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An Engineering Analysis of Three Manually Operated Sunflower Seed Oil Presses Manufactured in Tanzania

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Experimental Results and Plots

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INTRODUCTION

In October, 1987, I conducted for ATI an engineering study of three hand-operated sunflower seed oil presses which were being produced by Themi Engineering in Arusha, Tanzania. These are:

1. An 80-Ton Scissor Jack Press designed by IPI in Dar-es-Salaam.

2. A 20-Ton Scissor Jack Press also designed by IPI.

3. A small Ram Type Press designed by Carl Bielenberg of A.T.I.

I was to assess the three presses (all being manufactured by Themi Farm Implements) in terms of:

1. Function - how well designed are they to generate mechanical force for the extraction of oil from sunflower seed.

2. Durability - how susceptible are they to failure due to stresses generated during operation or due to poor maintenance practices.

3. Appropriateness - how appropriate are they as far as the materials used, ease of manufacture, ease of operation, output, cost, safety etc. and how appropriate are they for use by rural or peri-urban poor.

For each machine I was to consider both the original designs, as specified in the drawings supplied by IPI and also any design changes which had subsequently been made by Themi or IPI. This report was to include any recommendations for new design changes and suggestions on where and how each machine can best be utilized.

In order to properly complete this study it was necessary to carry out a complete engineering force and stress analysis on each of the three machines. Without such an analysis it would not be possible to properly evaluate the machines for function or durability.

First, however, I had to know the forces applied to the machines during operation. These forces could only be determined by conducting experiments and taking measurements on the machines themselves. I proceeded to Arusha to begin my work. There I discovered a number of things which affected the subsequent course of my study.

First, I found that Themi was no longer producing the 80-Ton Press, nor were they planning to resume production of this

press. However, they were still making and testing the 20-Ton Presses and they had started designing and building an (intermediate size) 40-Ton Scissor Jack Press.

Second, Lynn Schlueter, Lutheran World Relief[1] Representative, told me that in addition to Themi, the Bielenberg Ram Press is being manufactured by Nand Singh Ltd., and United Engineering, both in the town of Arusha. All three manufacturers produce slightly different models of the press, but the Themi and the United designs are quite similar. At the time of my visit, the project was ordering machines from both Nand Singh and Themi. For reasons of quality control, Lynn preferred the Nand Singh machines. The United machines were slightly more expensive and none had been ordered in quite some time. I was asked to examine the Nand Singh press and to try out specific design changes to either reduce the cost and/or increase the efficiency of the Ram Press. I agreed to include both tasks.

The 80-Ton Press was still being produced by IPI in Dar-es-Salaam and I decided that it would still be worthwhile to include in my study despite the fact that Themi was no longer manufacturing them. However, to save time I examined an 80-Ton Press manufactured by IPI that is being operated in Arusha town by a Mr. Ferry Klemann. There are no operating Themi 80-Ton Presses within easy access of Arusha.

I chose not to work with a new 40-Ton Press. It was still in the early stages of design and the analysis would be very similar to that of the 20- and 80-Ton Presses.

I carried out experiments to determine the forces during operation for each of the four presses and also ran an experiment to compare the yield from the 20-Ton Press using decorticated seed with the yield for the same 20-Ton Press using undecorticated seed. I demonstrated that even in the large batch presses, increased pressure can fully compensate for the lack of decortication and other preprocessing.

I suggested design changes for the Ram Press and supervised the construction of a new prototype at the Nand Singh workshop. This and other prototype design changes were tested and the results were encouraging.

This report is divided into four chapters. The first chapter outlines the general philosophy and approach as well as the specific design criteria and analytical and experimental techniques used for the analysis. The following three chapters discuss the 20-Ton Press, the 80-Ton Press and the Ram Press. The details of the stress analysis are included in the Appendices.

In each of the last three chapters, I have included a brief description of the press, general comments and observations about the presses, descriptions of the experiments carried out

[1] Lutheran World Relief is the implementing organization for the ATI Oil Press Project in Tanzania.

and their results, tabulated summaries of the stress analysis, and my conclusions and recommendations.

CHAPTER 1

APPROACH TECHNIQUES AND DESIGN CRITERIA

1.1 INTRODUCTION

In normal engineering practice the task of determining all of the stresses in a manually operated press would be simple. Electronic strain gauges would be attached to the parts in question and the stresses would be measured directly as the press was operated. However, no strain gauges were available and even the device previously used to measure the frame elongation instead could not be located. In this situation, a combination of simple experimental and analytical techniques had to be used.

Because all three presses analyzed here are fixed displacement, rather than fixed force machines, (a hydraulic jack is a fixed force machine) it is the mechanical properties of the compressed sunflower seed which determine the forces and thus the stresses in the machines' components. In fact, the essential mechanical property is the force versus displacement (or stress vs. strain) relationship for the compressed sunflower seed. If you compress a sponge by 10 mm you obviously generate much less force than if you try to compress a steel rod by that same 10 mm.

All three presses have variable theoretical mechanical advantages (defined as: the force generated by the plunger divided by the force applied to the handle) which approach infinity at the end of their compression strokes. Theoretically, each is capable of generating incredibly high forces (approaching infinity) while requiring very little effort. In reality the generated forces will be much lower because the seed cannot withstand such a high force and will simply deform instead.

For the purpose of this analysis I devised a simple technique for approximating the actual forces on the seed as a function of plunger displacement. First, I used 50 Kg. spring balances to measure the force applied to the handle of the press during normal operation. Then I calculated the generated forces by using an analytical formulation of the press's mechanical advantage as a function of plunger position. Once the generated forces are known the complete stress analysis can be completed. This technique is discussed in more detail in the following section.

There are many unknown quantities involved with any exercise in engineering design and analysis; quite a number of approximations and assumptions have to be made. In the case of a Scissor Jack Press the coefficients of friction in the screw threads and the thrust collar are unknowns, as are the material properties of the mild steel used in the

construction. Other unknowns include the relationship between oil output, pressure and time for the compressed sunflower seed and the exact mechanisms of friction and slippage between the seed and the cage of the Ram Press.

In a "perfect world" one would like to carry out a series of laboratory experiments to determine the properties of the materials used to make the presses and the properties of the compressed sunflower seed. Instead one has to make do with the equipment and information at hand.

In the following subsections I describe how I have tackled some of these questions, the approximations and assumptions I have made, and the general procedures I have used to complete the stress analysis.

1.2. EXPERIMENTAL DETERMINATION OF FORCE VS. DISPLACEMENT CURVES

As explained in the previous section, before one can do a proper stress analysis of a fixed displacement oil press one needs to know the relationship between the force generated on the seed and the displacement of the plunger. In this section I describe a simple technique for determining this relationship. This technique can be usefully adapted in the field for stress analysis of any other type of fixed displacement, hand operated pressing device.

1. First, a theoretical calculation of the mechanical advantage of the device as a function of both handle and plunger position must be made. This usually involves simple geometry and force analysis but it is important to include frictional forces and any other losses one can account for analytically.

2. Then a simple spring balance is attached to the handle or lever in a measured position, the press is loaded for normal operating conditions and the operating force is applied to the other end of the spring balance instead of directly to the handle. In this way the applied load can be measured directly from the spring balance.

3. The applied load is recorded as a function of piston displacement during a single piston stroke.

4. The theoretical calculation of mechanical advantage as a function of displacement is used to calculate the generated force as a function of displacement. This is the result you need for the stress analysis.

This technique can work equally well on the Scissor Jack Presses, with the spring balance attached to the end of the turning handle, as it can on the small ram press with the

balance on the end of the lever arm.

A few observations and cautions are useful here:

- For presses such as these for which the mechanical advantage approaches infinity at the end of the compression stroke, the applied force on the handle reaches a maximum value somewhere in the middle of the stroke and then decreases again at the end of the stroke, at which point the generated force reaches a maximum. Both of these maximums are important for the stress analysis calculations. These stresses in the handle are largest when the applied load is largest and the stresses in the cage and other components are largest when the generated force is largest.

- Because the mechanical advantage approaches infinity, one cannot trust the final applied load readings. Near the end of the compression stroke these values quickly approach zero.

- In these tests, the force vs. displacement curves for pressing sunflower seeds are very time dependent. If seed is left for a time under a fixed piston displacement, oil will be squeezed out of the seed and the pressure will drop. If seed is pressed very slowly the forces and thus the stresses in the machine will be substantially lower than if it is done at a high speed. For the purposes of design criteria it is best to press the seed at a speed as fast or faster than is done in actual practice. If one designs for these forces by incorporating an appropriate safety factor, one can be sure the machine will work well even if it is grossly overloaded.

- There are many different types of sunflower seed; all have a slightly different force vs. displacement curve. Even a varying moisture content in a given type of seed may substantially effect the results.

I have carried out this test on the Themi 20-Ton Press, the IPI 80-Ton Press and on both a Themi and a Nand Singh Ram Press. I used the readily available, fairly soft shelled, sunflower seeds provided by Lynn Schlueter for the Ram Press tests and used decorticated and pretreated sunflower seed on the 20- and 80-ton presses. I have plotted the results as generated force versus piston displacement and the plots are included in the appendices (page A).

The data from these experiments provide a good starting point for a sensible stress analysis and a more complete understanding of the working of the three presses.

1.3 DESIGN PHILOSOPHY

When designing "appropriate technologies" for use by the rural or peri-urban poor in Third World countries, it is advisable to adhere to one of two design criteria. Either the devices should be so simple in design and structure that they can be easily repaired and maintained by the users and less sophisticated mechanics found in the rural areas, or, they should be designed so that they are unlikely to break down over a useful lifetime if operated and maintained in a "normal" manner. Experience has shown that "normal" usage by the rural poor may be quite different from what the design engineer had in mind. One will often find three people pulling with all their might on a handle ostensibly designed for a single operator. The second option means that the device should be appropriately overdesigned.

All three manually operated oil presses examined in this report are of sufficiently complex design and structure that they can only be properly repaired by a competent mechanic working in a well equipped workshop where replacement materials are readily available. Therefore, the oil presses need to be appropriately overdesigned.

I have chosen what I feel to be appropriate design criteria for the given situation and I have evaluated the chosen designs by these criteria. Engineering analysis and design are inexact arts and many judgments and approximations have to be made. I have used my engineering knowledge and my experience working in small East African workshops and with urban and rural poor users to make these judgments.

1.4 MATERIAL PROPERTIES AND DESIGN CRITERIA

The predominant material used in the manufacture of the three oil presses is the locally available mild steel. In general in Tanzania the supply of steel is not very reliable and it may come from many different sources. Both the prices and the available sections may vary from week to week. It is also true that the quality of the steel (except in the cases of specially imported alloys) is not very well known and not very reliable. No proper testing program of material properties is possible in the local setting and even if it were, new tests would have to be conducted on each consignment of steel.

For these reasons, I have chosen quite conservative values for the expected material strength properties of the mild steel. For the yield strength, the stress above which permanent deformation will occur under uniaxial tension, I have chosen 200MPa. I have also chosen the conservative Maximum Shear Stress (or Tresca) Yield Condition as the appropriate failure theory. This theory predicts that failure (defined as permanent deformation or cracking) will occur when the

maximum shear stress in the component equals or exceeds half the value of the uniaxial failure stress.

Fatigue-failure causes most failures in machine components. Cyclic loads below the yield level cause microcracks to grow, eventually resulting in component failure due to cracking. For mild steel one can design for virtually infinite fatigue life if one keeps the maximum stress values below what is called the endurance limit. The endurance limit gives the maximum allowable stress value for the special case of fully reversed cyclic loading, i.e. a loading cycle which goes from full tension to full compression and back.

The endurance limit for mild steel can be taken as one-half of the ultimate strength of the material. The ultimate strength is the stress at which material failure due to cracking will occur under uniaxial stresses and can conservatively be taken to be 300MPa for mild steel. This puts the endurance limit for fully reversed cyclic loading at 150 MPa.

For the oil presses examined in this study, we find that the loads applied to the components are not "fully reversed" loads but rather they go from zero to a maximum value and then back to zero. We shall call this zero-to-max. cyclic loading. One can analytically account for this difference by using the conservative Soderberg Criteria to modify the endurance limit. For mild steel subjected to zero-to-max cyclic loading the modified fatigue limit, (i.e. the maximum allowed stress for infinite life,) is found to be 172 MPa.

A place where there is a sharp change in the surface geometry of a stressed component is known as a stress concentration point. The localized stresses around this concentrator will be substantially higher than the average stresses in the rest of the component. These high localized stresses can cause the growth of fatigue cracks; this must be considered in the design. Examples of stress concentrators are notches, screw threads and holes. For fatigue design purposes one assumes that these stress concentrators cause a decrease in the material's fatigue limit by a factor known as the fatigue-stress-concentration-factor. For notches and threads cut in mild steel components, this factor can be taken to have a value of 2.0. This indicates a factor of 2 reduction in allowable stress values for these components.

After accounting for all the above factors, there are still many unknowns and uncertainties involved with machine design. The actual stresses are never accurately known. There is a large scatter in material properties even in what is nominally the same material. There are machining faults and pre-existing microcracks. There are unknown residual stresses and environmentally induced effects such as corrosion. To account for these and many other uncertainties, it is common practice, in any design work, to

use a factor-of-safety. The factor-of-safety is a constant factor by which the allowable stresses are reduced to account for all the uncertainties. I have chosen the factor of safety to be 1.25.

To summarize, the material properties for mild steel that I have used to judge the designs are the following:

S_u = Ultimate Strength = 300 MPa = Stress to cause fracture in uniaxial tension.

S_y = Yield Stress = 200 MPa = Stress to cause permanent deformation in uniaxial tension.

S_e = Endurance Limit = 150 MPa = Maximum stress for infinite life for fully-reversed cyclic loading

S_f = Fatigue Limit = 172 MPa = Maximum stress for infinite life for zero-to-max. cyclic loading.

The design criteria that I have chosen give:

r_y = 160 MPa = Maximum allowable tensile stress for static loads.

t_y = 80 MPa = Maximum allowable shear stress for static loads.

r_f = 138 MPa = Maximum allowable tensile stress for zero-to-max. cyclic loading of smooth components.

t_f = 69 MPa = Maximum allowable shear stress of zero-to-max. cyclic loading of smooth components.

r_{fc} = 69 MPa = Maximum allowable tensile stress for zero-to-max. cyclic loading of components with stress concentrations.

t_{fc} = 35 MPa = Maximum allowable shear stress for zero-to-max. cyclic loading of components with stress concentrations.

Apart from the mild steel, the other material used in the construction of these presses is an alloy steel called ST70. This material, which is used for the spindle and the pins on the Scissor Jack Presses, has an ultimate strength of 700 MPa.

For design purposes we can take:

r_F = 320 MPa
t_F = 160 MPa
r_{fc} = 160 MPa
t_{fc} = 80 MPa

As I have said, I have chosen conservative values for the material properties and have used conservative design criteria. However, given the large number of uncertainties mentioned above I feel that these are the appropriate criteria for doing mechanical design in Tanzania.

When it is found that the calculated stresses in a given component exceed the chosen design criteria it does not mean that each and every component built in that way will inevitably fail when used. Rather it indicates that there is a high likelihood that at least some of them might fail and that in my opinion the given component is underdesigned. Such is the nature of mechanical design.

1.5 SPECIAL CONSIDERATIONS FOR SCISSOR JACK PRESS ANALYSIS

The proper force and stress analysis of a Scissor Jack Press involves a large number of uncertainties. Other than the already discussed problems of knowing the material properties and properties of sunflower seed under compression it is also vital to know the values of the coefficients of friction of the screw threads and the thrust collar and to know something about the distribution of the loads over the screw threads of the nut. Stress concentrations in the spindle and nut must also be considered.

In Equation 1.5 of Appendix 1 we have determined the theoretical mechanical advantage of a Scissor Jack Press. It is a function of: the length of the handle (l); the geometry of the spindle and thrust collar; the coefficients of friction of the threads (f) and of the thrust collar (fc); and the angle between the scissor arms and the vertical (i.e. the position of the plunger).

For a given press design in a given configuration the only unknowns in Equation 1.5 are the coefficients of friction in the screw threads (f) and in the thrust collar (fc). The value of the coefficient of friction between two lubricated machined metal surfaces such as between the spindle threads and the nut, or between the thrust washer and the thrust piece, can vary from a value of 0.15 to 0.30 depending on the surface finish and on the quality and amount of lubrication. The coefficient of friction (fc) of a thrust ball bearing is about 0.01. Equation 1.5 shows that the exact values of f and fc have a very large effect on the resulting mechanical advantage. The smaller the coefficients of friction the larger the mechanical advantage, and for the spindle

geometries used here, a variation by a factor of 2 in coefficients of friction, leads to a variation by a factor of about 1.7 in mechanical advantage. This means an uncertainty by this same factor in all the calculated forces and stresses. This is a problem common to the analysis of any device with a turning screw thread.

Although replacing a thrust collar with a thrust ball bearing, as Themi has done, may reduce the effort required for operating the press, it also increases the mechanical advantage and in so doing increases the possibility of overloading the press. As such, it is a two edged sword.

In general for the purpose of designs one would only consider the worst possible case (i.e. f = 0.15); however, it is sometimes useful to determine just how large an effect a given uncertainty leads to. In this case I have chosen to carry out all the calculations for both f=0.15 and f=0.30.

There are also large frictional forces in the pivot pins of the press. These forces will act to lower the mechanical advantage (the efficiency) of the press and lead to less pressure on the seeds and thus lower stresses in the main frame. Because the magnitude of these effects is not easy to predict, I have not included them in my analysis.

The determination of the distribution of loads over the screw threads of a nut is a difficult problem that depends on the relative material properties of the spindle and the nut and on the specific geometries of both the threads and the nut itself. It is the distribution of these loads that determines the ultimate strength of the nut and spindle combination. If the entire load is taken up by the first thread only, then the nut will be very weak; if the load is evenly distributed between all the threads, the nut will be very strong. Because both the nut and the spindle tend to deform under the applied load, the true distribution lies somewhere between those two extremes. For a straight non-tapered nut like the ones on the 20 and 80-Ton Presses, most of the load is taken up by the first 4 or 5 full threads. In this analysis I have made the approximation that 90% of the full load is supported by the first 4 full threads of the nut. A very long nut does not provide any advantage; 7 or 8 full threads is probably more than adequate.

