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**HIGH-CYCLE FATIGUE TESTING
OF TURBINE BLADES UNDER
MULTIAXIAL LOADING**



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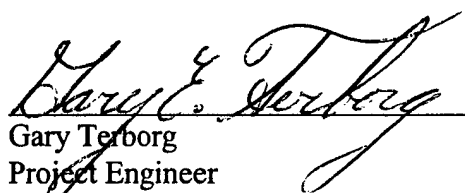
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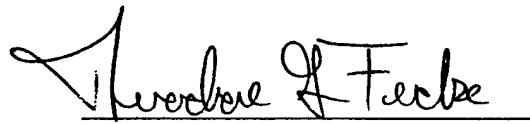
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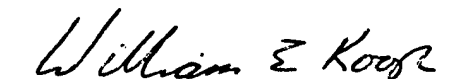
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14. ABSTRACT (Maximum 200 Words) Gas turbine engine blades are subjected to a centrifugal force in the radial direction and high frequency transverse excitation under operating conditions. A multiaxial loading machine was proposed in the present SBIR program to perform high-cycle fatigue testing on gas turbine engine blades. A GE F110 second stage fan blade was selected as the specimen for initial testing and design studies. The blade was modeled and analyzed using finite element analysis to determine the loading parameters required to simulate the operating conditions. These parameters were used as guidelines in designing the machine. The test frame, hydraulic components, control system, and data acquisition system have been integrated successfully into a fully functional multiaxial loading machine. The multiaxial loading machine has three individual hydraulic actuators. One actuator is mounted in the vertical axis and is used to simulate the radial load on the blade. The other two actuators are mounted in the horizontal axis and are used to apply a transverse bending or torsion load to the blade at high frequencies. The multiaxial high frequency testing capability of the machine was demonstrated by running tests on trial specimen and an actual GE F110 second stage fan blade.					
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1.0 INTRODUCTION

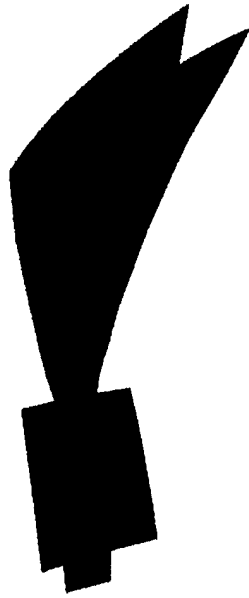
Modern gas turbine engines need to maintain a balance between high performance, affordability, and design robustness. The rotating components of the turbine engine, such as fan blades and turbine blades, are subjected to high revolutions per minute during operation. As a result, one of the most common modes of failure in engine components is fatigue. This means that to improve the robustness of components, the fatigue behavior would have to be improved.

Component fatigue behavior can be improved by improving either the material property, the component geometry, or both. Material characteristics are typically studied by testing coupons. The effect of component geometry is lost in coupon tests. Finite element analysis can be used to simulate the actual component behavior to a certain extent. The best method would be to test the actual component by subjecting it to conditions similar to the operating conditions.

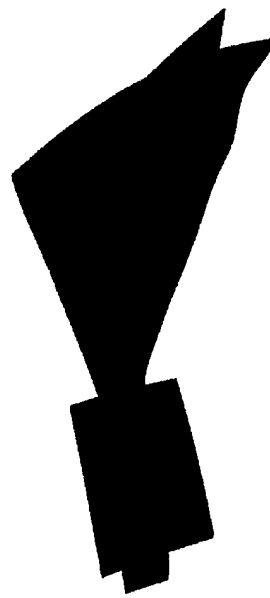
In a gas turbine engine, the fan blades and turbine blades rotate at high revolutions per minute during operation. The blades are subjected to a radial centrifugal force due to this rotation. Gas turbines have alternating stator and rotor blades. The stator blades guide the gas onto the rotor blades. As a rotor blade advances from one stator blade to the next, the gas pressure on the blade decreases and increases again. This results in the application of a cyclic load to the rotor blade. The blade also vibrates at its modal frequency under suitable conditions. These periodic loads result in a vibratory loading on the blade.

This vibratory loading causes high-cycle fatigue failure in engine blades. To study and improve the fatigue life of gas turbine engine blades, a new test methodology has been proposed. According to this method, the blade will be loaded multiaxially during testing, to simulate the actual operating conditions. A prototype biaxial testing machine was proposed and demonstrated in Phase I of this program. The biaxial machine had two hydraulic actuators perpendicular to each other. This enabled the machine to apply both a radial load to simulate the centrifugal force and a cyclic transverse load to simulate the vibratory loading. The single transverse actuator facilitated the application of bending loads.

Vibration of turbine blades include both bending and torsion modes as shown in Figure 1. The biaxial machine could not be used to apply torsion loads to the blade, as that required two actuators in the transverse direction. A multiaxial testing machine was proposed, designed and demonstrated in Phase II. The multiaxial machine had three hydraulic actuators, a large actuator along the vertical axis, and two small actuators along the horizontal axis. The large actuator was used to apply the radial centrifugal force. The two small horizontal actuators were used to apply the vibratory loading. The two small side actuators can be offset independent of each other. This enabled the machine to apply not only bending loads but also torque to the blade. The application of bending and torsion loads to the blade is explained in detail in Section 3.1.



Bending (First mode)



Torsion (Third mode)

Figure 1. Bending and torsion of gas turbine fan blade

2.0 PROTOTYPE TEST SYSTEM WITH BIAXIAL LOADING CAPABILITY

2.1 Design of a Biaxial Loading Test Frame

In Phase I, an existing four-post test frame was modified to test a gas turbine engine fan blade under biaxial loading. A four-post die set was designed and fabricated. It was placed between the top and bottom platforms, and was mounted on the load train. A vertical plate was mounted on the side of the die set, on which a second hydraulic actuator was mounted. Figure 2 shows the schematic of the biaxial loading machine. Figure 3 shows a photograph of the completed biaxial loading machine. The vertical actuator was used to apply the steady radial load to simulate the actual centrifugal load on the blade. The horizontal actuator was used to apply the fatigue loading to simulate the actual vibratory load. Both actuators were controlled by a Intelaken DDC4000 controller. The load and time data were collected in a computerized data acquisition system. Labtech Notebook was the data acquisition and analysis software used.

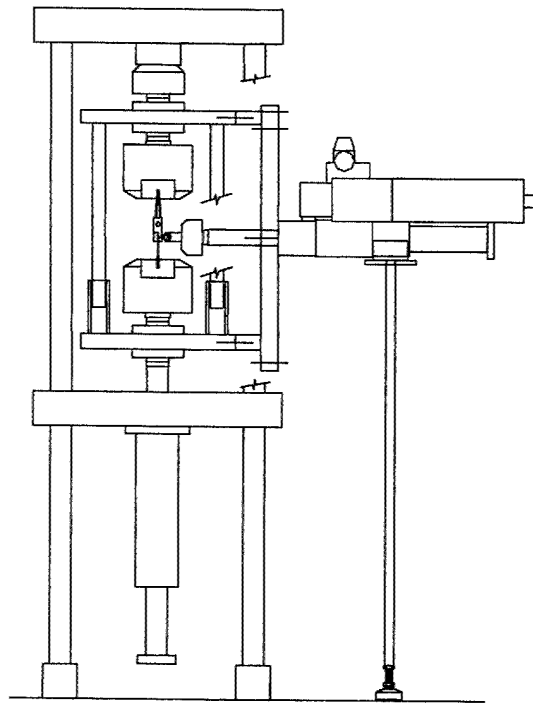


Figure 2. Schematic of biaxial loading machine



Figure 3. Photograph of biaxial loading machine

A gripping system was designed and manufactured to apply the biaxial loading to the blade. A schematic of the gripping system is shown in Figure 4. Transverse fatigue loading was applied to the blade through a rod-end bearing connection on the grip surface. The rod-end bearing was connected to a clevis, which was in turn connected to the piston of the horizontal actuator. The rod-end bearing allows transverse fatigue loading to be applied to the test specimen and at the same time allows small rotation around the gripping. This extra degree of freedom will allow the test specimen to bend under transverse loading.

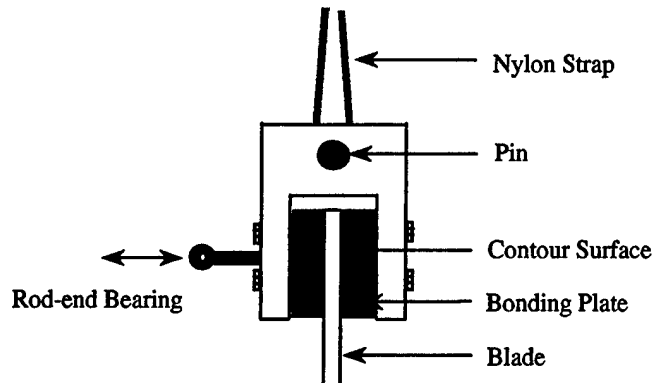


Figure 4. Blade gripping system for biaxial machine

A nylon strap was used to connect the fixture to the hydraulic grips. The nylon strap served as the flexible connection that prevented interference between the axial and lateral actuators. The gripping system had two bonding plates whose internal contours conformed to the surface of the blade. A photograph of the complete setup is shown in Figure 5.

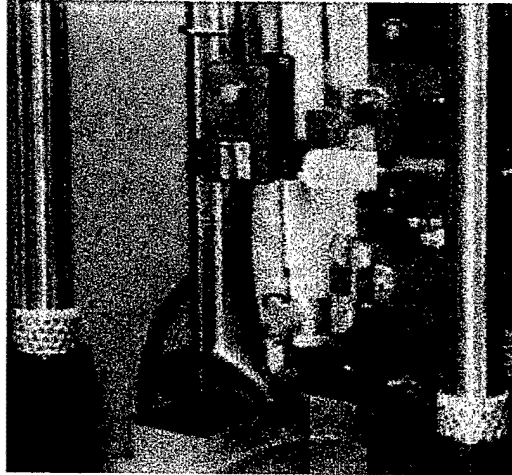


Figure 5. Photograph of biaxial loading of GE F110 fan blade

A broach block was used to hold the turbine blade specimen in place. The broach block resembles a part of the turbine rotor disk. The dovetail of the blade slides into the block, and the blade was tightened against the block by two screws from underneath. The broach block was connected to the vertical actuator directly.

2.2 Biaxial Test of a GE Engine Fan Blade

Trial specimens made of Ti-6-4 were tested initially on the biaxial loading machine. The trial specimen was tapered and gripped over the taper. The specimen and the grip were bonded together using an adhesive (JB Weld). The load versus displacement curve for adhesive bonding without bolted pressure is shown in Figure 6. For a 1 square inch bonding area, the tensile load reached about 1800 lb before the grip detached from the specimen.

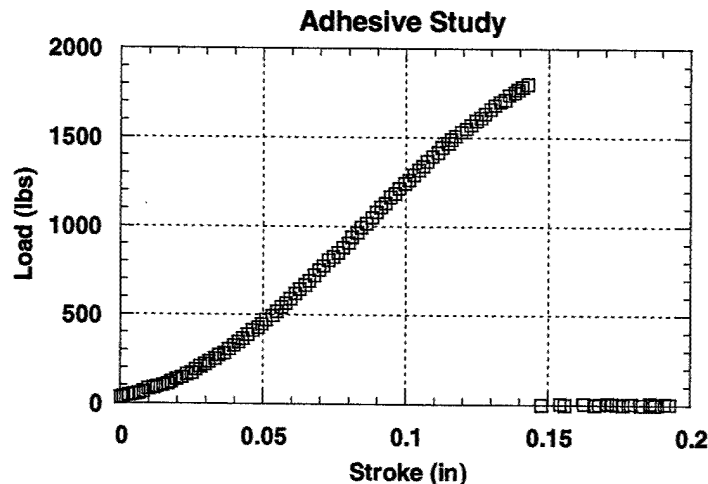


Figure 6. Tensile test of trial specimen using adhesive bonding without bolted pressure

One more static test on trial taper specimen was performed with bolted pressure on the specimen. The assembly carried more than 6500-lb tensile load without failing. The nylon strap was also subjected to a tensile test, to determine its tensile strength. Figure 7 shows that the strap can carry more than 3500-lb static load without failing.

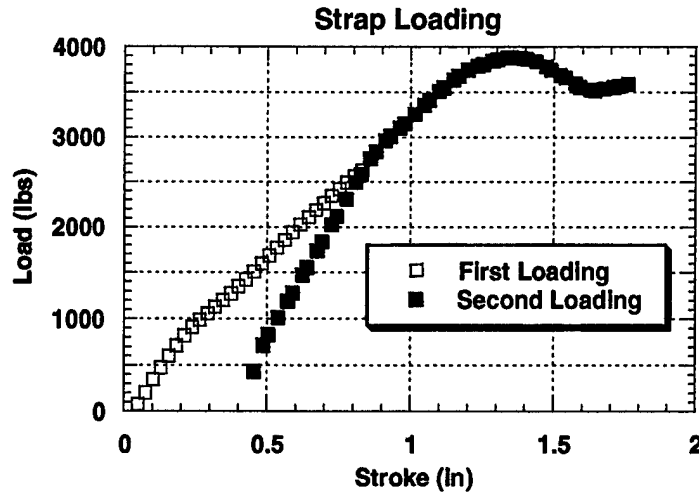


Figure 7. Tensile test of strap

Tensile-tensile fatigue test was performed on taper trial specimen with both adhesive bonding and bolted pressure. A typical load versus displacement curve is shown in Figure 8. The fatigue test, at a frequency of 2 Hz, was stopped after more than 2,000,000 cycles. Both the bonding area and the nylon strap were in very good condition without any visual failure. The trial specimen assembly was then used to test the biaxial loading capability of the test frame. Both actuators worked well with the controller and the data acquisition system.

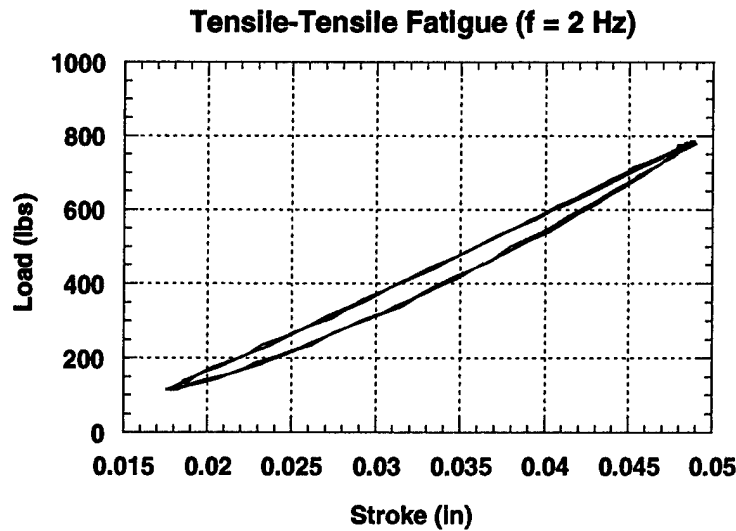


Figure 8. Fatigue test of trial specimen

Biaxial tests were performed on a GE F110 second stage fan blade. Grip plates were bonded to the blade using the tested adhesive. Six (0,45,90) rosette strain gages were placed on the blade. The strain gage readings were recorded using the data acquisition system. Various combinations of radial and transverse fatigue loads were applied to the blade through the gripping system. The finite element analysis results provided a guideline for the range of loads. Radial loads in the range of 1000 lb to 2000 lb and transverse loads in the range of 400 lb to 800 lb were applied. Figure 9 shows the fatigue load progression during a typical biaxial test performed on the blade.

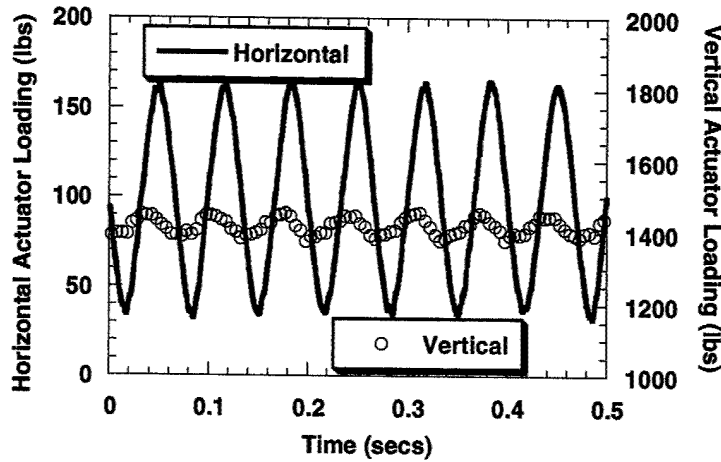


Figure 9. Biaxial fatigue loading of blade

The purpose of this preliminary testing was to examine the strain distribution in the fan blade specimen. Therefore, only a low frequency was used to apply the transverse fatigue loads. The transverse fatigue loads were applied at frequencies of 0.1 and 0.25 Hz. A radial load of 1200 lb and transverse loads of 400, 500, 600, and 800 lb were used. The load profiles for the tests done at 0.25 Hz are shown in Figures 10 through 13.

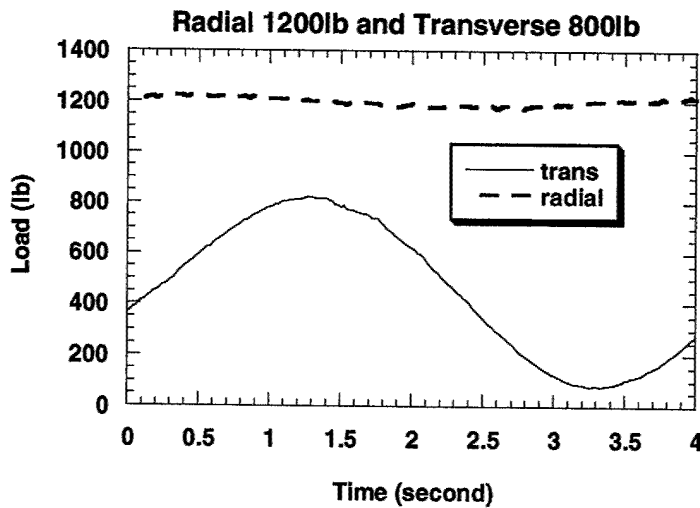


Figure 10. Biaxial loading of blade radial 1200 lb and transverse 800 lb

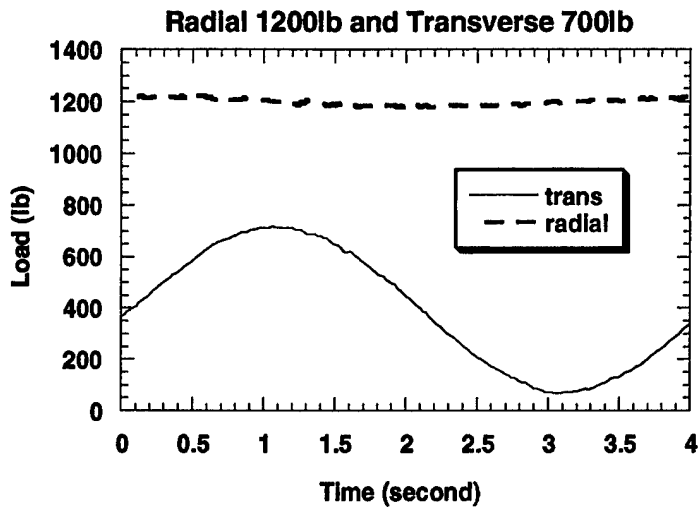


Figure 11. Biaxial loading of blade radial 1200 lb and transverse 700 lb

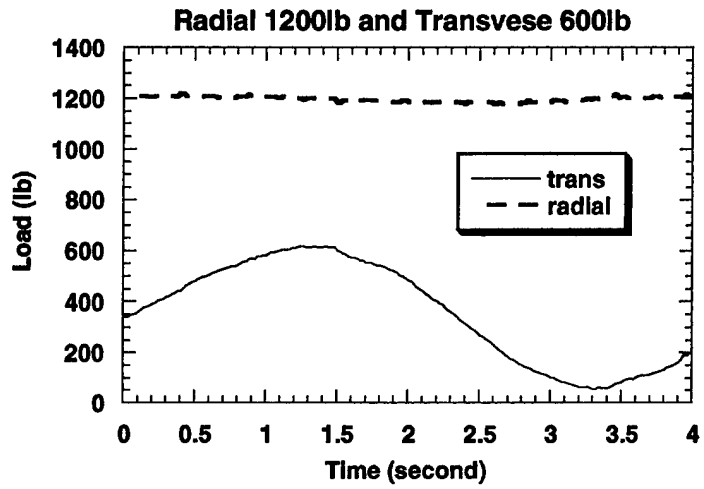


Figure 12. Biaxial loading of blade radial 1200 lb and transverse 600 lb

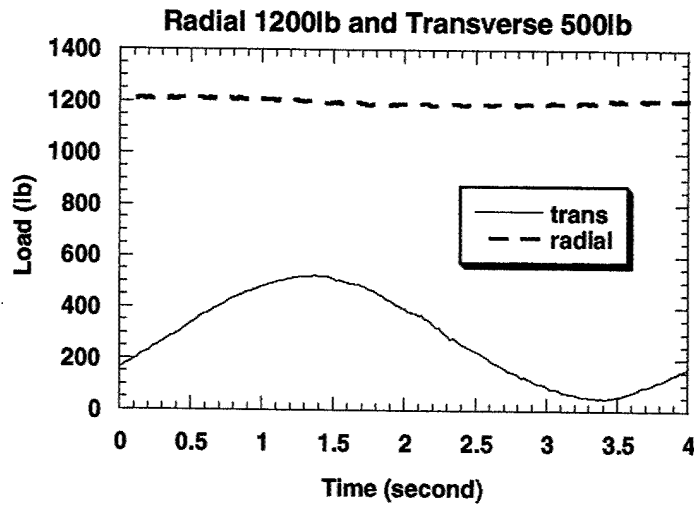


Figure 13. Biaxial loading of blade radial 1200 lb and transverse 500 lb

The radial stresses obtained during the experiments performed at 0.1 Hz are presented in Figure 14, along with the radial stresses predicted by finite element analysis. The experimental values of the radial stresses were obtained from the strain gage closest to the leading edge and the root of the blade. The correlation between the experimental data and the analysis data indicates that biaxial loading can be effectively used to simulate the stress distribution in the blade under operating conditions.

