blades when in doubt.

As evidence that this lack of data results in more blade replacement and reworking than is absolutely necessary are the many engines where blade damage is first discovered at normal overhaul; some of the blades in these engines have undoubtedly operated many hours after sustaining the damage. A good basis for inspecting blades could provide substantial savings of money spent every year for overhauling engines and rejecting damaged blades that either represented no danger to the aircraft or could have been reworked to remove the deleterious effect without complete disassembly of the engine. Part of this cost is due to the fact that many axial-flow engines are so constructed that special presses are required to disassemble and reassemble the compressor and that these presses are usually found at only a few major overhaul bases in the country.

If the safety of the crew and aircraft is to be weighed intelligently against the cost of rejecting or reworking damaged blades, there must be more accurate information available than that included in the present inspection specifications. This information should give recognition to the fatigue strength, nick location, and notch sensitivity of the blade material and to the vibratory stress level of the blades during engine operation. Since most compressor blade materials are relatively ductile, nicks would have no stress-raising effect on the steady-state stresses and, if no vibrations were present, would not adversely affect the lives of the blades. Even moderate vibrations might not be dangerous to damaged blades if the endurance limit of the blade material is high and the notch sensitivity is low.

The investigation reported herein was designed to devise an economical method by which the fatigue strengths of nicked blades could be determined and compared with the maximum-vibratory-stress level measured for the blades in the engine. Such a method would take into account the location, size, and stress-concentration effect of the damage, and the fatigue strength of the material. Experimental work included strain-gage measurements of the vibratory stress distribution in the blades, laboratory infliction of nick damage on blade edges, and fatigue testing of these blades. On the basis of the experimental results, a limit-vibratory-stress chart for the damaged blades was constructed. As a corollary of this work, some means of reworking nicked blades were studied.

The method of predicting damaged blade strength was verified by fatigue testing first-stage compressor rotor blades from a widely used axial-flow engine.

DEFINITIONS

Specific terms used in this report are defined as follows:

Critical-edge location	location of maximum vibratory stress along edge of an undamaged blade.
Critical point	location of maximum vibratory stress in an undamaged blade.
Limit stress	maximum-vibratory-stress level at which the blade can be run without failing at any given damage location for a desired number of cycles of vibration. This stress will always be measured at the critical point for present report.
Local stress	peak vibratory stress at damage location.
Nominal stress	vibratory stress at a damage location that would exist if damage were not present.
Safe vibratory stress	maximum vibratory stress at which a blade will not fail at a given number of cycles, regardless of where it is damaged; also the limit stress for the worst damage location. This stress will always be measured at the critical point for present report.
Stress concentration factor	local divided by nominal stress.

BASIS OF METHOD

At first consideration, the quantitative determination of the effect of blade damage might appear to involve a large amount of testing, since the type, shape, and location of the damage should be included. Fortunately, reasonable assumptions can be made to reduce enormously the amount of testing involved and still produce satisfactory evaluations with respect to engineering.

A first simplification is to limit the study to nicks that occur only on the leading or trailing edges. Reference 1 reports that any damage on the pressure surfaces away from the edges did not adversely affect the fatigue life. This fact is primarily due to the lower vibratory stresses on the pressure face, which is closer to the neutral axis for bending vibrations (because of the blade camber) than are the edges or high camber region of the suction face. Damage on the suction surfaces is extremely rare. Dents are not considered herein because, although they do affect the fatigue strength, it was found in reference 1 that dents can be straightened and most of the lost fatigue strength restored.

A second simplification is to select a standarized nick for evaluation of damage. While the type of nicks encountered in practice vary considerably in shape, depth, and jaggedness, the previous investigation (ref. 1) has indicated that these factors are of secondary importance in affecting the fatigue strength of the blade once a reasonable size of nick has been inflicted on the blade. For example, the stress-concentration factor due to the nicks was found to vary little for depths ranging from 0.030 to 0.100 inch. It is not practical to study large nicks, since the impact energy needed to produce them is likely to cause instantaneous failure of the blade in the engine, or at the least to deform the blade to such an extent that it must be replaced because of aerodynamic considerations. This investigation was confined to small nicks up to a 1/16-inch depth, since these nicks are the most common type of foreign-object damage to blades.

A third simplification arises because a nick of a given shape will produce the same stress concentration at a particular stress, regardless of the location along the leading or trailing edge. In most compressor blades, there will be a size effect due to the taper, but this effect should be small. Thus, if the damaging effect of a nick is evaluated at one point along the leading edge, its effect at another point can be determined from a consideration of the nominal stresses at the two points when the blade is vibrating at a given tip amplitude. In practice, instead of measuring the tip deflection, the stress at the critical point necessary to attain this tip deflection will be measured. An integral part of the program, therefore, involves an experimental determination of the nominal-vibratory-stress distribution along the leading and trailing edges and the relating of these stresses to the critical point.

On the basis of the foregoing assumptions, it follows that a relatively small amount of experimental evaluation of the effect of nicks in a single location, coupled with an analytical extension of these results, can completely predict the effects on blade fatigue life of nicks at any edge location. First, however, a criterion must be determined for telling whether or not a blade is damaged by the presence of a nick. In most compressor blades, the normal point of failure of freely vibrating blades without nicks is on the suction surface at a point of maximum camber. Thus, while a nick at this location would always be deleterious to the life of the blade, nicks that occur in other locations would not be damaging if the blade continued to fail at the critical point. Only if the stress concentration of the nick causes the local vibratory stress to exceed the vibratory stress needed for failure at the critical point in an undamaged blade under the same exciting force would failure result at the nick and thereby damage the blade. If the nick is in such a location that, even with its stress concentration, the resulting local stress under vibration is still lower than that at the critical point, there is no need to reject the blade because of the presence of the nick.

The basis used in this report for evaluating the damage effect of nicks is a chart that essentially differentiates the regions of the blade where nicks can cause premature failure from those regions in which the presence of nicks would not affect the blade life. For most applications the charts will be designed for the blade to withstand an infinite number of cycles of vibration, since in the majority of engines, particularly for manned aircraft, the blade will attain 10^7 cycles before the first overhaul. For the blades studied in the present report, only an infinite blade life was considered in constructing the charts. The curves obtained thus far are independent of the vibratory stresses at which the blades operate in the engine. The vibratory conditions in the engine now are compared with the damage curves from the laboratory fatigue tests and, on the basis of the comparison, the decision to accept or reject the blades is made. The fatigue strength of the blades can sometimes be increased by reworking the nicks; for these cases new curves can be constructed by the same procedure. Usually the most critical mode of vibration in compressors is first bending, and the blades used in this investigation were vibrated in this mode. It is possible that in some cases the mode in which a blade has its most critical vibrations in an engine is other than first bending. This will not affect the method reported herein as long as the blades are vibrated under the same conditions in the laboratory.

No tensile loads simulating the steady-state stresses in the engine were added in the vibration tests. At the speeds at which the maximum vibrations normally occur in engines, the steady-state stresses are low. The effect of the steady-state stresses on the fatigue strength is taken into account by means of a Goodman diagram (ref. 2).

USE OF LIMIT CHARTS

The final result of this method is a chart showing the maximum allowable vibratory stresses at various span locations along the leading and trailing edges for a range of nick depth. As an example, a chart is shown in figure 1 with two limit curves. Curve 1 is for blades nicked at the edges; curve 2 is for the same nicks reworked. The horizontal lines represent hypothetical maximum-vibratory-stress levels measured at the critical point of the airfoils in a vibration survey of the engine conducted under the most severe engine operating condition. These stresses should be based on the most severe of the vibratory surveys of four or five engines of the same model, since there is likely to be some variation among engines. The vibration surveys could be run either by the engine manufacturer or the major overhaul bases.

The use of the limit charts can be illustrated from figure 1. If A is the vibratory stress level that the blades reach in engine operation, an inspector could pass safely all the damaged blades, since the line is below both curve 1 and 2. For vibratory stress level B, he would require reworking of all blades with damage in the range between locations v and w on curve 1 and would pass all others without reworking. For vibratory stress level C, he would reject all blades damaged between locations x and y, since the rework limit curve falls below the line in this region. Between y and z the damage can be reworked, and beyond z the damaged blades should be accepted without reworking. The charts can be further refined by having two sets of limit curves: one for normal long time engine operation and the other for short time operation under certain emergency conditions such as in a combat zone where, in the interests of maximum utilization, higher blade stresses for a short period of time can be tolerated.

PROCEDURE FOR OBTAINING LIMIT CURVES

The step-by-step procedure for obtaining limit curves is described as follows:

(1) Find the location of the maximum vibratory stress in the undamaged airfoil by vibrating several blades to failure. The maximum stress or failure location will be known as the critical point throughout this report and, except for the nominal and local stresses at the nick, all stress levels will refer to this point. Any other point could actually be used as a reference, but the critical point was chosen since it is the most informative to the engineer.

(2) Mount a strain gage on another undamaged blade at the critical point and measure the stresses for different tip amplitudes of vibration. Plot a curve of tip deflection against vibratory stress. From this curve the vibratory stress level for any other blade can be closely approximated merely by measuring the blade tip deflection.