Large stress concentrations in both the nut and the spindle of a Scissor Jack Press are caused by the small radiuses of curvature at the roots of the square thread and the machined notches. To account for these stress concentrations we must compare the maximum calculated stresses in the spindle and the nut to the maximum allowable stresses for cyclic loading of components with stress concentrators, for example, r_{fc} = 70 MPa and t_{fc} = 35 MPa.

CHAPTER 2

20-TON PRESS

2.1 DESCRIPTION OF PRESS

This oil press is a manually operated Scissor Jack Press that has been rated by IPI to generate a maximum force of 20 tons. It is a batch type press that stands 1670mm high and has a 847mm long threaded spindle. It basically consists of a vertically oriented, screw operated, scissor jack mechanism that is suspended inside a mainframe and acts against it to push a plunger into an oil-pressing cage. The pressing cage is made of closely spaced vertical slats and is 225mm deep and 150mm in diameter. The scissor jack mechanism consists of four 445mm long pivoted scissor arms, a nut piece, a thrust piece and the spindle. The spindle is turned by a handle with four 850mm long turning arms. The main frame consists of two vertical columns and a bottom and top cross piece against which the press acts. Both the nut and the thrust pieces are designed with two removable pins so that the press can easily be disassembled.

During operation the turning arms are used to rotate the spindle clockwise; this pulls the nut and the thrust piece together, and through the action of the scissor arms pushes the plunger into the oil-pressing cage. The press is designed to be used on preprocessed sunflower seed which has been decorticated, winnowed, rolled and scorched.

As the pressing cage is being filled with the preprocessed seed, three thin pressing plates are introduced between layers of the seed. These plates are designed to facilitate the outward flow of oil during the pressing. The cage is filled to capacity and the seed is compressed until the first oil starts to flow. At this point the plunger is backed out and more seed and another pressing plate are added to refill the cage. The pressing is then continued and the oil is squeezed out of the seed. The pressing is stopped at the point when the seed itself starts to be extruded through the slats of the cage. At this point the plunger is backed out of the cage and the cage is lifted up and supported so that the plunger can now be used to expel the spent seed cake.

2.2 GENERAL OBSERVATIONS

At the time I visited Themi Engineering they had manufactured 10 20-Ton Presses. Three were operating in the field. The one in Marangu was being used to press coconut; the one in Arusha was being used to make fruit juice and the one in Rambo was pressing peanuts and occasionally sunflower seed. Themi was field testing five (sunflower seed) presses before delivering them to buyers.

The managers of Themi prefer the 20-Ton Press to the 80-Ton Press. They said it is easier to transport and repair and easier and faster to operate. The cycle time for a single pressing is 20 minutes instead of half an hour. It is also less expensive. The basic press sells for 35,000Tsh (Tanzania Shillings) ($US 374) . The pre-processing equipment costs an additional 98,000Tsh ($US 1,047). (see breakdown in Chapter 3, p. 23).

On examining the Themi made presses I found that manufacturing quality control was quite poor. The equivalent components on different presses all had slightly different measurements. The length of the four scissor arms on a single press varied by as much as 15mm. Most of the pressing cages were twisted and some of the welds were poorly made. In general I found that cutting/drilling guides and welding jigs either had not been made, or had not been used.

The Scissor Jack Press design is particularly vulnerable to inaccurate cutting and welding. Small misalignments caused by different length or skewed components can cause very large unwanted eccentric forces.

2.3 DESIGN CHANGES MADE BY THEMI

I compared the 20-Ton Presses made by Themi to the IPI drawings of the "Model III Reinforcement Pressure 20-Ton Press". There were only a few structural differences:

1. The spindle diameter has been reduced from 40mm to 38mm;

2. The thrust washer has been replaced by a thrust ball bearing;

3. There are three reinforcing rings on the cage as opposed to five in the drawings (one at the top, one at the bottom and one in the middle); their cross section has been increased 10mm by 20mm to 10mm by 25mm;

4. The scissor arms are made from 50mm x 10mm bar sections instead of the 40mm x 10mm sections shown in the drawings;

5. The solid round bars used for the turning handles have been increased from 20mm to 23mm in diameter; and

6. The piston guide bars have been reduced in size and an extra guide has been added for them in the middle of the frame.

2.4 EXPERIMENTAL DETERMINATION OF GENERATED FORCE vs. POSITION

This experiment was performed on each of two of the 20-Ton Presses built by Themi. The seed was fully preprocessed and was pressed as described above. The press cages were fully loaded and pressed by 75mm at which point the first oil was produced. This was at a total strain of 0.33 millimeters. The cages were then refilled to the top and the experiments were started. Using the procedure outlined in Section 1.2 the applied forces were measured as a function of plunger displacement. The mechanical advantages of the presses as a function of angle Ø were calculated using Equation 1.5 in Appendix 1. Because the Themi presses use thrust ball bearings instead of thrust collars, the coefficient of friction, f_c, was taken to be 0.01.

In Table 1.0 of Appendix 1 the original data and the results of the calculations are given for the cases of f=0.15 and f=0.30. We see that the pressing was continued up to a point where the angle Ø was 12°. This corresponds to a total strain on the seed (including the first pressing and accounting for the four pressing plates) of 0.62. At this point the seed cake began to be extruded through the slats of the cage and the pressing was stopped. The applied force (P) remained very low over the first 80mm of this compression stroke; it then increased rapidly with further pressing to reach a final value of 31kgs force in one press and 27kgs in the other.

The results of generated force vs. displacement during the pressing (as calculated with f=0.15 and f=0.30) are plotted in Figure 1.1. Similar to the applied forces, the generated forces remain at a low value until the displacement reaches about 75mm when they increase rapidly with further pressing to reach a maximum value of about 31,000 kgs or 307,000 N.

In the stress analysis in the following section, I have assumed that the maximum force applied by the operator to the handle is 31kgs or 300 N and that the maximum piston displacement reached corresponds to an angle of $Ø=12^{o}$. Judging from my tests these seem to be the "normal" operating values. However, the limiting factor in these tests was that the seed cake started to be extruded through the gaps in the cage and not that further compression became too difficult or that the bottom position of the piston was reached. The force at which the seed cake starts extruding is dependent on the properties of the particular seed and on the degree of the preprocessing. Thus during actual use in the field it is likely that higher forces will be applied and lower piston positions will be reached causing higher forces and higher stresses in the press. Nonetheless in this analysis I have chosen to use the "normal" conditions. This choice is justifiable for two reasons. First, mechanical losses that have not been adequately accounted for in my theoretical

analysis (i.e. friction in the pins) act to reduce the mechanical advantage. Second, even these low values cause the press to be overloaded and thus there is no need to pick higher values[1].

2.5 FORCE AND STRESS ANALYSIS

This section contains a tabulated summary of the force and stress analysis for the Themi 20-Ton Press.

DIMENSIONS OF THEMI 20-TON PRESS

l	=	length of turning handles	=	850 mm
R	=	length of scissor arms	=	450 mm
d_m	=	mean diameter of spindle	=	34.9 mm
d_i	=	root diameter of spindle	=	31.6 mm
d_o	=	outside diameter of spindle	=	38 mm
pit	=	pitch of spindle threads	=	6.35 mm
tanu	=	pit (helix angle of threads)	=	0.0579
tanu	=	$\frac{pit}{\pi d_m}$ (u=helix angle of threads)		
d_{mc}	=	mean collar diameter	=	50 mm
D	=	Diameter of plunger	=	150 mm

For the following stress analysis we take:

P = maximum force applied to the end of the handle = 300N

f_c = coefficient of friction for thrust ball bearing = 0.01

max = value of the angle 0 at maximum compression = 12°

The analysis is done for two cases:

1. f = coefficient of friction in threads = 0.30
2. f = 0.15

and the calculated stresses are compared to the maximum allowable stresses as defined in Section 1.4.

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1 If one were to design a new Scissor Jack Press rather than evaluate the existing model one could extrapolate the force vs. displacement curves up to a displacement corresponding to the bottom possible piston position: in this case X=115mm at $0=2.5^o$ giving F=40,000 kgs. (392,000 N) for f=0.15 and F=23,800 kgs. (233,000 N) for f=0.30. These would be the appropriate maximum loads assuming that the correct amount of sunflower seed was put into the cage and that the press was screwed down to its lowest possible position. Because in this part of the stroke the mechanical advantage rises more steeply as a function of plunger displacement than does the generated force, the necessary applied load in the final position would be only about 8 kgs.

FORCE ANALYSIS

	f = 0.30	f = 0.15
Torque on the Spindle (T) (Equation 1.3)	255,000 N-mm	255,000 N-mm
Tensile load on the Spindle (L) (Equation 1.4)	38,600 N	65,200 N
Generated Force at the Plunger (F) (Equation 1.5)	182,000 N	307,000 N
Generated Pressure in the Cage (P) (Equation 1.6)	10.3 MPa	17.4 MPa
Theoretical Mechanical Advantage (F/P)	607	1,020

STRESS ANALYSIS

	MAXIMUM ALLOWABLE STRESS	STRESS IF f = 0.30	STRESS IF f = 0.15
Stress in Spindle			
Tensile Stress (Eq. 1.7)		40 MPa	68 MPa
Shear Stress (Eq. 1.8)		29 MPa	29 MPa
Maximum Shear Stress (Eq. 1.9)	35 MPa	36 MPa	45 MPa
Stresses in Spindle Threads (t = 3.2mm, w = 3.05mm)			
Bending Stress (Eq. 1.10)		90 MPa	152 MPa
Shear Stress (Eq. 1.11)		29 MPa	48 MPa
Maximum Shear Stress	35 MPa	54 MPa	90 MPa
Stresses in Nut Threads			
Bending Stress (Eq. 1.10)		75 MPa	127 MPa

Shear Stress (Eq. 1.11)		24 MPa	40 MPa
Maximum Shear Stress	35 MPa	45 MPa	75 MPa
Stress in Scissor Arms A = 2(10) (50) + 5(50) = 1,250mm2			
Compressive Stress (Eq. 1.13 & Eq. 1.14)	160 MPA	74 MPa	123 MPa
Stresses Around Pin Holes (D = 25mm)			
Bearing Stress in Arm Holes (a = 10mm) (Eq. 1.15)	160 MPa	185 MPa	340 MPa
Bearing Stress in Plunger Slab	160 MPa	227 MPa	384 MPa
Stresses in Pins (D- 25mm, c = 50mm)	(Pins are made from ST70 Steel)		
Shear Stress (Eq. 1.17)	160 MPa	95 MPa	160 MPa
Bending Stress (Eq. 1.18)	320 MPa	378 MPa	622 MPa
Stresses in Main Frame (Eq. 1.19 & Eq. 1.20)			
Tensile Stress in Columns (A = 800mm2)	138 MPa	114 MPa	192 MPa
Bending Stress in Top Cross Bar (l' = 50mm, l = 270mm) (b = 40mm, h = 100mm)	138 MPa	221 MPa	282 MPa
Bending Stress in Bottom Cross Bar (l' = 175mm, l = 270mm) (b = 40mm, h = 75mm)	138 MPa	221 MPa	373 MPa
Stresses in Turning Handle (I = π , r = 11.5mm)			
Bending Stress (Eq. 1.21)	138 MPa	213 MPa	213 MPa

Stresses in Pressing Cage
(Eq. 1.22 & Eq. 1.24)

Bending Stress in Slats (l = 105mm, h = 10mm)	138 MPa	568 MPa	960 MPa
Hoop Stress in Tension Rings (w = 10mm, t = 25mm)	138 MPa	324 MPa	548 MPa
Proper Tension Ring Separation		62 mm	48 mm
Proper Tension Ring Cross Section (Eq. 1.25)		347 mm2 (19mmx19mm)	454 mm2 (21mmx21m

In almost every instance in the above stress analysis, the stresses are higher than the allowable values, even for f=0.30. The highest stresses occur in the pressing cage which is substantially underdesigned. A number of the cages on the presses being tested at Themi already had bent slats. In the original drawings by IPI there were five tension rings with 45mm separations and an average cross section of 260mm^2; Themi replaced these with three rings with 250mm cross sections. The IPI design was adequate; the Themi design is unacceptable.

The bearing stresses around the pin holes, especially those in the plunger assembly, are too large and some deformation of the holes might be expected. Also any lubrication of the pins would be squeezed out and high frictional forces would result. The bending stresses in the pins are also too high and a fatigue fracture is likely. The stresses in the columns of the main frame are not critical if f=0.3 but a fatigue failure could be expected if f=0.15. Both the top and bottom crossbars are underdesigned although the stress value in the bottom bar is slightly exaggerated.

The stresses in the spindle are unlikely to cause complete fracture in the near future but the spindle threads are in danger of stripping. In a screw, the spindle threads are more likely to strip than are the nut threads. Even though the turning handles have been increased in diameter from the original design the stresses are still too high and bending or fatigue failure is likely.

In general the press is significantly underdesigned even if one assumes "normal" use and that f=0.3. If f is, in fact, less than 0.3, and if the full length of the plunger stroke is utilized (which will lead to higher stresses) or if the press is otherwise overloaded, severe failures including the stripping of the spindle threads and the bursting of the cage can be expected.

The original IPI design had a thicker spindle (40mm instead of 38mm), and a stronger cage both of which were design improvements. It also had a thrust washer instead of a thrust ball bearing. Using the bearing instead of the large washer has two disadvantages. If the same amount of seed is compressed to the same strain, larger applied forces are needed on the handles, and the handle arms are underdesigned and will bend. If the same force is applied to the handle, lower forces and thus lower stresses are generated and less oil can be extracted from the seed.

The scissor jack design is very susceptible to failures due to eccentric loads caused by inaccurate machining and assembly. This fact and the fact that the 20-ton Press is underdesigned were clearly demonstrated by the experiments described in the following section.

Later in this report, I will analyze whether the Scissor Jack Press is the most appropriate design for pressing sunflower seed.

2.6 THE YIELD FROM PRESSING DECORTICATED AND UNDECORTICATED SEED AND IMPLICATIONS FOR FUTURE PRESS DESIGN

In this experiment I wanted to demonstrate that if enough pressure is used the 20-Ton Press can extract oil even from undecorticated sunflower seed. I wanted to compare the yield for a "normal" pressing of preprocessed seed with the yield from pressing the undecorticated seed. I also wanted to demonstrate that the 20-Ton Press is underdesigned.

First, I watched the workers decorticate, winnow, roll and scorch 5 kgs. of whole seed. Using a hand operated decorticator and the wind, it took about 15 minutes to decorticate and winnow the seed. Now only 3.75 kgs. of the original 5 kgs. of seed remained. The efficiency of the decorticator was about 80%. The rolling took 5 minutes and the scorching another 30 minutes. Finally 3.5 kgs. of the preprocessed seed was pressed using the normal procedure, as described in Section 2.1. The total yield was 1.0 liter of oil. The entire process, from whole seed to oil, had taken 70 minutes. The oil produced per kilogram of whole undecorticated seed was 0.21 liters/kg.

In the second test I spread the whole undecorticated seeds out in the sun to preheat them slightly, but did no other preprocessing. I only loaded small amounts of seed into the press at a time adding an extra pressing plate on top of each layer and pressing it fully before adding the next layer of seed. After pressing three layers of seed the oil started to flow from the cage (at this point the total strain on the seed was 0.68) and after pressing two more layers the machine bent due to an eccentric load caused by misalignment during manufacture. I only used the press in the bottom half of its stroke, where it has its highest mechanical advantage, and the forces applied to the handle never exceeded about 20 kgs. During the actual pressing the angle Ø (between the scissor arms and the vertical) never exceeded 20° and at the end of each stroke it reached 2.5^{o}, the lowest position the press's geometry will allow.

When the pressing was stopped because of the bent machine, a total of 2.55 kgs. of seed had been pressed yielding a total of 0.65 liters of oil. This gives a yield, in liters of oil per kilogram of whole seed, of 0.25 liters/kg, almost 20% higher than the yield obtained from the normal process. The final pressure applied to the seed can be approximated by using Equation 1.5 for the 20-Ton Press with $f = 0.3$, $Ø = 2.5$ and $P = 20$ kgs. We find a maximum force of 580,000N and a maximum pressure of 33 MPa. The maximum total strain on the seed at this final point was 0.78. This compares to a normal pressing where the maximum pressure was found to be 10.3 MPa.

(if f = 0.3) and total strain was 0.63. Because the shells had not been removed from the seed, seed cake was not extruded through the cage even at this very high pressure.

This experiment demonstrated that a higher pressure can substitute for preprocessing of the seed. This holds true even for a large diameter batch press.

The pressure in an oil press serves two functions: 1) it crushes the seed and squeezes the oil out of it, and 2) it pushes the oil through the seed cake and out of the press. This second action is not caused by the pressure itself, but rather by the gradient in the pressure; the change in pressure over some distance.

Pressure is defined as force per unit area. For a given force generated by a press, the resulting pressure in the cage will be higher for a small diameter cage and lower for a large diameter cage. In order to build up a higher pressure we have to have a higher strain i.e. we have to change the height of the seed column by a greater percentage.

In these oil presses, the pressure outside the cage is zero while the pressure at the center of the cage is a maximum. Because the gradient is defined as the change in pressure divided by the distance we can increase the gradient by either increasing the pressure in the cage or by decreasing the cage diameter. Either way, the oil will flow out of the cage more easily.

I conclude that a better oil press should have a smaller diameter cage and a longer stroke. The small diameter cage allows the user to generate higher pressures and still press with lower forces (thus the press doesn't have to be so strong). At the same time it increases the pressure gradient and makes it easier for the oil to flow out of the cage. The longer stroke, meanwhile, is needed to generate high enough strains on the seed so that the pressure will be sufficient to crush even undecorticated seed and to squeeze out the oil. A longer stroke also means that more seed can be pressed in one pressing.

