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Success by Design: Progress Through Failure Analysis

MFPG 21st Meeting

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MFPG
Success by Design: Progress Through Failure Analysis

Proceedings of the 21st Meeting of the
Mechanical Failures Prevention Group,
held at the National Bureau of Standards,
Gaithersburg, Maryland, November 7-8, 1974

Edited by

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Metallurgy Division
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Washington, D.C. 20234

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FOREWORD

The 21st meeting of the Mechanical Failures Prevention Group was held November 7-8, 1974, at the National Bureau of Standards in Gaithersburg, Maryland. The program was organized by the MFPG Design committee under the chairmanship of Mr. Robert E. Maringer of Battelle, Columbus Labs. The Design committee, the session chairmen, and particularly the speakers are to be commended for the program.

The papers in these Proceedings are presented as submitted by the authors on camera ready copy, except for some minor editorial changes. In addition to the papers, the Proceedings include the discussions of the talks and a panel discussion. The discussions were recorded at the meeting and have been edited to improve readability.

Appreciation is extended to Mr. T. Robert Shives and Mr. William A. Willard of the NBS Metallurgy Division for their editing, organization, and preparation of the Proceedings, to Mrs. Sara R. Torrence of the NBS Office of Information Activities for the meeting arrangements, to Mr. Harry C. Burnett of the Metallurgy Division for general coordination and registration, to Mr. Ronald B. Johnson of the NBS Institute for Materials Research and Mr. Paul M. Fleming of the Metallurgy Division for handling financial matters, and to the entire staffs of the Metallurgy Division and Institute for Materials Research for their assistance in many ways. Special thanks are accorded to Mrs. Marian L. Slusser and Mrs. Bronny Webb of the Metallurgy Division for their diligent efforts in transcribing the recorded discussions.

ELIO PASSAGLIA
Executive Secretary, MFPG

Chief, Metallurgy Division
National Bureau of Standards

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* Indicates speaker when a paper had more than one author.

The following registrants gave presentations at the Symposium, but did not submit a manuscript for publication:

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John C. New, "Test Philosophy for Unmanned Spacecraft", Session I

Alan H. Schoem, "View of the Consumer Product Safety Commission", Session III

ABSTRACT

These Proceedings consist of a group of twenty two submitted papers and discussions from the 21st meeting of the Mechanical Failures Prevention Group which was held at the National Bureau of Standards in Gaithersburg, Maryland on November 7-8, 1974. The central theme of the Proceedings was improvement in design through failure analysis. Emphasis was on design philosophy, the use of failure analysis case studies as an educational tool, successful redesign through failure analysis, and design oversights.

Key words: Design; engineering education; failure analysis; failure analysis case histories; failure prevention; reliability; safety.

UNITS AND SYMBOLS

Customary United States units and symbols appear in many of the papers in these Proceedings. The participants in the 21st meeting of the Mechanical Failures Prevention Group have used the established units and symbols commonly employed in their professional fields. However, as an aid to the reader in increasing familiarity with and usage of the metric system of units (SI), the following references are given:

NBS Special Publication, SP330, 1974 Edition, "The International System of Units."
ISO International Standard 1000 (1973 Edition), "SI Units and Recommendations for Use of Their Multiples."
E380-74 ASTM Metric Practice Guide (American National Standard Z210.1).

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SESSION I

DESIGN PHILOSOPHY

Chairmen: R.E.Maringer
Battelle Memorial Institute

D.D.Fuller
Columbia University

CREATIVE STUDENT ENGINEERING DESIGN

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The Creative Engineering Design Display is an activity of the Engineering Design Graphics Division of ASEE. The growth of this activity in this particular division is the result of several causes, any one of which might be debated by someone. Members of the division might argue that it grew out of an attempt to re-emphasize the design aspects of engineering in an otherwise science-heavy curriculum. A few might counter that it was an attempt to pump some life into what they consider dull graphics courses.

Seriously, however, Alfred Whitehead advocated that universities and their courses could not exist simply to disseminate information, but had to explore the possibilities and implications of that knowledge. Many individuals in the division felt that their drawing courses of the mid-sixties had to be altered to meet this criterion. Since engineering graphics is a language for conveying ideas, it was concluded that it could be naturally combined with a course in basic conceptual engineering design. The concept of open-ended problems had been established earlier, and it seemed to be only a short step to consider problem-solving methods. By doing this, several contributions might be realized. Students would gain an appreciation for the role of engineering graphics in their careers and for the courses included in various curricula. Students would also have an opportunity to fulfill their desire to be creative, which is probably why they selected engineering in the first place. Finally, this early experience would motivate and tend to reduce attrition among promising students.

In 1967, division chairman Edward Jacunski appointed Percy Hill to develop a summer school in creative design in conjunction with the 1967 ASEE conference at Michigan State University. Many people, too numerous to mention, contributed to the success of that school, but Henry Fuchs and Peter Bulkely should be recognized for their invaluable work. Many schools were represented and incorporated these concepts into their courses soon after that summer school. Professor Jacunski's successor as division chairman, Gene Paré, felt that the results of that summer school ought to be displayed, and created the committee establishing the first creative engineering design display for the 1968 ASEE conference held at UCLA. The society accepted the display and its basic concepts immediately and enthusiastically, and it has been a major event at every conference since. Judges from educational and industrial sectors are invited to determine the winning entries. They are enthusiastic about the display and are eager to return.

Initially, the display was limited to freshmen since the bulk of the courses represented in the division were being taught at the freshman level. However, many members within and outside the division who were directing their efforts toward creative design at other academic levels expressed a great deal of interest and today there are six categories for entries: Freshman, Sophomore, Junior, Senior, Graduate and Co-op. As a result of the efforts of Borah Kreimer and Ralph Blanchard of Northeastern University, financial assistance has been obtained to make several prizes available and guarantee the continuity of the committee's operation, which requires a large annual budget for mailing, etc. All students represented in the display receive a certificate on parchment; entries winning first place in each category are awarded $100, and the sponsoring school receives a plaque.

What the entries represent

The projects exhibit a broad range of subject material and sophistication, from bicycle wheel retainers to biomedical devices and operational computers. The students represented often show a surprising ability to handle material far beyond their academic level when properly challenged. Almost all the entries represent the conclusions made in solving an open-ended problem "situation", in which the instructor offers guidance and suggestions, but not rigid control. Entries are usually made by teams of students, and include a formal written and graphic report; frequently an extremely well-executed model or prototype. Study of the entries discloses the influence of the summer school. Teams almost always follow a series of steps, or design process, which converges upon a satisfactory solution. While no single set of steps is used, the processes exhibit certain commonalities. In general, the teams must probably do five things.

First, analyze the causes, consequences, and limits of occurrence in order to define the true nature of the problem represented by the situation.

Second, investigate the market for existing solutions, and augment their knowledge where obviously needed data is missing. The instructor usually assists the typical freshman team at this point, suggesting source material or experts in the field, and by asking questions that the team cannot answer and suggesting a way of determining the answer.

Third, assemble and synthesize a wide variety of solutions to the problem. Creativity is generally stressed; brainstorming sessions are held and methods of idea-promotion are discussed. The rationale here seems to be that the existence of the problem, when solutions are available, indicates that none of the solutions are wholly satisfactory and the problem is not completely understood or has not been adequately investigated; a new and creative approach is needed.

Fourth, logically select the best solution from those they have generated, and defend that selection, frequently during an oral presentation.

Fifth, describe their solution well, demonstrating their understanding and ability to create a high quality formal report using graphics and report writing.

Lee Harrisburger says that every audience is motivated by personal interests - the "What's in it for me?" syndrome. What does this have to do with the Mechanical Failures Prevention Group?

Students who have been exposed to this open-ended design experience have a "mature" attitude toward design. They are aware that many solutions may exist for any problem, and that any new solution must compete in the marketplace with established and future competition. They are conscious of the many facets of a design that must be considered in formulating it; reliability, production cost, safety, ease of operation, failure modes of all kinds, human factors, and so on. They are unlikely to accept blindly a pat definition of a problem without questioning its validity or at least examining it. They are generally very motivated about design; they have had at least one experience where they tried to "put it all together" on a rational basis, succeeded to one degree or another, and can benefit from their first mistakes.

Having received a rather classical and frequently tedious education in mechanical engineering, I join the many associates with whom I have discussed this topic in saying that I wish we had been exposed to this material in our freshman year. Many of the things we are teaching freshmen now had to be found out the hard way <u>after</u> graduation and at no small expense to the employer or the productive time of the employee.

I believe that the work exhibited in these displays indicates that there is a new crop of young engineers becoming available who are very conscious of problem-solving techniques and who will be able to perform competent engineering design work much earlier because of this exposure.

DISCUSSION

B. P. Bardes, Bimba Manufacturing Company: One of the results from offering a case study program to freshmen and sophomores is that these people have not been told that they do not know how to solve the problem. Therefore, they just go at it with the best they've got, and very frequently they come up with not only workable, but elegant solutions.

C. O. Smith: Very true. Arizona State introduces a design project to their freshman engineering students the day that they walk onto the campus--not towards the end of their freshman year. This program has been in existence for ten years. The results have been quite good, particularly in lowering the attrition rate. After the design projects have been completed, a panel of experts is assembled to judge the students' efforts. I had the pleasure of serving on that panel at one time. The primary consideration in the judging is method--not the attractiveness or degree of sophistication of the specific project. In my opinion, and in the opinion of many of my colleagues in many institutions, method or approach is again considered to be the critical aspect.

R. E. Maringer, Battelle Memorial Institute: I think this kind of approach to education puts a little fun into it for the students.

C. O. Smith: And a different kind of frustration.

R. E. Maringer: I have heard that viewpoint expressed by other people who complain bitterly about having all the rough courses in the first semester so you can flunk out all the people who really are not appropriate, instead of having those courses in the first semester which will really grab the students' interest and get them enthusiastic before you show them how tough life is.

THE DESIGNER'S CONTRIBUTION IN THE DEVELOPMENT OF NEW PRODUCTS

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Abstract

The designer's main responsibility is in the aesthetics of the product, but he often can make a contribution in other areas as well. A designer is trained to look at problems differently from engineers concerned with the specifics of function and cost benefit analysis. The designer is interested generally in the product and in its larger context particularly as it relates to user needs.

The best way to illustrate that contention is to present a project that I worked on while employed as an Industrial Designer at the Westinghouse Electric Corporate Design Center. The project was the Tampa Airport Shuttle System - often called Skybus. I arrived on the project after specifications and contracts were signed. My job was to work on a detail that was causing a problem and in desperation it was given to design to see what we could "come up with". The problem was a door located at the ends of the vehicle. After looking at the concept for a few days it became quite obvious that the problem was not with the door but with its location. I will discuss with slides my method of analysis and conclusions, which resulted in a total revision of the specifications of the vehicle and of the architecture of the building.

For the program Success by Design; Progress Through Failure Analysis, I would like to begin my presentation, "The Designer's Contribution in the Development of New Products" by making four points that cover some explanatory aspects of industrial design.

Traditionally, the Industrial Designer's primary responsibility in product design is aesthetics. Aesthetics remains a prime concern of the Industrial Designer but new approaches to problem solving are changing this traditional approach.

1. By training, the Industrial Designer is conditioned to be sensitive to user needs and depending on the individual designer the concern for user needs often takes precedence over the traditional approach to design. Functional analysis with respect to user need (or performance specification) has become the criteria for making the aesthetic decisions in product design.

2. The Industrial Designer can make significant contributions to product performace if he is a member of the product development team. If the Industrial Designer is brought into the picture after major concept decisions have been made and tooling has begun, he can only make cosmetic contributions. This is true of all professions, but the Industrial Designer (through his own fault) has been considered more of a decorator of concepts than a contributor to the concept.

3. One method that the Industrial Designer uses in his approach to design is a search for simplicity: simplicity in production and simplicity for the user.

4. Pertinent to the subject of this conference the Industrial Designer can, through analysis of user need, (a) help prevent performance failure and (b) reduce the possibility of mechanical failure by the elimination of mechanisms in favor of the development of simple and direct use of the product. This is usually done with innovative use of existing or known technology.

I would like to illustrate these points with a brief review of a project that I got entangled in as a young and recent addition to the Design Department at the Westinghouse Electric Corp. in Pittsburgh, PA. Although that was in 1965, the system has only recently been in operation.

The project was the Tampa Airport Passenger Shuttle System (often called Skybus) which began operations in April, 1971. The Tampa System has eight vehicles and a total track length of 7,600 ft., with an annual average mileage at 382,000 and about 13 million passengers. The operations and maintenance cost per month is $23,000, with a total system investment of eight and one quarter million dollars. Cost per passenger is around $.07.

Back in 1965 shortly after July, the month I began working at the design center, I was given the assignment to investigate and design a door for the proposed vehicle at Tampa. After discussing the project briefly with my boss he left for a 3 week vacation. It soon became clear that the specifications and contact for an end loaded vehicle would cause many mechanical problems; but a more serious situation was that it could not do the job that it was expected in the time that it was expected.

Total turn around was to be 140 seconds per cycle. A cycle was defined as; 50 people loading, 40 sec. travel, 100 people unload/load, 40 sec. travel, 50 people unload.[1] In 60 seconds 200 people were expected to pass through a door that could not be greater than 7½ feet. There were penalties in the contract if the system could not meet the requirements. The time requirements seemed totally unrealistic particularly with the added problem of computer controlled docking and some vague ideas on the population that would be using the airport. At the end of the third week my boss returned expecting to take the drawings or even a model of the door to a pre-arranged meeting with the hopeful engineers. Of course my boss was less than pleased when instead of a model, I handed him a paper with a critique of the end loaded proposal and my alternate solution which was a double side, six door vehicle. The engineers were equally disappointed and dismissed my proposal saying that the contract specifications were signed.

I would like to show some slides of the Tampa project which may illustrate the design process, that was used to change their minds. I will begin with the original end loaded proposal.

Figure 1 is an artists concept of the original proposal which was taken from an advertisement in the September 1965 Scientific American Magazine.[2] It is obscure but the drawing indicates some sort of folding door at the ends of the vehicle. I was given the job to make the end loading concept work by designing a door that would magically solve the problems posed by the concept. As I began working on the project a number of problems began bothering me. Most important was the fact that the passenger would have to pass over a space between the vehicle and the building. From our experience with elevators we knew that people are uncomfortable with this kind of situation. People prefer to step onto a platform that feels more secure, like a platform parallel with the direction of movement.

[1]Raised figures indicate literature references at the end of this presentation.

Figure 1

Figure 2

Another was the fact that the power lines were located directly below, which is shown in figure 2. A number of controls and mechanisms would be required to dock the vehicle but that was mechanically possible if it was necessary to do it that way. It was a "technologically limited" problem. A problem that was "societally limited" and not solvable was the fact that people simply could not move as fast as they were expected in the time that was given through the door that was proposed. Figures 3 and 4 is a diagram of the problem of the problem of people unloading and loading in the end loaded concept and the double side loader. The double side load concept separates unloading passengers from the loading passengers. Six doors were used to cut the time needed to load and unload. Figure 5 shows one of the doors on the current installation with information for passengers of the system. Figure 6 is a view of the vehicle showing doors on both sides of the vehicle.

Returning to the end loaded concept for a moment, figure 7 shows the full scale mock-up of the end loaded concept which was built to attempt to show that people could pass through the door in the required time. This study was conducted at the South Park site in Pittsburgh, PA.[3] The structure was then shipped to Tampa to try and demonstrate to the Tampa people that the concept would work. Needless to say the proponents of the end loaded system finally demonstrated to themselves that their concept would not work. Besides the

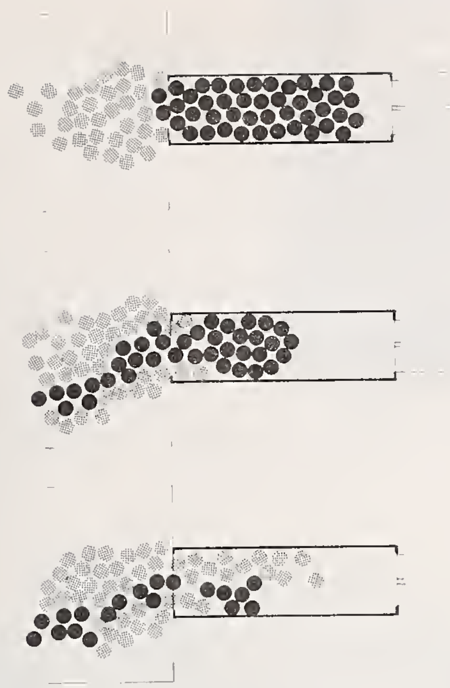

Figure 3

Figure 4

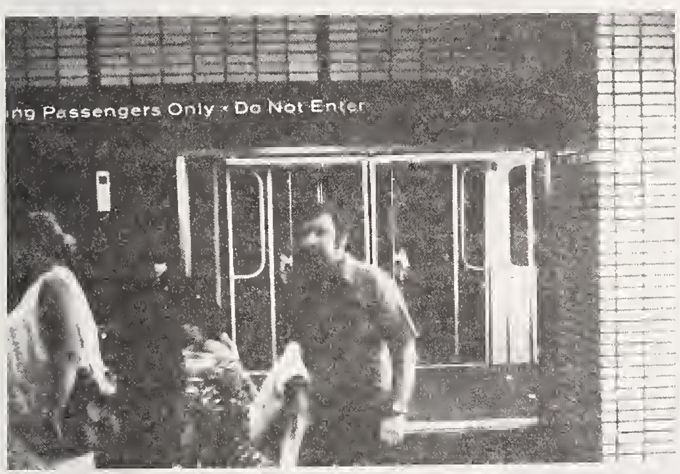

Figure 5

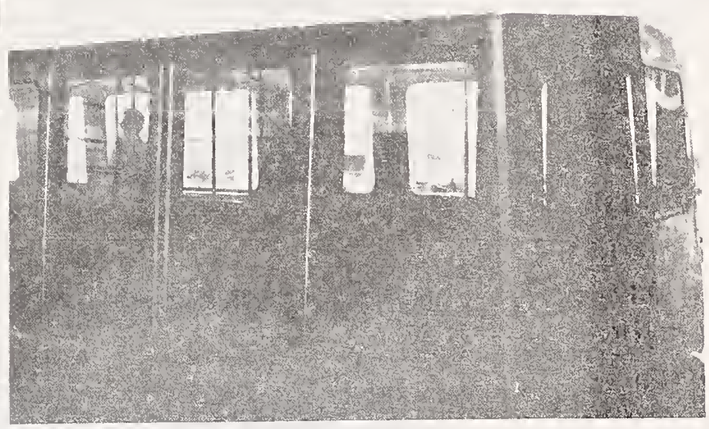

Figure 6

time and gap problems the end loaded concept posed a number of other serious design problems. (1) If the vehicle stopped short of the dock or the vehicle was on fire the passengers could not get out through the doors. They would have to break the windows.(2) If the vehicle made an emergency zero deceleration period stop, which it was programmed to do, the 100 passengers would be forced against the door. With that kind of force on it, the door in all probability would fail and the passengers would continue out of the vehicle.

Figure 7

Figure 8 shows what a passenger would see in that case; figure 9 is another view.

Figure 8

Figure 9

If any of you have used the Tampa Shuttle System, you probably remember that it is one large box which can hold 100 people. I mentioned earlier that the vehicle could make an abrupt stop which it did once during normal operations. All of the passengers were forced against the end of the vehicle, and some passengers were seriously injured. I had recommended that the vehicle be separated into three compartments to minimize the effects on the passengers in the event of a quick stop. An additional benefit was that one section could be equipped for use by the elderly and infirmed passengers. Additionally, internal bracing of the vehicle would be possible which would increase the structural strength, and it would also be lighter and cost less to build. I mention that because it demonstrates point #1 that I made in the beginning which I would now like to briefly develop.

First I would like to explain that after the concept phase was completed the Tampa project was taken from the design center and given to consulting firm. The decision to ignore the report and to keep the vehicle open rather than to have compartments was based on aesthetics and not on user need. This decision was made with management level power over engineering. There was a breakdown in the Product Development team. The results of course were a number of injured passengers and a number of law suits against the manufacturer and owner. Further costs will be incurred in the event of other accidents which will lead to the eventual conversion to a compartmented vehicle. I believe that it is necessary to maintain teamwork throughout the project, and that all final decisions be based on user need.

I would now like to conclude with figure 10, a photo of the final vehicle and installation and by quoting from a letter by the project engineer which summarizes the important points. "The Industrial Design Department contributed significantly in the conceptual design phase of the project . . . and was instrumental in providing information and a design concept of a double side load vehicle which established an efficient station design . . . The double side system eliminates passenger congestion and maximizes passenger movement, with a reduction in time for system turn around. Additionally it made the control system design significantly easier since it was not necessary to dead-end the car to a mating platform at each end of travel."[4]

Figure 10

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2. Scientific American Magazine, September 1965.

3. Westinghouse Electric Corporation, Pittsburgh, PA, Tampa Report Number 65-202-01, Tampa/South Park Passenger Flow Study, December 10, 1965.

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Tampa International Airport Passenger Shuttle System, WTD 71-521A, Westinghouse Electric Corporation, Transportation Division, E. Pittsburgh, PA, 15112, December 1971.

DISCUSSION

R. Lenich, Caterpillar Tractor Company: In industrial design, do you school the students in design features such as stress risers that could lead to failures?

J. J. Lesko: At Purdue we have a joint program with the Mechanical Engineering and other engineering schools. It is our philosophy to develop dialogue and interaction among the various engineering departments. For instance, last year our junior-senior project was a vehicle design, using hydrogen fuel because of the energy crisis. We had engineering students acting as consultants to the industrial design group and the design group acting as consultants to the engineers much like it would happen in industry. We made major decisions in our design process that were based on information from students in other departments. There were a number of significant design changes based on information that was given from engineering to design, and also from design to engineering. We do not feel that designers should be second rate engineers, nor do we think that engineers should be second rate designers. We feel that it is more important to develop the proper teamwork so that when a student graduates and goes into a real-world situation in design, he will have enough experience to communicate with an industrial engineer.

G. Wagner, Westinghouse Electric Corporation: A designer must consider the trade-offs between safety and cost. If three people are injured out of a total of 30 million handled by a system, is it worth changing the design to eliminate these three injuries?

J. J. Lesko: I do not think the designer should be conditioned to accept any injury. His primary role is as an idealist to say we want zero injuries. He will be in constant battle because he has no control over the funding of the project.

G. Wagner: That turns into a circular argument in terms of safety. You can always hypothesize, but at some time you have to face realistically the problems created in your field and in industrial engineering.

J. J. Lesko: On every design development team, there should be a consumer's representative who is constantly saying, "I don't want to get hurt regardless of the cost to you. I do not care what I have to pay, but if I use your system I do not want to get hurt." He may be unrealistic and idealistic, but I think in the long run he will save the company quite a bit of money if he avoids law suits against the company.

SESSION II

CASE STUDIES

Chairmen: J.E. Stern

NASA, Goddard Space Flight Center

J.A. Houck

National Transportation Safety Board

SUCCESSFUL REDESIGN AND LAUNCH PERFORMANCE OF THE ERTS/NIMBUS ADAPTER AFTER AN ACCELERATION TEST FAILURE

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Abstract

Both the Earth Resources Technology Satellite (ERTS) and the Nimbus meteorological satellite play vital roles in the efforts of the National Aeronautics and Space Administration to focus the capabilities of space on the needs of man.

Success of these two programs is, therefore, of the utmost importance. This paper describes the redesign effort and subsequent successful launch performance of the ERTS/Nimbus spacecraft structure after a catastrophic failure was encountered during the design qualification acceleration test on an engineering model spacecraft.

The derivation of the test loads, the configuration of the test article, and the failure sequence are included. A description is given of the effort to reconstruct the spacecraft so as to glean clues as to the cause of the failure, and of the static load test which was conducted on another launch vehicle adapter to pinpoint the precise failure mechanism.

Whereas the prior mathematical model has shown adequate margin for the test condition, information gained from the static load test was used to correct the mathematical model of the adapter so that the stress analysis accurately predicted the failure which had occurred. This sequence is fully discussed in the paper.

A description of the adapter redesign, the rationale for the redesign approach taken, and the successful performance of the redesigned structure during a design qualification static load test and launch are discussed.

INTRODUCTION

Both the Earth Resources Technology Satellite (ERTS) and the Nimbus meteorological satellite programs play vital roles in the efforts of the National Aeronautics and Space Administration to focus the capabilities of space on the needs of earth.

Since its launch in July 1972, ERTS-1 has facilitated studies of water quality control, land use management, and related areas of oceanography and agronomy previously impossible without the improved resolution of its current state-of-the-art sensors. From its near-polar orbit, it scans the entire surface of the earth once every 18 days providing spectral information acquired in 4 different infrared bands. From such data, analysis of seasonal variations is made possible, thereby facilitating predictions of water resource availability and crop yield as well as other vital environmental data.

At the same time, the Nimbus series of spacecraft continues to serve as the principal program for remote-sensing research on which are developed the sensors used in the operational weather satellite program of the National Oceanographic and Atmospheric Administration (NOAA). Over 50 nations currently rely on the data transmitted from NOAA's satellites to provide timely, accurate weather prediction. With sensors currently under development for Nimbus, prediction capability of two weeks or longer is anticipated within the next decade.

Timely and successful launches of spacecraft for both the ERTS and Nimbus programs are therefore of the utmost importance, despite problems which might be encountered during spacecraft development.

BACKGROUND

With the exception of the complement of scientific instruments, the ERTS and Nimbus spacecraft are nearly identical. The two programs are managed by the Goddard Space Flight Center. Both spacecraft are fabricated and integrated by the General Electric Company at their Valley Forge Space Technology Center, and both utilize the two-stage Delta launch vehicle to achieve orbit. The spacecraft contractor has the responsibility for the structure, power, control, command, and data handling subsystems as well as for integration of the scientific instruments (developed separately by NASA and provided as government furnished equipment) and the subsequent system level performance tests and environmental tests. It is, however, the structure subsystem that is of primary interest in this paper.

The ERTS/Nimbus spacecraft structure is depicted schematically in Figure 1. The upper portion of the structure consists of the attitude control box whose base plate is one-inch thick aluminum honeycomb. The base plate is stiffened from above by a series of channel members attached in a cruciform shape. The control subsystem electronic components are mounted between the arms of the cruciform, and the pneumatics tank and thrusters are mounted to the top of the channel members.

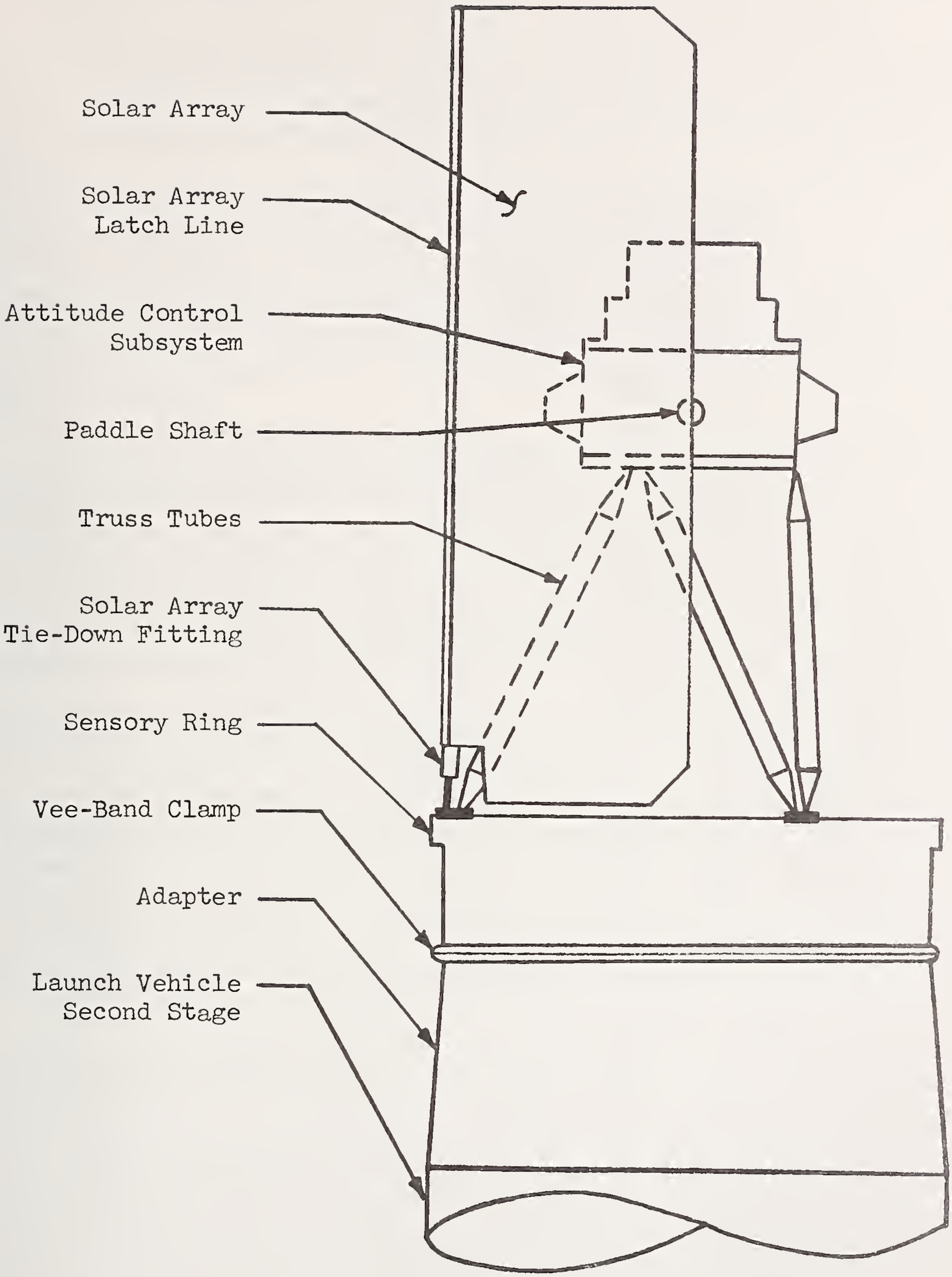


Figure 1. ERTS/Nimbus Structural Schematic

Two large solar arrays (paddles) are supported from opposite sides of the control box via hollow aluminum shafts. The solar array structure is one-half-inch aluminum honeycomb covered on one side only with solar cells. In orbit, these paddles extend outward from the control box, but during launch they are folded inward with their outboard edges joined along the latch line. During launch, additional support for thrust axis loads is provided to the arrays by the paddle tie-down fitting.

The control box is attached to the sensory ring by means of six tubular aluminum legs arranged in a truss-like fashion. Two truss tubes attach to each of three hard points on a triangular frame that is mounted to the underside of the control box honeycomb base plate. Similarly, two truss tubes attach to each of three hard points located 120 degrees apart on the upper side of the sensory ring. Uniball joints are utilized at both ends of the truss tubes to eliminate bending loads in the tubes themselves.

The sensory ring is a toroidal shell, rectangular in cross-section, that houses the major portion of the spacecraft electronics in a series of 18 bays into which the toroid is subdivided. The sensory ring structure consists basically of 18 machined aluminum separator castings joined by means of 4 aluminum rings, 2 on the outer edge of the shell and 2 on the inner edge, one of each at both the upper and lower planes of the sensory ring. Extending across the center of the toroid is an aluminum cross-beam structure which supports the complement of scientific instruments. The instruments themselves extend downward inside the shell of the adapter.

Both the lower outboard ring of the sensory ring and the upper ring of the adapter are machined surfaces joined by a tensioned vee-band clamp. After powered flight, two bolt cutters release this clamp and, by means of push off springs located in the adapter, the spacecraft is separated from the launch vehicle at this plane. The adapter itself is a frame and skin structure. The frame consists of 4 aluminum rings, the upper and lower rings are machined and the 2 intermediate ones are bent-up channel sections, and 11 aluminum longerons, 8 main ones above each of the 8 bolt locations that attach the spacecraft to the launch vehicle and 3 localized longerons. The entire frame is covered by a metallic skin which is part aluminum and part magnesium.

The above general description was provided to familiarize the reader with the ERTS/Nimbus structure. The description of the adapter is necessarily vague because it is that which failed during the acceleration test of a structural development model spacecraft. Detailed descriptions of the original adapter design, as well as of the failure analysis and the redesign effort are provided in later portions of this paper. The design of the remaining portions of the ERTS/Nimbus structure are as described above and were not revised as a result of the failure of the adapter.

TEST PHILOSOPHY

In accordance with the test philosophy of the Goddard Space Flight Center, the design of a spacecraft structure is considered qualified only after it has successfully demonstrated the capability of withstanding loads equal to one and one half times the maximum anticipated launch loads. These anticipated launch loads are computed by conducting a flight dynamic loads analysis in which a mathematical model of the spacecraft coupled with one of the launch vehicle is excited by a series of transients representative of the various load producing events expected to occur during launch. Response predictions are made at each spacecraft mass point to each of the various forcing functions. These responses are then integrated to obtain loads at each major interface of the spacecraft structure for each loading condition. Events typically included in this flight dynamics loads analysis are first stage ignition and lift-off, wind gust loading, first stage shutdown, first and second stage separation, and second stage ignition and shutdown.

For the Delta launch vehicle used by both ERTS and Nimbus, there is an additional event termed POGO for which an analysis is conducted. On the Delta vehicle, POGO is caused by propellant feed line oscillations that couple with the launch vehicle structure and induce vibratory response of the entire launch vehicle at its first longitudinal or "accordian" mode. This large dynamic load is coupled with high compressive static loads because it occurs immediately prior to first stage burnout. Goddard test philosophy defines the flight acceptance level at the mean plus two sigma value. For a normal distribution, this equates to the 97.7 percent probability level. Observed data from previous Delta missions had established that the POGO vibration level at the spacecraft/launch vehicle interface was 4.0 G at approximately 20 Hz. The simultaneous steady-state acceleration at the appropriate flight time was approximately 6.0 G. The lateral excitation of the launch vehicle at this flight time is negligible. The only lateral response that must be considered therefore is that resulting from coupling through the spacecraft structure due to the longitudinal excitation. In the case of ERTS and Nimbus, the coupling is appreciable due partly to the asymmetric spacecraft configuration and partly to the fact that several lateral resonant modes of the spacecraft lie in the POGO frequency range.

As a result of a preliminary loads analysis for both the ERTS-1 and Nimbus-5 missions conducted by the spacecraft contractor, it was determined that the POGO condition was a qualification load condition for both missions. After adding the factor of 1.5 to the predicted flight loads, the values shown in Table 1 were defined as the qualification loads for both spacecraft at each of the four major interfaces. While the final predicted set of launch loads computed by the launch vehicle contractor were significantly lower than the ones shown in Table 1, it should be noted that the loads used for the test were the best available estimates at the time. The reduction in the final loads was brought about primarily because of reduced lateral axis coupled response of the spacecraft to the thrust

TABLE 1

ERTS-1/Nimbus-5 Design Qualification Loads

Interface	Type Load	Magnitude	
		Nimbus-5	ERTS-1
ACS/Truss	Shear (lb)	2,478	1,695
	Compression (lb)	7,054	5,757
	Bending (in.-lb)	31,319	20,553
Truss/ Sensory Ring	Shear (lb)	2,465	1,680
	Compression (lb)	7,831	5,578
	Bending (in.-lb)	161,308	91,762
Sensory Ring/ Adapter	Shear (lb)	7,923	4,934
	Compression (lb)	32,531	30,636
	Bending (in.-lb)	242,298	132,282
Adapter/ Launch Vehicle	Shear (lb)	7,914	4,917
	Compression (lb)	33,613	31,710
	Bending (in.-lb)	434,712	239,682

axis POGO oscillation. The control box and the truss tubes had previously been qualified to these loads, the control box during a controls subsystem acceleration test, and the truss tubes in a separate load test. It was decided that the remainder of the structure would be qualified by conducting an acceleration test of a structural development model spacecraft using the Goddard Space Flight Center's Launch Phase Simulator facility.

Because of the structural similarity of the two spacecraft, a decision was made to conduct one test to the worst-case loads, those defined for Nimbus-5, and thereby qualify the structure for use on both missions.

The spacecraft model used for the test was actually configured with ERTS peculiar secondary structure. The Nimbus peculiar hardware was qualified either by similarity or else during separate component-level acceleration tests. All structure was flight type hardware; the other subsystems were mass and center-of-gravity simulations only. There was no functional electronic hardware on board the spacecraft.

Because the test setup utilized a pure static load to simulate a combined dynamic and static load condition, it was impossible to precisely duplicate the desired load at each structural interface simultaneously. The simulation adequacy was greatly enhanced, however, by redistributing some of the spacecraft weight. One hundred pounds was removed from the sensory ring bays and 230 pounds of ballast was added to the control box baseplate. Thus the actual test article, which weighed approximately 2100 pounds, was 130 pounds heavier than the ERTS-1 flight spacecraft and approximately 400 pounds heavier than the Nimbus-5 flight spacecraft.

Table 2 defines the 100-percent test load at each of the three interfaces that were to be qualified during the acceleration test and compares it to the desired qualification load on a percentage basis. As can be seen, the bending and compression loads were reasonably well matched at all interfaces but, in order to achieve this, the shear loads were substantially above those desired. Nevertheless, all parties involved agreed that the planned test was acceptable.

TABLE 2

Planned 100-Percent Loads versus Design Qualification Loads

Interface	Type Load	Planned 100% Test Load	Test Load/Nimbus-5 Qualification Load
Truss/ Sensory Ring	Shear (lb)	3,007	1.22
	Compression (lb)	8,144	1.04
	Bending (in.-lb)	153,243	0.95
Sensory Ring/ Adapter	Shear (lb)	10,854	1.37
	Compression (lb)	31,555	0.97
	Bending (in.-lb)	244,721	1.01
Adapter/ Launch Vehicle	Shear (lb)	11,555	1.46
	Compression (lb)	33,613	1.00
	Bending (in.-lb)	512,920	1.18

RESULTS OF PRE-TEST STRUCTURAL ANALYSIS

Prior to the acceleration test, the adapter was the subject of a rigorous structural analysis for the expected test loads and was found to have sufficient margin to survive loads 50 percent higher than the qualification level test loads. The critical stresses for the applied test loads had been calculated using the GE-proprietary finite element structural computer routine, "Mechanical Analysis of Space Structures" (MASS).

Basically, this routine requires the input of the geometric and stiffness properties of the various structural elements, such as the geometric coordinates of the structural nodes defining the boundaries of the structural elements, the definition of the element as to its boundary nodes, its type (beam or plate element, either straight or curved), and the element physical and cross-sectional properties. A special input allows the fixity or release in any of 12 degrees of freedom for beam elements and 24 degrees of freedom for 4-sided plate elements. The loading conditions may be input as any of a system of concentrated loads or moments applied at the structural nodes, uniformly-distributed loads or moments applied to beam elements, or uniformly-distributed pressures to plate elements, or any combination thereof. Restraints to the structure may be applied by either inputing fixity or by the use of linear spring rates applied in any of six degrees of freedom at preselected nodes. Output consists of nodal deformations, internal loads, stresses for each load component, and an effective combined stress at the ends of each beam element or at the center of the edges of each plate element.

The original primary structure of the ERTS-1/Nimbus-5 adapter consisted of a truncated, conical shell 60 inches in diameter at the lower launch vehicle mounting ring, 24 inches high, and tapering inward at 4-1/2 degrees toward the spacecraft separation ring. The shell was formed of 0.032-inch magnesium skin, reinforced by nine full length longerons, two partial length longerons, and four circumferential rings. The 11 longerons were 0.040-inch 7075-T6 aluminum alloy rolled hat sections. The upper and lower rings were machined from 2014-T652 aluminum alloy forgings. The intermediate rings were 0.032-inch 7075-T6 aluminum alloy formed hat sections, and primarily used for skin panel stiffening and for local shear redistribution. Four structural access doors were installed in the large center band skin panels. Launch vehicle mounting was provided by eight bolt locations in the lower ring at the intersections of longerons designated numbers 2, 3, 4, 5, 6, 7, 9, and 11. It should be noted that structural symmetry about the X-axis has only been broken by longeron number 9 which has been shortened by the installation of an interchangeable structural RF plate to which antenna reradiators had been mounted. Structural continuity has been provided by full length framing longerons numbers 8 and 10, and local skin doubling adjacent to longeron 9.

The original analytical computer model defining this structure (illustrated in Figure 2) consisted of 159 nodal points, 182 beam elements, and 45 plate elements. The upper ring was modeled using 34 nodal joints, including 5 solely to provide applied-loading points corresponding to the structure above, and 34 curved beam elements. The two intermediate stiffening rings and the lower mounting ring were modeled using 29, 26, and 25 curved beam elements, respectively. The nine full length longitudinal members were each modeled using six straight beam elements. The partial length longitudinal members designated as longerons numbers 1 and 9 were modeled using four and two straight beam elements, respectively. Additionally, in the longitudinal direction, there were four separation spring brackets each modeled by five straight beam elements. The skin areas defined by the rings and longitudinal members were represented by 45 curved, trapezoidal plate elements.

The following potentially unconservative assumptions, the effects of which on the structure will be discussed in greater detail later, had been utilized in order to expedite the analysis:

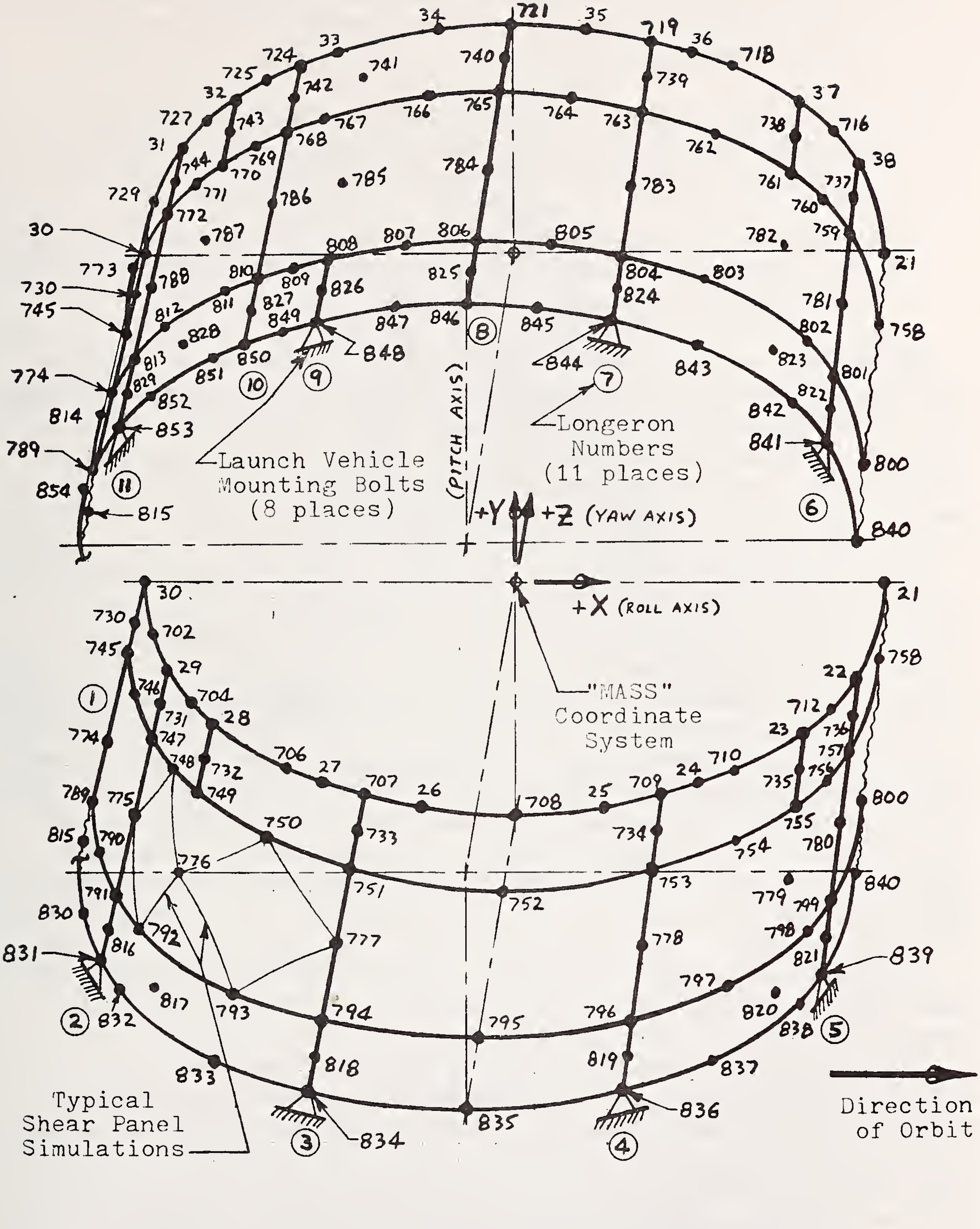


Figure 2. ERTS/Nimbus MASS Computer Model of Adapter Structure

- Longeron loads were considered to act at the longeron centroid. That is, the model did not permit any eccentricities, or the resulting bending moments, to exist.

- Panels do not buckle. The MASS computer program is limited to a purely linear analysis and, therefore, cannot account for either panel buckling or any secondary effects due to buckling.

- To account for relatively low compressive buckling allowables on large skin panels, center band adapter panels were made capable of carrying shear, but not compression.

The adapter analytical model was loaded directly at the upper ring (at 18 nodal points designated by the numbers 21 to 38) by reactions calculated to have occurred at the separation ring of the ERTS-1 spacecraft, above, when subjected to the qualification level test loads. The method of obtaining these spacecraft reactions, and their resulting applied adapter loads, will be described below. Restraint to the adapter model was provided by the input of three linear degrees of fixity at each of the eight mounting bolt locations.

The ERTS-1/Nimbus-5 spacecraft analytical computer simulations are of interest to this paper only because they provide an insight into the manner in which the applied spacecraft loads were redistributed to the adapter model. Briefly, the spacecraft structure may be considered as having two major components: the six-member space framework known as the truss assembly and the more complex sensory ring, which houses the electronic modules and earth-viewing sensors. The truss assembly is composed of six pin-ended tubes interconnecting three points on the Attitude Control Subsystem (ACS) with three points on the sensory ring. The sensory ring is basically a toroidal structure comprising 18 separator castings connected by four circumferential rings and a sheet metal web system. This structure is further stiffened by a cross-beam system paralleling the Y-axis, which in turn, supports four tape recorders and two major sensors.

The spacecraft computer simulations consist basically of two models, the first being identical for both ERTS and Nimbus, and the second being partly identical and partly unique to each spacecraft in the ERTS/Nimbus series.

Figure 3 defines, in detail, the truss assembly mathematical model, as well as, giving an overall view of the assembly of the various components of the spacecraft and adapter. The truss assembly supports the ACS, including the major portion of the mass of the solar array. Because the truss assembly was statically determinant, it was not included in the overall spacecraft computer model; however, based on a unit load type of computer analysis, the truss assembly provides concentrated loads to be applied at three locations of the sensory ring emanating from applied ACS and solar array inertia loads.

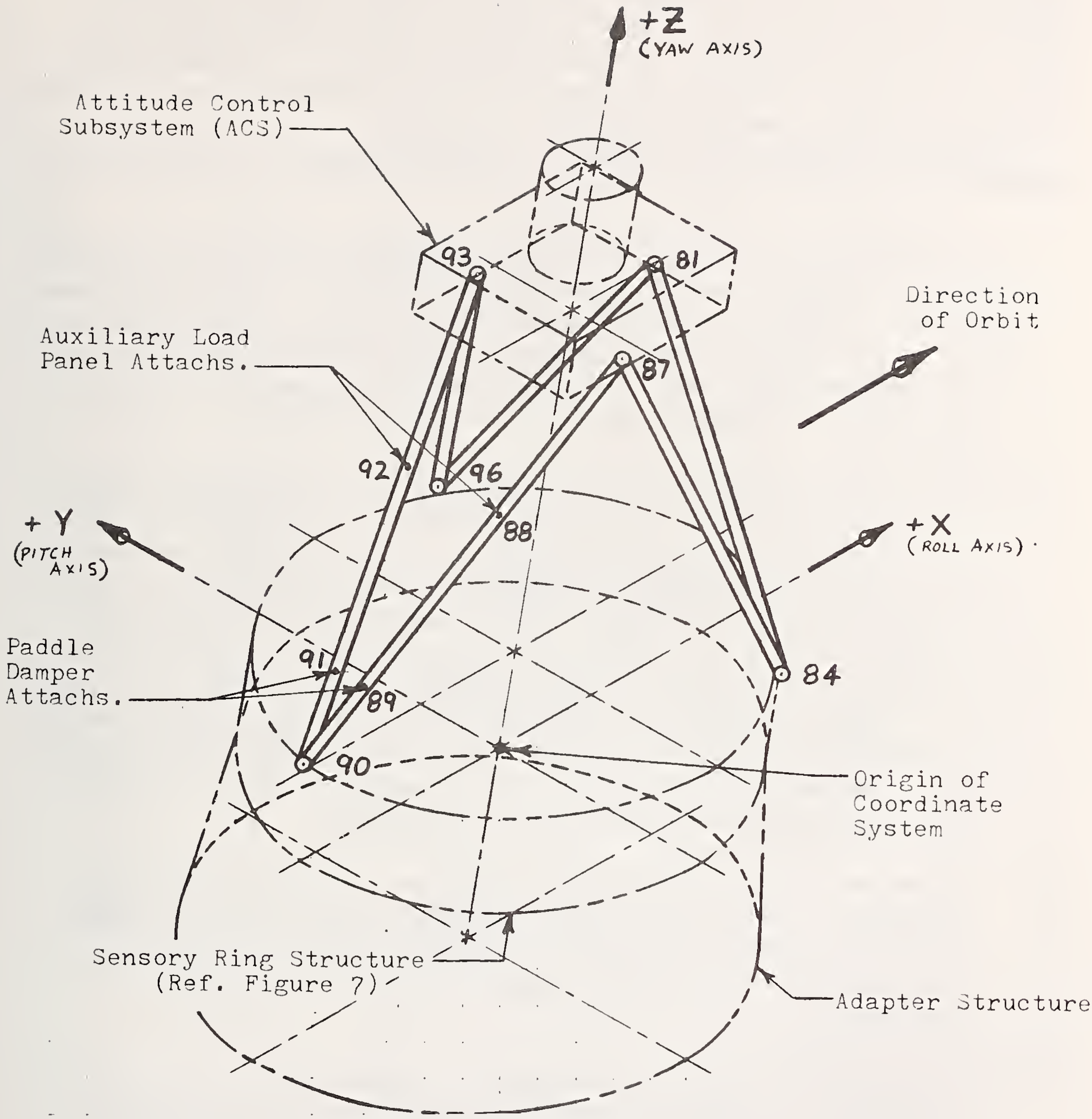


Figure 3. ERTS MASS Computer Model of Truss Assembly

The sensory ring MASS computer model consists of two parts: the torus, identical for both ERTS and Nimbus, and the crossbeam assembly, which is unique to each spacecraft. Figure 4 defines, in detail, the 137 node, torus structural model, but, in the interest of clarity, only the basic outline of the ERTS-1 100-node crossbeam structural model. An ERTS-1 configured structural development model spacecraft was the subject of both the structural analysis and the ensuing acceleration test.

In the performance of the analysis of the sensory ring, the following considerations and assumptions were made:

- The truss-induced loads described were applied eccentrically to three rigid connector elements at nodes 84, 90, and 96. These rigid connectors represented the truss support fittings which were mounted to the three separator castings designated, counting clockwise from +X-axis, as numbers 4, 10, and 16. These separators transmitted the loads, basically undiminished, to the adapter below.

- The sensory ring-induced loads resulted from accelerations acting on the various masses within the sensory ring. These included major mass items, such as the tape recorders, the externally-mounted (wide-band video tape recorders (WBVTR)) electronics package, the orbit adjust system (OAS), and the sensors, and distributed masses, such as the structure, harnessing, thermal control system, insulation, antennas, and electronic modules installed in the 18 torus bays. These local and distributed acceleration loads were then allocated to various torus and crossbeam nodal points, accounting for local eccentricities, by a rigid-body type of analysis. These values then were the applied loads utilized in the sensory ring MASS analysis.

- Reactions to the complex loading system were provided in the thrust direction (Z-axis) by the input of 18 single degree-of-freedom spring rates to the separation ring at the base of each separator casting. These values each had been calculated based on local adapter stiffness.

- Reactions in the shear direction (any combination of X-axis or Y-axis components) have been provided by the input, around the periphery of the torus separation ring, of a series of uniformly-distributed bearing loads varying in accordance with the function $1/2\ (1 + \text{Cos}\ \theta)$. These distributed loadings represent, in a sense, the bearing reactions provided by the relatively flexible adapter upper ring when clamped to the torus. Theory indicated that this distribution would produce no in-plane bending in the adapter ring, only hoop tension.

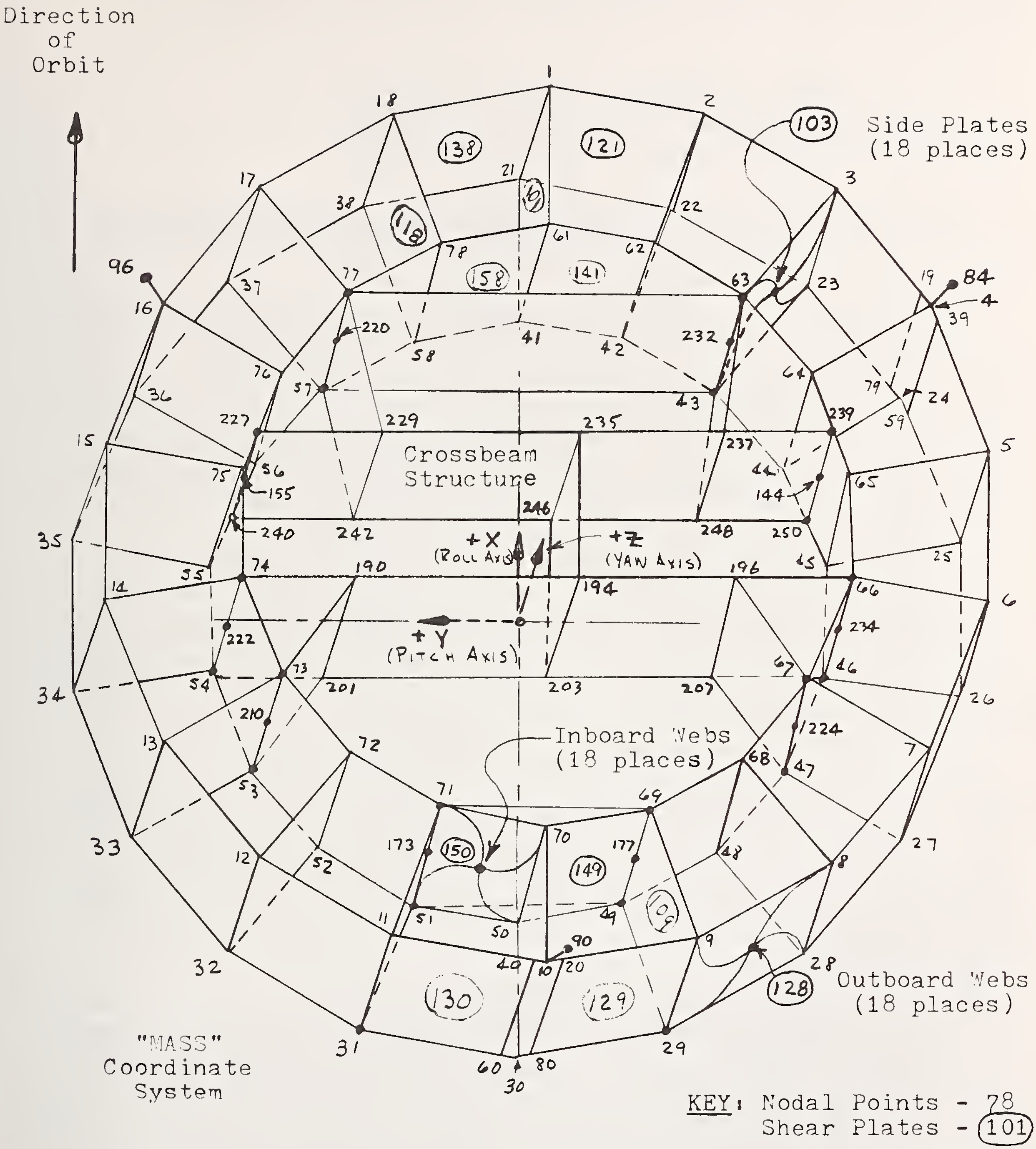


Figure 4. ERTS MASS Computer Model of Sensory Ring Structure

Based on the evaluation of the anticipated flight loads, an acceleration test to be performed with the spacecraft in three orientations was selected to qualify both the ERTS-1 and Nimbus-5 spacecraft. The test load conditions were defined as:

- -16 g thrust (Z-Axis) and +5.5 g roll (+X-axis)
- -16 g thrust (Z-Axis) and -5.5 g roll (-X-axis)
- -16 g thrust (Z-Axis) and +3.0 g pitch (+Y-axis)

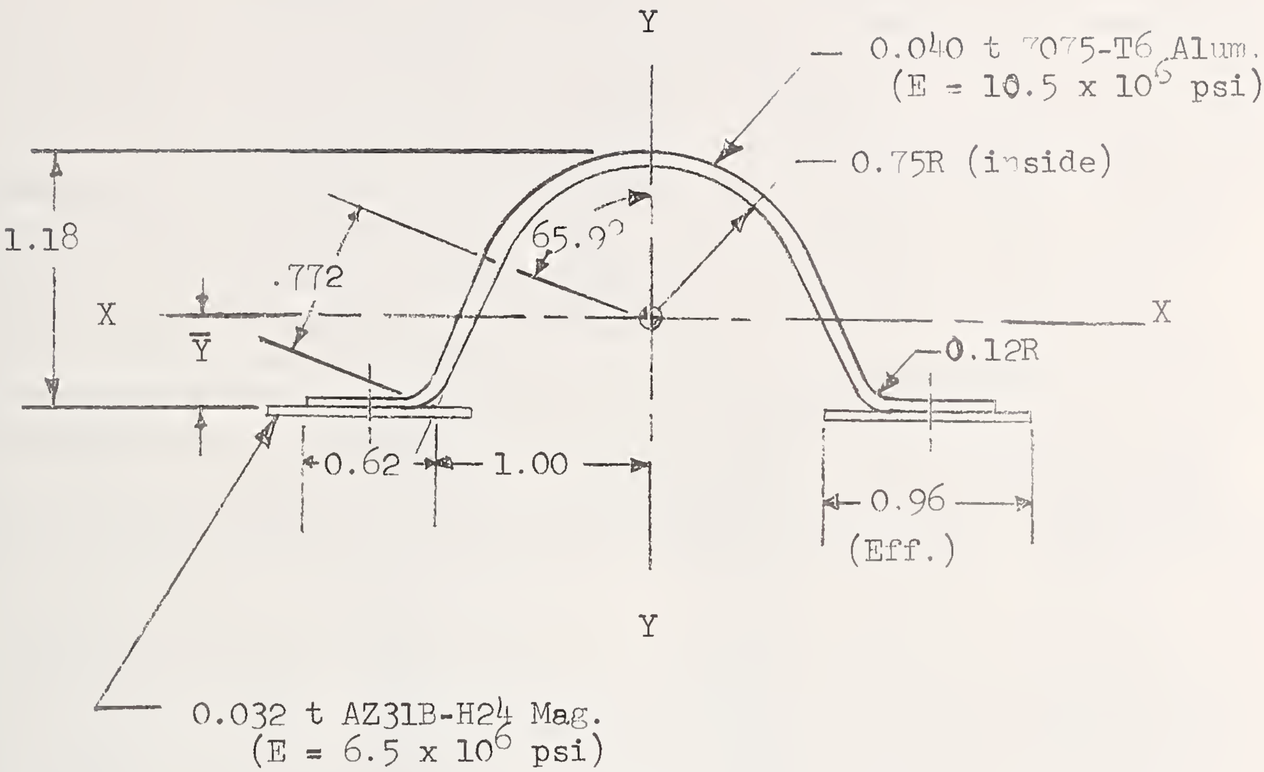
In the sign convention presented, minus thrust denotes a downward load, and plus roll and pitch denote loads applied to the forward (direction of orbit) and right directions, respectively.