(3) Mount a series of strain gages spanwise, as shown in figure 2, on both the pressure and suction faces as close to the leading and trailing edges as possible. Measure the vibratory stress distribution in the spanwise direction at the edges for any tip deflection. These edge stresses will, henceforth, be called nominal stresses. The ratio of the nominal stress to the stress at the critical point is then plotted against spanwise location. This stress distribution will remain the same as long

as the stress at every location is within the elastic range. There will be two curves (pressure and suction faces) for each of the edges; only the one that gives the largest stresses for each edge will be used.

(4) From the nominal stress curves of step 3, the edge location where the largest vibratory stress occurs can be determined. Nicks should be inflicted at this location to a reasonable depth (greater than 0.020 in.). These blades are then fatigue tested at various blade vibratory stress levels (as measured at the critical point), and a curve is drawn at the lower limit of the scatter in order to allow for the worst possible conditions. The fatigue limit or safe stress level, at which the blade will not fail, is the flat portion of this curve. In the present investigation the blades were fatigue tested to 10^8 cycles before tests were discontinued, if the blades had not failed in that time.

(5) Plot a chart of limit stress against location, with a line representing the maximum vibratory stress that the blade will experience in the engine drawn horizontally across the graph. The equation for the limit stress is:

$$S_{limit,x} = S_{limit,e} \frac{\left(\frac{S_{nominal}}{S_{critical}}\right)_e}{\left(\frac{S_{nominal}}{S_{critical}}\right)_x} \tag{1}$$

where

$S_{limit,x}$	limit vibratory stress (measured at critical point) for nick at any location x
$S_{limit,e}$	limit vibratory stress (measured at critical point) for nick at critical-edge location
$\left(\frac{S_{nominal}}{S_{critical}}\right)_{x \text{ or } e}$	ratio of nominal vibratory stress (at location x or at critical-edge location) to vibratory stress at critical point

This equation follows from the assumption that, if the stress-concentration factor due to the nick has been evaluated at one location for a certain nominal stress, it must be the same at any other nick location if the same nominal stress is reached. Thus, the limit stress for each point is a function of the tip deflection required to reach that nominal stress. Instead of actually measuring the tip deflection, the stress at the critical point for that deflection is being measured. The ratio of the limit stresses for the two nick locations is merely the ratio of the respective stresses at the critical point required to give the same nominal stress at each nick location. The safe stress level ($S_{limit,e}$ for an infinite blade life) is found from step 4 and

$$\left(\frac{S_{nominal}}{S_{critical}}\right)_{x \text{ or } e}$$ from step 3.

In some cases it might be desirable to have limit curves based on a finite blade life, that is, some number of cycles to failure to the left of the knee of the S-N curve plotted from the data from step 4. Equation (1) will still apply for this case except that $S_{limit,e}$, instead of being the safe stress level, will be the vibratory stress that will give the finite blade life that is required.

(6) Repeat step 4 for reworked damage. The reworking can be accomplished by filing out a generous radius to the deepest penetration of the nick with all corners in this notch rounded off. The filed-out area should be finished by polishing. The remainder of the procedure is the same; the rework limit curve should also be plotted on the limit chart.

(7) On the limit chart, plot as a horizontal line the maximum-vibratory-stress level (at the critical point) the blades undergo during engine operation, as determined from vibration surveys of the engine model. This vibratory stress level should be corrected, as explained later, for the effect of the steady-state stresses on the fatigue strength of the damaged blades. The intersection points of the horizontal line with the limit curves determine in which regions damage can be tolerated.

APPARATUS

The blades used to verify the method were designed for use in the first compressor rotor stage of the same production jet engine considered in reference 1 and were new when received. The blade material, which had an average hardness of Rockwell C-21, was not the same as that used for the blades of reference 1; the chief difference between the two materials is in the fatigue strength and notch sensitivity, both being higher for the blades used in this investigation. The S-N curve for the blade material used herein, as determined from fatigue tests of undamaged blades, is shown in figure 3.

Damage was inflicted on the blade edges in the laboratory by shearing material out to a desired depth with a "V" striking head attached to the hammer of a pendulum-type impact machine, so that the blade was struck on the face near the edge. The method using the equipment shown in figure 4 simulates very closely the actual mechanism of damage to compressor blades from foreign objects during engine service.

The blades were fatigue tested with the pneumatic exciters and recording apparatus described in reference 1. Tip deflections of the vibrating blades were measured with an optical comparator.

RESULTS AND DISCUSSION

The critical point for the undamaged blades was found to be on the suction surface at the thickest part of the cross section at a location 28.5 percent of the span distance from the base. The distribution of vibratory stresses along airfoil elements 3/16 inch from the edges is shown in figure 5 in nondimensional form, where the ratio of edge vibratory stress (nominal stress) to the vibratory stress level at the critical point is plotted against the span location. Only the pressure-face curves were used in the remainder of the investigation, since these gave the maximum nominal stresses along the edges. The highest edge stress was 64 percent of the vibratory stress level (as measured at the critical point) and occurred 28.5 percent of the spanwise distance from the base on the leading edge.

Nicks were inflicted at the critical-edge location to depths varying from 1/32 to 1/16 inch, the range of depth being the result of lack of control in impacting the blades. The results of fatigue testing these blades are shown in figure 6. In this figure, data for blades that did not fail when first fatigue tested and were subsequently retested at higher stresses are shown in solid symbols. For the sake of speeding up the fatigue testing, it is probably adequate to stop the testing at 10^7 cycles if a definite fatigue limit has been reached, instead of continuing the tests to 10^8 cycles as is normally done to establish fatigue limits. Since the natural frequency of these blades was about 200 cycles per second, only 15 hours would have been required to reach 10^7 cycles, whereas it took nearly a week to attain 10^8 cycles.

The rather large degree of scatter in figure 6 shows the need for testing a sufficient number of blades. A curve is drawn at the lower limit of the scatter. The fatigue limit of this curve, or the safe stress level, occurs at approximately ±23,000 psi. It is apparent that, if the vibratory stresses in the blades during engine operation are always kept below this allowable vibratory stress for the damaged blades, there will not be any danger of fatigue failure. The validity of the method was checked by inflicting nicks at other locations on the blades and fatigue testing them; the results were compared with the predicted limit stresses for these locations.

The limit vibratory stresses for various locations along the leading and trailing edges of the airfoil were determined from equation (1) and plotted in figure 7 against the log of cycles to failure. It can be seen that only 1 out of 68 failure data points in figure 7 falls below the limit lines, the deviation being 2 percent. The vast majority of data points fell far above the limit lines. In addition to the locations shown in figure 7, a number of blades were damaged at approximately 6.25 and 73 percent of the spanwise distance from the base at both the leading and the trailing edges. None of these blades failed in fatigue at the nicks. All failures started near or at the critical point of the undamaged blades and at combinations of stresses and cycles that fell on the S-N curve of the blade material in figure 3. The limit vibratory stress for the 6.25-percent location cannot be calculated, since the stress distribution curves of figure 5 were not carried far enough and the curves change too rapidly in this region to be extrapolated. For the 73-percent location, the limit stress would be ±63,000 and ±58,500 psi at 10^7 cycles for the leading and trailing edges, respectively. Since these blades failed effectively as undamaged blades at high stresses and low numbers of cycles, they would have actually failed at ±80,000 psi (the fatigue limit from fig. 3) in 10^7 cycles. On the basis of these fatigue results, it would appear that the limit curves are a conservative basis for inspecting blades.

The series of limit vibratory stresses calculated by means of equation (1) for various damage locations (as shown in fig. 7) can be plotted as in figure 8. Here it is assumed that the blades must undergo at least 10^7 cycles of repeated stressing, and the limit stresses are taken from figure 7 for 10^7 cycles.

In order to apply the limit chart of figure 8 to engine operating conditions, the maximum vibratory stress that the undamaged blade undergoes in the engine must also be plotted. The maximum-vibratory-stress level in these blades measured under engine operating conditions in a static test stand is reported in reference 3 as ±16,000 psi. This value, however, was measured by a strain gage on the suction surface near the airfoil base. This stress would correspond to ±23,500 psi for the maximum-vibratory-stress location in the airfoil (the critical point).

To obtain the correct point of intersection between the limit curve and the measured-vibratory-stress line, the limit vibratory stress should be corrected for the effect of the steady-state stresses by the use of a modified Goodman diagram, as shown in figure 9. The ultimate tensile strength used in the construction of the diagram was found by tensile testing a specimen cut from a compressor blade. The endurance limit used was found from figure 3. The steady-state stresses for the blade used (centrifugal and gas bending) are reported in reference 1 as 15,400 psi. The maximum working stress from figure 9 is then 84,000 psi. The allowable vibratory stress for undamaged blades not to fail is 84,000 - 15,400 psi or ±68,600 psi. This represents a 14-percent decrease at every value for the limit curves of figure 8. The same points of intersection between the limit curve and the measured-vibratory-stress line can be obtained in an easier manner; however, by applying a correction to the measured-vibratory-stress level in the engine at the maximum stress location by dividing it by 86 percent and keeping the same limit stresses. Thus, the maximum vibratory stress based on engine operation becomes ±27,300 psi.

