

IMPROVEMENT OF AN ARTIFICIAL
SAFETY KNEE JOINT

BY

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A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE DEGREE OF

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in the Department
of
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A thesis submitted to the Faculty of Graduate Studies of
the University of Manitoba in partial fulfillment of the requirements
of the degree of

MASTER OF SCIENCE

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ABSTRACT

"Locking" or "safety" prosthetic knee joints have been used for many years to improve the stability of above-knee amputees. In general, these knee joints attempt to prevent unexpected buckling of the prosthesis. The purpose of this thesis is to improve the degree of safety provided by a successful, commercial safety knee, the Otto Bock Orthopaedische K.G. model 3R15.

Firstly, a parameter called the critical offset distance, δ_{CR} is introduced to provide a quantitative means of measuring the degree of safety of a safety-style knee joint. An equation is developed to calculate this distance for the 3R15 knee. This equation is used to identify geometric parameters which could improve, in principle, the critical offset distance for the 3R15 knee. The possibility of modifying each parameter is considered and an initial redesign is proposed. A finite element technique is employed to estimate the critical offset distance and the corresponding maximum stresses for both the existing 3R15, and the initial redesign. A series of refinements of the initial redesign are then introduced until it is found that further refinements had a negligible effect on improving δ_{CR} . Maximum stresses for the redesigned knee are not increased with respect to the original knee. The refined design is then proposed as an "optimized" version of the

original artificial knee. Finally, the effects of using titanium instead of steel for the locking mechanism of both the original and "optimized" knee joints are investigated.

It is found that the redesigned steel knee increases the critical offset by approximately 2.0 to 3.0 mm, depending on the body weight of the amputee and the radial clearance of the locking mechanism. The increase of the critical offset is found to be greatest for lighter body weights and large radial clearances of the locking mechanism. The use of titanium in place of steel for the locking mechanism of either the original or redesigned knee is found to greatly increase the critical offset. This increase is attributable to the lower elastic modulus of titanium. The greatest increase of the critical offset is found with the titanium version of the redesigned knee, followed respectively by the titanium version of the original knee and the steel version of the redesigned knee.

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CHAPTER 1

1.1 INTRODUCTION

Walking is a simple, natural process that requires little conscious effort for a normal person. However, walking demands not only physical effort but concentration and conscious control of movements for the lower limb amputee with a prosthesis. To prevent a lower limb amputee from tiring and, thereby, impairing control of movement and the ability to walk, it is important that the amputee's prosthesis be designed to minimize the concentration and effort required to walk [1]. This thesis will investigate the redesign of an existing, commercially available artificial knee joint to decrease the effort and control required to walk on the prosthesis of which the joint is a part.

To help understand the operation of a lower limb prosthesis, consider a person whose leg has been amputated above the knee. This person, who is generally called an above-knee (A.K.) amputee, is missing two major joints, the ankle and the knee. To enable the A.K. amputee to ambulate, the present practice is to fabricate a prosthesis like that shown in Figure 1.1. The prosthesis consists of four basic parts--the foot, lower shank, knee joint, and custom-fitted stump socket. This typical combination of components gives an amputee the ability to walk.

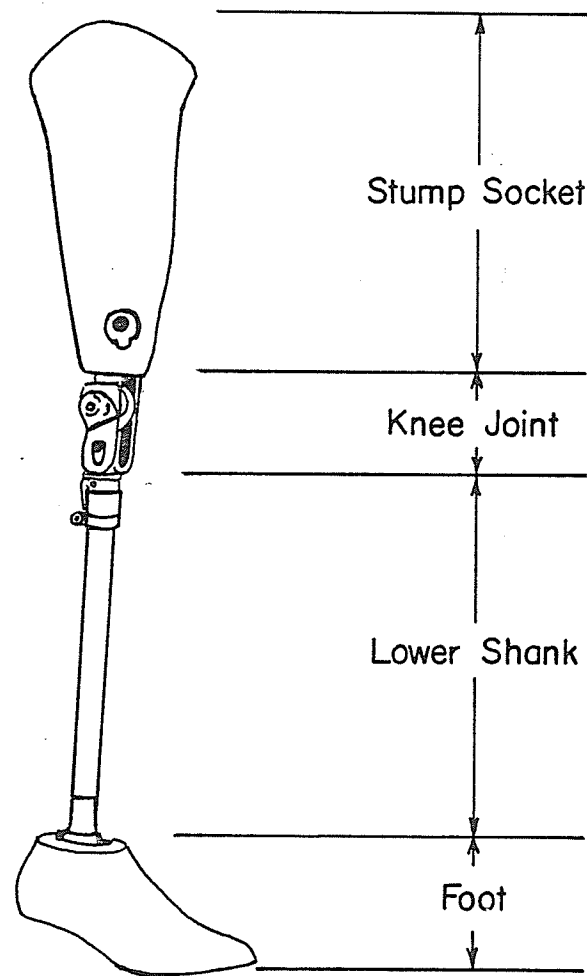


Figure 1.1 : Modular Endoskeletal A.K. Prosthesis

Regardless of the specific design, all A.K. prostheses must meet five major goals:

- (i) Amputee comfort: Obviously, the comfort of the amputee is of great importance. Proper socket construction and the correct alignment of the centres of rotation (or pivot points for simple hinge-style joints) of the prosthesis are the major determinants [2] in ensuring maximum comfort. Each socket is custom made so that the fit and comfort depend largely upon the skill of the prosthetist.
- (ii) Energy efficient use: Although a prosthesis allows an A.K. amputee to walk, the gait pattern is not normal [3]. The pelvis must be raised higher on the amputated side (compared to the sound leg) during the swing phase in order for the artificial foot to clear the ground. This difference results in an abnormal gait and a greater expenditure of energy. Proper alignment of the knee joint and proper component design can help to significantly reduce this extra expenditure of energy.
- (iii) Stability: The prosthesis must not buckle whilst it bears the weight of the body. An unexpected fall due to buckling (instability) of the prosthesis could result in serious injury to the amputee. However, there is a compromise between the conflicting needs of greater stability and less expenditure of energy.

Generally, a more stable alignment of the leg increases the pelvic tilt required for the swing-through of the leg. This, in turn, increases the expenditure of energy [1]. Excessive stability is avoided, therefore, whenever possible. The requirements for stability will be examined later in Section 1.2.2.

- (iv) Cosmetics: The finished prosthesis should have a natural appearance to enhance its social acceptability. This need restricts the size and placement of the components used to make the prosthesis. Foam covers and flesh-coloured stockings are often used to disguise the mechanical structure of a prosthesis. Quiet and clean operation are also required.
- (v) Durability: A relatively active amputee can subject a prosthesis to over a million steps per year. Consequently, the use of durable components is essential. Failure of any one component could result in serious injury, particularly if the amputee was to fall unexpectedly. Generally, simple mechanisms are more mechanically reliable than complex ones which have a greater number of parts and, therefore, have a correspondingly greater potential for failure. Simple components are often used even when more

sophisticated (but also more complex) designs are available because of the decreased risk of failure.

Attaining the preceding five goals presents a challenge to both the engineer and the prosthetist. While the prosthetist works to assemble each prosthesis to best suit the needs of the individual, the engineer must design durable components which are readily adjustable to allow function over a wide range of requirements. Well designed components can improve the quality of life for an amputee and it is the job of the engineer to provide good designs which are within the economic reach of the average amputee.

1.2 PROSTHETICS AND BIOMECHANICS BACKGROUND

1.2.1 Basic Approaches to A.K. Prosthetics

The prosthesis shown in Figure 1.1 is called a modular endoskeletal prosthesis and it is currently the most common type of A.K. prosthesis. Fabrication simply involves the linking of appropriate standard commercially available components, such as feet and knee joints, with structural tubing which is cut to a suitable length. The upper end of the knee joint is laminated into a stump socket that is custom-made to suit the amputee. The advantages of the modular approach are quick assembly, ease of adjustment, relatively low cost, and component interchangeability. The

knee joint which will be redesigned in this thesis is part of an endoskeletal prosthetic system.

An exoskeletal prosthesis, such as that shown in Figure 1.2, is an alternative to the endoskeletal prosthesis. Structural support is provided then by a custom laminated composite shell or wooden structure rather than the central tube common to endoskeletal prostheses. Although a stronger and lighter prosthesis can be fabricated by using this technique, alignment changes are difficult and construction is much more labour-intensive. Cost is also usually higher (primarily due to the labour intensive construction) compared to that of the endoskeletal prosthesis. These disadvantages have resulted in the endoskeletal prosthesis becoming more popular than the exoskeletal prosthesis.

1.2.2 Knee Joint Stability and Instability

"Knee instability is buckling or flexing of the prosthetic knee in the stance phase of walking when the amputee depends upon the prosthesis to support his weight [2, p.83]." The concept of knee joint stability is understood most easily by considering Figure 1.3 in which an A.K. amputee is shown standing with the body weight on the prosthetic leg. Instability will occur if the centre of gravity (C.G.) of the whole body is posterior (behind) to the centre of rotation of the knee. This case is

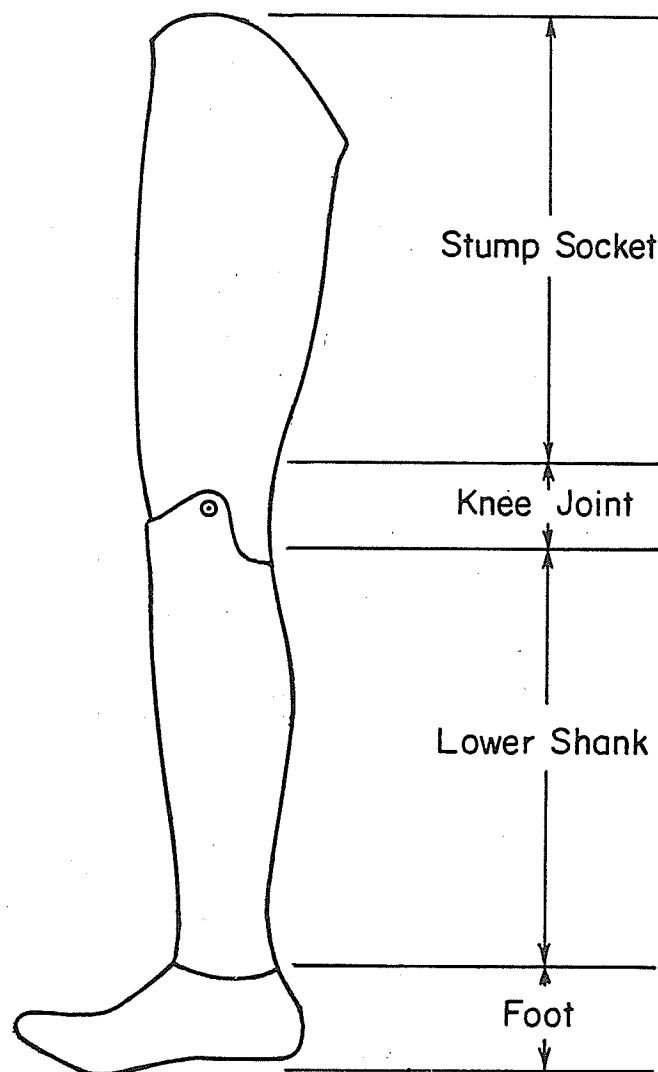


Figure 1.2 : Exoskeletal Prosthesis

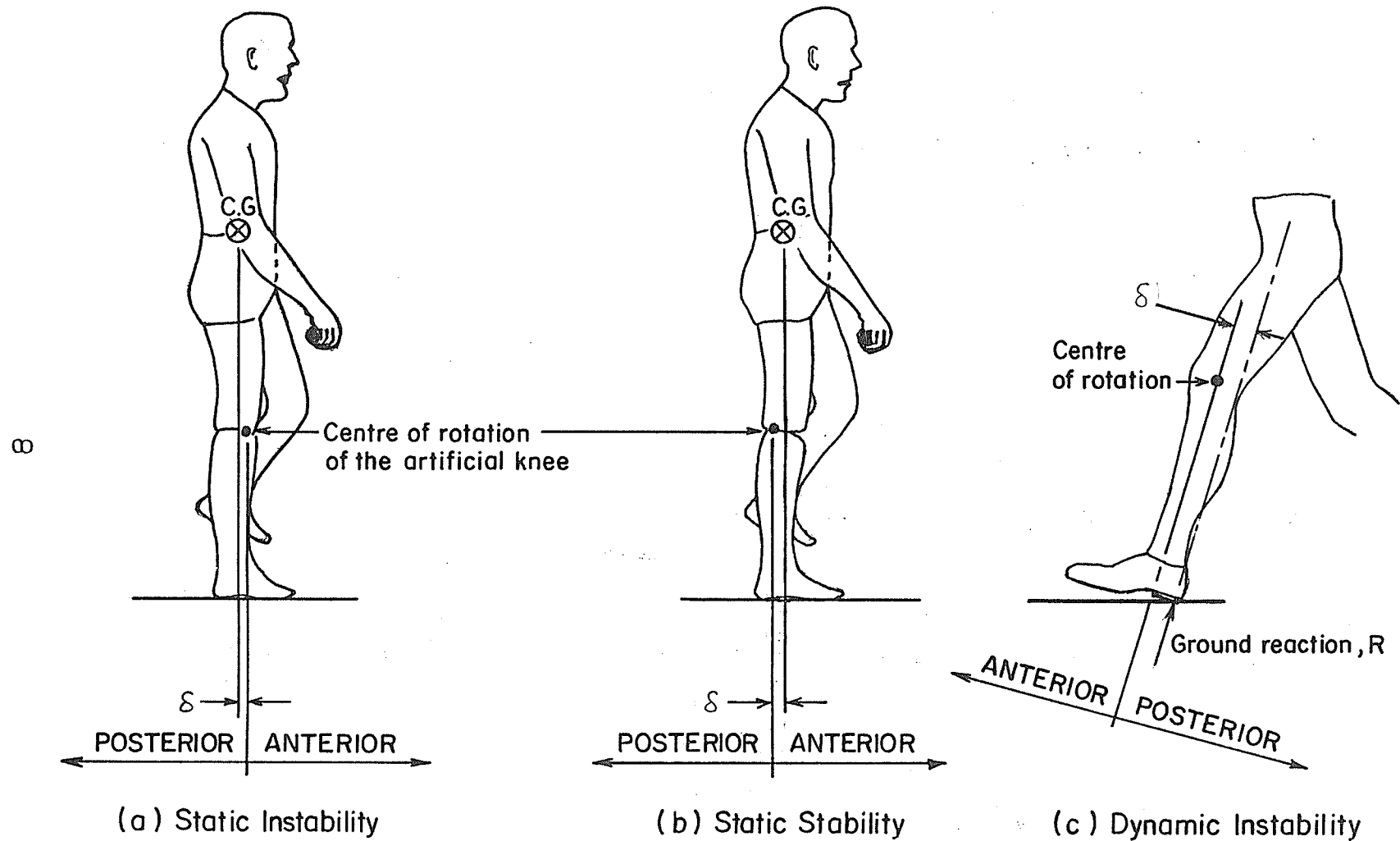


Figure 1.3 : Stability and Instability

illustrated in Figure 1.3(a) where a moment tending to buckle the knee about its pivot point is created when the variable " Δ " is posterior to the pivot point. The amputee will fall when this condition occurs. When " Δ " is as shown in Figure 1.3(b), a mechanical stop, commonly called an extension stop, provides a normal stance by preventing the knee from extending beyond the normal standing position.

The basic concepts of knee joint stability or instability can be extended to walking. However, the analysis then becomes much more complex because the line of action of the ground reaction constantly changes throughout the gait cycle. As in the static case, however, the knee is stable only when the line of action of the ground reaction is anterior to a knee's pivot point. A line of action of the ground reaction which, as illustrated in Figure 1.3(c), is posterior to the centre of rotation of the knee causes instability. A detailed analysis of the mechanics of dynamic gait can be found in references [1, 4, 5, and 6].

1.2.3 Ensuring Knee Joint Stability

An amputee can resist instability of the artificial knee by dynamically extending the stump about the hip. This action holds the knee in extension while the body is accelerated in the forward direction. As shown in Figure 1.4, the hamstring muscles then contract to force the stump

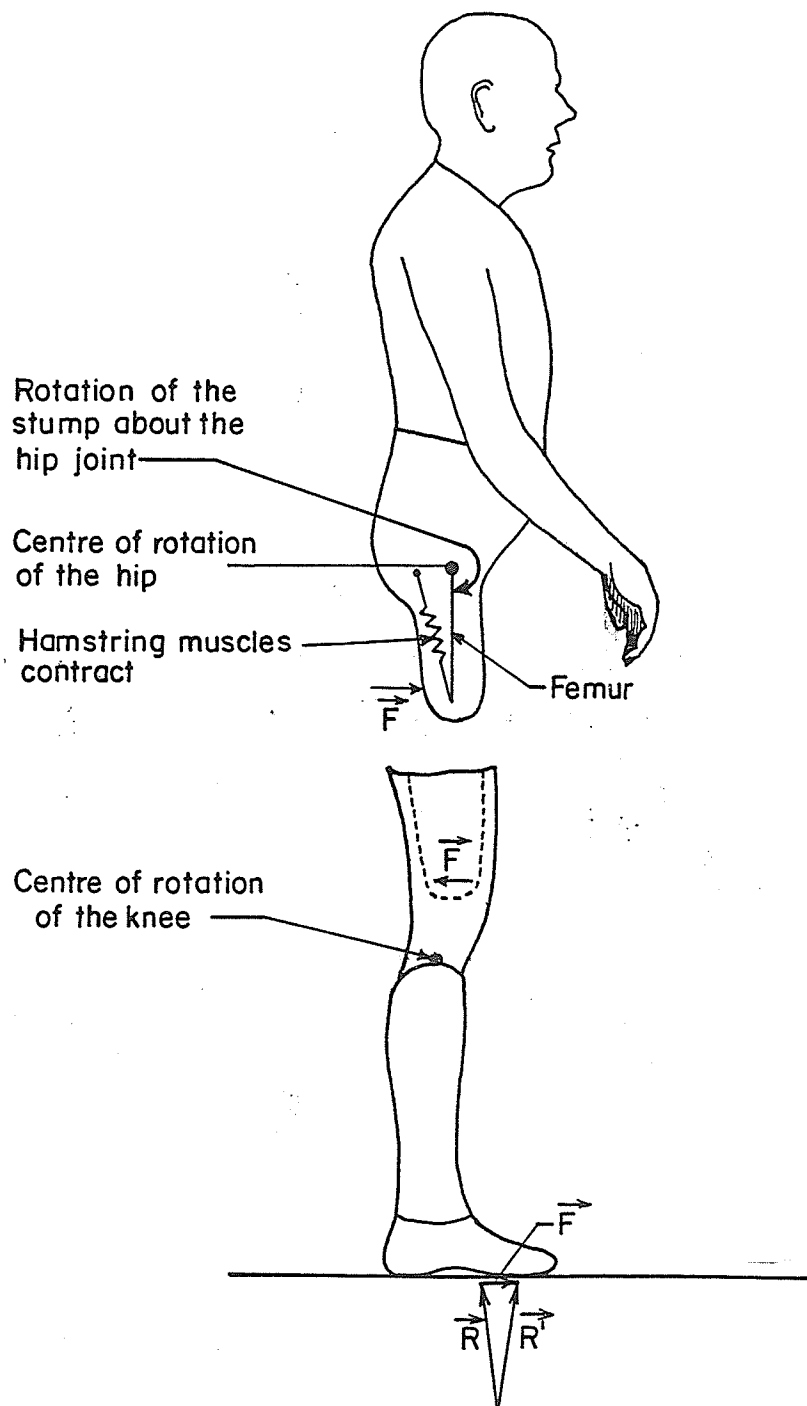


Figure 1.4 : Voluntary Control of Stability

to rotate about the hip joint. Consequently the bottom portion of the stump is forced against the back of the socket of the prosthesis. The resulting contact force between the stump and socket, $\vec{F}$, acts as shown in relation to the stump and socket. This contact force $\vec{F}$ accelerates the body in the forward direction. When $\vec{F}$ is added vectorially to $\vec{R}$, the ground reaction on the prosthesis before extension of the stump, the original ground reaction becomes modified to $\vec{R}'$. The line of action of $\vec{R}'$ is anterior or in front of the line of action of $\vec{R}$. As discussed previously, a larger anterior distance from the centre of rotation of the knee to the line of action of the ground reaction increases stability. Hence, greater stability has resulted from the dynamic extension of the stump. This method of voluntary control is most effective with strong amputees because they are able to exert larger forces than more sedentary amputees.

Although all A.K. amputees fitted with a prosthesis have enough strength to exercise some degree of voluntary control, many elderly or less aggressive amputees are unable to extend their stump with sufficient force to prevent instability at all times. This is especially true if the amputee becomes tired or encounters rough terrain. The lack of voluntary control can be partially counteracted by increasing the alignment stability. Alignment stability involves statically adjusting the centre of rotation of the

knee so that it is posterior to the line of action of the ground reaction. As the posterior distance from the knee's centre of rotation to the line of the ground reaction increases, the knee becomes more stable because a moment is created which tends to move the knee joint into a fully extended position.

"Excessive knee stability is a condition in which the knee of the prosthesis is so resistant to being flexed that the amputee has difficulty in 'breaking' it at toe-off.* This results in high energy consumption and a very unnatural appearing gait.[2]" Excessive stability results when a knee's rotational centre is aligned too far posterior to the line of action of the ground reaction. The objective in aligning an artificial knee is not only to provide alignment stability to prevent the amputee from falling but also to ensure that minimal energy is consumed in a close to natural gait.

The stability of a knee can be enhanced mechanically by employing specially designed knee joints. There are three basic types of such knees. The simplest type is the manual locking knee which, as its name implies, has a manually operated mechanism to lock the knee in the extended position. The prosthesis basically functions as a "peg leg" in the locked position. The second style of knee joint is

*Toe-off is the stage in the gait cycle at which the toe leaves the floor.

the polycentric knee which is designed to enhance dynamic stability. The socket and shank sections of this type of knee are joined by a four bar linkage as illustrated in Figure 1.5. This linkage creates a moving centre of rotation similar to that of a human knee. The locus of the knee's centre of rotation is controlled by the geometry of the linkage. Polycentric knees can provide a very natural gait, relatively low energy use, and greater stability during weight bearing than a simple, single axis knee joint. Their disadvantages are increased size, greater mechanical complexity and high decelerations as the knee nears the extension stop.

The third type of artificial knee is the "locking" or "safety" one which is designed to enhance knee stability. This knee functions as a normal, single axis knee during the swing phase of gait. However, the knee becomes frictionally "locked" in extension during weight bearing. The locking mechanism is typically a split ring which tightens around the knee axis when the weight is placed on the knee. The locking feature allows less alignment stability but still provides greater overall stability than that required by a typical single axis knee. As discussed previously, a lower alignment stability gives a more natural gait and a smaller expenditure of energy.

"Safety" knees generally provide the greatest stability of presently available commercial knee joints. Their use is

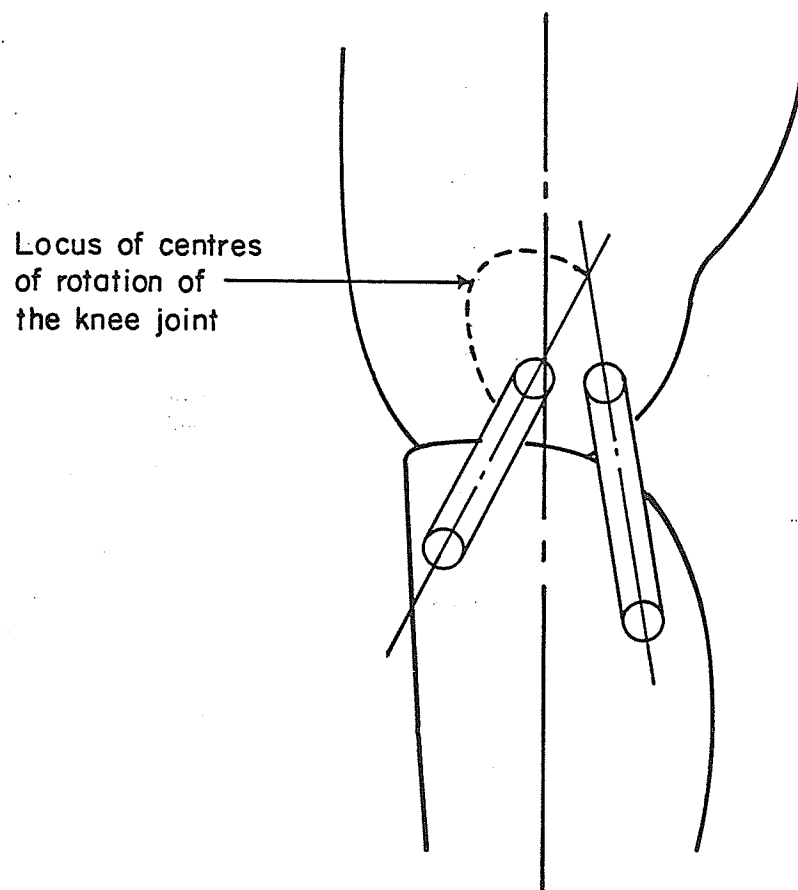


Figure 1.5*: Polycentric Knee Mechanism

* Reproduced from Reference [1], p.287

restricted generally to geriatric or weak-muscled amputees who are unable to exert the voluntary control required by a normal, single axis knee. The development of safety knee joints has allowed the use of a prosthesis by amputees who may otherwise be confined to a wheelchair because they lack the muscle control and stamina required to operate a conventional prosthesis.

The intent of this thesis is to improve the design of an existing, highly successful commercial safety knee, the Otto Bock Orthopedic Ltd. model 3R15, to provide greater stability and ease of use.

CHAPTER 2

2.1 The Otto Bock Model 3R15 Safety Knee

The model "3R15" single axis modular safety knee is made by Otto Bock Orthopedic Industry of West Germany. It was introduced in 1974 and has since become one of the most widely used safety knees in the world. Its compact and simple design, coupled with a relatively low cost, are the factors contributing to its popularity. The 3R15 knee is one of the component parts of Otto Bock's modular endoskeletal prosthetic system.

2.2 FUNCTION OF THE 3R15

The principle of operation of the 3R15 is illustrated in Figure 2.1(a). (The terminology used in this figure and later will correspond to that employed by Otto Bock K.G.) The knee has three major components: (i) a central "swing block" with a split bore centered at point 'A' and a closed bore at point 'B'; (ii) an "upper fork" which attaches to the swing block with a pin joint located at 'B'; and (iii) a "lower fork" which attaches to the swing block through a pin joint at 'A'. The pin joint at 'A', which is the centre of rotation of the knee, is normally a free swinging joint with a mechanical extension limit provided by the contact between

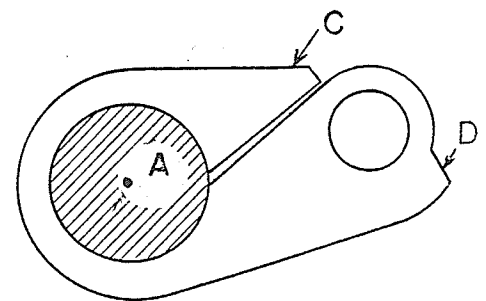
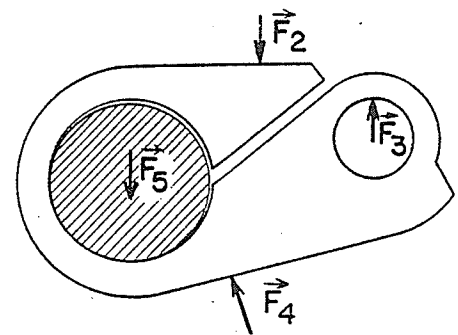
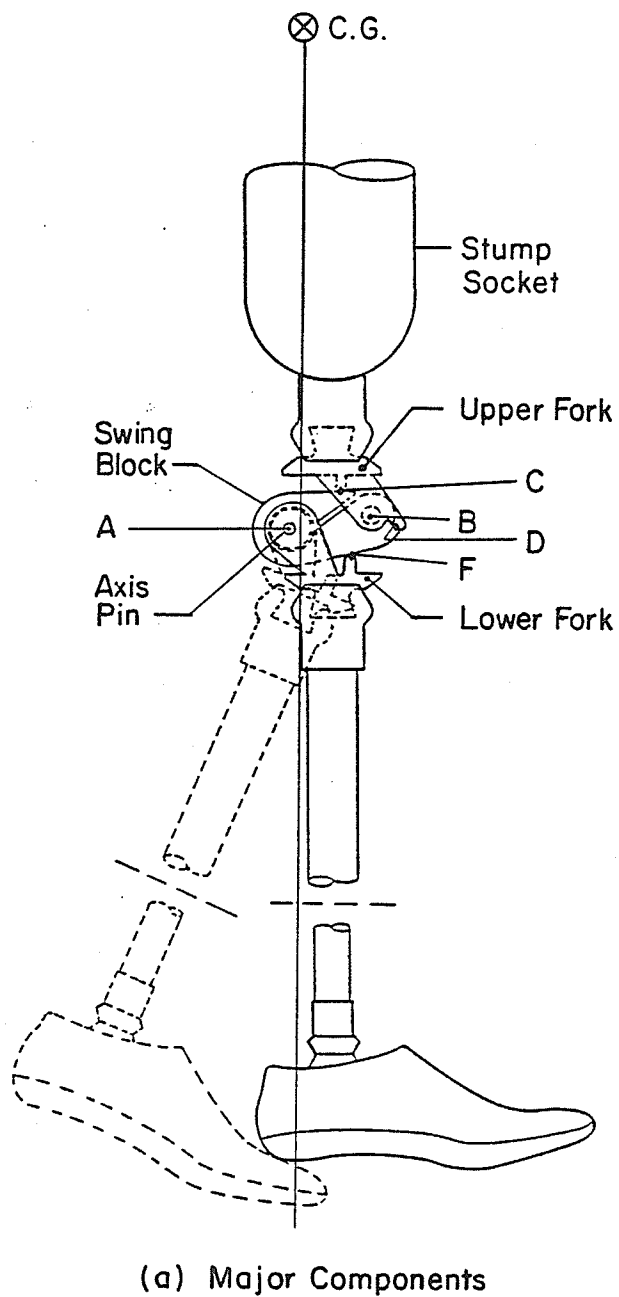


Figure 2.1 : Principle of Operation of the Model 3R15 Knee

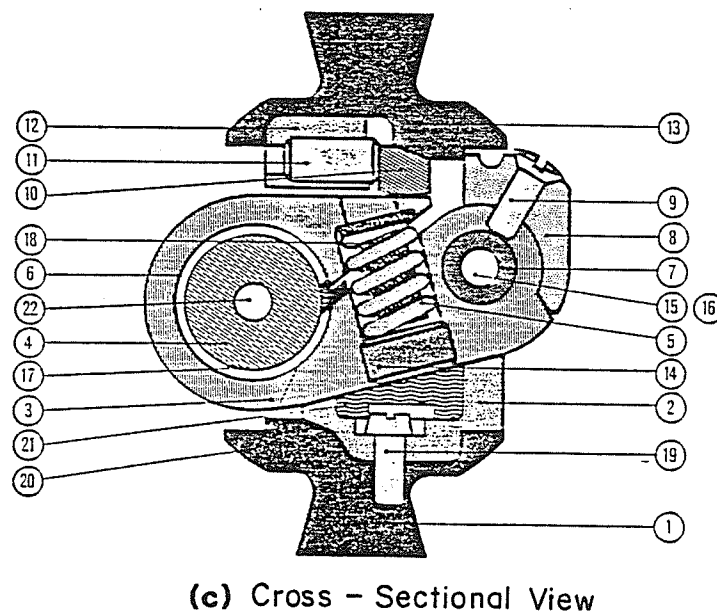
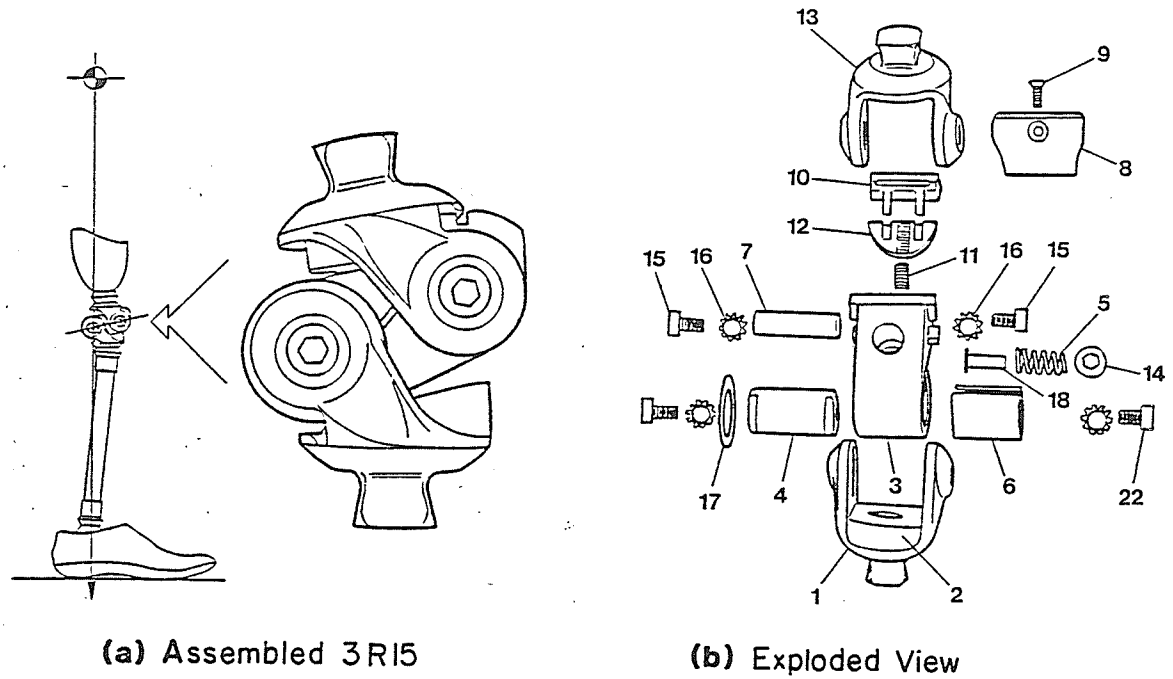
the lower fork and the swing block at point 'F'. This contact occurs during stable weight bearing.

The pin joint at 'B' has a very restricted range of motion. The upper fork is constructed so that contact is made normally between the fork and the swing block at both point 'C' and 'D' in Figure 2.1(a). In practice, this simultaneous contact is achieved by placing an adjustable wedge at point 'C'. The wedge would completely restrict the joint from any rotational freedom if the swing block were perfectly rigid. In practice, however, the swing block flexes slightly. The effects of this flexing will be discussed next.

Consider a stationary amputee whose weight is borne on the prosthesis. The reaction forces on the swing block and the pin at 'A' act as shown in Figure 2.1(b). These forces cause the split bore of the swing block to flex and tighten around the pin at 'A', as illustrated in Figure 2.1(c). The resulting frictional braking effect, called stance phase control, restricts the rotation of the lower fork. The upper fork rotates slightly as the upper split portion of the swing block deflects which, in turn, causes a small amount of sliding at point 'C' and loss of contact at point 'D'. The braking effect increases in proportion to the weight applied on the prosthesis. When the weight is removed, the bore of the swing block returns to its original diameter and the knee is able to swing freely.

Figures 2.2(a) through (c) serve to illustrate the function of the 3R15 knee in greater detail. Figure 2.2(a) portrays the actual 3R15 and its use in a typical prosthesis. Figure 2.2(b) is an exploded view of the various parts of the 3R15 while Figure 2.2(c) gives a cross-sectional view of an assembled 3R15.

The knee's function is controlled by two adjustments which tailor its operation to the needs of the amputee. First, the radial clearance between the axis pin and swing block and, thus, the amount of friction for swing phase control can be adjusted by using the mechanism consisting of items 10, 11, and 12 in Figure 2.2(c). By tightening the set screw (item 11), the pressure block (item 10) is wedged between the upper fork (item 13) and the swing block. This process causes the split bore of the swing block to tighten around the axis pin and bushing which, in turn, increases the friction on the axis pin. Adjustment of the set screw is used to adjust the swing phase characteristics of the prosthesis. Tightening the set screw increases friction which decreases the rate of the free swing of the prosthesis. The second adjustment relates to the stance phase control which can be modified by utilizing the brake screw or item 14 in Figure 2.2(c). Tightening the brake screw compresses a spring (item 5) which pushes open the split bore of the swing block. The spring then opposes the



Parts

- (1) Lower Joint Fork 4G97
- (2) Knee Stop 4G37
- (3) Swing Block 4G20
- (4) Axis 4G28
- (5) Spring 4L7 = 2
- (6) Brake Bushing 4B74
- (7) Swing Axis 4A44
- (8) Plastic Mount 21Y72
- (9) Screw 501S27 = M4 X 10
- (10) Pressure Block 4G26
- (11) Friction Screw 506G3 = M6 X 12
- (12) Plastic Block 4G38
- (13) Upper Joint Fork 4G30
- (14) Brake Screw 4Z32
- (15) Axis Screw 4Z33
- (16) Tooth Lock Washer 507S13 = 6,4
- (17) Cup Spring 513T4 = 27,7 X 17,5 X 0,4
(Brass Washer 4Z48)
- (18) Spring Nucleus 4R36
- (19) Self Tapping Screw 501B4 = M4 X 8
- (20) Safety Device for Bushing 4G31
- (21) Bumper 4Z46
- (22) Axis Screw 4Z47

Figure 2.2 : The Model 3R15 Knee

clamping of the axis pin resulting from the amputee placing weight on the prosthesis.

The brake screw is left loose (i.e. the spring is uncompressed) to obtain maximum locking of the knee. Tightening the brake screw decreases the locking effect for those amputees who do not require maximum locking. Maximum locking may not be desirable, for example, for an amputee who regularly walks on sloping ground because descending an incline is difficult with a locked knee.

2.3 THE CRITICAL OFFSET DISTANCE

A new parameter called the critical offset distance, δ_{CR} , will be defined in this section. It will be used as a measure of the safety factor provided by locking-type knee joints. To date, there has been no quantitative basis proposed to measure the safety factor of locking-style knees. Commercial success and qualitative user reports are the existing gauges of a knee's safety.

For an A.K. amputee standing still on any prosthesis, δ_{CR} would represent the greatest posterior distance of the amputee's centre of gravity from the knee's centre of rotation without the knee buckling. In the more general case of walking, δ_{CR} represents the distance by which the line of action of the ground reaction can pass posterior to the knee's centre of rotation before the knee will buckle.

A larger δ_{CR} corresponds to a "safer" knee. The δ_{CR} would be zero for a simple, non-locking knee.

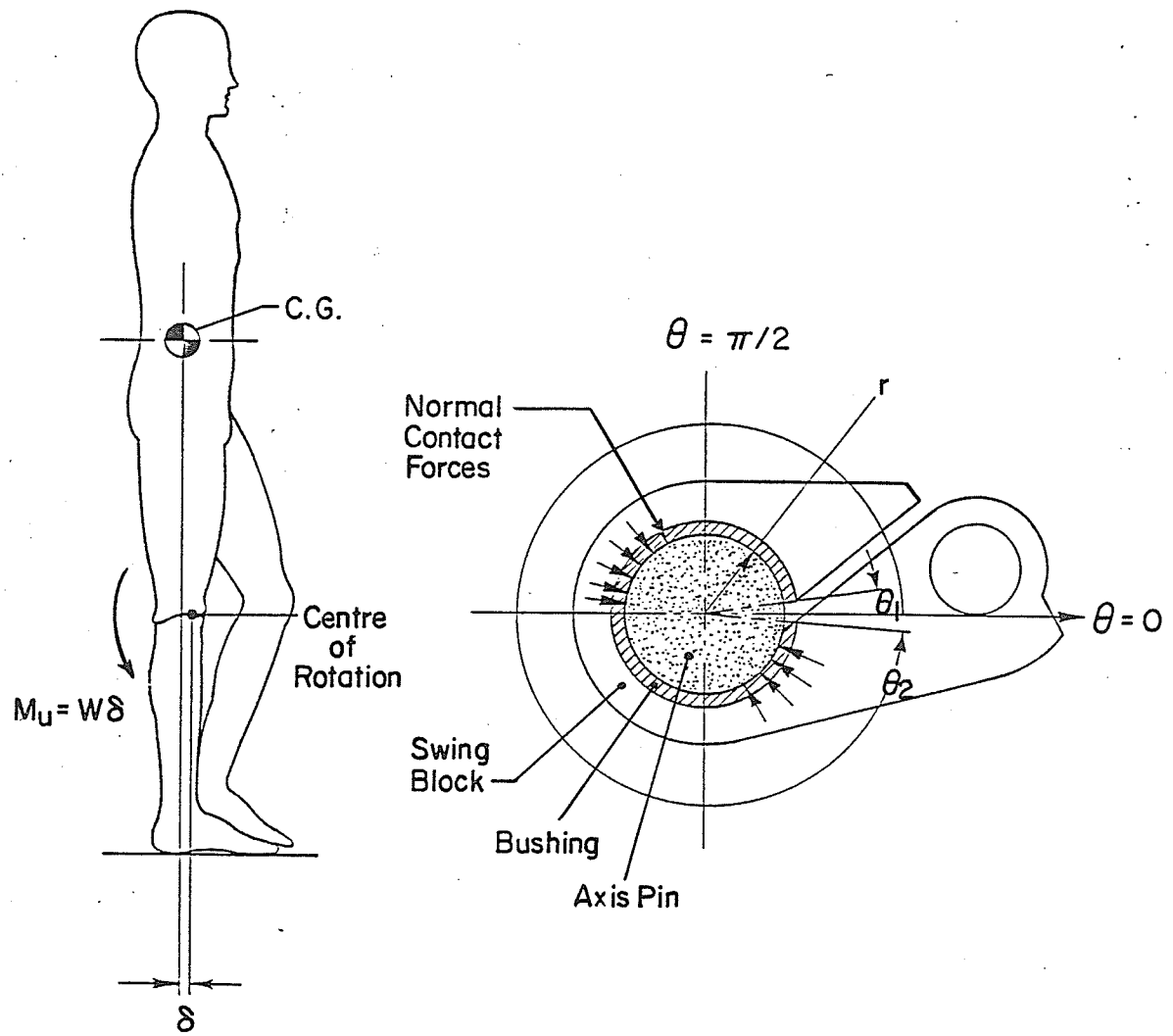
To better understand the effect of δ_{CR} , consider the configuration of a single axis knee joint shown in Figure 2.3(a). The static alignment of the prosthesis in this figure is such that the centre of gravity of the amputee is posterior to the knee's centre of rotation by an offset " δ ". This offset would create instability in a normal single axis knee. However, for a safety-style knee joint, clamping of the knee's pivot pin tends to prevent any uncontrolled rotation (or buckling) caused by the offset centre of gravity. The offset at which the braking or clamping of the pivot pin of the safety knee is overcome by the rotational torque caused by the offset of the amputee's C.G. is called the critical offset distance, δ_{CR} . An offset larger than δ_{CR} will be unstable. Thus, the δ_{CR} is a measure of the safety factor provided by the locking feature of the knee joint.

Two counteracting factors must be considered when determining δ_{CR} . For a stationary amputee bearing all the body weight on the prosthesis as illustrated in Figure 2.3(a), these factors are:

- (i) The magnitude of the unbalanced moment of the body weight, M_U , about the knee pivot. M_U is given by

$$M_U = W \delta \quad (2.1)$$

where W is the weight of the amputee; and



(a) Offset of the Static Weight Line (b) Clamping of the Knee's Axis

Figure 2.3 : The Critical Offset Distance

δ is the actual posterior offset of the centre of gravity of the amputee with respect to the centre of rotation of the knee.

- (ii) The frictional or locking torque, which is created by the contact between the swing block, bronze brushing, and the axis pin shown in Figure 2.3(b). By assuming simple Coulomb friction, the magnitude, M_f , of this locking torque is given by

$$M_f = \int_{\theta_1}^{\theta_2} r u F_c (\theta, W) d\theta \quad (2.2)$$

where

r is the radius of the axis pin

u is the static coefficient of friction between the axis pin and bronze bushing;

F_c is the magnitude of the normal contact force (as indicated in Figure 2.3(b)) between the axis pin and bronze bushing; and

θ_1, θ_2 are the limits of the included angle of the split bronze bushing as illustrated in Figure 2.3(b).

Physically, relation (2.2) corresponds to the torque amplitude which would have to be applied to the clamped axis pin before the knee would rotate. This value is called the locking torque of the knee.

For equilibrium at the point of impending instability,

$$M_{CR} = W \delta_{CR} \quad (2.3)$$

where δ_{CR} is the critical offset; and
 M_{CR} is the locking torque or unbalanced
moment at the point of instability.

Thus, the locking torque of the knee, given by equation
(2.2), must equal the unbalanced moment, M_{CR} , at this point.
By combining equations (2.2) and (2.3),

$$W\delta_{CR} = \int_{\theta_1}^{\theta_2} r u F_C(\theta, W) d\theta. \quad (2.4)$$

Because u is approximately constant [7], equation (2.4)
can be rewritten as

$$\delta_{CR} = \frac{ur}{W} \int_{\theta_1}^{\theta_2} F_C(\theta, W) d\theta. \quad (2.5)$$

A closed form solution of equation (2.5) is not
possible because the function $F_C(\theta, W)$ is not known
explicitly. However, a basis for an approximation is given
by the discretized form

$$\delta_{CR} = \frac{ur}{W} \sum_{i=1}^k (F_{C_i} \Delta\theta_i) \quad (2.6)$$

where F_{C_i} is the average normal contact force on the
 i th section of the axis pin;

$\Delta\theta_i$ is the discrete angular increment; and

k is the number of discrete increments.

The accuracy of equation (2.6) will depend, of course, on
the coarseness of the assumed discretization of the i
subscripted variables. Equation (2.6) will be employed in
Chapter 3 to estimate the critical offset distance.

2.4 OPTIONS TO IMPROVE THE 3R15

The 3R15 knee joint was designed specifically to provide an increased margin of stability (or safety factor) for the amputee. A new parameter, δ_{CR} , was proposed in Section 2.3 to measure this increased margin of stability and equation (2.6) was developed as a means of calculating δ_{CR} . Equation (2.6) can be inspected to determine those parameters which could lead to a beneficially large δ_{CR} . It can be seen that δ_{CR} can be increased, for a given body weight W , in the following three ways:

- (i) The coefficient of friction, u , between the swing block and the axis pin can be increased;
 - (ii) the radius of the axis pin, r , can be enlarged; or
 - (iii) the total contact force, F_{TOT} , can be increased,
- where

$$F_{TOT} = \sum_{i=1}^k (F_{Ci} \Delta \theta_i) . \quad (2.7)$$

The option of increasing the coefficient of friction is not practically viable because a larger u would impair the free swing of the prosthesis and would probably increase wear on the moving parts. The second option of increasing the radius of the axis pin is feasible as long as the size and weight of the knee are not increased significantly. The third option of increasing the contact force, F_{TOT} , is also a practicable alternative which can be accomplished in seve-

ral ways. The F_{TOT} depends upon the following three main factors:

- (i) the rigidity of the swing block;
- (ii) the magnitude and placement of the forces applied to the swing block (which are determined, in turn, by the geometry of the upper fork and swing block); and
- (iii) the nominal clearance between the bore of the swing block, the bushing, and the axis pin.

The rigidity of the swing block refers to its resistance to flexing. A physical explanation of how the clamping occurs was given in Section 2.2. Basically, the front web section of the swing block illustrated in Figure 2.4 flexes as the amputee's weight is placed on the knee joint. The flexing causes the gap between the swing block, bushing, and axis pin to decrease until contact occurs. This contact constrains further deflections at the contact points and serves to clamp the axis pin which is prevented, therefore, from rotating by the action of friction. The total clamping force on the axis pin, F_{TOT} , depends upon the rigidity of the swing block. For example, if the swing block was perfectly rigid, F_{TOT} would invariably be zero because there would be no contact at all and, hence, no clamping action.

Forces on the swing block are influenced by the geometry of the block itself and the upper fork. These geometries determine the manner in which the body weight is

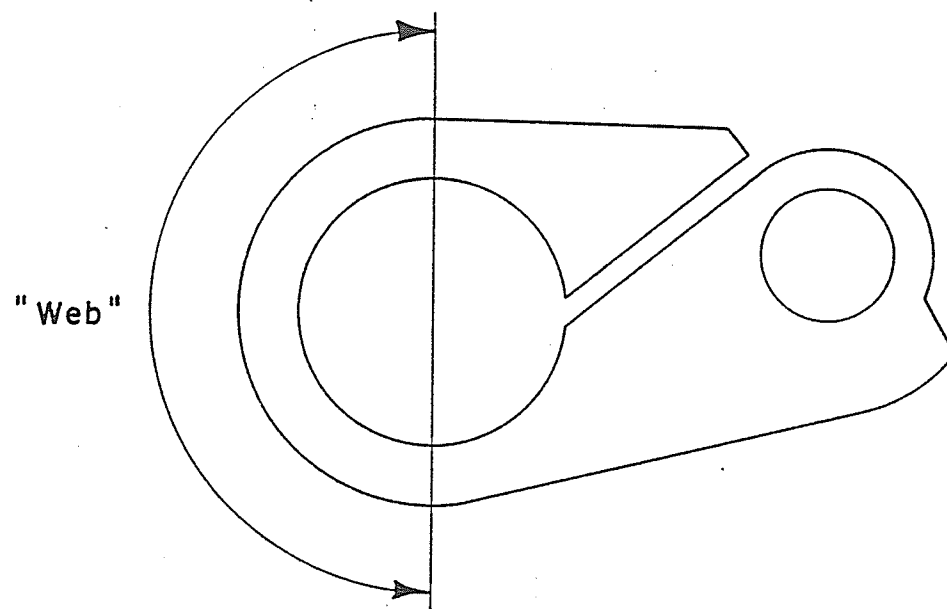


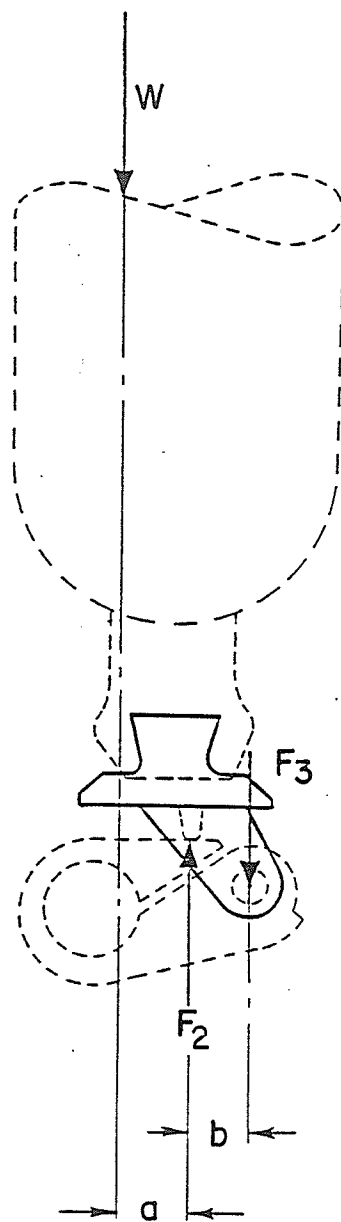
Figure 2.4 : The "Web" Section of the Swing Block

transferred to the swing block. The upper fork of the knee basically acts as a force multiplication mechanism as suggested in Figure 2.5. Forces with magnitudes F_2 and F_3 in this figure are the reactions which tend to force the swing block to flex and clamp around the axis pin when the amputee's weight, W , is applied. The F_2 and F_3 would grow if the ratio a/b was to be increased which would enlarge, in turn, the amplitude of the clamping forces, F_{TOT} , on the axis pin. The contact force could also be increased by decreasing the nominal clearance between the bore of the swing block, the bushing, and the axis pin. With a smaller gap, the swing block would have to deflect less to make contact, thus requiring a smaller force.

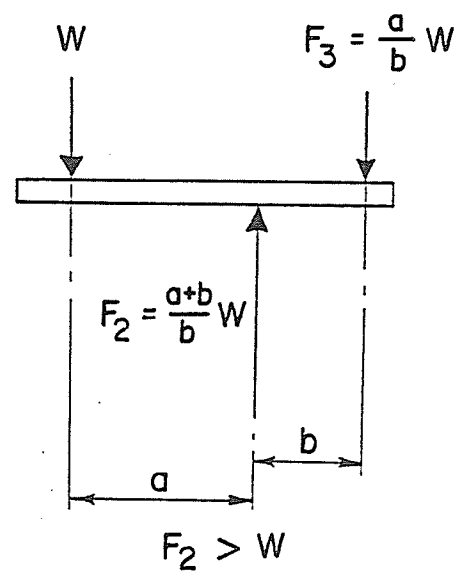
Based on the foregoing discussion, the magnitude F_{TOT} can be increased by making the swing block less rigid, by modifying the geometry of the knee to magnify the forces causing the swing block to tighten around the axis pin; or by decreasing the nominal clearance between the swing block, bushing and axis pin.