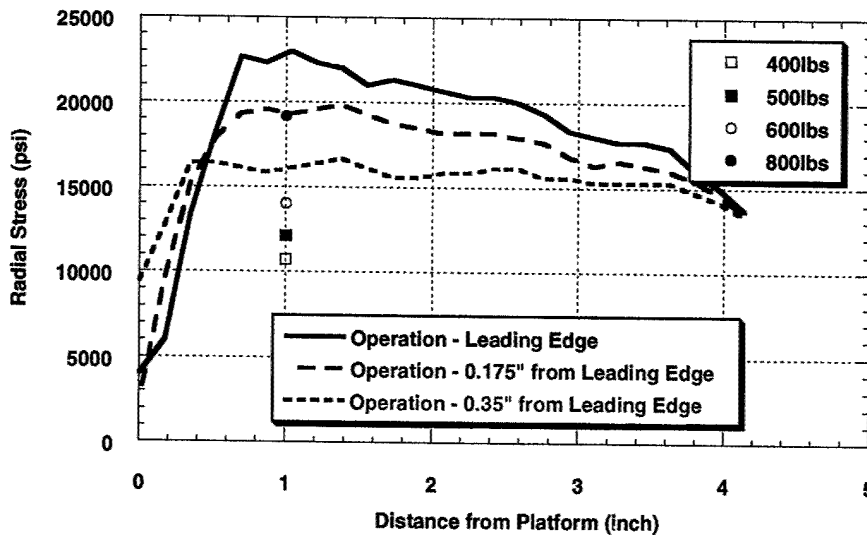


Figure 14. Comparison of experimental data with the operating conditions predicted by finite element analysis

3.0 DESIGN OF THE MULTIAXIAL LOADING TEST SYSTEM

The multiaxial loading test system includes the following subsystems:

- (1) Main test frame, including an enclosed chamber with four side walls;
- (2) Main test frame support, including two I-beam pedestals reinforced by gussets;
- (3) Auxiliary test frame support, including a four-post die set and other mounting adaptors;
- (4) Servohydraulic components, including hydraulic service manifold, two high-frequency actuators (1.1 kip, 400 Hz), and one main actuator (11 kip); and
- (5) Multiaxial digital control and data acquisition system, including the control console and a computer workstation.

The design features of the above subsystems will be discussed in paragraphs 3.1 and 3.2.

3.1 Concept of Bending and Torsion Loading

Since vibration of turbine engine blades under service conditions includes bending and torsion modes, two actuators have been designed into the test system to simulate the bending and torsional vibration loading. A schematic of the top view of the transverse actuators in relation to the blade is shown in Figure 15.

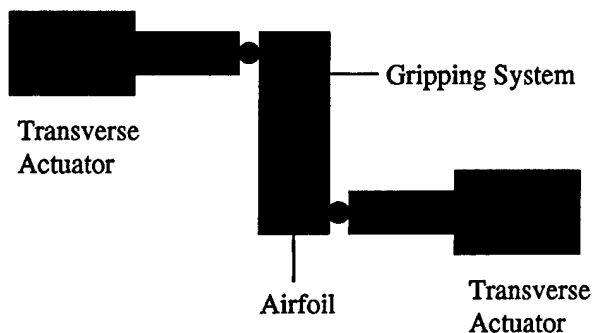


Figure 15. Transverse loading in the multiaxial machine

The blade is subjected to torsion loading when the transverse actuators move in-phase, that is when both actuators move either in or out at the same time. Figure 16 shows a schematic of the torsion loading. The blade is subjected to bending loading when the actuators move out-of-phase, that is one actuator moves in when the other moves out or vice versa. Figure 17 shows a schematic of the bending loading.

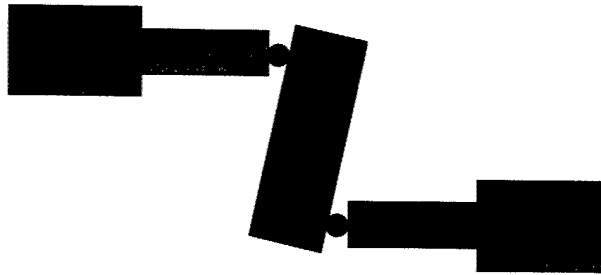


Figure 16. Torsion loading

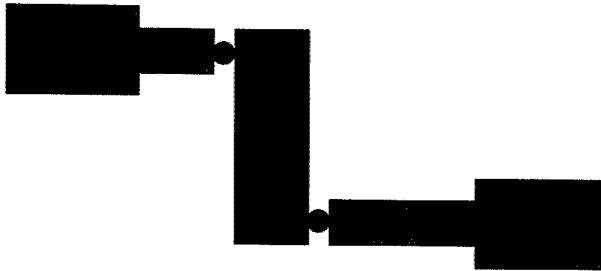


Figure 17. Bending loading

3.2 Design of the Gripping System

A new gripping system was designed and manufactured to apply the multiaxial loading to the blade. Figure 18 shows the three-dimensional model of the gripping and loading system developed during the design stage of Phase II. The broach block used in Phase I was used after some modifications. The dovetail of the blade slides into the dovetail of the broach block, and the blade is tightened against the block by two screws from underneath. To access these two screws, the broach block has to be removed from the machine every time the blade has to be removed. In Phase I, the broach block was connected to the moving die plate by a threaded rod. Since the position moved every time the broach block was removed from the machine, the broach block had to be set up again each time

An adapter was added to the base of the broach block in Phase II. The adapter was connected to the stationary die plate by a threaded rod. The broach block was located on the adapter plate by two dowel pins, and it was attached to the adapter plate by four bolts. This facilitated the easy assembly and removal of the broach block and the blade for each test. The dowel pins ensured that the broach block is returned to the same position as before it was removed from the machine. This eliminates the need for setting up the broach block between tests on same or similar blades.

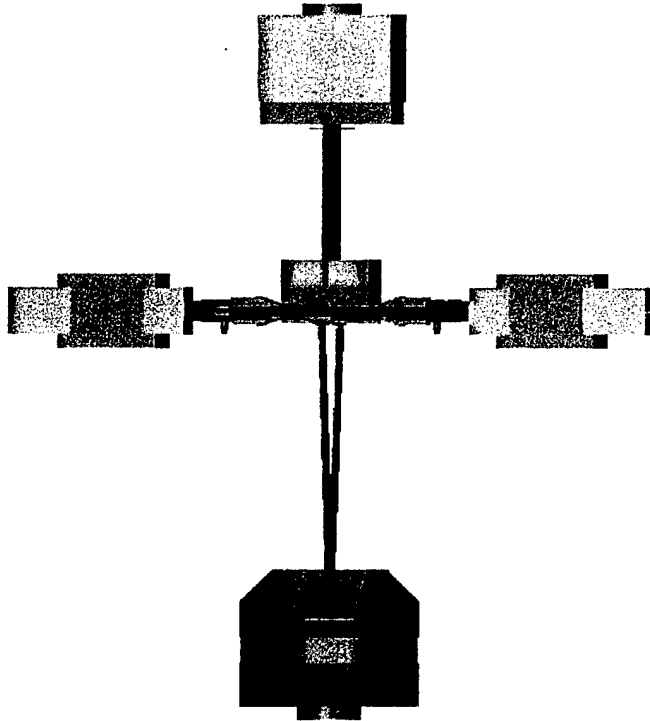


Figure 18. Three-dimensional model of multi-axial loading mechanism

The strap is gripped by a hydraulic wedge grip. In Phase I, the hydraulic grip was mounted on the top (fixed) die plate, while the broach block and specimen were mounted on the bottom (movable) die plate attached to the vertical actuator. As a result, the position of the specimen relative to the transverse actuator changed whenever the vertical actuator moved and the hydraulic power was switched off. In the new multi-axial test frame, the hydraulic grip is mounted on the bottom (movable) die plate attached to the vertical actuator. The broach block and the specimen are mounted on the top (fixed) die plate. This ensures that the relative distance between the specimen and the transverse actuators remains undisturbed by the movements of the vertical actuator. This also enables quick change over from one test to another on same or similar blades, by eliminating changes in the critical locating dimensions.

A load cell of 2000-lb capacity supplied by Sensotec is mounted on the piston rod of each transverse actuator using an adapter. A clevis is attached to the load cell. The clevis is connected to the grip pin using two rod-end bearings. The grip pin extends from one end of the grip to the other end. When the piston rod of a lateral actuator moves forward, it pushes one end of the grip pin forward via the clevis and the rod-end bearings. This results in the application of a force to the specimen in the direction of the actuator movement. The rod-end bearings and the clevis joints allow movement of the grip in the vertical direction to a certain extent. A three-dimensional model of the transverse loading mechanism is shown in Figure 19. A photograph of the GE F110 gas turbine engine fan blade subjected to multi-axial loading using the gripping system is shown in Figure 20.

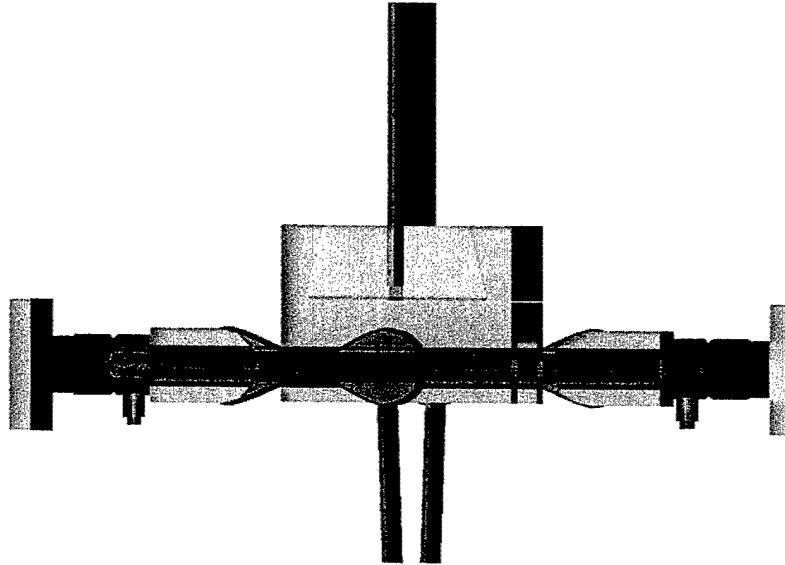


Figure 19. Transverse loading of blade

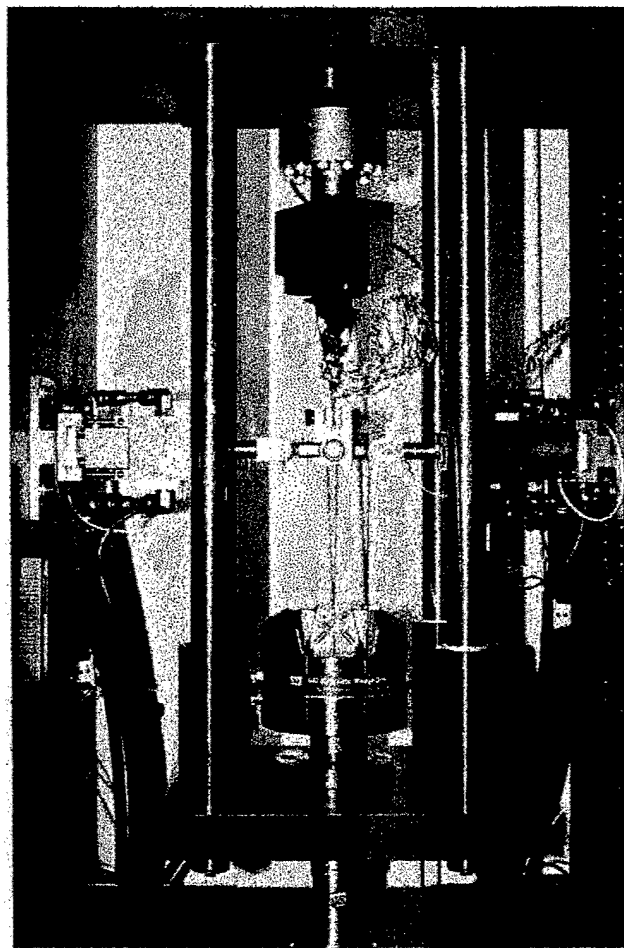


Figure 20. Photograph of multi-axial loading setup

3.3 Hydraulic Components and Control System

The hydraulic control system and the actuators were procured from Instron Schenk Testing Systems. A Labtronic 8800 multi-axis digital control console, manufactured by Instron, is used to control all the three actuators. It monitors the load and displacement conditions while performing high-speed data acquisition. Labtronic 8800 has multistation capability. The Labtronic 8800 control console is connected to a personal computer running Windows NT. A software called RS Console is used for interacting with the Labtronic 8800 control console. RS Console can be used for setup, waveform generation, and setting of limits. It uses a wizard to provide simple easy-to-use instructions for complex operations. The software also includes multiple, live displays for digital readout of data. RS Console has function generators that can be interlocked with phase control. This enables the maintenance of phase relations between the actuators during cyclic loading.

The two lateral actuators are of type PLF7D supplied by Instron. Each actuator has a capacity of 1100 lbs (5.2 kN) and 20-mm stroke. These actuators can apply high-frequency loading up to 400 Hz. Standard fatigue rated actuators using high-pressure rod seals experienced a banding problem resulting in early failure. When the dynamic stroke is too small to carry fresh oil under the seals, the oil film breaks down and results in damage to the rod. The PLF7D servo hydraulic actuators are designed, built, and optimized for high-frequency operation. Hydrostatic bearings and laminar high-pressure seals allow sustained high-frequency, short-stroke operation. The actuator does not have any elastomeric seals in contact with the piston rod during operation. The single rod seal that prevents external leakage when turned off is retracted from the rod when operating. A suction pump is used to scavenge the leakage oil during operation.

The vertical actuator is a labyrinth bearing pedestal base actuator supplied by Instron. The actuator has a dynamic force rating of +/- 11,000 lbs (50kN) and +/- 50-mm stroke. The rod diameter is 63.5 mm and the actuator stall force is 63 kN. The load in the vertical direction is measured by a dynamic load cell of 11,000 lb (50 kN) capacity.

All three actuators are connected to servovalves. The servovalves are connected to a hydraulic service manifold (HSM). The hydraulic service manifold is in turn connected to the hydraulic power supply. Accumulators are provided in the hydraulic service manifold to enable high-frequency operation of the actuators.

3.4 Design of the Multiaxial Test Frame

The major structural components of the multiaxial test frame are a main chamber with four side walls, two I-beam pedestals, and a four-post die set. Figure 21 shows a three-dimensional model of the multiaxial test frame.

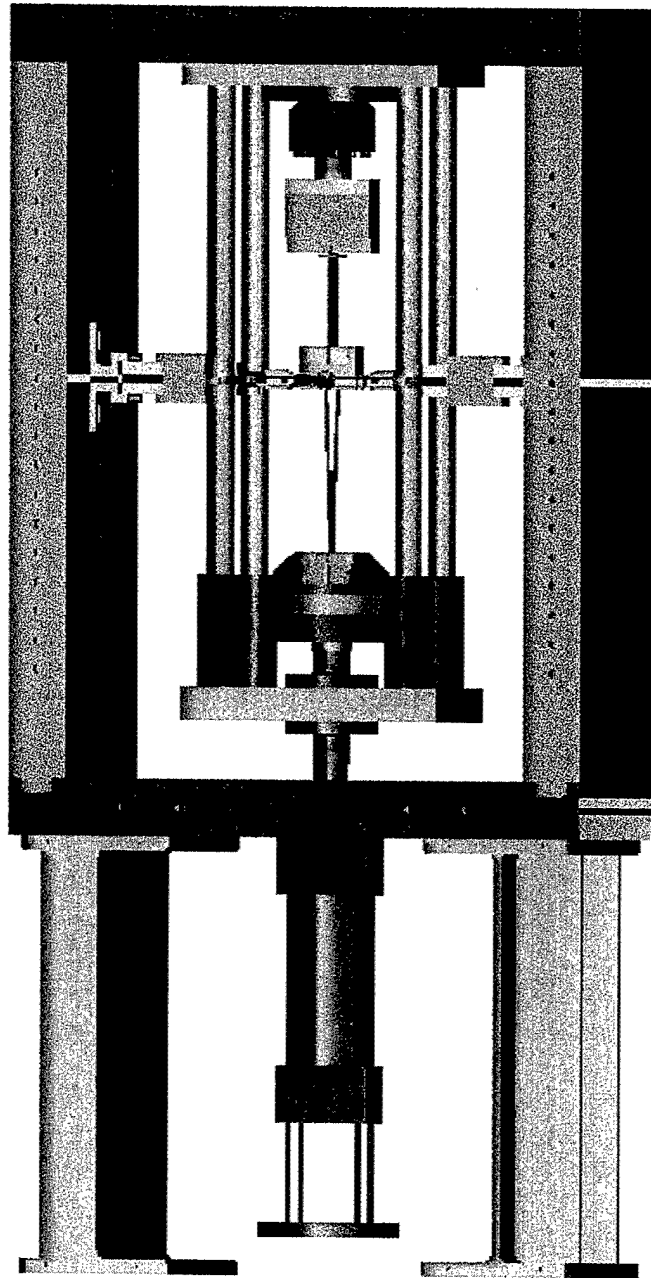


Figure 21. Multi-axial test frame

A photograph of the complete multiaxial test frame is shown in Figure 22. The test frame and the hydraulic service manifold are located inside an enclosed test chamber as shown in Figure 23. The control system and the personal computer that acts as the interface with the user are located outside the chamber. This isolation of the test frame helps to reduce the noise pollution of the surroundings. The I-beam pedestals of the machine are bolted to a metallic test bed using T-bolts. This prevents the machine from moving due to vibrations during high-frequency tests. The I-beams are separated from the test bed by neoprene pads. The neoprene pads reduce the transfer of vibrations from the test frame to the test bed.



Figure 22. Photograph of multiaxial machine



Figure 23. Multiaxial machine and operator

The side frame walls of the test frame have three rows of threaded holes. The actuator fixtures are bolted to the side frame walls using these holes. This is shown in Figure 24. The rows of holes allow the actuator fixtures to be moved and positioned both in the vertical and horizontal directions. This enables the positioning of the lateral actuators at different points with respect to the specimen. The base of the lateral actuator slides inside a slot in the actuator fixture and the actuator is bolted to the fixture. This allows the actuator to be moved along the slot to a certain extent for quick adjustments. The actuator fixture has slots that are used to bolt it to the side frame wall. These slots allow the fixture to be moved in the vertical direction to a certain extent for quick adjustments.

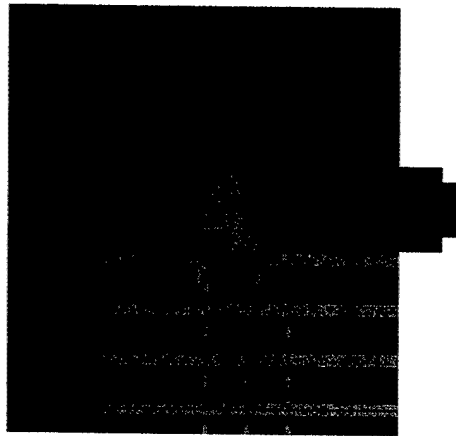


Figure 24. Positioning of the transverse actuator

The main chamber of the multiaxial test frame consists of four frame walls that are 4 inches thick. The construction of the main chamber is shown in Figure 25. The top and the bottom frame walls have machined grooves. The side frame walls have matching machined projections that rest in these grooves. This reduces the movement of the side frame walls due to the forces from the transverse actuators. This also reduces the transfer of the transverse forces to the bolts holding the walls together. A row of threaded holes is provided on the front and back sides of the frame walls to facilitate easy attachment of ancillary equipment such as measuring devices and cameras.