On the basis of figure 8, for an expected vibratory stress of ±27,300 psi, all blades with edge nicks up to 1/16 inch deep, up to 50 percent of the distance from the base on the trailing edge and 52 percent on the leading edge, would be rejected. Blades with nicks at all other locations would be accepted.

The results of fatigue testing a number of blades with reworked nicks are shown in figure 10 and are compared with the fatigue tests for unworked blades that were replotted from figure 6. The reworked blades showed an increase in fatigue strength over unworked blades. Shot peening the area around the nicks proved inferior to filing the nicks to a 1/2-inch radius. For the limited amount of testing done, the safe vibratory stress for the reworked blades was roughly twice that for the blades with unworked nicks.

The limit curves for reworked blades are also shown in figure 8. The reworked-blade limit curve was based on a safe stress level of ±45,000 psi (from the limited data available) from figure 10 for the filed and polished nicks. It can be seen that all the blades with damage at the reject locations of the unworked limit curve could be reworked to make them perfectly safe for continued engine operation. No reworking need be done to damaged blades that would have been accepted on the basis of the unworked-blade limit curve, since unnecessary reworking is a waste of time, money, and manpower. The reworked-blade limit curve indicates when reworking is practicable. All reworking can be done on the fully bladed compressor without dismantling the compressor and removing the blades. If parts of the rework limit curve are below the vibratory stress line, all blades with damage in this region should be replaced. The efficiency of reworking damage probably depends largely on the blade material; the more notch-sensitive it is, the greater would be the improvement in fatigue strength.

It must be emphasized that the limit chart must be redetermined every time a change is made either to the blade shape or material. For certain nonferrous materials (such as aluminum) that generally have no sharply defined endurance limit, this method will not be applicable, since in these cases there cannot be a definite safe stress level at which either nicked or unnicked blades would never fail in fatigue. For these materials, because of the indeterminate fatigue limit, no nick should be tolerated and any damaged blade should be replaced for the cases where a long fatigue life is required. For a finite life, the alternative method at the end of step 5 in the section on procedure can be used for these blade materials.

CONCLUSIONS

A procedure has been worked out for drawing up limit charts that indicate whether a blade damaged by foreign objects during engine operation should be accepted or rejected.

1. On the basis of experimental results on compressor rotor blades, the method is judged to be a realistic criterion for inspecting damaged blades.

2. Filing out to a reasonable radius and polishing nicks in the blades restored some of the fatigue strength, although the extent of this improvement probably would vary with the blade material. Shot peening appeared to be inferior to filing as a means of reworking nicks.

3. If the maximum vibratory stress measured for the blade in the engine does not exceed the safe-vibratory-stress level, there is no danger of fatigue failure of any damaged blade, regardless of the location of the damage.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, June 19, 1958

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2. Marin, Joseph: Mechanical Properties of Materials and Design. McGraw-Hill Book Co., Inc., 1942.

3. Hanson, Morgan P., Johnson, Donald F., and Meyer, André J., Jr.: Compressor-Blade Vibration and Performance in a J47-23 Turbojet Engine Under Conditions of Rotating Stall. NACA RM E54L20a, 1955.

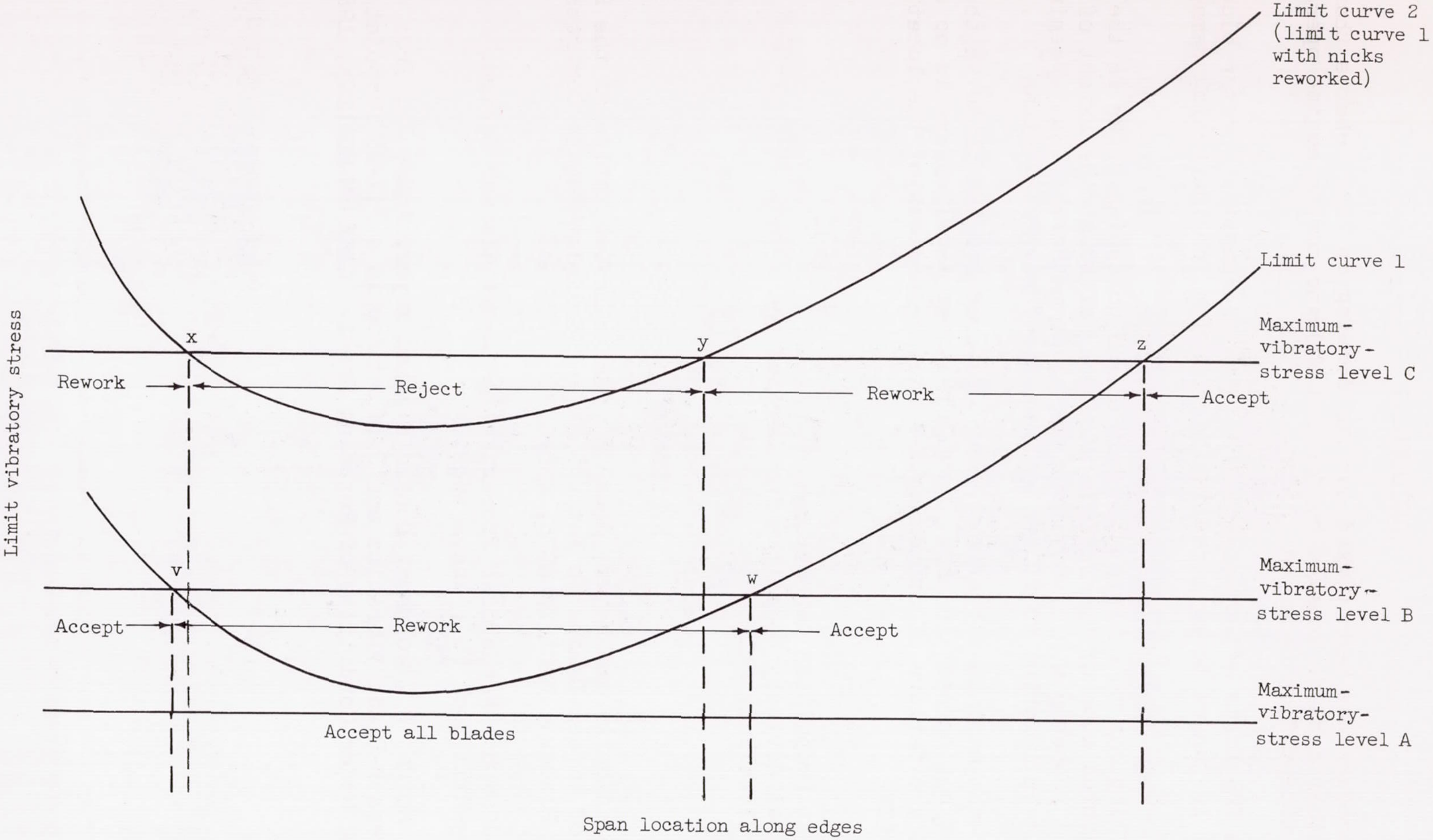


Figure 1. - Sample limit chart.