Inertia loads were calculated at the center-of-gravity of the various major mass items such as the solar array, ACS, truss, orbit adjust, sensors, and distributed masses of the sensory ring for each of the three conditions. These loads were resolved into the following interface loads: Solar Array Tie-Down/Sensory Ring, ACS/Truss, Truss/Sensory Ring, Sensory Ring/Adapter, and Adapter/Launch Vehicle. From a comparison of these loads, two conditions were selected as being the more critical (-16 g thrust/+5.5 g roll and -16 g thrust/-5.5 g roll), and were subjected to further analysis.

The interface loads were distributed to the various nodal points of the sensory ring MASS model in the manner outlined, and the sensory ring structural analysis was completed. From this analysis, reactions calculated at the intersections of the 18 separators and the lower-outboard (separation) ring and the reacting bearing distributions at the lower-outboard ring were extracted and then used as the applied loads to the adapter MASS model.

It was apparent from an examination of the computer printouts, that the areas of the adapter immediately beneath the sensory ring separators numbers 4 and 16 (truss assembly support castings) were critical for the +Roll/-Thrust (+X/-Z) conditions, and the area immediately beneath separator number 10 (truss assembly and solar array tie-down support casting) was critical for the -Roll/-Thrust (-X/-Z) condition. More specifically, these areas may be identified as longerons numbers 4 and 5, 6 and 7, and 11 through 2, and their immediately adjacent skin panels.

The most critical stresses and their resulting margins of safety from these areas have been calculated and summarized. In Figure 5 the critical section of a typical longeron is defined, and the crown and attachment flanges are immediately identifiable. Table 3 presents a summary of the longeron stresses and margins. Because the computer analysis neglected thrust loads in adjacent skin panels, these results have been adjusted to reflect an effective amount of skin acting. The most critical margin for the actually-applied test condition (+Roll/-Thrust) resulted at the attachment flange of longeron number 4, the point of



Section Properties:

$A = .218 \text{ in.}^2$

$\bar{y} = .431 \text{ in.}$

$I_{xx} = .0435 \text{ in.}^4$

Crippling Allowables:

at crown- $F_{cr} = 58{,}500$ psi

at flange- $F_{cr} = 44{,}700$ psi

Figure 5. Properties of Adapter Longerons

TABLE 3

ERTS-1/Nimbus-5 Critical Margins of Safety
From Pretest Analysis

Longeron Number	Critical Condition	Stress in Crown (lb/in.^{2})	Stress in Flange (lb/in.^{2})	Margin of Safety*
2	-X/-Z	38,820	27,810	+0.01
4†	+X/-Z	22,740	27,370	+0.09
5†	+X/-Z	25,680	24,520	+0.22
6	+X/-Z	24,560	23,300	+0.28
7	+X/-Z	28,950	23,960	+0.24
11	-X/-Z	32,000	26,300	+0.13

Skin Panel** Location	Critical Condition	Shear Stress (lb/in.^{2})	Margin of Safety††
1 & 2	-X/-Z	4012	+0.24
2 & 3	-X/-Z	4453	+0.12
3 & 4	-X/-Z	3643	+0.37
4 & 5†	+X/-Z	4920	+0.02
6 & 7	+X/-Z	4757	+0.05
7 & 8	+X/-Z	4602	+0.72
10 & 11	-X/-Z	2483	+1.01
11 & 1	-X/-Z	3376	+0.48

*Based on crippling allowables of 58,500 psi and 44,700 psi at crown and flange, respectively. Ultimate factor of safety equals 1.50.

**Critical panels are in wide center band between longerons designated.

†Area of acceleration test failure.

††Based on initial panel buckling allowable of approximately 5000 psi at limit load, except panel numbers 7 and 8 (7910 psi) which have flanged doubler.

actual failure. It should be noted that the most critical margin results occurred at longeron number 2 for the reverse axis condition (-Roll/-Thrust). This is also indicative of a potential failure at this point if this axis had been run first. Also shown in Table 3 are the more critical center band skin panels and their resulting margins of safety. In this case the most critical panel lies between longerons 4 and 5 at the point of test failure.

ACCELERATION TEST DESCRIPTION

The acceleration test was performed using the Goddard Launch Phase Simulator (LPS). This unique facility was designed for conducting combined environmental tests and has capabilities for simultaneously applying steady-state acceleration, vibration, acoustic noise, and atmospheric venting profiles in an automatically controlled real-time sequence. At the time of the qualification test of the ERTS structural development model in mid-1971, however, the LPS vibration system was not operational when mounted on the centrifuge arm, thus forcing the test to be a pure static acceleration test.

An overall view of the centrifuge is given in Figure 6. The nominal radius to the base of the test item is approximately 65 feet, and capability exists for accelerating a 5000-pound test item to 30 G at which time the arm is rotating at approximately 40 rpm. The total rotating weight is 465,000 pounds. During testing, the test item is enclosed in the test chamber, a 15-foot diameter, 20-foot-long shell which also houses the acoustic noise and atmospheric venting systems. In preparation for test, the test item is mated to the LPS end cap while in its normal orientation. A specially designed 6 degree-of-freedom LPS loading vehicle then picks up the end cap, pitches it so that the thrust axis of the test item is essentially horizontal, and places the end cap against the test chamber where it is bolted into place for test. Figure 7 shows the ERTS structural development model on the end cap about to be mated to the test chamber by the loading vehicle.

The actual acceleration magnitudes of the test did not correspond directly to any particular flight condition but rather they were calculated so as to produce the proper qualification loads. To induce these loads in the ERTS structural development model spacecraft, the required accelerations were 16.0 G along the thrust axis and 5.5 G along the lateral axis. To accomplish this multi-axis acceleration, the spacecraft thrust axis was rotated to a position 15.6 degrees below the horizontal (see Figure 7) by mounting the spacecraft on a variable angle tilt fixture specifically designed for LPS use. At this angle when applying a centrifugal acceleration of 16.9 G and including the effects of gravity, the loads of Table 2 would result.

Monitoring instrumentation during the test was not extensive since it had been viewed as a go/no go test. Although the applied acceleration is regulated by controlling the centrifuge spin rate, there were four accelerometers on the spacecraft. Two accelerometers were at the top of the solar paddle latch line and two were at the spacecraft center-of-gravity on a bracket 4 inches above the top of the sensory ring. At each location, one accelerometer measured the applied thrust acceleration and the other measured the lateral acceleration. More importantly, there were 19 strain gages on the structure. Of these, six were located on the three separator castings of the sensory ring immediately below the hard points at which the truss tubes are supported. The remaining 13 gages were mounted on the adapter structure; three legged, rosette gages

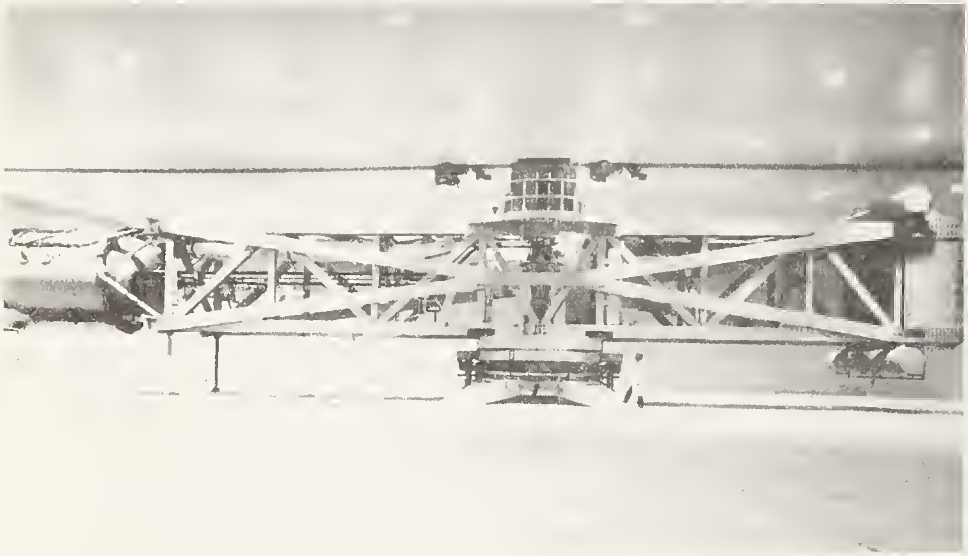


Figure 6. Overall View of Launch Phase Simulator

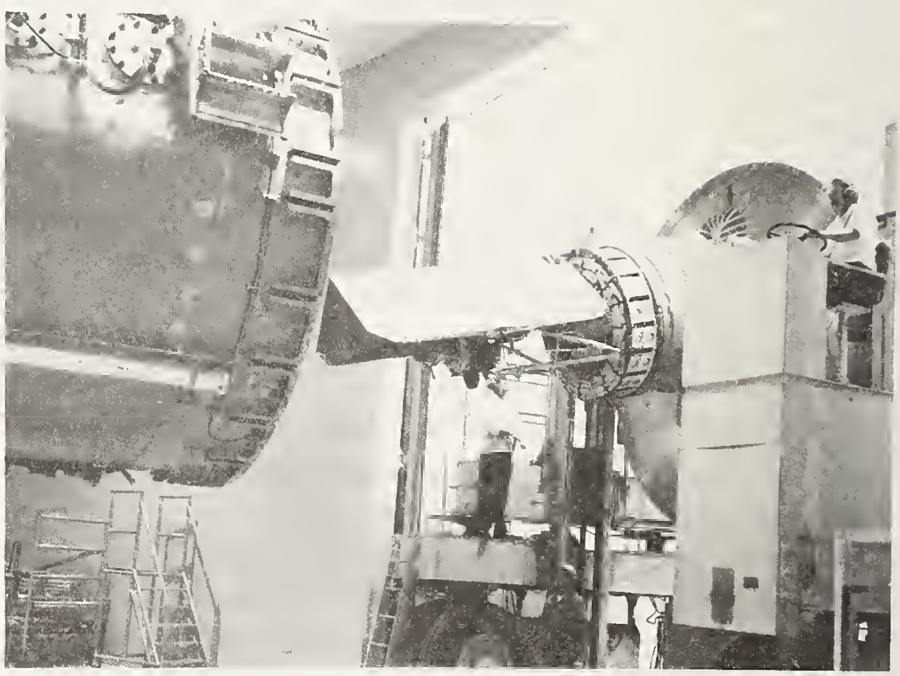


Figure 7. ERTS Structural Development Model Entering LPS Test Chamber

were mounted on each of three selected skin panels, and single gages were mounted on each of four longerons. These gages were located so as to measure strains in members whose design had been modified as a direct result of the loading condition to which the structure was being qualified in this acceleration test. All 23 instrumentation channels were monitored in real time on an oscillograph and simultaneously recorded on magnetic tape for subsequent additional data analysis.

During the test, the acceleration level was held constant at 25, 50, and 75 percent of full level in order that the measured strains could be compared with those predicted for that loading condition. It was noticed from these intermediate readings that the loads in the skin panels were higher than predicted and those in the longerons were lower than predicted. The real-time decision, reached while the centrifuge was running, was that the differences between the measured and predicted strains was not significant enough to warrant stopping the test, and the authorization was given to proceed to the 100 percent test level. At 94 percent of full load, the spacecraft collapsed, instantly falling to the bottom of and being forced against the end of the test chamber, leaving only the lower ring of the adapter still bolted to the test fixture. The centrifuge was shut down at once, and the failure analysis commenced immediately. A view from within the test chamber of the untouched wreckage is shown in Figure 8.

FAILURE INVESTIGATION

The failure had occurred at 94 percent of the qualification level, although the adapter stress analysis had indicated no expected failure of the adapter below 150 percent of qualification levels. The fact that the adapter had failed at less than two-thirds of the predicted allowable loads led to the initiation of an extensive failure analysis that consisted of the wreckage analysis, a material analysis, a review of the adapter history, revision and review of the stress analysis, and a static test program, each of which are discussed in following sections of this paper.

Wreckage Analysis

Initial inspection of the wreckage had indicated that the failure most likely initiated in the +X, +Y quadrant of the adapter. Video tape pictures also indicated that a hesitation occurred just after the start of collapse. These pictures, however, were not adequate enough to determine the direction of initial failure or whether this direction changed after the noted hesitation.

In an attempt to reconstruct the failure sequence, a detailed study of the wreckage was performed. This consisted of physically arranging the pieces of the wreckage into their original positions and studying the damage and markings on the major spacecraft components and on the test fixture. Due to the large degree of fragmentation of the compression side of the failed adapter, the attempt

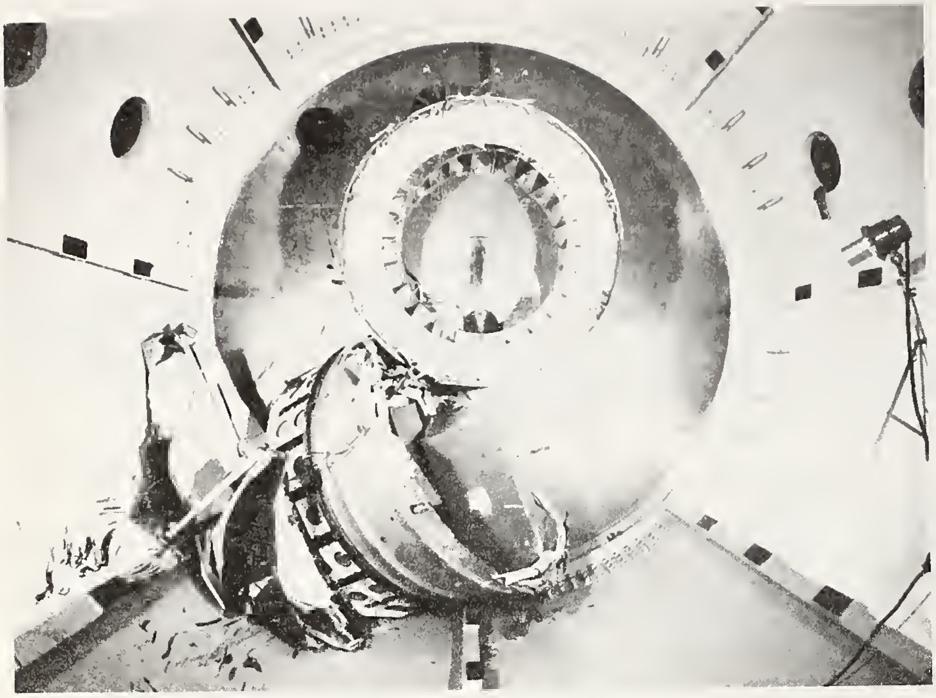


Figure 8. ERTS Structural Development Model in LPS Test Chamber after Failure

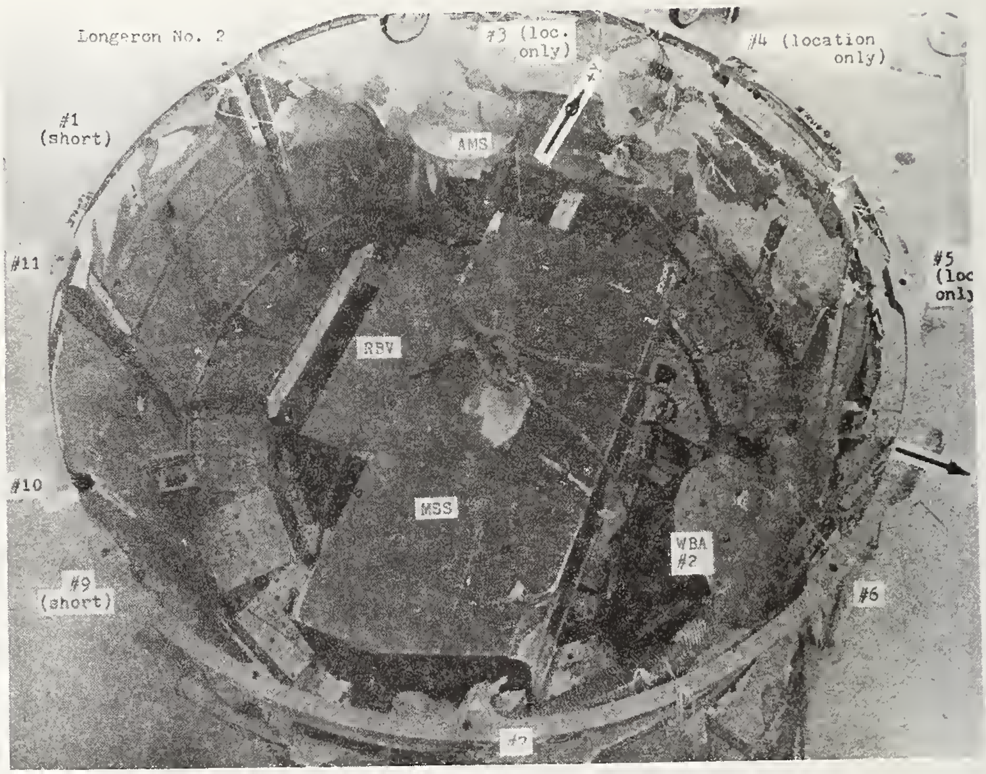


Figure 9. Failed Adapter with Damaged Components in Place

to reassemble it was not as successful as was hoped. However, some conclusions could be drawn from examination of the failed adapter, reassembled as shown in Figure 9.

- The spacecraft collapsed in compression generally in the +X direction inclined toward longeron 5.

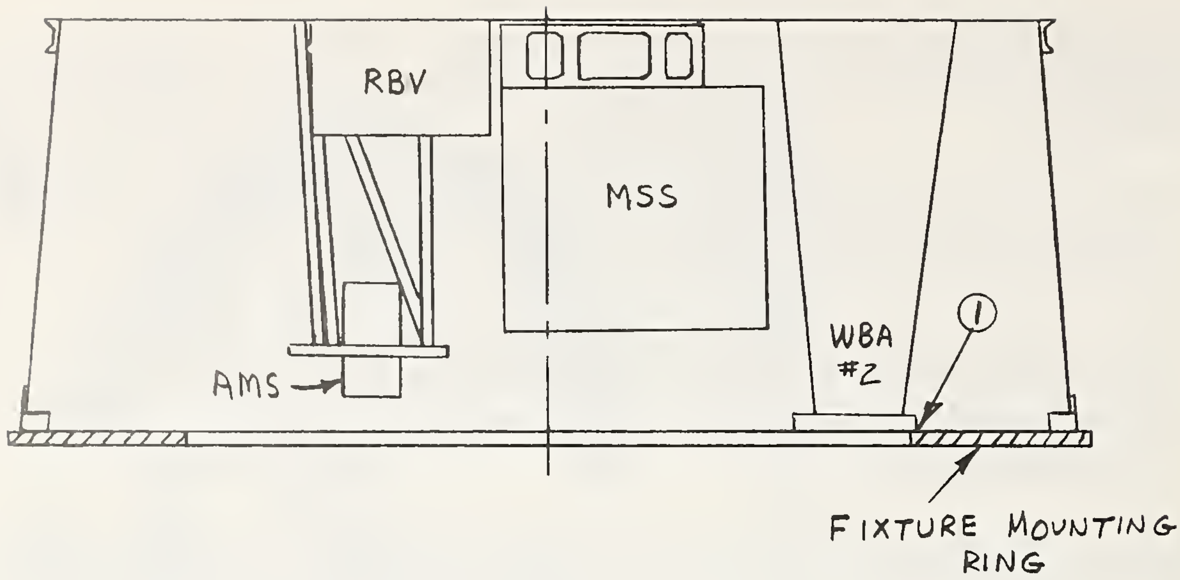
- The -X side of the adapter experienced primarily a tension failure of the skin at the lower adapter flange, indicating that this was a secondary failure (this skin was theoretically still in compression prior to failure).

A study of the damaged hardware was made to identify the probable sequence of events that occurred subsequent to the initial failure and also to test the credibility of some theories of initial failure events.

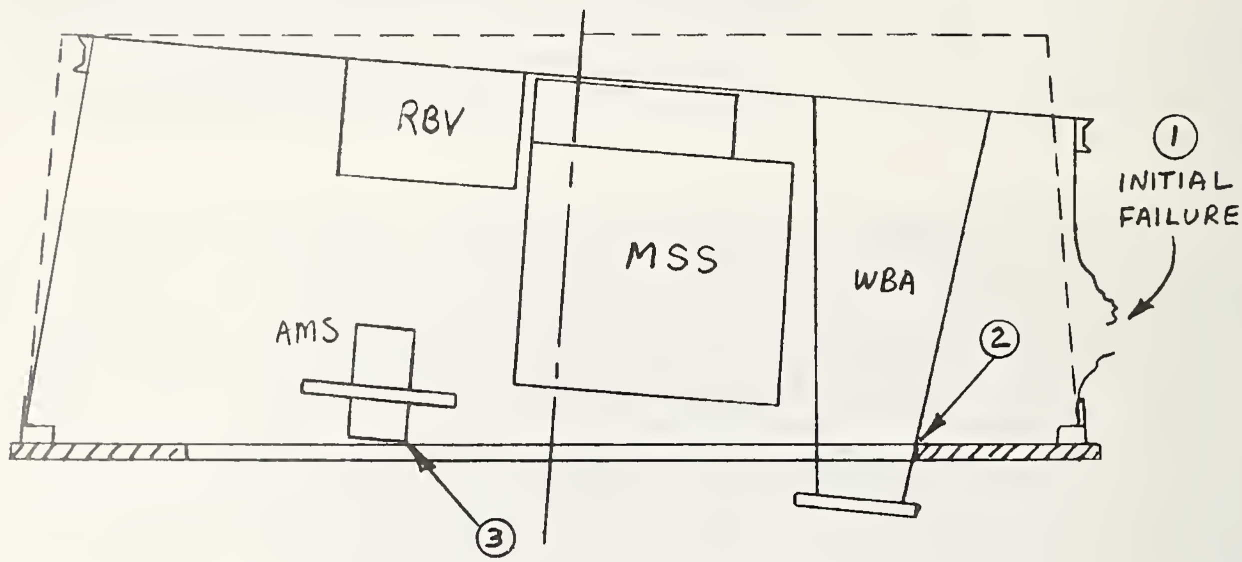
Through study of the damage to the adapter and the components mounted within the adapter, and inspection of the markings on the test fixture mounting ring, a sequence of failure has been developed which has appeared to be valid. The locations of the significant components relative to the adapter and to the fixture mounting ring are shown in Figures 9 and 10a. The derived failure sequence is as follows:

1. An initial compression failure of the adapter occurs in the +X, +Y quadrant (Figure 10b). The vehicle starts to rotate about an axis which is approximately parallel to the Y-axis and which intersects the -X axis near the periphery of the adapter. The wideband antenna (WBA) No. 2 impacts the fixture mounting ring immediately; an analysis of the geometry had shown that the nominal clearance with the ring is essentially zero. As the vehicle starts to rotate, the lower end of the Attitude Measurement Sensor (AMS) contacts the fixture ring leaving a mark. The WBA is pushed inboard until the antenna bends and slips below the fixture ring.

2. The vehicle continues to rotate until the Multispectral Scanner (MSS) comes in contact with the fixture mounting ring, as shown in Figure 10c. This is confirmed by the damage to the +X, -Y corner of the MSS as shown in Figure 9. The corner of the MSS acts as a new fulcrum for the spacecraft and, coupled with the increased moment due to rotation of the vehicle, causes a tension failure of the adapter skin on the -X side. The WBA mount is torn loose at its base, but remains trapped between the fixture and the MSS. The AMS tears loose from its truss mount and falls to the bottom of the fixture cavity.



a. ① WBA in contact with ring (prior to failure)

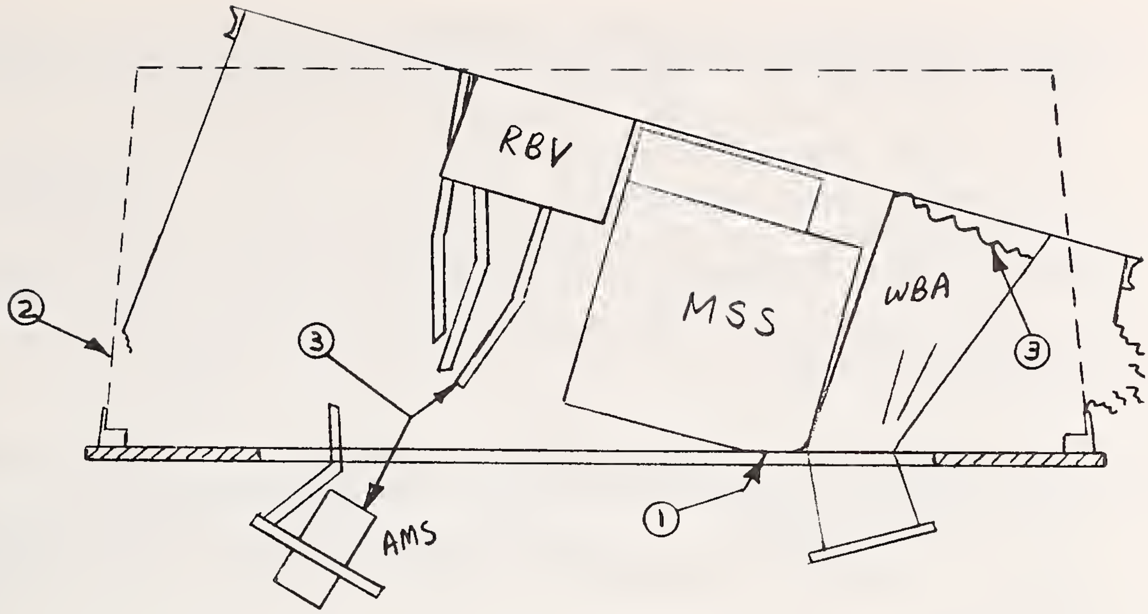


b. ① Adapter Fails

② WBA bends inboard and snaps through fixture ring. Support contacts ring.

③ AMS contacts fixture ring

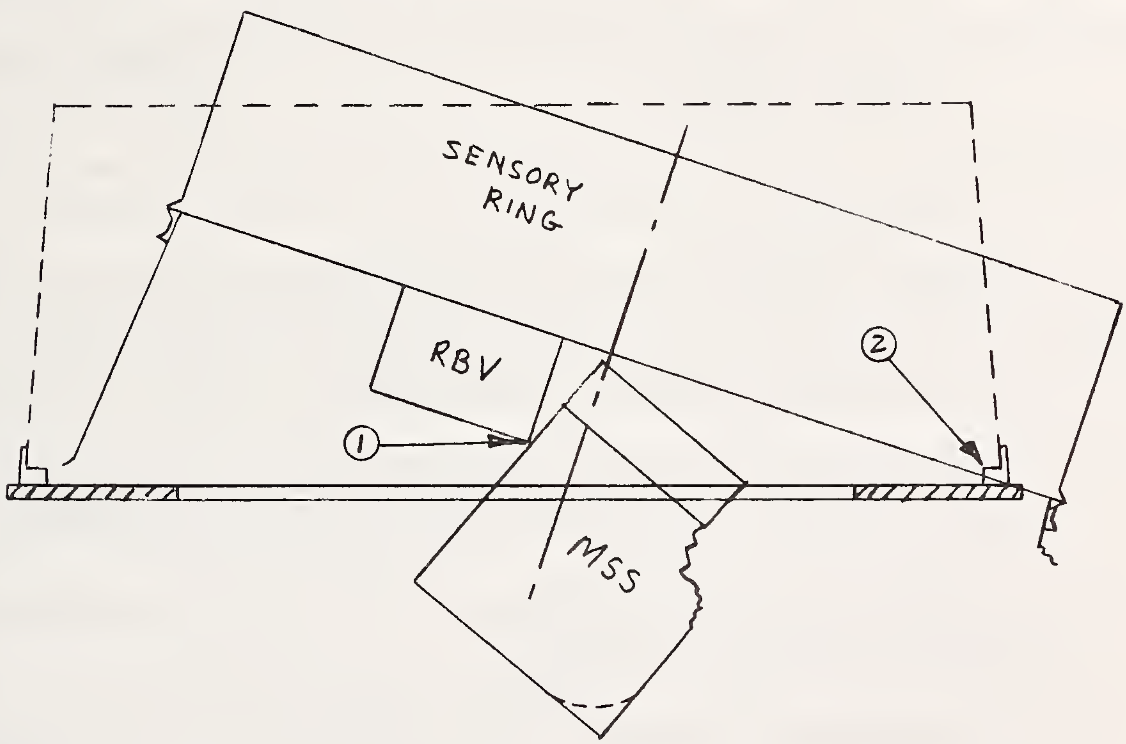
Figure 10. Failure Sequence



c. ① Spacecraft rotates about MSS contact

② Adapter skin tears on tension side

③ WBA and AMS are torn loose



d. ① MSS rolls back into RBV

② Sensory ring contacts fixture damaging castings and crossbeam

Figure 10 (continued). Failure Sequence

3. The increasing tilt of the spacecraft causes it to slide off the corner of the MSS as shown in Figure 10d. This results in a second MSS contact with the fixture ring. This time, the upper +X, +Y corner hits the ring, breaking off a large piece of the wooden MSS simulation, and exposing a steel mounting bar which also gashes the ring. The MSS is torn loose from its mounting when the steel mounting bar hits the fixture. It then rotates back into the Return Beam Vidicon (RBV) simulation as verified by matching the RBV with the gouges on the -X side of the MSS.

4. With the MSS out of the way, the sensory ring comes in contact with the fixture ring (Figure 10d), damaging the crossbeam and separator castings of bays 2 and 3. The spacecraft then rolls over the edge of fixture and comes to rest against the cover and wall of the centrifuge chamber.

Four theories were advanced to explain the initial failure of the adapter. These are discussed in the following paragraphs as is the conclusion that was drawn, based on the evidence noted, as to the validity of the theory presented.

<u>Theory No. 1</u>: The MSS mounting bolts failed and the MSS impacted adapter, initiating failure.

<u>Evidence</u>

a. Video tape showed that the MSS stayed with spacecraft. If the MSS had come loose initially, it would have fallen into the fixture cavity.

b. No zinc chromate marks on the MSS indicate no adapter contact.

c. Mounting bolts have shown a shear-type failure. Bearing damage to holes in the sensory ring have indicated a CCW in-plane torque.

d. Damage on the lower -Y, +X corner of the MSS indicated that the spacecraft rolled about this point.

e. Stress analysis had indicated a high margin of safety on the mounting.

<u>Conclusion</u>: The MSS did <u>not</u> initiate adapter failure.

Theory No. 2: The AMS broke loose, impacting the adapter and initiating failure.

Evidence

a. No zinc chromate marks on the AMS indicate no adapter contact.

b. The buckled condition of the mounting legs and markings on the fixture ring indicated the AMS was in place when it contacted the fixture.

c. Stress analysis had indicated a high margin of safety on the mounting.

Conclusion: The AMS did not initiate adapter failure.

Theory No. 3: Interference of the WBA with the fixture ring caused it to break loose, impacting the adapter and initiating failure.

Evidence

a. No interference was noted by the test crew during spacecraft assembly to the fixture.

b. The head of WBA bolt was coined into fixture ring. Coin tests showed it required 5,000 to 10,000 lb to duplicate the indentation.

c. Gap checks at the attachment pads, with 2,100 lb of spacecraft weight on them, indicated less than 0.003-inch gap at any point. This would have been greater if interference was sufficient to cause coining loads.

d. The WBA mount would have acted as an additional adapter longeron, supporting the spacecraft.

Conclusion: The WBA interference did not cause initial failure.

Theory No. 4: Insufficient adapter strength.

Evidence

a. The sequence of failure as previously described supports this theory.

b. Failure occurred on the most highly loaded side (+X).

c. Longerons in failure area evidenced very low margins of safety in pretest structural analysis.

d. There is no evidence of other failure before adapter collapse that can be substantiated.

e. When included, the effects of the eccentricity of loading in the adapter longeron analysis, as will be shown, indicated the existence of failing stresses.

Conclusion: The adapter was understrength for the acceleration test loads and initial failure was a compression collapse of the adapter primary structure.

Adapter Materials Analysis

As part of the failure analysis, it was considered necessary to verify the mechanical properties of the materials used in the failed adapter. To this end, test samples were obtained from the skin and longerons of the failed adapter and tested for tensile strength and elongation. The results have indicated that the tested materials met or exceeded the material specification minimum requirements. The surface pitting discovered in the magnesium skin was probably present when the adapter was manufactured and may be typical of the material. No indications of significant corrosion were found. From this investigation, it was concluded that the adapter material properties were satisfactory and did not contribute to the failure.

Adapter History

The adapter that failed in the acceleration test was Serial No. 001 and had seen many uses and modifications since its initial fabrication. This led to the possibility that the adapter failure could be attributed to a weakness caused by its prior experiences. To evaluate this possibility, a review of the adapter history was performed. Usage on the adapter prior to the acceleration test failure was as follows:

1. The adapter had experienced a total of about 2 hours of vibration testing, or approximately 1 million cycles of loading at various levels. An analysis of the theoretical fatigue damage was not performed because it would have been difficult and costly to review all the test data and convert it accurately to the required stress/cycles data. Additionally, inspection of the failed adapter disclosed no evidence of fatigue. The conclusion was that fatigue was probably not the primary cause of the failure.

2. Open holes (0.20 in. diameter) existed in the crowns of longerons 4, 5, and 6 due to the removal of Nimbus-B crossbeam simulation. Longeron 4 had four holes near the bottom, longeron 5 had six holes near the bottom, and longeron 6 had seven holes near the center. Because the acceleration test data and the subsequent analysis review showed the longeron crown to be a low stress region, the conclusion was reached that the open holes had no significant influence on the adapter failure.

3. X-rays and subsequent sectioning of rivets revealed that rivet holes had been damaged when the reinforcing angles had been added to the tops of longerons 1, 2, 4, 7, and 11 and to the bottom of longerons 2 and 11. The damage had been caused by the drilling out of the rivets and the resulting voids are randomly oriented about the rivet circumference. Because the voids were randomly distributed and they did not exist at the bottom of longerons 4, 5, 6, and 7, where the adapter failure apparently started, it was concluded that this did not contribute significantly to the adapter failure.

A check of the Product Assurance records for the adapter showed that all changes, discrepancies, and material review actions had been satisfactorily completed prior to the acceleration test. It was concluded that the adapter used for the acceleration test was adequately representative of the flight adapter and did not fail due to the effects of previous use.

<u>Stress Analysis Review and Revision</u>

Prior to the acceleration test, the adapter had been analyzed for the expected test loads and was found to have sufficient margin to survive loads at least 50 percent higher than the test loads. The adapter failure, and the absence of any other obvious deficiency in the adapter, brought the validity of the analysis into question. The impact of analytical model assumptions on the adapter loads and the acceleration test failure was investigated.

The analytical model assumes all of the longeron centroids to be in the plane of the skin. The geometric centroid is actually located 0.46 inches inboard of the outer skin surface.

The MASS computer model was reconfigured to allow the actual eccentricities of longerons 4 and 5 (area of failure) to become effective as shown in Figure 11, and the -Z, +X acceleration test condition was rerun. The following longeron stresses were calculated for the critical longeron 5, midway between the lower frame (hat section) and the lower mounting ring:

- Maximum skin stress = 31,590 psi, compression.

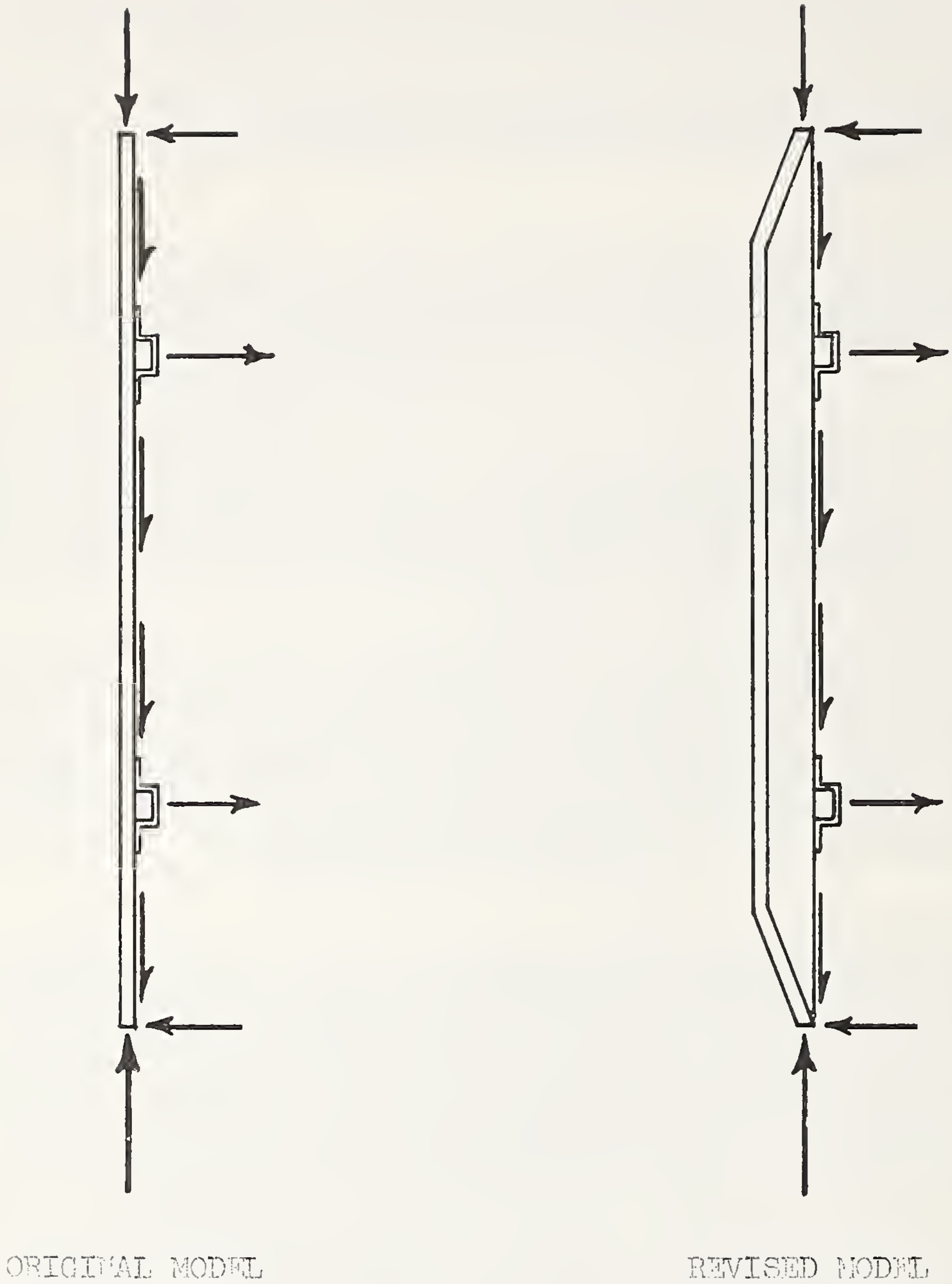


Figure 11. Comparison of Original and Revised Longeron Computer Models

- Maximum longeron stress = 49,650 psi, compression.

- Minimum longeron stress = 1,300 psi, tension.

The minimum stress occurred in the crown of the hat-section and the maximum stress occurred in the longeron at the skin attachment flange. It became apparent at this point that the low longeron crown strains measured during the acceleration test were probably due to this longeron eccentricity. This in turn indicated the presence of the high compression stresses on the skin side of the longeron as shown above.

Previous analysis of the longeron had indicated a longeron crippling allowable of approximately 38,200 psi for a bending load which induces compression on the skin side of the longeron. This indicated that longeron load eccentricity was a strong contributor to the adapter failure, and its absence in the analysis was significantly unconservative.

In an effort to evaluate the potential effects on the adapter primary structure of one or more panels reaching a buckled or otherwise ineffective state, the following shotgun approach was taken:

- The panel or panels considered in a shear buckled state were removed in their entirety from the analytical model. Note that in the model these panels were already ineffective in compression.

- The critical -Z, +X acceleration test condition was rerun on MASS.

- The effects on the adjacent skin panels and framing structural members were extracted from the MASS printouts and evaluated.

This is a rather gross approach, and represents an overly conservative analysis, in that the skin shear-carrying capability prior to its buckling is ignored.

The results of this analytical study indicated an insufficient increase in shear loading in adjacent skin panels to precipitate a catastrophic failure such as was witnessed. For example, an ultraconservative approach in which four panels were considered completely ineffective (neglecting the skin shear-carrying capability prior to initial buckle) resulted in the following increase in the critical panel shear stress:

- Initial panel shear = 3250 psi (all panels effective)

- Final panel shear = 3910 psi (four panels ineffective)

The shear stress in the critical panel increased only 20 percent due to the elimination of four adjacent panels. Considering the effects of initial shear-carrying capability of these eliminated panels would bring the panel shear stress to a point between 3250 psi and 3910 psi. Because initial shear buckling for the panel had been estimated to be 5000 psi, considering the stiffening angles already installed, it was felt that panel buckling in itself was not the cause of failure. Therefore, further analysis was not pursued since the static test would verify or modify this conclusion.

All of the center panels (between hat section frames) were considered in the MASS program model to be ineffective in compression, carrying only shear. This assumption would be conservative for the longerons, because all of the axial loads would be carried by them. However, this assumption is unconservative for the panels. Shear buckling allowables are greatly reduced in the presence of compression, resulting in higher panel buckling loads being applied to the longerons, reducing their load-carrying capability.

It was felt that compression buckling did occur in many of the adapter panels during the acceleration test; however, it was not considered practical to try to analytically predict this effect and it was evaluated in the static test.

From the failure analysis activities that were performed prior to the static test of the adapter, several preliminary conclusions were derived. The subsequent static test program would provide the necessary verification or correction of these conclusions. The preliminary conclusions were:

- The adapter used in the acceleration test was adequately representative of the design. Its strength had not been significantly reduced due to its history, its configuration, or its material properties.

- Failure initiated in the adapter; all other failures were secondary.

- Adapter failure was caused by the initial collapse of either longeron 4 or 5, the adjacent skin panels, or a combination of these.

- Initial longeron failure was probably, at least partially, the result of the eccentricity between the load and the longeron centroid, causing high combined bending and axial compressive stresses in the skin and longeron legs.

- The panels adjacent to the longerons probably had buckled in combined shear and compression. The buckling also significantly contributed to bending in the longerons.

ADAPTER STATIC TEST

During the failure analysis activities previously described, it became obvious that the true failure mode could only be determined and verified by performing a static test of the adapter. The basic objectives of this adapter static test were threefold:

- To identify the failure mode and the initial failure point as they occurred during the acceleration test on the NASA centrifuge.
- To provide sufficient data to update and improve the analytical model of the adapter.
- To determine the load paths and distributions within the adapter.

To accomplish these objectives, it was necessary to have a test article that was a near duplicate of the one that failed in the acceleration test. This required the assembly of major spacecraft elements that would have the same stiffness and geometric characteristics as the ballasted ERTS SDM. In addition, the adapter used would have to be configured exactly the same as the failed adapter. To achieve these goals, the following hardware (Figure 12) was assembled as the test article:

- ACS - In this case, exact simulation was not imperative and a dummy ACS was used that consisted of an aluminum loading plate to which the upper ends of the truss members were attached, similar to flight configuration.
- Truss - The truss members were flight quality and identical to those used on the ERTS SDM.
- Sensory Ring - To obtain a proper stiffness representation, the Nimbus Mechanical Test Model (MTM) sensory ring was used, including the existing Nimbus leaded wood dummy bay modules.
- Separation Clamp - A flight band was used, installed in accordance with normal procedures.
- Adapter - It was most important to simulate this item exactly. Therefore, a new adapter was obtained and was reworked, resulting in a configuration which duplicated in all essential details the adapter that failed in the acceleration test.

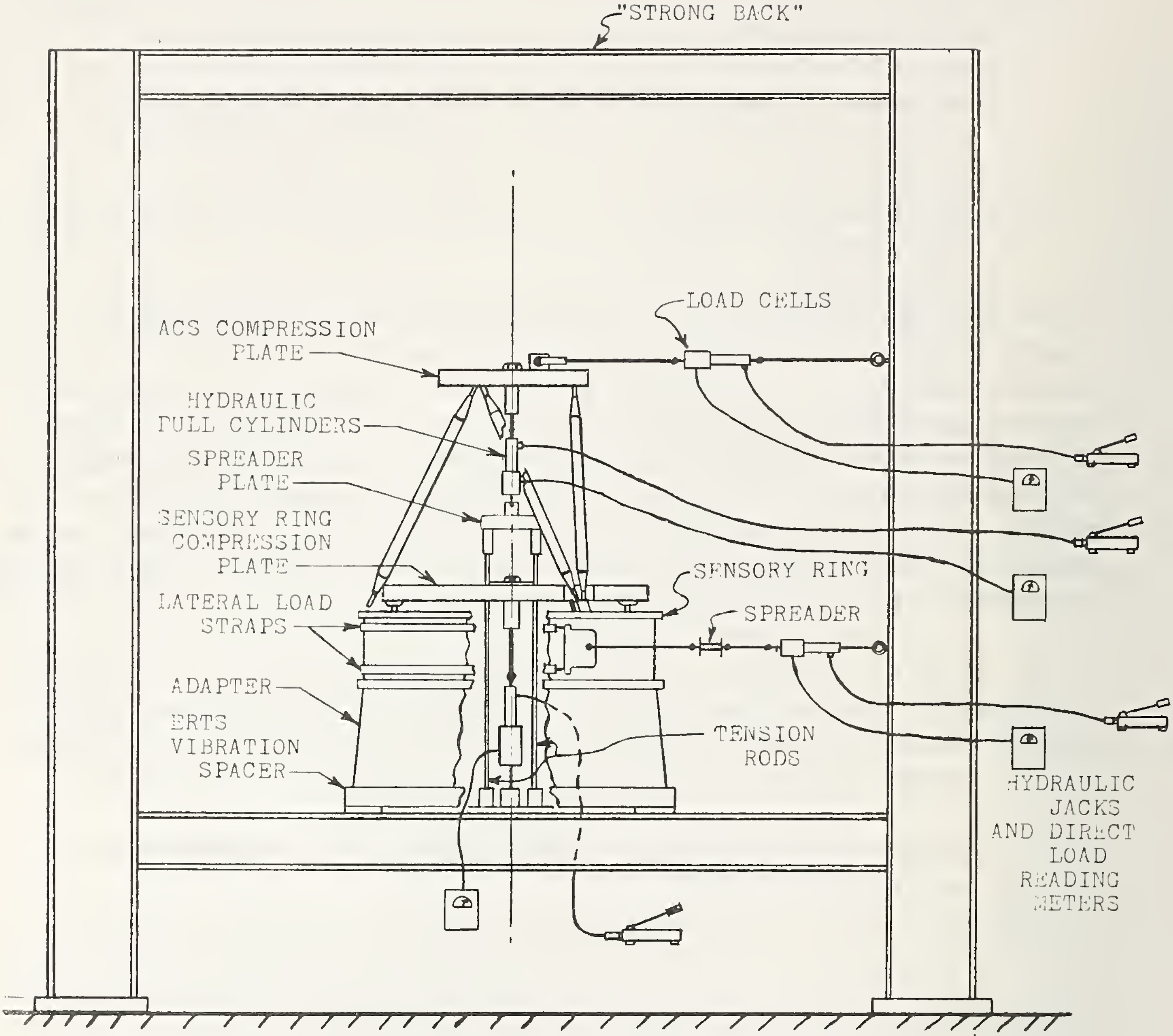


Figure 12. Schematic of Static Test Set-up

The test plan evolved in response to the specified objectives and the approaches selected to achieve them. The following is a brief discussion of some of the decisions and rationale that went into the construction of this test plan.

Adapter Instrumentation

Figure 13 indicates the adapter instrumentation provided. Gages 1 through 40, 120, and 121 were rectangular strain rosettes, and gages 71 through 90 were axial bridges. All others were axial strain gages.

This extensive instrumentation of the adapter evolved as a result of several requirements. The primary requirement was that to determine the internal adapter loads it is necessary to know the externally applied loads and reactions. That is, the load distributions at the sensory ring/adapter and adapter/launch vehicle interfaces must be determined. No direct measurement of these loads during the combined load test was considered feasible, so indirect methods were necessary.

As the means of establishing the sensory ring/adapter interface loads, all of the upper skin panels were instrumented with strain rosettes. Similarly, all of the lower skin panels were instrumented with strain rosettes as a means for determining the shear reactions at the eight booster attachment points. The axial (thrusts) loads at the eight booster attachment points were determined by calibrating the longerons. This was done during a separate test, the axial element test, in which an axial load was applied through the sensory ring and the reactions were measured with load washers while the corresponding longeron strains were being recorded. In later tests, the longeron strains could be used to establish the thrust component of the pad reactions. This established the requirement for instrumenting all of the longerons.

It was desired that correlation with acceleration test data be obtained from the static test. To achieve this, similar instrumentation was provided at the same locations as that provided on the ERTS SDM for the acceleration test. This consisted of (Figure 13) rosettes 20, 21, and 28, and longeron gages 61, 63, 66, and 70. It was suspected that the instrumented panels buckled during the acceleration test, prior to the failure. Since panel gages are only applied to one side of the skin, a buckle in the region of the gage could produce unintelligible readings. To assess the effects of skin bending on rosette readings, and to help in the interpretation of the rosette data, gages 20 and 21 were backed up leg for leg on the inside of the skin with gages 120 and 121, respectively. These locations also coincide with gages from the acceleration test. The interpretation of data obtained from strain rosettes under complex loading is not precise or well understood. A shear element test was included in the program. The intent of this test was to apply known loads to an area of the adapter where the resulting panel shear loads could be fairly well predicted and then to compare these predictions with the resulting

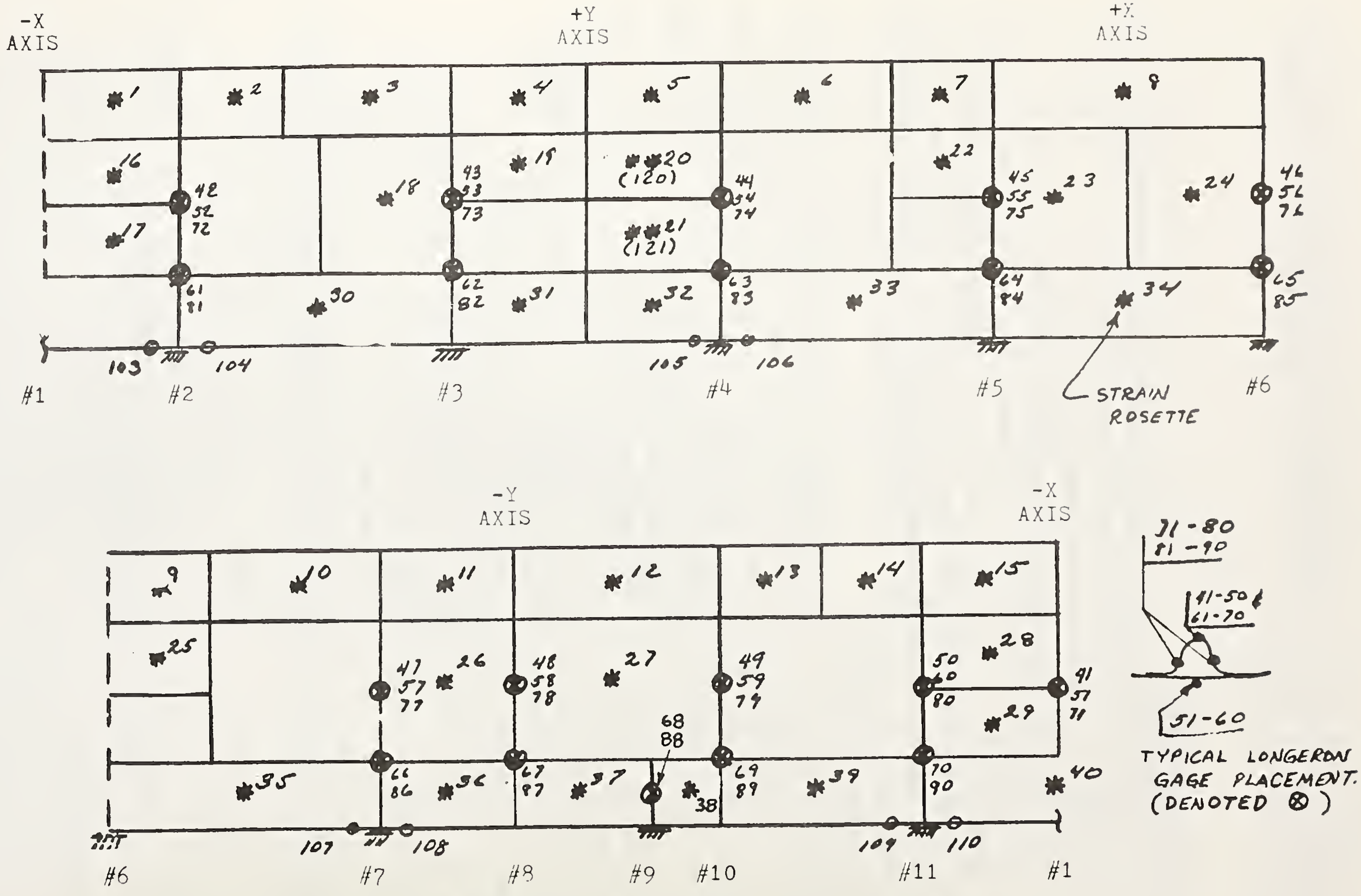


Figure 13. Adapter Strain Gage Locations during Static Test

strain rosette data. Results of this test would then be helpful in the interpretation of the strain rosette data to be obtained during subsequent testing.

Loads and Load Schedules

Since it was planned that the test adapter would be returned to the fabricator for rebuilding for future use as a prototype adapter, no significant damage was to be permitted during the static tests. This placed restrictions on both the load levels and the sequence of loading.

The static test loads were to be representative of the loads which the vehicle experienced during the acceleration test. A set of thrust and lateral loads and a loading geometry were selected which would develop loads at the spacecraft interfaces that were approximately equal to those at 100 percent of the acceleration test load levels. However, because failure was not to be risked, the test loads were to be restricted to approximately two-thirds of these levels. Table 4 defines the static test interface loads and their relationship to the acceleration test loads.

The test loading schedules were selected to minimize the risk of an unexpected failure by building up to maximum values in an orderly fashion. This was accomplished by loading the sensory ring only in thrust, then loading the sensory ring and ACS in thrust only, and finally combining lateral loads and thrust loads up to the maximum test values. These final loads were applied in the following sequence: sensory thrust, ACS thrust, sensory lateral, and ACS lateral.

The -Z, +X loading condition in which the failure occurred was chosen for the first static test in order to establish the failure mode as soon as possible so that the adapter redesign could proceed. This was to be followed by the -Z, -X condition, which was possibly the most critical, and then by the -Z, +Y condition, considered to be less critical than the other two.

The test loads shown in Table 4 account for the weight of the test article.

Analytical Predictions

Prior to the actual performance of the adapter static test, a complete set of analytical stress predictions was generated, with which to correlate the measured strain data. At the time of preparation of this analysis, the loading sequence and maximum levels had not yet been established. Therefore the sequence of the pretest analytical effort was as follows:

1. Calculation of required maximum applied test loads to duplicate the acceleration test.

2. Preparation of unit stress analyses for each of the load components as required by 1, above.

TABLE 4

Static Test Load Combinations Analyzed

Interface	Load	-Z, +X			
		Accel. Test Loads	Static Test Loads		
			100%	Test	Static/Accel
ACS/TRUSS	$F_{lat.}$	2,899	3,007	2,000	0.69
	F_z	-7,830	-8,127	-6,156	0.79
	$M_{lat.}$	14,757	12,100	8,010	0.54
TRUSS/SENSORY	$F_{lat.}$	3,007	3,007	2,000	0.67
	F_z	-8,144	-8,144	-6,173	0.76
	$M_{lat.}$	154,846	154,840	103,000	0.67
SENSORY/ADAPTER	$F_{lat.}$	10,854	10,854	7,500	0.69
	F_z	-31,555	-31,507	-22,443	0.71
	$M_{lat.}$	264,274	264,240	178,300	0.70
ADAPTER/BASE	$F_{lat.}$	11,555	10,854	7,500	0.65
	F_z	-33,613	-31,555	-22,491	0.67
	$M_{lat.}$	532,474	524,740	358,200	0.67
<u>APPLIED LOADS</u>	Z_{acs}		-7,971	-6,000	
(Add weight of spacecraft for total Z loads)	$Z_{s/r}$		-22,093	-15,000	
	LAT_{acs}		3,007	2,000	
	$LAT_{s/r}$		7,847	5,500	
LAT. LOAD APPLICATION					
(In. above sensory ring)	z_{acs}		51.5	51.5	
(in. above sep. plane)	$z_{s/r}$		8.96	8.96	

TABLE 5

Applied Static Test Loads

Interface	Load Component	(ALL VALUES IN KILOPOUNDS)							
		Thrust Only			Combined Thrust & Shear				
		1	2	3	1	2	3	4	5
Thrust - Sensory	F1Z	-7.5	-7.5	-15.0	-7.5	-7.5	-15.0	-15.0	-15.0
Thrust - ACS	F2Z	0	-6.0	-6.0	-3.0	-3.0	-3.0	-6.0	-6.0
Lateral - Sensory	F1X	0	0	0	1.0	3.3	3.3	3.3	5.5
Lateral - ACS	F2X	0	0	0	0	1.2	1.2	1.2	2.0

3. Combination of the above unit stress analyses to produce predictions representing the actual test loading sequence and maximum levels.