The design implications of these conclusions are that the pressing <u>cage</u> has to be stronger, in order to withstand the higher pressures, but the <u>press</u> can be substantially weaker because the forces will be less. A scissor jack press, most efficient for producing high forces with a short stroke length, would probably no longer be the right design choice. Simpler designs such as simple screw presses should be considered.

A rough back-of-the-envelope calculation suggests that a simple screw press with a 50mm diameter spindle, a 720mm long stroke, and a 1000mm long by 60mm diameter cage, could produce half as much oil per hour pressing undecorticated

seed as the 20-Ton Press pressing fully preprocessed seed. It could cost less than half the price and no preprocessing equipment would be needed.

Section 3.8 of Chapter 3 discusses the pros and cons of using a scissor jack press to press sunflower seed oil.

CHAPTER 3

80-TON PRESS

3.1 DESCRIPTION OF PRESS

This press is a very large manually operated Scissor-Jack Press, similar in design to the 20-Ton Press. It has a total height of 3.65 meters and weighs 900 kgs. Like the 20-Ton Press it is a batch type press that consists of a vertically oriented, screw operated, scissor jack mechanism which is suspended inside a mainframe and acts against it to push a plunger into an oil pressing cage.

The slatted pressing cage is 555mm deep and 298mm in diameter. The scissor jack mechanism consists of four 1105mm long pivoted scissor arms, a nut piece, a thrust piece and a 1725mm long, 40mm diameter threaded spindle. The mainframe is made up of two columns and a bottom and top crossbar against which the press acts. This press, similar to the 20-Ton Press, is designed to be used on sunflower seed that has been decorticated, winnowed, rolled and scorched before it is pressed. A single operator turns the spindle with a handle which has four 1400mm long turning arms.

The turning arms are used to rotate the spindle in a clockwise direction. The nut and the thrust piece are drawn together; through the action of the four pivoted scissor arms, the plunger is forced down into the pressing cage.

3.2 GENERAL OBSERVATIONS

When I visited Themi Engineering (October, 1987) they were no longer making the 80-Ton Presses. They had not manufactured any for a number of months and had decided not to make any more.

Themi had sold 10 of the 12 80-ton presses they had manufactured. Only 4 were still operating. Some presses had broken down and some were not being used. However, no actual follow-up reporting had been undertaken.

Themi engineers felt that the presses were too big and too heavy, and thus too dangerous to install and operate. During an installation, one worker had lost a couple of fingers; another narrowly escaped having his legs crushed. The 80-Ton Presses were difficult to sell, to transport and to repair.

The 80-ton Press was priced at 80,000Tsh ($US 855). However, it is designed to be used as part of a package with a decorticator/ winnower costing 60,000Tsh ($US 640), a roller costing 20,000Tsh ($US 210), and a scorcher costing 18,000Tsh ($US 190), for a total cost of 178,000Tsh ($US 1900) minus a small discount.

3.3 THE THEMI BUILT PRESS

Themi felt that the changes they had made to the original IPI design had solved most of the presses' engineering problems with the exception of the nut (the nut's threads are still being stripped). However, they agreed that a proper engineering stress analysis needed to be conducted.

Themi made these changes in the original design:

1. Using a high strength carbon steel alloy for the spindle. ST70 (with an ultimate strength of 700 MPa) was substituted for the original mild steel. IPI also made this change on their later presses;

2. Redesigning the nut piece and the thrust piece so that each had two removable pins by which to attach and remove the scissor arms. In the IPI design a single fixed pin was used which only allows for disassembly by breaking a weld in the scissor arms. Only one press had been made with the new nut design and, although easier to assemble and disassemble, it was not yet clear if it might cause other problems;

3. Redesigning the cage by eliminating the hinges and clasps so that it no longer opened and closed. The hinges had a tendency to break and Themi also wanted to reduce the cost. Themi also discontinued the use of the center (oil draining) pipe in the cage; and

4. Replacing the thrust washer with a thrust ball bearing.

Because there were no operational, Themi made, 80-Ton Presses in Arusha, I decided to examine an operational 80-Ton Press made by IPI.

3.4 THE IPI BUILT PRESS

This press had been operated by Ferry Klemann in Arusha town for the past 21 months. For 18 of those months the press was used for 6 pressings per day, five days per week. Klemann is friendly with the people at IPI and they have helped to repair his equipment each time it has broken down. Klemann has kept careful records.

Since Klemann has been using the press, both the original mild steel spindle and its replacement have failed due to cracking. They were eventually replaced with a high carbon steel spindle. The threads of the original nut were stripped out and IPI modified it so that a new replaceable nut could be attached by two bolts onto the outside face of the old nut. This modification solved the problem of having to cut through a weld every time the nut needed replacing. Since that time two of the new, mild steel attachable nuts have

stripped out (one only lasted for a few days). When I arrived the press was not operational; a new nut was being made from medium strength steel. Other problems included: the wearing out of the thrust washer, a top pivot pin that became loose and worked its way out of the holes, worn out locks on the cage that had to be welded, and broken hinges on the cage that had to be repaired using extra material.

3.5 DESIGN CHANGES IN IPI PRESS

Apart from the modified nut and spindle there are a number of other differences between the Klemann press and the original IPI drawings.

1. The pressing cage on Klemann's press is made in two parts with a bottom hinged cage that is 285mm high, with 4 separate locks, and a top sheet metal cage extension which is 270mm high. The oil removal is now done in a single pressing starting with the cage and its extension both full of seed, instead of in two or three stages as was done in the past.

2. The mainframe upright members have been changed from 100mm x 50mm x 6mm channel covered by a 100mm x 6mm bar stock to rectangular columns made up of two 100mm x 16mm bars and two 50mm x 16mm bars welded to form a hollow section.

3. The pins have been changed from 40mm diameters to 50mm diameters and the top scissor arms have been reinforced around each pin hole by the addition of a 200mm long 19mm x 80mm solid section. Also the web pieces in these top arms have been changed from 19mm x 100mm sections to 10mm x 100mm sections.

4. The vertical places in the plunger assembly have been changed from 150mm x 19mm sections to 200mm x 25mm sections to compensate for the larger pin hole sizes.

5. The turning handles have been changed from bent 25mm solid rods to a stronger combination of 750mm long 40mm x 20mm solid sections with 660mm long sections of 35mm diameter pipe on the end. Diagonal reinforcing braces of the 40mm x 20mm bar stock have also been added.

Smaller design changes have also been made. In general, the major changes sought to increase the strength of the highly stressed components of the press. This in itself suggests that the initial design was substantially under-designed. Judging from the aforementioned failures of the nut threads, thrust washer, cage hinges etc. it would appear that some major design flaws are still evident.

In May of 1986, IPI published an instruction manual for its

oil processing equipment. In this manual it states that the torque applied to the spindle of the press should never exceed 300N-m. The manual points out that this is the equivalent of hanging a 21.5kg load at the end of the handle, and warns that if this load is exceeded the spindle might break. It suggests that if the load exceeds this value the processing should be stopped and the cake removed even if it still has a high oil content.

It is highly unlikely that the new high strength steel spindle would break at this low torque. (The nut on the other hand might be a problem). However, the idea of setting a very low limit on the allowable load to be applied to a manually turned handle is contrary to actual field conditions. This is especially true when the maximum allowed load is less than one third of an average man's weight, and when there is no way to determine this applied load.

Ferry Klemann expected that the men using his machine regularly exceeded this applied load. In the experiment I carried out, the maximum load applied to the handle by the operator was 49 kgs., more than twice the recommended maximum value.

3.6 IPI PRESS IN OPERATION

Shortly before I left Arusha the new attachable nut, made from medium strength steel, was completed and I watched and recorded data for a single seed processing and oil pressing cycle of Klemann's press.

Four men work Klemann's press. They use an electric powered decorticator - winnower, a manually operated roller and a scorcher to preprocess the seed. Twenty-two kgs. of preprocessed seed are pressed in one pressing and the average yield is 8.1 liters of oil. In an 8 hour day the four workers can complete 6 full pressing cycles, yielding almost 49 liters of oil per day.

The 22 kgs. of preprocessed seed corresponds to approximately 36.7 kgs. of whole undecorticated seed. Thus, the average oil yield in liters of oil per kilogram of whole seed is 0.22 liters/kg.

I used the spring balance to measure the force applied to the handle as a function of the plunger position in the manner described in Section 1.2. Unfortunately, at some point during the loading cycle the split retaining ring broke (this holds the thrust piece in place). It had been assembled incorrectly when the new nut was put on the press. It is impossible to determine the exact effect of the broken ring. However, it probably increased the frictional forces in the threads and thus increased the required applied forces on the handles.

Although they did not know about the broken ring the operators stopped the compression stroke at an earlier point than usual. "Because of incomplete scorching", a larger than normal amount of the compressed seed was being extruded through the gaps in the pressing cage. (Although the operator mentioned that the required force seemed a bit higher than usual for that part of the compression stroke, they later assured me that it was probably no higher than the usual maximum applied force during a normal pressing.)

At the point where the compression stroke was stopped, the applied force on the handle (which had been increasing monotonically) had reached a maximum value of 49 kgs. The final angle (Ø) between the scissor arms and the vertical was 9.3^{o}, corresponding to a total plunger displacement of 323mm and (accounting for the pressing plates thickness) a total strain on the seed of 0.61.

The operators told me that in a normal pressing the scissor arms reach a final angle of 5.2^{o} to the vertical, corresponding to a plunger displacement of 342mm and a total strain on the seed of 0.64. This is the maximum displacement that the geometry of the press will allow.

Because the retaining ring broke in the middle of this experiment, the results were not very useful. There was no point in plotting them as was done for the 20-Ton Press experiments. Instead, I decided to rely on the judgment of the operators to determine the design values for completing the analysis. They told me that the maximum applied force was about equivalent to the 49 kgs. applied in this test and that it usually occurred at a later point in the pressing. I chose to assume that the maximum force applied to the handle was 500 Newtons and that this value occurred in the final bottom press position where the angle Ø was 5.2^{o}. In the following section these are the values used to complete the stress analysis. The analysis is done for both the two extreme values of the coefficients of friction f = 0.3 and f = 0.15.

3.7 FORCE AND STRESS ANALYSIS

This section contains a tabulated summary of the force and stress analysis for the IPI 80-Ton Press. The equations used for the analysis are given in Appendix 1, and the design values are justified in the previous section. The stresses are calculated for the two extreme values of the coefficients of friction, f = 0.3 and f = 0.15, and the allowable stresses for each as determined in Chapter 1 are given for comparison.

DIMENSIONS OF IPI 80-TON PRESS:

l	=	length of turning handle	=	1380.0 mm
R	=	length of scissor arms	=	1025.0 mm
d_m	=	mean diameter of spindle	=	36.8 mm
d_i	=	root diameter of spindle	=	33.6 mm
d_o	=	outside diameter of spindle	=	40.0 mm
pit	=	pitch of spindle thread	=	6.35mm
tanu	=	$\frac{\text{pitch}}{\pi d_m}$ (helix angle of thread)	=	0.0349
d_{mc}	=	mean collar diameter	=	50.0 mm
D	=	plunger diameter	=	290.0 mm

For the following stress analysis we take:

P	=	maximum force applied to end of the handle	=	500.0 N
ϕ_{max}	=	value of Ø at maximum compression	=	5.2^{o}

The analysis is done for two values of the coefficients of friction

1. $f = f_c = 0.30$

2. $f = f_c = 0.15$

and the calculated stresses are compared to the maximum allowable stresses as defined in Section 1.4.

FORCE ANALYSIS

	$f = f_c = 0.30$	$f = f_c = 0.15$
Torque on Spindle (T) (Equation 1.3)	690,000 Nmm	690,000 Nmm
Tensile Load on the Spindle (L) (Equation 1.4)	49,000 N	91,000 N
Generated Force at the Plunger (F) (Equation 1.1)	536,000 N	1,003,000 N
Generated Pressure in the Cage (P) (Equation 1.6)	8.1 MPa	15.2 MPa

	MAXIMUM ALLOWABLE STRESS	STRESS IF f = 0.30	STRESS IF f = 0.15
Stress in Spindle			
Tensile Stress (Eq. 1.7)	(Spindle Made from ST70 Steel)	46 MPa	85 MPa
Shear Stress (Eq. 1.8)		33 MPa	36 MPa
Maximum Shear Stress (Eq. 1.9)	80 MPa	40 MPa	56 MPa
Stress in Spindle Threads (t = 3.2mm, w = 3.05mm)			
Bending Stress (Eq. 1.10)		108 MPa	200 MPa
Shear Stress (Eq. 1.11)		34 MPa	64 MPa
Maximum Shear Stress (Eq. 1.12)	80 MPa	64 MPa	119 MPa

Stresses in Nut Threads			
Bending Stress (Eq. 1.10)	(Mild Steel Nut)	91 MPa	168 MPa
Shear Stress (Eq. 1.11)		29 MPa	53 MPa
Maximum Shear Stress (Eq. 1.12)	35 MPa	54 MPa	100 MPa
Stress in Scissor Arms A = 2(20) (80) + 20(60) = 4,400mm2			
Compressive Stress (Eq. 1.13 & Eq. 1.14)	160 MPA	61 MPa	113 MPa
Stresses Around Pin Holes (D = 50mm)			
Bearing Stress in Arm Holes (Eq. 1.15) (a = 20 mm)	160 MPa	133 MPa	264 MPa
Bearing Stress in Plunger Slab	160 MPa	54 MPa	100 MPa
Stresses in Pins (D = 50mm, c = 60mm)			
Shear Stress (Eq. 1.17)	69 MPa	67 MPa	125 MPa
Bending Stress (Eq. 1.18)	138 MPa	162 MPa	301 MPa
Stresses in Main Frame (Eq. 1.19 & Eq. 1.20)			
Tensile Stress in Columns A= 2(50) (16) = 2(100)(16)=4800mm2	138 MPa	112 MPa	208 MPa
Bending Stress in Top Cross Bar l' = 620mm, h = 200mm, b = 80mm, l = 620mm	138 MPa	143 MPa	268 MPa
Bending Stress in Bottom Cross Bar l' = 620mm, l' = 300mm, h = 160mm, b = 80mm	138 MPa	185 MPa	345 MPa

<u>Stresses in Turning Handle</u> (Eq. 1.21)			
Bending Stress (l = 1380mm, I = 107,000mm^4, c = 20mm)	138 MPa	129 MPa	129 MPa
<u>Stresses in Pressing Cage</u> (Eq. 1.22 & Eq. 1.24)			
Bending Stress in Slats (h = 12mm, l = 95mm)	138 MPa	254 MPa	476 MPa
Hoop Stress in Tension Rings (w = 29mm, t = 25mm)	138 MPa	151 MPa	283 MPa
Proper Hoop Separation (Eq. 1.23)		70mm	51mm
Proper Hoop Cross Section (Eq. 1.25)		585mm2 (25mmx25mm)	800mm2 (28mmx28mm)

The above stress analysis demonstrates that many components of the press are underdesigned even if one assumes the highest value of coefficient of friction f. In particular the pressing cage and the nut threads have stresses which are much too high and thus they are prone to failure. Both components have failed in the Klemann press. The spindle also has high stresses which would cause failure in mild steel. This failure was seen in the early presses but the substitution of the high carbon ST70 has solved the problem. The bending and shear stresses in the pins are also too high as are the bending stresses in the top and bottom cross bars of the frame.

If we examine the original IPI or the Themi designs we find that they have all the same problems as the new IPI design, and, in addition the columns at the mainframe and the turning handle arms are both substantially underdesigned and failures are likely.

The recent replacement of the mild steel nut by a medium carbon steel nut will result in higher allowable stresses, but a less even distribution of loads over the nut threads will do little to alleviate the problem of the threads stripping.

In conclusion, even the improved IPI 80-Ton Press is substantially underdesigned, and is highly susceptible to mechanical failures due to fatigue cracking or yielding.

3.8 DESIGN PROBLEMS OF A SCISSOR JACK OIL PRESS

This section discusses the general pros and cons of the Scissor Jack Press Design. These comments are relevant to both the 80-Ton Press discussed in this chapter and to the 20-Ton Press discussed in Chapter 2.

First, we examine how well suited the Scissor Jack design is for pressing sunflower seed. From Equation 1.5 we see that the mechanical advantage of a Scissor Jack press varies as one over tan Ø. This means that it starts at zero for Ø equal to 90° and then increases very slowly with decreasing Ø until Ø equals about 10° or 15° after which point it increases very rapidly, towards infinity for Ø equal to zero.

In our experiments on the 20-Ton Press we saw that because the cage has a large diameter, a large force is required to compress the seed. This force, like the mechanical advantage, remains at a low value over most of the compression stroke and then increases over the last portion of the stroke (see Figure 1.1). Unfortunately, this increase starts earlier and is less steep than the increase in the mechanical advantage. If these two curves were the same shape the required applied force (P) would remain at a constant low value throughout most of the compression stroke.

As it is, P can only be kept to a minimum value by operating the press in the range where its mechanical advantage is high, i.e., for small values of the angle Ø. This means that the length of the plunger stroke is limited. From Equation 1.2 we find that the plunger displacement is proportional to R, the length of the scissor arms, and from our experiments we see that large plunger displacements (large strains) are needed to generate high pressures. Because a Scissor Jack Press must have long scissor arms in order to efficiently press sunflower seed, both the presses examined here are large and expensive. The large size is not needed to withstand the high forces but rather to allow for a long stroke while still operating in a range where the mechanical advantage remains high. The 80-Ton Press is larger than the 20-Ton Press because, the larger diameter cage requires higher forces to develop the same pressure, and thus a higher mechanical advantage is needed over the length of the stroke. This requires that the press be operated over a smaller angular range; thus to get a long enough stroke length, a longer scissor arm length is needed.

A Scissor Jack Press is most useful for applying small loads over a long stroke length or for applying very high loads over a very short stroke length. To press sunflower seed in a batch press one needs to apply higher forces over a longer stroke length. The only way to do this with a Scissor Jack Press is to make a very large one, and to operate it only in the bottom portion of the stroke; the result is a heavy and expensive press.

A closer examination of Figure 1.1 shows that if slightly less seed is pressed and if the press is allowed to go all the way to the end of its stroke Ø=0, then higher pressures can be reached with less applied force. However, this option makes it too easy to overload the press. Because the press is designed to use preprocessed seed, the higher pressure will cause the seed to be extruded through the cage slats.