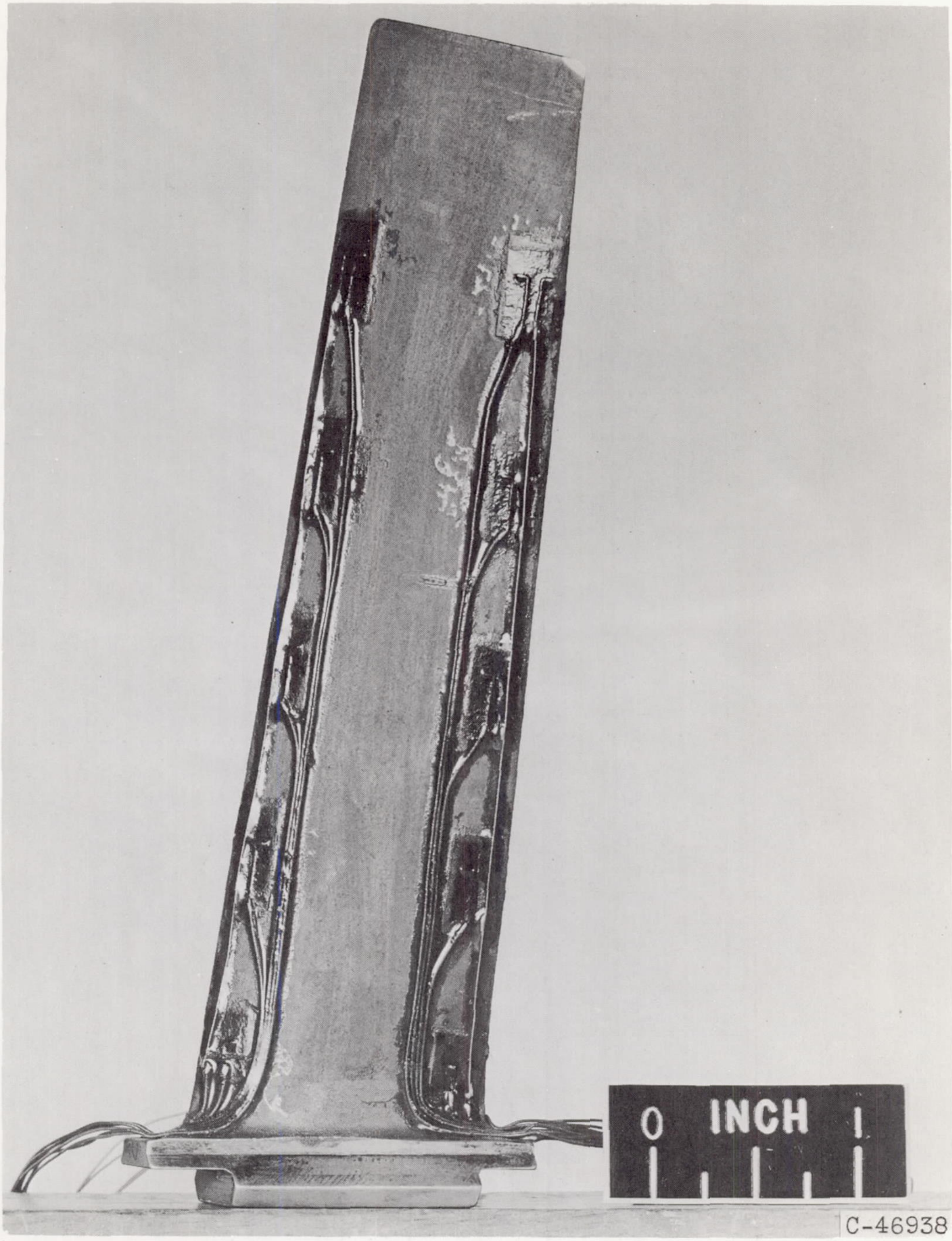


Figure 2. - Instrumentation for measurement of nominal-vibratory-stress distribution near edges.

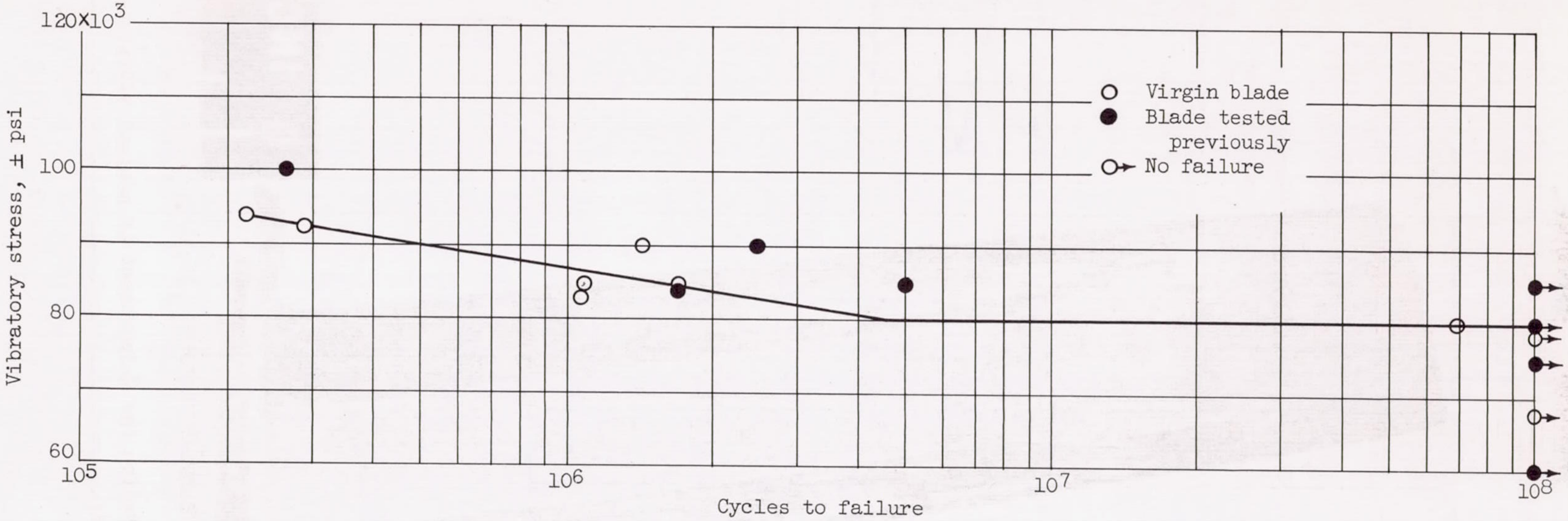


Figure 3. - S-N Curve for blade material. (All blades tested were undamaged.)

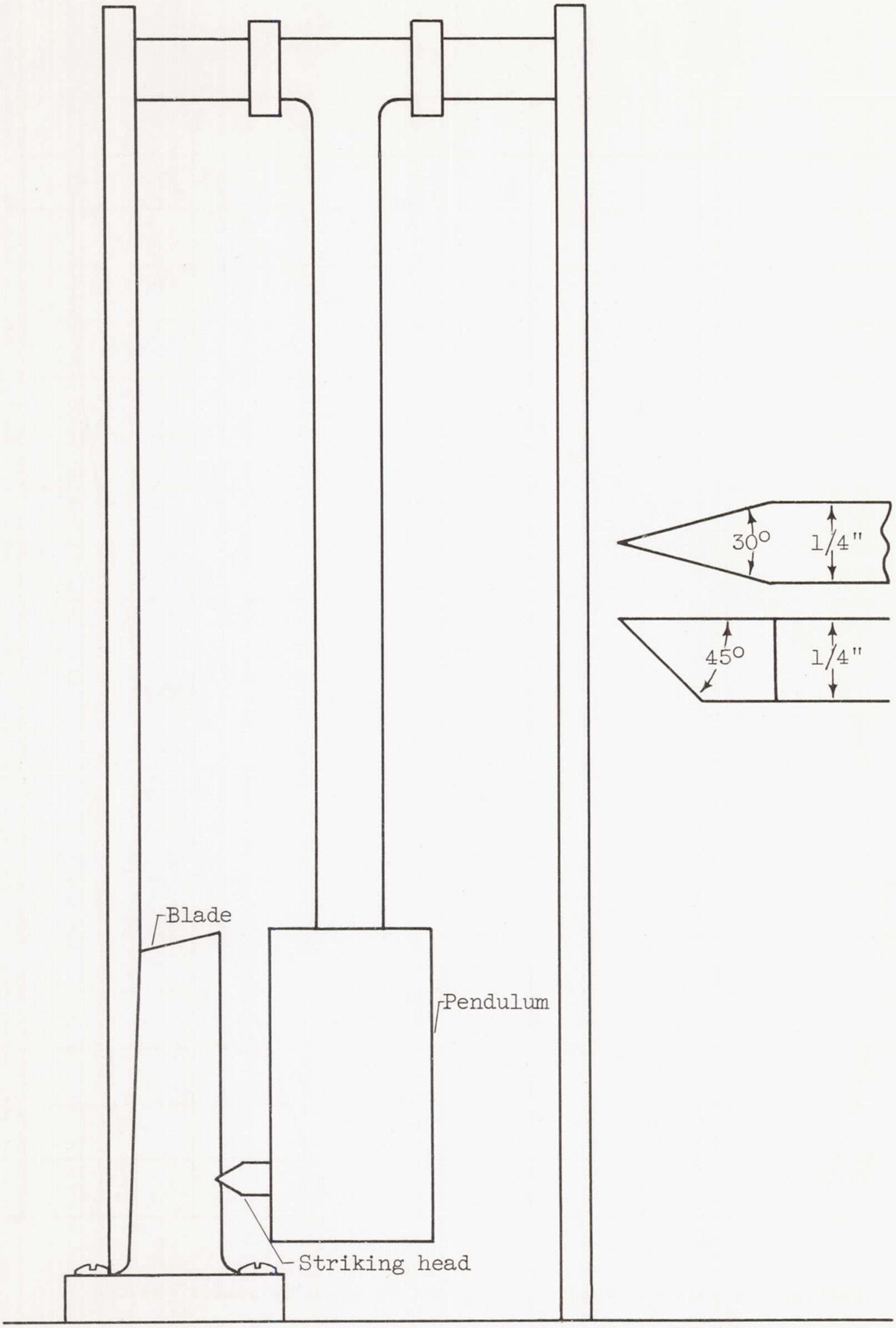


Figure 4. - Apparatus for inflicting damage on blade by shearing with "V" striking head.