Table 4 shows the resulting interface loads from the + roll acceleration test (+5.5g roll lateral and -16.0g thrust), which had been designed to attain 1.5 times the flight loads. Also shown are the four applied static test loads, the points of application, and the resulting interface loads which nearly duplicate the acceleration test loads.

The adapter primary structure was analyzed in depth using MASS. Four load cases were analyzed, each representing one of the (100 percent) test loads of Table 4. The definition of these load cases is as follows (adjustment has been allowed for weight of spacecraft components):

- Applied unit sensory ring thrust load
- Applied unit ACS thrust load
- Applied unit sensory ring lateral roll load (applied at a point 8.96 inches above the separation plane)
- Applied unit ACS lateral roll load (applied at a point 4.03 inches above the truss interface, 51.5 inches above the sensory ring)

Thrust loads were applied to the adapter separation ring at 18 points, representing the bases of each of the sensory ring separator castings. Magnitudes of the 18 thrust loads were adjusted to represent the total thrust loads and bending moments for the above four load cases, as redistributed by the sensory ring stiffness characteristics.

Sensory ring shear loads were transmitted to the adapter separation ring as a distributed bearing load varying in accordance with the function 1/2 $(1 + \cos\theta)$. As noted previously, this distribution will produce no bending in the adapter ring, only hoop tension, consistent with the actual condition of a stiff sensory section separation ring bearing on a relatively flexible adapter ring.

At the time the maximum test loads and testing sequence were finalized, the unit stress analysis cases were combined in such a manner as to duplicate these test loads. Predicted values were calculated for combined compression and bending stresses in the longerons (total stress on the crown, centroid, and skin elements of each longeron section) and for shear stresses in the skin. These predictions were prepared for every instrumented longeron and skin panel section as defined in Figure 13. Stresses were predicted for each of the loading combinations and sequences outlined in Table 5.

The analytical model utilized in the prestatic test analysis was modified from that used in the preacceleration test analysis as follows:

- Longerons 2, 4, 5, 6, 7, and 11 (the areas of maximum longeron loads for any of the three loading axes) were reconfigured to simulate the eccentricity of their centroids from the point of application of the loads at the upper and lower rings. This change was expected to eliminate the major portion of the unconservatism existing in the previous analysis.

- The section properties of longerons 2 and 11, in full, and longerons 1, 4, and 7, locally, were modified to simulate the aluminum angles added to these members prior to the acceleration test.

- The center bay (between hat section frames) skin panels were unchanged; that is, they were capable of carrying shear, but not compression, in order to account for relatively low compression buckling allowables at these locations. This assumption is somewhat conservative for the longerons, but unconservative for the skin panels.

Data Reduction

The reduction and analysis of data recorded during the earlier test phases (specifically, the shear element test and the axial element test) led to the development of two methods for evaluating the data output. A brief description of the development of these methods is outlined in the following paragraphs.

First, it was necessary to establish a method of reducing rosette strain data to values usable in comparison with standard analytical procedures, that is, to stresses oriented with the edges of a rectangular plate. The strains were recorded in three gages oriented with one gage vertical, one diagonal at 45°, and one horizontal. This gage arrangement is described in the literature as a rectangular rosette. The interpretation of strain rosettes in a complex stress field (a combination of biaxial and shearing stresses) is not well understood, in that the orientation of the principal stresses with respect to the rosette or the edges of the plate is not precise. A study of the literature and a comparison of measured results with a known loading distribution, the shear element test, led to the development of a logical and suitable method of evaluation later used in reducing data from the thrust load test and the combined thrust and lateral load test.

In contrast to the strain rosette data reduction, the reduction and analysis of the longeron data is straightforward. The strains recorded by three channels of data were converted to stresses. The channels represented were at the crown, the centroid (axial bridge), and the skin intermediate to the attachment flanges, respectively. From these data were extracted the maximum stress in the longeron attachment flanges, the axial load, and the longeron bending moment.

Data Analysis and Interpretation

Test data was recorded in the following forms:

- Strain gages and shear rosettes to measure internal stresses
- Load cells to measure input loads
- Load washers to measure base reaction forces
- Dial gages to measure selected deformations

Most significant of the strain data recorded were the strain-versus-load curves plotted for longerons 4 and 5 (area of failure) during the thrust and combined shear and thrust testing. In these plots, strain readings were spotted during the ongoing tests, against preplotted analytically predicted curves for five channels of strain information per longeron. Typical curves have been reproduced for longeron 4 as Figure 14 for the combined sensory and ACS thrust static test and as Figure 15 for the combined shear and thrust static test. The most obvious fact, upon reviewing these curves, is that the test values are, in all cases, significantly below the predicted values. This leads to the obvious conclusion that the skin panels are carrying considerably more axial load (pure thrust in Figure 14 and combined thrust and bending-induced axial load in Figure 15) than analytically predicted. This conclusion is verified by examination of the data reduction for skin panel axial stresses, indicating significant stress levels in the panels adjacent to longerons 4 and 5. In particular, for the combined test, data indicate very high compressive stresses (probably greater than buckling) for rosettes 5, 22, 33, 120, and 121. High tensile stresses in rosette 20 combined with the high compression evident in rosette 120 (on the reverse side of the panel containing rosette 20) indicates high local plate bending which is a very good indication of a buckled state.

Three other observations on the longeron strain-load curves are subject to interpretation as follows:

1. The test-derived longeron axial stress curves, particularly for the combined test, draw closer to the predicted curves at the higher applied-load ranges.
   Conclusion: Panel buckling reduces skin efficiency, allowing longeron to carry a larger proportion of the load.

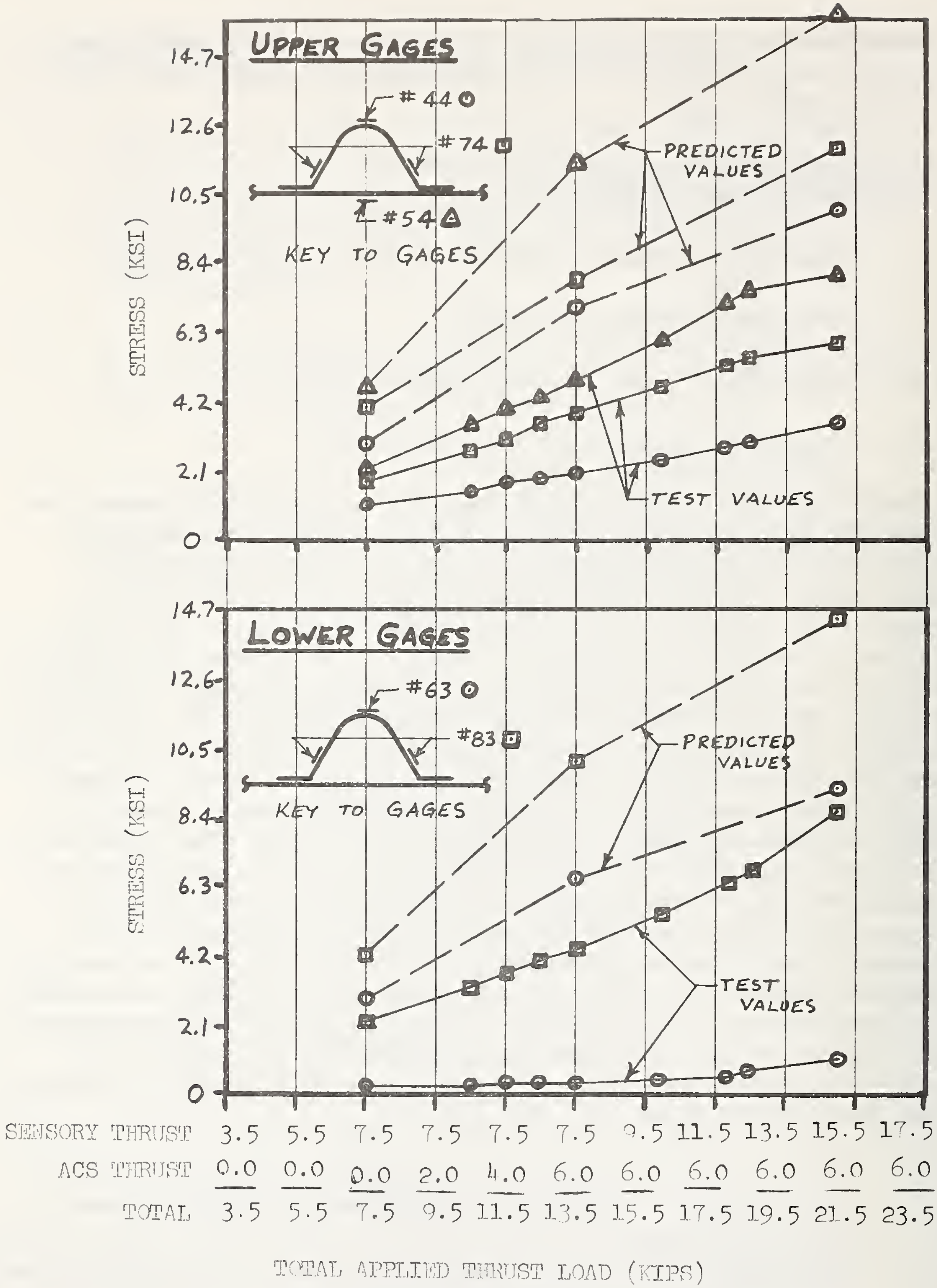


Figure 14. Longeron No. 4 Stresses during Combined Sensory and ACS Thrust Static Test

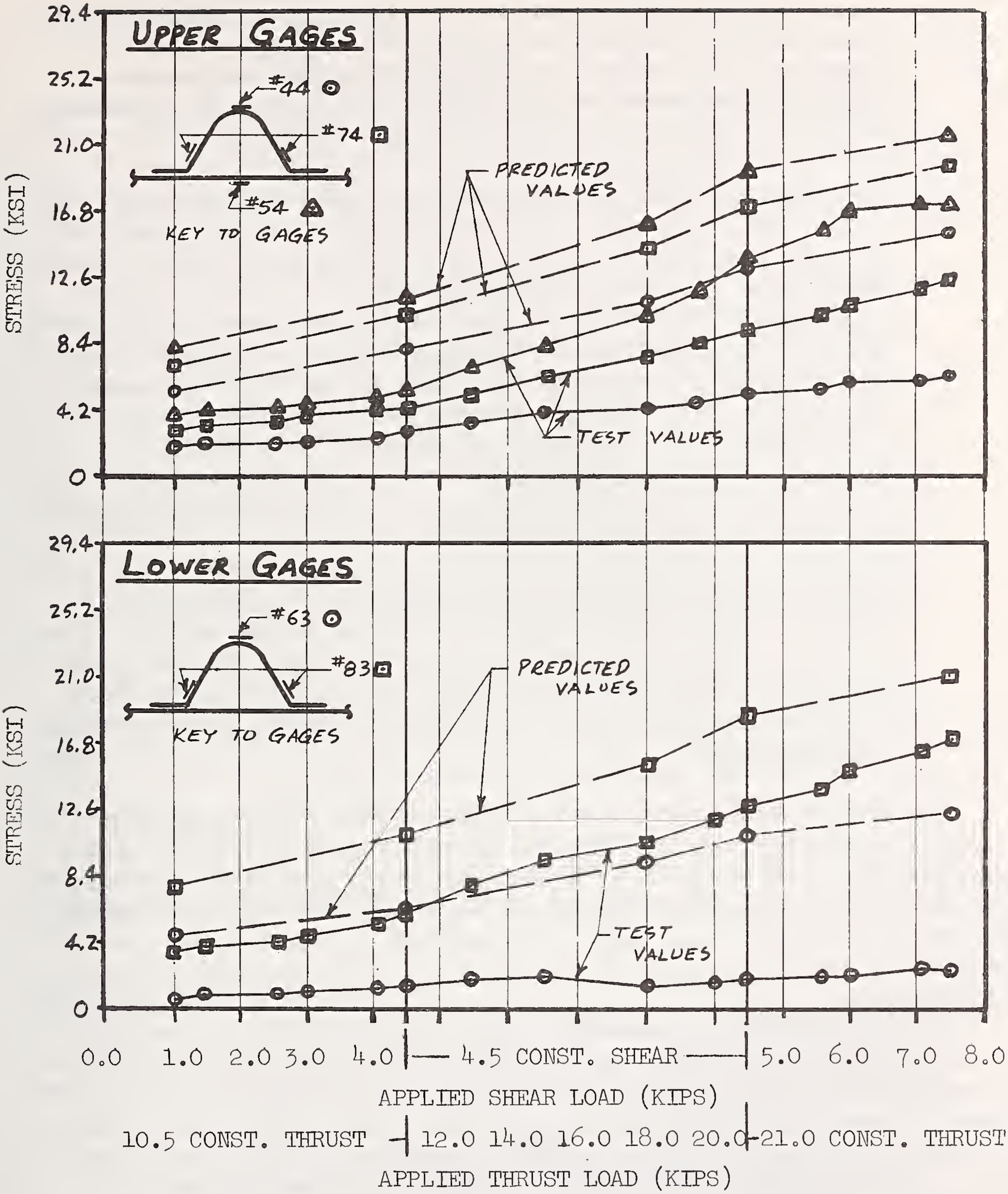


Figure 15. Longeron No. 4 Stresses during Combined Shear and Thrust Static Test

2. The test-derived longeron skin stress curves "tail-off" and, in some cases, decrease, in the higher applied load ranges.
   Conclusion: The stress in the longeron flange does not decrease, rather, the skin between the two rows of rivets, reaching its buckling limit, has a considerably reduced efficiency in compression.

3. The test-derived longeron crown stress curves are at very low values. The spread between the crown stress and axial stress curves are generally greater than predicted. Both of these phenomena are most significant at the lower level of gages.
   Conclusion: The crown gage location utilized during the acceleration test was erroneously chosen, because this is the point of minimum longeron stress. A better choice would have been the axial bridge (the 70 and 80 series of gages as delineated by Figures 14 and 15). As indicated above, the skin gage produces erratic results as its efficiency varies with applied load. The optimum for maximum stress would be on the attachment flange, but local disturbances due to riveting makes this impractical.
   Conclusion: The longeron bending moment is even greater than predicted using an eccentric representation of the longeron in the computer model. This is especially true near the base of the longeron, additionally pointing out the necessity of a "bathtub-type" fitting at this prime load-carrying point. A fitting of this type would remove the eccentricity in the load path and make the longeron stress nearly pure axial.

Not included are curves plotted for longerons 6 and 7, symmetrically opposite to longerons 5 and 4, respectively, which theoretically should show the same loading characteristics. It was apparent during testing, in performing tasks 5 and 6, that strain readings for longerons 6 and 7 were substantially less than for 5 and 4, respectively. This phenomenon, which has become known as the load bias, is probably attributable to the stiffness characteristics of the adapter.

The interpretations presented here tend to support the theory of failure as concluded in this report—forced crippling at either longeron 4 or 5. This is evidenced by the high bending moment at the base of the longeron which, when combined with axial stress, results in locally high compression stresses in the longeron flange. High compression stresses in the lower skin panels, when resulting in a compression buckle, could then have been the trigger that completed the failure.

Results of Static Test Program

The static test program was stopped after completing only the first loading condition (-Z, +X). The two remaining test conditions planned were deleted for two reasons: (1) the test data obtained were considered to be sufficient to achieve the basic objectives of the test program; and (2) the adapter design being tested would never be flown and the redesigned adapter would receive a complete qualification static and dynamic test. The static tests of the adapter redesign would also permit obtaining any additional data that might be necessary.

The static test results verified the failure mode that was previously proposed. That is, the adapter failure occurred when longeron 4 collapsed in forced crippling due to the combined effects of eccentric loading and panel buckling. Although no failure of the static test adapter was permitted, severe and permanent buckling of the panels adjacent to the lower ends of longerons 4, 5, 6, and 7 was evident. These buckles carried the longeron flanges with them as shown in Figure 16. As the applied loads increase, the buckle becomes deeper, until it forces its way through the radius of the flange (Point A, Figure 16). When this happens, load redistribution occurs which causes the total collapse of the longeron in compression and bending, buckling inward.

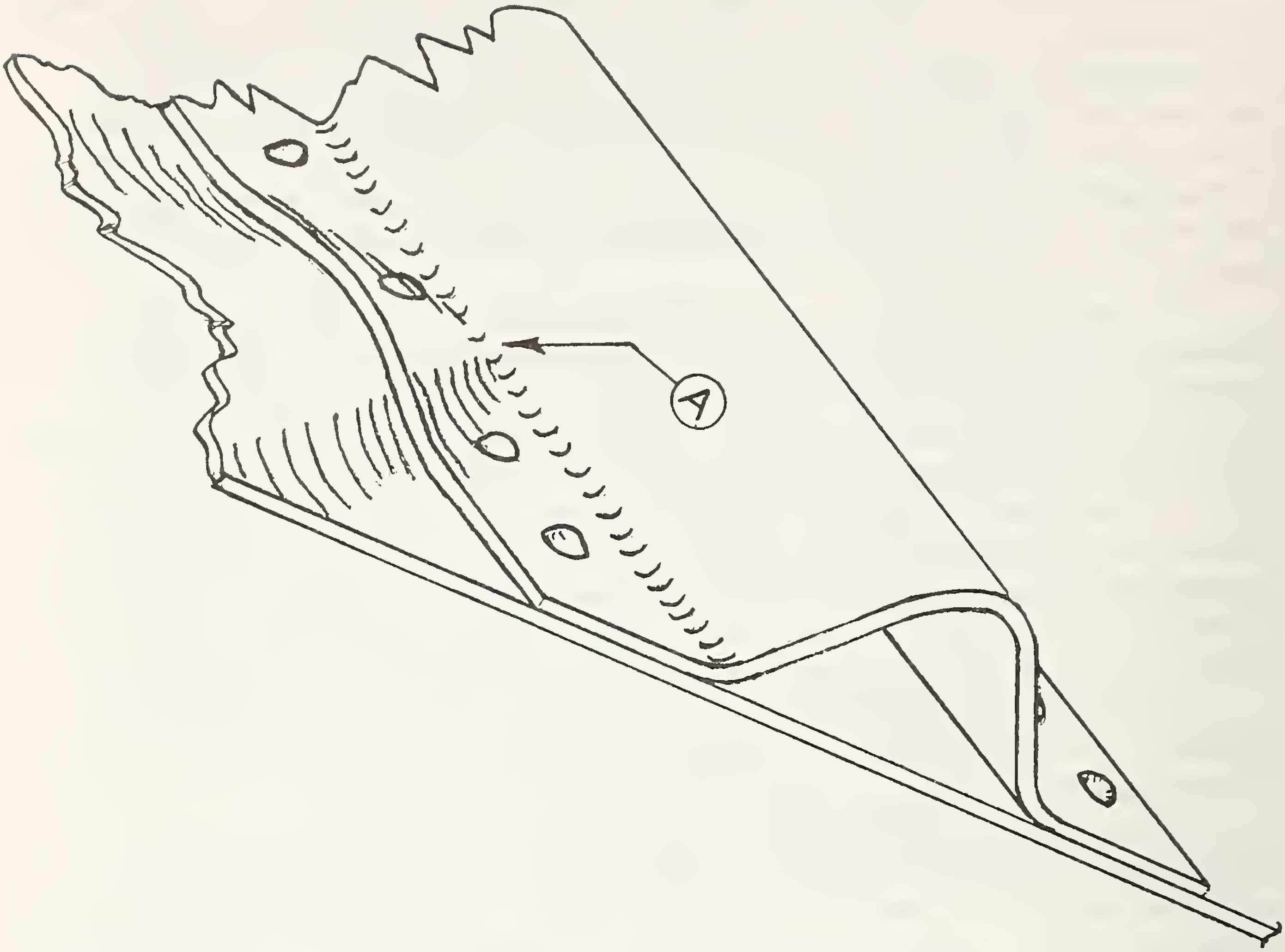
From the static test results, it appears that longeron 4 collapsed first, rapidly followed by longeron 5, which would be near its critical loading when 4 collapsed and dumped its load into the adjacent longerons.

The static test results also indicated the existence of an unexpected bias or anomaly in the load distribution in the adapter and the reactions at the adapter base. This bias resulted in much higher reactions at longerons 4 and 5 than at 7 and 6, respectively, although theoretically, they should have been close to equal. Additional tests were run to try to identify the source of this anomaly; it appeared to be a property of the adapter. The identification of this property was not pursued, but it is thought to be associated with the stiffness characteristics of the -X, -Y quadrant as related to the rest of the adapter.

ADAPTER REDESIGN EFFORT

The nature of the causes of the adapter failure set the requirements for the redesign. After evaluations of the wreckage analysis and the revised stress analysis, the following requirements for redesign were established:

- The first requirement was to reduce or eliminate the eccentric loading of the longerons. This was accomplished by adding "bathtub" fittings between the longerons and the lower ring at the eight mounting points, as shown in Figure 17.



1. Lower panel adjacent to longeron buckles, carrying longeron flange with it.

2. As load increases, buckle becomes deeper and progresses through radius at (A).

3. This side of longeron dumps its load into rest of member. Other leg cripples similarly.

4. Longeron fails in compression and bending, buckling inward.

Figure 16. Longeron Failure Mode

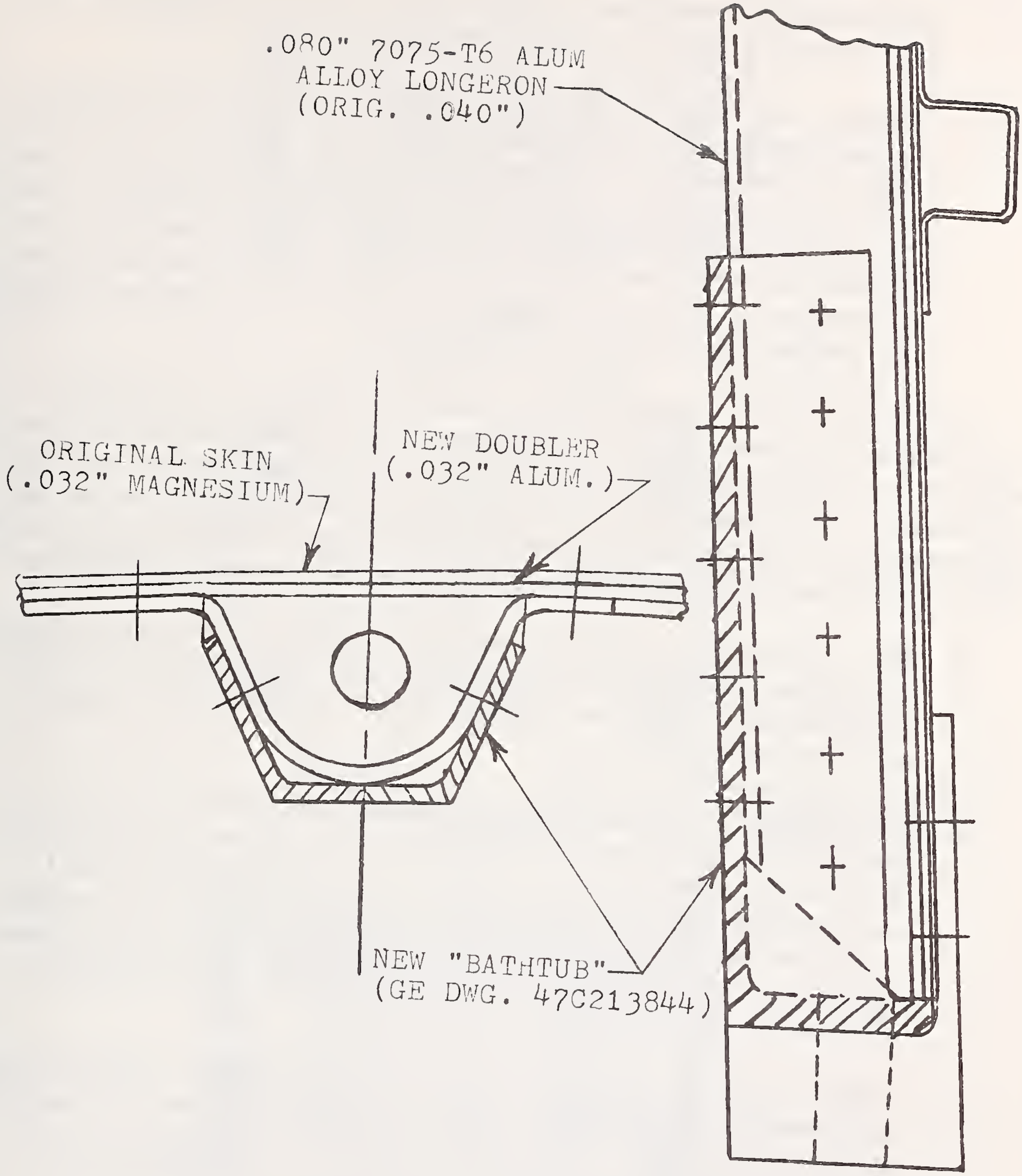


Figure 17. Design Modifications to Longeron

- The longeron thickness was doubled to 0.080 in. in order to negate a requirement for local beefups at the upper and lower ring attachments, also having the effect of lowering the general level of stress.

- The third requirement was to reduce or eliminate the panel buckling that apparently contributed to the failure of the longeron. This was effected in the redesign by riveting 0.032 aluminum doublers to areas of the 0.032 magnesium adapter skin, as indicated in Figure 18. All of the changes were designed so that existing adapters could be readily reworked to the new configuration. As a result, although almost all of the existing skin area required the doublers, it was not replaced by a single aluminum skin.

Because practical considerations dictated that the adapter redesign be accomplished at a time prior to the static test, the redesign details were established using only the results of the wreckage analysis and the revised structural analysis. This necessitated the use of an overlapping design, since the quantitative effects of the various possible weaknesses indicated were not as yet firmly established. The rationale of this overlapping design was as follows:

- Skin doublers were added to ensure every panel would be stable under a maximum combination of axial compression and shear in order to eliminate the trigger that initiated the failure, tentatively diagnosed as forced crippling.

- The bathtub fittings were added at the mounting points in order to eliminate the local eccentricity between the longeron centroid and the flange of the mounting ring. This would virtually eliminate the induced bending which allowed the skin and longeron stresses to peak at the skin attachment flange, having the effect of lowering compressive stresses at the skins immediately adjacent, thus reducing the cause of the skin buckling.

- Rather than add local reinforcements to key longeron ring attachments, as dictated by prior analysis and because longerons had to be removed to allow skin doubling, it was believed that increasing the overall thickness of the longerons would reduce the overall stress level, again reducing the possibility of local crippling.

QUALIFICATION OF REDESIGNED ADAPTER

Rather than repeat the acceleration test to qualify the new adapter, it was recommended that it be qualified by static test to both ERTS-1 and Nimbus-5

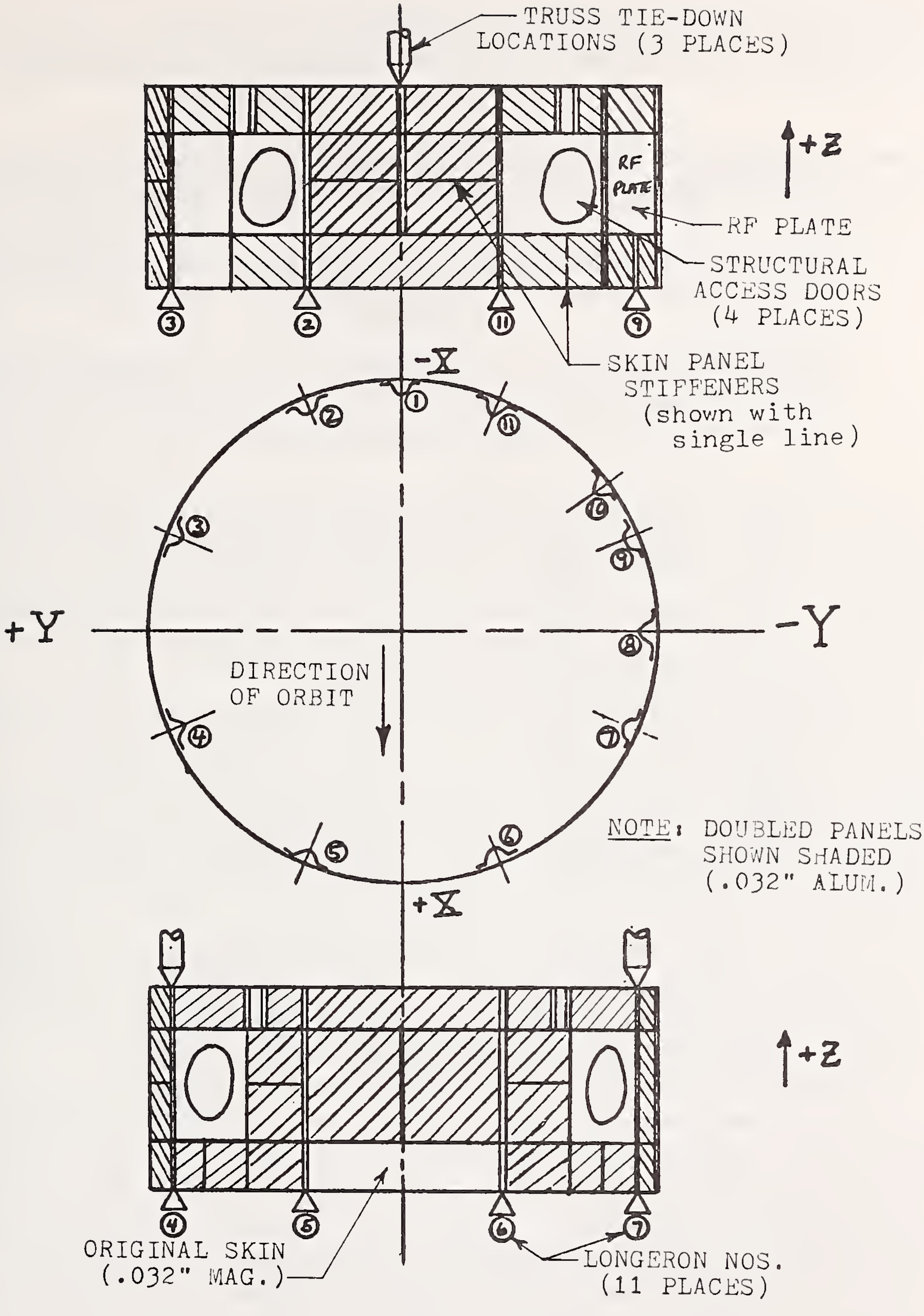


Figure 18. Design Modifications to Adapter Skin

load levels (MECO-6) for three lateral load directions (+X, -X, -Y). Sufficient instrumentation was provided to preclude unexpected failure, to support analytical modeling, and to identify the load distributions. The basic test objectives were:

- To qualify the prototype flight adapter, as modified, to 1.5 times the then anticipated Nimbus-5 and ERTS-1 launch vehicle flight loads.

- To provide complete data for verification and improvement, if required, of the adapter structural analytical model.

- To provide sufficient data to establish an experimentally-derived six degree-of-freedom stiffness matrix for the adapter, for frequency correlation between static and dynamic testing.

The qualification static test article, procedure, and instrumentation were virtually identical with those for the failure analysis static test (see Figures 12 and 13). The qualification test article representation of the ACS, truss, sensory ring, and separation band were the same articles as those utilized in the failure analysis test. The redesign adapter was the adapter utilized in the previous test reworked to the new configuration.

The instrumentation was very similar to that previously shown in Figure 13, but with additional axial strain gage locations on the key longerons. For instance, on longeron 4, instead of a total of 5 gages at 2 levels, there was a total of 17 gages at 7 levels, the additional levels predominantly between the circumferential skin stiffening rings. It was felt that these additional gages would be an indication of the shear lag; that is, they would enable an estimation of the relative load distribution between skin and longeron. These additional gages exceeded the capacity of the recording equipment; therefore, the gages were placed in three patch groups, one of which was recorded for each axis of test determined by which group would be most active. Dial gages to measure upper ring deformations were strategically located.

Qualification level loads were to be applied to the adapter representative of both ERTS-1 and Nimbus-5 flight loads for both roll (X) and pitch (Y) lateral axes. Tables 6 and 7 define the required test loads, their resulting interface loads, and the relationship to the various flight interface loads for roll and pitch axes, respectively. The columns labeled RATIO should indicate 1.50 to provide the proper safety factor on qualification. Again loading schedules were selected to build up to maximum values in an orderly manner to minimize the risk of unexpected results. Basically, thrust loads were first applied, then lateral loads were built up to provide maximum combined test levels.

TABLE 6

Comparison of ERTS-1/Nimbus-5 Roll/Yaw Qualification Loads and Flight Loads for POGO Condition

INTERFACE	LOAD	ERTS-1 FLIGHT	ERTS-1 TEST	ERTS-1 RATIO	NIMBUS-5 FLIGHT	NIMBUS-5 TEST	NIMBUS-5 RATIO
Applied Test Loads							
Thrust-Sensory	F1Z		25,900 (1)			25,700 (1)	
Thrust-ACS	F2Z		5,900 (2)			7,800 (2)	
Lateral-Sensory	F1X		2,200 (3)			4,600 (3)	
Lateral-ACS	F2X		1,600 (4)			3,100 (4)	
Interface Loads							
ACS/Truss	F_x	908	1,600	1.76	1618	3100	1.91
	F_z	3,824	5,900	1.54	4684	7800	1.66
	M_y	7,265	6,450	0.89	22193	12500	0.56
Truss/Sensory	F_x	908	1,600	1.76	1618	3100	1.91
	F_z	3,916	5,900	1.50	5233	7800	1.49
	M_y	54,899	82,400	1.50	107330	159600	1.49
Sensory/Adapter	F_x	2,542	3,800	1.49	5140	7700	1.50
	F_z	20,573	31,800	1.54	21747	33500	1.54
	M_y	87,674	130,300	1.49	172548	256700	1.49
Adapter/Base	F_x	2,545	3,800	1.49	5147	7700	1.49
	F_z	21,190	31,800	1.50	22367	33500	1.50
	M_y	148,674	221,500	1.49	296298	441500	1.49

NOTES:
(1) Indicated load includes 1500 lb. weight of sensory ring and loading fixtures.
(2) Indicated load includes 200 lb. weight of loading fixtures.
(3) Lateral sensory ring load applied by one loading strap at a plane 12.33 in. above separation plane.
(4) Lateral ACS load applied at a point 4.03 in. above Truss Interface.

TABLE 7

Comparison of ERTS-1/Nimbus-5 Pitch/Yaw Qualification Loads and Flight Loads for POGO Condition

INTERFACE	LOAD	ERTS-1 FLIGHT	ERTS-1 TEST	ERTS-1 RATIO	NIMBUS-5 FLIGHT	NIMBUS-5 TEST	NIMBUS-5 RATIO
Applied Test Loads							
Thrust-Sensory	F1Z		26,000 (1)			25,700 (1)	
Thrust-ACS	F2Z		6,000 (2)			7,900 (2)	
Lateral-Sensory	F1Y		2,100 (3)			1,280 (3)	
Lateral-ACS	F2Y		1,040 (4)			500 (4)	
Interface Loads							
ACS/Truss	F_y	660	1,040	1.57	298	500	1.68
	F_z	3,824	6,000	1.56	4,684	7,900	1.68
	M_x	4,429	4,190	0.95	3,051	2,020	0.66
Truss/Sensory	F_y	660	1,040	1.57	298	500	1.68
	F_z	3,916	6,000	1.53	5,233	7,900	1.51
	M_x	35,779	53,600	1.50	17,206	25,800	1.50
Sensory/Adapter	F_y	2,081	3,140	1.51	1,197	1,780	1.49
	F_z	20,573	32,000	1.55	21,747	33,600	1.54
	M_x	54,387	81,500	1.50	27,417	41,000	1.49
Adapter/Base	F_y	2,083	3,140	1.50	1,196	1,780	1.49
	F_z	21,190	32,000	1.51	22,367	33,600	1.50
	M_x	104,334	156,800	1.50	56,121	83,700	1.49

NOTES:
(1) Indicated load includes 1500 lb. weight of Sensory Ring and Loading Fixtures.
(2) Indicated Load includes 200 lb. weight of Loading Fixtures.
(3) Lateral Sensory Ring load applied by two loading straps with centroid of applied load 6.85 in. above separation plane.
(4) Lateral ACS load applied at a point 4.03 in. above Truss Interface.

Prior to the qualification static test of the modified adapter, a complete set of stress predictions was generated in order to correlate with measured strain data. The structural analysis was performed in a manner very similar to that for the original static test: calculation of maximum required test loads, preparation of four unit load stress analyses (one each for ACS-applied thrust, sensory-applied thrust, ACS-applied shear, and sensory-applied shear), and combination of the unit analyses in a manner to represent the applied loads schedule. The following changes in the analyses in a manner to represent the applied loads schedule. The following changes in the analytical computer model were required:

- Longerons were reconfigured to eliminate the eccentricities at their bases by assuming that the mounting bolts were located at the longeron centroid, extended. This assumption now makes the mounting points eccentric to the lower ring centroid, as in actuality.

- Longeron section properties were increased representing increased thicknesses.

- Skin panel thicknesses were increased representing doubled areas.

- Axial load capability was added to all skin panel simulations.

As could be anticipated, with the overlapping design modifications, the general stress levels were determined to be rather low. With the increased crippling and buckling allowables, as a result of the thickened longerons and doubled skin panels, there was no evidence of potential crippling stresses nor initial panel buckling. All margins of safety were found to be substantial.

Static qualification testing was successfully completed. All of the test objectives were met, post test inspections indicated no failures nor permanent deformations, and reduction of data indicated no excessive measured deformations or stresses. As a result of the testing program the following conclusions were drawn.

- The adapter primary structure was qualified to both Nimbus-5 and ERTS-1 qualification level loads as defined. Actual test loads and a comparison with predicted flight loads are summarized in Tables 6 and 7.

- Strain data obtained, although only reduced to a limited extent, indicated that the static test tends to verify the analytically-predicted load paths and stress levels and provided sufficient data to enable prediction of loads and stresses for future loading conditions.

- Dial gage information obtained allowed construction of a 6-by-6 stiffness matrix of the adapter which verified stiffness characteristics found from dynamic testing.

Because the qualification tests proceeded to a successful conclusion without event, it was felt that only a limited reduction of strain data was required. One basic observation, however, was apparent. Although stress predictions were substantially more accurate than previously noted, the measured stress levels in the longerons were still lower than anticipated. Obviously, the MASS computer routine did not accurately predict the thrust, nor the bending-induced thrust, load distribution between skin and longerons. In a practical sense, the utilization of the MASS routine requires the breaking into panel segments, only those areas bounded by structural members, such as: longerons, rings, or other stiffening members. This technique has the effect of softening the panel simulations; that is, the panel simulations were too large relative to the stiffnesses of the framing members.

In addition, it was noted that a good correlation between computed skin panel stresses and measured rosette strain gage data was not readily obtainable. These results, coupled with other observations obtained during the failure analysis efforts, have led to the following conclusion: Axial and shearing stresses, in a skin panel adjacent to highly loaded framing members, vary too rapidly across its span to be either accurately measured using a single strain rosette, nor calculated using a single plate element.

CONCLUSIONS

Based on the failure analysis results, it was concluded that the acceleration test failure was caused by inadequate adapter strength which was not predicted by the pretest structural analysis. Adapter failure was caused by the initial collapse of longeron 4, as a result of forced crippling, due to the combination of panel buckling forces with the basic compression and bending loads. This has been verified by both analysis and test.

Additionally, it was concluded from this failure analysis that the adapter failure during the acceleration test was predictable by analysis if longeron eccentricities and panel buckling effects could have been fully simulated in the structural analytical model. In this case, it was not possible to completely simulate both the local eccentricities and the skin buckling effects; therefore, the computerized analysis had to be supplemented by additional analysis to properly account for these effects. The structural analytical model has been modified to include all local eccentricities as completely as possible and, when supplemented by appropriate manual analysis, has produced results identifying the mode of failure in agreement with the results of the subsequent static test.

Finally, it became apparent during the failure analysis and subsequent static testing that analytical techniques must be further sophisticated in order to more readily predict a highly local failure in a structure loaded nearly to capacity. Sophistication of local failure predictions would:

- Attempt to include in the analytical model the effects of all local eccentricities, particularly those at major load-carrying attachments. If not included in the analytical model, these effects must be considered in a separate supplementary analysis.

- Include the relative effects of the compression-carrying capability of all skin panels or other predominently shear-carrying members. The relative axial stiffnesses of the longerons and the skin panels have been assessed by the static test results. In this specific case, even with full compression-carrying capability included, static testing indicated that the skin still carries more axial loading than predicted by the MASS analysis. Because, to a large extent, practical considerations dictate the modeling techniques used in panel simulation, the effect is one of tending to soften the panel representation. Again, a supplementary analysis appears to be a logical solution.

- Include a local analysis which would assess the effects of a combination of local skin buckling adjacent to a highly-loaded compression-carrying primary structural member. In lieu of this, ensure that all panel members will not evidence initial buckling in combined compression and shear at ultimate rather than limit loads. Such a shotgun approach was felt to be necessary in cases where overlapping designs—thickened longerons and doubled skin panels—were utilized.

It has been found that relatively small eccentricities, while not radically altering the overall structure load distribution, may have a significant effect on a local area. It has been the authors' experience that, if due to practical considerations on computer modeling, an eccentricity exists or panels have been simulated too coarsely, an acceptable solution is to break out a small section of the model to be more accurately simulated. Loadings may be provided by the original computer printout as internal loads at the boundaries of the supplemental model. Such loads may then be more properly redistributed fully evaluating the local effects. This technique, which provides the solution to all three items above, has already been incorporated in the structural analyses of several areas of ERTS and Nimbus series spacecraft components.

In summation, using the redesigned configuration of the adapter, as outlined in this report, ERTS-1 was successfully launched into orbit in July 1972, and in December 1972 Nimbus-5 successfully attained orbit. After successfully exceeding their design lifetimes in orbit, both spacecraft are currently operating in reduced modes, but still returning useful information to earth stations.

Being readied for launch in early 1975 are two more in these series of spacecraft—ERTS-B and Nimbus-F (to be designated ERTS-2 and Nimbus-6 after successful launch). Both spacecraft have been fully qualified and utilize flight adapters structurally identical to that described.

DISCUSSION

J. J. Scialdone, NASA, Goddard Space Flight Center: Does the computer program now include buckling instability?

V. T. Sweet: As we use it, it does not. I understand the developer of our computer program has managed to mathematically simulate this phenomenon, but this simulation has not been incorporated into our program. If we want to predict buckling, we have to do it in a series of stages rather than have it taken care of mathematically.

J. J. Scialdone: Did you try the same program or a different computation with the NASTRAN program, or was it not available at that time?

V. T. Sweet: At the time of failure we considered using NASTRAN, but that approach was given up because it was too costly to completely simulate the model. We had already made plans to do a static test and we thought that would be sufficient to determine what the actual cause was. In the future we may use a program similar to NASTRAN for such a problem.

G. Wagner, Westinghouse Electric Corporation: How did the nominal values of strength that you used to get your factors of safety compare with the actual measured values in the failure analysis?

V. T. Sweet: They were quite similar.

AGRICULTURAL EQUIPMENT FOR UNDERDEVELOPED COUNTRIES

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Equipment designed for underdeveloped countries should be simple to adjust and operate. Where possible it should also be simple and easy to construct using locally available craftsmen as well as materials. Malfunctions, although undesirable, can be tolerated provided they can be rectified locally. What cannot be tolerated are long delays while a part is shipped in from some distant point. Also, machines requiring special service tools or skills are difficult to maintain.

Because the safety of operators, passengers or community is not usually at stake in the malfunction of simple agricultural devices, mechanical failure does not have the same meaning as on airplanes, automobiles, space crafts and nuclear power stations. Malfunction, especially of critical components, in these machines often results in serious and immediate jeopardy of life. Malfunction in simple agricultural machines does not usually constitute such a hazard. If the machine cannot be quickly repaired the jeopardy can be expected later in the form of increased labor to manually perform the machine's task or loss of the goods or services which the machine produced. In extreme cases, especially in tribal societies, this could mean famine.

In addition to simplicity the over-riding design philosophy is that the machine should be built as much as possible by local craftmen using local materials. If they can build a new one they should be able to repair an old one.

Shafts and bearings are items that sometimes must be supplied from "outside". However, it has been shown that slow speed shafts may be made of wood and fitted into wood bearings. Shafts of several hundred rpm are usually made of steel but may be fitted into hardwood bearings soaked in oil. The desirability of wood as a building or construction material is obvious both from the standpoint of availability as well as the ability of local craftmen to work and shape it. Village blacksmiths in many remote areas are able to work metal quite effectively and produce the metal brackets, wear surfaces, hinges, etc., needed.

Because wood cannot be welded like steel or other metals, care must be exercised in fastening parts together. Nails, bolts and screws work well provided sufficient shear surface is provided. Otherwise the fastener tends to cut through or pull out of the wood. Mortise and tenon joints of various types in which part of one member is inserted in an opening in another have been used for centuries and overcome many of the disadvantages of nails, bolts and screws. They should be used as extensively as possible. Modern technology has produced excellent glues but these are often not available in underdeveloped or remote areas. Use of available glues in load bearing applications should be limited to mortise and tenons where they can keep these joints from coming open under load reversals (tensile loads). Reinforcing high stress areas with metal straps, corner to brackets, etc., is a useful practice.

Over the past several years the writer has designed several agricultural machines for use in underdeveloped countries. This work has been done in response to requests from VITA (Volunteers for International Technical Assistance) which serves as a clearing house between technical cooperators and the person requesting assistance. A few of the devices designed are described below.

Animal Powered Pump

Figure 1 shows a mechanism for applying animal power to the handle of an existing man-operated pump. It consists of a beam, mounted on a vertical shaft, to which a draft animal can be hitched and constrained to move in a circle. The shaft drives a beveled gear and pinion set to which a crank arm is attached. A connecting rod between the crank arm and the pump handle provides the required reciprocating motion.

This design is more difficult to repair than it should be as the gear set would not be available in many remote areas. Some ideas have been advanced which would provide the necessary motion without the gears. It has appeal because it applied animal power to deep wells where the Persian Wheel is not applicable.

Coffee Bean Sheller

A machine to remove the pulp and outer covering from coffee beans is shown in Figure 2. It consists of upper and lower wooden counter-rotating disks fitted with interdigitating spikes. A feed hopper supplies the beans to the shelling area through the hollow shaft of the upper disk. Bearings may be of wood. The support frame (not shown) would also be of wood for ease of construction and repair.

Tobacco Twister

A tobacco processing machine, designed to twist leaves of chewing tobacco into a rope, is shown in Figure 3. It is man powered by means of a treadle and looks like a heavy duty spinning wheel. Again, the construction is of wood with a minimum of complex detail. Leaves with midribs removed, are arranged on the table, twisted by the spindle and then wound on the drum. The machine should operate trouble free if properly constructed, but if malfunctions do occur they can be easily repaired. As this machine converts a bulky crop into a product which can be stored, transported and marketed it could be a valuable asset to a tribe or other small production group.

Twine Spinning Machine

Various types of naturally growing fiber can be twisted into twine or rope by the machine shown in Figure 4 which was originally designed to spin a particular variety of palm leaves. It is very similar to the tobacco twisting machine except that it operates at a higher rpm in recognition of the greater number of twists required. Also the twisted material can be wound onto the spool without stopping the machine's rotation. When the tube and spool rotate together no windup occurs. When the spool is stopped, by hand pressure, for example, the twisted thread is drawn through the tube and wound on the spool. A slight drag introduced between the spool and frame makes for automatic windup.

Pedal Powered Thresher

In many areas of the world human beings are used as a source of mechanical power because motors or engines are not readily available and animal applications may not be easy to make. Figure 5 shows a pedal operated thresher which is significantly more efficient than man powered flails or animals stomping out the grain with their hooves. Pedal actuation was selected because the legs are stronger than the arms. In order to conserve power and energy, only the heads of grain are allowed to enter the threshing mechanism.

The machine is built on a wooden frame, Figure 6, which is mortised together. The threshing mechanism consists of a wooden barrel-like container inside which a series of bars, mounted on disks at each end, is rotated. These bars pass in close proximity to fixed bars in the bottom of the barrel-like container and strip or thresh the grain from the heads. The straw is then withdrawn by the operator and the grain falls through slots into a tray.

Everything except the shafts, the threshing bar surfaces and the drive belt can be made from wood and the belt can be made of local rope or leather. Shafts turn in oiled wooden bearings but a metal sleeve or antifriction bearing could be added if available. Also, it would be possible to mechanical power the machine wherever motors or engines are available. An additional pedal set could be added to divide the load between two people.

A cleaning attachment was added, Figure 7, to remove the chaff from the threshed grain. This consisted of a belt driven wooden propeller fan arranged to blow air across the stream of grain as it falls into the collecting tray.

In all of these applications the emphasis has been on simplicity of design, ease of construction using local materials and workmen, and the capacity to be repaired without requiring service parts from "outside." Although the tools described cannot increase productivity enough to make the user competetive with mechanized agriculture, they may support the production of small amounts of goods for trade purposes which is a first and necessary step toward further mechanization. Also, these simple devices could well be the first step in the development of a ladder of technological skills which would pave the way for the introduction of more complex and more productive devices. Much depends on the ease with which machine failure can be prevented.

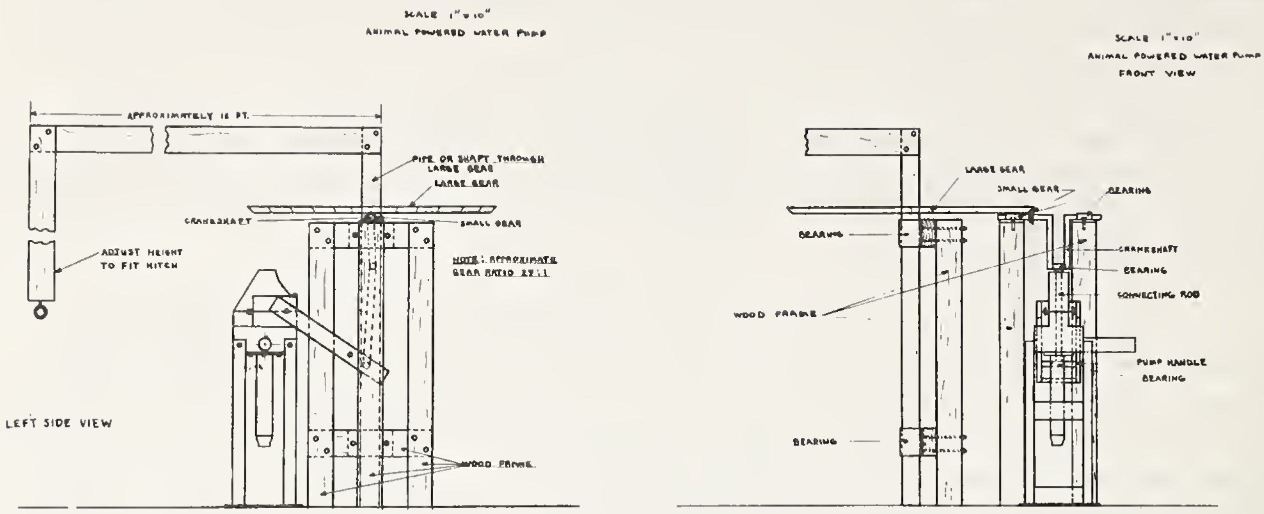

Figure 1. Animal powered reciprocating pump.

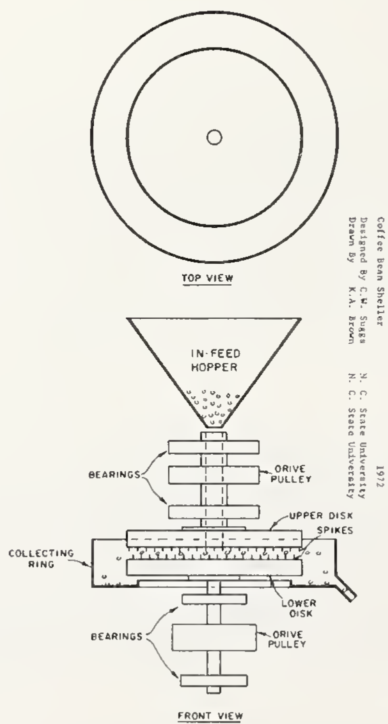

Figure 2. Coffee bean depulper.

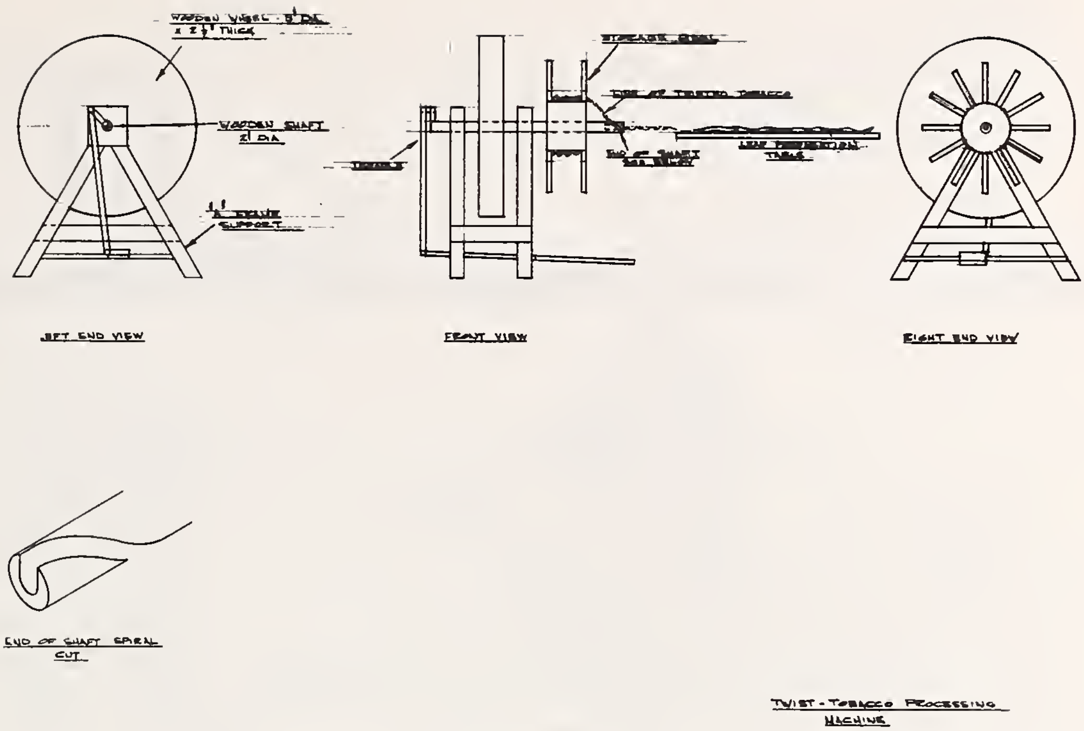

Figure 3. Machine for twisting cured tobacco leaves into a rope.

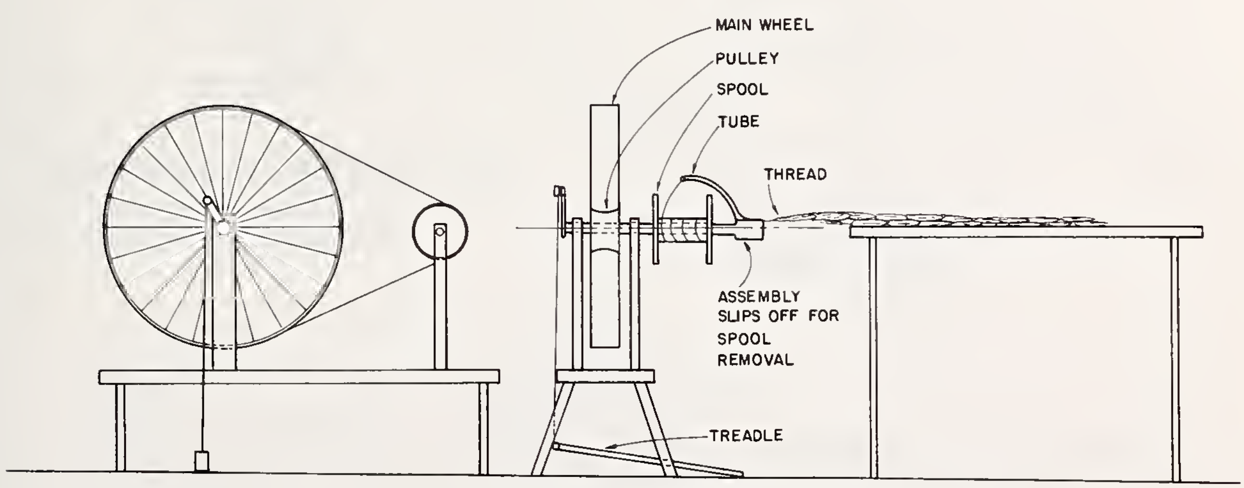

Figure 4. Twine making machine.

plans for a soybean thresher by a VITA volunteer

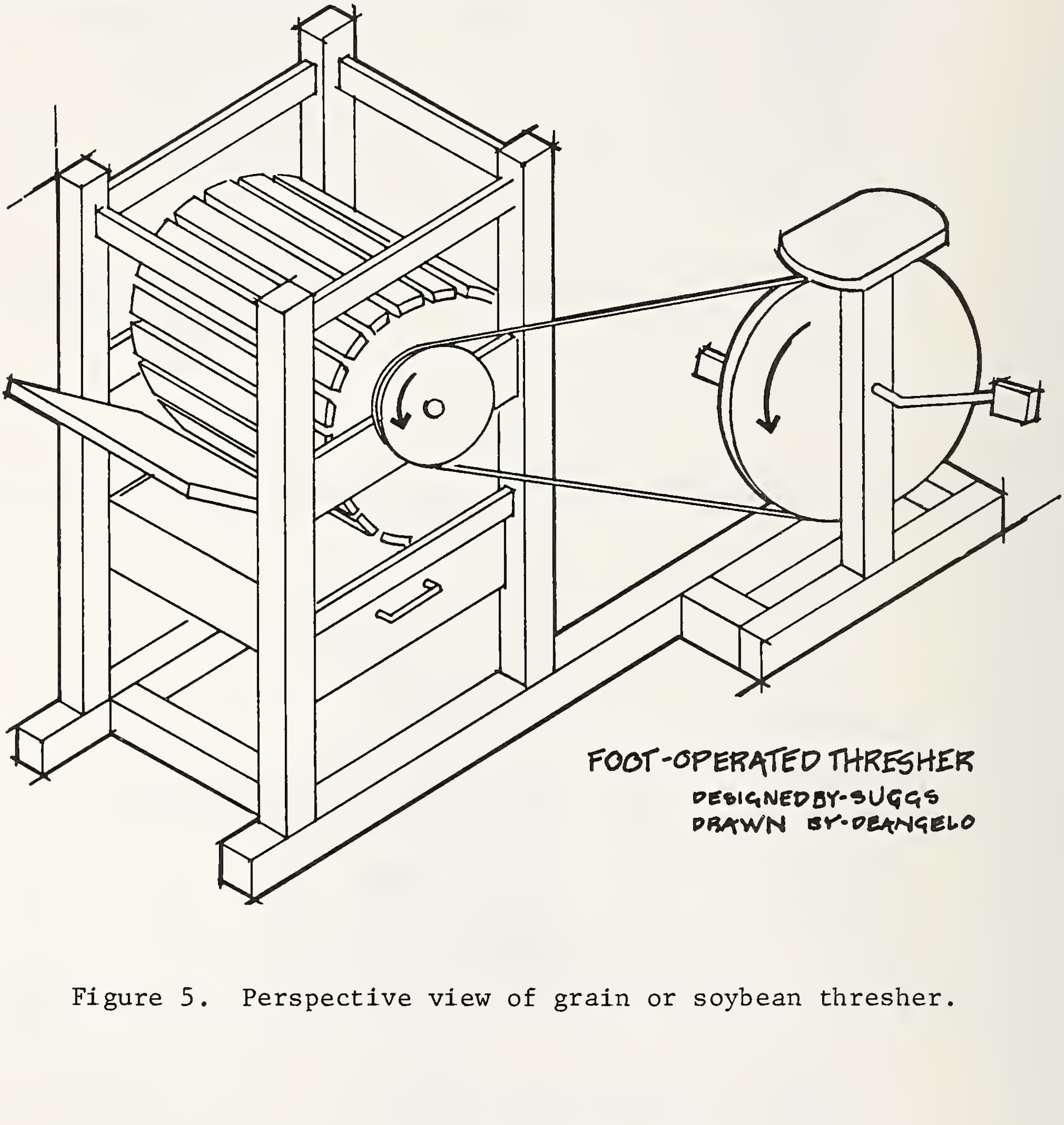

Figure 5. Perspective view of grain or soybean thresher.

