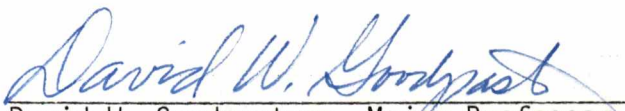


To the Graduate Council:

I am submitting herewith a dissertation written by Louis Richard Deken entitled "Structural Life Assessment of Turbomachinery Components." I have examined the final copy of this dissertation for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Doctor of Philosophy, with a major in Civil Engineering.


David W. Goodpasture, Major Professor

We have read this dissertation
and recommend its acceptance:







Accepted for the Council:


Vice Provost
and Dean of The Graduate School

STRUCTURAL LIFE ASSESSMENT OF
TURBOMACHINERY COMPONENTS

A Dissertation

Presented for the

Doctor of Philosophy

Degree

The University of Tennessee, Knoxville

Louis Richard Deken

August 1986

ACKNOWLEDGMENT

In the course of this dissertation research, a number of people have assisted me generously. In particular, I am indebted to Mr. H. Jones (Sverdrup Technology) for his support in the instrumentation of the XS-2 exhauster, Mr. T. W. Cromer, Sverdrup Technology program manager, for his assistance in the data acquisition and compilation of the XS-2 exhauster's history, Dr. D. W. Goodpasture and Dr. M. A. Wright for their advice in the analysis of the test data and the determination of the life assessment analysis procedure.

This work was funded by the United States Air Force and completed at the Arnold Engineering Development Center. The work was completed to reduce maintenance costs of turbomachinery in the various test facilities at the Arnold Air Force Station. It is to their funding and interest in this work that I am deeply indebted.

ABSTRACT

The development of a life assessment analysis procedure is described for turbomachines in the Engine Test Facility (ETF) of the U. S. Air Force, Arnold Engineering Development Center (AEDC). Both the crack initiation and retirement-for-cause (RFC) approaches for life assessment analysis were evaluated for ETF turbomachines. The RFC approach was selected, and the fracture mechanics analysis code BIGIF was acquired for use with the analysis procedure. The life assessment analysis procedure is described in detail along with a demonstration case using an ETF plant exhaustor compressor. Also described is the exhaustor test program in which operational test data were acquired for the life assessment analysis of a first stage exhaustor blade. A prototype noninterference blade-tip deflection measurement system developed at AEDC was used for the exhaustor test program.

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CHAPTER I

INTRODUCTION

The Engine Test Facility (ETF) of the U. S. Air Force, Arnold Engineering Development Center (AEDC), consists of turbine engine and rocket engine altitude test cells and plant equipment for supplying, conditioning, and exhausting air for the test cells. Major plant equipment includes centrifugal and axial flow turbomachines which are used as air compressors and exhausters. These turbomachines have been operating for approximately 25 to 30 years, and the projected service lives of these machines have not been determined. The inspection and maintenance schedules for the machines are currently based on their service history. These turbomachines have experienced various structural reliability problems such as bearing failures, fatigue cracking and failures of vanes and blades, and erosion and corrosion of blade leading edges. Replacement costs for these turbomachine components are high, and the loss of testing capability due to the repair or replacement of a turbomachine is significant. Therefore, a service life prediction procedure is required for turbomachinery components which will optimize inspection and/or replacement practices.

Historically, the limiting factor on service life of ETF turbomachinery is the failure of a turbomachine critical component. The structural life may be limited by a fatigue failure of a component such as a disk, vane, or blade. These failures result from the application of fluctuating loads over a period of time. By design, these loads are

below static failure. Turbomachine cyclic loads are caused by start up and shut down, normal operation load fluctuations, and more severe aerodynamic load fluctuations due to abnormal operation such as surges.

Techniques and codes have been developed for prediction of fatigue lives for structural components. However, the selection of a life assessment analysis procedure for a turbomachine is dependent on its operating characteristics, environment, and the load/stress measurement techniques which can be employed. This report summarizes the literature search and describes the life assessment analysis procedure developed for the AEDC ETF turbomachines. The description includes 1) different approaches for life assessment analysis that were examined, 2) the selection of the techniques and codes appropriate for AEDC ETF turbomachinery, and 3) the life assessment analysis procedure and its demonstration on a selected ETF plant exhaust compressor blade. Also described in this dissertation is the exhaustor test program in which operational test data were acquired on the exhaustor blades. Finally, conclusions and recommendations of the study will be presented.

CHAPTER II

LITERATURE SEARCH ON STRUCTURAL LIFE ASSESSMENT ANALYSIS

Life assessment analysis is the process of determining the structural characteristics and fatigue service life of components subjected to cyclic loads. An estimate of service lives can improve the maintenance and inspection programs for turbomachines. The total fatigue life of a component such as a blade or disk is directly related to the cyclic loading it experiences. If a crack is initiated due to this loading, the crack will continue to grow or propagate until the component fails. Prediction or determination of crack initiation and crack propagation lives are extremely important to avoid fatigue failures.

Many life prediction models used in industry at the present time use the principles of crack initiation, probabilistic methods, and fracture mechanics. The elastic theory commonly referred to as Linear Elastic Fracture Mechanics, LEFM, is the most widely used for fatigues; however, new advances have been made in inelastic analysis. The non-linear methods include Crack-Opening-Displacement, COD, Crack-Tip-Opening-Displacement, CTOD, and the J-Integral. The popularity of LEFM is due to the experimental data readily available and the ease of analysis. The non-linear methods, on the other hand, have very limited experimental data available.

Crack Initiation

The traditional approach to turbomachinery life assessment

analysis has been the prediction of fatigue crack initiation lives. This approach is based on the alternating stress-versus-cycles to failure (S-N) techniques. Initially, smooth or notched test specimens are polished so that all surface defects are removed. These specimens are cyclic tested (unloaded-loaded) to failure, and the resulting cyclic lives for various alternating stresses are plotted as S-N curves. These curves are used to predict the crack initiation lives for the turbomachine components.

The prediction of fatigue crack initiation life of a component requires the determination of the alternating stress ranges (maximum stress - minimum stress) and mean stresses (one-half of the sum of the maximum and minimum stresses) encountered during major load cycles of the turbomachine. The fatigue life estimates for each major load cycle are determined from fatigue life curves such as S-N curves using the predicted stress ranges and mean stresses. Finally, a linear cumulative fatigue damage model, based on a technique such as Miner's rule [16]¹, combines the predicted fatigue lives and determines a fatigue crack damage parameter for the turbomachine components. The Miner's rule for cumulative fatigue damage is one of the most widely used techniques because of the simplicity of the approach and because other techniques have not been universally applicable to all materials at all stress levels [42]. The Miner's rule is defined as

¹Numbers in brackets refer to similarly numbered references in the List of References.

$$D_j = n_i/N_i \text{ and } D = \sum D_j \leq 1 \quad (2-1)$$

where D_j is the j^{th} relative damage due to n_i cycles at a load condition with a simple-cyclic life of N_i . By this method, failure may occur when $D = 1$.

Prediction techniques for crack initiation lives are satisfactory from safety and reliability viewpoints; however, these predictions have resulted in overly conservative estimates of structural life. For example, the Air Force [2] has removed aircraft turbine engine disks from service when they accrue analytically predicted fatigue lifetimes. The fatigue lifetimes are based on a statistical analysis of fatigue data indicating the cyclic life at which one out of 1000 disks will have a fatigue-induced crack of approximately 0.03 inch in length. The 999 remaining disks which are also removed at the same predicted lifetime could have considerably more useful life. The uncertainties of the prediction techniques used for disk crack initiation lives are due to significant scatter in fatigue data, stress analysis errors, poor load definition, and manufacturing imperfections. These uncertainties require conservative crack initiation life estimates to preclude cataclysmic disk, and possible engine, failures.

Retirement-For-Cause

The expense of retiring 999 engine disks when only one disk might have a detectable crack has caused the Air Force [2] to develop a new approach for retiring engine disks, called retirement-for-cause (RFC). The RFC approach is a procedure, based on fracture mechanics and non-destructive inspections, which would allow the remaining disks to be

used in an engine until their individual safe fatigue lives are exhausted. The adoption of the RFC philosophy has resulted in significant cost savings through reduced replacement costs for turbine engine disks. This RFC approach could be adapted to turbomachinery.

The retirement-for-cause procedure requires the turbomachine components to be inspected and returns the component to service if cracks are not detected. The procedure is repeated until a crack has been detected, at which time the component is retired for cause. The return-to-service (RTS) interval is the time between inspections and is determined by a fracture mechanics calculation of the remaining propagation life (N_p). The propagation life is the time required to grow a crack from one small enough to have been missed during inspection to a final crack size too large to be tolerated in service. The RTS intervals are based on two broad technologies: applied fracture mechanics and nondestructive evaluation (NDE). Fracture mechanics must provide an accurate crack propagation prediction for the component should it pass NDE with a crack just below the inspection limit. The NDE must provide the means of screening components with flaws that could cause component failure within an economically feasible RTS interval.

The RTS intervals are determined by conducting a life-cycle-cost (LCC) analysis to determine the most economical safety factor (SF) to apply to N_p . The RTS interval is defined as

$$\text{RTS interval} = N_p / \text{SF} \quad (2-2)$$

Cost versus SF is plotted for each turbomachine component (blades, disks, etc.) and combined to determine the most economical interval for inspection of the turbomachine.

Non-destructive evaluation techniques for AEDC turbomachines were evaluated with the support of a consultant (Southwest Research Institute). The consultant recommended that the magnetic particle inspection technique be used for surface inspections (blade surfaces) and the ultrasonic inspection technique be used for volumetric inspections (rotor bore and disk rim welds). The magnetic particle technique is being used at AEDC for inspection of various structural components. Volumetric inspections are not being made at the present time.

Fatigue and Crack Growth

In studying fracture data we find most of it was developed from constant amplitude cyclic loads. Real world load conditions, however, are usually of a random nature. It was soon discovered that a single overload in stress could increase fatigue life significantly [4]. This effect has been referred to as crack retardation or temporary arrest of fatigue cracks. Some models proposed to explain these effects include 1) residual compressive stresses at or near the crack tip and their associated yield zone interactions, 2) crack closure, 3) crack tip blunting, and 4) crack tip strain hardening. Figure 1² graphically shows several crack retardation curves developed experimentally [4]. Curve "a" is a typical constant load amplitude plot while "b", "c",

²Figures and tables are found in the Appendix.

and "d" show different types of retardation effects due to a single overload. Curve "b" shows an immediate decrease in crack growth (retardation) occurring immediately after the applied overload. Curve "c" shows an increase in crack growth occurring immediately after the overload with the decrease delayed for a period of time. Curve "d" shows the retardation is lost due to a delayed increase in crack growth. Crack retardation appears to occur if the mean load level, P_l (max), is approximately 40 to 60 percent of the overload cycle, P_h (max), as shown in Figure 2. All materials currently tested appear to have this behavior. Retardation, however, is affected by the thickness, yield stress, and to some degree the crack depth and geometry. In general, a single tensile overload can increase the structural life significantly. The inclusion of a compressive overload, however, may reduce the life. Compressive loads appear to reduce the effect of beneficial residual stress caused by tensile overloads. In some cases, a compressive overload can reduce the structural life to a level less than that for constant amplitude loads.

Other factors which affect fatigue life and retardation include material, component geometry, type of loading and its sequence, rate of loading, crack geometry, crack front orientation, crack tip blunting, crack closure, residual stresses and strains, loading magnitude, and environmental conditions. The inclusion of all of these parameters is obviously a complex task. The exact effect of many of these at the present time is only speculation or empirical. The study of many of these effects can be found in several of the references listed in this dissertation.

If a structure is subjected to a moist environment, the mechanism of corrosion fatigue or stress corrosion cracking [45,47] can affect the life. A typical plot for the effect on the stress corrosion crack propagation rate is shown in Figure 3. This mechanism is of more concern in low-cycle fatigue situations such as in offshore oil platforms. In this case, crack tip corrosion reactions may cause crack blunting or they may wedge the cracks open with corrosion products which reduce the structural life.

It appears that good data for the corrosion mechanism are hard to obtain experimentally. Due to the time required for the corrosion process, as well as the actual agents involved, the data can only be treated as a probable trend. Many of the tests performed are actually accelerated tests. It is generally believed that the results obtained are conservative.

Grain size [48] of the material can have a significant effect on structural life, particularly on a microscopic scale. The grain size affects the slip motion, crack nucleation, and crack growth stages of the fatigue process. Grain size is usually the result of the material manufacturing process.

A lot of study has been concentrated on the effect of the load spectrum as can be seen from References [1], [5], [12], [18], [19], [26], [28], [30], [31], [33], [37], [39], and [54]. These papers cover such variables as load sequence effects, variable amplitude loading, blocked load data from spectrum loads, editing of load spectrum and load cycle counting methods.

Most of the data generated in these papers were acquired to predict structural life of aircraft and, to a smaller degree, the piping and pressure vessels of the nuclear industry. The Air Force's maintenance program [3] has the specific purpose of defining and scheduling of structural maintenance actions; inspections, modifications, and retirement. This is accomplished through the use of potential crack growth curves developed for specific locations in the aircraft. These curves are derived from assumed load spectrums for planned missions. The basic scheme consists of the development of a plot of crack length (by inspection) versus actual time of aircraft usage, $t_{a/c}$. This is then compared to a plot of projected crack length versus planned usage time, t_{mu} . The comparison of these plots, Figure 4, can give a projected service life for the aircraft. If service life is defined as

$$t_L = t_{mu*} - t_{mu} \quad (2-3)$$

we can define the time L to reach a critical crack length a_f by

$$L = t_L / U_p \quad (2-4)$$

where U_p is the planned monthly utilization rate. In using these curves, it was discovered that the random variation in potential initial crack length, flying rates, and usage can result in significant errors in the estimated date the potential crack will reach its critical size. This meant maintenance inspections were sometimes not needed, and when the inspections were performed it prevented use of the aircraft for an extended period of time. More accurate models for structural life prediction were required.

In general, the analysis of load history curves is complex. A cycle-by-cycle analysis would be considered the most desirable but is only possible for simple load spectrums. Since most load spectrums have a large frequency of overloads and a large variability in load amplitudes, this simple approach becomes too cumbersome in terms of amount of data and computation time.

The "rainflow" method [18] is generally considered the most desirable method of counting load cycles. This method is usually conservative but in some cases a modified form may be preferred. If a load spectrum becomes extremely complex, it may be necessary to define a set of sequenced loads (blocked loads). Standardized load spectrums have been defined in the aircraft industry for the design of specific classes of aircraft. The automotive engineers have also developed standard load spectrums for the design of various automotive components.

Probabilistic Methods

A review of the papers which describe the probabilistic methods of structural life prediction indicates that limited data are available for this type of analysis. The majority of the work has been associated with piping, pressure vessels, and aircraft structures. Many of the methods develop probability of failure functions based on past histories. Other approaches use a statistical analysis of similar structures. These methods obviously require extensive amounts of data to give reliable life predictions. Probabilistic or statistical models are included in References [10], [13], [21], [23], [25], [29], [32], [35], [40], [43], [46], and [51].

Payne [40] defines two types of failure for a structure subjected to repeatedly applied loads. The first type occurs after the formation of the initial crack. In this case the fatigue crack extends progressively as the life continues with a progressive reduction in the residual strength of the structure. There is an increasing probability at each successive application of a service load that the structure will fail statically. This type of failure is called static fracture by fatigue. Failure by fatigue fracture occurs if the structure survives the risk of failure as static fracture, and the crack eventually reaches some length, a_f , at which failure occurs under steady mean load.

The use of a survival probability function for these failures requires acceptable threshold values of survival rates. These rates must be determined from previous experience with similar structures. If this experience is not available, rates must be assumed and modified as experience is gained.

Another approach to the probabilistic methods is described by Lidiard [32]. His approach is a theoretical synthesis of failure probabilistics made from a knowledge of the statistical distributions of the parameters known to cause failure. It is assumed that a structure initially contains one or more cracks in its critically stressed regions. Given $N_0(a)$, a function which represents the initial average number of cracks in a critical region, Lidiard developed two equations for the determination of the probability of failure, P_f .

$$P_f = \int_0^h dy \int_y^n dx \int_0^x N_0(a) \phi_t(x,a) f(y) da \quad (2-5)$$

In the linear elastic case, $f(y)$ is replaced by the corresponding function for toughness, K_{IC} , which gives

$$P_f = \int_0^\infty f(K_{IC}) dK_{IC} \int_y^h dx \int_0^x N_0(a) \phi_t(x,a) da \quad (2-6)$$

where $\phi_t(x,a)$ is a transfer function which represents the probability that a crack initially of depth a has depth x after time t ; $f(y)$ is the probability distribution function for a critical crack size under a stress σ ; h is the thickness; y is the critical crack depth; and x is the instantaneous crack depth.

To apply this method to a particular structure, estimates of the probability distribution function $N_0(a)$, $f(K_{IC})$, $\phi_t(x,a)$, and $f(y)$ must be made. At the present time, it appears these estimates are purely observational or empirical and in most cases very sketchy. The parameter $N_0(a)$ is written as the product of two functions $A(a)$ and $B(a)$. $A(a)$ is the size distribution function of cracks resulting from manufacture and $B(a)$ is the probability of not detecting these cracks in the subsequent pre-service inspections.

$$N_0(a) = A(a)B(a) \quad (2-7)$$

The difference by researchers in the estimates of $A(a)$ and $B(a)$ varies significantly. The toughness distribution can be defined for most

materials with better accuracy than $N_0(a)$. One approach to define $f(K_{IC})$ is from Charpy v-notch data, defined by the relationship

$$(K_{IC}/\sigma_y)^2 = 5[(C_V/\sigma_y) - 0.05] \quad (2-8)$$

Although somewhat difficult to obtain, the transfer function $\phi_t(x,a)$ can be determined by the integration of the crack growth law

$$da/dN = C(\Delta K_I)^n \quad (2-9)$$

and averaging the solution over all possible transient stresses and all possible values of the material constants "C" and "n".

A study of the parameters involved in Lidiard's method shows the failure probabilities are sensitive to the crack size function $N_0(a)$ and the crack growth rates. This means a good understanding is needed of the laws for crack growth as well as the initial crack distribution. The failure probability is not particularly sensitive to the accuracy of the stress-intensity function $f(K_{IC})$.

The final failure probability of a structure is simply the sum of the separate failure probabilities for the component parts. These calculations can easily be used to show the relative importance of the various parts of the structure.

Fracture Mechanics Methods

The fracture mechanics approach is by far the most popular method of life assessment analysis. Experimental data were used to develop a wide range of fracture mechanics models. Most models show variations in the principles which cannot be used as a general theory for fracture mechanics. In fact, a theoretical approach seems to be very complex,

if not impossible, to formulate. As a result, the general principles are empirical and do give conservative results for most materials.

The crack propagation concept of fatigue is illustrated schematically in Figure 5. Cracks grow from an initial defect size, a_i , to a size determined to be "critical for failure", a_f , as a function of the number of cycles, N , of repeated loads [15]. Basic to the crack propagation concept of fatigue is the concept that the per cycle rate of growth of a crack da/dN can be expressed as a function of the crack-tip stress-intensity factor K . Given a mathematical definition of the growth process and boundary conditions over which this growth process occurs, the fatigue crack growth rate of a component can be calculated. Such an approach to fatigue, called fracture mechanics, places emphasis on the crack growth rate characteristics of metals. Both linear elastic and nonlinear approaches for fracture mechanics analysis were investigated during the development of the AEDC life assessment analysis procedure.

The linear elastic approach for fracture mechanics is based on a crack-tip model shown schematically in Figure 6. This model considers metal under tensile stress in the presence of a crack to have a small zone of plastically strained material at the crack tip, even though the stresses in the surrounding bulk material are below yield [15]. An important restriction to the use of linear elastic fracture mechanics is that the plastic zone at the crack tip must be small relative to the geometrical dimensions of the specimen or part [24]. A proposed limiting condition for linear elastic fracture mechanics is that

nominal stresses in the crack plane must be less than 80 percent of the material yield strength.

Three modes of loading [7] shown in Figure 7 are defined for a crack, but Mode I loading (opening mode) has been shown to be the most critical. Mixed mode loading, however, can cause significant changes in the fatigue life. Under Mode I cyclic loading, the crack is pulled open and then closed. There is an abrupt rise in stress at the crack tip as the crack is opened, Figure 6. The stress intensity factor, K , is a parameter which describes this abrupt rise in stress from the nominal stress levels to levels above yield within the plastic zone, at the crack tip [15]. Dimensionally, K is directly proportional to the product of nominal stress (σ) and the square root of crack length (a),

$$K \propto \sigma(\pi a)^{1/2} \quad (2-10)$$

and has the units of Ksi- $\sqrt{\text{In}}$. The exact expression of proportionality is dependent upon crack geometry and, for many typical cracks, can be determined from handbooks. Critical values of K refer to the condition when a crack extends in a rapid (unstable) manner without an increase in nominal tensile stress [24]. Critical values of K , denoted with a subscript c , are defined as

$$K_c \propto \sigma_c(\pi a_c)^{1/2} \quad (2-11)$$

where σ_c is the applied nominal stress and a_c is the crack length at instability. K_c is called fracture toughness and depends on the material, temperature strain rate, environment, thickness, and, to a lesser extent, crack length. The K_c parameter can be used to obtain critical crack sizes for fracture under a given cyclic loading. The

parameter which defines the threshold conditions for fatigue crack growth (the stress intensity level at which no crack growth occurs) is the stress intensity threshold, K_{th} . Once a value of K is calculated, its significance is geometry-independent [15], an important characteristic of linear-elastic fracture mechanics.

For the assessment of fatigue crack propagation lives, fracture mechanics assumes that a crack of some initial size, a_i , can grow to some final size, a_f , under the action of cyclic loads. The initial crack size is usually defined on the basis of nondestructive evaluation capabilities or by prior inspection limitations. The final crack size is usually defined by the fracture toughness or a predetermined allowable crack size [52]. The crack growth rate (da/dN) is a function of the stress intensity factor

$$da/dN = F(K) \quad (2-12)$$

The crack propagation life (N_p) of a structure under cyclic loading can be determined by rearranging and integrating Equation (2-12).

$$\int_0^N dN = N_p = \int_{a_i}^{a_f} [F(K)]^{-1} da \quad (2-13)$$

Since most engineering structures experience a variety of loads, a composite load history can be constructed which consists of major load cycles. Crack growth increment, Δa_k , can be computed for each major load cycle and then summed to obtain the crack growth, Δp , for the load spectrum as defined by

$$\Delta p = \sum_{k=1}^n \Delta a_k \quad (2-14)$$

where n is the total number of major load cycles in the load spectrum. The fatigue crack propagation life can be defined by repeating the load history so that the crack will grow from a_i to a_f .

The relationship which describes the fatigue crack propagation characteristics, shown schematically in Figure 8a, relates crack growth rate (da/dN) to the crack-tip stress-intensity factor range ($\Delta K = K_{\max} - K_{\min}$) [15]. A standard method to characterize the da/dN versus ΔK curve is to test a pre-cracked laboratory specimen. The specimen is cyclically loaded and the crack length is periodically measured and recorded along with the number of load cycles. The crack length versus cycles is determined as shown in Figure 8b. The slope of the curve, da/dN , can be computed at any crack length, a . The stress intensity factor K can be calculated for the same crack length, a , and da/dN versus ΔK , can then be plotted as shown in Figure 8a. The overall shape of the curve is sigmoidal with a region of stable crack growth (Region 2) connecting the upper and lower thresholds. The lower threshold of the curve (Region 1) defines the region of nonpropagating fatigue crack conditions, and the upper threshold (Region 3) defines the region of rapid, unstable, crack growth associated with the onset of fracture. The da/dN versus ΔK curve is independent of loading and geometry, and can be used for general life analysis.

Empirical relations to express da/dN behavior have been proposed to fit the data represented by Figure 8a. The earliest and most well-

known relation is a power-law relationship called the Paris rule [55] which has the form

$$da/dN = C(\Delta K)^n \quad (2-15)$$

where C and n are material constants determined from relevant data.

The advantages of the Paris rule are: 1) it is simple in form (straight line on a log-log plot) and 2) it fits experimental data well in the middle region of ΔK . One major disadvantage of the Paris rule is that it does not account for the influence of the stress ratio (R = Minimum stress/Maximum stress) on fatigue crack propagation. Another expression developed by Forman [55] accounts for stress ratio effects. The Forman expression has the form

$$da/dN = (C\Delta K)^n / [K_c(1 - R) - \Delta K] \quad (2-16)$$

where C and n are material constants (different from the Paris rule), and K_c is either the plane strain fracture toughness as originally proposed by Forman or can serve as an additional parameter for the data fit.

Interaction effects due to variable amplitude loadings has added additional complication into the crack growth rate equation. In this case, the Willenborg model and the Wheeler model [53] can be used to account for interaction effects by

$$\Delta a_i = \beta_i (da/dN)_{ca} \quad (2-17)$$

where Δa_i is the incremental crack growth; $(da/dN)_{ca}$ is the constant amplitude crack growth rate; β_i is factor accounting for interactions. The β_i parameter is assumed to depend on plastic zone sizes associated

with load cycle i and the preceding cycles. As a result, crack growth delays are obtained after a peak load, but negative interaction (crack growth acceleration) cannot be predicted.

From a crack closure approach [20], Equation (2-17) can be replaced by

$$\Delta a_i = (da/dN)_{ca} = f(\Delta K_{eff,i}) \quad (2-18)$$

$$\Delta K_{eff,i} = C \Delta S_{eff} (\pi a)^{1/2} \quad (2-19)$$

$$\Delta S_{eff} = S_{max} - S_{op} \quad \text{if } S_{min} < S_{op} \quad (2-20)$$

$$\Delta S_{eff} = S_{max} - S_{min} \quad \text{if } S_{min} > S_{op} \quad (2-21)$$

where S_{op} is the crack opening stress; S_{max} and S_{min} are maximum and minimum spectrum stresses.

The equation gives a relationship between crack rate and ΔK_{eff} as proposed by Elber. He also extended this method [19] of life prediction to the use of constant amplitude data. The equivalent number of constant amplitude cycles, N_{eq} , is defined by the expression

$$N_{eq} = \Sigma [(S_i - S_{e,i})^n / (1-\alpha)^n (S_{max} - S_B)^n] \quad (2-22)$$

$$\alpha = (S_{op} - S_B) / (S_{max} - S_B) \quad (2-23)$$

where S_i is the stress in the i th excursion of the random spectrum; $S_{e,i}$ is the effective minimum stress; S_{op} is the crack opening stress; S_B is the lowest minimum stress.

This constant-amplitude concept was based on the assumption that crack-opening stresses remain essentially constant during short random load sequences. Based on Elber's experimental results, this crack

closure model gives results in good agreement with tests. This concept simplifies crack growth calculations for complex random loads to an analysis of equivalent constant amplitude load cycles, to which data are readily available for many materials.

Another crack growth rate relation called the effective stress intensity range concept [37] defines crack growth by the equation

$$da/dN = C(\Delta K_{eff})^n / (K_c - K_{max}) \quad (2-24)$$

where ΔK_{eff} is the effective stress intensity range; $(K_{max} - K_{res})$; K_{res} is the residual stress intensity; K_c is the critical stress intensity; K_{max} is the maximum stress intensity; and C and n are constants obtained from the constant amplitude power law. The philosophy of using ΔK_{eff} is symbolically shown in Figure 9. The use of ΔK_{eff} in this way accounts for the possible load sequence effects due to crack retardation, decrease of retardation by sufficiently large compressive overloads, the acceleration of crack growth by gross yielding in compression (as K_{res} goes to zero).

Additional life prediction models can be found in References [11], [17], [22], [27], [44], and many others previously discussed. Many of these models have similarities to those discussed above while others are more elaborate. A few models are based on so-called "damage" parameters. These parameters are used as a guide in evaluating the remaining life.

The elliptical surface crack approximates many cracks found in engineering components and has received widespread analytical, numerical, and experimental analysis. The surface semi-elliptical crack in

a finite thickness solid shown in Figure 10a is very common in fatigue. A general expression for the Mode I semi-elliptical surface crack in a finite thickness plate [24] is

$$K = [\sigma(\pi a)^{1/2}/\theta] M_f M_b [\sin^2 \beta + (a/c)^2 \cos^2 \beta]^{1/4} \quad (2-25)$$

where θ is a complete elliptical integral of the second kind and depends on the crack aspect ratio, a/c , β is the angle shown in Figure 10b, and M_f and M_b are the respective front face and back face correction factors which are functions of β .

Except for some simple crack geometries, the determination of K as a function of crack depth is a complex problem when the body dimensions are finite and the stress distribution is varying in a nonlinear fashion. One approach for calculation of K is called the influence function (IF) method [55]. The prominent feature in the formulation of the IF method is the use of an elastic superposition principle to reduce a difficult crack problem to

1. the problem of calculating the stresses in the uncracked structure across the locus of each crack face and
2. the problem of calculating K where the crack face is loaded so that the uncracked stresses are cancelled, achieving the proper traction-free crack face boundary conditions.