2.5 DESIGN CONSTRAINTS

The intent in this thesis is to optimize the function of the Otto Bock 3R15 safety knee. However, any improvement in function is only academic if the resulting design is not viable commercially. Discussions were held, therefore, with



(a) Forces on the Upper Fork



(b) Equivalent Force System

Figure 2.5 : Force Multiplication Through the Upper Fork

both prosthetists and the manufacturer of the knee joint to set realistic constraints. Four basic design constraints became evident from these discussions. The first constraint was that the durability of the present knee joint must not be compromised. The second was that the size, cost, and weight of the knee should not increase and, ideally, all these parameters should decrease. The third requirement was to ensure that stresses in the redesigned knee would not exceed comparable ones in the current knee. The fourth and final requirement was introduced to ensure that the least number of manufactured components would need modification. Any design change would require, in all likelihood, new tooling. Understandably, the manufacturer desired to keep tooling expenditures (which can be typically around \$10,000.00 per new part) to a minimum. Furthermore, changes would increase inventory costs because the manufacturer would have to stock parts for both the existing and the improved designs.

2.6 DESIGN APPROACH

Based on the constraints described in the last section, it seemed that the options of increasing the flexibility of the swing block and changing the clearance between the swing block and the axis pin were the most feasible. The last option of enlarging the axis pin was eliminated because it

would have entailed not only changing the present tooling for the axis pin, bushing, and swing block, but the change would also increase the overall size of the joint. Furthermore, modifying the geometry of the upper fork and the swing block to increase the clamping forces was also rejected because, again, new tooling would have been needed.

The design process, which will be detailed in Chapter 3, was based on the following general approach:

- (i) The magnitudes of the contact forces (F_{TOT}) and stresses in the present swing block were calculated for various body weights and radial clearances by using a finite element model.
- (ii) The calculated contact forces were substituted into equation (2.6) to determine $\hat{c}_{CR}$.
- (iii) Based on the stress distribution in the model of the present swing block, material was removed progressively from the swing block to increase its flexibility. The ensuing contact forces, stresses, and the resulting $\hat{c}_{CR}$ of the new swing block were calculated by employing the finite element technique.
- (iv) An iteration was performed to achieve the optimum design. The optimum design is the one in which the swing block is as flexible as possible without increasing the maximum stresses in the swing block.

This computational approach allows a more economical and relatively quick investigation to be conducted, compared

to an experimental approach, to determine the effect of design modifications. It is especially suitable in the present situation where an existing part can be analyzed initially to assess the validity of the computer modelling which will be only slightly perturbed subsequently.

CHAPTER 3

3.1 Finite Element Modeling of the 3R15

The general purpose commercially available finite element program "ANSYS" [8] was used to model the behaviour of the swing block and axis pin for various loadings and radial clearances. The model was employed to obtain estimates of the stress field and contact forces on the axis pin for the existing and redesigned versions of the knee joint.

3.1.1 Two-Dimensional Approximation

A drawing, which illustrates the overall size and shape of the swing block, is given in Figure 3.1. From the figure, it can be seen that the large bore which houses the axis pin and the bushing as well as the slot which allows the bore to tighten around the axis pin, extend through the thickness of the swing block. The axis pin and bushing contact across the entire thickness of the swing block when the block is loaded. In addition, the loads on the swing block are distributed uniformly across its width. Consequently, the clamping action of the swing block is basically in two dimensions because the clamping mechanism does not vary along the width of the swing block.

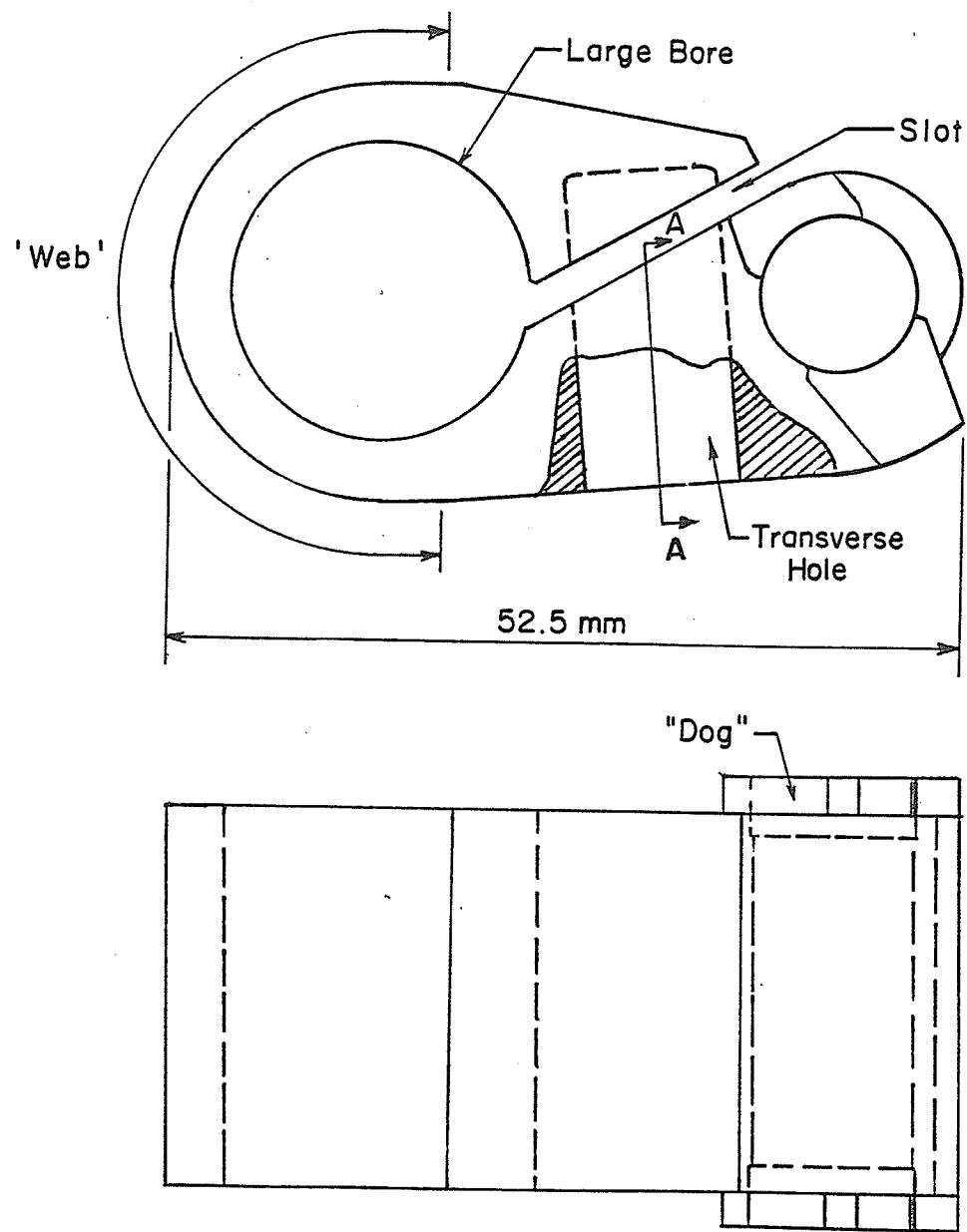


Figure 3.1 : The Original Swing Block

The only exceptions to the two-dimensional nature of the swing block are a transverse hole in the central region and the "dogs" which extend from the sides of the block. The transverse hole holds the brake screw and spring for the stance control adjustment discussed in Section 2.2. The dogs serve simply as stops to prevent the rotation of the upper fork. These two exceptions to the otherwise planar geometry of the swing block have little effect on the two-dimensional clamping action of the swing block around the axis pin. Indeed, an approximate calculation based on comparing the moment of inertia of the cross section of the web to that of the region of the transverse hole indicates that the latter is approximately nineteen times stiffer in bending than the web. Thus, the deflections that cause clamping of the axis pin depend mainly on the web. The relatively rigid region around the transverse hole contributes very little to the clamping. Furthermore, the "dogs" have a negligible effect on the flexibility of the swing block because they are far removed from the web. Hence, the neglect of the dogs and the transverse hole in a two-dimensional modelling of the swing block will have little effect on the clamping force and subsequently calculated δ_{CR} . The stress distribution in the region of the hole will be affected, however, by using a two-dimensional model. The region most affected will be at section A-A of Figure 3.1 because the hole decreases the

cross-sectional area of the swing block by a maximum 48% or so at that section. Therefore, the actual stresses around the hole will be about twice as large as those predicted by the two-dimensional model. However, it will be shown later that stresses around the hole of the two-dimensional model are, at most, about an inconsequential one third of the maximum stress in the web.

The two-dimensional model is much more economical than a three-dimensional model in terms of computer time. As an approximate estimate, the simplified two-dimensional model would probably require only 10 to 25% of the CPU time of an analogous three-dimensional model.

3.1.2 Plane Strain Approximation

Plane strain is normally assumed when a planar type of structure is "thick" in relation to its cross-sectional area. Plane stress, on the other hand, is appropriate when the structure is "thin" in relation to its cross-sectional area. A "thick" structure tends to constrain thickness changes, which creates a three-dimensional stress state more than does a thin structure. The swing block does not truly fit either category because its main body is relatively thin, with a thickness:area ratio of about 1, whereas the web section has a fairly "thick" thickness:area ratio of about 5.6. Initial analyses were performed, therefore, by

using both the plane strain and the plane stress assumptions for the finite element model to quantitatively investigate the effects of the two assumptions. It was found that there was almost no difference. Maximum principal and effective stresses and contact forces with the plane strain assumption were within 0.5 percent of those obtained by assuming plane stress. As well, the stress distributions for the maximum principal and effective stresses were visually identical for both cases. In practice, therefore, it appears to make little difference whether plane stress or plane strain is assumed. The plain strain approximation was chosen for all the remaining analyses because of the relatively high thickness to area ratio for the more important web section.

3.1.3 Loading and Scaling

The case of a stationary, standing amputee bearing weight on the prosthesis was chosen as a basis of comparison for all analyses. Although the case of a walking amputee could have been investigated, the choice of the stage in the walking cycle and the magnitude of the horizontal component of the floor reaction force would have been quite arbitrary. The dynamic case of a walking amputee could be estimated from the results of the static analysis. If, for example, the peak dynamic loading of a prosthesis was, on average,

30% greater than the body weight, then the dynamic case could be estimated by using results for a static case of 1.3 times the actual body weight. The static case was chosen as the basis of comparison to simplify the derivation of the equilibrium equations and to simplify the construction of future test equipment to physically determine δ_{CR} . Because δ_{CR} has been proposed as a basis for comparing and evaluating the effectiveness of locking-type knee joints, it is very important that the test conditions for measuring δ_{CR} be easily reproducible. This is so for the assumed static standing case. It was also assumed that the alignment of the knee joint is such that the weight passes directly through the centre of rotation of the knee. This allowed the critical offset distance to be estimated by using equation (2.6). This particular alignment was chosen because it represented the point of imminent instability for no locking of the knee ($\delta_{CR}=0$), which in turn ensured physical stability of the finite element model for all cases. It is important to emphasize that δ_{CR} calculated for the case of the body weight passing through the centre of rotation of the knee is an approximation. This approximation arises because the weight line is actually offset from the centre of rotation by δ_{CR} when the knee is at the point of instability. However, δ_{CR} is not known a priori, so that a time consuming and expensive iterative approach involving a series of corrections subsequent to an

initial guess would be necessary to obtain more accurate results. This procedure was not done because a relative (as opposed to an absolute) comparison of the existing and redesigned knee joints yielded trends of improvement using only one iteration with an initial guess of $\delta_{CR}=0$ for all cases. The comparison is subject to the inherent assumption that the degree of error is consistent for both designs for the same conditions of loading and radial clearance.

The derivation of the equations of equilibrium and the resulting loads for the knee joint under the two previously described assumptions are given in Appendix A. A scale factor of 1/25 was applied to all loads when they were input to ANSYS because the model of the swing block was only 1mm thick while the physical width of the swing block is 25mm. The choice of a 1mm thick "slice" of the swing block was arbitrary. To avoid confusion, however, all values given in this thesis will refer to actual physical loads or results rather than to the scaled loads or results.

3.1.4 Use of Gap Elements

Gap elements [8] were used to model the radial clearance between the swing block and the axis pin. These elements acted essentially as almost rigid springs at the points where the swing block came into contact with the axis pin. No forces were transmitted, of course, where a gap

existed. The contact stiffness or spring rate of the gap elements was specified to be 1×10^7 N/mm. The contact stiffness was chosen to be approximately ten times the stiffness of the surrounding elements as per reference [8]. An initial gap equal to the radial clearance was assigned to each element for all cases. The elements were frictionless so that the forces at the points of contact were strictly normal to the surfaces of the swing block and the axis pin, as shown in Figure 3.2. The magnitude of the total normal contact force on each element, F_N , can be substituted directly into equation (2.6) so that

$$\delta_{CR} \approx \frac{ur}{W} \sum_{i=1}^k F_{N_i} \quad (3.1)$$

where F_{N_i} is the amplitude of the total normal contact force on the i^{th} element;

i is the element number; and

k is the number of gap elements employed.

As explained in [8], the gap elements require an iterative solution. The number of iterations required for convergence depends upon the number of elements used. A larger number of gap elements requires a greater solution time so that it is desirable to use a minimum number of gap elements. For example, doubling the number of gap elements could increase the solution time by a factor of around four. The split bronze bushing (item 6 of Figure 2.2(c)) and the axis pin were not modelled individually, therefore, in order to decrease the number of gap elements. The combined

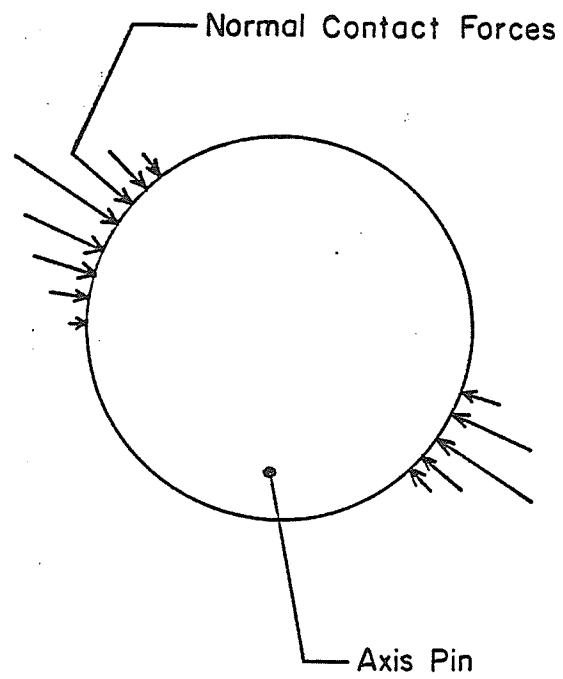
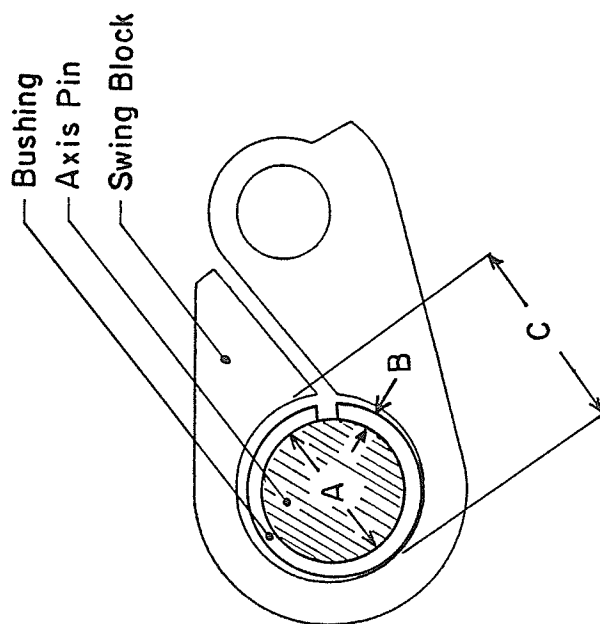


Figure 3.2 : Normal Forces on the Axis Pin

effects of the bushing and axis pin were approximated by using the enlarged version of the axis pin shown in Figure 3.3. This approximation is reasonable because the split bushing is essentially a spacer which is simply clamped between the axis pin and swing block. The same clamping action would occur if the bushing were eliminated and replaced with a larger diameter axis pin. In practice, however, the bushing is needed to prevent steel to steel contact.

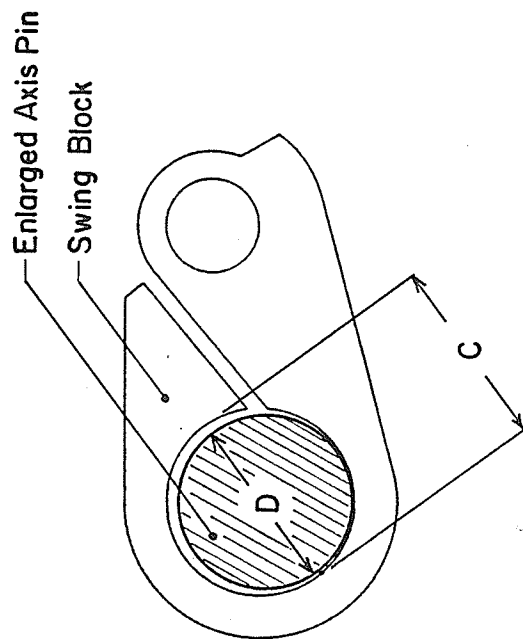
3.1.5 The Actual Finite Element Model

The automated meshing capabilities of ANSYS were employed to generate the finite element mesh. This feature was found to be very useful. A change in the radial clearance or body weight simply involved editing a few lines of the basic input file. The complete finite element model for the knee is shown in Figure 3.4. A sample input file used to create such a model is given more conveniently in Appendix B. The particular discretization illustrated uses a total of 188 quadrilateral (STIF42) and 332 triangular (STIF42) elements for the swing block while the enlarged axis pin was modelled with 120 triangular (STIF42) elements. The radial clearance was modelled by employing 48 gap (STIF52) elements. This idealization was derived by using an iterative approach. A model was created initially which



$$\text{Radial Clearance} = \frac{1}{2} [C - 2B - A]$$

(a) Actual Axis and Bushing



$$D = A + 2B$$

$$\text{Radial Clearance} = \frac{1}{2} [C - D]$$

(b) Finite Element Model with Enlarged Axis Pin, No Bushing

Figure 3.3 : Enlarged Axis Pin Approximation

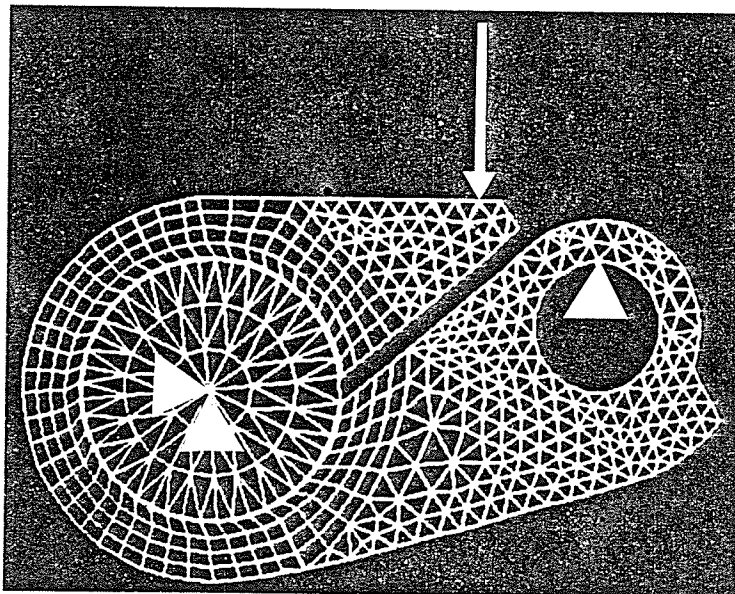


Figure 3.4 : Actual Finite Element Model of the Original Swing Block

employed approximately twice as many elements as the model presented in Figure 3.4. The model was analyzed for a test case using a radial clearance of 0.05mm and a body weight of 750N. and the results were recorded. A new model was created which used somewhat fewer elements and the new results were compared to those of the initial model. This process was repeated until the progressive reduction in the number of elements affected the very first model's results by more than about 4%. At this point it was decided to use the model immediately preceding the coarsest discretization. The results for the model of Figure 3.4 varied by less than 1% from those for the initial model while the time for the computer solution was reduced by more than a factor of two.

3.2 Results and Discussion For The Original 3R15

3.2.1 Effects of Varying the Radial Clearance

The effects of varying the radial clearance between the steel swing block and the axis pin as well as the effects of changing the applied load on the knee joint were investigated. Initially, the total load or body weight W , shown acting on the knee joint in Figure 3.5, was held constant at 750N (or 169 lbs.) while the radial clearance was varied from 0.01mm to 0.17mm in increments of 0.02mm. The applied load of 750N is within the normal weight range

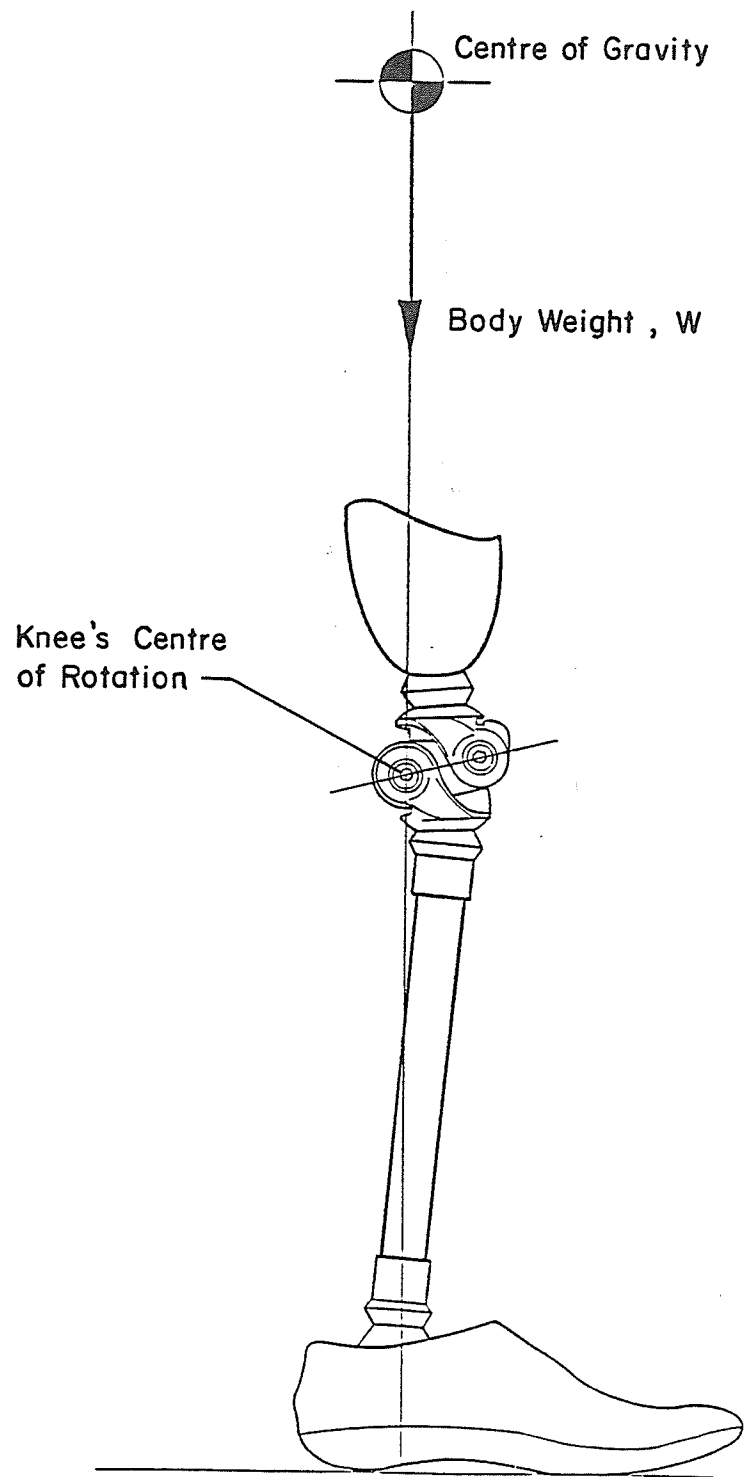


Figure 3.5 : Body Weight on the Knee Joint

of most amputees and the radial clearances investigated extend well above and below the average clearance of about 0.05mm for a real knee. The results for these initial cases are presented in Figures 3.6 through 3.9 and also in Figures C.1 through C.9 of Appendix C. Discussion will be limited at this time to the results for the "original" swing block which are invariably denoted by an "x" in the figures.

Figure 3.6 shows the variation of the sum of the normal forces acting on the gap elements ($\sum_{i=1}^k F_{N_i}$ in equation (3.1)) with a changing radial clearance. As discussed in Section 3.1.4, the sum of the normal forces on the gap elements physically represents the total contact or clamping force of the swing block on the axis pin. One would expect this sum to decrease as the radial clearance increases because larger deflections (and, therefore, larger forces) are required then to initiate contact between the swing block and the axis pin. This expectation is verified by Figure 3.6 where it can be seen that the contact forces decrease as the radial clearance increases.

Figures 3.7(a) and (b) present the maximum principal stress and effective stress, respectively, for the various radial clearances. The maximum stresses can be seen to increase quite rapidly as the radial clearance increases. This trend is attributable again to the greater deflections of the swing block before contact with the axis pin at larger radial clearances.

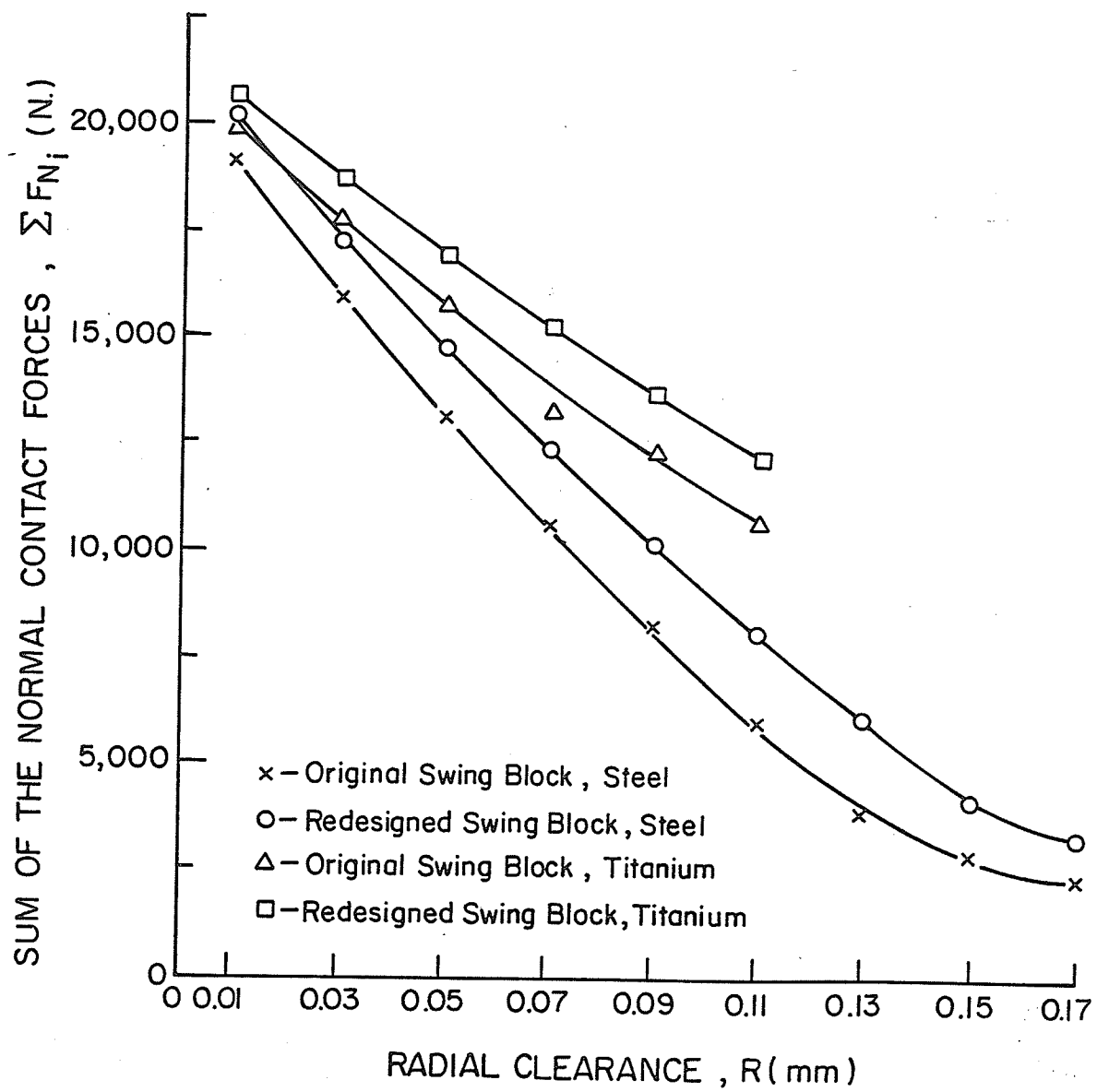


Figure 3.6 : Contact Force Versus Radial Clearance, Load = 750 N.

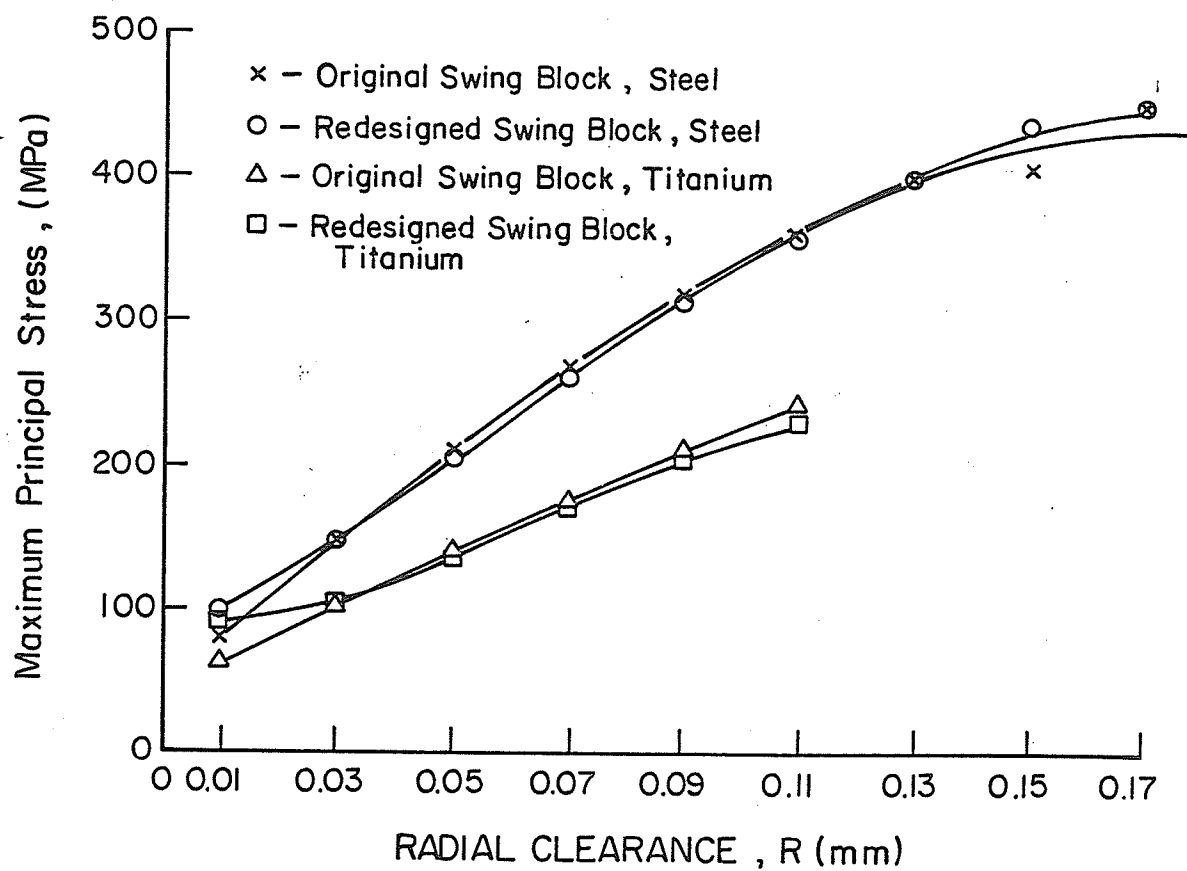


Figure 3.7 (a) : Maximum Principal Stress Versus Radial Clearance,
Load = 750 N.

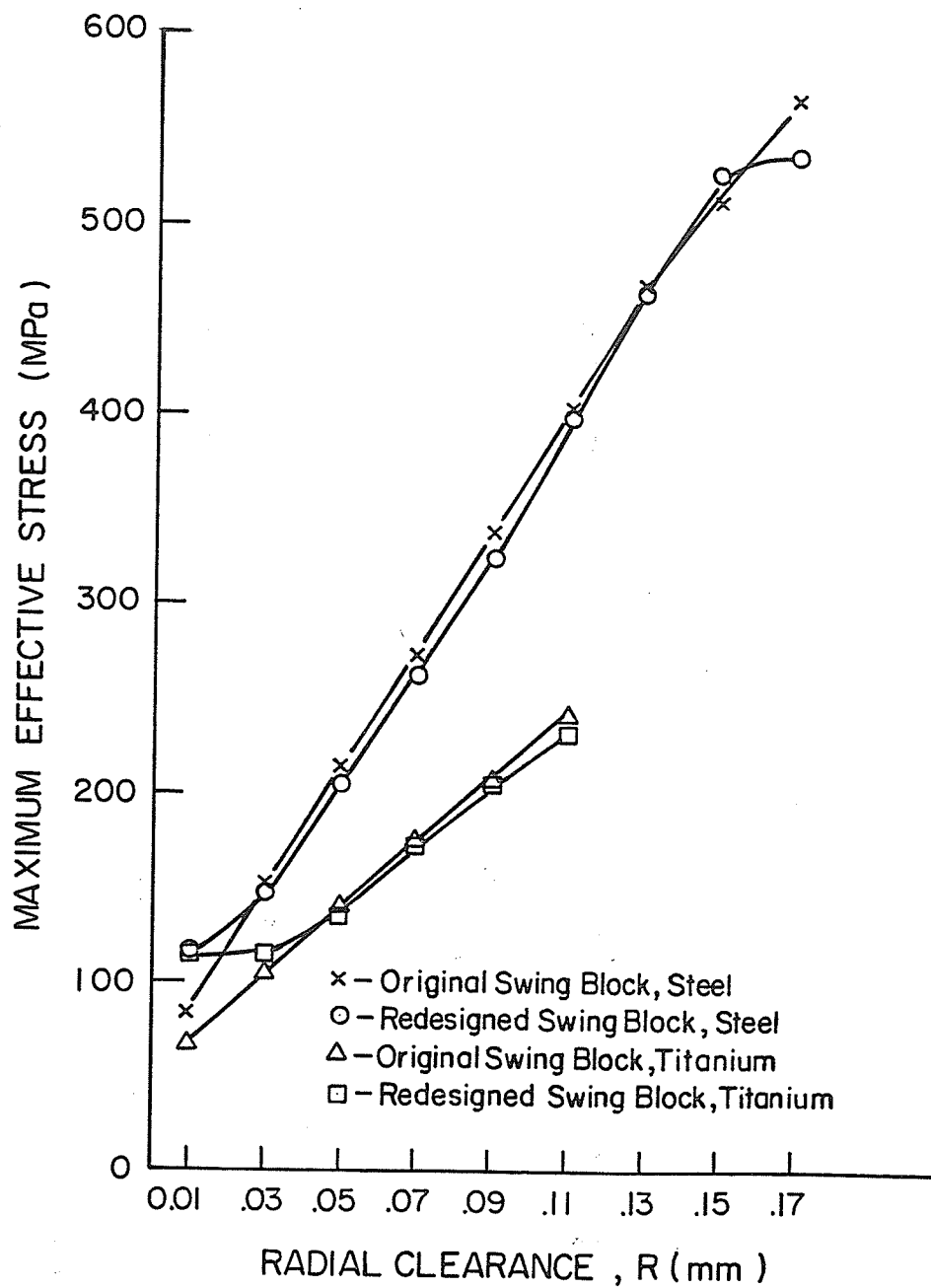


Figure 3.7 (b) : Maximum Effective Stress Versus Radial Clearance ,
Load = 750N

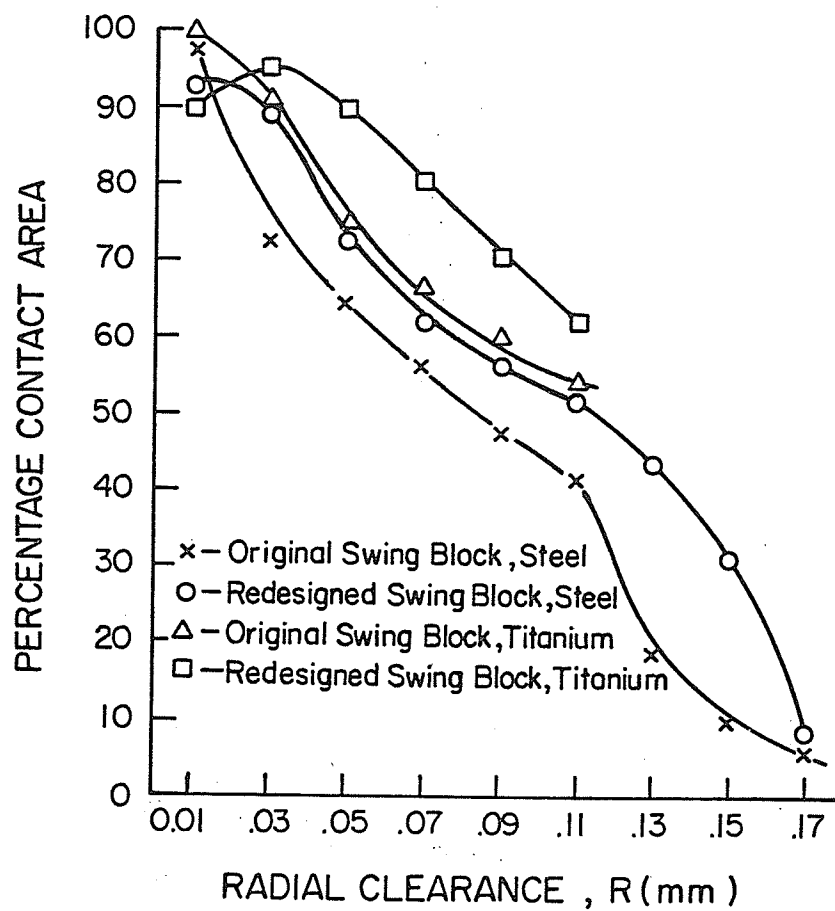


Figure 3.8 : Percentage Contact Area Versus Radial Clearance ,
Load = 750 N

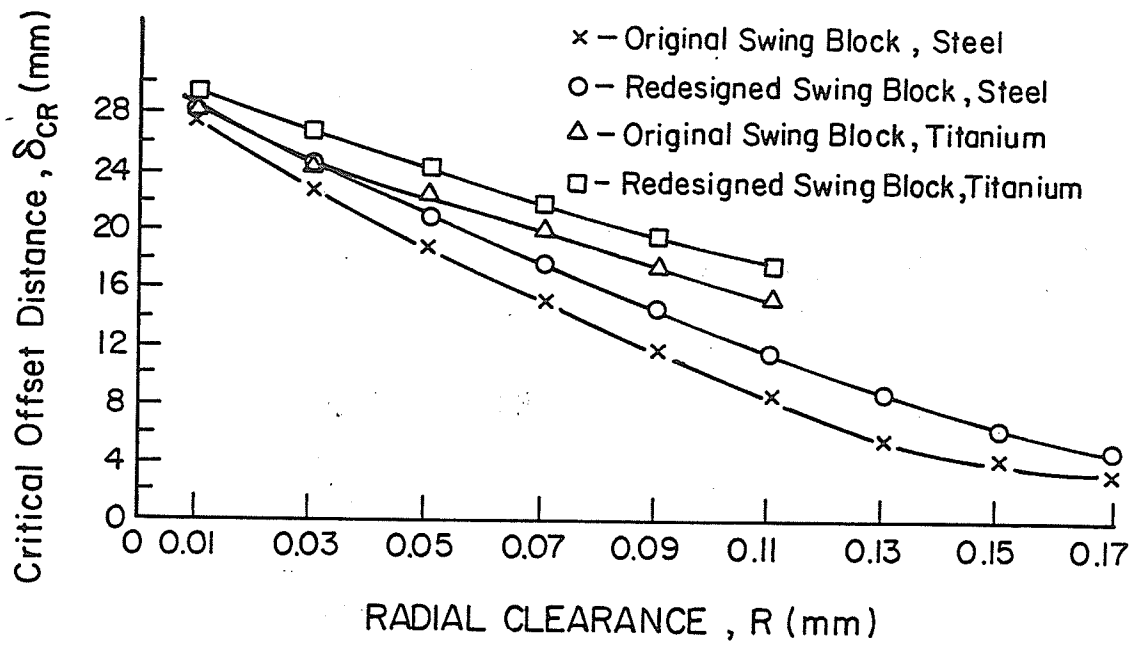


Figure 3.9 : Critical Offset Distance Versus Radial Clearance,
Load = 750 N.

Figure 3.8 illustrates how the percentage contact area between the swing block and axis pin varies with the different radial clearances. Physically, this contact area represents the percentage of the total surface area of the axis pin which is in contact with the swing block. It was calculated by simply dividing the number of contacting gap elements by the total number of gap elements and multiplying by 100%. Generally, the area of contact decreases quite rapidly as the radial clearance increases. This trend is expected because, as the swing block deflects, the initially round inner bore surrounding the axis pin becomes slightly oval shaped and, therefore, cannot make complete contact with the round axis pin. The axis pin deflects more at greater radial clearances so that the bore becomes increasingly out-of-round and the contact area decreases accordingly.

Figure 3.9 gives the critical offset distance, δ_{CR} , as a function of the radial clearance. It can be seen that δ_{CR} decreases quite rapidly as the radial clearance increases. This trend is expected because, as for the variation of the summed normal contact forces in Figure 3.6, larger deflections (and, therefore, larger forces) are required then to initiate contact between the swing block and the axis pin for larger radial clearances. From Figure 3.9, it can be concluded that the radial clearance should be kept as small as possible to increase a given artificial knee joint's δ_{CR} .

Figures C.1 through C.9, given for convenience in Appendix C, present the equivalent and principal stress distributions in the swing block for the cases of Figure 3.6 through 3.9. Each colour represents a different stress level; dark blue corresponds to the regions of lowest stress, light blue shows the next higher stress level, etc. The coloured numbers on the side of the figure correspond to the stress levels, in MPa, of the various contours. Letters "MX" and "MN" shown in these figures represent the points of maximum and minimum stress, respectively. The deflections of the swing block are shown full scale.

It can be seen that, except for the largest radial clearances over 0.13mm, the maximum stresses tend to be at the top and bottom sections of the web where the web blends into the main body of the swing block. Furthermore, the stresses at the top and bottom of the web are very nearly equal even though the maximum stress (denoted by "MX") is generally located at the web's bottom. As the radial clearance increases, the maximum stresses increase correspondingly and eventually become more uniform throughout the web section. When the radial clearance is greater than about 0.13mm, the point of maximum stress is located at the centre of the web section. This difference occurs because of the then almost complete loss of clamping contact between the swing block and the axis pin.

It can also be seen that stresses in the region of the transverse hole (section A-A of Figure 3.1) are typically about 1/4 of the maximum stress, although for radial clearances of less than 0.05mm the proportion is about 1/3. As was discussed in Section 3.1.1, stresses around the hole in the physical swing block, then, would be approximately 1/2 to 2/3 of the maximum stress. The higher proportion for radial clearances under 0.05mm does not indicate that the swing block is more likely to fail because, even though the proportion has increased, the magnitude of the stresses is much less than that for greater radial clearances.

The main conclusion to be drawn from Figures 3.6 through 3.9 and Figures C.1 through C.9 is that the radial clearance has a pronounced effect on the function of the knee joint. As the radial clearance increases, the maximum stresses grow rapidly whereas, on the other hand, δ_{CR} decreases. Physically, this trend implies that, as the radial clearance increases, the knee does not lock as well and, also, it is more likely to be subject to fatigue failure. Therefore, the radial clearance should be kept as small as possible.

3.2.2 Effects of Varying the Body Weight

The effects on the knee joint of varying the applied body weight were investigated in a second set of analyses.

The radial clearance was held constant at 0.05mm whilst the total load on the knee was varied from 250N to 2000N in increments of 250N. The radial clearance of 0.05mm was derived by averaging the measured clearances of about five of the existing 3R15 knee joints. The loads from 250N to 2000N comprehensively encompass the weights of A.K. amputees. Ensuing results are summarized in Figures 3.10 through 3.13 and in Figures C.10 through C.17 of Appendix C. They are presented in the same format as before except that the abscissa is now the applied load rather than the radial clearance.

Figure 3.10 indicates that there is basically a linear relationship between the sum of the normal contact forces and the applied load for weights greater than 250N. However, for loads under 250N, the relationship is not linear. A simple physical test of the existing swing block indicates that initial contact occurs for weight of approximately 50N. This is indicated on the figure by the dashed line extension of the curve for the existing swing block. The nonlinear section of the curve is expected because the weight is resisted only by the flexing of the swing block before contact occurs. As the load increases after the swing block contacts the axis pin, part of the load is resisted by additional flexing of the swing block while part is resisted by compression of the axis pin. As the load is further increased (over about 250N.), additional

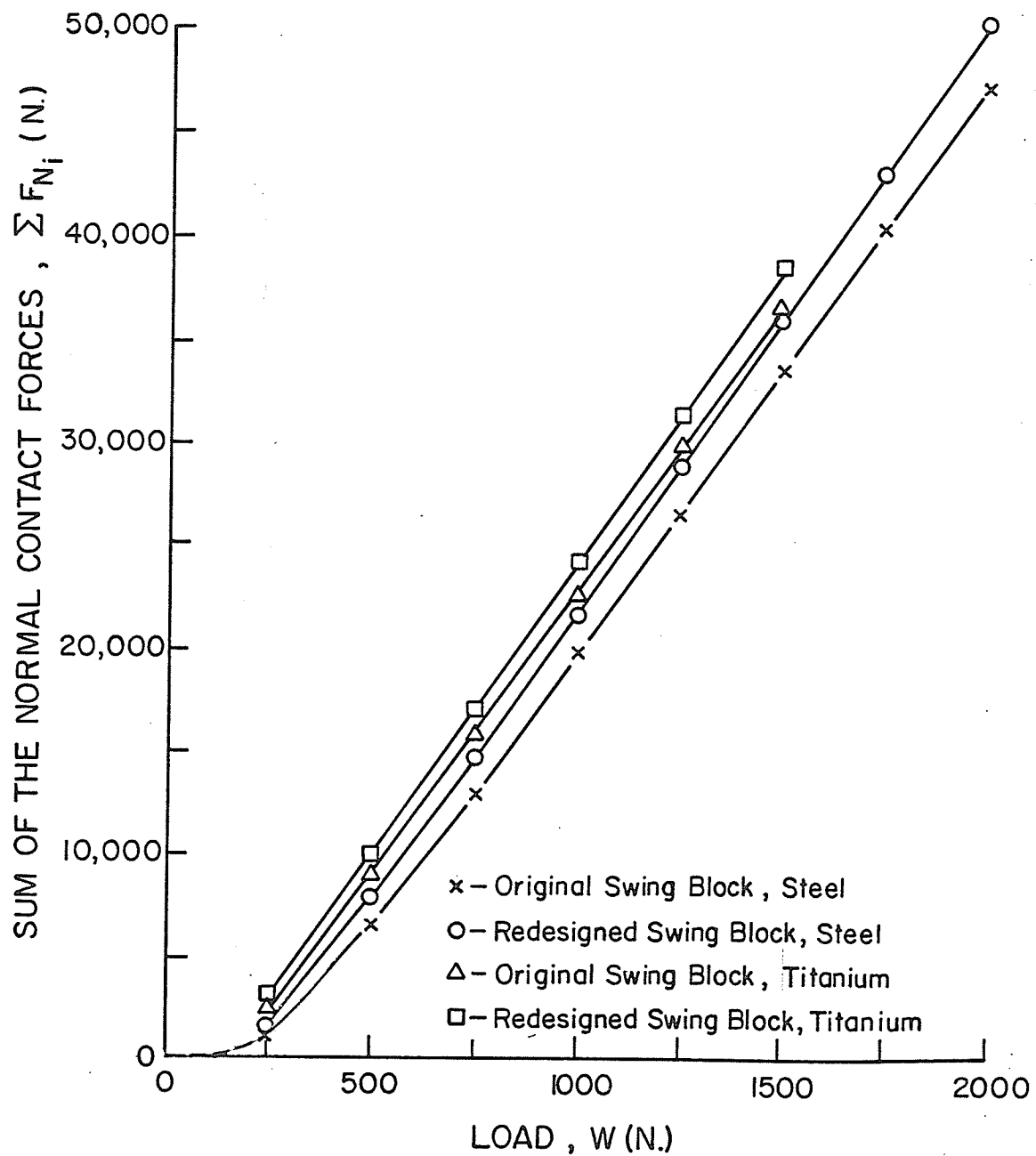


Figure 3.10 : Contact Force Versus Load , Radial Clearance = 0.05 mm

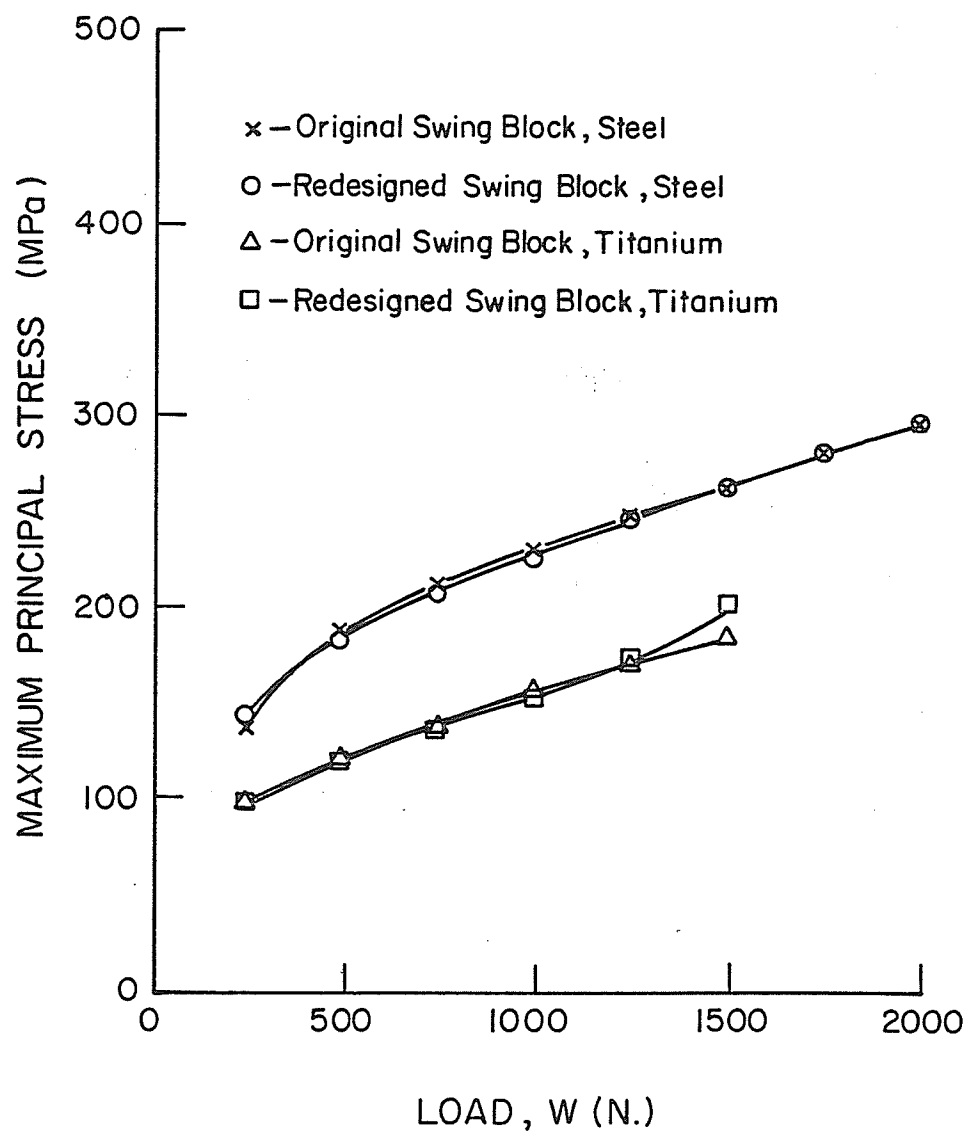


Figure 3.11 (a): Maximum Principal Stress Versus Load ,
Radial Clearance = 0.05 mm

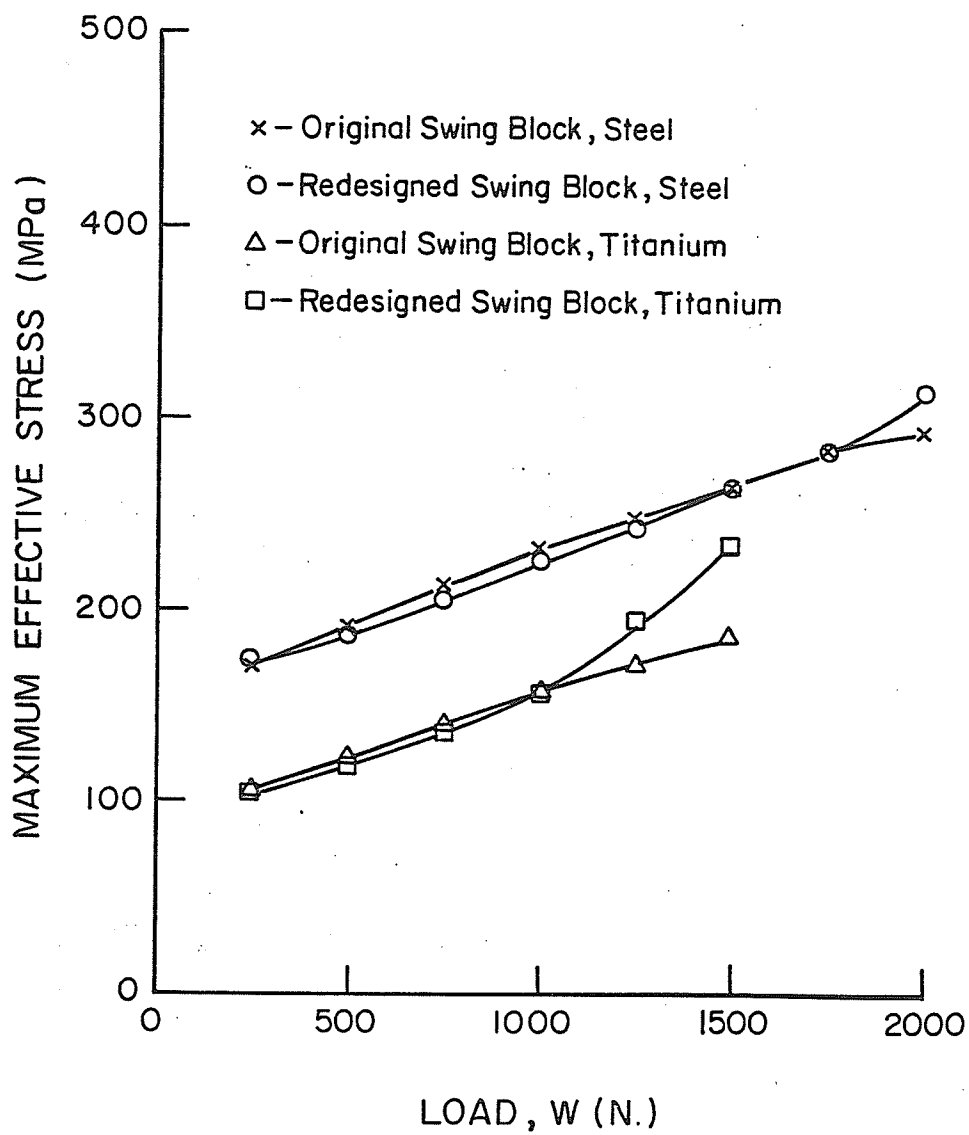


Figure 3.11(b) : Maximum Effective Stress Versus Load, Radial Clearance = 0.05 mm

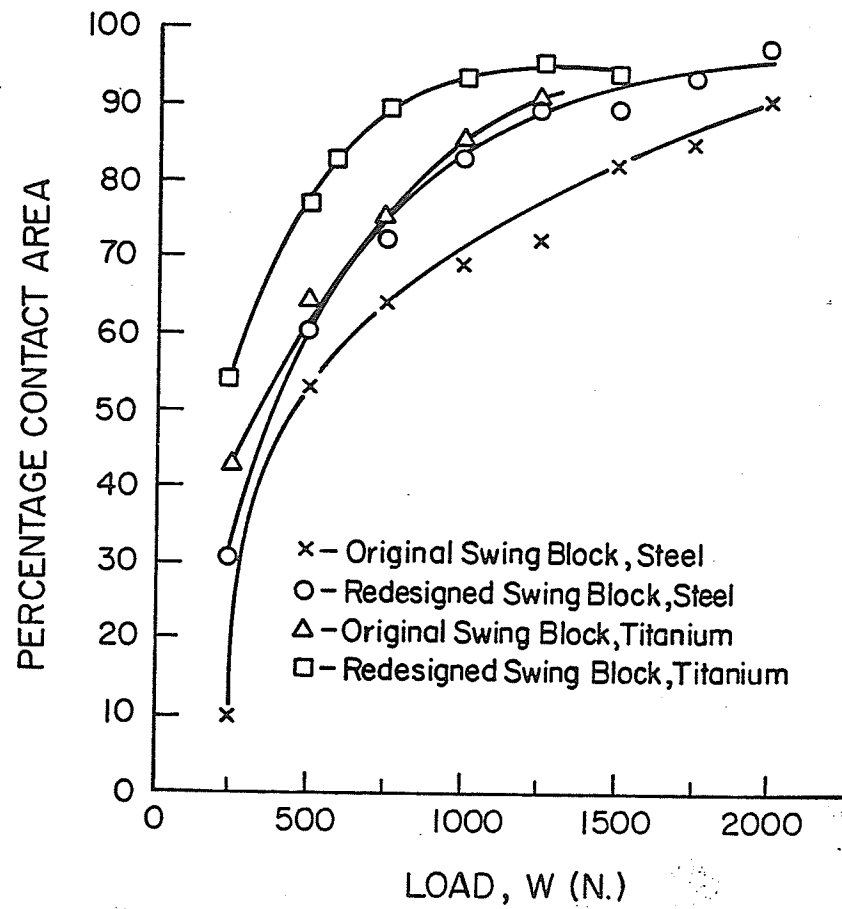


Figure 3.12 : Percentage Contact Area Versus Load ,
Radial Clearance = 0.05 mm.