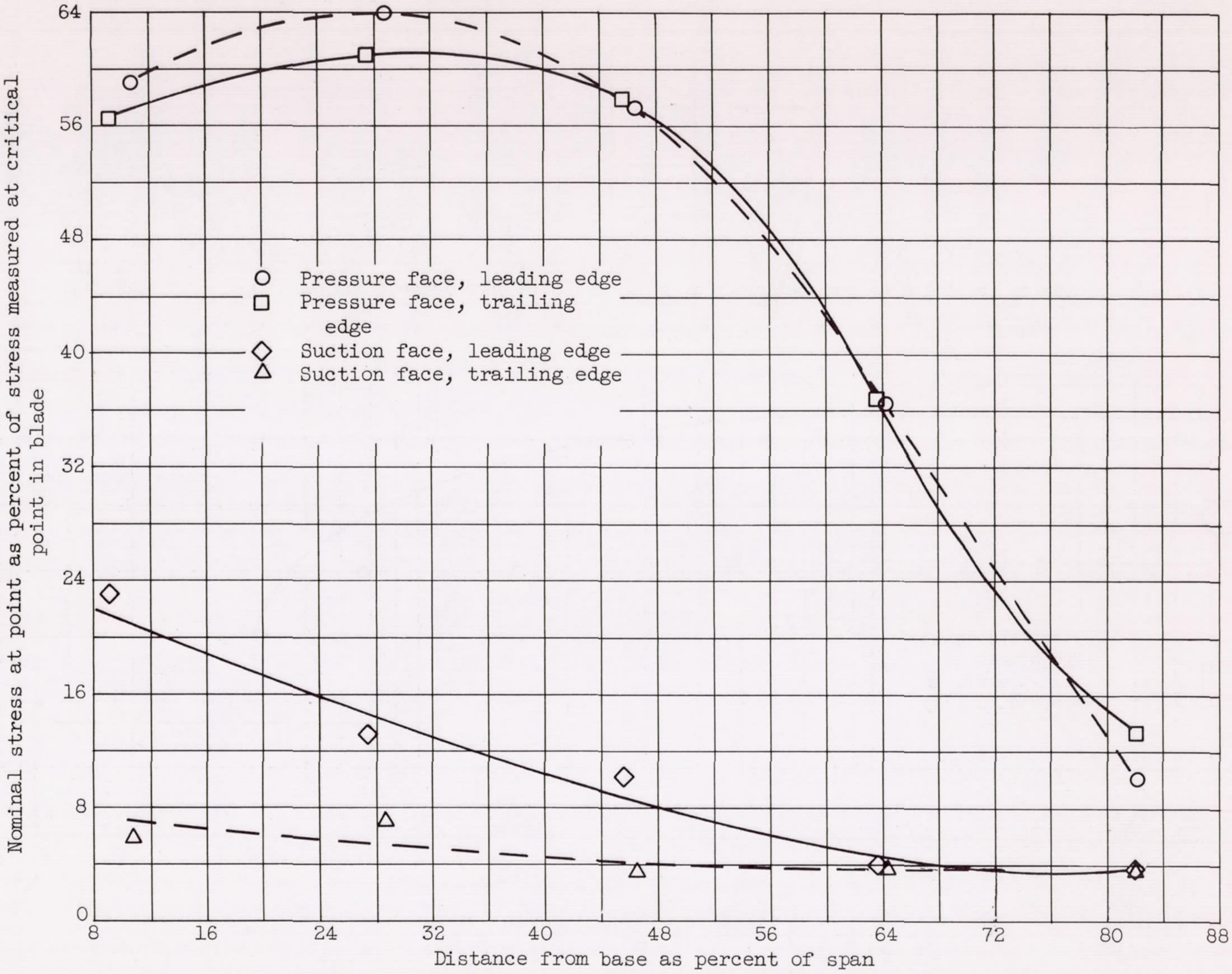


Figure 5. - Vibratory stress distribution near airfoil edges.

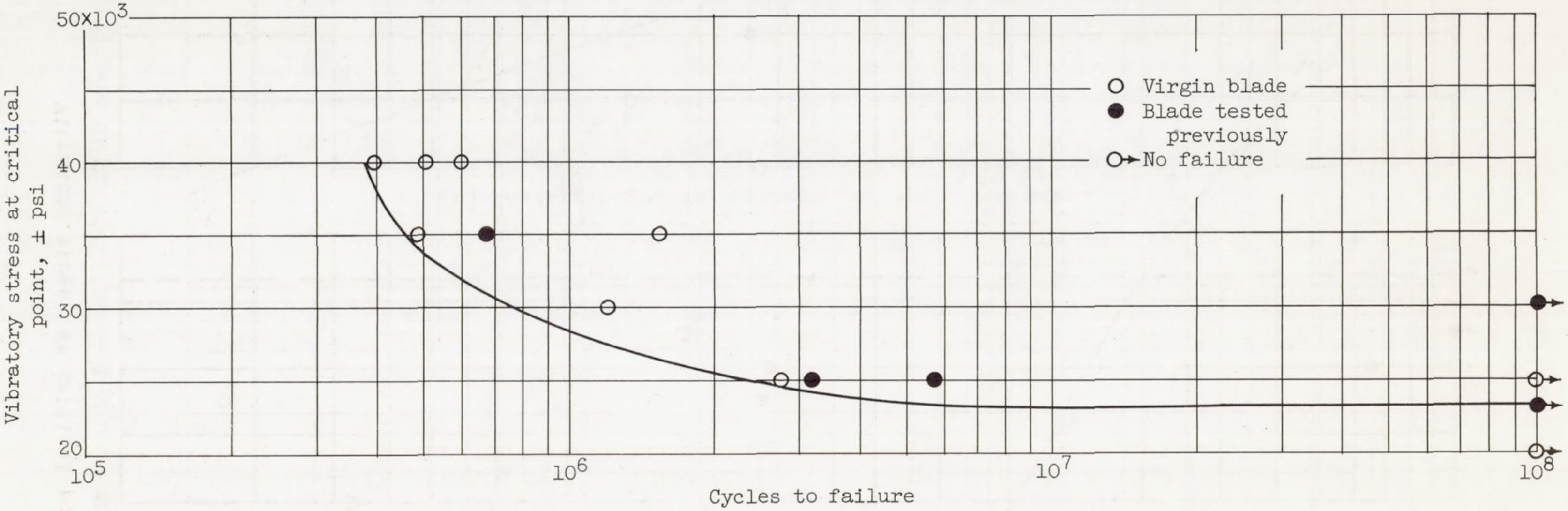
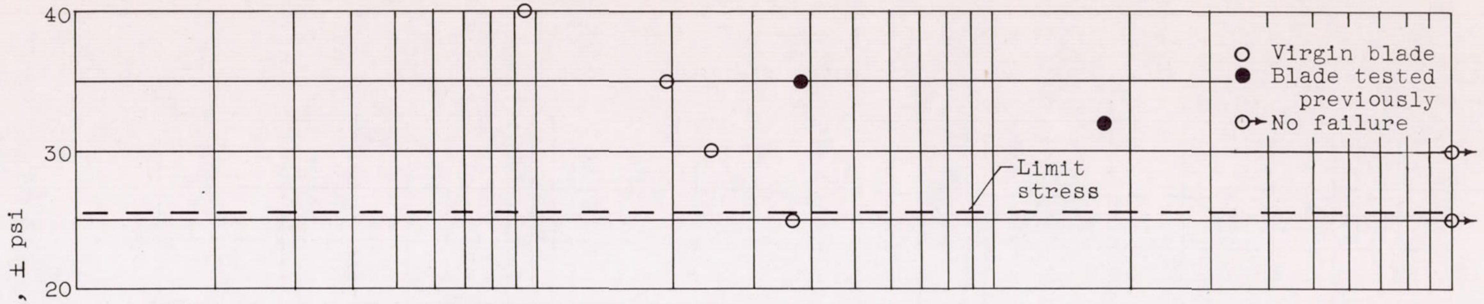
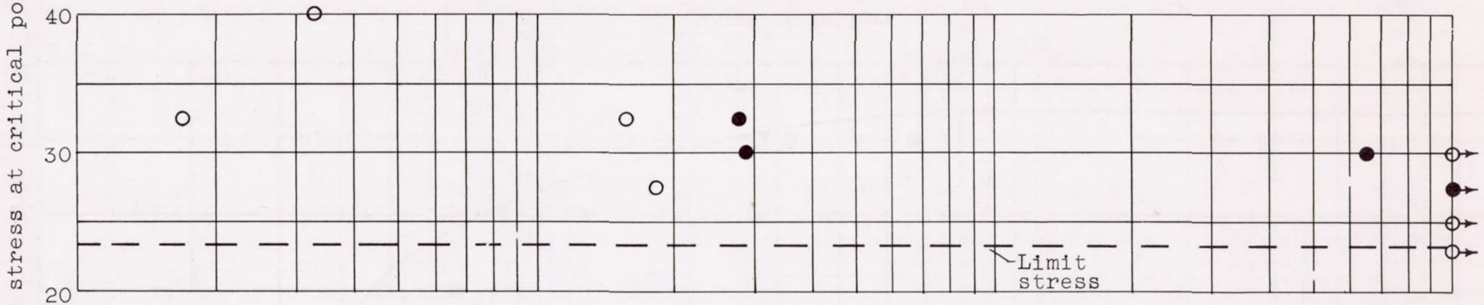