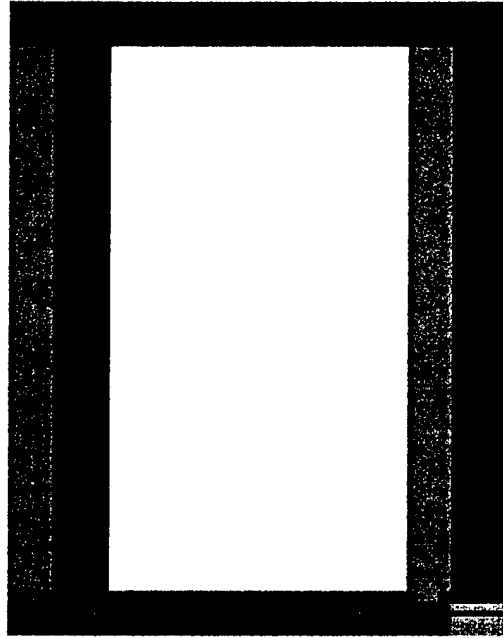


Figure 25. Main chamber

A four-post die set is used in the vertical load train. The four-post die set is used to minimize the effects of the lateral loading on the piston rod of the vertical actuator. The top plate of the die set is bolted to the top frame wall. The load cell that measures the load in the vertical axis is bolted to the top plate. The broach block is attached to the broach block adapter plate, which in turn is attached to the load cell. The hydraulic grip used to grip the strap is mounted to the bottom die plate using a threaded rod. The bottom die plate is attached to the piston rod of the vertical actuator. When the piston rod of the vertical actuator is moved downwards, the strap is pulled resulting in a vertical load on the specimen. The four-post die set is shown in Figure 26.

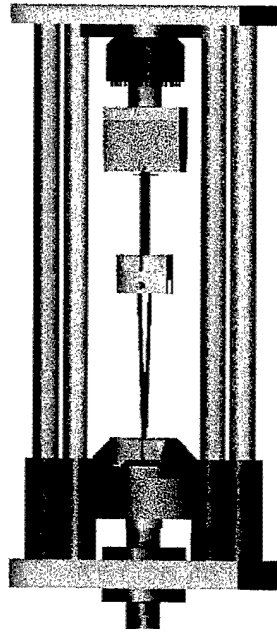


Figure 26. Four-post die set

The actual dimensions of the manufactured parts and the materials used to manufacture them are shown in the detailed drawings attached as Appendix A.

4.0 FINITE ELEMENT ANALYSIS OF THE TEST FRAME

The final design of the machine frame was derived after several design iterations. Finite element analysis was used extensively to validate each major design decision. The different finite element analysis used during the design stage and their results are explained in sections 4.1, 4.2 and 4.3.

4.1 Modal Analysis

As the machine is expected to operate at high frequencies, modal analysis was performed on the machine frame and the pedestal to estimate their natural frequencies. The analysis was performed on the frame, and the frame and pedestal combined. The properties of generic steel were used for the analysis. The material properties used were:

Young's modulus: 30×10^6 psi
Poisson's ratio: 0.29
Density: 0.7317×10^{-3} lbf s²/in⁴

Different types of pedestals were studied during the design phase. The different configurations analyzed are shown in Figures 27 and 28. The results for three configurations, namely, frame without pedestal (Configuration 1), frame with I-beam pedestal (Configuration 2), and frame with I-beam pedestal stiffened by gussets (Configuration 3) are listed in Table 1. These three configurations were selected for further consideration due to ease of manufacture. The results for the remaining configurations are listed in Table 2.

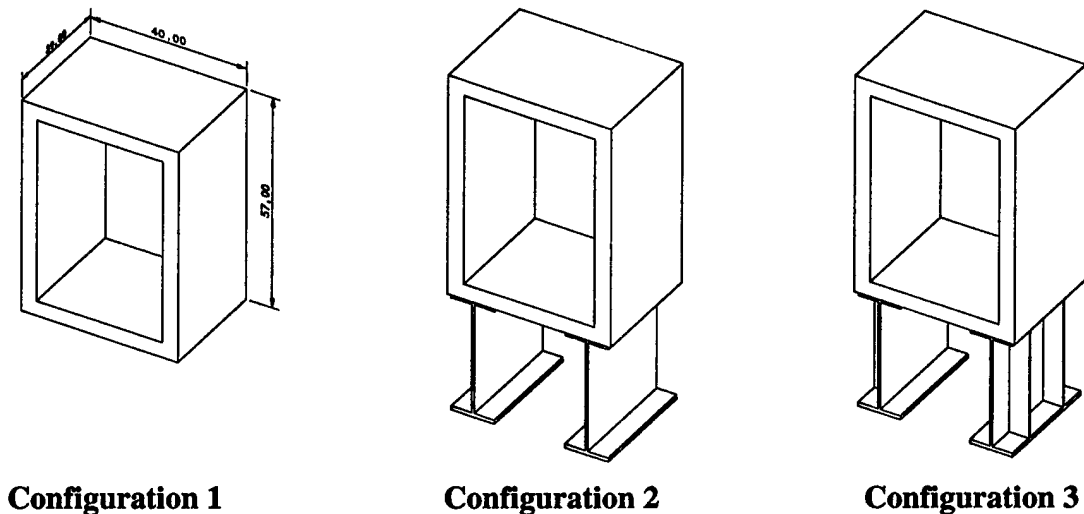
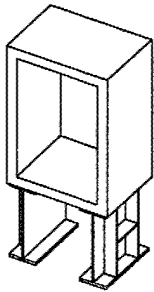


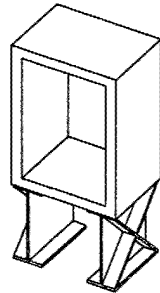
Figure 27. Configurations (1-3) subjected to modal analysis

Table 1. Natural frequencies of frame, pedestal combinations 1-3

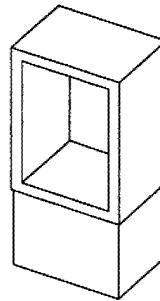
Mode	Natural Frequency (Hz)		
	Only Frame: Config. 1	Frame + Pedestal	
		No Gusset : Config. 2	Two Gussets : Config. 3
1	51	5	26
2	182	37	40
3	213	58	60
4	219	61	85
5	304	156	171
6	397	165	227
7	432	166	251
8	438	196	263
9	618	196	264
10	722	224	315



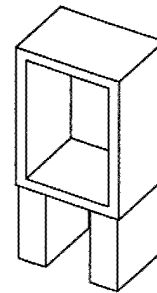
Configuration 4



Configuration 5



Configuration 6



Configuration 7

Figure 28. Configurations (4-7) subjected to modal analysis

Table 2. Natural frequencies of frame, pedestal combinations 4 - 7

	Natural Frequency (Hz)			
	Configuration 4	Configuration 5	Configuration 6	Configuration 7
Mode 1	27	23	42	45
Mode 2	41	40	60	116
Mode 3	60	67	112	149
Mode 4	86	76	149	212
Mode 5	171	166	153	215
Mode 6	229	249	183	371
Mode 7	253	255	194	388
Mode 8	264	266	196	420
Mode 9	266	292	207	428
Mode 10	324	322	242	469

4.2 Normal Mode Dynamic Analysis

Normal mode dynamic analysis was performed on the main chamber of the machine frame (four frame walls). The mode 1 natural frequency of 51 Hz coincides with the lateral loading axis. Lateral loading at this frequency may cause resonance. The results of the normal mode dynamic analysis for different assumptions of damping are listed below:

Load: 1000 lb on each lateral actuator at 51 Hz

Maximum displacement for different levels of assumed damping:

No damping: 3.5 inches

0.001 (Steel): 0.8 inch

0.03 (Typical): 0.04 inch

0.05 (Bolted joints): 0.02 inch

The frequency versus displacement graph for the chamber assuming no damping at all is shown in Figure 29. The frequency versus displacement graph for the chamber assuming the damping of solid steel is shown in Figure 30. Figures 31 and 32 show the frequency versus displacement graphs for typical damping usually assumed for a solid and for bolted joints. It is seen that the maximum displacement for bolted joints is the least in the above list. As bolted joints assemble the frame, the response is expected to be closer to the lower end of the above values. More damping is expected with the addition of the die set and internal components to the frame.

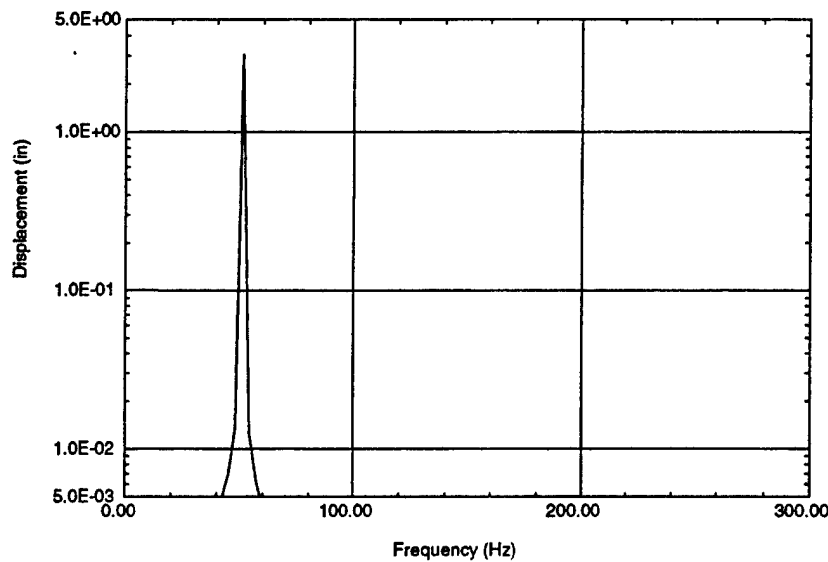


Figure 29. Frequency response function (no damping)

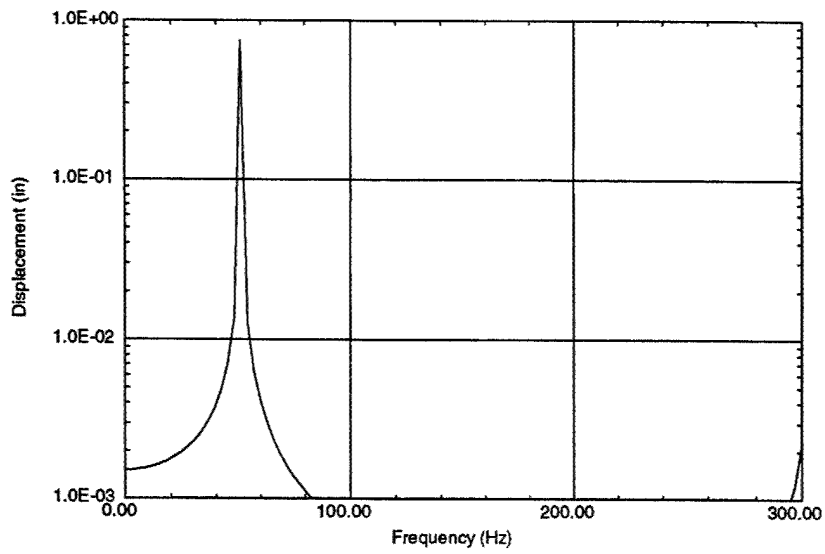


Figure 30. Frequency response function (0.001 damping factor)

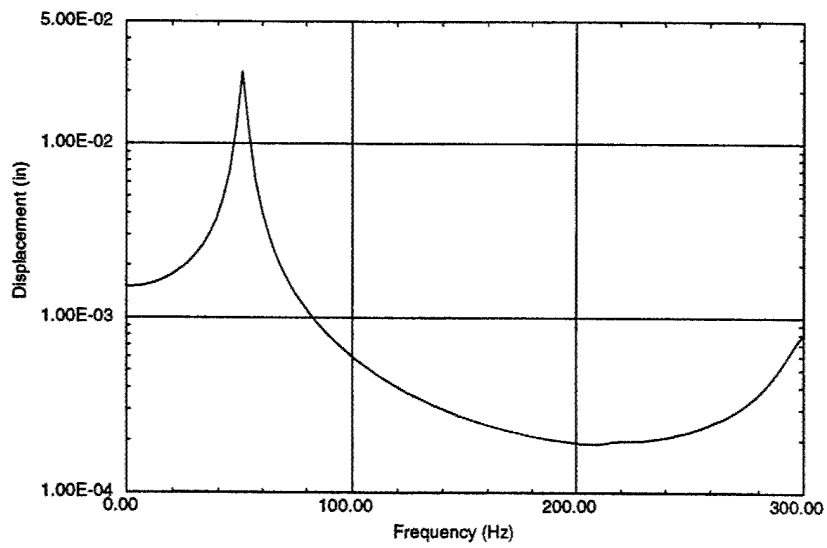


Figure 31. Frequency response function (0.03 damping factor)

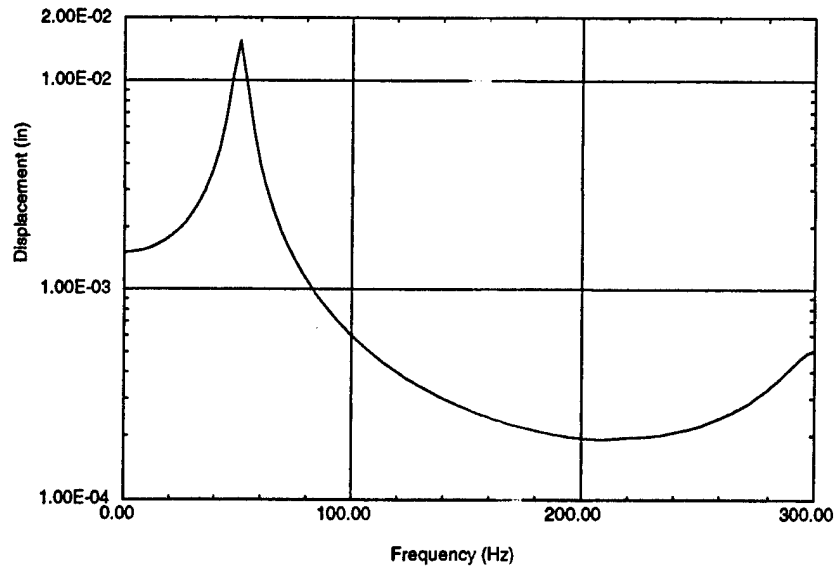


Figure 32. Frequency response function (0.05 damping factor)

4.3 Buckling Analysis

The I-beam pedestal was subjected to buckling analysis to evaluate its susceptibility to buckling. The machine frame with all attachments is expected to weigh about 7,000 lb. The expected load on each of the two beams is 3,500 lb. The buckling loads for the I-beam pedestal without stiffening, stiffened by one gusset, and stiffened by two gussets are listed in Table 3. It is seen that the expected load is significantly lower than the minimum buckling load for the following cases. Stiffening of the pedestal seems to be redundant from a buckling point of view, but it serves the purpose of increasing the natural frequencies of the structure. As a result, the I-beams were designed to have two gussets each.

Table 3. Buckling loads of the I-beam pedestal

Mode	Buckling Load (lb)		
	No Stiffening	One Gusset	Two Gussets
1	36,000	464,000	2,711,000
2	33,000	889,000	2,891,000
3	40,000	1,065,000	2,997,000

5.0 FINITE ELEMENT MODELING OF A GE ENGINE FAN BLADE UNDER MULTIAXIAL LOADING

5.1 Modal Analysis

Finite element analysis of a GE F110 turbofan second stage fan blade was performed using I-DEAS Master Series software. The blade was analyzed initially using a finite element mesh made of 387 parabolic brick elements. This model will be referred as Model-1 in the following discussion. Toward the end of the project another model using finer elements was developed to ratify the results obtained using Model-1. This model consisted of 12,520 parabolic brick elements. This model will be referred as Model-2 in the following discussion. The two models are shown in Figures 33 and 34. Apart from the size of the elements, the main differences are (i) the blade is modeled using a single layer of elements in Model-1, while Model-2 uses two layers, and (ii) Model-1 preserves the sharp sides of the blades, while the sharp edges are flattened in Model-2 to facilitate meshing. Both models were subjected to modal analysis. The properties of Ti-6-4 were used to model the blade. The material properties used were:

Young's modulus: 16×10^6 psi

Poisson's ratio: 0.33

Density: 0.4211×10^{-3} lbf s²/in⁴

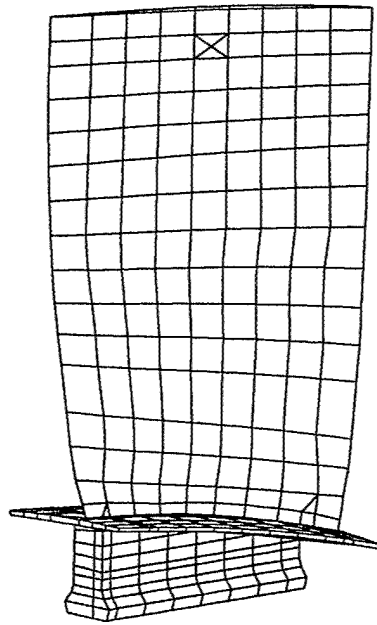


Figure 33. Model-1

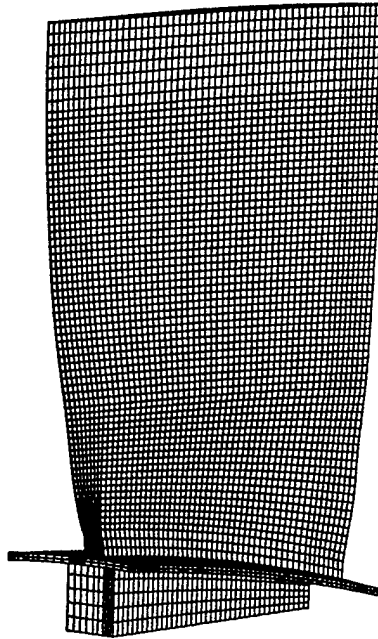


Figure 34. Model-2

The results of the modal analysis are shown in Table 4. It is seen that the modal frequencies obtained correlate well with the experimental bench test data (data provided by GE) for both models. This validates the accuracy of the two finite element models used in the analysis.

Table 4. Comparison of bench test and the predicted modal frequency

Mode	Bench Test Frequency (Hz)	FE Model Frequencies (Hz)	
		Model-1	Model-2
1	239	236	236
2	648	648	649
3	1212	1168	1193
4	1458	1504	1502
5	1930	1994	1937
6	2317	2306	2266
7	2407	2460	2397

5.2 Modeling of Multiaxial Loading on the Fan Blade

Finite element analysis was performed on a GE F110 turbofan second stage fan blade with different configurations of multiaxial loading. Most of the analysis was performed using the Model-1 finite element mesh. Some have been performed using both Model-1 and Model-2 finite element meshes. The objective of this analysis was to match the stress pattern experienced by the blade during operation, by varying the positions and magnitudes of the individual axial loads. During operation, the blade experiences a steady stress state due to the gas pressure and centrifugal force, and a varying stress state due to vibration.

The stress values during operation were predicted using finite element analysis by GE, and are shown in Figures 35 and 36. The pressure side refers to the front surface of the blade, while the suction side refers to the back surface of the blade. Figure 35 shows the steady state radial stress on the pressure side at 8400 rpm. Figure 36 shows the radial stress due to Mode-1 vibrations on the pressure side.

