RESEARCH AND DESIGN OF 5.4-INCH DIAMETER MOLD ROTARY PRESS FOR COAL LOG AND BIOMASS COMPACTION

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by

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The undersigned, appointed by the Dean of the Graduate Faculty, have examined the thesis entitled

RESEARCH AND DESIGN OF 5.4-INCH DIAMETER MOLD ROTARY PRESS FOR COAL LOG AND BIOMASS COMPACTION

presented by **Kang Xue**

a candidate for the degree of Master of Science

and hereby certify that in their opinion it is worthy of acceptance.

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ABSTRACT

The research and development (R&D) of coal log compaction machines is very important for commercializing the new technology of coal log pipeline (CLP). The CLP technology will be successful commercially only if the machine can produce high-quality, low-cost coal logs at a high production rate.

In this thesis, a commercial scale rotary press for compacting coal logs and biomass logs is designed. This machine is not a small, conventional tablet rotary press used by the pharmaceutical industry. It is one of the largest rotary presses in the world. This machine provides a maximum of 250-ton compaction force, and can produce coal logs of 5.4 inches in diameter and 10 inches in length. The designed production rate is 3,600 logs per hour or one log per second. Because of the large friction force and the long stroke, the design of such a machine is very challenging. In order to satisfy all design requirements and overcome technical problems, the critical components such as the cam, the punches, the turret, the feeding system, and the power train are carefully analyzed and designed. The design also incorporates special features learned from coal log compaction studies such as the backpressure during ejection of logs from mold. The cost estimation for this design is also conducted. As a result of the preliminary design, a set of assemblies, subassemblies, and main part drawings is provided with this thesis.

TABLE OF CONTENTS

Page

LIST OF TABLES

LIST OF FIGURES

Page

Page

Chapter 1

INTRODUCTION

The coal log pipeline (CLP) technology was invented by Drs. Henry Liu and Thomas R. Marrero in 1990 at University of Missouri-Columbia. CLP involves compacting coal into cylindrical forms (logs), and transporting the coal logs through an under-ground pipeline from a coal mine to one or more power plants or seaports, by using water as the carrier fluid. This technology requires innovative compaction machines to produce strong coal logs. Since no such machines are commercially available, research and development on coal log fabrication machines is an important part of the CLP technology development.

After eight years of intensive research and development, the CLP technology is approaching commercialization. A CLP pilot plant is under construction at the Field Station of the Capsule Pipeline Research Center, University of Missouri-Columbia (Figs. 1.1 and 1.2). Based on the success of the research and development work on CLP with 1.9-inch-diameter coal logs, the first coal log fabrication machine (Fig 1.3) for producing 5.4-inch-diameter coal logs was designed. This machine was a 250-ton hydraulic press (Lin *et al.*, 1996; Lin *et al.*, 1997a, Lin *et al.*, 1997b). The machine was constructed by Gundlach Company in 1997, and installed in the Coal Log Pilot Plant (Fig. 1.4). This coal log fabrication press has two hydraulic cylinders; each can generate 508,000 pound force with a maximum piston speed of 15 inches/sec. The maximum production rate is 180 coal logs/hour (or 3 logs/min). It is automatically controlled by a programmable logic control (PLC) with manual override. The machine can be controlled to undertake various fabrication procedures, such as different compaction and ejection speeds, and a wide range of compaction forces (up to 250 ton). This machine can also handle different materials in making 5.4-inch diameter logs. It has been used to study the scale-up of the CLP technology, and to conduct the research and development work of compacting biomass materials such as saw dust and waste paper. Important technical information for designing the next generation compaction machine was collected using this machine.

After the successful field tests of the CLP technology in a 5-mile commercial underground pipeline in Conway, Kansas (Liu, 1997), a second generation coal log and biomass compaction machine was conceived based on the rotary press concept.

The advantage of using rotary press concept for coal log compaction machine is to increase the output rate without reducing the compaction time for each coal log. Rotary press does not use hydraulic or pneumatic forces; it is based entirely on cam and follower mechanism. The conceptual design of a 30-ton rotary press for fast compaction of 1.9-inch coal logs was conducted (Xue, 1995a). The main reason that commercial rotary presses cannot be used for coal log compaction is that the required height of the coal logs is much larger than their diameters. The major differences between a commercial tablet rotary press (e.g., Gladiator Rotary Press) and the 1.9-inch coal log rotary press are listed in Table 1.1.

Due to the considerable increase in the maximum fill length, the dies of the rotary press for coal logs are significantly longer than those for making medicine tablets. This

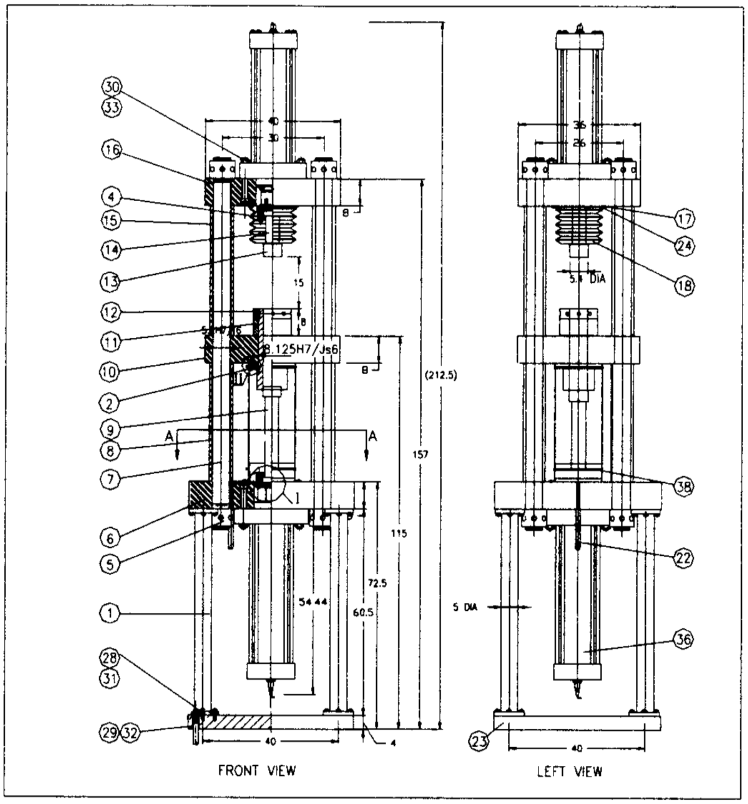


Figure 1.3 Structure of 250-Ton Hydraulic Press

causes the height and diameter of the turret to increase accordingly to satisfy the requirement of the compaction profile, and to produce strong compact logs. The basic structure of the 30-ton coal log rotary press is shown in Fig. 1.5.

The middle section of the machine is a rotating turret, where the compaction molds are located. The turret is driven by a large worm gear. The optimal design of the worm gears for this rotary press was studied using Design Optimization Tools software (Lin *et al.*, 1997).

Based on the experience gained from designing two compaction machines and testing data of making 5.4-inch coal logs using the 250-hydraulic coal log press, a large rotary press can be design and constructed for commercial coal log pipelines. This

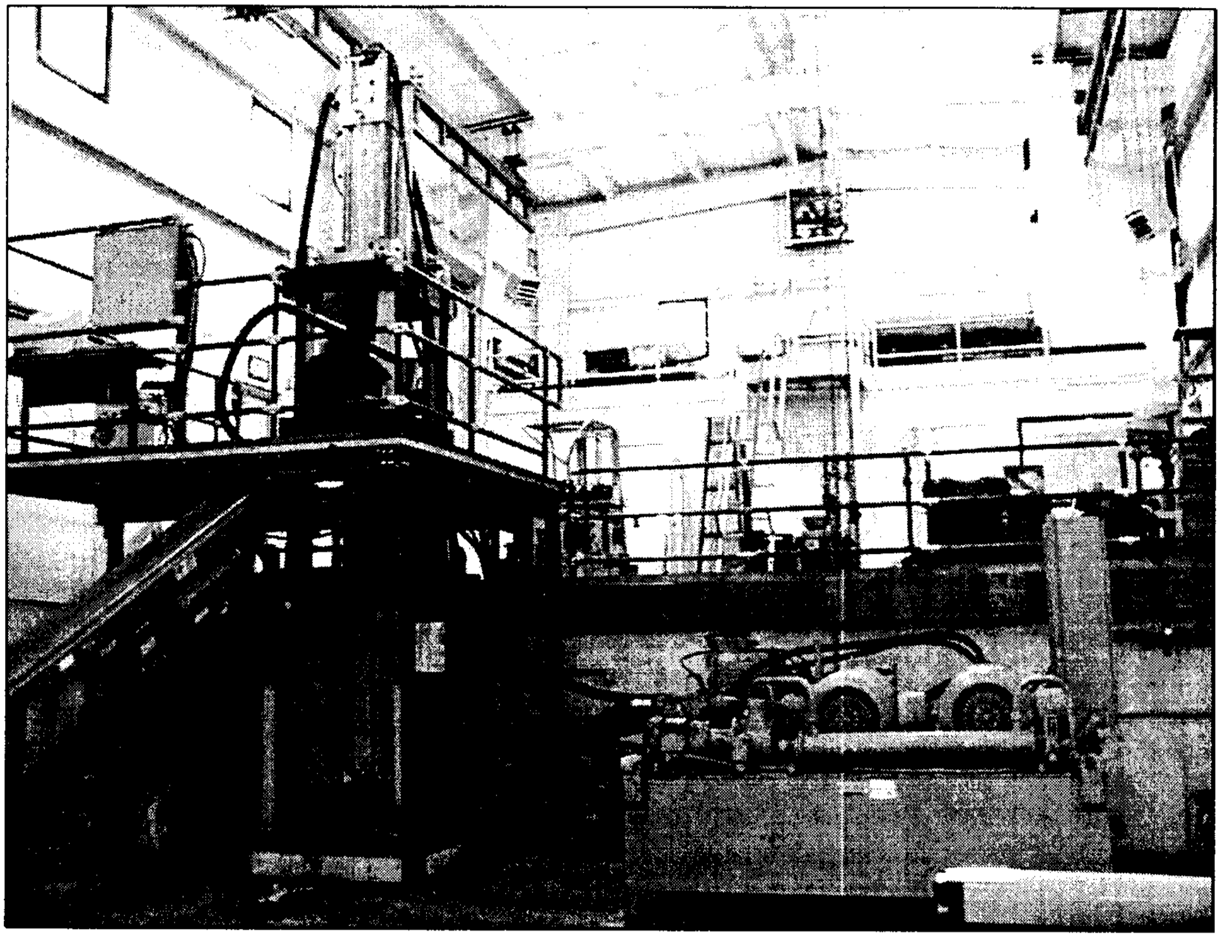


Figure 1.4 250-Ton Hydraulic Press Inside Field Station of CPRC

machine can also be used to compact and de-water other loose or powdered materials into logs, so that they can be handled, transported and utilized more efficiently. The objective of this thesis is to design a rotary press for fabricating 5.4"-diameter coal logs. The technical specification of this rotary press is given in Table 1.1. The concepts, calculations, and drawings for the design of the key components are included in this thesis.

Table 1.1 Comparative Specifications of Tablet and Coal Log Rotary Presses

Specifications	Vector Gladiator	Coal Log Rotary Presses	
		1.9" log	5.4" log
die stations	15	30	30
maximum output rate (tablet, log/min)	255	60	60
maximum turret speed (rpm)	17	2	2
tablet/log diameter (in)	3	1.9	5.4
maximum fill depth (in)	3	6.84	19.44
maximum force (lbs.)	80,000	80,505	458,044
Motor drive with variable speeds (H.P.)	25	30	584
Length of upper punch (inch)	15.875	19	44
Length of lower punch (inch)	20.875	20	55.5
Length of die (inch)	3.25	7.6	21