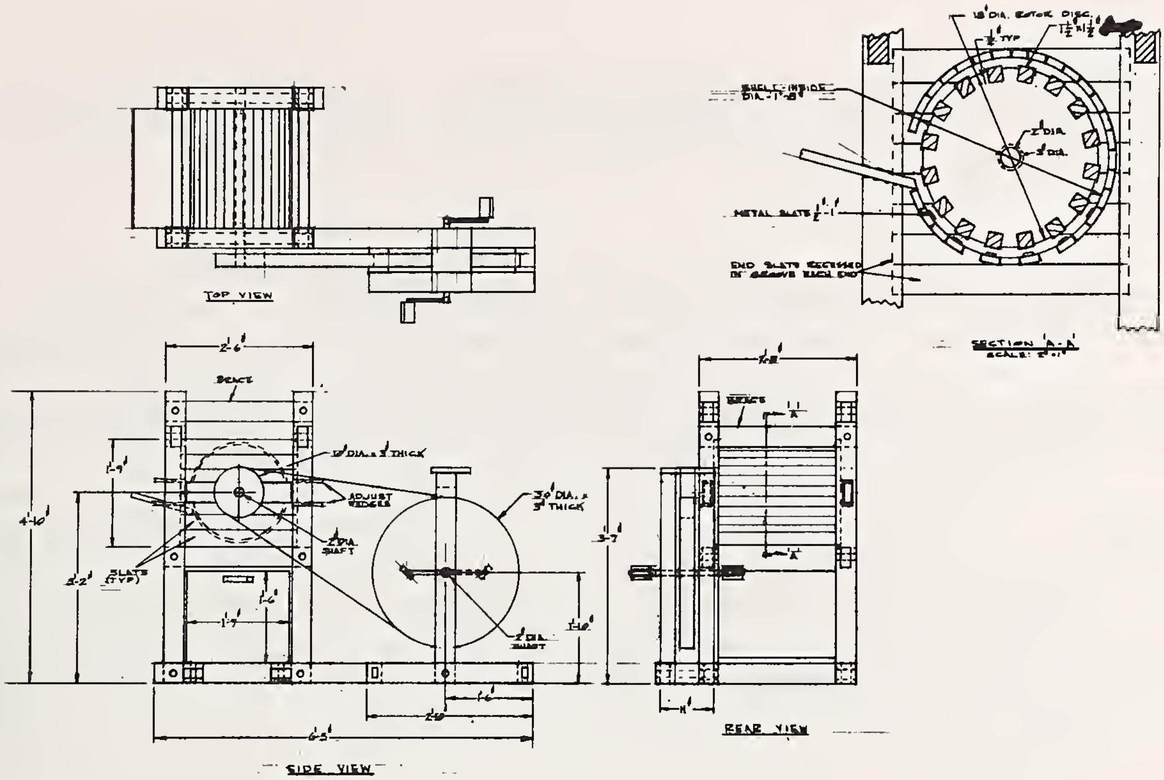

Figure 6. Drive and threshing mechanism showing slots for threshed grain to fall through.

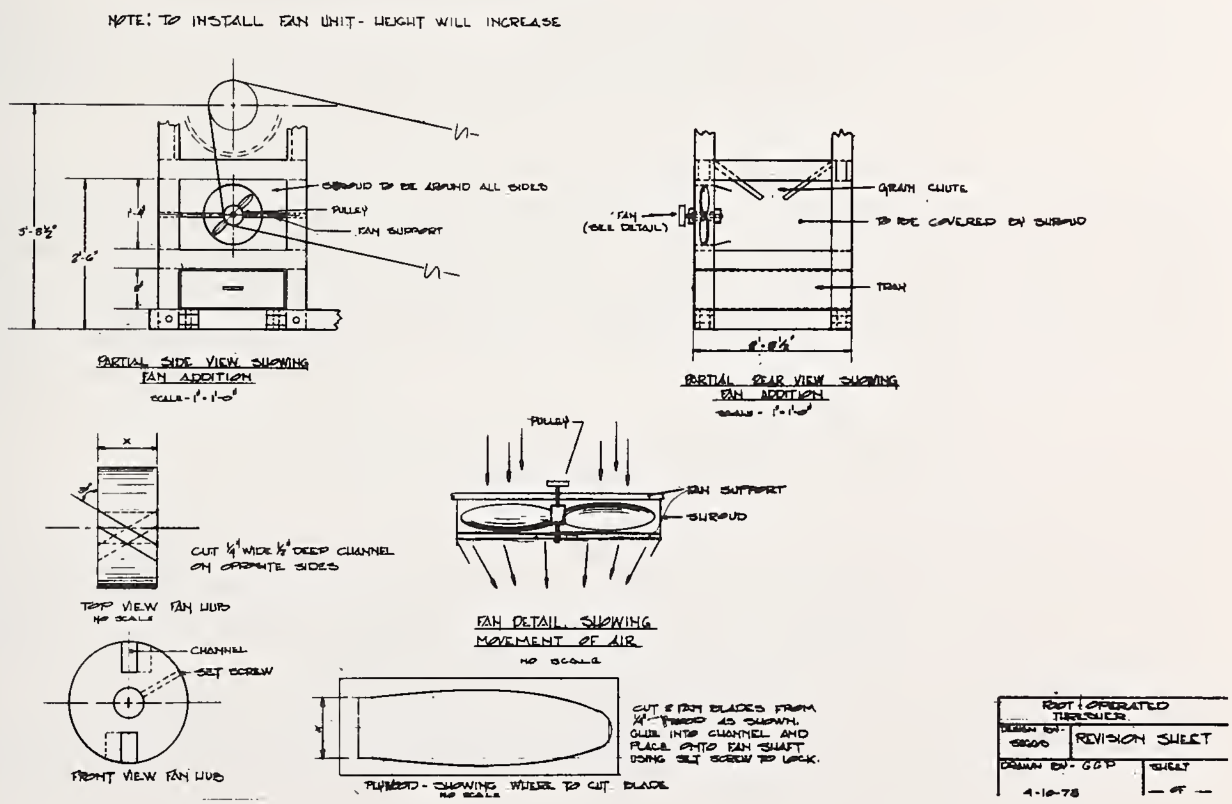

Figure 7. Design of cleaning fan for thresher.

DISCUSSION

R. M. Thomson, National Bureau of Standards: Agricultural design is probably several hundred thousand years old. I wonder if design can still be improved on. Do you find yourself going back to old designs and putting things together?

C. W. Suggs: Sometimes. Designs are available for a great deal of the equipment that is needed in some of the underdeveloped areas. Usually adaptation is necessary because the conditions are different.

J. Oroshnik, U. S. Navy: I am curious to know how the coffee grower ultimately separated the coffee bean husk from the bean itself.

C. W. Suggs: I do not know. I wish that it was possible to have more feedback from the people in the field so we could see how they make use of a design. This system was not complete; it only carried the bean part way. From there, a floatation technique would be needed to separate the hull from the bean. We had to assume that the operator had available some way of doing this.

SAFETY SUCCESS THROUGH DESIGN

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The U.S. Army Agency for Aviation Safety has the mission of determining ways and means to enhance the combat capabilities of the U.S. Army through the conservation of aviation resources. The aviation safety effort to be discussed here involves the use of computer stored mishap data, the research and analysis of that data, and the application of the results of that research.

We are convinced that if the hazards that produce accidents can be absolutely identified, then design, training and supervision can reduce the potential of these hazards becoming cause factors. With this conviction we want to get plugged into design and training cycles as early as possible. The past solution to safety problems has been "fly-fix-fly." This has been replaced through the use of the system safety process which uses not only problems from similar aircraft, but problems from similar subsystems and analysis of the new design.

The order of precedence in the resolution of hazards includes the realization that not all hazardous conditions can be "designed out." If the presence of a hazard must be accepted then safety devices, warning devices or special procedures have to be developed.

The first example of the application of this process involves the flight helmet used by aviation crew members. Injury data for the calendar year 1967 to 1969 shows that nearly one in four fatalities was due to head injury (Fig 1). The vast majority of these people were wearing helmets. Why then was the head so vulnerable? The helmet became dislodged during the accident.

Army Aviation Accident Fatalities
Calendar Years 67-69

Fatal Head Injury	175
Total Non Thermal Fatalities	766
Percentage as Head Injury	22.8%
Fatal Thermal Injuries	328
Total Fatal Injuries	1094
Percentage as Thermal	30%

Figure 1

SPH-4
(new)

APH-5
(old)

Figure 2

SPH-4
(new)

APH-5
(old)

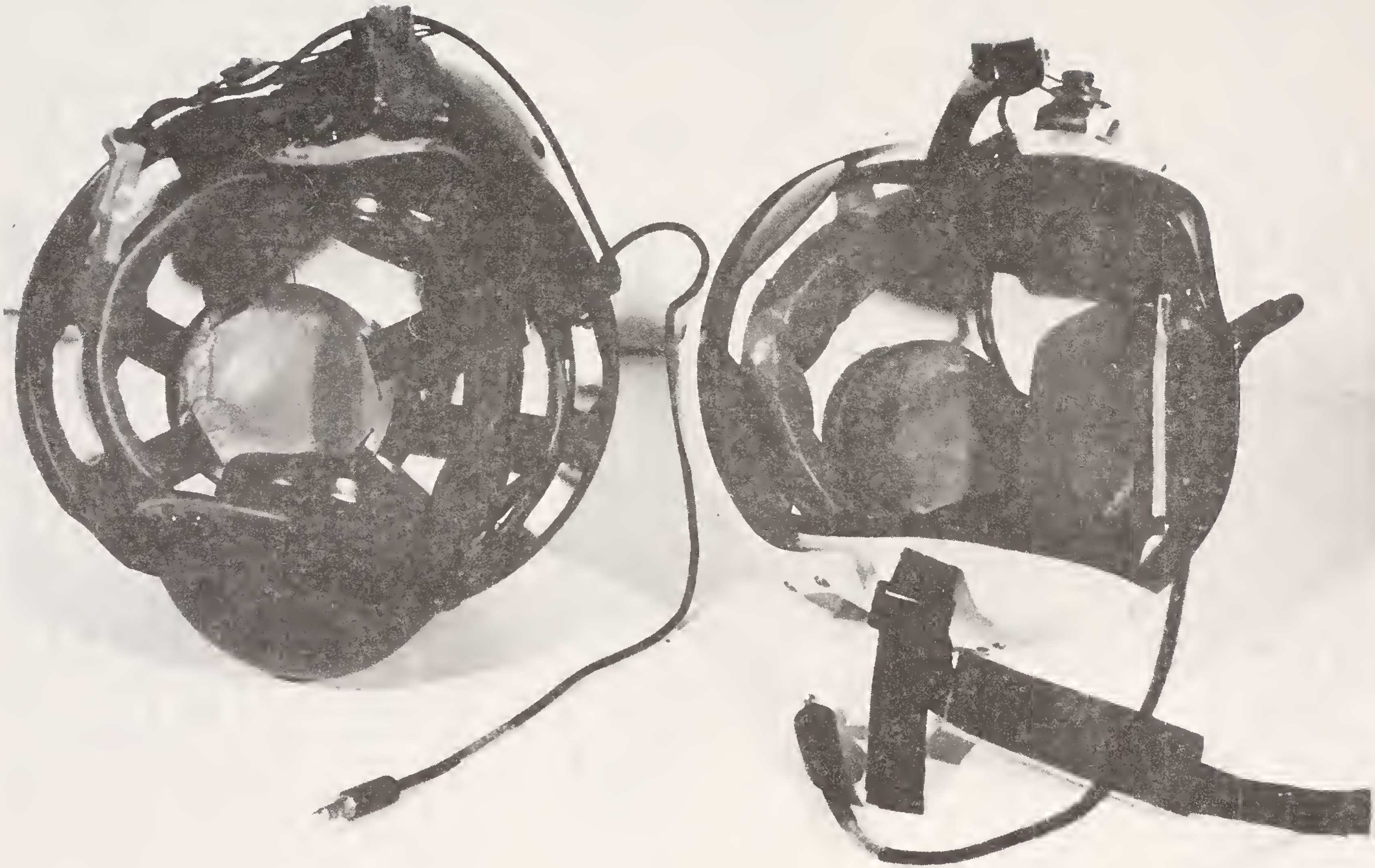

Figure 3

Substantial redesign of the helmet was accomplished. The general appearance wasn't changed except for an "earmuff" look (Fig 2). The major changes are inside. The new helmet uses a suspension system not unlike a football helmet (Fig 3). The suspension system attaches to the head and the helmet is attached to the suspension system. Injury data from fiscal years 1972 to 1974 shows only one in 10 fatalities due to head injury (Figs 4 & 5). This is a substantial improvement achieved through well founded design and design criteria.

Army Aviation Accident Fatalities
Fiscal Years 72-74

Fatal Head Injury	38
Total Non Thermal Fatalities	364
Percentage as Head Injury	10.4%

Figure 4

Army Aviation Accident Injury Data

	Old Helmet		New Helmet	
	No.	Percent	No.	Percent
Personnel Wearing Helmet	160		739	
Helmet Dislodged from Blow	5	3.12%	19	2.57%
Helmet Dislodged from Deceleration	11	6.89%	7	0.95%
Helmet Contributed to Injury	9	3.75%	6	1.22%

Figure 5

Another example of safety achievement through design relates to the fact that 30% of the accident fatalities are thermal. Aircraft carry fuel. Fuel burns. After the violence of a crash impact, fuel can be ignited by a variety of ignition sources and engulf the entire aircraft in a fraction of a second. The weakness of the fuel cells, fittings and lines resulted in conceptualization and design of crashworthy fuel systems. This includes fuel cells that are relatively rupture proof, fuel fittings that are self sealing at breakaway, and fuel lines that pass loads on to the breakaway fittings. Again the design that is oriented towards the environment - the crash environment - has produced some impressive results. The occurrence of thermal fatalities in impact-force survivable accidents has gone to zero (Fig 6). In the System Safety Logic Process this represents a "safety device." There are cost and weight penalties associated with these crashworthy systems, but the savings in resources have been overwhelming.

Crash Injury Data
Survivable Accidents

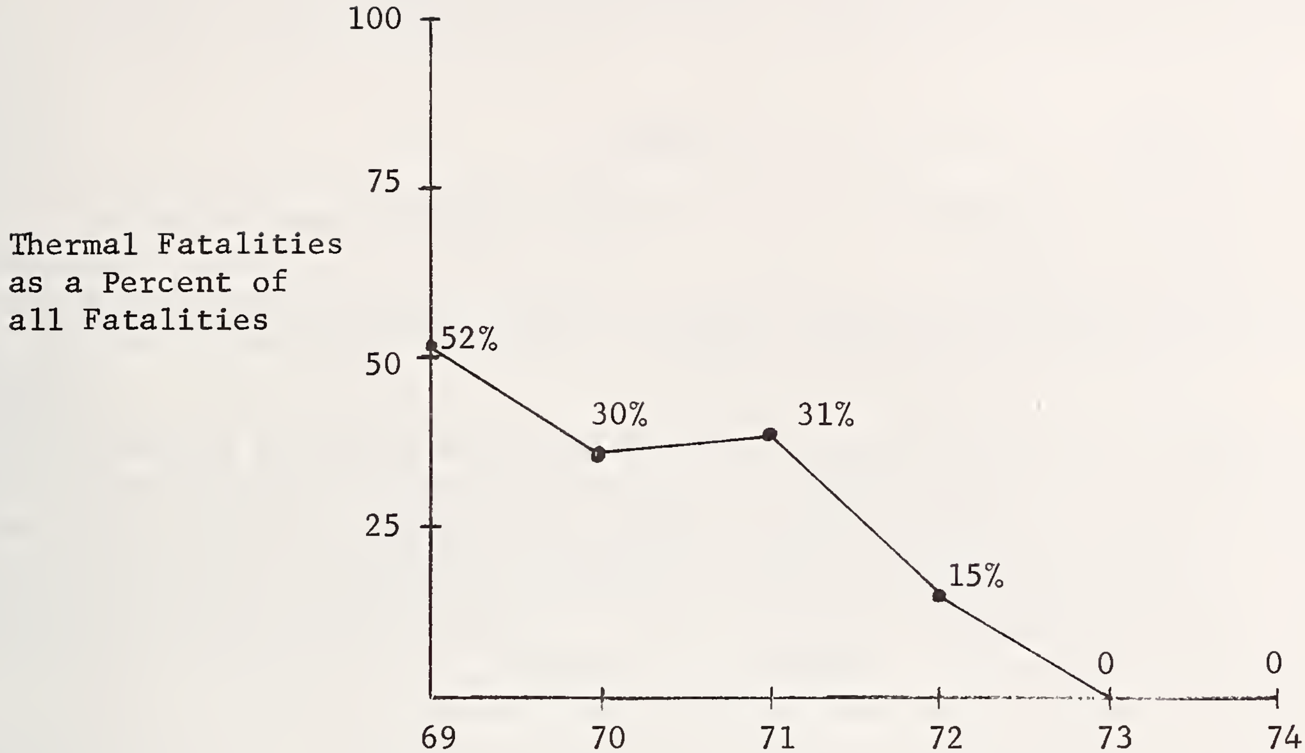

Figure 6

There is more work not yet proven. Crashworthy structures are to be a part of the new Utility Tactical Transport Aircraft System (UTTAS) and the Advanced Attack Helicopter (AAH). Hydraulic systems and power train systems are being designed with an eye towards the prevention of the mission abort failures as well as the catastrophic failures.

There are many areas in which safety success can be achieved by component design. We in the Army have engineering capabilities. We use these capabilities to sort out the problems. But we rely very heavily on industry to apply the principles of design engineering to achieve the needed safety success. Identification of problems in terms of criteria that can be "design criteria" is challenging, but it produces results. It's the only way to go.

(Opinions stated above are not to be construed as an official Department of the Army position unless so designated by other authorized documents.)

DISCUSSION

R. E. Maringer, Battelle Memorial Institute: Has the crashworthy fuel system technology been applied to the automotive industry?

D. W. Logan: To the best of my knowledge, no. However, some of the automotive manufacturers have positioned fuel tanks in less vulnerable positions rather than building them of flexible material as the Army is requiring.

R. E. Maringer: To civilian aircraft?

D. W. Logan: I believe that it has not been applied to other than crop duster aircraft.

R. Lenich, Caterpillar Tractor Company: Would you elaborate on the Tele-torg method vs the torque wrench?

D. W. Logan: The Tele-torg bolt that Boeing selected has a hollowed out shaft. A pin goes all the way through the head to the threaded end of the bolt. Then the bolt is filled with fluid and a glass cap is put on top. As the bolt is stretched by tightening a nut, the pin that is against the glass window is pulled away from the window and fluid covers the head of the pin. So, it indicates the preload in tension and not torque. From what I understand, the Tele-torg bolt is reusable.

RELIABILITY, POLLUTANTS AND ALUMINUM RAW MATERIAL

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SUMMARY

This paper describes how subtle pollutants from the storage atmosphere led to a sizeable corrosion condition in aluminum tubing raw stock that was suspected of adversely affecting the reliability of the end products. This paper is divided into the following major topic segments. A description of the factors contributing to the corrosion condition is first given. This is followed by a discussion of the effects of the corrosion condition on the reliability of the products. Next is a review of the planned actions to resolve the condition. Specifics of the test program are then discussed which include the different types of testing performed and the test results. At this point enough data is available to draw some basic conclusions to conclude the paper.

The investigation covered the following three basic types of aluminum tubing in various heat treated conditions: (1) 2024, (2) 6061, and (3) 5052. The 2024 which has a high copper content for aluminum tubing and is used for more structural applications was the most severely damaged by the corrosion and as a result required the most corrective measures.

DESCRIPTION OF CONDITIONS

The fact that subtle atmospheric pollutants infiltrated our aluminum raw stock area to the degree that corrective measures became necessary establishes the basis for this paper. Seldom do we have great concern over such a simple thing as the apparent normal environment of the storage area for raw materials. Warehouses across the country being somewhat open to the elements and shipments being made in open trucks result in a tendency to accept raw material storage with less concern than we reserve for completed end products. We were concerned enough with our raw material storage, however, that new dry and ventilated warehouse facilities were provided at the beginning of the production line. Yet under these conditions pollutants in the form of corroded tubing were detected in the production cycle. Tracking back through the production steps the corrosion was traced to the raw material stores area. Investigation in the raw material stores area revealed that dust particles settled on the aluminum tubing while stacked vertically in racks. The particles were on both the inside and the outside of the tubing. The large diameters had more inside than did the small diameter tubing. The dust particles absorbed humidity from the air along with just enough fume traces from material cleaning and processing dip tanks located in the next building bay to be corrosive. The shape of each corrosion pit followed the shape of a dust particle resembling a tiny worm track. For the most part they were minute shallow surface pits no larger than a dust particle. The density of dust particles settling on the surface of the aluminum determined the lateral extent of corrosion. A review of air ventilation movement disclosed that air coming in from the roof ventilation could induce fume traces. For, on occasion, the wind could be from the proper direction to move vented processing fume traces outside the building across the ventilation openings above the aluminum storage area and down into and onto the raw stock. Such movement could allow the fume traces to be trapped by the dust particles on the aluminum. The corrosion appeared minute and negligible in most cases, even though some aluminum pieces did give the appearance of being somewhat heavily worm tracked. This in itself would have little to no effect on the reliability of the end products which include such things as air conditioning ducts, hydraulic lines (high and low pressure), drain lines, hot air flow ducts, push pull tubes for control systems, structural braces and other things. Samples of the tubing were subjected to laboratory analysis to determine the extent and effect. The analysis revealed shallow surface pitting which was no real concern for alarm provided the corrosion was arrested prior to finish application. However, in addition to the surface pitting some intergranular corrosion was detected which was considered to be severe in nature for critical applications. Intergranular corrosion, if not arrested, will go deep into the metal. It progresses along between the grains and eventually destroys its structural properties resulting in part failure. Critical applications in our case were such things as high pressure hydraulic and pneumatic lines and control system linkage members (push pull rods). The intergranular corrosion was detected in all three types of tubing analyzed but was the most prevalent in the 2024 due to its copper content. In those cases where the intergranular corrosion was found, it most usually started in the bottom of a corrosion pit and extended down into the aluminum. The paths of corrosion were very evident in specimens that were cross sectioned and polished for examination. The findings from these analyses resulted in the establishment of a plan of action to prevent an impact on the reliability of the end products.

EFFECTS ON RELIABILITY

A major concern created by the presence of the corrosion was the potential effect on the reliability of the end products. This effect would be a reduction in reliability due to part failures. The presence of slight surface pits would not affect the functional use of the aluminum tubing parts. However, the threat of the presence of intergranular corrosion caused a classification of all affected aluminum parts into the two categories; critical and non-critical. It was decided that a test program to thoroughly measure the extent and effects of corrosion on the three aluminum types involved was necessary to evaluate corroded parts in the critical category for final disposition.

It was felt that intergranular corrosion could damage the structural characteristics of the aluminum causing it to fail under load. It was concluded that such failures would occur after the parts had been in service for a period of time but before they should reach their life expectancy. High pressures and/or physical loads would be seen by the critical parts. The non-critical parts were such things as low pressure air flow ducts, return lines and light load bearing members. As such they would not be stressed to the point that they would be likely to fail.

Another concern was to determine when the corrosion started and how far it reached into production. A special study was made in this direction to determine the most likely production lots affected. This portion of the investigation keyed from a likely date that the corrosion could have started. Since the cause was associated with the new storage area, the date it was put into use service became a key in determining part lots that could be affected. The condition was primarily isolated to parts in work and in stock. To eliminate this apparent threat to product reliability, a plan of action was decided upon.

PLANNED ACTIONS

It became evident that an overall plan of action was necessary to cover all major elements of the condition. It would be necessary to purge the raw materials receiving and storage area, review parts in work on the production line and evaluate the effect on completed parts for possible rework or replacement action. A detail plan evolved which encompassed the following major actions.

Stock Purge

The entire aluminum tubing raw materials stores area was purged. New receipts were inspected during this time and some corrosion was found. However, it was not to the extent of that in stock. This amounted to the immediate rejection of approximately 620,000 feet of aluminum tubing which had exhibited corrosion under the visual inspection of a 10X hand magnifying glass. Test samples sent to the materials processing lab were cross sectioned and examined under magnifications up to 1000X to confirm intergranular corrosion. This material was held until there was enough valid test data to make a reasonable disposition. Approximately one-third of this tubing was ultimately reclaimed.

All critical tubes were pulled from stock, stripped of paint, re-inspected and corroded parts were rejected. Half of all non-critical painted tubes and all non-painted non-critical tubes underwent similar treatment to evaluate the extent of the corrosion and to evaluate if it was necessary to inspect all of the non-critical painted parts. The other half of the painted non-critical tubes were held and ultimately not reinspected due to subsequent faborable test results which indicated that chemical cleaning prior to fabrication and the part finish (anodize and paint) had arrested any corrosion. The effects of corrosion on these parts was not of sufficient magnitude to cause part failure. All tubes in process of fabrication were reinspected and the corroded parts rejected. All rejected raw material and parts were reviewed for the proper disposition.

Tubing Control Board

A special tubing control board was established to recommend disposition to Material Review Board (MRB) for rejected tubes and raw material.

The board members consisted of:

1. Design Groups
2. Tooling
3. Product Assurance
4. Process Control - assisted by:
    a. Inspection
    b. Manufacturing, and
    c. Manufacturing Control

The board reviewed all rejected tubes from manufacturing and made a documented recommendation to MRB for disposition. These dispositions were based on results from the test program taking into account the material type and the application of the part by the board members. When necessary, the control board would request special testing (i.e., sample sectioning and inspection under 500X - 1000X magnification) on lots of parts to aid in making their recommended dispositions. This board was established in the factory area and ultimately reviewed approximately 60,000 parts of which approximately 17 percent were scrapped by MRB and 83% were returned to production as acceptable for use. The scrappage occurred primarily on those corroded tubes that were to be used in what was considered to be a critical application. Critical applications were considered to be such usage as high pressure lines and load bearing members of structural or linkage mechanisms.

Over half of the raw material was ultimately scrapped due to the extent of the corrosion, for only that material that was considered to be usable for both critical and non-critical applications was retained. It was chemically cleaned in dip tanks to arrest the active corrosion and placed in polyurethane bags for restocking.

Industry Conference

Experts from the leading aluminum companies were called upon to review the condition for recommendations and opinions. It was the opinion of the aluminum companies that chemical cleaning in dip tanks and subsequent anodine/paint finishing would permanently arrest the corrosion. However, they recommended a testing program to determine: (1) if the corrosion was in fact arrested; (2) the effects of the corrosion on the structural characteristics of the aluminum tubing; and (3) the effects of weather environment on a large sample of different aluminum finishes on the different aluminum types. These recommendations were subsequently implemented with the development of a test program on the aluminum tubing.

TEST PROGRAM

The test program associated with the corroded aluminum tubing condition consisted of the following types of tests performed on several heat treated conditions of the three types of aluminum involved; (1) 2024, (2) 5052, and (3) 6061. Each type of test and its results are discussed separately. The results of these tests shaped the actions taken to prevent detrimental effects on the reliability of the products.

Intergranular Attack Examination (Visual-Microscopic)

Raw stock, partially fabricated, and completed production parts were examined to determine:

1. If corrosion was present
2. Type of corrosion
3. Depth of corrosion

A total of 735 parts was selected for this examination. The samples were from both raw materials and production parts.

Each specimen was sectioned at areas of maximum appearing corrosion, with up to four sections per part. These sections were mounted, polished, and visually studied. Each section had from zero to four corrosive areas, thus giving up to 12-16 pits per part to evaluate, with the maximum depth being the recorded figure.

All studies were made at magnifications up to 500X, and the intergranular suspects verified at 1000X magnification. The results are shown in the following table:

Material	Total Specimens	Intergranular Corrosion	Surface Corrosion	No Corrosion
2024	52	33	15	4
5052	501	50	416	35
6061	182	64	99	19
TOTAL	735	147	530	58

The maximum depth of corrosion per specimen is shown in the following table.

IG = Intergranular
Surf = Surface
Depth in inches.

Corrosion Depth	2024 IG	2024 Surf	5052 IG	5052 Surf	6061 IG	6061 Surf
Surface	0	7	1	181	23	71
- .0011	6	5	12	119	13	14
.0011 - .0020	24	3	32	110	19	14
.0021 - .0030	3	0	3	5	7	0
.0031 - .0040	0	0	2	1	2	0
	33	15	50	416	64	99
TOTALS	48		466		163	
			TOTAL 677			

Depth summary. The above table measurements show that of the 677 specimens evidencing corrosion, 654 had maximum corrosion depths of 0.0020 inch or less, and no depths greater than 0.0040 inch.

A percentage breakdown of corrosion depth of the 677 specimens is as follows:

Corrosion Depth	No. Specimens	% of Total
Surface	283	42
- .0010	169	25
.0011 - .0020	202	30
.0021 - .0030	18	2.6
.0031 - .0040	5	0.7

Conclusions. Of the 2024 aluminum specimens exhibiting surface corrosion 68% of it was intergranular in nature while 39% of the 6061 was intergranular and only 12% of the 5052 was intergranular. As for depth, 97% of all corrosion was less than .002" deep.

Salt Spray

The purpose of this test was to determine if active corrosion can be arrested by the chemical film process (MIL-C-5541) and Type I anodize process (MIL-A-8625A) and if tensile and elongation properties were affected. It was very important to know this to support decisions of what to do with completed parts.

Specimens. Pitted and corroded tubes were selected from raw stock and cut into 8 inch lengths.

No. Specimens	Type Aluminum	Surface Condition
3	2024-0	These nine specimens were
3	5052-0	tested for control - no
3	6061-T6	salt spray exposure
5	2024-0	Anodized PS 74.02-3
5	5052-0	Anodized
5	6061-T6	Anodized
5	2024-0	Chemical film PS 74.02-2
5	5052-0	Chemical film
5	6061-T6	Chemical film
2	2024-0	No surface processes
2	5052-0	No surface processes
2	6061-T6	No surface processes

Procedure. Five specimens from each alloy and process were placed vertically in the salt spray (per QQ-M-151) along with two untreated specimens of each alloy. All specimens were removed at the same time after 240 hours exposure. The physical properties - ultimate tensile strength and percent elongation - were determined on each specimen along with controls which were not subjected to the salt spray test. The results of the test are shown in the following summary table.

Salt Spray Test Summary			
240 Hours QQ-M-151			
		Physical Properties	
Material	Finish	Tensile % Loss	Elong. % Loss
2024-0	Bare	-10.7	-25.0
	Alodined	- 2.1	-11.0
	Anodized	Nil	- 8.7
5052-0	Bare	Nil	Nil
	Alodined	Nil	Nil
	Anodized	Nil	Nil
6061-T6	Bare	Nil	Nil
	Alodined	Nil	Nil
	Anodized	Nil	Nil

Conclusions. The chemical filmed and anodized 5052-0 and 6061-T6 alloys showed little or no change in physical properties after 240 hours of exposure to the salt spray test. Minor additional corrosion occurred on the non-treated (bare) 5052-0 and 6061-T6 alloys, but with no measurable effect on their physical properties.

The 2024-0 alloy showed a significant loss in physical properties.

Recommendations:

1. 5052-0 and 6061-T6 tubing showing evidence of corrosion was satisfactory for non-critical and critical use.

2. All 2024-0/2024-T3 tubing showing evidence of corrosion in raw stock, in process of fabrication, or finished unpainted parts was recommended for disposition by accept/reject criteria to be established to prevent a degradation of reliability in certain applications.

   NOTE: 2024-T3 (final condition of fabricated parts) was slightly more susceptible to corrosion than 2024-0.

Verification. Technical experts from two aluminum tubing producing companies reviewed in-house processing, cleaning, alodining, and anodizing procedures and verified that such processing arrested the corrosion.

Tensile - Compression Tests

Specimens. Specimens of 2024-T3, 5052-0 and 6061-T6 corroded and non-corroded tubing were taken from production stores and tested as representative of structural and hydraulic return line (low pressure) tubing.

Procedure. Tensile and compression tests were conducted in a Baldwin Lima Hamilton Universal test machine of 120,000 pounds capacity, equipped with an autographic load-deformation recorder. Tensile yield (0.2 percent offset) was measured with a T-1M extensometer clamped to the tubing. Elongation was measured over a 2 inch gage length. Full section tensile specimens were tested in accordance with Method 211 of Federal Test Method Std. 151, July 17, 1956.

Compression specimens were tested per ASTM Method E9-67. A spherically seated upper compression head was used. The yield strength was measured with a PC-1M compressometer using the 0.2 percent offset method.

Results. The tensile and compressive properties of the tested tubing, both new and corroded, are shown in the following summary table:

Material	F_{ty},KSI	F_{tu},KSI	% Elong	F_{cy},KSI
5052-0 New	12.5	28.7	---	12.6
5052-0 Corroded	11.7	26.8	---	11.6
Corroded vs New, %	- 6.4	- 6.6	---	- 7.9
2024-T3 New	50.2	70.9	20	53.2
2024-T3 Corroded	50.6	71.3	16	49.7
Corroded vs New, %	+ 0.8	+ 0.6	-20	- 6.6
6061-T6 New	41.5	48.0	20.3	40.3
6061-T6 Corroded	39.4	47.6	20.7	42.9
Corroded vs New, %	- 5.1	- 0.8	+ 2.0	+ 6.5

The tensile properties met the strength requirements of the applicable AMS specifications, 4071G, 4082H and 4086H, except for one low elongation value on the 2024-T3. The tensile strength requirements of WW-T-700/3C for 2024-T3 and /4C for 5052-0 and /6D for 6061-T6 were also met.

Relative tensile and compressive yield strength values between the two types of tubing (good and corroded) were:

1. 5052-0 aluminum, corroded, 6.4 percent lower tensile yield and 7.9 percent lower compressive yield strength.

2. 2024-T3 aluminum, corroded, .8 percent higher tensile yield and 6.6 percent lower compressive yield strength.

3. 6061-T6 aluminum, corroded, 5.1 percent lower tensile yield and 6.5 percent higher compressive yield strength.

Since different manufacturing lots of material were evaluated, this spread could represent normal variation between production runs.

Conclusions. Corroded specimens averaged about 5 percent lower yield and ultimate values than non-corroded specimens. This difference was too small to be considered significant especially in view of the fact that the corroded and non-corroded specimens came from different manufacturing lots.

Recommendations. No restrictions were recommended to be imposed on any of the three alloys from the standpoint of yield-ultimate physical property degradation. The 20 percent reduction in the 2024-T3 elongation was considered to be significant, and it was recommended that it be considered in the overall evaluation of the usability of 2024-T3 to prevent a degradation in reliability for certain parts due to their application.

Fatigue Tests - Rotating Beam

Specimens. Total specimens tested:

16	3/8 OD x .022	Corroded	6061-T6
12	3/8 OD x .022	Good	6061-T6
4	3/4 OD x .028	Corroded	5052-0
6	3/4 OD x .028	Good	5052-0
5	3/4 OD x .049	Corroded	6061-T6
9	3/4 OD x .049	Good	6061-T6
25	Total	Corroded	
27	Total	Good	

Procedure. All 52 specimens were tested using the cantilever beam method with free end rotary motion (MIL-F-18280B, para. 4.8.7.1).

Test results. The rotating beam test applies uniform stress around the circumference of the tube. Despite the fact that maximum stress was applied to areas with known defects, all specimens (27 good and 25 corroded) failed in the tube flare. There was no significant difference in the number of cycles to failure of the good and corroded specimens.

Conclusions. Existing corrosion was not sufficient to cause a tubing failure outside the flare area, and the corrosion damage in the flare area is insignificant and not sufficient to cause early fatigue failure.

Recommendations. No restrictions were imposed on 5052-0 and/or 6061-T6 tubing from the standpoint of fatigue as determined by the rotating beam test. No significant effect was expected on reliability of the parts from this aspect.

Fatigue Tests - Tension-Tension

Specimens. Total specimens tested:

6	3/4 OD x .049	2024-T3	Good
1	3/4 OD x .049	2024-T3	Good (MPG)
3	3/4 OD x .058	2024-T3	Corroded
3	3/4 OD x .058	2024-T3	Corroded
1	3/4 OD x .058	2024-T3	Corroded (MPG)
2	3/4 OD x .049	6061-T6	Good
2	3/4 OD x .049	6061-T6	Corroded
2	3/4 OD x .049	6061-T6	Corroded

MPG: With molded plastic on ends of specimen.

Procedure. Specimens were tested in a Baldwin SF-1U fatigue machine with a 5:1 multiplying fixture. Cyclic rate was 30 cps. The specimens were fitted with a rounded-tip, steel mandrel inside the tubing and a smooth collet outside on each grip end. This placed the tube walls in compression between the mandrel and the collet, affording a firm grip on each end of the tube. Each specimen was loaded to 24,000 - 4,800 - 24,000 psi stress (each wall thickness).

Test results. About 2/3 of the corroded 2024-T3 specimens tested failed at the junction of the collet grips and the test section. In contrast only one of the non-corroded 2024-T3 tubes failed at the collet. Although failures at the collet are undesirable and represent more severe stress conditions than found in normal fatigue tests, the results of the corroded vs. non-corroded are directly comparable since they were run identically.

Average Fatigue Life
2024-T3 Tubing
24,000 psi maximum
R = 0.2 Fatigue Stress

Corroded	288,000 cycles
Non-Corroded	1,578,000
Non-Corroded (1)	553,000

(1) With 0.003 deep surface imperfections from drawing during manufacturing.

Surface crack failures were typical fatigue failures, and in spite of failure originating at the collet interface, many fatigue cracks were present beyond the collet clamping area. Fatigue cracks did start from corrosion on the tube. The corrosion was of both types, intergranular and pitting corrosion.

Limited fatigue testing on 6061-T6 tubing was also conducted.

Conclusions. 2024-T3 fatigue life loss was:

82 percent corroded vs. non-corroded (good surface)
48 percent corroded vs. non-corroded (.003 surface imperfections)

No significant difference in corroded and non-corroded 6061-T6 tubing was observed. Alodining seemed to benefit the fatigue strength of the 6061-T6 corroded tubing.

Recommendations. Due to the significant percentage loss of the fatigue strength of the corroded 2024-T3 tubing, action was recommended to establish accept/reject criteria for the corroded 2024-T3 tubing to prevent a degradation in reliability where fatigue strength is a necessity.

Sea Shore Environment

Purpose. The purpose of this test was to represent an accelerated life test to determine if being exposed to extreme environments would result in corrosion which could affect the life expectancy of the part, thus reducing its reliability.

Procedure. A sixteen (16) month sea shore environment test was conducted by a leading aluminum company along the Atlantic Coast to determine the effect of a sea shore environment on aluminum tubing specimens with different types of finishes. Two groups of specimens were tested. One group was located 80 feet from the shore line and the other group 800 feet from the shore line. These specimens were left exposed to the environment for a 16 month period. There were multiple specimens of the three types of aluminum: (1) 2024, (2) 5052, and (3) 6061 at both the 80 and 800 feet levels. Each aluminum type at each level had three specimens with the following type of finish condition:

Bare
Alodine 1200
1200 + Epoxy
1200 + Epoxy + Acrylic
Anodized
Anodize + Epoxy
Anodize + Epoxy + Acrylic

Test results. With regard to the 800 feet level test group of specimens, only the bare aluminum and Alodine 1200 coating variations developed corrosion as a result of the 16 month exposure - both considered superficial. All other specimens were unattacked.

With regards to the 80 foot level test group of specimens pitting-type general corrosion developed on the bare control and Alodine 1200 coated tubes. The pitting on the 80 foot samples is more severe and general than on the 800 foot tubes previously reported. No other coating variation showed pitting.

Conclusions. These tests conclude that corrosion was arrested by the surface finish normally applied to the end product and such finishes should sustain the part during its normal life. Alodine 1200 is not normally used as a finish coat. After weeding out the parts recommended from the test program, those remaining were considered to be adequate to meet the reliability requirements and their finish coats should guard them from corrosion during their life expectancy.

CONCLUSIONS

In reviewing the factors leading to the corrosion condition and the subsequent investigation and corrective measures, a number of basic conclusions have been drawn. Following is a listing of these conclusions:

1. Storage location and environment are very important to prevent corrosion in aluminum raw material and should be kept pollutant free and under close surveillance.

2. Aluminum tubing should be adequately inspected upon receipt to assure that corroded material is not introduced into inventory, especially since it may be housed in warehouses for an extended period of time prior to delivery.

3. Precautions should be taken to protect aluminum raw stock while in inventory such as covering with a protective coating of oil and/or storing in polyethelene bags.

4. Some types of aluminum tubing such as 2024 may require special handling and storage due to the significant changes that corrosion has on its strength characteristics to prevent adverse effects on desired reliability.

5. Issues from stock should be inspected closely for corrosion prior to release to production, especially where the application is critical in nature.

DISCUSSION

H. O. Fuchs, Stanford University: Who found the corrosion first?

V. D. Matney: It was found by an inspector in the production area who was looking a little closer than inspectors normally look.

ROTOR BURST PROTECTION PROGRAM

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Introduction

A seemingly irreducible number of uncontained gas turbine engine rotor bursts occur each year in U.S. commercial aviation. The potential for catastrophy that can be associated with these events has prompted NASA to sponsor the Rotor Burst Protection Program. This program was developed and is being conducted by the Naval Air Propulsion Test Center in conjunction with the Massachusetts Institute of Technology. The basic goal of the program is to develop criteria and data for the design of optimum light-weight devices that can be used on aircraft to protect passengers and vital parts of the aircraft structure from the lethal and devastating fragments that are generated when a gas turbine engine rotor bursts.

The intent of this paper is to acquaint you with the RBPP by:

- Explaining what motivates us to pursue this effort.
- Describing the test facilities that were developed and are being used to implement the program.
- Presenting some of the experimental results that have been produced.

Motivation for the RBPP

Impetus and motivation for the RBPP stems from the statistics of the rotor failure situation or problem in U.S. commercial aviation. The data that will be presented on this problem comes from analysis of Flight Standards Service Difficulty Reports (SDR) that are published daily by the Department of Transportation, Federal Aviation Administration (F.A.A.).

It has been stated that an irreducible number of uncontained rotor bursts occur each year. In fact, the data shown in figure 1 indicates that on the average 32 uncontained rotor bursts occur each year. A more detailed presentation of the statistics on rotor burst for the year 1972 are shown in figure 2. Here, the data is presented in terms of what part of the engine was affected, and how many rotor failures and bursts occurred. A rotor burst being defined as a failure that produced fragments. These data indicate that 196 rotor failures were experienced in 1972. These failures accounted for approximately 7% of the 2854 shut downs that were experienced by the gas turbine powered U.S. commercial aircraft fleet. The data shown in figure 3 further characterizes the rotor burst problem by identifying: what types of fragments are being generated; where in the engine bursts occur; and what percentages of the bursts are uncontained. This type of data serves to establish where in the engine the burst problems exist and what type of fragment has the most potential for doing damage and therefore must be protected against. This in a sense directs our efforts so that the most critical aspects and locations of the rotor burst problem are addressed by the program. The conclusion that we've drawn from these and other more detailed data is that rotor burst in commercial aviation is a relatively sizeable problem with potentially serious consequences - at stake is the welfare and safety of literally thousands of airline passengers.

The question that we've addressed ourselves to is this: What can be done to minimize or eliminate the hazards and risks that are attendant to rotor burst? As we see it there are two basic alternatives: The level of safety that is needed can be achieved either through improved reliability or protection. We've chosen to pursue that goal which involves developing methods of providing light-weight protection. This position has been adopted, because the statistics of rotor burst show that some limit to reliability has been reached and we can expect to experience some minimum number of rotor failures each year. And, in order to ensure safety some measure of protection must be provided.

Test Facilities for the RBPP

From the outset, it was recognized that extensive experimentation and testing would be needed to meet the goals of the RBPP. Meeting these goals would involve:

- Characterizing the burst and fragment control processes.
- Conducting parametric studies to establish functional relationships between significant process variables.
- Evaluating the effectiveness of various rotor burst protection devices and configurations.

To accomplish these tasks a Rotor Spin Facility was designed and constructed at the NAPTC. This facility is shown in figure 4. It consists of a control and data acquisition area which houses the controls and instrumentation used for test; and a test area which contains the spin chambers and auxiliary equipment such as the vacuum and lubrication pumps.

- The facility has two spin chambers. The smaller chamber, which is shown in figure 5 can accomodate rotors up to 40 inches in diameter and has a working height of 32 inches. The large chamber, shown in figure 6, is comprised of a heavy walled (1 inch thick) vacuum vessel that is protected by a 5 inch thick laminated steel inner liner. The working space in this chamber is 10 feet in diameter with a height of 6 feet. It was designed to accomodate rotors from the largest aircraft engines that are made. This chamber has ports on its walls for instrumentation feed-thru and optical access. In a normal test situation the rotor to be burst is suspended vertically from an air turbine drive motor which mounts on the chamber lid as is shown. A family of air turbine motors are available to produce rotor speeds up to 150000 rpm. To minimize the power required to accelerate the rotors to failure, the chambers are evacuated to produce a vacuum of approximately 5 mm Hg. The main data acquisition systems used for test, aside from those used to monitor facility operational variables such as rotor speed, chamber pressure and the like, are the impact strain measuring and high-speed photo- instrumentation systems. The strain measuring system which consists of balanced bridge millivolt signal conditioning equipment and two dual beam oscilloscopes, is used to measure and record the strains that are induced in the containment and deflection devices as a result of rotor fragment impact. The high-speed photo system is comprised of a continuous framing camera and photo lighting unit. The camera is capable of producing 225 pictures as a framing rate of 35000 frames per second. The lighting unit has an output of 12 million beam candlepower.

These are just some of the salient features of the Spin Facility.

The Development of Design Criteria

The experimental development of rotor fragment protection design criteria, which is what this program is all about, has progressed through two distinct phases. The first phase involved the conduct of exploratory tests that were performed to determine what mechanisms were involved in the rotor fragment containment and deflection processes; and also to establish what variables significantly influenced these processes. The second and current phase involves the conduct of systematic experimentation to generate data for the design of fragment containment rings. The results of some of the earlier exploratory tests are shown in figures 7 through 12. These results are in the form of high-speed photographs that depict the ring and fragment behavior and interactions that result from the impact phenomena and processes associated with rotor fragment containment and deflection.

The systematic rotor burst containment experimentation evolved from the results of the exploratory testing, which provided insight and an understanding of how certain variables significantly influenced the rotor fragment containment process. From this information a program was developed that aims at establishing a functional relationship between a measure of containment ring capability and the significant variables that are involved in the containment process. These relationships, once established, constitute the criteria needed for design. The significant variables for fragment containment, as determined through experimental observation are:

Rotor:	Diameter	Speed
	Length	Material
	Tip-to-hub-ratio	Number and type of fragments
	Weight/inertia	

Ring: Diameter
Radial thickness
Axial length
Material

For purposes of developing the design criteria and the test scheme needed to generate this criteria, a dependent variable was formulated which measures the containment capability of the ring. This variable was called the specific contained fragment energy (SCFE) and is derived by dividing the rotor energy at burst by the weight of the ring required to contain this fragment energy. This variable is a combination of several of the significant variables, namely: The rotor mass inertia and speed; and a gross description of the ring in terms of its weight. The remaining significant variables become independent or experimental variables; that is, factors that are varied from test to test to determine what influence these variations have on the containment capability of a ring.

Conceptually the relationship and therefore the design criteria developed through experiment would take the form shown in figure 13. Here the SCFE is plotted against the number of pie sector fragments generated at burst and the ring/rotor diameter. The rotor to ring axial length ratio is the parameter. The relationships between these variables are being derived experimentally by varying the: Rotor diameter; Number of fragments; Ring axial length; and Ring radial thickness. These relationships, when fully established, will provide all of the information that is needed to design an optimum weight ring for the containment of rotor burst fragments.

THE INCIDENCE OF UNCONTAINED ROTOR BURSTS IN U.S. COMMERCIAL AVIATION

1962 - 1972

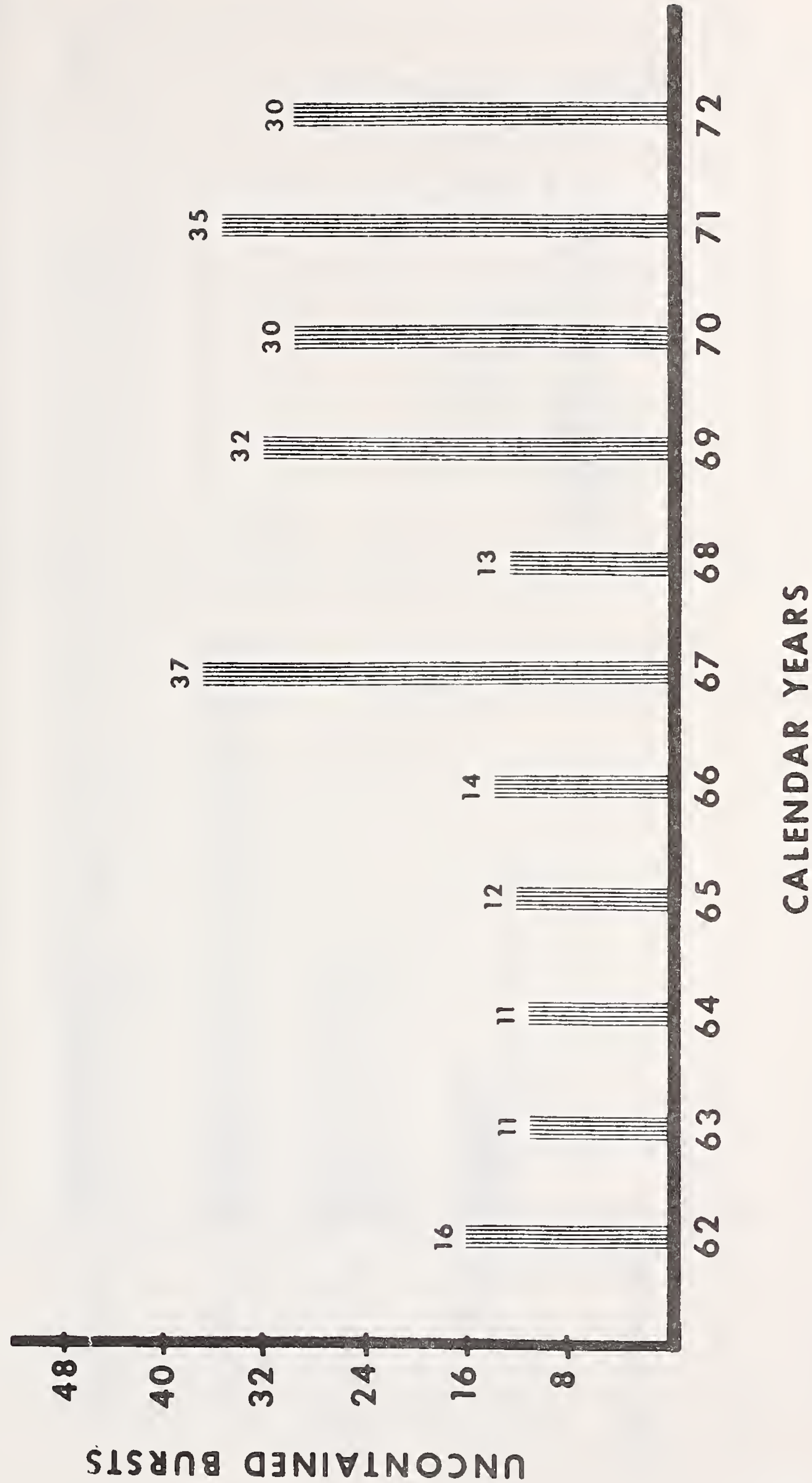


FIGURE 1

INCIDENCE OF ROTOR FAILURE/BURST IN U.S. COMMERCIAL AVIATION 1972

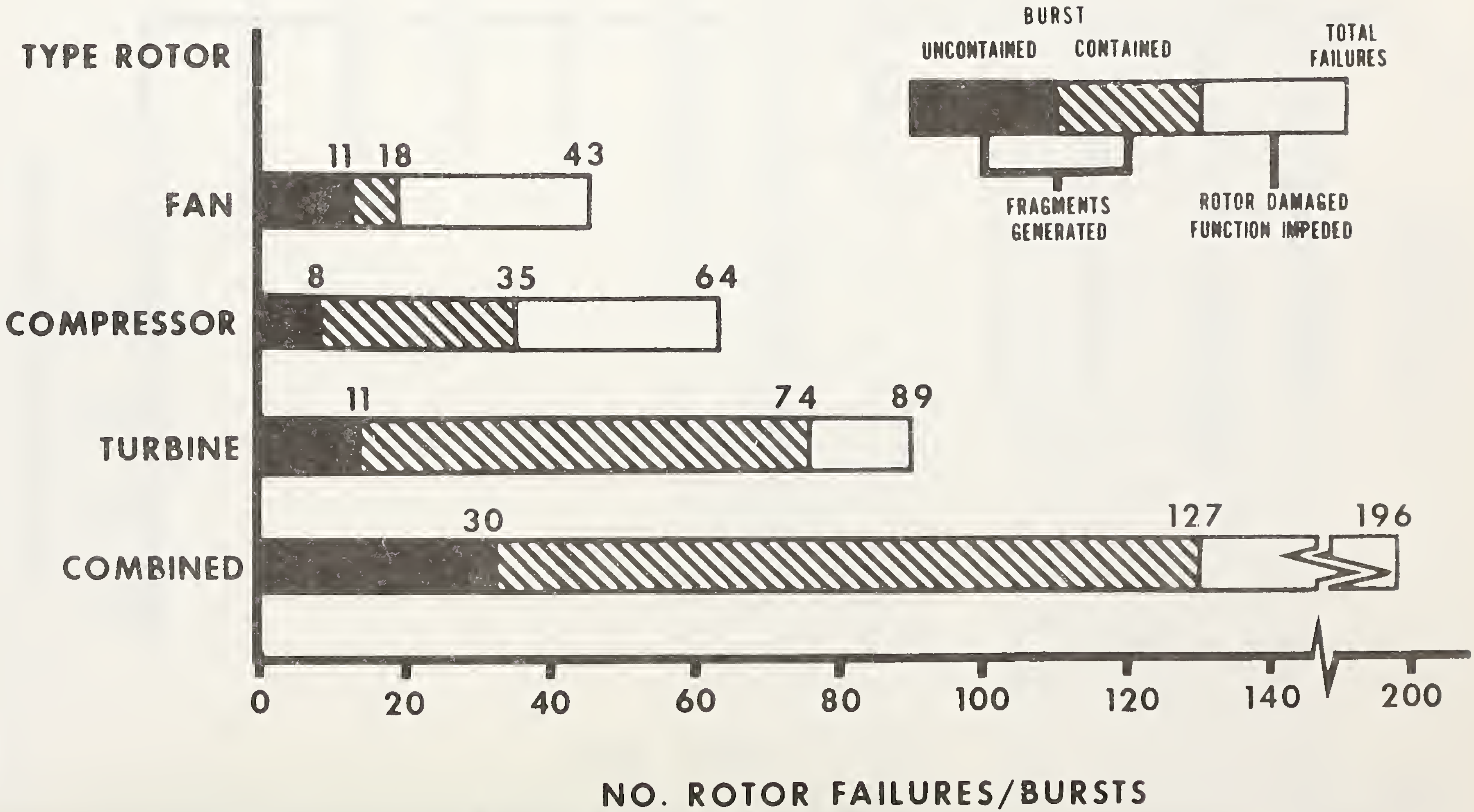


FIGURE 2

COMPONENT AND FRAGMENT TYPE DISTRIBUTIONS FOR CONTAINED AND UNCONTAINED ROTOR BURSTS[1] - 1972

ENGINE ROTOR COMPONENT	TYPE OF FRAGMENT GENERATED						TOTALS	
	DISK		RIM		BLADE			
	TF	UCF	TF	UCF	TF	UCF	TF	UCF
FAN	1	1	0	0	17	10	18	11
COMPRESSOR	2	2	0	0	33	6	35	8
TURBINE	2	2	1	1	71	8	74	11
TOTALS	5	5	1	1	121	24	127	30