At the top of the stroke the scissor arms are spread apart; if the cage is too long they can't fit into the cage. This is why the cage on the 20-Ton Press has to be refilled in the middle of the pressing in order to build up enough pressure (enough strain). This problem is very easily solved by extending the length of the plunger plates by 75mm and lowering the cage with respect to the scissor mechanism by the same amount.

The result of this change would be a longer cage and a single instead of double pressing. This would almost halve the time needed per pressing but would also increase both the height and the cost of the press. The maximum required applied force on the handle would remain about the same, but the potential for unwanted eccentric loads would increase.

The Scissor Jack Presses are designed to be operated by one man, who pushes or pulls the turning handles around their horizontal axis. He is in a fairly comfortable standing position and can make effective use of both his strong leg muscles, by pushing, or his weight, by hanging or pulling on the handles. In the Scissor Jack design a large displacement of the nut piece corresponds to a small displacement of the plunger and a large amount of the input energy is used to fight friction. Therefore a single pressing using the 20-Ton Press takes 15 minutes and half an hour for the 80-Ton Press. The operator may become a bit tired. However, because of the large amount of seed preprocessing required and the difficulty and time required to remove the seed cake and reload the cage, the operator can expect quite a long rest between pressings. His overall work for the day will not be physically exhausting. Only six pressings are made per day with the 80-Ton Press and probably no more than 12 can be made with the 20-Ton Press.

A Scissor Jack design, is in general, very sensitive to inaccuracies in machining and misalignments during assembly. Small dimensional errors can cause large unwanted eccentric forces and bending movements. This is especially true if the scissor arms are of slightly different lengths. Therefore proper cutting and drilling gauges and welding jigs should be used. The proper manufacture of the spindle and the nut are also difficult operations which require an accurate long bed lathe and a practiced, careful machinist.

Because of the high stresses in the spindle threads and in the pins, they should be made of a high strength steel alloy. The un-availability of such an alloy in Tanzania on a regular basis may cause production delays and the cost may become prohibitive. Replacing these components with thicker mild steel components would, in theory, be possible but the large diameters required and the resulting additional frictional forces would not make this practical.

The design of the nut on a Scissor Jack Press is problematical. Ideally, it is best if the top and bottom scissor arms are attached to the nut by a single pin, in the same plane as the spindle as in the original IPI design for the 80-Ton Press. This arrangement insures that there are no bending movements applied to the nut and so none are passed on to the screw threads. Such bending movements could arise from inaccurate machining or assembly of the scissor arms and would act to increase both the stresses in the threads and the frictional forces. However, this design requires a substantial amount of machining and does not allow for the use of a removable pin. This in turn complicates the attachment of the scissor arms to the nut. The least complicated solution to this problem involves making a permanent attachment by welding together the scissor arm flanges only after they have been assembled over the nut pin. Although this solution is used on the original 80-Ton design,

it means that the press cannot be disassembled without cutting through the weld on the scissor arms. This has become a particularly serious problem because the threads of the nuts have been stripping out and there is no way to replace the nut without carrying a large part of the press to Arusha to have the scissor arm welds broken, the nut replaced and the arms re-welded. One possible solution to this problem is to design the scissor arms so that they can be bolted instead of welded together, but this involves additional time and cost in the manufacture. A better solution is what IPI did on the Ferry Klemann press after the nut stripped out. A detachable nut is bolted by two bolts to the outside of the old stripped nut piece; if and when this new nut strips it can be easily removed and replaced.

On the 20-Ton Press, and incidentally, also on the new Themi 40-Ton Press, the nut is attached to the scissor arms by two removable pins; one above and one below the spindle. Although this solves the problem of disassembly of the press and replacement of the nut, this design is very susceptible to bending movements causing increased stresses and frictional forces as described in the last paragraph. The bending movements can be minimized by putting the two pin holes as close to the spindle hole as possible. I noticed that the new 40-Ton Press had a particularly large nut piece with a large pin separation. Given Themi's lack of quality control procedures and assembly tolerances, this is likely to cause problems.

It is difficult to accurately determine the stresses in the Scissor Jack Press; they vary considerably depending on the degree of lubrication and surface finish of the threads. Also, large stress concentrations in the nut and spindle threads are likely to cause fatigue failures; these are very difficult to properly account for. For those reasons it is difficult to optimize the press designs; the only solution is to overdesign them assuming the worst case. The result is again a heavier and more expensive press.

Analyzing the stresses for the spindle and the nut is very difficult. As demonstrated by the experience of the 80-Ton Press, these are the components most prone to failure. There are, however, a number of simple design changes which can lessen these problems. The square threads used in these designs have sharp corners which cause high stress concentrations and they also have larger than necessary bending stresses because of their uniform cross sections. A tapered thread cross section, wider at the root and narrower at the tip, could help to reduce both of these problems. The other major problem is the uneven distribution of the load between the threads of the nut. This problem can be at least partially alleviated by making a tapered nut which is narrower on the highly stressed end and wider on the other end. In this way, the nut will deform in the same way as the spindle and the load will be more evenly spread between the

nut's threads. A longer nut can thus be effectively used to reduce the stresses in the nut threads, and nut thread tear-out will be less likely. Although these two steps can help to strengthen the nut, their exact effects are not easily predictable analytically (without a computer), and the only practical solution is to overdesign the components. Using a high carbon steel for both the nut and the spindle is a possible but expensive alternative.

A sunflower seed oil press is designed to produce as much oil as possible in a given amount of time, for as little money as possible. In the case of the Scissor Jack Presses, time was saved by making the volumes of the pressing cages as large as possible by giving them large diameters. The larger the cage the more oil you can get per pressing. The large diameters mean that high forces are needed to provide enough pressure to extract the oil from the seed, and thus large and expensive presses are required. But, despite the high forces produced by the large presses, the large diameter cages ensure that the pressures are still too low to extract the oil without much preprocessing of the seed. This preprocessing requires much extra expense and extra time and thus the original reason for having a large cage has been defeated.

This thought process suggests a smaller, cheaper press with a smaller diameter but longer and stronger cage. This combination could produce higher pressures and could eliminate the need for any preprocessing thus saving a lot of time and money. More pressing would be needed to get the same amount of oil from a single press but the cost and required work force would be much less, and a number of the smaller presses could be bought and operated simultaneously. An additional advantage of such a system would be that if one press broke down the repair or replacement cost would be affordable and there would be no interruption of oil production.

3.9 CONCLUSIONS ON 20- AND 80-TON PRESSES

Using a Scissor Jack Press presents a number of inherent difficulties and complications when extracting oil from sunflower seeds. Its potentially very strong mechanical advantage can only be utilized in a very large press. The design is sensitive to inaccurate machining and assembly, and it requires specialized materials and machining. The nut design is problematical and the threads of both the nut and bolt are subject to high stress concentrations likely to cause fatigue failures. The exact stresses in the press are highly dependent on the coefficients of friction in the threads, and are difficult to predict; The only way to guard against failure is to vastly overdesign all the components; this results in a very heavy and expensive press.

Finally, the large cage diameters used to maximize the oil

output per pressing lead to high forces and low pressures which require heavy expensive presses and large amounts of expensive preprocessing of the seeds.

The stress analysis has shown that both the 20-Ton and the 80-Ton Presses examined in this study are substantially underdesigned. Now it appears that the Scissor Jack design itself may in fact not be the most appropriate for a manually operated sunflower seed oil press.

CHAPTER 4

BIELENBERG RAM PRESS

4.1 DESCRIPTION OF THE PRESS

This small (100 kgs. weight) continuous throughput, manually operated sunflower seed press was designed by Carl Bielenberg of ATI in June, 1985. A small piston is moved back and forth inside a cylinder by the action of a hand-operated lever arm; on each back stroke a small amount of sunflower seed is fed into the cylinder from a hopper; on each forward stroke the seed is compressed by the piston and pushed into, and eventually through, a seed cake filled pressing cage. The pivoted operating lever is connected to the end of the piston by two tie arms, one on either side of the press.

As the operator rotates the lever downward, the piston is pulled into the cage by the tie arms and the seeds in the cylinder and in the cage are compressed. At some point during the stroke the static frictional forces between the seed cake and the cage are overcome and further movement of the piston forces an amount of the spent cake to be expelled from the far end of the cage. The lever arm is then rotated upward; this returns the piston to its original position and allows more seed to drop from the hopper into the now empty cylinder. This back and forth action is continued until all the seed is pressed.

The pressing cage is composed of closely spaced slats and is conical in shape. The narrow end which is 50mm in diameter is connected to the compression cylinder and the wide end is left open. At the wide, open end, of the cage there is an adjustable restriction cone that can be screwed in and out to control the area of the cage opening. This in turn controls the frictional forces and thus the pressure build-up in the cage.

In general, this press is designed to be used with whole undecorticated sunflower seed (prior decortication is recommended for very hard seed). The small diameter cylinder and cage permit high pressures to be generated and make it easier for the oil to seep through the seed cake and out of the cage.

The only recommended pretreatment of the undecorticated seed is to warm it in the sun for 5 to 10 minutes. Because it is undecorticated, the pressed oil contains a fine black powder, which is scraped off the shells during pressing. Thus the oil must be properly settled (about 2 days) or filtered before it can be used.

4.2 THEMI RAM PRESS

Themi has manufactured two different models of the Ram Press. Their first model was based on Carl's original design but recent models incorporate quite a number of design changes. Many of these changes can be traced to the press design developed by United Engineering. Themi's retail price for the Ram Press has increased from 15,000Tsh ($US 160) to 25,000Tsh ($US 265). Major differences between the new Themi design and Carl's original design as based on his drawings and reports include:

- The tie arms have been increased in size from 16mm diameter round bars to 28mm hexagonal bars, and large diameter brass lined bushings, which fit over the ends of the two end pins, have replaced the nuts at each end of the arms. The arm lengths which have been increased by 30mm can still be adjusted by screwing the threaded ends of the rods into nuts on the bushings. This major, expensive design change was apparently made to overcome the problem of the bending tie arms.

- The choke ring has been discarded completely. Four bolts now hold the large end of the cage to what appears to be a 25mm long segment of straight 100mm diameter pipe. This pipe is welded between a 13mm thick, machined annulus, connecting flange on the cage end, and the 19mm thick end plate, on the other end. A conscious decision may have been made that this change serves the same purpose as the ring -- or Themi may not have understood the proper function of the choke ring and added the section simply to imitate the original design. In either case, it seems an expensive and quite possibly unnecessary addition to the design.

- The 8mm wide machined cage tension rings have been replaced by 13mm wide by 25mm thick (radial measurement) machined annuluses, which require a lot more machining.

- The two end pins have been increased in length and the center to center separation of the tie arms is now 250mm instead of 170mm. It is not clear why this change was made.

- Brass bushings have been introduced into the pivot holes at the bottom end of the handle.

- The handle is made from a 50mm x 50mm x 2mm rectangular hollow section instead of the suggested 44mm o.d. x 3mm wall thickness pipe and its length has been reduced from 1600mm to 1360mm.

- The distance between the handle pivot pin and the tie arm connection pin has been increased from 80mm in

Carl's design to 100mm in Themi's.

- The adjustable restriction cone assembly has been redesigned so that the 20mm thick nut plate is no longer welded to the mainframe side plates. Instead it can slide into and be removed from a slot made up of small 10mm x 10mm bars which are welded to the side plates. (I did not quite understand why they felt that it should be removable.)

  The replaceable head on the end of the piston is no longer used; the piston instead has been increased in length.

In general many of these design changes seem to have been introduced with little regard for either engineering or cost-benefit analysis. Compared to Carl's design drawings (August 13, 1985), the mechanical advantage of the press has inadvertently been reduced by more than 30% and the bending stresses in the end pins have been increased by as much as 40%. The required amount of machining has also been increased seemingly for no good reason. It is not clear if a press was ever made to correspond exactly to the drawings of August, 1985, but this original design has a number of useful features that seem to have been discarded in more recent designs.

4.3 NAND SINGH RAM PRESS DESIGN

The Nand Singh Press presently sells for 30,000Tsh ($US 320). The main differences between this design and the Themi design are the following:

- The tie arms are made from the same 28mm hexagonal rod but instead of having the large bushings on the end they have been machined down to 16mm diameters at the ends and threaded. Two nuts are then used to secure them to the end pins which have been drilled through and ground flat as in Carl's design. The distance between the tie arms is 200mm.

- The distance between the bottom pivot on the handle and the tie arm attachment pin is 85mm. This gives it a higher mechanical advantage and a shorter piston stroke than the Themi Press.

- The length of the handle has been reduced by 60mm to a length of 1300mm and no brass bushings are being used on the pivots.

- The cage tension rings are made from forged 13mm x 13mm solid bar bent in a ring and welded together at the ends.

4.4 OTHER DESIGN CHANGES

When I arrived in Arusha, I found that the Oilseed Project Director was experimenting with a number of new design changes for the Ram Press. A visiting engineer from the U.K. had suggested using a shorter cage and trying to increase the pressure inside the cage by using a longer restriction cone that would extend further into the cage. Nand Singh had already constructed a press with a shorter cage. The cage length had been reduced from 200mm to 135mm, and tests had shown that there was very little if any difference in press performance and oil production rate. This was encouraging because a shorter cage is also cheaper. I had my doubts whether the longer cone would work, but a long thin cone was made. We found that even with the cone backed out to its farthest position the frictional forces were too high and the press could not be operated.

I decided to include the short cage press in my analysis, and I did measurements of force versus displacement for it and for one of the first improved Themi Presses.

4.5 MANUFACTURE OF THE PRESSES

During my stay in Arusha, I was able to watch Themi assemble a Ram Press. As I had observed for the 20-Ton Presses, the quality control was very poor. When the press was tested, its performance was unacceptable. The gaps between the slats (Themi is now making these slats on their new milling machine) of the cage were inconsistent in size; some were so large that the wet seed cake was extruded through them even at very low pressures. The brass bushings in the handle pivot were too tight so that the bushings turned with the shaft, and the other pivot holes were very loose causing a lot of sideways play in the handle. Other components were cut to the wrong sizes and the welding was very inconsistent: in some places large forces were supported by small spot welds.

Part of the problem can be traced to the lack of final production drawings for the design of the Ram Press. Other problems are caused by the lack of cutting/drilling guides and welding jigs and by the inconsistent staff supervision and management. Themi does not seem enthused about the Ram Press. One staff member commented that he didn't feel it was the type of technology on which they should focus. He preferred the larger 20-Ton scissor jack presses because they have a "higher", "more visible", oil production rate. Another complaint was that the Ram Press takes too much hard work to operate.

The Ram Presses made by Nand Singh were of a consistently high quality and finish. Although Nand Singh is a very small and not very well equipped workshop (an old lathe, a hand

held drill, and a homemade power hacksaw), its manager and owner, Mr. Autar Singh Neote, is a well trained and careful mechanic. He has keen interest in the Ram Press and keeps a close eye on the production work. Mr. Neote claims that over half the cost of the Nand Singh Press results from machining costs. His design requires substantially less machine work than does the new Themi design.

This comparison is especially interesting when one notes that Themi has a very large, very well equipped workshop, (one of the best equipped I have seen in East Africa), staffed by several well qualified mechanical engineers. Their workshop, however, is drastically under-utilized.

4.6 THE RAM PRESS IN ACTION

The Project Director keeps some twenty Ram Presses at his house and employs two young men to press oil. I was able to watch these presses at work and to use the presses myself.

My first impression was that the press took a lot of work. It seems to take two average-size men (men are somewhat smaller in Tanzania than in the U.S.) to operate it normally, one pulling and one pushing on the handle. Because the handle is quite short (1.3 meters) and rotates through 90° between a vertical position and a position parallel to the ground, it is not easy to operate it so that it efficiently uses a person's strength and weight. If the press is pulled from the back, the operator is pulling backward and not at right angles to the handle as intended. At this point, the second person starts to push; this person at least can make efficient use of his weight. When the handle is close to the ground the only easy way to apply that last little bit of force is to step on the handle. The operator is standing up and has to bend down again all the way to the ground to lift the handle up to its original position. A single stroke requires a backward pull by one person, a combination pull (very ineffective at this point) and push by two people, a final push or stamp by one or two people and finally a full deep knee or bent back lift.

This is not an easy, smooth operation. When the pressure is set a bit lower and the press is operated by a single person, it requires a backward pull down to a crouching position (very ineffective by this point) and then a stand-up-lean-over-and-push, with a final stamp or push and then the full body lift. This may be slightly easier and smoother, but it is not something most people would like to do 8 hours a day. During the hardest part of the stroke (somewhere between the pulling and pushing phase), the required applied force can reach as much as 100 kgs.

Despite the above observations, most people in the rural areas seem to be happy to do the required work. Only in a

few instances have people returned the machine or refused to use it because it is too much work.

According to the Project Director's records, the average oil output from the Ram Press is about 12 liters of oil per 50 kg. bag of seed or 0.24 liters/kg. of seed. Actual output can vary from 0.20 to 0.34 liters/kg. depending on the quality of the seed used. In a day's work of six hours, 2 or 3 people can press between 50 kgs. and 70 kgs. of seed. So the daily oil output varies from approximately 10 liters to approximately 23 liters/day depending on the seed quality and the work force.

4.7 FIELD REPORTS ON RAM PRESS

At present at least 12 Ram Presses are being used in rural Tanzanian Villages; 10 were made by Themi and 2 by Nand Singh. Overall Themi has produced 14 Presses, Nand Singh 16, and United Engineering 7. Most of these have been purchased by the Village Oil Project and are stored at the Project Director's house before being sold to new buyers. Most of the Themi Presses in the field are of the old design, but a few are of the new design.

A number of problems have been reported about the use of the presses in the field. One person returned it because it was too much work. The major mechanical problems, at least with the old, original design, seem to be that the tie arms and cage slats are bending at the small, higher pressure end of the cage. These problems seem to be quite common; during my 2 week visit a number of machines were brought in from the field to be repaired.