Many factors must be considered in a fracture mechanics analysis, such as loading spectrum, stress analysis, initial crack size, stress ratio, crack growth retardation, and environment. Accurate assessments of fatigue crack propagation lives require a knowledge of the number, amplitude, and sequence of major load cycles. In crack propagation,

accurate predictions of both surface stresses and stress gradients through the material are very important. The initial crack size used in the calculation of crack propagation life strongly influences the final results (a smaller initial crack gives a longer fatigue life). The effect of stress ratio on crack propagation is second order. Crack growth retardation, another factor which can be of secondary importance, has a potentially beneficial influence on fatigue crack propagation life resulting from the hysteresis effects of periodic tensile overloads. Environments can have a detrimental effect on fatigue crack propagation life. In most structural alloys, severe environments such as sea water or elevated temperatures significantly accelerate fatigue crack growth.

The LEFM method is based on the assumption that the crack tip elastic stress yield is uniquely described by the stress intensity factor, K . This means the crack tip plastic zone is small compared to the crack size. If the plastic zone is large, the plasticity effects become significant and stress intensity factor loses its original significance. In linear elastic fracture mechanics, the size of the plastic zone [7] ahead of the crack tip and using Irwin's plastic zone correction, Figure 11, is defined by

$$r_p = 2r_p^* \quad (2-26)$$

$$r_p^* = (K_I)^2 / 2\pi(\sigma_y)^2 \quad (2-27)$$

where r_p is the diameter of the assumed circular plastic zone ahead of the crack tip.

The assumption for the Dugdale crack, Figure 12, gives similar results

$$r_p = \pi K^2 / 8 \sigma_y^2 \quad (2-28)$$

If the plastic zone is large compared to the crack size, the application of neither of the above expressions as a correction for the plastic zone is valid. Another case where LEFM does not apply is for materials with very high toughness.

The assumption for the "Crack Tip Opening Displacement" method, CTOD, is that fracture takes place at the exceedance of a critical CTOD [7]. According to the Wells' criterion, Figure 13, the general expression for "Crack Opening Displacement", COD, based on elastic theory is defined as

$$COD = 4\sigma(2ra_{eff})^{1/2} \quad (2-29)$$

from which the expression for CTOD becomes

$$CTOD = 4(K_I)^2 / E\sigma_y \quad (2-30)$$

or based on the Dugdale approach (plane strain)

$$CTOD = G_I / \lambda \sigma_y = (K_I)^2 (1 - \mu^2) / E \lambda \sigma_y \quad (2-31)$$

where G_I is the energy release rate and λ is an experimentally determined material parameter. The use of the CTOD criterion relies on the assumption that a critical CTOD can be established as a material constant. However, $CTOD_c$ determinations are difficult to obtain experimentally. The values determined are very sensitive to the exact location used for the crack tip opening measurements.

Another approach to the elastic-plastic problem is the J-integral concept [7,34,36]. For the special case of linear elastic materials, J reduces to the strain energy release rate where

$$J = G = K^2/E \quad (2-32)$$

For plastic deformation J loses its potential energy interpretation but still serves as a measure of the crack-tip strain field. The general expression for the J-integral is the line integral defined by

$$J = \int_L (Wdy - T(\partial y/\partial x))ds \quad (2-33)$$

where L is an arbitrary contour around the crack tip; T is surface tractions; and $\partial y/\partial x$ is strain.

$$W = \int_0^{\epsilon} \sigma_{ij} d\epsilon_{ij} \quad (2-34)$$

Musuva and Radon [36] used the J-integral concept to calculate the crack growth rate for low alloy steel specimens. Based on their work, the crack growth rate is described as

$$A = 2^{n+1}/4E^n \quad (2-35)$$

$$B = (\Delta J_{\text{eff}} - \Delta J_{\text{c,th}})/(I_{\text{on}}(\epsilon_f)^{n+1}(\sigma_y)^{1-n}) \quad (2-36)$$

$$da/dN = AB + C(J_{\text{max}})^m \quad (2-37)$$

where ϵ_f is the true fracture strain; $\Delta J_{\text{c,th}}$ is the critical threshold value for a non-propagating crack; ΔJ_{eff} is an effective ΔJ at which the crack tip is opened and the material can deform, it is assumed deformation does not take place when the crack is closed; I_{on} is a function of the strain hardening exponent, n ; and C and m are material constants determined experimentally.

The J-integral concept at present appears to be limited in use. This is primarily due to limited availability of data and a somewhat more complex analysis.

The study of crack-tip stress fields has also involved the use of finite element methods. This method requires detailed modeling of the crack-tip geometry and uses special crack-tip elements which include the singularity effect caused by a small crack-tip radius. The standard isoparametric element is ideal if the mid-node is shifted to the quarter point. This will automatically create the singularity which is built into many of the special elements. These elements can be used in elastic as well as elastic-plastic analysis.

CHAPTER III

STRUCTURAL LIFE ASSESSMENT ANALYSIS PROCEDURE

A review and evaluation of various approaches for life assessment analysis was conducted. A review of the turbomachinery systems in the AEDC Engine Test Facility (ETF) was also conducted for selection of the best available approach for life assessment analysis of the ETF turbo-machines. The crack initiation, probabilistic, and retirement-for-cause approaches previously discussed for life assessment analysis were evaluated.

Selection of Life Assessment Analysis Approach

Typically, the crack initiation approach for life assessment analysis results in overly conservative estimates of service lives. The life-cycle-costs based on this approach would be large because the turbomachine component replacement intervals would be based on these service lives. The retirement-for-cause approach for life assessment analysis would allow the turbomachine components to be used until their individual safe lives are exhausted. Typically, this approach would lengthen the component replacement interval, resulting in lower life-cycle costs. Therefore, the retirement-for-cause approach was selected for development of a life assessment analysis procedure for AEDC turbomachinery.

The retirement-for-cause approach is based on fracture mechanics. Both linear elastic and nonlinear approaches for fracture mechanics were investigated. The nonlinear approach is not widely used in design

analysis and material characterization data are limited. The linear elastic approach is widely used in design analysis and material characterization data are available for a wide range of materials. The operational and maintenance history of ETF turbomachines shows few fatigue failures. Based on this fact, it is assumed that the normal operating stresses in the ETF turbomachine components are below yield. Therefore, the linear elastic fracture mechanics approach was selected for development of the life assessment analysis procedure for AEDC turbomachinery.

Description of Analysis Procedure

The life assessment analysis procedure for AEDC turbomachinery consists of four major interrelated parts, with each part composed of several tasks. The following is a brief description of the four major parts:

1. Operational Loads - Typically, the operational loads are determined from test data acquired on the turbomachine. These data include rotor speeds, metal temperatures, and stress. The test data and calculated loads are used in the stress analysis of the turbomachine components.
2. Stress Analysis - Stress analysis models are constructed of critical turbomachine components such as disks and blades. The operational test data, along with the calculated loads, are input to the stress models for determination of steady-state and dynamic stresses. These stress data are then used in a fatigue analysis of the

components. Also, strain gage data are used to validate the stress models.

3. Fatigue Analysis - Fracture mechanics analysis models of critical regions of the turbomachine components are constructed. These models are used in a fatigue analysis to compute the crack propagation lives, N_p . The steady-state and dynamic stress data are formulated into a load history which can be input to the fracture mechanics models. The computed crack propagation lives are used in the determination of the inspection interval for the turbomachine.
4. Inspection Interval - The inspection interval for the turbomachine is determined by conducting a life-cycle cost analysis. The purpose of this analysis is to determine the most economical safety factor (SF) to apply to the computed crack propagation lives. The inspection interval is then determined by N_p/SF .

Each step of the analysis procedure should be completed in the proper order to ensure that the information is available for subsequent steps. A more detailed description of each part of the analysis procedure is contained in the following chapters.

CHAPTER IV

COMPONENT OPERATIONAL LOAD ANALYSIS

The first major part of the life assessment analysis procedure is the determination of the operational loads for the turbomachine components. Typically, these loads are determined from operational test data acquired on the turbomachine. These data include rotor speeds, metal temperatures, and steady-state and dynamic stresses. These data are acquired during start-up and shutdown, steady-state, transient, and surge operation of a turbomachine. A surge condition will result in rapid load fluctuations of possibly large amplitudes. The time span for a surge condition should be determined so that an accurate representation in the total load spectrum can be constructed. It is anticipated that a surge period may result in accelerated crack growth and a reduction in structural life. The test data and calculated loads are used in the stress analysis of the turbomachine components (i.e., disks and blades). If test data are not available, then the operational loads could be estimated. However, the accuracy of the stress analysis and subsequent fatigue analysis is heavily dependent on the accuracy of the operational loads which can only be verified by test data.

The rotor speed data are input to the blade and disk stress models for calculation of centrifugal stresses. The metal temperature data are used to determine the thermal gradients in the turbomachine components. If the temperature data indicate significant thermal gradients, then heat-transfer models should be constructed to determine

detailed temperature distributions for a thermal stress analysis. Additional operational data such as pressures and air temperatures are needed for an accurate thermal and stress analysis. Dynamic strain data are used to determine the blade natural frequencies and modes of vibration. These data are also used to determine the component vibratory and surge loads and component stresses. Finally, the computed loads are formulated into a load spectrum which is assumed to be repeated over the life of the turbomachine.

Turbomachine operational test data can be acquired using conventional instrumentation and data acquisition systems. Typically, turbomachine blade dynamic strain data are acquired using strain gages attached to the blade surface. The strain gage data are transmitted from the blade to the data storage system via sliprings and/or telemetry. There are significant problems associated with the installation, operation, life expectancy, and cost of strain-gage systems. The installation of strain gages and connecting leads could require the disassembly of the turbomachine rotors which is expensive. The operating life expectancy of the strain-gage system is limited, with failures of the connecting leads being the major failure mode. Redundant strain gages are required to minimize the loss of data. Finally, the overall cost of a strain-gage system can be very expensive. Because of the problems and associated costs of strain-gage systems, an alternate approach for acquiring blade dynamic strain data was investigated. This alternate approach is based on a prototype noninterference strain measurement system developed for turbine engine compressor blades [34].

Two special purpose computer codes were developed at AEDC for processing and analyzing blade-tip deflection test data. A Dynamic Analysis Program (DAP) was developed to analyze a vibration spectrum and determine the dominant frequencies in a time spectrum. The DAP code can be used to determine blade-tip deflection amplitudes and frequencies, which can be used in the fracture mechanics analysis of turbo-machine components. Also, a Load History (LOADHIST) program was developed to convert the random blade-tip deflection data into a blocked deflection data spectrum suitable for fracture mechanics analysis.

Dynamic Analysis Program (DAP)

Vibrations of several different natural frequencies can exist simultaneously in a structure. A Dynamic Analysis Program was developed to analyze a vibration spectrum and determine excited natural modes. This program determines the dominate frequencies in a time spectrum.

Fourier has shown that any periodic motion can be represented by a series of sines and cosines which are harmonically related [38]. If a vibration spectrum, $x(t)$, is assumed to be a periodic function of period T , it can be represented by the Fourier series

$$x(t) = A_0 + \sum_{i=1}^{\infty} [A_i \cos(2\pi it/T) + B_i \sin(2\pi it/T)] \quad (4-1)$$

where A_i and B_i are constants and t is instantaneous time. If both sides of this equation are multiplied by $\cos(2\pi it/T)$ or $\sin(2\pi it/T)$ and integrated over the period T , the Fourier coefficients, A_i and B_i , can

be determined. These equations reduce to

$$A_i = (2/T) \int_{-T/2}^{T/2} x(t) \cos (2\pi i t/T) dt \quad (4-2)$$

$$B_i = (2/T) \int_{-T/2}^{T/2} x(t) \sin (2\pi i t/T) dt \quad (4-3)$$

$$A_0 = (1/T) \int_{-T/2}^{T/2} x(t) dt \quad (4-4)$$

DAP uses a Fast Fourier Transform (FFT) algorithm developed by Cooley [14] for the solution of Discrete Fourier Transform (DFT). The coefficients are then used to determine a Fourier spectrum (amplitude versus frequency) which represents the original time spectrum. The frequencies ω_i , (Hz) are computed by

$$\omega_i = i/N\Delta \quad i = 1, 2, 3, \dots, N/2 \quad (4-5)$$

where N is the number of data points in a discrete time series and Δ is the time sampling interval. The amplitude, C_i , and phase, θ_i , can be computed by the relations

$$C_i = 2(A_i^2 + B_i^2)^{1/2} \quad i = 1, 2, 3, \dots, N/2 \quad (4-6)$$

$$\theta_i = \tan^{-1} (B_i/A_i) \quad i = 1, 2, 3, \dots, N/2 \quad (4-7)$$

It can be shown that the coefficients calculated by the DFT are correct Fourier coefficients for frequencies up to

$$\omega = 1/2\Delta \quad (4-8)$$

Higher frequencies will result in Fourier Coefficients which are repetitions of those for the lower frequencies. On the other hand, if there are frequencies above $1/2\Delta$ present in the vibration spectrum, they will produce a distortion in the amplitude-frequency plot, Figure 14, which is called aliasing. The high frequency components contribute to the amplitude by distorting the Fourier coefficients calculated by the DFT for frequencies below $1/2\Delta$. The frequency $1/2\Delta$ Hz, is called the Nyquist frequency (folding frequency) and represents the maximum frequency that can be detected from data sampled at a time interval Δ .

Since it is assumed by the DFT that the vibration spectrum is periodic over a period T , the vibration spectrum period is represented by the data span from t_i through t_n as a single cycle. A data window can be applied which will smooth the discontinuity at the two extremities of the spectrum.

The DAP code multiplies the data by either a rectangular window or a cosine taper data window weighting function. The rectangular data window assumes zero end points on the spectrum while the cosine taper data window [38] assumes a weighting function as

$$WF(t) = (1/2)[1 - \cos(10\pi t/T)] \quad \text{when } 0 < t < T/10 \quad (4-9)$$

$$WF(t) = 1.0 \quad \text{when } T/10 \leq t \leq 9T/10 \quad (4-10)$$

$$WF(t) = 1/2\{1 + \cos[10\pi(t - 9T)/10]/T\} \quad \text{when } 9T/10 < t < T \quad (4-11)$$

The resulting DFT is divided by the factor

$$F = 1/N \sum_{i=1}^N WF^2(t_i) \quad (4-12)$$

which corrects for the loss of amplitude caused by applying the cosine data taper.

Load History Program (LOADHIST)

The program LOADHIST was developed to analyze vibration deflection data and determine a cyclic spectrum for a fracture mechanics analysis. The basic assumption of LOADHIST is that the data represent a component's load history. The program determines the number, magnitude, and mean value of deflection cycles represented in the data. Once the cycle counting is complete, the number of deflection cycles is grouped into constant amplitude blocks. This grouping minimizes computational requirements for the crack propagation analysis.

There are several methods available to count load cycles from a given load spectrum. Some of these methods include range counting, level crossing counting, and rainflow counting [50]. Range counting simply computes the difference between adjacent data peaks and valleys. A matrix containing ranges and numbers of occurrences is developed. Level crossing counting produces a histogram containing range and number of occurrences. In this case, a count is determined according to preselected levels above and below zero. There are many variations to these methods. Some retain the mean value of the range while others do not.

Rainflow counting is a procedure which defines a range-mean-count matrix similar to a simple range counting method. The difference is that rainflow counting does not pair adjacent peaks and valleys in the spectrum. This method attempts to pair the peak and valley combinations

which would define a closed stress-strain loop if the variable amplitude history were applied to a uniaxial test bar. Rainflow is the method incorporated into the LOADHIST program. The algorithm used in LOADHIST is as follows: if the blade-tip displacement versus time data are plotted with the displacement plotted horizontally and time vertically as shown in Figure 15, it resembles water flowing over the top of all individual spikes as on a series of roof tops. In all cases, the principle flow path is determined by the flow path with the largest maximum and smallest minimum displacement values. Flow paths are restricted in that they are not permitted to cross or join another flow path. The maximum and minimum values of a given path determine its displacement range and mean values. For example, in Figure 15 flow path one represents one load cycle varying from zero to 24 mils; flow path five, one cycle varying from 12 to 23 mils; and flow path seven, one cycle varying from 13 to 17 mils. The algorithm will preserve sequence effects in the order shown in Figure 15.

Since the BIGIF code requires blocks of constant amplitude data to minimize computational requirements, it was necessary to block the results of the rainflow analysis. To do this, the root-mean-squared (RMS) approach proposed by Barsom [41] was used. Barsom found that random amplitude data could be treated as constant amplitude data. In general, this is true as long as the load cycle amplitudes are away from the threshold level for crack growth. The RMS maximum and minimum values from the data were therefore used as the constant amplitude load spectrum for the fracture mechanics analysis. In order to somewhat

preserve the sequence of the data, the data were divided into 10 increments. Each increment was then treated as a load spectrum itself. The rainflow analysis was made on each increment to determine maximum and minimum values for the imbedded cycles. The maximum displacement range was subdivided into five blocks of constant amplitude data. The magnitude of the transients in each block was the RMS value of the maximum and minimum values obtained by the rainflow analysis. Likewise, the total number of cycles was the actual number of cycles computed for each block.

CHAPTER V

STRESS ANALYSIS

The second major part of the life assessment analysis procedure is the determination of the operational stresses in critical turbomachine components such as blades and disks. Before any prediction of structural life can be made, the state of stress for the component under consideration must be known.

Finite element stress analysis models are constructed of critical turbomachine components for determination of operational stresses. These stress models can be used for three basic functions in the life assessment analysis. The first is to identify high stress regions in a component. These high stress regions may be the result of geometric stress concentrations such as bolt holes and fillet radii or the method of load application such as concentrated loads. These regions may also be critical locations for existing flaws or cracks. The finite element models must include enough detail (mesh refinement) that stress concentrations can be determined. Second, the stress models are used to determine natural frequencies, mode shapes, and modal participation factors. The mode shapes define the deflected shape of the component for the corresponding natural frequency. The modal participation factors identify the degree to which the individual modes contribute to the response of the component, and can therefore be used to identify which modes of vibration should be included in the analysis. This information is useful in the analysis of the turbomachine

operational test data and can help in analyzing the behavior of the component under load fluctuations. The final function of the stress models is to define the stress distribution in a region of a component assumed to contain a crack. A stress distribution in the region of the crack is required to define a stress field for a fracture mechanics analysis. If significant changes in geometry occur in the regions assumed to contain a crack, the finite element model mesh may have to be refined. Inaccuracies in the computed stress distribution, which is a function of mesh refinement, can cause significant errors in the life assessment analysis.

The finite element stress models are used to compute the steady-state (mean) and dynamic (alternating) stresses in the turbomachine components. The steady-state stresses are usually due to centrifugal and thermal loads. These steady-state stresses are the mean stress level to which alternating stresses are applied for a fracture mechanics analysis. The dynamic stresses are caused by vibratory loads such as forced excitations due to a turbomachine surge. Once the natural frequencies and mode shapes of a component are determined, the dynamic deflection test data can be used to determine the dynamic stresses. Dynamic displacements resulting from one or more of its dominate modes of vibration are assumed at a point on the component where the data are acquired. A stress distribution for the assumed displacement and vibratory modes is computed. An approximation of the dynamic stress distribution can then be determined from the component deflection data by

$$\sigma_D = \sigma_C \Delta_M / \Delta_A \quad (5-1)$$

where σ_D is the dynamic stress, σ_C the computed stress, Δ_M the measured displacement, and Δ_A the assumed displacement to determine σ_C . The dynamic stresses are then added to the steady-state stresses to determine the stress fields for a fatigue analysis.

Turbomachine component stress models can be constructed using computer codes available at AEDC. Of the codes available at AEDC, only the SAP6 structural analysis program had the capabilities for both steady-state stress analysis and dynamic analysis of turbomachine components. Since codes were not available at AEDC for dynamic stress analysis of deflection data, the dynamic stress program (DSTRESS) was developed for analysis of turbomachine components.

Structural Analysis Program (SAP6)

The SAP6 code can be used to determine stress distributions of components under load. SAP6 [56] is a proprietary structural analysis program which was developed at the University of Southern California. The code is a large general purpose finite element program for static and dynamic analysis of continuous elastic structures. This code has been used extensively by AEDC and others for the analysis of structural components.

The finite element approach relies on the direct stiffness method for structural analysis. This method makes a fundamental assumption that the structure can be satisfactorily represented by an assembly of discrete elements. These elements have simplified elastic properties and are connected to represent the continuous structure.

The program first calculates the stiffness of each element and then sums all of the element stiffnesses and loads into a system of simultaneous equations. The code then solves for the unknown displacements and element loads throughout the structure. The element forces are related to the corresponding displacements by the matrix equation

$$\{P\} = [K]\{U\} \quad (5-2)$$

The properties associated with the stiffness matrix, $[K]$, include geometry and mechanical properties. The load matrix, $\{P\}$, includes forces, moments, density, accelerations, pressures and temperatures. The displacement matrix, $\{U\}$, contains either known or unknown displacements and rotations.

The SAP6 code can also be used to determine the natural frequencies and mode shapes for a structure. The code uses the relationships for free vibration of an undamped n degree-of-freedom system. The equations of motion for this system can be expressed in matrix form as

$$[M]\{\ddot{X}\} + [K]\{X\} = \{0\} \quad (5-3)$$

where $[M]$ is the mass matrix, $[K]$ the stiffness matrix, $\{X\}$ the displacement matrix and $\{\ddot{X}\}$ the matrix of accelerations. If we assume harmonic motion, $\ddot{X} = (-\omega^2)X$, and premultiply by M^{-1} , we obtain the relation

$$[M^{-1}K - (\omega_i)^2 I]\{X\} = \{0\} \quad (5-4)$$

Letting $A = M^{-1}K$, we obtain the characteristic equation for the system as the determinate

$$\left| A - (\omega_i)^2 I \right| = 0 \quad (5-5)$$

where I is the identity matrix. The roots of this equation, $(\omega_i)^2$, are called the eigenvalues and ω_i represents the natural frequencies of the system. Upon substitution of the eigenvalues into Equation (5-4), the eigenvectors, x_i , are obtained for the system of equations. The eigenvectors represent the mode shapes for the corresponding natural frequencies of the system. SAP6 can employ either an eigenvalue extraction method or an approximate method (Rayleigh Ritz) to determine the frequencies and mode shapes for the structure.

The SAP6 code contains a library of 1-D, 2-D, and 3-D elements. Different types of elements can be combined as needed to define the structure. All of the elements can be subjected to thermal and inertial loadings. To apply inertial loading, SAP6 calculates the total mass and distributes it equally at the vertices of the elements.

For development of finite element models, a general purpose Interactive Finite Element Mesh Generation Program (IFEMGP) was developed. This program allows the user to generate a finite element model interactively from a time-sharing graphics terminal. The program uses interpretive commands to either define or generate elements and nodes. The model can be plotted in its entirety or in parts as desired to check on its accuracy. Modifications and corrections can be made to the model at any time. The nodes and elements are saved in permanent files which are formatted for the SAP6 code. IFEMGP can also be used to build a SAP6 data input file.

Dynamic Stress Program (DSTRESS)

The dynamic stresses contained in a structure can cause significant load fluctuations that result in crack propagation of existing flaws. For analysis of these stresses, a program was developed to transform measured displacement data to stress. The Dynamic Stress Program (DSTRESS) is based on the concept of finite element analysis. The program can use the same finite element models as constructed for the SAP6 program.

The dynamic equation of motion [49] is defined as

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{F(x,t)\} \quad (5-6)$$

where, in this case, the damping matrix $[C]$ and velocity vector $\{\dot{X}\}$ are included. The set of simultaneous equations given by Equation (5-6), may be uncoupled if real normal modes with viscous damping is considered. In this case the damping is assumed proportional to the mass and stiffness matrices by the expression

$$C = \alpha M + \beta K \quad (5-7)$$

where α and β are constants.

The set of equations contained in Equation (5-6) can be uncoupled by transforming the equations to their principal coordinates. The displacements written in terms of the system eigenvectors are

$$\{X\} = [N]\{\phi\} \quad (5-8)$$

where $[N]$ is the matrix of eigenvectors satisfying the equation

$$[\omega^2][M][N] = [K][N] \quad (5-9)$$

and $\{\phi\}$ is the vector of generalized coordinates. $[\omega^2]$ is the diagonal matrix of system eigenvalues. The matrix $[N]$ contains one of the system eigenvectors (mode shape vectors), which is mass orthonormalized to the remaining eigenvectors. In this case,

$$[N]^T[M][N] = [I] \quad (5-10)$$

where $[I]$ is the identity matrix.

The uncoupled set of equations representing Equation (5-6) can be written as

$$\ddot{\{\phi\}} + [2\omega\xi]\{\dot{\phi}\} + [\omega^2]\{\phi\} = [N]^T\{F(x,t)\} \quad (5-11)$$

The forcing function $F(x,t)$ will generally excite all the modes of the structure. This results through the modal participation factors. The degree of severity to which a particular mode is excited depends on both the spectral content and the spatial distribution of the forcing functions. If the magnitude of the forcing function is proportional to a particular mode shape

$$\{F(x,t)\} = a(t)\{N\}_i \quad (5-12)$$

then a single mode of vibration can be excited. The function $a(t)$ is the time-dependent proportionality factor and $\{N\}_i$ is the i th, eigenvector. If the tip displacement of a blade is known at only one point, it is necessary to assume which mode of vibration is dominant. Using this mode of vibration, the displacement of the finite element model can be obtained from the expression

$$\{X\} = \{N\}_i \Delta_{\text{tip}} / N_{i,\text{tip}} \quad (5-13)$$

where Δ_{tip} is the measured tip displacement, and $N_{i,tip}$ is the value of the i th mode shape at the tip measurement location. Once the node point displacements are known, the complete stress distribution can be obtained by the expression

$$\{\sigma\} = [D][B]\{X\}_i \quad (5-14)$$

where $\{\sigma\}$ is the stress vectors, $[D]$ is the matrix of elastic material coefficients, $[B]$ is a matrix relating the element node point displacements to the strain vectors, and $\{X\}_i$ is the vector of node point displacements for the i th element.

The SAP6 finite element program can be used to obtain the frequencies (eigenvalues), mode shapes (eigenvectors), and the $[D]$ and $[B]$ matrices. These parameters are then used either by DSTRESS or as a guide in determining the mode of vibration and the dynamic stress distribution.

Since the spectral content of dynamic loads may contain many frequency components, the spatial distribution of the loads will not correspond to a particular mode. Since more than one mode will be present, the selection of the dominant mode shapes should be made with care.

It is possible that the tip displacement can be approximated very well by considering one mode shape. The stresses, however, are proportional to the displacement gradients within the structure. The response of the neglected modes may not influence the displacement significantly at a particular node, but may be influential at other nodes away from the region.

The major advantage of the DSTRESS code is that a complete stress field can be determined from a limited amount of data. The dynamic stress field can then be used in a life assessment analysis.

CHAPTER VI

FATIGUE ANALYSIS

The third major part of the life assessment analysis procedure is a fatigue analysis of the critical turbomachine components. Linear elastic fracture mechanics models are constructed of the critical components for determination of the crack propagation lives. Various types of crack geometries are assumed for different regions of the critical components, and crack propagation lives are computed for each case. Using the fracture mechanics approach for fatigue analysis, the definition of failure is based on a predetermined critical crack size. This crack size can be based on the critical stress intensity factor, material thickness, and cross-sectional area required for static rupture. The initial crack size for use in the fracture mechanics analysis should be the smallest crack that could be found with a high degree of confidence using NDE techniques.

The operating stress spectrum used for the fatigue analysis can be determined from the turbomachine component measured deflection data and stress analysis. The steady-state stress spectrum is determined directly from the stress models. The dynamic stress spectrum is determined from both the stress models and the dynamic data.

The two approaches for analyzing the deflection data will result in different dynamic stress spectra. The approach using the LOADHIST code gives a dynamic stress spectrum based on random deflection data. These data can be modified to represent a blocked load spectrum for

the component. Various periods of operation (start-up, shutdown, normal operation, surge, etc.) are included in this spectrum. The blocked spectrum (blocks of constant amplitude data required by the fracture mechanics code) minimizes the computational requirements compared to a cycle-by-cycle analysis. The DSTRESS code and the preprocessor for the fracture mechanics code converts the blocked deflection spectrum into a dynamic stress spectrum. The approach using the DAP code gives a dynamic stress spectrum based on an amplitude versus frequency analysis of the same random data. The deflection amplitude of the dominant frequencies is converted to dynamic stresses using the DSTRESS code. A dynamic stress spectrum is then constructed using the dynamic stresses and frequencies. The steady-state distribution was used with both approaches to determine the final dynamic stress distribution for a fracture mechanics analysis.