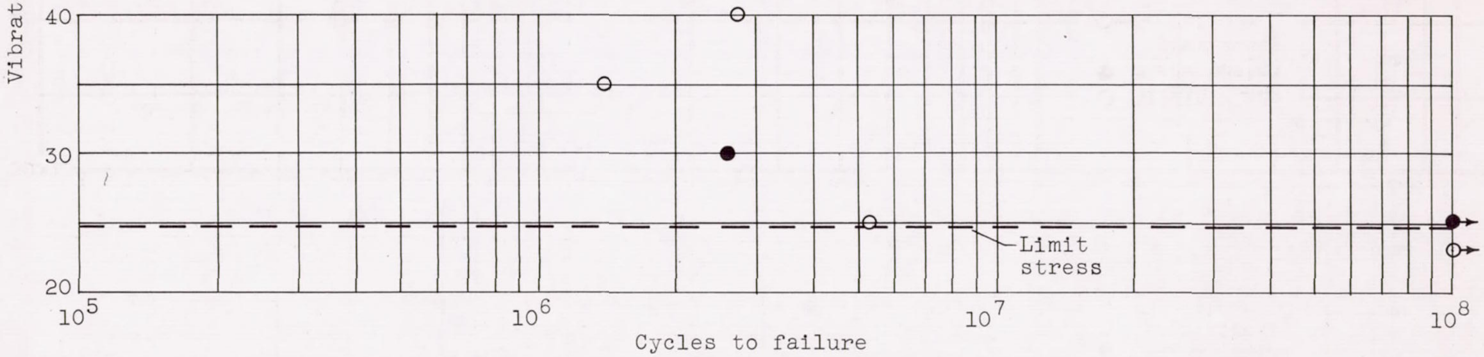
Figure 6. - S-N Curve for leading-edge nick 1/32 to 1/16 inch deep and 28.5 percent of span from base.



(a) Leading edge, 10.7 percent of span from base.

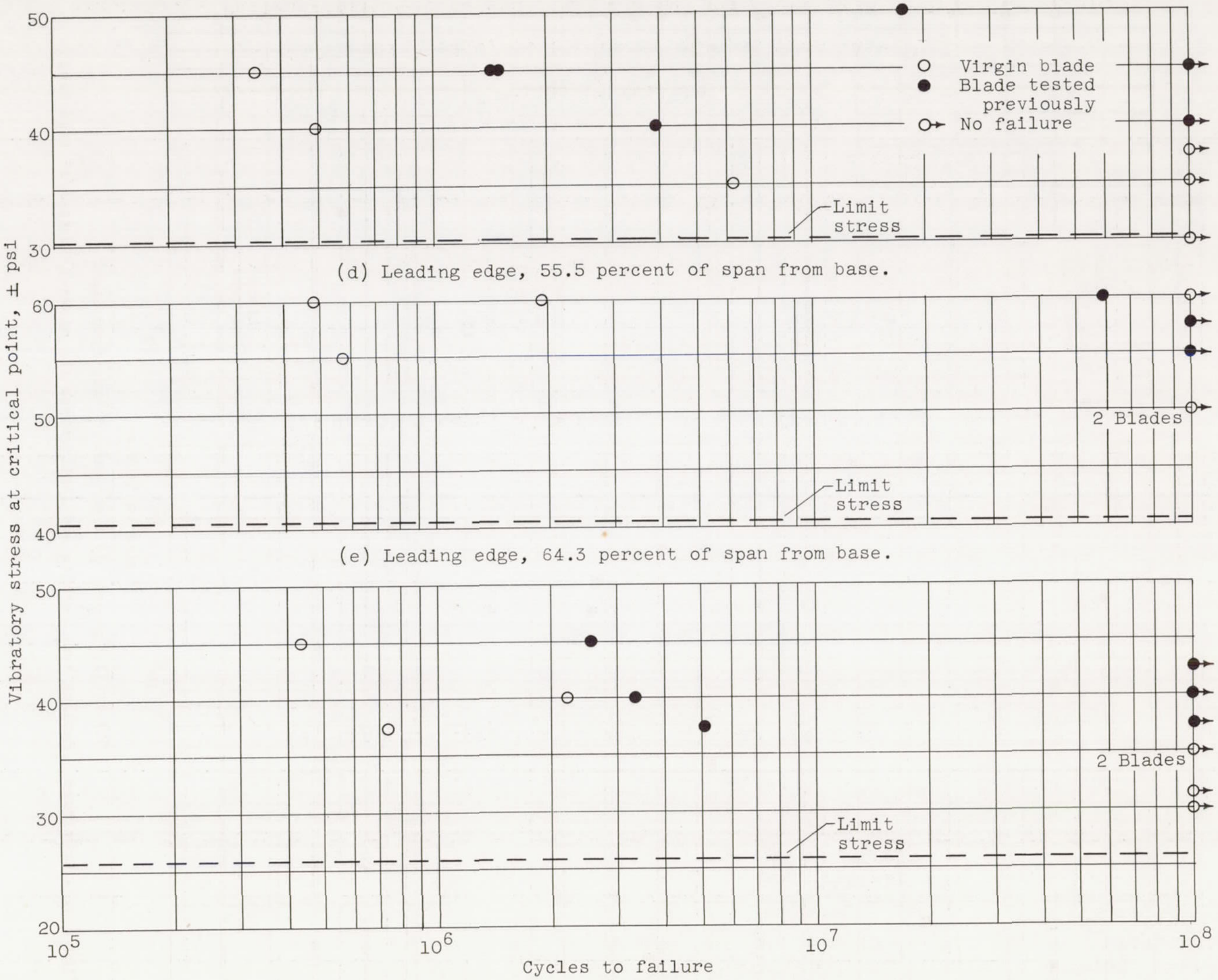


(b) Leading edge, 37.5 percent of span from base.



(c) Leading edge, 46.5 percent of span from base.

Figure 7. - Fatigue test results and limit stress for nicks 1/32 to 1/16 inch deep.

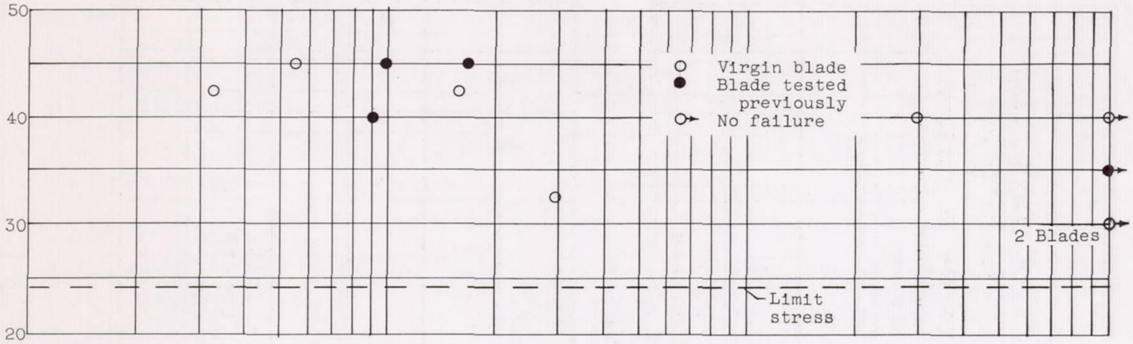


(d) Leading edge, 55.5 percent of span from base.

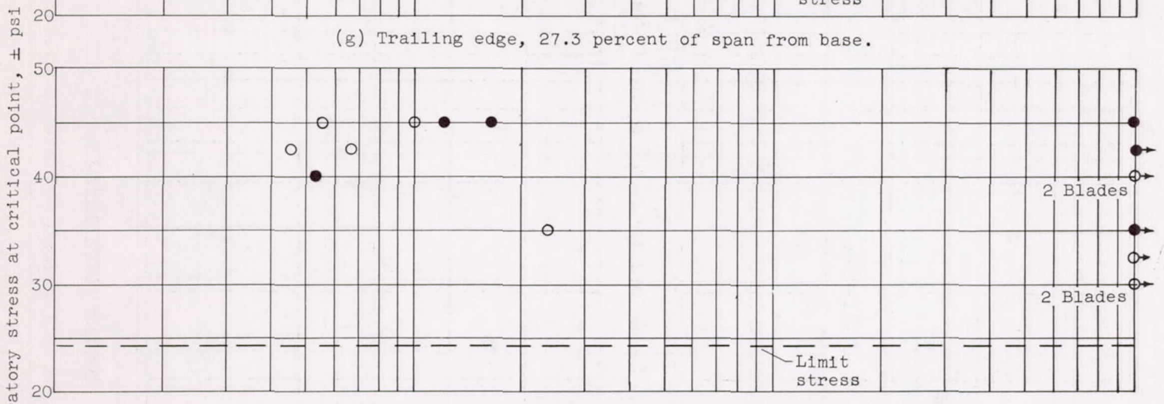
(e) Leading edge, 64.3 percent of span from base.

(f) Trailing edge, 9.1 percent of span from base.