4.8 DETERMINATION OF FORCE VERSUS DISPLACEMENT

I conducted experiments to determine the relationship between generated force and piston displacement both on newly designed Themi press and on the Nand Singh press that was built with a short cage. The cages and the lengths of the piston strokes were different on the two presses. The Themi Press has a full piston stroke of 107mm which starts at $\emptyset = 95°$ and goes to $\emptyset = -3°$. The Nand Singh Press has a stroke of 84mm that starts at $\emptyset = 90°$ and goes to $\emptyset = -3°$. I did two separate experiments on each of the presses.

From Equation 2 in Appendix 2 we can see that the theoretical mechanical advantage (defined as the ratio between the generated force (F) and the applied force (P) of a Ram Press is directly proportional to the length of the handle (L) and inversely proportional to the distance (D) between the handle pivot pin and the tie arm pivot pin (the pivot separation). It is also a function of the angle ($\emptyset$) between the lever handle and the horizontal and the angle (θ)

between the tie arms and the horizontal. Equation 2.4 gives the angle Φ in terms of the angle Ø and the pivot offset (r) as defined in the sketch. Finally, the piston displacement (X) is given by Equation 2.5 - in terms of the tie arm length (R) the pivot separation (D) and Øo and Φo, the angles at the start of the compression stroke.

If the presses' dimensions are known one can use a programmable calculator to determine the mechanical advantage as a function of the piston displacement, X. This is the analytical result that we need for our experimental determination of the forces in the press.

Both the Themi Press and the Nand Singh short-cage press were properly primed and were operated normally for some minutes before the experiments began. Two 50 kg. spring balances were attached to the end of each handle, the applied forces were measured as a function of piston displacement and the generated forces were determined as described above and in Section 1.2.

Table 2.1 of Appendix 2 gives the results for the two experiments done on the Themi Ram Press, and Table 2.2 gives the results for the two experiments done on the Nand Singh Press. Included in the tables are the piston displacement (X), the two angles Ø and Φ, the mechanical advantage (F/P), the experimentally measured applied forces (P), and the calculated generated forces (F).

The mechanical advantages start at 13.5 for the Themi Press and at 15.3 for the Nand Singh Press. They increase with displacement slowly at first, then in the last 6 or 7mm of the stroke they climb very quickly toward infinity. For any given angle of the lever handle (Ø) the mechanical advantage of the Nand Singh press is about 12% higher than that of the Themi press, primarily because of the smaller pivot separation (D).

The forces applied to the lever of the Themi Press increased as a function of piston displacement until they reached a maximum value at X = 85mm and Ø = 38, thereafter they decreased. The maximum applied force was 65 kgs in the first test, and 74 kgs in the second. The exact magnitude of this force will vary depending on the setting of the restriction cone.

The forces applied to the Nand Singh press also increased as a function of piston displacement and reached a maximum value at X = 65 and Ø = 38° for the first test and X = 70mm and Ø = 32° for the second. Thereafter the applied forces decreased. The maximum applied force for both pressings was 95 kgs.

The generated forces reached maximum values in the same positions as did the applied forces. The maximum generated

forces in the Themi Press were 1500 kgs. and 1710 kgs. respectively for the two experiments; in the Nand Singh Press they were 2800 kgs. and 2400 kgs. in each of the two tests. These maximum applied and generated forces are needed for the stress analysis.

4.9 WHAT HAPPENS INSIDE A RAM PRESS AND ITS IMPLICATION FOR DESIGN

In this section we take a closer look at the experimental results, present a model of what is happening inside the Ram press, and examine the design implications of these results. The forces (F) in Kgs force generated on the seed are plotted versus piston displacement in (mm) for all four experiments in figure 2.1. In all cases the generated forces increase with displacement up to a maximum value and then quite suddenly drop off.

The curves for the Themi Press drop off to a local minimum value at 95mm and then increase again at the end of the stroke. Unfortunately, because of our somewhat crude method of measuring the applied forces and because the calculated mechanical advantages increase rapidly in this same region we cannot ascertain if this second peak is an actual effect.

The forces in the Nand Singh Press increase much more steeply as a function of displacement than those in the Themi Press.

To explain what is physically happening inside the presses, I will present a simple model which explains both the general shapes of the curves and the differences between the curves for the two machines.

In the first part of its compression stroke the piston is compressing the new seed which has just been dropped from the hopper into the cylinder. At first this uncompressed seed offers very little resistance to the piston and the generated force is very low, but as it becomes more strained the seed becomes harder to compress and the force increases. This first part of the curve has the same general shape as for seed compressed in the 20-Ton Press (see Figure A1.1). Basically the new seed is being compressed between the moving piston and the already compressed seed that is inside the cage. The new seed is being highly strained (changing in length), while the much stiffer old, already compressed seed is hardly being strained at all. The static frictional forces between the cage and the seed cake are stopping the seed cake from moving and no seed is being expelled. At some point, the newly pressed seed becomes just as stiff, or hard, as the old seed and subsequent straining is distributed throughout the whole seed cake. Shortly after this point the generated force from the piston overcomes the forces of static friction and the whole seed cake moves forward together as a lump, causing some seed cake at the open end to be expelled.

The peak force reached in the cycle corresponds to the point where the force from the piston just equals the sum total of the forces of static friction between the seed cake and the cage and between the cake and the restriction cone. Once this force has been reached, the cake starts to move and the generated force drops sharply -- the dynamic forces of friction now are substantially lower than the static forces. The high initial force needed to overcome the static force of friction momentarily accelerates the seed cake forward. Because the piston is moving very slowly it can't keep up and the seed comes to rest again allowing the static coefficients of friction to take over once more. By now the generated force has been reduced and can no longer overcome the friction forces. With subsequent compression the force builds up again and the whole process can be repeated.

Although this is an oversimplified description of what actually happens, this slip-and-stick phenomenon is commonly seen whenever a soft, high friction, elastic solid is pushed slowly across a hard surface. Trying to push a rolled up ball of toilet paper very slowly and evenly through a cardboard toilet roll will demonstrate this effect.

The observed second build-up of force in the Themi Press is thus easily explained. This second peak could be of a slightly different height than the first, because the new configuration of the seed cake will have slightly changed the sum total of the frictional forces. This slip-and-stick phenomenon can also explain a previously reported observation, by Carl Bielenberg, that in a press with a long piston stroke the seed was only expelled during the middle of the stroke while much of the oil was produced at the end of the stroke. The seed cake was expelled and then a second pressure peak was built up which caused the oil to be expelled. The data for the Nand Singh Press reveals that the piston stroke appears to be too short to build up this second peak.

If we consider the first portion of the pressing curve to represent the compression of the small cylinder of new seed, as explained above, then we would expect the curves derived from the two different presses to be almost the same in this region. In fact, because the two cylinders of uncompressed seed are of different lengths, we would really expect the plots of force versus cylinder strain (defined as the piston displacement divided by the maximum stroke length) to be almost the same. Figure A2.2 plots the generated force versus cylinder strain for the four experiments. The curves for the two different machines are very similar up to a cylinder strain of about 0.48. This value corresponds to a piston displacement of 50mm in the Themi Press and of 40mm in the Nand Singh Press.

As explained earlier, for any subsequent piston displacement the conclusion is that the mechanical properties for the new seed are similar to the rest of the seed cake; the displacement will be taken up as strains over the whole seed cake instead of just over the new seed. Thus, for compressions beyond this point, a press with a longer seed cage can be expected to generate less back force, (less strain) for a given piston displacement, than a press with a shorter cage. This reasoning explains why the force versus displacement curve for the long cage Themi Press rises less steeply than the curve for the shorter cage Nand Singh Press.

The value of the maximum generated force is determined by the sum of all the frictional forces between the cage and the seed and between the restriction cone and the seed. In theory a longer cage will increase the forces and a flared cage will decrease them. However, in practice the position of the restriction cone seems to have the greatest effect. The other limiting factor on this force is whether the press has a large enough mechanical advantage so that a single operator can overcome the frictional forces and expel the seed cake from the cage.

The shorter the piston stroke, the steeper the initial rise is in the generated force as a function of displacement. In the limit of a very short stroke it might take two or three strokes before the maximum pressure is reached and before any seed cake is expelled. The shorter the stroke, the more strokes are needed to press a given amount of seed. Force rises more steeply with displacement for a shorter seed cage. However, a shorter cage also means lower frictional forces. This might cause a problem if it can't be fully compensated for by using the restriction cone or by reducing the cage taper. Other fears are that if the cage is too short not all the oil will have been squeezed out of the seed cake before it is expelled or alternatively that some of the oil may inadvertently be squeezed out of the end of the cage instead of out of the sides. These two effects are difficult to estimate because they both depend on unknown properties of the compressed seed. How important is the time held at pressure to the extraction efficiency and how shallow a pressure gradient is needed to prevent the flow of oil through the seed cake? More experiments would have to be done to answer these questions before the piston stroke and cage length can be optimized.

Let us briefly now look at the pressure distribution inside the cylinder and the seed cage. The highest pressure is generated in the cylinder at the end of the piston and can be calculated by dividing the generated force by the area of the piston. This pressure is almost constant over the cross section of the cylinder. Because of the low frictional forces in the oil filled cylinder there should be very little drop in pressure over the cylinder length, so the seed cake at the beginning of the seed cage is also subjected to this

high pressure. But here, because of the gaps in the cage there is a pressure gradient over the cross section of the cage. In fact, the exact pressure distribution becomes very complicated. It is in the cylinder and in this high pressure area that most of the oil extraction probably takes place.

Between this point and the far open end of the cage the average pressure on any cross section decreases, finally reaching zero at the open end. This drop in pressure (pressure gradient) is due to two effects. One is that the high frictional forces between the cage walls and any one cross-sectional slice of the seed cake (which incidentally vary as a function of the pressure) support some of the applied load and stop it from being fully transferred to the next adjacent cross-sectional slice of the cake. Thus, the pressure decreases within each subsequent slice. The second reason for the pressure gradient is that the cage is flared and its cross-sectional area increases as you move away from the piston.

As a result the applied load is spread over a larger area and the resulting average pressure is reduced. On any given cross section there is also a pressure gradient between its middle and its outside edge. It is this gradient that causes the oil to flow out of the cage.

The effect of the restriction cone is to increase the back pressure on the seed cake. Its action can be explained in two ways. First, it reduces the effective cross-sectional area of the final section of the seed cage and thus increases the pressure and hence the frictional forces in this same area. Second, it generates its own high pressure region in the same way that the piston does. When the seed cake tries to move, the cone presses against it and a back force and an area of increased pressure are generated. Theoretically (if all the seed had similar properties) one could design a press with a straight seed cage and no restriction cone. One would only have to pick the right length for the cage so that the frictional forces were high enough to create the required pressure for oil extraction but low enough so that they could still be overcome by the maximum generated force in the press. Because one can't judge the relative impact of the above pressure controlling effects, it is difficult to predict how long or short this straight cage would have to be, and whether oil would come out of its open end. However, the idea of trying a shorter, straight cage with the incorporation of a smaller restriction cone is certainly suggested by this analysis, and could be a much cheaper alternative. Further experimentation along these lines is required.

The above analyses and descriptions of what happens inside a Ram Press are certainly oversimplified. I have not accounted for the time dependence of the oil extraction or for the effect of the oil itself. Nonetheless, this information is

useful for suggesting possible design changes in the Ram Press.

4.10 FORCE AND STRESS ANALYSIS

For any given type of sunflower seed the magnitude of the peak generated force in a Ram Press is determined by the value of the static frictional forces between the seed and the press walls, the setting of the restriction cone, the strength of the operator and the mechanical advantage of the press. The position of the peak is also determined by the length of the piston stroke and the length of the pressing cage. For both the Themi and the Nand Singh presses, force occurred when the angle Ø was somewhere between 30° and 40°. At this position the mechanical advantages of the two presses are still quite low and thus they become the limiting criteria for the forces. If, after this same amount of compression, the angle Ø had been at 10° instead of 35°, the mechanical advantages would have been much higher and the restriction cone could have been screwed in deeper to allow for a much higher generated force peak, with a lesser amount of effort on the handle. As it is, a very large applied force is required (up to 100kgs) to reach a moderately high peak in generated force. The maximum possible value of generated force in these presses is determined by the product of the maximum possible applied force and the mechanical advantage at the peak. In this case the peak is at Ø = 38° for the Themi Press and at Ø = 32° for the Nand Singh Press. The fact that this peak occurs at almost the same angle for the two presses is a coincidence caused by the similarity of ratios of cylinder length to cage length in the two presses.

This stress analysis assumes the maximum applied force to be 130kgs or 1275 Newtons. Two men pushing or a single man jumping could apply a load of this size to the end of the handle. Such rough treatment by the press operators, although not recommended, is not unlikely. In this section, I use the analysis given in Appendix 2 to determine some of the crucial forces and stresses generated in both the Themi and the Nand Singh Presses. The maximum allowable stresses as determined in Chapter 1 are given for comparison.

Dimensions:

			Themi	Nand Singh
L	=	Length of Lever Handle	1360mm	1300mm
D	=	Pin Separation on Handle	100mm	85mm
R	=	Length of Tie Arms	730mm	675mm
r	=	Handle Pivot Offset	35mm	35mm
Length of Cage			200mm	135mm
Length at Piston Stroke			107mm	85mm
$\emptyset_o$	=	Lever Angle at Start of Stroke	95°	90°
$\emptyset_o$	=	Lever Angle at End of Stroke	-3°	-3°
$\emptyset_m$	=	$\emptyset$ at Maximum Generated Force	38°	32°
Φ_m	=	Φ at Maximum Generated Force	2.0	0.9

			Themi Press	Nand Singh Press
Force Analysis:				
P	=	Maximum Applied Force	1275 N	1275 N
F/P	=	Mechanical Advantage at $\emptyset_m$ (Eq. 2.2)	23.1	29.6
F	=	Maximum Generated Force (Eq. 2.2)	29,450 N	37,700 N
T	=	Maximum Tension in Tie Arms (Eq. 2.6)	14,740 N	18,850 N
P	=	Maximum Pressure in Cylinder (Eq. 2.5) ($r_{)\Sigma}$ = 25mm)	15 MPa	19.2 MPa

The above values of maximum forces are used for the stress analysis. In some cases, to give the minimum allowable dimensions for a given component, the equation is solved in reverse. These are the values that could be used if the press is redesigned.

STRESS ANALYSIS

	Maximum Allowable Stresses	Themi Press	Nand Singh Press
Stress in Tie Arms			
		(r=14mm)	(r=8mm) (at ends)
Tensile Stress (Eq. 2.7)		24MPa	94MPa
Bending Stress (Eq. 2.8) r_p = 20mm		(f=0.2 brass bushing) 27MPa	(f=0.3 no bushing) 281MPa
Total Stress ($r_T + r_B$)	69MPa (stress concentration in threads)	51MPa	375MPa
Minimum Allowable Tie Arm Radius		13mm	15mm
Stresses in Pins			
Bending Stress (Eq. 2.9) (r=20mm) (l'=60mm)	138MPa	(l=250mm) 258MPa	(l=200mm) 255MPa
Shearing Stress (Eq. 2.10) (r=20mm)	69MPa	12MPa	15MPa
Bearing Stresses (w = 20mm)	160MPa	18MPa	24MPa
Stresses in Lever Arm	(Hole is stress concentration)		
Bending Stress at Pivot Arm Pin (Eq. 2.12) (b=40mm, h=70mm)	69MPa	50MPa	47MPa
Bending Stress at Bottom End of Lever Handle (Eq. 2.13) (h_o=50mm, h_i=45mm)	139MPa	(L'=1200mm) 213MPa	(L'=1160mm) 206MPa
Stresses in Pressing Cage			
Bending Stress in Slats (Eq. 2.14) (h=9mm, l=50mm)	138MPa	213MPa	296MPa
Hoop Stress in First Tension Ring (Eq. 2.16) (r-27mm)	138MPa	(t=23mm, w=13mm) 69MPa	t=w=10mm) 259MPa

Maximum Allowable Distance Before First Tension Ring (Eq. 2.15)	38mm	34mm
Minimum Allowable Cross-Sectional Area of Tension Rings at Maximum Allowable Separation (Eq. 2.17)	111mm2 (10mmx11mm)	128mm2 (10mmx13mm)

A number of things can be learned from the above force and stress analysis. The mechanical advantage and thus the generated forces and the pressure are higher for the Nand Singh Press than they are for the Themi Press. As a result most, but not all, of the stresses are also higher.

The stresses in the newly strengthened Themi tie arms are below the maximum allowable values while the stresses in the Nand Singh Press are too large by a factor of 5.4 (in fact the ends of these arms were bent during the Arusha test). However, the calculation of the minimum allowable radius for the tie arms showed that they bent because the 28mm hexagonal rods had been machined down to fit 16mm diameter nuts. If 30mm diameter rods had been used instead and had been threaded for 30mm diameter nuts there would have been no problem. This would have been a much cheaper alternative than the changes Themi made using the large, brass lined bushings.

The bending stresses in the pin at the piston end are too large in both presses and a bending or fatigue failure would be expected. Because of the shorter pins in the Nand Singh design the stresses are lower than for the Themi design. This is so despite the higher compression forces in the Nand Singh Press. If the tie arm separation were reduced to 170mm as in Carl's original drawings, the stresses would be reduced to 164MPa in the Themi Press and to 210MPa in the Nand Singh Press. At this separation a 42mm pin diameter in the Themi Press and a 46mm diameter pin in the Nand Singh Press would bring the stresses down to an acceptable level.

Because the bending stresses are also too large at the bottom of the lever arm in both presses, failure is expected. This problem could be most easily solved by welding a 50mm x 6mm x 300mm bar to each side of the bottom end of the arms. The stresses in the cage are also too high; quite a number of presses have been returned from the field with bent slats. Reducing the distance between the front of the cage and the first tension ring should solve this problem. The appropriate correct distances would be 38mm for the Themi Press and 34mm for the Nand Singh Press. The distance between subsequent tension rings could be slightly longer as the pressure in the cage is lower further away from the cylinder.

The tension rings on the Themi Press are overdesigned and those on the Nand Singh Press are underdesigned. A 12mm x

12mm square bar bent and properly welded into a circle would be a cheap and adequately strong alternative if the rings were spaced as suggested above.