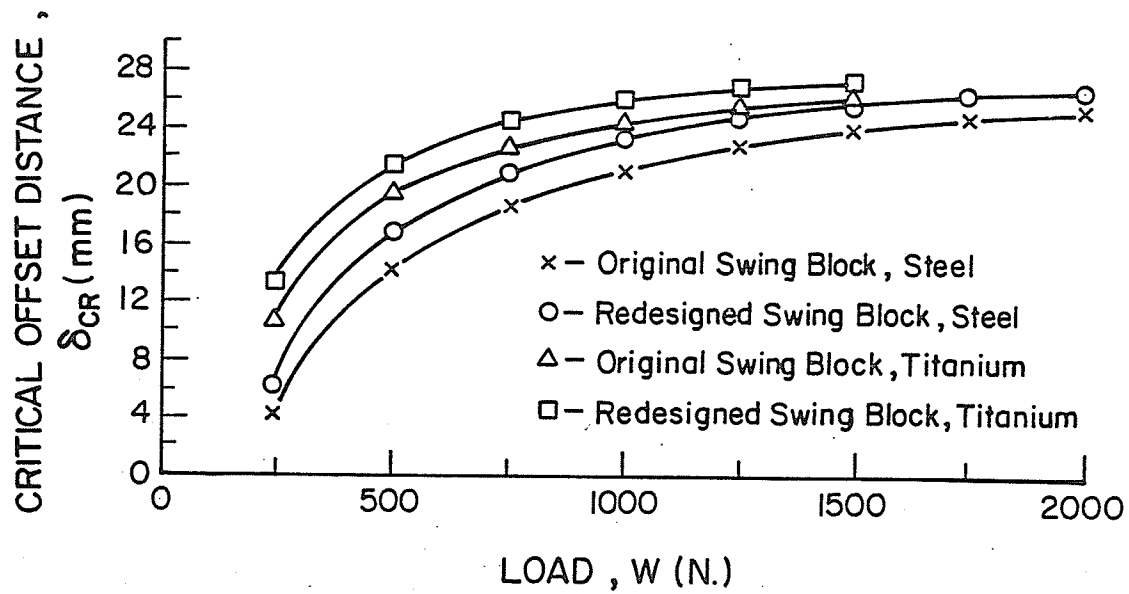


Figure 3.13 : Critical Offset Distance Versus Load ,
Radial Clearance = 0.05 mm

deflection of the swing block becomes negligible and the increase in the applied load is resisted by a proportionate increase of the contact forces between the axis pin and the swing block.

Figures 3.11 (a) and (b) illustrate the variation of the maximum principal and effective stresses, respectively, with a changing applied load. It is interesting to note that the stresses grow as the load increases but not in direct proportion. For example, doubling the load or body weight from 750N to 1500N does not double the maximum principal stress in Figure 3.11(a) but, rather, it increases from 212 MPa to 265 MPa or by only 25%. Thus, the load has less effect on the stress level than probably would have been anticipated. This diminished effect can be attributed to the increased contact shown in Figure 3.12 between the swing block and the axis pin as the load increases. This additional contact, of course, helps to decrease the maximum stress.

Figure 3.13 shows the variation of the critical offset distance, $\hat{c}_{CR}$, as the load or body weight is increased. The $\hat{c}_{CR}$ can be seen to grow very quickly for loads up to about 750N. This rapid rise is expected because a smaller percentage of a larger applied load is needed to close the constant 0.05mm gap for clamping to occur. This leaves an ever-increasing percentage of the larger load to be resisted by the contacting axis pin.

Figures C.10 through C.17 in Appendix C present the equivalent and principal stress distributions in the swing block. It was found again that, except for the lightest loading of 250N., the maximum stresses tended to occur at the top and bottom sections of the web. Furthermore, the stresses in these two areas were very nearly equal for a given load.

The main conclusion to be drawn from Figures 3.10 through 3.13 and Figures C.10 through C.17 is that δ_{CR} is very dependent upon the loading (particularly for weights less than about 1000N.) whereas the maximum stresses are not so dependent on the loading.

3.3 The Design Optimization Process

The previously computed results for the existing swing block form the basis for comparing the behaviour of slightly modified designs. The redesigned knee should not fail so long as the new maximum stresses are not greater than the analogous values for the existing, well proven knee. Variations in the critical offset distance with changing load and radial clearance will provide a comparative yet quantitative measure of the safety factor of the redesigned knee joint. A larger δ_{CR} for the redesigned knee would indicate an improvement. The comparisons are valid, of course, only if the analysis of the not radically redesigned

knee is performed for the same assumptions, materials, and loads as those for the existing knee.

3.3.1 The Initial Redesign

The goal of the redesign process is to increase the flexibility of the swing block so that it clamps more tightly around the axis pin during the bearing of the body weight. It was noted that the flexibility of the swing block depends primarily upon the web section because the rest of the swing block is relatively rigid. Increasing the effective length of the web or decreasing its thickness should significantly increase the flexibility and, therefore, the clamping of the axis pin.

Removing material (steel) from the swing block in the areas shown in Figure 3.14 would extend the length of the web section. A close study of the stress distributions presented in Figures C.1 through C.17 of Appendix C suggests that such material could be removed safely because stresses are generally low in the region in question. Unfortunately, the effects of removing material from other areas of the swing block are not quite so obvious. On one hand, any localized removal of material would be expected to increase stresses in that region. On the other hand, however, the improved flexibility should increase contact between the swing block and the axis pin which should produce, in turn,

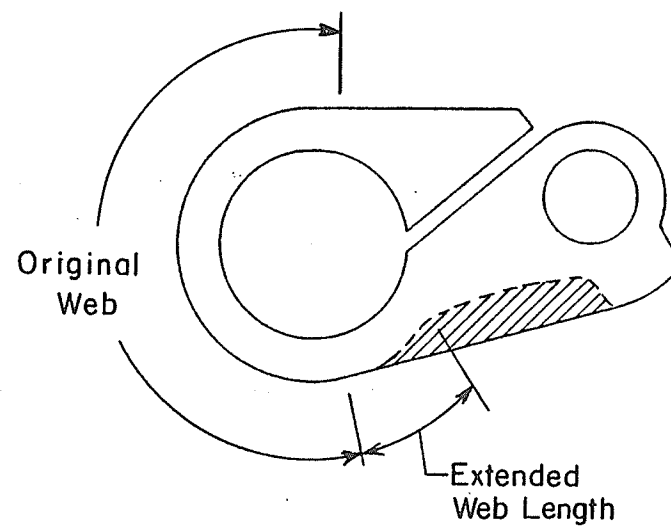
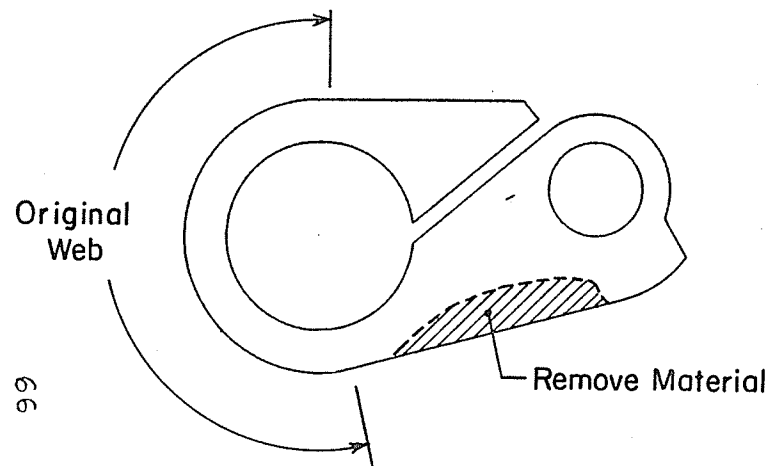


Figure 3.14 : Increasing the Length of the Web

lower bending stresses in the critical web region. An iterative approach tempered, of course, with engineering judgement appeared to be the simplest method of resolving this conflict and, hence, this was the approach adopted.

The first trials involved experimenting with the geometry of the swing block along the lines illustrated in Figure 3.15(a). The round hole in the central section of the swing block arose from a redesign of the stance phase control mechanism. The compression spring mechanism of the existing artificial knee was replaced with the split ring design as indicated in Figure 3.15(b). This substitution was done with the intention of eliminating the large transverse hole for the compression spring to ensure that the region of the swing block containing the transverse hole would not be weakened by material removal. Material was removed from the top and bottom sections of the swing block to lengthen the web and the web thickness was also decreased to provide more flexibility in the swing block. The rationale behind this initial redesign was to trim away probably excessive material to determine the extent an increased flexibility would negate a rise in the stresses.

Several representative finite element analyses of the new design suggested that the critical offset distance, δ_{CR} , increased significantly, especially at the lowest loading of 250N. Unfortunately, the stresses increased significantly too, particularly in the areas shown in Figure 3.15(c). The

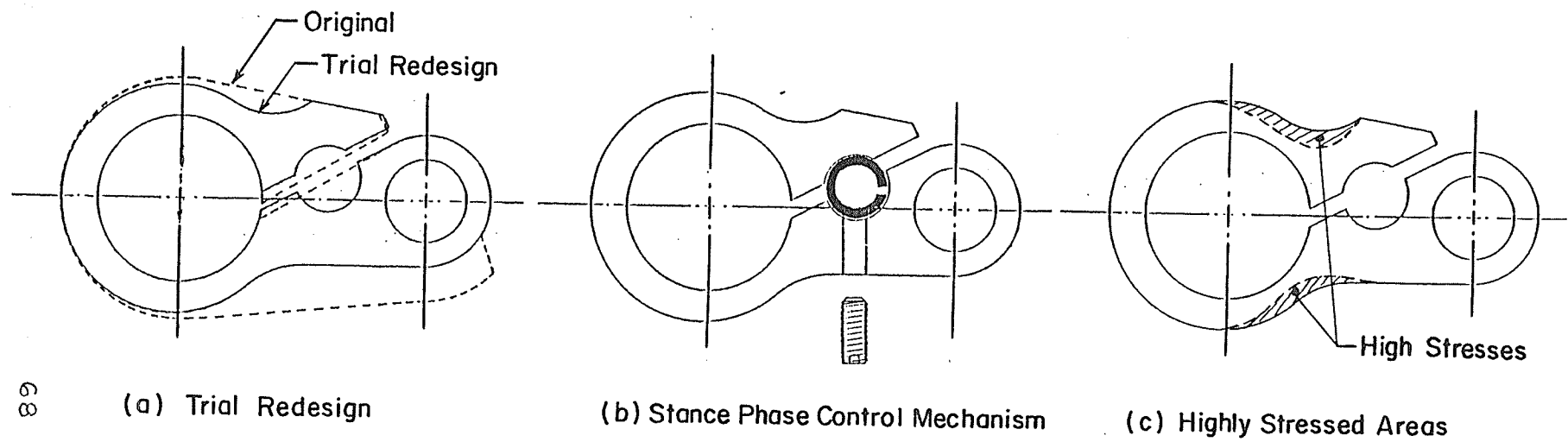


Figure 3.15

new stresses, which were about 20% to 30% higher than those for the original design, were due to excessive material removal from the web. The initial trials indicated that δ_{CR} generally increased by a greater percentage than did the stresses. However, the removal of some material is feasible although too much material had obviously been eliminated in the initial redesign.

The second iteration of the design process involved adding material to the initial design as shown in Figure 3.16. Finite element analyses of this new design indicated that δ_{CR} increased with respect to the existing swing block but was not quite as large as with the first trial design. The additional material reduced stresses with respect to the first design iteration, although they were still about 15-25% greater than the maximum stresses in the original swing block. The highest stresses were in area 1 of Figure 3.16 and they changed by under five percent with the additional material. This indicated that more material was required.

Based on these results, the design of Figure 3.16 was modified as shown in Figure 3.17. The finite element results for this design indicated a slight decrease of δ_{CR} with respect to the previous design. However, δ_{CR} was still greater than that for the original swing block. The stresses in area 2 of Figure 3.17 were within about 10% of the maximum stresses in the existing swing block. However, the stresses in area 1 continued to be of primary concern

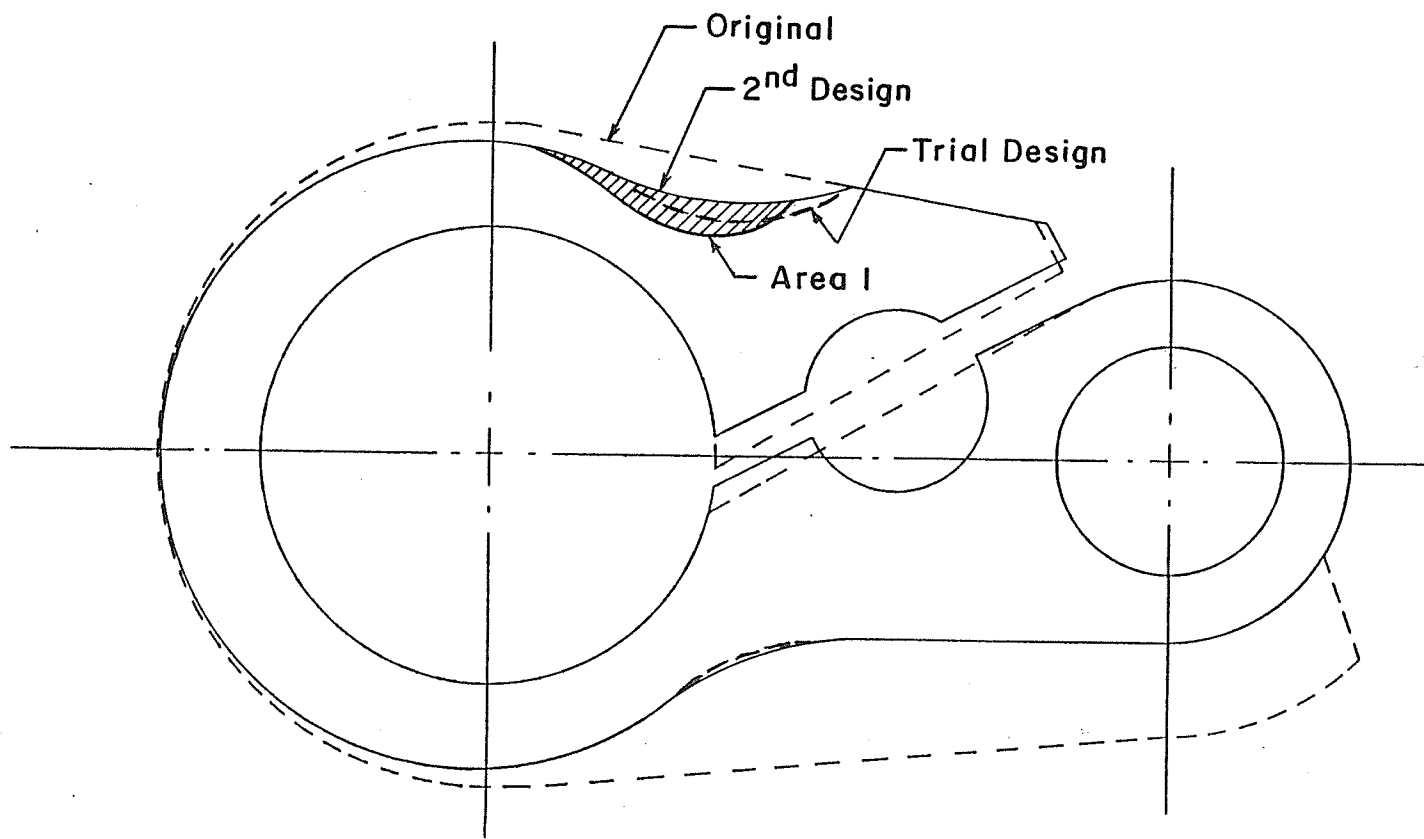


Figure 3.16 : Second Design Iteration

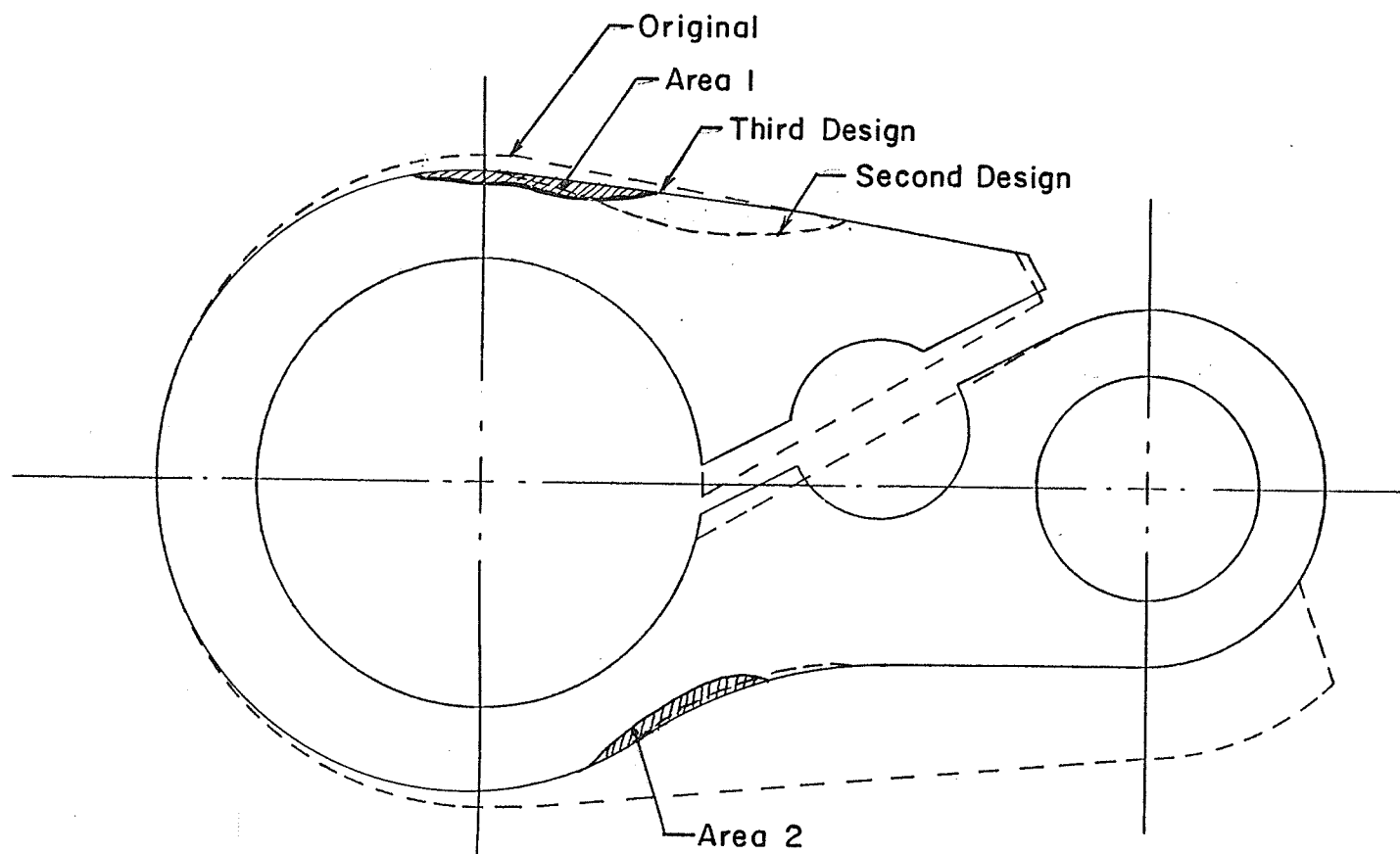


Figure 3.17 : Third Design Iteration

because the additional material had reduced stresses only marginally (less than 5%). This suggested, again, that more material was required in area 1 in particular and, to a lesser extent, in area 2.

After several iterations of successively adding material to area 1 only, it was found that any material removal from this region with respect to the existing swing block tended to increase the maximum stresses significantly whereas δ_{CR} was increased only marginally. It was decided, therefore, to leave this section of the swing block unchanged.

About eight combinations of the blended radii R_1 and R_2 shown in Figure 3.18 were investigated before arriving at the configuration illustrated. Both the stresses and δ_{CR} were found to be sensitive to small additions or deletions of material caused by varying R_1 and R_2 . A change in R_1 of as little as 0.2mm could affect values for the maximum stresses and δ_{CR} by up to 10%. The finite element analysis of the design of Figure 3.18 indicated that the maximum stresses were kept within about three percent of those in the existing swing block for identical conditions while δ_{CR} was found to increase by an average of about 2.5mm. However, a problem regarding deflections was discovered when radial clearances greater than about 0.09mm between the swing block and axis pin were encountered. As shown in Figure 3.19, the round hole for the split ring can deform

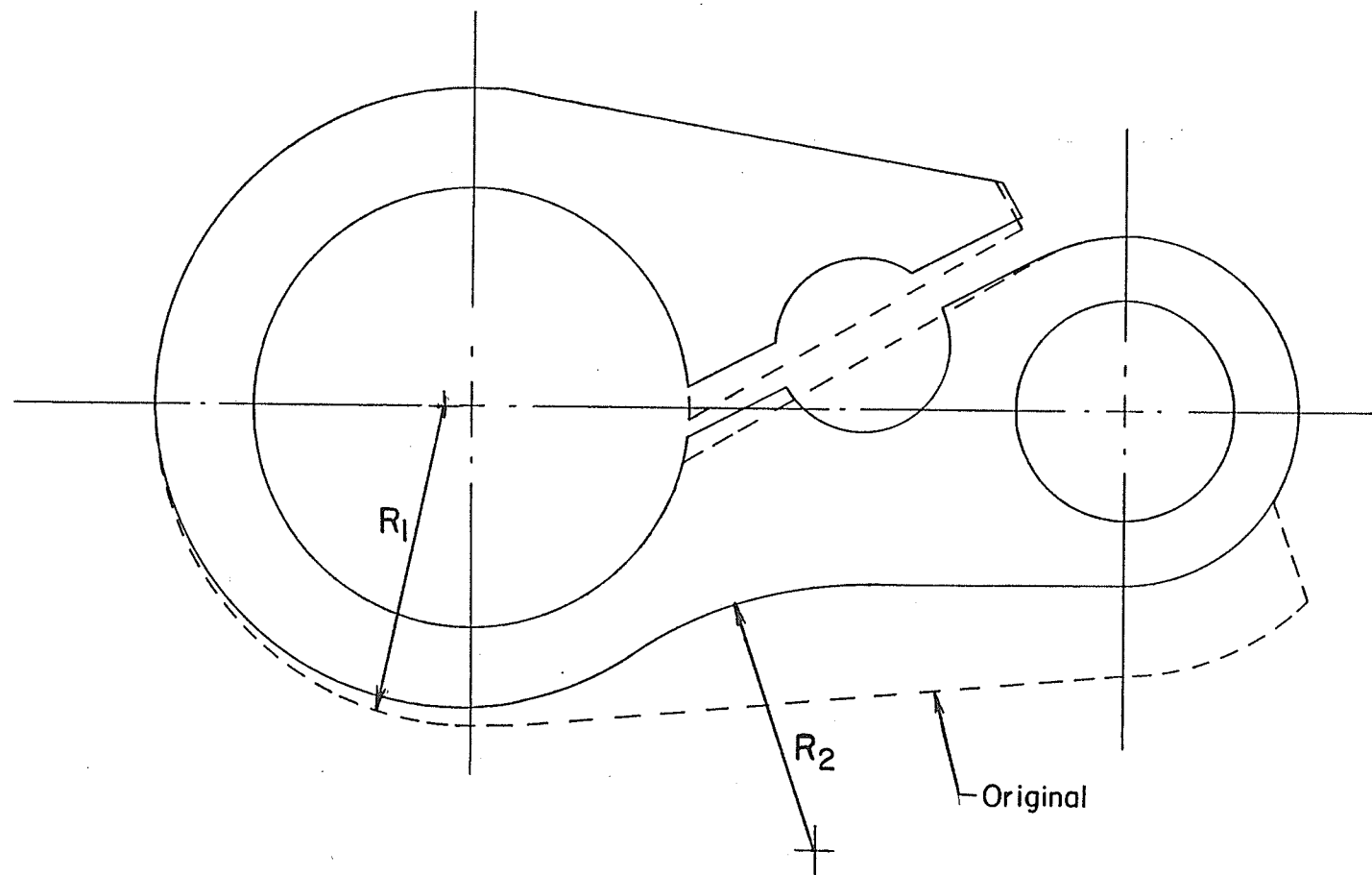


Figure 3.18 : Possible Design

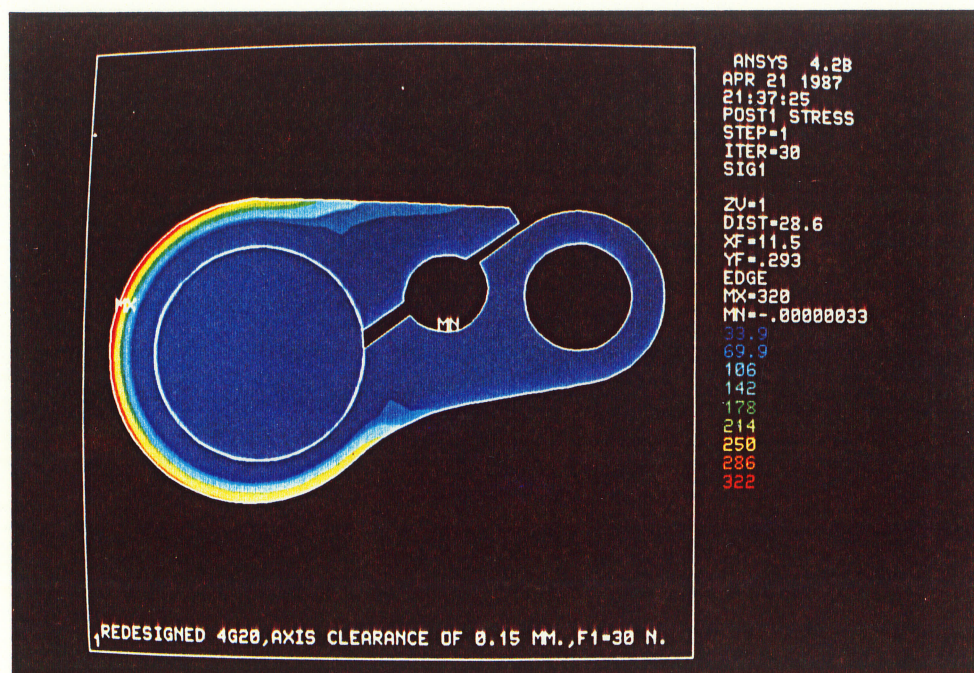


Figure 3.19 : Excessive Deformation of the Hole

THE QUALITY OF THIS MICROFICHE
IS HEAVILY DEPENDENT UPON THE
QUALITY OF THE THESIS SUBMITTED
FOR MICROFILMING.

UNFORTUNATELY THE COLOURED
ILLUSTRATIONS OF THIS THESIS
CAN ONLY YIELD DIFFERENT TONES
OF GREY.

LA QUALITE DE CETTE MICROFICHE
DEPEND GRANDEMENT DE LA QUALITE DE LA
THESE SOUMISE AU MICROFILMAGE.

MALHEUREUSEMENT, LES DIFFERENTES
ILLUSTRATIONS EN COULEURS DE CETTE
THESE NE PEUVENT DONNER QUE DES
TEINTES DE GRIS.

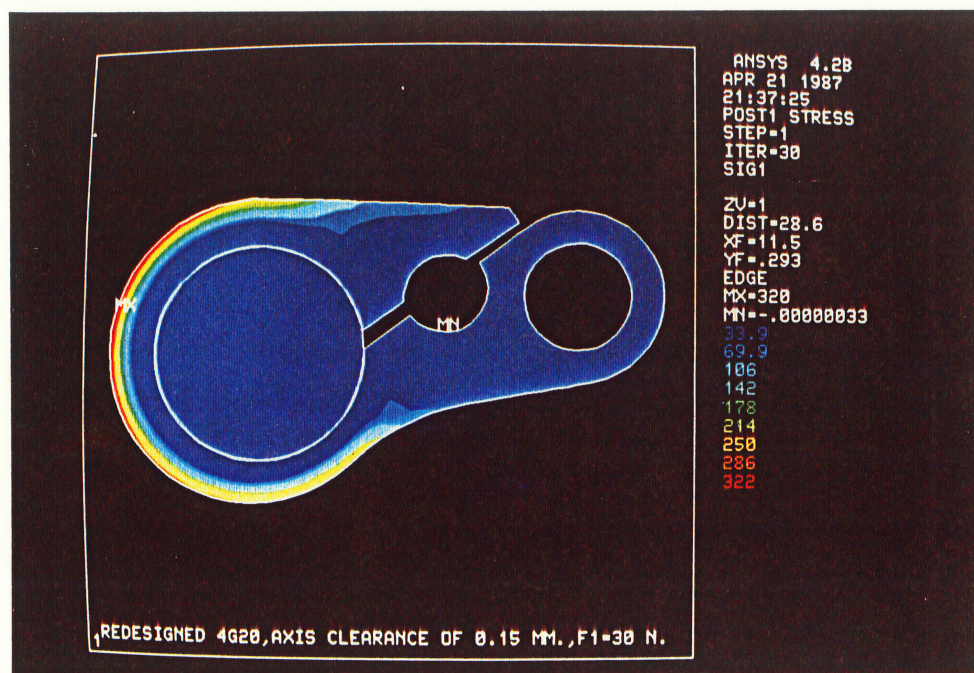


Figure 3.19 : Excessive Deformation of the Hole

greatly under a load of even 750N. which could result in permanent deformation or even damage of the split ring. This approach was discarded, therefore, in favour of the design shown in Figure 3.20.

The redesigned swing block of Figure 3.20 reverts to using a down-sized version of the original compression spring mechanism for the stance phase control adjustment. As well, the web geometry was modified to give a more uniform stress distribution. From Figure 3.19, it can be seen that the highest stresses occurred where the bottom section of the web blends into the main body of the swing block. The lowest stresses in the web are in the region adjacent to the highest stresses. The redesign shown in Figure 3.20 was introduced to more uniformly distribute stresses in the swing block.

3.3.2 The Final Design and Its Analysis

Figure 3.20 is the finalized redesign. Over a dozen intermediate design iterations were performed to arrive at the given geometry. Various combinations of blended radii defining the outside of the web region were analyzed. Both δ_{CR} and the stress distribution were very sensitive to even small changes in the web thickness. The final iterations involved changing the web thickness by as little as 0.1mm to ensure that the stresses were distributed reasonably

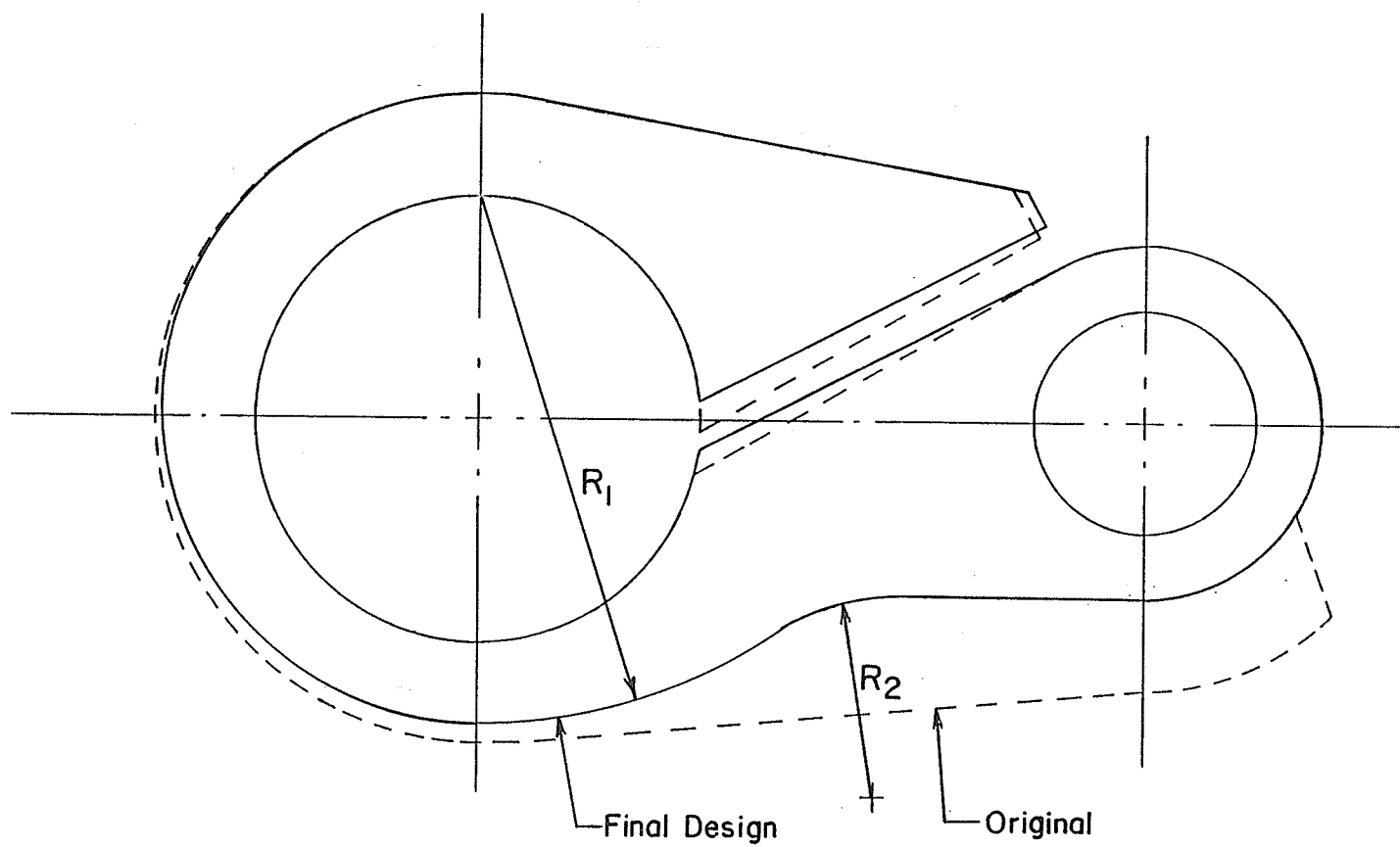


Figure 3.20 : Final Redesigned Swing Block

uniformly and that they would not exceed present maximum values.

For comparison, the new design was analyzed for the same series of loading and radial clearances as performed for the existing swing block. The results of these analyses are summarized in Figures 3.6 to 3.13 and stress distributions are shown in Figures D.1 to D.17 of Appendix D. The format for these figures is identical to that used in Section 3.2. Graphical results for the new design have been overlaid on the same curves presented previously for the existing swing block. The curves for the new design are marked with a "o" while the existing swing block is denoted by a 'x'. (Curves denoted by a ' Δ ' or a ' $\square$ ' will be discussed later in Section 3.5.)

The new maximum stresses plotted in Appendix D indicate that they, like the analogous values for the original design, tend to occur at the top and bottom sections of the web where the web blends into the main body of the swing block. Again, the stresses at the top and bottom of the web are always very nearly equal although the absolute maximum stress (labelled 'MX') is generally located at the top of the web. On the other hand, the absolute maximum stress is generally at the bottom of the web for the existing design. This difference is of little concern, however, because a variation of the stresses by only one or two percent would

move the point of maximum stress from the top of the web to the bottom.

Figures 3.7(a) and 3.11(a) provide a graphical comparison of the maximum principal stresses for the final and existing steel designs. It can be seen that, regardless of the radial clearance or the load, the maximum stresses are comparable. Although the largest discrepancies occur for extreme values of radial clearance or body weight, the differences are small, representing at most a variation of about 30 MPa.

Figures 3.6, 3.8, 3.10 and 3.12 show that the final redesign produces somewhat larger total contact forces and contact areas compared to those of the original design. The increase is generally most apparent in Figure 3.6 at the larger radial clearances, R . This trend with increasing R is reasonable because the swing block has to flex more to make contact as the radial clearance grows. Consequently, the increased flexibility of the final design would have less effect for small clearances. This behaviour is reflected in Figures 3.6 and 3.8 where it can be seen that separation of the the curves for the final and existing designs reduces as the radial clearance decreases.

Figure 3.9 and 3.13 show that the critical offset distance for the final design is beneficially increased, relative to that of the original swing block, for all combinations of applied load and radial clearances. The

largest percentage improvement in δ_{CR} is noted for light loads between about 250N to 500N and for radial clearances greater than 0.07mm. For example, with respect to Figure 3.13, δ_{CR} increases by about 40% for a weight of 250N but by only 10% for a weight of 1000N. The actual increase in δ_{CR} is fairly constant, however, and is typically in the range of 2.0 to 3.0mm for most cases. Therefore, the trends generally follow those of the existing swing block which is expected because the general mechanical functioning of the knee has not been altered.

3.4 A Comparison of Titanium and Steel

The intent of this thesis is to improve the Otto Bock model 3R15 steel knee joint by modifying the geometry of the swing block to increase its flexibility. The possibility of changing materials to improve the function of the knee has not been considered yet. However, Otto Bock recently introduced the model 3R35 knee joint which is geometrically similar to the model 3R15 but it has a swing block made of titanium rather than steel. Consequently, it was decided to investigate the merits of such a material substitution.

The titanium swing block offers two distinct advantages over the original steel version. First, the density of titanium is only 56% that of steel which results in a much lighter block. More important, however, is the fact that

titanium's elastic modulus is only 52% that of steel. The titanium swing block, therefore, should be much more flexible than the steel version which should result in an improved critical offset distance and, hence, a "safer" knee. The cost of the titanium knee, however, is much higher than that of the steel version. A typical titanium casting for a knee component costs up to six times that of the equivalent steel casting.

For the purposes of comparison, it was decided to analyze both the original and the final redesigned swing blocks when using titanium. Identical finite element models to the steel versions were employed and the same series of loading and radial clearances were generally investigated. However, the two largest loads and the three largest radial clearances were not run to save computer costs. These latter cases are well outside the normal range of function of the knee. The only change in the original ANSYS input files was to change the elastic modulus to correspond to that of titanium.

Results of the titanium analyses are summarized in Figures 3.6 through 3.13, Figures E.1 through E.12 given in Appendix E, and Figures F.1 through F.12 presented in Appendix F. The presentation is identical to that used previously. The distribution of the stresses (see Appendices E and F) in the titanium versions of the existing and redesigned swing blocks are similar to the corresponding

steel ones. However, the maximum stresses shown in Figure 3.7 and 3.11 for the titanium models are 30-40% lower than those in the analogous steel models. This reduction can be attributed to the higher flexibility of titanium (compared to steel) which, in turn, results in greater contact between the swing block and axis pin. The larger contact (illustrated by the curves of Figures 3.8 and 3.12) leads to smaller bending stresses in the web.

The critical offset distance is enhanced greatly by the material switch to titanium. As illustrated in Figure 3.9 and 3.13, δ_{CR} for the titanium models is greater for all combinations of loading and radial clearances than for the equivalent steel models. The improvement is especially noticeable for loads between 250N to 500N and at radial clearances greater than about 0.05mm. It is attributable, again, to the greater flexibility of titanium. Thus, the titanium version of the existing swing block is functionally superior to the steel version for both the original and redesigned swing blocks. On the whole, however, the titanium version of the redesigned swing block provides the greatest degree of safety of any of the designs investigated. This is evidenced by the curves of Figures 3.9 and 3.13 where it can be seen that δ_{CR} is always greatest for the redesigned titanium swing block. Consequently, the use of titanium for the swing block is

seen to be an excellent choice of material providing the additional cost can be justified.

3.5 Fatigue Testing

Although a finite element analysis of the redesigned swing block has been performed, physical testing is always required as the final proof of success or failure of a new design. Therefore, both the existing 3R15 knee and a knee using the redesigned steel swing block were subjected to classical fatigue tests [9] using the apparatus illustrated in Figure 3.21. The tests simulated the same geometric conditions of loading used in the finite element analysis. The prototype of the redesigned swing block was machined from a heat-treated cast block of the same material as that used in the production swing block.

Both artificial knees were tested for a total load of 1500N. and the radial clearance was invariably 0.05mm. The load was applied and removed at a rate of three cycles per second and this procedure was continued until a total of 1.5 million cycles was completed for each knee. If a steel specimen does not fail during the first million cycles at a given load, then it should survive indefinitely, in all likelihood, at that load [9]. The 1.5×10^6 cycles of loading should be sufficient, therefore, to establish whether the swing block will fail at the given load.

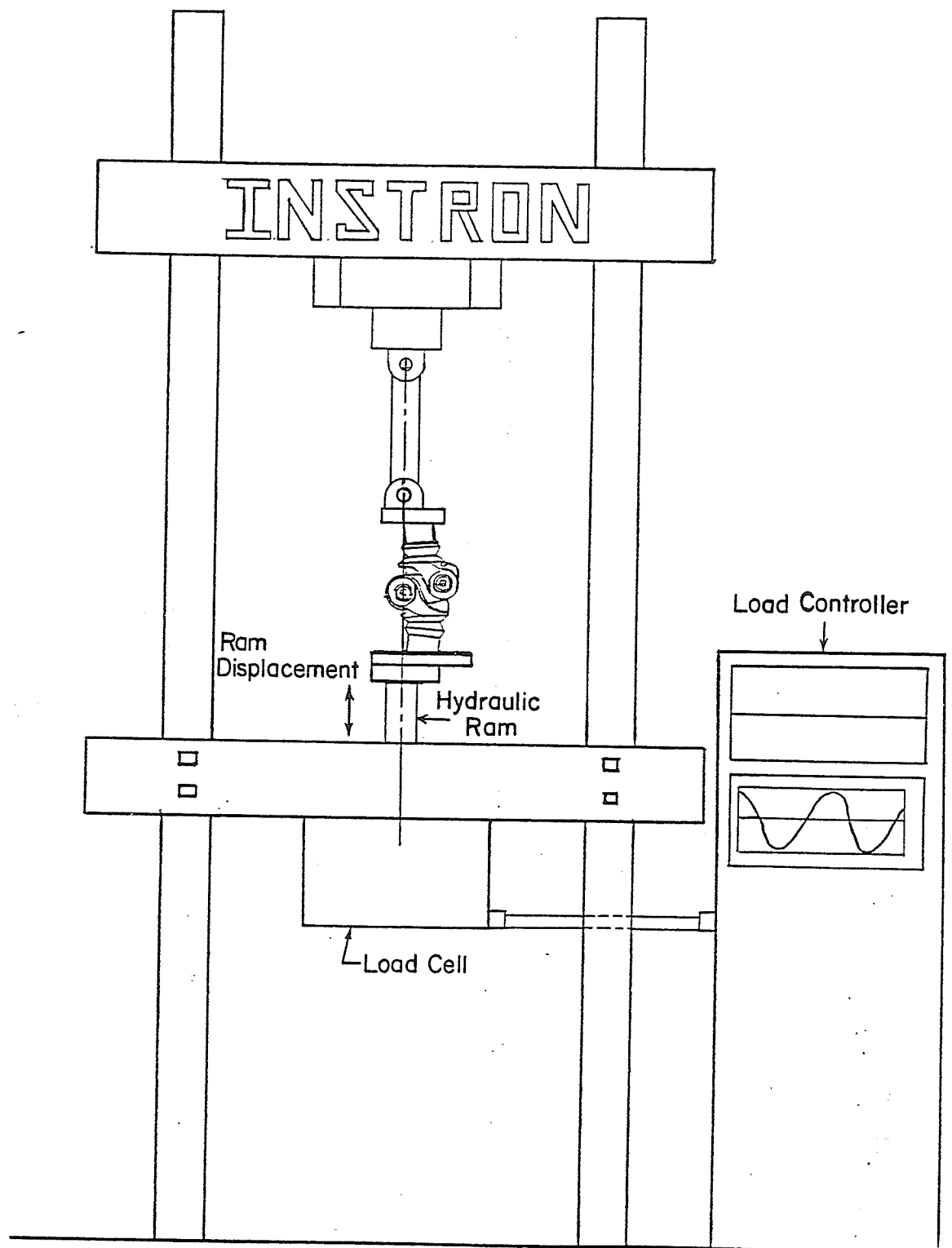


Figure 3.21 : Fatigue Test Apparatus

The results of the tests were good. As expected, neither the existing nor the redesigned swing block failed. The results tend to validate the finite element results which predicted that maximum stress would be below the fatigue limit of the steel. A fatigue test of a single specimen is certainly not conclusive, however, and a number of specimens (at least 5 to 10) should be tested [9]. However, financial constraints precluded the testing of more than one specimen in this thesis. Further physical testing would be recommended before actually adopting the redesigned swing block.

CHAPTER 4

4.1 Conclusions and Recommendations

- (1) The redesigned steel swing block detailed in Chapter 3 significantly improves the functional safety of the Otto Bock model 3R15 knee joint. The improvement is especially noticeable for loads under 500N and for radial clearances over about 0.03mm between the swing block and the axis pin. It is recommended, however, that the new design be subjected to further fatigue testing and clinical evaluation before replacing the existing swing block.
- (2) The finite element program ANSYS proved to be a valuable design tool. The iterative approach of analyzing and redesigning the swing block resulted in an improved design. However, this procedure was time consuming during the final stages of refinement. It is possible that an automated process could have been employed during the final stages although the iterative approach is still felt to be the most expedient initially. This is because all factors affecting a new design cannot usually be anticipated until the design is near finalization. For example, the too large deflections when using the split ring design of Section

3.3.1 were not anticipated during the initial design stage.

- (3) The use of titanium rather than steel for the swing block significantly improved the performance of the knee joint. The redesigned titanium swing block yielded the greatest improvement in δ_{CR} compared to the existing steel design. For example, δ_{CR} increased by about 50% for a load of 500N and a radial clearance of 0.05mm for the redesigned titanium swing block. The titanium version of the original swing block was also very good, with δ_{CR} increasing by around 35% for the same load and clearance conditions. It is not recommended, however, that a titanium version of the redesigned swing block be produced because further improvement is possible. The swing block was optimized under the assumption that it would be made of steel. When the final redesigned version using titanium was analyzed, it was found that the maximum stresses decreased considerably compared to the analogous steel version. Coupled with the greater fatigue strength of titanium, these lower stresses suggest that more material could be removed to arrive at a more optimal titanium swing block.