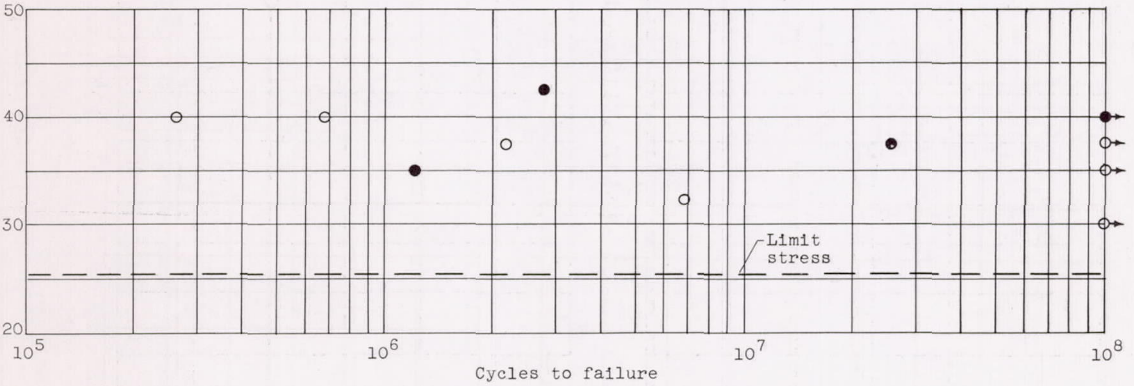
Figure 7. - Continued. Fatigue test results and limit stress for nicks 1/32 to 1/16 inch deep.



(g) Trailing edge, 27.3 percent of span from base.

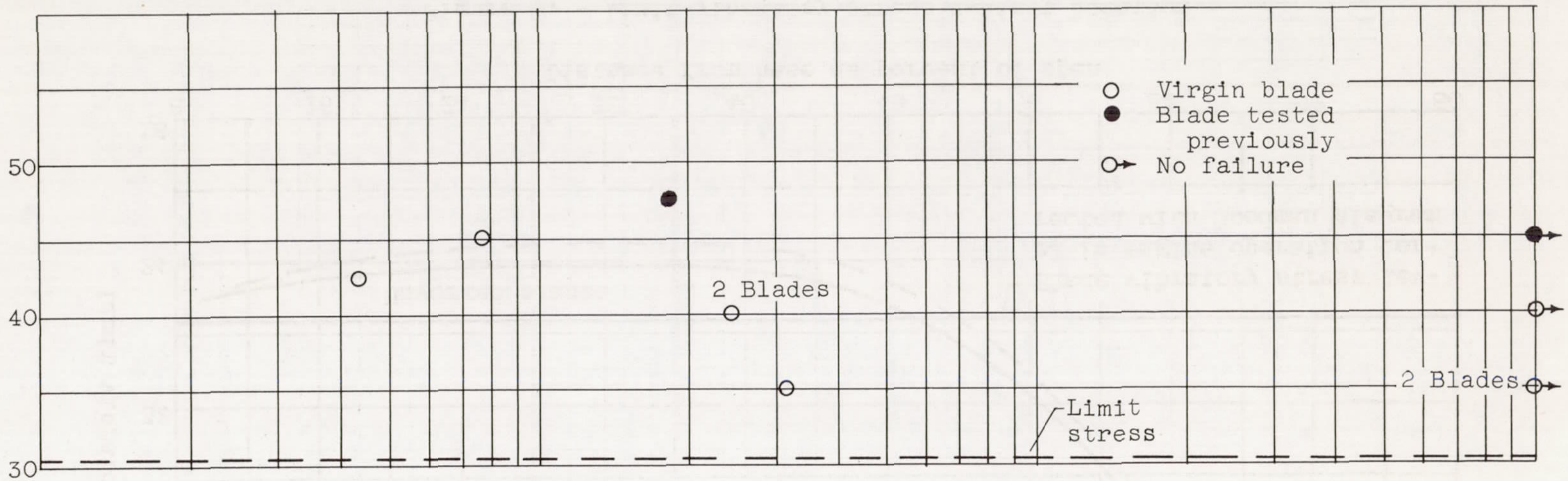


(h) Trailing edge, 36.4 percent of span from base.

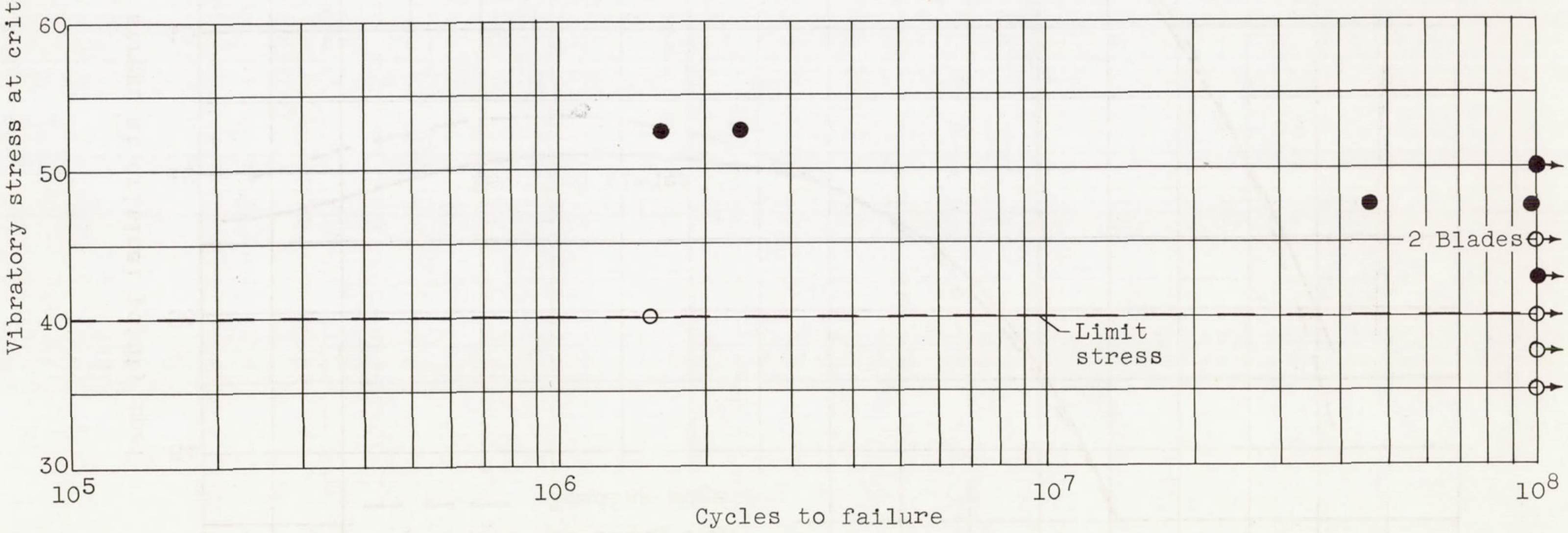


(i) Trailing edge, 45.5 percent of span from base.

Figure 7. - Continued. Fatigue test results and limit stress for nicks 1/32 to 1/16 inch deep.



(j) Trailing edge, 54.5 percent of span from base.



(k) Trailing edge, 63.6 percent of span from base.

Figure 7. - Concluded. Fatigue test results and limit stress for nicks 1/32 to 1/16 inch deep.