Four fracture mechanics analysis computer codes that could be acquired for use at AEDC were evaluated. These analysis codes include the Hyperbolic Sine code, CRACKS code, CRACKGRO Code, and BIGIF code. The Hyperbolic Sine code was developed for analyzing laboratory material test specimens and would require major modifications for analysis of turbomachine components. The CRACKS code was developed for analyzing aircraft structures and requires large quantities of input data. The CRACKGRO code is a streamlined version of the CRACKS code, and has significant limitations regarding the types of structures and crack geometries it can handle. Both the CRACKS and CRACKGRO codes would require significant modifications for analysis of turbomachine

components. The BIGIF code is commercially available and was developed by Failure Analysis Associates of Palo Alto, California. This code can be used to analyze many types of structures. It has been used to analyze turbine engine rotating components, and would require little if any modifications for analysis of AEDC turbomachine components. Therefore, the BIGIF code was acquired for development of the turbomachinery life assessment analysis procedure.

Fracture Mechanics Program (BIGIF)

The BIGIF (Boundary Integral Equation Generated Influence Functions) program [55] is a linear elastic fracture mechanics code which can be used to determine crack growth lives of structures for a wide range of two- and three-dimensional stress fields, cracks, and structural geometries. BIGIF uses many of the fracture mechanics principles previously described in the calculation of crack growth due to cyclic loads.

BIGIF can account for general tensile elastic stress gradients due to notches, holes, temperature gradients, and residual stress fields in the vicinity of the crack. The simultaneous effects of several cyclic load transients can be modeled even though they may have different stress fields. Also, BIGIF can account for first-order plasticity effects on subcritical crack growth provided that the inelastic deformation is contained enough to provide a near-linear load displacement curve.

The calculation of stress intensity factors by BIGIF are made by using the Influence Function (IF) method [55]. The strengths of the

IF method are that

1. the application of elastic superposition allows the use of the "uncracked" stress distributions in the K analysis and,
2. the influence function is invariant with stress, can be computed for very simple loading states, and provides a method to calculate the effect of the crack in redistributing any stress field.

The influence function, h , is a function of position x along a crack, the displacement boundary condition u , and geometry of the structure and crack g . The calculation of K for the general class of two-dimensional problems in Mode I cracking (Figure 7) is

$$K = \int_L h(x,u,g)\sigma(x) dx \quad (6-1)$$

where L is the crack line and $\sigma(x)$ is the uncracked stress distribution normal to the crack face. Once h has been determined for a crack geometry, the value of K can be computed by the integration of Equation (6-1). For three-dimensional problems, K is a function of the crack front position. In this case, Equation (6-1) is replaced with more complex formulas with integration over the crack surface area or face. The IF approach is to define one or more local averages of K ($\bar{K}$) over the defined portions of the crack front. These local averages are also related to and may be calculated directly from the strain energy release rates resulting from defined crack growth and shape changes. Each dimension or parameter used to describe crack growth or shape change is called a degree-of-freedom (DOF) on K . When the crack front

is discretized into enough DOF, the $\bar{K}$ values will equal corresponding local values of K . A major feature of the If method directly responsible for achieving high numerical accuracy for K computations is that a simple stress field is used to generate the crack opening displacement data necessary for calculation of h . Also, the boundary-integral equation method which was used in BIGIF code [6] is a very expedient method for determining the required crack opening displacement data.

The BIGIF program contains a large library of typical crack models that can be accessed by the user. Information supplied by the user includes: material properties, uncracked stresses, choice of crack model, initial and final crack sizes, and amplitude and number of load cycles in each transient. The user is given a choice between several crack growth rate models which include: Paris law, Forman's rule, and some modifications to these models which account for near threshold effects and stress-ratio effects.

A preprocessor was developed to convert the data from the DSTRESS and LOADHIST codes to the BIGIF format. This program also combines the steady-state and dynamic stresses for the BIGIF analysis. The preprocessor provides a free-format mode of input for all data and constructs a formatted data file.

CHAPTER VII

INSPECTION INTERVAL

The final part of the life assessment analysis procedure is the determination of an inspection interval for the turbomachine. An inspection interval can be defined as time or cycles (N_p) required to grow a crack from an initial size after inspection to a predetermined critical size with a safety factor (SF)

$$\text{Inspection Interval} = N_p / \text{SF} \quad (7-1)$$

As mentioned previously, the safety factor is determined by conducting a life-cycle-cost (LCC) analysis for the turbomachine components. A factor that should be considered in the LCC analysis is the repair and/or replacement costs for the turbomachine in the event of a structural failure of a component and resulting damage to the other components. Other factors that should be included in the LCC analysis are inspection costs and the costs associated with engine test delays for facility turbomachine repair or replacement.

The crack propagation rate for a component is heavily dependent on the types and numbers of major load events (i.e., starts and stops, surges) encountered by the turbomachine. Therefore, if accurate records are kept on major load events experienced by the turbomachine, an accumulated crack growth for the component could be computed based on these loads. This method would allow a floating inspection period which is modified for actual usage. This could result in extended

inspection intervals for operating periods with no surges while a period of many surges would require a shortened inspection interval.

CHAPTER VIII

DEMONSTRATION CASE FOR LIFE ASSESSMENT

ANALYSIS PROCEDURE

A review and evaluation of ETF turbomachines were conducted for selection of a demonstration case for the life assessment analysis procedure. The ETF basic plant has four centrifugal flow air compressors and six centrifugal flow exhauster compressors. These turbomachines were manufactured by the Brown-Boveri Corporation, and were designed in the 1930's. There is very limited design information available on these compressors, and no detailed component drawings. Also, these compressors have been operating for approximately 30 years with few if any structural failures. The ETF addition plant has three axial flow air compressors and two axial flow exhauster compressors. These turbomachines were manufactured by Allis Chalmers Corporation, and were designed in the 1940's. A partial set of detailed drawings was available on the ETF addition plant exhausters (XS-1 and XS-2). The XS-2 exhauster was scheduled for a teardown inspection and maintenance period during the same time frame that this project was conducted. This teardown inspection would allow time for instrumentation to be installed in the XS-2 exhauster. Therefore, the ETF XS-2 exhauster was selected as a demonstration case for the AEDC turbomachinery life assessment analysis procedure.

Description of XS-2 Exhauster

The exhauster gas system for the ETF addition plant consists of

two identical exhauster compressors (XS-1 and XS-2). The exhauster system pumps exhaust gases from turbine engines and rocket engines during testing. These exhausters can operate in parallel and discharge to atmosphere or be operated in series with the ETF basic plant exhaust system. The XS-2 exhauster is an Allis Chalmers Model VA-3307 axial flow compressor driven by a 24,500 HP synchronous electric motor. The exhauster has a seven-stage, welded drum rotor with 69 blades per stage, six rows of variable stator vanes, and one row each of inlet guide vanes and flow straightening vanes with 70 blades per row. A cross-sectional schematic of the XS-2 exhauster is shown in Figure 16. The operating characteristics of the exhauster are:

Rotational Speed	1450 RPM
Design Airflow	910,000 CFM
Maximum Discharge Temperature	480°F
Maximum Inlet Temperature	190°F
Maximum Inlet Pressure	10 PSIA
Maximum Discharge Pressure	14.4 PSIA
Design Point Pressure Ratio	2.6
Design Point Inlet Pressure	5.53 PSIA
Design Point Discharge Pressure	14.4 PSIA
Surge Pressure Ratio	3.1

The XS-2 exhauster preventive maintenance program includes an inspection of the exhauster blades approximately every 2000 operating hours. An average operating year for XS-2 includes approximately 100 operating periods and 700 operating hours. The remainder of this

chapter is a summary of the significant events in the operational and maintenance history of XS-1 and XS-2 exhausters.

The assembly and erection of the exhauster system were completed in 1958, and the performance and acceptance tests were completed in 1959. The first inspection of the XS-2 exhauster was conducted in 1959. Considerable erosion of the first four stages of rotor blades was discovered, and some of the inlet guide vanes and discharge straightening vanes were cracked. In 1960, a consultant from the Ohio State University recommended that the exhauster be rebladed with 403 stainless steel blades.

During the early part of 1961, a fatigue failure of a first-stage rotor blade occurred in the XS-1 exhauster. The failed blade propagated through the entire machine and destroyed all rotor and stator blades after the first stage. Some time after the failure in XS-1, the XS-2 exhauster was inspected to determine the condition of its rotor blades. A metallurgical inspection found a crack in the first-stage rotor blade and in a flow straightener blade. Five first-stage blade attachment sections were removed from XS-1 and installed in XS-2, and the remaining first-stage blades in XS-2 were burned-off and blade roots were ground smooth. The XS-2 machine had limited operation until XS-1 and XS-2 were rebladed in 1962 with 403 stainless steel blades.

In mid-1963, excessive vibration was noticed during operation of XS-2. Inspection of the machine revealed an excessive rub condition of the stub portion of the drive shaft. A representative from Allis Chalmers determined that the cause of the vibration was a shaft

alignment problem due to machining out the shaft eccentricity. The shaft was straightened and aligned by Allis Chalmers.

In early 1965, XS-2 experienced extensive blade damage resulting from water injection due to faulty barometric well operation. There were three broken stainless steel rotor blades in the first stage and one broken carbon steel stator blade (original blade) in the first row. The remaining damage consisted of bent blades for 50 percent of the stator blades and 70 percent of rotor blades. The carbon steel stator and rotor blades removed when XS-2 was rebladed in 1962 were reworked and installed in the second through seventh stages of XS-2. New stainless steel rotor blades were installed in the first row of XS-2. The damaged stainless steel rotor blades were reworked and stored as spare parts. The installation of the blades and reassembly of XS-2 were completed in mid-1966. XS-2 was operated with a combination of stainless steel and carbon steel blades.

The XS-1 exhaustor experienced a bearing failure in 1974 which resulted in a warped drive shaft. The drive shaft was straightened and cracks were machined out by the Norfolk Naval Shipyard. The XS-1 was down from 1974 to 1979.

The rotor was removed from XS-2 in the fall of 1982 for repair of one of its main bearings. During a magnetic particle inspection of the exhaustor rotor, two cracked carbon steel blades were found in the seventh stage of the rotor. A replacement set of 403 stainless steel blades had been acquired in 1977, and these blades were installed in the XS-2 rotor. Installation of the blades and reassembly of XS-2

were completed in late 1982. The XS-2 has been operating since early 1983 with a new set of stainless steel blades.

XS-2 Exhauster Life Assessment Analysis Procedure

The AEDC turbomachinery life assessment analysis procedure was applied to the XS-2 exhauster. A first-stage rotor blade was selected as a demonstration case for the analysis procedure. The reasons for selecting this blade are 1) a first-stage blade would be more convenient for instrumenting to acquire operational data, 2) a complete set of detailed drawings was available on the front stages of the rotor and not available on the rear stages, and 3) the first-stage blade was longer than the others, and its centrifugal stresses would be greater. The seven-stage disk drum rotor was considered in the preliminary analysis of the XS-2. An axisymmetric stress model was constructed of the disk drum rotor, and the predicted steady-state stresses were well below yield and below the blade stresses. Also, the surge loads in the disk drum rotor would be significantly smaller than the surge loads in the first-stage rotor blade. Therefore, only the first-stage rotor blade was considered for the demonstration case of the life assessment analysis procedure.

The XS-2 first-stage rotor blade is a forged, Type 403 stainless steel blade. A sketch of the blade is shown in Figure 17. The overall length of the blade is 16.9 inches with an airfoil height of 13.4 inches and an attachment root height of 3.5 inches. Other important blade dimensions are shown in Figure 17. The hole in the blade attachment root is for weight saving, and the blade total weight is 19.7

pounds. The blade twist from the root to tip is approximately 21.13 degrees.

The basic steps used in the life assessment analysis of the XS-2 first-stage blade are summarized in Figure 18 and are described as follows:

1. Acquire Dynamic Data - Using a prototype noninterference deflection measurement system, blade-tip dynamic deflection data were acquired on the first-stage blade. The dynamic data were acquired during both normal and surge operation of the turbomachine, and were used in the determination of blade dynamic (surge) stresses. In addition, rotor speed data were acquired and used in the blade stress analysis for calculation of steady-state stresses. From an assessment of limited temperature information on the exhaustor, it was assumed that blade thermal gradients were small and metal temperature test data would not be required.
2. Perform Frequency Analysis - A frequency analysis of the blade-tip dynamic deflection data was conducted using the DAP code. This analysis identified the significant deflection amplitudes associated with blade natural modes. Deflection amplitudes were determined for both normal and surge operation of the exhaustor. This amplitude versus frequency information was used in the determination of blade dynamic (surge) stresses and in the fracture mechanics analysis of the blade.

3. Determine Deflection Spectrum - The blade-tip dynamic deflection data were converted into a blocked deflection data spectrum using the LOADHIST code. Both normal and surge operational data were included in the data spectrum for the fracture mechanics analysis.
4. Construct Finite Element Models - Two three-dimensional, finite element models were constructed of the blade using the IFEMGP code. One of the models was used in the steady-state and dynamic stress and vibratory analysis of the blade. The second model was used in the steady-state stress analysis of blade attachment root.
5. Perform Steady-State Stress Analysis - A steady-state stress analysis of the blade and attachment root was conducted using the blade finite element models and the SAP6 code. The primary load used in the stress analysis was the centrifugal load resulting from the blade rotational speed. The steady-state stress analysis determined the maximum stresses and stress distributions in the blade and attachment root. The blade steady-state stresses serve as the mean stress level to which the dynamic (alternating) stresses are applied for the blade fatigue analysis.
6. Perform Vibratory Analysis - A vibratory analysis of the blade was conducted using the blade finite element model and the SAP6 code. The vibratory analysis determined the blade natural frequencies, mode shapes, and modal participation

factors. The mode shapes define the deflected shapes of the blade for the corresponding natural frequencies, and the participation factors identify which blade mode of vibration contributes the most to the total response of the blade. This information was used in determination of the blade dynamic stresses.

7. Determine Dynamic Stresses - The blade dynamic stress distribution was determined using the blade finite element model, the DSTRESS code, and the blade-tip dynamic deflection data. The dynamic stress distribution was determined for both random vibratory and surge loads. The dynamic (alternating) stresses were then added to the steady-state (mean) stresses to determine the stress fields for the blade fatigue analysis.
8. Perform Fatigue Analysis - A fatigue analysis of the blade was conducted using the computed stress distributions and the BIGIF code. The steady-state and dynamic stress distributions were used to construct two load history representations for the fracture mechanics analysis. The first consisted of the combined steady-state and dynamic stress spectrum based on the random deflection test data (LOADHIST analysis). The second consisted of the combined steady-state and dynamic stress spectrum based on the amplitude versus frequency analysis (DAP analysis) of the same deflection data. The critical stress region (combined high steady-state and dynamic stress) of the blade was the airfoil root; therefore, linear elastic fracture mechanics models were constructed

of this region. The number of XS-2 operating hours to grow a crack to a critical size was determined for different initial crack sizes, airfoil root locations, load history blocks, and number of surge events per operating period.

9. Recommend Inspection Interval - An approach was developed for determining the inspection interval for the XS-2 exhaustor based on the blade fatigue life analysis and an acceptable safety factor.

CHAPTER IX

XS-2 TEST PROGRAM

The objective of the test program was to acquire operational data on the XS-2 exhaustor rotor for determination of dynamic stresses in the first-stage rotor blade. A prototype noninterference stress measurement system (NSMS) developed for turbine engine compressor blades [34] was used for the test program. The NSMS was used to detect the passing of blade tips, and the detected signals were converted to blade-tip deflections using special processing algorithms. Tip deflection data were displayed on-line for monitoring the blade activity, and were also recorded on magnetic tape for off-line data reduction and analysis.

Data Instrumentation System

The configuration for the test program instrumentation system is shown schematically in Figure 19. Five blade-tip sensors were mounted in the exhaustor case on the periphery of the first stage at 15 degree spacings. Each sensor detected the passing of all 69 blade tips of the first-stage blade. Signals from the sensors were detected by the photomultiplier tubes (PMT) and then routed to the NSMS in the exhaustor control room for conditioning and processing. A one-per-revolution (OPR) sensor located approximately 50 feet upstream of first-stage rotor was used to detect the rotor shaft position.

The OPR signal was processed in the NSMS to determine blade location. The inability to locate the OPR sensor at the exhaustor inlet

could have caused measurement error in the blade-tip deflection data. An analysis was performed to determine the torsional deflection of the exhaustor drive shaft between the OPR sensor and first-stage rotor. The computed torsional deflection at 100 percent torque was approximately three mils. During exhaustor surge, the drive shaft will load and unload due to blade stalling resulting in shaft oscillations. However, the maximum peak-to-peak torsional deflections of the drive shaft would be six mils which is below the sensitivity of the data measurement system.

A representative drawing of the blade-tip sensing system is shown in Figure 20. The light source was connected to a bifurcated arm of the fiber-optic cable. The light was transmitted through the input fibers of the cable and focused by a set of lenses on the blade tips. The reflected blade-tip signal was transmitted through the output fibers to the PMT for detection and signal amplification before processing.

The NSMS data acquisition software had the capability of acquiring data in two modes, nonintegral and integral. The nonintegral mode occurs when the blade vibration frequencies are not an integer multiple of rotor speed and the integral case occurs when the blade vibrations are an integer multiple of rotor speed. In the nonintegral mode, data were recorded from one blade-tip sensor for a selection of any five blades per revolution. In the integral mode, data were recorded from four blade-tip sensors for a selection of any four blades, at the rate of one blade per revolution. Due to hardware and software problems, data were acquired primarily in the nonintegral mode.

Data Acquisition

When the sensors were installed prior to starting the exhauster, the sensors became cloudy during the exhauster start-up and initial operation because of water and dirt contamination. The contaminated sensors resulted in low level, noisy blade-tip signals. To minimize the exposure to contamination during start-up, the sensors were installed after the exhauster reached synchronous speed. Also, only those sensors needed for the predetermined mode of data acquisition were inserted, leaving spare sensors for replacement any time water or dirt contaminated the sensors. Each sensor used was disassembled and cleaned after the test in preparation for the next run.

The NSMS required that adjustments be made to the processor unit for sequencing one per revolution and blade-tip timing signals needed for proper system operation. Once the system was initialized, data could be recorded at any time. Real-time data displays informed the operator of blade activity.

Blade-tip deflection data were acquired during eight operational periods of the exhauster. Initially, transient data points were started at the beginning of a major event such as a surge or rocket firing. After failing to acquire data during a surge, it was decided to take transient data points of approximately five minutes in duration throughout an operating period. Using this approach, nonintegral data were acquired during both normal and surge operation of the exhauster at synchronous speed. Also, data were acquired during spooldown of the exhauster. The most comprehensive transient data

point which included presurge, surge, and postsurge activity was selected for the life assessment analysis of the XS-2 blade.

The prototype NSMS could only resolve blade deflections to approximately six mils because of the large diameter (130 inches) of the bladed disk rotor and the number of blades (69 per stage). The NSMS maximum capacity for data samples is 100,000 counts per revolution, with a restriction of a whole number of counts between adjacent blades. The sampling rate was set for 1000 counts between blades at synchronous rotor speed. Also, the approximate distance between adjacent blades was six inches. Therefore, the resolution of blade-tip deflections was six mils (6 inches per 1000 counts).

The XS-2 low rotor speed and large rotor diameter resulted in a low data sampling rate for the NSMS. The 24 revolutions per second rotor speed resulted in a maximum frequency window of 12 Hz for acquisition of blade-tip deflection data. Therefore, the blade frequencies measured by the NSMS were aliased if vibrational frequencies larger than 12 Hz were present.

CHAPTER X

RESULTS AND DISCUSSION

The AEDC turbomachinery life assessment analysis procedure was applied to the XS-2 first-stage rotor blade. Included in the life analysis procedure were 1) an analysis of the blade-tip dynamic deflection data, 2) a stress analysis of the blade, including attachment root, 3) a fatigue analysis of the blade airfoil and attachment root using fracture mechanics models, and 4) an approach for determining the inspection interval for the exhaustor.

XS-2 Test Data Analysis

Deflection data were acquired for several blades on the XS-2 exhaustor first-stage rotor. A review of the data revealed similar trends for all blades. The data appeared to have a low cycle wave superimposed with high cycle random data, Figure 21. The low cycle wave continued throughout the transient data and increased in amplitude at the start of the surge period. This was characteristic of all surge data. Insufficient surge data were obtained to conclude this would always be the case. A surge period of 35 seconds was defined for this analysis (Figure 21). A 15-second interval of this data contained the dominant surge event.

Histograms were constructed using the LOADHIST code to show additional characteristics of the data shown in Figure 21. The histograms were similar to that of a sine wave with random data, Figure 22a. This shape was consistent with data taken at any sub-interval chosen

from the spectrum except the surge period. The surge period histogram is similar to the binomial distribution normally obtained for random data, Figure 22b.

Since the instrumentation sensors measured tip deflections from calibrated positions of the blade, all data represents vibrations about the steady-state deflected position. It also represents vibrations that are not in phase with the rotational speed of the machine (nonintegral data). If vibrations were in phase with speed (integral data), the blade would be in the same deflected position each time a data point was taken. Nonintegral data will appear either as random data or low cycles within the spectrum. To impose a false low cycle within the spectrum, the blade tip must have a slight shift in its cyclic position relative to the sensor during each revolution. This could be the case of the low cycle wave shown in the data spectrum. There are other possible reasons for the low cycle wave, such as the influence of vibration frequencies in the rotor shaft, rotor, exhaust casing, and the sensitivity of the sensor and measurement system.

XS-2 Blade Stress and Dynamic Analysis

For the life assessment analysis of the XS-2 blade, an accurate representation of the steady-state and dynamic stress fields for various regions in the blade were required. To perform this analysis, the finite element (FE) approach was used. Finite element models were constructed of the blade for use in both the steady-state (mean) and dynamic analysis. This analysis determined the mean and dynamic stress fields associated with any desired region of the blade.

The SAP6 code has a library of element types including an eight-noded hexahedron which was chosen for the blade models. This solid element can be collapsed to a six-noded wedge when the geometry requires its use. The hexahedron allows a linear variation in stress while the wedge allows only constant stress. Since the hexahedron can calculate stresses at two locations in the element (centroid, face centers) compared to a centroidal stress only for the wedge, the hexahedron was used as the primary element in the models.

This element has three degrees-of-freedom (DOF) associated with each node. These DOF include the displacements along the x, y, and z coordinate axis. Stress output for the element can be specified for a given face or centroid of the element. The stresses on the face of the element are given in the plane of the face and in the element local coordinate system while the centroid stresses are in the global coordinate system.

Two three-dimensional, FE models of the XS-2 exhaustor blade were constructed. The first model contained more mesh refinement in the airfoil region and a coarse representation of the base while the second model had more mesh refinement in the base attachment region and a coarse airfoil representation. This was necessary due to a SAP6 limitation in the total number of DOF associated with a model (maximum of 9999 DOF). Steady-state stresses were determined for each model. The models were then used to determine dynamic characteristics of the blade as required for the program DSTRESS.

The model shown in Figure 23a concentrated the mesh refinement in the airfoil root region of the blade. This model contained 3632 nodes

and 2770 elements. The base of the blade was assumed to be a thick plate of constant depth. Restraints were applied to the outer regions of the base in the global x , y , and z directions. These restraints simulated the connection of the blade section to the attachment region. The central region of the base, which would normally be over the weight saving hole in the attachment region, was not restrained.

The blade airfoil was modeled to minimize the element aspect ratio in the airfoil region. More accurate stress predictions can then be made in the airfoil root region of the blade. The airfoil root fillet was also included in the model.

The model shown in Figure 23b concentrated the mesh refinement in the attachment region of the blade. This model contained 3228 nodes and 2426 elements. The airfoil of the blade had a coarse mesh, thus allowing a refined mesh in the attachment region while staying within the DOF limitations of the SAP6 code. The restraints applied to the attachment region were chosen according to the results of a preliminary analysis assuming centrifugal loading only. A preliminary execution of the model had the x -axis face restrained in the x -direction, y -axis face restrained in the y -direction, and the z -axis face restrained in the z -direction. The results of this run were used to determine which zones of the attachment region were in compression. The restraints on the tension regions were removed because tensile loads cannot be transferred across the interface of the blade attachment region and the disk rim slot. The final restraints applied to the attachment region, as shown in Figure 23b, included radial restraints on the attachment

contact planes, axial restraints on the top and a portion of the bottom flanges, and tangential restraints on one face of the attachment region.

The purpose of the detailed attachment region model was to give more accurate stress predictions in this region of the blade. This model can be used to obtain stress profiles throughout the attachment region. The airfoil region has a coarse mesh and does not include the airfoil root fillet; however, it does represent all of the mass of the airfoil region required for the centrifugal loading of the blade attachment region.

The steady-state loading conditions used for the blade analysis included a centrifugal loading at 1440 RPM and a uniform temperature of 200°F. Blade steady-state pressure loads were examined using the COCODEC compressor design code [8] and found to be insignificant. Therefore, the centrifugal loading was the dominant loading on the blade and the steady-state pressure loads were not included in the analysis.

The SAP6 steady-state analysis indicated that the highest stress distribution was just above the fillet in the airfoil root of the blade model. The radial stress profiles obtained along the chord of the plane, Figure 24, varied from a maximum of 22.5 Ksi at the trailing edge and a maximum of 18.0 Ksi at the leading edge of the blade. Typically, the radial stress distribution had similar characteristics for higher span locations of the blade. The stress amplitude varied to zero at the blade tip.

Stress profiles through the thickness of the blade, Figure 25, show a radial surface stress which decreases toward the center and returns to a higher stress at the opposite surface. This stress distribution appears to be typical except toward the leading and trailing edges of the blade. The critical region chosen for a fatigue analysis of the airfoil is above the root fillet and has the steady-state stress distribution shown in Figure 25.

The SAP6 code was also used for analysis of the attachment region. In this case, several areas were identified as possible high stress regions. The radial stress in all cases was dominant for the attachment region as well as the airfoil section.

The stress profile determined around the weight saving hole in the attachment region, Figure 26, varied from a minimum stress of -6.6 to a maximum stress of 31.6 Ksi. In the 360-degree loop the stress profile had two cycles of an alternating stress pattern. The point of maximum stress (31.6 Ksi) around the hole was investigated in a horizontal and vertical projection to the outer surfaces (Figure 27). In both cases the stresses dropped below 15.0 Ksi at the outer surfaces. A stress profile was also determined through the webs just above the contact surface, Figure 28. In this case, the maximum stress approached 62 Ksi and was concentrated at the outer surface of the web. The load was much greater in one web compared to the other indicating a significant effect due to the tangential component of the load vector or possibly temperature effect due to model restraints. The 62 Ksi peak stress was investigated further by determining the stress profile

extending from the bottom to the top of the attachment region, Figure 29. This profile shows that the maximum stress peak at 62.1 Ksi was at the fillet region next to the contact plane. The logical plane of the attachment region for a fatigue analysis is web B shown in Figure 28.

The choice of regions for the fatigue analysis, however, must also include maximum stress regions from a dynamic analysis. If the dynamic stress is much larger than the steady-state stress, it may control the combined maximum stress.

The dynamic analysis of the blade required a knowledge of the deflected shape of the blade for various natural frequencies. It was determined that the first five mode shapes would cover the dominant modes of vibrations. The SAP6 code and the blade FE model shown in Figure 23a were used to perform a force response analysis which determined natural frequencies, mode shapes, and modal participation factors for the first five modes of vibration. The results of this analysis, Table 1, includes frequencies varying from 177.0 Hz to 1993 Hz. The basic mode shapes included first and second bending and first and second torsion. The modal participation factors varied from 0.1097 to -0.00242 with the first bending mode having the largest response (a factor of 1.75 over the second largest response). The x-direction gave the largest participation factors and coincides with the direction of the blade-tip measurement system. This is the tangential (circumferential) direction of the blade motion. The z-direction (axial) gave the second largest factors and was 48 percent of the x-direction modal

participation factors for the first mode of vibration. The y-direction (radial) factors were significantly less than the others.

The blade-tip measurement sensor was positioned over the geometric axis of twist for the blade. This axis also appeared to be the axis of untwist and the rotational axis for the torsional mode of vibration of the blade. A single sensor would not detect much torsional deflection but would provide bending information. To confirm the influence of the torsional axis of twist, the IFEMGP program was used to plot the FE predicted torsional mode shapes over the undeformed shape. These plots gave very little shift in the geometric axis of twist. The participation factors predict that the torsional modes do not contribute as significantly to the blade response as the bending modes. The first bending modal participation factor is 6.4 times larger than the first torsional factor; as a result, the torsional modes were excluded from the analysis.

For determination of dynamic stresses, the program DSTRESS was given a mode shape and an x-displacement defined for the point on the blade at which the deflection data were taken. This analysis was made by assuming a 10 mil displacement for the selected point. The dynamic stress distribution corresponding to the actual data was then based on a ratio of the actual displacement to the 10 mil displacement. The stress spectrum for the fatigue analysis consisted of the superposition of the steady-state and dynamic stresses.

The first bending mode stresses through the airfoil for the assumed 10-mil displacement gave a bending stress variation of

approximately four Ksi tension to four Ksi compression, Figure 30. The first bending mode was assumed to be the deflected shape of the blade as predicted by the participation factors.