4.11 A NEW DESIGN FOR THE RAM PRESS

Although the Themi and Nand Singh designs can be modified slightly so that the presses will not fail during use, the problems in the design of the Ram Press are more basic.

In the preceding sections I have argued that the press requires too much force to operate and that it is not designed to make optimum use of a man's weight and strength. The operation is somewhat awkward and the required forces are up to 100kgs. I have also demonstrated that it is the variation of the mechanical advantage with piston position that causes the press to require so much force. At the point in the compression stroke when the required generated force is high, the mechanical advantage of the press is still quite low, and a large applied force is needed. I have also shown with my experiments on undecorticated seed in the 20-Ton Press that higher pressure alone can effectively increase the yield of oil. (Of course at some point there are diminishing returns.) This means in order to produce higher pressures with less applied force we need to maximize the mechanical advantage of the press. We also need to change the position of the handle lever during operation so that a man's weight and strength can be used to greater advantage.

In order to carry out these tasks we must first examine closely Equation 2.2 for the mechanical advantage of a Ram Press. The mechanical advantage is proportional to L, the handle length, and inversely proportional to D, the distance between the handle pivot pin and the tie arm pin. This immediately suggests a longer handle and a shorter D. The mechanical advantage is also a somewhat complex function of the angle Ø, the pivot offset r, the tie arm length R, and again the distance D. This function is made explicit by combining Equation 2.2 and Equation 2.3. This function can only be properly evaluated and optimized with the help of a computer, but the basics can be determined more easily.

The mechanical advantage is not very sensitive to variations in either R or r but it can be shown with a programmable calculator that for any given piston displacement, X, one can increase the mechanical advantage by as much as 10% by reducing r from 35mm (its value in both presses) to zero. This is our first suggested design change.

The variation with angle Ø gives the largest change of the mechanical advantage. Approximately, it varies as one over sine. This means that it starts off increasing slowly as Ø decreases from 90° to 20° to and it then increases very rapidly towards infinity as Ø decreases from 20° to 0°. With

both the Themi and Nand Singh design the maximum generated force is reached at Ø = 38° and the high mechanical advantage for Ø less than 20° is wasted. What this suggests from a design point of view is that perhaps we should make a complete piston stroke go from, say, Ø = 60° to Ø = 0° instead of from Ø = 90° to Ø = 0°. In this way, at the point of maximum force the angle will be, say, 15° instead of 38° and the corresponding mechanical advantage could be 2.5 times higher.

In order to test this idea I rigged the two presses so that the handle rotation was restricted to go from 74° to 0° for the Nand Singh Press (a stroke length of 58mm) and from 58° to 0° for the Themi Press (a stroke length of 45mm). This shorter stroke length meant less seed was compressed on each stroke but it also meant a shorter movement of the lever arm for each stroke. The length of the tie arms had to be slightly adjusted to allow the seed to fall into the hopper on the back stroke. The result was that much higher force, so higher pressure, was generated in the press (this was evidenced by the bending of the tie arms on the Nand Singh Press) and required applied force was substantially less. Unfortunately, I could not conduct a conclusive controlled comparison of the oil yield between this and the original design. In the few tests we did, we found that the yield from the Themi Press with the shorter stroke length and resultant higher pressure, was 25% higher than the yield with the longer stroke. For the Nand Singh Press we found going to a shorter stroke increased the yield by about 8%. However, the tests were conducted on different days under slightly different humidity conditions, etc.

The only problem with this modified press was that the handle lever was being pushed from waist height to the ground and then had to be lifted with a full-back or deep-knee bend. It was only slightly less awkward than the motion in the original design. In order to solve this problem and at the same time increase the mechanical advantage still further, a new handle head was designed which has a removable handle pivot pin and the tie arm pivot pin. Now at the start of the stroke the handle is at an angle of 100° to the horizontal while the angle Ø between the two pins and the horizontal is only 50°. In the final position the handle is 50° to the horizontal and the angle Ø° is zero. The pin separation (D) was kept at 100mm. The removable handle is 2 meters long and has three crossbars on which to pull, one in the middle, one at the end and one halfway between. At the start of the stroke required force is minimal and the bottom crossbar is pulled backward. As the operator walks and pulls backward, and as the required force increases, he steps his hands up to each subsequent bar as if climbing a ladder. In this way his mechanical advantage increases as more force is needed and he can apply his weight by hanging on the top crossbar to get through the hardest position. In the final position the end of the handle is at chest height and can simply be pushed

back across to its original position.

A prototype of this new handle and pivot design was built under my supervision at the Nand Singh workshop and tested. We put it on the short caged Nand Singh Press. The results were very encouraging. The amount of force required was greatly reduced; so much so that a single 50Kg (5' 4") woman could operate the press with no difficulty. She suggested that we add the third handle crossbar. Although there was no time for a controlled test, the oil yield appeared, on that first pressing, to be about 10% to 15% higher than previous yields, using the same seed as the original press. Even the Project Director and his two laborers, who had been using the old press for a couple of years, agreed that the required motion was much smoother and put much less strain on the lower back. This handle is designed to be operated by a single person; because of its position during operation it is unlikely two or more people can jump on it and overload it.

I recommend that the angle between the lever arm and the pins should be reduced to 40° on any subsequent models. This will drop the position of maximum required force by 10° and allow for better utilization of the operator's weight. There are a number of other design changes that should be made on this initial prototype before it is properly field tested. Also, the full stress analysis should be carried out and the rest of the press be appropriately redesigned to accept the higher forces.

A number of other smaller changes were also made on this new prototype. The 28mm hexagonal tie arms were threaded for 25mm nuts instead of 16mm nuts, to increase their strength and also to reduce the machining costs. The end pin through the piston was increased in diameter from 40mm to 50mm, to stop it from bending, and the piston's length was changed to allow for the shorter stroke length and the new geometry. Another significant change was to roughen the end of the piston with 4mm deep sharp ridges. The ridges will produce localized higher forces on the seed and hopefully facilitate the breaking of the shell. Again, there was no time to properly evaluate the effectiveness of this change but such an evaluation should be conducted. We noticed that the shorter piston stroke and higher pressure tended to leave some oil in the cylinder after the back stroke. This problem could be easily solved by putting a small drainage hole in the bottom of the cylinder.

Rough sketches of the new handle design and how it works with the press are shown in Appendix 3. This is only a first prototype. Since I finished this work in Arusha, The Project Director has informed me that he has further tested the prototype and is still very happy with its performance compared to the old designs.

A number of other design changes should be tried on the Ram Press before the design is finalized. As Carl Bielenberg has mentioned in early reports and as I have talked about in an earlier section, the shape and length of the pressing cage can be optimized for maximum oil production at the lowest cost. Also, a number of other cost saving simplifications in manufacturing processes and design (i.e. eliminating the brass bushings or "restriction" pipe, etc.) can be explored.

4.12 CONCLUSIONS ON THE RAM PRESS

In this Chapter I have discussed two models of the Bielenberg Ram Press; one made by Themi Engineering and one by Nand Singh Ltd. Experiments have been carried out to measure the forces and a complete force and stress analysis has been done on each press. The mechanics of the press have been explained and these results have been used to suggest design changes. A new prototype has been built and tested and the results have been very encouraging. Until now the design of the Ram Press has not been optimized and it does not make very efficient use of manpower to generate the required forces. Nonetheless, its small diameter cage and continuous throughout capacity allow it to effectively squeeze oil from the sunflower seed. The suggested design changes will allow higher forces to be generated with less effort and will more efficiently use the operator's strength and weight.

Neither of the two designs is very durable. Each has components which are underdesigned and may fail after heavy usage. However, minor changes can be made to the designs to solve these problems. The Themi Press is also likely to fail because of incomplete welding, poor adherence to tolerances, and lack of quality control procedures. If the suggested design changes are made to reduce the stresses on the weak components, then little maintenance or repair of the presses should be required other than regular cleaning and oiling of the bushings. Also, if these changes are made, the presses should be able to withstand occasional heavy overloads with no ill effects.

The only part of the press which is difficult to manufacture is the tapered cage, but even this can be done in a small, sparsely equipped workshop as is demonstrated by Nand Singh. The most important factor in obtaining a quality press appears to be the dedication of the staff and not the quality of the available tools.

With careful design the price of the presses should be able to be reduced. Some parts are overdesigned, others are unneccessarily complex and others are machined when simpler techniques such as forging could serve. It is recommended that experiments be tried using a shorter straight cage with smaller restriction cone. If this works, manufacturing costs

could again be reduced.

The biggest complaints about the Ram Press are that it takes too much force to operate and that it has a relatively low daily output of dirty oil. If the design changes recommended in this report and demonstrated in the new prototype are followed, I believe the first of these problems will be solved.

The daily output varies from 10 liters to 25 liters depending on the seed used and the dedication of the work force. This compares to 50 liters per day for the 80-Ton Press. On the other hand, the average yield from the 80-ton Press is only 0.22 liters per kg of whole seed while for the Ram Press it is 0.25 liters per kg. With the suggested design changes a higher pressure will be reached and an even higher yield will be expected. Also, since the press will require less effort to operate, workers should be easier to recruit. In principal, the press might even be operated by women.

Dirty oil can be left standing to settle for a few days or be filtered through a paper filter.

The press is lightweight, cheap, easy to transport and install and it requires no seed pre-processing. Even if only the suggested modifications are made to increase its strength, it is a very appropriate device for the rural and semi-urban poor. If, on the other hand, the ideas demonstrated in the new prototype are properly followed up and an improved press is produced, I would recommend it even for large scale urban production of oil, where 3 or 4 presses could be operating in the same plant. If the presses are sold, as is, with no modification, many small repairs may be necessary in the years to come.

Because they can't be repaired locally in the rural areas the presses will have to be transported back to Arusha. Because they are lightweight and easily transportable this should not cause too great a hardship to the customers. However, the appropriate design changes should be made before more presses are sold.

In summary, I feel that if the required design changes are made the Bielenberg Sunflower Seed Oil Ram Press is a very appropriate press to be manufactured and operated in Tanzania.

APPENDIX 1

FORCE AND STRESS ANALYSIS FOR SCISSOR JACK PRESSES

Definitions:

Forces

P	=	Force applied to end of handle during operation (Newtons)
F	=	Force generated by press on the seed (Newtons)
L	=	Tensile Load in Spindle (Newtons)
S	=	Compressive Load in Scissor Arms (Newtons)

Torque and Moment

T = Torque M = Bending Moment

Dimensions

l	=	Length of handle arms and other dimensions (mm)
R	=	Length of scissor arms (mm)
X	=	Displacement of Plunger (mm)
D	=	Diameter of Pressing Plunger (mm)
d_m	=	Mean diameter of spindle (mm)
r_m	=	Mean radius of spindle (mm)
d_i	=	Root diameter of spindle (mm)
pit.	=	Pitch of threads on spindle (mm)
d_c	=	Mean diameter of thrust collar or bearing (mm)

Angles

Ø	=	Angle between scissor arms and vertical (degrees)
Øo	=	Angle Ø at top of plunger stroke
u	=	Helix angle of spindle ($\tan u = \frac{(pit)}{\pi d_m}$)

Coefficients of Friction

f = Coefficient of friction between the nut and the spindle threads.

f_c = Coefficient of friction between thrust collar or thrust bearing and thrust piece.

Moments of Inertia

I = Moment of Inertia = $\frac{bh^3}{12}$ for rectangular cross sections

b = Width of cross section
h = height of cross section

I = $J/2$ for circular cross section
J = Polar Moment of Inertia = $\frac{\pi r^4}{2}$

r = radius of cross section

Pressure and Stresses

P = pressure on seeds in cage (MPa)

r = tensile or compressive stress (MPa)

t = shear stress (MPa)

FORCE ANALYSIS

From Figure 1 we find that:

$$F = < \frac{\cos\phi}{\sin\phi} \qquad \text{(Eq. 1.1)}$$

$$X = 2R\,(\cos\phi_o - \cos\phi) \qquad \text{(Eq. 1.2)}$$

The torque applied to the spindle must equal the sum of the torque from the threads and the torque from the thrust collar. The following equation can be derived from a simple force analysis for a square thread spindle with a thrust collar.

$$T = PL = < \frac{(dm\,(f + and)}{2\,(1-f\tan u)} + \frac{dc\,fc)}{2} \qquad \text{(Eq. 1.3)}$$

$$< = \frac{2Pl}{dm\,\frac{(f + \tan u)}{(1- f\tan u)} + d_c f_c} \qquad \text{(Eq. 1.4)}$$

The Theoretical Mechanical Advantage is:

$$\frac{F}{P} = \frac{21(\cos\phi)}{dm\,\frac{(f + \tan u)}{(1- f\tan u)} + d_c f_c \quad (\sin\phi)} \qquad \text{(Eq. 1.5)}$$

The Pressure Inside the Cage is:

$$p = \frac{F}{\pi(D)2\,(2)} \qquad \text{(Eq. 1.6)}$$

STRESS ANALYSIS

Stresses in Spindle

$$\text{Tensile Stress} = r = \frac{L}{\pi(r_m)2} \qquad \text{(Eq. 1.7)}$$

Torque in Spindle between nut and thrust piece:

$$T = PL - L \frac{(dc)}{(2)} f_c$$

Shear Stress: $t = \frac{Trm}{J} = \frac{2T}{\pi r_m^3}$ (Eq. 1.8)

Maximum Shear Stress: (Eq. 1.9)

$$tmax = \sqrt{\left(\frac{r}{2}\right)^2 + t^2}$$

Stresses in Threads

It is assumed that 90% of the load is carried by the first 4 full threads.

Definitions:
- w = width of thread
- t = depth of thread
- d = do for nut
- d = di for spindle

Approximate Bending Stress at Root of Threads:

$$r = \frac{(0.9)}{(4)} \frac{3t}{\pi dw^2}$$ (Eq. 1.10)

Approximate Shear Stress at Root of Threads:

$$t = \frac{(0.9)}{(4)} \frac{(L)}{\pi dw}$$ (Eq. 1.11)

Maximum Shear Stress at Root of Threads:

$$tmax = \sqrt{\left(\frac{r}{2}\right)^2 + t^2}$$ (Eq. 1.12)

Stress in Scissor Arms

Compressive Force in Arms:

$$S = \frac{L}{2\sin\phi}$$ (Eq. 1.13)

Compressive Stress in Arms: A = X-sectional area of arms

$$r = \frac{S}{A}$$ (Eq. 1.14)

Stresses Around Pin Holes

D = Pin Diameter
A = Thickness of Arm or Plunger Slab

Bearing Stress in Arm Holes:

$$r = \frac{S}{2a} \qquad \text{(Eq. 1.15)}$$

Bearing Stress in Plunger Slab:

$$r = \frac{F}{2aD} \qquad \text{(Eq. 1.16)}$$

Stresses in Pins

Shear Stress:

$$t = \frac{2S}{\pi D2} \qquad \text{(Eq. 1.17)}$$

Bending Stress:

$$r = \frac{4Sc}{\pi D^3} \qquad \text{(Eq. 1.18)}$$

Stresses in Main Frame

Tensile Stress in Columns: A = X-sectional area

$$r = \frac{F}{2A} \qquad \text{(Eq. 1.19)}$$

Bending Stress in Top and bottom Cross Bars:

l = length of Cross Bar
l' = length of distributed land
b = width of Cross Bar
h = height of Cross Bar

$$r = \frac{Mh/2}{I} = 3/2 \frac{F(l-l1/2)}{2bh2} \qquad \text{(Eq. 1.20)}$$

This calculation overestimates the stress in the bottom cross-bar, because in reality the moment of inertia (I) is increased by the circular base plate of the pressing cage.

Stresses in Turning Handles

The maximum stresses are bending stresses at the spindle end of the handle, they are:

$$r = \frac{Plc}{I} \qquad \text{(Eq. 1.21)}$$

where c is the half thickness of the handle and P is the force applied to the handle.

Stresses in the Pressing Cage

The problem of accurately determining the stresses in the pressing cage is difficult, but there are a number of approximations that can be made.

- We assume that the pressure inside the cage is constant and that it has a magnitude equal to the generated force divided by the cross-sectional area of the plunger $p = \frac{F}{\pi(\frac{D}{2})^2}$

- We thus take the slats to be loaded uniformly by the pressure on the inside and to be supported on the outside by the tension rings.

- After determining the stresses in the slats and the tension rings we can use the same formulations to find the maximum allowable tension ring spacing and the minimum allowable tension ring diversions for a given slat size.

Bending Stresses in Slats:

The problem of bending stresses in a multiply supported uniformly loaded slat is an indeterminate problem. Hence, we choose to approximate the bending stresses by considering a length of slat that is held at each end by fixed supports with a separation (1) equivalent to the tension ring separations. This should give reasonable approximations of the maximum bending moments and thus of the maximum stresses.

The total distribution load on the slat is the maximum pressure p, times the exposed area of the slat. (Load) = plb where b is the width of the slat. The maximum moment is at the ends and has a value of:

$$M = \frac{(Load)\ l}{12}$$

Bending Stress in Slats:

$$Q = \frac{(Load)\ l}{2bh2} = \frac{pl2}{2h2} \qquad \text{(Eq. 1.22)}$$

where h = thickness of the slats

The Maximum Allowable Separation Between Tension Rings:

$$l\ max = \frac{v2r_allow_h2}{P} \qquad \text{(Eq. 1.23)}$$

Hoop Stress in Tension Rings

Each ring must support a tensile load equivalent to the pressure load over the area between the rings:

$$Load = {}_pDl = 2rwt$$

The Hoop Stress in the Rings:

$$r = \frac{{}_pDl}{2wt} \qquad \text{(Eq. 1.24)}$$

D is the cage diameter, w is the width of the ring, and t is the thickness of the ring.