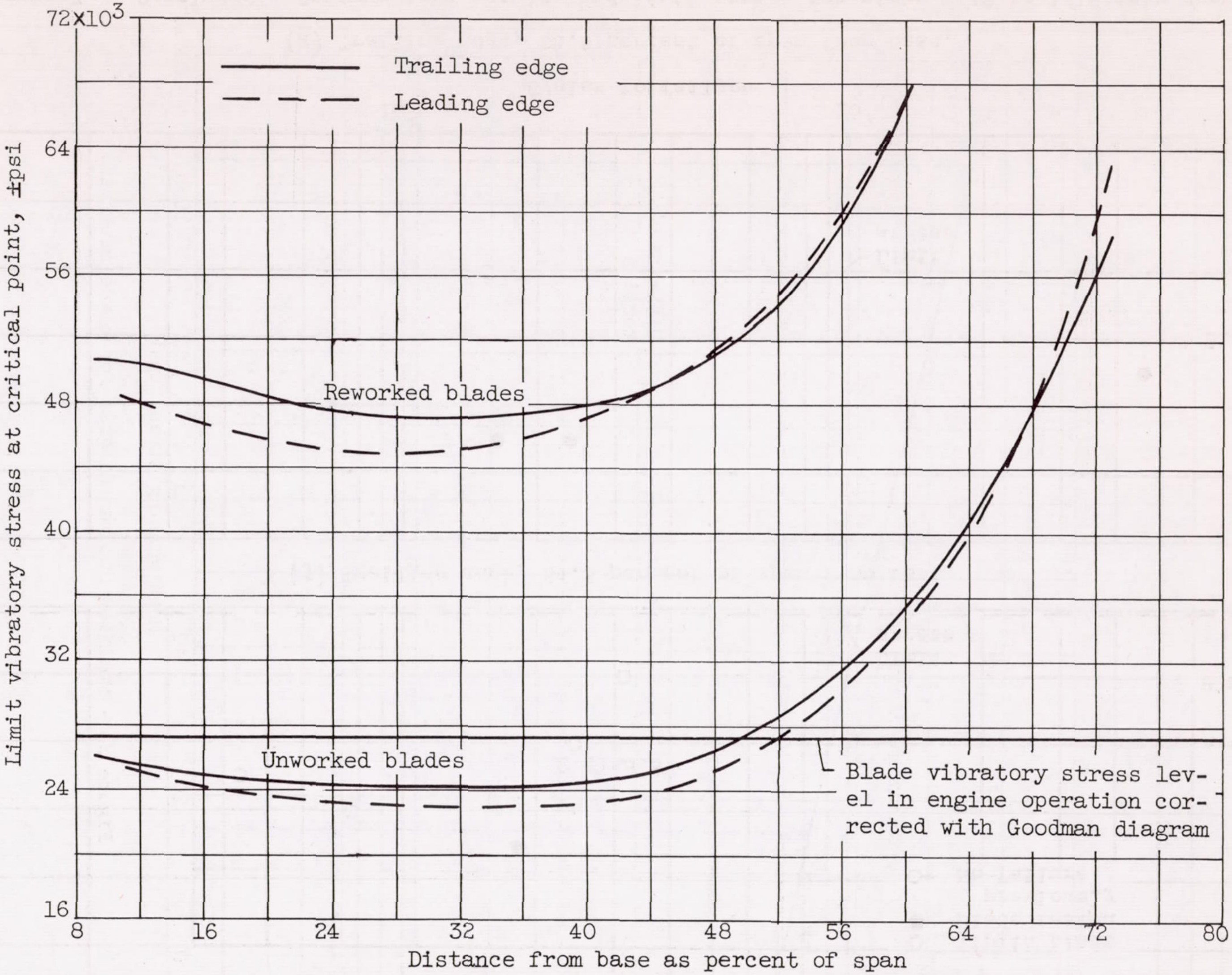


Figure 8. - Limit vibratory stress against location.

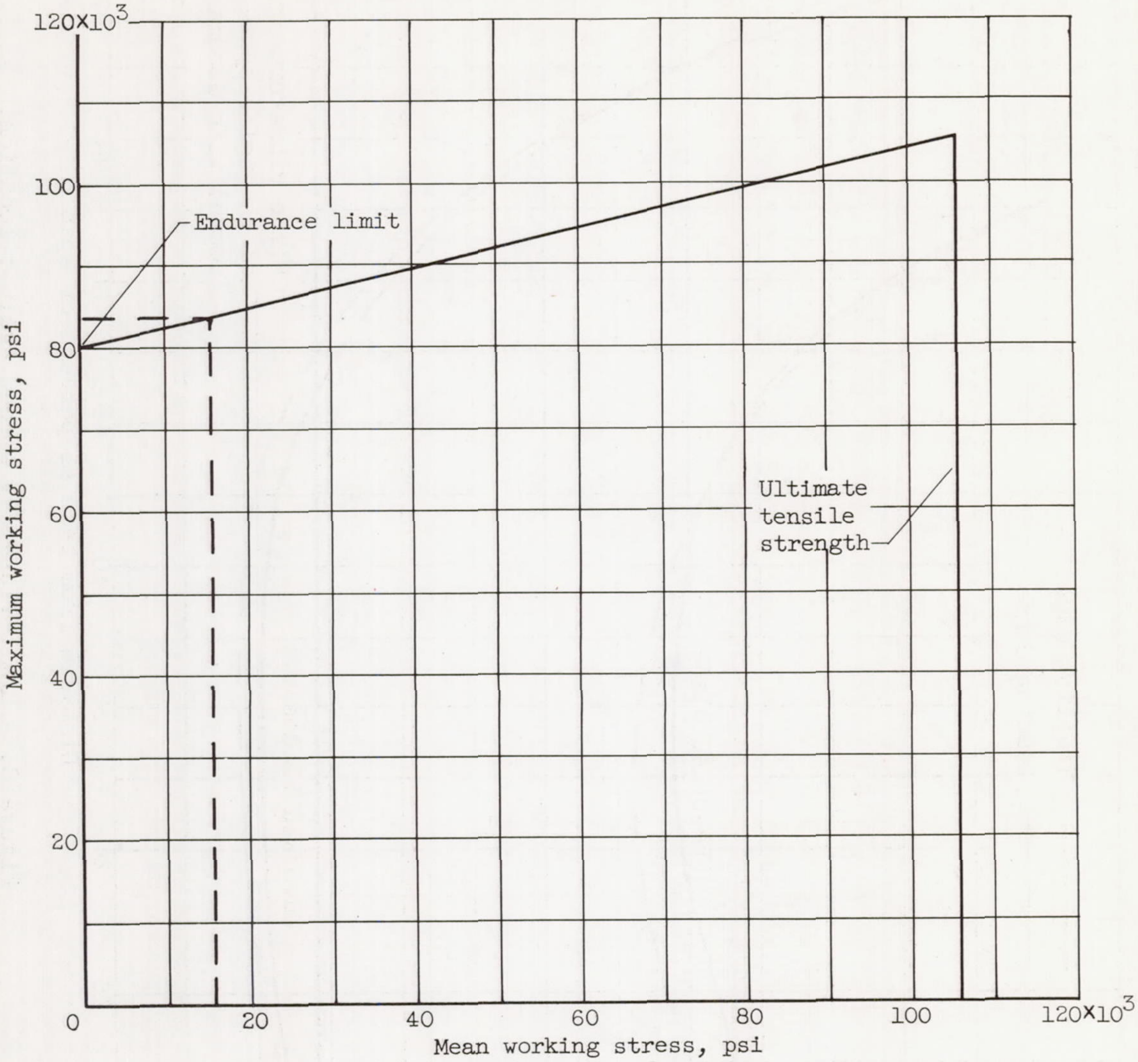


Figure 9. - Modified Goodman diagram for blade material.

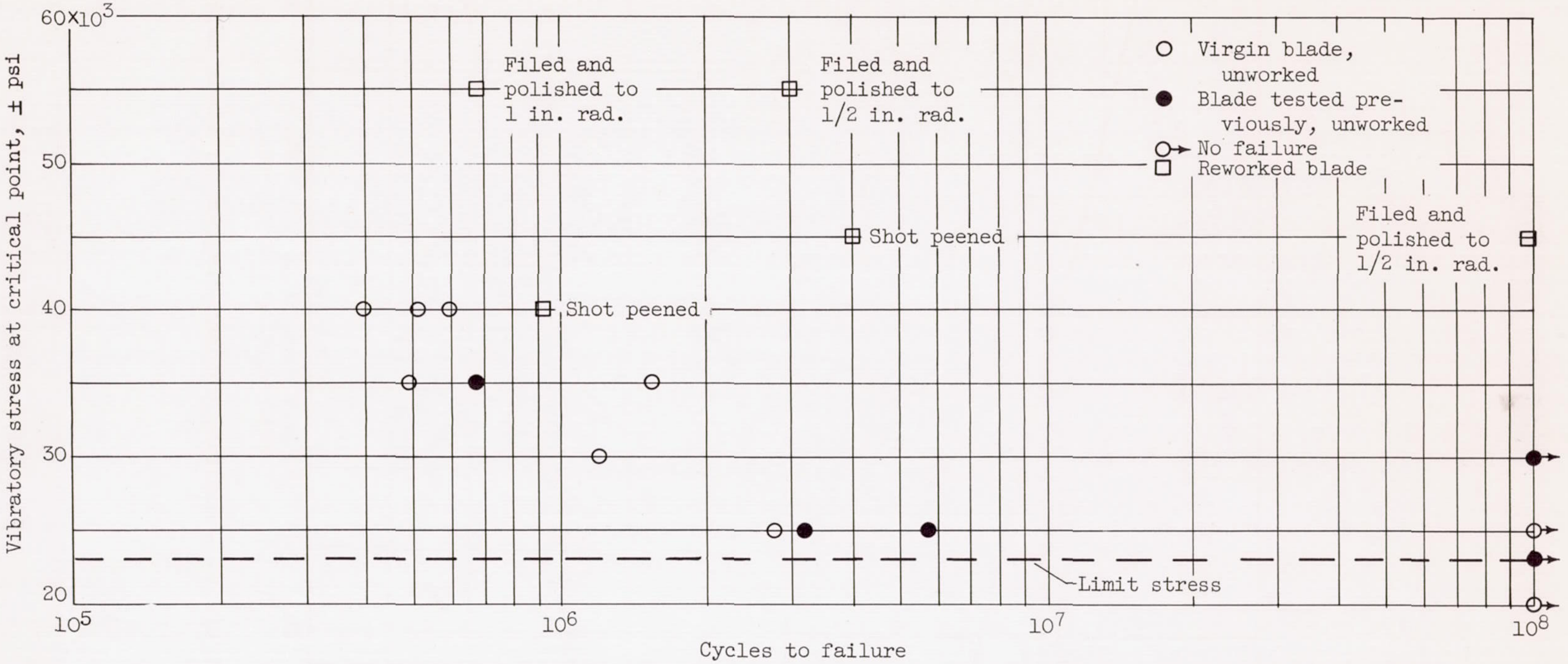


Figure 10. - Comparison of fatigue test results of reworked and unworked leading-edge nicks 1/32 to 1/16 inch deep and 28.5 percent of span from base.

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