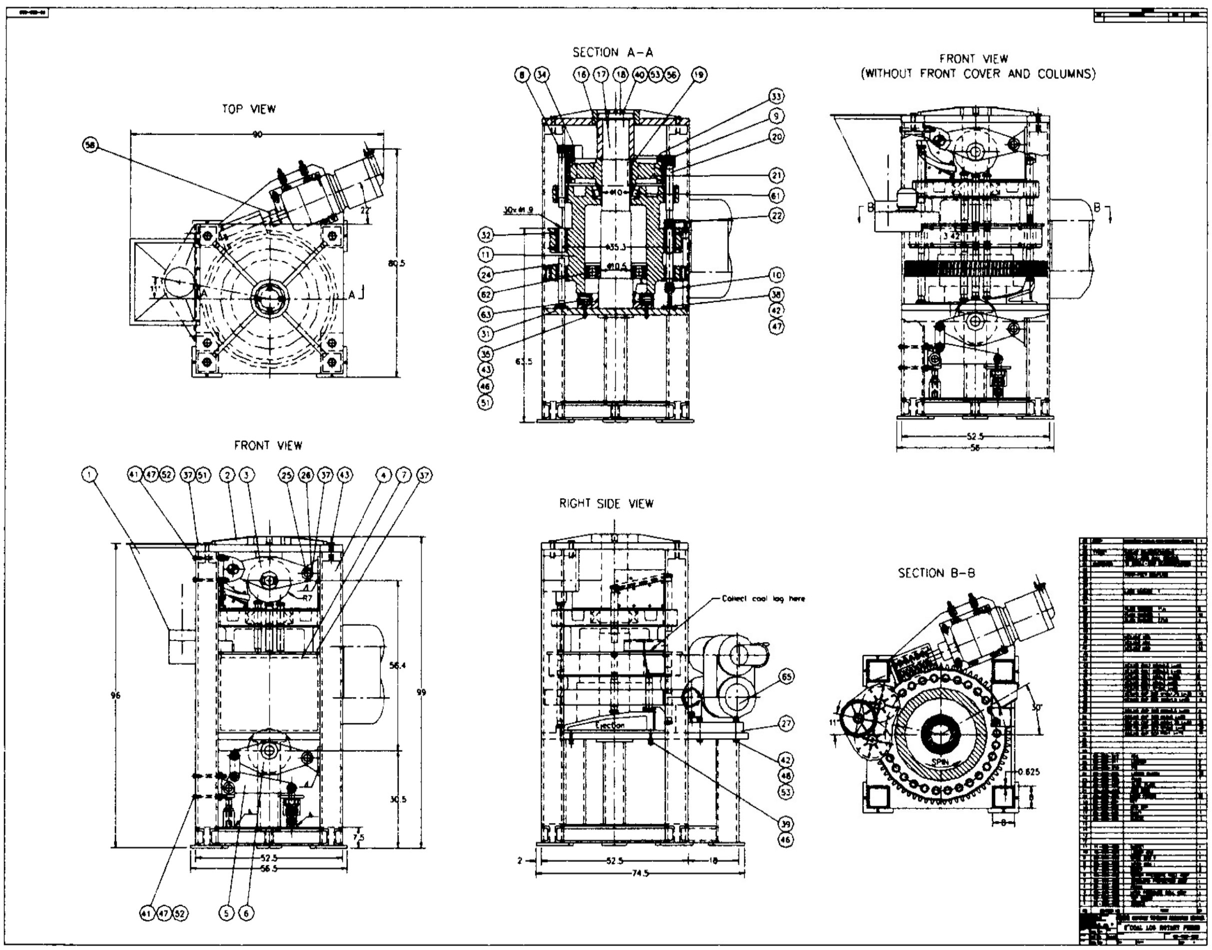


Figure 1.5 Conceptual Design of 30-Ton Rotary Press for CPRC

Chapter 2

LITERATURE REVIEW

Rotary presses are often used in medicine tablet production and in machine part fabrication of powder metallurgy industry, where high output rates are required. Conventional rotary presses for pharmaceutical industry can meet the requirement of a fast production rate up to 222,000 tablets per hour (e.g. MANESTY NOVAPRESS 2). The rotary machine achieves high production rate by using many punches and die sets, fitted around the periphery of a rotating table, which is also called turret (Fig. 2.1). The dies are filled at one or multi-locations on the machine as the turret rotates. The punch heads are brought in turn between pairs of stationary rollers, where the punch heads are forced to move towards each other, and exert a compressive force on the materials in the dies.

The action of the rotary presses is different from that of single-punch compaction machines in several ways. In a rotary press, both punches (pistons) move into the die (mold) together as the rollers move on the cams. In contrast, in a single punch compaction machine, there is only one punch (piston) moving to compact the powders. Figure 2.2 shows a single-sided rotary press which has one fill station and one pair of compression rolls to produce one compact per mold in each revolution of the turret. By having two fill stations, double-sided rotary presses (Fig. 2.3) can produce two compacts per mold per revolution. Figure 2.4 presents the typical operating sequence of a rotary

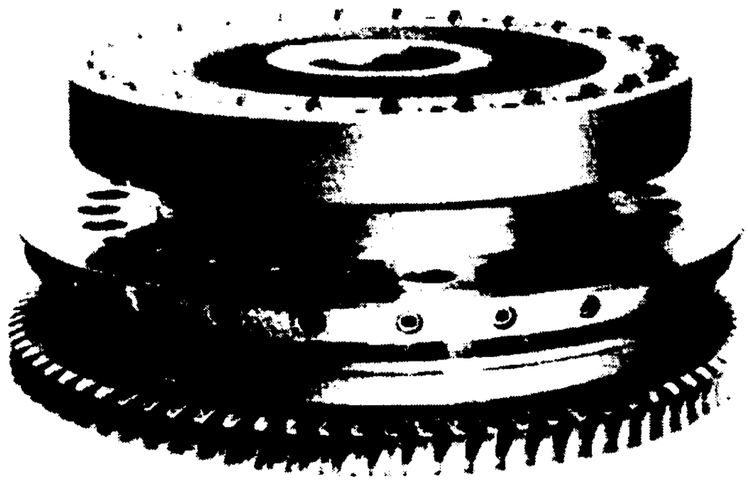


Figure 2.1 Turret (Powder Metallurgy Equipment Manual, 1977)

press. The major components of a typical rotary press are shown in Fig. 2.5. Basically, they consist of a feeding system, a rotating turret with upper and lower punches, an upper and a lower cam systems which regulate the motions of the punches, and the upper and lower compression rollers. Figure 2.6 shows the overload protection mechanism of a rotary press.

During compacting operation, the material filled into the mold is compressed by the two punches passing between two compression rollers, which are opposite in vertical direction. This produces the effect of double-ended compaction, which is beneficial to the quality of the compacts. Some high-output tablet rotary presses have two pairs of compression rolls. All these machines had odd numbers of dies to avoid two pairs of rolls reaching the maximum force at the same time. The largest presses were reported to have up to 79 dies mounted on one turret (Marshall and Rudnic, 1990). The working procedure for all rotary press consists of the following steps (see Fig. 2.4):

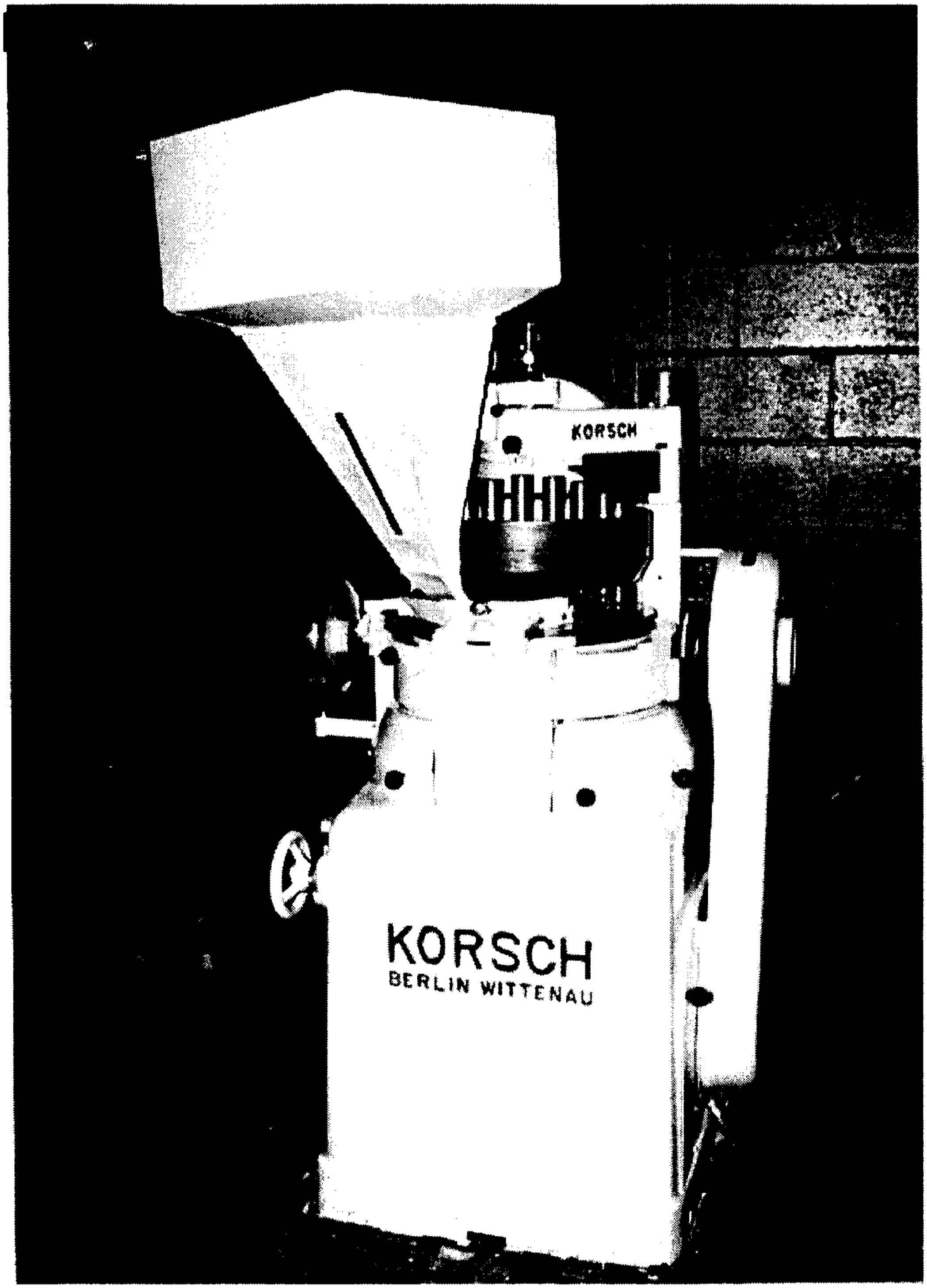


Figure 2.2 Single-sided Rotary Press (Snyder, 1967)

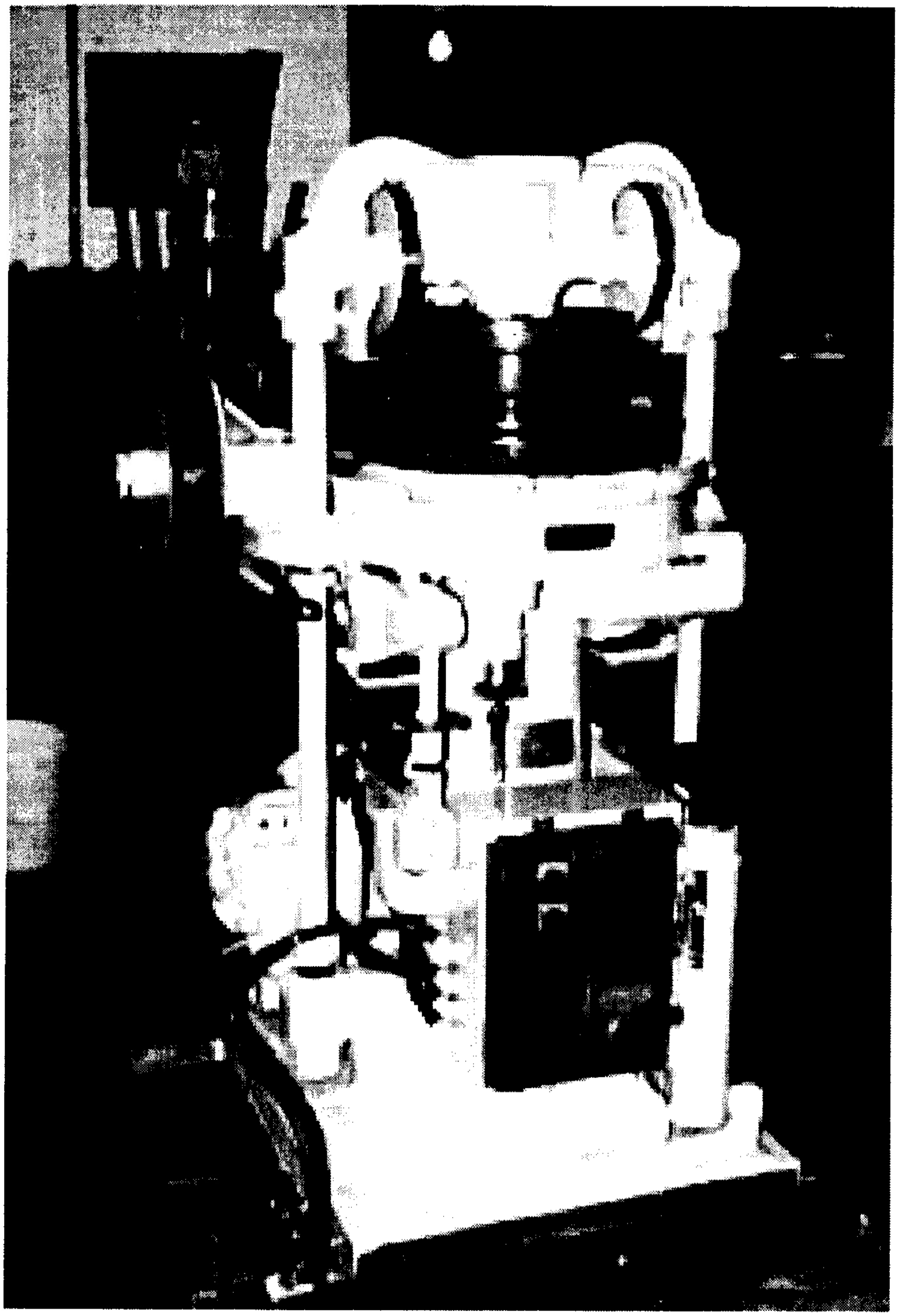


Figure 2.3 Double-sided Rotary Press Turret (Snyder, 1967)