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APPENDIX "A" - STATIC LOADING OF THE SWING BLOCK

The present derivation of the equilibrium equations assume that the amputee is standing still, not walking. This assumption removes the horizontal ground reaction force component from the equations, as suggested by Figure A.2. Figure A.2 shows a free body diagram of only the body and upper fork of the 3R15 knee under the conditions given in Figure A.1. For moment and force equilibrium it is required that

$$F_2 = \frac{a+b}{b} F_1 \quad (A1)$$

and

$$F_3 = \frac{a}{b} F_1 \quad (A2)$$

where F_1 , F_2 and F_3 are the amplitudes of the corresponding vectors $\vec{F}_1$, $\vec{F}_2$ and $\vec{F}_3$ shown in Figure A.1. The figure also defines distances a and b .

To determine the forces applied to the swing block, consider the free body diagram presented in Figure A.3. Forces $\vec{F}_2$ and $\vec{F}_3$ are transferred from the upper fork to the swing block. Their magnitudes are given by equations (A1) and (A2) above. Force $\vec{F}_5$ is the load on the axis pin while $\vec{F}_4$ results from contact between the swing block and the extension stop. The $\vec{F}_4$ must be perpendicular to the swing block's surface as a result of a sliding contact condition. Written in vector notation, the forces are:

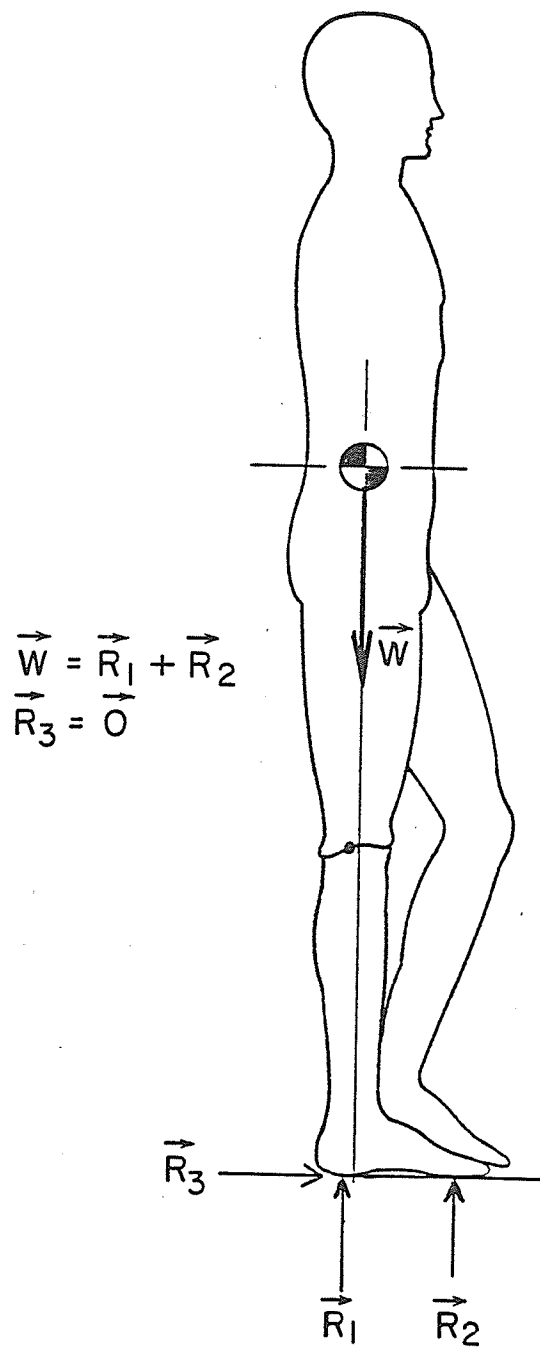


Figure A.1 : Standing Amputee

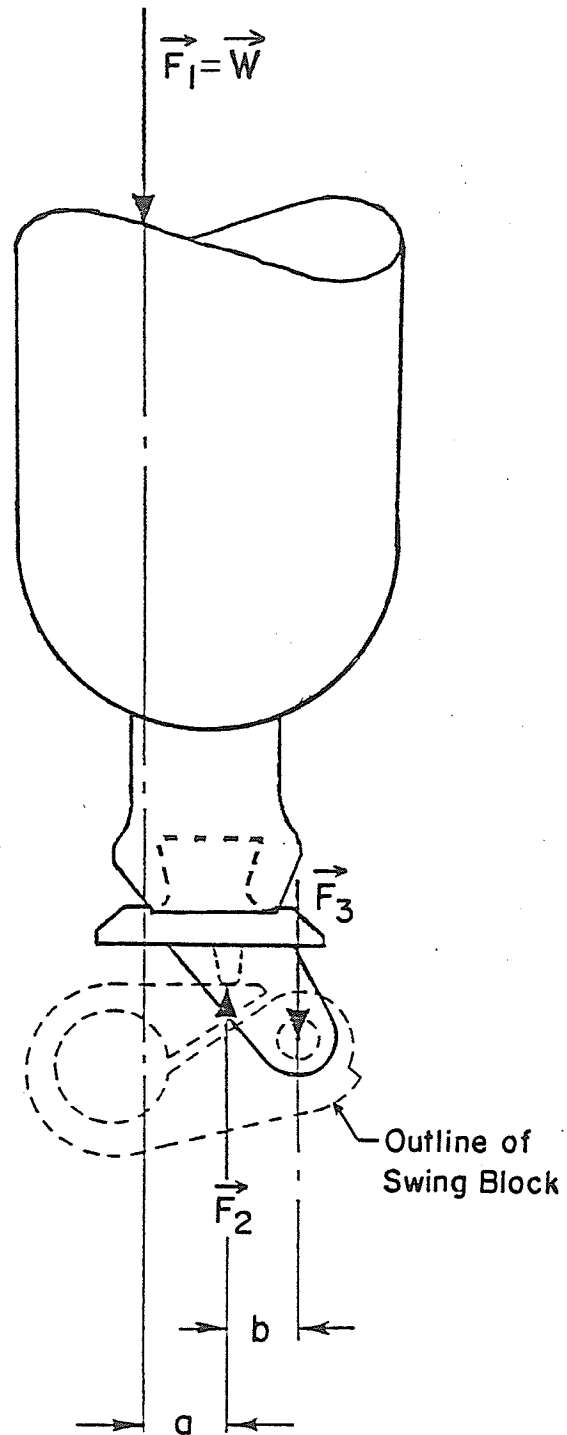


Figure A.2 : Free Body Diagram of the Upper Fork and Body

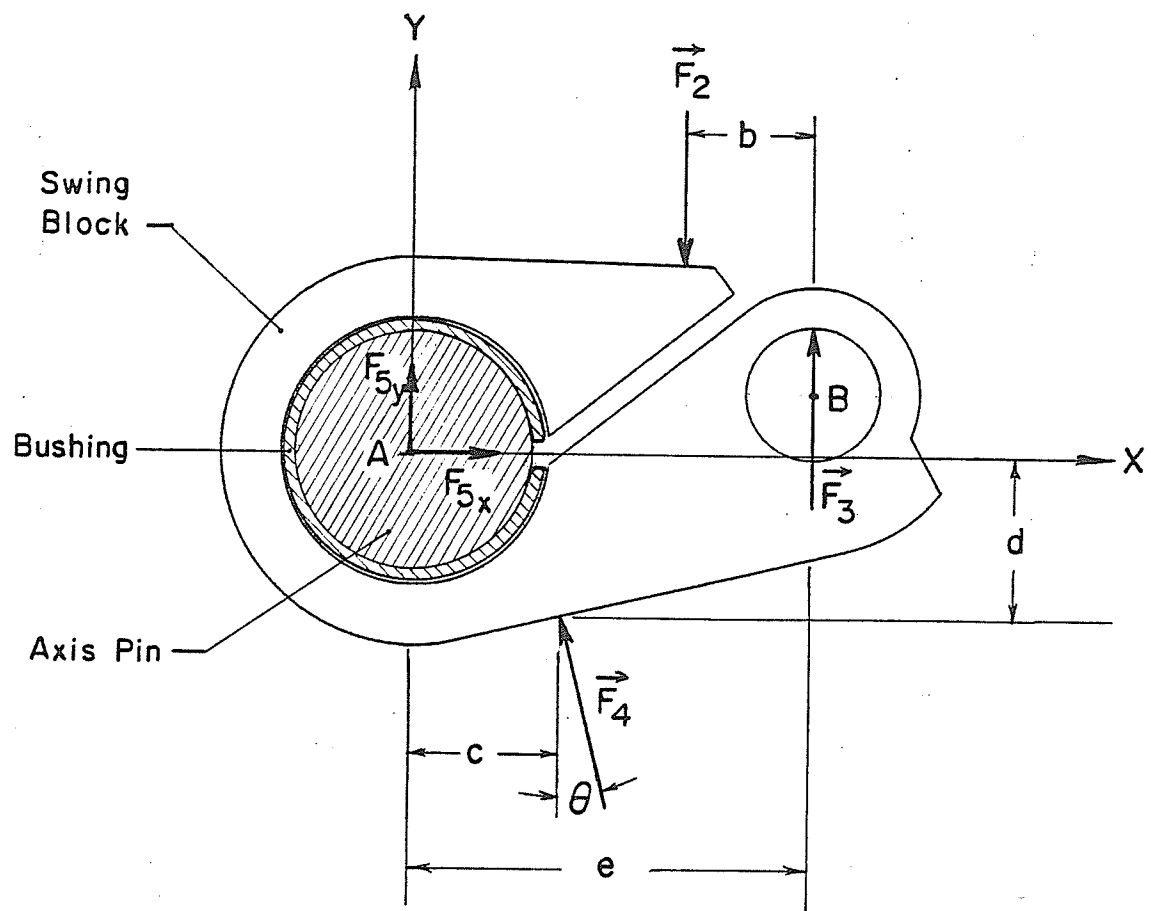


Figure A.3 : Free Body Diagram of the Swing Block with the Bushing and the Axis Pin

$$\vec{F}_2 = -\frac{(a+b)}{b} F_1 \hat{j} \quad (A3)$$

$$\vec{F}_3 = \frac{a}{b} F_1 \hat{j} \quad (A4)$$

$$\vec{F}_4 = -F_4 \sin\theta \hat{i} + F_4 \cos\theta \hat{j} \quad (A5)$$

$$\vec{F}_5 = F_{5x} \hat{i} + F_{5y} \hat{j} \quad (A6)$$

where $\hat{i}$ and $\hat{j}$ are unit magnitude vectors in the X and Y directions, respectively.

For moment equilibrium about point B of Figure A.3,

$$\sum M_B = 0 = [(c\hat{i}-d\hat{j}) \times (-F_4 \sin\theta \hat{i} + F_4 \cos\theta \hat{j})] + [(e-b)\hat{i} \times (-\frac{(a+b)}{b} F_1) \hat{j}] + [(e\hat{i} \times \frac{a}{b} F_1) \hat{j}]. \quad (A7)$$

Expanding and simplifying the last equation gives

$$F_4 = \frac{e - (a+b)}{c \cos\theta - d \sin\theta} F_1. \quad (A8)$$

For force equilibrium in the X direction

$$\sum F_x = 0 = -F_4 \sin\theta + F_{5x} \quad (A9)$$

or

$$F_{5x} = F_4 \sin\theta \quad (A10)$$

where F_{5x} is the resolved component of $\vec{F}_5$ in the X direction.

Combining equation (A8) and (A10) gives

$$F_{5x} = \frac{e - (a+b)}{c \cos\theta - d \sin\theta} F_1 \sin\theta. \quad (A11)$$

Similarly, for equilibrium in the Y-direction

$$\sum F_y = 0 = - \frac{a+b}{b} F_1 + \frac{a}{b} F_1 + F_4 \cos\theta + F_{5y} . \quad (A12)$$

By combining equations (A8) and A12) and simplifying,

$$F_{5y} = \left[1 - \frac{e - (a+b)}{c \cos\theta - d \sin\theta} \cos\theta \right] F_1 . \quad (A13)$$

(It should be reiterated that equations (A1) through (A13) are valid only under the assumption that a static body weight is applied to the prosthesis so that there is no horizontal component to the ground reaction force.)

The second assumption that will be made is that the knee joint is aligned so that the line of action of the body weight passes through the centre of rotation of the knee joint (point A) in Figure A.3. This alignment is within the recommended range for a safety knee. In a non-locking knee, this alignment would be the point of imminent instability. For the present case, this assumption requires that

$$a + b = e . \quad (A14)$$

As well, it is known from the physical geometry of the knee that

$$b \approx 9.535\text{mm} \quad (A15)$$

$$\text{and} \quad e \approx 29.544\text{mm} \quad (A16.a)$$

so that from equation (A14)

$$a \approx 20.009\text{mm} . \quad (A16.b)$$

By substituting equations (A14) through (A16) into equations (A3) through (A6), (A8), (A11) and (A13), it can be shown that approximately

$$\vec{F}_2 = -3.098 F_1 \hat{j} \quad (A16)$$

$$\vec{F}_3 = 2.098 F_1 \hat{j} \quad (A17)$$

$$\vec{F}_4 = \vec{0} \quad (A18)$$

and

$$\vec{F}_5 = F_1 \hat{j} . \quad (A19)$$

Equations (A16) through (A19) give the loads on the swing block in terms of the body weight on the prosthesis under the assumptions that: (i) the horizontal ground reaction force component is zero; and (ii) the line of the body weight passes through the centre of rotation of the knee joint.

APPENDIX B - Sample Input File to ANSYS

```
/TITLE,ORIGINAL 4G20,INSTABILITY ANALYSIS
ALBC,1
/SHOW,9600,110,1,1
/MDIV,5
KAN,0
KNL,0
ET,1,42,0,0,2
R,1,0
EX,1,2.05E5
NUXY,0.33
KAY,6,0
```

```
LOCAL,12,1,0,0,0,10.0
CSYS,12
N,1,10.0,-2.6843
N,48,10.0,-14.415
N,24,10.0,167.745357
FILL,1,24,
NGEN,2,23,2,24,1,0.0,170.429657
N,49,11.125,0.967429
N,96,11.125,-8.99435
KNOD,7,49
KNOD,6,96
NGEN,2,48,2,47,1,1.125,3.651729
KNOD,2,61
KNOD,3,73
KNOD,4,85
KNOD,5,88
KNOD,8,60
KNOD,30,57
E,50,2,1,49
EGEN,47,1,1,1,1,
```

```
LOCAL,11,0,0.0,0.0,0.0,10.0
K,1,0.0,0.0
K,9,0.0,14.5
K,10,-14.5,0.0
K,11,0.0,-14.5
K,12,5.0,-14.5
K,13,15.3189,0.7609
K,14,15.5251,2.7275
K,15,1.32298,14.5
K,16,23.9299,10.5138
K,17,25.2,8.31384
K,18,26.0,6.9282
K,19,30.0,8.0
K,20,38.0,0.0
K,21,38.4296,-8.5406
K,22,31.0459,-11.9543
K,23,36.8348,-4.15926
```

K,24,30.0,-4.95
K,25,25.05,0.0
K,26,27.5250,4.28683
K,27,30.0,4.95
K,28,34.95,0.0
K,29,30.0,0.0
K,31,8.5850,13.2201

CSYS,0
KMODIF,27,29.5442,10.1594

LARC,9,10,1,14.5
LARC,10,11,1,14.5
L,11,12
L,12,22
LARC,22,21,29,12.0
L,23,21
LARC,23,20,29,8.0
LARC,20,19,29,8.0
LARC,19,18,29,8.0
L,18,13
L,13,6
LARC,6,5,1,11.125,8
LARC,5,4,1,11.125,3
LARC,4,3,1,11.125,12
LARC,3,2,1,11.125,12
LARC,2,8,1,11.125,1
LARC,8,30,1,11.125,3
LARC,30,7,1,11.125,8
L,7,14
L,14,17
L,17,16
L,16,31
L,31,15
L,15,9
L,15,8
L,9,2
L,10,3
L,4,11
L,5,12
LARC,12,13,1,21.0
L,13,22
L,22,24
L,22,23
L,23,28
L,28,20
L,27,19
L,18,26
L,13,25
LARC,26,25,29,4.95
LARC,25,24,29,4.95

LARC,24,28,29,4.95
LARC,28,27,29,4.95
LARC,27,26,29,4.95
A,9,10,3,2
A,3,10,11,4
A,4,11,12,5
A,5,12,13,6
A,7,14,31,30
A,15,8,30,31
A,9,2,8,15
A,31,14,17,16
A,13,25,26,18
A,13,22,24,25
A,13,12,22,13
A,22,21,23,22
A,24,22,23,28
A,28,23,20,28
A,19,27,28,20
A,18,26,27,19

LMODIF,26,9,2,3
LMODIF,19,7,14,3
AMESH,1,7,1
ELSIZE,1.75,0,1

AMESH,8
ELSIZ,1.75,0,1
AMESH,9,16,1

CSYS,12
NGEN,2,800,1,48,1,-0.070
NGEN,2,50,801,848,2,-3.3
NGEN,2,50,851,898,4,-3.3
N,1000,0.0,0.0
E,851,801,802
EGEN,24,2,611
E,853,851,802
EGEN,23,2,635
E,853,802,803
EGEN,23,2,658

E,901,851,853
EGEN,12,4,681
E,905,853,855
EGEN,11,4,693
E,905,901,853
EGEN,11,4,704
E,1000,901,905
E,1000,905,909
E,1000,909,913
E,1000,913,917

E,1000,917,921
E,1000,921,925
E,1000,925,929
E,1000,929,933
E,1000,933,937
E,1000,937,941
E,1000,941,945
E,1000,945,901
E,901,897,851
E,901,945,897
E,851,848,801
E,851,897,848

ET,2,52,0,0,1
R,2,1.0E7,0.07,0.0
MU,2,0.0
TYPE,2
MAT,2
REAL,2
E,1,801
EGEN,48,1,731

CSYS,0
WSORT,X

/COM APPLY BOUNDARY CONDITIONS HERE

CSYS,0
D,505,UY,0.0
D,1000,ALL,0.0
CNVR,.01
f,298,fy,-92.9524
iter,-30,0,30
lwrite

LSRSEL,LINE,12,18
NLINE,1

NASEL,NODE,49,96
MERGE,.5
NALL
LSALL

AFWRIT

THE QUALITY OF THIS MICROFICHE
IS HEAVILY DEPENDENT UPON THE
QUALITY OF THE THESIS SUBMITTED
FOR MICROFILMING.

UNFORTUNATELY THE COLOURED
ILLUSTRATIONS OF THIS THESIS
CAN ONLY YIELD DIFFERENT TONES
OF GREY.

LA QUALITE DE CETTE MICROFICHE
DEPEND GRANDEMENT DE LA QUALITE DE LA
THESE SOUMISE AU MICROFILMAGE.

MALHEUREUSEMENT, LES DIFFERENTES
ILLUSTRATIONS EN COULEURS DE CETTE
THESE NE PEUVENT DONNER QUE DES
TEINTES DE GRIS.

Appendix C: Equivalent and Principal Stress Distributions For The Original Steel Swing Block.

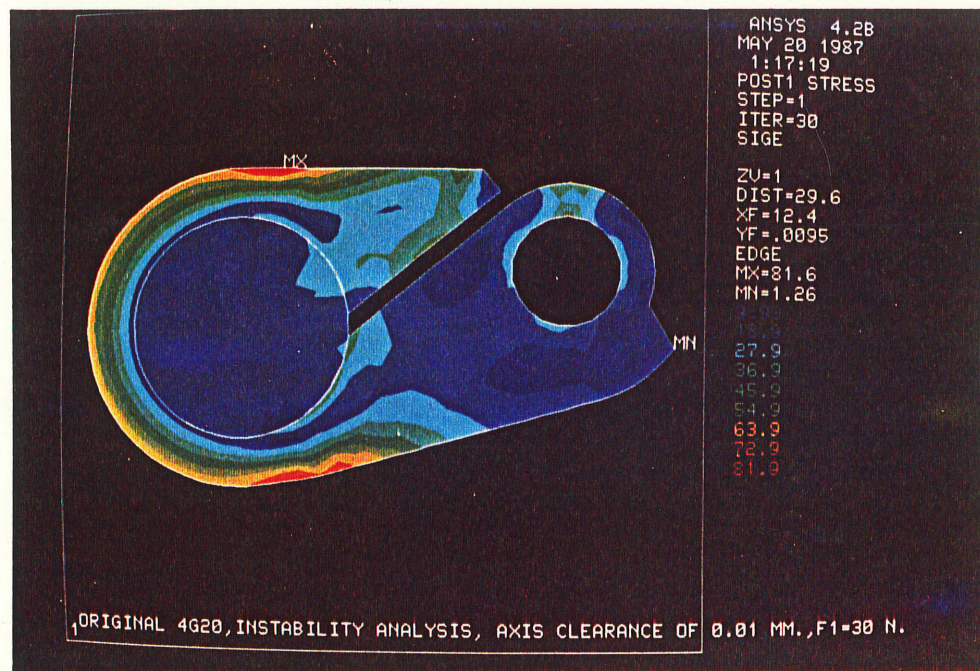


Figure C.1(a): Equivalent Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

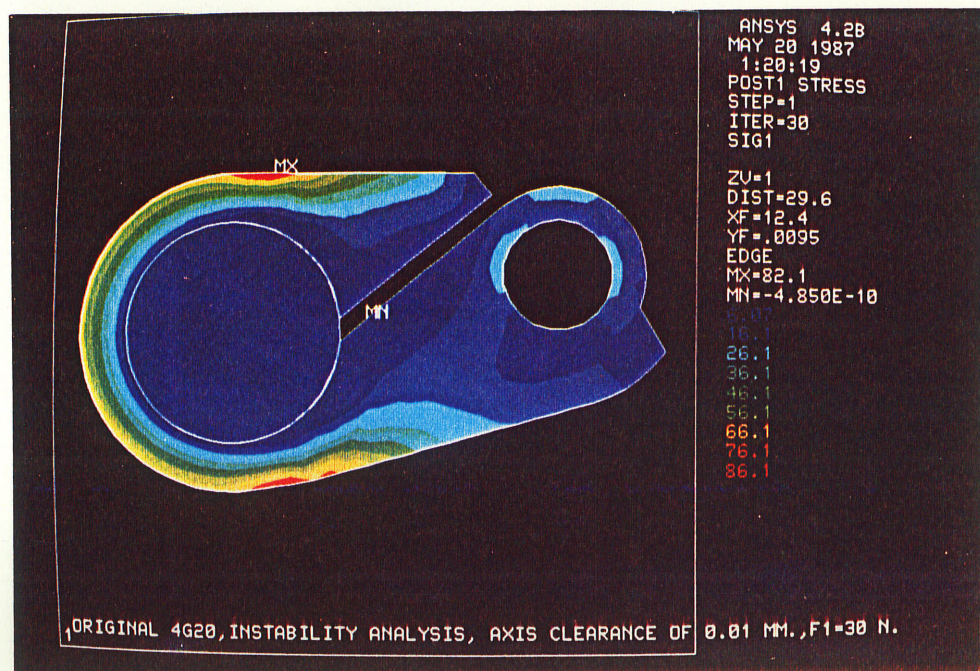


Figure C.1(b): Principal Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

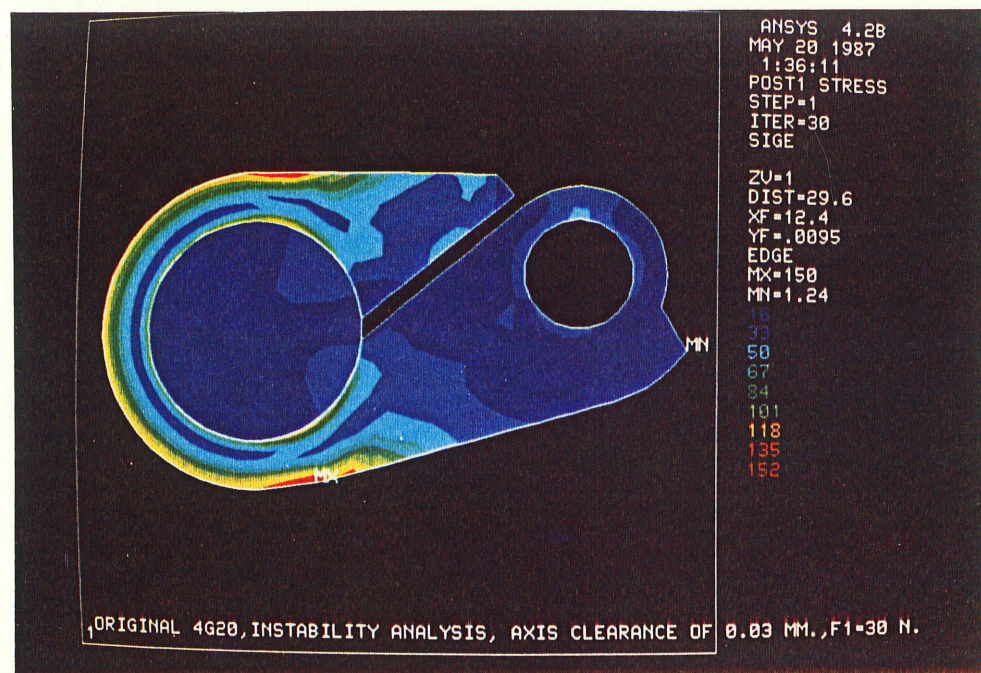


Figure C.2(a): Equivalent Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

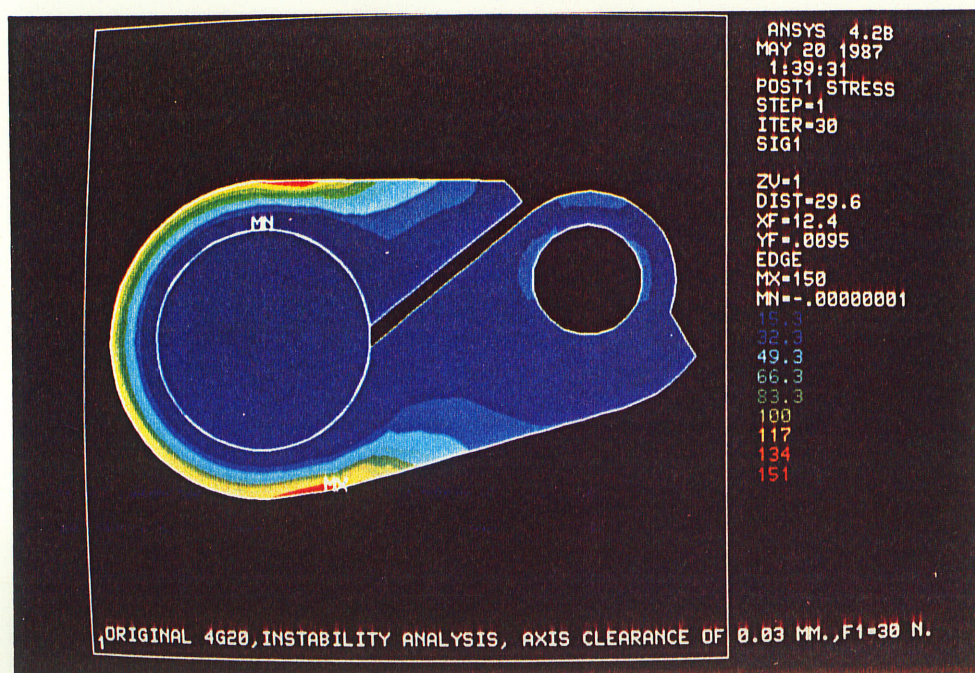


Figure C.2(b): Principal Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

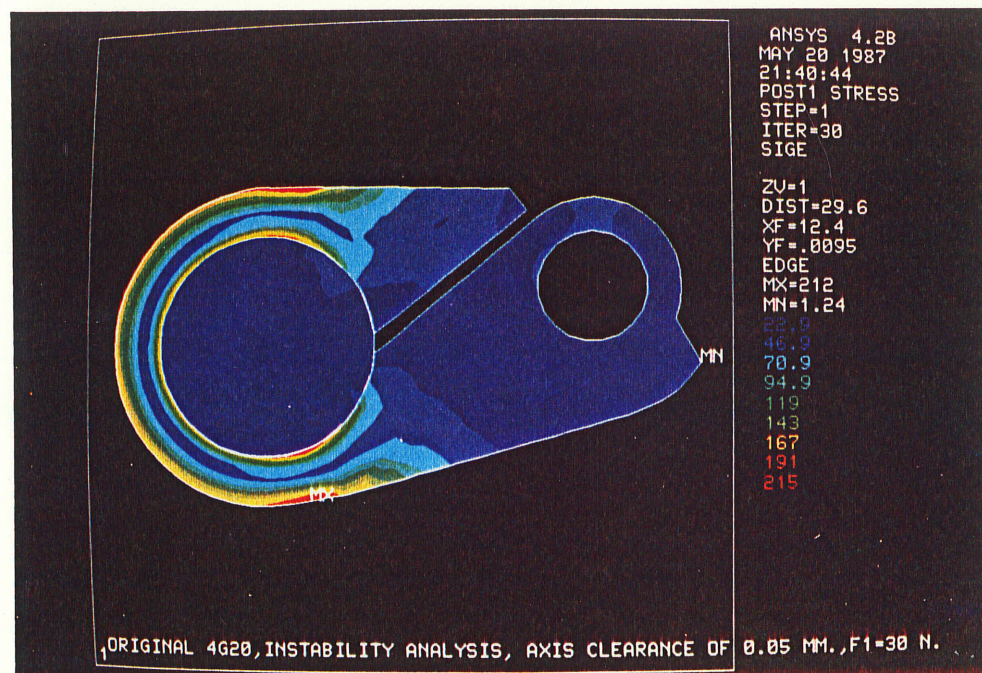


Figure C.3(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

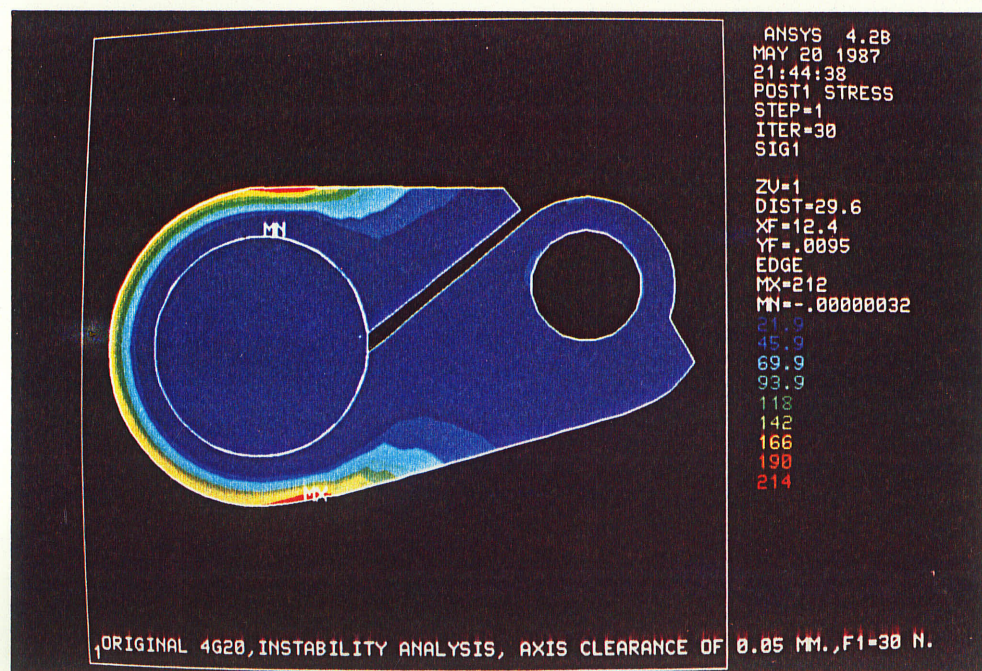


Figure C.3(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

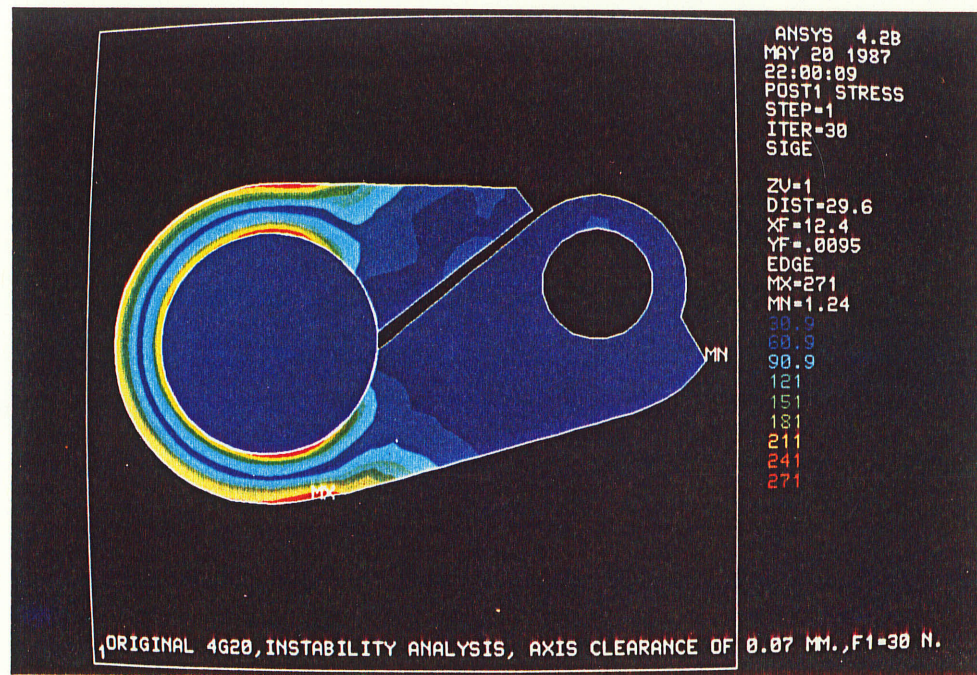


Figure C.4(a): Equivalent Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

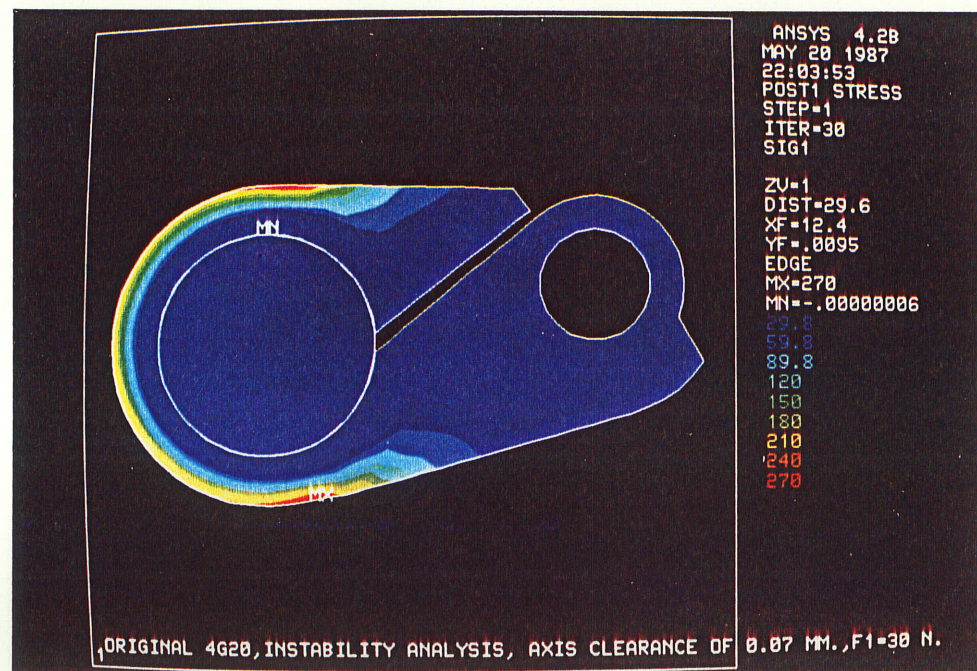


Figure C.4(b): Principal Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

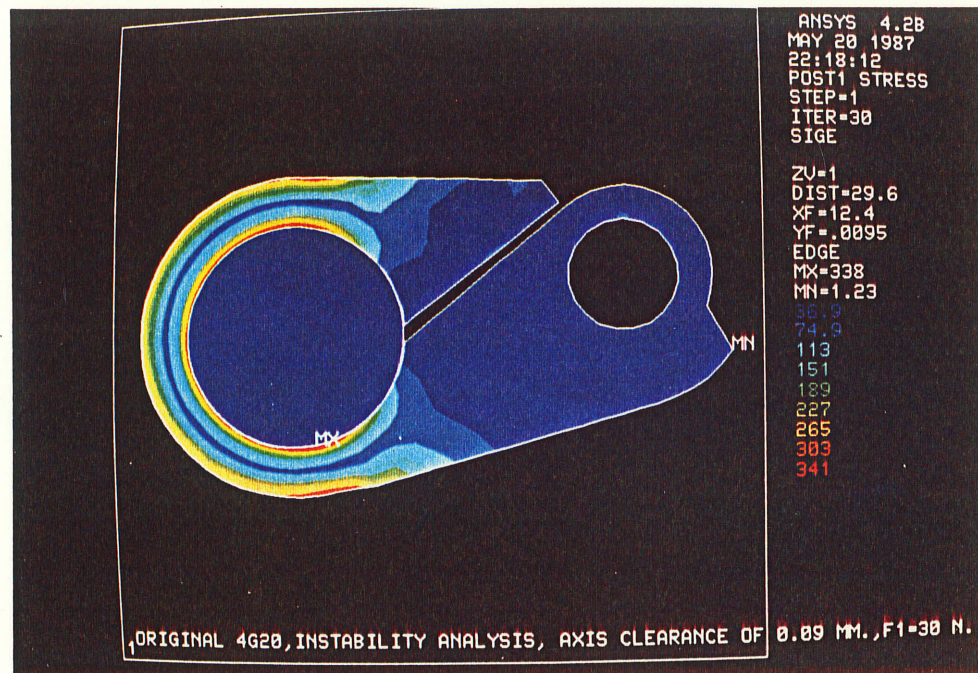


Figure C.5(a): Equivalent Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

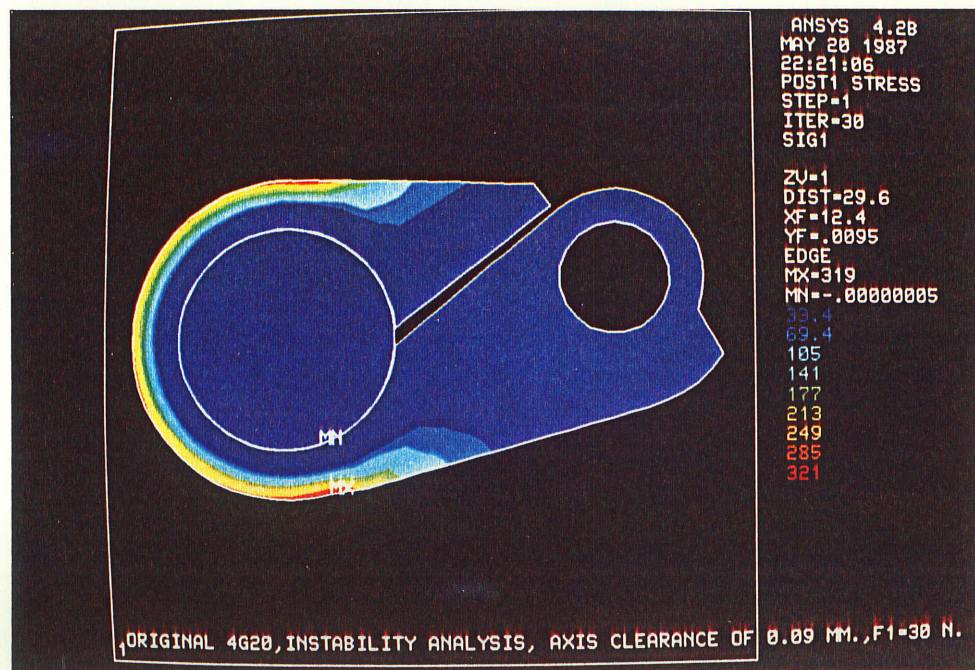


Figure C.5(b): Principal Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

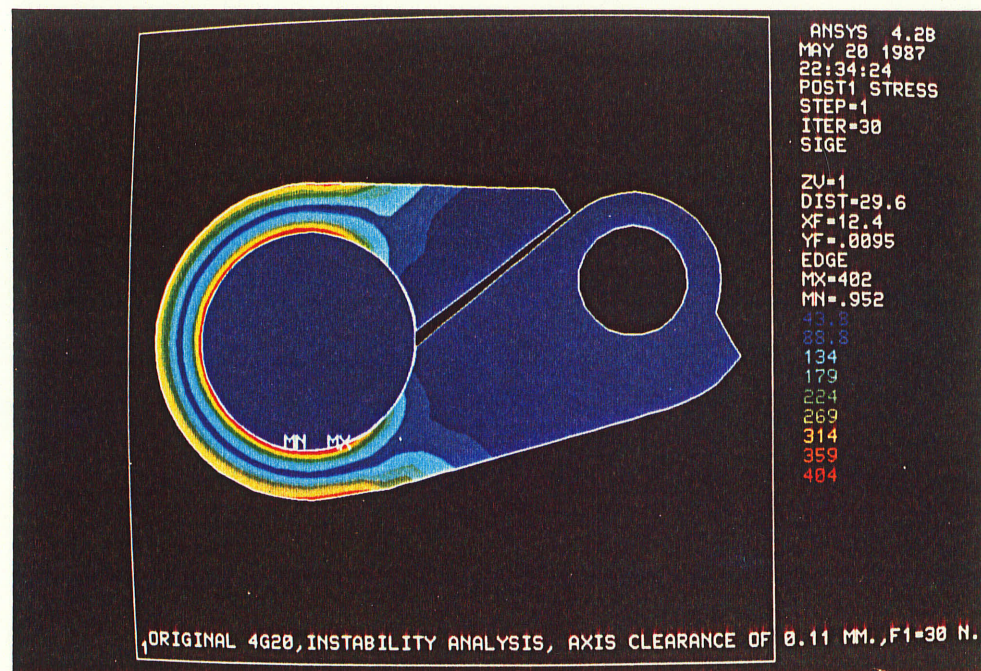


Figure C.6(a): Equivalent Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

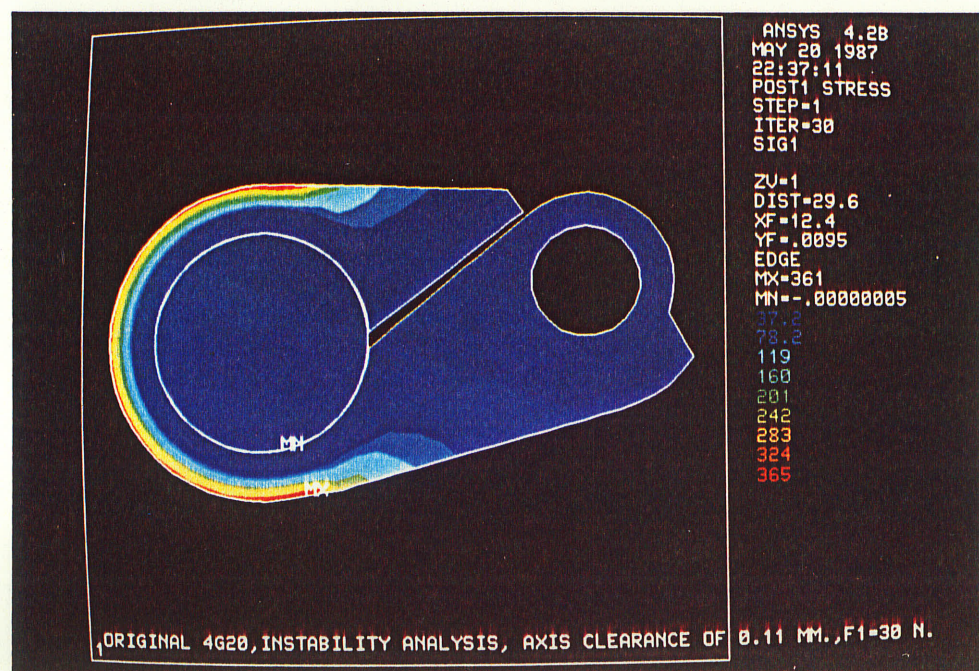


Figure C.6(b): Principal Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

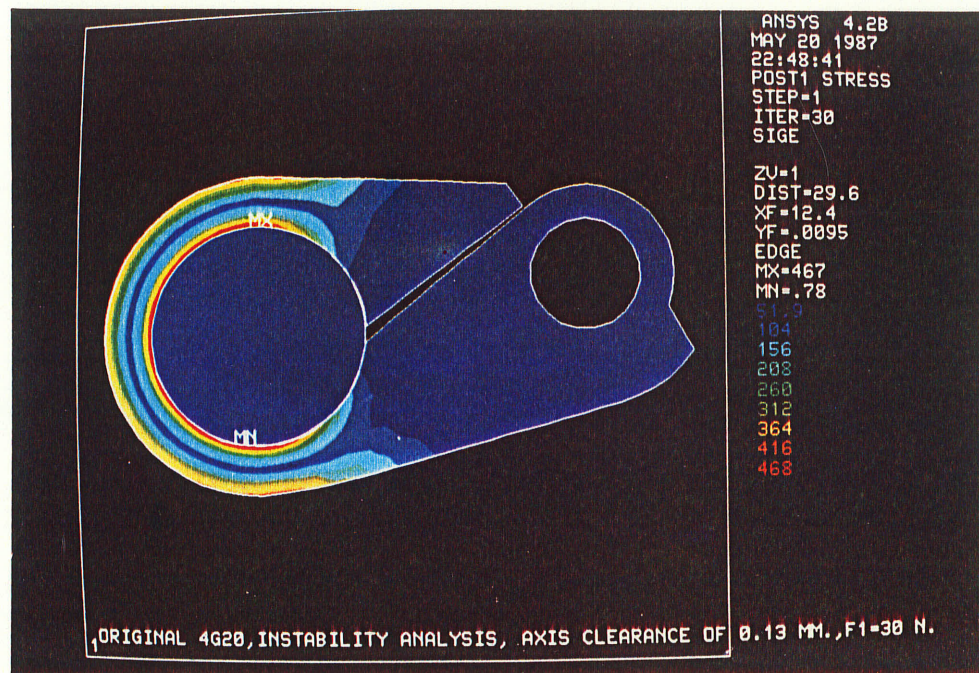


Figure C.7(a): Equivalent Stress Distribution, $R=0.13\text{mm}$, $W=750\text{N}$.

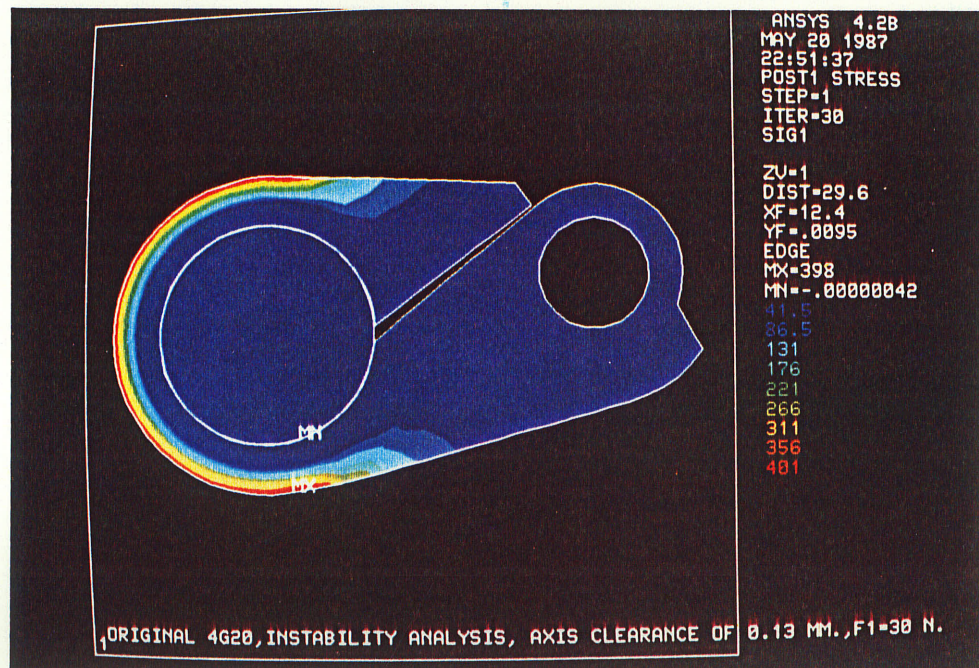


Figure C.7(b): Principal Stress Distribution, $R=0.13\text{mm}$, $W=750\text{N}$.

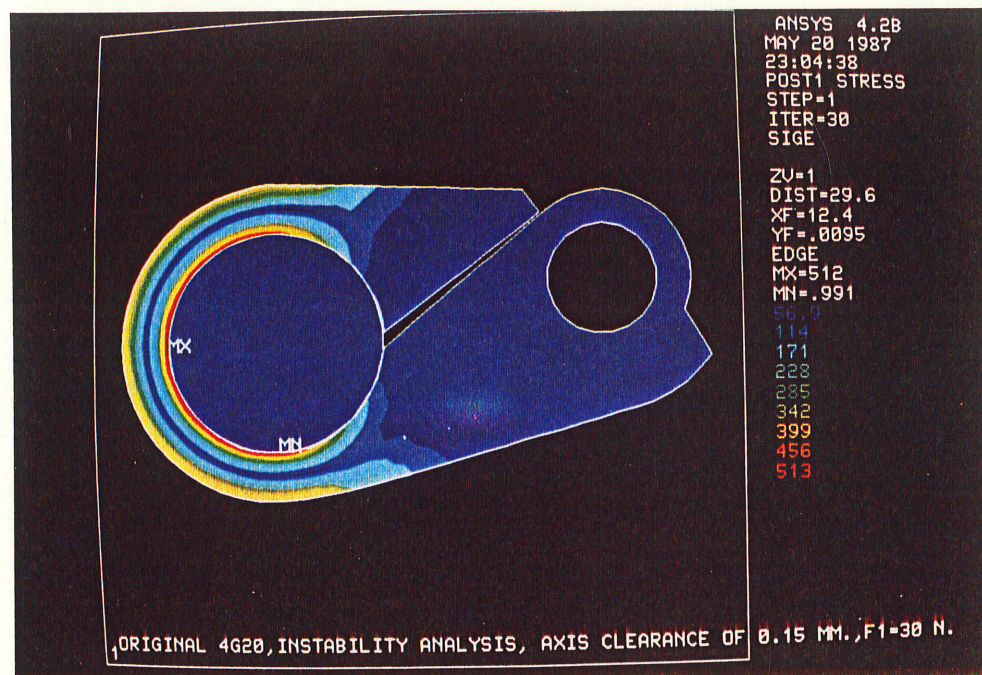


Figure C.8(a): Equivalent Stress Distribution, $R=0.15\text{mm}$, $W=750\text{N}$.

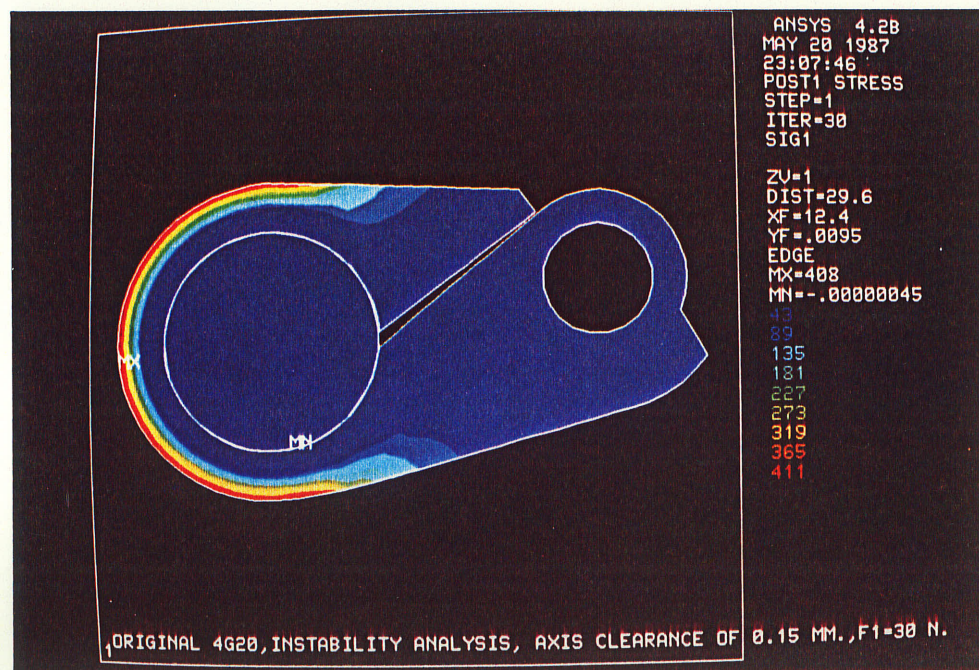


Figure C.8(b): Principal Stress Distribution, $R=0.15\text{mm}$, $W=750\text{N}$.