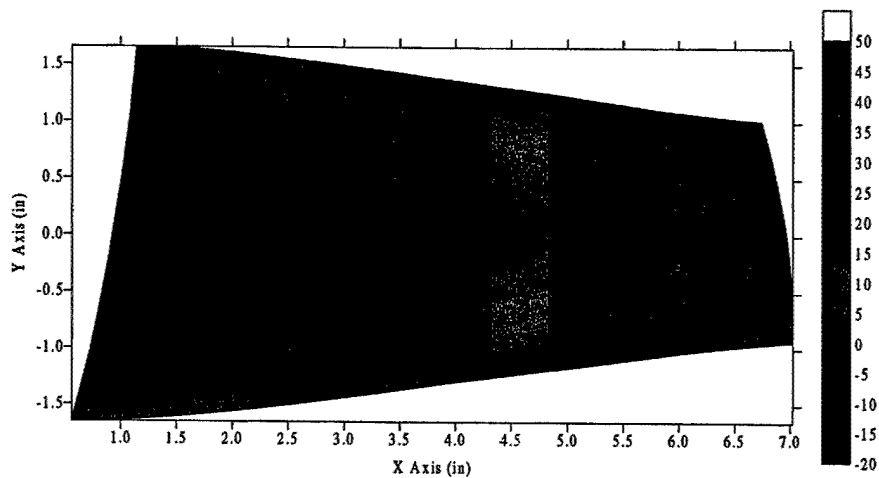


Figure 35. Steady State Radial Stress on the Pressure Side
Contours show radial stress in Ksi

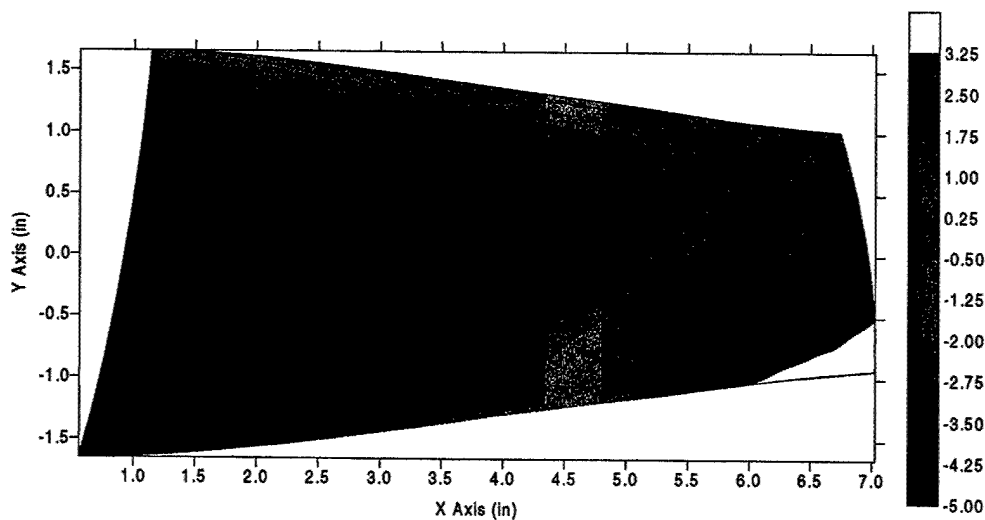


Figure 36. Radial Stress on the Pressure Side due to Mode-1 Vibrations
Contours show radial stress in Ksi

Figures 37 and 38 show the radial stress when both the steady state and Mode 1 vibratory stresses are superimposed. Corresponding radial stress along the leading edge of the blade is shown in Figure 39. The radial stress at a given point fluctuates between the top and the bottom curves due to vibration. The middle curve represents the steady state radial stress.

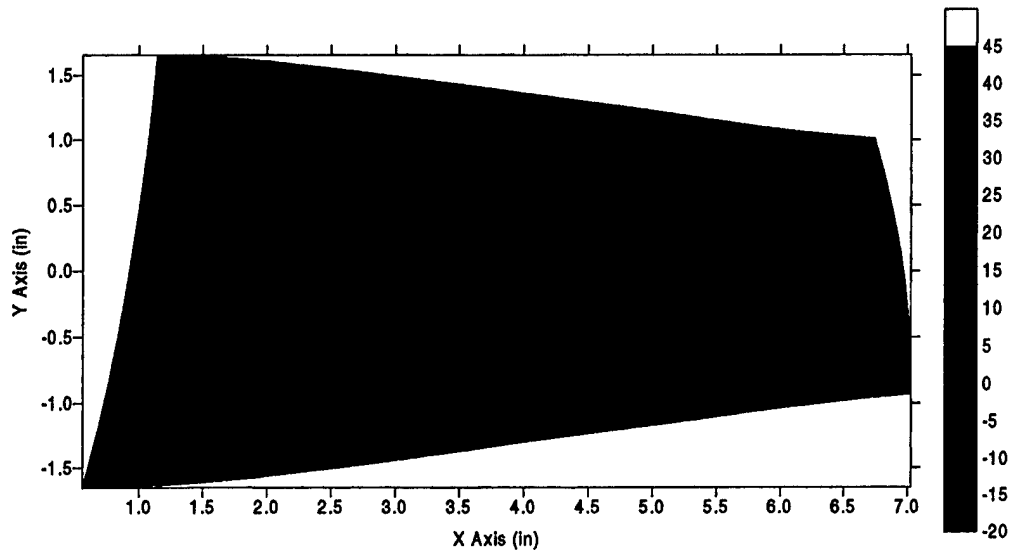


Figure 37. Pressure Side: Steady state radial stress + Mode-1 radial Stress Contours show radial stress in Ksi

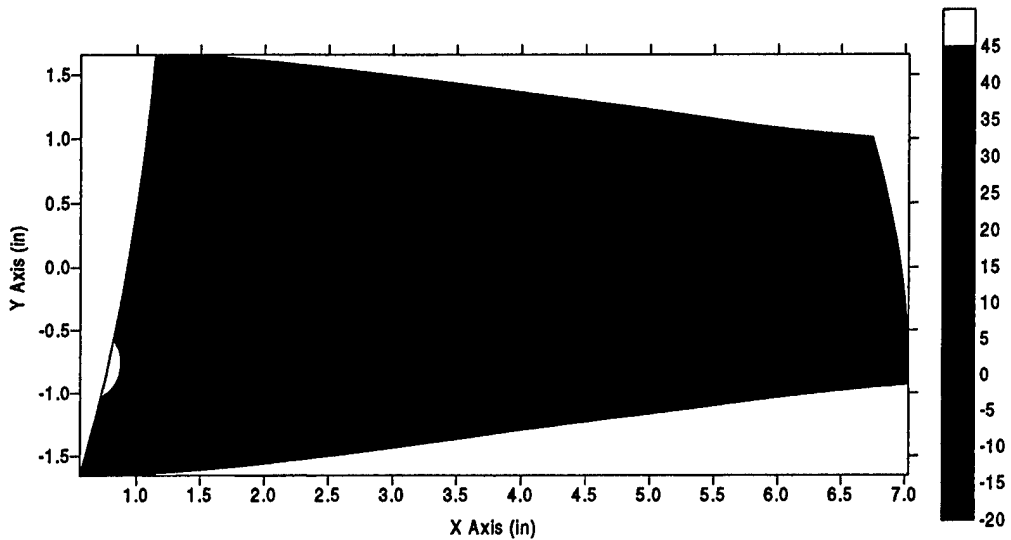


Figure 38. Pressure Side: Steady state radial stress - Mode-1 radial stress Contours show radial stress in Ksi

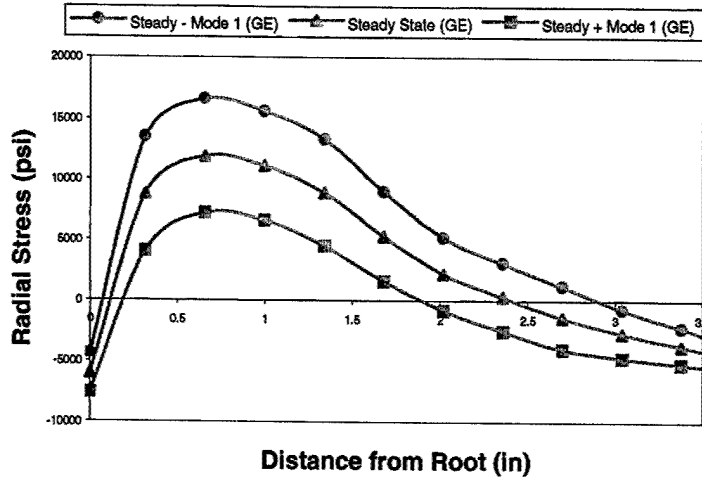


Figure 39. Radial stress along the leading edge (GE Data)

The material properties used in the analysis are listed below:
Blade (Ti-6-4):

Young's modulus: 16×10^6 psi

Poisson's ratio: 0.33

Density: 0.4211×10^{-3} lbf s²/in⁴

Grip (Steel):

Young's modulus: 30×10^6 psi

Poisson's ratio: 0.33

The finite element analysis of the fan blade was performed in two configurations. One set of analysis was performed assuming only transverse loading using the two transverse actuators. Another set of analysis was performed after adding a vertical load apart from the two transverse actuators.

In the first set of analysis with only transverse loading, four different grip designs were studied. They were (i) single grip at the top of the blade, (ii) split (two piece) grip at the top of the blade, (iii) single grip at a distance of 4 inches from the root, and (iv) split (two piece) grip at a distance of 4 inches from the root. The designs are shown in Figures 40 through 43. Loads were applied to the ends of the grips as shown in Table 5.

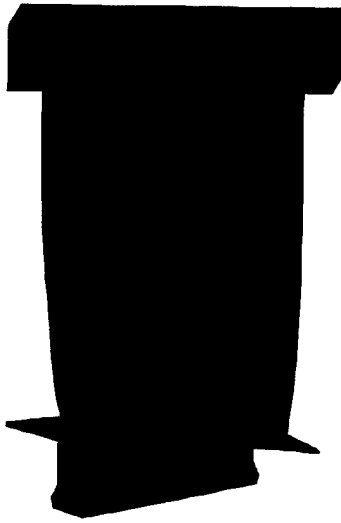


Figure 40. Single grip at the top of the blade

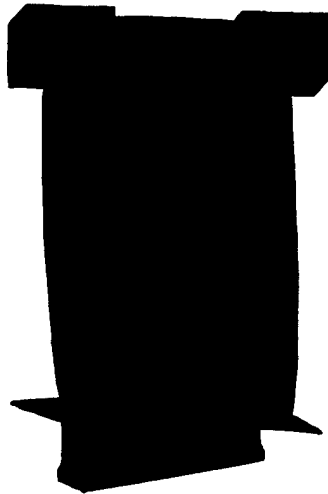


Figure 41. Split grip at the top of the blade



Figure 42. Single grip at the middle of the blade (4 inches from root)

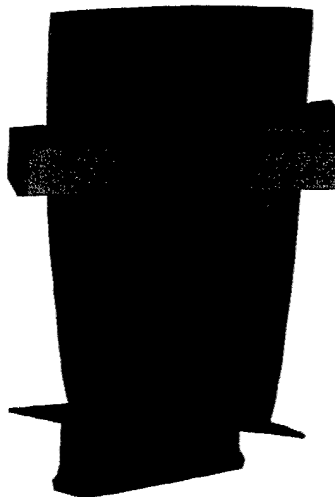


Figure 43. Split grip at the middle of the blade (4 inches from the root)

Table 5. Lateral actuators loading configuration

	Load (lb)		Loading Point
	Trailing Edge	Leading Edge	
Single grip at top	-37.5 ↔ +37.5	75	0.25" outside the edge of blade
Split grip at top	-37.5 ↔ +37.5	75	0.25" outside the edge of blade
Single grip at 4" from root	-50 ↔ +50	100	0.25" outside the edge of blade
Split grip at 4" from root	-50 ↔ +50	100	0.25" outside the edge of blade

The radial stresses along the leading edge for the above cases are shown in Figures 44 through 47. It is seen that the radial stresses along the leading edge follow the prediction by GE more closely as the grip is moved towards the root.

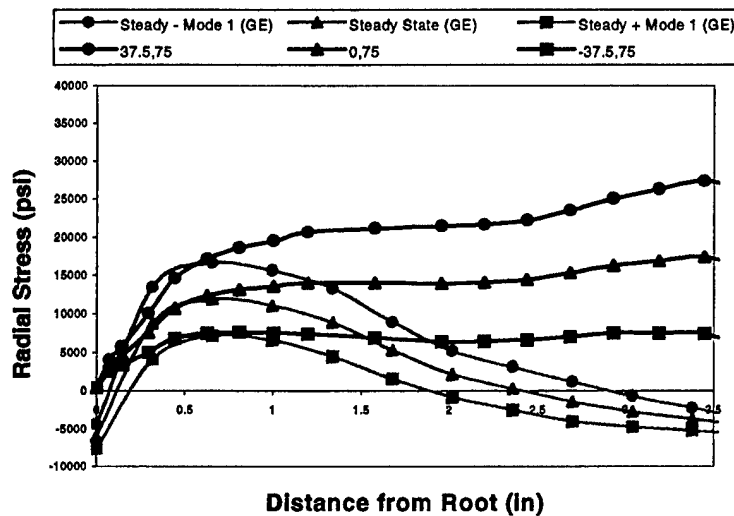


Figure 44. Radial stress along the leading edge of the blade with a single grip at the top of the blade

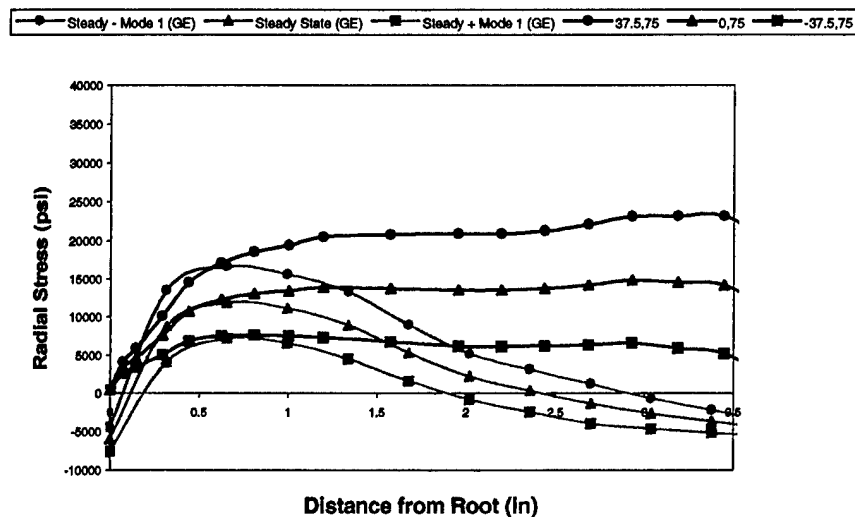


Figure 45. Radial stress along the leading edge of the blade with split grips at the top of the blade

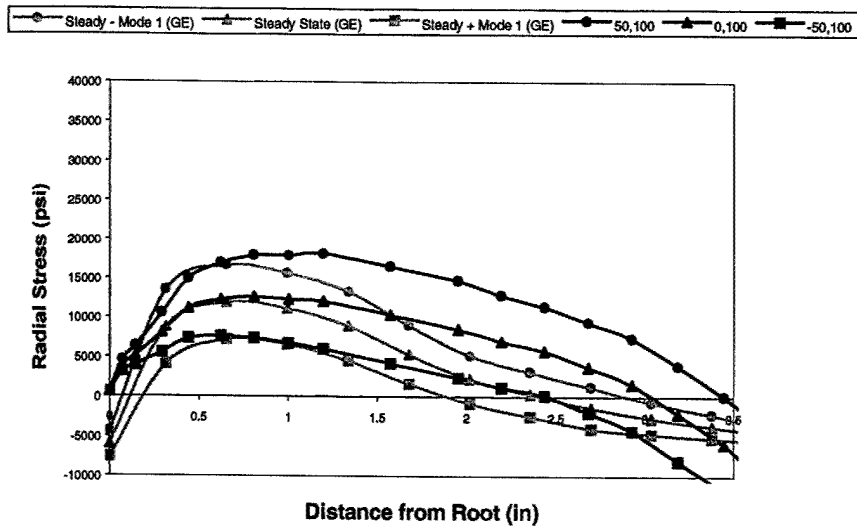


Figure 46. Radial stress along the leading edge of the blade with a single grip at the middle of the blade (4 inches from root)

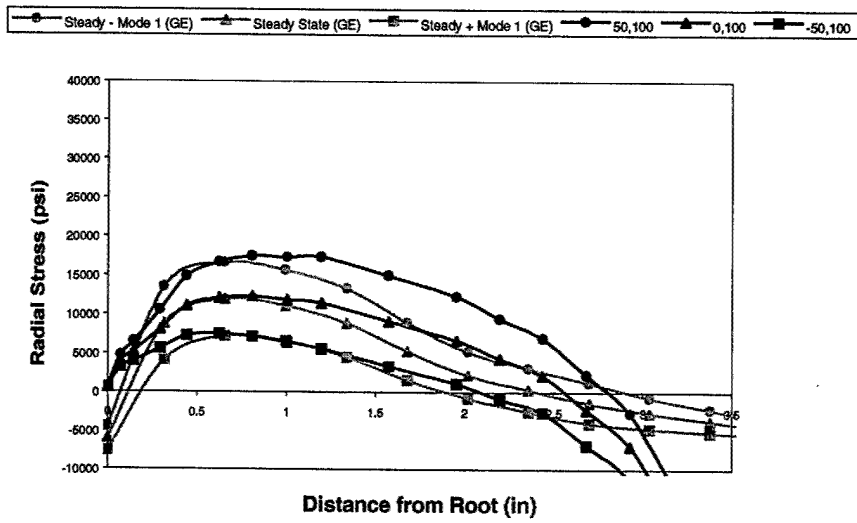


Figure 47. Radial stress along the leading edge of the blade with split grips at the middle of the blade (4 inches from root)

The cyclic displacements of the lateral actuators for the above loading conditions are shown in Figures 48 through 51. Figure 48 shows that by employing a single grip instead of a split grip at the top of the blade, the displacement required at the leading edge actuator can be reduced by about 22 percent, and the displacement required at the trailing edge can be reduced by about 1 percent.

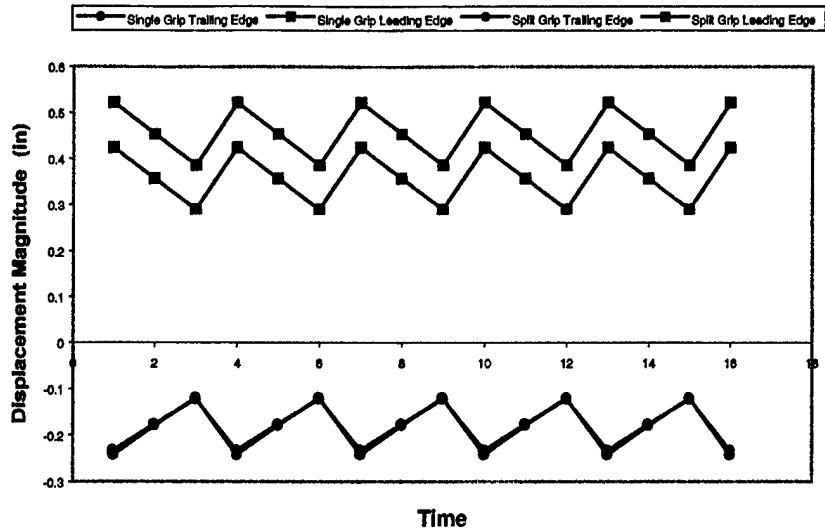


Figure 48. Comparison of actuator displacements between single and split grips at the top of the blade

Figure 49 shows that by using a single grip at about 4 inches from the root of the blade instead of a split grip will reduce displacement at the leading edge by about 30 percent and the displacement at the trailing edge by about 7 percent. These facts favor a single grip over a split grip, as reduction in the displacement of the actuators will enable moving to higher frequencies during testing.

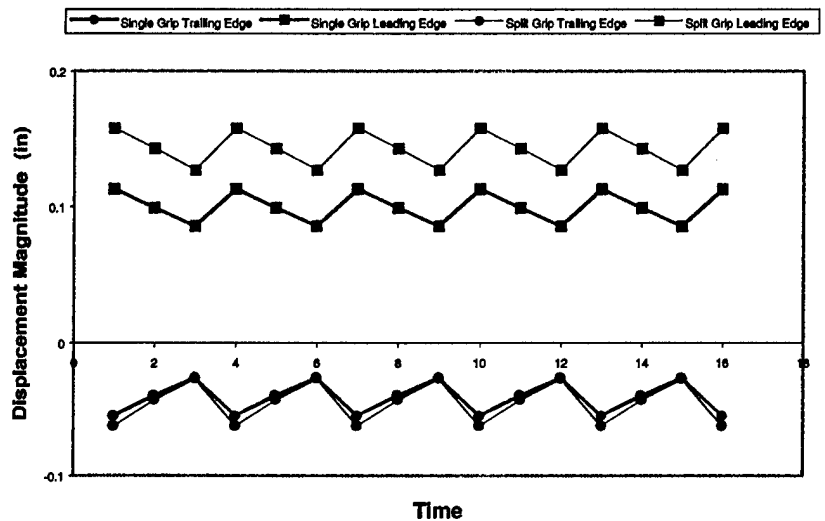


Figure 49. Comparison of actuator displacements between single and split grips at the middle of the blade (4 inches from the root)

Figure 50 shows that by moving a single grip from the top of the blade to about 4 inches from the root, the displacement required at the leading edge actuator can be reduced by about 72 percent, and the displacement required at the trailing edge can be reduced by about 77 percent. The above facts indicate that a single grip located closer to the root is desirable.