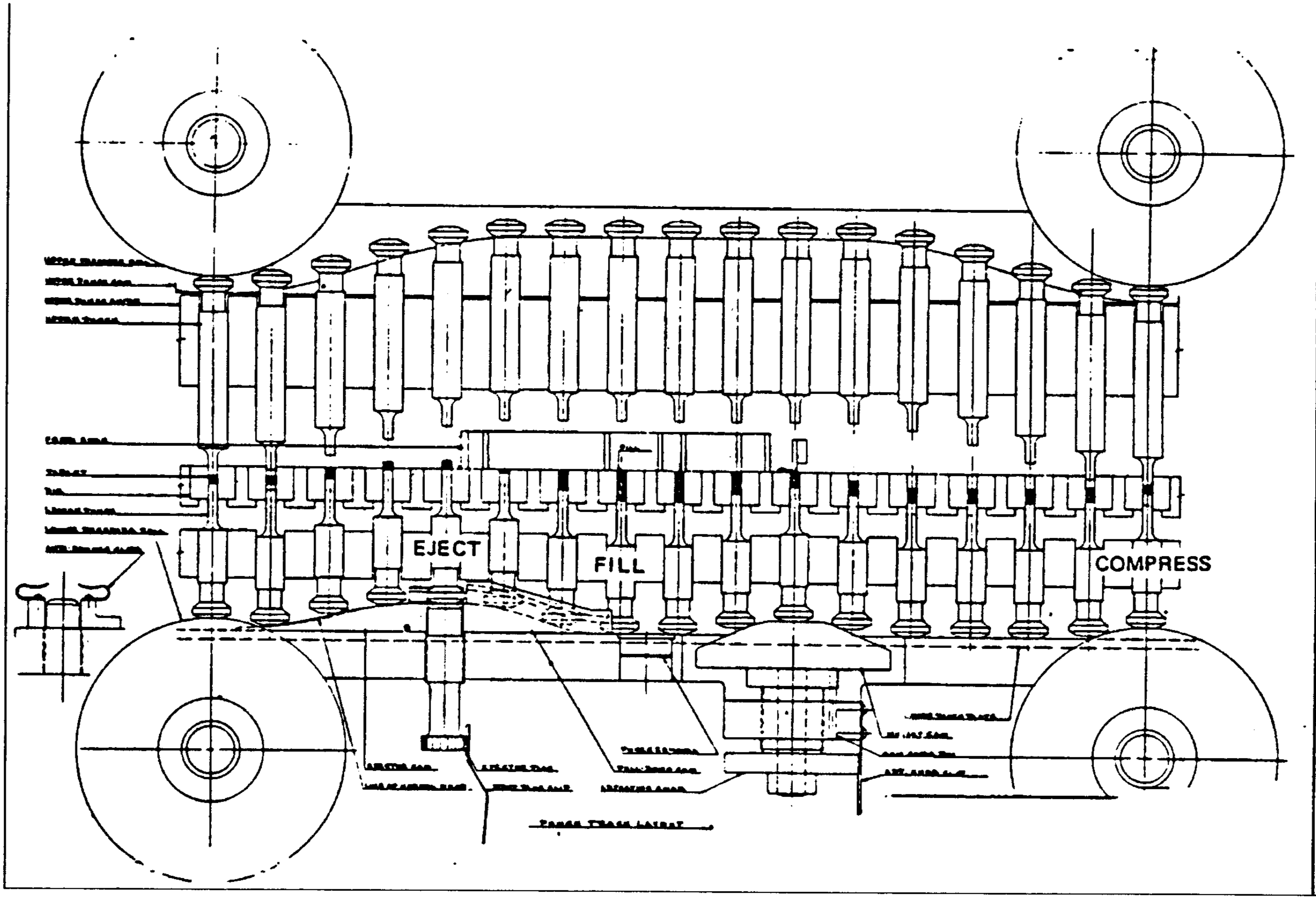


Figure 2.4 Typical Operating Sequence of a Rotary Press (Pietsch, 1991)

(1) Feeding material into the mold

(2) Removing excess material

(3) Compacting the material in the mold cavity

(4) Ejecting the compact from the mold

(5) Transferring the compact to the collecting chute

A tablet rotary press manufactured by Cherry-Burrell Corporation (Model No. 270-18) can make tablets of 2"-diameter with 40,000 lbs of the maximum force (Fig. 2.7). This model has 18 die stations and uses a 7.5 HP motor. The power is delivered to the main drive through bevel gears. The output rate is 325 tablets/min, which gives a rotating speed of 18 rpm (Cherry-Burrell Corporation, 1969). The Gladiator Rotary Compacting Press manufactured by Vector Corporation (Marion, Iowa) can make tablets of 3"-diameter with 80,000 lbs. of maximum force. This model has 15 die stations and a 25 HP motor. The maximum output rate is 255 tablets/min, which indicates a rotating speed of 17 rpm.

A review by Cooper and Rees (1972) pointed out that the technology of tablet formulation and production is ultimately limited by the compression machinery available. The basic principle of tablet machinery has been changed very little for decades. Some improvements in the design of rotary tablet press have been reported, but none of them involve new basic concept.

Swartz (1969) reviewed the specifications of the newer high speed rotary presses available in the United States. Problems resulting from the reduced dwell time of the granulation or powder in the dies were discussed; they involve forced feeding and pre-compression. Maekawa *et al.* (1971) described a modified rotary machine in which the

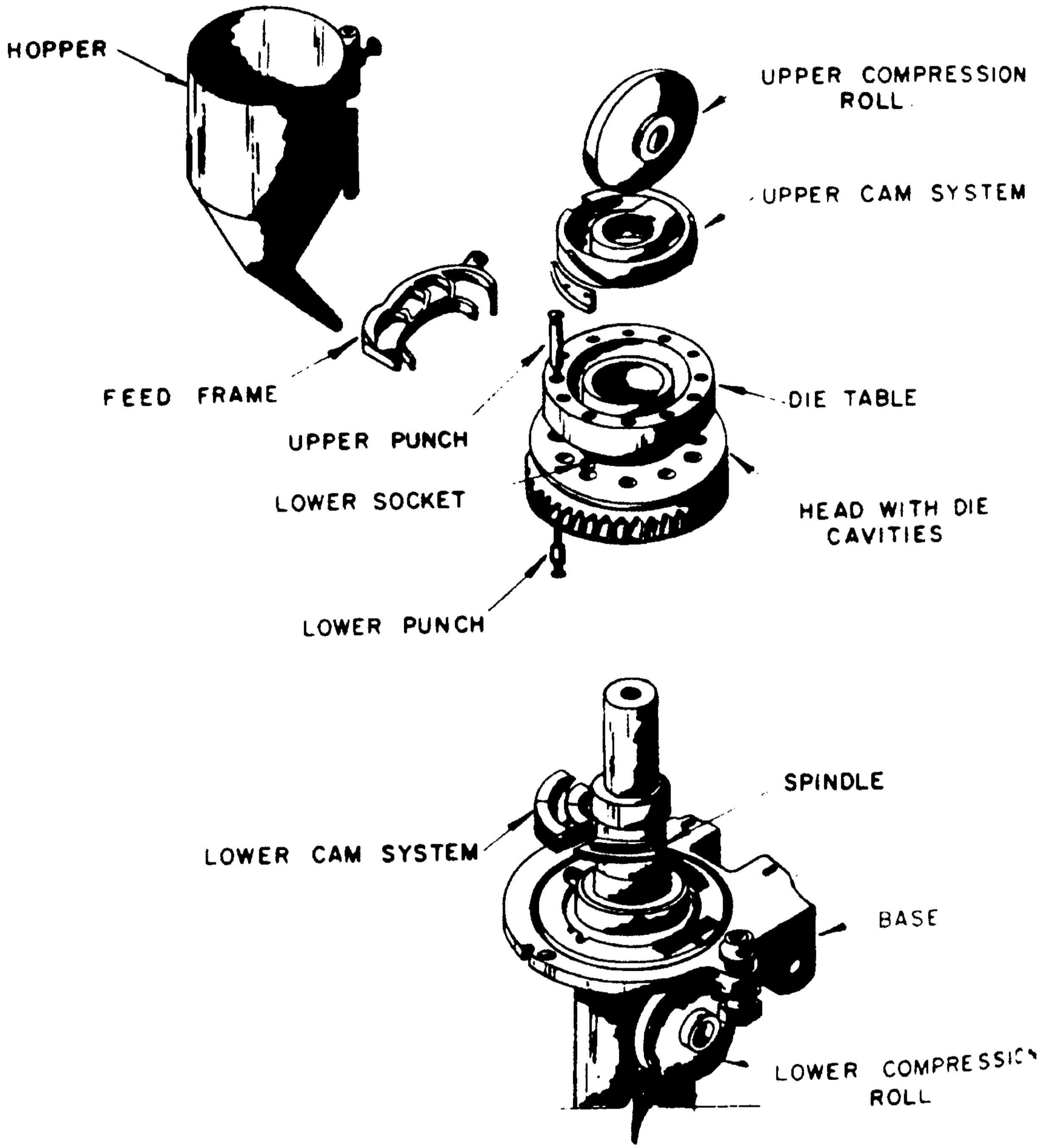


Figure 2.5 Main Components of A Rotary Press (Snyder, 1967)

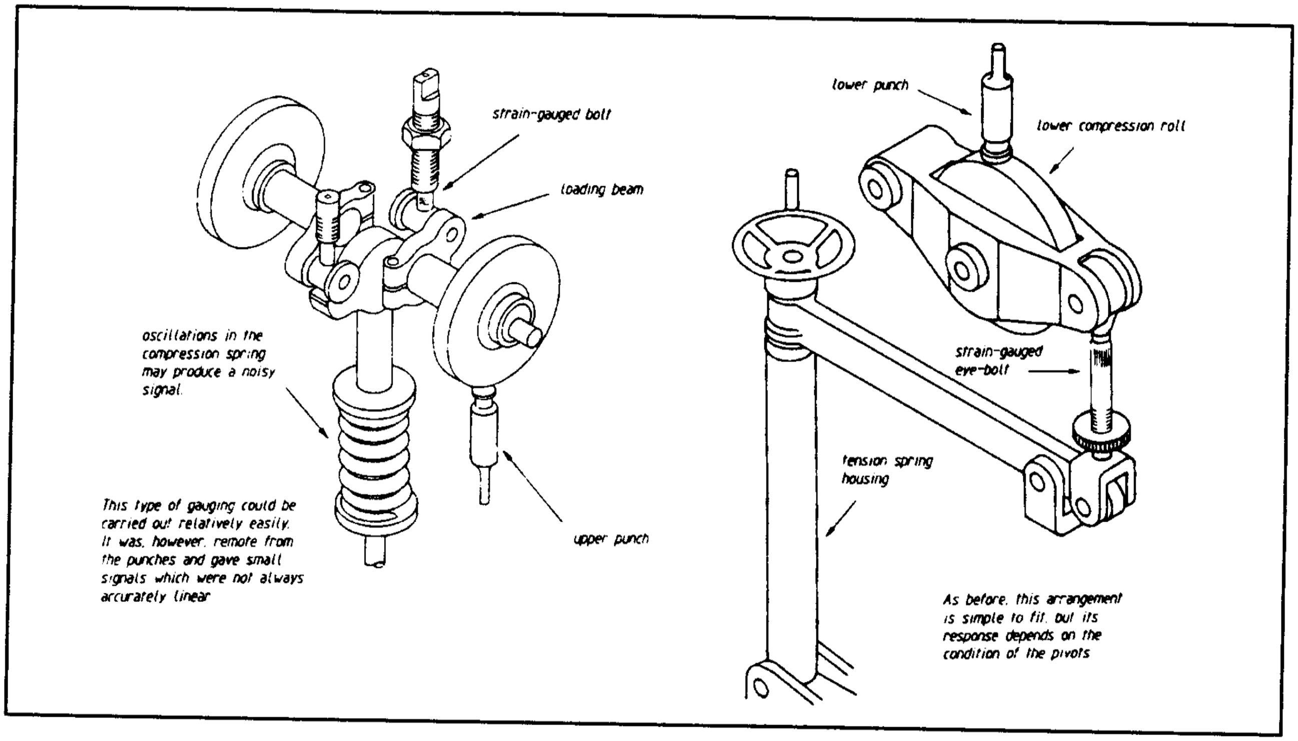


Figure 2.6 Overload Protection Mechanism of A Tablet Press (Ridgeway-Watt, 1988)

compression rollers are mounted obliquely, and not perpendicular to the die table. It was claimed that this arrangement reduces the rate of compaction, increases the dwell time, and minimizes the wear of the punches and compression rollers.

Rotary compacting presses used for powder metallurgy can be furnished commercially in tonnage ranges from tens to hundreds of tons. The tonnage capacity is calculated from the following equation (Powder Metallurgy Equipment Association, 1977):

$$T = P A \tag{2.1}$$

where T is the press load in tons, P is the maximum compaction pressure in psi, and A is the projected area of one punch. Therefore, this formula dose not indicate the total axial load of a rotary press, but the load resulting from compaction of one die station.