In general, the profiles determined for the dynamic stresses in the attachment region gave stresses less than one Ksi. The conclusion was that dynamic stresses in the airfoil are much greater than the attachment region. Therefore, the airfoil root is the most critical location for crack growth in the blade, and it is the region that should control the structural life of the blade.

XS-2 Blade Fatigue Analysis

The fatigue analysis of the XS-2 blade was conducted using the computed steady-state and dynamic stress distributions and the BIGIF fracture mechanics codes. Limited fatigue property information for the blade material (Type 403 stainless steel) was available at AEDC. Therefore, Failure Analysis Associates (FaAA) of Palo Alto, California was contracted to determine the appropriate da/dn data for the analysis.

The data spectrum shown in Figure 21 was used with the LOADHIST program to determine a blocked load spectrum to represent the random data. The data covered a time span of 220 seconds and had a maximum deflection range of 628 mils. Due to the process of data blocking, some of the shift in the data was removed which reduced the maximum deflection range to 537 mils. This deflection, however, includes errors due to lateral blade shifts, pressure fluctuations, and response time errors associated with the data measurement system. For a

conservative analysis, no attempt was made to remove these errors. The final blocked random data determined by LOADHIST are shown in Table 2.

For the fracture mechanics analysis, the data were considered in a conservative manner by assuming it to be random at the first bending mode. In this way, the data were used as shown in Figure 21. However, it is doubtful that the maximum surge amplitude shown in the deflection spectrum can be considered true dynamic deflection. Some of this amplitude results from the errors associated with the data measurement system.

The DAP program was used to determine the dominant frequencies in the data. The analysis of several blades gave consistent results. The data shown in Figure 21 were analyzed, resulting in the amplitude-frequency plots shown in Figures 31 and 32, for two different time spans in the spectrum. The maximum frequency shown in these plots is 12 Hz (results from the sampling rate of 24 points per second) which represents the Nyquist frequency (folding frequency) associated with the data. It is also obvious from these plots that if the blade frequencies are more than 12 Hz, they must be represented as lower frequencies in the plots (i.e., fold back to lower frequencies in the Fourier Analysis due to aliasing).

As noted previously, the error associated with the data measurement system is approximately six mils. The amplitude shown in the amplitude-frequency plots is half peak-to-peak. Therefore, deflection data less than 12 mils peak-to-peak are considered noise associated with the data measurement system. Due to the system error, there are

only four dominant frequencies associated with the data. These frequencies include 0.1, 5.0, 8.9, and 11.9 Hz which represent excited natural modes of the blade.

The finite element results shown in Table 1 were folded back to the frequency range of the DAP analysis. This analysis predicts the first bending mode (177.0 Hz) to be 9.0 Hz on the 12.0 Hz frequency scale. The frequency-amplitude plot of Figure 32 gives a peak amplitude of approximately eight mils at 8.9 Hz. When comparing this amplitude with other frequencies there is little deviation in amplitude except at 0.1 Hz. Modal participation factors predict the response expected from the natural modes of vibration. The factors shown in Table 1 predict the amplitude of the first bending mode to be 1.75 times larger than the next largest amplitude for the direction of displacement measurements (x-direction). Since the DAP analysis gives an amplitude of 18 mils at 0.1 Hz (Figure 32), it is concluded that this was the first bending mode of vibration. Also, the participation factors predict that 5.0 Hz is combined bending and torsion, 8.9 Hz is second bending, and 11.9 Hz is the first torsional mode.

The deviation between the blade natural frequencies predicted by SAP6 and DAP could be due to 1) stress stiffening of the blade due to centrifugal forces, 2) the imposed boundary conditions on the finite element blade model, and 3) the overall mesh refinement of the model. The SAP6 program does not compute stress stiffening for the blade under centrifugal load. It is doubtful that the finite element model can be constrained exactly as the blade is restrained in the disk. The element size chosen for the model also has an effect on the final

results. Previous experience with other blade models using the solid element is that larger element sizes will increase the calculated natural frequencies. In general, these differences are small; however, a two percent error in the calculated frequencies (i.e., two percent of 177 is 3.5 Hz for the first mode of vibration) can result in a 29 percent error in the amplitude-frequency plots constructed by DAP ($3.5/12.0 = 0.29$). In this case, the Nyquist frequency has a significant effect on the location of the natural frequencies in the amplitude-frequency plots computed by DAP when comparing with calculated values. As a result, the participation factors were used to determine the modes of vibration from the data analysis.

If the surge time increment is included in the analysis, DAP predicts the same frequencies as it did for normal operation. However, two additional frequencies occurred which are associated with the surge, Figure 32. These low frequencies of 0.25 and 0.5 Hz have amplitudes of approximately 14 mils. To reduce the averaging effect of the DAP program, a small block of data (22-second intervals) was used to step through the entire spectrum. In this case, the amplitude of all frequencies except those which appeared during the surge had constant amplitudes. For example, Figure 33 covers the time period 140 to 161 seconds and has a maximum surge amplitude of 62 mils. Figure 34 covers the time period of 150 to 171 seconds and has a maximum surge amplitude of 55 mils. Since these frequencies are in the same region of the amplitude-frequency plot as the previous first bending mode, it is assumed they are excitations of that mode.

The amplitude data shown in Figures 31 through 34 represent half the peak-to-peak values. As a result, DAP predicts a maximum deflection of 124 mils compared to the random data prediction of 537 mils.

If 537 mils were used in a static analysis (first mode bending), the stresses predicted would exceed the steady-state stress levels significantly. Since in some cases the stresses would approach the yield stress for the 403 stainless steel, this deflection appears unrealistic. The stress predictions based on 124 mils would be less than yield but would reach stress levels comparable to the steady-state analysis. Based on the past performance of the XS-2 blades, the DAP predictions appear to be more reasonable. Although life assessment calculations were made with both deflection predictions, the DAP 124-mil prediction was concluded to be the most realistic. This conclusion is supported by the fact that the older blades in the exhaustor were of carbon steel with a much lower yield stress than the present blades. These blades would not have performed structurally if the 537 mil deflections occurred.

Two approaches were used in constructing fracture mechanics models of the XS-2 blade. The first approach used the random load spectrum determined from the LOADHIST analysis, and the second approach used the FFT load spectrum determined from the DAP analysis.

Failure Analysis Associates (FaAA) conducted a literature search for appropriate da/dn data for the fracture mechanics models. From their study, FaAA recommended the modified Paris rule option of BIGIF to account for near-threshold and stress ratio (R) effects. The

material properties recommended were determined at the same stress ratio ($R = 0.1$), and they included: $C = 8.343(10)^{-10}$, $n = 2.529$, and $\Delta K_{th} = 7.0 \text{ Ksi-}\sqrt{\text{In}}$ for typical values. For a worse case analysis, they recommended: $C = 1.969(10)^{-9}$, $n = 2.54$, and $\Delta K_{th} = 5.0 \text{ Ksi-}\sqrt{\text{In}}$. Because of the controlled fabrication requirements for the blades, the typical material property values were used for the fatigue analysis.

The BIGIF option based on a modified Paris rule which accounts for near-threshold and stress ratio effects was used. The stress fields used in the analysis were the superposition of the steady-state and dynamic stresses for first bending mode. Since the random data approach has transients with stresses above yield, a study was made with BIGIF code to determine if the elastic-plastic redistribution of stresses would result in significant differences in the life predictions. The result of this study was that a residual stress was created that should be included in the superposition of dynamic and steady-state stresses. Although the residual stress was small, the crack propagation life was increased. Exclusion of the elastic-plastic redistribution was determined to enhance the conservatism of the analysis.

The initial crack size (a_i) was based on current NDE inspection practices (magnetic-particle technique) for the XS-2 exhaustor. Inspection is normally done in place. According to AEDC inspection personnel, the smallest crack length that can be found depends on daylight conditions. On a cloudy day, they can find a 1/16-inch crack with a high degree of confidence. However, on a sunny day, they can only find a 1/8-inch crack with confidence. Sunny day conditions make

it difficult to detect a crack because there is not sufficient contrast between the sunlight and the ultraviolet light used to illuminate the magnetic particle penetrate. Initial crack lengths of both 1/16-inch and 1/8-inch were used in the fracture mechanics analysis.

Various final crack sizes (a_f) were arbitrarily chosen and checked for structural stability under the steady-state loading condition. Choosing a final crack size was not difficult. In nearly all cases, an infinite crack propagation rate was approached as the final size was reached. This meant that the crack propagation life remaining to reach either the rupture crack size or the critical crack size was very short. For this reason, a two-inch crack length or a crack depth equal to the thickness of the blade was specified as the final crack size. If the critical crack size was less than the chosen final crack size, the BIGIF model execution was terminated with the message "the critical crack size was reached". The inspection interval is then determined from the computed crack propagation life and an appropriate safety factor.

The random data constructed with the LOADHIST program was used to confirm that a short fatigue life would be obtained for an elliptical surface crack in the airfoil root. Using the BIGIF program and the random data approach, the predicted crack propagation lives varied from 99 to 18,640 hours of operation. Assuming a crack length to depth ratio of 2.5, the shorter life was obtained for a_i of 1/8-inch length and 0.05-inch depth (sunny day inspection condition). If a_i was 1/16-inch length and 0.025-inch depth (cloudy day or shaded inspection

conditions), the life was predicted to be 735 hours. If the depth of the 1/16 crack was 0.005 inch, the life is increased to 18,640 hours. Although this approach is considered very conservative, an important characteristic is shown. This analysis shows the benefit of performing XS-2 blade inspections under cloudy or shaded conditions. The smaller the detectable crack, the larger the crack propagation life. The difference in predicted life for an a_i of 1/8-inch and 1/16-inch was a factor of seven. Another characteristic in this case was the significant increase in predicted life due to the assumption of a much smaller crack depth (18,640 hours).

The random data approach was also used to predict the structural life in the attachment root region of the blade. The dynamic stresses in the attachment root, however, were very low while the steady-state stress was much higher. Using the random data in the life calculations for the attachment region, very long lives were predicted. It was concluded that the attachment region has an infinite life when compared to the airfoil root region. Therefore, the life of the blade is dependent on a surface crack in the airfoil root region. Other types of cracks were investigated; however, the elliptical surface crack predicted the shorter life.

The FFT amplitude data predicted by the DAP analysis were used to investigate the effect of the surge frequency. Only surge data were used in this analysis, which assume the normal operational data had an insignificant effect. It was assumed that the primary mode of vibration was the first bending mode at 177 Hz. The surge period was

assumed to be of 15-second duration. The initial crack sizes were based on the inspection limitation previously discussed. The crack was assumed to be an elliptical surface crack growing through the blade. The exhauster operating period was assumed to be 8 hours. The number of surges per operating period was assumed to be one per period and one per ten periods. Three different cases were investigated in this study.

The first case was based on the assumption that all surge cycles were at a constant amplitude with a frequency of 177 Hz. This would be a very conservative approach when compared to the frequencies determined from the DAP analysis (0.25 and 0.5 Hz). This assumption, based on zero damping of the surge pulses, gave the shortest life predictions of all cases studied, Table 3. The maximum life predicted was 2520 hours of operation for a_i of 1/16-inch and the one surge per period. If the surge was assumed to occur once in ten periods, the life was increased to 25,200 hours.

The next case assumed a more realistic load cycle. The surge vibrations were assumed to have a large amplitude spike occurring at a 5.0 Hz frequency (surge frequency). This spike was assumed to have nonlinear damping which reduced vibration amplitude to an insignificant level before the next spike occurred. This cycling continued throughout the surge period. For this case, the life predictions were much larger. For one surge per period, the a_i of 1/16-inch gave a predicted life of 61,400 hours. One surge per 10 test periods resulted in a predicted life of 581,000 hours.

The third case was the same as the previous case except the surge spike agreed more closely with the frequency predicted from the data. For this case, the surge frequency was 0.25 Hz at the maximum amplitude. This amplitude was damped out as in the previous case. Again, the predicted life was increased significantly over previous cases. The results of the three surge comparisons are shown in Table 3. The most critical parameter for calculation of crack propagation life was the assumed surge frequency, and the second most critical parameter was the number of surges per test period.

The final blade life predictions were based on the DAP analysis for reasons as mentioned previously. Since the airfoil root region was the critical location for a crack, the DAP load predictions were applied to this region only. Since the dominate mode of vibration is the first bending mode, it was used as the deflected shape for the life assessment analysis. The amplitude of the surge spike was 124 mils peak-to-peak and the normal operational amplitude was 36 mils peak-to-peak. The maximum surge amplitude was applied at a frequency of 0.5 Hz (no damping was assumed between spikes since damping could not be verified from the data) and the normal operational amplitude was applied at 177 Hz. The startup and shutdown cycle were also included in this analysis. With the potential of two inspection tolerances, both initial crack sizes were used. The critical type of crack used in this analysis was the elliptical surface crack. A typical operating period was assumed to be of eight-hour duration with the occurrence of one surge per period. The mean stress level was assumed to be the steady-

state stress level for all cases. The dynamic stresses determined by DAP and DSTRESS for the first bending mode were superimposed on the steady-state stresses to define the stress field for the life analysis.

The results of this analysis are shown in Figure 35 as a plot of initial crack depth versus cycles to failure. Two curves are shown for the initial crack sizes which represent the different inspection conditions. The plot shows a significant advantage to the establishment of an inspection procedure which results in the detection of the smaller cracks and their depths. An accurate prediction of crack depth is as important as the crack length. However, the crack depth is a difficult parameter to determine.

CHAPTER XI

XS-2 INSPECTION INTERVAL

An inspection interval for the XS-2 blades can be determined from the predicted fatigue lives, appropriate safety factor, and a life-cycle-cost (LCC) analysis for the exhauster. A plot of crack size versus predicted life as shown in Figure 35 can be used to determine the inspection interval. The factors that affect the predicted life (such as initial crack size, final crack size, and the number of surges per operating period) must be evaluated before the inspection interval can be established. According to Failure Analysis Associates, a safety factor of 2.0 is considered nonconservative and a factor of 6.0 is considered too conservative.

An LCC analysis should be conducted on the exhauster to determine the appropriate safety factor. A rigorous LCC analysis was not conducted with this study. This analysis could be performed by engineers responsible for the maintenance of the XS-2 exhauster. Factors that should be considered in the analysis were discussed previously.

Another approach for determining the inspection interval is to monitor the operation of the exhauster. Additional analysis was performed using the BIGIF model to illustrate this concept. An elliptical surface crack with an initial length of 1/8-inch and depth of 0.05-inch was assumed to exist at the blade airfoil root. The deflection amplitudes and surge frequency determined from the DAP analysis were used. It was further assumed that the exhauster operation was divided into five segments differing by the number of surges per operating

period. The assumed sequence was: 1) one surge per period, 2) five surges per period, 3) one surge per ten periods, 4) one surge per five periods, and 5) one surge per period. The results of this analysis are shown in Figure 36 as a plot of crack depth versus expended life. It is evident from this plot that 1) the increased surge activity causes a significant increase in the crack growth and 2) the crack size is heavily dependent on the number of surges. If the number and duration of surges are recorded, then the exhaustor inspection interval could be based on this information. Once an assumed crack reaches a predetermined size the exhaustor would be inspected. This approach would result in a floating inspection interval. The inspection interval estimate can be changed according to actual exhaustor operation rather than projected operation.

Calculation of an inspection interval for the exhaustor blades was performed based on the predicted blade structural life of 70,000 hours shown in Figure 36. An estimated cost of reblading the XS-2 exhaustor is \$1,250,000, and an estimated cost of inspecting the blades is \$5000. Based on these costs, a very conservative safety factor of six was applied to the predicted blade structural life to determine an inspection interval of approximately 12,000 hours. This would be a six-fold increase over the current inspection interval. However, the blade inspection interval could be significantly greater or less, depending on the blade operational scenario, crack characteristics (crack type and initial size), and safety factor.

CHAPTER XII

CONCLUSIONS AND RECOMMENDATIONS

A life assessment analysis procedure was developed for AEDC turbo-machines. This procedure was based on the retirement-for-cause approach for life assessment analysis which has been used by the Air Force for aircraft turbine engine disks. The fracture mechanics analysis code BIGIF was acquired for use with the analysis procedure. The life assessment analysis procedure was demonstrated using the first-stage blade of an exhaust compressor (XS-2) in the ETF plant. Operational test data were acquired on the XS-2 blade for determination of blade dynamic stresses. A prototype noninterference blade-tip deflection measurement system developed at AEDC was used for the exhaust test program.

As part of developing the life analysis procedure, special purpose computer codes were also developed for analyzing the blade-tip dynamic deflection data and determining blade dynamic stresses. The Dynamic Analysis Program (DAP) converted the blade-tip deflection data into a vibration spectrum and determined excited natural modes. The Load History (LOADHIST) code converted the random blade-tip deflection data into a blocked deflection data spectrum suitable for a fracture mechanics analysis. The Dynamic Stress (DSTRESS) code transformed the measured blade-tip deflection data to dynamic stresses.

The life assessment analysis procedure was used to predict a structural life for the XS-2 blade and to perform parametric studies of different variables on the predicted blade life. Also, an approach was

presented for determining the blade inspection interval based on the predicted structural life. This approach was used to determine an inspection interval for the XS-2 blade which was a six-fold increase over the current inspection interval.

It was concluded from the XS-2 blade life assessment analysis that the most significant load condition was a surge event. Also concluded was that the blade structural life prediction and associated blade inspection interval could be significantly greater or less, depending on the operational scenario, crack characteristics, and inspection safety factor.

Several important items were identified during the life assessment analysis of the XS-2 blade. The more significant of these include 1) load history, 2) dynamic stress distribution, 3) initial crack sizes, 4) material fatigue properties, and 5) safety factor for inspections.

The data measurement system used to acquire load history data had very limited capabilities. Only nonintegral blade-tip deflection data were acquired. Integral data could not be acquired because of software and hardware problems. The frequency window (12 Hz) of the measurement system made it difficult to accurately determine the blade mode of vibration associated with the data. Also, the surge frequency of the exhaustor could not be determined precisely from the data. However, the blade dynamic stress distributions were estimated from the nonintegral deflection data. It is not known if the stresses caused by integral vibrations were larger than nonintegral vibrations. Also,

the estimated dynamic stresses were not verified because strain gage data were not taken. Strain gage data are recommended in order to improve the accuracy associated with the NSMS deflection data.

Two significant factors in the determination of crack propagation lives were initial crack size and the material fatigue properties. The initial crack sizes varied in length from 1/16- to 1/8-inch, and the difference in computed lives were typically two-fold. Also, typical values for fatigue parameters da/dn , C , and n were used for the analysis. If worst case values were used the difference in computed lives would be much greater than the difference due to initial crack sizes.

The purpose of inspection safety factors is to allow for the uncertainties in the life prediction of the blade. The risk associated with a particular inspection interval can be related to this safety factor. Both risk and maintenance costs should be considered in the determination of the appropriate safety factor.

Based on the results of the XS-2 blade life analysis, the following recommendations are made to improve the XS-2 maintenance program at AEDC:

1. A floating inspection interval should be established for turbomachinery components based on a) the number and duration of surge events, b) an appropriate inspection safety factor, c) the smallest detectable crack, and d) improved operational test data for determination of blade dynamic stresses.

2. The number and duration of exhauster surges should be documented. This information could be used to lengthen or shorten the inspection interval.
3. A more rigorous life-cycle-cost analysis of the XS-2 exhauster should be performed to determine the appropriate safety factor for the inspection interval. If this analysis is not performed, a safety factor of five is recommended for the inspection interval.
4. The blade inspections should be conducted in more favorable conditions such as using a canopy to shield sunlight from the blades and with the best equipment available to improve the detection of smaller cracks. A longer crack propagation life and inspection interval depends on the smallest crack which can be detected and removed from service.
5. The instrumentation and data acquisition system used to acquire blade-tip deflection data should be improved and expanded. Both integral and nonintegral dynamic data should be acquired. Also, the frequency capability for the measurement system should be expanded from 12 Hz to a minimum of 2000 Hz. If a noninterference blade-tip deflection measurement system is used for future programs, the method for converting deflection data to dynamic stress should be validated using strain gage data. For future programs, it is recommended that the data measurement system have these expanded capabilities.

This life assessment analysis procedure can be applied or adapted to other structural components. The procedure can be applied to any type of machine or structure that experiences cyclic loads. Operational load data can be acquired on both new and old machines or structures. This data along with structural component drawings, material characterization data, and other design data could then be used for a life assessment analysis. Maintenance and inspection programs for these components can then be based on this analysis along with service experience.

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APPENDIX

Table 1. XS-2 Blade Natural Frequencies, Mode Shapes, and
Modal Participation Factors

Mode	Frequencies	Modal Participation Factors			Description
		X-Direction	Y-Direction	Z-Direction	
1	177.0 Hz	0.10970	-0.00017	-0.05214	1st Bending (about weak axis of blade)
2	644.2 Hz	0.01710	0.00429	0.02551	1st Torsion
3	888.3 Hz	0.06283	-0.00215	0.09355	Bending & Torsion
4	1354.0 Hz	0.04448	0.00034	-0.06004	2nd Bending
5	1993.0 Hz	-0.00241	-0.003127	-0.00476	2nd Torsion

Table 2. XS-2 Blade Random Deflection Data for
Life Assessment Analysis

Block	$\Delta_{\min}$ (Mils)	$\Delta_{\max}$ (Mils)	No. Cycles/Hour
1	-95.15	62.28	20
2	-101.09	41.51	20
3	-53.05	56.63	41
4	-14.93	93.52	82
5	-80.12	23.74	41
6	-70.39	29.66	123
7	-17.14	73.36	184
8	-72.07	17.80	164
9	-56.53	32.60	41
10	-48.50	35.84	205
11	1.31	83.59	266
12	-60.19	17.41	348
13	-82.13	-5.63	327
14	-66.67	8.03	286
15	-7.25	62.85	614
16	-59.28	9.33	675
17	-62.66	5.66	368
18	-48.50	13.49	695
19	8.79	70.69	880
20	-51.48	7.94	1,002
21	-57.12	2.17	941
22	-65.97	-9.41	941
23	-49.04	5.17	655
24	2.90	56.79	634
25	-45.80	4.48	491
26	-31.11	10.21	492
27	18.47	58.42	511
28	-44.85	-5.13	450
29	-39.66	-0.93	552
30	-39.77	-2.77	266
31	11.93	46.78	307
32	-29.79	-15.83	1,002
33	-34.18	-20.89	2,187
34	28.54	41.27	1,575
35	32.83	45.50	1,575
36	-24.69	-12.15	2,066
37	-33.16	-20.88	1,555
38	-40.70	-28.88	1,943
39	-28.14	-17.14	1,657

Table 2. Concluded

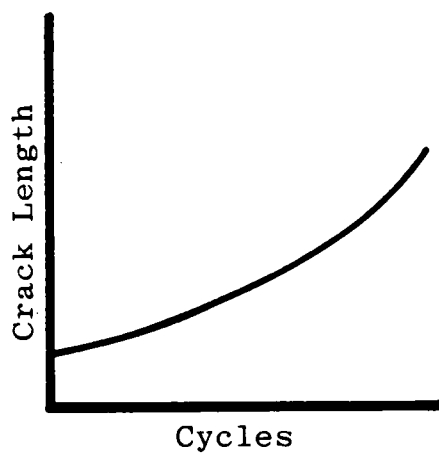
Block	$\Delta_{\min}$ (Mils)	$\Delta_{\max}$ (Mils)	No. Cycles/Hour
40*	-197.45	339.69	0.250
41*	-1.32	380.25	0.250
42*	-59.50	178.18	0.125
43*	-89.21	124.70	0.125
44*	-29.79	94.99	0.125
45*	-77.31	11.82	0.250
46*	-33.09	29.34	8.950
47*	42.93	79.19	19.625
48*	-12.11	2.89	12
49	(Startup, shutdown, steady-state load)		0.125
50**	-20.00	20.00	558,000

*Surge data

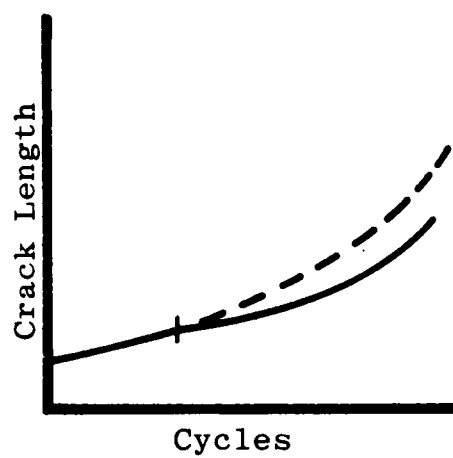
**1st bending mode

Table 3. Comparison of XS-2 Blade Life Predictions

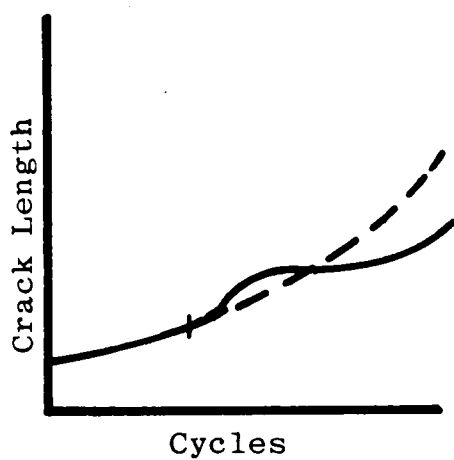
Crack Length x Depth	Number of Surges per Test Period	Surge Frequency	Life (Hours)
1/8 x 0.100	1.0	177	1,073
1/8 x 0.050	1.0	177	1,312
1/16 x 0.025	1.0	177	2,520
1/16 x 0.008	1.0	177	94,240
1/8 x 0.100	0.1	177	10,728
1/8 x 0.050	0.1	177	13,128
1/16 x 0.025	0.1	177	25,192
1/16 x 0.008	0.1	177	942,400
1/8 x 0.100	1.0	5	15,976
1/8 x 0.050	1.0	5	22,016
1/16 x 0.025	1.0	5	61,456
1/16 x 0.008	1.0	5	3,119,000
1/8 x 0.100	0.1	5	153,440
1/8 x 0.050	0.1	5	210,240
1/16 x 0.025	0.1	5	580,880
1/16 x 0.008	0.1	5	29,248,000
1/8 x 0.100	1.0	0.25	299,520
1/8 x 0.050	1.0	0.25	411,440
1/16 x 0.025	1.0	0.25	1,152,800
1/16 x 0.008	1.0	0.25	58,488,000
1/8 x 0.100	0.1	0.25	1,830,000
1/8 x 0.050	0.1	0.25	2,356,800
1/16 x 0.025	0.1	0.25	5,362,400
1/16 x 0.008	0.1	0.25	234,720,000