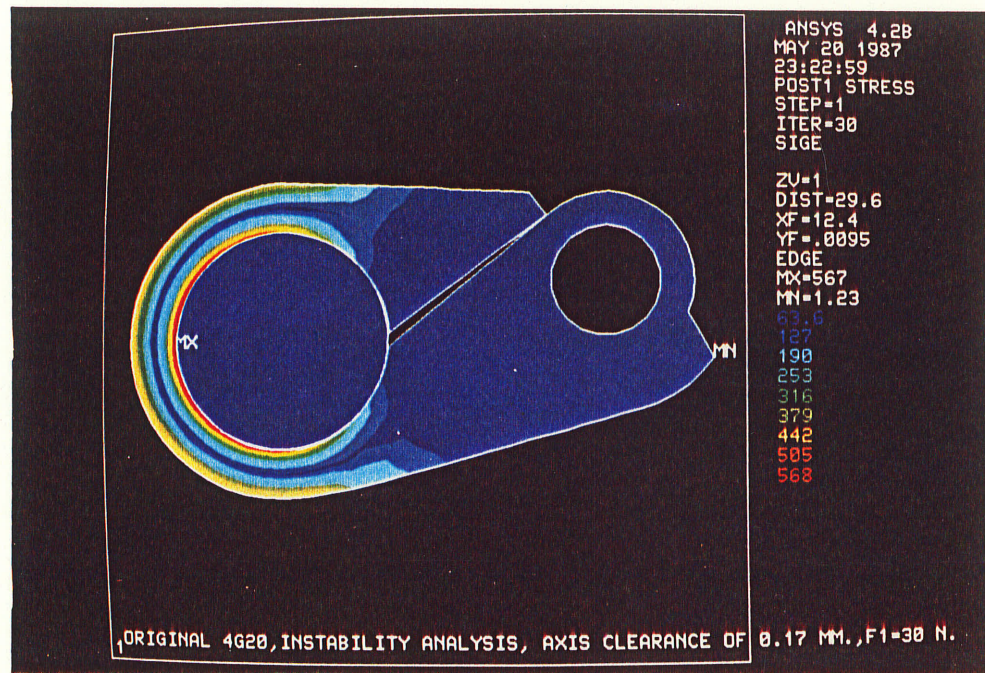


Figure C.9(a): Equivalent Stress Distribution, $R=0.17\text{mm}$, $W=750\text{N}$.

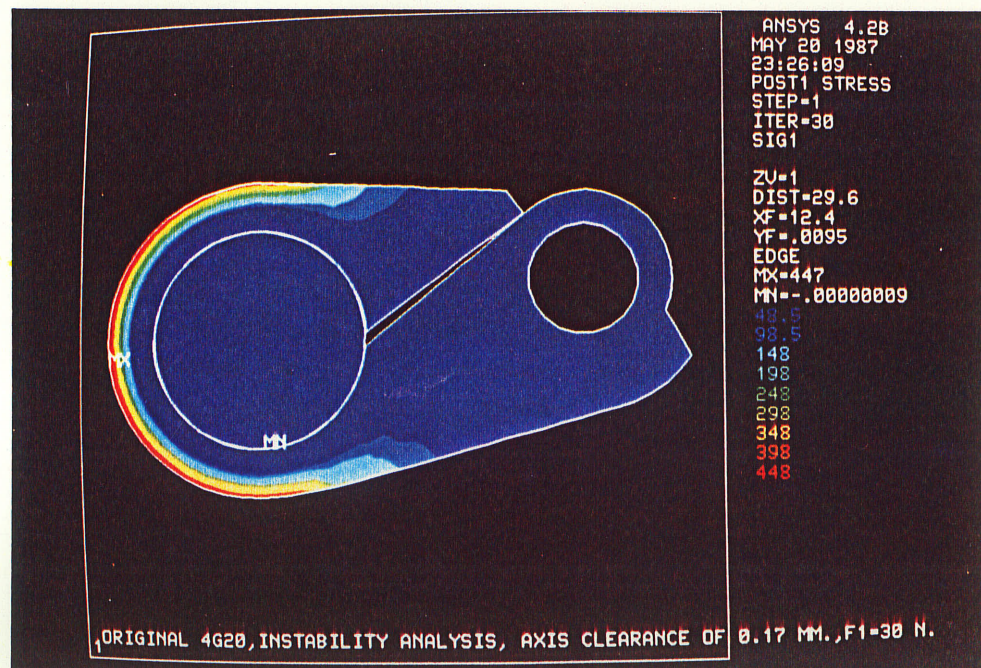


Figure C.9(b): Principal Stress Distribution, $R=0.17\text{mm}$, $W=750\text{N}$.

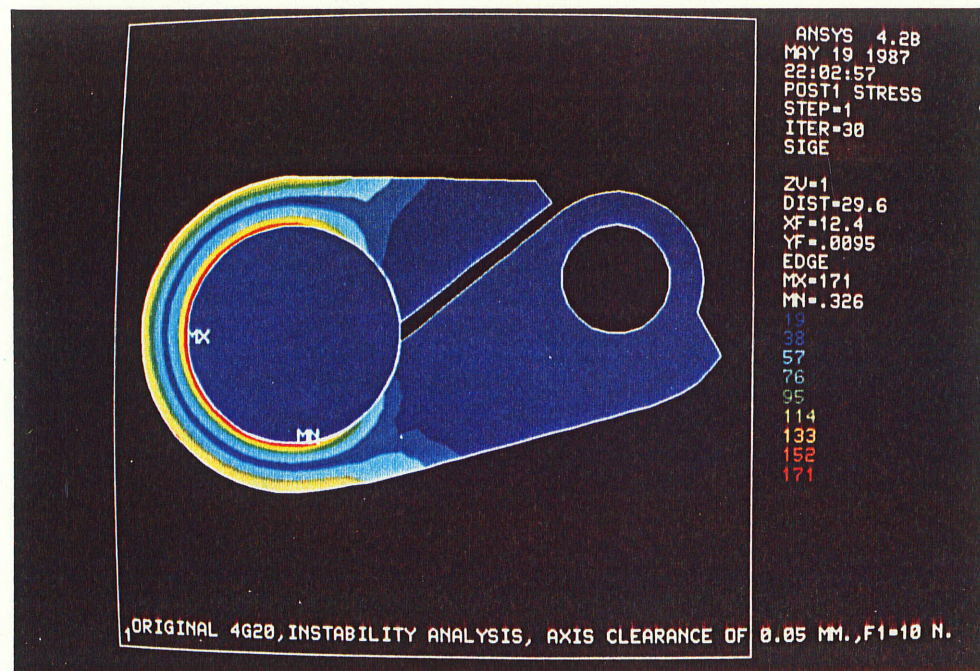


Figure C.10(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

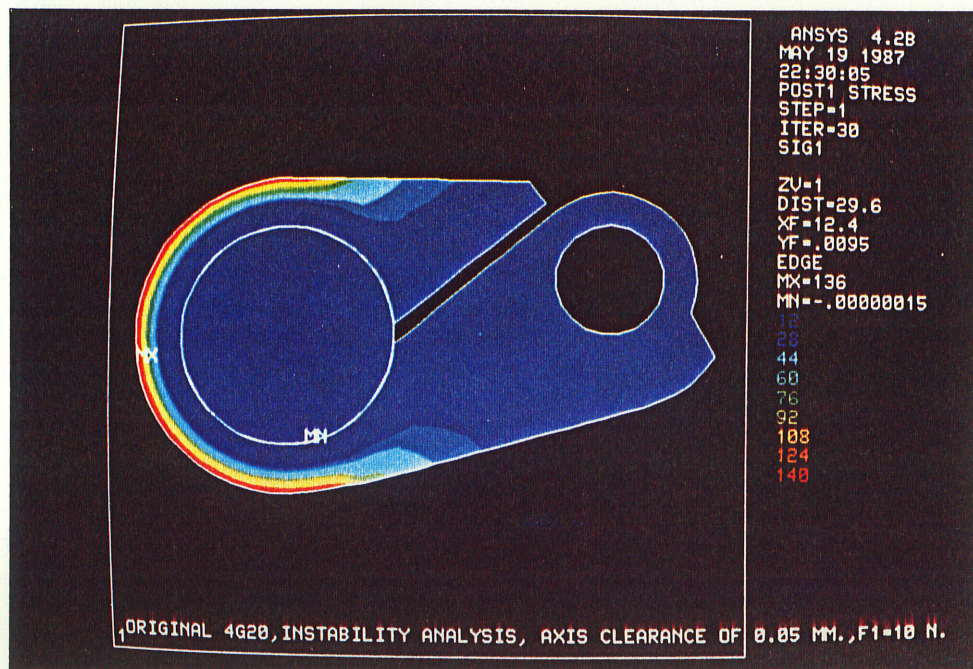


Figure C.10(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

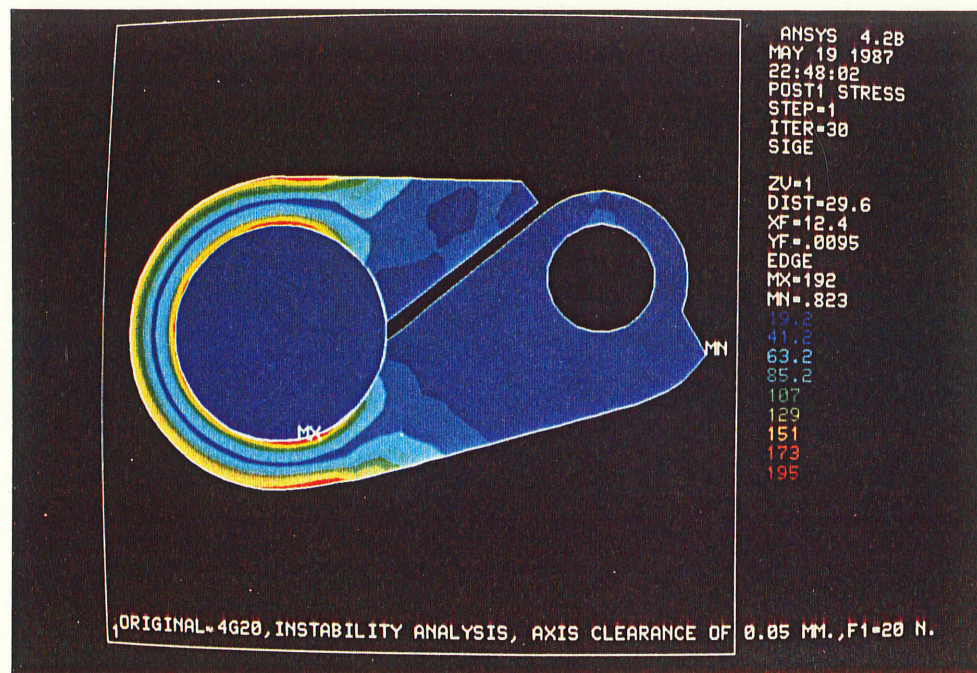


Figure C.11(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

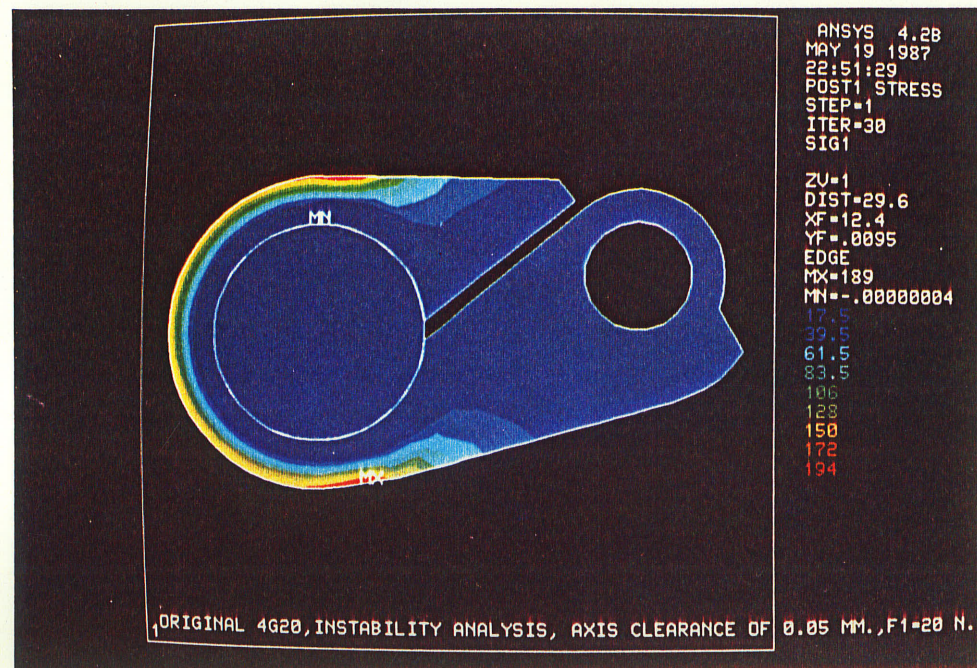


Figure C.11(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

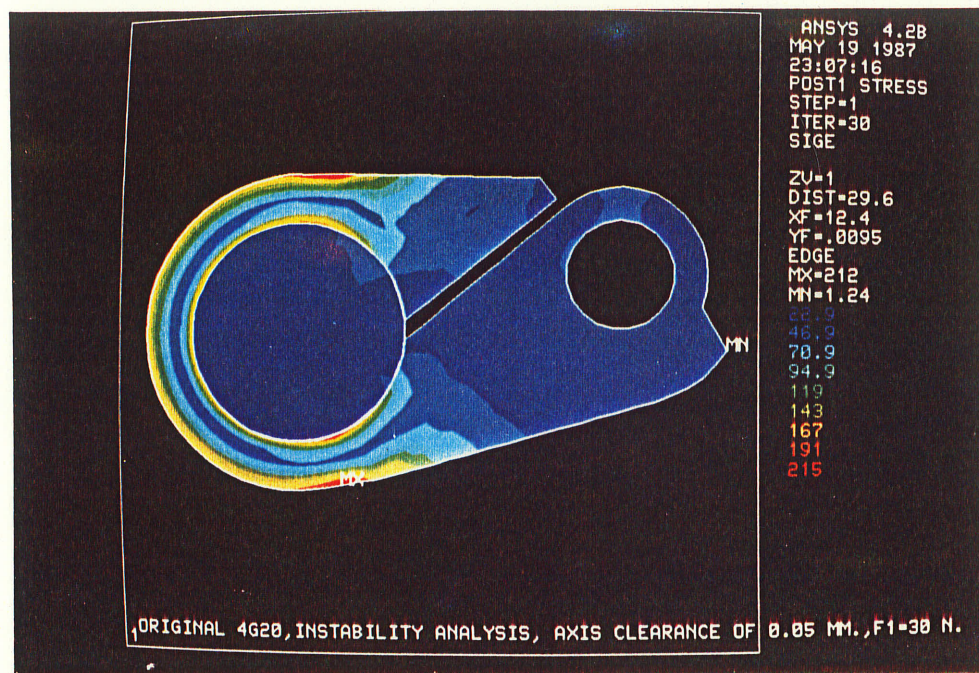


Figure C.12(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

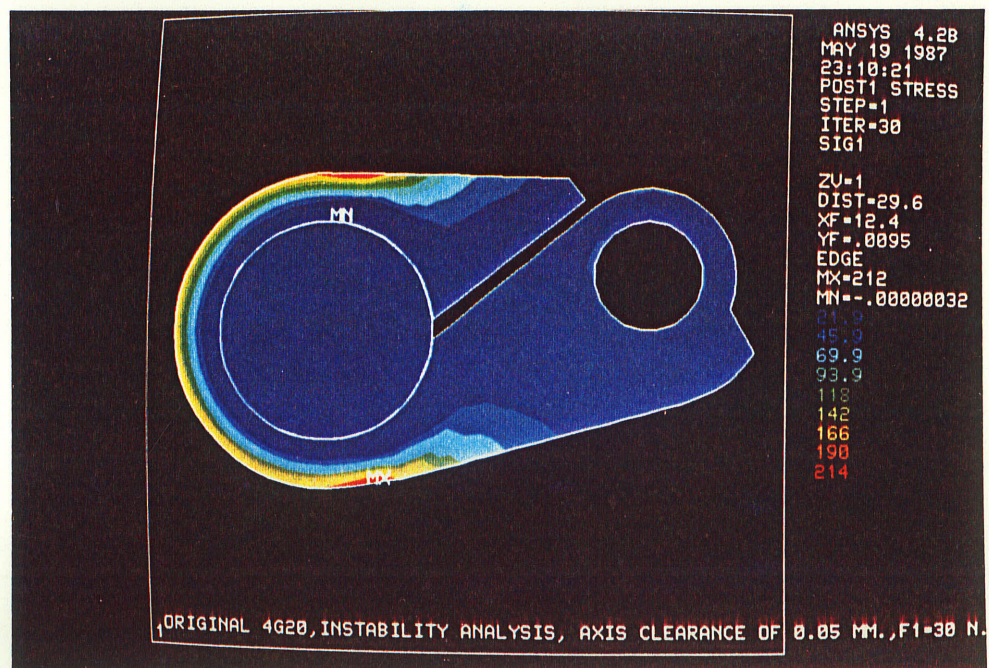


Figure C12(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

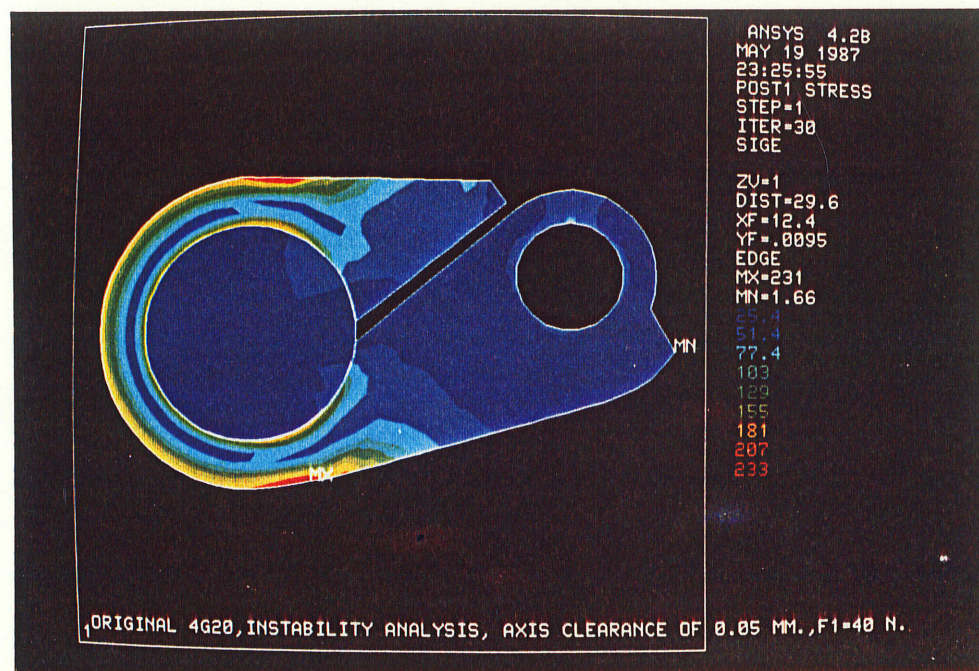


Figure C.13(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

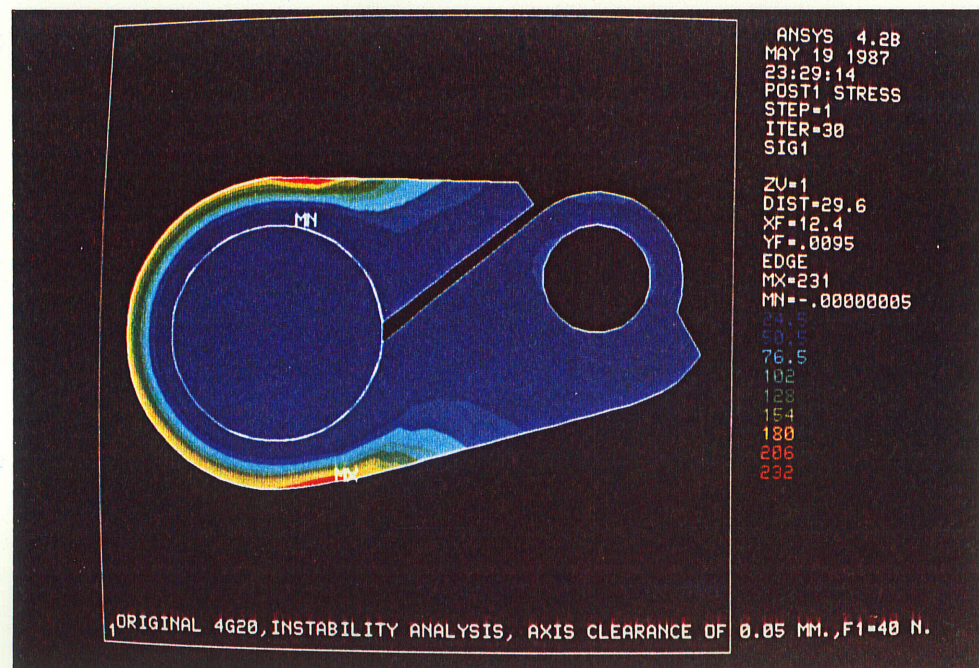


Figure C.13(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

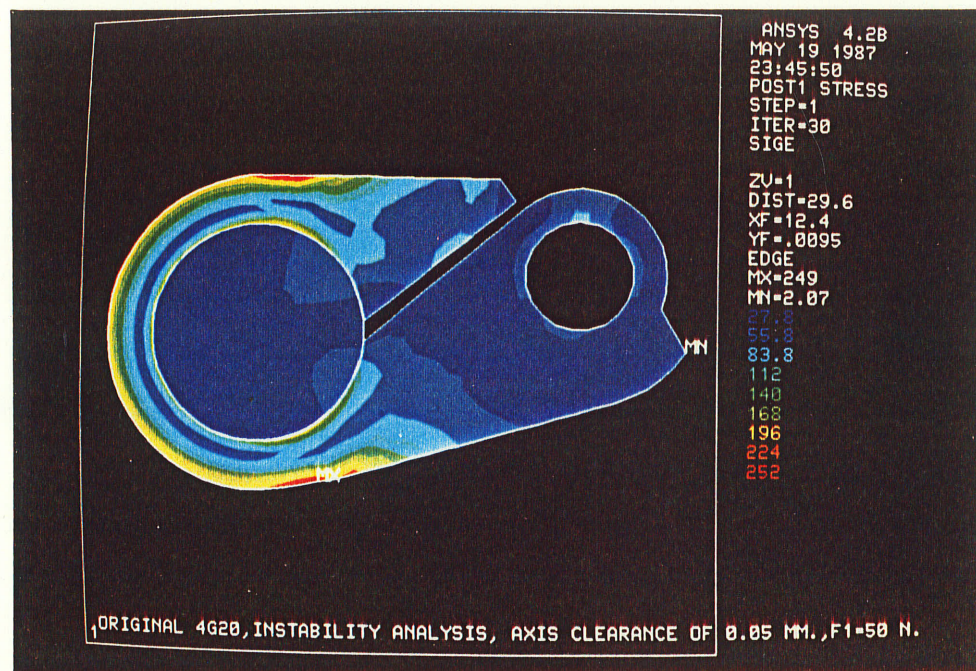


Figure C.14(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1250\text{N}$.

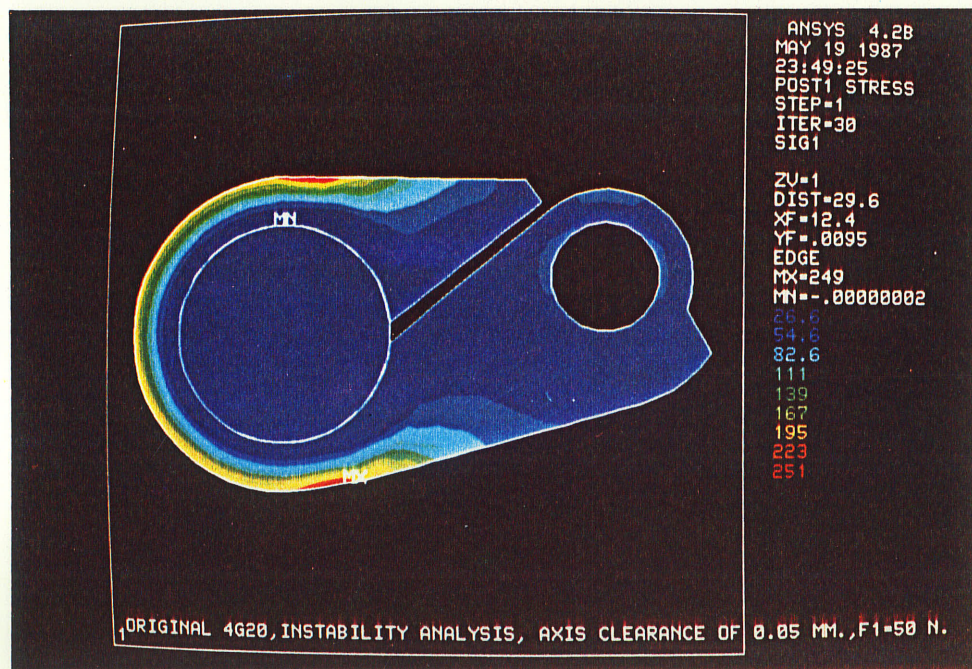


Figure C.14(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1250\text{N}$.

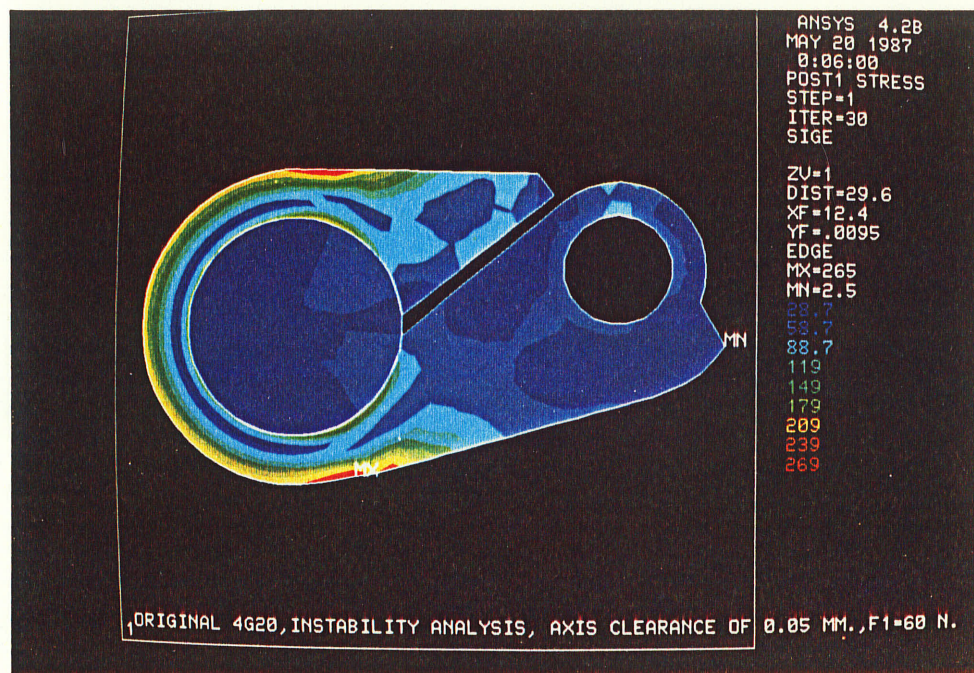


Figure C.15(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.

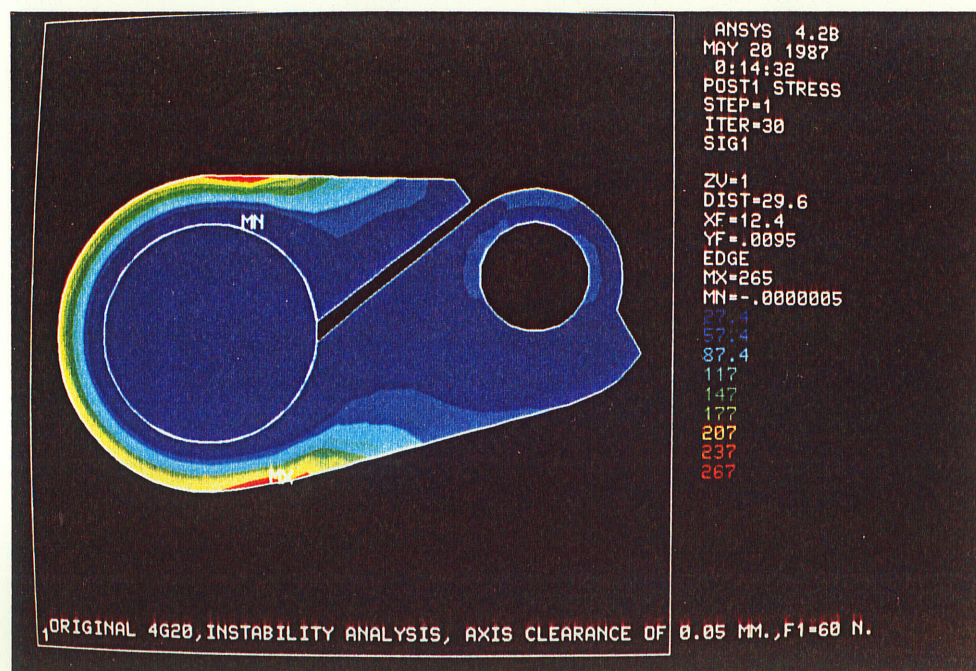


Figure C.15(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.

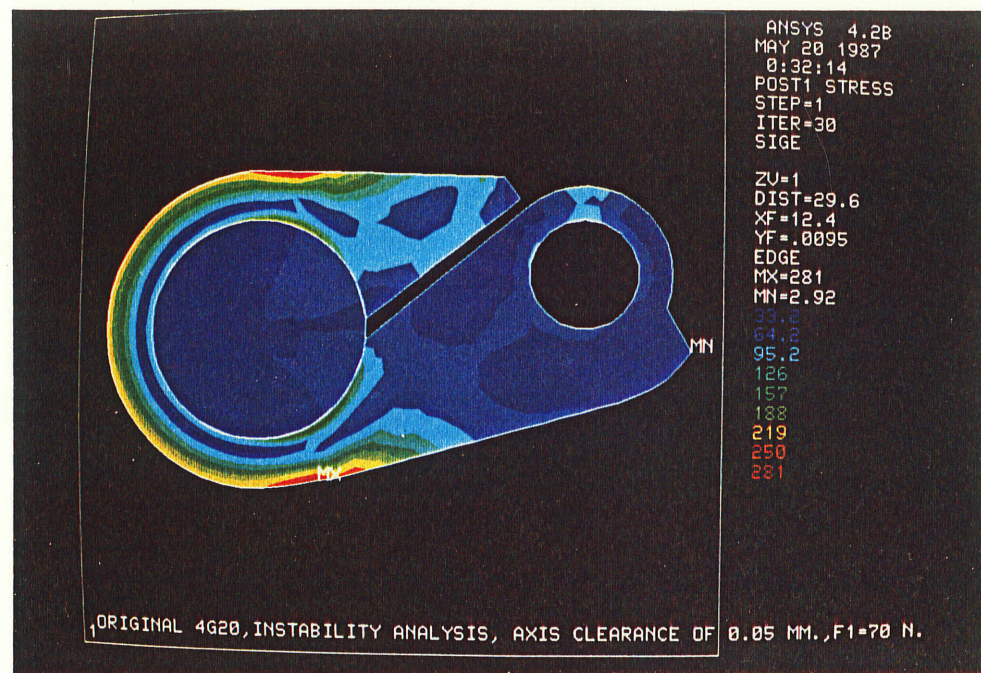


Figure C.16(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1750\text{N}$.

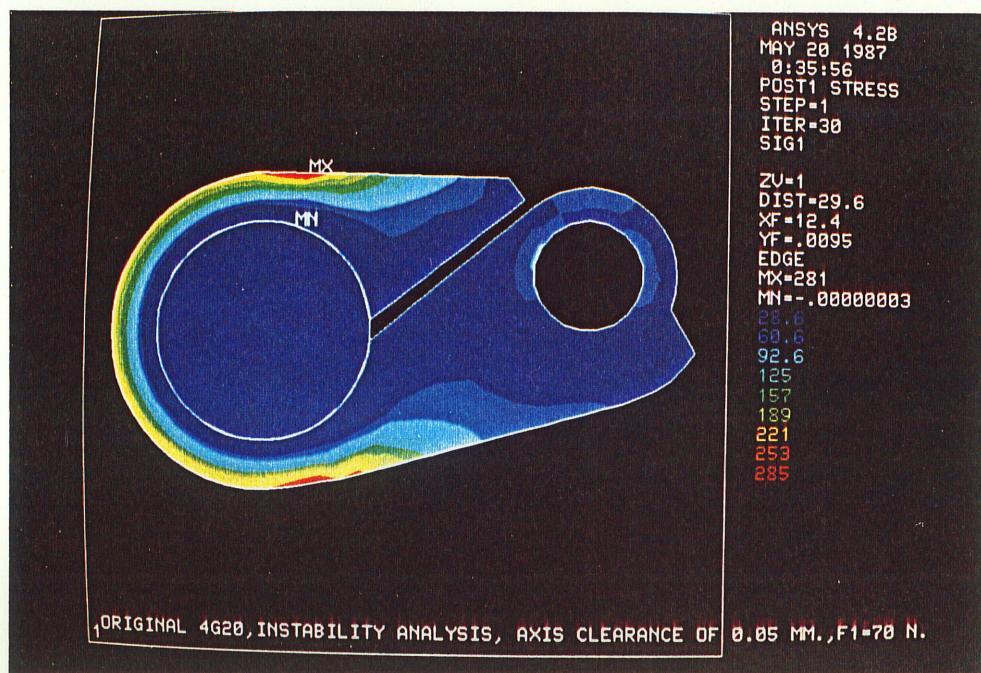


Figure C.16(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1750\text{N}$.

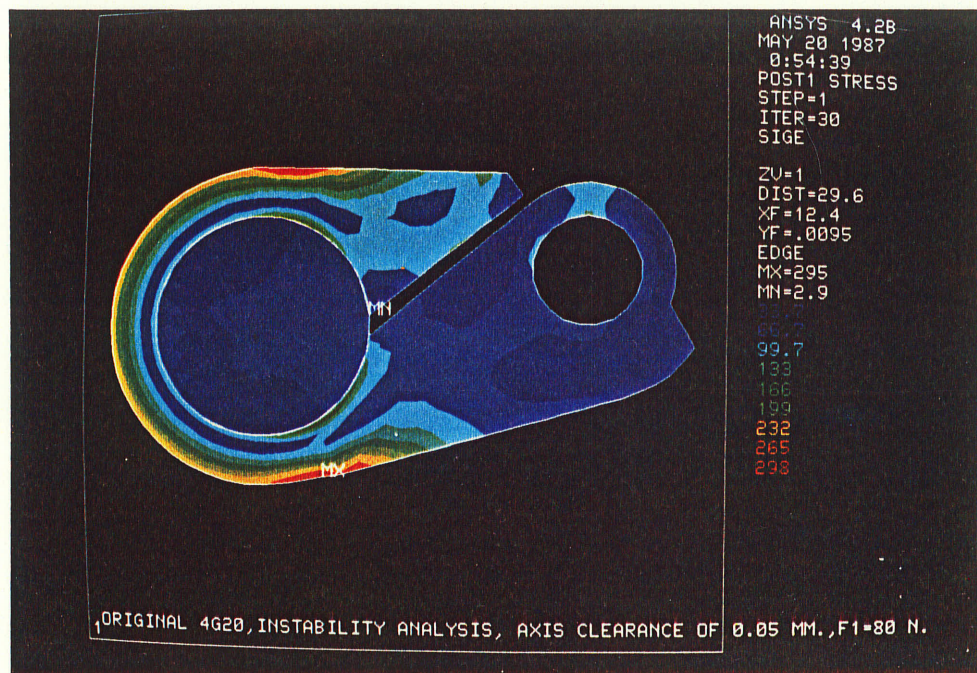


Figure C.17(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=2000\text{N}$.

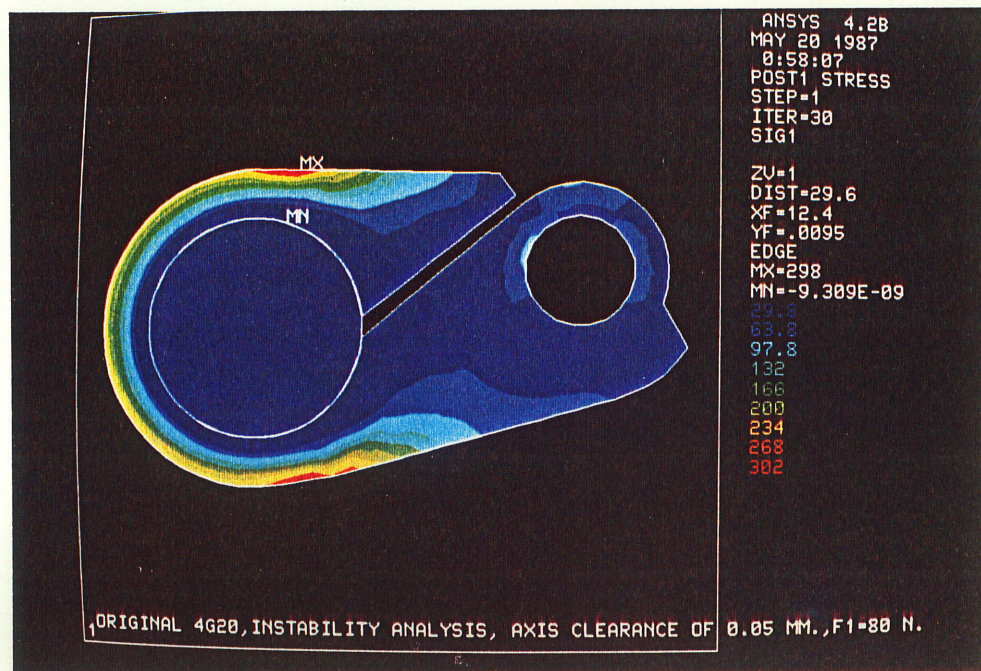


Figure C.17(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=2000\text{N}$.

Appendix D: Equivalent and Principal Stress Distributions for the Redesigned Steel Swing Block.

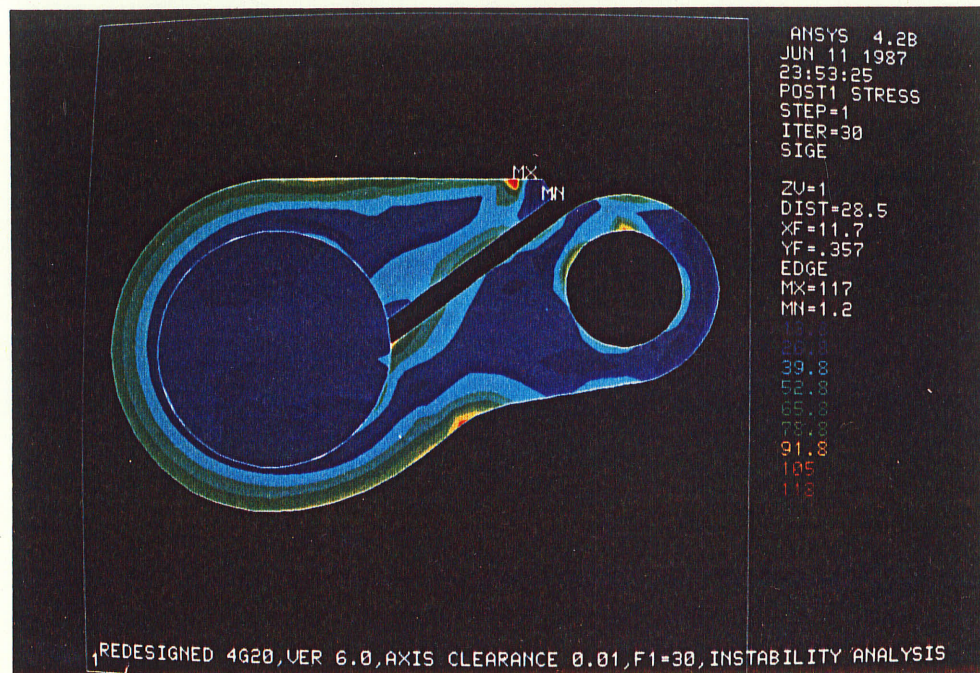


Figure D.1(a): Equivalent Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

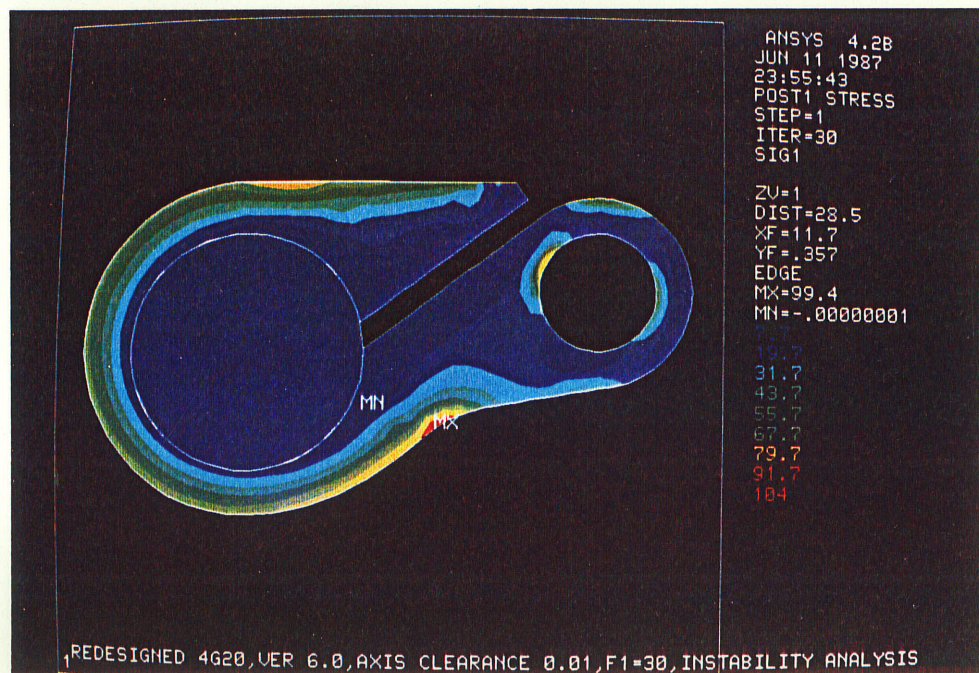


Figure D.1(b): Principal Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

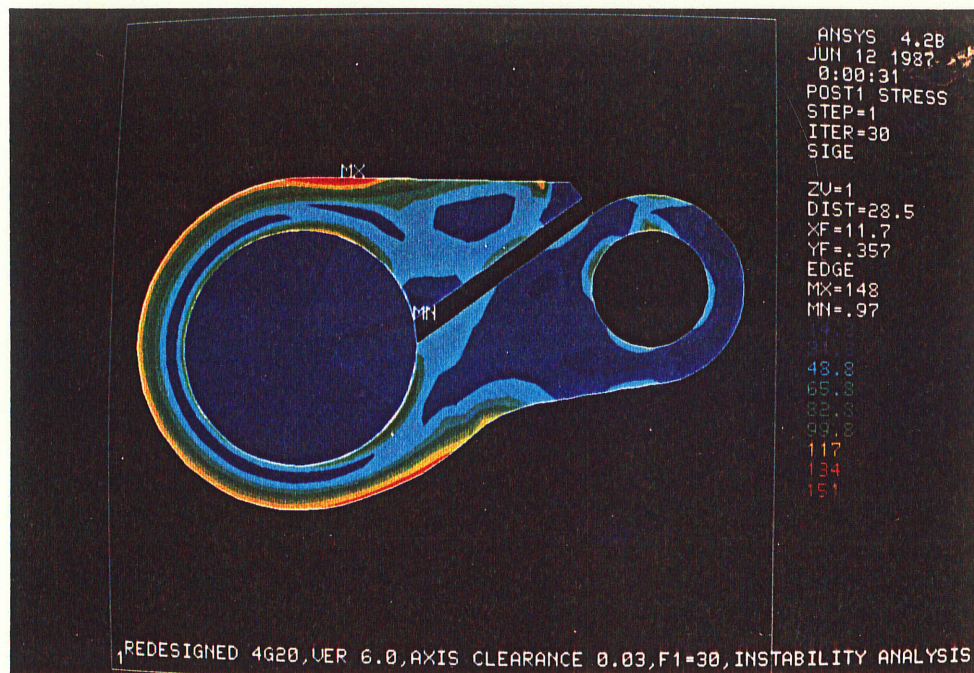


Figure D.2(a): Equivalent Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

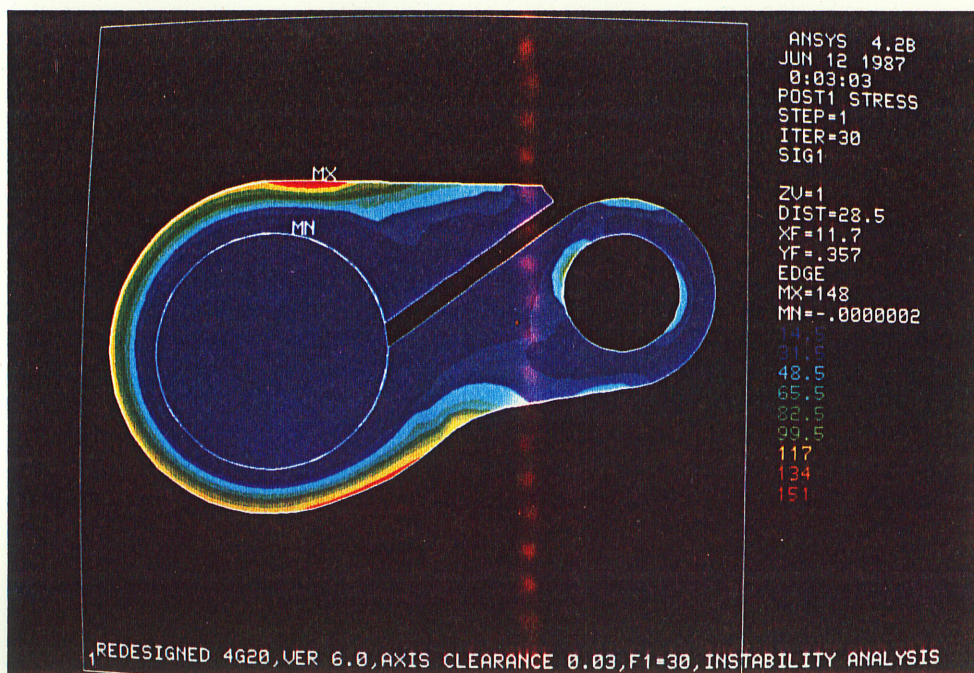


Figure D.2(b): Principal Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

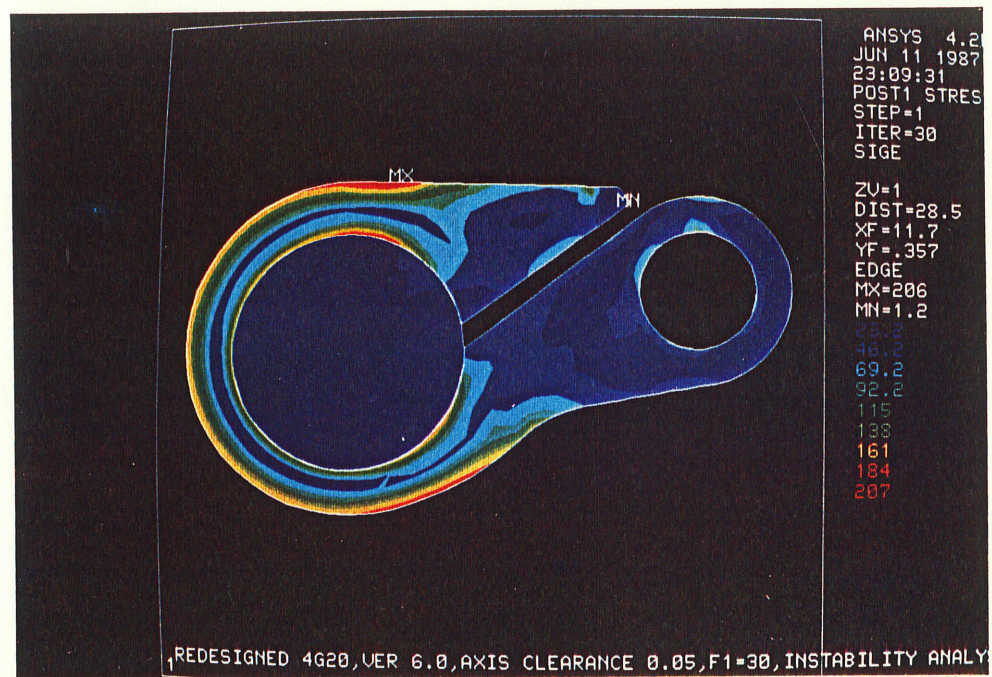


Figure D.3(a): Equivalent Stress Distribution. $R=0.05\text{mm}$, $W=750\text{N}$.