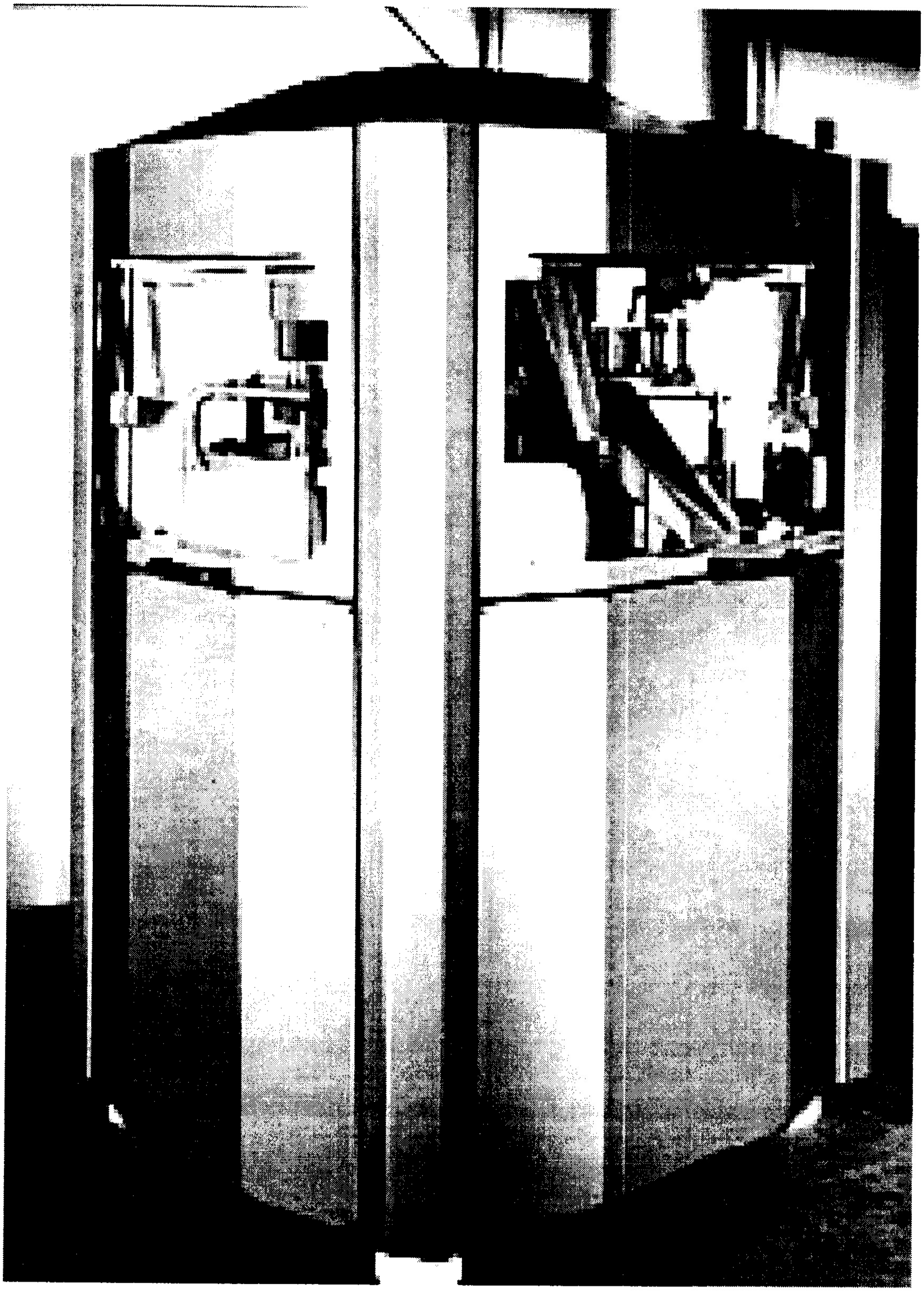


Figure 2.7 A Large Diameter Tablet Press (http://www.moindustriees.com)

The first Stokes rotary press for powder metal was developed more than 70 years ago. It produced about 200 copper-graphite brushes per minute, to meet the need of a major carbon manufacturer for high volume production of motor starters. Early models were 15- to 20-ton machines, but by the late 1930s, 30- to 100-ton presses were in use (Snyder, 1967). Powder metallurgy rotary presses having 6 in-diameter dies were mentioned in the literature (Powder Metallurgy Equipment Manual, 1977). Fill depths were reported to range from 11/16 inch to 5 inches.

Die and punch designs are important to any rotary press design. The design of a multi-spring floating die compaction system to simulate double-ended compaction for plastic powders was reported by Crawford *et al.* (1989). This design used three linear levels of springs to model the load-deformation characteristics of the given material and provided an economic way of compacting powders using a single hydraulic ram but with the benefits of the double-ended compaction.

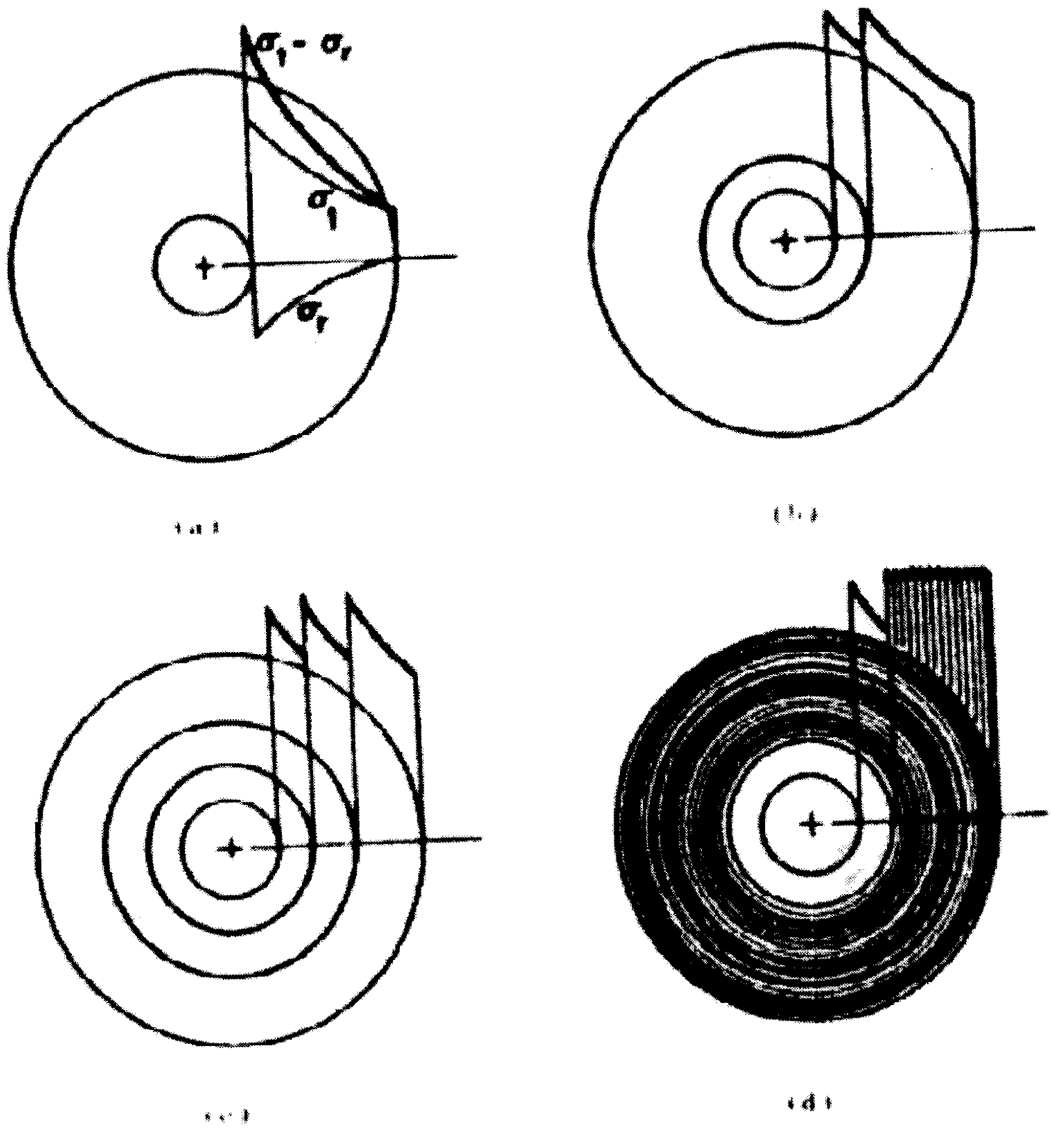
Shrink rings, or stress rings, are commonly used in the fabrication of powder compaction dies to increase the allowable compaction pressures for a given die material, so that the required outer dimensions of the die assemblies can be reduced. According to Kuhn (1978), a die can be considered as a thick-walled cylinder under internal pressure. The pressure generates radial and tangential (hoop) stresses. The resulting stress is the greatest at the inside surface of the die and diminishes toward the outer surface. The use of a stress ring (Fig. 2.8b) shrunk fit around the die puts the die under a radial compression, which increases material utilization by reducing the stress differences. Stress ring materials usually have higher ductility and toughness than the die insert materials. The same principle can be extended to double stress rings (Fig. 2.8c), and/or to multi-stress rings, as in a ribbon-wound die design (Fig. 2.8d). The optimization procedures of stress ring die design for powder compaction were discussed in detail by Kuhn (1978).

Theoretical estimation of punch velocities and displacement of single-punch and rotary tablet machines was reported by Charlton and Newton (1984). Their analysis can be applied to machines with different dimensions and operating speeds, but it requires modifications if the punch head design on the rotary machine was different significantly.

A later study by Armstrong and Palfrey (1987) discussed the punch velocities during compaction process using an eccentric tablet press. In their study, the actual velocities of the upper punch were compared with those predicted by equations describing the movement of the punch in an empty die. Up to the maximum punch movement, the actual speeds during compaction were invariably less than the predicted ones. As the punch withdraws from the die, its speed can exceed that predicted from the theory, due to the elastic expansion of the tablet assisting punch ejection.

A study of tableting tool life records, accumulated over a period of 8 years, was published by Swartz and Anschel (1968). The purpose of this study was to develop and define the relationships between tool performance and the operating variables that exist in routine tablet production. One conclusion important to rotary press design is that small working tolerances for tools certainly helped to extend the life of the tools.

Based on the above review and the specifications for the coal-log rotary press machine, it seems possible to use the basic principles similar to the Vector Gladiator rotary press machine to design a rotary press for making 5.4"(diameter) × 10" (length) coal and biomass compacts.



(a) one-piece die; (b) die insert with one stress ring; (c) die insert with two stress rings; (d) ribbon-wound or wire wound die insert.

Figure 2.8 Cross sectional view of die loading and stresses (Kuhn, 1978)

Chapter 3

DESIGN ANALYSES AND CALCULATION

In machine design, major design parameters must be analyzed and determined first. Then, these parameters will be used to determine the size and material of major components of the machine. This general approach is also used in the design of the 5.4" rotary press.

3.1 Cam Profile Design

In general, cams are divided into uniform motion cams and accelerated motion cams. The uniform motion cam moves the follower at a constant speed from the beginning to the end of the stroke. The type of cams is suitable for moderate-speed machinery. Its major disadvantage is the sudden changes in acceleration at the beginning and the end of the stroke. A cycloidal motion cam curve produces no abrupt changes in acceleration and is often used in high-speed machinery because of the low noise, vibration and wear (Green *et al.*, 1996).

3.1.1 Displacement Diagrams

Cam design begins with the displacement diagram. There are two cam-and-follower systems for the 250-ton rotary press: one is above the molds and the other is below the molds. The cams are stationary, and during the turret rotation, the punches slide up and down along the cam surfaces to perform the compaction, maximum pressure dwell, ejection, return, back pressure, and material filling dwell functions. In order to make the punches move according to the needed displacement and speed, two simple displacement diagrams were determined (Fig. 3.1).

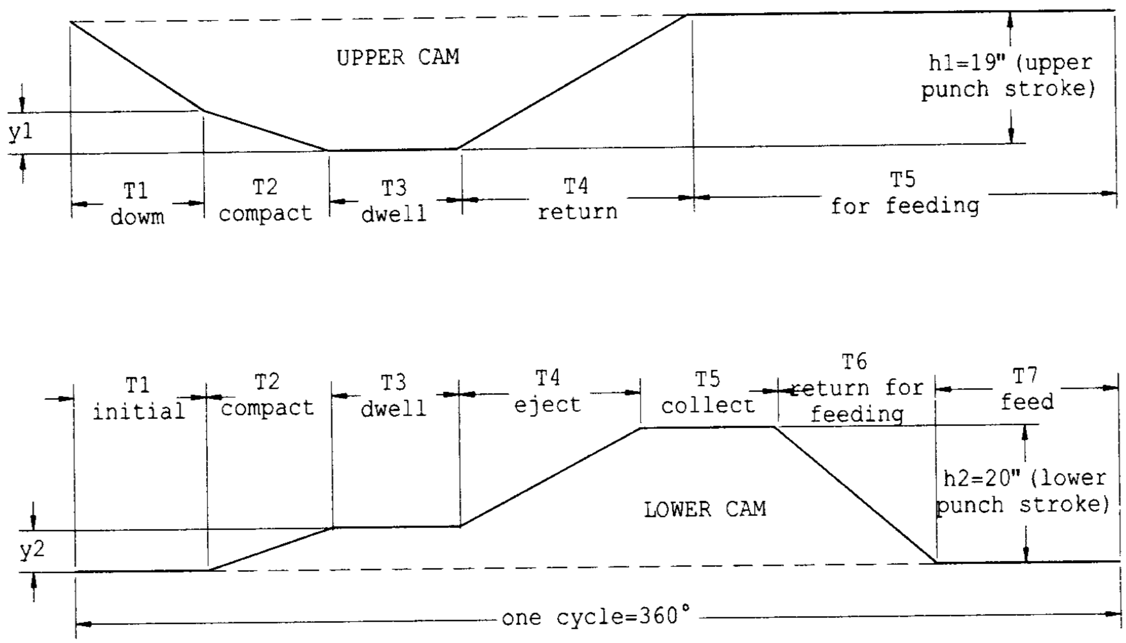


Figure 3.1 Simple Displacement vs Time Diagrams of Cams Designed for the 5.4" Rotary Press

One cycle corresponds to the follower sliding or rolling through a whole revolution of the cam. The horizontal distances T_1, T_2, T_3, T_4 T_5, T_6, and T_7 are expressed in units of time (e.g. seconds). The vertical distance, h, represents the maximum stroke of the follower. In our case, the follower is the punch (piston). To analyze the action of a cam it is necessary to study its displacement, velocity, and acceleration as a time function. The velocity and acceleration are based on the first and second time-derivatives of the equation describing the time-displacement curve:

$$y = h(t) \tag{3.1}$$

$$v = \frac{dh(t)}{dt} \tag{3.2}$$

$$a = \frac{dv}{dt} \tag{3.3}$$

Calculation of the Filling Time

In order to determine the time-displacement curve of the punch, the time required to fill the mold (i.e., filling time) needs to be analyzed first. The flow rate of the powder through an orifice in the bottom of a storage container is reported to be (Orr, 1966):

$$w = \frac{\pi}{4} C \cdot \rho \cdot \left[\frac{g \cdot D^5}{2 \cdot \tan \beta} \right]^{0.5} \tag{3.4}$$

where w is the mass flow rate (kg/s); D is the orifice diameter (m); β is the angle of repose of the material; ρ is the density of the powder; and C is the discharge coefficient, which usually varies between 0.5 and 0.7 for large bunkers, and g is the gravitational acceleration (9.815 m/s^2). This equation is applicable only if the ratio of the particle diameter to the orifice diameter is less than 0.1.