(1) FAILURES THAT PRODUCED FRAGMENTS

TF - TOTAL FAILURES

UCF - UNCONTAINED FAILURES

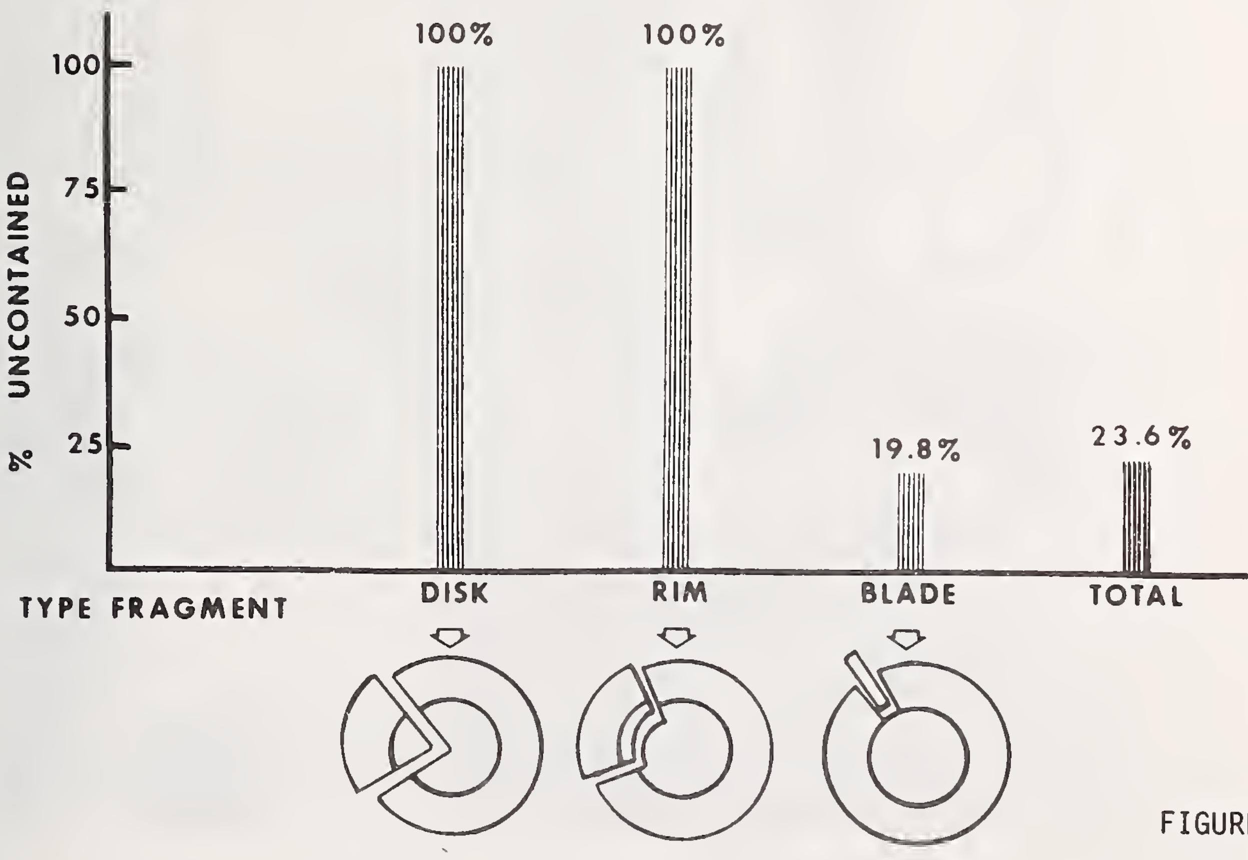
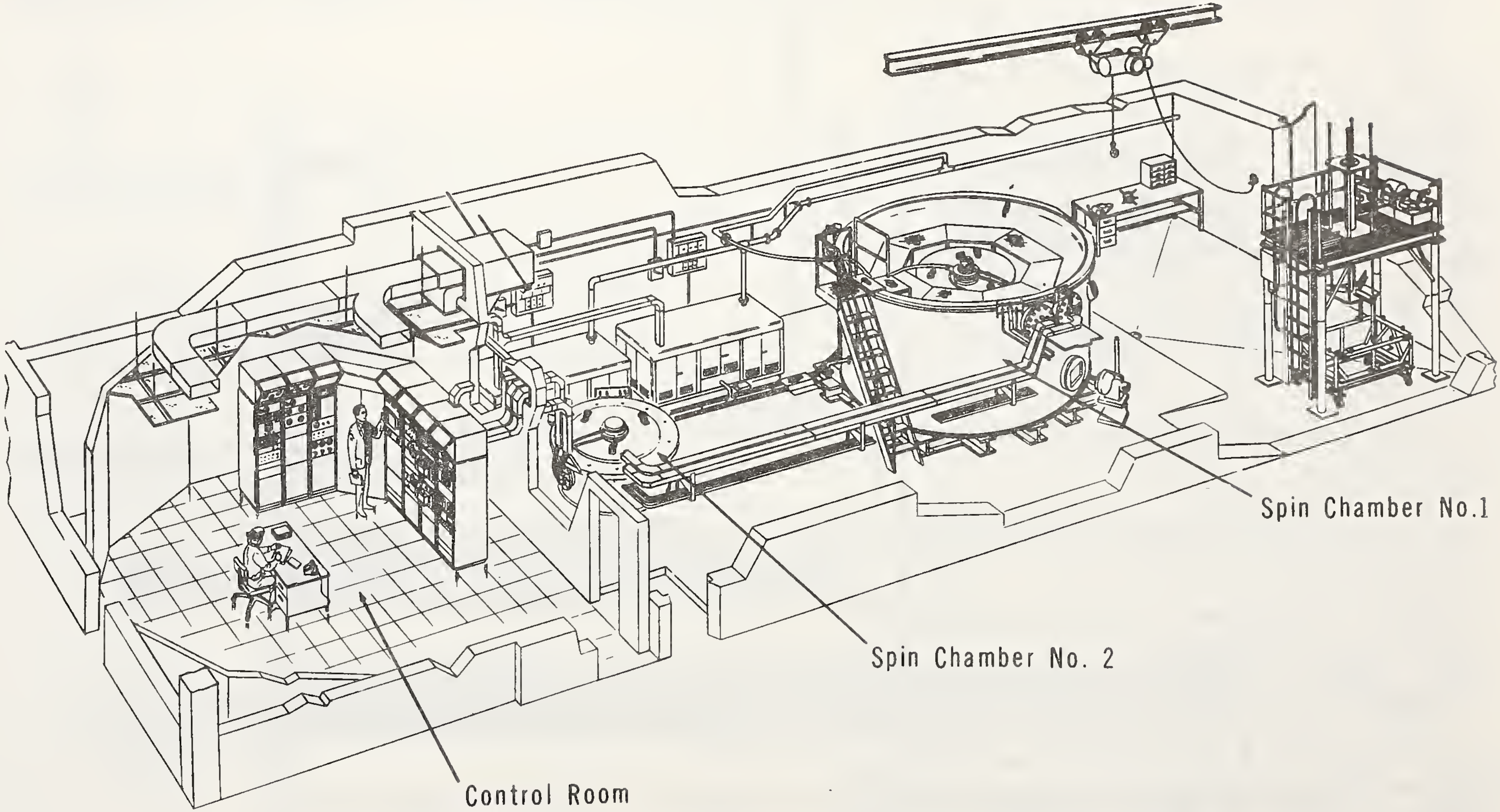


FIGURE 3



CONTAINMENT EVALUATION FACILITY

Figure 4

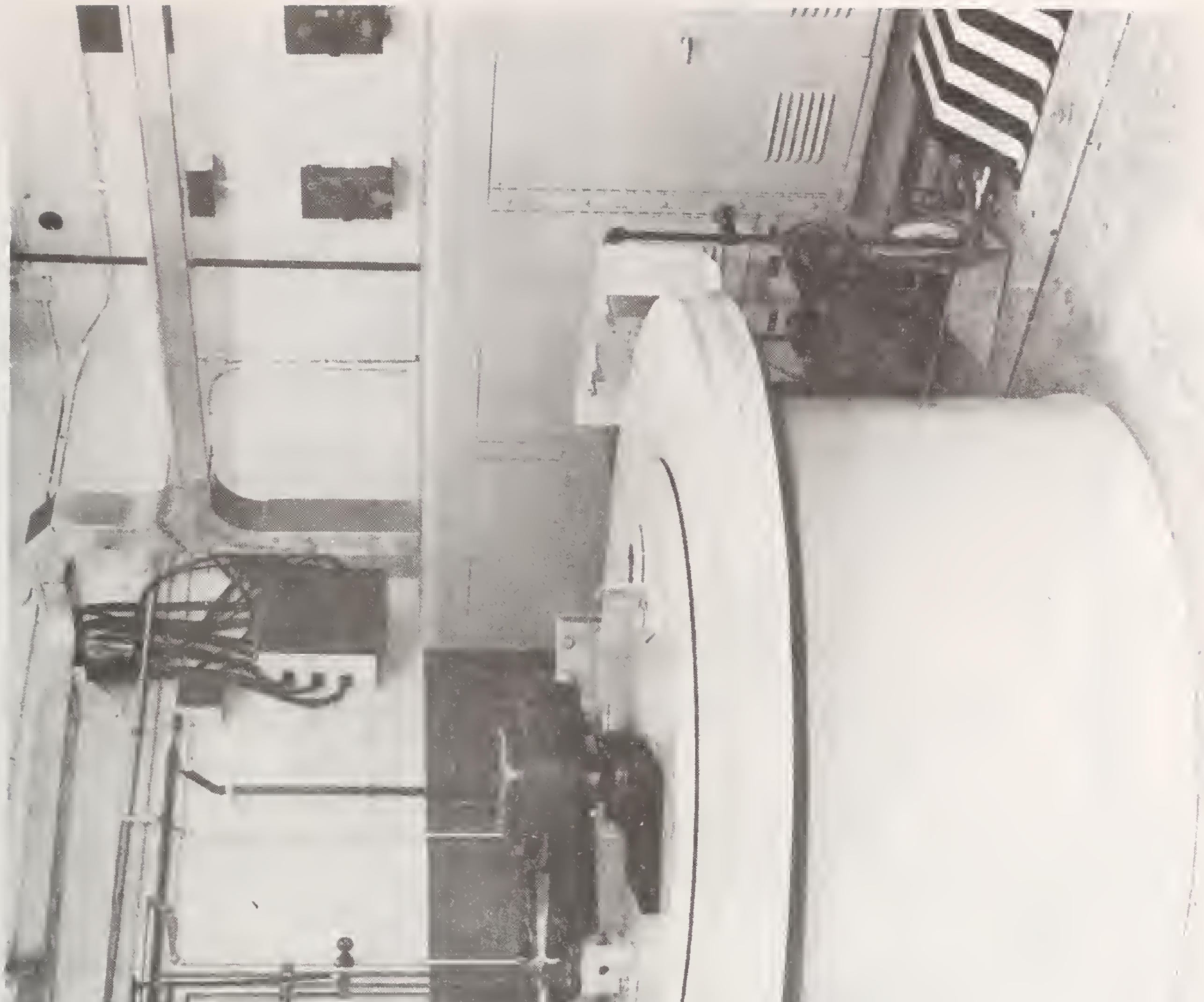
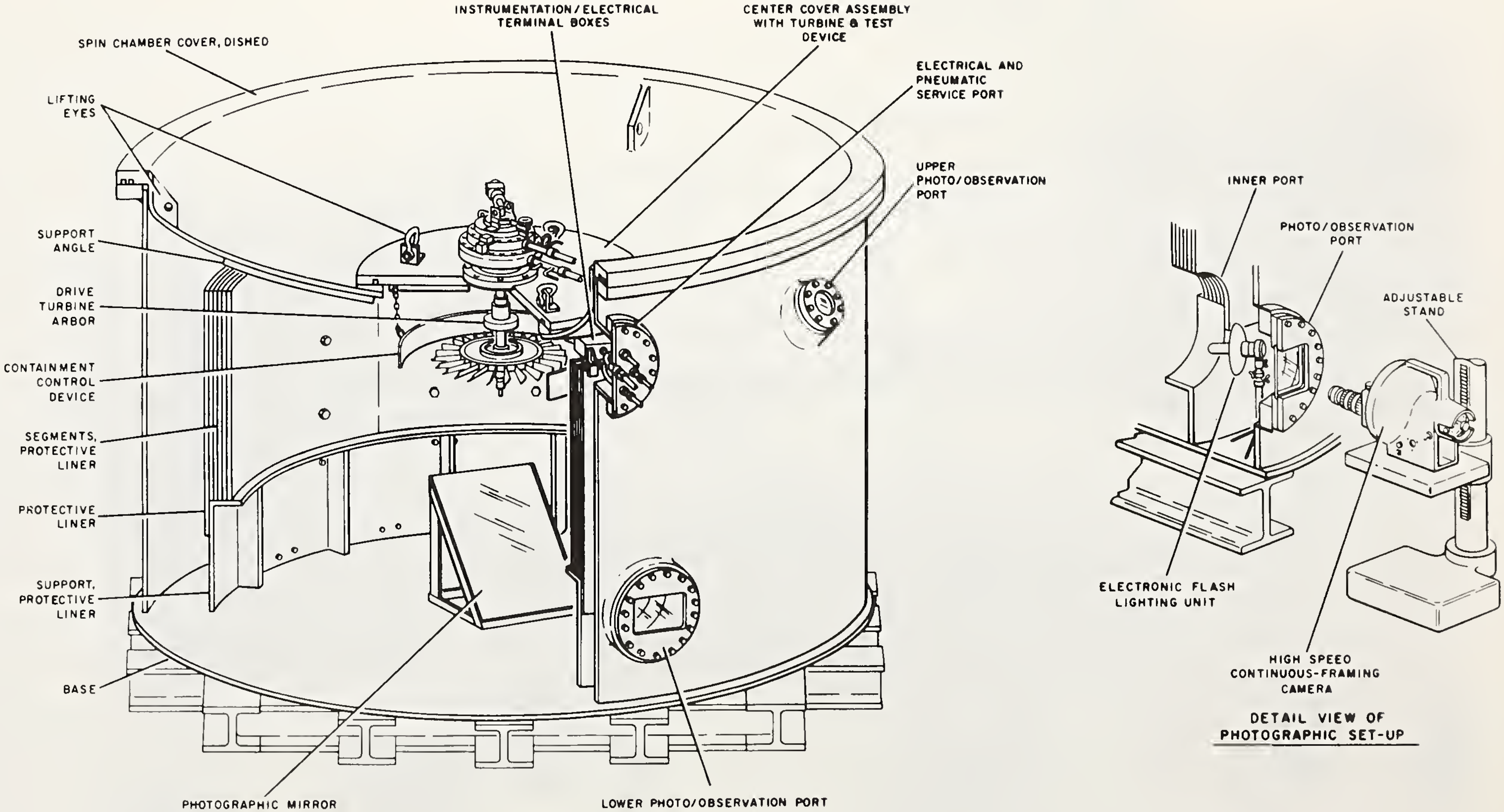


FIGURE 5

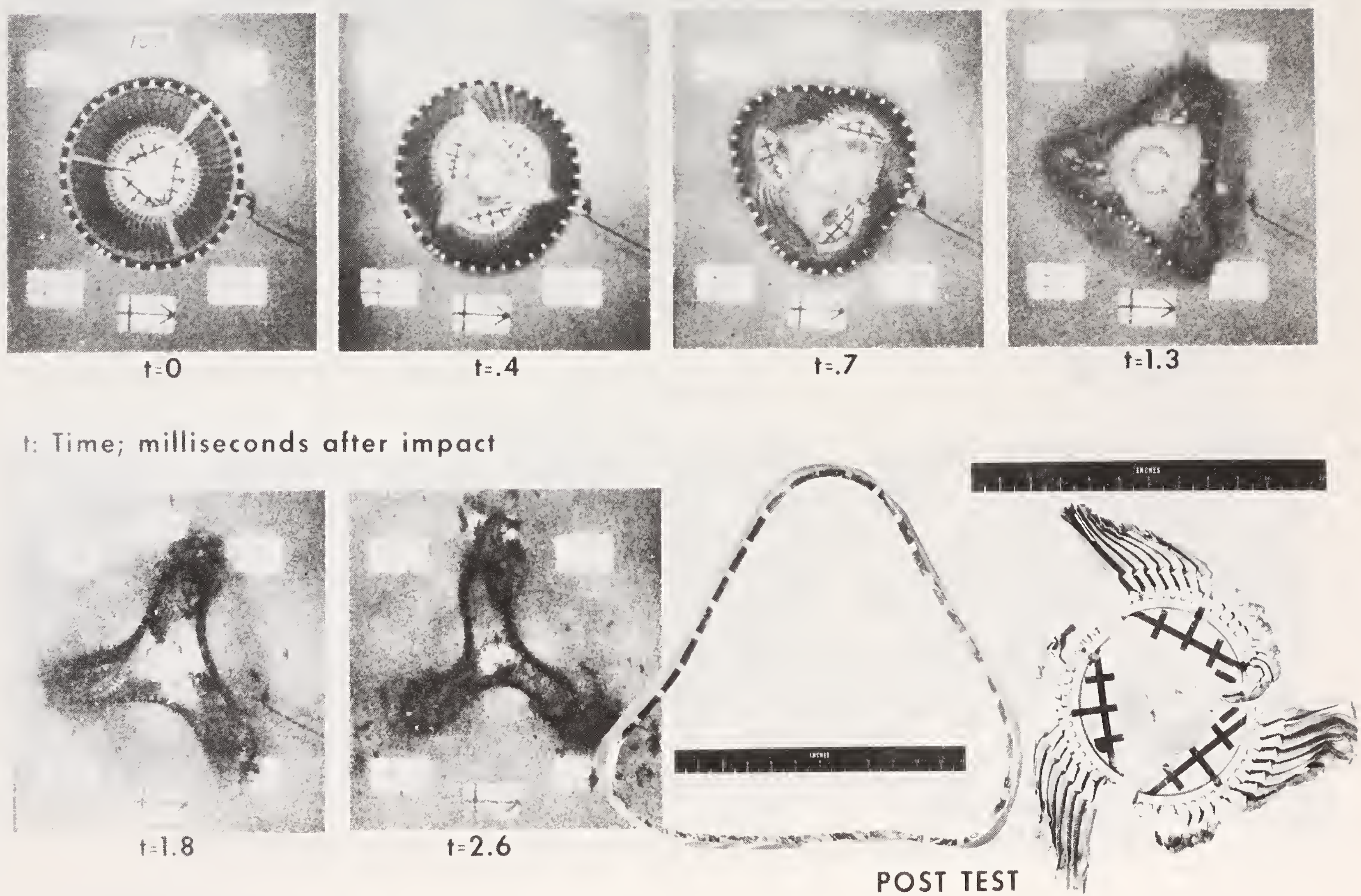


NAPTC Spin Chamber No.1

FIGURE 6

EXPERIMENT 100 -

3 FRAGMENT ROTOR BURST INTO A TRIP STEEL RING

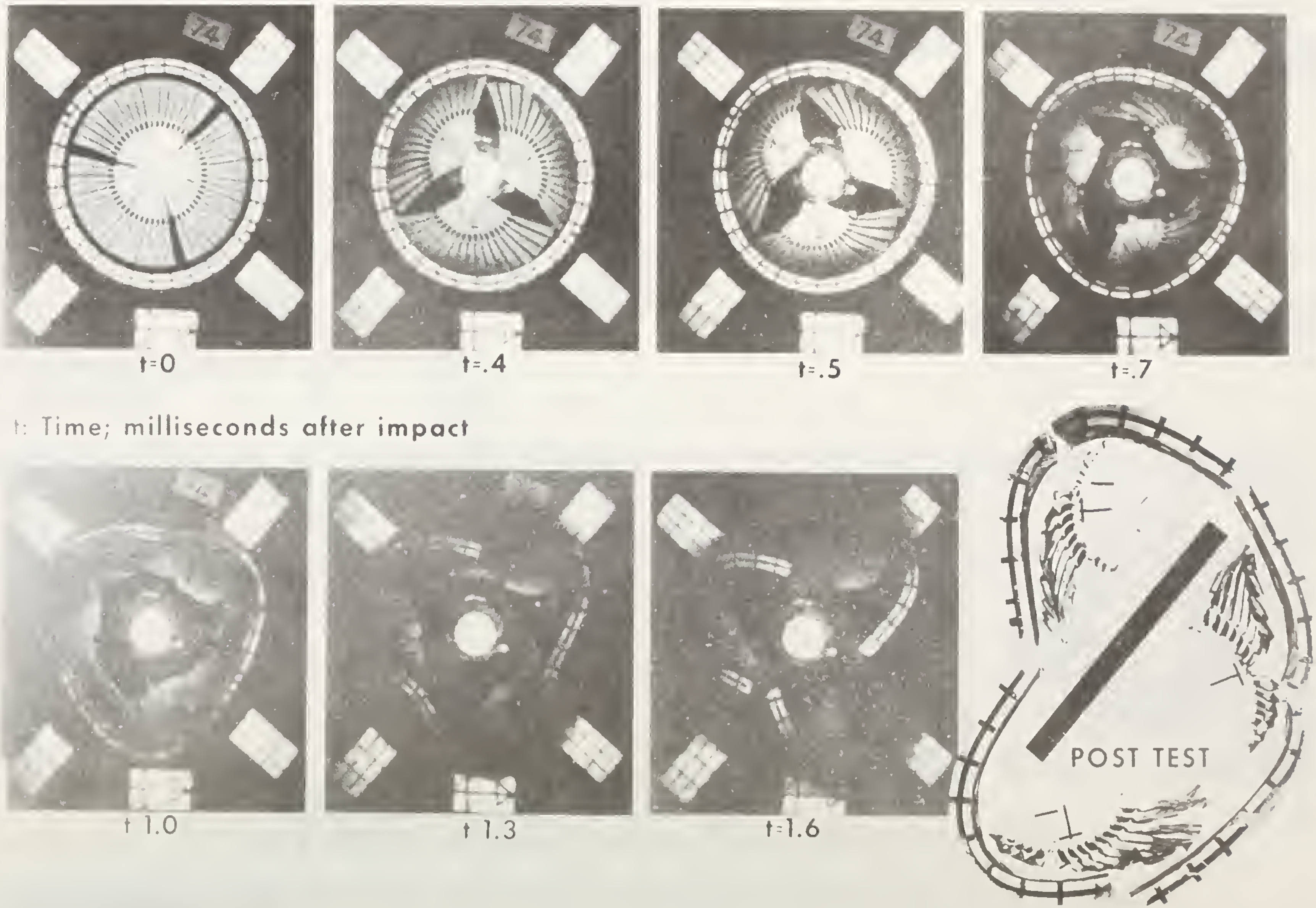


t: Time; milliseconds after impact

FIGURE 7

EXPERIMENT 74 -

3 FRAGMENT ROTOR BURST INTO A 2024-T4 ALUMINUM RING

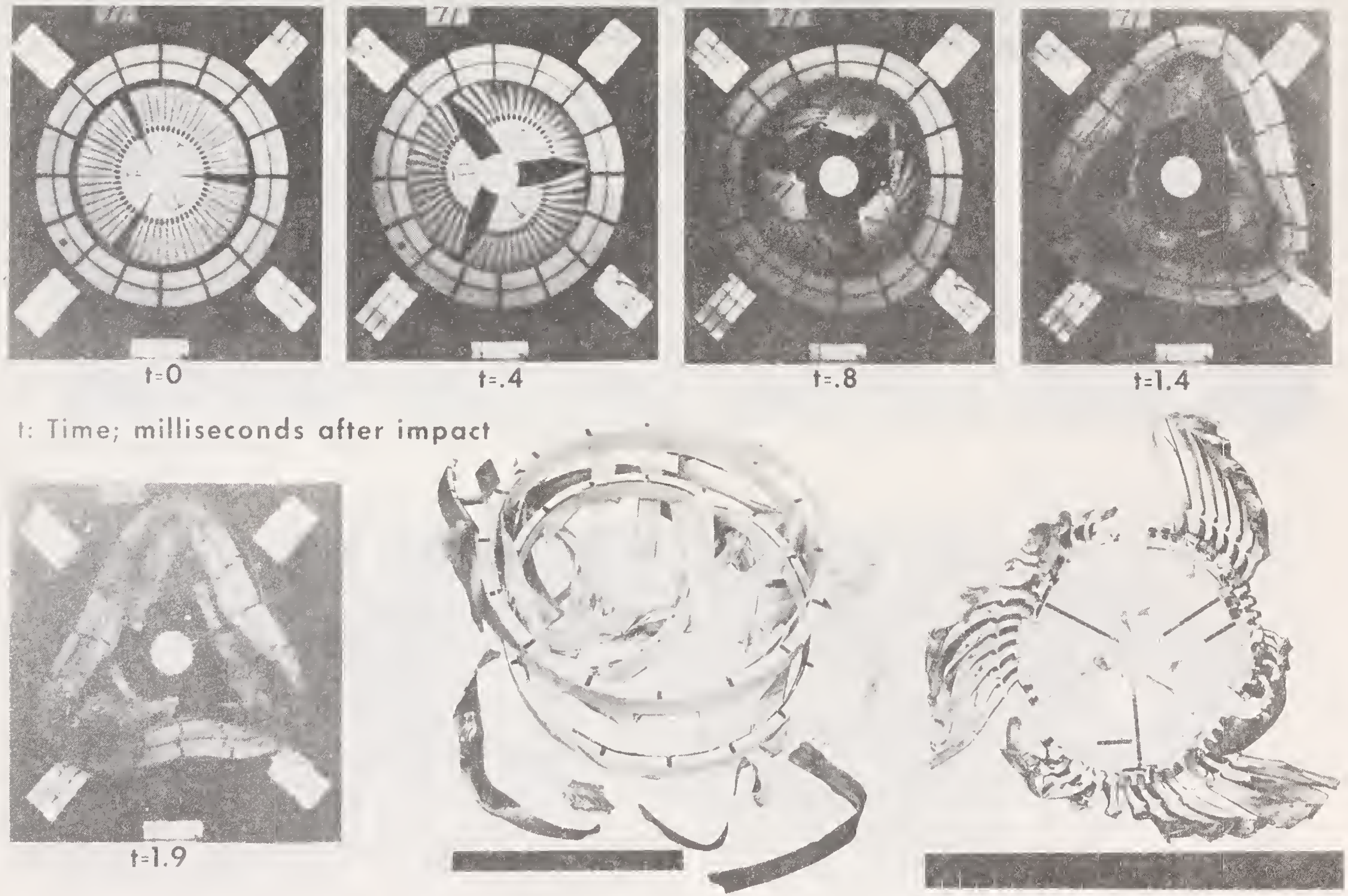


t: Time; milliseconds after impact

FIGURE 8

EXPERIMENT 71 -

3 FRAGMENT ROTOR BURST INTO A BALLISTIC NYLON WITH STEEL LINER RING



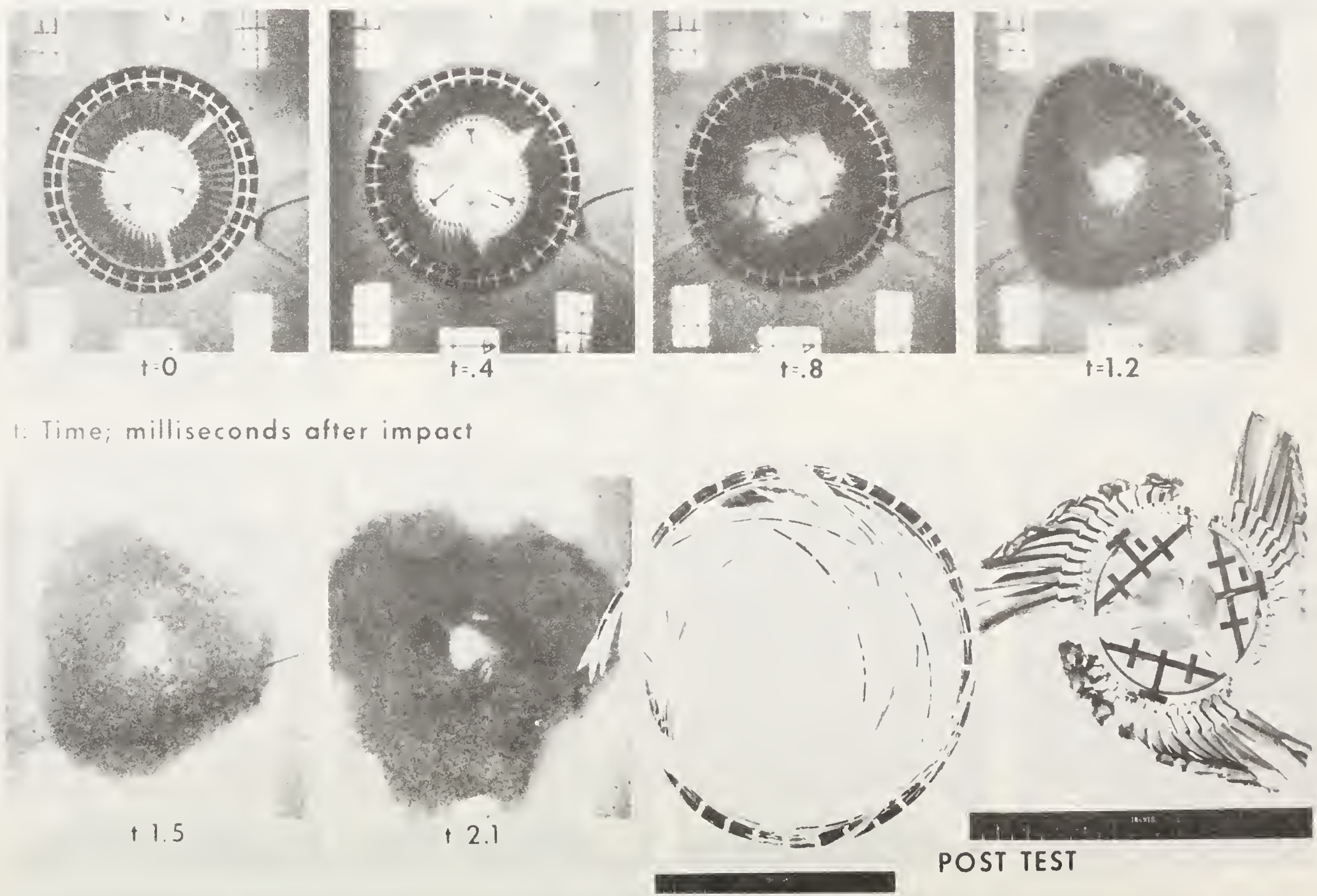
t: Time; milliseconds after impact

POST TEST

FIGURE 9

EXPERIMENT 86 -

3 FRAGMENT ROTOR BURST INTO A FILAMENT WOUND FIBERGLASS RING



t: Time; milliseconds after impact

FIGURE 10

EXPERIMENT 109 -

SINGLE BLADE BURST INTO A 6061-T6 ALUMINUM RING

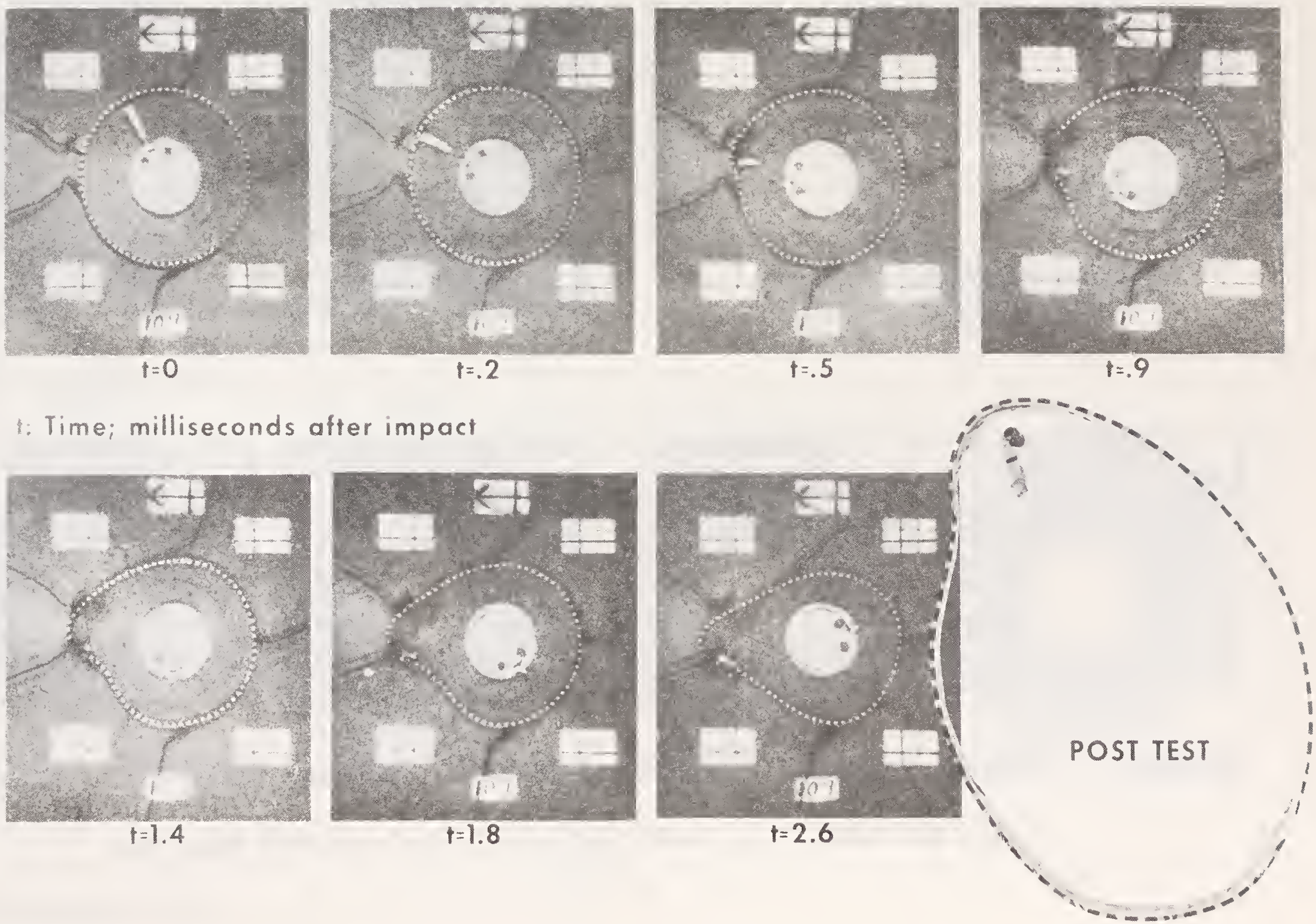


FIGURE 11

EXPERIMENT 114 -

ROTOR BLADE BURST INTO A 6061-T6 ALUMINUM RING

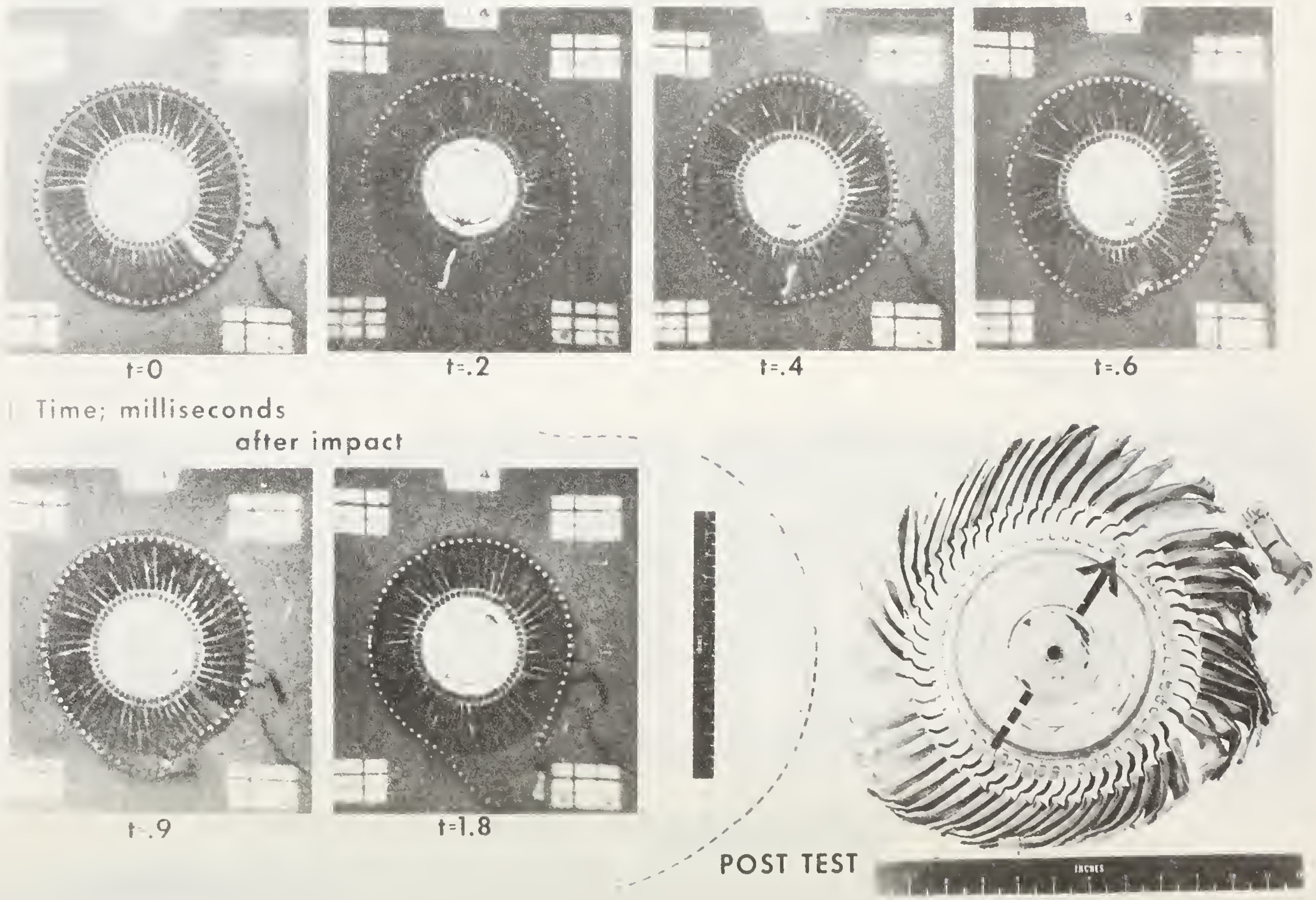
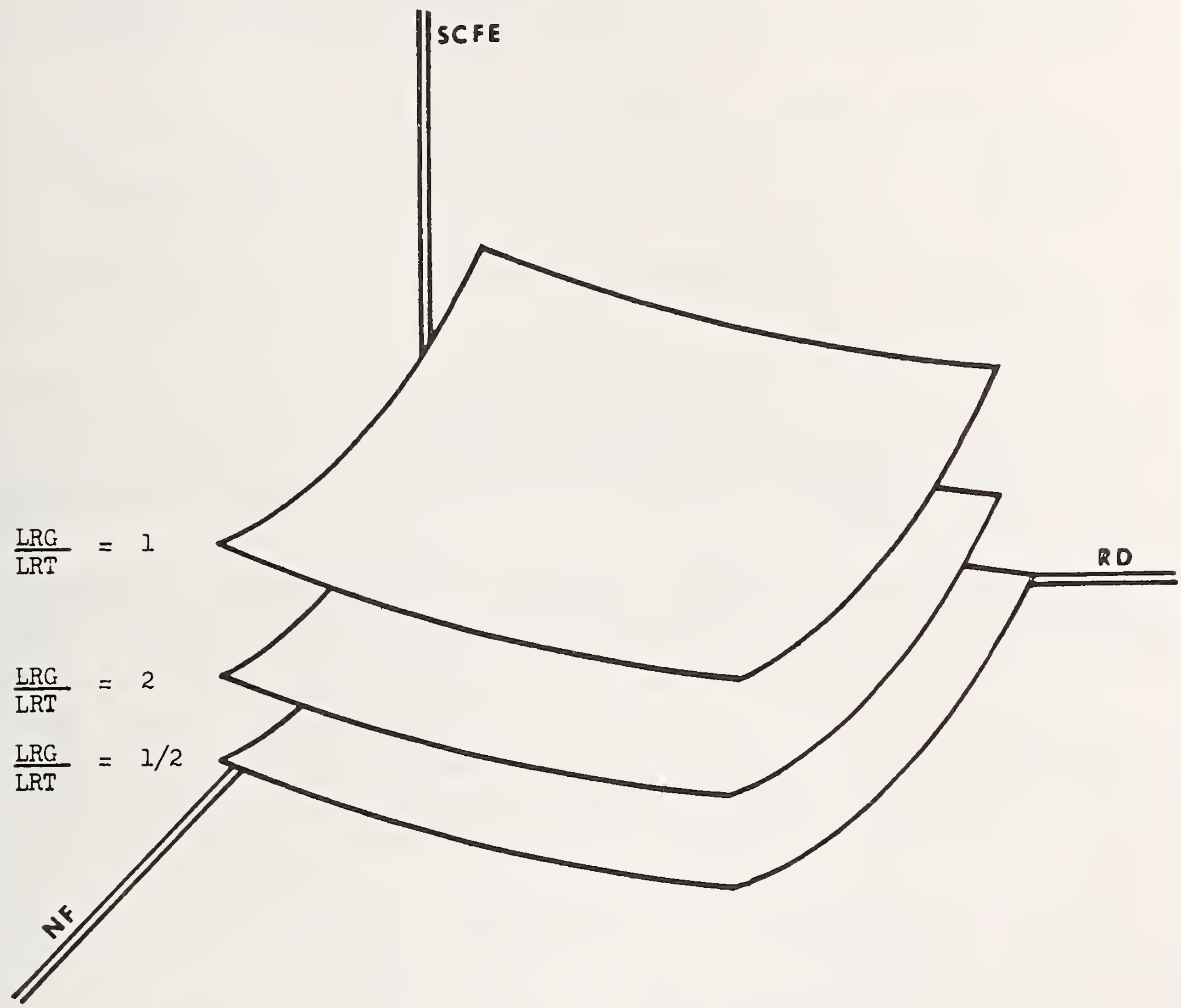


FIGURE 12

SCFE Specific Contained Fragment Energy
RD - Ring (Rotor) Diameter
NF - Number of Fragments
LRG Ring Axial Length
LRT Rotor Axial Length



ROTOR BURST CONTAINMENT, FUNCTIONAL RELATIONSHIP, OR DESIGN CURVES (CONCEPTUAL)

FIGURE 13

DISCUSSION

J. J. Scialdone, NASA, Goddard Space Flight Center: Do you simulate the temperature?

G. J. Mangano: No, we are not considering temperature. Later on in the program we will examine the effect of temperature. Right now, we are just trying to develop information that will give us a gross idea of what is involved in providing containment protection for these aircraft engines.

J. J. Scialdone: Would it change the energy at first?

G. J. Mangano: It won't change the energy at first, but it will probably change the behavior of the rings or devices that we use.

H. S. Link, U. S. Steel Research Laboratory: Are there any systems for containment?

G. J. Mangano: Yes, for blade containment. Containment is required for blade failures in most commercial engines and in some military engines. But close to 20% of those types of failures are uncontained despite FAA regulations. That is why we are alarmed and conducting a program such as this.

R. Lenich, Caterpillar Tractor Company: Was any previous work done by turbine manufacturers in regard to containment or was this the first effort?

G. J. Mangano: This is probably the most comprehensive effort, but it is not the first. The nature of the aircraft engine manufacturing business is such that when a problem develops, only that specific problem is treated. Therefore, no generalized data are available for design.

R. Lenich: Are you suggesting that action be taken to prevent the wheel burst in the first place?

G. J. Mangano: In some cases, that is done. In other more complicated cases where the problem cannot be solved through design, containment protection must be provided.

G. Wagner, Westinghouse Electric Corporation: Were there any surprises in the evaluation concept from MIT?

G. J. Mangano: I just presented the experimental aspects of this program. In fact, MIT has a very ambitious structural dynamics program, and hopefully at the end of our efforts, their theoretical programs will become the design criteria used by the engine manufacturers.

W. R. McWhirter, Jr., Naval Ship Research & Development Center: Did you implant the faults?

G. J. Mangano: Yes, in these experiments we are examining the effect of the number and shape of the fragments. So we do implant faults so that the rotors fail in the desired manner.

W. R. McWhirter, Jr.: Do you document what kind of fault you implant?

G. J. Mangano: Yes.

W. R. McWhirter, Jr.: Have you done any work in trend analysis for failure prediction?

G. J. Mangano: We do collect data on the causes of failures and when in the flight envelope they occur.

P. Hallick, Federal Aviation Administration: Insofar as the monitoring is concerned, when the disk failure occurs, it develops so rapidly that we do not have the real capability for detecting incipient failure. Failure is almost instantaneous. Our approach has been in design and testing to limit the service life of the disk to a number of cycles, hours, etc. At this time, the disk is removed from service and sent back to the manufacturer for further testing to determine the remaining life.

AIRFRAME "CRASHWORTHINESS" EXPERIMENTS

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When technical accuracy, reliability, and human performance can be adequately monitored to detect degradation before final failure, then there is a basis to prevent accidents, and in airplane design or operation, the airframe is called safe. We believe there is a concomitant need, namely a need to initiate remedial action. That is: The degradations, the failures need to be assessed collectively so as to change the design standards, where necessary.

For the most part, aviation safety efforts are concentrated on assuring successful flight through the design and construction of airplanes, airports, and navigational aids, and the day-to-day operation and maintenance of them. The airworthiness and safe operation of the airplane are related primarily to it physically. Going beyond hardware integrity and safe operation, however, is the consideration of assuring the protection of those onboard the airplane, the passengers and crew, once degradation becomes a real possibility.

Aviation safety, therefore, is aimed at a twofold objective: First, and primary, safety of the airplane relative to the operating environment, and second, safety of the occupants relative to the airplane.

The integrity of the airplane, its reliability, maintainability, performance, and factors of cost and operation are considered in the design development from the outset. In recent years, more attention has been given to the protection of persons onboard the airplane from the minor or so-called "survivable" crash landing environment. This has been a direct result in large measure to the overall assessment of accidents.

Changes in Federal Regulations have stemmed from the results of the investigations of accidents, research by both industry and government, and recommendations from many interested groups. Amendments FAR 25-32 of May 1972 reflect crashworthiness standards that were the result, in large part, of a major government/industry research program.

The present wide-body transport airplanes--B-747, DC-10, and L-1011--were designed to these upgraded standards. These airplanes reflect many design improvements, particularly, in the cabin interiors. The remodeling or "new look" of the older narrow-body transports, such as the B-707, DC-8, B-727, and DC-9, reflect these improved interiors. The success of these new designs is the result of many factors, such as improved materials. There is one area of improvement which is being verified by tests at the NASA Johnson Space Center, specifically, full-scale aircraft cabin flammability tests of improved fire-resistant materials. These tests are outlined in NASA Technical Memorandum X58141, June 1974.

As background it should be pointed out that for many years, tests for flammability standards for cabin materials remained relatively unchanged, calling for a simple horizontal bunsen burner test, with a maximum burn rate of 4 inches per minute. This was intended primarily to get the extremely fast burning materials out of the cabin and to prevent the start of a major fire from common minor cabin sources, such as dropping a lighted cigarette.

The work started in the mid-1960's, however, aimed far beyond this simple objective. It began to view cabin materials as a source of potential fuel in a major cabin fire, and sought to assess the fire hazards of this potential from the standpoints of not only flammability, but of smoke and toxic gas emission as well, all of which affect passenger survival. The work was concentrated initially on the primary hazard of flame propagation, complemented by work on the secondary hazards of smoke and gas emission. Flame was considered to be the main threat, since it in itself is lethal and consumes life supporting oxygen within the cabin. This priority on flame not only sought directly to reduce a primary hazard, but to reduce the secondary hazards as well by reducing a major gas generating heat source which might be present in a cabin fire, the burning cabin materials themselves. Full-scale cabin fire tests conducted by NASA have confirmed that this was a prudent action, and that the newer fire retardent materials, by virtue of their overall lower potential for fire spread, do give the cabin interior a lower potential for smoke and gas emission during the first few minutes of a cabin fire.

Several graphs in the NASA report show how the pre-1968 materials generated heat, smoke and carbon monoxide, and correspondingly depleted oxygen more rapidly and significantly than the newer materials which are available. They point out that the use of the improved materials provides additional safety during aircraft cabin fires and that substantial ignition sources are required to ignite such improved materials. These tests are continuing.

Another program of interest concerns the mechanical failure concept from an airplane structural point of view. It is a general joint crashworthiness program. One problem has been the use of static design criteria for crash conditions. In actuality, crash conditions are dynamic, and survivability depends on the dynamic response of the structure in relation to the occupants. Another problem is that mathematical methods used in dynamic analyses generally are in the elastic range but only marginally valid beyond the elastic range.

In order to develop the techniques for treatment of dynamic structures in both elastic and plastic ranges, a joint FAA, NASA, and GAMA (General Aviation Manufacturers Association) program has been established which encompasses the following phases:

1. Man-seat Model - Dynamic Criteria
2. Airplane - Math Model
3. Analytical Technique - Computer Models

The program seeks to achieve a means of realistically predicting failure modes in the early design stages of an aircraft so that design variations may be studied to optimize the structure for crashworthiness. The mathematical modeling will result in programs that will take into account the plastic deformations of structural elements and the redistributions of internal loadings which result from these deformations. The programs will lead to choices of materials and structural shapes that will most efficiently dissipate the kinetic energy of motion, thus protecting the occupants as far as possible from severe decelerations.

Development of this program requires knowledge of force/time inputs to the structure from the impact. Thus the effects of resistency of the impacting surface becomes part of the problem. It also requires the determination of peak shock waves through the structure from the points of impact of the occupants. To obtain and verify this information, full-scale crash tests are being conducted at NASA.

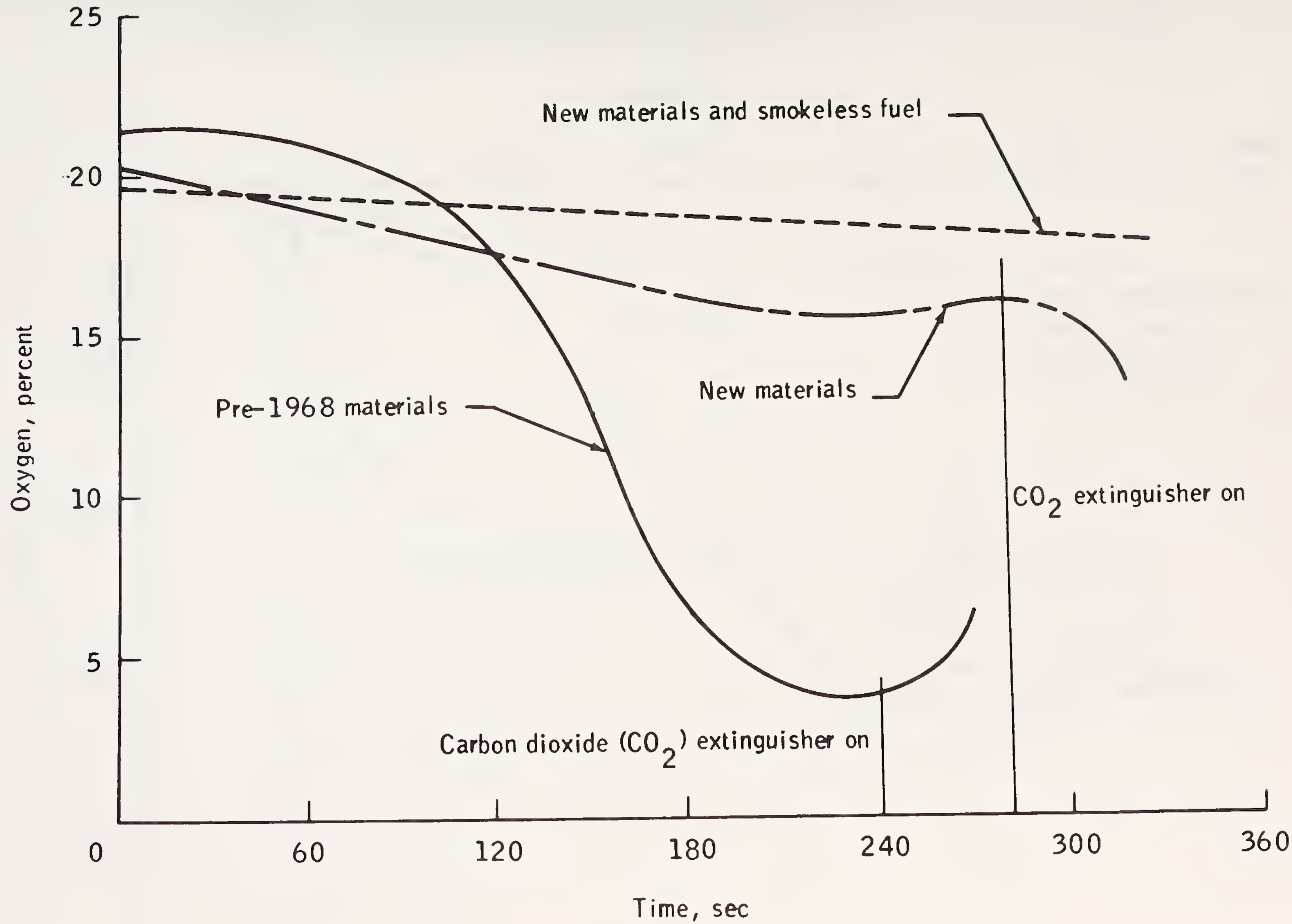

Figure 1. Oxygen Concentration.

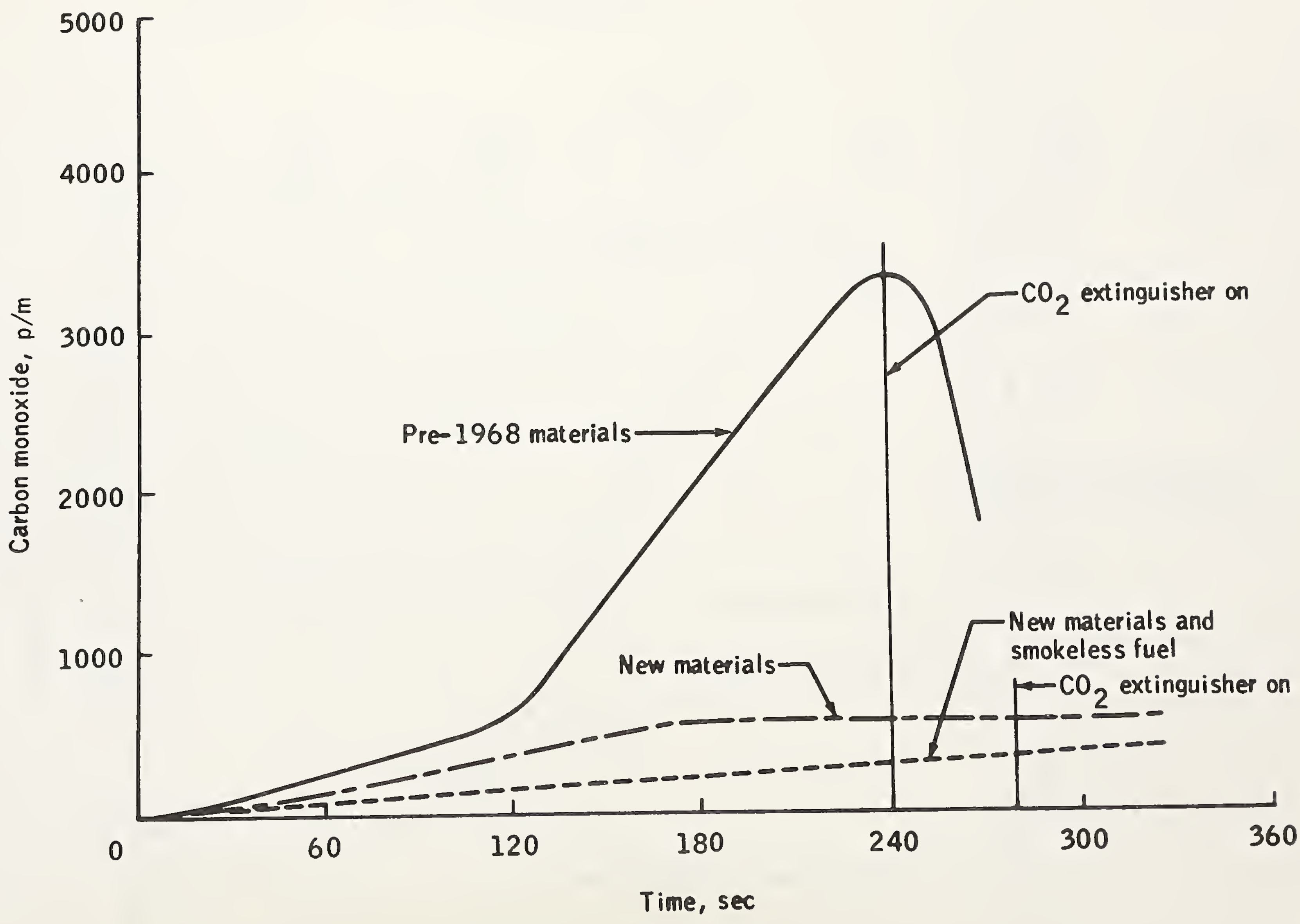

Figure 2. Carbon Monoxide Concentration.

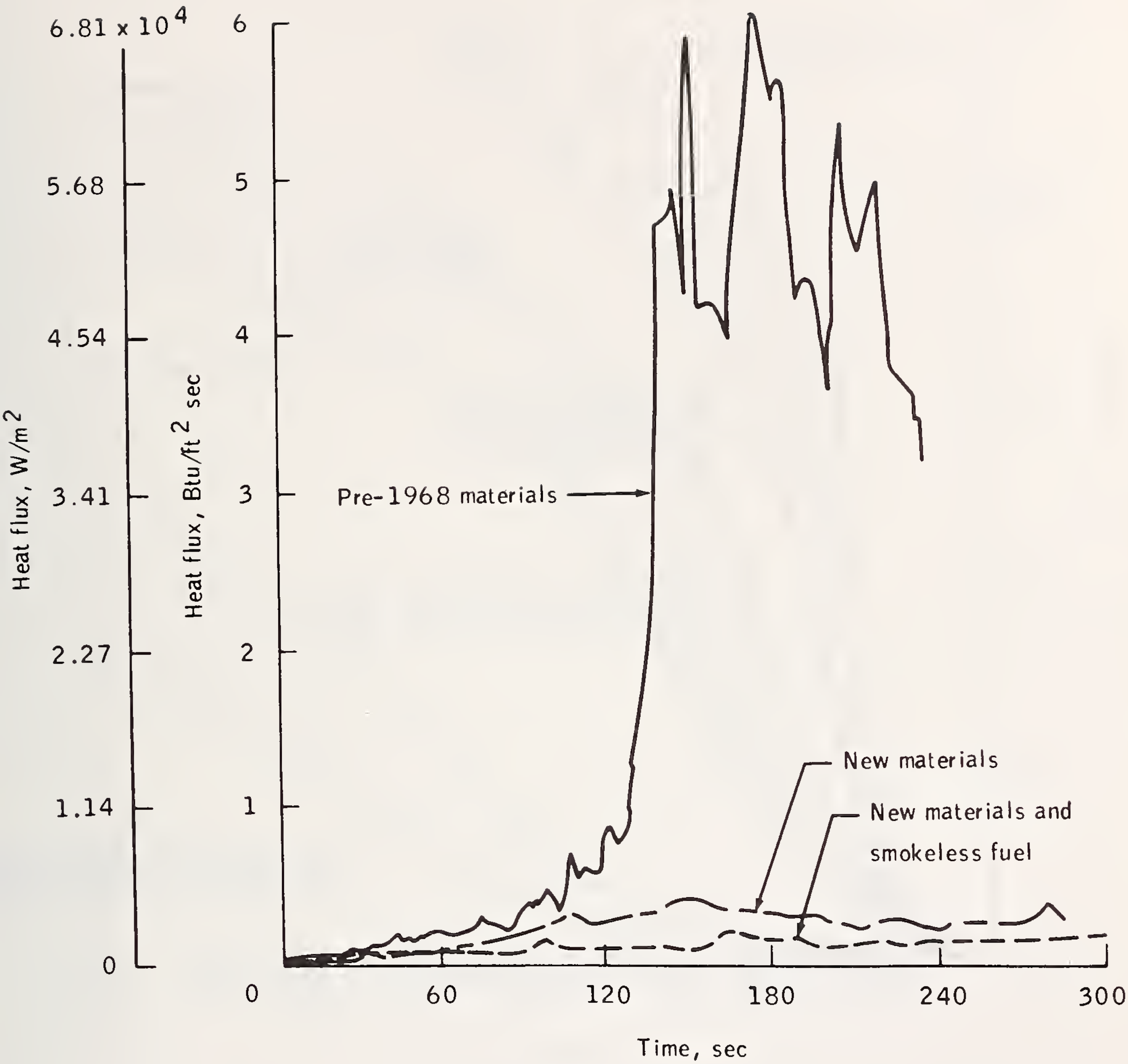

Figure 3. Heat Flux at Center of Test Section, Center Aisle.