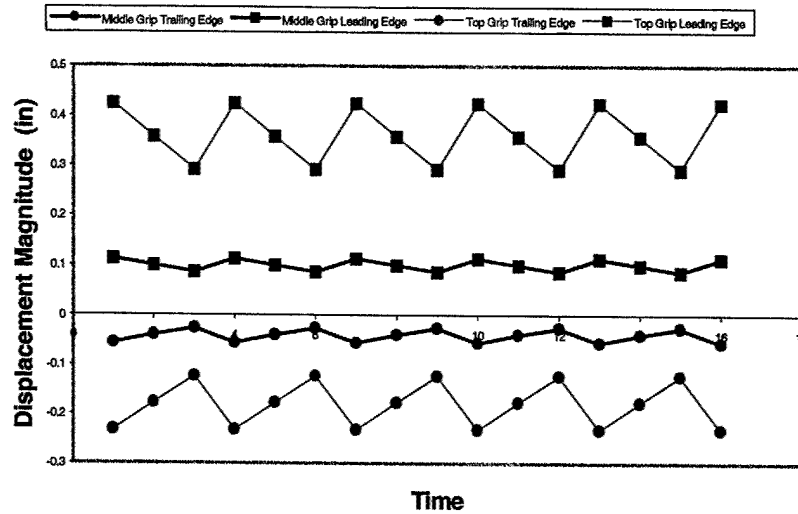


Figure 50. Comparison of actuator displacements between single grips at the top and at the middle of the blade (4 inches from the root)

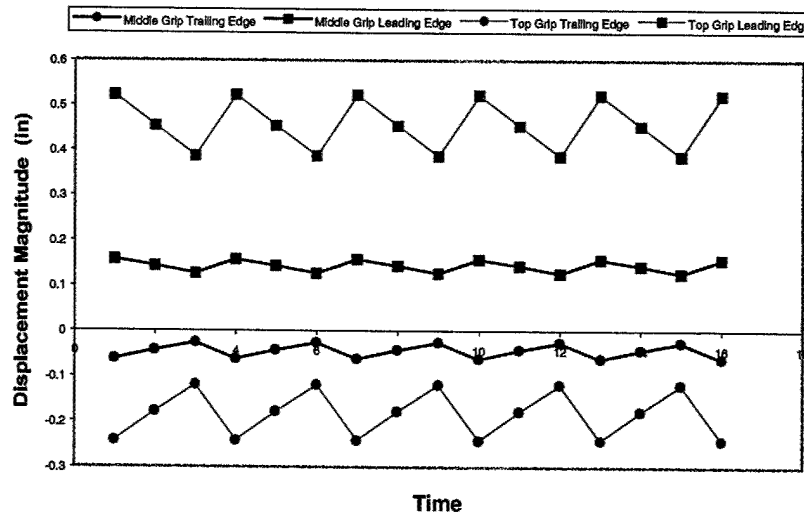


Figure 51: Comparison of actuator displacements between split grips at top and middle

To verify the results obtained from the above models, the single grip at the top case was also simulated using Model-2. The finite element mesh of Model-2 is shown in Figure 52. The radial stresses along the leading edge of the blade for these simulations are shown in Figure 53.

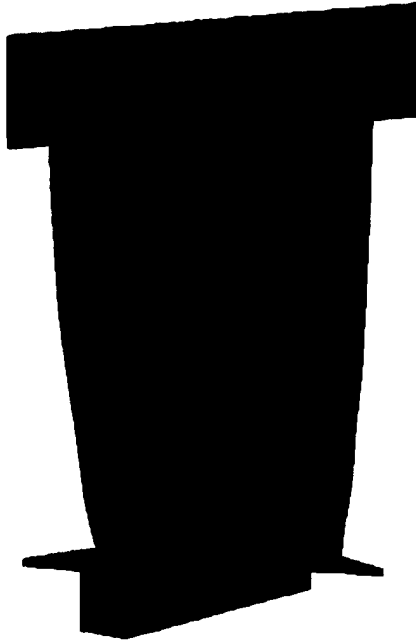


Figure 52. Single grip at the top of the blade (Model-2)

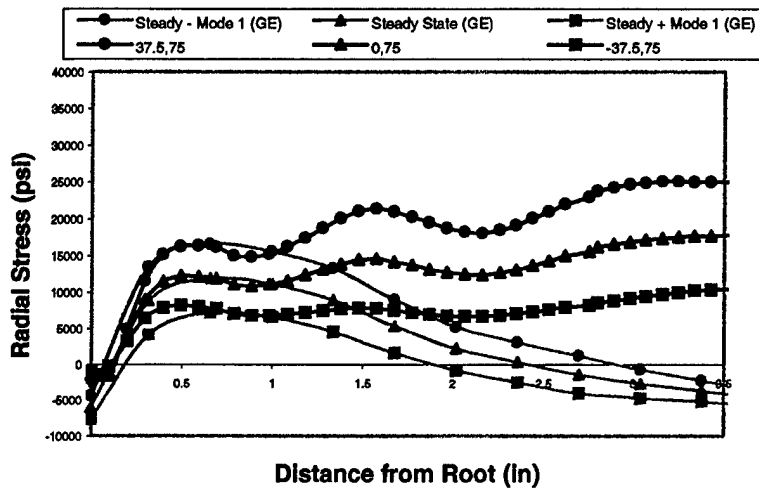


Figure 53. Radial stress along the leading edge of the blade with a single grip at the top of the blade (Model-2)

The blade was also analyzed by adding a strap to impart vertical loading apart from the transverse loading. The finite element model is shown in Figure 54. The default strap properties used for simulation were: strap modulus = $2/3$ of blade modulus, length = 3.625 inches, width = 1.82 inches and thickness = 0.046 inch. The strap was modeled using quadrilateral parabolic thin shell elements.

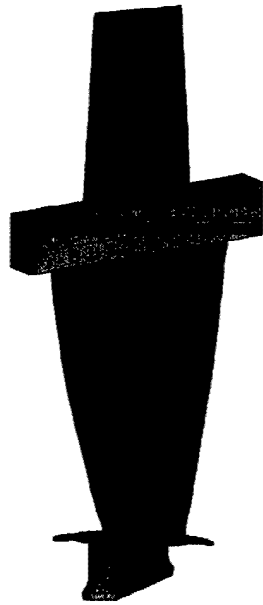


Figure 54. Single grip at the top of the blade with strap

The radial stresses along the leading edge for the above simulation is shown in Figure 55. It is seen that the addition of the strap to the transverse loading imposes a compressive stress along the leading edge. This is due to the fact that the addition of the strap moves the model from a cantilever to a fixed-fixed state. This result does not reflect the actual mechanism accurately. In the actual grip, the single piece of strap slides around the grip pin. This provides a certain amount of freedom for the grip to move in the transverse direction.

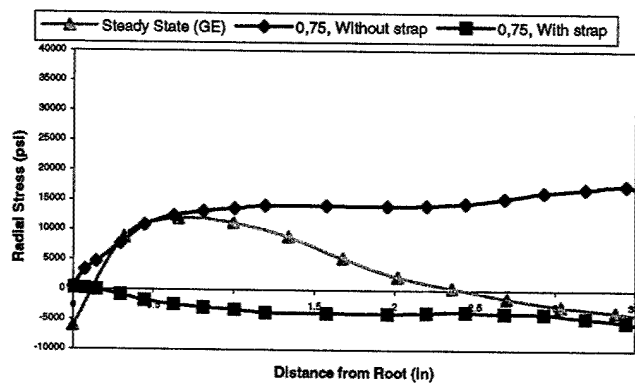


Figure 55. Radial stress along the leading edge of the blade with a single grip at the top of the blade and a vertical strap

6.0 HIGH-CYCLE FATIGUE TESTING OF A GE ENGINE FAN BLADE

The performance of the multiaxial test system was evaluated initially by conducting trial runs to frequencies as high as 350 Hz. High-frequency fatigue loading was applied to a tapered trial specimen made of titanium alloy to examine the performance of the gripping system. It was found that the gripping system was adequate for the high-cycle fatigue testing.

The control system was fine tuned to the requirements of the high-cycle fatigue testing of the turbine blades. Both stroke and load controls were setup in the system. The load control should be used first to calibrate the test at the beginning of the test. After the stroke magnitudes are determined for a certain force input, the control should be switched to stroke control, which allows more accurate control and input at high frequencies.

A GE F110 second stage fan blade was then installed in the test frame. Multiaxial fatigue loading with frequencies up to 100 Hz was applied to the specimen, initially, to examine the performance of the test system. After ensuring the satisfactory performance of the test system, high-frequency loading was applied to the blade. The tests were performed under displacement control, and the displacement amplitude varied from 0.01 inch to 0.015 inch. The loading frequency varied from 100 Hz to 200 Hz. The blade has been subjected to approximately 53 million loading cycles from December 1999 to July 2000 under inplane, bending and torsion loading conditions.

The test frame and all the critical joints were examined periodically for fatigue crack developments. No major problems have been encountered so far in the high-cycle fatigue testing of the fan blade. The blade was also examined periodically for fatigue damage. High-frequency fatigue test of the GE F110 second stage fan blade is still under progress.

7.0 SUMMARY

In Phase II of this SBIR research program, a multiaxial loading machine was proposed to test gas turbine engine blades at high frequencies. The multiaxial loading machine has three individual hydraulic actuators. One actuator is mounted in the vertical axis and is used to simulate the radial load on the blade. The other two actuators are mounted in the horizontal axis and are used to apply a bending or torsion load to the blade at high frequencies. This enables the machine to apply both loads to the blade simultaneously.

Finite element analysis of a GE F110 second fan blade was performed to determine the magnitude, location, and direction of the loads required to simulate the conditions faced by the blade in the turbine during actual operation. The loading requirements obtained from the analysis served as a guideline in selecting the hydraulic components and controls. The loading requirements also dictated the design of the test frame and the gripping system. Static and dynamic finite element analysis was used to ensure that the test frame withstood the rigors of high-frequency testing.

The parts of the test frame were designed, manufactured, and assembled using stringent quality standards. The test frame, hydraulic components, control system, and data acquisition system were integrated successfully into a fully functional multiaxial loading machine. The multiaxial high-frequency testing capability of the machine was demonstrated by running tests on trial specimen and an actual GE F110 second stage fan blade. The blade has been subjected to more than 53 million cycles up to a frequency of 200 Hz without any signs of damage in the blade.

A video was made showing the multiaxial loading machine in action and the concept behind its evolution. An instruction manual for operating the machine has also been written.

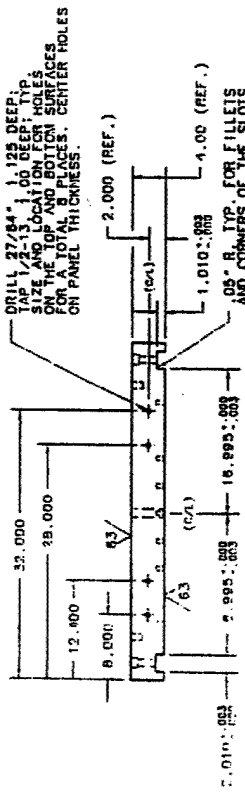
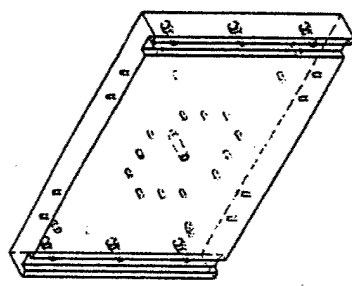
APPENDIX A

Part Drawings of the Test Frame

4 3 2 1

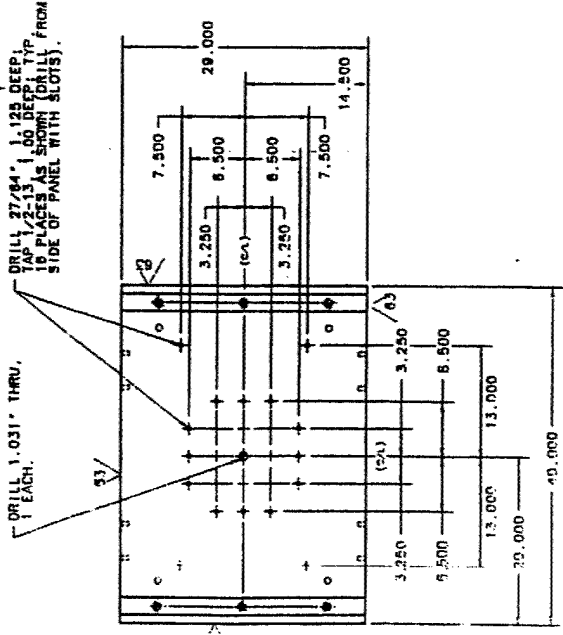
ZONE	REV	DESCRIPTION	DATE	APPROVED
D-1	A	SURFACE FINISH NOTE (2 EA)	05-12-89	M. XIE
D-1	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-2	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-3	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-4	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-5	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-6	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-7	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-8	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-9	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-10	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-11	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-12	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-13	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-14	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-15	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-16	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-17	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-18	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-19	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-20	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-21	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-22	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-23	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-24	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-25	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-26	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-27	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-28	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-29	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-30	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-31	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-32	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-33	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-34	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-35	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-36	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-37	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-38	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-39	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-40	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-41	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-42	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-43	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-44	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-45	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-46	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-47	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-48	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-49	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE
D-50	A	SCALE NOTATION ON DIMENSION	05-12-89	M. XIE

NOTES:
 - MATERIALS IS TO BE 4" BLANCHARD GROUND STEEL.
 - QUANTITY OF 1 EACH REQUIRED.
 - PANEL IS TO RECEIVE FLASH NICKEL PLATING AFTER MACHINING. (NICKEL PLATING THICKNESS -.002).
 - REMOVE ALL SHARP EDGES AND CORNERS.

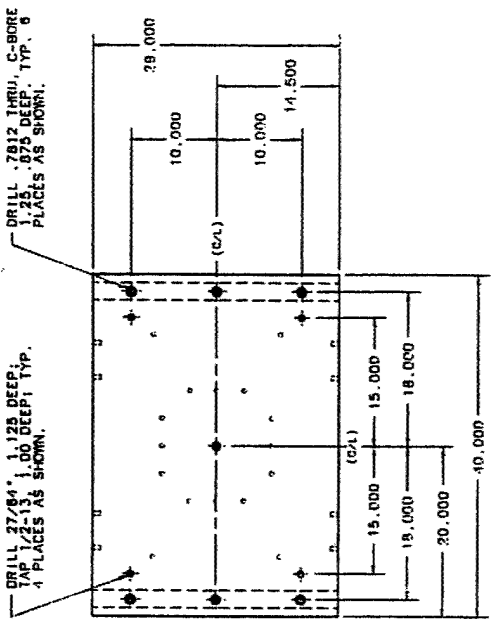


NOTE: SLOT DIMENSIONS ARE TYP. FOR BOTH SLOTS. SLOT WALLS ARE TO BE PARALLEL WITH EACH OTHER AND PARALLEL WITH THE PANEL SIDE WALLS TO WITHIN ±.001.

TOP VIEW



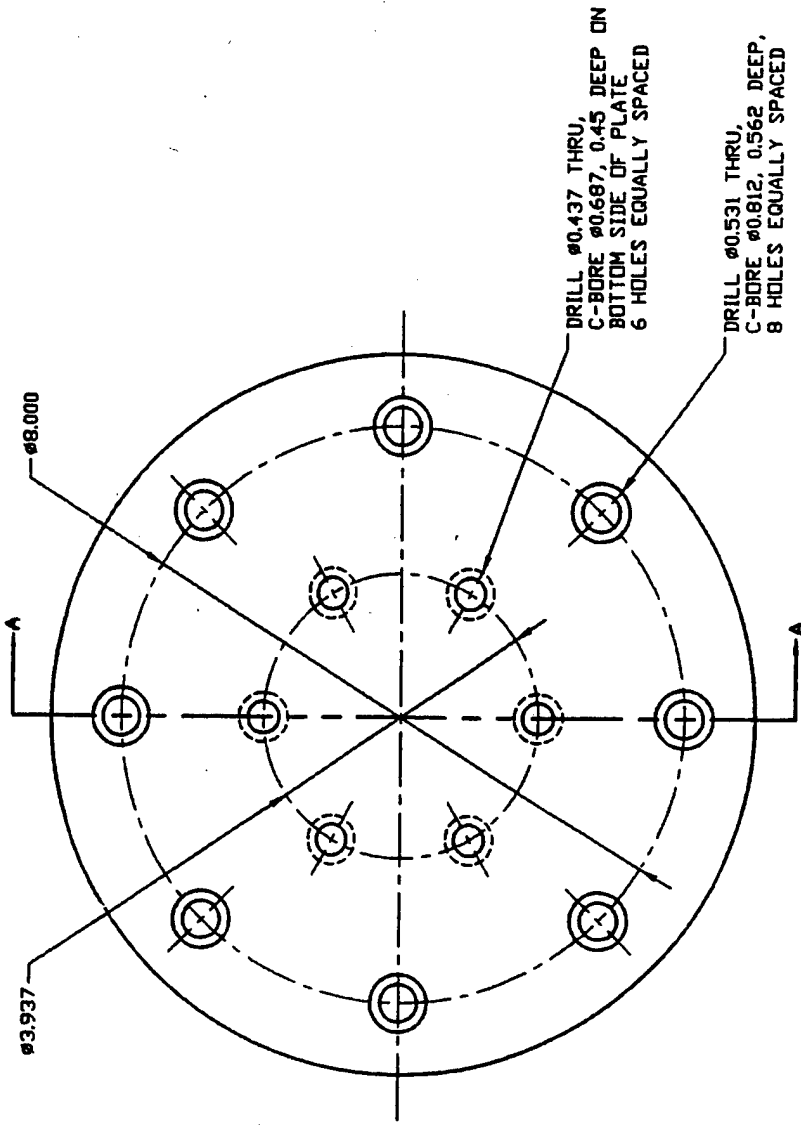
FRONT VIEW



BACK VIEW

REV	DATE	DESCRIPTION	APPROVED
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2	05/12/89	REVISIONS	M. XIE
3	05/12/89	REVISIONS	M. XIE
4	05/12/89	REVISIONS	M. XIE
5	05/12/89	REVISIONS	M. XIE
6	05/12/89	REVISIONS	M. XIE
7	05/12/89	REVISIONS	M. XIE
8	05/12/89	REVISIONS	M. XIE
9	05/12/89	REVISIONS	M. XIE
10	05/12/89	REVISIONS	M. XIE
11	05/12/89	REVISIONS	M. XIE
12	05/12/89	REVISIONS	M. XIE
13	05/12/89	REVISIONS	M. XIE
14	05/12/89	REVISIONS	M. XIE
15	05/12/89	REVISIONS	M. XIE
16	05/12/89	REVISIONS	M. XIE
17	05/12/89	REVISIONS	M. XIE
18	05/12/89	REVISIONS	M. XIE
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24	05/12/89	REVISIONS	M. XIE
25	05/12/89	REVISIONS	M. XIE
26	05/12/89	REVISIONS	M. XIE
27	05/12/89	REVISIONS	M. XIE
28	05/12/89	REVISIONS	M. XIE
29	05/12/89	REVISIONS	M. XIE
30	05/12/89	REVISIONS	M. XIE
31	05/12/89	REVISIONS	M. XIE
32	05/12/89	REVISIONS	M. XIE
33	05/12/89	REVISIONS	M. XIE
34	05/12/89	REVISIONS	M. XIE
35	05/12/89	REVISIONS	M. XIE
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44	05/12/89	REVISIONS	M. XIE
45	05/12/89	REVISIONS	M. XIE
46	05/12/89	REVISIONS	M. XIE
47	05/12/89	REVISIONS	M. XIE
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49	05/12/89	REVISIONS	M. XIE
50	05/12/89	REVISIONS	M. XIE