Minimum Allowable Ring Cross Sectional Area:

$$(Wt)min = \frac{{}_pDl}{2r\ allow} \qquad \text{(Eq. 1.25)}$$

Table 2.1

Force vs. Displacement Data for Nand Singh Press with Short Cage

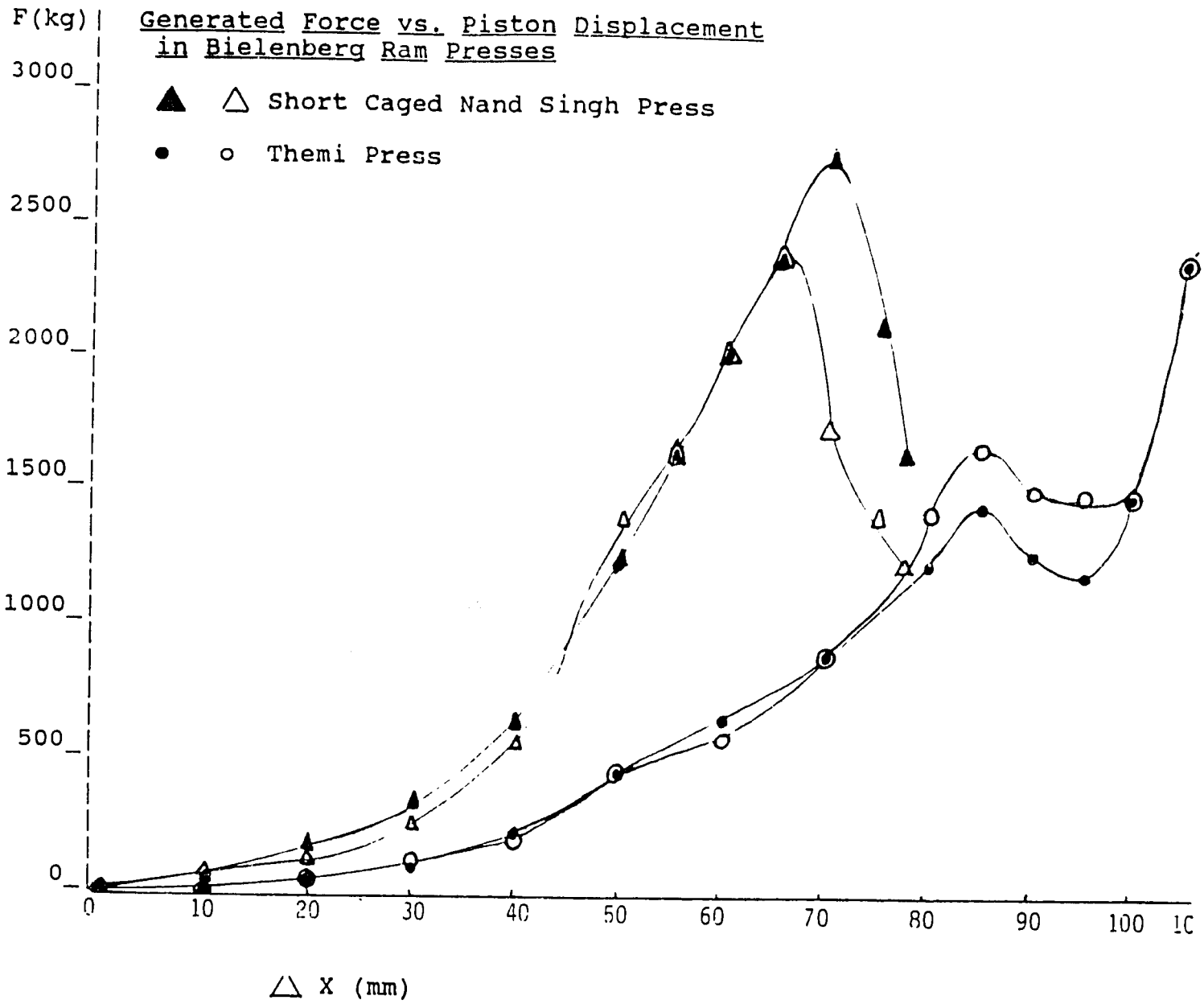
$\triangle X$(mm)	θ	ϕ	F/P	Test 1 P(kg)	Test 1 F(Kg)	Test 2 P(Kg)	Test 2 F(Kg)
0	90°	4.25°	15.3	0	0	0	
10	83°	4.1°	15.6	5	78	4	62
20	76°	4.0°	16.0	8	128	12	192
30	69°	3.8°	16.8	16	268	20	336
40	61°	3.3°	18.1	32	580	36	650
50	53°	2.8°	19.9	72	1430	65	1290
55	49°	2.5°	21.1	80	1685	80	1685
60	44°	2.0°	22.9	90	2060	90	2060
65	38°	1.5°	25.7	95	2440	95	2440
70	32°	.85	29.5	95	2800	60	1770
75	25°	.1°	36.3	60	2180	40	1450
77.5	21°	-.4°	41.9	40	1680	30	1260
80	16°	-1.0°	52.3				
84	0°	-3.0°	302.				
84.2	-3°	-3.35°	2500°				

Table 2.2

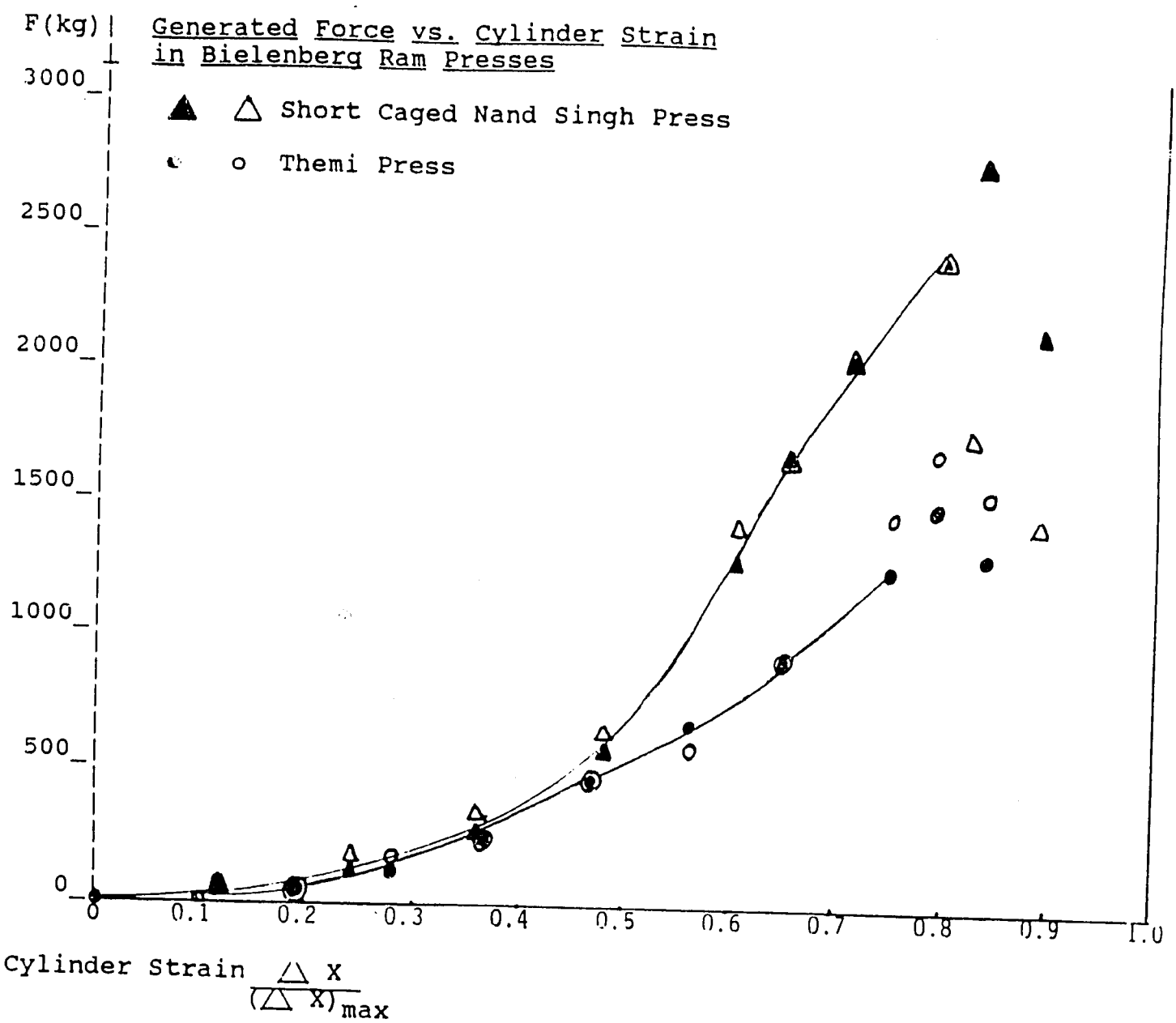
Force vs. Displacement Data for Themi Ram Press

$\triangle X$(mm)	θ	ϕ	F/P	Test 1 P(kg)	Test 1 F(Kg)	Test 2 P(Kg)	Test 2 F(Kg)
0	5.1°	95°	13.5	0	0		
10	5.05°	89°	13.6	0	0		
20	5.0°	84°	13.8	4	55	0	0
30	4.85°	78°	14.1	8	113	10	142
40	4.7°	71.5°	14.8	16	236	14	206
50	4.4°	65°	15.5	30	467	30	470
60	3.9°	58°	16.8	40	670	35	586
70	3.3°	51°	18.4	50	918	50	918
80	2.6°	43°	21.0	60	1260	70	1470
85	2.0°	38°	23.1	65	1500	74	1710
90	1.5°	33°	26.0	50	1300	60	1561
95	0.8°	27°	30.8	40	1232	50	1540
100	- .2°	19°	42.2	36	1520	36	1520
105	-1.6°	8°	81.5	30	2445	30	2445
106.6	-2.7°	0°	288.4	0		20	
106.8	-3.16°	-3°	4860.				

(Figure 2.1)

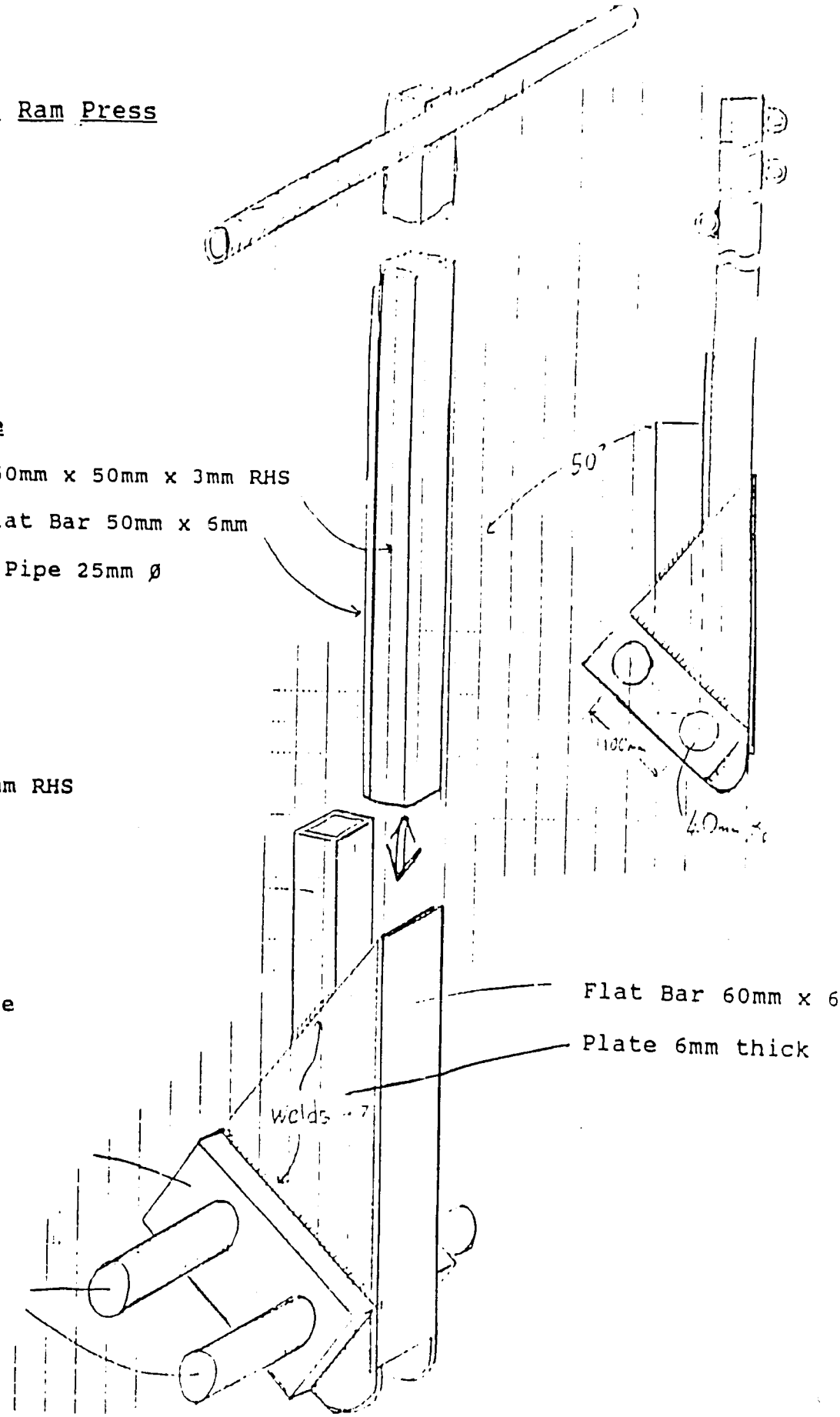

F(kg)
Generated Force vs. Piston Displacement in Bielenberg Ram Presses
Short Caged Nand Singh Press
Themi Press
3000
2500
2000
1500
1000
500
0
0
10
20
30
40
50
60
70
80
90
100
X (mm)

(Figure 2.2)


F(kg)
Generated Force vs. Cylinder Strain in Bielenberg Ram Presses
Short Caged Nand Singh Press
Themi Press
3000
2500
2000
1500
1000
500
0
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
Cylinder Strain $\frac{\Delta X}{(\Delta X)_{max}}$

APPENDIX III

New Prototype Ram Press



Removable Handle

Handle - 2mm x 50mm x 50mm x 3mm RHS

Reinforcing - Flat Bar 50mm x 6mm

3 Cross Barrs - Pipe 25mm ø

Handle Support

50mm x 50mm x 3mm RHS

2 Bushings

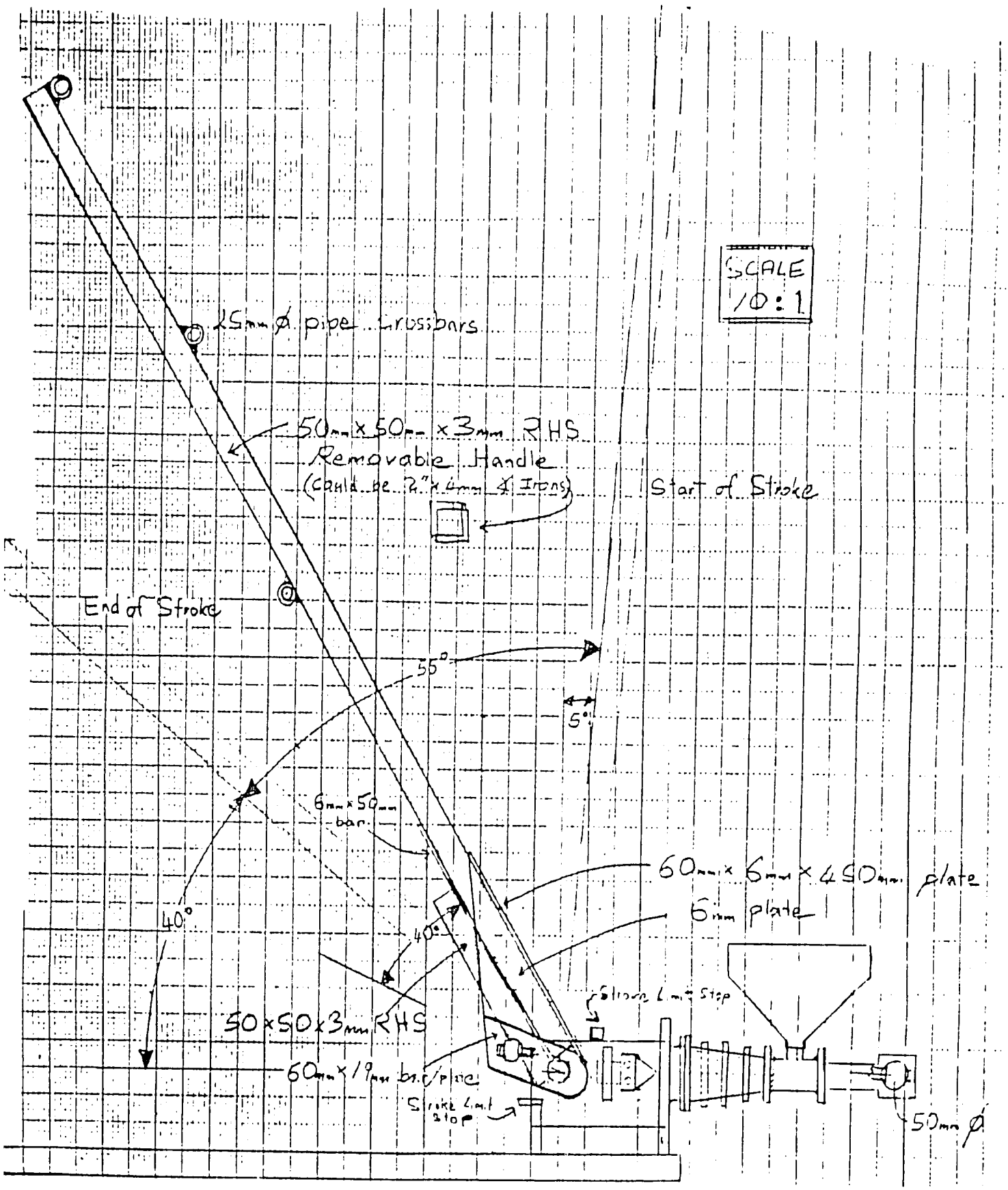
Flat Bar or Plate

19mm x 60mm

Removable Pins

40mm øo

ROUGH SKETCH OF HANDLE AND HEAD PROTOTYPE MADE AT NAND SINGH



ROUGH SKETCH OF IMPROVED PROTOTYPE FOR RAM PRESS

(Head angle can probably be reduced to 35_o instead of 40_o)