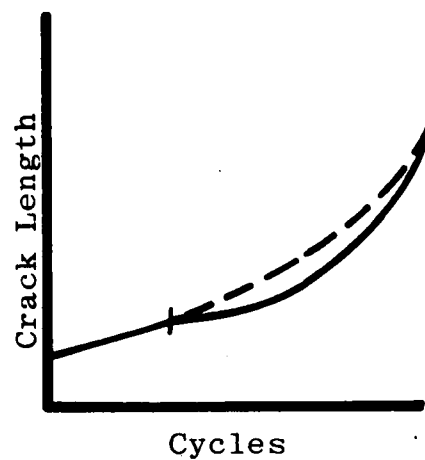
a. Constant Amplitude



b. Retardation



c. Delayed Retardation



d. Lost Retardation

Figure 1. Types of Crack Growth Retardation Behavior

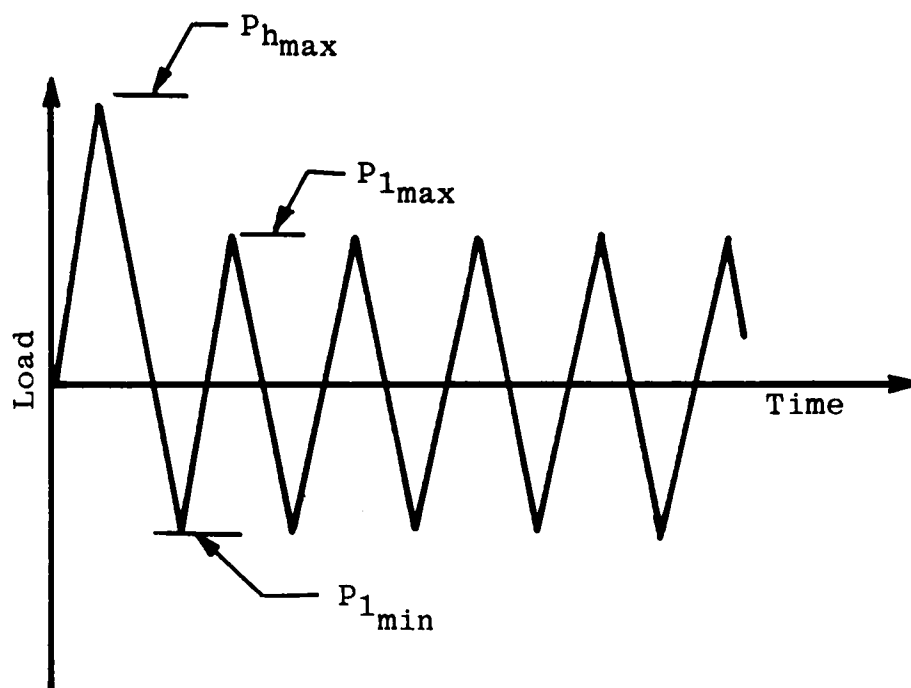


Figure 2. Tension Overload Cycle Causing Crack Retardation

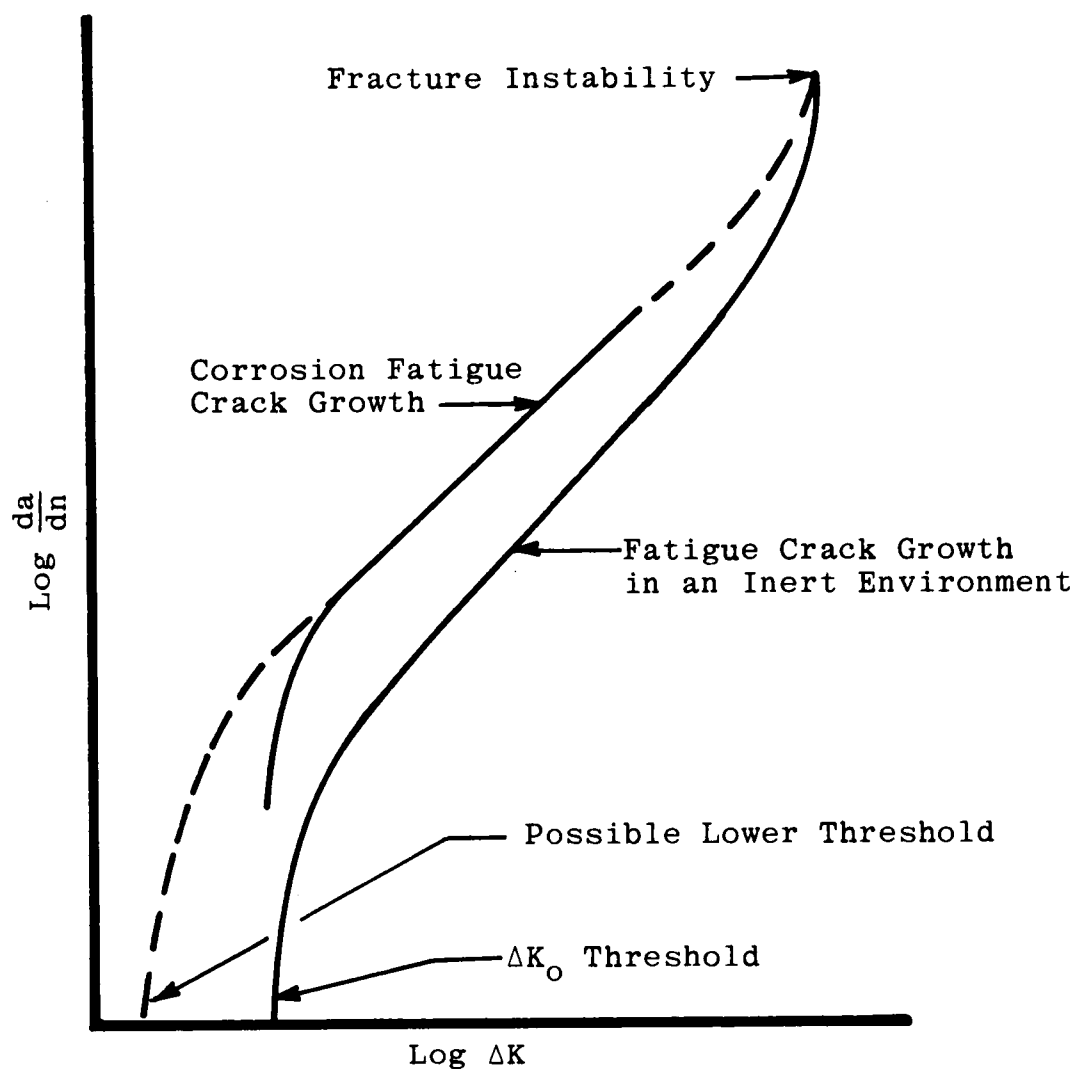


Figure 3. Corrosion Fatigue Crack Propagation Rate as a Function of the Cyclic Crack Stress Intensity Factor

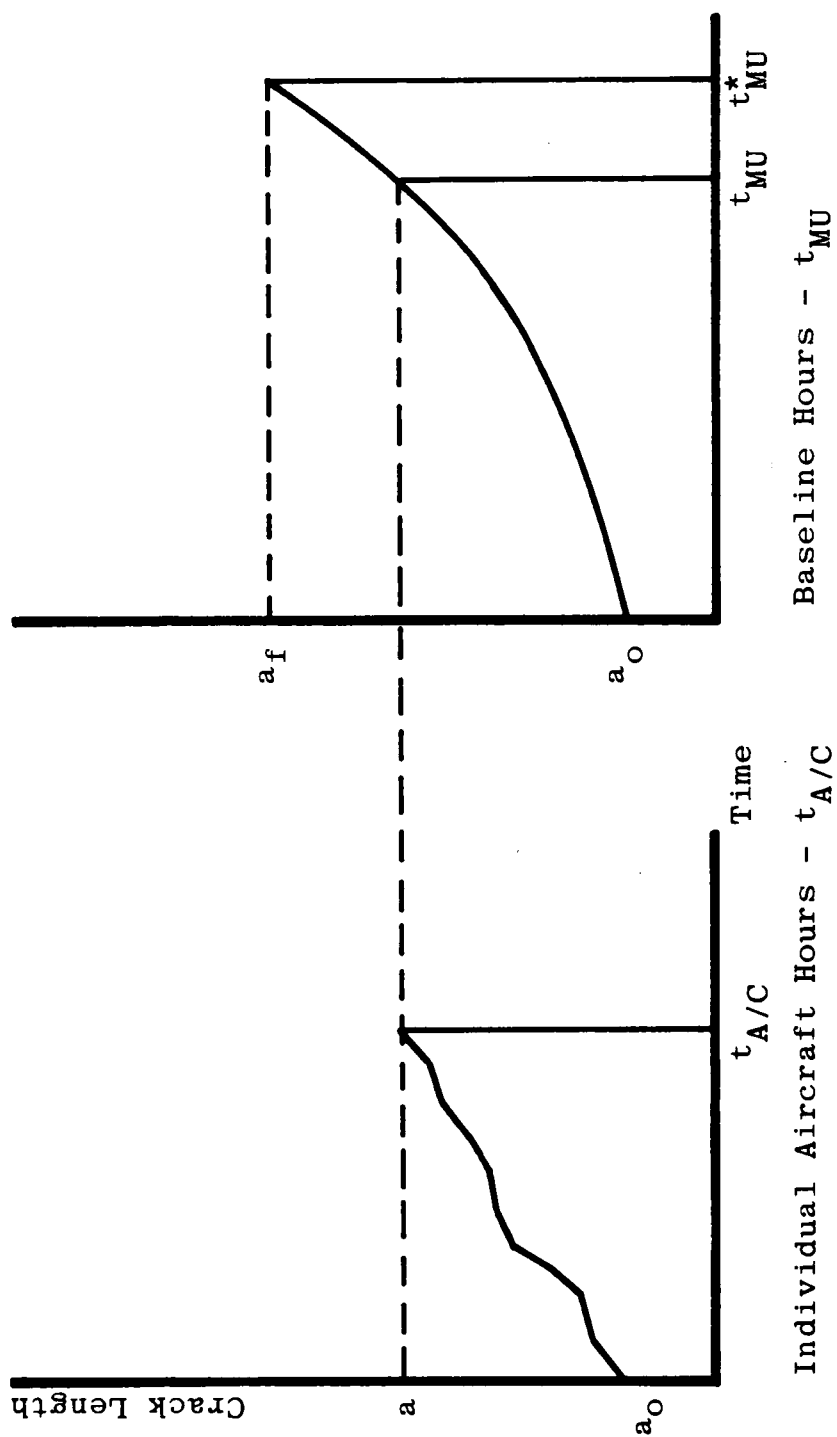


Figure 4. Relating Individual Aircraft Usage from Crack Growth Curves for Specified Critical Point

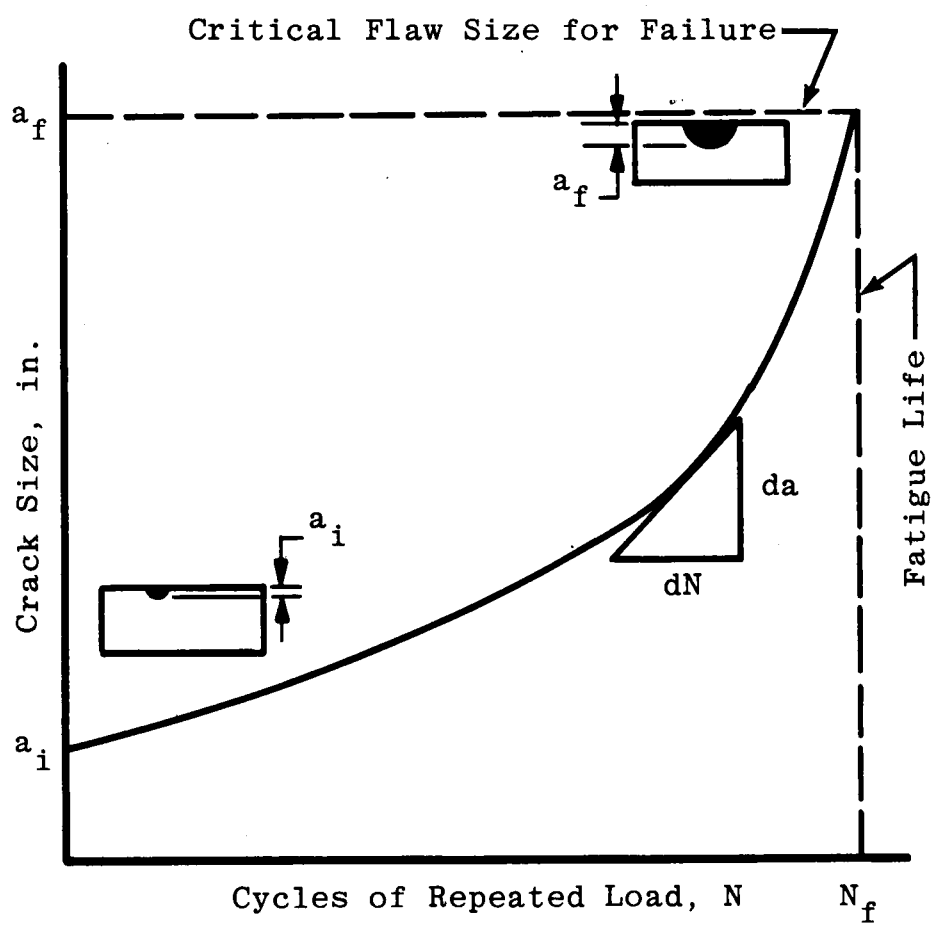


Figure 5. Schematic Illustrating Crack Propagation Concept of Fatigue

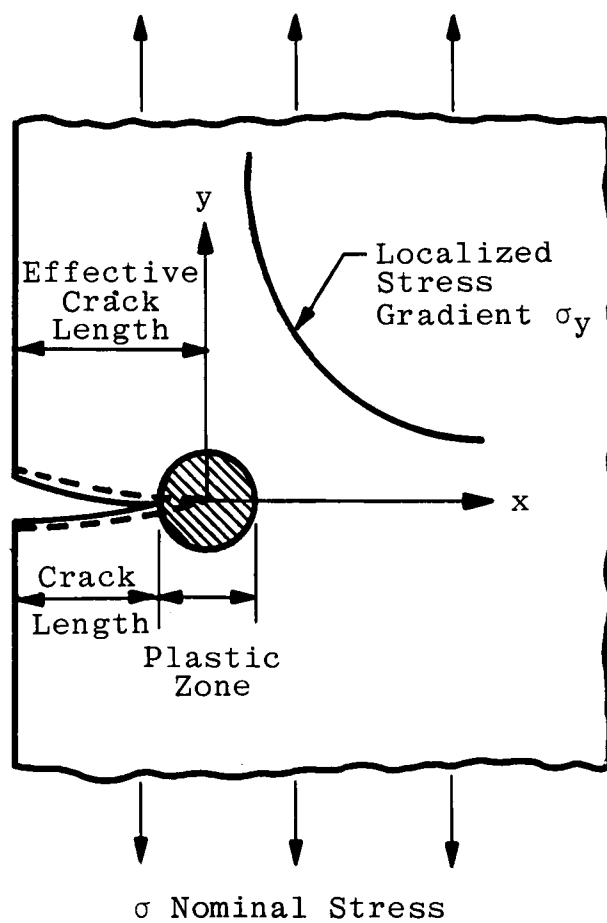


Figure 6. Linear Elastic Fracture Mechanics Crack Tip Model

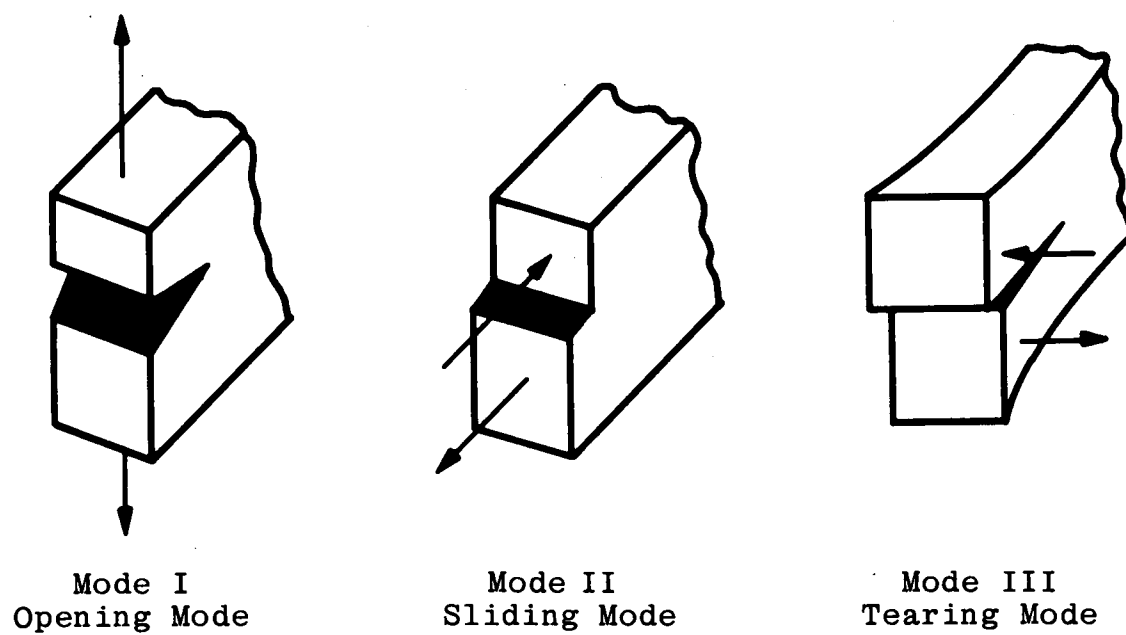
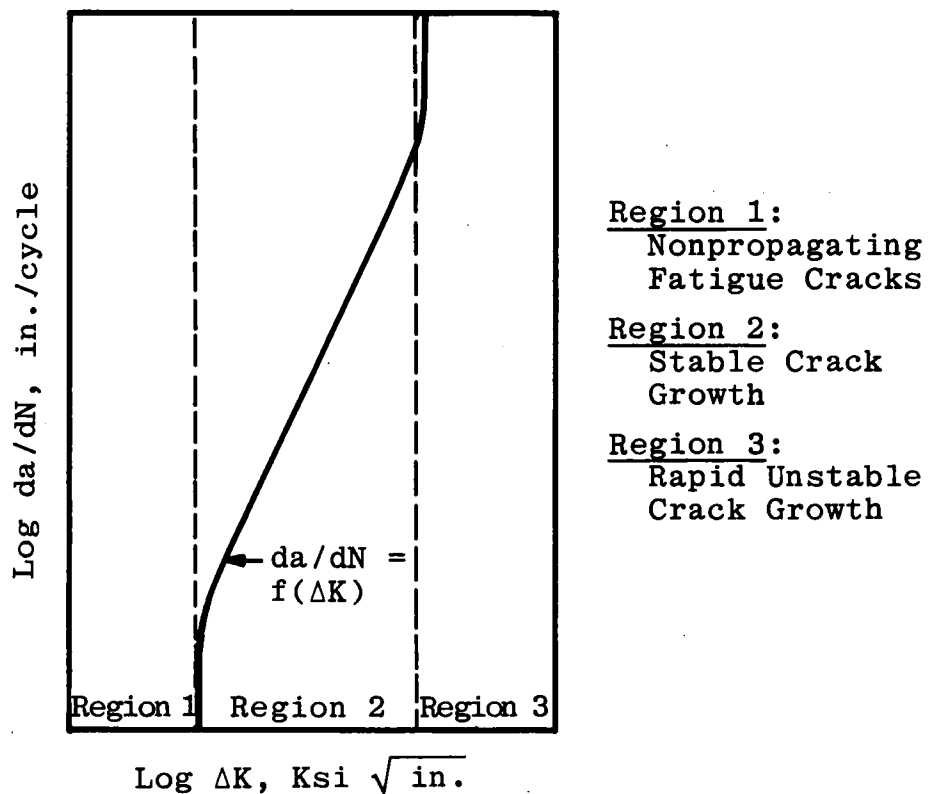
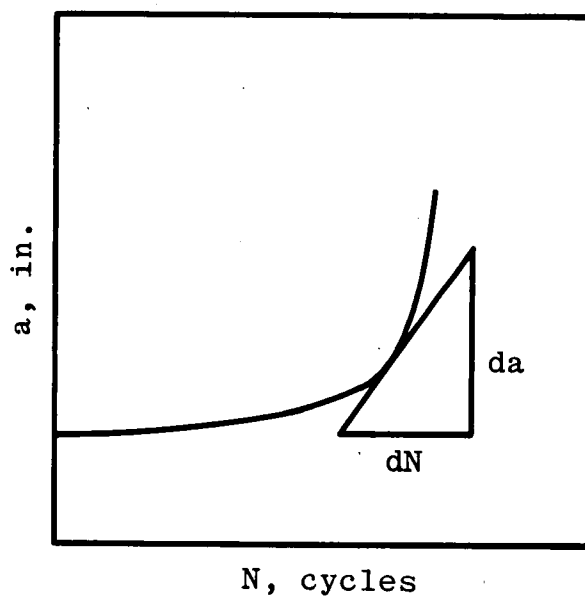


Figure 7. Three Modes of Crack Loading



a. Crack Growth Representation



b. Crack Length versus Applied Cycles

Figure 8. Schematic of Fatigue Crack Growth Data

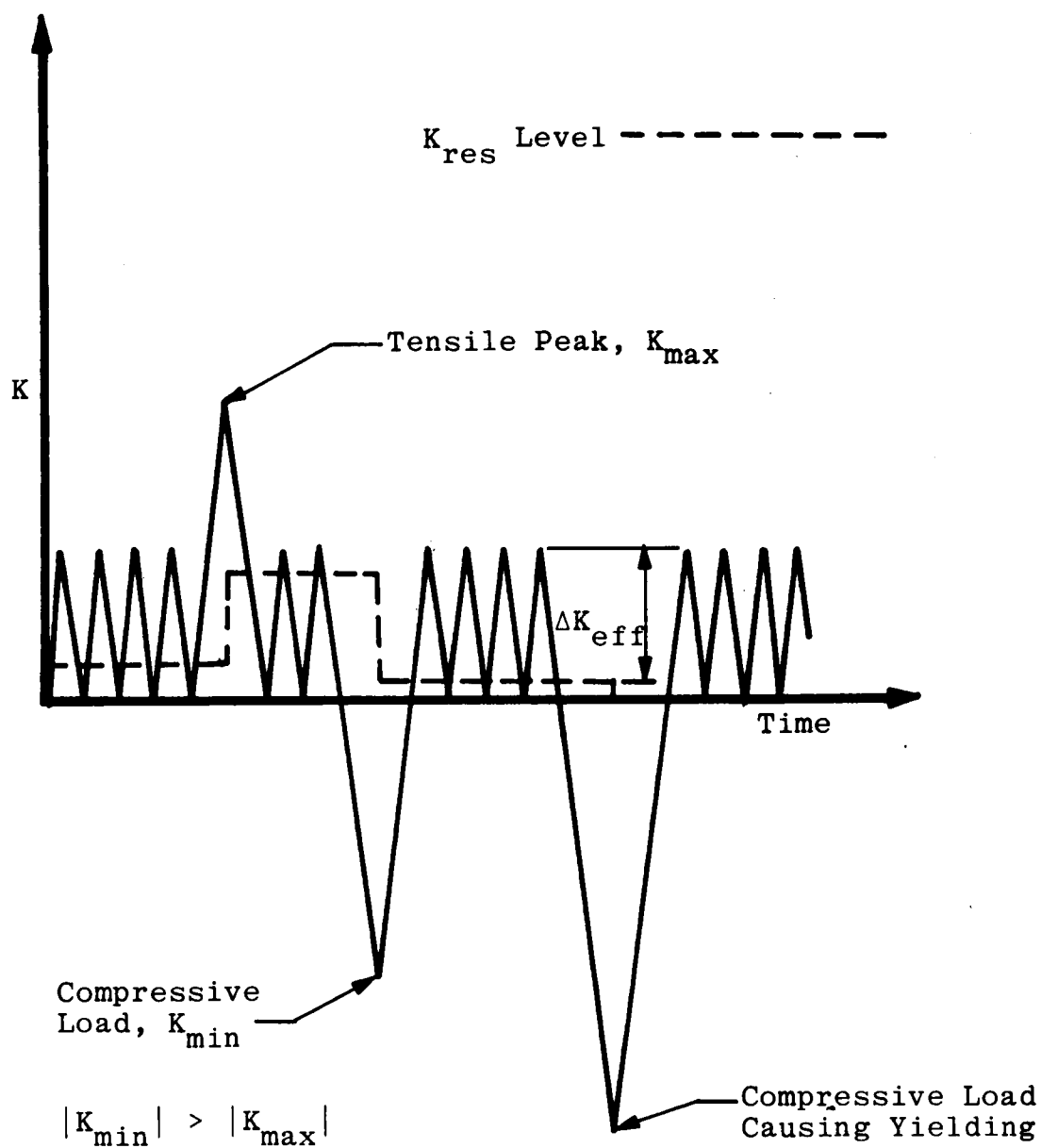
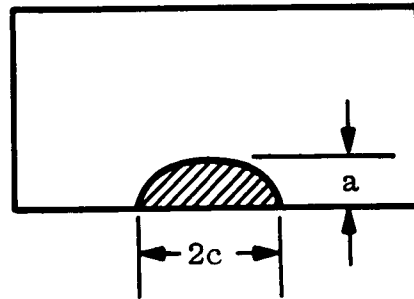
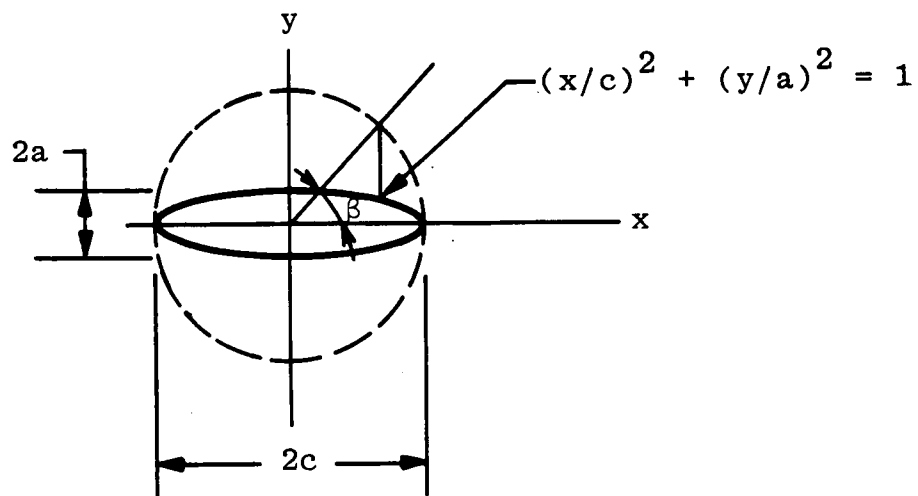


Figure 9. An Effective Stress Intensity Range Concept to Account for Sequence Effects in Variable Amplitude Loading



a. Surface Half-Elliptical Crack



b. Elliptical Crack Parameters-

Figure 10. Surface Semielliptical Crack in a Finite-Thickness Solid

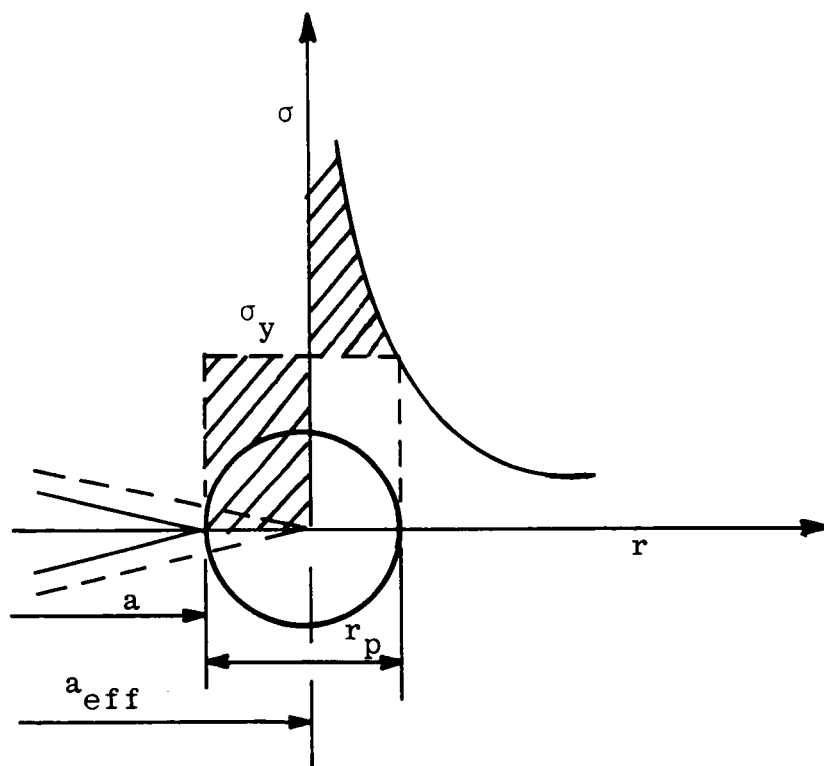
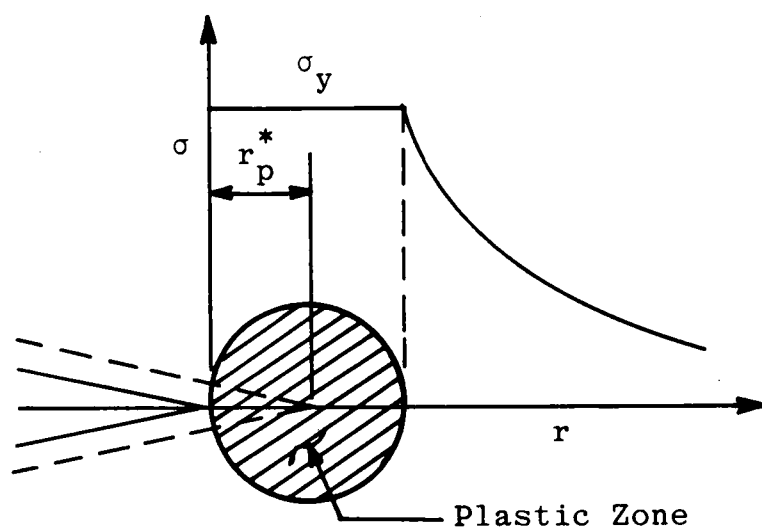