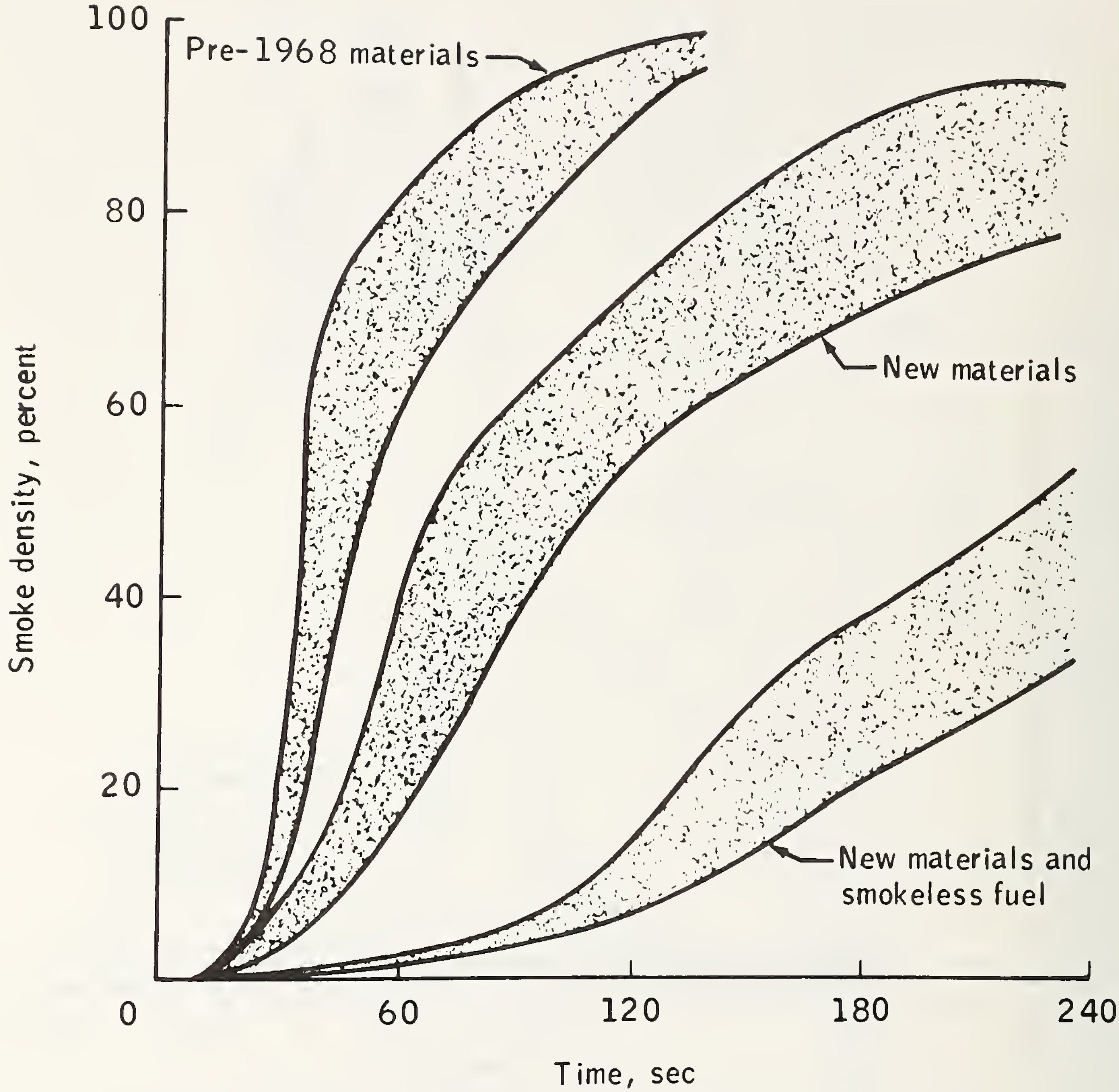

Figure 4. Minimum and Maximum Smoke Density Levels.

SESSION II

CASE STUDIES (continued)

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Detroit Diesel,Allison Division

B.R.Noton
Battelle Memorial Institute

REDESIGN & ASSEMBLY OF ANTI-FRICTION BEARING HOUSINGS FOR IMPROVED LIFE

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Project Officer
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ABSTRACT

A survey of anti-friction bearing failures in marine equipment, has shown that 95% of the failures occurred before the predicted mean service life had been reached. The principal causes of premature failure were bearing duress created by deficiencies in the design and assembly stages as well as, damage caused during the initial break-in periods. The investigation has also established that rotor rundown time (RDT) is a reliable indicator of the quality of assembly workmanship and the effectiveness of the break-in runs. This paper describes the revisions performed on a 25 HP motor driven pump and discusses the implementation of RDT as a quality control tool. Redesigning in conjunction with controlled assembly procedures, have resulted in significant improvements in equipment service life.

INTRODUCTION

The Naval Engineering Test Establishment (NETE) has conducted an investigation of Anti-Friction bearing failures on behalf of the Canadian Forces (CF). The purpose of this study was to ascertain the principal causes of premature failure of anti-friction bearing assemblies incorporated in the auxiliary equipment installed in the destroyer fleet of the Sea Element of the CF.

In all, about 1,500 in-service bearing failures from 10 types of mechanical and electrical equipment were analyzed in detail over a period of four years. This analysis showed that 95% of the observed defects were the direct result of improper assembly practices and inadequate knowledge of anti-friction bearing technology. It should be pointed out that these findings substantiate to a large extent the conclusions of surveys conducted independently by a number of bearing manufacturers or other types of industrial machinery (1,2)[1]. Therefore, it would appear that in many applications, installed bearing life is only a fraction of the attainable service life predicted by the bearing designers. From an economic standpoint, the inability of technical personnel to obtain consistently the rated service life from bearing assemblies is a particularly costly and wasteful practice.

A study of the failure patterns obtained in our investigation, has shown that in the majority of cases, a considerable improvement in equipment reliability could be achieved if the bearing housings were redesigned and assembled under rigidly controlled conditions. The general aim of these changes is to assure that the bearing/grease/housing com-

[1] Figures in parentheses indicate literature references at the end of this presentation.

bination would be subjected to a force system that is within its design limits. Such modifications have been successfully field tested for seven of the ten units included in this study and have resulted in substantial increases in equipment service life.

During the evaluation stage of the proposed revisions and various alternatives which were considered, rotor rundown time (RDT) was found to be a reliable indicator of assembly workmanship and the effectiveness of the initial running in period. It was also established that good correlation existed between attained service life and initial rotor rundown time. In addition, this parameter was used successfully to predict the remaining useful life of the system through periodic monitoring after the equipment was returned to normal operation. In view of these findings, it would appear that measurements of RDT, reinforced with measurements of temperature and vibration levels, would constitute a promising method of predicting incipient defects and preventing catastrophic failures.

This paper discusses the changes in design and assembly procedures which were adopted in the case of a 25 HP motor driven, seawater pump with sealed, grease packed, lubricated for life anti-friction ball bearings of extra wide design. Ancillary changes involved redesigning the motor casing components to ensure continued submerged operation and modifications to the pump bushing in order to minimize corrosion effects. The implementation of assembly and maintenance procedures for anti-friction bearing assemblies using rotor RDT measurements is also illustrated.

REDESIGNING FOR IMPROVED BEARING LIFE

Description of the Equipment

The principal causes of anti-friction bearing failures that this investigation encompassed, were the result of fabrication and assembly faults which in some cases were compounded by subjecting the unit to applications outside its design range. It was found that in all the units analyzed, substantial improvement to bearing reliability was achieved by introducing design modifications which ensured that the anti-friction bearings were subjected only to the force environment they were intended to accommodate.

In order to attain the desired clearances during the operating cycle of the equipment and to avoid damage to the bearings upon installation and during the initial break in runs, it became necessary to redefine and revise the existing fitting and assembly procedures. One of the first pieces of equipment to be modified in this manner was a 25 HP motor driven seawater pump.

A general view of the pump unit as originally manufactured, is shown on Fig. 1. The motor free and drive end anti-friction bearings have been designated as No. 1 and No. 2 respectively. The steady bearing between motor and pump, which was eventually eliminated, is the No. 3 bearing. The pump bearings are identified as No. 4 and No. 5 with the bottom bushing being the No. 5. The main function of this pump is to supply seawater at 100 psi at a rate of 40 tons per hour to the ship's fire mains. It is also used to draw from various watertight compartments and discharge overboard. As can be seen, the induction type 25 HP 440V three phase motor is enclosed in an air bell to permit operation while submerged to a maximum depth of 13 feet of water above the pump seating.

History of Failures

Examination of 23 failed motor bearings received over a two year period, indicated that failure was primarily due to outer ring rotation and excessive radial thrust loads. The high thrust loads were caused by improper installation, as well as malfunction of the flexible coupling between motor and pump. Some bearings failed due to overheating but it is not possible to ascertain whether overheating was caused by improper lubrication, severe misalignment, or because the ambient temperature exceeded the design value.

The pump end has a double row anti-friction bearing at position No. 4 and a steel backed DU bushing at the lower support. Analysis of the 28 failed pump bearings received, showed that the majority of failures was due to overspeed, inner ring rotation, misalignment and excessive thrust. Inner ring rotation and misalignment are defects which stem directly from installation and assembly errors and partly from inadequate support of the bearing inside the housing. The latter emits a certain amount of skewing and excessive thrust loading to take place, particularly when the hold-down bolt pressure is uneven. Also, excessive thrust may be generated by incorrect fitment of the coupling.

Another major cause of pump failures was attributed to malfunctioning of the cooling water system brought about by clogging of the strainer. This affects the supply of coolant to the pump seal and the bottom bush. It should be noted that the bottom bush in the original design is dead-ended with no flow through provision.

Available information indicated that the average operating life of this unit was 500 hours. Breakdowns were equally distributed between motor and pump ends. A review of the performance of the subject equipment together with an analysis of the failure patterns exhibited by the bearings received, revealed that design improvements were required in the following areas:

(a) Motor bearing housings in order to eliminate outer ring rotation and bearing distortion,

(b) Flexible coupling and rotor support arrangement so that a greater degree of misalignment could be accommodated without subjecting bearings to excessive loads,

(c) Pump bearing housings and cooling water circulation system so that adequate flow of coolant could be maintained, and

(d) Air bell enclosure and seals to retain operating capability under submerged conditions without overheating of the motor bearings.

The design changes in each of these areas will be described in the following sections.

Modifications to the Air Bell

The existing air bell enclosure fulfils the submergence requirements but at the same time it severely limits accessibility to the motor components. Normal servicing of motor parts is cumbersome and relatively costly even for minor repairs. Furthermore, temperature measurements taken inside the motor casing showed that under full load conditions, the enclosure was the cause of excessive local overheating which in turn caused bearing and electrical winding breakdowns.

The submerged operation capability was retained by remachining the motor end covers true to the casing centerline to $\pm$ 0.001", sealing all bolt recesses with a joint compound and fabricating a special electrical cable entry box. A relatively inexpensive lip seal was introduced at the shaft opening in the drive end cover to make the motor assembly watertight.

These revisions rendered the air bell enclosure obsolete and thus the overall unit weight was reduced by approximately 640 lb. while still retaining the capability of operating under submerged conditions. Accessibility to motor components was improved and the careful refacing of the end covers and motor casing made it possible to obtain better alignment between motor and pump.

Modifications to the Coupling Arrangement

The severe misalignment which was evident in the bearings examined, was attributed to malfunction of the coupling and to the three point support of the rotor which made aligning the motor shaft to the required tolerances very difficult to achieve.

It was decided to eliminate the steady bearing altogether and design a new motor skirt. This change also entailed manufacturing a new motor shaft which was about 12" shorter than the original item. In addition, a new coupling, with greater misalignment tolerance and simpler construction was specified. The end effect of these changes was that alignment between motor and pump could be easily controlled to within $\pm$ 0.002".

Modification to the Pump End

Failures of the pump end were caused by excessive misalignment, overspeed and inner ring rotation of the No. 4 bearing and by inadequate coolant flow to the pump seal and the No. 5 bushing.

The cooling water piping was altered by routing the flow through the bottom end of the pump casing and discharging into the inlet chamber, providing separate supply and return lines to and from the Durametallic seal, introducing a larger strainer and approximately doubling the water flow rate. These changes improved coolant supply to the pump seal and to the bottom bushing.

At position No. 4, a double row angular contact ball bearing had been specified by the supplier. Such an arrangement is not conducive to the high degree of misalignment that appears to arise during operation of this unit. The bearing was replaced by a single row, shielded, deep groove, cartridge type ball bearing which can accommodate misalignment to a greater degree and also withstand higher operating speeds. A number of dimensional changes were made to the bearing housing and associated components to ensure that the final assembly satisfied the tolerances recommended by the bearing manufacturers for this application.

At position No. 5, the steel backed DU bushing had exhibited a substantial rate of corrosion mainly due to the use of materials with different electrolytic potentials. The DU bushing was replaced with a

Ferrobestos bushing and the 316 S/S shaft sleeve was ceramic coated. The revised arrangement effectively isolates all dissimilar materials resulting in a much lower rate of corrosion of the non-sacrificial components.

A further modification, not related to bearing performance, was implemented at this time. This involved ceramic coating the whirl chamber and impeller. Accurate data on erosion rates were not readily available, primarily because these units very seldom operated for periods long enough to yield such information. However, some pumps did exhibit significant erosion at the cut-off point and of the impeller. Consequently, the whirl chamber casting and impeller were ceramic coated and tested on one trial unit. Evidence to date suggests that this modification should be extended to all pumps of this type.

Modifications to the Motor

Service failures of the motor were attributed to a combination of high ambient temperatures and a number of assembly and operational deficiencies which caused the bearings to perform under duress. The high ambient temperature problem was eliminated by removal of the air bell and improved coolant flow resulting from changes to the cooling system piping.

Bearing operating conditions were improved by remachining the housings and shaft. These changes assured that:

(a) All machined surfaces such as casing faces, cover spigots and bearing housings were parallel and true to the rotor center line. Perpendicularity of spigots to the housing centerline was held to $\pm$ 0.001".

(b) The drive end bearing cap spigot was fitted to acquire 0.005" interference. Since bearing width tolerances permit a variation of 0.005", this allowed the maintainer to achieve a clamping interference on the bearing of 0 to 0.002" without distorting the outer ring.

(c) The bearing housing radial tolerances were increased to an ISO G6 fit at the free end to allow unhibited axial travel accommodating whatever temperature growth occurred under full load conditions.

(d) A load spring washer was inserted at the free end bearing to arrest outer ring rotation which would be aggravated by the increase in the diametral clearances of the new housing. The load spring washer is expected to reduce ball skid at this bearing. An axial preload of 100 lb was used.

A general view of the modified unit incorporating all the revisions described in the preceding paragraphs, is shown in Fig. 2.

Test Results

The modifications outlined in this paper were implemented into one motor/pump unit and field tested. The motor operated for 14 675 hours, an improvement of 30 times over the previous mean time to bearing failure. The reason for removing the motor from service was to permit a detailed examination of the bearings and grease, especially, in view of the fact that the service acquired was almost three times the originally estimated life. Inspection of the motor bearings indicated that both were in excellent condition. Some minor fretting corrosion on the outer ring was noted at the top bearing and it is suspected that this was caused by axial movement generated by thermal growth of the rotor in an unlubricated housing.

During this time, the pump end was replaced three times giving a mean life of 5000 hours. The cause of failure was severe erosion or corrosion of the impeller and casing brought about by the presence of materials with widely differing electrolytic potentials. A pump with a ceramic coated impeller and whirl chamber coupled with an aluminum bronze shaft has been subsequently tested, and after 6000 hours showed no visual evidence of significant deterioration. Preliminary results appear to indicate that these modifications will make it possible for the pump end to reach a service life comparable to that of the prime mover.

ASSEMBLY OF ANTI-FRICTION BEARING HOUSINGS

Anti-Friction Bearing Run-in Procedure

Running in procedures for grease lubricated anti-friction bearings have not been standardized throughout industry. As a rule, such procedures vary from nothing at all to up to four hours of continuous running under no load at full voltage. Experimental data for a number of motor designs, show that with across-the-line starting under no load, the rotor attains full operating speed within 250 milliseconds. Such high rates of acceleration cause the inner ring to skid on the rolling elements. Skidding occurs because the unchannelled grease creates enough friction to inhibit the rolling elements from rotating normally and consequently, flat spots are formed. This has been repeatedly substantiated through detailed visual inspection of bearings subjected to across-the-line under no load start up procedures. Microscopic examination of the rolling elements revealed the presence of flat spots and temper rings. Once the grease has been properly channelled, localized friction is significantly reduced and hence the rolling elements are no longer prevented from rotating on the inner ring. However, the damage caused by the initial application of high start up torques is irreparable and invariably leads to premature failure.

In order to guarantee that start up torques do not set up unacceptable localized friction forces, we adopted a two stage, low voltage start combined with consequentive run-in periods of 1 minute, 5 minutes, 10 minutes, 20 minutes, 30 minutes and finally 1 hour. The purpose of the gradual increase in the running time is to ensure that the build up of frictional heat which is generated during the initial grease channelling process, will not be permitted to reach a level that might cause damage to the bearing. The two-step controller assures that acceleration rates are low and hence, initial ball skid will be minimized.

Rotor RDT As A Quality Control Tool

Measurements of rotor RDT during the running in tests have provided an excellent indication of the effectiveness of the breaking in procedures and of the quality of the bearing assembly workmanship. Rundown time curves obtained during the break in trials of the 25 HP motor discussed in this paper are shown in Fig. 3.

The motor shaft RDT increases relatively steeply at the beginning as a function of the total running time (TRT). The slope of the increase becomes gradually less and one would expect the RDT to asymptotically approach a constant value which would depend on the bearing design and the lubricating characteristics of the employed grease. However, preliminary measurements indicate that the rotor RDT is a continuously increasing function of the TRT, provided that the bearing is not subjected to any duress. This behavior may be typical of the medium temperature type CGSB 3-GP-691 grease (equivalent to US MIL-G-24139 and British XG-274) and bearing combination that was used exclusively in our tests. Further experimental work is required to determine whether this phenomenon applies to other bearing designs and grease combinations.

The gradual increase noted in the rotor RDT can be attributed to the reduction of frictional losses expended within the moving parts of the system. Frictional losses diminish as operating clearances are attained and hence RDT increases. Test units which exhibit a rotor RDT to TRT relationship similar to that depicted on Figure No. 3 are considered to have been installed correctly and the grease/seal/rolling element combination broken in without any damage to the system components.

Subsequent decreases in the RDT indicate that the bearing is not operating as freely as it had been. Such decreases are a direct measure of the formation of defects which bring about an increase in the system friction. Fig. 4 shows typical curves of RDT vs TRT obtained from equipment in service. In the cases where bearing defects had developed, a significant decrease in RDT was observed. Measurements of RDT therefore, could also be used to monitor the condition of bearings in operation and to provide an accurate method of establishing the amount of remaining service life in a particular piece of equipment. Admittedly, such measurements are somewhat cumbersome in that they require temporary removal of the equipment from service and partial disassembly of the coupling. However, our experience has shown that bearing defects influence RDT much before there is a significant change in ultrasonic or vibration levels. It is expected therefore, that in cases where RDT measurements could be implemented, such a monitoring tool would be a much more reliable indicator of bearing performance and relative condition than is potentially possible with monitoring programs based on measurements of other parameters.

SUMMARY

This study of anti-friction bearing failures, has shown that 95% of the bearings investigated had failed well before the potential bearing life had been attained. The primary causes of these failures were bearing duress created by deficiencies in assembly procedures, operation of bearings in a force environment that they were not intended to accommodate and lastly, damage caused during the initial run-in period.

The first two factors can be eliminated by selecting the appropriate bearing for each application, remachining housings and revising

installation procedures to ensure that the operational environment conforms to the bearing manufacturers' specifications. Removal of the third source of failures requires the use of controlled running in procedures. Break-in runs must be conducted under conditions which will avoid local overheating and guarantee the gradual channelling of grease.

Measurements of rotor RDT time have been successfully used in assessing the effectiveness of the running in tests and the quality of assembly workmanship. The RDT also provides an accurate measure of the remaining useful life of the bearing assembly.

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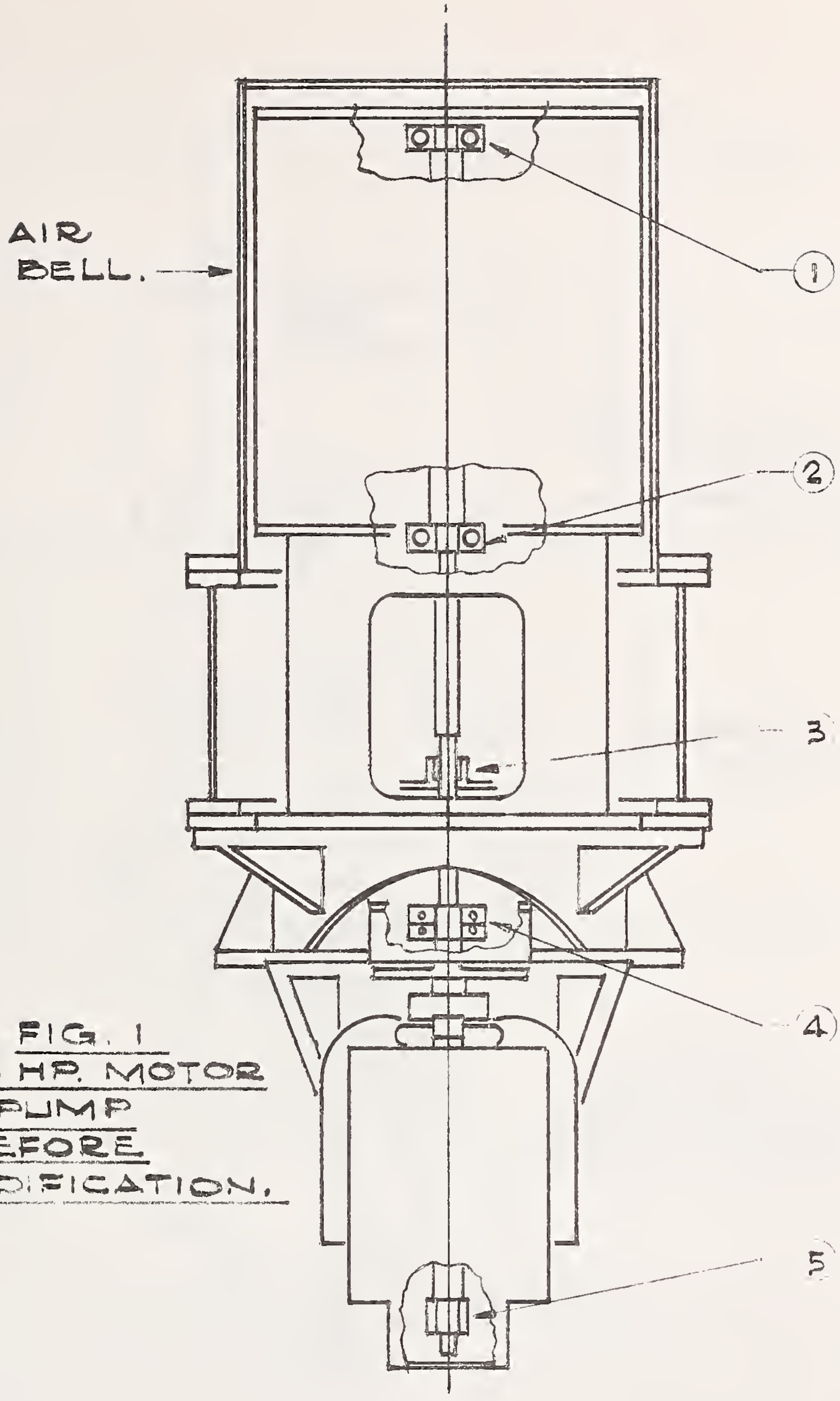

FIG. 1
25 HP. MOTOR
& PUMP
BEFORE
MODIFICATION.

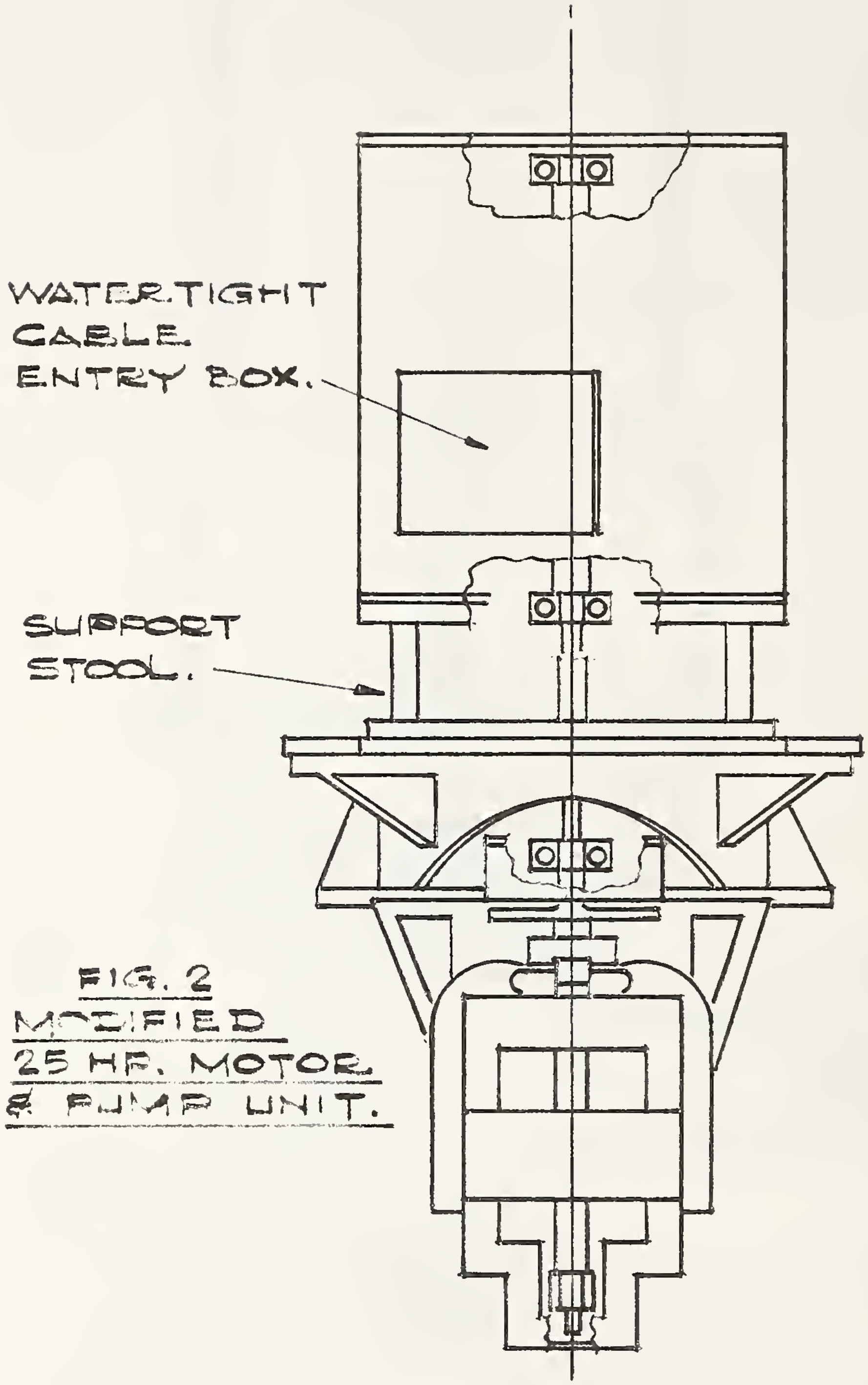

FIG. 2
MODIFIED
25 HP. MOTOR
& PUMP UNIT.

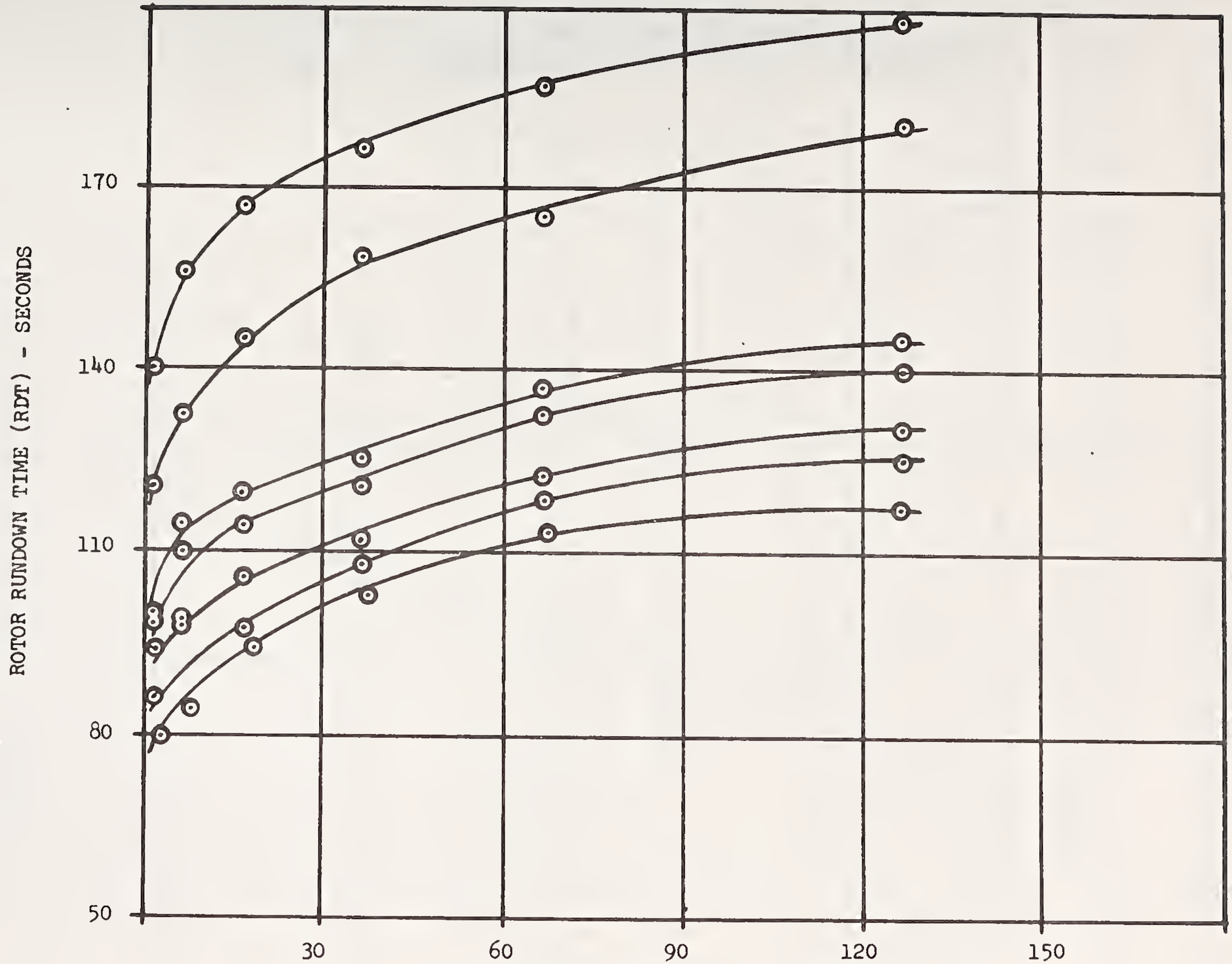

FIG.3 BEHAVIOR OF ROTOR RDT FOR A 25HP MOTOR DURING BEARING BREAK-IN PERIOD

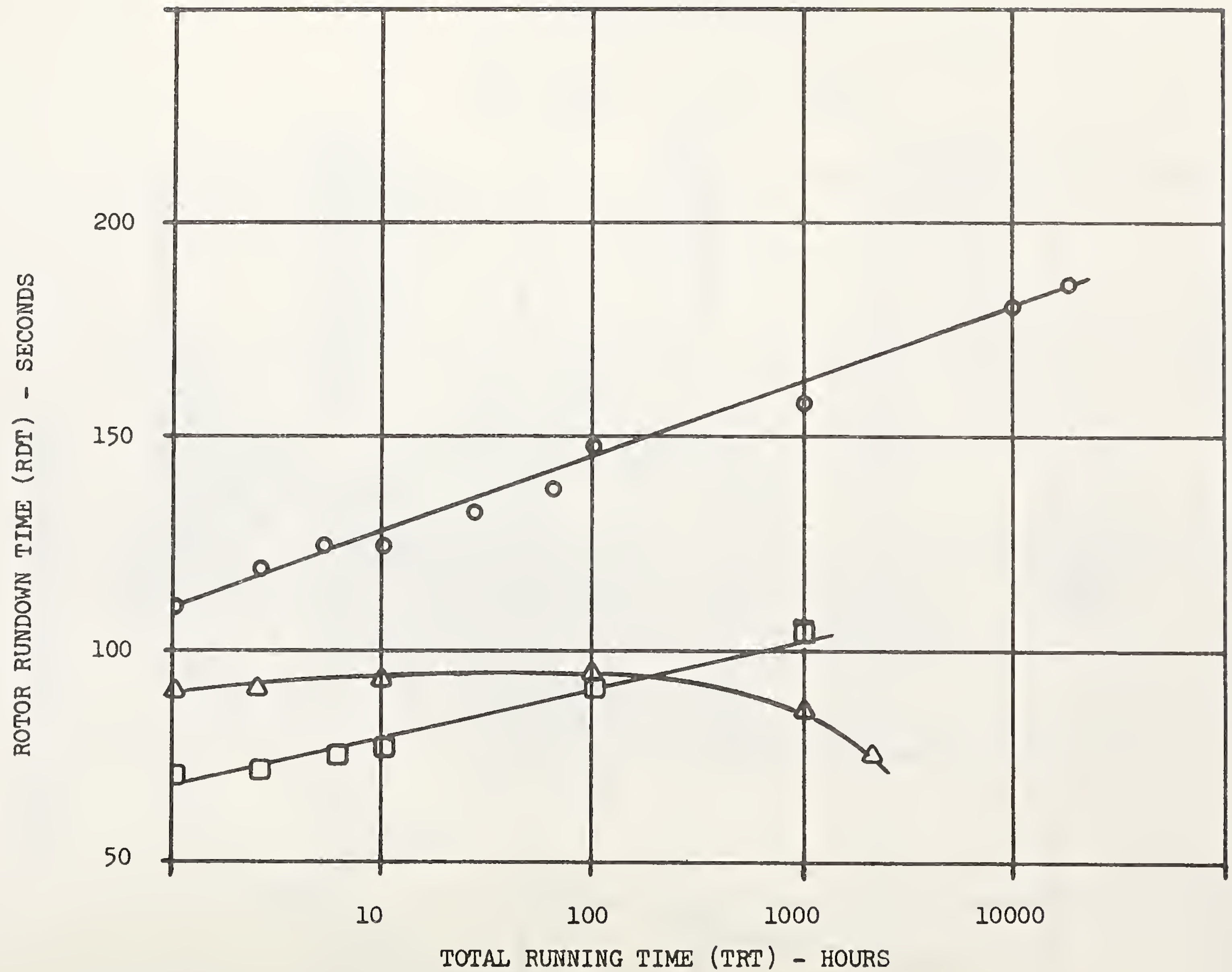

FIG.4 ROTOR RDT OF A 25HP MOTOR AS A FUNCTION OF TOTAL OPERATING TIME

DISCUSSION

R. Lenich, Caterpillar Tractor Company: How did the coupling perform?

D. C. Watson: Excellently. Some have been in service over 30,000 hours and we have not replaced a component yet.

J. E. Stern, NASA, Goddard Space Flight Center: What prompted you to investigate bearing failures? Were there case histories of systems failing or did you have a regular maintenance schedule?

D. C. Watson: There were two reasons. Firstly, somebody at headquarters thought the Armed Forces, the Navy in particular, were going through too many bearings. Secondly, I did not like the instruction books that called for opening a unit after 9 months of service. I knew perfectly well that when the units were opened to see why they were running well, the bearings were being changed. Or if the bearings were not changed, they were being damaged. Bearings are in very short supply and our whole concept was to try to cut the wastage of bearings.

R. L. Lundeen, Boeing Commercial Airplane Company: Can the run-down test technique be used on oil-lubricated bearings?

D. C. Watson: I have used the run-down time concept on gas turbine generator sets. People are also trying this concept in traction motors in the locomotive field, and I believe they have cut bearing failures by 75%. It can be used to check for misalignment in a gas turbine unit; it can also be used on roller bearings as well as on grease-lubricated ball bearings.

CASE HISTORY OF FAILURES IN A HAMMER MILL

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When I learned that the Conference "Success by Design: Progress through Failure Analysis" was to be organized by The Mechanical Failures Prevention Group, I thought a brief presentation about the failure of a few components used in High Speed Mills might be of general interest to this elite group. In this presentation, I will talk about the failure of hammers and screens and some of the steps taken to reduce the frequency of failure.

Hammer Mills are generally defined as Impact Mills, using either Fixed or Swing hammers, with the tip of the hammers operating in close proximity to a screen, that either partially or completely surrounds the hammers. Basic mill design permits continuous size reduction and mixing of many products ranging from viscous liquids and slurries to dry solids.

In a Hammer Mill, the factors that influence the Power, Capacity, etc., are:
(a) RPM of the machine: The material breaks down by impact and attrition. When the speed is too low, we don't get the desired end results and the machine may get clogged, whereas high speeds produce fine material and waste energy. The tip speed is a function of the diameter and the RPM whereas the centrifugal force is a function of the diameter and RPM^2.
(b) Screen and its area: Screen is used for classification. Certain minimum screen area is required to efficiently handle the material.

These Mills are available in two configurations -- Horizontal and Vertical. The Mills referred to here are the Horizontal Mills. The basic principle is illustrated in Fig. 1 which shows a horizontal shaft with fixed hammers and a screen for classification. Hammers apply a large tensile force to the infeed material and hurl the material against the breaker plate, followed by strong attrition forces. Normally, the hammers are so placed on the rotor so as to wipe the whole area of the screen. A variety of blade, rotor, screen, and throat options make possible an extremely wide range of processing applications.

The various types of blades and feed throats used on the above-mentioned Mills are illustrated in Fig. 2 and Fig. 3.

Before I go into detail relative to the main subject matter, a few relevant specifications of the Mills would not be out of place.

Mill Model No.	Max. RPM	No. of Blades (Old Design)	Screen Area Sq. In.	Calculated Tip Speed-Max. F.P.M.
T-6	9600	14	105	28,000
TS-12	7200	30	210	21,000
MS-12	6250	38	305	27,500
KS-20	5400	48 or 72	650	29,300
KS-32	3600	72	1040	19,500

These Mills have found a wide application in processing tomato products. They are used in "Catsup" processing for the most part to improve consistency in solids particle size, but in some lines, the Mill pre-grinds the feed material to a fine particle size to increase the yield. On "Sauces", they are used to improve or increase viscosity. Another application of the Mills is to reclaim pulp from the tomato finishers. Stating generally, the Mill as used in the tomato industry has provided a homogenizing effect and also grinding of the solids.

For application in Tomatoes Industry, the need has been for the screen with a hole size of 0.023". The size of the screen has to be carefully selected for proper retention time and the size of the product passing through it. These screens have normally been made of two layers -- the top layer 0.015" thick with a hole size of 0.023" and an open area of 25% and a back-up plate 0.109" thick with one inch perforated holes and 60% open area. These two screens were spot welded at various places to form a composite. The top screen has mostly been Inconel, but Stainless, Monel and Nickel screens have also been used. None of them had given satisfactory performance.

The "failure of the screen" terminology used here means rupture or breakage of the screening material to allow larger sized particles to pass through.

Some of the problems encountered with the screens have been that the thin screen material at the top was pushed through the backing plate as if a very hard product was being run and driven through it. It appeared that the internal pressure was too great and hence, the failure. In some cases, because of uneven feed and surge load, there was a large fluctuation of pressure and hence, the failure. In such cases, failure appeared always on the longitudinal side of the screen. It was analyzed that this type of failure might be due to the fatigue cracking where the screen had been welded to the backing plate. These cracks were often patched with silver solder. It was thought that Monel screens during processing, might have failed due to the ductility of the screening material. These screens had averaged 2 weeks and in some cases, lasted only one day.

To overcome some of the failure problems as mentioned above, one piece screen 1/8" thick was designed. The top half of the thickness was with 0.023" hole diameters and in the lower half the hole size was enlarged to 1/16" to ease the material out. This screen was about 7-8 times more expensive and had to be imported from Europe for one test trial. No one in U.S.A. could manufacture it and the cost of importing it was prohibitive -hence, this approach was shelved. Other alternative design of using small hole openings of 1/2" - 3/8" hole size, instead of 1" hole diameter in the backing plate were tried. These have proved to be fairly successful in preventing frequent failures and bulging of the top thin screen. Another approach that was tried to reduce or eliminate the screen problem was to put longitudinal liners on top of the thin screen over each rib of the backing plate. This meant that when spot welding, the thin screening material would be sand wiched between the two thicker screens. It also meant that the spot welding would be indirect. This seemed to help in some instances.

The second and major part of this talk is related to the failure of hammers and knife bars. The blades or hammers were mounted on the rotor via a knife bar. Knife bar and hammers are shown in Figure 4. The earlier designs called for mounting of the hammers in the knife bar as illustrated and then forcing the bar into the head.

The hammer or the knives were first pressed into the knife bar and then these bars were pressed into the dovetail slot into the head. The dimensions and the tolerances of these mating parts is very critical. Any gap would cause possible contamination. Another concern (valid for high speed operation) was that when the fit was not perfect it might affect the balance of the machine. Gauging of the bar and the slot is a time-consuming job. The specifications also called for 10,000 lbs. to 30,000 lbs. force for pressing operation. The bars were built up by chrome plating if the force required for pressing was less than 10,000 lbs. and the material was removed when the press fit was too tight and required more than 30,000 lbs. of force for assembly.

Before going into any analysis about the repeated failure of the different types of hammers, the following modes of failure were most predominately noticed. Figures 5 and 6 illustrate the different modes of failure.

1. Complete wearing off on the top part of the hammer resulting in rounding at

the top edge. (Illustrated in Figures 7 and 8).

2. Pitting -- Erosion at the top part of the hammer (illustrated in Figs. 9 and 10).

3. Cracking at the bottom part of the hammer where there is a change of section or even breaking of the complete hammer. (illustrated in Figs. 11 and 12).

4. Initiation of cracks at the base where the hold down screws are mounted. (Illustrated in Figs. 13 and 14).

In the early stages of design, the criteria for the selection of hammer and bar material was to have good wear and strength characteristics. The two principal metallurgical methods used to make stronger stainless steels are by "Precipitation Hardening" and, by the addition of strengthening elements like Nitrogen. 17-4 PH Precipitation hardening steel was extensively recommended and used in most applications for hammers. The important advantage of this steel is that it can be formed into required shapes in their annealed, soft condition and then heat-treated to higher strength after forming. The dimensional changes from hardening are very slight. In general, corrosion resistance of these steels is generally inferior to the basic stainless steel grade 304 and the cost is also higher. The hammers heat-treated to 44 R_c have about 190,000 PSI Ultimate strength and 60 ft. lbs. of Izod Impact strength.

The selection of 17-4 PH steel was not a bad choice if wear and strength were the only criteria of selection of material. This was not true, as noticed later. During the operation of these mills, repeated failure due to plain wear, pitting and erosion and also breaking of the complete hammers were experienced. If one hammer broke during processing, it would hit all other hammers causing damage to the rotor, screen and other hammers. Figure 15 illustrates the damage done to the knife bar and other hammers because of breakage. The initiation of cracks and, hence the breaking at the bottom part of the blade or hammer, where there is a change of section from the dovetail to flat, could be attributed to the stress concentration. Provision of a general radius might have solved the problem. The breaking or cracking of the rest of the hammers could be attributed to low impact-strength of the hammers at such high value of hardness. When one broken hammer or any extraneous material came in contact with other blades rotating at 5000 RPM, it would either crack the blades or completely break these because of low impact strength ---- thus damaging the complete rotor.

The pitting - erosion resistance of 17-4 PH steel is also poor and failure due to this is quite often noticed in liquid slurry applications. The wearing or rounding of the hammer was only at the top where the centrifugal force is maximum. When the hammer wears off, it would lose its effectiveness of size reduction. In one instance, when the 17-4 PH blade failed, die penetration examination was also performed to find if the knives were defective, but were found to be metallurgically sound and well made. 17-4 PH steel hammer should have had a life of at least 200 - 300 hours, but many of them failed much earlier.

Due to repeated failures, because of one reason or another, the following knife materials were also tested. Hammers made of 440-C stainless steel hardened to Rockwell 56-58 were tested without any success. (These were supposedly perfect metallurgically.) Another type of knife that was tried had carbide insert and these were (a) 17-4 PH with a type K-6 flat insert, (b) 17-4 PH with type 703-45° rear insert, and (c) 304 SS with type 703 flat insert. The failure of the carbide knives is also well illustrated in the Figure 16.

One of the design modifications done was the addition of jack screws below the hammers. Later on, it was found that in some hammers the cracks had initiated at the places where the screws were being forced against the hammer. Also, the "area" of the dovetail where the hammers and the knife bar or the knife bar and the rotor was corroded. This could be attributed to the poor corrosion resistance of the material.

As mentioned above, the hammers would break or wear and would need replacement, but the problem encountered in the design was that due to frequent failures of the blades, and hence in resembling all the components the fit of the mating parts could not be maintained. With frequent blade changes, these bars had badly scarred in pressing them out of the rotor. It was also found that the manufacturing and assembly of new or replacement hammers and bars was quite expensive. As it was virtually impossible to eliminate failures which had been caused because of one reason or the other, another approach was tried to at least reduce the frequency of failure, the initial cost and the cost of replacement.

After analyzing years of data and considering the availability, strength, wear characteristics, weldability price, corrosion-erosion characteristics, a new approach of using stamped hammers made of 316 stainless steel and hard-faced with stellite 6 was tried. (Figure 17) These hammers were so designed that they could be used on the existing machines in the field. Hammers were installed on the rotor just by sliding them in the dovetail slot (originally meant for the knife bar). They were grouped together along with spacers (also stamped) to provide the desired configuration and held in position by end locks. The cost of hammers, hard-facing and assembly on the rotor was significantly lower as compared to the original design of heat-treated machined hammers and bars. This approach met with tremendous success in the field. Some of the stamped hammers also cracked in the field. (The initiation of crack is illustrated in Fig. 18.) But the breakage of these did not cause catastrophic failures of the whole machine because of the ductility of the material from which the other hammers were made. When one hammer broke, it was replaced within minutes and the mill was back in operation. Hard-facing the hammers improved the wear and pitting characteristics. The stamped hammers could again be hard-faced if the material wore off.

By using stamped hammers, the knife bars have been eliminated. As the stamped hammers slide in the dovetail slot, therefore there are no close tolerances to be maintained.

Another solution undertaken to overcome the problem of breakage of the hammers has been to make an integral one-piece knife and the bar from the same material. This is the most expensive approach taken as the different knife-bar blocks mounted on the circumference of the mill are all different and each of them have to be individually machined and hardened. This has worked quite well in one application, but costs of manufacturing are prohibitive.

In conclusion, I would just like to add that partial success was achieved after repeated failures, but one thing to bear in mind is that the manufacturer and the user were willing to cooperate at all stages and the project was not abandoned. If a similar attitude is adopted, all parties concerned would gain tremendously in better understanding of the process, and hence help in designing better machines for the future.

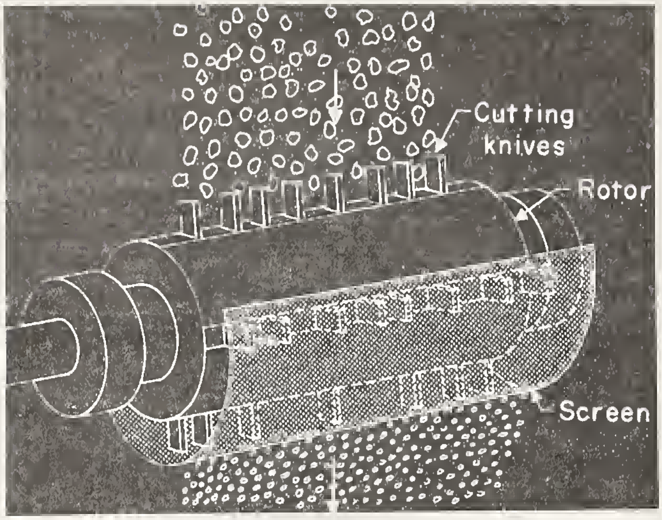


Figure 1: Schematic diagram showing the principle of a high speed impact mill.

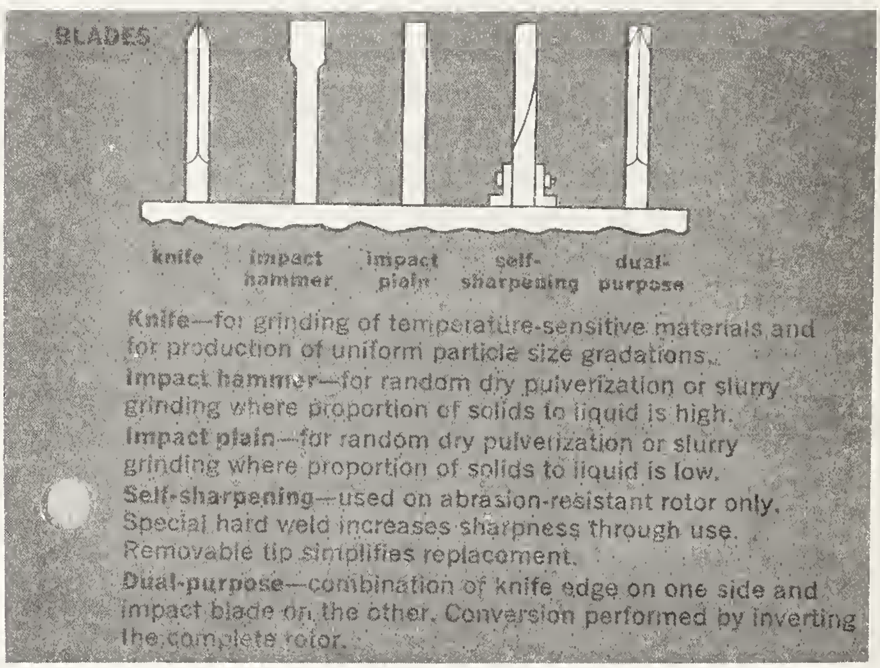


Figure 2: The various types of blades used in a mill

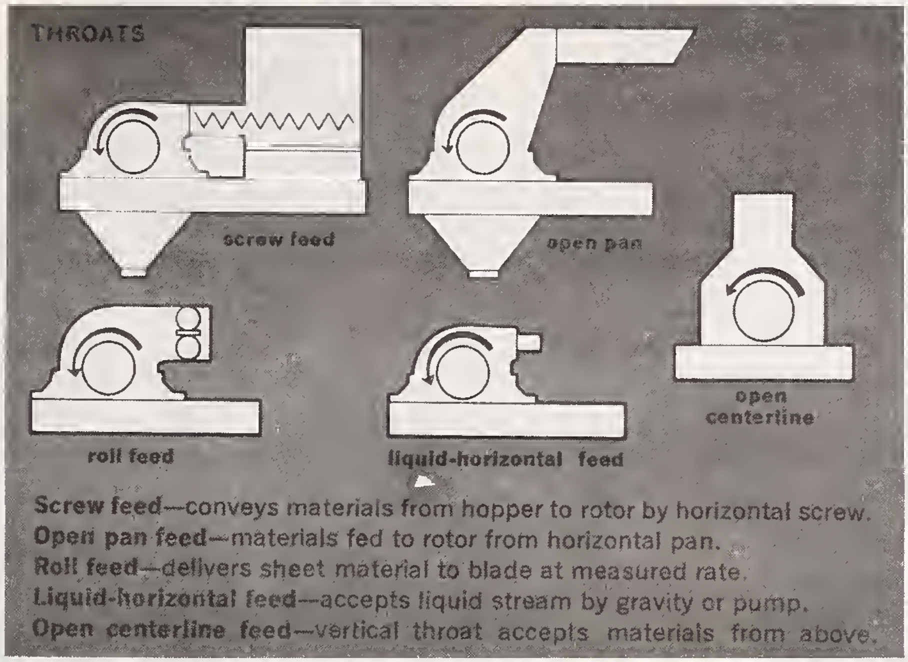


Figure 3: The various types of feed throats used with the Mills.

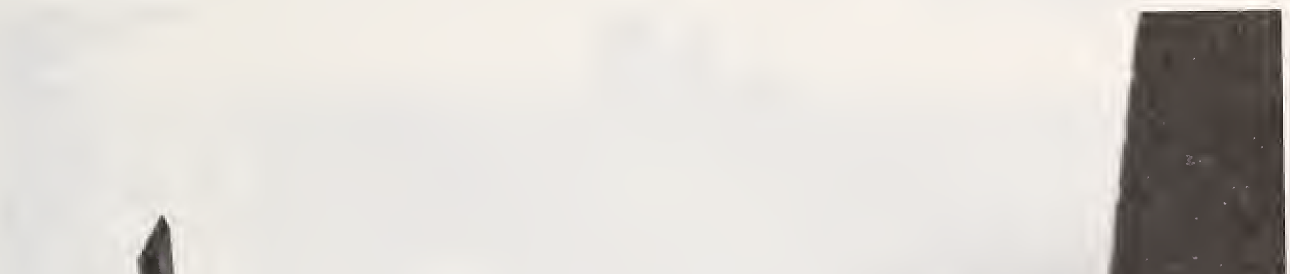
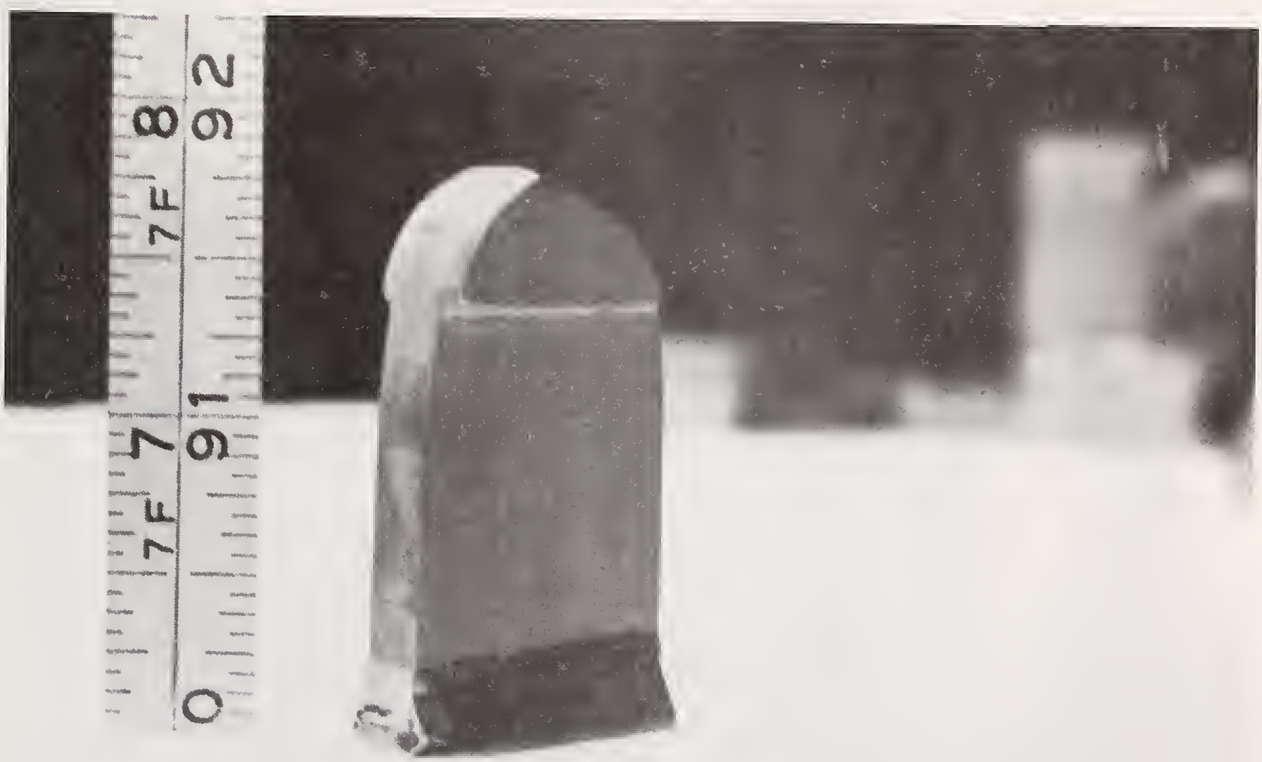
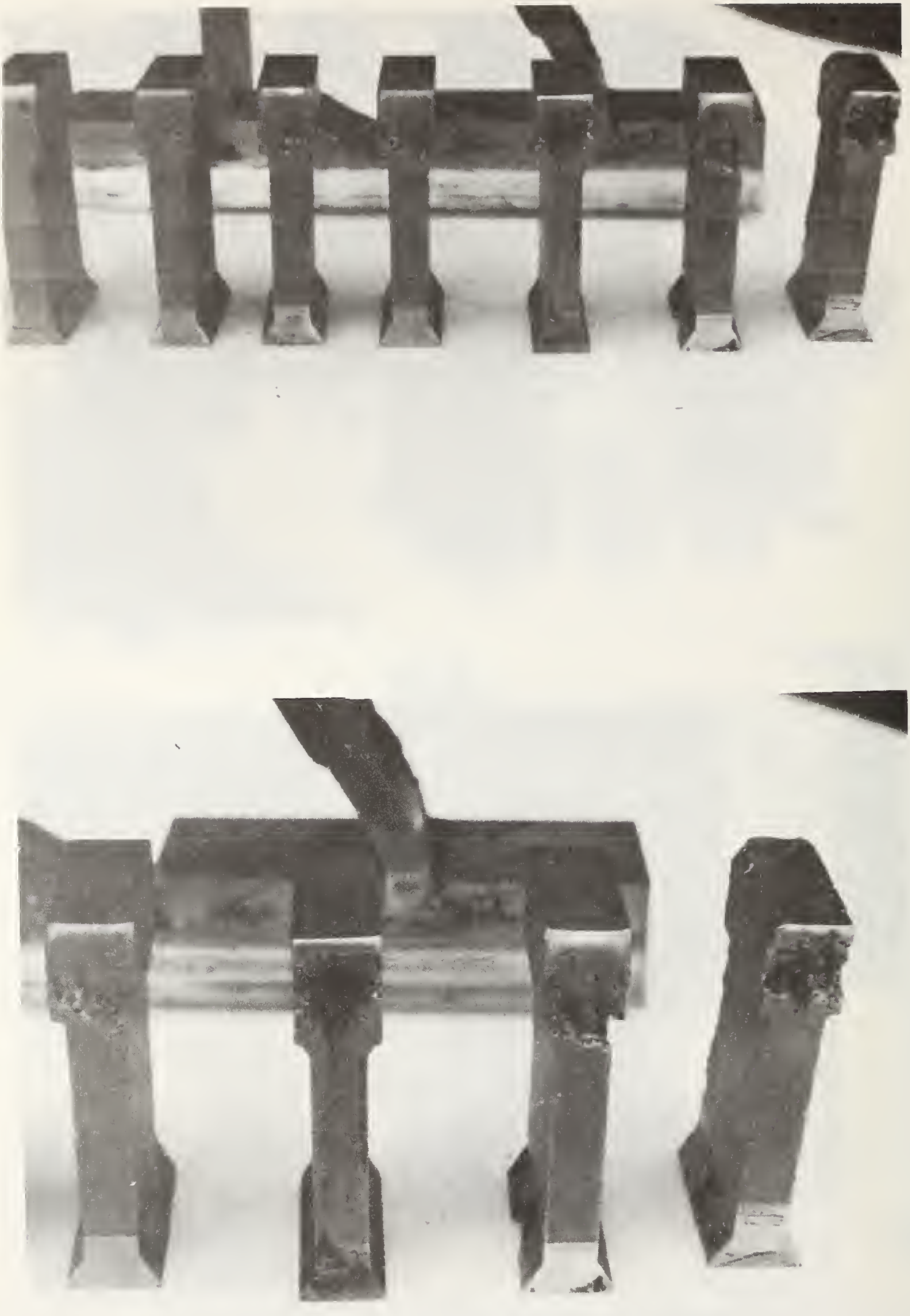


Figure 4: The Knife bar and the different types of hammers used.