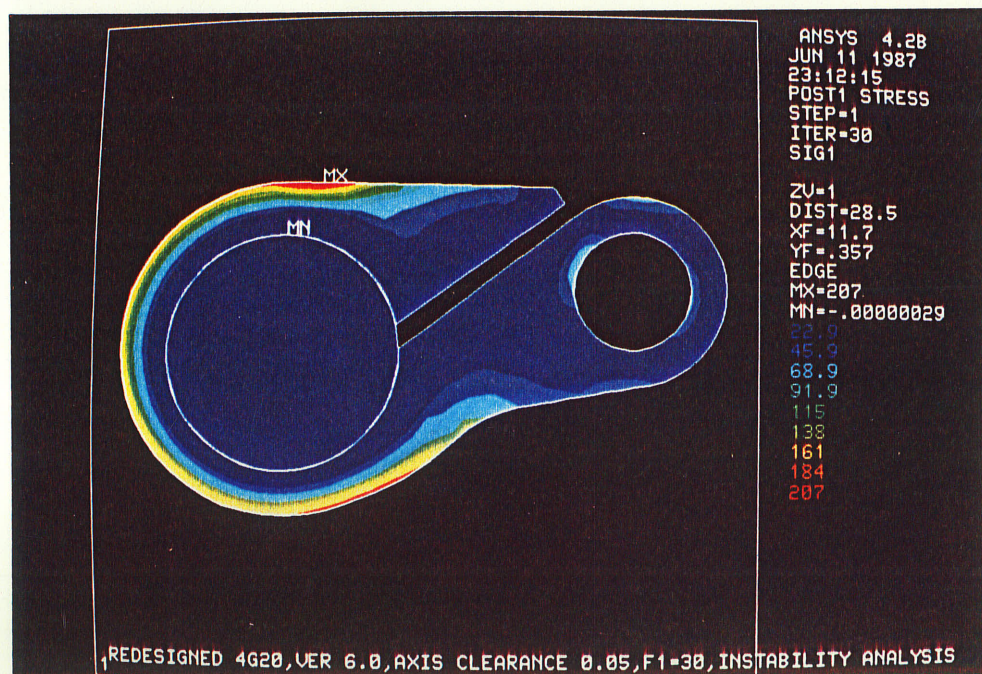


Figure D.3(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

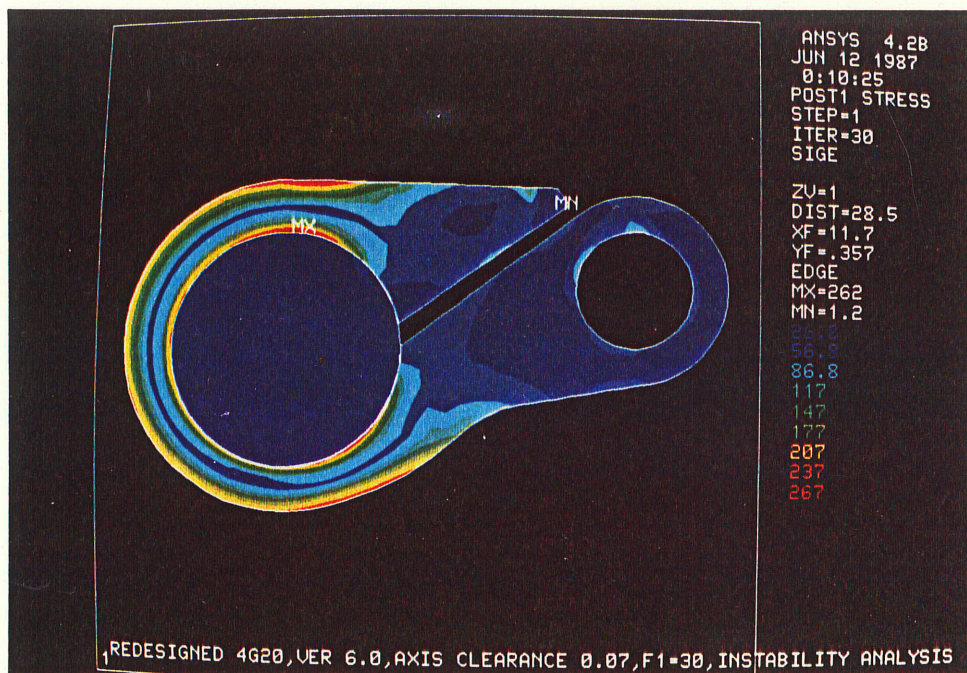


Figure D.4(a): Equivalent Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

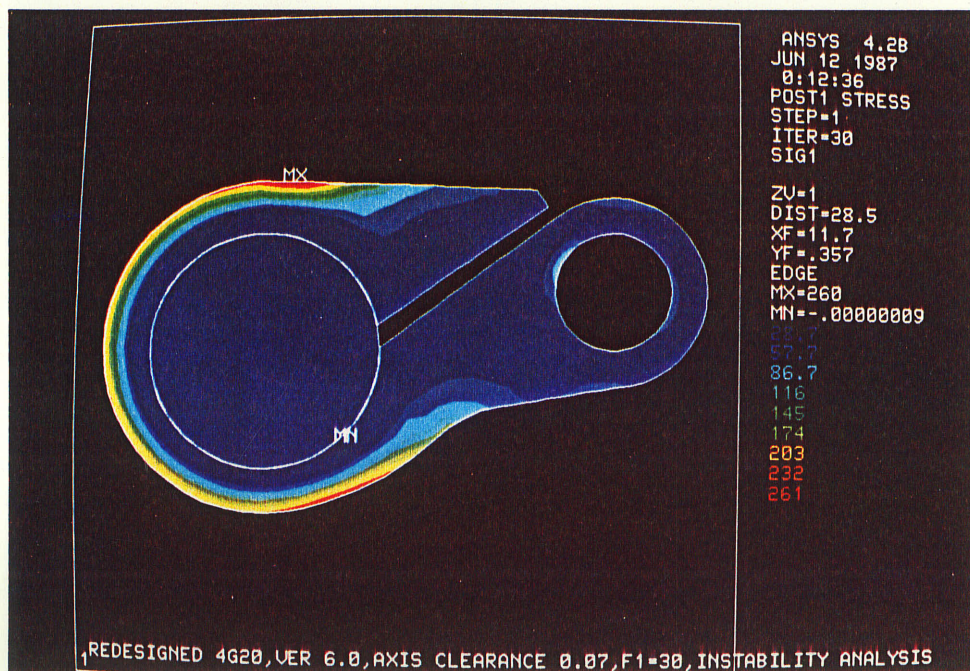


Figure D.4b): Principal Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

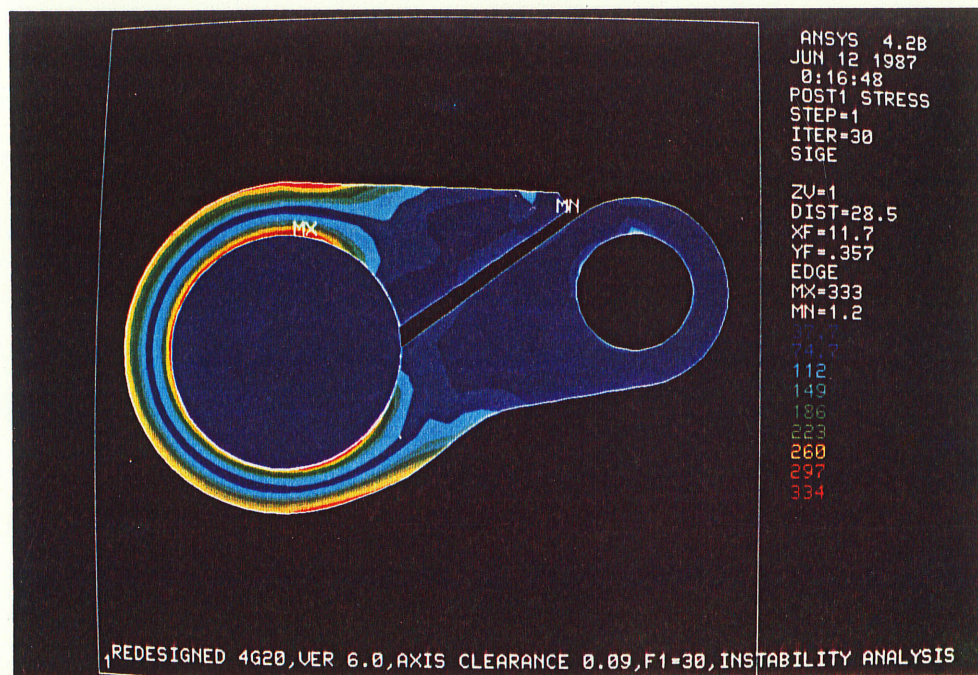


Figure D.5(a): Equivalent Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

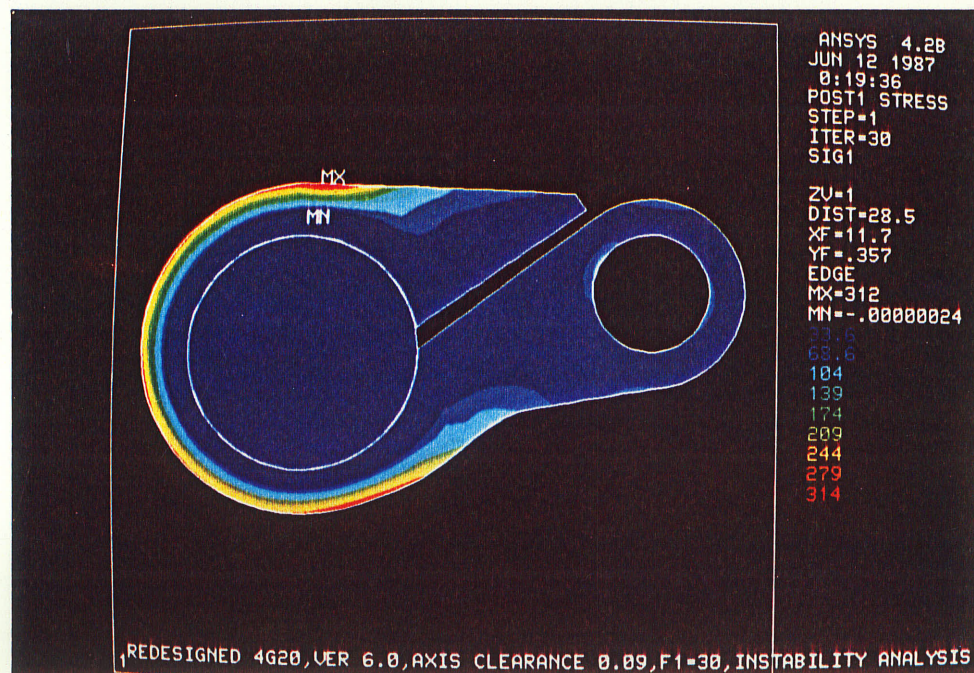


Figure D.5(b): Principal Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

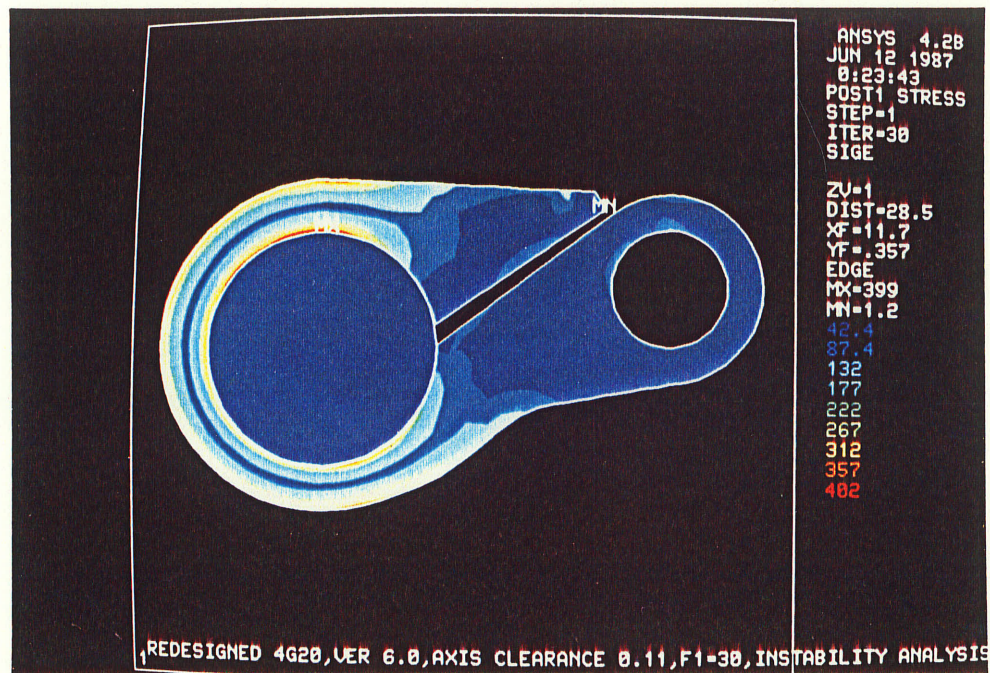


Figure D.6(a): Equivalent Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

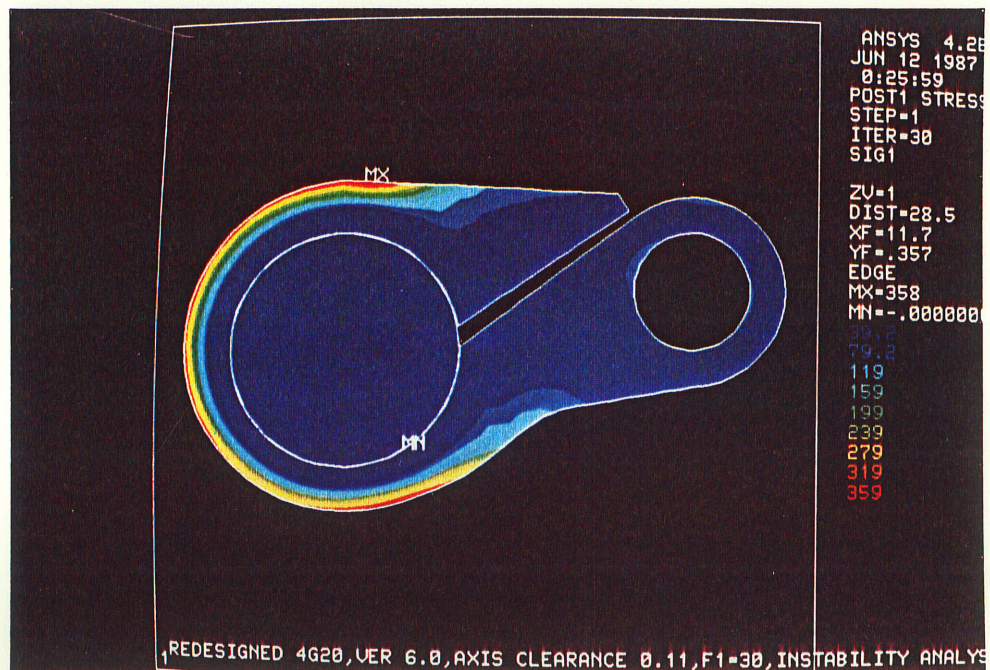


Figure D.6(b): Principal Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

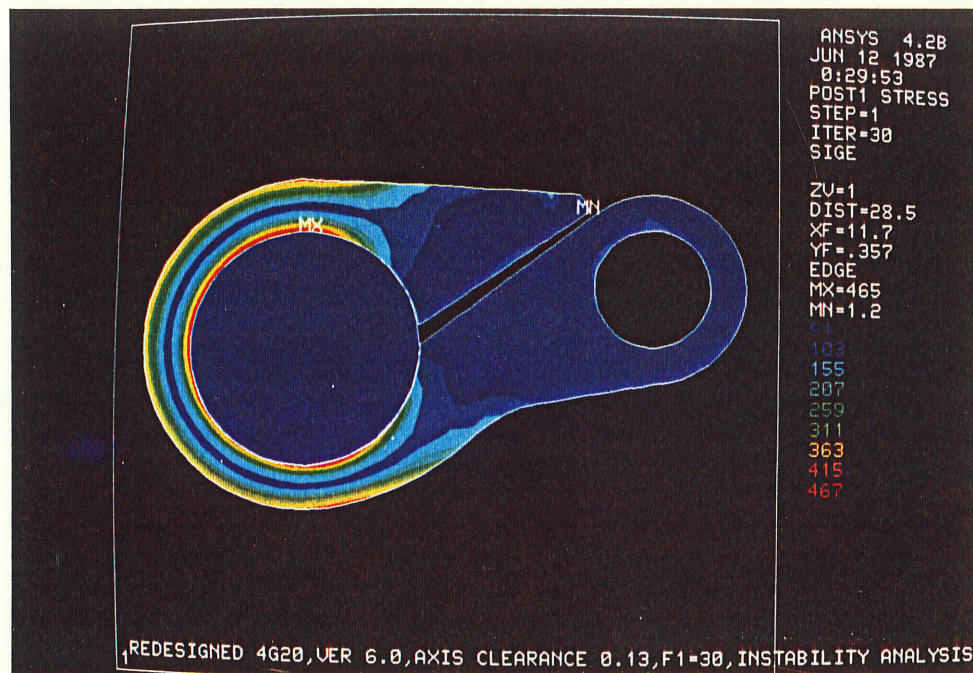


Figure D.7(a): Equivalent Stress Distribution, $R=0.13\text{mm}$, $W=750\text{N}$.

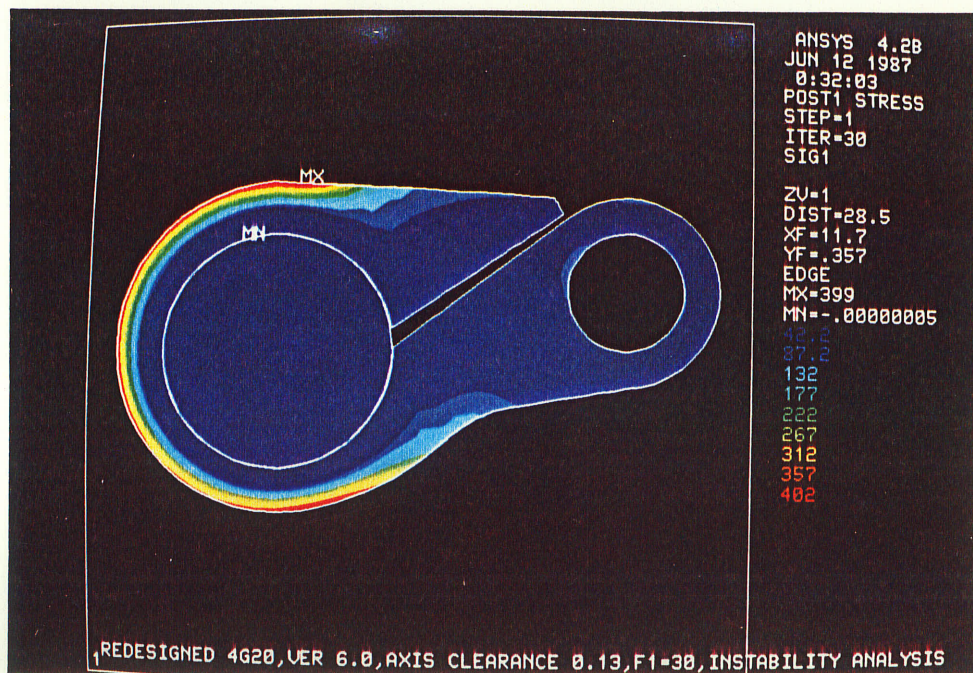


Figure D.7(b): Principal Stress Distribution, $R=0.13\text{mm}$, $W=750\text{N}$.

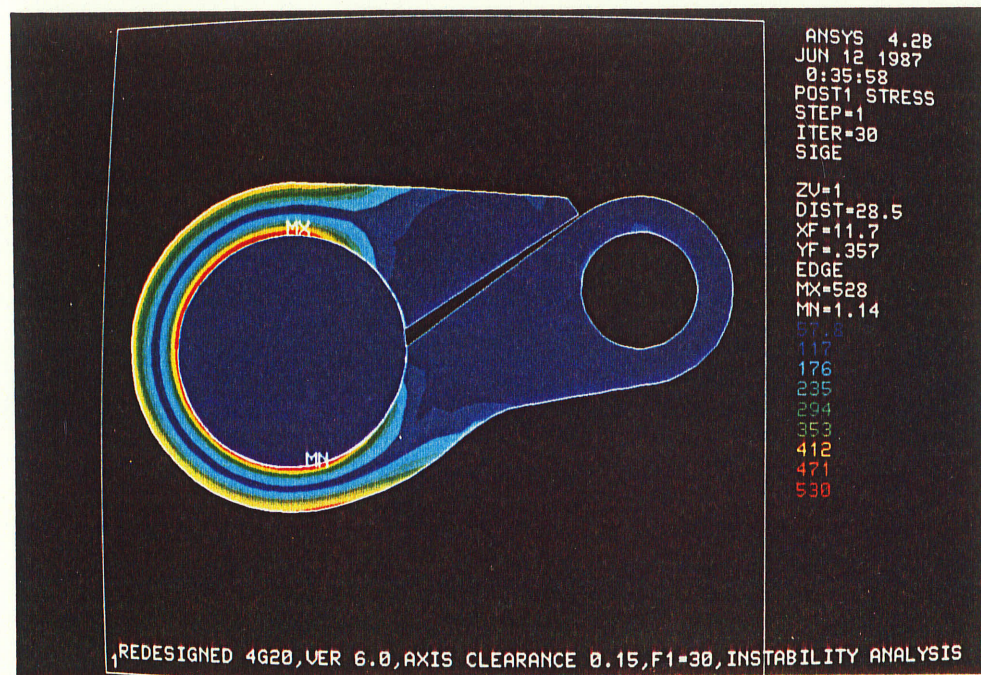


Figure D.8(a): Equivalent Stress Distribution, $R=0.15\text{mm}$, $W=750\text{N}$.

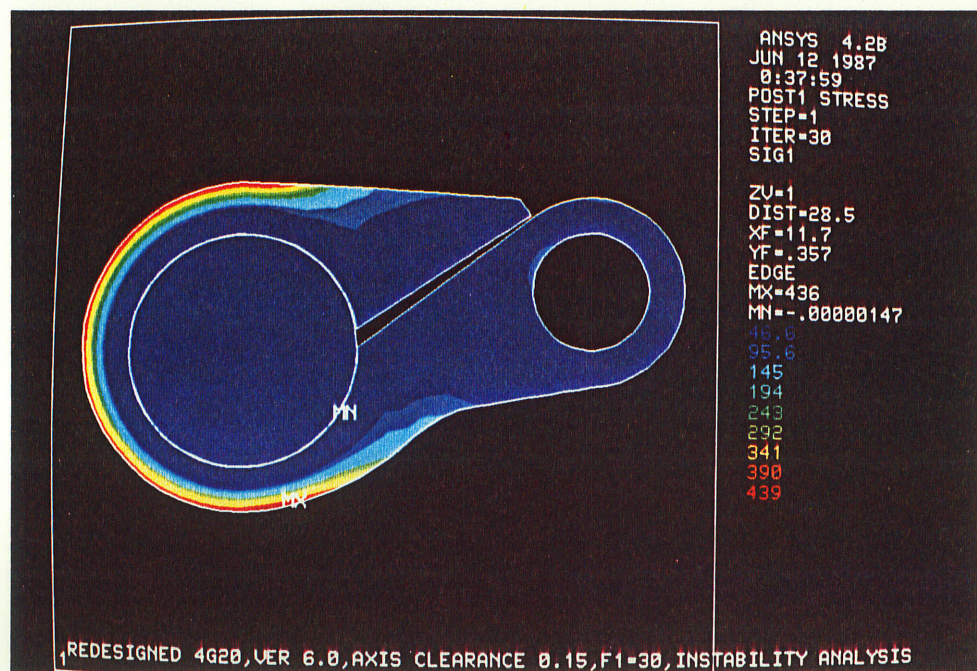


Figure D.8(b): Principal Stress Distribution, $R=0.15\text{mm}$, $W=750\text{N}$.

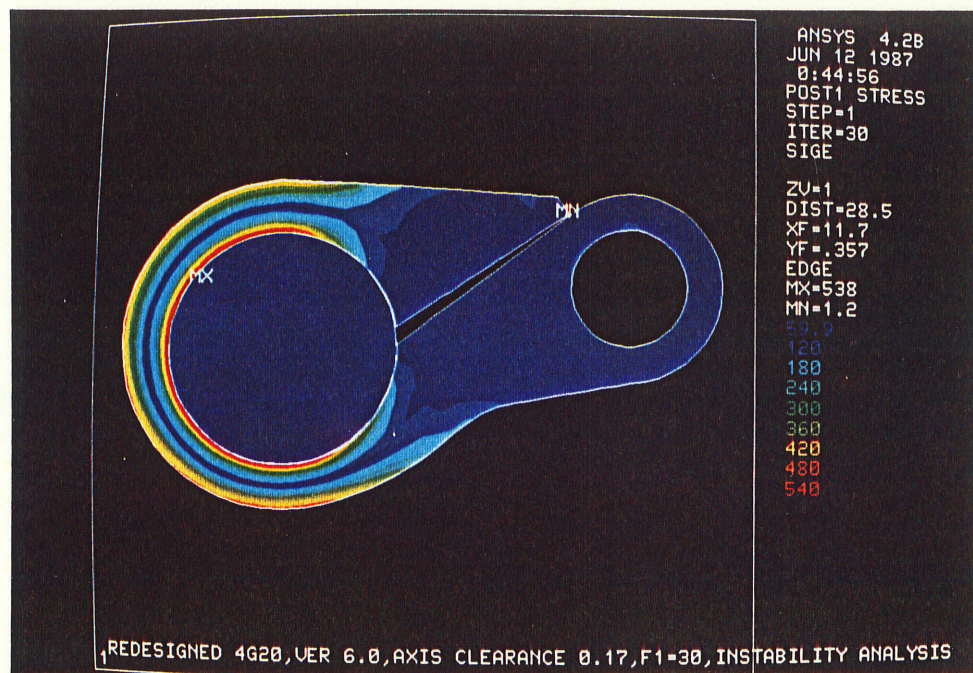


Figure D.9(a): Equivalent Stress Distribution, $R=0.17\text{mm}$, $W=750\text{N}$.

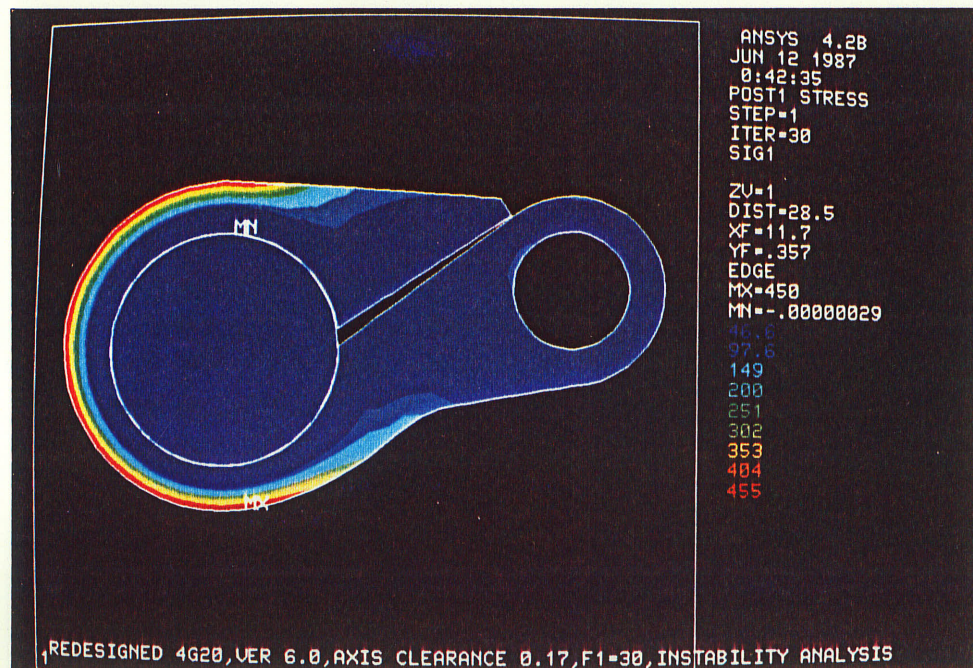


Figure D.9(b): Principal Stress Distribution, $R=0.17\text{mm}$, $W=750\text{N}$.

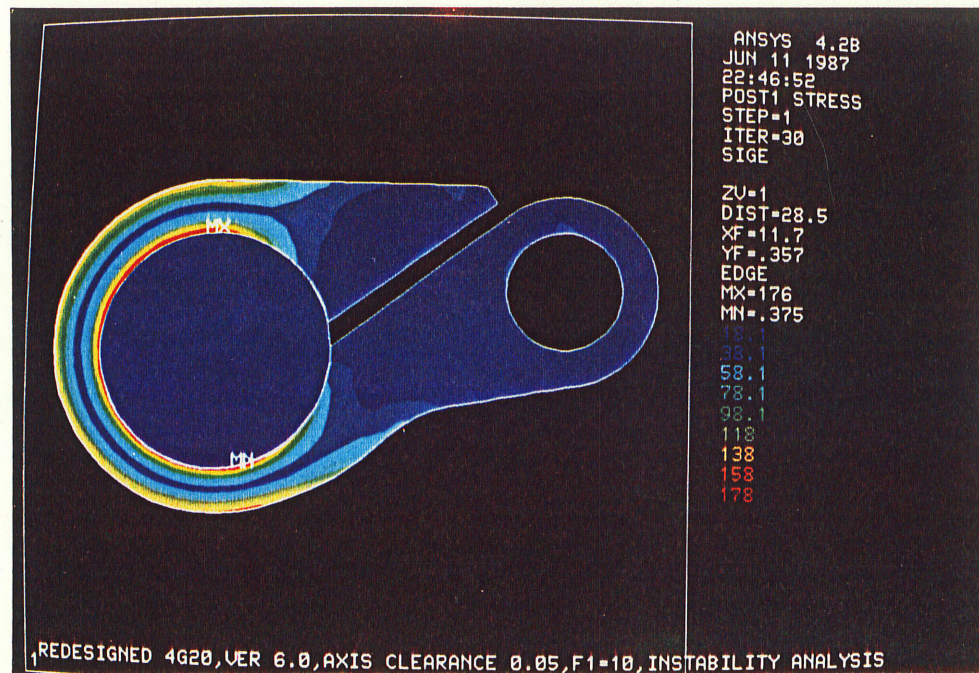


Figure D.10(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

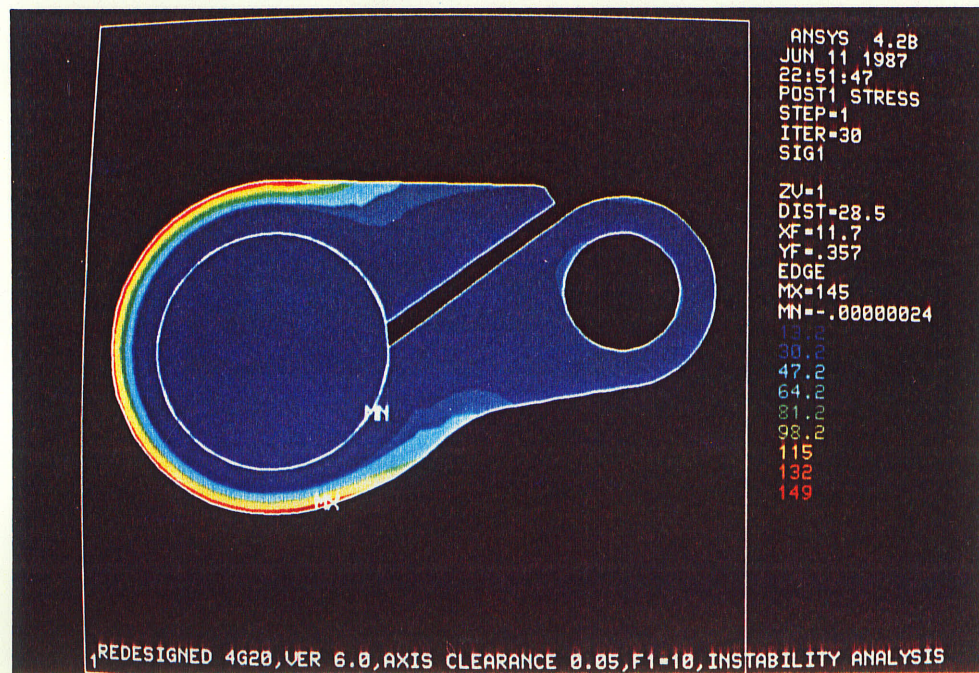


Figure D.10(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

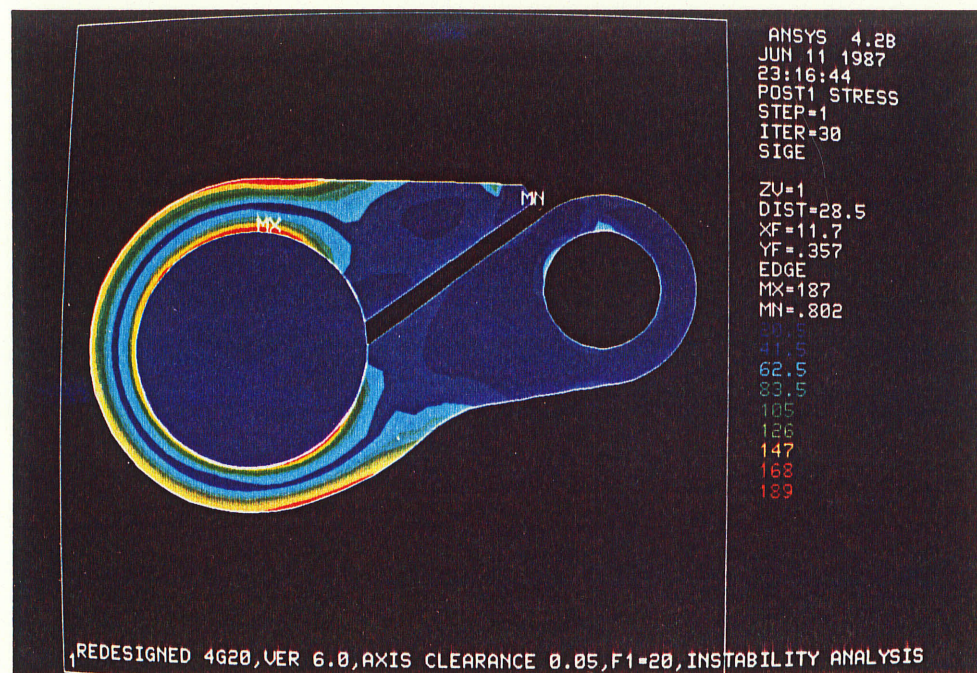


Figure D.11(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

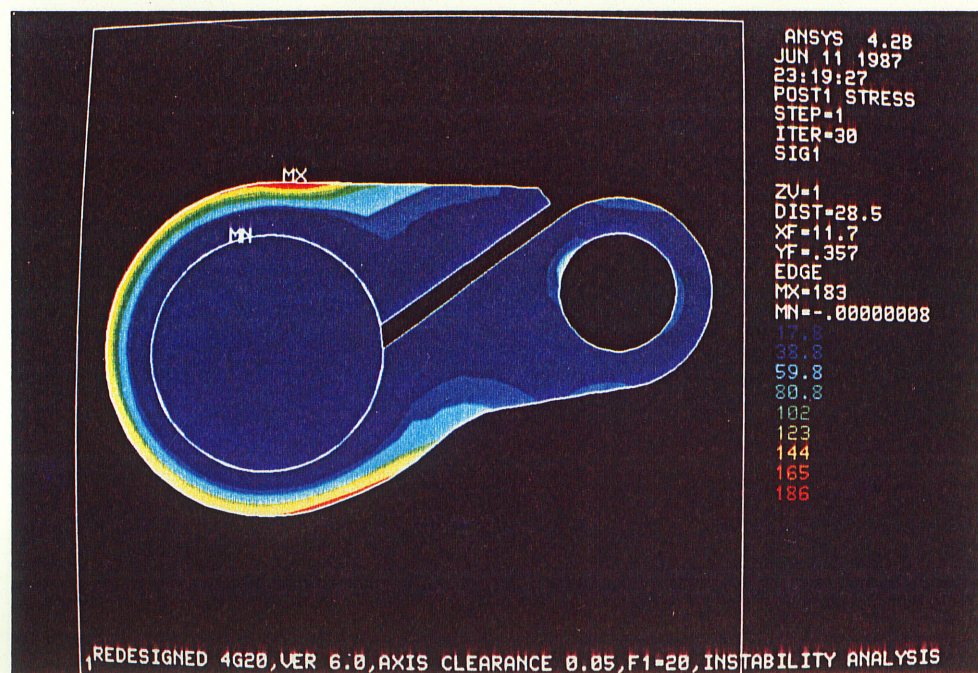


Figure D.11(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

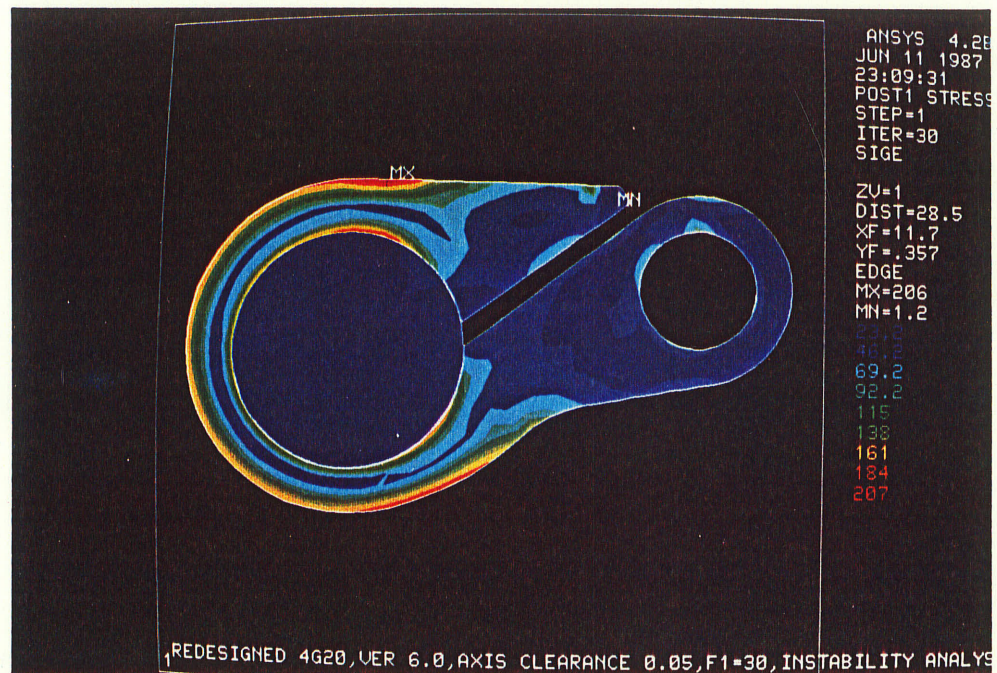


Figure D.12(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

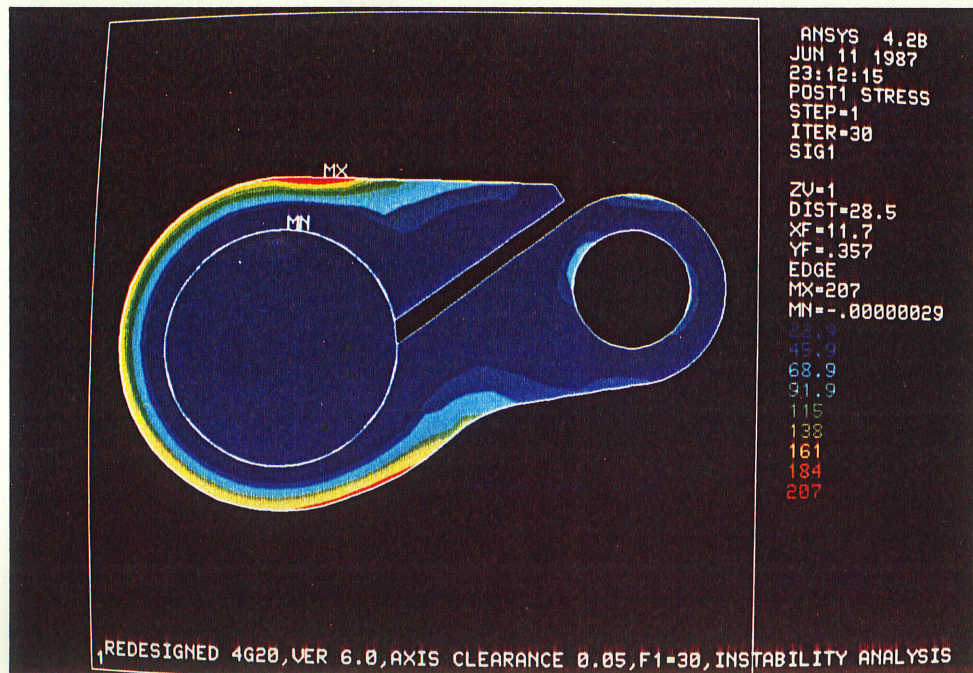


Figure D.12(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

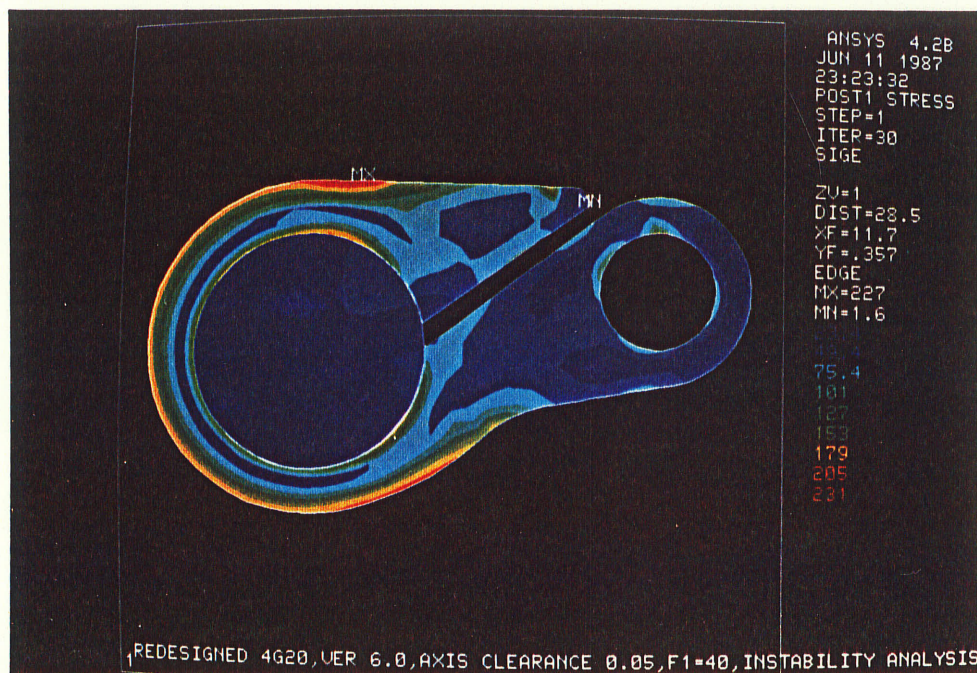


Figure D.13(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

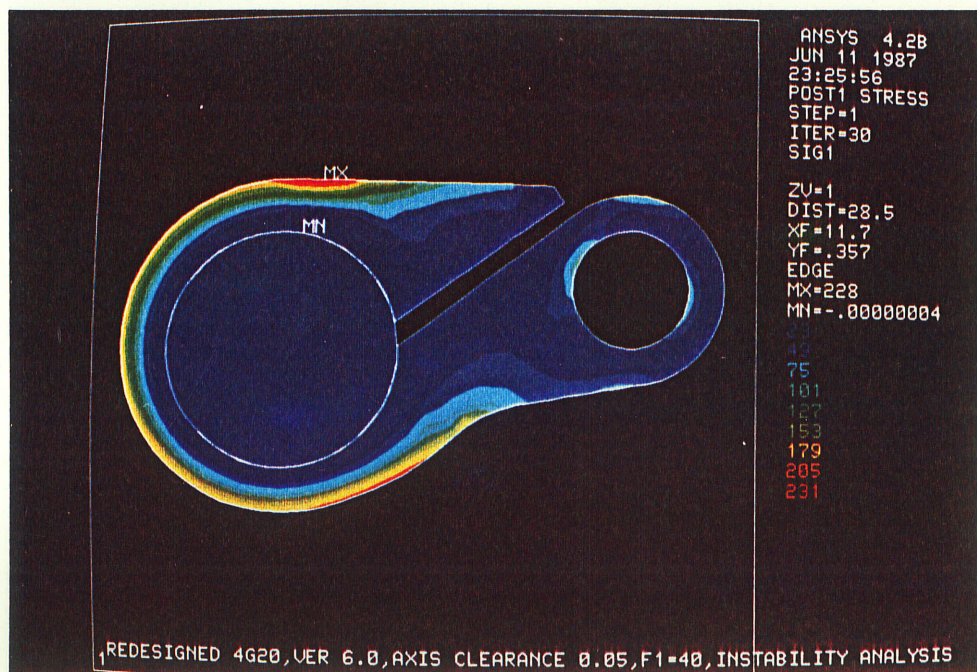


Figure D.13(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

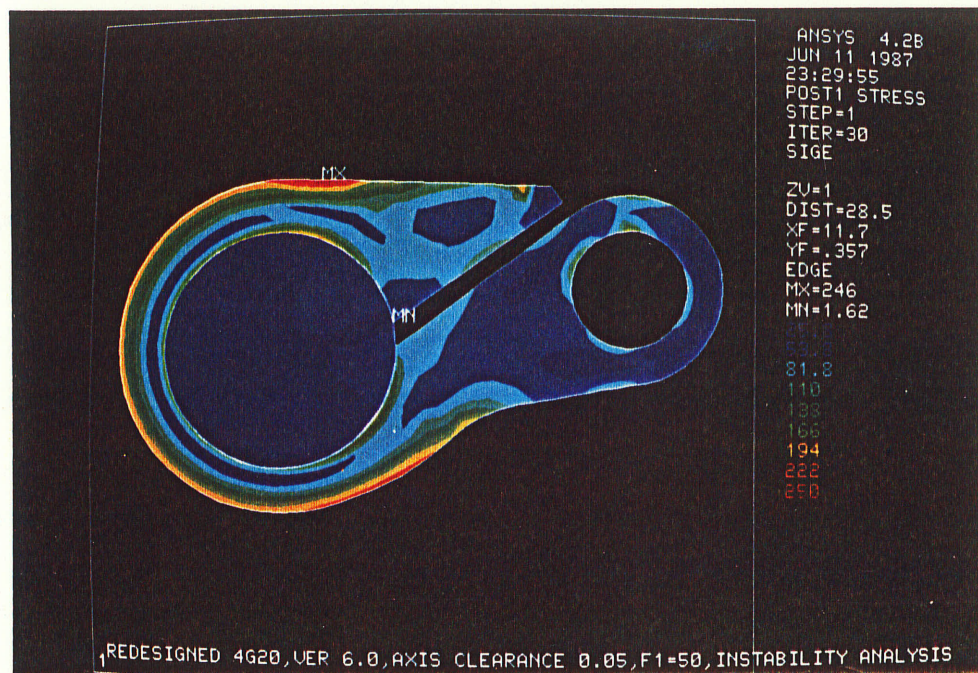


Figure D.14(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1250\text{N}$.

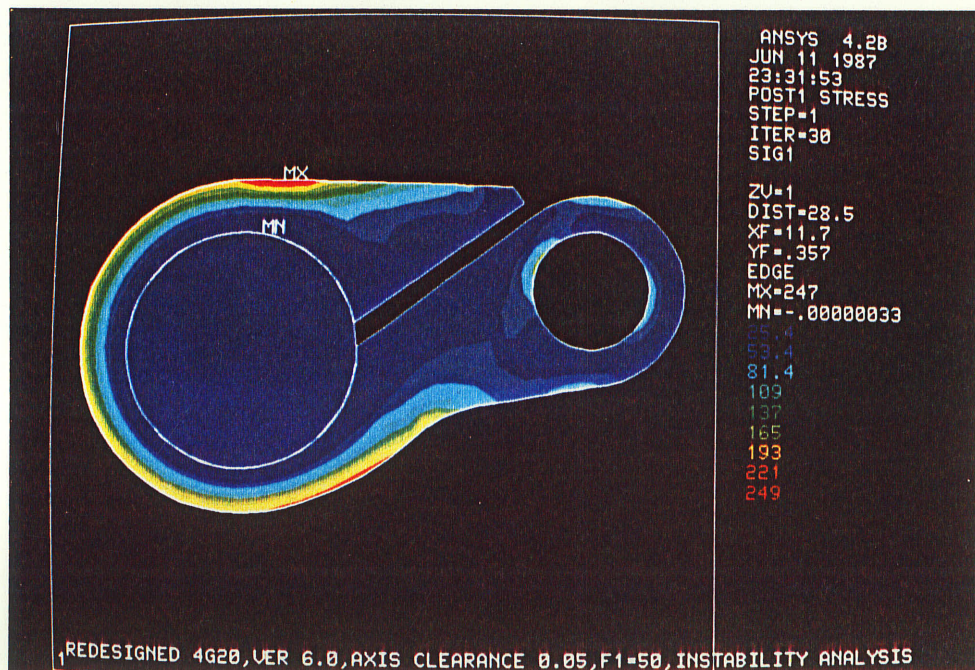


Figure D.14(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1250\text{N}$.

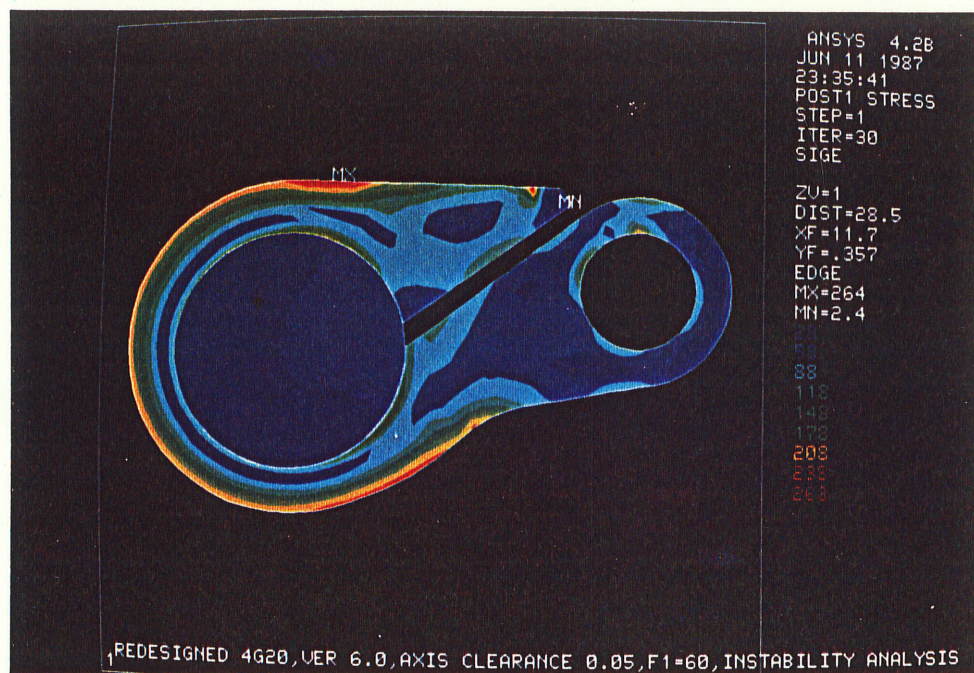


Figure D.15(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.

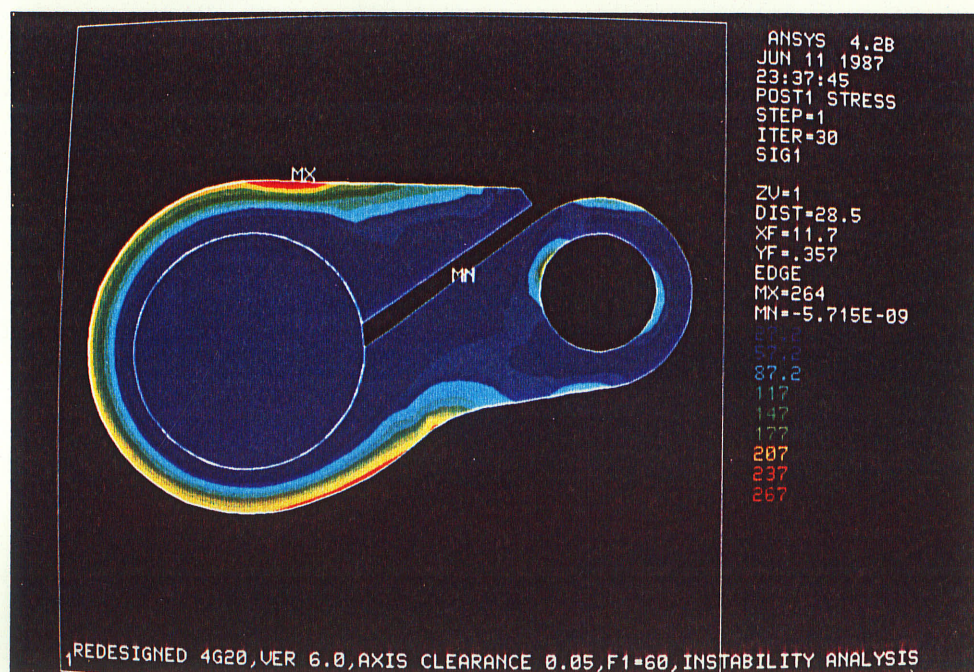


Figure D.15(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.

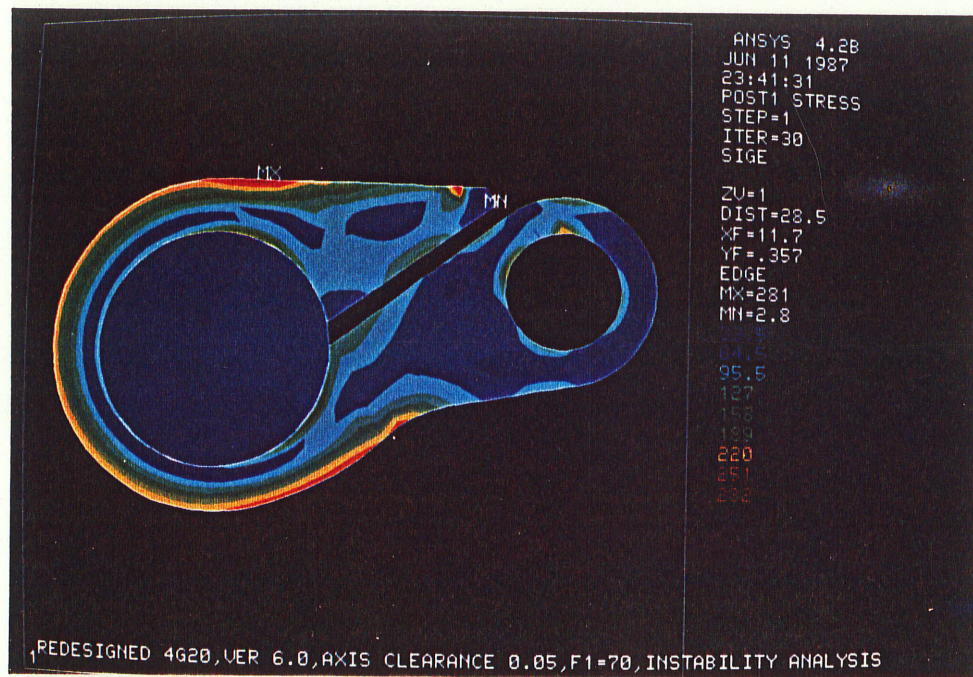


Figure D.16(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1750\text{N}$.