Based on the above equation, the equation for the filling time can be derived as:

$$t_f = \frac{\pi \cdot D^2 \cdot H \cdot \rho}{4 \cdot w} = \left[\frac{2 \tan \beta}{g \cdot D} \right]^{0.5} \cdot \frac{H}{C} \tag{3.5}$$

where t_f is the filling time (sec), and H is the height of the loaded coal.

The angle of repose and the discharge coefficient were measured for Mettiki coal

(Xue, 1995b). Raw coal was screened using a 30 mesh sieve. The angle of repose was measured with 200 g coal or coal mixture at three different moisture concentrations. Three measurements were made for each sample mixture. The results are given in Table 3.1. The discharge coefficient C was estimated for a 1.75" diameter mold. The time needed to fill this mold is about 6 seconds. Using the largest angle of repose obtained, C is determined to be about 0.1. Using the angle of repose $\beta = 47°$, the discharge coefficient $C = 0.1$, the mold diameter $D = 5.4$ inches, and the height of the mold $H = 20.0$ inches, the minimum filling time required is $t_f = 6.4$ sec approximately.

Table 3.1 The Angle of repose for Mettiki coal

	Mattiki Coal	Mettiki +H_2O	Mettiki + H_2O +3% Binder
β (°)	38	45	47
Moisture	5%	7.6%	7.2%

Compaction Time Determination

By analyzing the laboratory data collected from the 250-ton hydraulic press during coal log compaction, we can estimate a suitable compaction time for this new design.

According to Fig. 3.2, which shows the experimentally collected data of compaction cylinder pressure as a function of piston displacement, the time used for moving the top piston from the initial position to the top surface of the mold was about 1 second. The piston speed during this period was about 15.7 in/s. The ejection speed of the bottom piston was about 4.13 in/s, and the ejection time was about 4.12 seconds. Before ejection, the top piston moved upwards 1 second earlier than the bottom piston. The whole compaction and ejection cycle was approximately 20 seconds (Fig 3.2), which was the time counted from the top piston starting to move downward to the bottom piston finishing ejection. Therefore, the compacting time of the 250-ton hydraulic press was 20 – 1 – 4.12 – 1 =13.88 seconds. In the design of the rotary press, the time distribution on the driving cams is limited by the pressure angle and the material feeding section requirement. Therefore, the time intervals for the upper cam are determined (Fig. 3.1) as follow:

T_1 is the time for the upper punch to move from the initial position down to the top of the mold = 2.266 sec.

T_2 is the compaction time = 6.734 sec

T_3 is the dwell time = 3 sec

T_4 is the time for the upper punch to go back to the initial position = 8 sec.

T_5 is the time for feeding = 10 sec

The total stroke of the upper cam is

$$H_1 = 19 \text{ in.}$$

The compaction stroke is

$$y_1 = 5 \text{ in.}$$

For the lower cam, the time divided in one cycle is shown as follow:

T_1 is the time for the upper punch to move from the initial position down to the top of the mold = 2.266 sec.

T_2 is the compaction time = 6.734 sec.

T_3 is the dwell time = 3 sec.

T_4 is the ejection time of the lower punch = 6 sec.

T_5 is the time for log collection = 3.5 sec

T_6 is the time for the lower punch to go back and the feeding starts = 5 sec

T_7 is the time for feeding = 3 sec

The sum of T_6 and T_7 is 8 seconds > t_f = 6.4 seconds (the required minimum filling time). The total stroke of the lower cam is

$$H_2 = 20 \text{ in.}$$

The compaction stroke is

$$y_2 = 5 \text{ in.}$$

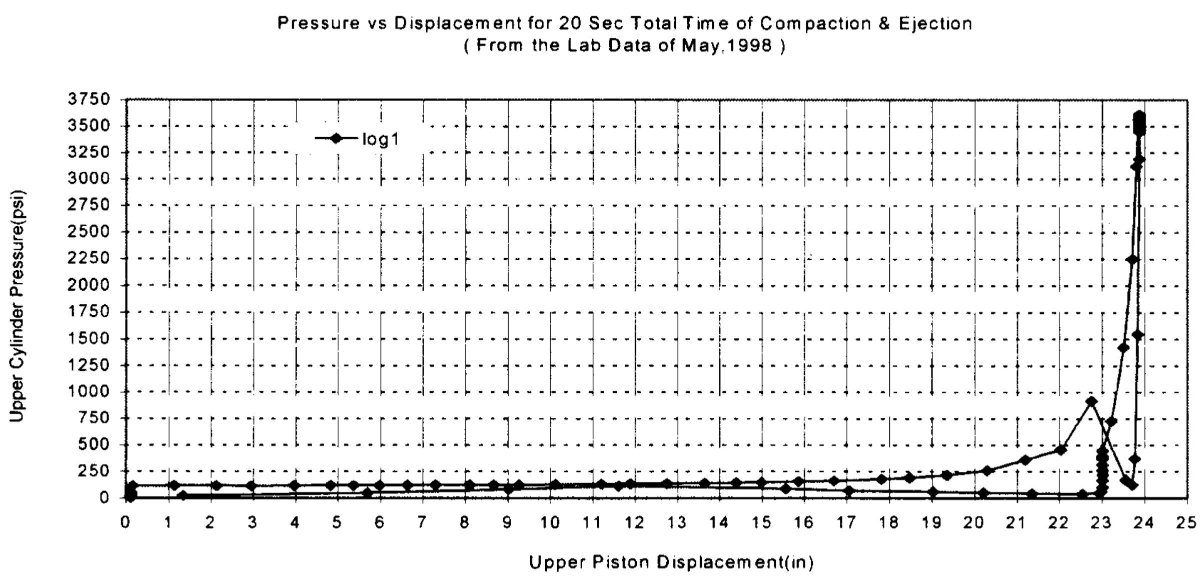


Figure 3.2 Upper Piston Displacement vs Pressure

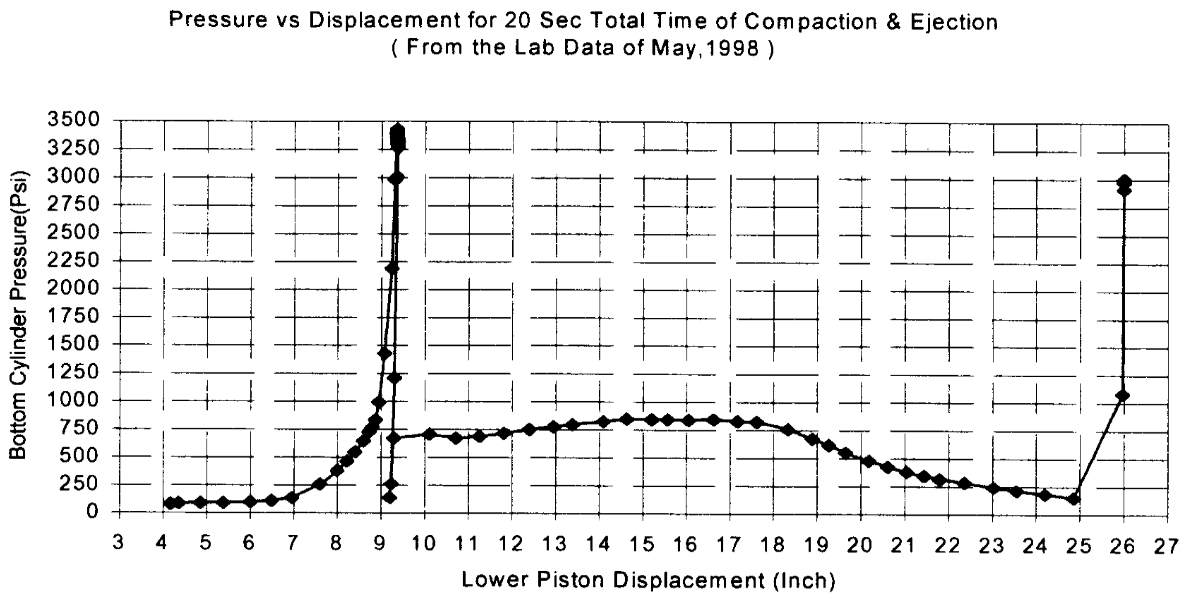


Figure 3.3 Lower Piston Displacement vs Pressure

The pressure-displacement curves during the compaction of a 5.4" coal log using the 250 ton hydraulic press are presented in Figs. 3.2 and 3.3. The total stroke of the top piston is 24 inches, the distance between the front of the piston head to the top surface of the mold is 15 inches (Fig 3.1). The total compaction length is 9 inches. There is no pressure on the top piston until it goes down for 17.5 inches. The pressure goes up at the last 10% of the total stroke, and the pressure on the top piston increases to the maximum of 20,000 psi within the last 2 inches of the displacement.

3.1.2 Cam Displacement Curve Design

A variety of cam curves are available for moving the punches up and down, four displacement curves are of the greatest utility in cam design. They are: (1) constant-velocity motion, (2) parabolic motion, (3) simple harmonic motion, and (4) cycloid motion. In our case of 5.4-inchrotatry press, the cam handles heavy load, long stroke, and slow rotation. Therefore, the parabolic motion cam curve is used at the start and the end slow rotation. Therefore, the parabolic motion cam curve is used at the start and the end of the compacting section and eject section. The most important advantage of this curve is that for a given angle of rotation and rise, it produces the smallest possible acceleration to avoid impact at the beginning and the end of the stroke (Machinery Handbook, 25th ed.). Between the two parabolic motion curves, there is a straight line graph shown in Fig 3.4. The straight line graph has important advantage of uniform velocity. This is desirable so that many cams used constant velocity in design. The angle of the straight line was determined from the angular speed of the turret, the radius of mold rotating, and the stroke of the punch on each section. Because this rotary press has plenty of room to make a longer follower travel in one cycle, the cam can be designed with low pressure angles to reduce the resistant forces.

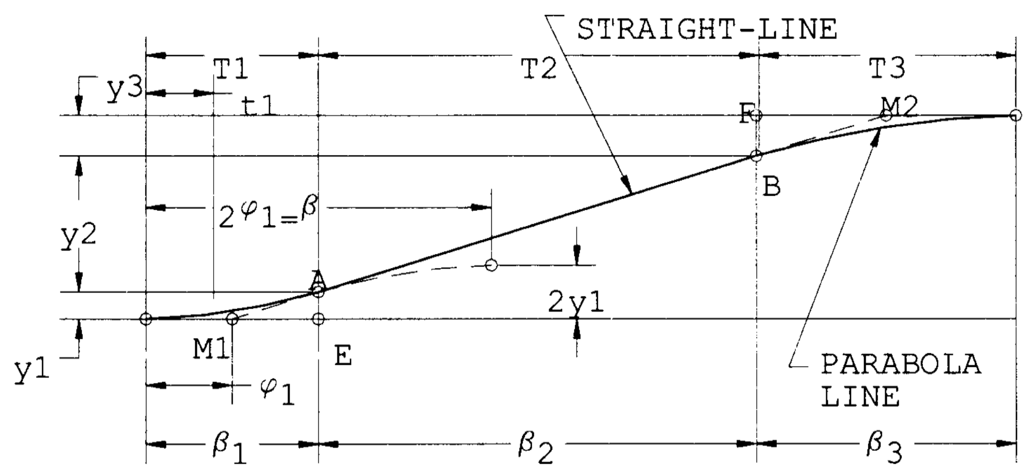


Figure 3.4 Matching a Parabola at Each End of Straight Line Displacement

The design of the cam curves for the 5.4" rotary press is shown in Fig. 3.6. The upper cam curve is divided into five sections (Fig. 3.1). Section I is for the upper punch traveling from the initial position down to the mold. Section II is for compaction, and

Section III is for dwell which puts the compact under the maximum pressure for a period of time. Section IV is for the upper punch traveling back to the starting position, and Section V is for collecting logs and feeding the material.

In Section I, the turret rotation angle is 27.2°, and its horizontal linear length is 21.356 inches. The punch traveling time is 2.266 seconds. The cam curve starts with a parabolic shape, then it connects with a 40° line to the end of this section. These curves are combined to provide a punch displacement from a height of 19 inches to a height of 5 inches. The equation of displacement is shown below.

1) Equation for the parabolic curve:

$$y = h - 4h_1(t/T_1)^2 = h - 4h_1(\varphi/2\varphi_1)^2 \tag{3.6}$$

$$0 \le t \le 0.5$$

$$0 \le \varphi \le 6°$$

where $T_1 = 1$ sec; $\varphi_1 = 6°$; $h = 19$ in, the total stroke of the upper punch ; $h_1 = 2.356$ in.