Figures 5 & 6: The different modes of failures of various hammers.



Figures 7 & 8: Complete wearing off on the top part of the hammer.



Figures 9&10 : Pitting-Erosion at the top part of the hammer.

Figures 11&12 : Cracked and Broken Hammers.

Figures 13&14 : Hammers broken at the base.

Figure 15 : Damage done to the knife bar and other hammers because of breakage.

Figure 16 : Failure of the hammers with carbide insert.

Figure 17 : Stamped Hammers with and without the Stellite hard facing. The bent hammer shows the ductility of stainless 316.

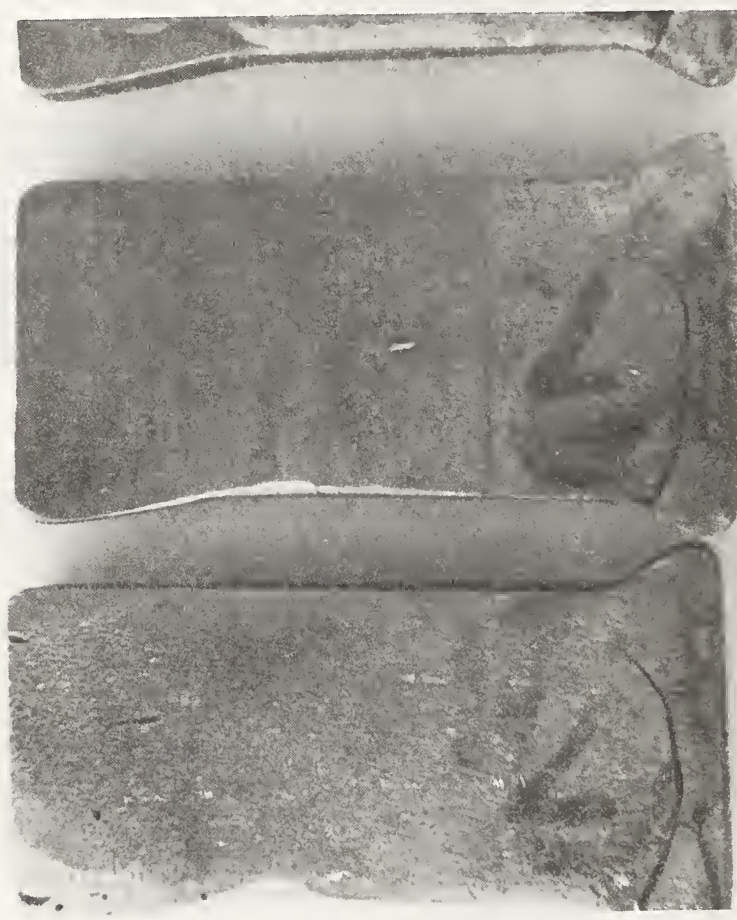


Figure 18 : Crack initiation prior to complete breakage in stamped hammers.

SOLUTION OF AN ART RESTORATION PROBLEM

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In Washington, D. C., behind the Lincoln Memorial, there are four large equestrian statues. Two of them, representing the "Arts of War", front on the Arlington Memorial Bridge, and are named "Valor" and "Sacrifice". Two others, representing the "Arts of Peace", front the entrance to Rock Creek Drive, and are named "Aspiration and Literature" and "Music and Harvest". The groups were originally sculpted in plaster by the American sculptors James Earle Fraser and Leo Friedlander. The statues as they now stand were gifts of the Italian Government to the people of the United States, presented in thanks for the Freedom Train sent to Italy by the U. S. after the second World War. They were installed in their present locations in 1951, and were dedicated by President Truman.

The statues were cast in bronze by four Italian foundries, located in Milan (Valor), Florence (Sacrifice), Rome (Aspiration and Literature) and Naples (Music and Harvest). There are two interesting facts about their fabrication. First, they were cast in bronze by use of the lost wax process, a process well-known and widely used in Italy since Benvenuto Cellini wrote his treatise on the casting of his "Perseus" by this process. Second, the statues were gilded using the "fire-gilding" process, in which an amalgam of gold and mercury is put on to the statues and the mercury subsequently boiled off by heating with torches.

By 1970 the statues had begun to show definite signs of deterioration, particularly "Music and Harvest" and "Aspiration and Literature". Instead of their original gold color, the statues were turning dark-grey-green, evidence that the gold gilding had deteriorated and was not carrying out its function of preventing corrosion of the underlying bronze. "Valor" and "Sacrifice" were by comparison much better, but they also were showing signs of deterioration.

In 1970, the National Capital Parks, which has the responsibility of maintaining the statues in the Nation's Capital, asked the National Bureau of Standards what needed to be done to restore the statues. After a preliminary investigation of the problem, the following options were laid out.

1. Do nothing, in which case the statues would continue to deteriorate, but eventually reach a final stage of patina. What the condition would be like in this final stage, and how long the process would take, could not be predicted because of the presence of the gold coating.

2. Sandblast off the gold coating and let the statues assume a natural bronze patina.

If the gold color had to be preserved, the following options were available.

1. Clean the statues and paint with gold gilt paint. Such paint would have to be replaced every 5 years.

2. Clean the surfaces and apply gold leaf.

3. Clean the surfaces thoroughly and plate with gold by the "brush-plating" technique, a technique normally used for industrial purposes. This would require that all the surfaces to be repaired be free of pits and pores that would be the source of future corrosion.

In any of the cases above, structural repairs would have to be carried out as necessary.

The National Capital Parks chose the last option, and restoration work was begun in the fall of 1971. The work fell under two broad categories: structural repair, and surface restoration.

Structural Repair

The various parts of the bronze castings had been bolted together by steel bolts, and steel angle irons had been used in the interior of the statues to provide structural rigidity. Unfortunately, the statues were set on hollow plinths that were open to subterranean chambers. This provided excellent opportunity for condensation of moisture on the inside surfaces of the statues, causing classic galvanic corrosion between the steel and the bronze. This had proceeded to the point where an incipiently dangerous situation had developed with respect to the bolts holding the various parts of the statues together, many of which had corroded almost in two. The steel angle irons had corroded to the point that they had very little structural strength left.

The correction of these conditions was straightforward, but required considerable work. All the steel bolts and angle irons were removed and replaced with bronze counterparts, and the statues are now structurally sound.

There were, however, other structural defects. There were numerous cracks in the castings, some very large indeed, being over two feet long. Moreover, the abutment between the pieces of the statues had in some cases not been completely sealed. These and the cracks provided openings from the inside to the outside, and gypsum left from the casting molds on the inside worked out through these openings, adding to the corrosion problems. To correct these conditions, the insides of the statues were thoroughly cleaned, and the cracks filled with 97% Sn-3% Ag solder. Welding proved to be unsatisfactory. The largest cracks were reinforced by strapping.

Surface Restoration

The surface restoration proved to be the most difficult and the most challenging. First the surface was sandblasted, using care to preserve the surface detail left by the artists. After the surface was cleaned and thus available for close inspection, it was found that the castings were of exceedingly poor quality. This was particularly true for "Music and Harvest" and "Aspiration and Literature". They were very porous and in many places hundreds of pits ranging in size from less than one millimeter to several millimeters covered the surface. Some of these were deep enough to penetrate the wall. These were drilled out and filled with bronze rod, or the tin-silver solder. However, filling in all the pits was impractical, and protection of this surface by a gold plating was impossible. What was done, therefore, was to plate the statues using the brush technique with a 500 microinch coating of nickel followed by a 160 microinch coating of gold. Then the exterior was covered with a clear protective lacquer especially formulated for the protection of bronzes.

Conclusion

The restoration was esthetically successful. The statues recovered their heroic character, and were a pleasing warm gold color. The technical success of the restoration remains still to be determined. It is expected that the lacquer coating will last at least five years, and then may require another coat. A few additional pits that penetrated the wall of the statues have been found and repaired. Some discoloration apparently due to insufficient washing away of the plating solution has been found and corrected. Only minor and easily correctable evidence of corrosion has been seen, and in that sense, at least, the restoration has been a technical success.

DISCUSSION

R. L. Lundeen, Boeing Commercial Airplane Company: Did you plug the hole in the bottom of the statues?

E. Passaglia: There was a lot of discussion about whether to plug the hole in the bottom. Once we got all the gypsum cleaned out and replaced all the steel bolts with bronze, all we would have is a bronze surface. Then there would be no need to plug the hole. Some people considered putting heaters inside the statues to keep them dry. What we did, in fact, was to fill up with epoxy compounds places where water would collect so that the water now runs down the inside of the plinth. The inside of one of the statues was coated with a polysulfide. In my view, that was not really necessary because bronze is supposed to be able to take weather. If it patinas on the inside, you do not really care.

R. Lenich, Caterpillar Tractor Company: How long did the restoration process take?

E. Passaglia: The whole process took about 60 days. It was all done with unskilled workmen. There was really nothing tremendously difficult about it. Sandblasting, of course, required a certain amount of skill--buffing you have to learn. Brush plating can be taught to people very quickly. The most skilled people there were probably the solderers and the welders. The work was all done by National Park Service personnel.

One of the problems that occurred was that no one realized that these plating solutions would turn dark under sunlight. The way the Navy uses the brush plating technique is to plate objects and then put them out into the air where rain water washes off any remaining plating solution. On the statues, a coating of lacquer was applied after plating. This coating prohibited the removal of the excess solution and allowed the statues to turn dark. I am quite confident that now that this has been removed, the statues will remain bright. In the places where there was no plating solution, there was no evident deterioration after 3 years.

J. K. L. Bajaj, Envirotech Corporation: What was the cost of restoration?

E. Passaglia: That I do not know.

J. J. Scialdone, NASA, Goddard Space Flight Center: What was the effect of hydrocarbons on corrosion?

E. Passaglia: This is a complicated problem. Patinization of bronzes is one of those areas where art meets science. There are people who know what conditions and what kinds of alloys are needed to obtain certain colors. I do not know of anything in the scientific literature about this. I would not expect hydrocarbons to cause any significant problem. However, sulfides could be a very serious problem in that they would result in the formation of a black patina rather than the desirable green patina. No one feels that traffic fumes had anything to do with the condition of these statues. The deterioration of these statues was simply caused by galvanic corrosion between the gold covering and the copper substrates because the gold finish did not completely cover the bronze.

ELIMINATION OF FAILURES OF U-BOLTS IN FARM TRACTOR DUAL WHEELS

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When farm tractors are to be used on sticky or wet soils, many farmers will equip their tractors with dual driving wheels. One common design for these dual wheels includes a semicircular hub and two or more U-bolts which are used to fasten the wheel to the axle. Shortly after introducing a new line of dual wheels, one manufacturer experienced a number of failures of U-bolts after only a few hours of service.

Cold finished 1 1/8" diameter AISI 1045 steel was used for these U-bolts. The bars were cut to length, the threads were cut, the blanks were bent to shape, the U-bolts were oil hardened and tempered to R_C30 and then the U-bolts were rebent if necessary.

Without exception, the failures occurred through the threads of the U-bolts, immediately adjacent to one of the nuts used to tighten the U-bolt in place. The fracture surfaces exhibited positive evidence of fatigue. An analysis of the service loading indicated that there were three components of the loading. There was a constant preload, at a level determined by the farmer who installed the dual wheel. There was a periodic variation as the hub and U-bolts alternately bore the weight of the tractor. There was also a load caused by the widely varying torque applied by the power train to the drive wheel. It was concluded that the composite loading could logically cause fatigue failures.

Metallurgical examination of several failed U-bolts indicated that only a shallow case of each bolt contained the desired tempered martensite microstructure. The interior of the bolt was a mixture of ferrite and pearlite. For immediate relief of the failure problem, the specified material was changed to AISI 4140, which has sufficient hardenability to obtain the desired tempered martensite microstructure throughout the U-bolt. The material change prevented service failures, although the cost of the alloy steel was greater than that of the plain carbon steel it replaced.

The U-bolt failures also initiated a general review of the manufacture of this item. As a result of this review the entire operation was changed. The material was changed to warm drawn AISI 1527, 1.070" diameter. The bar stock was cut to length, threads were rolled and the blanks were then stored. Blanks were bent to size only to meet orders. No heat treatment was required. No U-bolt made in this manner has failed in the two years since the production methods were changed.

Besides totally eliminating service failures, the new production methods have substantially reduced manufacturing costs. The weight per foot of the steel is approximately 10 per cent less. Roll threading is faster than cutting threads (and produces a thread more resistant to fatigue). By cutting all blanks to the same length, that of the longest U-bolt, only one item was kept in inventory, instead of 12 different U-bolts. Storing blanks rather than U-bolts greatly reduced the required storage space. And the costs of heat treatment, transportation and rebending were eliminated.

The original U-bolt design evolved from smaller U-bolts used on smaller tractors, where the hardenability of AISI 1045 was adequate. Only because the part designed by evolution failed was the general review started. In the review the engineers examined the interrelations between the function and design of the U-bolt, its material and the methods by which it was made. From this systems approach to design, a far superior product at a far lower cost was achieved. The success of the redesigned product may be judged by considering that every competitor in this market has since adopted the same material, manufacturing methods and design.

STRUCTURAL IN-FLIGHT WING FAILURES

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ABSTRACT

An overview of the types and number of structural in-flight wing failures occurring within the 10-year period of 1964 through 1973 is presented. Particular attention is given to failures caused by fatigue fractures of wing components. Some case histories involving those fatigue failures of the wing which relate to design are presented.

INTRODUCTION

The Congress of the United States recognized the importance of aircraft accident investigation as a means of preventing accidents when it passed the Civil Aeronautics Act of 1938. This Act gave the Civil Aeronautics Board the authority and the responsibility for investigating and determining the cause of all United States civil aircraft accidents. In 1966 the Transportation Act created the National Transportation Safety Board, to whom the Civil Aeronautics Board's accident investigation functions and other safety activities were transferred. Since this paper deals with the period of 1964 through 1973, information on accidents was, in part, taken from records and files accumulated by the Civil Aeronautics Board but now maintained by the National Transportation Safety Board.

The Safety Board conducts its own investigations of all air carrier and air taxi accidents, most fatal light plane and helicopter accidents, as well as other accidents or incidents that may be of special interest because of their safety implications. The Safety Board has delegated the authority to investigate most nonfatal light plane (under 12,500 lbs. gross weight) accidents to the Federal Aviation Administration, but retains the responsibility for determining the probable cause of these accidents. To assist in the determination of probable cause in accidents or incidents involving a material failure, the Safety Board maintains a metallurgical laboratory in Washington, D. C.

This paper will be confined to United States-registered fixed-wing aircraft having in-flight structural failures of the wing (within the 10-year period of 1964 through 1973). Descriptions of failures precipitated by fatigue fractures of wing components and some case histories where design was a contributing factor will be presented.

FAILURES IN GENERAL

Data complied by the National Transportation Safety Board for 1964 through 1973 disclosed 170 accidents involving United States-registered civil aircraft that were caused by structural failure of the wing. Table 1 shows a breakdown of these accidents by failure type.

TABLE 1

IN-FLIGHT WING STRUCTURAL FAILURES (1964 THRU 1973)

FAILURE TYPE		NUMBER OF ACCIDENTS
OVERLOAD		117
UNKNOWN		29
FATIGUE		15
FABRIC FIBERGLASS OR WOOD		7
CORROSION DETERIORATION		2
	TOTAL	170

The failures included in Table 1 are partial or total wing separations, folding without separation, wing support structure failures, and deformations due to aerodynamic loads (such as wrinkling of the wing skin or partial breaking of structure.)

Failures listed as "UNKNOWN" include those where the wreckage was destroyed by fire or lost at sea, and those not reported as being caused by overload, fatigue, stress corrosion or corrosion. Listed as fabric, fiberglass or wood were failures known to have been caused by separation or breakage of these nonmetallic components. All the fatigue fractures shown were substantiated by examination in the NTSB metallurgical laboratory.

FATIGUE FAILURES OF THE WING

Table 2 lists chronologically the 15 accidents attributed to fatigue fractures of wing components. Also shown are the probable cause(s) and/or contributing factor(s) for each of the failures.

TABLE 2

IN-FLIGHT FAILURES CAUSED BY FATIGUE FRACTURES OF WING COMPONENTS (1964 THRU 1973)

ACCIDENT DATE	AIRCRAFT INVOLVED	PROBABLE CAUSES AND/OR CONTRIBUTING FACTORS
06/17/64	BEECHCRAFT C-18S	INHERENT DESIGN; TURBULENT WEATHER
08/15/66	BEECHCRAFT C-18S	INHERENT DESIGN; POOR WELD
02/28/67	BEECHCRAFT C-45H	WELD CRACK
04/28/67	BEECHCRAFT E-18S	INHERENT DESIGN
09/27/67	AERO COMMANDER 560 E	SUBSTANDARD QUALITY CONTROL (overlapping drilled holes)
12/02/68	FAIRCHILD F-27B	INHERENT DESIGN; INADEQUATE INSPECTION; TURBULENCE
01/28/70	DEHAVILLAND DH-104	INHERENT DESIGN; INADEQUATE REPLACEMENT TIME REQUIREMENT
05/06/71	DEHAVILLAND DH-104-7AXC	POOR/INADEQUATE DESIGN; SUBSTANDARD QUALITY CONTROL
08/21/71	HELIO AIRCRAFT H-295	INHERENT DESIGN; TURBULENT WEATHER
12/04/71	PIPER J4A	OVERSTRESS FROM PRIOR ACCIDENT
06/22/72	BEECHCRAFT E-18S	INHERENT DESIGN; INADEQUATE INSPECTION
04/19/73	BEECHCRAFT E-18S	INHERENT DESIGN; INADEQUATE INSPECTION
08/24/73	PIPER PA-25	INHERENT DESIGN
10/01/73	PIPER PA-25	SUBSTANDARD QUALITY CONTROL (overlapping drilled holes)
10/18/73	PIPER PA-25	SUBSTANDARD QUALITY CONTROL (inadequate material strength, improperly drilled hole); UNAPPROVED MODIFICATION

Inherent design is shown as a cause or contributing factor to the failure when the design itself does not appear adequate to prevent fatigue cracking under the normal flight profile loading conditions of the aircraft. It should be pointed out that all the aircraft shown in Table 2 were not required by the existing regulations to be designed and substantiated in fatigue at the time of their certification.

Ten of the fifteen accidents involving fatigue fractures occurred on aircraft types having more than one failure; Beechcraft 18 series (5 failures), Piper PA-25 (3 failures), and DeHavilland 104 type (2 failures).

Some examples of fatigue failures which show design as a cause or contributing factor are illustrated in the following examples.

BEECHCRAFT E-18S ACCIDENTS

A good example of failures attributed to inherent design where a change in design of the original part is impractical is the Beechcraft 18 series accidents of April 28, 1967, June 22, 1972, and April 19, 1973. All three accidents occurred as a result of fatigue fractures almost identical in nature.

In the accident of June 22, 1972, the failure of the wing was precipitated by a fatigue fracture through the lower spar cap that initiated at the toe of a weld used to secure a gusset plate to the top side of the lower spar cap (see arrows "O" in Figures 1 and 2). The spar cap is an elliptical shaped 4130 steel tube procured to MIL-T-6736A and heat treated to 180-200 KSI tensile strength range. MIL-T-6736A allows up to 0.15 inch total decarburization on the outside diameter of the tube for the wall thickness specified (0.10 inch). Decarburization such as this lowers the tensile strength of the surface of the spar cap to about 80 KSI; more than half of the internal material strength of the spar cap. This low strength at the surface and the geometric stress riser at the toe of the weld contributed to fracture initiation in fatigue. The spars, as they were designed and manufactured, were inherently susceptible to fatigue cracking. Changing the original design in these cases was not a practical solution, and inspections were performed periodically to detect cracking before ultimate failure occurred.

X-ray radiographic inspections were performed on the aircraft involved in the accidents on June 22, 1972, and April 19, 1973, with no cracks being reported. Post accident examination of the X-ray films disclosed that substantial cracks were in the spars during some of these inspections and should have been detected during the inspection. The probable cause of these accidents were, therefore, attributed to inadequate inspection.

As a result of the April 19, 1973, accident, the FAA issued an amendment to the existing inspection Airworthiness Directive[1] requiring modification of the spar within 600 service hours. The modification consisted of attaching steel straps to increase the structural integrity of the area.

HELIO AIRCRAFT H-295, AUGUST 21, 1971

The original part which failed was redesigned after the Helio aircraft H-295 accident on August 21, 1971. That was the steel main spar carry-through fitting shown by arrows "F" in Figures 3 and 4.

Multiple fatigue cracks had originated in the inside surface of the fitting along the line where the inner wall changes from a cylindrical to a conical surface near the closed end of the tubular section. This area is at the bottom of a longitudinal 1.25-inch diameter drilled hole. The fatigue cracks had propagated through the wall thickness and across approximately 40 percent of the cross sectional area of the tubular section before the fitting failed completely (shown in Figure 4).

The points where the fatigue fracture originated were in a rough drilled surface where the edge of the drill had left a sharp corner at the change in section thickness near the bottom of the hole. The manufacturing drawing for the fitting showed a requirement for a 0.062-inch radius and 125-microinch RMS surface finish in this area. The metallurgical examination on the failed fitting, however, indicated that no attempt had been made to form a true radius or improve the surface finish after the drilling operation. These discrepancies probably contributed to the cause of the fatigue fracture. A subsequent fatigue analysis of the primary aircraft structure indicated that the steel carry-through fittings were subjected to high stress levels and the development of fatigue cracks and, therefore, had a limited fatigue life.[2]

As a result of this investigation, the National Transportation Safety Board issued a safety recommendation[3] to the FAA to require inspections of fittings of the same design for cracks, adequate internal fillets, and surface roughness in the internal fillet area at the earliest practical time by an FAA-approved method. Also recommended was the replacement of fittings with an improved design if possible, at the time of this inspection. A suggested design change to greatly increase the structural integrity of the fitting was to decrease the depth of the 1.25-inch hole to move the 0.62-inch radius to an area of less stress. Such a design change, however, would require the manufacturer to change materials because the original material chosen (4130 steel, 180-200 KSI heat treatment) could not be heated to the strength level required.

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[1]Raised figures indicate literature references at the end of this presentation.

The manufacturer did, however, change the design incorporating the original material and heat treatment. This was done by changing the conical shaped bottom of the hole to a hemispherical shape, thereby decreasing the stress intensity in the area. The hole depth was also decreased somewhat to increase the cross-sectional area near the bottom of the hole. The newly designed fitting was incorporated in the new assembly which contained factory installed carry-through reinforcing straps.

The FAA, as a result of this accident, issued an Airworthiness Directive[4] (AD) requiring a gamma ray inspection of the fittings within 10 hours time in service, and thereafter at 100-hour intervals, on all Helio Model H-250, H-295, H-391, H-391B, H-395, H-395A land airplanes with 3,000 or more hours of service, and all float equipped planes of these models with 1,500 or more hours in service.

The AD further required that, within 350 hours in service after the initial inspection, carry-through reinforcing straps were to be installed in the field. With the installation of these straps, the repetitive inspection interval was increased from 100 hours to 500 hours. The AD also provided that if a new assembly with the newly designed fitting was installed, then the inspections would no longer be required.

DEHAVILLAND DH-104-7AXC, MAY 6, 1971

Apache Airlines DeHavilland Dove 104-7AXC crashed into a cotton field on May 6, 1971, after separation of the right wing.[5] The failure was the result of a fatigue fracture in the wing attachment fitting (see Figures 5 and 6). Origination of the fatigue fracture was in an area of fretting and corrosion in the hole for the main wing-to-fuselage attachment bolt.

The aircraft involved was originally a standard DeHavilland Dove which had been substantially modified by Von Carstedt Corporation, C-Air, Long Beach, California. The modification consisted primarily of increasing the fuselage length to accommodate more passengers and installing more powerfull engines for increased speed. To accomplish this, the fuel tanks in the wing were relocated and the main wing attachment fittings were redesigned. The design was approved by the Federal Aviation Administration and Supplemental Type Certificate SA-1747WE for the aircraft was issued July 23, 1968.

The redesigned wing attachment fitting consisted of increasing the tensile strength and cross-sectional area of the fitting over that of the original Dove fitting to accommodate the increased load expected due to the modifications. The new fitting was made of 4130 steel heat treated to 180-200 KSI tensile strength (originally fitting had 175 KSI tensile strength).

Since the new fitting was to be structurally similar to the original fitting, having similar internal stress, the design was approved without a requirement for substantiating fatigue tests. The fatigue life of the Von Carstedt fitting was predicated upon the life of the original DeHavilland fitting, provided that the new fitting maintained the same precise tolerances and joint-sealing procedures employed in substantiating the life of the original DeHavilland fitting.

The aircraft that crashed, N4922V, had accumulated a total service time of 5,593 hours and was the first production aircraft modified by Von Carstedt under STC SA-1747WE.

Hardness measurements in the failure area gave results indicating the fitting had an average tensile strength of 157 KSI, well below the engineering drawing requirements (180-200 KSI). The alloy chosen for the fitting was 4130 steel known to have low hardenability properties. According to Military Handbook 5A, used in the design of the modification, a part fabricated from 4130 alloy steel with the size and geometry of the fitting could not be consistently hardened throughout the section thickness to attain the specified tensile strength required in the engineering drawing; tables in the handbook indicated that 4340 steel would have been the preferred alloy. The hardnesses obtained on the failed fitting were found consistent with hardenability data supplied for the material certification for the barstock used in fabricating the fitting. Although this data was available to the designers and checkers at the time of the design change, the drawing was approved.

The stress analysis submitted to and approved by the Federal Aviation Administration noted that the service life of the new fitting was predicated upon maintenance of the original DeHavilland tolerances. The engineering drawing checked and approved by a designated engineering representative of the FAA, however, specified a tolerance allowing an 0.0022 inch greater diametrical clearance than that specified in the fatigue analysis.

The conformity inspections of the aircraft during manufacture of N4922V were performed by an FAA manufacturing inspector for the local district office. The inspector testified in a public hearing on the accident that he had rejected the fitting on the basis of its strength. The part was rejected because a hardness test on another part from the same heat treat lot was not within its hardness specifications and, therefore, the entire lot was rejected. The inspector did not, however, follow up to assure compliance with his request for a subsequent inspection to determine if the part was properly heat treated. Although the procedures used for the ultimate acceptance of the wing attach fitting were never determined, the fitting was subsequently installed and the aircraft was certificated.

Shortly after the accident, the Federal Aviation Administration, in an emergency order, suspended the airworthiness certificates on the remaining four aircraft modified by Von Carstedt. These aircraft remained grounded until October 22, 1971, when a Supplemental Type Certificate SA-2348WE was issued. The STC modified the aircraft to incorporate a lower wing-to-wing strap that could carry the total load should the lower wing joint fail (fail safe). This was performed, of course, only after inspecting each fitting to assure that they were crack free.

REFERENCES

1. Federal Aviation Administration Airworthiness Directive 72-20-5, Amendment 39-1526 as amended by Amendment 39-1632. Amendment 39-1632 became effective May 7, 1973.

2. Federal Aviation Administration letter to owners of Helio Model H-250, H-295, H-391, H-391B, H-395 or H-395A airplanes dated October 5, 1971.

3. National Transportation Safety Board Recommendation A-7 -46 and 47 to the Federal Aviation Administration adopted September 23, 1971, and issued October 4, 1971.

4. Federal Aviation Administration Airworthiness Directive 71-21-11 Amendment 39-1320 effective October 16, 1971.

5. Accident Report NTSB-AAR-72-19, Apache Airlines at Coolidge, Arizona, 5/6/71 (available from NTIS, Springfield, Virginia, 22161).

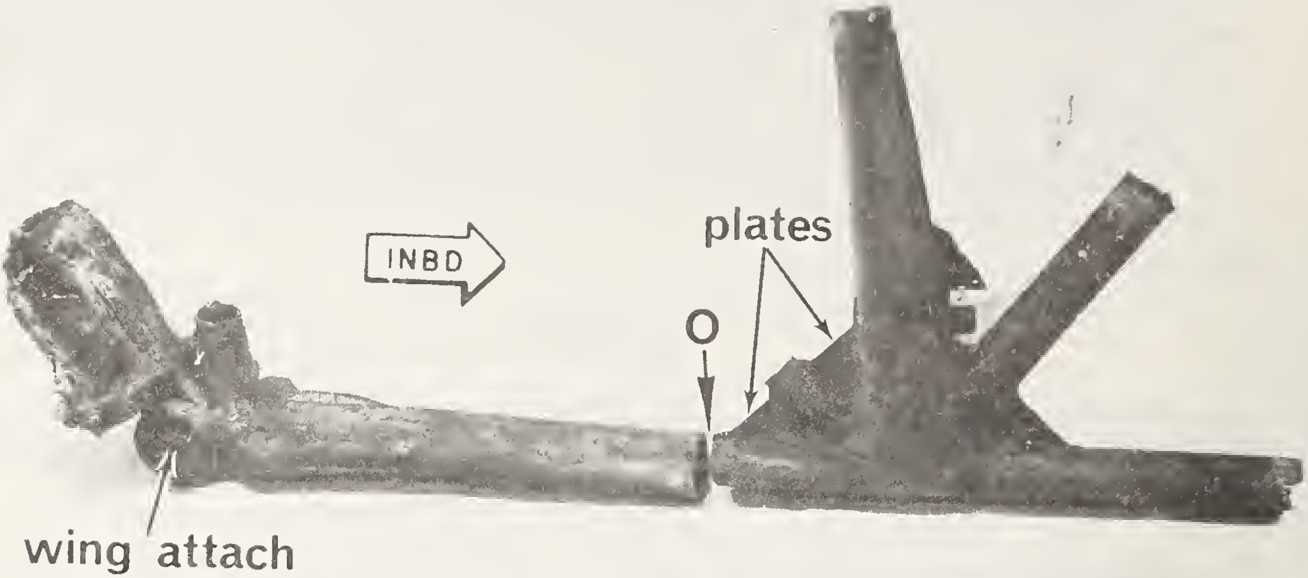

Figure 1. Overall view of portion of center section truss removed from accident of Beechcraft E-18S, June 22, 1972. Arrow "O" denotes fatigue fracture origin in lower spar cap.

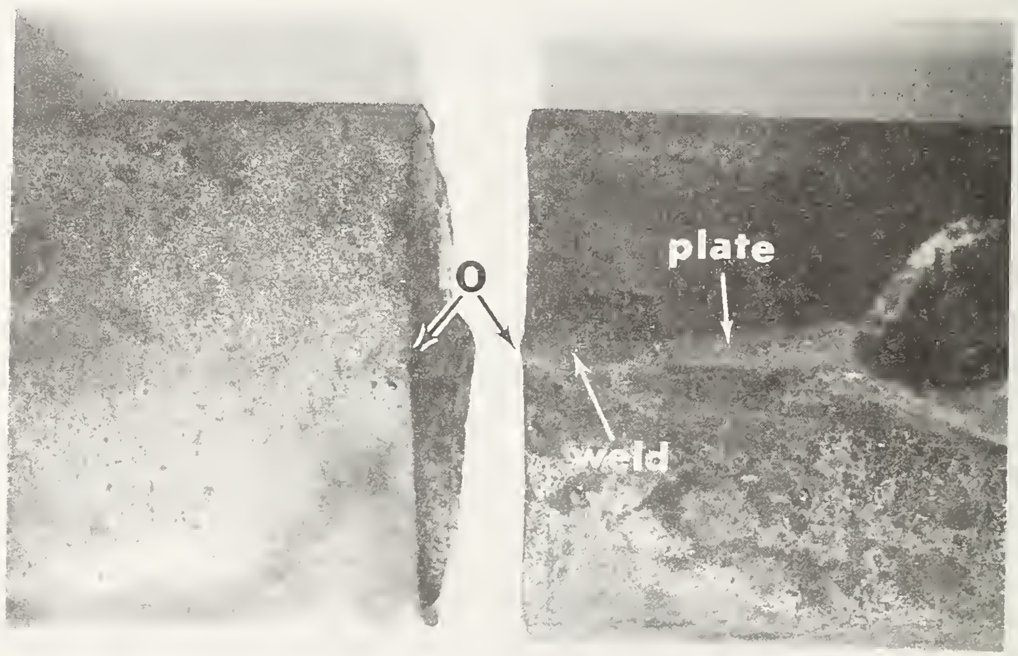

Figure 2. View of fracture origin showing relation to weld for gusset plate. Accident Beechcraft E18S, June 22, 1972.

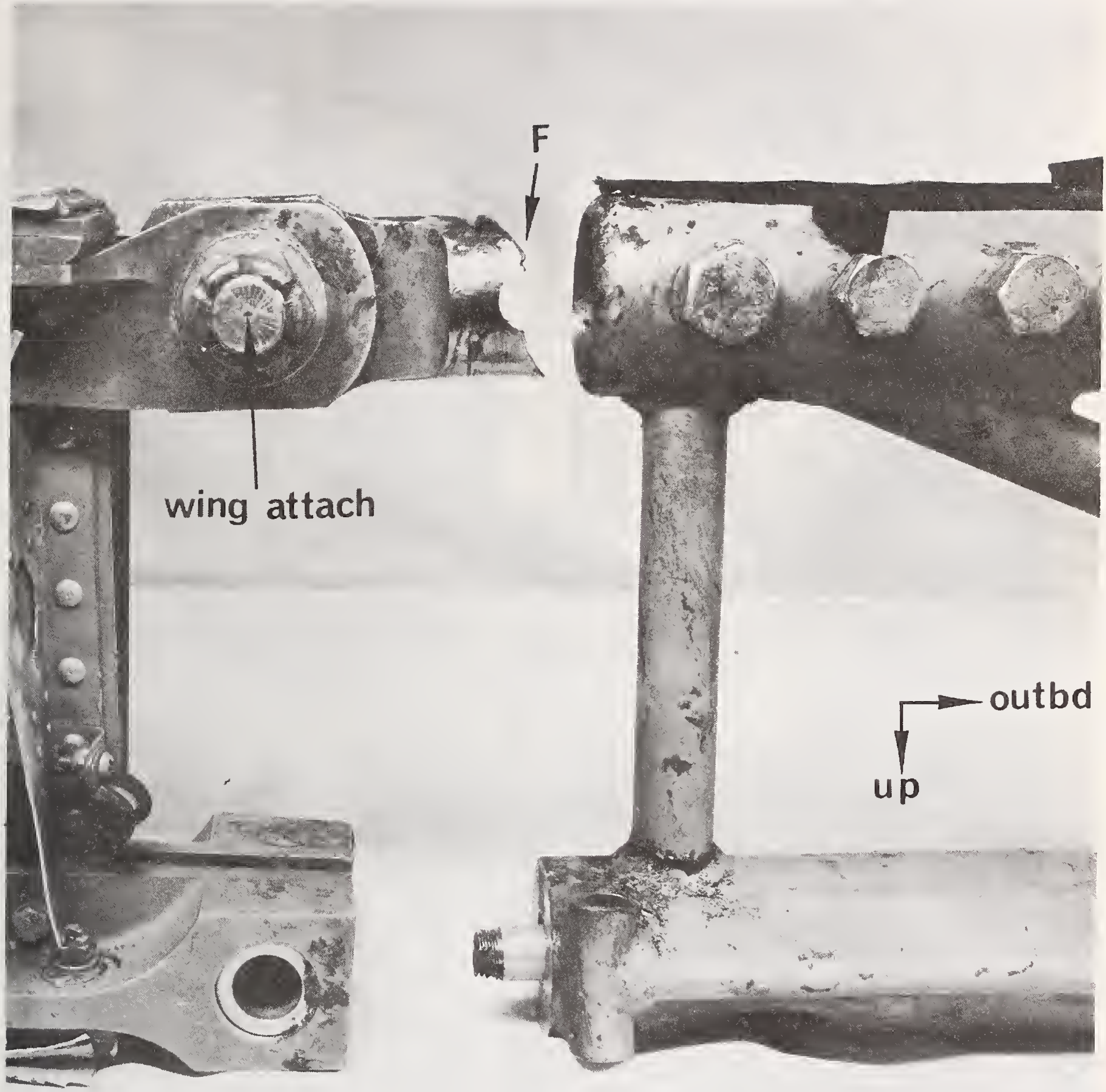

Figure 3. Fatigue fractured fitting, arrow "F", in relation to mating structure. Accident Helio aircraft H-295 on August 21, 1971.

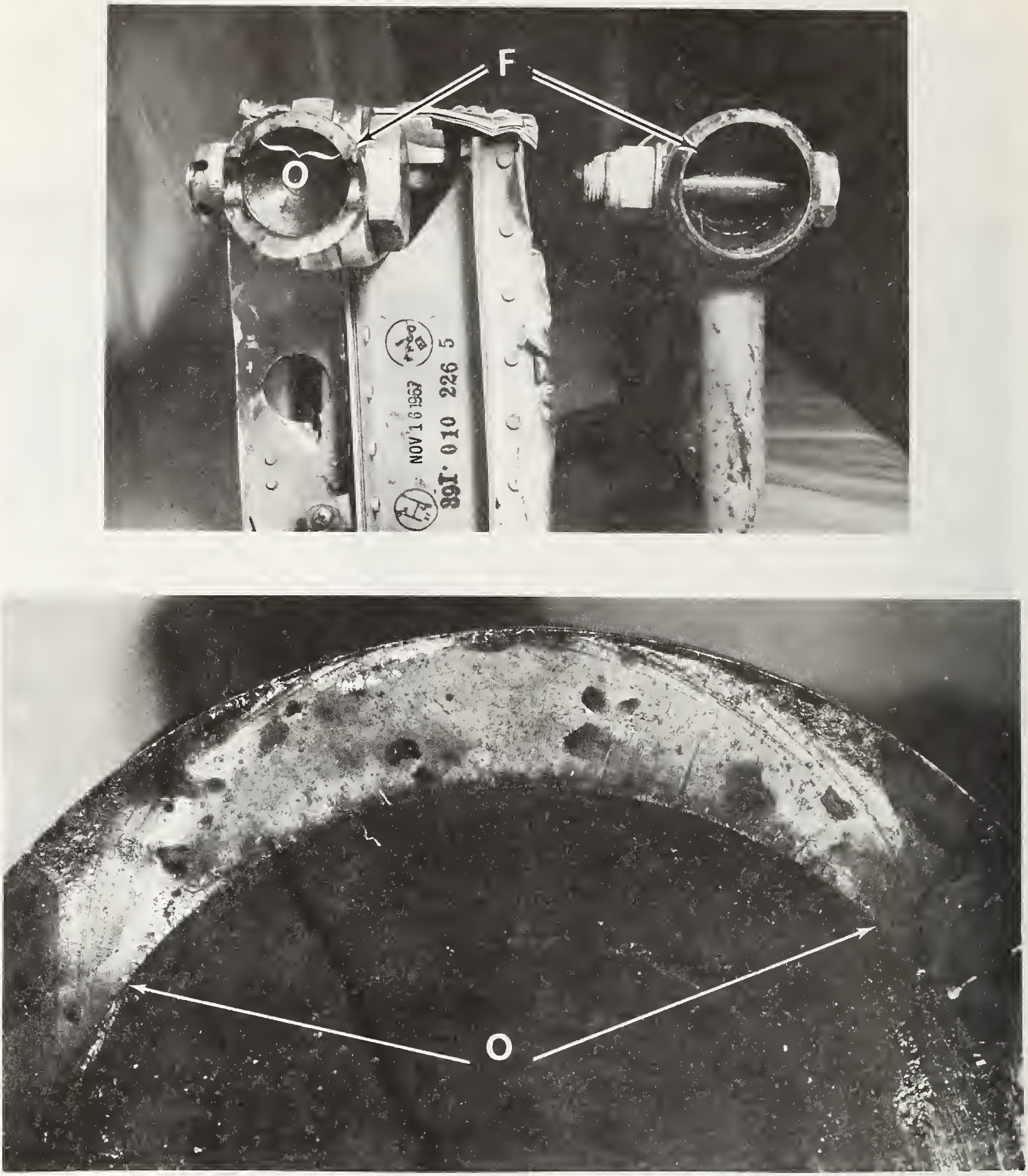

Figure 4. Overall and closeup views of fractured fitting. The fatigue fracture originated from numerous origins on the inside diameter of the hole shown by bracket "O" in top photograph which is the same as that shown between arrows "O" in the bottom photograph. Accident Helio aircraft H-295 on August 21, 1971.

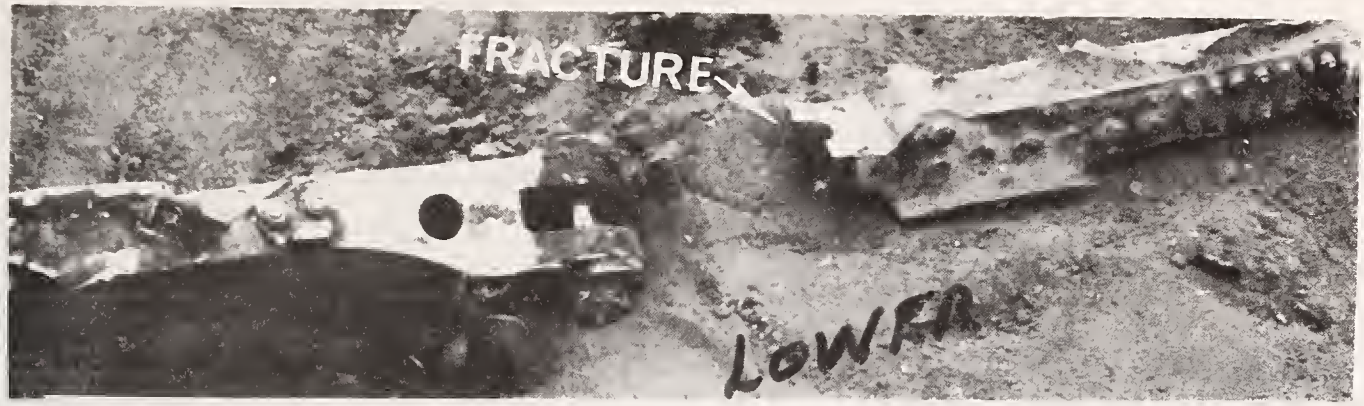

Figure 5. Lower wing root attachment area as laid out at the accident site. Carry through spar is shown on left and fractured wing attachment assembly is shown on right. Accident DeHavilland DH-104-7AXC, June 6, 1971.

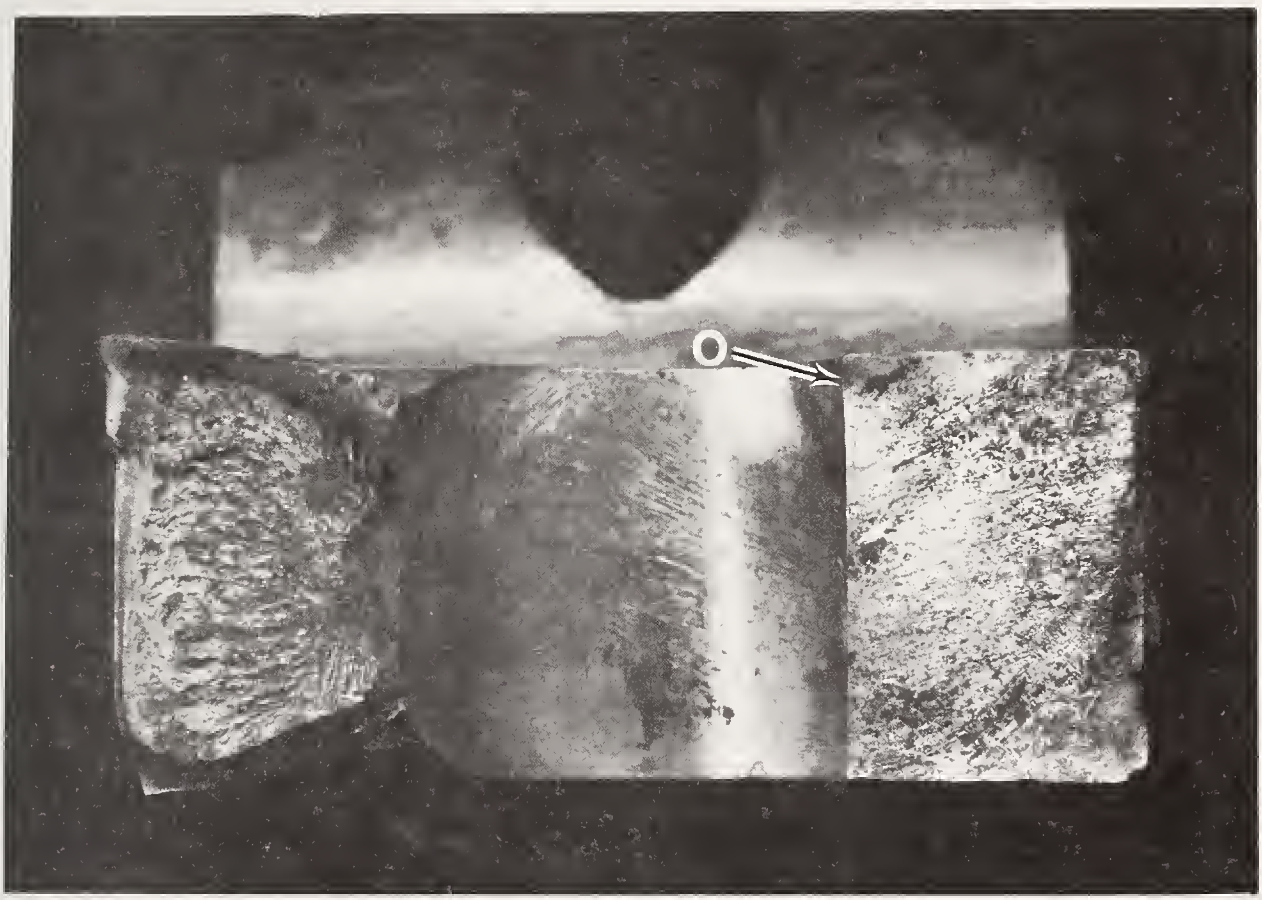

Figure 6. Closeup view of fracture surface of lower wing attach fitting. Arrow "O" denotes origin of fatigue fracture. Accident DeHavilland DH-104-7AXC, June 6, 1971.

DISCUSSION

R. Lenich, Caterpillar Tractor Company: Was the overload failure due to an over gross weight condition or due to aerodynamic overloading?

M. L. Marx: Aerodynamic overloading.

J. Stern, NASA, Goddard Space Flight Center: You mentioned that a probable cause of failure is inherent design. Would you explain what you mean by inherent design?

M. L. Marx: It is a term that I use to describe parts that, as designed and manufactured, are inherently susceptible to fatigue failure.

J. A. Bennett, National Bureau of Standards: The Beech-18 keeps turning up at these meetings. At the time that we looked at the Beech-18 accident in 1964, we referred back to the 1947 accident. The whole approach was to try to save these aircraft by radiographic inspection. This approach obviously did not work. At that time there was complete refusal to even consider modifications such as grinding off the weld reinforcements or shot peening the area to improve the fatigue strength. Is that still the case?

M. L. Marx: Modifications were made eventually. In the past, modifications were resisted because it was felt that inspection requirements would be adequate. In the investigation of the last accident, we found that the cracks were growing very slowly. These cracks could be seen on radiographs taken as early as 6 years prior to the accident. As it turned out, the inspections that were performed were sometimes adequate to detect the cracks, but the people that were reading the radiographs just didn't see them.

H. Ferris, Hughes Helicopters: It looks to me that design is blamed for failures whereas the failures really appear to be related to quality assurance. What was done about the decarburization which caused the strength to be half of what it should have been?

M. L. Marx: The design permitted up to 0.15 in decarburization.

H. Ferris: I thought that 4130 steel heat treated to 180 to 200,000 psi was specified.

M. L. Marx: Right.

H. Ferris: But that was not obtained.

M. L. Marx: The material was heat treated to the proper strength. These aircraft were designed in about 1925, and at that time, the applicable design specifications permitted a certain amount of decarburization.

USE of CASES in ENGINEERING EDUCATION
THEIR SPECIAL VALUE

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Case studies such as we have heard in these sessions can be of immense value in educating engineers, particularly when they become case histories suitable for use in classrooms. Their contribution as a supplement to the other media which we use is unique.

Let us compare the work of learning engineering to the work of learning a language: We find that in engineering education the greatest fraction of time is spent in learning the scientific background. In learning languages this would correspond to grammar, the general rules of the language. A smaller fraction of time is spent in design projects or other project work. In language learning this corresponds to composition in the foreign language. A very small amount of time, frequently none at all, is spent in observation and discussion of the work done by engineers. In language studies this corresponds to reading and discussion of the literature written in the foreign language.

Engineering education is unique, I believe, in omitting from most of its curricula the study of the works of practitioners. Language students study literature, music students listen to recorded music, art students go to museums.

The place of case histories in engineering corresponds to the place of literature in the study of language or to that of records in the study of music.

An engineering case history is an account of an engineering job as it was actually encountered, told with the circumstances of time constraints, resource limits, and personalities which influenced the outcome of the job. A technical report which tells only the problem and its solution is not an engineering case: It omits the circumstances and all the efforts which might have been avoided if foresight had been as good as hindsight. Cases taken from their own experience were often told by professors who were also practicing engineers. Today very few engineering teachers have had responsible engineering experience.

Our collection of written case histories provides material which can supplement lectures, problem sessions, laboratories, and projects by providing snapshots of outside reality for study in schools. These written case histories have some advantages over oral accounts by experienced engineers: They can be studied more leisurely, they can be discussed more critically, and they can easily be used as sources of problems and role-playing situations.

The ASEE-Stanford Case Library is a central clearing house for engineering cases. It was started ten years ago with an NSF grant to the late Professor John Arnold and is now supported by a few industrial sponsors. It has about 200 case histories available in pamphlet form for distribution to students. These pamphlets have been used in classes by well over 100 schools. Almost as many schools keep bound volumes of the case collection available as research material.

The sources of case histories are varied: For instance, both Professor Smith and Professor Kardos have contributed to the collection. Some of their cases are taken from their own experience, others are skillfully assembled from historical accounts or from newspaper articles. One of these deals with the collapse of false work under a bridge which led to the deaths of many people. It is a fine example of the many uses to which a single case can be put: I have used it in a class on failure prevention for problems in the analysis of buckling. Professor Kardos has used it to discuss professional ethics. Others have used it to bring out concepts in liability and in engineering organization. Each of these different uses is enhanced by the fact that the problem which we analyze is embedded in the circumstances of the case.

Professors are not the only contributors of cases. Graduate students have written excellent cases under the supervision of their professors and practicing engineers have contributed case histories from their experience. We have also made available some previously published case histories, for instance, some of the incidents reported by the Flight Safety Foundation, and Mr. Graham's article on the development of his variable speed transmission which he entitled, "Twelve Years to Discover the Obvious".

To me it seems obvious that the study of specific instances, reported with the circumstances surrounding them, is essential to an engineering education. This does NOT mean that cases should replace lectures, textbooks, laboratories and projects. Cases should supplement them.

When they first begin to study case histories, students often are reluctant to believe that good engineers can be so human as the cases show them, so fallible, so lacking in complete knowledge. To me, this reluctance indicates the need for case histories; it suggests their power to show engineering, as it really is, to engineers and to others, and it shows why we must restrict ourselves to true histories. We might perhaps invent better stories, but we could not overcome un-belief with them.

To sum it up: We need good true case histories. They can be used in many different ways. We have an engineering case library as a central clearing house for cases. We would like to receive good case histories from you and _your_ ideas on the use of cases in engineering education.

WHAT CAN BE LEARNED FROM CASES

G. Kardos
Carleton University
Ottawa, Canada

Introduction

For the last three years we have been running a core course in engineering design with the entire lecture period devoted to Case discussions. Because the use of Engineering Cases is still comparatively new, we have had the students make a course evaluation at the end of the term. The purpose of this evaluation is to provide a feedback as to what students felt they had learned from the Cases.

The Course

At Carleton University the four year engineering program consists of a three year common core and a final year of professional specialization in Civil, Electrical or Mechanical engineering. Because of this structure, the core classes must emphasize fundamentals. Engineering 82.323-Engineering Design Studies, was introduced in the second term of the junior year to give the students information about the design process and engineering practice.

To meet this objective it was decided that the course would not have a specific information content. The format of the course was two hours a week of Case discussion and one three hour laboratory period for design projects. The design projects were of the "make and break" type in which the students worked in teams on a competitive basis. The Cases used for the discussions were from the Stanford Case Library. Approximately eleven Cases are used during the term. A final written critical essay was required on one of the Cases.

The Case discussions were carried out in groups of approximately 25-30 students. Over the three years six different instructors were involved. Only two instructors participated all three years.

The specific objective of the course was laid out at the beginning as follows:

To give the student a realistic view of and various experiences in engineering design as actually carried out in industry.

To develop the students skills in using and appreciating the analytic tools he has and to develop his engineering judgement.

To demonstrate to the student how he may use his learning skills to tackle engineering situations beyond his immediately available analytical tools.

To give the student a real design experience with hardware feedback.

The Questions

The questions asked of the students in the evaluation are shown in Figure 1. This questionnaire was originally developed by Karl H. Vesper at Stanford University. In generating the original questionnaire Vesper had classified the questions into four categories: Skills (1-6), Habits (7-15), Knowledge (16-21) and Attitudes (22-29). Although these categories were retained they were not identified as such on the evaluation sheets.

In Vesper's original questionnaire students were asked to rate the Case method with respect to other teaching methods on a scale from 1 to 5. Although we found the questions the kind we wanted to ask, we found the rating method somewhat difficult to interpret. For our purposes we changed the rating to have the students indicate their relative agreement with the statements. The degree of agreement to be indicated as:

1. Disagree
2. Can't tell
3. Probably
4. Agree
5. Strongly agree.

Although both Cases and projects were rated, in this presentation I will confine my remarks to what the students felt they had learned from the Engineering Cases.

The Results

The results of the responses of 300 students over three years are shown in Figure 1.

For our purposes, we interpret a response of "Probably (3)" or better as showing that some learning of the type indicated had taken place. A response of "Disagree (1)" was interpreted as showing that learning of the type indicated had not taken place.

With the above interpretation, what did students feel they had learned from Cases?

Better than 80% of the students felt that Cases had probably:

increased their appreciation of what engineers do. (17)

increased their appreciation of criteria for judging engineering procedures and for evaluating design. (18)(19)

increased their ability to define practical problems and to spot key facts amid less relevant data. (9)(10).

increased their tendency to discriminate between fact and opinion and to search for more alternative solutions (2)(3).

In addition 70% of the students felt that Cases had probably:

increased their appreciation of Historical episodes as lessons.(16)

increased their appreciation of technical facts in engineering other than theory. (21)

developed their ability to view problems with perspective. (14)

developed their ability to sell their own ideas as well as increase their tolerance for other's ideas and errors. (15)(29)

More than 60% of the students, in addition to the above, felt that Cases had probably;

increased their ability to foresee consequences of alternative action, to think more carefully before speaking, to prescribe actions more specifically and developed their ability to communicate. (11)(6) (4)(8)

increased their knowledge of their own abilities and limitations.

Certain kinds of learning did not take place with our use of cases.

Forty percent of the students felt they had not learned anything about formulating idealized mathematical models of real problems or about how to manipulate such models for solutions from the Cases. (12)(13)

In addition, 30 percent of the students felt that Cases had not contributed to:

their ability in using unfamiliar tools or methods (7);
their self confidence (22);
their aesthetic sensitivity (26);
their enthusiasm or motivation toward engineers (25);

Learning From Cases

As a result of this evaluation we can say something about what can be learned from Cases as we have used them. Students can learn about what engineers do and about the criteria necessary to carry out a real engineering project. Students learn the necessity of properly identifying and defining the real problem and to look for alternate solutions. With Cases students learn to interact with their peers in a cooperative and tolerant manner toward the solution of a common engineering objective.