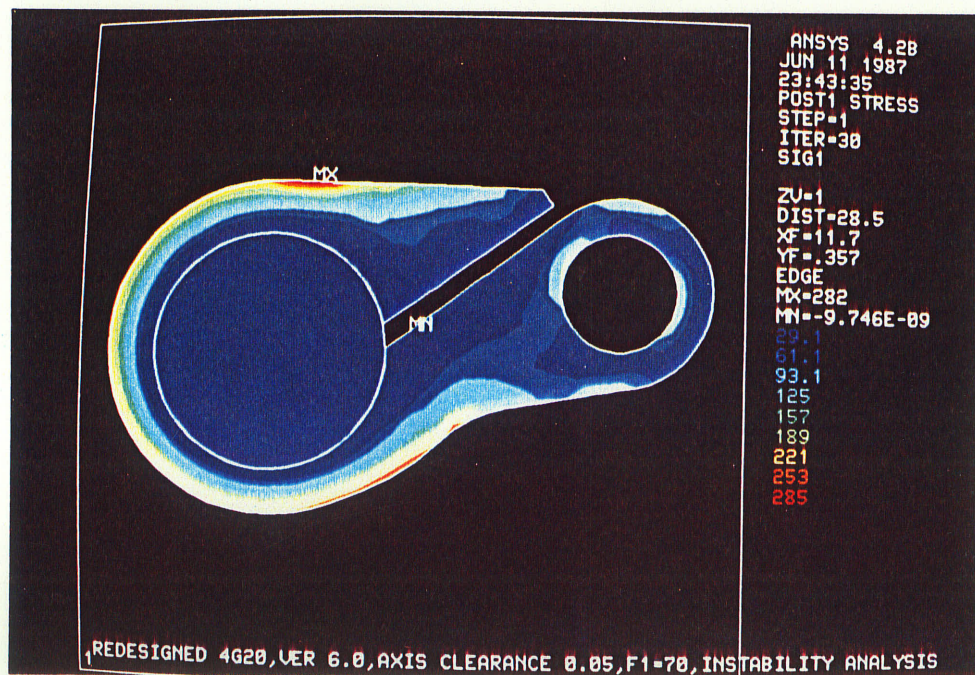


Figure D.16(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1750\text{N}$.

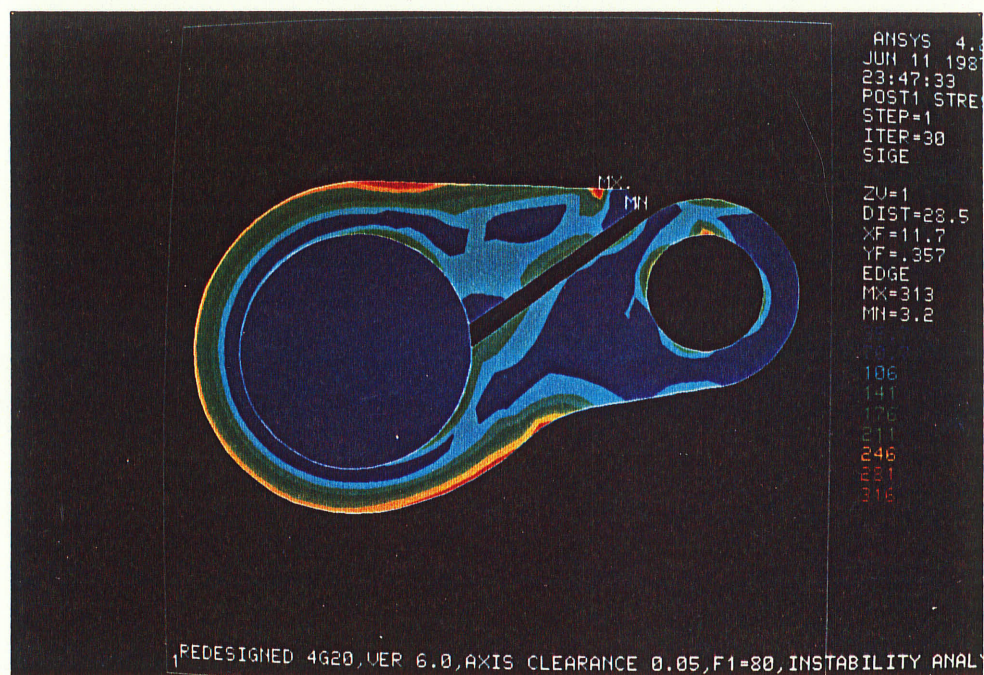


Figure D.17(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=2000\text{N}$.



Figure D17(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=2000\text{N}$.

Appendix E: Equivalent and Principal Stress Distributions For the Titanium Version of the Original Swing Block.

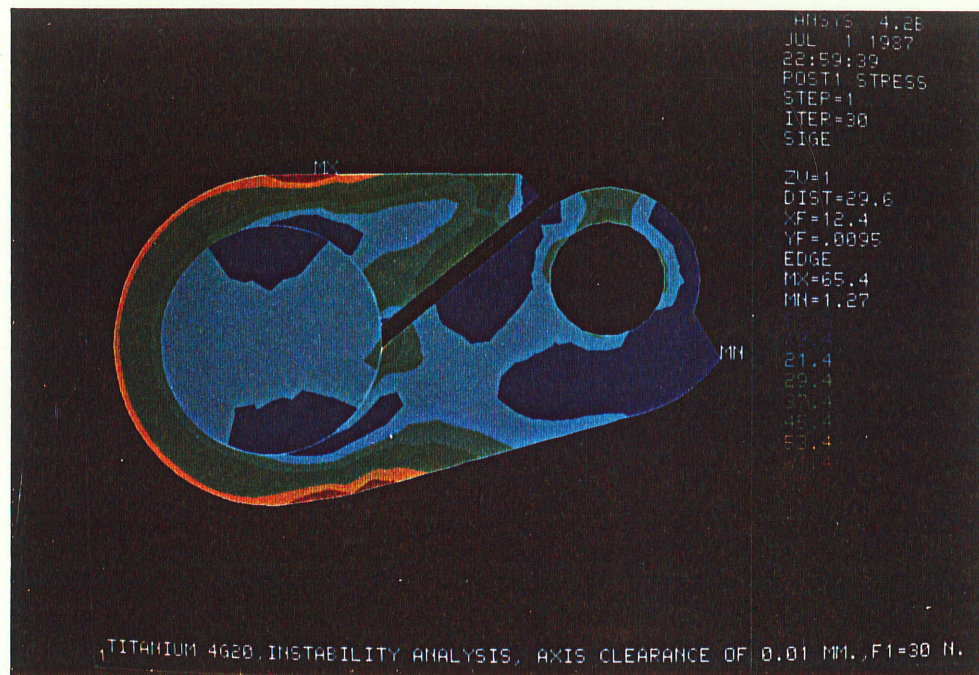


Figure E.1(a): Equivalent Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

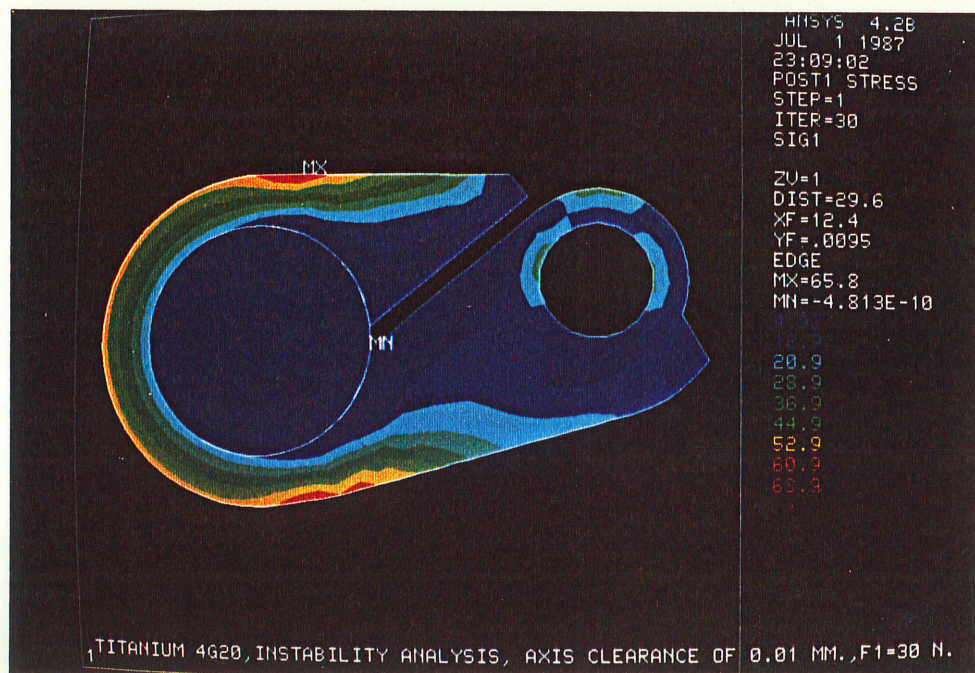


Figure E.1(b): Principal Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

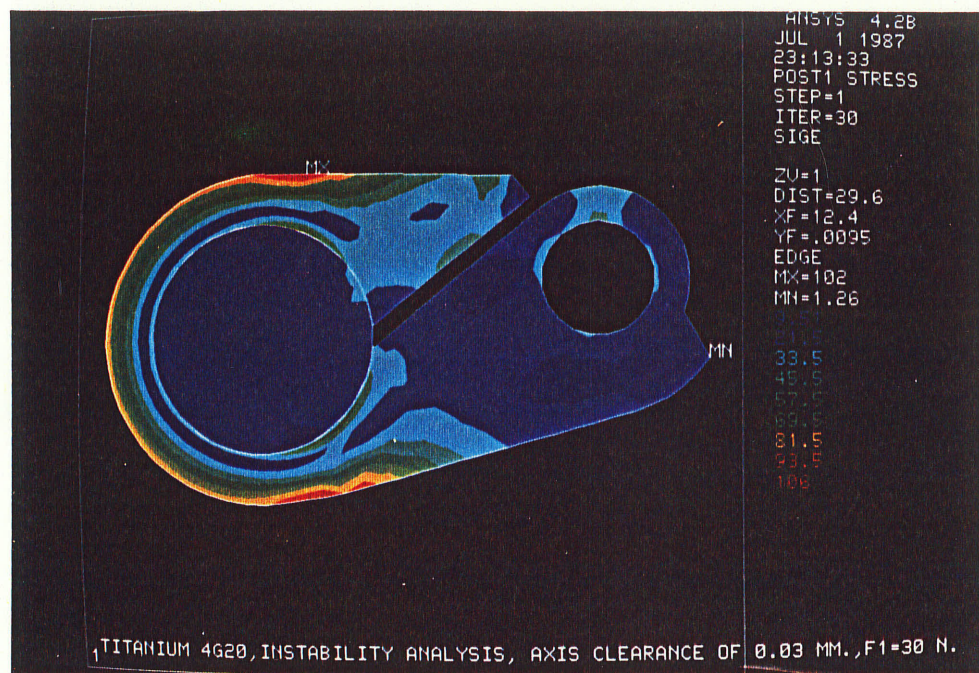


Figure E.2(a): Equivalent Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

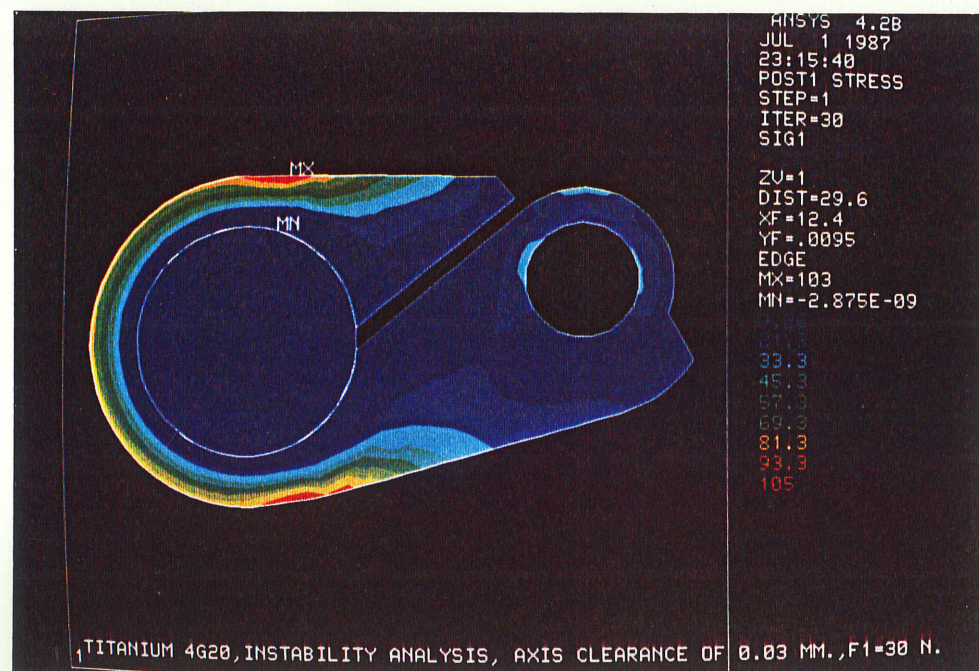


Figure E.2(b): Principal Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

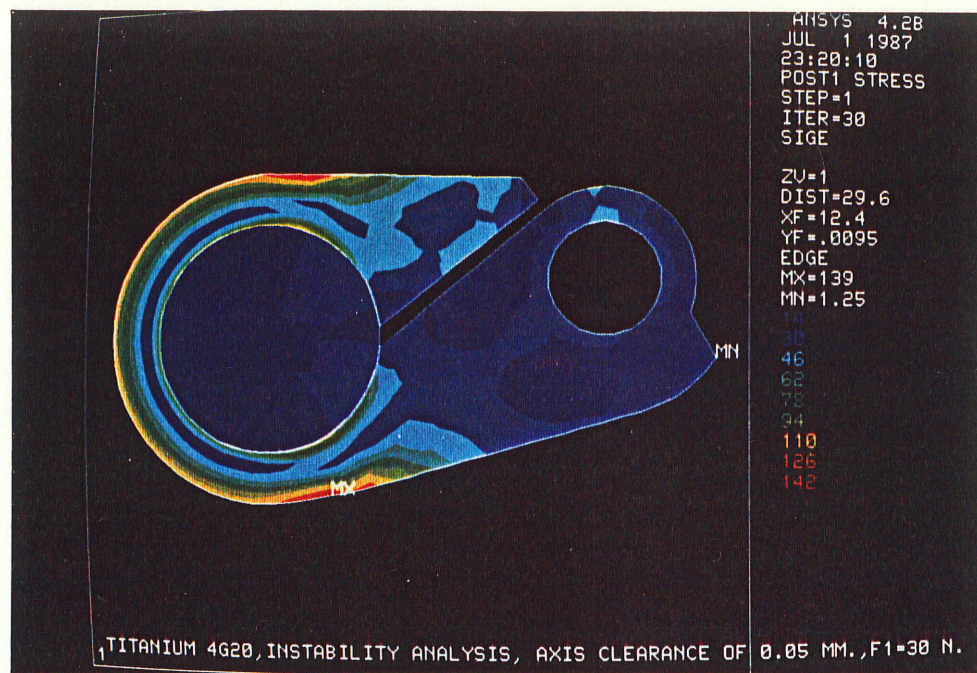


Figure E.3(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

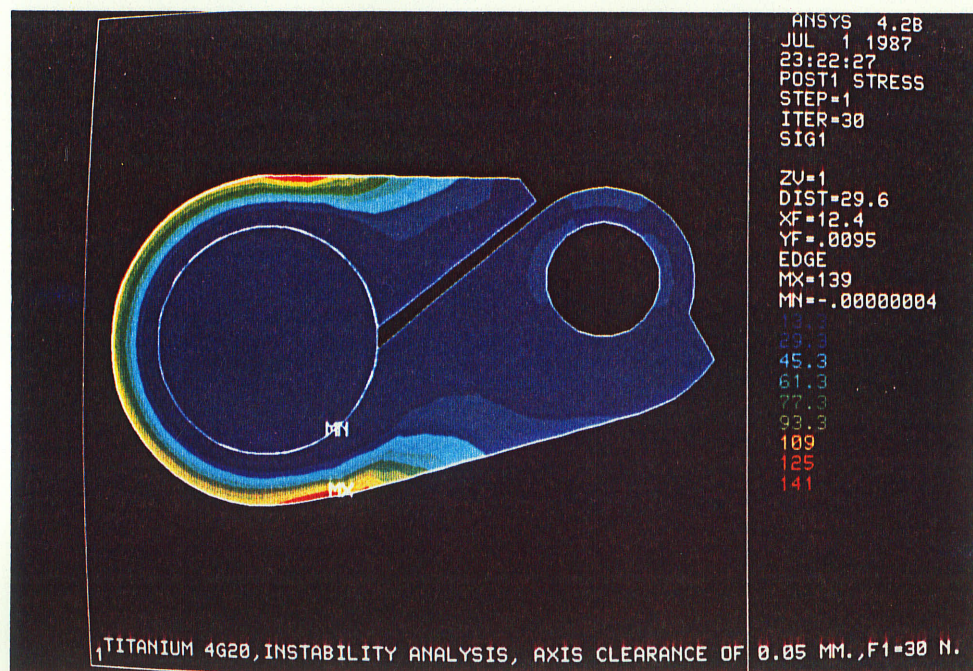


Figure E.3(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

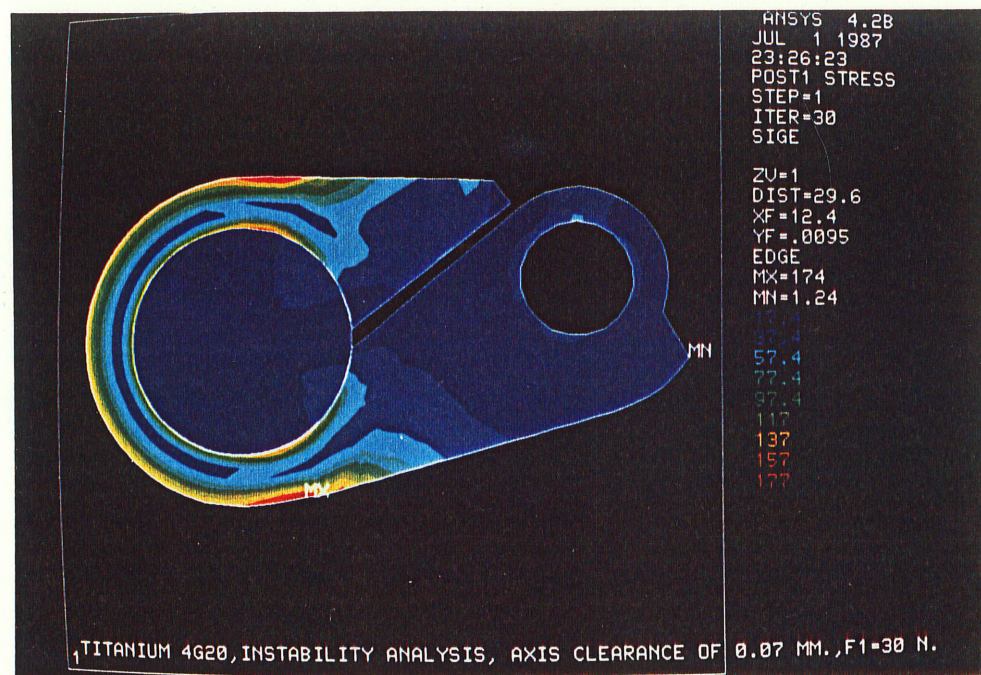


Figure E.4(a): Equivalent Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

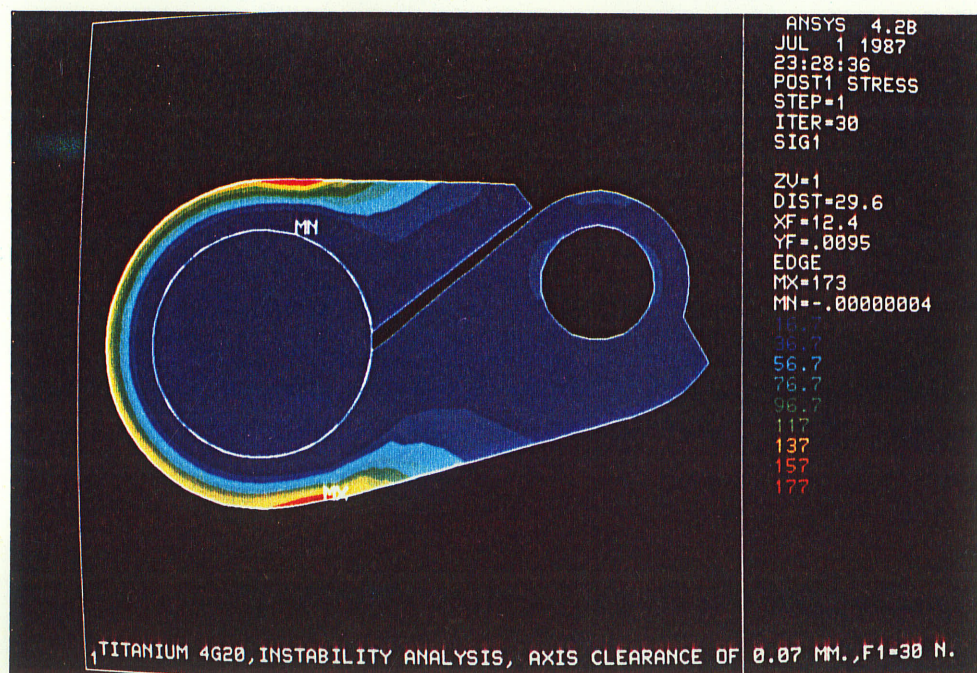


Figure E.4(b): Principal Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

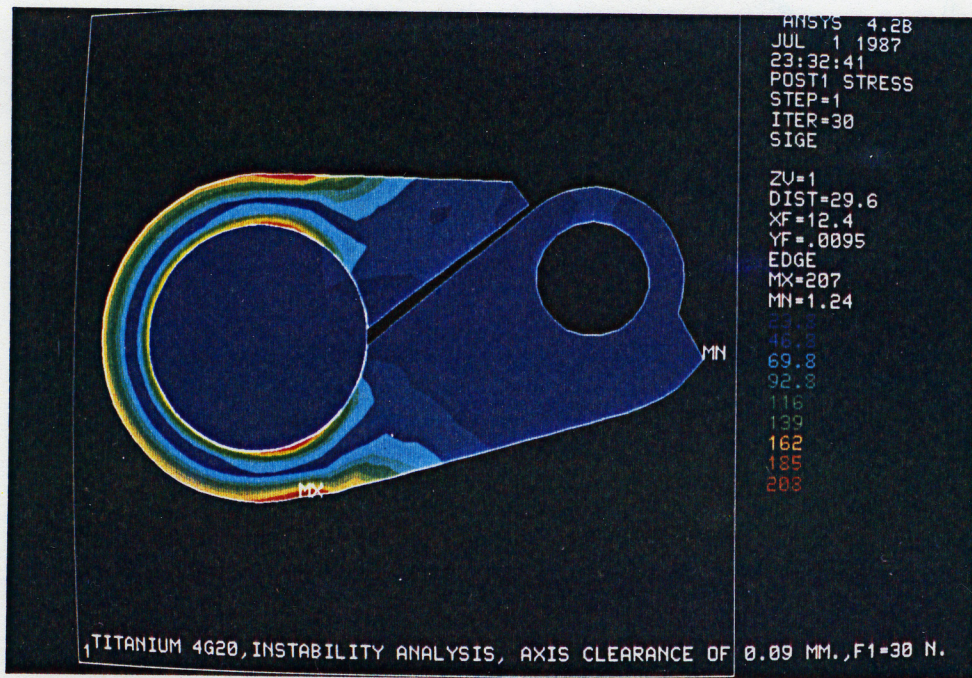


Figure E.5(a): Equivalent Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

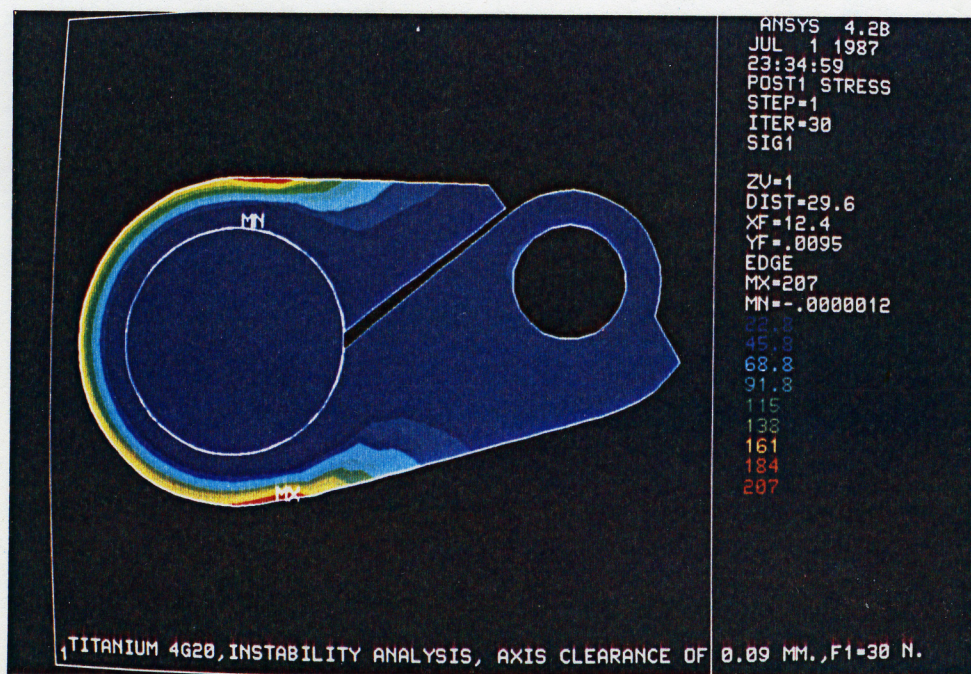


Figure E.5(b): Principal Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

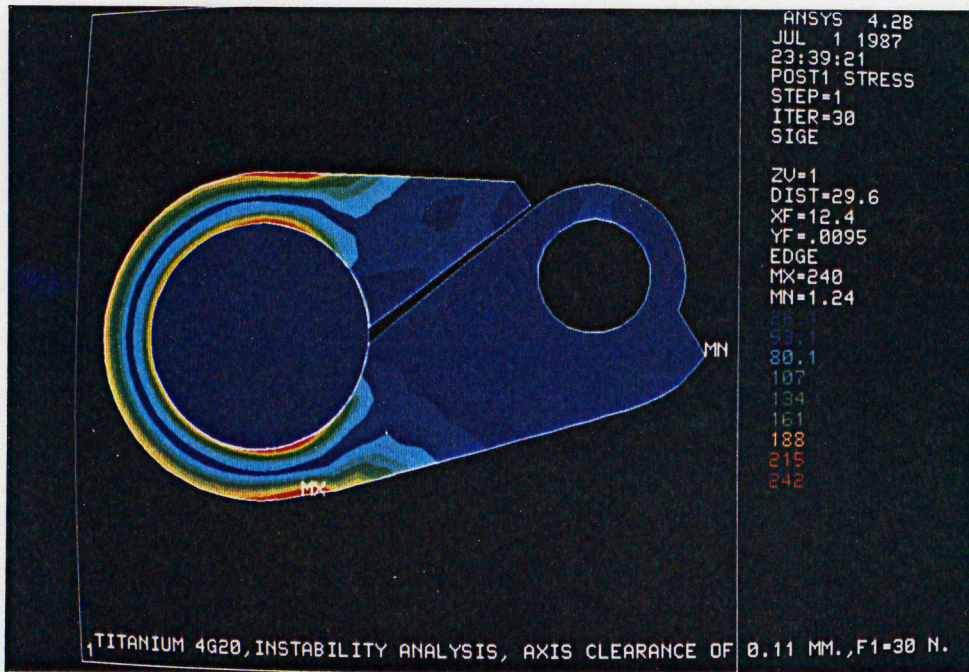


Figure E.6(a): Equivalent Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

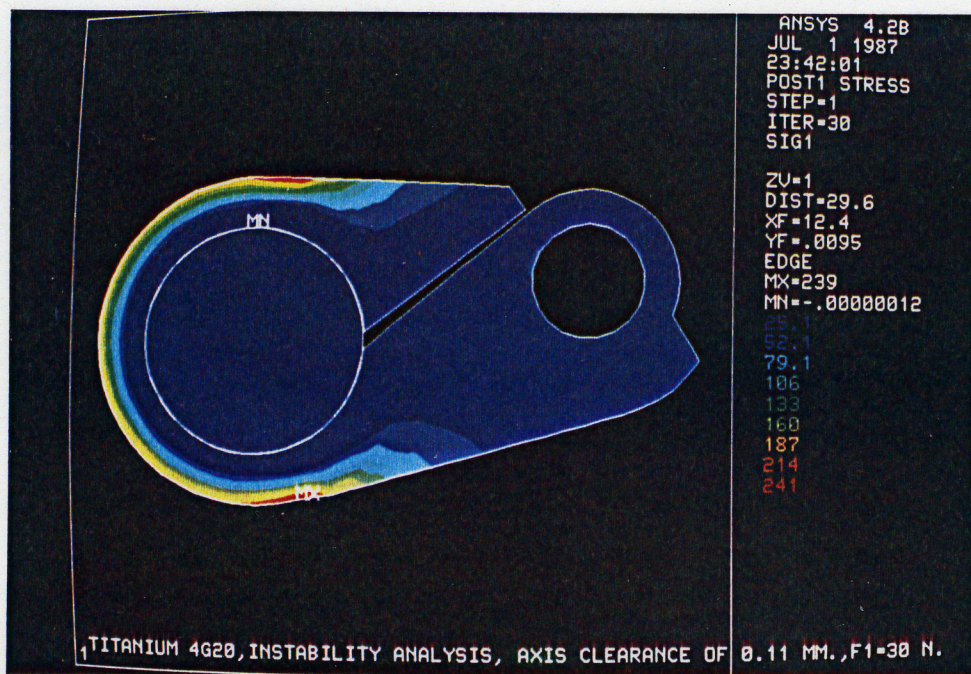


Figure E.6(b): Principal Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

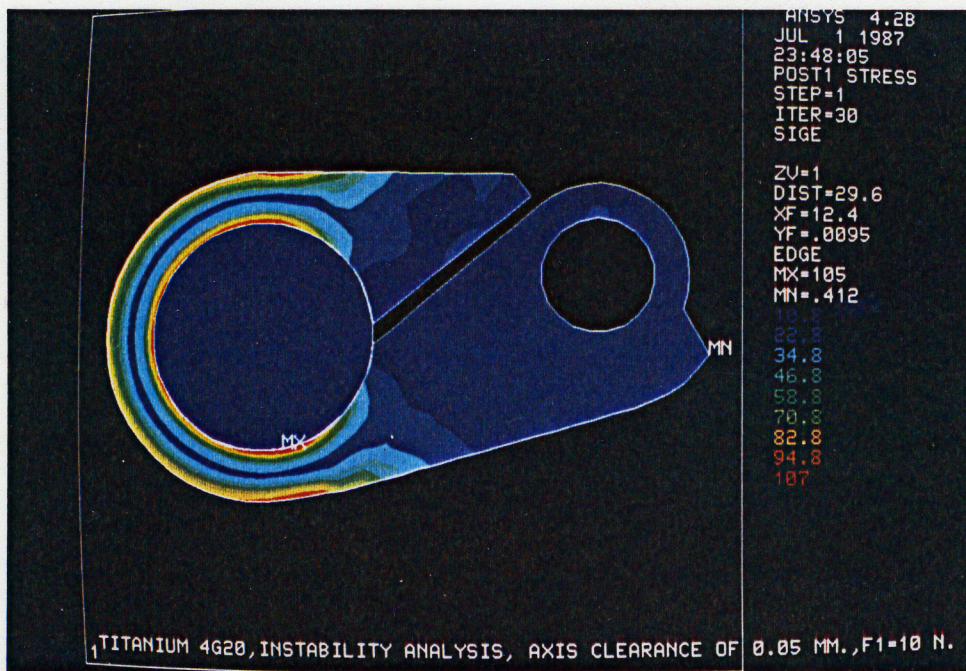


Figure E.7(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

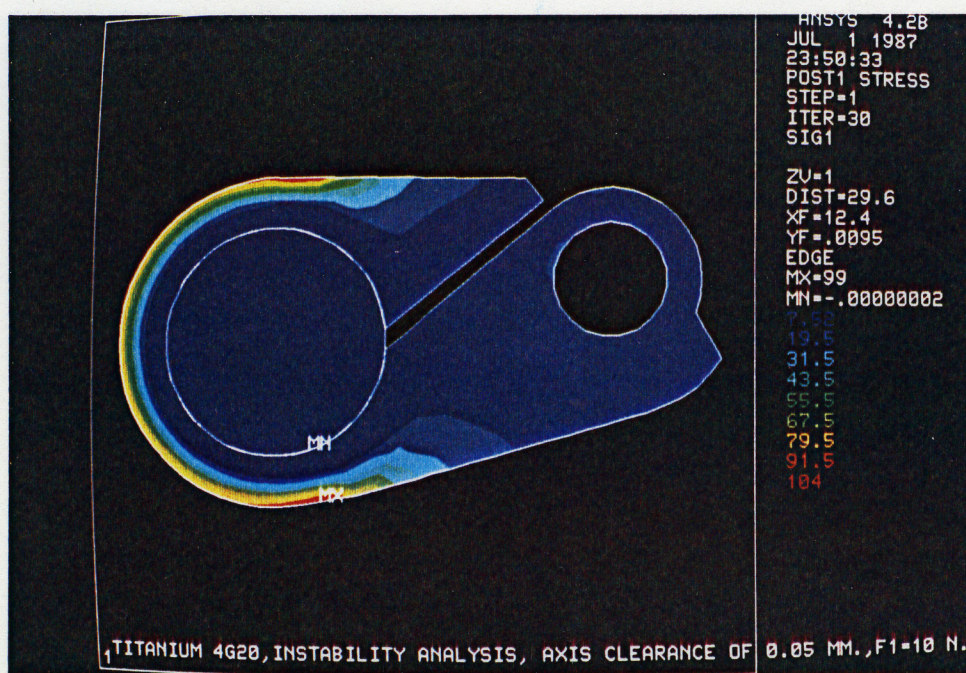


Figure E.7(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

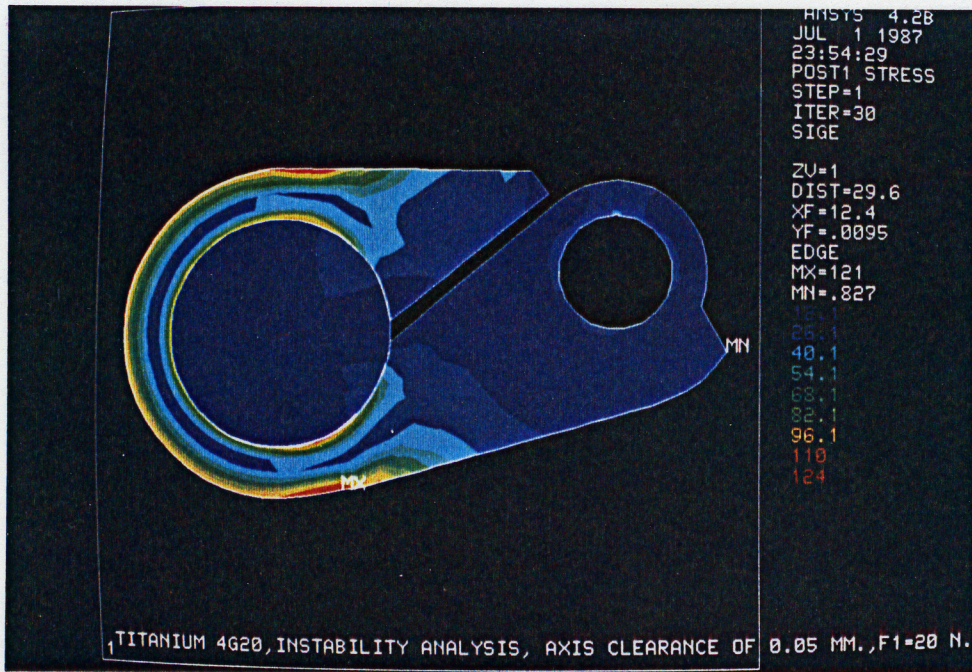


Figure E.8(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

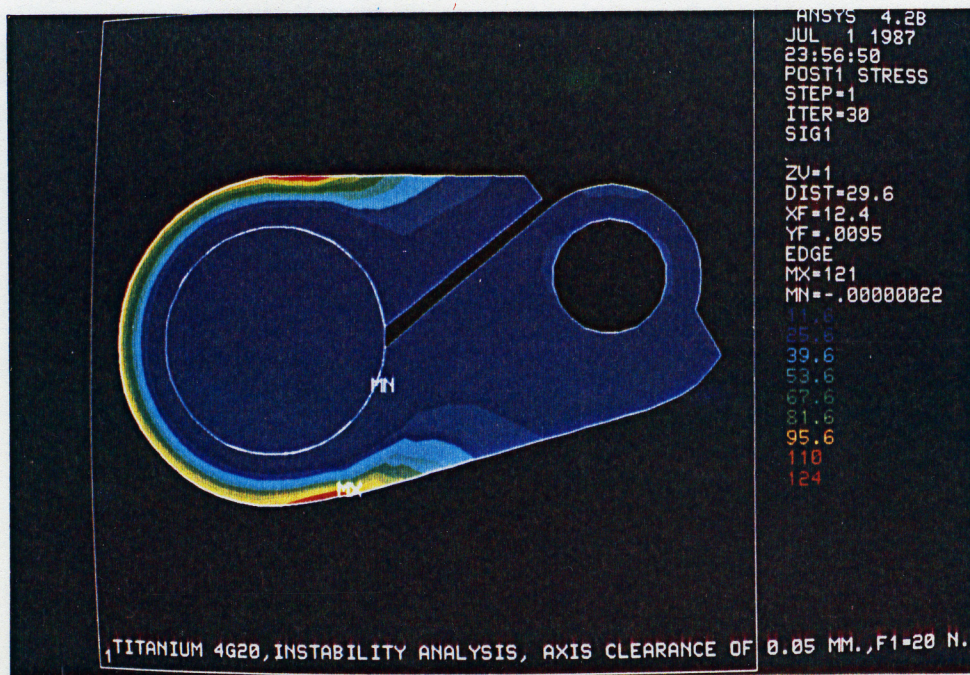


Figure E.8(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

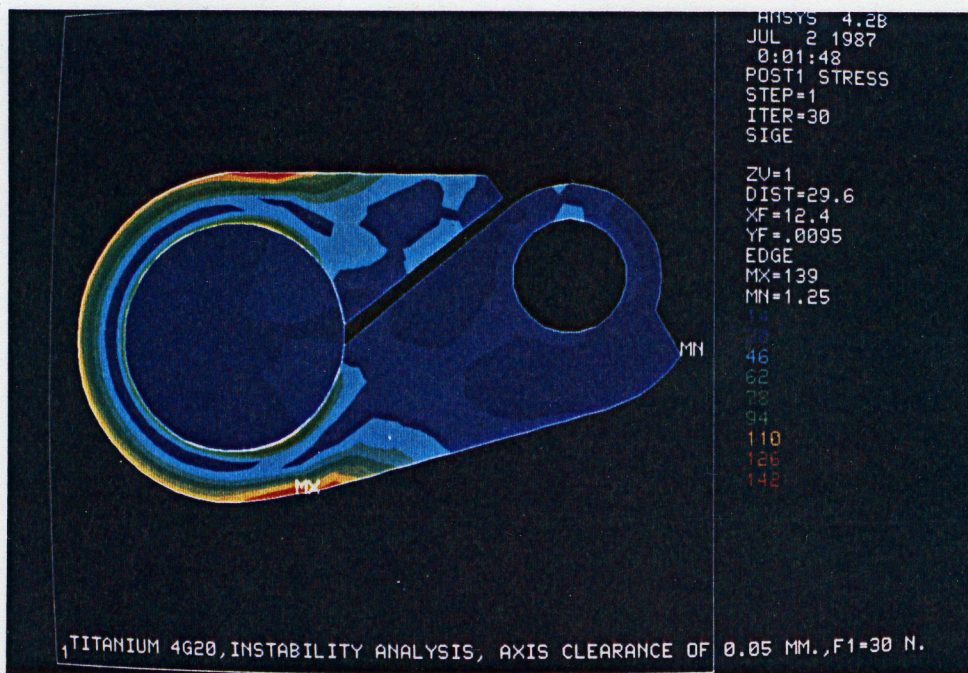


Figure E.9(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

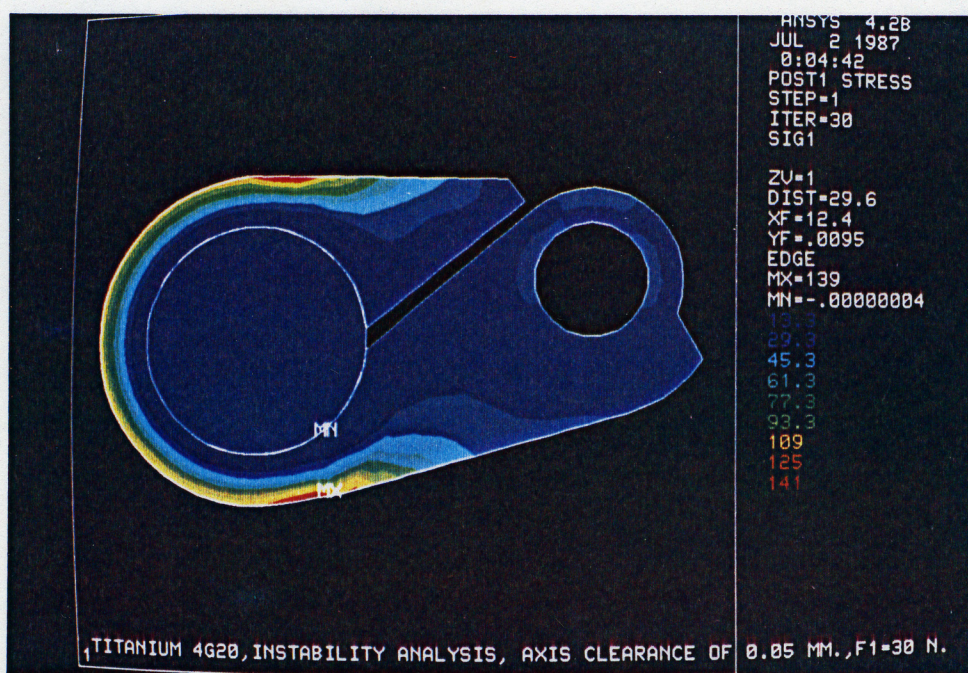


Figure E.9(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

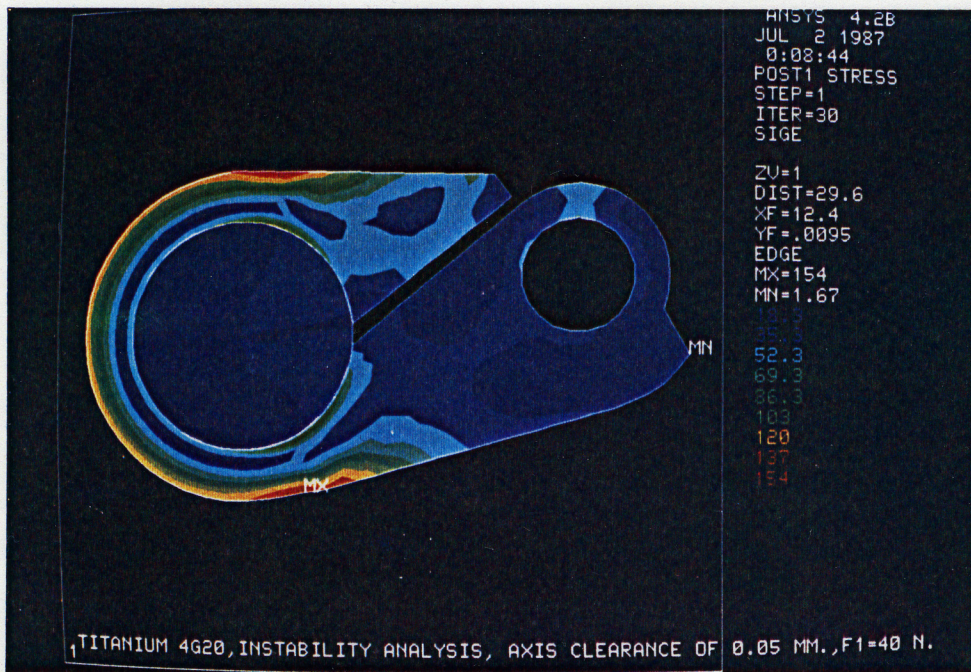


Figure E.10(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

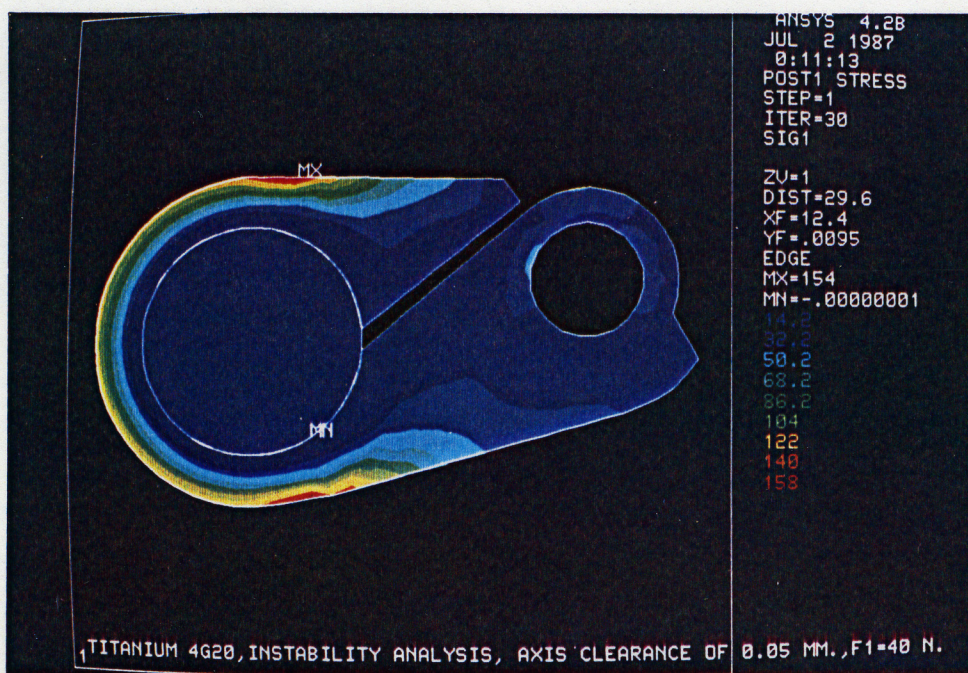


Figure E.10(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

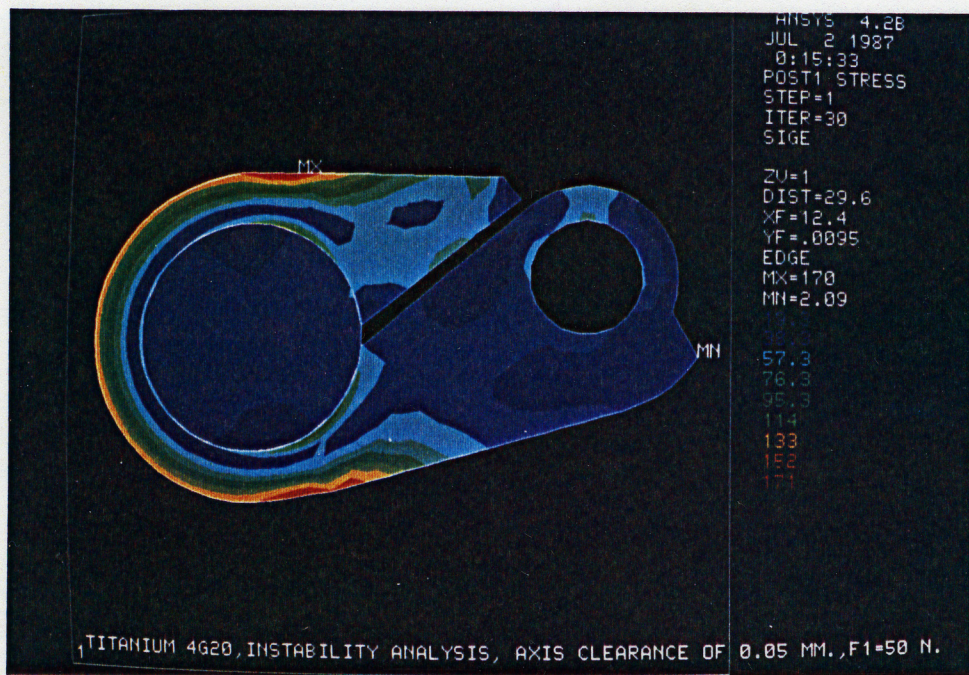


Figure E.11(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1250\text{N}$.

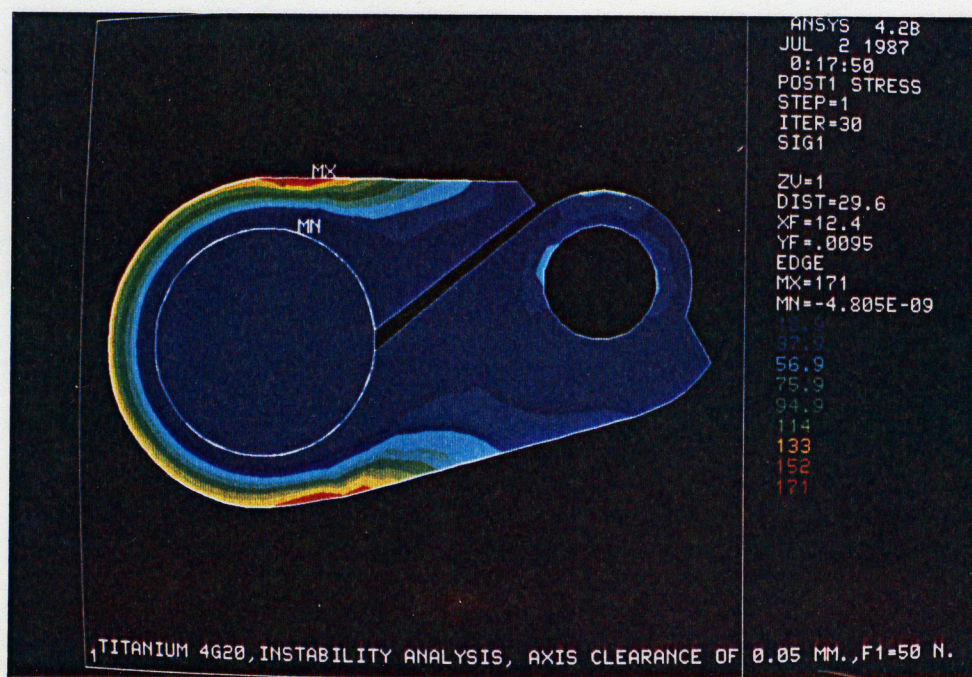


Figure E.11(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1250\text{N}$.

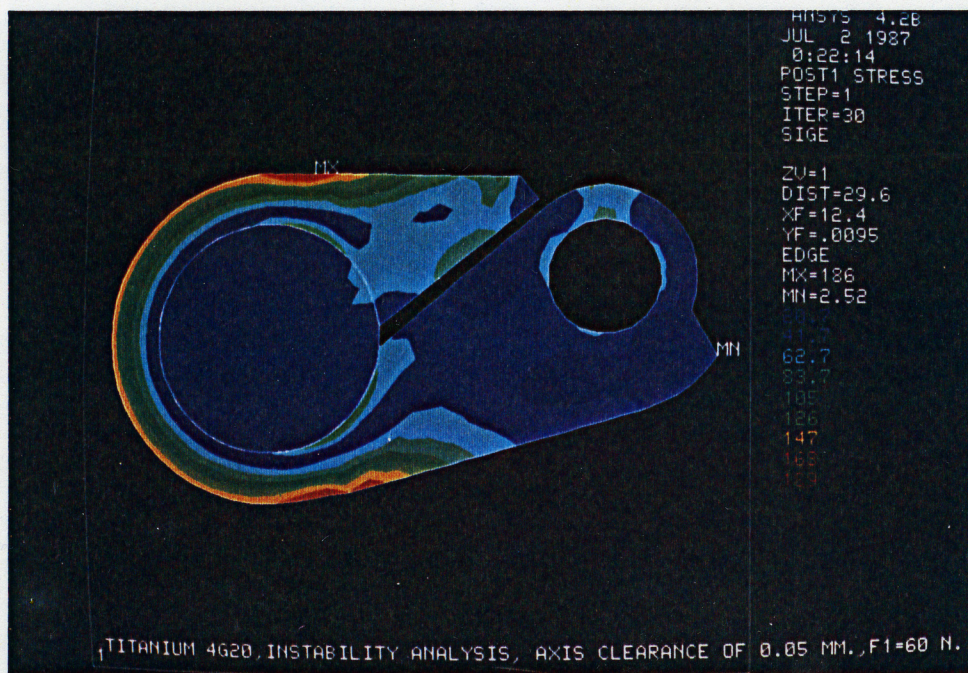


Figure E.12(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.