2) Equation for the curve of the 45° straight line:

$$\frac{y - y_1}{\varphi - \varphi_1} = \frac{y_2 - y_1}{\varphi_2 - \varphi_1} \tag{3.7}$$

Since $y_2 = 0$, Eq. 3.7 becomes

$$y = y_1\left(1 - \frac{\varphi - \varphi_1}{\varphi_2 - \varphi_1}\right) \tag{3.8}$$

$$6° \le \varphi \le 27.2°$$

where $\varphi_1 = 6°$; $\varphi_2 = 27.2°$; $y_1 = 19\text{-}h_1 = 16.644$ in.

In Section II, after the upper punch reaching the top surface of the mold, the pressure angle is changed to 5 degrees to reduce its speed for the compaction stage. The curve connects with the parabolic curve for smooth connection (Fig 3.5). About eight seconds later, the punch reaches the final compaction stage at the full stroke. The equation of motion for the third part of the upper curve is:

$$y_3=19-2h[1-((t_3-4)*90\pi/30)/T]^2 \qquad 4< t_3\leq 8; \qquad (3.9)$$

where, h = 3 in; T = 8 sec.

The lower punch starts its pre-compaction at the time when the upper punch reaches the mold. The pair of upper and lower punches gives the double action at the same time. The equations of motion for the lower punch are:

$$y_l = 1 \qquad 0\leq t_l\leq 2.8 \qquad (3.10)$$

$$y_2 =1+(t_2-2.8;)/\tan 10° \qquad 2.8 < t_2\leq 4 \qquad (3.11)$$

$$y_3= h[1-2(1-t/T)^2] \qquad 4< t_3\leq 8 \qquad (3.12)$$

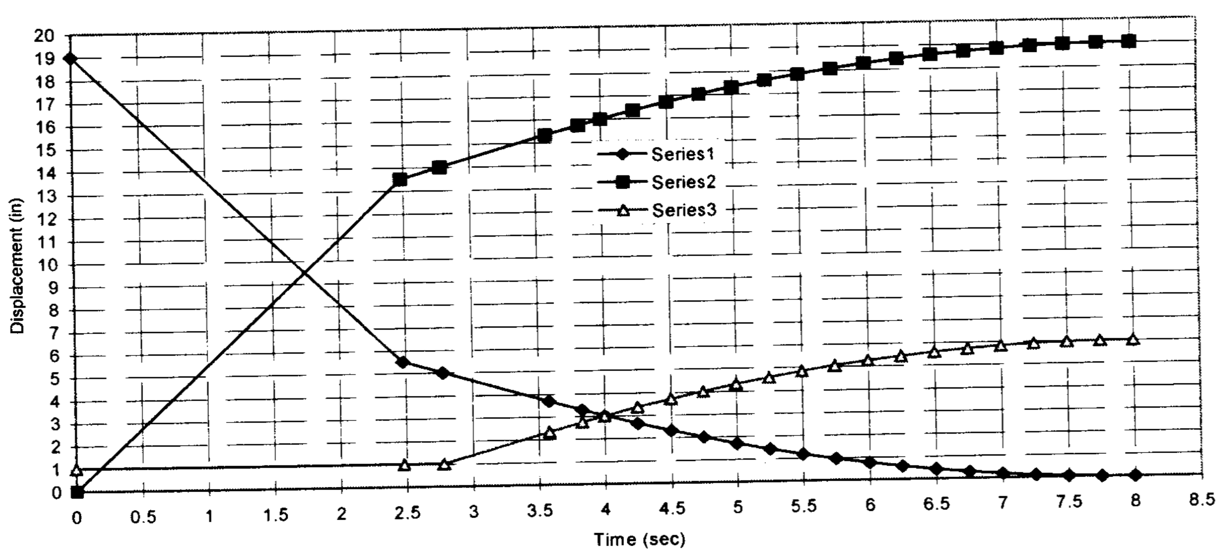


Figure 3.5 Displacement vs Time of the 5.4" Rotary Press

$$y=f(t)=2h\left(1-\frac{\phi}{\beta}\right)^2 \qquad (3.13)$$

$$v = 4h\frac{\omega}{\beta}\left(1 - \frac{\phi}{\beta}\right) \tag{3.14}$$

$$a = -4h\frac{\omega^2}{\beta^2} \tag{3.15}$$

$\phi = 41°$,

$\beta = 106°$,

$h = 7.3434$,

$\omega = 12°/\text{sec}$.

Therefore, $y = 5.5$ in

$$v = 4 \times 7.3434 \times \frac{41}{106}\left(1 - \frac{41}{106}\right) = 1.16 \quad \text{in/sec}$$

$$a = -4 \times 7.3434\frac{41^2}{106^2} = -4.39 \quad \text{in/sec}^2$$

where y is the displacement of the punch head in inches; v is the velocity of the punch head, inch /sec; a is the acceleration of the punch head, inch /sec^2; h is the maximum displacement of the punch head, inch; and t is the time for the cam to rotate, sec.

3.2 Cam Force Analysis and Driving Torque Calculation

After the displacement curves of the upper and lower cams are determined, the forces acting on the cam need to be calculated or otherwise determined. The main factors influencing cam forces are: (l) displacement and cam speed (forces due to acceleration): (2) dynamic forces due to backlash and flexibility: (3) linkage dimensions which affect weight and weight distribution; (4) pressure angle and friction forces.

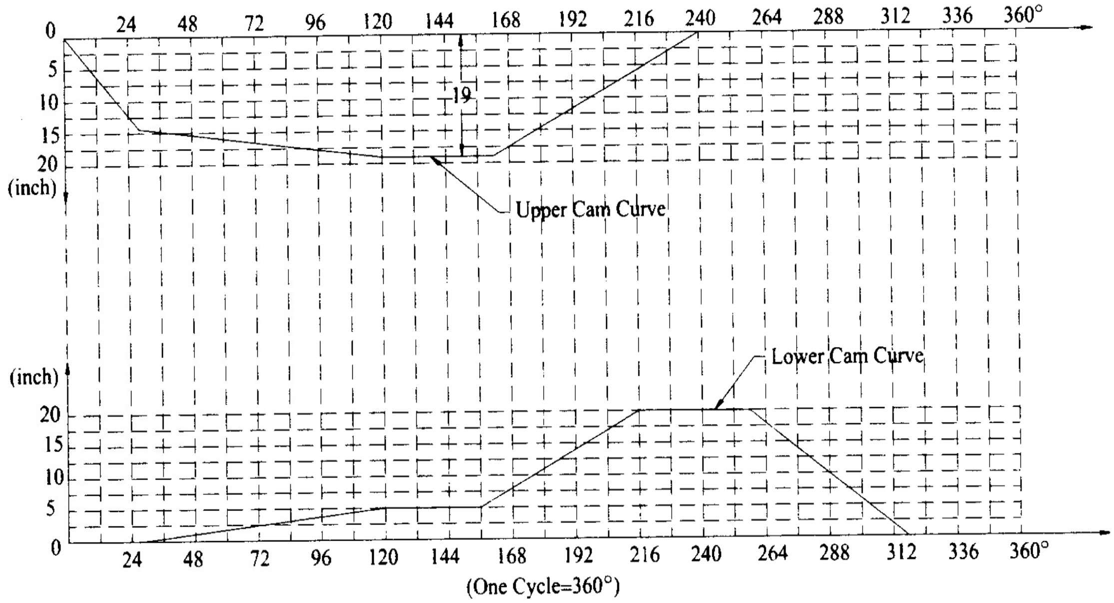


Figure 3.6 Cam Curves of the 5.4" Rotary Press

3.2.1 Cam Force.

The cam force (refers to Fig 3.6) can be calculated from the following formula (Machinery Handbook, ver. 25):

$$F_n = \frac{P}{\cos\alpha - \dfrac{\mu\sin\alpha}{l_2}\left(2l_1 + l_2 - \mu d\right)} \tag{3.16}$$

where, μ is the coefficient of friction in bushing = 0.05, P is the compacting force against the motion of the follower, α is the pressure angle, d is the diameter of the punch = 5.4 inches, l_1 is the distance between the roller center and the turret, l_2 is the length of the punch bushing.

For the force analysis, there are two cases to be considered. In Case I, during the 3 seconds of the maximum pressure dwell, the four pairs of punches are on the dwell section of cam (See Fig 3.8). During this period, one punch is on compaction stage, and is 12° ahead of the maximum pressure point. The other punch is at the eject section which is 12° after the maximum pressure point. At this moment, the four punches are under the maximum compaction force at the same time. Therefore, the driving force reaches the peak value in the Case I (Fig 3.8).

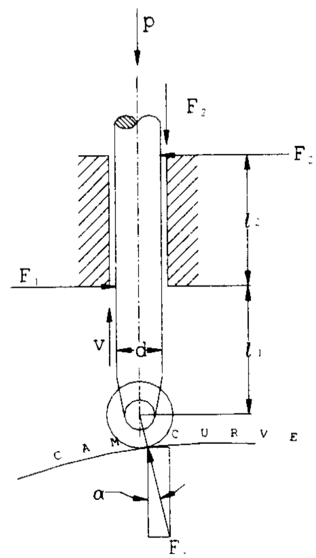


Figure 3.7 Cam and Punch Force Analysis

Case I (Fig. 3.8):

1) In the compaction section

The punch is 12 degree away from the dwell section, and the stroke from here to the maximum point is 0.41 inch. From the laboratory data (Fig. 3.2), the pressure of the top cylinder can be found to be 991 psi, and the pressure of the bottom cylinder can be found to be 726 psi. The upper punch compacting force at this point is:

$$P_u'' = 991(\text{psi}) \times \pi R^2(\text{cylinder area})$$

where, R = radius of the hydraulic cylinder = 6 inch.

$$P_u'' = 991 \times 6^2\pi = 112{,}079 \text{ lb.}$$

The lower punch compacting force at this point is

$$P_l'' = 725 \times 6^2\pi = 82{,}109 \text{ lb.}$$

The pressure angle at this point is $\alpha = 5°$.

For the upper punch, $l_1 = 4$ in, and $l_2 = 12$ in. To substitute these into Eq 3.16, the force on the upper punch is

$$F''_{nu1} = \frac{112079}{\cos 5 - \frac{0.05 \sin 5}{12}(2*4+12-0.05*5.4)} = 113{,}311\ lb$$

For the lower punch, $l_1 = 19$ in, and $l_2 = 34$ in. To substitute these into Eq 3.16, the force on the lower punch is

$$F''_{nl1} = \frac{82109}{\cos 5 - \frac{0.05 \sin 5}{34}(2*19+34-0.05*5.4)} = 83{,}187\quad lb$$

The average compacting force on both upper punch and lower punch is

$$F_{nl}'' = (F_{nul}'' + F_{nll}'')/2 = (114126+83966)/2 = 98{,}249 \text{ lb}$$

For the second punch pair, it is 24° before the dwell section. For the upper punch, $l_1 = 5.236$ in, and $l_2 = 35.763$ in, and $P = 100\pi R^2 = 11{,}310$ lb. To substitute these into Eq 3.16, the force on the upper punch is

$$F''_{nu2} = \frac{11310}{\cos 5 - \frac{0.05 \sin 5}{35.763}(2 * 5.236 + 35.736 - 0.05 * 5.4)} = 11{,}417 \; lb$$

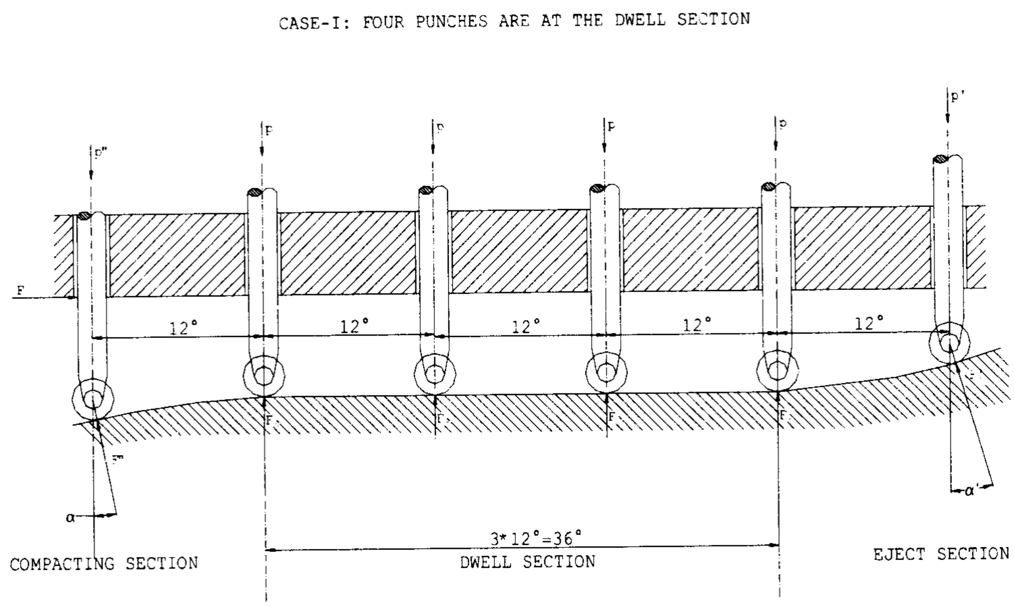


Fig 3.8 Cam Force Analysis Case I: Four punches on the Maximum Pressure Section

For the lower punch, l_l = 15.23 in, l_2 = 37.77 in, and P = 255 πR^2 = 22,620 lb. Substituting these into Eq 3.16, the force on the lower punch is

$$F''_{nl2} = \frac{22620}{\cos 5 - \frac{0.05 \sin 5}{37.77}(2 * 15.23 + 37.77 - 0.05 * 5.4)} = 22{,}886 \quad lb$$

$$F_{n2}'' = (F_{nu2}'' + F_{nl2}'')/2 = (11817+22886)/2 = 17{,}352 \;\; \text{lb}$$