We can also say something about what we have not been able to do with Engineering Cases.

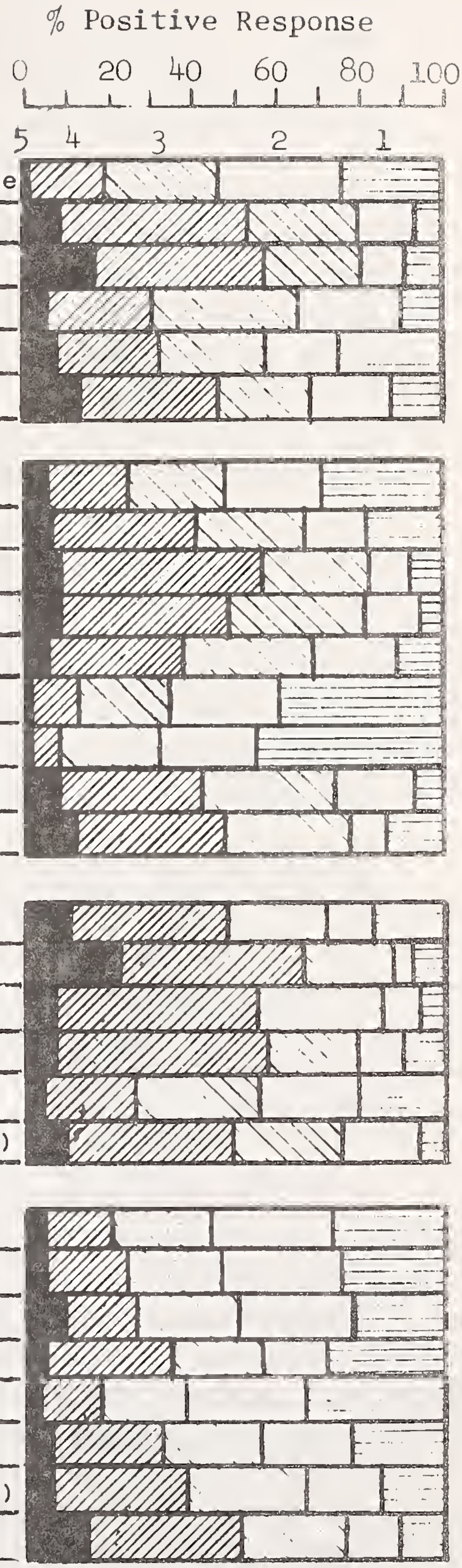
The students have not learned or appreciated the use of the mathematical and theoretical tools available to them. This may not be the fault of cases but in the manner in which we have applied them. In the future we plan to make changes which may correct this.

Cases have not been successful as a medium for motivation. This result is most unexpected. Instinct would suggest that cases would be highly motivational; for us this has not proved to be so. We have speculated about why, but we have been unable to come up with a reasonable explanation.

Figure 1

CARLETON UNIVERSITY
Faculty of Engineering

Engineering Design Studies
<u>Engineering Case Evaluation</u>
300 Responses



323 has increased my tendency to:

1 Reason quantitatively (use numbers) whenever possible
2 Discriminate between fact and opinion
3 Search for more alternative solutions
4 Prescribe action more specifically
5 Pay meticulous attention to detail
6 Think more carefully before speaking

323 has helped develop my ability of:

7 Using unfamiliar exercise tools or exercise methods
8 Communicating (writing or speaking or drawing)
9 Identifying and defining practical problems
10 Spotting key facts amid less relevant data
11 Foreseeing consequences of alternative actions
12 Formulating idealized math models of real problems
13 Manipulating and solving idealized math models
14 Viewing problems and complications with perspective
15 Selling ideas or arguing more persuasively

323 has increased my appreciation of:

16 Historical episodes as lessons
17 What engineers do
18 Criteria for judging engineering procedures
19 Criteria for evaluating designs
20 Mathematical, scientific, or engineering theory
21 Technical facts used in engineering (besides theory)

323 has increased my:

22 Self confidence
23 Perseverance in spite of setbacks
24 Concern with questions unanswered for me yet
25 Enthusiasm, motivation for course or engineering
26 Aesthetic sensitivity
27 Interpersonal sensitivity (feel for people)
28 Self knowledge (own abilities and limitations, etc.)
29 Tolerance for others' ideas and errors

STUDENT WRITTEN DESIGN CASE STUDIES

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Case studies are of immense value in the education of engineers, both in and out of the classroom. When they are written for classroom use, they become a unique supplement to the more traditional classroom tools of lecture, problem solving, etc. In using case studies or case histories, there is an opportunity for both writers and users to profit from them. It is recognized, however, that the potential profit for the writer is substantially greater than for the user.

As a consequence, I have offered a course at the University of Detroit called "Design by Case Study" at the graduate level. The objectives of the course are: (1) become acquainted with the case method; (2) gain greater insight into design and engineering practices; (3) assess the decision making process as it functioned in a series of cases studied; and (4) write a case study.

During the first five weeks (about 1/3) of the course, the class studied, prepared written discussions, and then verbally discussed a series of case studies. Some of these were from the Engineering Case Library at Stanford University. Others were student-written case studies generated at the University of California at Berkeley under the direction of Dr. R. F. Steidel.

During the next five-to-six week period, the class worked in small teams (usually three students per team) in the "real world". Each team worked with an engineer who was in responsible charge of a project in a local engineering organization. Both the engineer and the project were rather carefully selected in advance in concert with the cooperating organization, since the project engineer served as a "clinical professor". The instructor accompanied the team on its initial meeting with the project engineer at his place of work but not on subsequent visits. During this period, the class did not meet together but the instructor was available for consultation. Thus, the team had to function by itself under non-academic circumstances.

The first time this course was offered, the team visited the project engineer, listened to his exposition and asked questions (recording the discussion on tape). Between interviews, the team went through the results of the interviews and any file documents and drawings loaned to the team. The team members were expected to review the design from concept to completion (including field performance if possible) with particular attention to chronology, key decisions (and factors involved in reaching them), wrong directions taken as well as successful directions, and critical review of technical aspects.

The second time the course was offered, there was some shift in emphasis. This time, the team again met with the project engineer and discussed the background of the problem and arrived at a problem definition. The team then was expected to try to develop a conceptual solution. On each successive visit with the project engineer, the work of the team was reviewed by the project engineer and discussed. Additional information was presented to the team and they proceeded further with their conceptual solution (or substantially altered it, as appropriate). At the end of the series of visits, the project engineer told the team how the problem had actually been solved in practice.

In both modes of emphasis, the team had to prepare a written case study. A due date was established for a draft with copies being provided for the other students as well as the instructor. Each team was given time to prepare a critical review of the work of each of the other teams. The class then met with each team presenting its study and the other students asking pointed questions. Following this, each team had an opportunity to confer again with the project engineer, fill in gaps in the study, and develop a final written draft. At the end of the course, each team made a public oral presentation of its study. Faculty, project engineers, and other students were welcome at the final presentations.

This is not a "cake" course (for students or instructor) with each team producing a completed document of perhaps 50 to 80 pages, especially relating to a design project in an area in which they may have classroom background but little practical experience. The reaction of students, project engineers and instructor is that the students have had a valuable experience. It has been agreed that they gain a greater insight into real product development with its interplay of analysis, testing and decision making, than would be possible through academic study alone. In at least one case, the student team proposed a conceptual solution which the project engineer admitted had not occurred to the industrial team but which appeared superior to the one that had been developed.

Appendix

Student-Written Case Studies and Cooperating Companies

FAST - A New Friction and Wear Test for Brake Linings
Ford Motor Company, Dearborn, Michigan

Development of an All-Paper Asphalt Container
Owens-Illinois, Toledo, Ohio

Development of Military Industrial Standard Engines with Specific Consideration of the Governor
Teledyne Continental Motors, Warren, Michigan

Design and Development of a Brush Burning Plant
City of Detroit, Detroit, Michigan

First Harmonic Analyzer
Uniroyal, Detroit, Michigan

Automatic Cam Brake Adjuster
Eaton, Yale and Towne, Southfield, Michigan

Transformer Load Management Research Program
Detroit Edison, Detroit, Michigan

Design of the Pocket Select Gate for the Burroughs B9134-1 Reader Sorter
Burroughs Corporation, Detroit, Michigan

Evaluation of Passenger Car Eyellipse for Visibility Requirements
Ford Motor Company, Dearborn, Michigan

A New Air Flight Control Power System
Sperry Vickers, Troy, Michigan

These case studies have been reproduced and are available at cost.

TEACHER FEEDBACK ON THE USE OF ENGINEERING CASE STUDIES

Byron J. Pelan
Associate Professor of Mechanical Engineering
Department of Mechanical,
Industrial, and Aerospace Engineering
Rutgers University
New Brunswick, N. J. 08903

How do I as a teacher of mechanical engineering design and a former practicing engineer regard the engineering case method? Let me lead into my response by first briefly commenting on several considerations.

In recent years there has been a widespread movement in engineering education to expose the student to realistic engineering situations, sometimes contrived but often from real life. In part, our reason has been for stronger motivation. We have also believed that somehow we could then better prepare him as an engineer. I would be less than candid if I did not include our concern for an antidote to the attrition and disillusionment of some engineering students. There are other reasons, too, of course. Part of this widespread movement has been the institution of programs in which the student becomes an intelligent, active participant. The use of the engineering case method, the authentic involvement of students in industrial problems, and the selection of design projects that culminate in such useful hardware as prosthetic devices and unique instrumentation are instances of this movement. The particular approach adopted may depend upon several parameters, including time limitations, teacher preference, and industrial support as in the instance of authentic student involvement cited. However, the adoption of one approach does not preclude the adoption of another. Indeed, the use of more than one can have a reinforcing effect. A properly-selected case study can provide background for a design project and also motivate the student by adding the realistic ingredient of an actual engineering situation. My comments here are addressed to the case method, in particular to case problems and case histories that are available, not the writing of them. As you are aware, in a case problem the student is in the position of an engineer facing a problem to be solved whereas in a case history the student has the opportunity to see the problem and the outcome.

Primarily the teaching method to be used depends upon the teaching objectives. For example, to more nearly individualize instruction, self-paced instructional programs complete with special course material, including text, audio-visual tapes, and self-administered progress checks, are currently used by a number of colleges. Traditionally, although possibly not always with complete justification, many engineering courses have been taught using the lecture-recitation method. It is frequently the most viable method to employ, the economics alone being a major consideration. In such courses, some faculty, wishing to implement motivation of their students, have included case problems to illustrate and to add realism. For instance, case problems have been used in undergraduate courses as a supplement to conventional problems to show how practicing engineers attack real problems. Professor Hirsch of the U.S. Naval Academy has introduced a real case concerned with the dynamics of a recoil-operated, automatic weapon at a timely point in his first course in undergraduate engineering mechanics [1][1].

The most likely teaching objectives of the case method in engineering were investigated by Professors Vesper and Adams [2,3]. Three groups were polled: professors who had attended summer institutes where they had written cases and practiced on them, professors who taught case studies on the first-year graduate level, and, lastly, graduate students in that course. All groups placed high emphasis on these teaching objectives:

1. Knowledge of what engineers do and how they work
2. Enthusiasm, motivation for course or for engineering
3. Search for more alternate solutions
4. Identification and definition of practical problems
5. Discrimination between fact and opinion
6. Detection of key facts in the midst of less relevant data
7. Need to be more specific in prescribing action
8. Need to foresee the consequences of alternative actions

It is interesting to note that another survey in their investigation revealed that a second group of graduate students participating in a four-week summer institute composed entirely of cases agreed that two major effects of case instruction were exercise in thinking and increased realism [2].

Case studies may be used at different levels. It is not unusual to use the same case at several instructional levels: freshman, senior, and graduate. At the first level it may be used to expose the student to the real engineering situation and have him prepare engineering sketches of student-generated alternatives. The senior may find the case a justification for the time spent on fatigue failures in his stress-strain-strength course and may even have that delight of detecting the reason for the failure and of being able to competently recommend an alternative correction. For the graduate student, on the other hand, here is an opportunity to diagnose the case from an administrative viewpoint.

Cases have been assigned in a number of ways, as you have come to suspect: homework reading assignments, problem formulation, background, illustrations, research material, and so on. Normally, I prefer at some

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[1]Figures in brackets indicate the bibliography references at the end of this presentation.

point to have group discussion. Such a discussion may require more than one class period; for instance, it may be in sequential parts with the students not receiving the entire story until later. My students have had an opportunity to review each part prior to discussion. A group discussion gives the student an opportunity to do ~ and ideally should demand that he do ~ independent, constructive thinking and make positive contributions. Additionally, it provides the real environment of an engineering case as background. Initially I find that the attention of the class is divided between the case and myself, but the case and the developing discussion must become the prime focus of their attention. The case presents an opportunity for detached objectivity, free from the emotional involvement of the original participants. At times the student may be called upon to defend his position in open discussion. This is part of his opportunity to grow. This total experience, we hope, eases his transition from college to industry. Exposure to a number of cases is often desired. Ideally in such instances the point is approached where he does make independent, good judgments. But not all students can or do achieve this and not all faculty would be successful with case studies. Good cases in poor hands can be disastrous; poor cases in good hands still might be used effectively.

The role of the instructor is unique. In a good case activity, he is present, prepared to intervene as subtly as needed but nearly forgotten by the students. As Professor Kenneth Andrews of the Harvard Graduate Business School has stated in his paper entitled "The Role of the Instructor in the Case Method": "However his judgment prompts him to behave in the classroom, the instructor plays a multiple role. He is student, listener, and analyst. He is questioner, paraphraser, and minuteman lecturer. He plays these parts without costume changes, and he never steals the show from the rest of the cast" [4].

In conclusion, I would be delinquent if I did not mention that the teacher, too, benefits ~ and in many ways. First, he himself has had an opportunity to grow, for as he left the structured teacher-student relationship of the traditional lecture-recitation format, he also left the environment of almost predictable student response patterns. Now, he must play that role which will demand all of his skills as a teacher. Second, he has an opportunity himself to be exposed to different, often exciting engineering situations and thereby to add to his own professional experience.

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CASE STUDY ILLUSTRATION

(Illustration referred to at the closure of "Teacher Feedback on the Use of Engineering Case Studies")

I have a double purpose in taking a further moment of your time to describe one case: first, to show one way a case has been composed and, second, to hope this may suggest to you a problem that you have encountered that might be used for a case study.

Technically, the case* I have selected concerns the failure of the front-end ball bearing of an electric motor driving an aircraft pump. Figure 1 shows the condition of this bearing after 1800 hours of operation. This bearing and the one at the other end of the motor shaft were cooled by immersion in the circulating hydraulic fluid. Figure 2 shows a cross section of the pump and motor assembly. Considerable documentation has been provided for the study. For instance, Figure 3, which is my composite shows some of the typical data that has been included: there we see an excerpt of the specifications of the bearing from the manufacturer's catalog and also an excerpt of the specifications of the hydraulic fluid. Other documentation and commentaries re-establish in part a sense of the original atmosphere of concerns and urgency. The motor is but one of 300 on order. Each is specified by contract to have a failure-free life of 2500 hours at least. Yet, the first failure occurred during the life qualification test after the motor and pump had been connected. But, by then, tooling had been finished, production started, and some motors already had been made. The pump manufacturer had also started production. There were several conferences. The bearing manufacturer was consulted. Figure 4 shows a portion of the manufacturer's letter of analyses and recommendations. As a consequence, a different bearing was substituted on all units. Then again a failure ~ it was the substitute bearing. The case progresses rapidly. Outside consultants were engaged; there were differing opinions. Then another failure of the bearing occurs. Again questions are raised. Could it be that the pump produced an axial loading even though no reference was made to one in the original specifications? Could there have been a misalignment or a binding? Thus, the case develops and, as it does so, opportunities are presented to explore more than just the technical aspects if desired.

*Vesper, Karl H., "Task Corporation ~ Failure of a Ball Bearing" Stanford University Case Library No. ECL-14.

Exhibit 1 - 204SST5 Bearing After 1800 Hours Operation

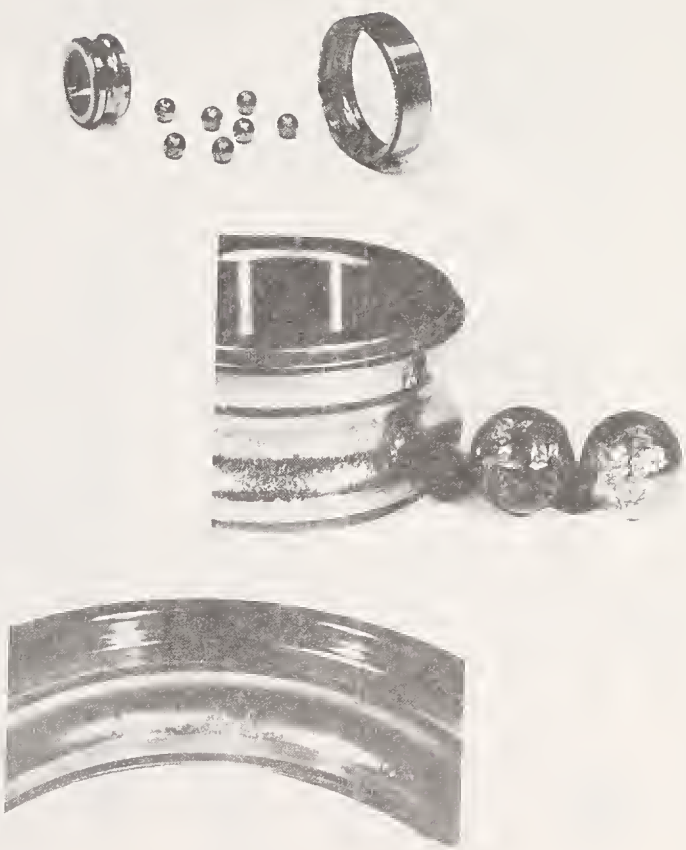
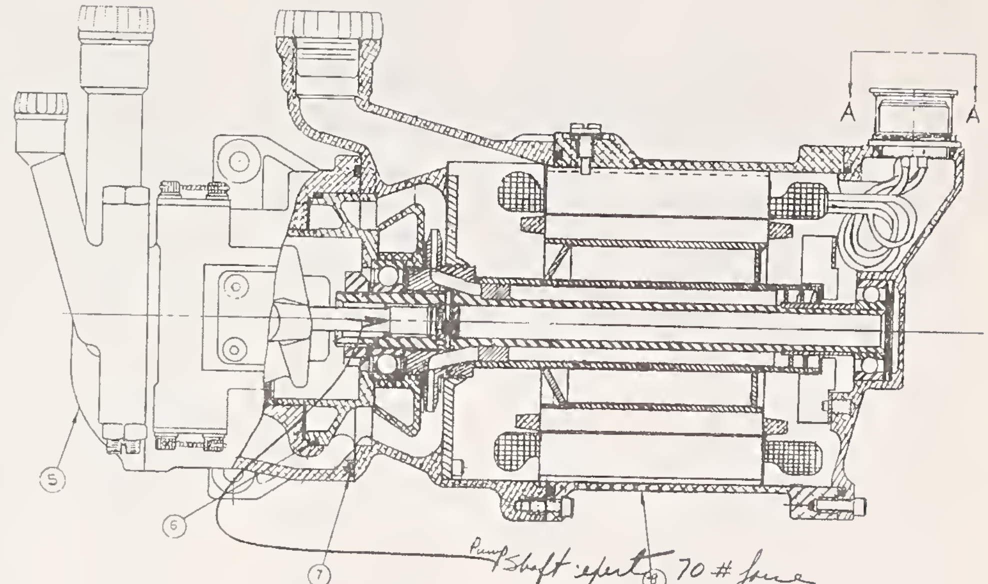


Fig. 1 Pump-end bearing after 1800 hours of operation



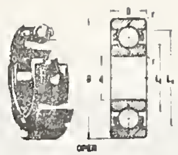
Pump shaft exerts 70 # force in direction shown on plug in motor shaft. (Picture slightly off.)

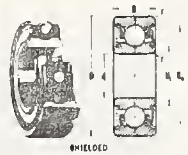
Fig. 2 Section view of motor and hydraulic pump

BARDEN PRECISION SPINDLE AND TURBINE BEARINGS

LOW TO
ULTRA HIGH SPEED

DEEP GROOVE
STEEL OR PHENOLIC
RETAINERS





AVAILABILITY
All sizes listed are normally available from stock in chrome steel.

DIMENSIONS

BORE mm	BORE inches	O.D. mm	O.D. inches	WIDTH mm	WIDTH inches	MAX FILLET RADIUS inches r	Steel Retainer Open	Steel Retainer Single shield	Steel Retainer Double shield	Phenolic Retainer Open	Phenolic Retainer Single shield	Phenolic Retainer Double shield	Other Dimensions inches L_1	L_2	O_1	U_1	APPROX WEIGHT pounds	BOT ...
15	5906	32	1 2598	9	3543	012				102T			798	1 053			07	10
15	5906	35	1 3780	11	4331	025	202K	202S	202SS	202T	202ST	202SST	815	1 153	755	1 223	10	20
15	5906	35	1.3780	12.70	5000	025					202STX1	202SSTX1			755	1 223	20	20
17	6693	35	1 3780	10	.3937	012	103K	103S	103SS	103T	103ST	103SST	895	1 153	835	1 215	09	10
17	6693	40	1 5748	12	4724	025	203K	203S	203SS	203T	203ST	203SST	952	1 292	890	1 372	20	203
20	7874	42	1 6535	12	4724	.025				104T	104ST	104SST	1 050	1 390	989	1 458	18	104
20	7874	47	1 8504	14	5512	040				204T	204ST	204SST	1 130	1 530	1 060	1 610	20	204
25	9843	47	1 8504	12	4724	025				105T			1 247	1 587			21	105

Exhibit 4 -

Properties

SKYDROL SPECIFICATIONS FOR NEW FLUIDS

Property	Skydrol 500A	Skydrol 7000
Appearance	Clear, purple liquid	Clear, green liquid
Neutralization number, mg KOH/gm.	0.10 maximum	0.20 maximum
Specific gravity 25°C./25°C.	1 060 to 1.066	1.080 to 1.086
Viscosity, CS		
@ 210°F (99°C.)	3 90 to 4.00	3.95 to 4.05
@ 100°F (38°C.)	11.30 to 12.10	15.00 to 16 00
Moisture, per cent	0.20	0.25

TYPICAL PHYSICAL AND CHEMICAL PROPERTIES

Property	Skydrol 500A	Skydrol 7000
Appearance	Clear, purple liquid	Clear, green fluid
Odor	Mild, pleasant	Mild, pleasant
Autogenous ignition Temperature, (ASTM D 286-58T)	Above 1100° F (593°C.)	1060°F. (571°C.)
Pour Point, (ASTM D97)	Below −85° F (−65°C)	Below −70°F (−57°C.)
Neutralization number, (ASTM D-974-58T)		

Fig. 3 Composite of excerpt from catalog of bearing specifications and excerpt of hydraulic fluid specifications

C O P Y

EXHIBIT I

THE BARDEN CORPORATION
Barden Precision Ball Bearings
Danbury, Connecticut

September 11, 1963

Mr. Elmer Ward, Chief Engineer
Task Corporation
1009 East Vermont
Anaheim, California

Dear Mr. Ward:

This will summarize our recent telephone conversations and letters regarding the failure of a Barden Precision 204SST5 bearing in a pump motor qualification unit and our recommendations to prevent recurrence of such failures. The pump runs at 6000 rpm in Skydrol at 150°F with a thrust load of 75 lbs on the 204SST5 bearing and 20 lbs on the opposed bearing, a 203SS5. The failure in the 204 size bearing was experienced at 1800 hours.

Examination of the 203SS5 bearing showed that it operated with normal contact angle in the presence of considerable contamination. There is also a very light running band where the bearing apparently operated under a reverse thrust condition. In spite of the evidence of

Fig. 4 Copy of portion of letter from bearing manufacturer

PANEL DISCUSSION: Use of Cases in Engineering Education

Panel Moderator: George Hurt, Detroit Diesel, Allison Division

Panel Members: Henry O. Fuchs, Stanford University
Geza Kardos, Carleton University
Byron Pelan, Rutgers University
C. O. Smith, University of Detroit

G. Hurt: Have you tracked any of these former students into industry and seen whether they could, or have, answered any of these same questions after they have had some experience?

H. O. Fuchs: Without deliberately tracking them, I have gotten some volunteer responses from students. One of them told me that the case course that he took had been more valuable to him than most of the other courses that he had taken. We got a similar response from another student who came back for a master's degree after working in industry.

C. O. Smith: In the graduate case course that I was talking about, we have some part-time students with 5 to 10 years experience. Without exception, these students said that the course was beneficial.

G. Kardos: My experience is the same. We have had this course for 3 years and it is a little too soon to get any direct feedback. But, anyone who has any working experience immediately recognizes this course as being useful. The working engineer seems to be much more responsive to the course than the new, green engineer.

R. E. Maringer, Battelle Memorial Institute: My feeling is that this is an ideal way of teaching. It takes a student who does not know anything about industrial life and gives him a little more of a feel for what he is going to be doing the rest of his life. It seems to me one way of judging whether the case studies approach is effective is how broadly it is used. Is it increasing in use; are there demands for these case studies?

H. O. Fuchs: Last year there were 7,000 copies of case histories requested from us at cost. The case studies approach is increasing in use. My guess is that it is used in half the schools, and in about 3% of the classes taught in those schools.

G. Kardos: Stanford has a very liberal policy on distributing case studies. If you want to buy it at cost, they will let you have it at cost. If you do not want to pay cost, they will be glad to save you the money by letting you reproduce it yourself. I am sure there are a lot of copies of case studies being used that we don't know about--there is no way of getting the feedback.

C. O. Smith: Bob Maringer used the term "ideal". My response is that it is not ideal--it is realistic. Teaching a course with case studies is not as easy, or as comfortable, for the instructor as a lecture course. Once you have lived with it for a while, it becomes more interesting. If you get good interaction among the students, you may learn something.

A. Wolff, Massachusetts Materials Research, Inc.: Do you use real industrial cases from your area so that the students can meet with the industrial people involved?

C. O. Smith: Yes, they have worked with people from the city of Detroit, Ford, Bendix and others.

A. Wolff: Do the students deal with the quality control managers and the technical managers?

C. O. Smith: They deal with whoever is responsible for the project.

G. Wagner, Westinghouse Electric Corporation: Do you precede these case history studies with some theory on the logic of design?

H. O. Fuchs: We at Stanford go easy on the logic of design. We do mention it in other courses, but we do not treat it as a separate subject.

G. Kardos: We do not precede the case studies with the logic of design. At the end of the case course, a student must turn in a critical evaluation of the design process from one of the more extensive cases against a written text book design process.

C. O. Smith: To try to teach the logic of design, per se, would be very dull.

G. Wagner: The natural extension of understanding the logic of design is recognizing that engineering is more certain in terms of uncertainties than otherwise. Again, the logical argument is that the student does not understand or appreciate that the nominal design is not necessarily the design that he gets at the end of the production line.

G. Kardos: In some of the essays that I mentioned previously, students often point out that the textbook design process was not followed. But the important thing is that the design worked.

G. Wagner: One other point, when an engineer is exposed to an actual industrial situation, he is rarely alone. Team development is a worthy consideration.

G. Kardos: We agree that team effort is important. We have team projects. When a project is completed, it is marked for performance against specifications by other members of the class.

C. O. Smith: I might add that we also require a design project at freshman and senior levels. The freshman projects consist of two or three-member teams. The senior project is an interdepartmental operation and has teams of three to six members.

SESSION III

OVERSIGHTS AND OVERVIEW

Chairmen: A.K.Wolff
Massachusetts Materials Research,Inc.

B.P.Bardes
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SOME THERMAL PROBLEMS IN THE DESIGN OF FLUID FILM BEARINGS

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The analysis of fluid film behavior in journal and thrust bearings is rather well developed and much useful material is presently available to permit the prediction of performance in such bearings. However, a common error is to concentrate on the fluid film itself and to more or less neglect the total bearing. The total bearing is the fluid film plus the bearing structure that contains it and which allows it to behave as fluid-film theory predicts that it should.

The mechanical design of the bearing does influence its ability to operate properly. Proper mechanical design is often overlooked, especially the effects of temperature and temperature gradients. This discussion will be concerned with these thermal aspects of bearing design.

The fluid film bearing is similar to all other common forms of bearings in that when operating it does have some friction. This friction produces an energy loss that appears as heat. Eventually this heat must be dissipated in some way through radiation, convection, conduction or evaporation until the rate of heat dissipation is just equal to the rate of heat generation. At that point thermal equilibrium will have been achieved and the steady state operating temperature of the bearing will have been established.

A complete bearing analysis must include, in general, a heat balance so that the leveling off temperature of the lubricant and the bearing materials can be predicted and also so that the thermal growth of the bearing package and possible distortion of the housing may be considered.

Details for doing this for both self-contained bearings and for externally-cooled bearings have been set forth in a number of texts on the subject, as for example, References (1) and (2). Because of the importance for industrial applications of the self-contained bearing I would like to make a few comments with special reference to this type.

For self-contained or self-cooled bearings, the housing and its supports should have sufficient bulk to produce an acceptable heat-dissipating surface. The housing surface of such bearings acts as a radiator to dissipate from the bearing by convection and radiation the heat generated by bearing, fluid-film friction. From a maintenance point of view accumulation of oily dirt on the housing is very undesirable. The extra insulating layer may upset the natural heat balance of the bearing and cause overheating. Many layers of paint may accomplish the same result. The heat-dissipating action of a bearing and its housing should never be modified without serious consideration.

In spite of all the obvious variables that enter the picture, the limits of size and speed of industrial bearings that have been successfully operated as self-contained bearings without auxiliary cooling are fairly clear and Fig. 1 represents these limits. Since the curve is based on typical designs it does have value as a rough design measure.

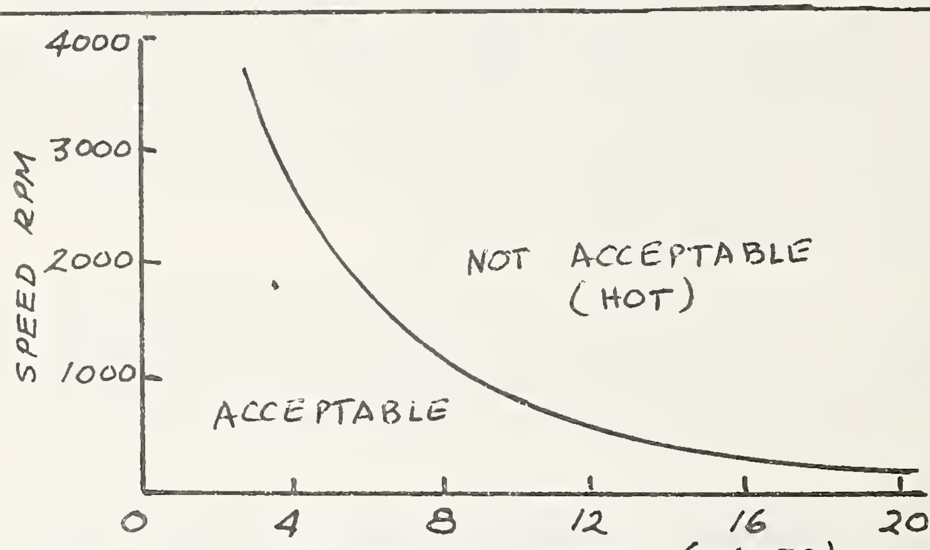

FIG. 1

A rather interesting example concerns a large pump to handle waste radioactive liquids. The unit had a shaft 6 inches in diameter turning at 1800 RPM. Its design incorporated standard Kingsburg journal and thrust bearings. Reliability requirments for the plant, that the pump was servicing, demanded that the unit continue to function for twenty-four hours in the event of failure of the circulating oil to the bearings. This means effectively that the bearings needed to have the cooling characteristics of self-contained bearings rather than those of bearings where the heat was carried away by circulating oil as they were originally designed.

The first attempt to do this resulted in failure. When the circulating oil was shut off, the bearing temperatures continued to rise until seizure took place. There was no hint of a leveling-off temperature being reached. Fig. 2. It seemed as though there could be no hope of ever reaching the goal of having a self-cooled bearing with this design concept.

Before going through a complete analysis and making a heat balance for these bearings we can spot the diameter of 6 inches and speed of 1800 RPM on Fig. 1 and observe that it falls right on the line. This is a quick but approximate indicator that this design concept could be self-contained.

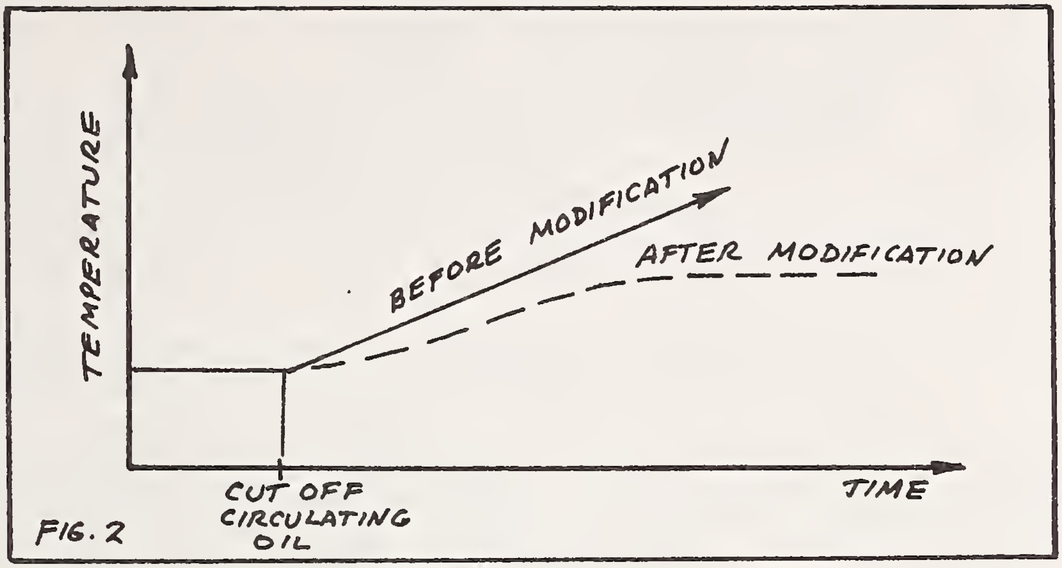

FIG. 2

Further investigation showed that the bearings were very lightly loaded and could be reduced in size without sacrifice of safety. This would reduce the shear area and heating. Accordingly the journal bearings were shortened to about 2/3 of their original length and every other shoe of the thrust bearings was removed. In addition, a new cover with extended ribs and fins was put on the bearing housing. With these modifications the bearing was converted to a truly self-contained, self-cooled bearing. The heat generation was reduced and at the same time the heat dissipation capacity of the bearing housing was increased by extending the surface. No further modification was needed, but if it were, an axial blower could have been attached tothe shaft, blowing air over the housing and thus increasing the rate of heat transfer from its surface.

Bearings operating at higher speeds than those indicated in Fig. 1 do need some form of external cooling. This is often accomplished by forced feeding of the lubricant through the bearing. However the hot lubricant on returning to the reservoir or sump must release its heat, in some manner, to the outside world. Thus the reservoir itself must be evaluated in terms of its heat dissipating capacity.

This is a continuing problem with automobile engines for example, especially when the car is driven at high speed. Cooling can only take place by flow of air over the surface of the crankcase as a result of the motion of the vehicle. If the crankcase is small, with a typical 4 quart capacity, the surface dissipation area is also small, especially if it is not ribbed, and the ability of the crankcase to dump heat into the atmosphere is limited. When super-powered cars are converted for use in stock car racing, under NASCAR regulations (National Association for Stock Car Auto Racing) it is necessary to about double the crankcase capacity and in addition, in many instances, add an oil cooler. Ref (3). Sports cars often have ribbed crankcases to increase the surface cooling area. To the best of the author's knowledge, no stock passenger car manufactured in this country has a crankcase oil cooler. The Mercedes-Benz Diesel engine passenger car has such a cooler and so does an MG sport sedan. Manufacturers of industrial equipment like tractors and trucks recognize this and use extra large crankcases. Transcontinental trucks frequently have separate oil cooling radiators and crankcase temperature control since bearing temperatures in truck engines, with no external cooling, have been measured as high as 310°F, even in the winter.

On rainy days with water spray from the tires and especially on snowy days where slush is being splashed on the crankcase, temperatures as low as 70°F have been measured in automobile crankcases even when driving at 40 to 50 MPH. On the other extreme, high-speed turnpike driving in the summer will show crankcase temperatures approaching 300°F, especially if the body sheet metal work tends to shield the crankcase. There have been instances where this has happened. Such elevated temperatures will tend to soften bearing materials and make them more prone to failure. The service life of oil lubricants is also sharply reduced, due to chemical deterioration.

The comments made above for conventional lubricants and bearing materials apply with some modification to more unusual kinds of fluid film lubrication where the lubricant may be water, acid, liquid refrigerants, molten metals, mercury, gas, etc.

Conditions of pressure and temperature must be established within the bearing package, the bearing film and where the lubricant leaves the bearing to be sure that vaporization or boiling will not occur. If the vaporization takes place within the film, cavitation may result; real cavitation, which can cause serious physical damage to the bearing materials, as the vapor bubbles are swept into areas of high pressure in the film and collapse. This has been a serious problem with mercury lubricated bearings in some of the space-power systems.

Material softening is usually not a problem with these unusual lubricants since conventional, low-melting point bearing materials are generally not employed for most of these systems. Instead, such things as Monel, stainless steel, Stellite, Alundum (al. oxide), tungsten carbide, al. bronze, etc., may be used.

Of course, vaporization can occur at low temperatures with volatile liquids such as alcohol, gasoline, or liquid refrigerants. We were running a journal bearing on Freon F-11 at room temperature conditions and fairly light loads. The Freon was being supplied to the bearing at 70°F. Vaporization occurred in the film so that the bearing was cooled. As a result the film temperature dropped to 32°F and frost accumulated on the outside of the bearing shell.

Gas, of course, does not vaporize at elevated temperatures and this is one of its advantages as used in gas-lubricated bearings. At the other end of the temperature spectrum, namely approaching the cryogenic level, gas may liquify. Thus with a change of state, either vaporization or if a gas vapor is used, condensation, there is a marked change in viscosity which means a large change in the performance characteristics of the bearing. This must be considered in design.

Temperature changes also mean variations in bearing clearances. Most bearing design calculations are based on idealized bearing clearances at room temperature. These are usually not the clearances that are found at operating remperatures. This may make a serious difference.

A very common problem is to find that the clearance has reduced as the unit reaches equilibrium temperature. If the clearance is reduced, the friction losses increase. At the same time, with a smaller clearance, the flow of lubricant is cut down, tending to raise the temperature. This in turn reduces the clearance still further, so that at times a condition of thermal instability is reached which can lead to failure.

The source of heat is in the film itself. The bearing housing has a greater capacity to get rid of heat than the journal. The amount of heat conducted through the journal is actually very small. Thus while the journal will increase in diameter as the temperature rises, and the bearing diameter will also increase, the increase in the bearing diameter will typically not be as great as that of the journal, so that there will be a loss of clearance. The prediction of just how much this will be requires a very careful thermoelastic study of the structure. However, in general, a slightly more generous value of cold clearance will usually take care of the problem.

For example, a large gas line compressor, with a shaft 5 inches in diameter, running at 5000 RPM, failed 3 sets of bearings. The failure pattern indicated loss of clearance. Design specifications, based on tolerances of manufacture for the shaft and bearings, called for a diametral clearance between 0.004 inches and 0.007 inches. Thus it was perfectly possible to start out with a cold clearance of 0.004 inches on a 5 inch diameter shaft operating at 5000 RPM. A loss of a mil or two in diametral clearance, would spell the difference between thermal stability and instability.

With a coefficient of thermal expansion of 6.5×10^{-6} per °F, a temperature difference between the bearing and shaft of 30°F would yield a loss of diametral clearance of $30 \times 6.5 \times 10^{-6} \times 5$, or about 1 mil. Thus one could be trying to operate this high speed bearing with an actual diametral clearance of 3 mils. Calculations would show that this is too small.

The recommended cold clearance was changed to 0.0007-0.010 inches and no further failures occurred. For the same hypothetical equivalent temperature differential, resulting in a loss of diametral clearance of 1 mil, the net minimum operating clearance would still be 0.006 inches or twice that achieved before.

Another similar example concerned a large centrifugal compressor pumping chlorine gas. Speed 12,000 RPM, shaft diameter 3.3 inches. Cold diametral clearance 0.004 inches. Bearing failures resulted from overheating. Cold clearances were opened up to 0.007-0.008 inches and eliminated these failures.

Another interesting example is the Haystack Antenna recently built by North American Aviation in Columbus for the Air Force in accordance with plans developed by the Lincoln Laboratory at M.I.T. Ref. (4) shows a combination thrust and journal bearing, 14 feet in diameter. Fig. 3. This is a hydrostatic, externally-pressured bearing. In order to maintain a proper balance between flow and pressure in the journal bearing pads, a radial clearance of 5 mils was desired. This imposed severe manufacturing problems, but consider the thermal condition! Suppose the temperature difference is 9°F between the inner journal and the cap. The change in diameter would be 14 X 12 X 6.5 X 10^{-6} X 9, or 10 mils. All of the clearances would be gone!

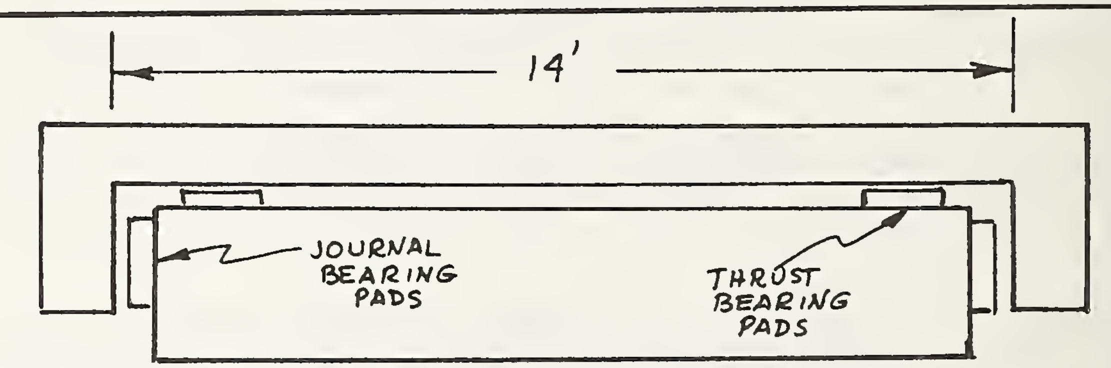

FIG. 3. SCHEMATIC CROSS SECTION OF BEARING

One suggestion was to place a ring of strip heaters around the circumference of the cap, and then if necessary, the cap could be heated to expand it and maintain the required hydrostatic bearing clearance. However, the secret of the solution to the problem was in the elastic distortion of the geometric elements. When under hydrostatic bearing pressure, the cap would expand and the inner journal would contract, thus tending to compensate in this way for loss of clearance due to thermal conditions. In a very oversimplified way, if we think of the cap as a thin ring, with a radial wall thickness of 3 inches, but with a diameter of 14 feet, and internal pressure of 1000 psi it will show an increase in diameter of about 0.150 inches. The real problem then was to manufacture the cap so that it could be assembled. After that, the hydrostatic pressure itself would maintain a clearance because of the elasticity of the structural elements themselves.

Sometimes it must be recognized that in a complex structural package subjected to various temperature gradients, the clearance of a bearing may increase for certain designs as the unit comes up to thermal equilibrium. As is well known, half-frequency whirl in journal bearings is sensitive to clearance. If the clearance increases by 10 or 20% certain bearing designs can become unstable and cause serious whirl problems. This has also been encountered in the space-power development program and has at times been a serious factor as it affects rotor dynamics and system stability. For this kind of inverse thermal problem the solution may well be to have a cold clearance which is smaller than the optimum value so as to allow for its increase under steady-state operating conditions.

It is clear then that for reliable design of fluid-film bearings, analysis of the film itself is necessary but not sufficient. In addition the contribution of the bearing structure or bearing package must be carefully evaluated. One of the most important aspects of this structural behavior may be concerned with thermal problems caused by both high temperature levels and temperature gradients. These should not be overlooked.

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NEVER OVERLOOK NOTCHES

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For several years, I have been consulting on analysis of failure of various items of hardware and machinery, mostly relating to products liability litigation. It has been my experience that at least 80% of these failures involve fatigue. I have yet to see a fatigue failure which did not start at a stress concentration of some description, i. e., a notch. In addition, I have observed some failures which did not involve fatigue but did initiate at a notch. In other words, <u>notches render hardware highly vulnerable to failure</u>.

Granted a biased viewpoint on my part, I find myself dismayed by what appears to be neglect or oversight of notches in design and/or manufacture. Certainly, knowledge of the effect of notches on stress concentration has been known for years. For example, Den Hartog (1) in discussing a stepped shaft with the larger diameter twice the smaller with a fillet radius between the two sections has said "if the fillet radius is made equal or better than 1/4 of the small shaft radius the stress-concentration factor is about 1.35 or better, which is low. Therefore more generous fillets than 1/4 are not practically justifiable. However, if the fillet corner is sharp, the stress-concentration factor becomes enormous." Elsewhere (2) he has said "stress concentration depends greatly on the local fillet radius of the reentrant corner: for a mathematically sharp corner (zero fillet radius) the stress becomes mathematically infinitely large, which in practice means very large, equal to the yield point of the material."

It so happens that I have in my personal library two textbooks (3, 4) published just prior to World War II which point out difficulties resulting from notches. One of these also refers to a paper by G. B. Jeffery in the Philosophical Transactions of the Royal Society in London, published in

1921. No doubt, a very little research effort would disclose even earlier publications. Comments on the effect of notches appear in a large number of textbooks and handbooks.

Notches come from a great variety of sources. In some cases, this appears to be a design oversight with much too small a fillet being indicated. In other cases, it appears to be a machining problem with too small a fillet resulting during fabrication. In other cases, notches result from surface defects in the material, such as inclusions in the surface or just below the surface. Machining marks, weld defects, and similar fabrication difficulties can be sources of notches and subsequent failure. Even quality control can be a source of difficulty as there are some cases on record in which the inspector's stamp effectively caused a notch which initiated failure.

It is extremely important to eliminate notches from all possible sources. This requires very careful design and manufacturing techniques with painstaking quality control. Production of finished hardware under rigid scrutiny from design to shipping will reduce (but probably never completely eliminate) the problems.

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INNOVATION OR RELIABILITY?

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Some of my friends wait until a new type of airplane has been in service for several months before they fly in it, and they do not buy a car until its engine design has been proven in service. As a rule, new designs are less reliable than old designs. Novelty and reliability compete for the available resources of time and money in engineering design. Are we achieving a balance between the two which is close to the most desirable balance? I doubt it.

Reliability means painstaking attention to detail. The pay-off is delayed; we must wait for service experience. If we succeed and all is well there is no triumph, just an absence of complaints. Innovation means creativity, the immediate thrill of having an idea and the continuing satisfaction of seeing the idea become reality. If the new idea succeeds the pay-off can be fairly quick and spectacular.

The pursuit of innovation is far more attractive than the pursuit of reliability. If we want to balance the two we should compensate the greater intrinsic rewards of innovation by greater social rewards of reliability. Is the inventor of the Rollamite worth more to us than a detail designer who produces consistently reliable designs? Do we have a surplus of good detail designers and a shortage of inventors? We act as if we did.

Universities can obtain support for innovations far more easily than support for the development of good concepts. We do have many courses in creative conceptual design but few in the design of reliable hardware. We do have courses in reliability engineering, but these are concerned with mathematical techniques for measuring reliability, or things like "The Use in Probabilistic Design of Probability Curves Generated by Maximizing the Shannon Entropy Function". They are not concerned with the achievement of reliability by careful design. Teachers are rewarded for new contributions to knowledge, not for teaching the importance of details.

Case histories of failures show that they are caused more often by overlooked details than by ignorance of advanced theories. It seems to me that in schools we should study such case histories and that in engineering practice we should give good detail designers more recognition and more pay, so that attention to detail becomes more attractive.

THE "MODEL" DESIGNER

by

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My own pet problems which show up consistently in design have to do with what I believe to be a fairly common misuse of "models". The misuse is not only by the designer himself, but also by those who transform the design criteria into the reality of the final product. These problems show up most prominently in precision equipment such as bearings, gyros, accelerometers, mirrors, etc.

A good example is the use of Hooke's Law which, in its simplest form, states that the strain is directly proportional to the stress over some stress range. As near as we can determine in the laboratory, there is <u>no</u> material which obeys Hooke's Law rigorously. This is to be expected, really, for Hooke's Law is simply a mathematical model. The usefulness of <u>any</u> mathematical model depends on how close it approximates reality. It is <u>never</u> more than a useful approximation, just as a doll is never more than an approximation of a human being. The design problem enters when needs require extreme accuracies under changing or difficult environmental conditions.

Let us suppose that a designer must make a structure that retains its symmetry under changing load conditions (a gyro is a good example). If he looks in the handbook for the elastic modulus of steel, he will find a number of about 30 million psi. In fact, the modulus of a given steel can vary 10 to 15 percent around this number, depending on how it is processed. There can be a significant difference in the modulus depending upon whether it is measured parallel to or transverse to the rolling direction of sheet. A structure made of strips cut at different orientations from a rolled sheet would, therefore, warp or tilt under stress. The modulus, obviously, is not the "constant" implied by the tables.

Let us suppose, however, that the designer succeeds in building the structure of components with adequate uniformity of modulus. It is not unreasonable that such a structure would be required to respond to imposed stresses in a manner that Hooke's Law would predict, with deviations from Hooke's Law being of the order of parts in 10^6. The design tables will often report tensile strengths, but these are useless for the purpose at hand. The tables will usually also report yield strengths, but these normally refer to the 0.2 percent offset yield. These mean 2000×10^{-6} strain at the yield stress value. The only value they have to the designer is that they indicate a stress which is much too high. On occasion, the tables will list the proportional limit or the elastic limit, but these values are extremely dependent on the sensitivity of the tools with which they are measured. Without a quantitative indicator of this sensitivity, these terms are meaningless. I believe, in fact, that the true elastic limit of all materials is precisely zero. The use of tabulated data requires both care and judgment.

Similar arguments can be made for virtually all of the "constants" with which the designer has to work. Thermal expansion coefficients, thermoelastic coefficients, electrical resistivity, density, and a host of other physical and mechanical parameters are amenable to mathematical modelling, and thus the designer can predict the effect of the environment on the response of the property of interest. But, the designer must remember that there is a real and philosophical difference between a model and reality. When you stretch such models to their limits, the agreement between the model and reality can fail completely.

While many of the mathematical models of reality are of undoubted value, some are a source of difficulty all by themselves, since they describe the wrong thing very accurately. Surface roughness measurement is a very illustrative example. This is a part of our manufacturing technology which has only just recently been attracting the attention it deserves.

Manufactured surfaces are normally described by a mathematical parameter which averages out all the hills and valleys and expresses them in a simple number. The problem is that when a surface is prepared, especially by conventional machinery, the surface layers of the materials are altered, often to a depth of many thousandths of an inch. These layers are usually highly deformed and contain severe residual stress gradients. It is readily demonstrable that these surface layers can have a more profound effect on the function of the part (in fatigue, for example) than the nature of the surface undulations. We achieve a sense of security by specifying a surface quantitatively, hoping thereby to be exerting quality controls on our product. But, in some cases, we might just as well specify that the machinist eat Wheaties for breakfast for all the good it does for the product.

The point I'm trying to make is simple. Mathematical models are just rather sophisticated "rules of thumb". They are easily (and often) stretched beyond their applicability, especially by computers (electronic and human) which do not take into account nature's delight in frustrating man's attempts to describe her simplistically.

THE MISUSE OF TENSILE STRENGTH AS A DESIGN PARAMETER

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One of my pet peeves is that engineers tend to treat tensile strength as the major design parameter of materials. So many specifications and codes base design stresses on tensile strength and yield strength. For example, most of the aircraft industry uses the limit load factor based on static loading to determine the factor of safety, which is the ratio of yield strength to load limit.

What we need to look at is, "What will the mode of failure be if this component or structure fails?" There might be more than one possible failure mode and we need to know what significant parameters to measure in order to determine the material's resistance to these particular modes of failure. The mode of failure might be low cycle fatigue, hydrogen embrittlement, stress corrosion cracking, etc. In many instances, of course, you do have to make some trade-offs between the properties of the material and various modes of failure. My point is that tensile strength is the least likely of the mechanical property parameters that will indicate resistance to failure.

Even in the case of wire, tensile strength may not be the major parameter, as evidenced by a failure some years ago of bridge wires. The tensile strength was too high which led to other problems. In general, increased tensile strength may lead to trouble because it is detrimental where the potential mode of failure might be brittle fracture, low cycle fatigue, stress corrosion cracking, etc. Realistically, a tensile stress-strain curve is no longer valid once a material has been subjected to fatigue loading. In some instances, the cyclic stress-strain curve will give more useful information than the static stress-strain curve.

Most young engineers come out of undergraduate engineering courses with an idealistic attitude toward tensile strength. They consider tensile strength to be a far more important parameter than it really is. It is unrealistic to approach design considering tensile strength and yield strength to be the only important parameters. Most standard specifications do require that, when purchasing a steel, the tensile strength, yield strength, and percentage elongation be specified. These are not really indicative of the inherent mechanical properties that are needed for service conditions. Material with a higher tensile strength may perform very poorly under certain service conditions, whereas a lower tensile strength material might perform satisfactorily. People in the aircraft industry think in terms of a strength-to-weight ratio, meaning usually good tensile strength or yield strength-to-weight ratio. This again is an inappropriate parameter.

Remember also that metal components may be subjected to a number of processing variables. They may be cold stretched, bent, stamped, machined, ground, welded, brazed, etc. On the way to becoming a finished product, they may undergo mechanical treatments, thermal treatments, and chemical treatments. We need to know the significant mechanical property(ies) in the localized zones of a component where the critical stresses occur. Most designers do not think much about the critical stress zones.

High strength weldments produce some special problems for the designer. The first thing that comes to mind is to use a high strength steel. The fatigue strength of a welded structure with a tensile strength of 130,000 psi may be no better than that of a structural grade steel with a tensile strength of 60,000 psi. Most engineers would not believe this--they have to try it and suffer a few failures before they understand what is going on. Remember that when operations such as welding are performed, the properties of the material may be vastly modified. These processes may result in residual stresses that are nearly equal to the yield strength of the material. Microvoids and other flaws are introduced into the material--the tensile strength in the region of the weld is lowered. The residual stresses and the flaws may lead to failure under cyclic loading. So there is a need to know the service environment, the range of anticipated loading, and the probable failure mode should failure occur. Forget tensile strength as a criterion for material performance.

In many instances, a component may deteriorate with time due to chemical corrosion, radiation, wear, etc. The original tensile strength cannot predict deterioration, but, for a given material, deterioration will be more rapid for higher tensile strengths than for lower tensile strengths.

DISCUSSION

A. Wolff, Massachusetts Materials Research, Inc.: Today, there is a great need for impact data, particularly on materials having transition temperatures, especially when they are being used in cryogenic applications. Published impact data are very difficult to relate to the real world. The direction of the notch, the effect of very minor residual impurities, processing, etc., make it very difficult to use this information. How can we better utilize impact data, or is there some alternative to using impact data to predict brittle transition failures?

T. J. Dolan: The fracture mechanics people have devised some rather standard tests for measuring toughness characteristics, primarily of the higher strength steels, which give us a good indication of how large a flaw can be sustained before there is danger of a sudden, running, rapid catastrophic fracture. They have also done quite a bit to understand how flaws grow from submicroscopic size either under sustained stressing or under cyclic loading. These are the things that are very valuable for the designer to know. Design must be approached assuming that all parts contain some flaws. We must know the smallest flaw that can be detected by inspection because design must take into consideration the fracture toughness characteristics of the material so that the designed part will perform satisfactorily either with no flaw growth, or with a gradual, calculable flaw growth.

A. Wolff: Then you feel that fracture toughness, even for the so-called garden variety of carbon steels, is going to have to become a part of specifications?

T. J. Dolan: It may well be so. A few years back, I examined a fracture in a gas transmission pipe. The fracture had propagated for 850 ft in what appeared to be a shear mode. It is well known that long, running cracks occur in a brittle mode. It turned out that this particular steel had very low Charpy impact values. It had a lot of manganese sulfide layers in it. These fracture characteristics could have been predicted if fracture toughness tests had been conducted before the pipeline had been installed.

R. L. Lundeen, Boeing Commercial Airplane Company: You said that you wanted to know what the smallest flaw was that could be detected. What you really want to know is what is the biggest flaw that can be overlooked.

T. J. Dolan: Definitely, right.

G. Kardos, Carleton University: The designer's principal problem is to make predictions based on the information available to him. Unfortunately, this information is often not a true reflection of the characteristics of the material.

T. J. Dolan: Data published in the producers bulletins are usually those attainable in the rolling direction. For low alloy, high strength steels you will find in the handbook tensile strengths, yield points, elongations, etc., in the direction of rolling. They do not tell you what the properties are transverse to the direction of rolling because these properties are often not as good as those attainable in the rolling direction. In the thickness direction, the properties may be terrible. That is why there are a lot of lamellar tearing failures in welds on structural components. These are things young designers are not aware of.

INSPECTION CONSIDERATION AT THE DESIGN STAGE*

John F. Erthal
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The designs of systems that require the assembly of parts and components which are expected to function for a length of time assisted by preventive maintenance and overhaul, must consider inspection and maintainability at the concept stage of the system. Provision for access to critical load carrying parts and components for inspection, primarily nondestructive inspection, must be considered at the design stage rather than an afterthought to a fixed configuration or design. This consideration demands continual dialogue and discussions among the designer, stress analyst, materials and quality assurance people. With such rapport, certain parts may even be redesigned without loss of efficiency to facilitate and accomodate nondestructive inspection. Nondestructive inspection during overhaul or periodic maintenance has been found to maximize safety and enhance reliability with the added feature of being cost effective. There is ample evidence and examples in the aircraft field to support this claim. Therefore, in the interest of reliability, safety, and reduced costs during operation and maintenance of an important system, it behooves us to take advantage of the advancing technology in the nondestructive testing and inspection discipline by consideration of its subsequent use at the earliest stage of system design.

*Letter from Mr. Erthal read by A. Wolff

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21st MEETING

Success by Design: Progress Through Failure Analysis

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