Figure 11. Irwin's Plastic Zone Correction Concept

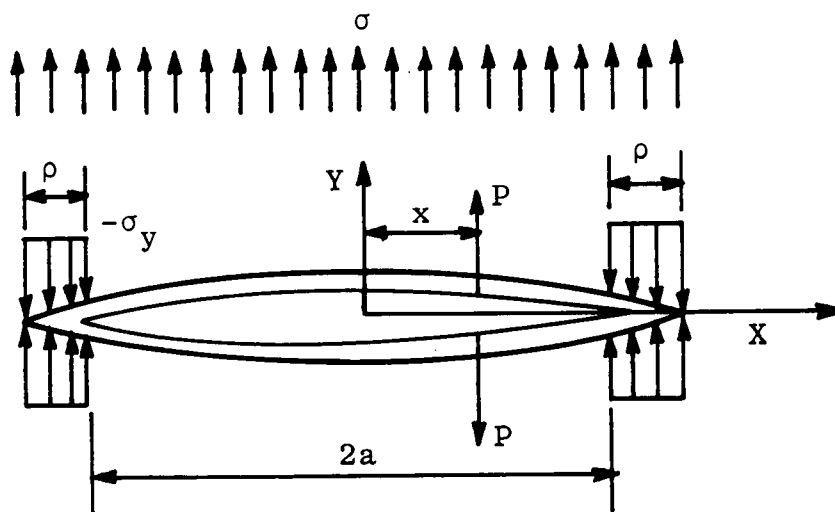


Figure 12. Dugdale Crack Model

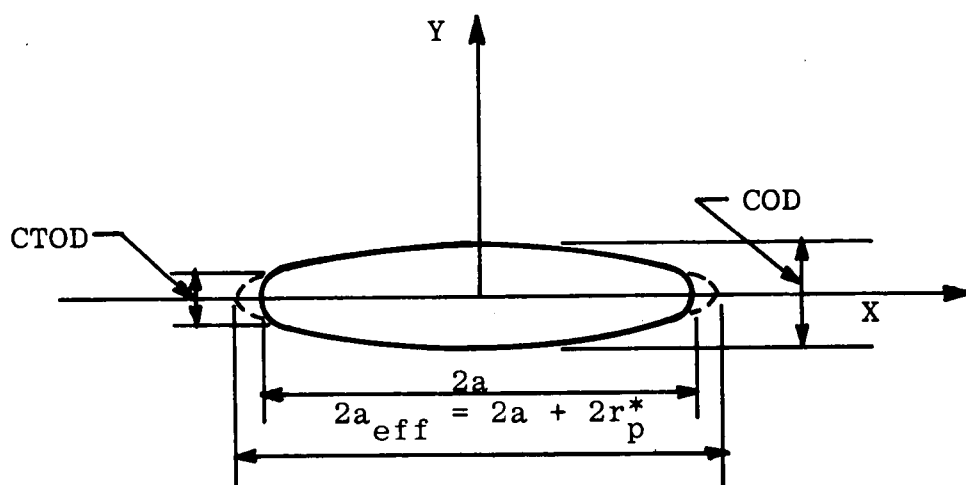


Figure 13. Crack Opening Displacement Model

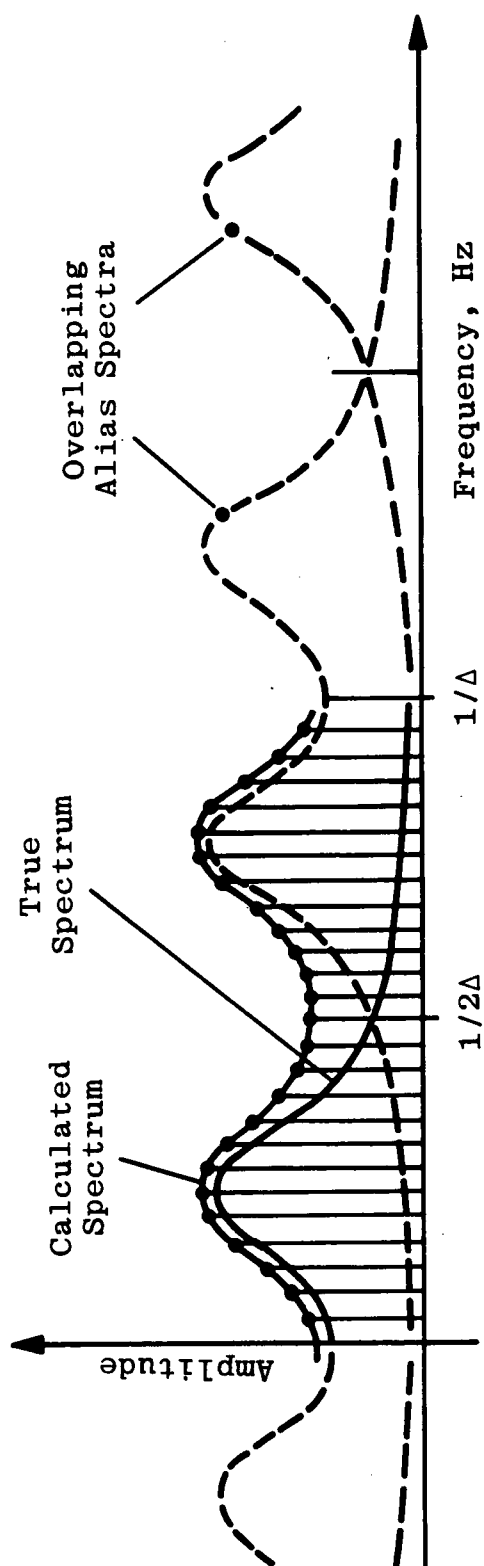


Figure 14. Aliasing Distortion for Frequencies More Than $1/2\Delta$ Hz

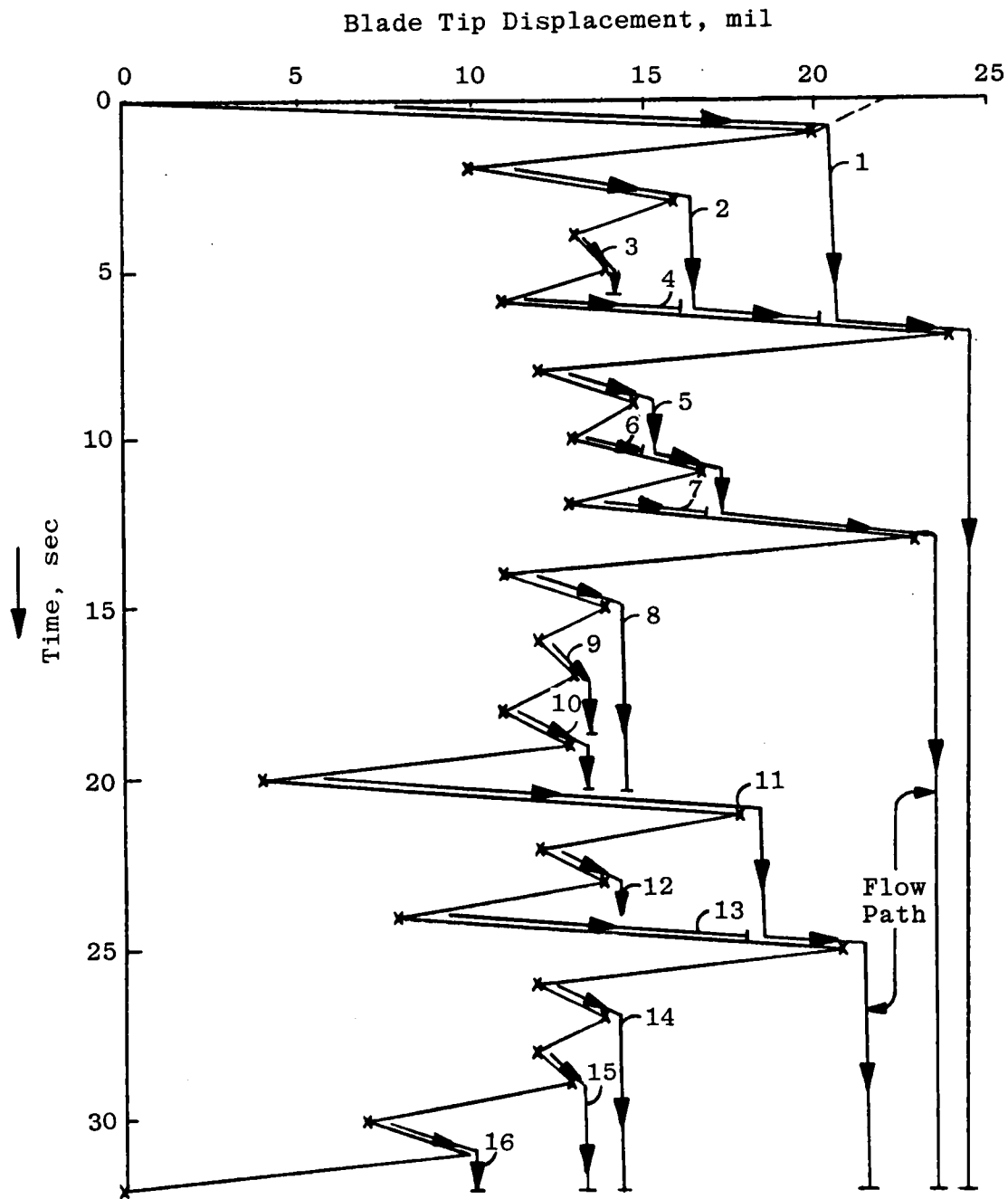
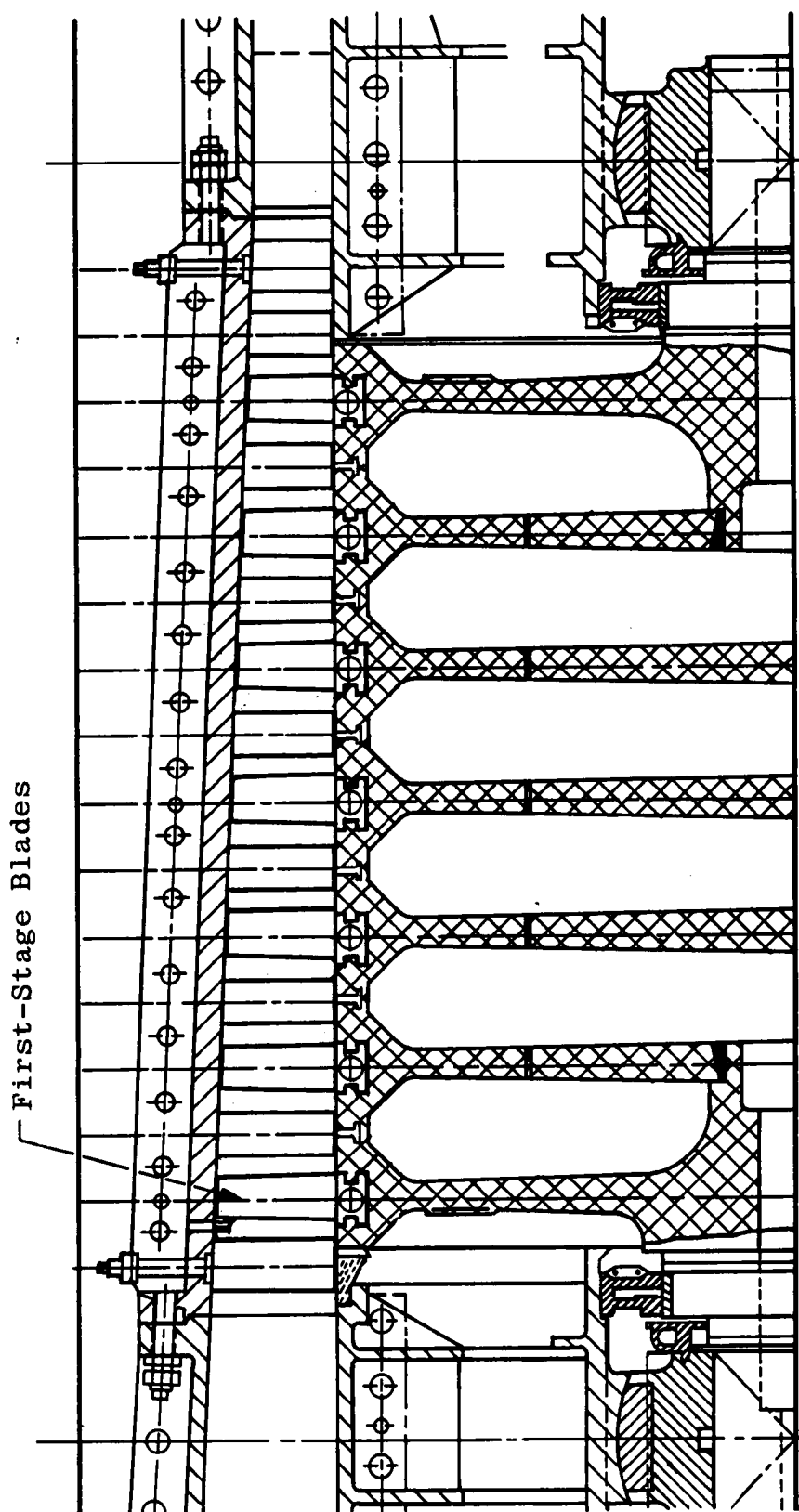
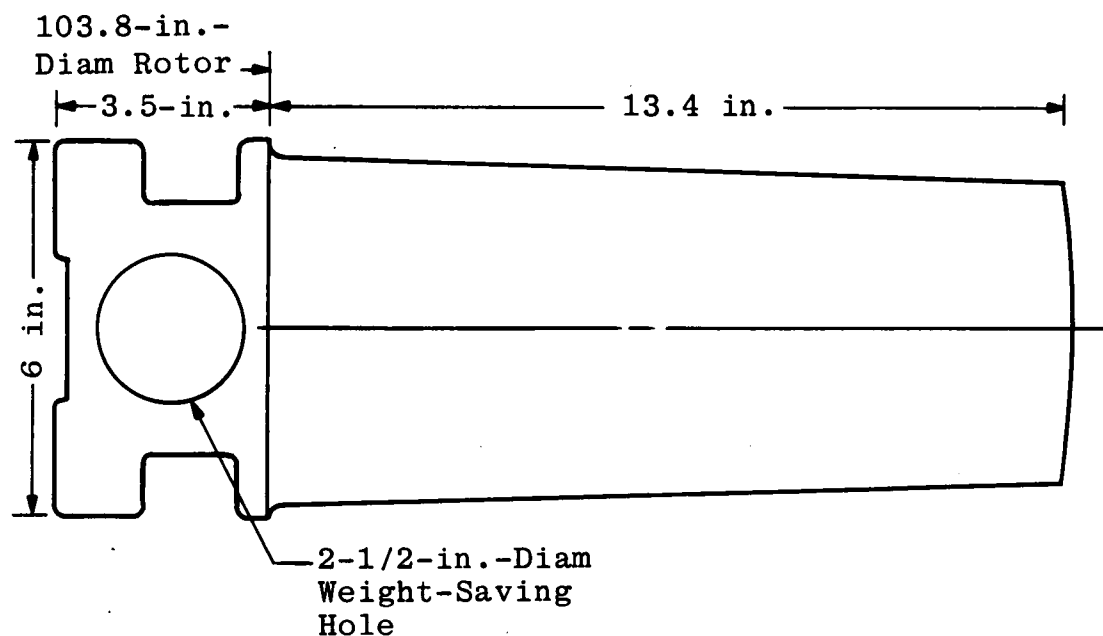


Figure 15. Rainflow Method of Counting Cycles

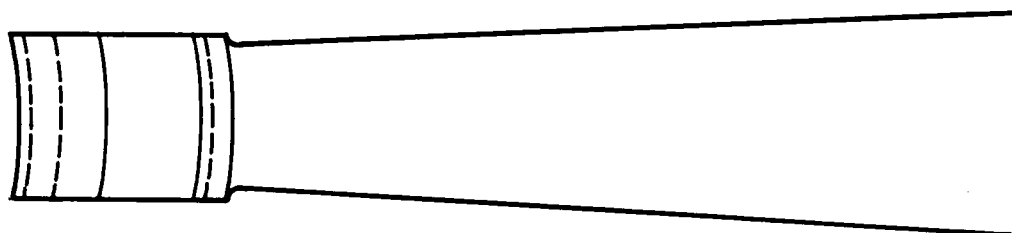


Note:  denotes the seven-stage drum rotor.

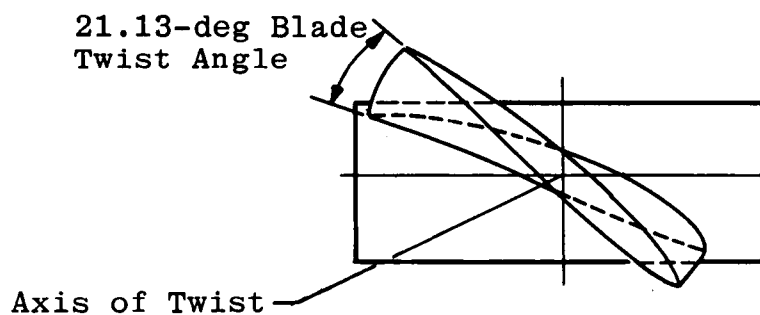
Figure 16. Cross-Sectional Schematic of the XS-2 Exhauster