2) The cam force on each punch dwell section

$$F_{nd} = P_{max} \tag{3.17}$$

$$P_{max} = Q \times S$$

where P_{max} is the maximum force on the punch in dwell section.

$$Q = \text{maximum compacting pressure} = 20{,}000 \text{ psi}$$

$$S = \text{area of punch head} = \pi r^2$$

$$\text{R} = 2.7 \text{ in}$$

$$F_{nd} = P_{max} = 20{,}000 \times 2.7^2 \times \pi = 458{,}044 \text{ lb}$$

3) The cam force in the ejection section

In the ejection section, the cam curve is a 15 degree straight line. The pressure angle is a constant α_e=15°, and the stroke h = 15 inch. The horizontal length of the cam is $15/\tan 15° = 55.98 \cong 56$ in. The turret rotation angle is 360×56/90π=71.276°. The angle between adjacent two punches is 12°, so that there are 5 punches on the ejection cam at the same time. In order to make good logs, it was determined that about 100 psi back pressure was needed to be applied to the punch head by the cams. There are five upper punches under the back pressure of 100 psi (Liu, group meeting of March, 1999) in this section for Case I. The force on each upper punch is

$$P_u' = 100 \times 2.7^2 \times \pi / 4 = 2290 \text{ lb} \tag{3.18}$$

$$\alpha = 15°;\ l_1 = 6.525 \text{ in};\quad l_2 = 34.475 \text{ in.}$$

From Eq 3.16, the force on the upper cam on eject section is

$$F_{neu} = \frac{5 \times P_u'}{\cos 15° - \frac{0.05 \sin 15°}{34.475}(2 \times 6.525 + 34.475 + 0.05 \times 5.4)} = 12{,}075 \quad lb$$

The eject force on the lower cam is

$$F_{nel} = \frac{5(P_e + P'_u + P_w)}{\cos\alpha°_e - \frac{0.05 \sin 15°}{l_2}(2l_1 + l_2 - 0.05 \times 5.4)} \tag{3.19}$$

where, l_1= (19+4)/2 =11.5 in, l_2 = (34 +49)/2 = 41.5 in, and P_e = average ejection force value which can be determined from Fig 3.3.

$$P_e = \frac{5.4^2 \pi}{4} \times \frac{900 + 300}{2} = 13{,}741 \quad lb$$

The P_w = punch gravity force

$$P_w = \frac{5.4^2 \pi}{4} \times 49 \times 491 \times \frac{1}{12^3} = 319 \quad lb$$

Substituting P_e and P_w into Eq 3.19.

$$F_{nel} = \frac{5 \times (13741 + 2290 + 319)}{\cos 15° - \frac{0.05 \sin 15°}{41.5}(2 \times 11.5 + 41.5 - 0.05 \times 5.4)} = 88{,}296 \quad lb$$

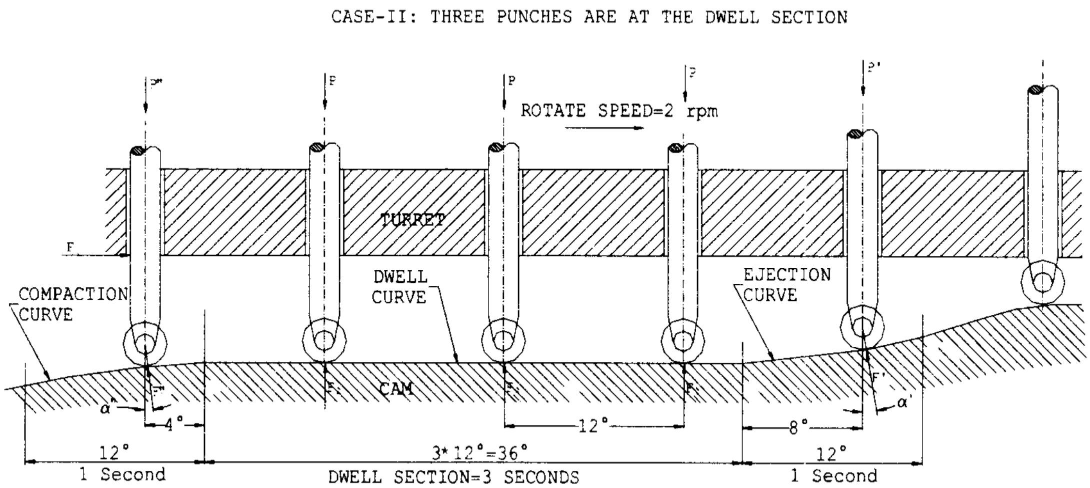


Figure 3.9 Cam Force Analysis case II

Case II (See Fig 3.9)

There are three pairs of punches on the dwell section, and two pairs of the punches are close to this section, one is 4° ahead of the beginning point of the dwell section, and the other one is 8° after the end of the dwell section. At this moment, only three pairs of the punches are under the maximum compaction force. Obviously, in Case II, the cam force is smaller than that in Case I. Therefore, it is unnecessary to calculate the cam force for Case II.

3.2.2 Cam Torque

The follower pressing against the cam causes resisting torque during the rise period and assisting torque during the return period. The maximum resisting torque was also calculated for Case I. The maximum resisting torque is used to determine the turret driving power. The maximum resisting torque can be calculated from the following formula:

(3.20)

where T_c is the total torque on the compacting section, T_d is the total torque on the dwell section, T_e is the total torque on the eject section, and T_f is the total torque on the feeding section.

Instantaneous torque values of a single punch on a rise cam may be calculated from following equation(machinery's handbook 25):

(3.21)

where N is the turret speed = 2 rpm, v is the velocity of the punch. Because at compacting section, the torque is the sum of the upper cam and the lower cam, computed torque from Eq 3.21 should be multiplied by 2.

At compacting section, $F_n = F_{n1}'' + F_{n2}''$=115,601 lb, v = πN/30*$tan5°$ = 0.825 in/s, α = 5°, and the cam torque is

$$T_c = \frac{30(F_{n1}"+F_{n2}")\cos 5° \times 2}{2\pi} \times 0.825 = 453{,}629 \quad lb-in$$

At the eject section, the cam torque is $F_n = F_{neu}+F_{nel}$ =100,370 lb,

v = πN/30*$tan15°$= 2.525 in/s, the α = 15°, and the cam torque is

$$T_e = \frac{30 \times 100{,}370 \cos 15°}{2\pi} \times 2.525 = 1{,}168{,}828 \quad lb-in$$

At the dwell section, the pressure angle α = 0, and cos0°=1. Since the dwell time is 3 seconds, there are four punches under the maximum compaction force. The total torque reached the maximum value at this moment. Therefore, the maximum torque is

$$\begin{aligned} T_d &= n\mu F_{nd} R = 8 \times 0.05 \times 458{,}044 \times 45 \\ &= 8{,}244{,}792 \quad lb-in \end{aligned} \tag{3.22}$$

where R is the distance from the turret center to the center of the mold.

In the dwell section, $F_{nd} = P_{max}$=458,044 lb,

where n is the number of punch pairs on dwell section, P_{max} is the maximum compacting force, and μ is the coefficient of rolling friction = 0.05.

At the feeding section,

$$\begin{aligned} T_f &= n\mu W_a R = 24 \times 0.05 \times 350 \times 45 \\ &= 18{,}900 \quad lb-in \end{aligned} \tag{3.23}$$

where W_a is the average weight of the upper and lower punches =350 lb.

Substituting the above value into Eq. 3.20, the maximum cam torque is

$$T_{\max} = 453{,}629 + 1{,}168{,}828 + 8{,}244{,}795 + 18900 = 9{,}886{,}152 \quad lb - in$$

$$= 823{,}846 \ \ lb\text{ - }ft$$

3.2.3 Driving Power

This section presents the power requirement of the rotary press for producing 5.4" diameter coal logs. The results of the analysis were used for designing the frame of the machine and determining the power required to drive the turret. Detailed calculations, including sketches, are as follows.

$$W_T = T_{max} \times \omega \times \mu \div k \tag{3.24}$$

where ω is the angular velocity = 0.20944 rad/s, which is obtained by assuming the turret speed to be 2 rpm; k is efficiency of gear box = 75%; and μ is coefficient of power spare = 1.4.

The turret driving power is calculated as

$$W_T = 823{,}846 \times 0.20944 \times 1.4 \div 0.75$$

$$= 322{,}086 \text{ ft- lbf/s} = 322{,}086 / 550 = 585 \text{ hp}$$

3.3 Cam Contact Stresses and Material

Because the roller on punch head takes very large compacting force from the cam during the highest pressure dwell section, the stress due to the pressure between them needs to be determined through calculations, and materials must be selected carefully. Their analysis and calculation are carried out in this section.

3.3.1 Cam Contact Stresses Calculation

The stresses at the cam and the punch surface must be calculated and suitable materials to withstand the stress must be selected. If the calculated maximum stress is too large, it will be necessary to change the cam design. Such changes may include:

(1) Increasing the cam size to decrease pressure angle and increase the radius of curvature;

(2) Changing to an offset or swing follower to reduce the pressure angle;

(3) Reducing the cam rotation speed to reduce inertia forces;

(4) Increasing the cam rise angle, β, during which the rise, h, occurs;

(5) Increasing the thickness of the cam, provided that deflections of the follower are small enough to maintain uniform loading across the width of the cam;

(6) Using a more suitable cam curve or modifying the cam curve at critical points.

When a roller follower is loaded against a cam, the compressive stress developed at the surface of the contact may be calculated from Eq. 3.25 (Machinery Handbook 25)

$$S_c = 1805\sqrt{\frac{F_n}{b}\left(\frac{1}{r_f} \pm \frac{1}{R_c}\right)} \tag{3.25}$$

where S_c = maximum calculated compressive stress, psi; F_n = normal load, lb; b = width of cam, inch; R_c = radius of curvature of cam surface, inch; and r_f = radius of roller follower, inch.

The plus sign in Eq. 3.25 is used in calculating the maximum compressive stress when the roller is in contact with the convex portion of the cam profile and the minus sign is used when the roller is in contact with the concave portion. The maximum contact stress occurs at the dwell section. In this section, the roller is in contact with the straight (flat) portion of the cam profile, $R_c=\infty$, and $1/R_c=0$. In practice, the greatest compressive stress occurs when the roller is in contact with that part of the cam profile which is convex and has the smallest radius of curvature.

In our case: the radius of the roller r_f= 4 in.,

the convex radius of the cam $R_c = \propto$ in.,

the width of cam b = 8 in.,

and the maximum load F_n = 458044 lb.

To find the maximum surface compressive stress, from Eq. 3.23,

$$S_c = 1805\sqrt{\frac{458044}{8}\left(\frac{1}{4}+0\right)} = 215{,}951 \quad psi$$

This calculated stress should be less than the allowable stress =226,000 psi (page 2100, Machinery's Handbook 25).

3.3.2 Cam Material Selection

In considering materials for cams, it is difficult to select any single material as being the best for every application. Often the choice is based on common practice or the machine-ability of the material rather than its strength. However, the failure of a cam or roller is commonly due to fatigue, so that an important factor to be considered is the endurance limits, and the relative hardness of the mating surfaces.

Base on individual cam application on the 5.4" rotary press, there are two kinds of material to be employed. The first one is SAE 4340 steel with 226 kpi (machinery's hand book 25, page 2100) compressive stress or SAE 1020 steel, carburized to 0.045 inch depth of the contact surface, 50-58 Rc surface hardness, and 226 kpi compressive stress. It is used for the high wear load cams where they are in the compaction section, the maximum pressure holding section, and the eject section compaction. The second one is PVC-U for the light load cams where they are in the filling section and return section. The PVC-U cam has some advantages: (1) easy to be machined, (2) quiet, (3) light weight , and (4) less cost.