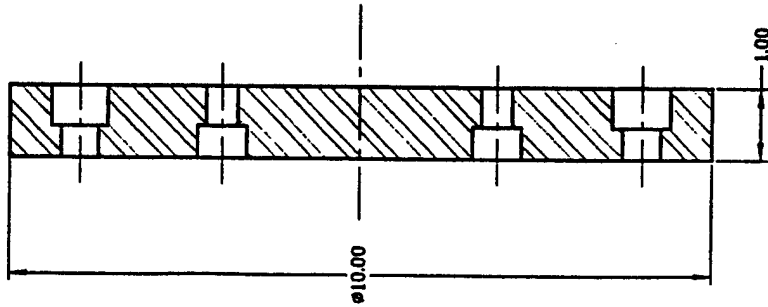
4 3 2 1



DRILL $\phi 0.437$ THRU,
C-BORE $\phi 0.687$, 0.45 DEEP ON
BOTTOM SIDE OF PLATE
6 HOLES EQUALLY SPACED

DRILL $\phi 0.531$ THRU,
C-BORE $\phi 0.812$, 0.562 DEEP,
8 HOLES EQUALLY SPACED

$\phi 3.937$ $\phi 8.000$



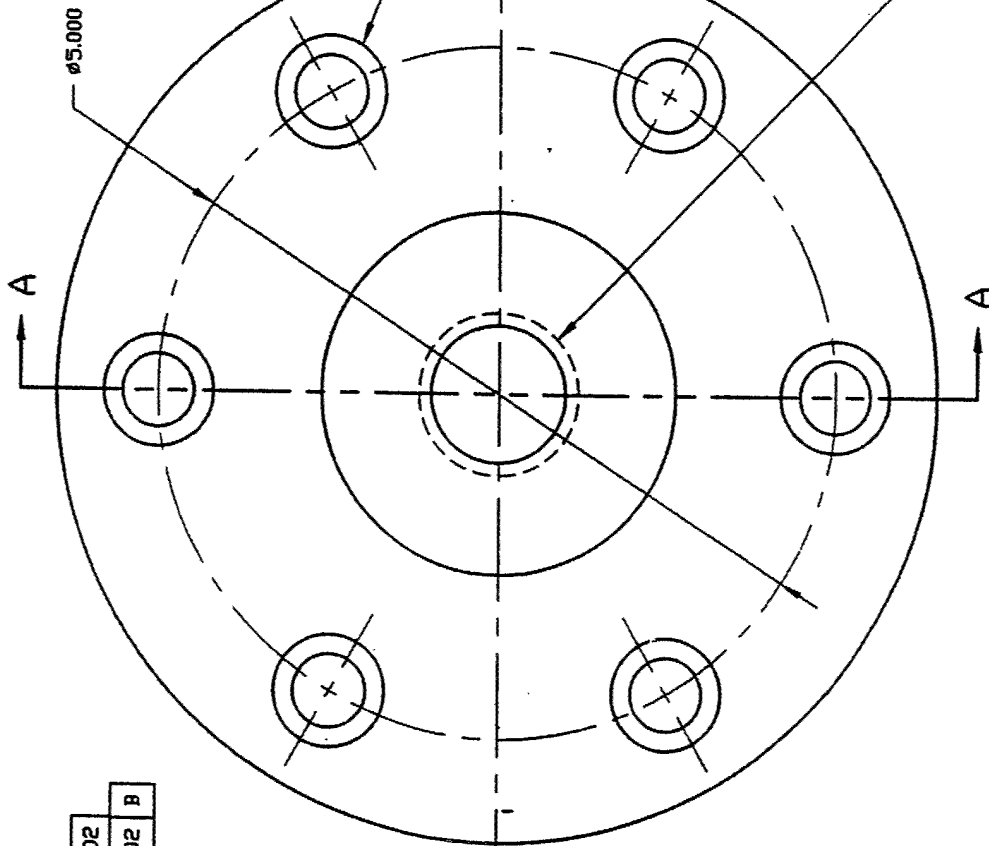
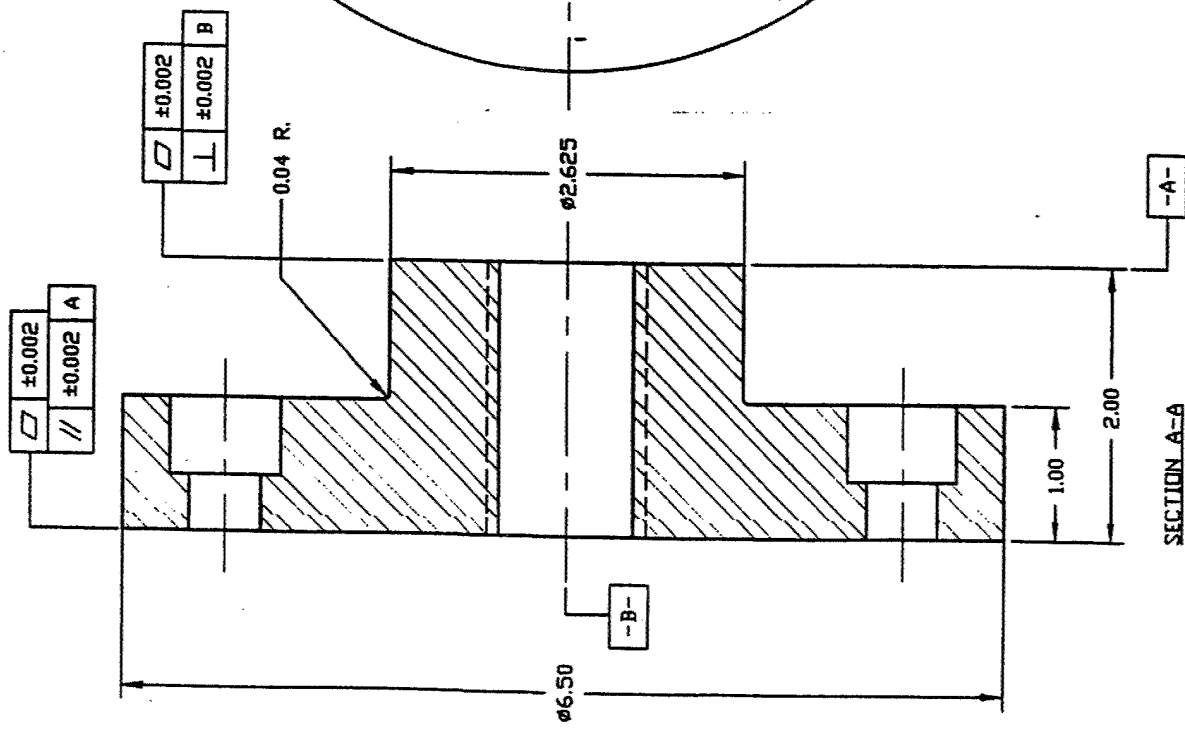
$\phi 10.00$

1.00

SECTION A-A

- NOTES:
- QUANTITY OF ONE REQUIRED.
 - MATERIAL IS STEEL.
 - BLACK OXIDE SURFACE TREATMENT REQUIRED.
 - REMOVE ALL SHARP EDGES.

ADTECH SYSTEMS RESEARCH INC.	
1342 NORTH FAIRFIELD RD., DAYTON, OH 45432-2898	
PH.# (937) 426-3329 FAX# (937) 426-8087	
DRAWING NAME: ADAPTER - TOP DIE PLATE	
DRAWING NUMBER: MATE-006	DATE: 08/10/1999
DIMENSION: INCHES	TOLERANCE: $\pm 0.003"$ OR AS SPECIFIED
QUANTITY: 1	FINISH: BLACK OXIDE
MATERIAL: STEEL	DRAWN BY: MOLMAN
APPROVED BY:	BRUNO VIE



DRILL $\phi 0.531$ THRU,
C-BORE $\phi 0.812$, 0.562 DEEP,
6 HOLES EQUALLY SPACED.

DRILL 28mm THRU,
TAP M30x2-LH THRU.

- NOTES:**
- QUANTITY OF ONE REQUIRED.
 - MATERIAL IS STEEL.
 - BLACK OXIDE SURFACE TREATMENT REQUIRED.
 - REMOVE ALL SHARP EDGES.

ADTECH SYSTEMS RESEARCH INC.

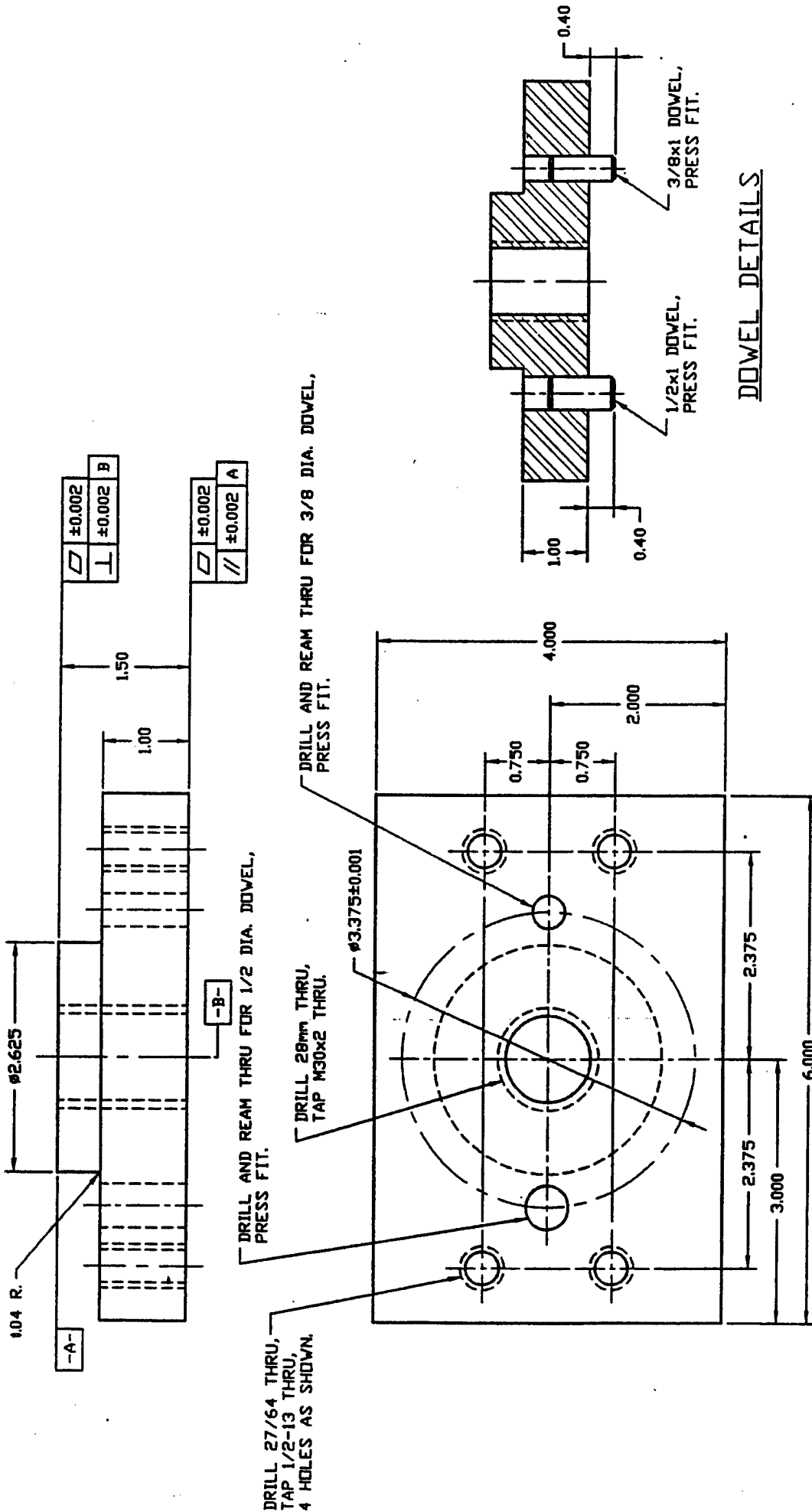
1342 NORTH FAIRFIELD RD., DAYTON, OH 45432-2698
PH.# (937) 426-3329 FAX# (937) 426-9087

DRAWING NAME:
ADAPTER - ACTUATOR

DRAWING NUMBER:
MATF-007

DATE:
08/21/1999

TOLERANCE: $\pm 0.003"$ OR AS SPECIFIED	QUANTITY: 1	MATERIAL: STEEL	FINISH: BLACK OXIDE	DRAWN BY: MOHAN	APPROVED BY: MING XIE
DIMENSION: INCHES					

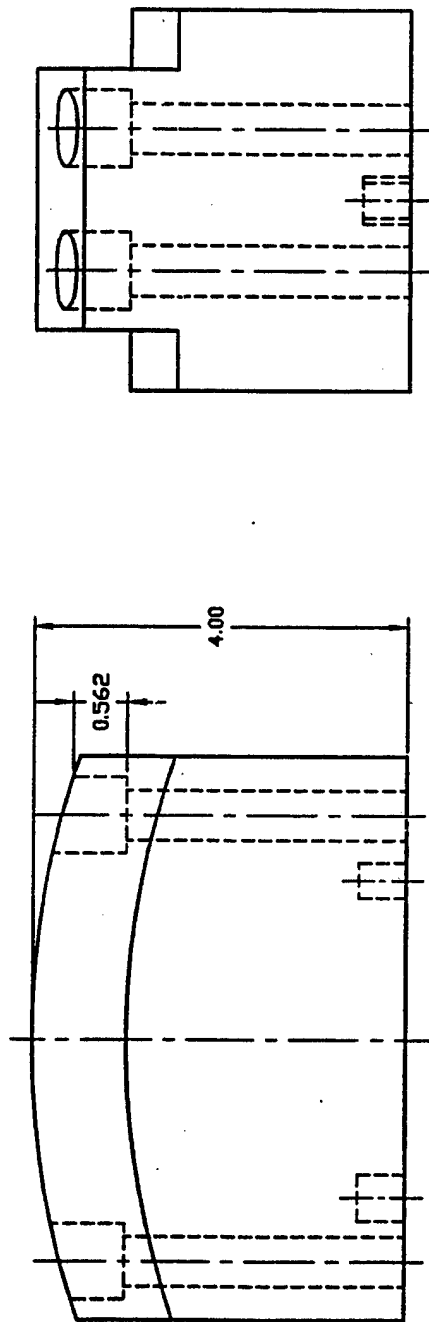


DOWEL DETAILS

- NOTES:**
- QUANTITY OF ONE REQUIRED.
 - MATERIAL IS STEEL.
 - BLACK OXIDE SURFACE TREATMENT REQUIRED.
 - SUPPLY PART WITH THE 1/2x1 AND 3/8x1 DOWELS IN POSITION.
 - HEIGHT OF DOWELS EXTENDING OUT OF THE BLOCK SHOULD BE 0.40".
 - REMOVE ALL SHARP EDGES.

ADTECH SYSTEMS RESEARCH INC.
 1342 NORTH FAIRFIELD RD., DAYTON, OH 45432-2698
 PH.# (937) 426-3329 FAX# (937) 426-8087
 DRAWING NAME:
 ADAPTER - BROACH BLOCK

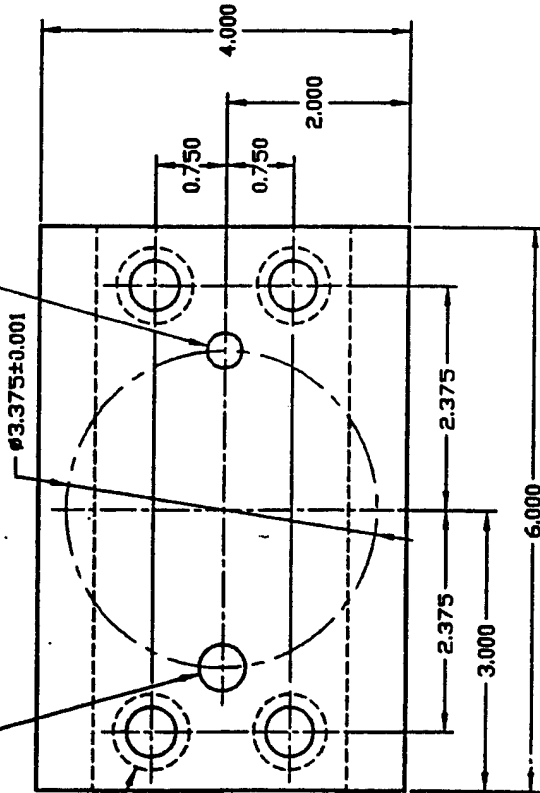
DRAWING NUMBER: MATF-008	DATE: 08/10/1999	DIMENSION: INCHES	TOLERANCE: ± 0.003" OR AS SPECIFIED	QUANTITY: 1	MATERIAL: STEEL	FINISH: BLACK OXIDE	DRAWN BY: MOHAN	APPROVED BY: MING XIE
-----------------------------	---------------------	----------------------	--	----------------	--------------------	------------------------	--------------------	--------------------------



DRILL $\phi 0.531$ THRU,
C-BORE $\phi 0.812$, 0.562 DEEP,
4 HOLES AS SHOWN.

DRILL AND REAM FOR 1/2 DIA. DOVEL,
0.5 DEEP, SLIDING FIT.

DRILL AND REAM FOR 3/8 DIA. DOVEL,
0.5 DEEP, SLIDING FIT.



- NOTES:**
- QUANTITY OF ONE REQUIRED.
 - MATERIAL IS TITANIUM ALLOY (TI-6-4).
 - PART WILL BE PROVIDED.
 - MACHINE 4 BOLT HOLES AND 2 DOVEL HOLES AS SHOWN.
 - PART HAS OTHER HOLES ALREADY. THEY ARE NOT SHOWN FOR CLARITY.
 - HOLE LOCATIONS HAVE TO MATCH WITH ADAPTER-BROACH BLOCK (DRG. # MATF-008).
 - PART HAS TO SLIDE IN AND OUT SMOOTHLY OVER ADAPTER-BROACH BLOCK DOVELS.
 - REMOVE SHARP EDGES.

ADTECH SYSTEMS RESEARCH INC.

1342 NORTH FAIRFIELD RD., DAYTON, OH 45432-2898
 PH.# (937) 428-3329 FAX# (937) 428-0087

DRAWING NAME:
BROACH BLOCK

DRAWING NUMBER:
MATF-010

DATE:
06/10/1999

DIMENSION:
INCHES

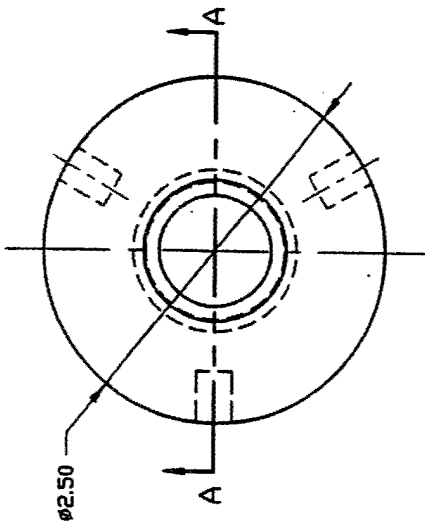
TOLERANCE:
 ± 0.003 " OR AS SPECIFIED

QUANTITY:
1

MATERIAL:
T.L.A.

DRAWN BY:
BROACH

APPROVED BY:

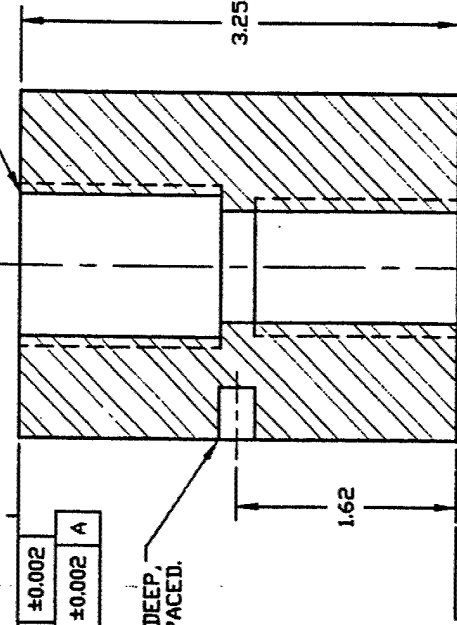


DRILL AND REAM 28mm, 1.50 DEEP,
TAP M30x2, 1.50 DEEP.

○	±0.002	B
⌘	±0.002	B

⌘	±0.002
//	±0.002 A

DRILL $\phi 0.266$, 0.375 DEEP,
3 HOLES EQUALLY SPACED.



⌘	±0.002
⊥	±0.002 B

DRILL AND REAM 15/16 THRU,
TAP 1-14, 1.50 DEEP.

SECTION A-A

NOTES:

- QUANTITY OF ONE REQUIRED.
- MATERIAL IS STEEL.
- THREADED HOLES SHOULD BE CONCENTRIC.
- BLACK OXIDE SURFACE TREATMENT REQUIRED.
- REMOVE ALL SHARP EDGES.

ADTECH SYSTEMS RESEARCH INC.

1342 NORTH FAIRFIELD RD., DAYTON, OH 45432-2898

PH.# (937) 426-3329 FAX# (937) 426-0087

DRAWING NAME:
ADAPTER - LOAD CELL

DRAWING NUMBER:
MATF-011

DATE:
06/10/1999

DIMENSION:
INCHES

TOLERANCE:
± 0.003" OR AS SPECIFIED

QUANTITY:
1

MATERIAL:
STEEL

FINISH:
BLACK OXIDE

DRAWN BY:
MOHAN

APPROVED BY:
MING XIE

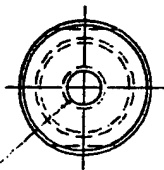
1 2 4

- NOTES:
- MATERIAL IS TO BE COLD ROLLED STEEL.
 - QUANTITY OF 2 EACH.
 - PART IS TO BE BLACK OXIDED.
 - REMOVE ALL SHARP EDGES.

REV. NO.		SH	REV.
C-2	A		ADDED WRENCH FLATS
DATE		APPROVED	
08-02-99		M. XIE	

REVISIONS

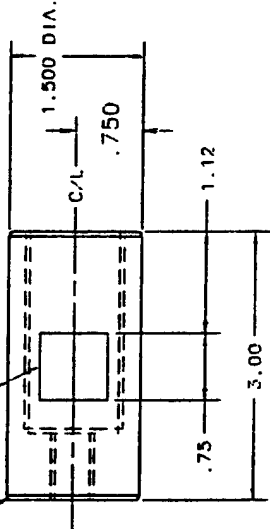
DRILL .01" THRU; TAP 3/8-24 THRU, 1 PLACE ONLY.