a. Circumferential View



b. Axial View



c. Top View

Figure 17. Sketch of XS-2 First-Stage Rotor Blade

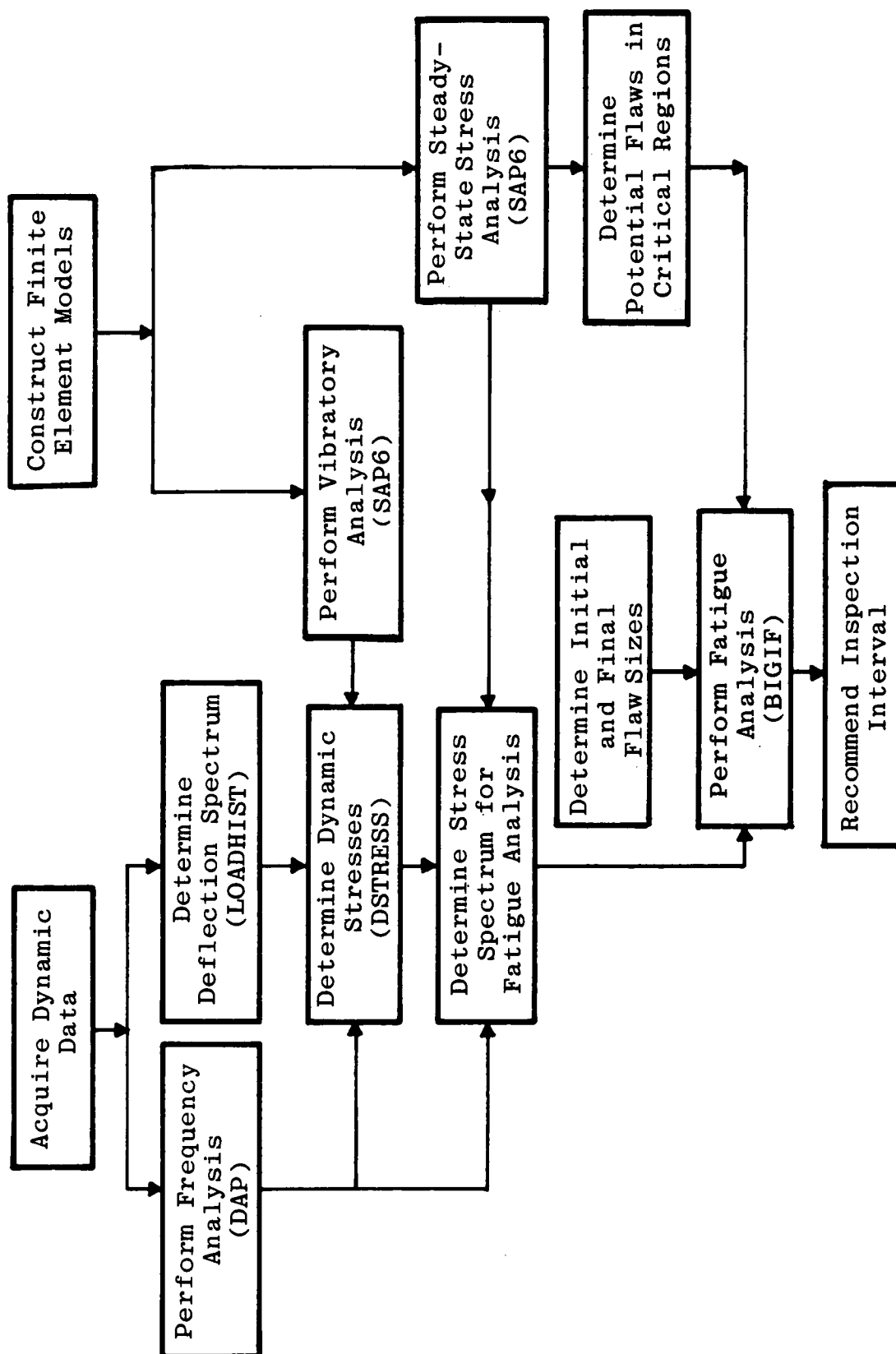


Figure 18. Life Assessment Analysis Procedure for XS-2 Blade

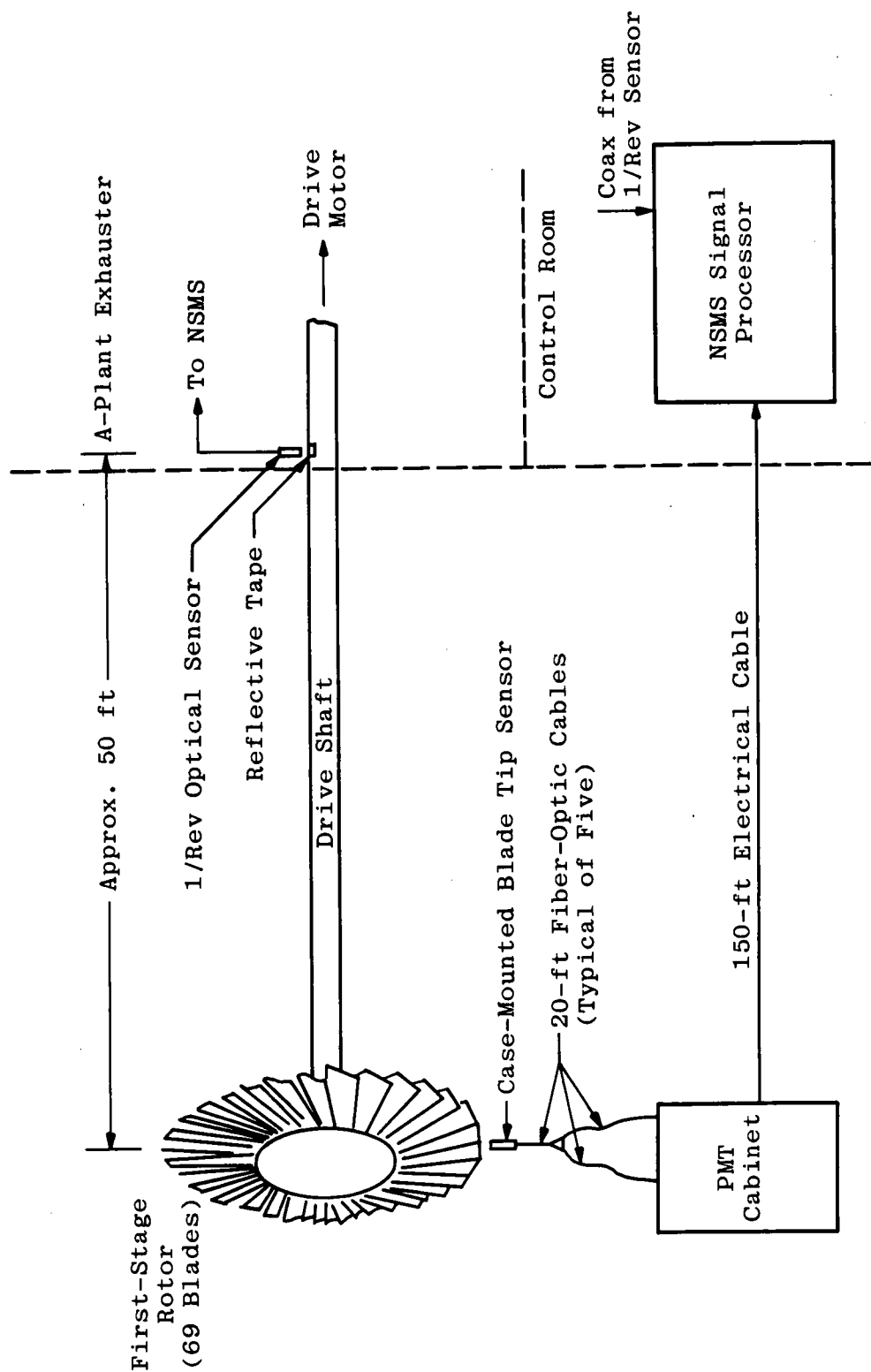


Figure 19. Schematic of XS-2 Test Program Instrumentation System

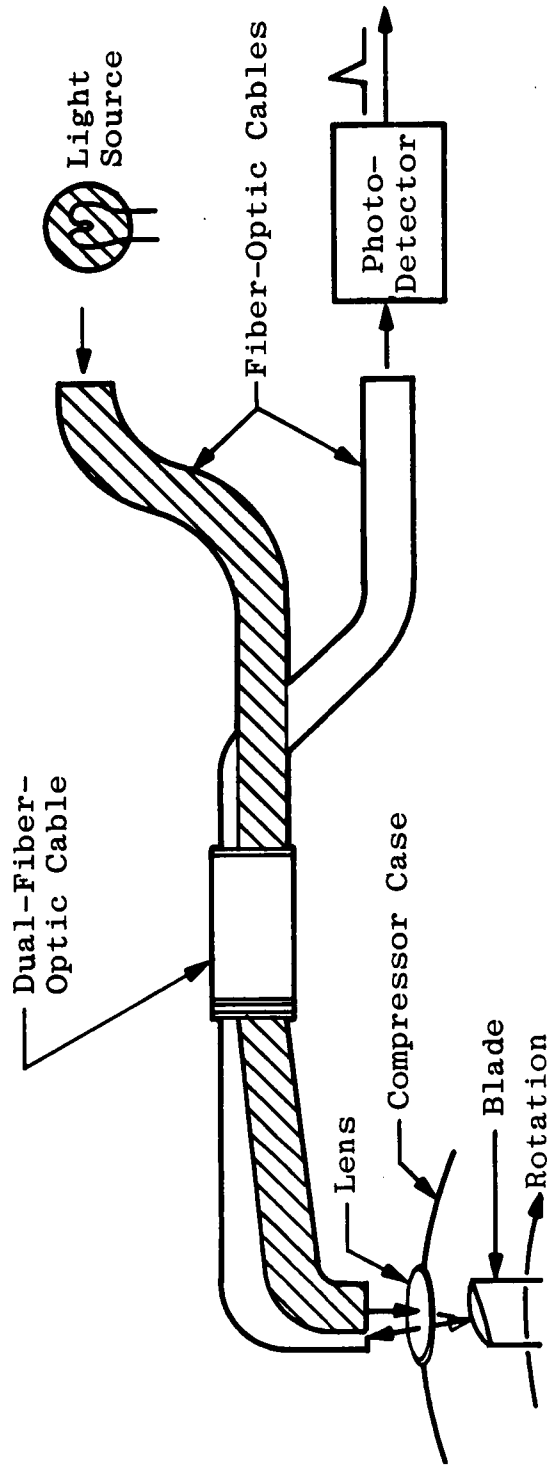


Figure 20. Schematic of Blade-Tip Sensing System

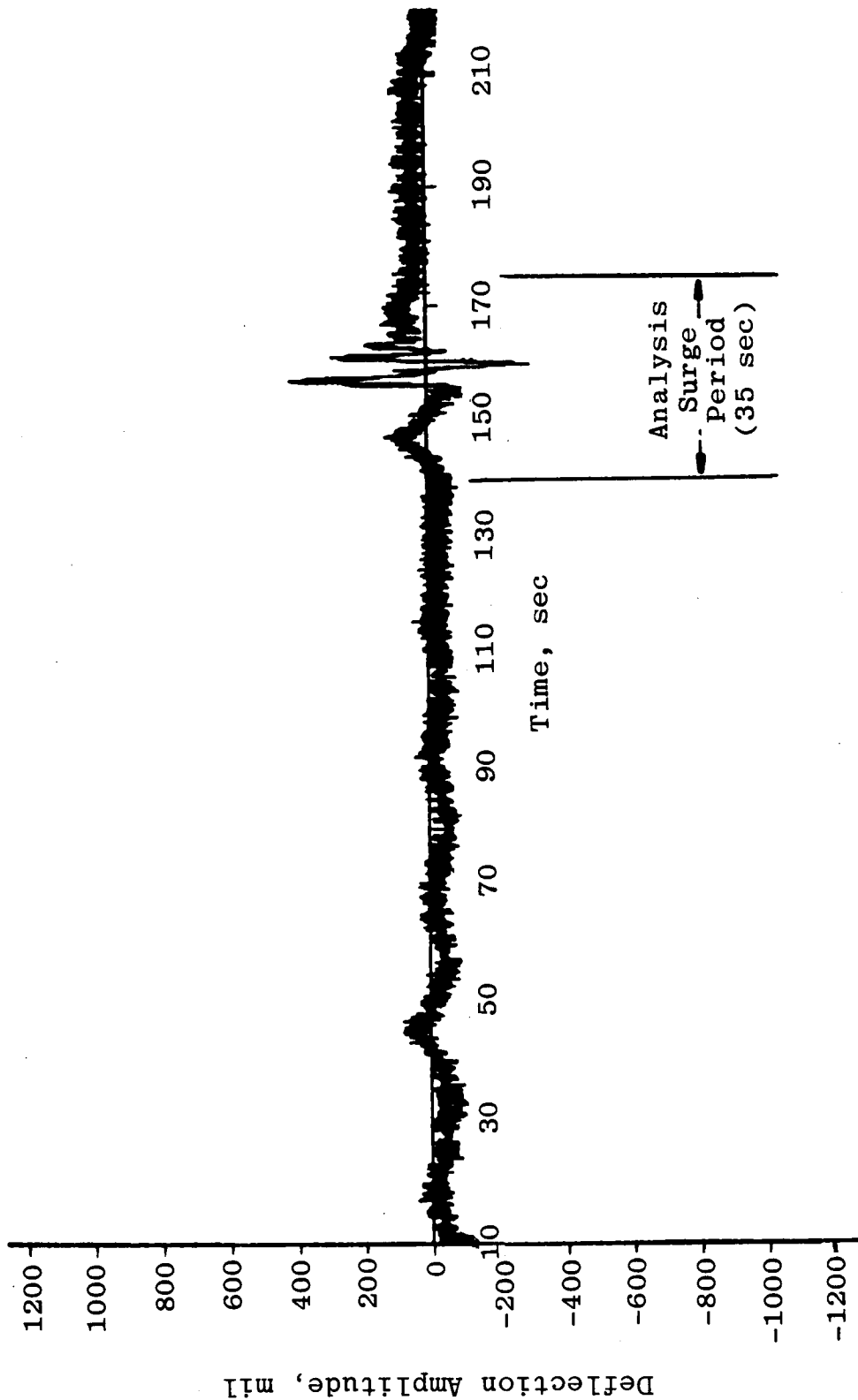
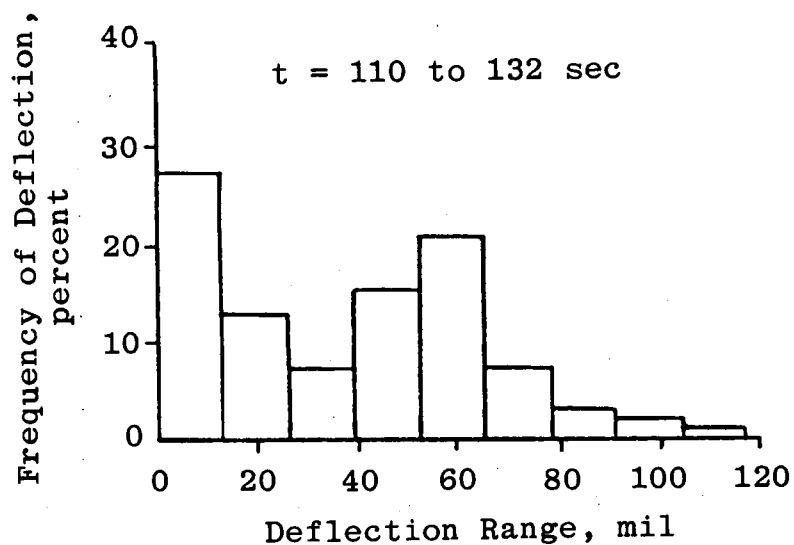
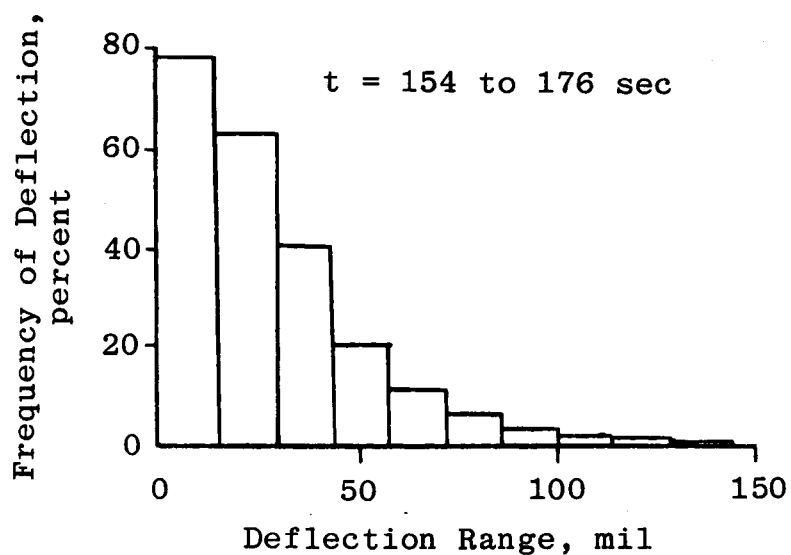


Figure 21. XS-2 Blade Typical Deflection Data Spectrum



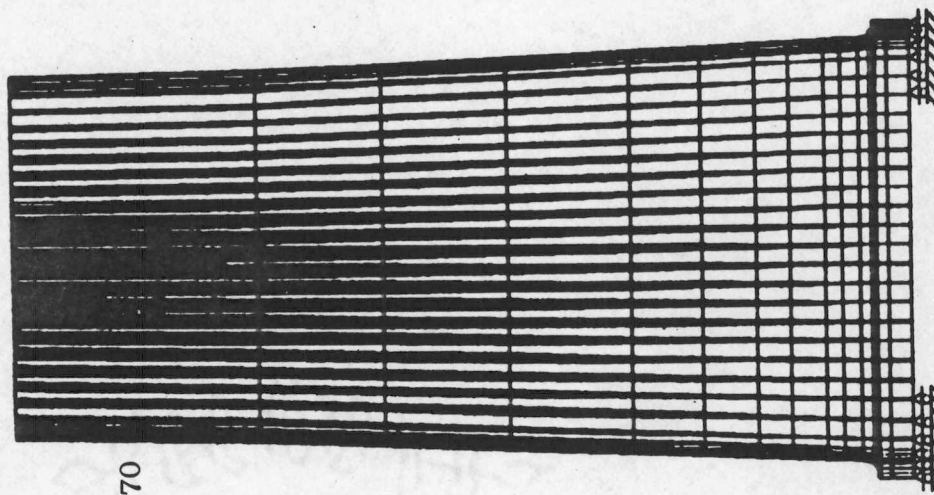
a. Typical Exhauster Operation



b. Exhauster Surge Operation

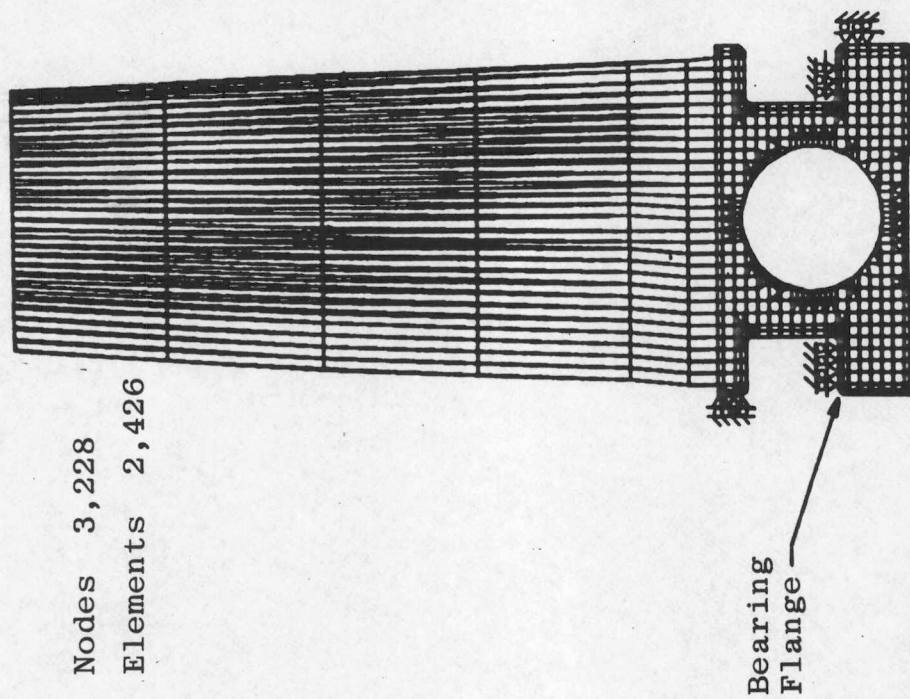
Figure 22. Histograms Constructed from XS-2 Blade-Tip Deflection Data

Nodes 3,632
Elements 2,770



a. Refined Airfoil
Model of XS-2 Blade

Nodes 3,228
Elements 2,426



Bearing
Flange

b. Blade Attachment
Model of XS-2 Blade

Figure 23. XS-2 Blade Finite Element Models

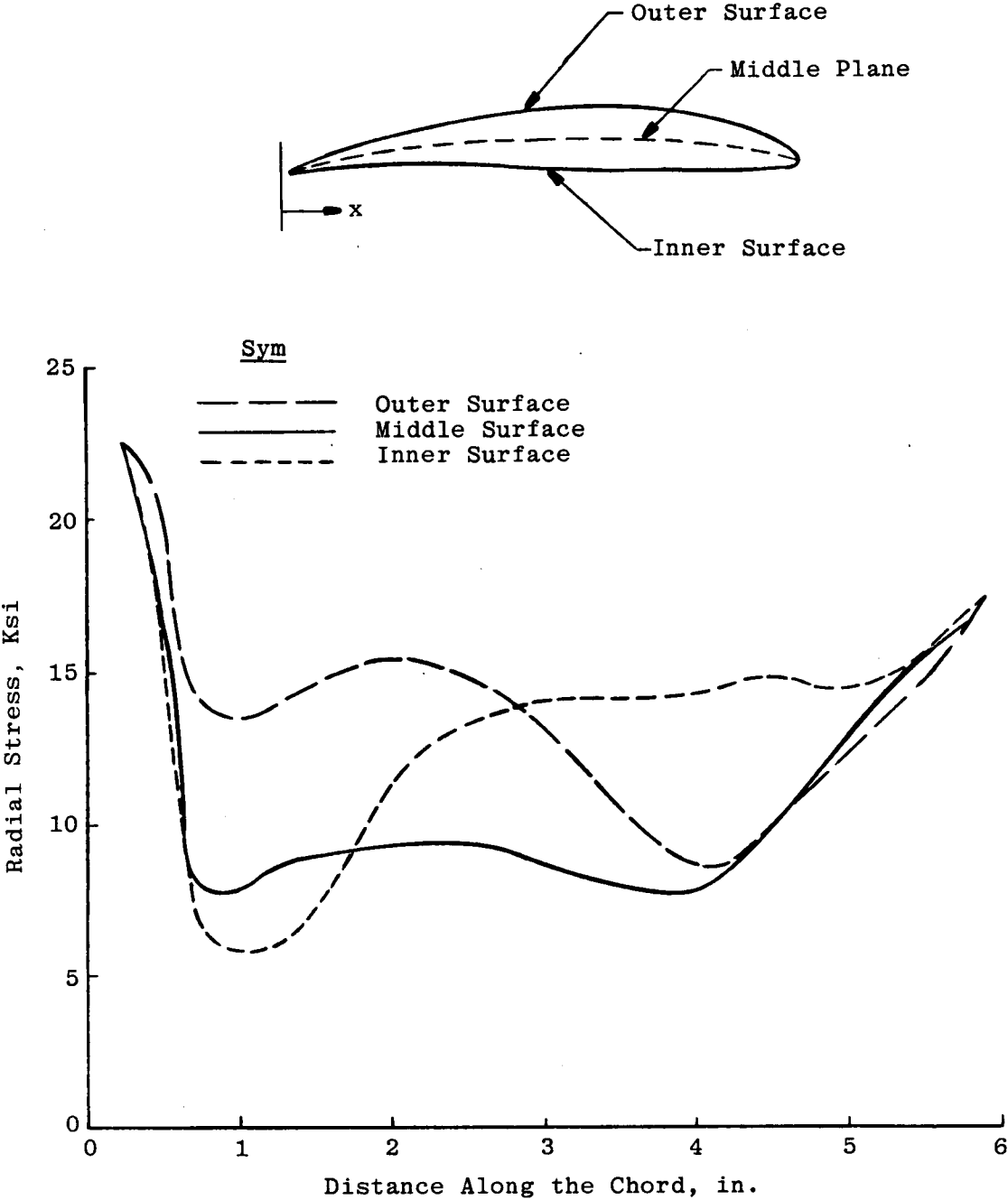


Figure 24. XS-2 Blade Steady-State Stress Distribution Along Blade Chord of Airfoil Root

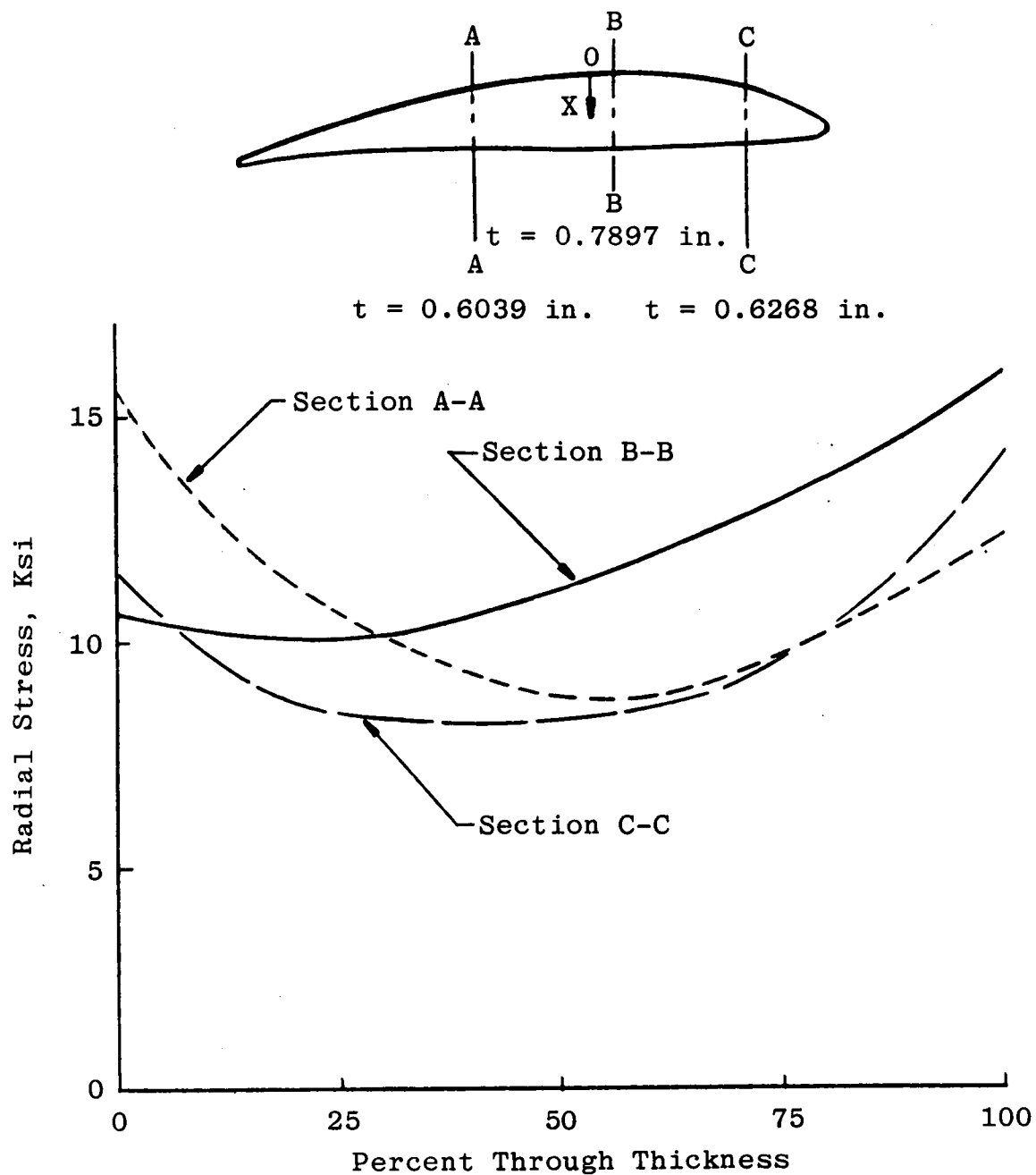


Figure 25. XS-2 Blade Steady-State Stress Distribution at Airfoil Root

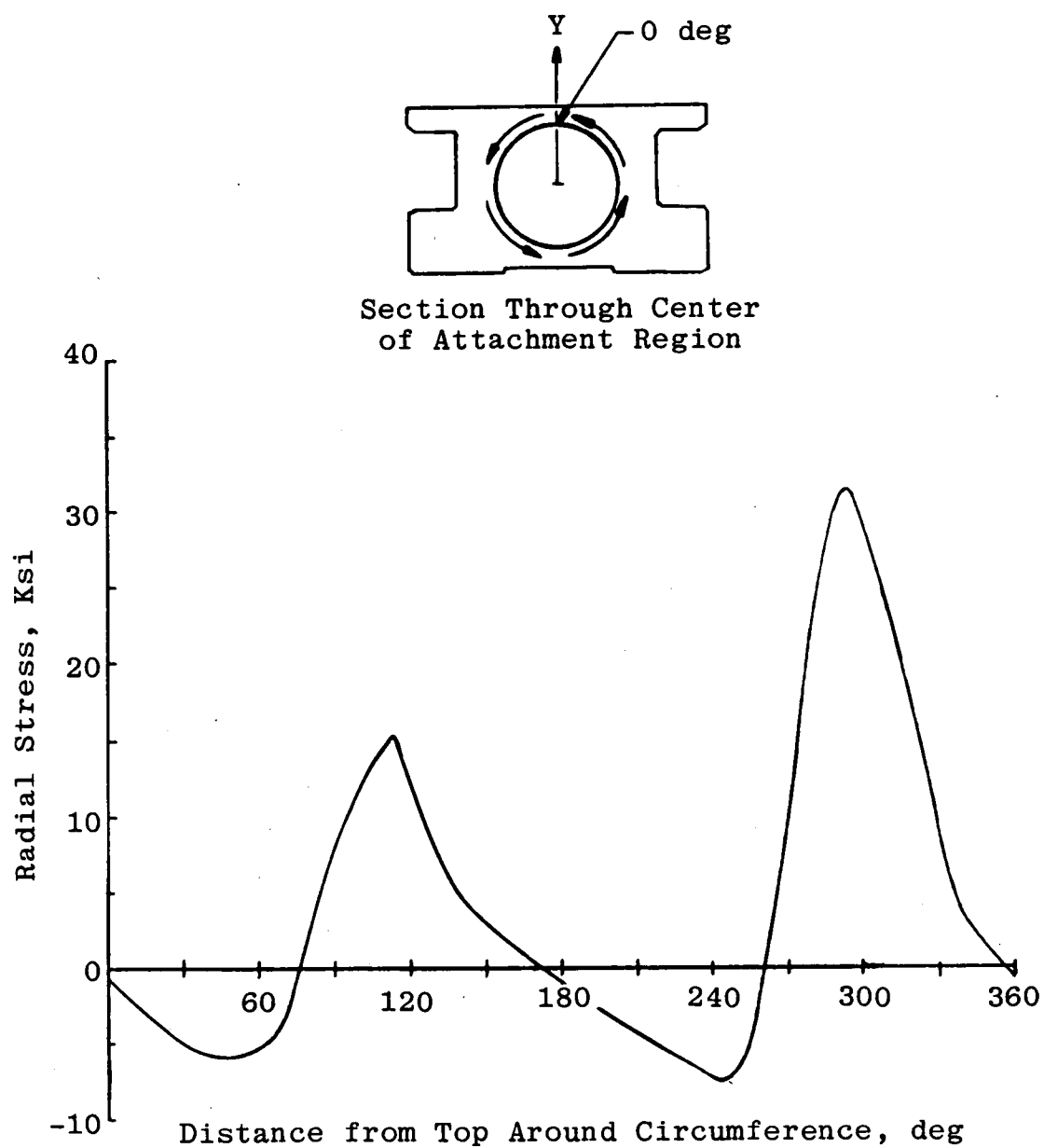


Figure 26. XS-2 Blade Steady-State Stress Distribution Around Hole in Attachment Region

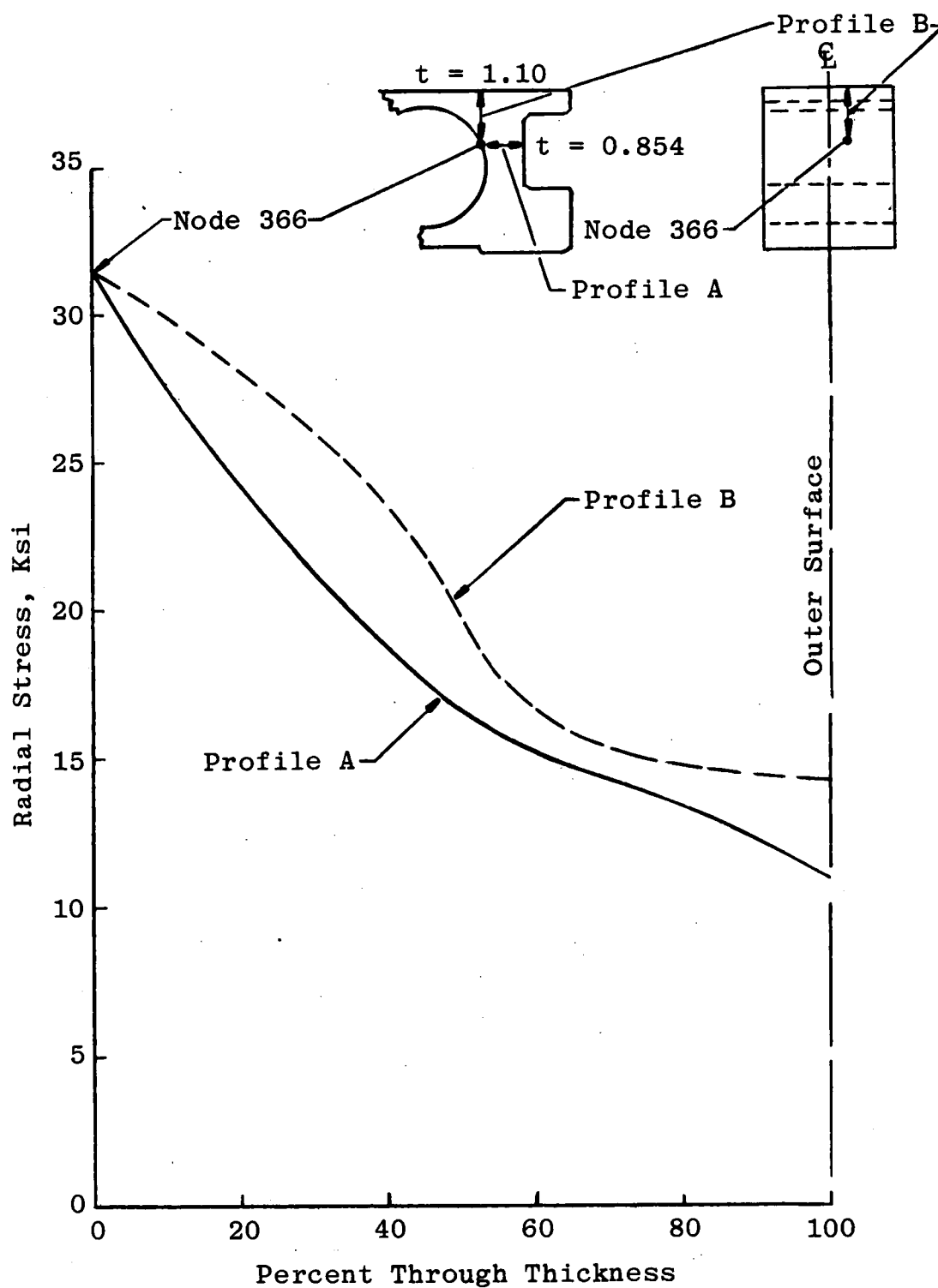


Figure 27. XS-2 Blade Steady-State Stress Distribution in Attachment Region

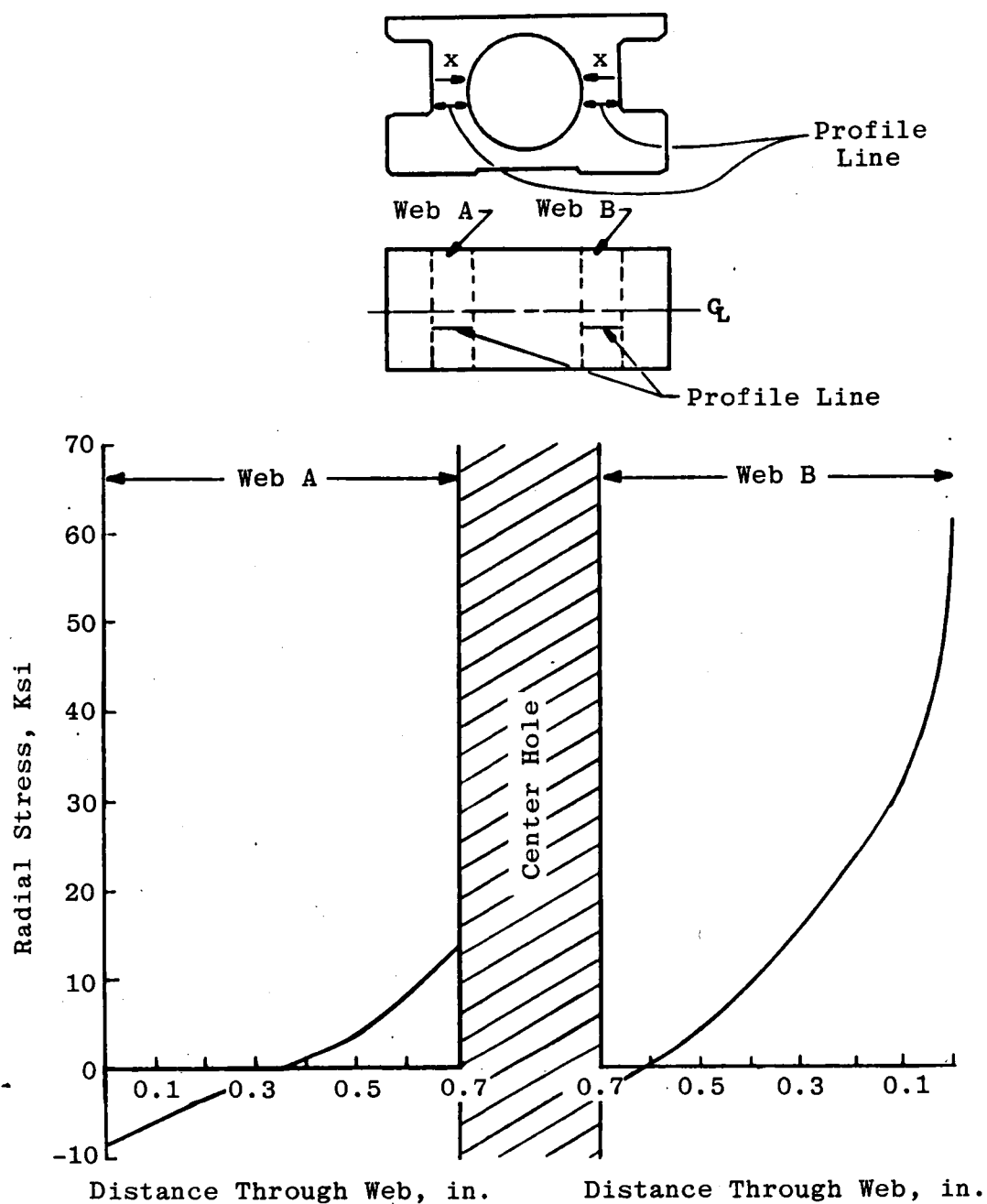


Figure 28. XS-2 Blade Steady-State Stress Distribution Through Webs of Attachment Region