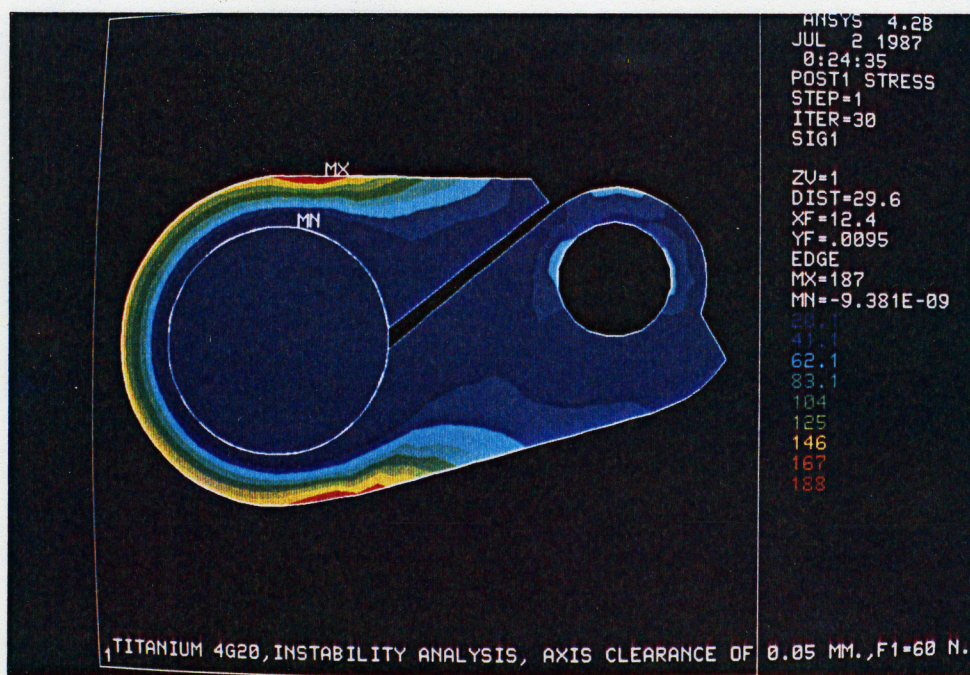


Figure E.12(b): Principal Stress Distribution: $R=0.05\text{mm}$, $W=1500\text{N}$.

Appendix F: Equivalent and Principal Stress Distributions For the Titanium Version of the Redesigned Swing Block.

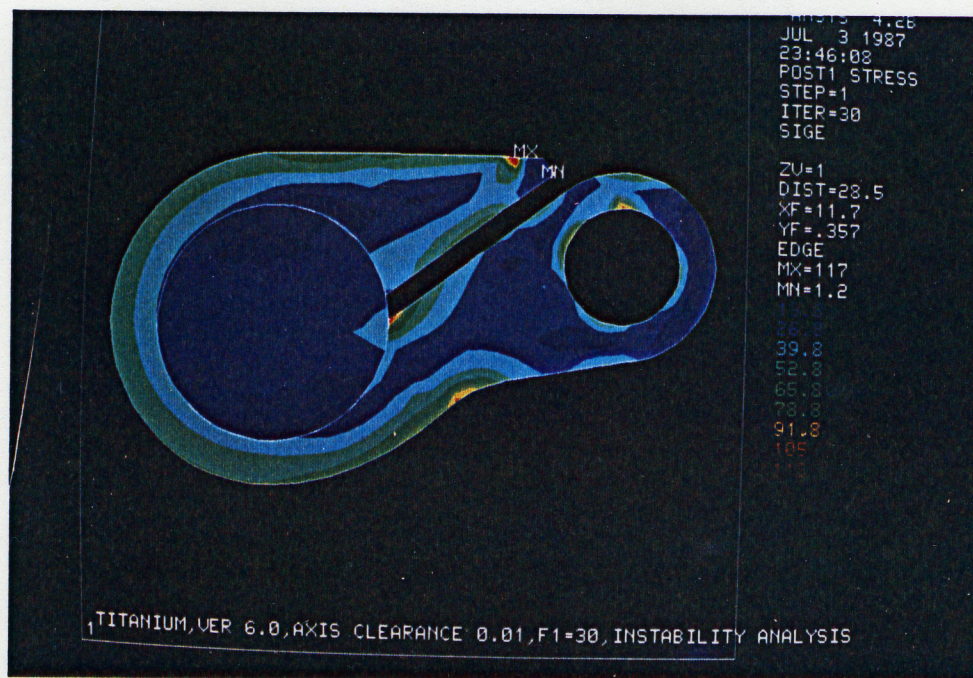


Figure F.1(a): Equivalent Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

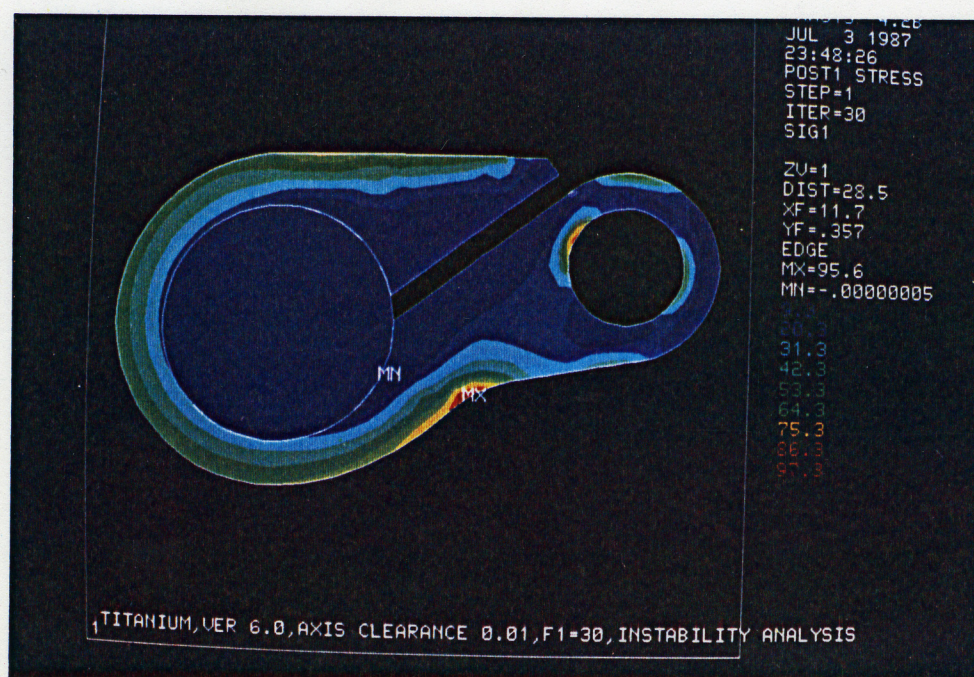


Figure F.1(b): Principal Stress Distribution, $R=0.01\text{mm}$, $W=750\text{N}$.

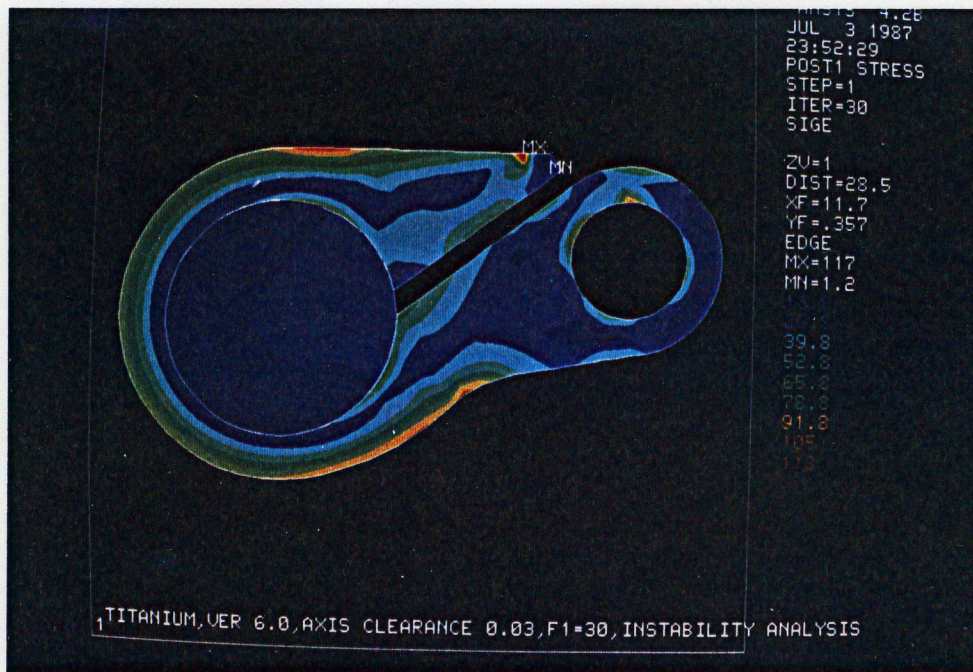


Figure F.2(a): Equivalent Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

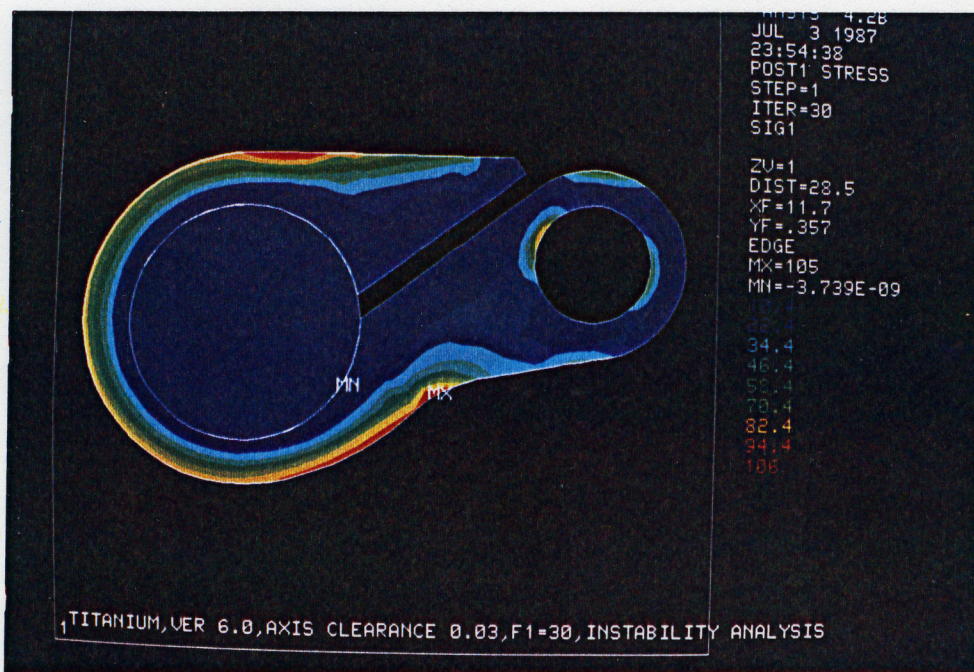


Figure F.2(b): Principal Stress Distribution, $R=0.03\text{mm}$, $W=750\text{N}$.

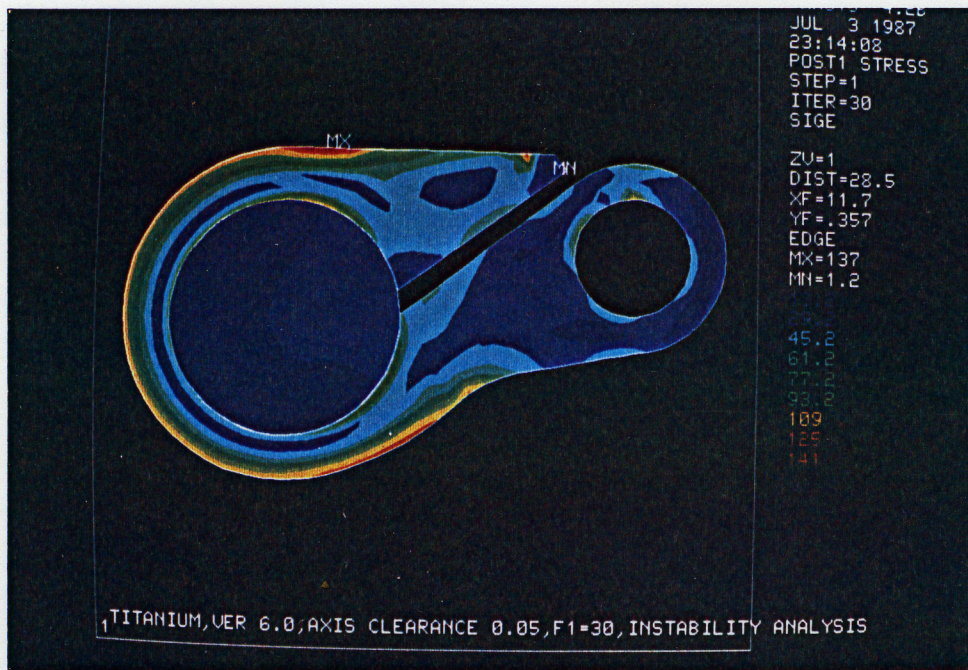


Figure F.3(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

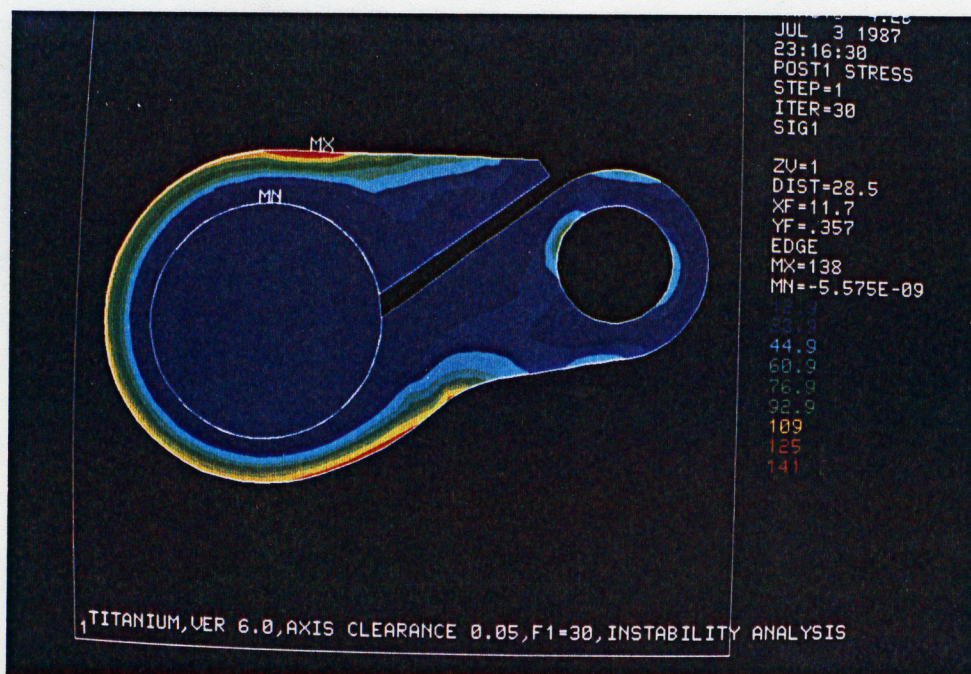


Figure F.3(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

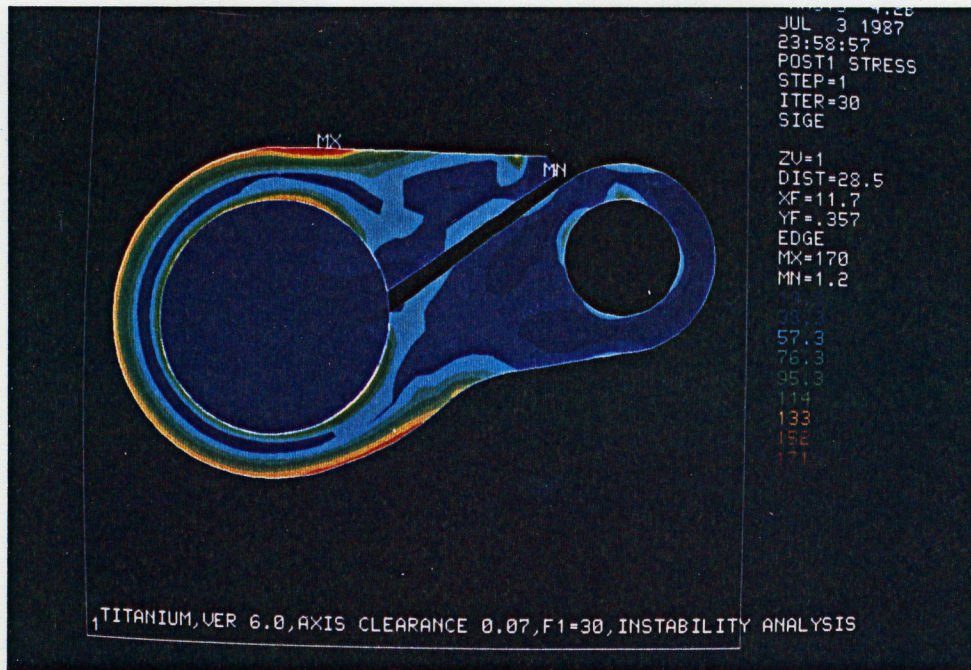


Figure F.4(a): Equivalent Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

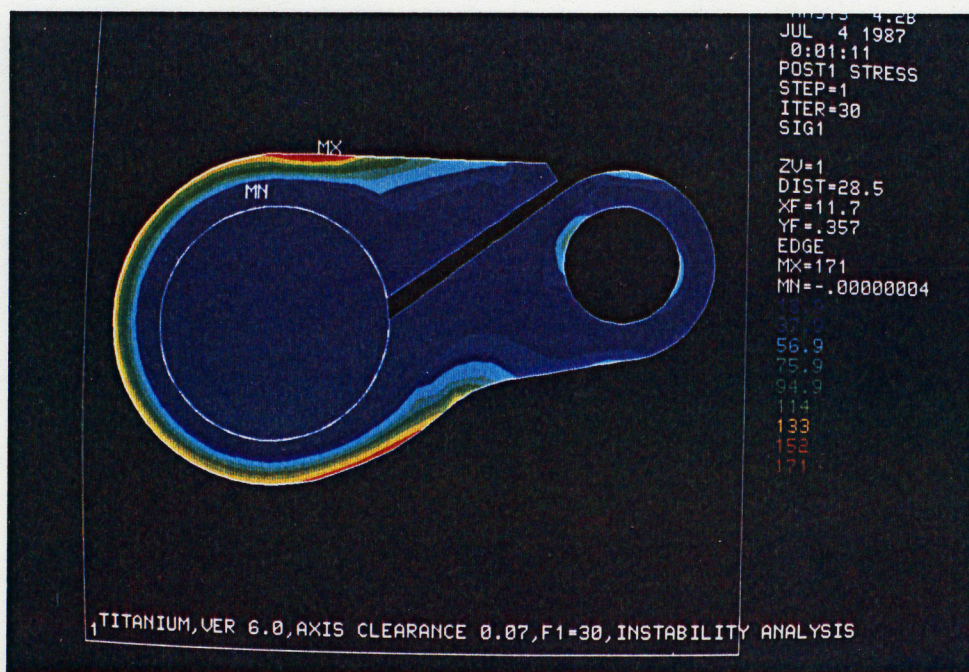


Figure F.4(b): Principal Stress Distribution, $R=0.07\text{mm}$, $W=750\text{N}$.

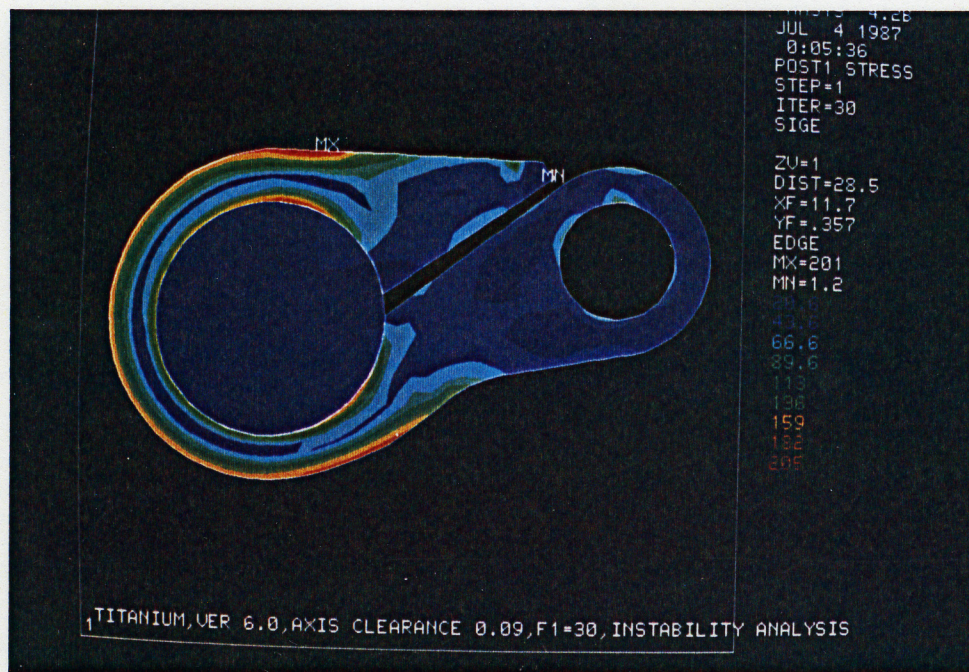


Figure F.5(a): Equivalent Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

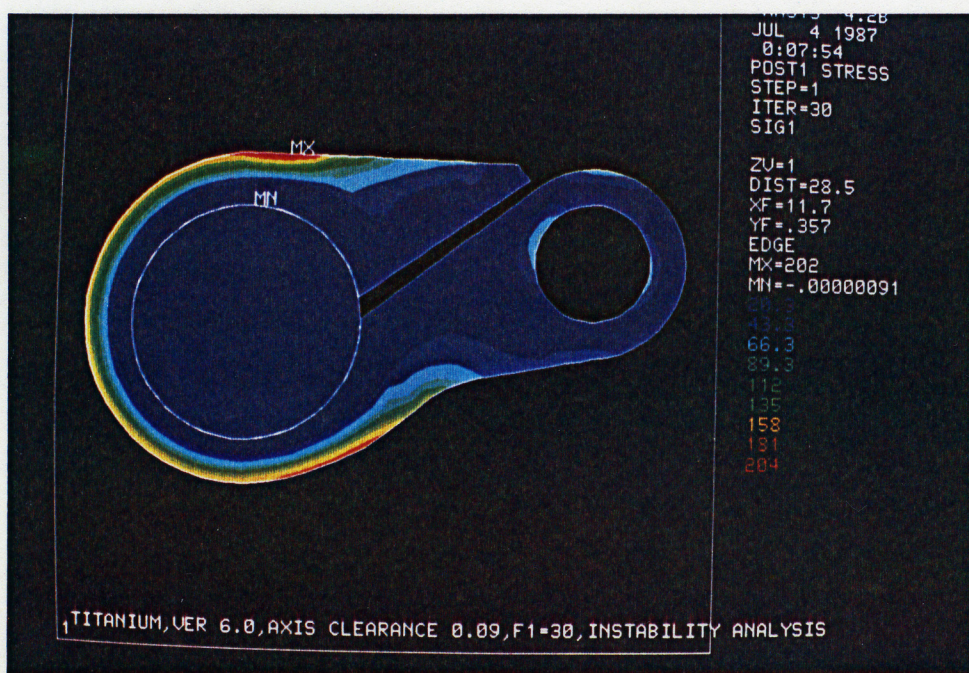


Figure F.5(b): Principal Stress Distribution, $R=0.09\text{mm}$, $W=750\text{N}$.

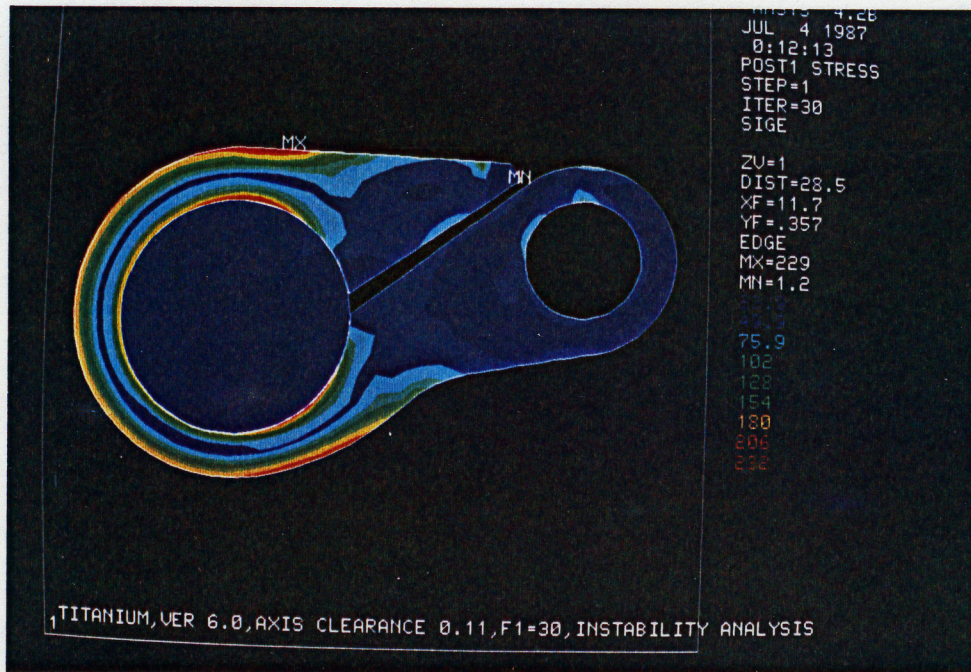


Figure F.6(a): Equivalent Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

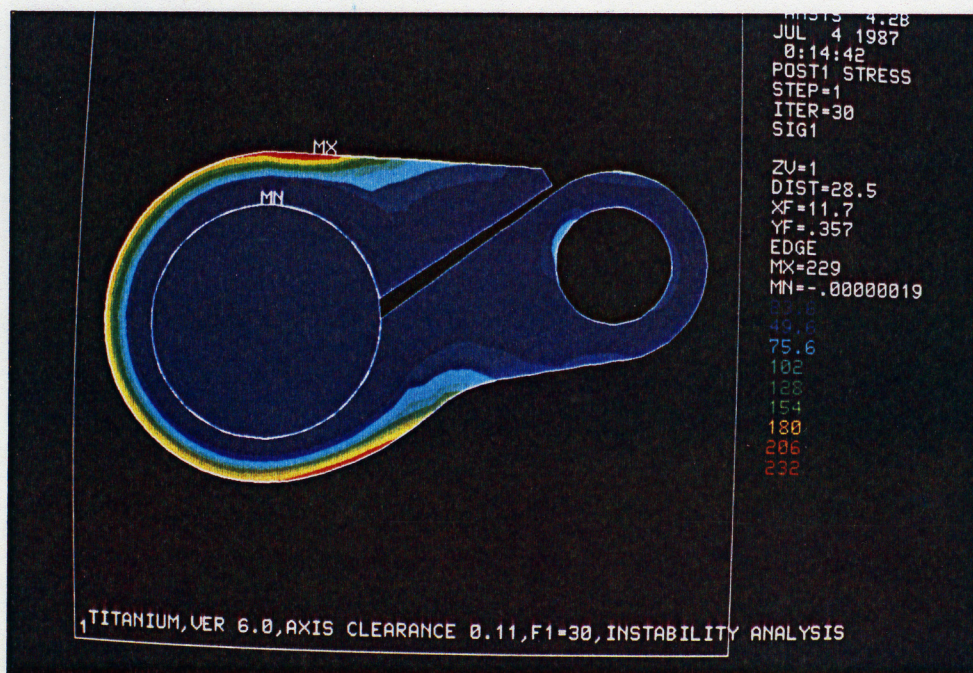


Figure F.6(b): Principal Stress Distribution, $R=0.11\text{mm}$, $W=750\text{N}$.

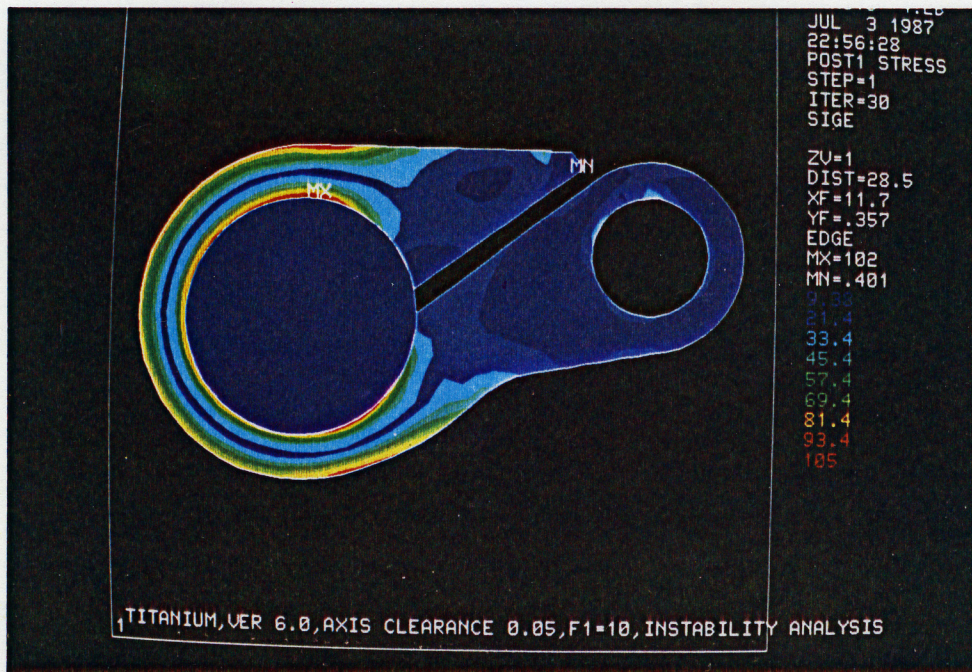


Figure F.7(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

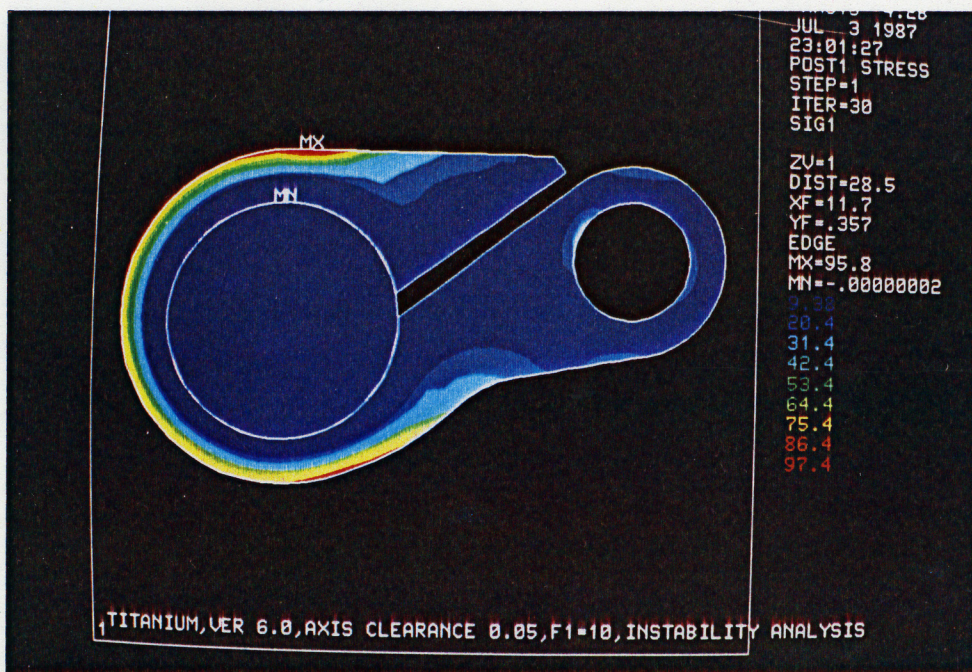


Figure F.7(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=250\text{N}$.

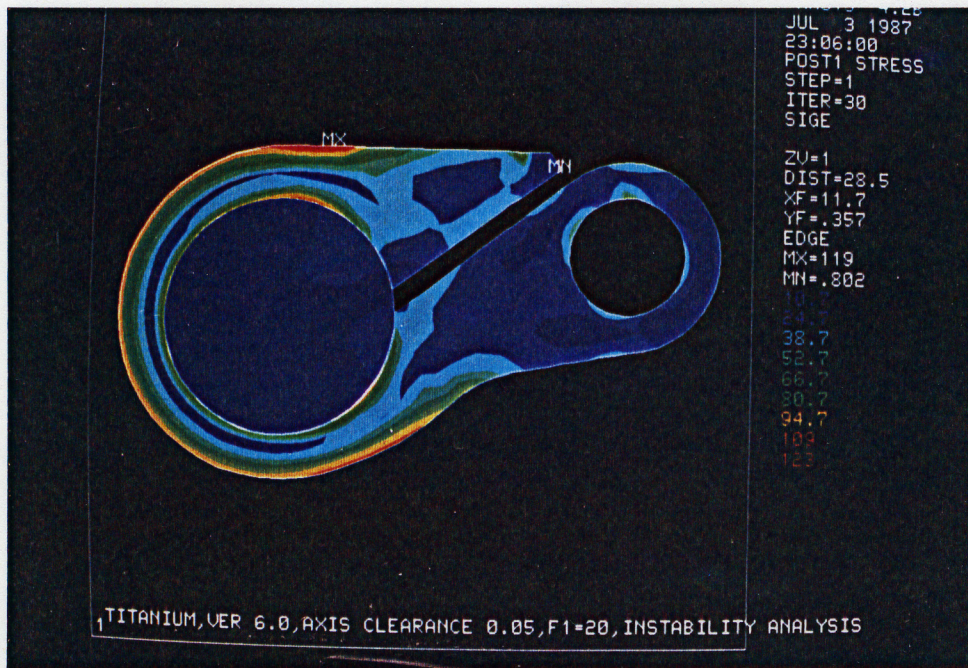


Figure F.8(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

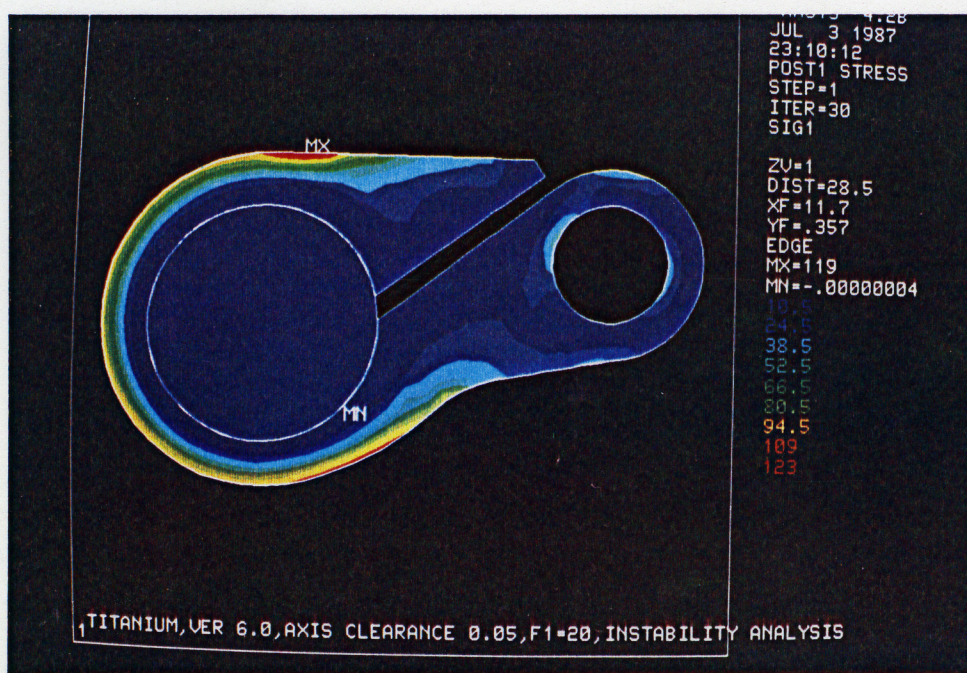


Figure F.8(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=500\text{N}$.

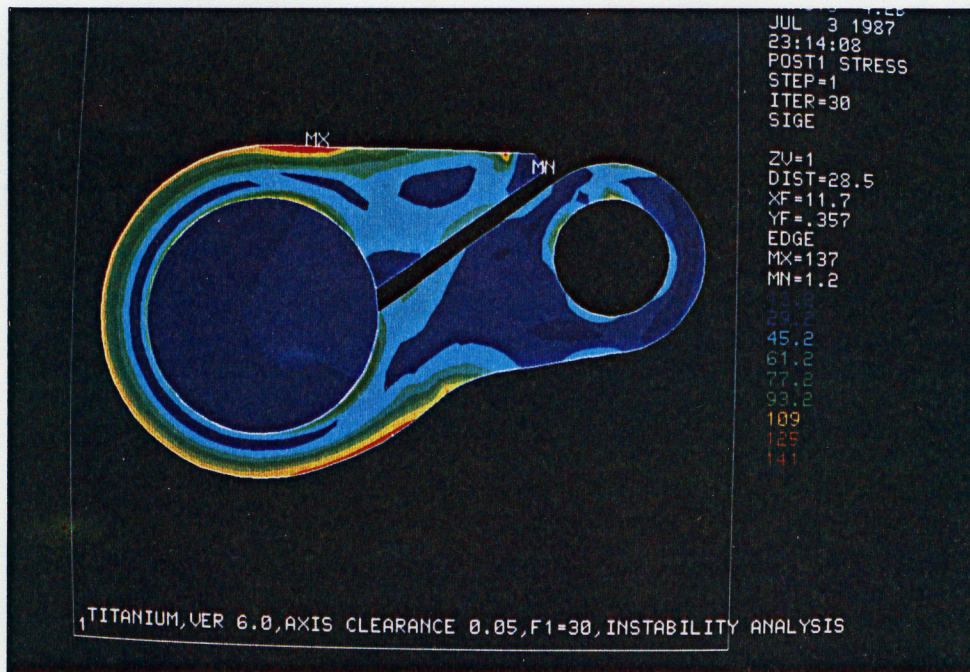


Figure F.9(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

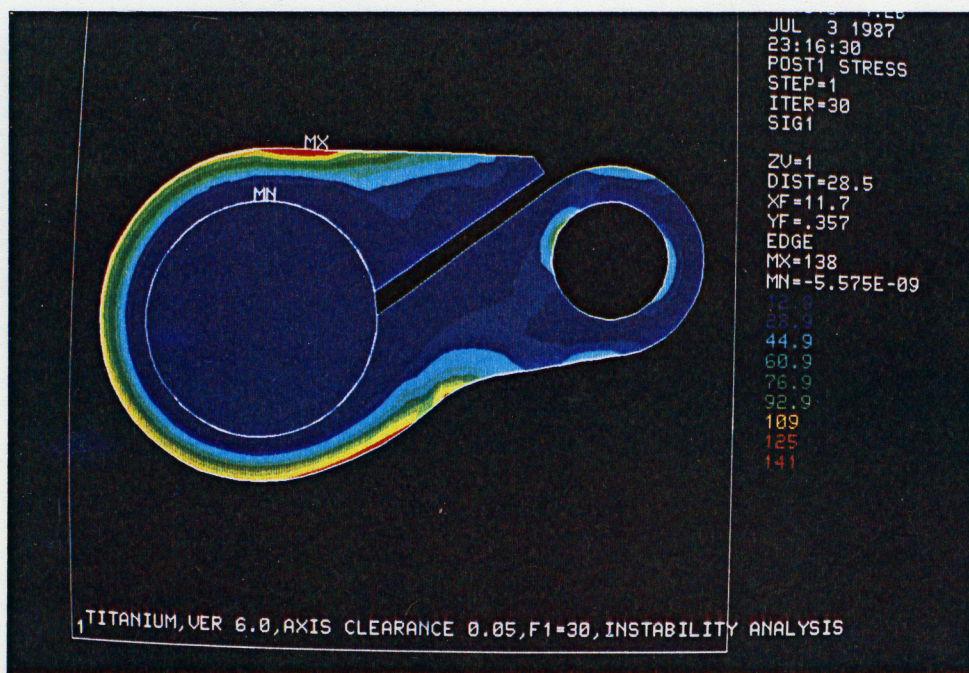


Figure F.9(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=750\text{N}$.

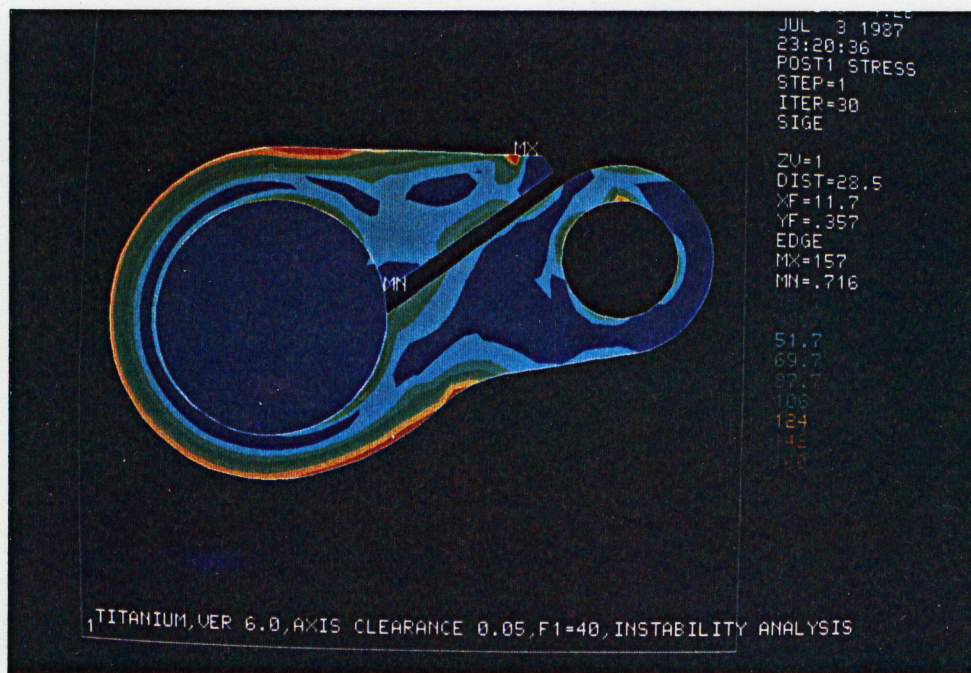


Figure F.10(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.

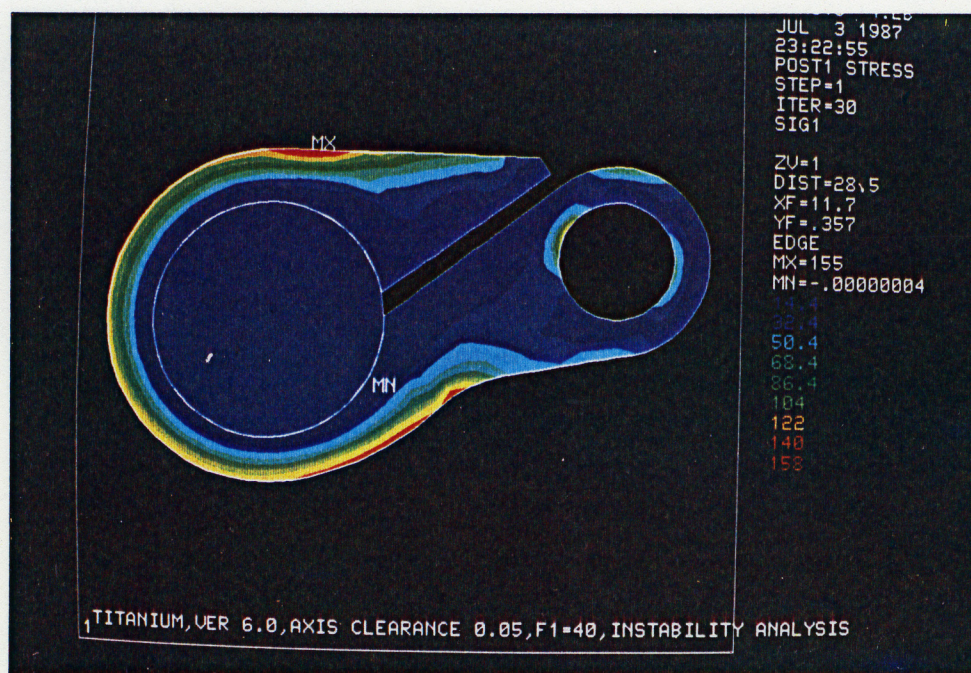
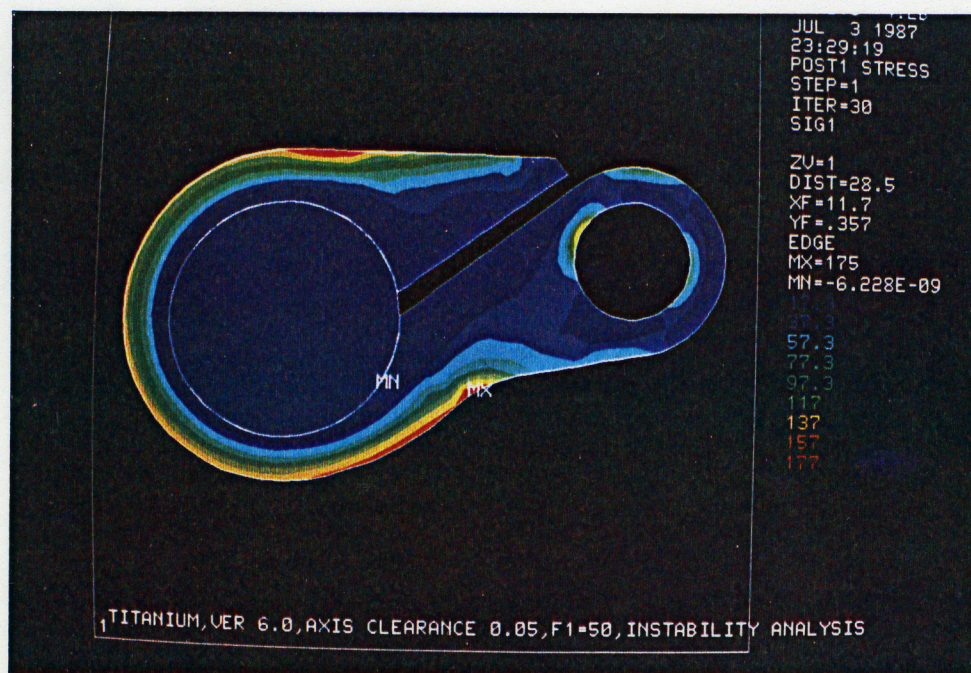
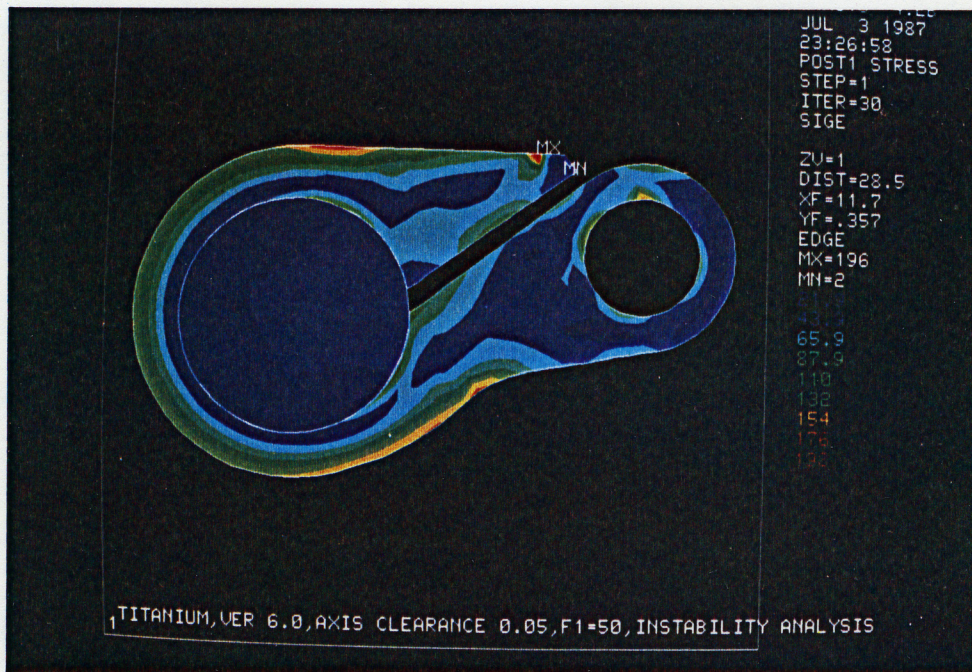


Figure F.10(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1000\text{N}$.



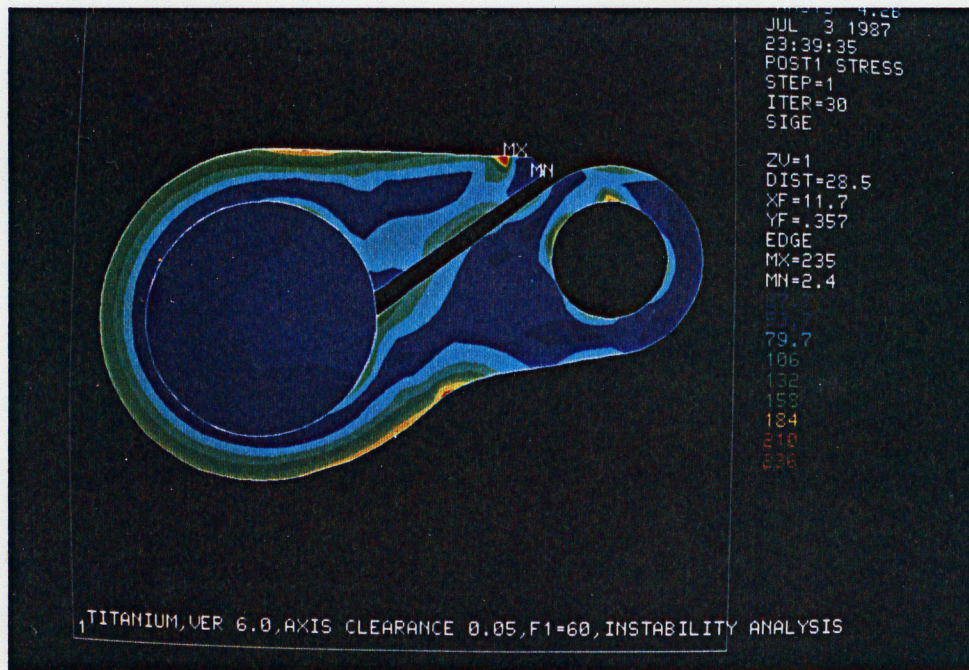


Figure F.12(a): Equivalent Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.

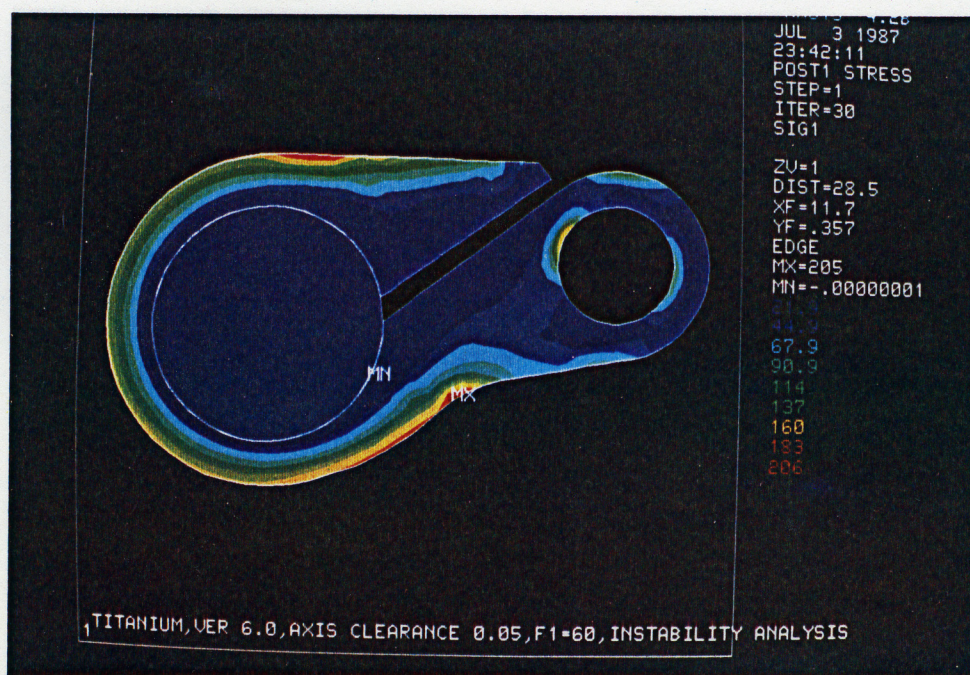


Figure F.12(b): Principal Stress Distribution, $R=0.05\text{mm}$, $W=1500\text{N}$.