LEFT SIDE VIEW

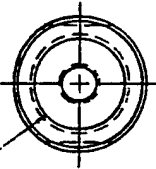
.05 RADIUS TYP. BOTH ENDS.

CUT WRENCH FLATS .10 DEEP. TYP. 2 PLACES 360 DEGREES APART.



FRONT VIEW

BORE 15/16" DIA. 2.25 DEEP. CHASE A. 1-14 THREAD. 2.20 DEEP.



RIGHT SIDE VIEW

QTY REQD	PART NO.	PART OR IDENTIFYING NO.	TEMPERATURE OR DESCRIPTION	PARTIAL SPECIFICATION	TITLE NO.
			ADTECH SYSTEMS RESEARCH INC		
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES			PARTS LIST		
FUNCTIONS			CONTRACT NO.		
TOLERANCES ARE:			DATE		
FRACTIONS .005			07-18-99		
DECIMALS .005			APPROVALS		
ANGLES 1 0.30°			BY: MARY BOWEN-CORST		
DO NOT SCALE DRAWING			CHECKED		
TITLES			DATE		
BLACK OXIDE (SEE NOTE)			MIMO XIE		
FINISH			SERIAL		
SIMILAR TO			MATERIAL		
MATERIAL			SCALE		
SCALE NONE			SHEET 1 OF 1		

CALIBRATION ADAPTOR

ADTECH SYSTEMS RESEARCH INC
BEAVERCREEK, OHIO
PH: 937-426-3329

DATE: 07-18-99
BY: MARY BOWEN-CORST
CHECKED: MIMO XIE
SERIAL: [blank]

SCALE: NONE
SHEET: 1 OF 1

1

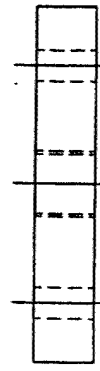
2

4

REV. NO.		SH		REV.	
ZONE		REV		DESCRIPTION	
B-3		A		CHANGED PLATE THICKNESS	
DATE		APPROVED		MATERIAL SPECIFICATION	
08-02-99		M. XIE			

NOTES:

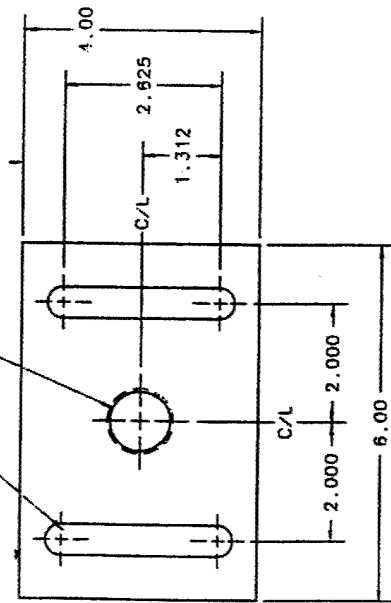
- MATERIAL IS TO BE COLD ROLLED STEEL.
- QUANTITY OF 2 EACH.
- REMOVE ALL SHARP EDGES.
- AFTER MACHINING PART IS TO BE BLACK OXIDED.



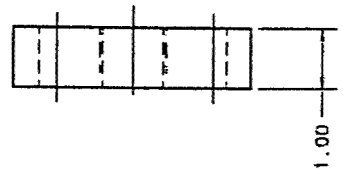
TOP VIEW

MILL SLOT .5312 DIA. THRU. TYP 2 PLACES EACH.

DRILL 15/16" THRU. TAP 1-14 THRU. TYP. 1 EACH.



FRONT VIEW



RIGHT SIDE VIEW

CITY	REQD	PART OR IDENTIFICATION NO.	NOMENCLATURE OR DESCRIPTION	MATERIAL SPECIFICATION	ITEM NO.
			PARTS LIST		
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES FRACTIONS DECIMALS ANGLES			ADTECH SYSTEMS RESEARCH INC.		
DO NOT SCALE DRAWING			BEAVERCREEK OHIO		
TREATMENT			PH: 937-428-3329		
APPROVALS			TITLE		
DRAWN			DATE		
CHECKED			07-16-99		
ISSUED			07-16-99		
BLACK OXIDE (SEE NOTE)			SIZE		
PART. NO.			B		
SCALE			NONE		
SHEET			1 OF 1		

1

2

4

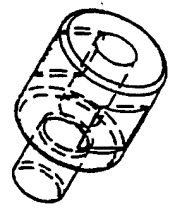
CALIBRATION PLATE

ADTECH SYSTEMS RESEARCH INC.
BEAVERCREEK OHIO
PH: 937-428-3329

4

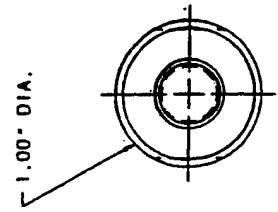
- NOTES:
- MATERIAL IS TO BE COLD ROLLED STEEL.
 - QUANTITY OF 2 EACH.
 - PARTS ARE TO BE BLACK OXIDED.
 - REMOVE ALL SHARP EDGES.

REVISIONS		DATE	APPROVED
ZONE	REV		

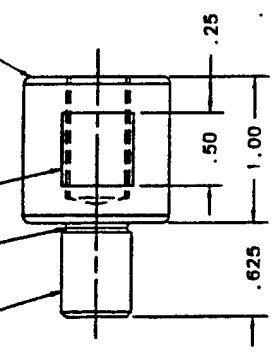


ISO VIEW

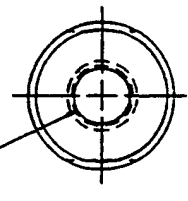
M12 THREAD
 CUT RADIUS RELIEF .05 R.
 CUT WRENCH FLAT .0625 DEEP TYP. 2 PLACES, SPACED 360 DEGREES APART.
 ROUND BOTH ENDS .05 R.



LEFT SIDE VIEW



FRONT VIEW

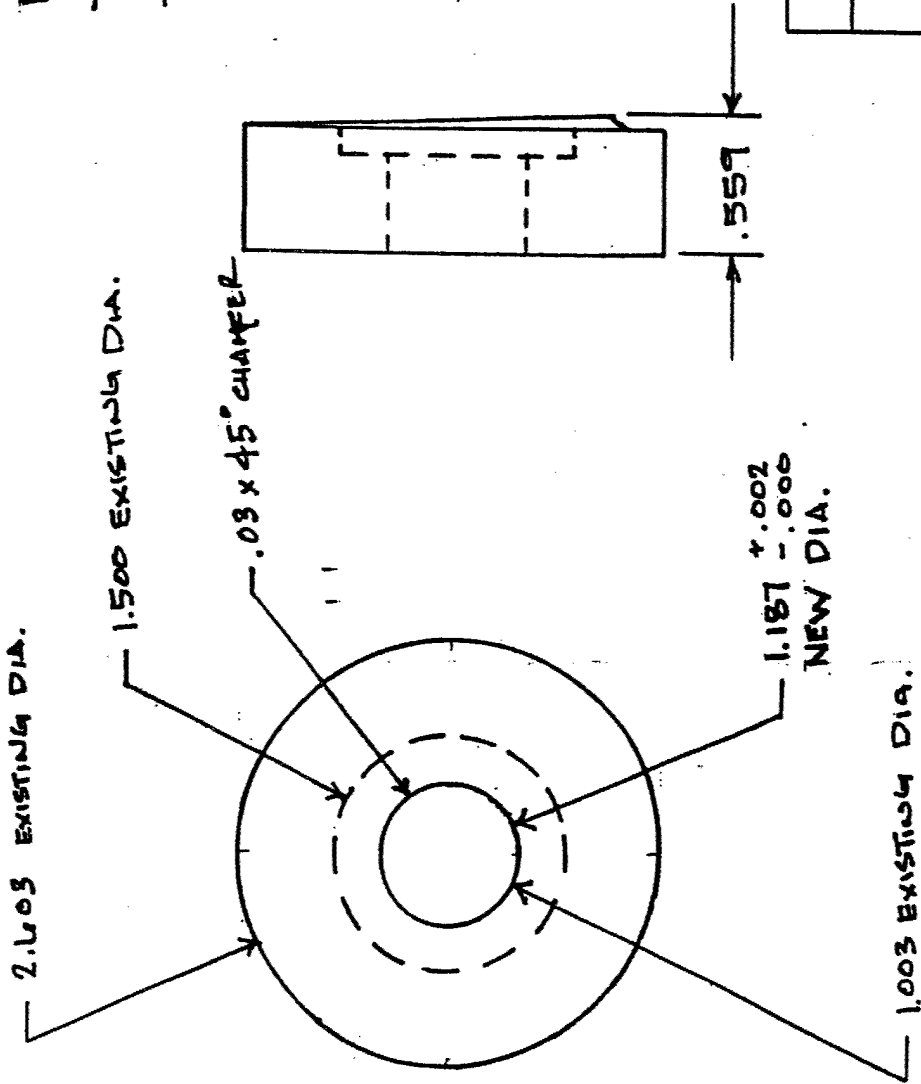


RIGHT SIDE VIEW

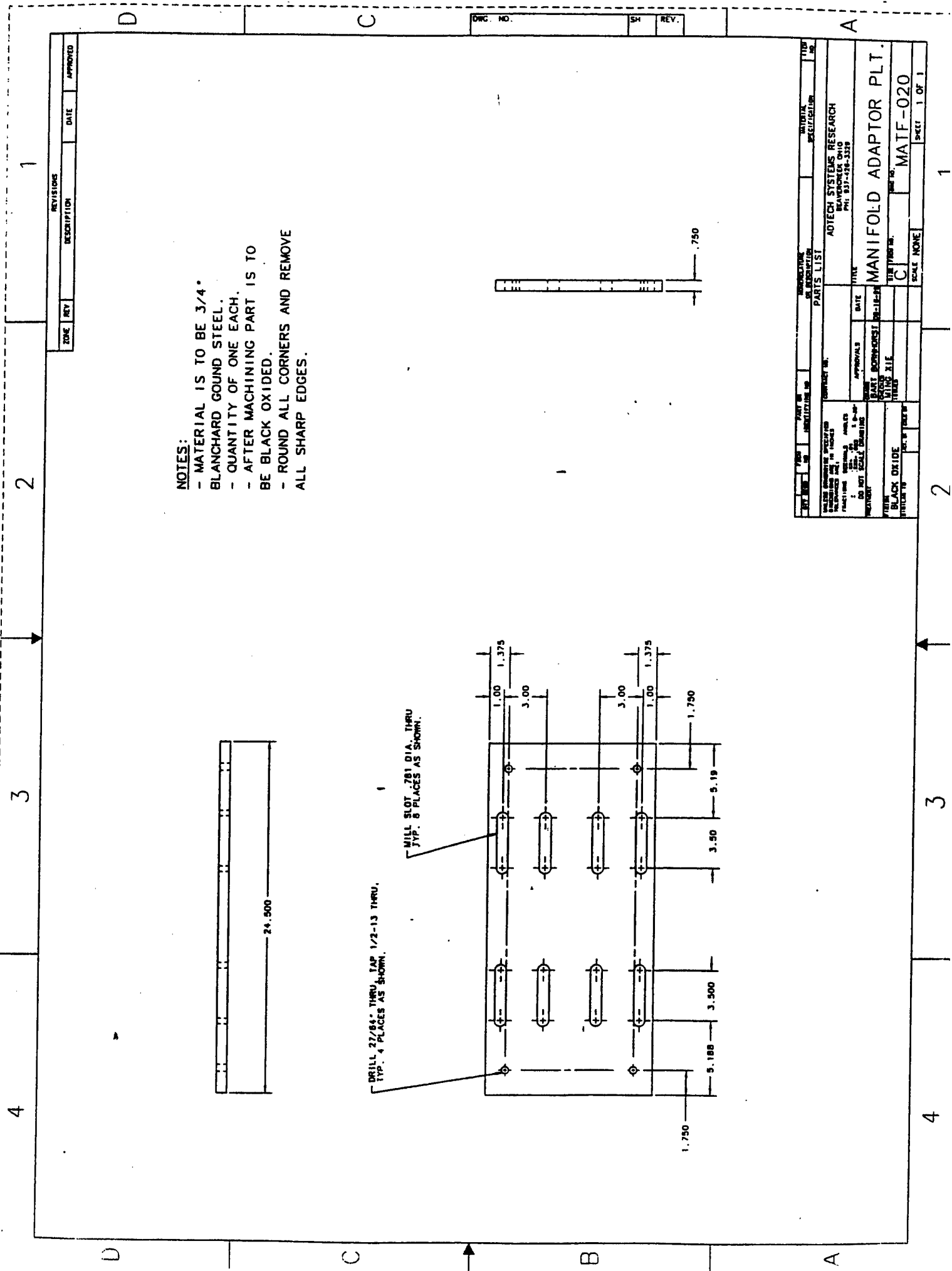
QTY. REQD.	ISSUE NO.	REV. IN LIBRIZATION NO.	REPRODUCTION OR REVISION	MATERIAL SPECIFICATION	ITEM NO.
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES FRACTIONS DECIMALS ANGLES .25 .50 .75 1 1.5 2 3 4 5 6 8 10 12 15 20 25 30 36 45 60 90 120 150 180 DO NOT SCALE DRAWING			PARTS LIST		
TREATMENT		APPROVALS	DATE	TITLE	
FINISH		BARBAR BART BORNHORST	07-27-89	LOAD CELL ADAPTOR	
SIMILAR TO		ORDERED BY	ISSUED	DRAW NO. MATF-018	
		SCALE NONE		SHEET 1 OF 1	

NOTES:

- SPIRAL WASHERS MATERIAL IS ALUMINUM. STEEL
- DRILL OUT CENTER HOLE DIA. AS SHOWN.
- QUANTITY OF 4 EACH.
- SPIRAL WASHERS ARE AN EXISTING ITEM.



TITLE: SPIRAL WASHER
DRAWN BY: BART BOEWHORST
DATE: 08-02-99
DRAWING NO: MATF-019
ADTECH SYSTEMS RES. BEAVERCREEK OH. 937-426-3329



- NOTES:**
- MATERIAL IS TO BE 3/4" BLANCHARD GROUND STEEL.
 - QUANTITY OF ONE EACH.
 - AFTER MACHINING PART IS TO BE BLACK OXIDED.
 - ROUND ALL CORNERS AND REMOVE ALL SHARP EDGES.

DRILL 27/64" THRU, TAP 1/2-13 THRU, TYP. 4 PLACES AS SHOWN.

MILL SLOT .781 DIA. THRU TYP. 6 PLACES AS SHOWN.

REV.	DATE	DESCRIPTION	APPROVED

DATE	REV.	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED

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REV. NO.	DATE	DESCRIPTION	APPROVED

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REV. NO.	DATE	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED

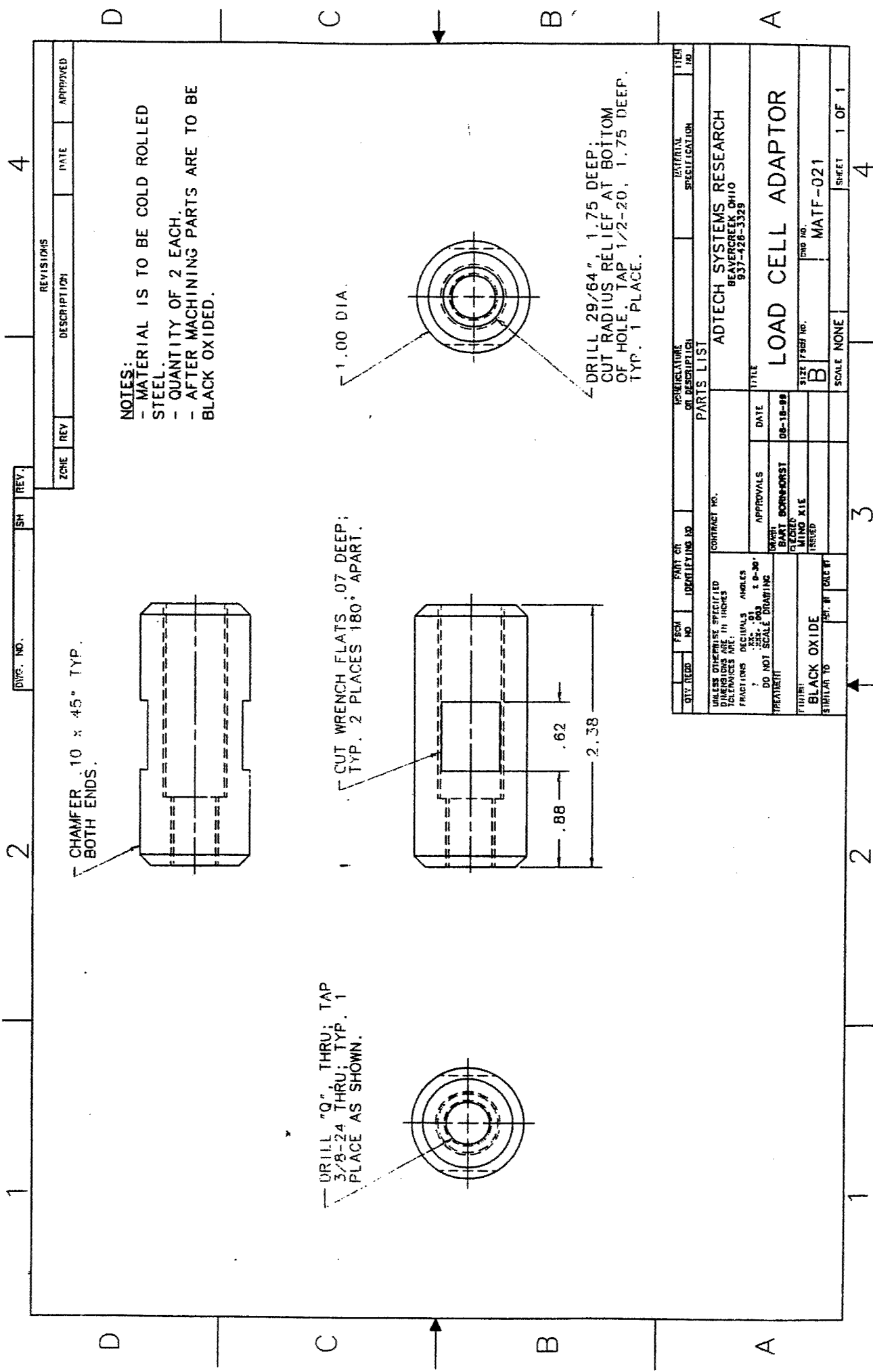
REV. NO.	DATE	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED

REV. NO.	DATE	DESCRIPTION	APPROVED



NOTES:
 - MATERIAL IS TO BE COLD ROLLED STEEL.
 - QUANTITY OF 2 EACH.
 - AFTER MACHINING PARTS ARE TO BE BLACK OXIDED.

Dwg. No.		SH		REV.		4	
REVISIONS				DATE		APPROVED	
ZONE	REV	DESCRIPTION					

QTY. REQD.	FORM NO.	PART OR IDENTIFYING NO.	INVENTOR	MATERIAL SPECIFICATION	ITEM NO.
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. FRACTIONS DECIMALS ANGLES DO NOT SCALE DRAWING.			PARTS LIST		
TREATMENT: BLACK OXIDE			ADTECH SYSTEMS RESEARCH BEAVERCREEK OHIO 937-426-3329		
CONTRACT NO.			TITLE		
APPROVALS			DATE		
DRAWN BY: BART BORNHORST			08-18-88		
CHECKED BY: MIND XIE			SIZE (FRESH NO.)		
TRIED			B		
SIMILAR TO			SCALE NONE		
1			3		
2			4		
1			1 OF 1		

NOTES:

- MATERIAL IS TO BE, 4140 TOOL STEEL.
- QUANTITY OF 2 EACH.
- REMOVE ALL SHARP EDGES.

$.500^{+0.000}_{-.001}$ DIA.

SNAP RING SLOT,
 $.04$ WIDTH X $.020$ DEEP,
 TYP. 2 PLACES AS SHOWN.