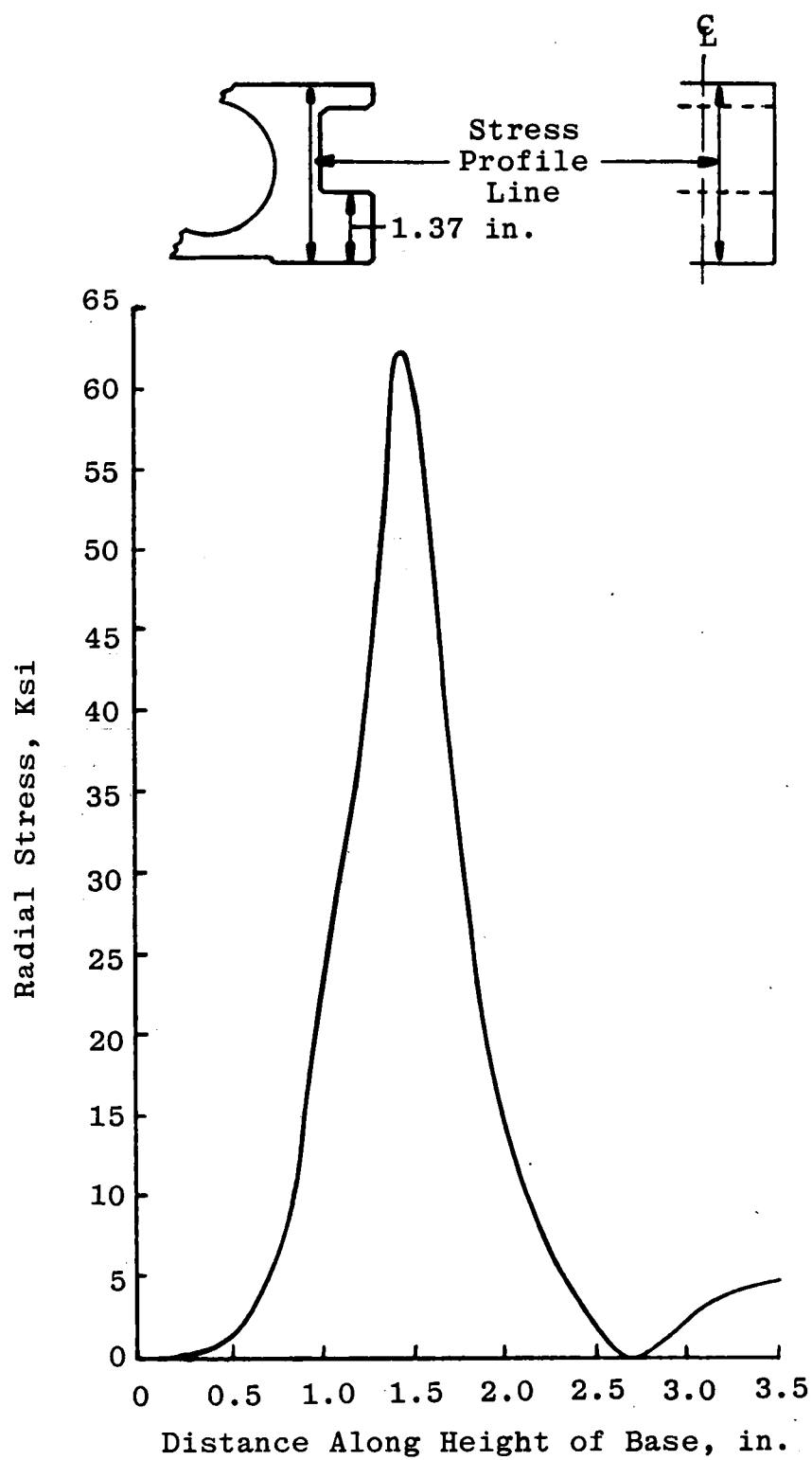


Figure 29. XS-2 Blade Steady-State Stress Distribution in Attachment Region

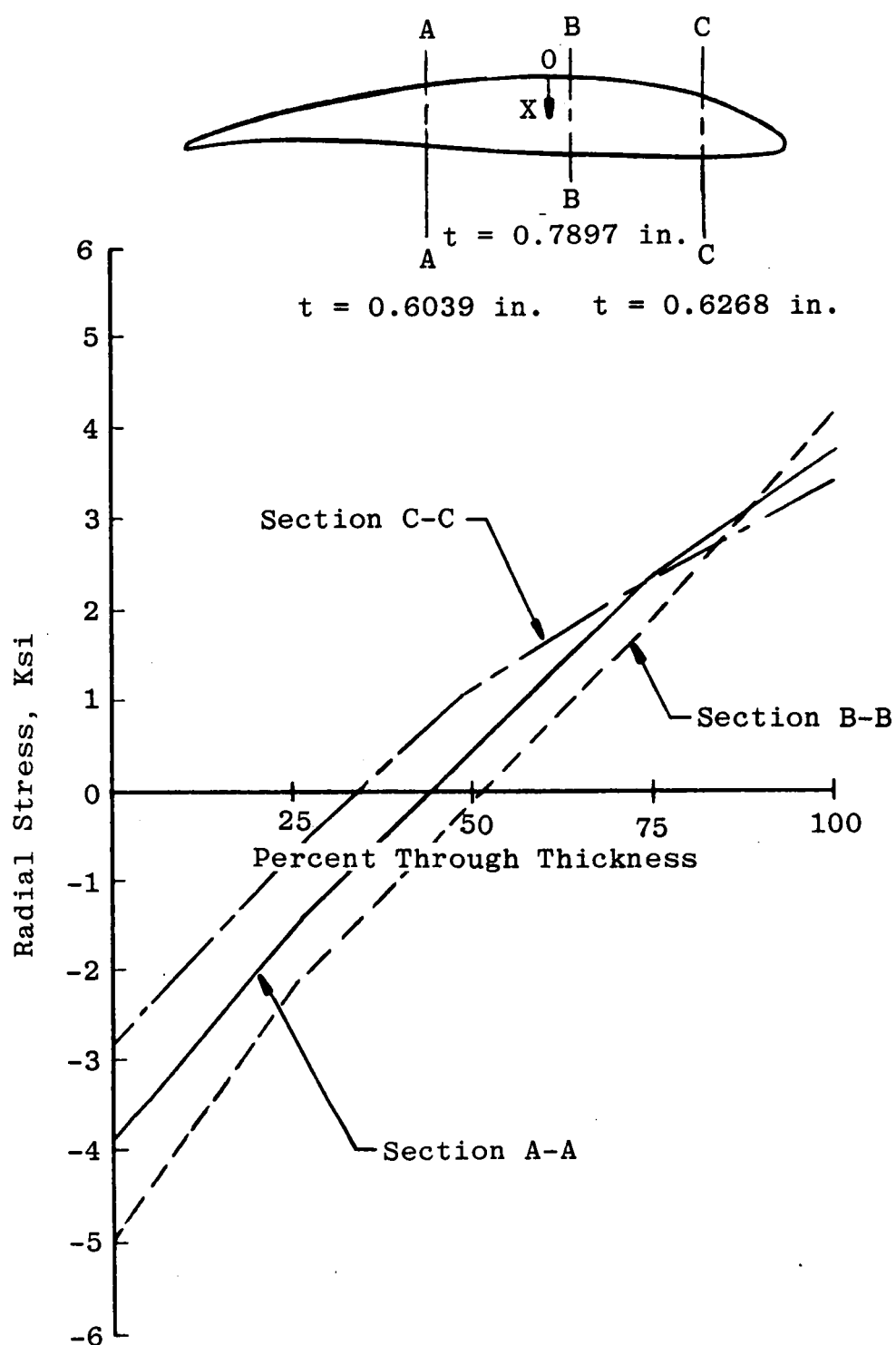


Figure 30. XS-2 Blade First Bending Mode Dynamic Stress Distribution Through Blade at Airfoil Root ($\Delta = 10$ mil)

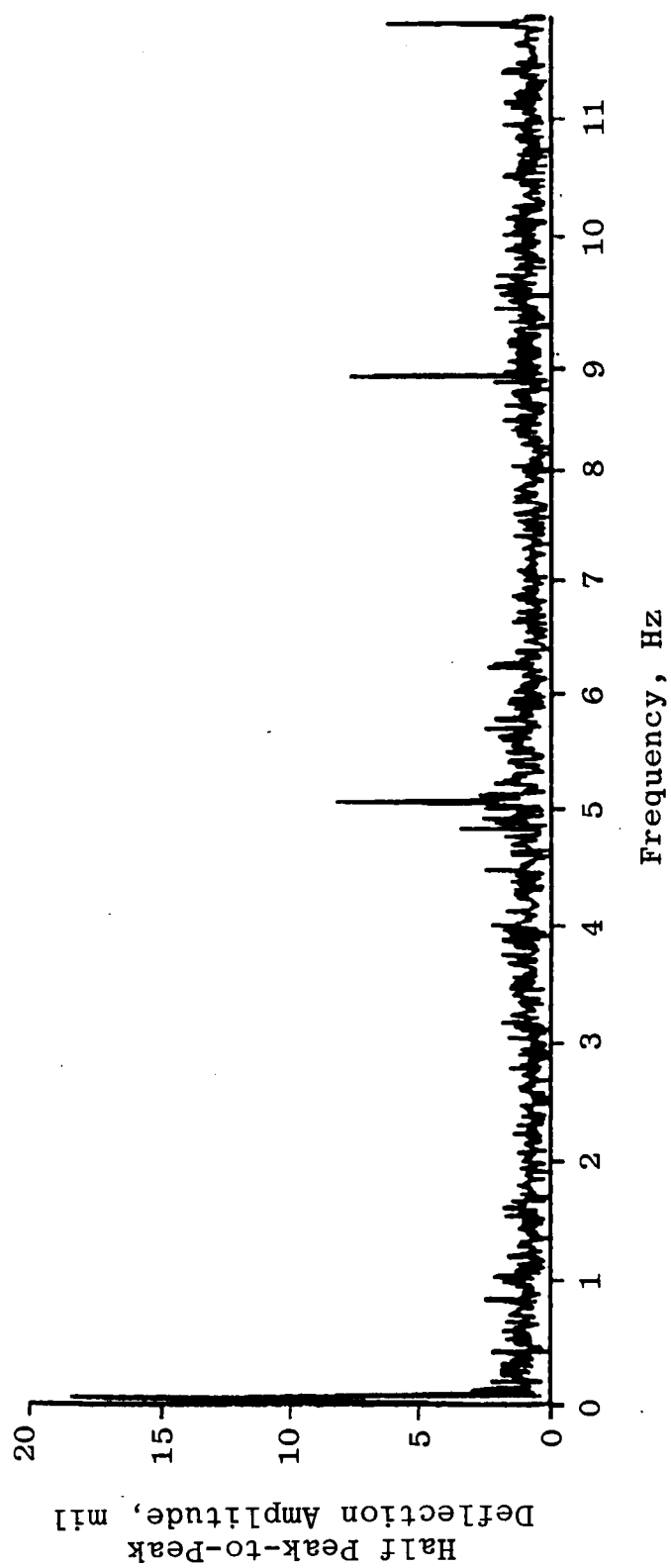


Figure 31. Amplitude-Frequency Results for XS-2 Blade Deflection Spectrum, Time = 10 to 95 Seconds

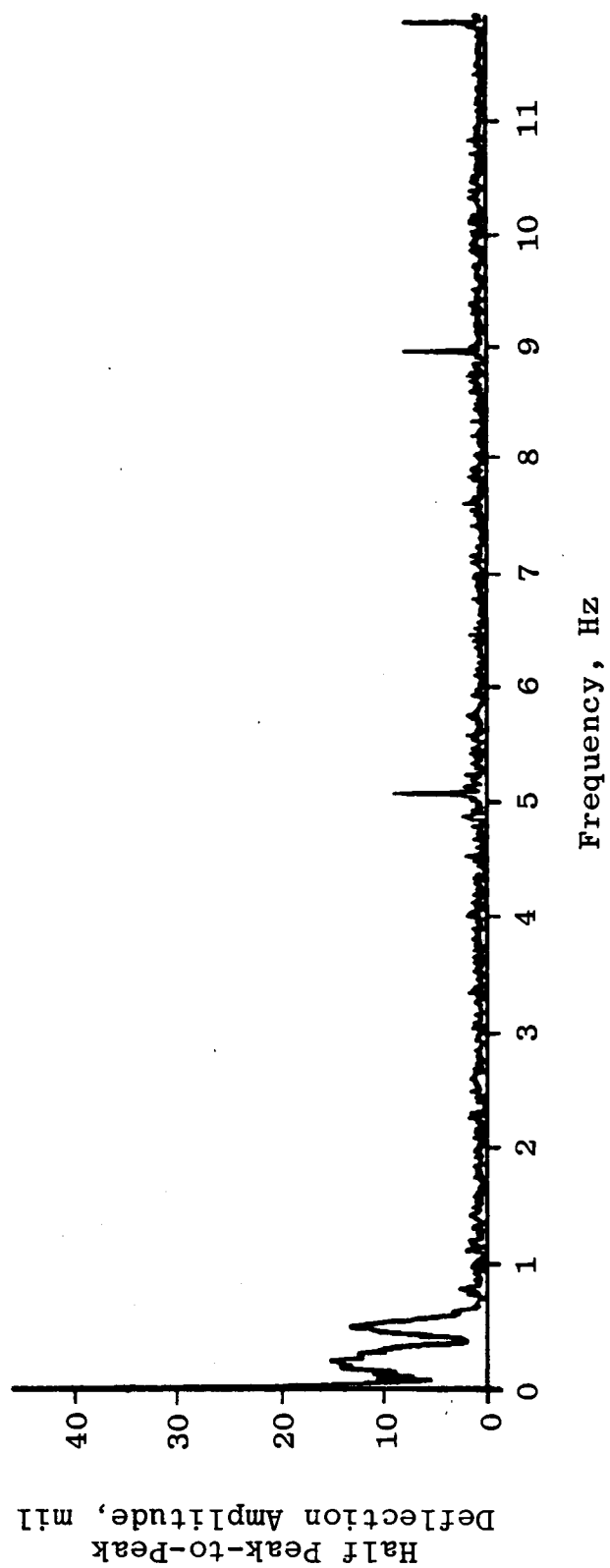


Figure 32. Amplitude-Frequency Results for XS-2 Blade Deflection Spectrum, Time = 10 to 180 Seconds (Normal Operation Plus Surge)

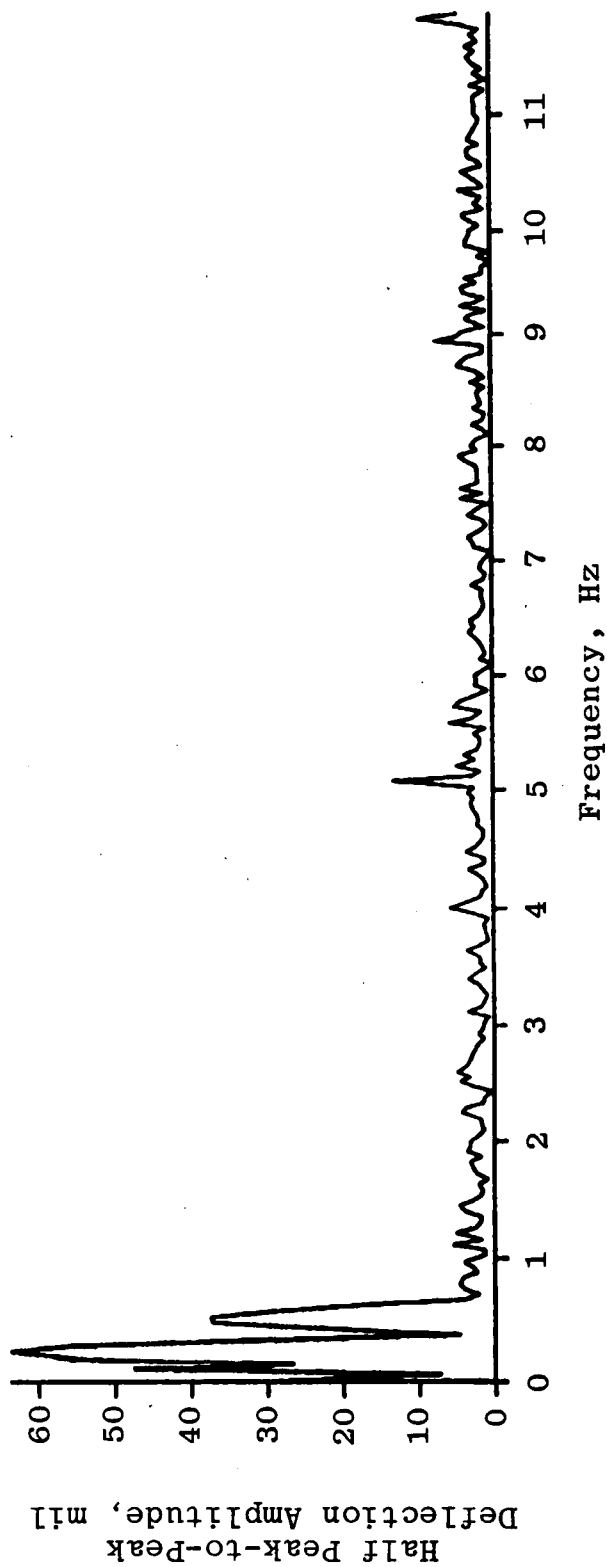


Figure 33. Amplitude-Frequency Results for XS-2 Blade Deflection Spectrum, Time = 140 to 161 Seconds (Surge)

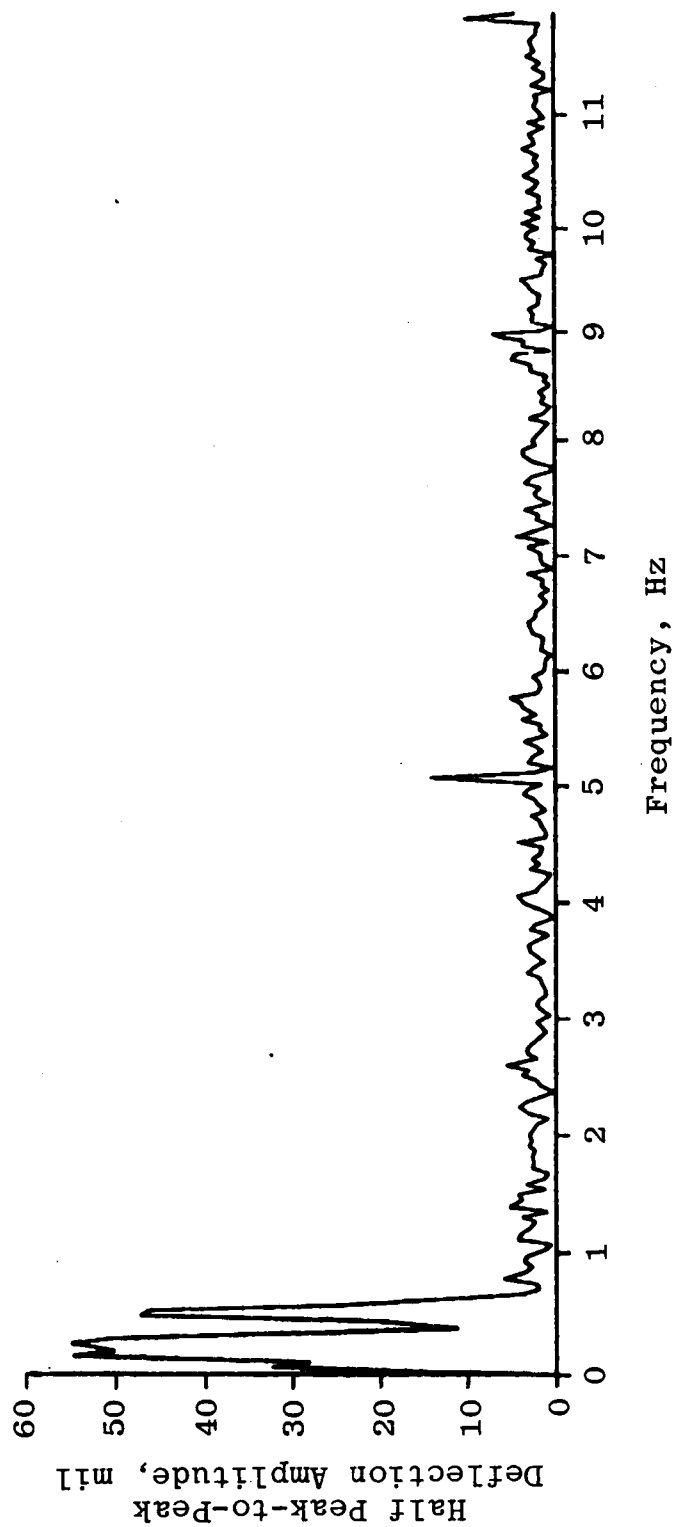


Figure 34. Amplitude-Frequency Results for XS-2 Blade Deflection Spectrum, Time = 150 to 171 Seconds (Surge)

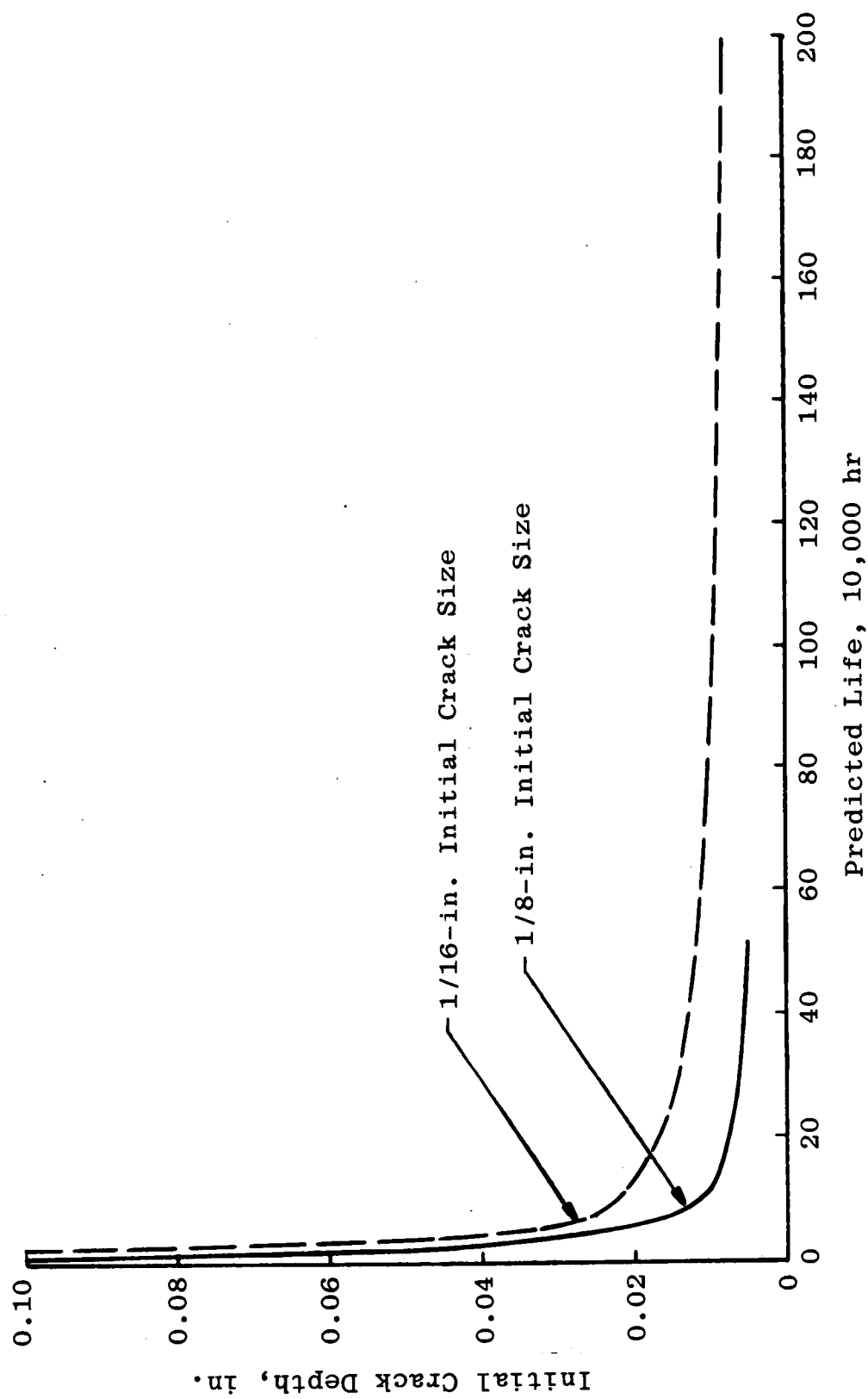


Figure 35. XS-2 Blade Structural Life Predictions

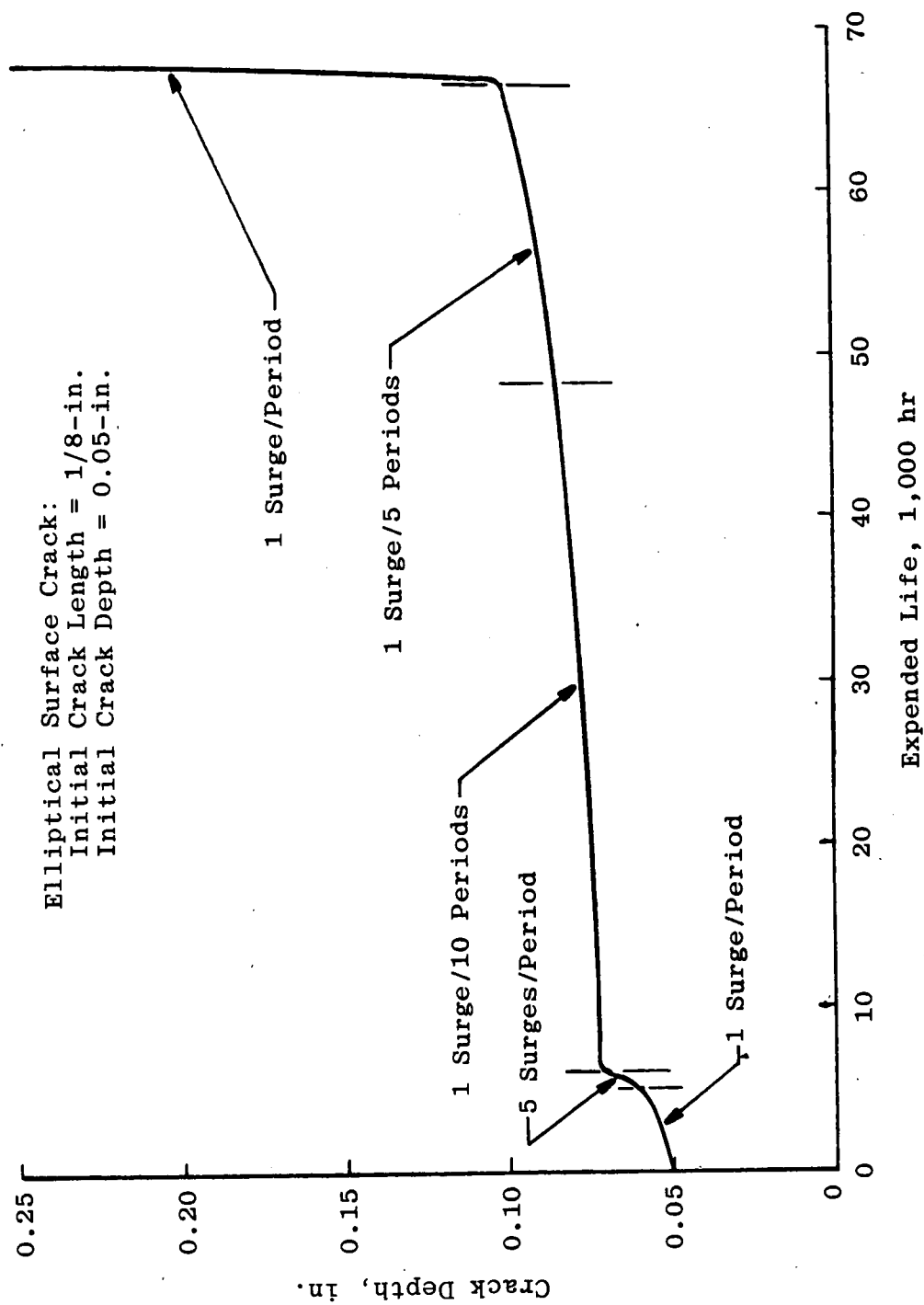


Figure 36. XS-2 Blade Structural Life Predictions Based on
An Operational Scenario

VITA

Louis Richard Deken was born in Qulin, Missouri on January 4, 1943. He attended elementary school in Glennonville, Missouri and graduated from Qulin High School in June 1961. He began study in a pre-engineering program at Southeast Missouri State College in September 1961. In 1963, he entered The University of Missouri at Rolla where he received a Bachelor of Science degree in Civil Engineering in August 1965.

After a brief employment with the Missouri State Highway Commission, he completed a four-year tour of duty with the United States Navy. He accepted employment with Sverdrup and Parcel & Associates, a consulting engineering firm, in St. Louis, Missouri in 1969, where he specialized in the structural design of bridges. During this employment, he attended The University of Missouri at Rolla where he received a Masters degree in Civil Engineering in 1973.

He attended Washington University in St. Louis, Missouri as a part-time student until he accepted a transfer in employment to Sverdrup Technology, Incorporated in Tullahoma, Tennessee in 1977. With Sverdrup Technology, Incorporated, he specialized in structural dynamics and finite element analysis. He continued graduate studies at The University of Tennessee Space Institute until 1982 when he received a graduate study grant from Sverdrup Technology and sponsored by the United States Air Force. At this time he entered The Graduate School of The University of Tennessee, Knoxville, where he received the degree of Doctor of Philosophy in Engineering in August 1986.

The author is a member of the National Society of Professional Engineers and the American Society of Civil Engineers. He will continue employment with Sverdrup Technology, Incorporated after graduation.