CALCULATION OF FORCE VS. POSITION IN IPI

20 TON PRESS

$$\alpha = \cos^{-1}\left(\frac{\overset{\boxed{2}}{\Delta X}}{2R} + \overset{\boxed{1}}{\cos\alpha_o}\right)$$

$$R = 445 \text{ mm} \quad \boxed{3}$$

(Seed precompressed to 75 mm in 225 mm deep cage)

ΔX (mm)	θ °	F/P	Experimental Result Machine ① $P_{①}$ (kgs)	$F_{①} = (217)P\left(\frac{\cos\alpha}{\sin\alpha}\right)$ (kgs)	Experimental Result Machine ② $P_{②}$ (kgs)	$F_{②} = (225)P\left(\frac{\cos}{\sin}\right.$ (kgs)
0	$X_o = 30.7$	365	0	0	0	0
10	29.4	386	.5	193	1	400
20	28	408	1	408	1	425
30	26.6	433	1.5	650	1	450
40	25.2	460	2	920	1	480
50	23.6	497	4	1,990	1	515
60	21.9	540	4	2,160	2	1,120
70	20.1	593	6	3,560	3	1,850
80	18.2	660	8	5,280	11	7,530
85	17.1	705	13	9,170	13	9,500
90	16	757	15	11,350	21	16,500
95	14.8	823	22	18,100	24	20,440
97.5	14.1	864	27	23,330	24	21,500
100	13.5	906	28	25,310	25	23,430
102.5	12.8	955			26	24,830
105	12.	1,020	31	31,650	27	28,580
110	10.4	1,182		L_{max}=6,730 kg.		L_{max}=6,080 kg
115	8.5	1,452				
120	5.9	2,100				
123	3.6	3,450		σ_{max}=18MPa		σ_{max}=16MP

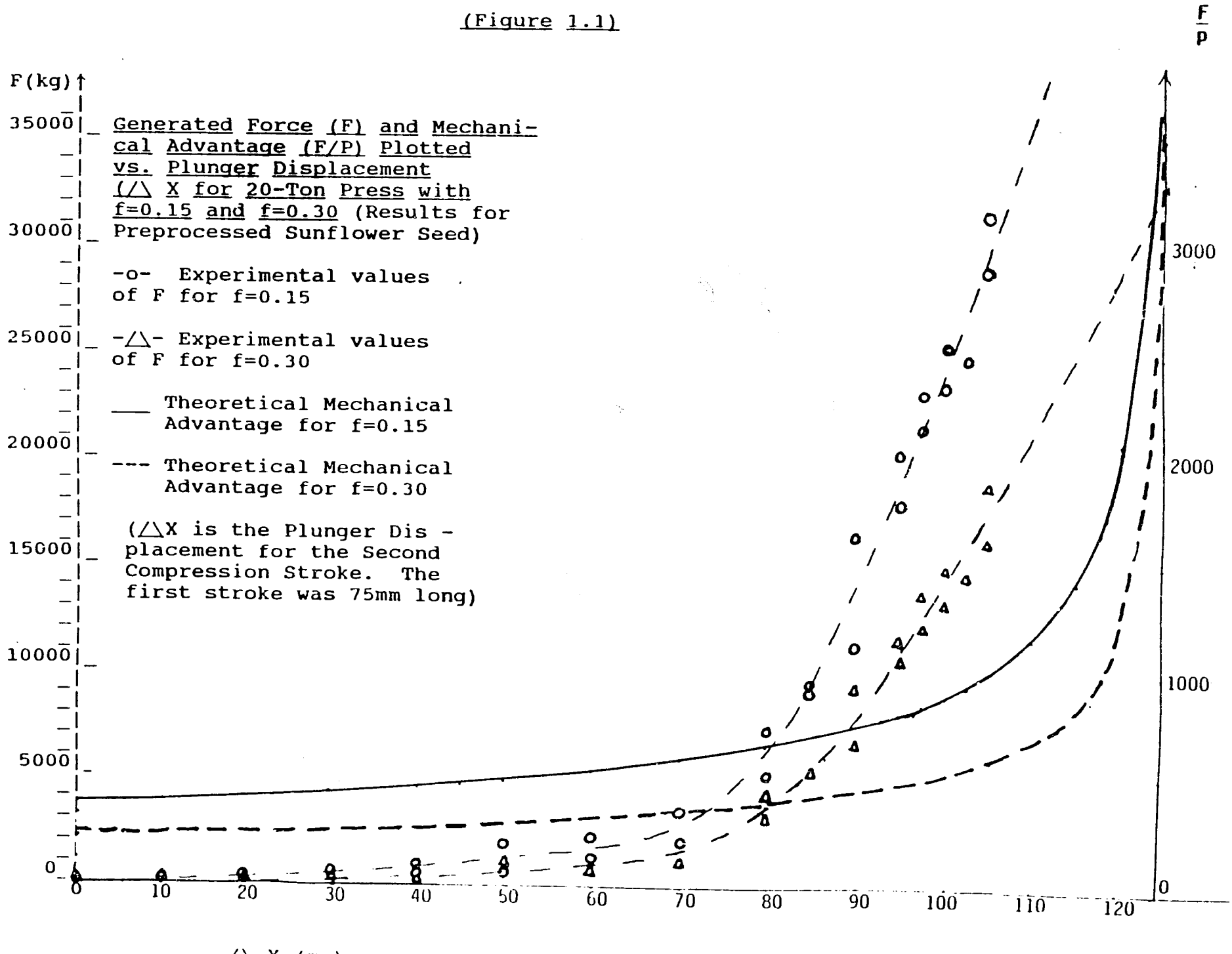
$$\text{Pressure} = \sigma = \frac{\text{Force}}{A} = \frac{\text{Force}}{\pi r^2} = \frac{F\ (9.8)}{\pi\ (75\text{mm})^2}$$

DATA OF FORCE VS. POSITION FOR 20-TON PRESS

with f = 0.3

			Test 1		Test 2	
X	O-	F/P	P_1	F_2	P_1	F_2
0	30.7	217	0	0	0	0
10	29.6		.5	114	1	228
20	28	242	1	242	1	242
30	26.6	257	1.5	385	1	257
40	25.2	273	2	547	1	273
50	23.6	293	4	1,180	1	294
60	21.9	320	4	1,280	2	640
70	20.1	352	6	2,110	3	1,050
80	18.2	391	8	3,130	11	4,300
85	17.1	418	13	5,440	13	5,440
90	16	449	15	6,730	21	9,420
95	14.8	487	22	10,700	26	11,700
97.5	14.1		27	13,800	24	12,300
100	13.5	536	28	15,000	25	13,400
102.5	12.8				26	14,700
105	12	605	31	18,800	27	16,300
110	10.4	701				
115	8.5	860				
120	5.9	1,245				
123	3.6	2,045				
124	2.3	3,170				

(Figure 1.1)


F(kg)
35000
30000
25000
20000
15000
10000
5000
0
F/P
3000
2000
1000
0
0
10
20
30
40
50
60
70
80
90
100
110
120
/\ X (mm)
Generated Force (F) and Mechanical Advantage (F/P) Plotted vs. Plunger Displacement (/\ X for 20-Ton Press with f=0.15 and f=0.30 (Results for Preprocessed Sunflower Seed)
-o- Experimental values of F for f=0.15
-/\- Experimental values of F for f=0.30
___ Theoretical Mechanical Advantage for f=0.15
--- Theoretical Mechanical Advantage for f=0.30
(/\X is the Plunger Displacement for the Second Compression Stroke. The first stroke was 75mm long)

APPENDIX 2

STRESS ANALYSIS FOR RAM PRESS

Definitions

Forces

P = force applied by operator on end of handle (kgs)(N)

F = generated force on seed (kgs)(N)

T = tension in tie arms (N)

Pressure and Stresses

P = pressure in seed cage (MPa)(N/mm^2)

σ = tensile or compressive stress (MPa)

τ = shear stress (MPa)

Moments

M = Generalized Bending Moment (N-mm)

Dimensions

L = length of compression handle

R = Length of tie arms (mm)

D = distance between bottom handle pivot and tie arm pivots (mm)

δ = vertical offset between center of piston and bottom handle pivot point (mm)

ΔX = displacement of piston (mm)

l.r.d. = generalized length, radius and diameters (mm)

Angles

θ = Angle between the handle lever and the horizontal (^{o})

ϕ = Angle between the tie arms and the piston (^{o})

$\theta_o = \theta$ at start of piston stroke

$\phi_o = \phi$ at start of piston stroke

Areas and Moments of Inertia

A = Cross-sectional area $(mm)^2$

I = Moment of Inertia = $\frac{bh^3}{12}$ for rectangular cross-sections $(mm)^4$

b = width

h = height $= \frac{\pi r^4}{4}$ for circular cross-sections $(mm)^4$

r = radius

<u>Coefficients of Friction</u> f = coefficient of friction

<u>Force of Displacement Analysis</u>

From Figure 2.1 we can sum the moments on the handle and find

$$PL = 2TR\sin(\theta-\phi)$$

also $F = 2T\cos\phi$ (Eq. 2.1)

so the theoretical mechanical advantage of the press is

$$\frac{F}{P} = \frac{L\downarrow\cos\phi}{D\uparrow\sin(\theta-\phi)\uparrow} \qquad \text{(Eq. 2.2)}$$

from the geometry:

$$R\sin\phi + \delta = D\sin\theta \qquad \text{(Eq. 2.3)}$$

so $\downarrow \phi = \sin^{-1}\left(\frac{D\sin\theta_o - \delta}{R\uparrow}\right)\downarrow$

the piston displacement:

$$\Delta X = R(\cos\phi_o - \cos\phi) - D(\cos\phi_o - \cos\phi) \qquad \text{(Eq. 2.4)}$$

where $\phi_o = \sin^{-1}\frac{(D\sin\phi_o - \delta)}{R}$

for a given press geometry one can now use a programmable calculator to solve for the generated force, F, if one knows the displacement, ΔX, the angle, ϕo, and the applied force, P.

<u>Pressure in the Cylinder</u> - r_c = radius of the cylinder

$$P = \frac{F}{\pi(r_c)^2} \qquad \text{(Eq. 2.5)}$$

<u>Stress Analysis</u>

<u>Stress in Tie Arms</u>

<u>Tensile Stress</u>

$$T = \frac{F}{2\cos\phi} \qquad \text{(Eq. 2.6)}$$

$$\sigma t - \frac{T}{(\pi r_2)} \qquad \text{(Eq. 2.7)}$$

r = tie arm radius

<u>Bending Stress</u>

This is caused by friction between the tie arm pin and the handle lever.

Total Frictional Force = 2 fT

M = Moment from the Friction on each tie arm = fTrp
rp = pin radius r = tie arm radius

$$\text{so} \quad \sigma b = \frac{Mrp}{I} = \frac{4fTrp}{\pi r^3} \qquad \text{(Eq. 2.8)}$$

<u>Total Stress</u>

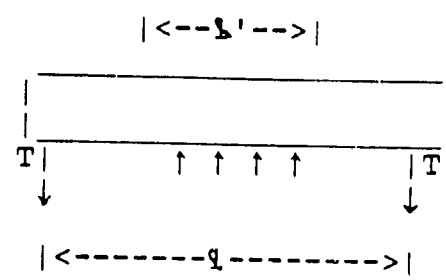
$$\sigma = \sigma t + \sigma b$$

<u>Stresses in Pins</u>

The pin with the highest bending stresses is the pin that passes through the piston.

<u>Bending Stresses</u>

$$M_{max} = \frac{T}{2}(\ell - \ell'/2)$$



$$\sigma = \frac{M(r)}{I} = \frac{2T(\ell - \ell'/2)}{\pi r^3} \qquad \text{(Eq. 2.9)}$$

The pin with the highest shearing stresses is the pin through the handle lever at the end of the tie arms.

<u>Shearing Stresses</u>

$$\tau = \frac{T}{\pi r^2} \qquad \text{(Eq. 2.10)}$$

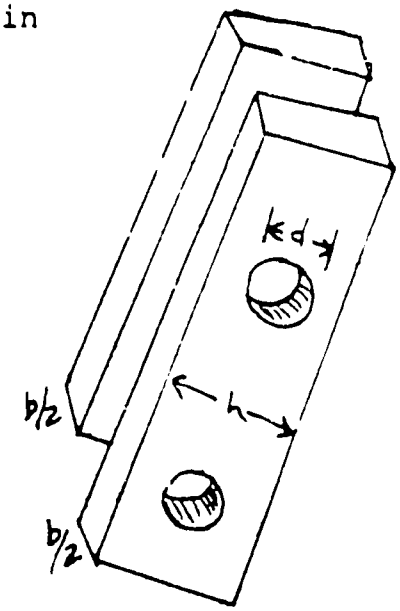
<u>Maximum Bearing Stresses</u> w = width of handle pin bashing

$$\sigma = \frac{T}{2rw} \qquad \text{(Eq. 2.11)}$$

Stresses in Lever Handle

The maximum stresses in the lever handle occur when the applied load P is in maximum.

Maximum Bending Moment is at the Pivot Arm Pin



$$M = P(L-R)$$

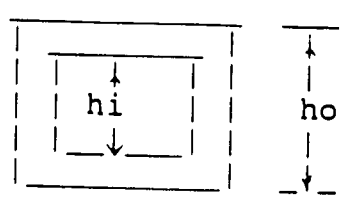
$$\sigma = \frac{Mc}{I} = \frac{6M}{bh^2} \qquad \text{(Eq. 2.12)}$$

In fact, because of the pivot pinhole, the stresses would be substantially higher than this value.

It is likely that the Maximum Bending Stresses will be at the bottom end of the square hollow section used for the lever handle.

$$M' = PL'$$

$$\sigma = \frac{Mc}{I}$$



where $I = \frac{ho^4 - hi^4}{12}$ and $c = \frac{ho}{2}$

The stress is: $\sigma = \frac{6PL'ho}{ho^4 - hi^4}$ (Eq. 2.13)

Stresses in Pressing Cage

As explained in Chapter 4, the exact distribution of pressures inside the ram press cage is not known, but we do know that the highest pressure is equal to the generated force divided by the piston's cross-sectional area, and that the highest pressure region is located at the small end of the cage. For the purpose of this analysis we shall take the pressure to be constant and equal to the highest value between the end of the cylinder and the first tension ring and we shall proceed as we did for the scissor jack presses in Appendix 1.

Bending Stresses in the Slats. From Appendix 1.

$$\sigma - \frac{pl^2}{2\,h^2} \qquad \text{(eq. 2.14)}$$

l = separation between cylinder and first tension ring
h = thickness of the slats
p = pressure in the cage

$$l_{max} \sqrt{\frac{2\ \sigma\ allow\ h^2}{P}} \qquad \text{(eq. 2.15)}$$

is the maximum allowable separation between tension rings

<u>Hoop Stress in Tension Rings</u>

$$\sigma = \frac{prl}{wt} \qquad \text{(eq. 2.16)}$$

r = average cage radius
t = thickness of the rings
w = width of the rings

$$(wt)_{min} = \frac{prl}{\sigma\ allow} \qquad \text{(eq. 2.17)}$$

gives the minimum allowable cross-sectional area of the tension rings.

APPENDIX 2

FORCE AND STRESS ANALYSIS FOR RAM PRESS

DEFINITIONS

Forces

P = force applied by operator on end of handle (Newtons)

F = generated force on seed (Newtons)

T = Tension in tie arms (Newtons)

Pressure and Stresses

P = pressure in seed cage (MPa) (N/mm)

= tensile or compressive stress (MPa)

= shear stress (MPa)

Moments

M = Generated Bending Moment (N-mm)

Dimensions

L = length of compression handle (mm)

R = length of tie arms (mm)

The Piston Displacement

where

(Eq. 2.4)

For a given press geometry, if one knows the displacement, the angle, O, and the applied force, P, one can now use a programmable calculator to solve for the generated force, F, the Pressure at the end of the Piston.

p = F (pr = piston radius)

(Eq. 2.5)

STRESS ANALYSIS

Stress in Tie Arms

Load: r = tie arm radius (Eq. 2.6)

Tensile Stress

(Eq. 2.7)

Bending Stress:

This is caused by friction between the tie arm pin and the handle lever.

Total Frictional Force = 2fT

M = Moment from the Friction on each tie arm = fTrp

r = pin radius (Eq. 2.8)

Total Stress:

Stresses in Pins

The pin with the highest bending stresses is the pin that passes through the piston.

Bending Stress:

The pin with the highest shearing stresses and highest bearing stresses is the tie arm pin.

Shearing Stresses:

(Eq. 2.10)

Bearing Stresses:

(Eq. 2.11)

Stresses in Lever Handle

The maximum stresses in the lever handle occur when the applied load P is a maximum

Maximum Bending Moment is at the Pivot Arm Pin:

(Eq. 2.12)

Bending Stress:

In fact, because of the pivot pinhole, the stresses would be substantially higher than this value.

It is likely that the Maximum Bending Stresses will be at the bottom end of the square hollow section used for the lever handle.

Bending Moment:

Moment of Inertia:

Bending Stress:

(Eq. 2.13)

Stresses in Pressing Cage

As explained in Chapter 4, the exact distribution of pressures inside the ram press cage is not known, but we do know that the highest pressure is equal to the generated force divided by the piston's cross-sectional area, and that the highest pressure region is located at the small end of the cage. For the purpose of this analysis we shall take the pressure to be constant and equal to its highest value between the first two tension rings and Appendix 1.

Bending Stresses in the Slats. (from Appendix 1)

(Eq. 2.14)

l = separation between tension rings
h = thickness of the slats
p = pressure in the cage

Maximum Allowable Separation Between Tension Rings:

(Eq. 2.15)

Hoop Stress in Tension Rings

(Eq. 2.16)

r = average cage radius
t = thickness of the rings
w = width of the rings

Minimum Allowable Ring Cross-sectional Area:

(Eq. 2.17)