32

$.12$

$6.00^{+0.010}_{-.000}$

4

REVISIONS				
ZONE	REV	DESCRIPTION	DATE	APPROVED
D-1	A	CHANGED MATERIAL SPEC.	09-01-99	MING XIE
D-1	A	REMOVED HEAT TREAT SPEC.	09-01-99	MING XIE
C-2	A	CHANGED SURFACE FINISH	09-01-99	MING XIE

QTY REQD	FRG. NO.	PART OR IDENTIFYING NO.	QUANTIFICATION	INVENTOR	DATE
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES FRACTIONS DECIMALS ANGLES					
: :		: : : .000 1 1/2			
CC NOT SCALE DRAWING					
DRAWN BY: BORDFORST					
CHECKED BY: MING XIE					
DATE: 09-28-99					
SCALE: NONE					
SHEET 1 OF 1					

ADTECH SYSTEMS RESEARCH
 BEAVER CREEK, OHIO
 PH: 937-428-3329

GRIP PIN

SIZE: B
 PART NO: MATF-022-A

4

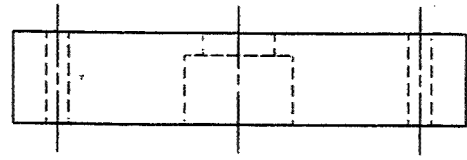
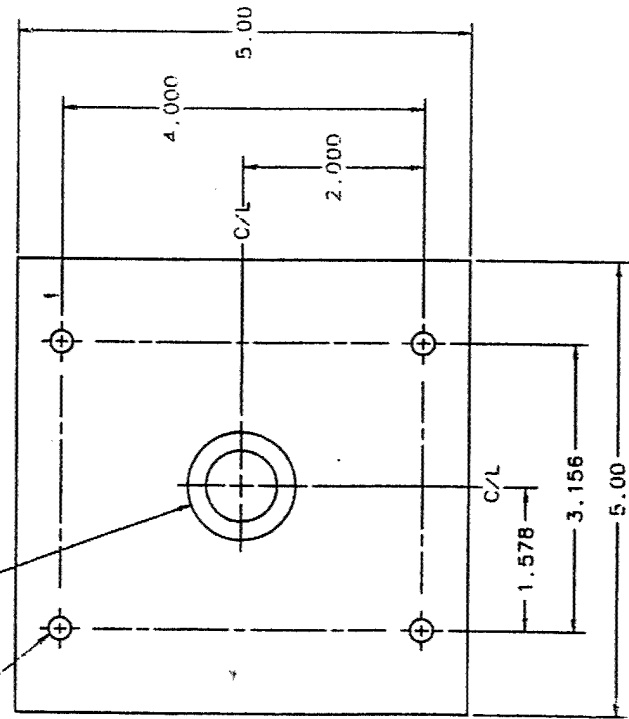
ZONE	REV	DESCRIPTION	DATE	APPROVED
------	-----	-------------	------	----------

- NOTES:
- MATERIAL IS TO BE COLD ROLLED STEEL.
 - QUANTITY OF 1 EACH.
 - AFTER MACHINING PART IS TO BE BLACK OXIDIZED.
 - REMOVE ALL SHARP CORNERS.



DRILL #7 THRU, TAP 1/4-20 THRU TYP. 4 PLACES, AS SHOWN.

DRILL .78 THRU, C-BORE DIA. 1.19, .78 DEEP. TYP. 1 PLACE AS SHOWN.



REV. NO.	DATE	DESCRIPTION	BY	CHK	REV.
PART OR IDENTIFICATION NO. PARTS LIST TITLE DATE APPROVALS PARTS ROOM/NO. 10-10-10 DRAWING FILE SCALE NONE SHEET 1 OF 1					
INSTRUCTIONS OR SPECIFICATIONS ADTECH SYSTEMS RESEARCH BEAVERTOWN OHIO PH: 837-886-3338					
SCVENGER PUMP ADAPTOR MATF-023					

1

2

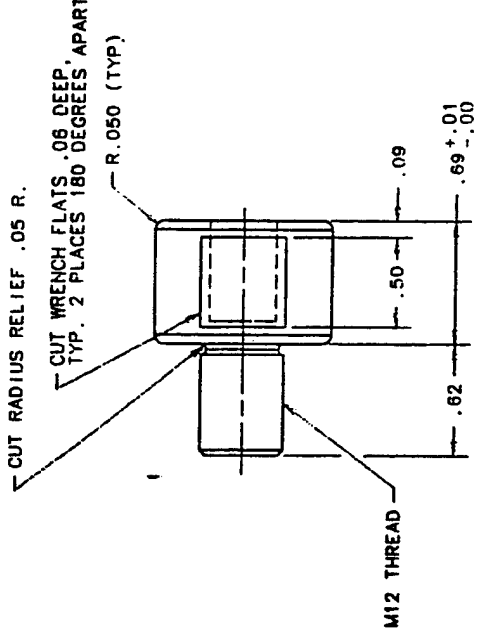
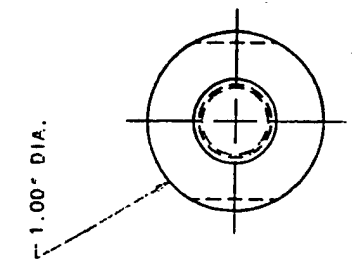
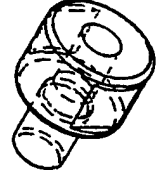
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NOTES:

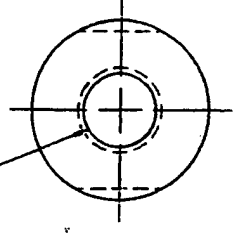
- MATERIAL IS TO BE 304 S.S...
- QUANTITY OF 2 EACH REQ'D.
- REMOVE ALL SHARP EDGES.

DWG. NO. SH REV.

REVISIONS		
ZONE	REV	DESCRIPTION
		DATE
		APPROVED



DRILL "O" .562 DEEP, FLAT BOTTOM
 DRILL; BOTTOM TAP 3/8-24. .562 DEEP;
 TYP. 1 PLACE.



D

C

B

A

CITY	FEET NO	PART OR IDENTIFYING NO	TEMPERATURE OR SPECIFICATION	ENTRITAL SPECIFICATION	TITLE
					ADTECH SYSTEMS RESEARCH BEAVERCREEK OHIO PH: 937-428-3329
PARTS LIST			TITLE		
CONTRACT NO.			DATE		
DRAWN			10-20-99		
CHECKED			BART BORNHORST		
ISSUED			MING XIE		
FINISH			SCALE NONE		
SIMILAR TO			PART NO. MATF-024		
REV. #			SCALE NONE		
REV. #			SHEET 1 OF 1		

1

2

3

4

A

B

C

D

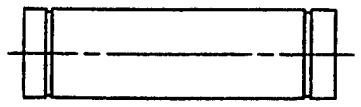
1

2

4

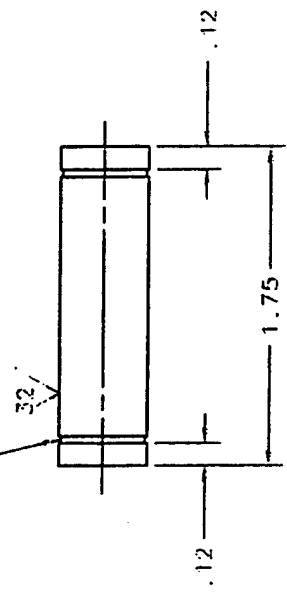
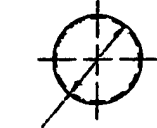
DWR. NO.		SH	REV.
REVISIONS			
ZONE	REV	DESCRIPTION	DATE
			APPROVED

NOTES:
 - MATERIAL IS TO BE 4140 TOOL STEEL.
 - QUANTITY OF 2 EACH.
 - REMOVE ALL SHARP EDGES.



SNAP RING SLOT .04 WIDTH X .020 DEEP. TYP. 2 PLACES AS SHOWN.

.500 ± .001



D

C

B

A

CITY	STATE	PART OR IDENTIFICATION NO.	REGULATORY OR DESCRIPTION	QUANTITY	UNIT NO.
PARTS LIST					
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES		CONTRACT NO.			
FRACTIONS	DECIMALS	ANGLE	DATE	TITLE	
3/16	.003	30°-30°	10-21-68	ADTECH SYSTEMS RESEARCH	
DO NOT SCALE DRAWING			APPROVALS	BEAVERCREEK OHIO	
			BY: BART BORNHORNST	PH: 937-428-3329	
			BY: MING XIE	CLEVIS PIN	
			BY: [blank]	SIZE (FSC) NO. MATF--026	
			BY: [blank]	SCALE NONE	
			BY: [blank]	SHEET 1 OF 1	

4

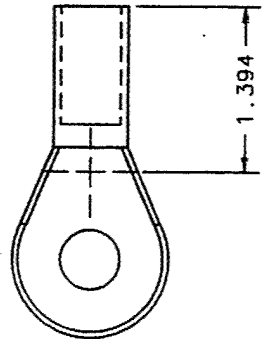
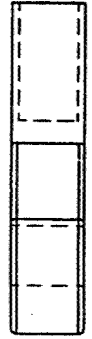
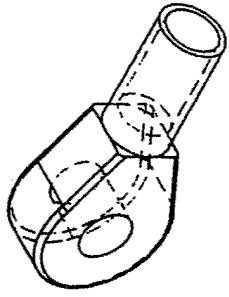
2

1

1 2 4

D C B A

FIG. NO.	SH	REV.	ZONE	REV	DESCRIPTION	DATE	APPROVED



NOTE:
(EXISTING DIM. = 2.125")

- NOTE:**
- PART IS EXISTING AND WILL BE FURNISHED TO THE VENDOR.
 - MATERIAL IS 303 S.S..
 - CUT THE LENGTH OF THE PART AS SHOWN.
 - QUANTITY OF 2 EACH.
 - ROUND ALL SHARP EDGES FROM CUTTING.

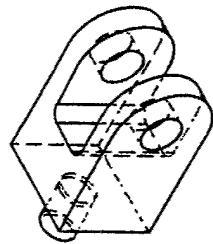
QTY REQD	FIG. NO.	PART OR IDENTIFYING NO.	QUANTITY	REVISIONS	DATE	APPROVED
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ARE:		PARTS LIST		ADTECH SYSTEMS RESEARCH BEAVERCREEK OHIO PH: 937-428-3329		
FRACTIONS	DECIMALS	ANGLES	DATE	TITLE		
± .0005	± .001	± 0°-30'	10-22-99	ROD END BEARING - FEMALE		
DO NOT SCALE DRAWING		APPROVALS	DATE	SIZE FIG. NO.		
		DRG. BORNHORST	10-22-99	B		
		CHKD. MING XIE		MATERIAL SPECIFICATION		
		ISSUED		NONE		
FINISH		PART. BY		SHEET 1 OF 1		
SUBMIT TO		DATE		4		

1 2 3 4

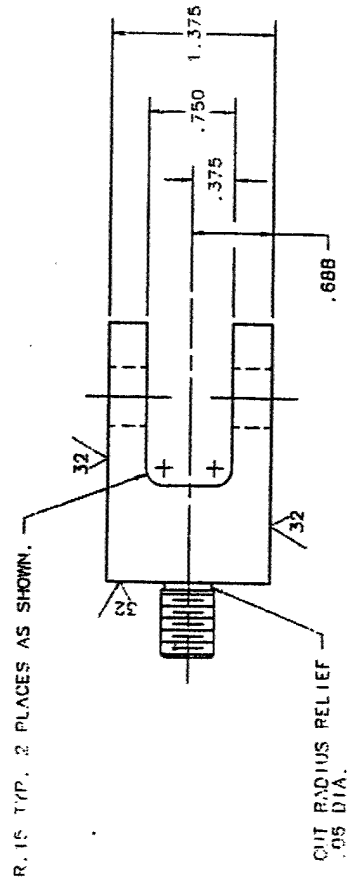
1 2 4

REV. NO.		DATE		APPROVED	
1					
2					
4					

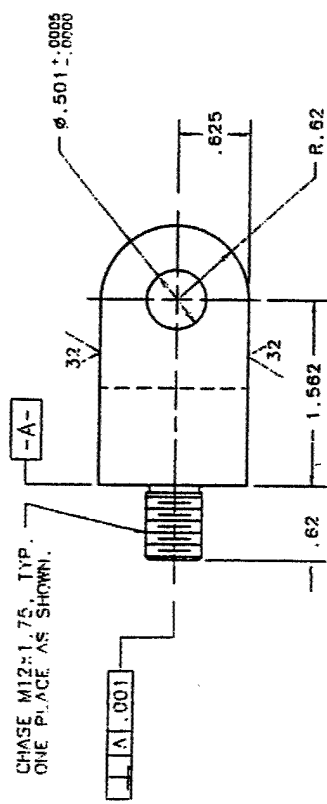
NOTES:
 - MATERIAL IS TO BE 304 S.S.
 - QUANTITY OF 2 EACH.
 - REMOVE ALL SHARP EDGES.



ISO VIEW

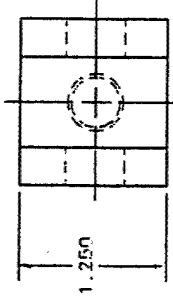


TOP VIEW



FRONT VIEW

RIGHT SIDE VIEW



QTY REQD	FSDI NO	PART OR IDENTIFYING NO.	ITERATION	TITLE
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ARE:		PARTS LIST		
1	FRACTIONS	DECIMALS	APPROX	DATE
2	± .005	± .0005	± 0.30°	11-20-88
TREATMENT		DRAWN BY		
		BART BORRHOEST		
FINISH		CHECKED BY		
		MUNG XIE		
SIMILAR TO		DATE		
		11-20-88		
SCALE		NONE		
SHEET		1 OF 1		

ADTECH SYSTEMS RESEARCH
 BEAVERCREEK OHIO
 P.O. 937-426-3329

GRIP CLEVIS

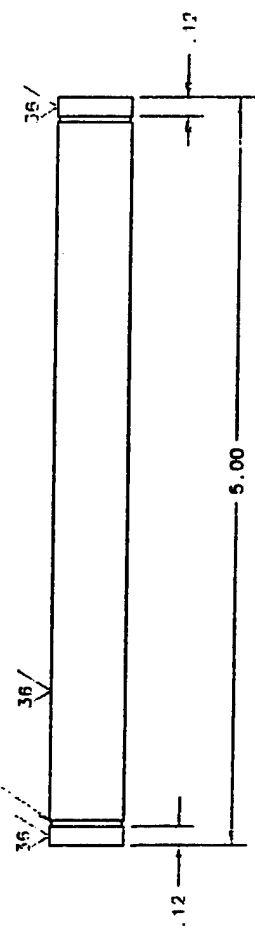
MATF-029

1 2 3 4

REVISIONS		DATE	APPROVED
ZONE	REV	DESCRIPTION	

NOTES:
 - PART IS EXISTING AND IS TO BE SHORTENED BY 1.00". (SEE ACCOMPANYING REF. DWG.).
 - MATERIAL IS BE 4140 TOOL STEEL.
 - QUANTITY OF 2 EACH.

SNAP RING SLOT
 .04 WIDTH X .02 DEEP
 TYP. 2 PLACES AS SHOWN.

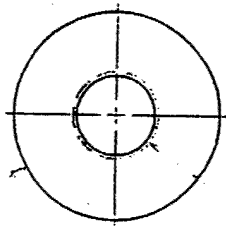


QTY REQ.	FRCH ID	PART OR IDENTIFICATION NO	ITERATION SPECIFICATION	ITERATION NO
		CONTRACT 12.		
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES FRACTIONS DECIMALS ANGLES		PARTS LIST		
2	36°	DATE	TITLE	
DO NOT SCALE DRAWING	APPROVAL	11-22-99	ADTECH SYSTEMS RESEARCH	
TREATMENT	BY		BEAVERCREEK OHIO	
	BY		PH: 937-428-3329	
	BY		GRIP PIN	
	BY		SIZE (SEE NO.)	
	BY		SCALE NONE	
	BY		SHEET 1 OF 1	

NOTES:

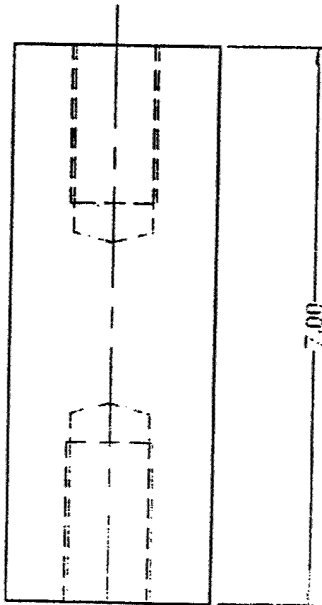
- MATERIAL IS TO BE 303 OR 304 S.S.
- QUANTITY OF 1 (ONE) EACH.
- REMOVE ALL SHARP EDGES.

0.63 DIA.



DRILL 15/16" 2.5" DEEP,
CHASE 1/14" THREAD, 2.1" DEEP.
CENTERS ARE TO BE PERPENDICULAR AND
CONCENTRIC TO EACH END OF THE PART ±.001
TYP. 2 PLACES AS SHOWN.

11



NOTE:
BOTH ENDS OF THE PART ARE TO BE
FLAT AND PARALLEL TO WITHIN ±.001.

ADTECH SYSTEMS RESEARCH INC.
1342 NORTH FAIRFIELD RD.
DAYTON, OHIO 45432-2698
513-426-3329

DRAWING NAME:
RAM EXTENDER

DRAWING #: MATF-031

DATE: 01-21-00

DRAWN BY:
BART B.

APPROVAL:
MING XIE

SURFACE: 32

MATERIAL:
SEE NOTE

TOLERANCE: " OR AS NOTED
+-.005"

APPENDIX B

Script for the Multiaxial High Frequency Test System Video

APPENDIX B

Script for the Multi-axial High Frequency Test System Video

Scene 1 – AdTech logo.

Scene 2 – Title : “A Multi-axial High Frequency Test System.”

Scene 3 – Graphics, excerpt from Phase I video (a turbine engine disk/blade assembly)

Narrator: In a typical turbine engine, rotating blades are subjected to both the centrifugal force and the periodic transverse excitation. This is a multi-axial loading condition also present during operation of many other high performance mechanical systems.

Scene 4 – Excerpt from Phase I video (prototype biaxial machine)

Narrator: In 1998, AdTech Systems Research Incorporated of Dayton, Ohio successfully developed a prototype biaxial loading test system. This four-post test frame is capable of applying mechanical loading on perpendicular axes through two servo hydraulic actuators. Its purpose was to demonstrate the feasibility of performing multi-axial loading simulation tests on turbine engine components.

Scene 5 – Computer and operator, zoom in to computer screen (finite element model)

Narrator: After the “proof of concept” demonstration of the prototype test frame, AdTech Systems Research continued to develop the multi-axial loading test methodology for advanced materials and structures, including high cycle fatigue testing of turbine engine blades.

Scene 6 – Graphics (bending and torsion modes of a deformed blade)

Narrator: Since vibration of turbine engine blades under service conditions includes bending and torsion modes, two actuators have been designed into the test system to simulate the combined bending and torsional vibration loading.

Scene 7 – Animation (two horizontal actuators are moving)

Narrator: When they move in-phase, the blade is subjected to torsion loading. When they move out-of-phase, the blade is subjected to bending loading.

Scene 8 – Animation (test frame, three actuators are being mounted)

Narrator: The new test frame design has three servo hydraulic actuators. Two movable small actuators, capable of high frequency loading, are mounted on the sidewall of the frame, and the main actuator is mounted on the bottom of the frame.

Scene 9 – Overview of the machine

Narrator: In September 1999, AdTech Systems Research completed design and manufacture, and installed the multi-axial test frame in the Air Force Research Laboratory's Turbine Engine Fatigue Facility at Wright-Patterson Air Force Base.

Scene 10 – Zoom in to the vibrating blade and gripping

Narrator: Multi-axial high cycle fatigue testing is being performed on a General Electric F110 turbine engine second stage fan blade. Two movable side actuators can apply a transverse fatigue loading to the blade. At the same time, the main actuator can apply a radial loading to the airfoil to simulate the centrifugal force.

Scene 11 – Control system, computer and the operator

Narrator: A digital three-station control system controls all three actuators, monitoring the load and displacement conditions while performing high-speed data acquisition. Two side actuators on the test frame, with 1000-pound capacity each, can apply high frequency loading up to 400 Hz, while the main actuator has an 11,000-pound loading capacity.

Scene 12 – Overview of the machine, then zoom in to one side actuator

Narrator: The test system can be used to test many other engineering structures under multi-axial high frequency loading. Two side actuators are movable and can apply loads to different locations of the test specimen.

Scene 13 – Computer screen shot (control software)

Narrator: The user-friendly control software makes it easy to setup and operate the test system. Three servo hydraulic actuators can be controlled individually or as a group.

Scene 14 – Computer screen, then overview of the machine

Narrator: This test system can be used in the design, development and analysis of operational gas turbine engine components, as well as evaluation of other advanced materials and structures. All currently available commercial test frames have only uniaxial loading capability. This multi-axial test system offers many new capabilities and will have important applications in both the research community and industry.

Scene 15 – Credits and acknowledgment (no narration)

Principal Engineer and Program Manager – Ming Xie, Ph.D.

Team Members – Bart Bornhorst, Mohan Balan and Norman Frey (consultant)

The development of the multiaxial high frequency test system was sponsored by the Turbine Engine Division of the Air Force Research Laboratory's Propulsion Directorate at the Wright-Patterson Air Force Base, Dayton, Ohio.

Air Force Technical Monitors and Support – Gary Terborg, Bruce Tavner and
Charles Cross, Ph.D.

Scene 16 – AdTech logo