3.4 WORM GEAR CALCULATION

After the turret driving force was determinate in the previous section, the worm gear calculation was pursued in this section. The worm gear set has two driving worms (Fig 3.10). A mechanical engineering design software, named TK Solver, was used to do the gear design calculation. It can carry out the design optimization throughout the calculation procedure. The output provides a whole set parameters of the worm gear pair. The results are showing as follows, the worm and gear set is showing in Fig. 3.11 and Figure 3.12. The calculation results and specification are shown in Table 3.2

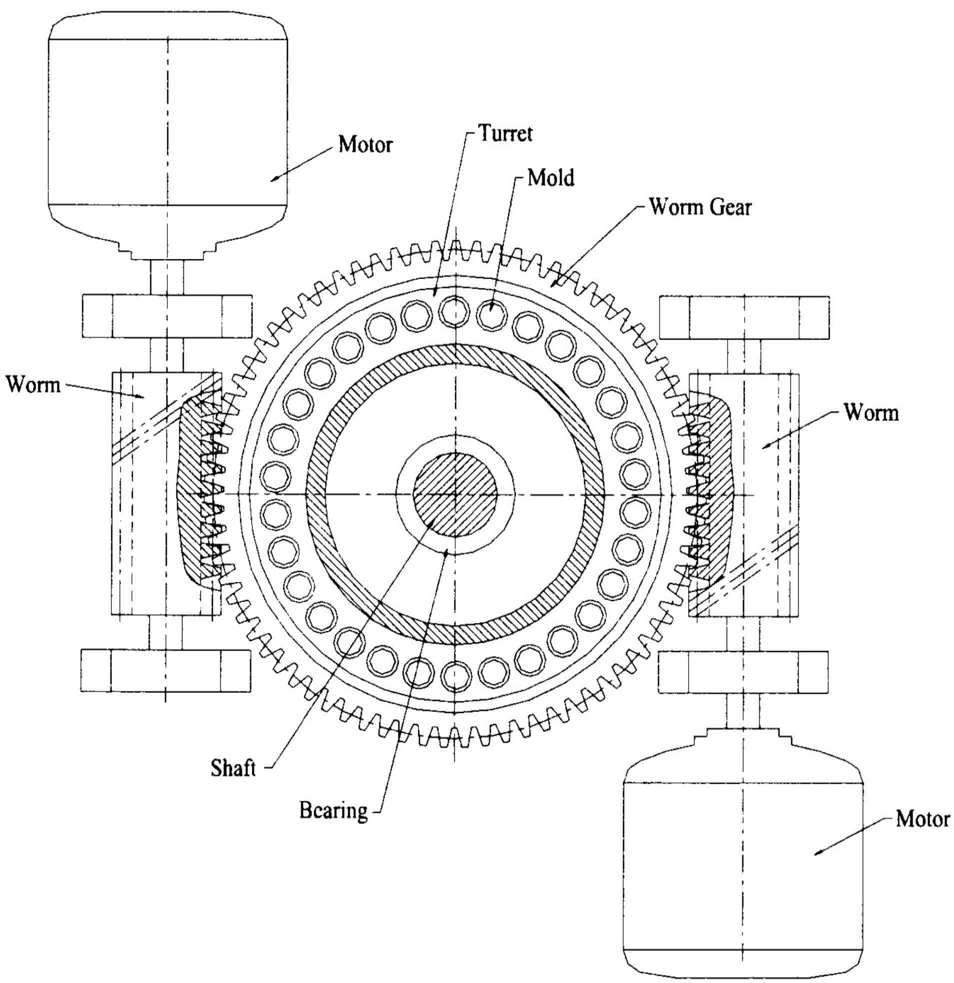
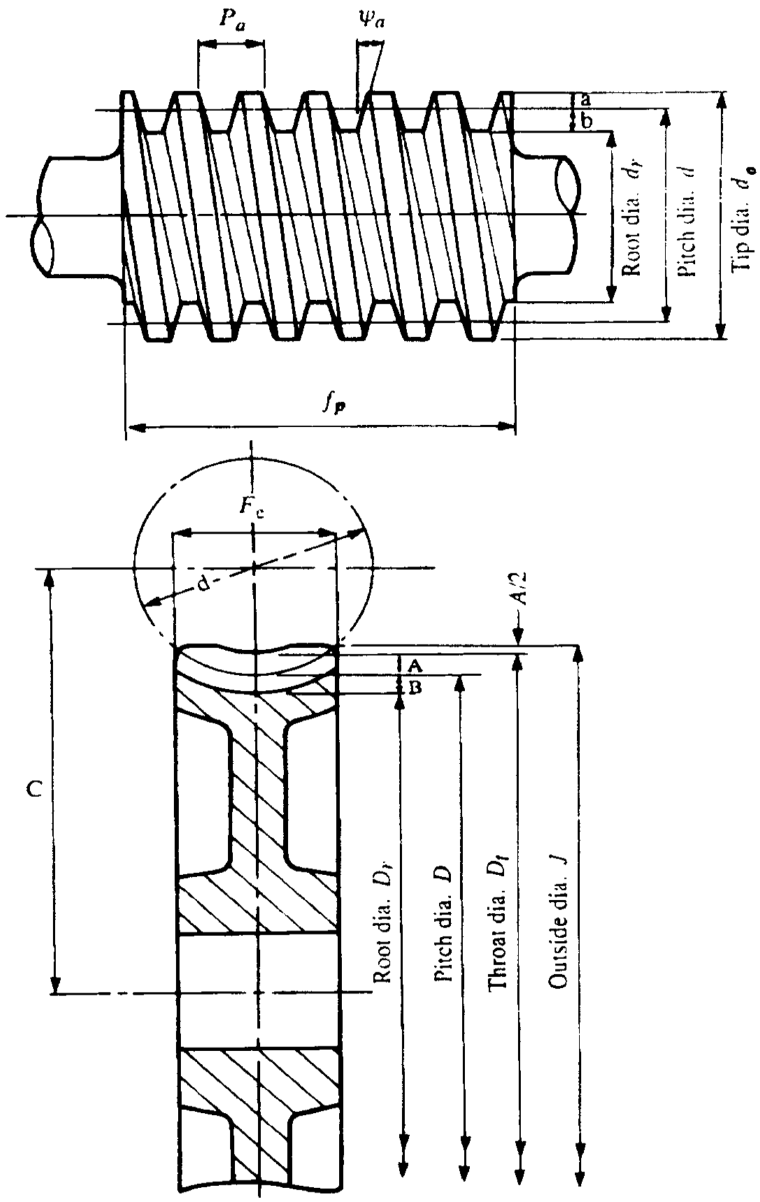


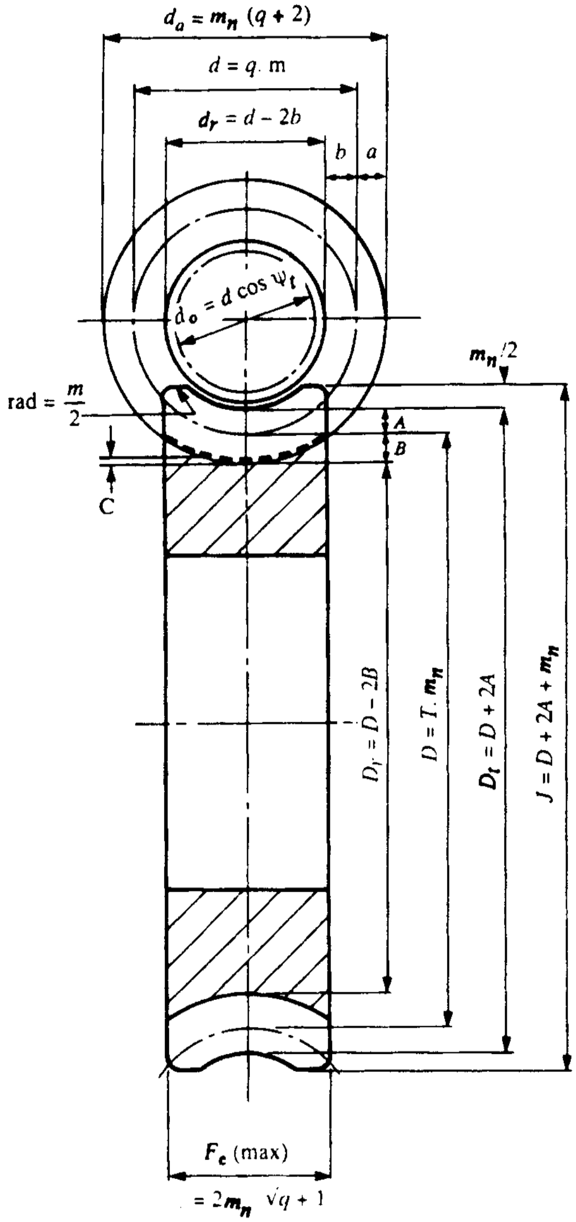
Figure 3.10 Worm Gear Set of the 5.4" Rotery Press



d_o = Outside diameter – worm
J = Outside diameter – wheel
d_r = Root diameter – worm
D_r = Root diameter – wheel
ψ_a = Axial pressure angle
f_p = Minimum facewidth –worm
F_e = Maximum facewidth – wheel

9 PC diameter – worm = Axial module × Diameter quotient

Figure 3.11 Typical Worm Gear Set View I (Alec, 1992)



$$m = \text{Axial module} = \frac{\text{Axial pitch}}{\pi}$$

16 Transverse pressure angle

$$= \tan^{-1}\left(\frac{\tan \text{ Normal PA}\sqrt{\text{No. of starts - worm}^2 + \text{Dia. quotient - worm}^2}}{\text{No. of starts - worm}}\right)$$

17 **Base diameter = PC diameter – worm × cos Transverse PA**

Figure 3.12 Typical Worm Gear Set -- View II (Alec, 1992)

Table 3.2 Parameters of Worm Gear Set

Sta	Input	Name	Output	Unit	Comment
	135.	n		rpm	worm input speed
		ngear	2.0	rpm	speed of wormgear
	20.0	phi		deg	pressure angle
	67.5	mG			gear ratio
	2.0 Nw	N_w			number of threads on worm
		Ng	135.0		number of teeth on worm gear
		Ngmin	21.0		min recomm # gear teeth
	63.000	C		in	center distance (assume for trial calc
		Dmean	108.939	in	mean gear diameter
		D	108.939	in	pitch diameter of gear
		V	605.7	ft/min	mesh velocity
		Wt	56,353.	lb	tangential load
		Wf	1,592.	lb	friction force
		Fe	11.431	in	effective face width
		Cm	.707		ratio correction factor
		Cv	338		velocity factor
		Cs	483.1		materials factor -chill cast bronze
		d	17.061	in	pitch (mean) diameter of worm
		d_min	12.511	in	min recomm worm pitch dia
		d_max	23.459	in	max recomm worm pitch dia
		add	.807	in	addendum
		ded	.934	in	dedendum
		clear	-.127	in	clearance
		do	18.675	in	outside diameter of worm
		px	2.535	in	axial pitch
		L	5.070	in	lead of worm
		lambda	5.404	deg	lead angle of worm
		lambto	2.702	deg	lead angle per worm thread (< 6)
		u	.026		coefficient of friction
		Tgear	3,069,5		in-lb rated torque on wormgear
		eff	76.93	%	efficiency of wormset
		powou	97.45	hp	output power available
		powlos	29.22	hp	power loss in mesh
		powin	126.67	hp	rated input power

3.5 Back Pressure Spring Calculation

The back pressure is put on the upper cam at the eject section for improving the quality of logs. Because the log is longer than the tablet, the compressive stress is built up inside of the log during the log compaction stage. When a log is ejected out of the mold without the back pressure, the elastic spring back will make a part of the log expend when it is ejected outside of the mold, which could crack the log. In conventional tablet rotary press, the pressure of the upper punch is released immediately after the tablet is compacted. There is no back pressure device designed for these machines. The back pressure application, invented by CPRC (Liu, 1999), is a new addition to the compacting technology. Therefore, a back pressure device is added to the 5.4-inch rotary press in this design. We use a group of spring to apply the back pressure to the upper piston.

The parameter of the back pressure spring calculation was done by using TK computer software (TK Solver, version 1.1). The results are shown in Table 3.3.

Table 3.3 Results and Specification of the Back Pressure Spring

Ta	Input	Name	Output	Unit	Comment
	.285	dens		lb/in^3	specific weight of steel
	386.4	g		in/sec^2	gravitational accel
	1,000.	rpm		rpm	excitation frequency
	1.150E7	G		psi	shear modulus of material
L	7.0	C			trial spring index
	.220	dia			trial wire diameter
		d	.207	in	available wire diameter (from List Fun
	1.000	y		in	deflection of spring
	15.	clash		%	% of deflection for clash allowance
					material conditions
	'music'	matl			one of 'music, 'oiltemp
					'hdrawn, 'chromev 'chromes
	'sqgrnd	end			one of 'plain,'pgrnd,'square,'sqgrnd
	'peen	surface			one of 'unpeen or 'peen
	'set	setflag			"set for a set spring - 'unset othe
	1.00	shift			number of shifts / day
	10.00	years			years of operation desired
		Ncycles	1.2E9		Cycles needed for desired life
					forces
	144.00	Fmax		lb	maximum applied force
	50.00	Fmin		lb	mimimum applied force
		Falt	47.00	lb	alternating force
		Fmean	97.00	lb	mean force
		Fshut	158.	lb	force at shut height
L		k	94.00	lb/in	spring rate
		ka	93.8	lb/in	spring rate with Na rounded coils
		N	9.23		no of active coils - exact
		Na	9.25		no of active coils - to nearest 1/4 co
		Ntot	11.25		no of total coils
		D	1.45	in	mean coil diameter
L		Dout	1.66	in	outside coil diameter
L		Din	1.24	in	inside coil diameter
L		Ks	1.07		static factor - for direct shear - Eq

Table 3.3 Results and Specification of the Back Pressure Spring (Continued)

L		Kw	1.21		Wahl Factor - direct shear + SC - Eq 1
L		Kc	1.13		Curvature SC factor = ratio of Kw to K
		tauinit	22,286.	psi	shear stress at installed length
		taustat	64,183.	psi	shear stress at Fmax for static loadin
		taushut	70,468.	psi	Stress at shut height
		taualt	23,714.	psi	Alternating shear stress for fatigue
		taumean	43,235.	psi	Mean shear stress for fatigue
		Sut	238,507.	psi	Ultimate tensile strength eq 13.3 & Ta
		Sus	159,800.	psi	Ultimate shear strength equation 13.4
		Ssy	143,104.	psi	shear yield based on Table 13-6
		Sew	67,500.	psi	wire endurance limit - equation 13.12
		Ses	68,043.	psi	fully-reversed endurance limit eq 13.1
L		Nf	1.79		safety factor - fatigue - eq. 13.14
L		Ns	2.23		safety factor - static loading at Fmax
L		Nshut	2.03		safety factor - shut height (yielding)
L		Lf	4.01	in	free length
L		Linstal	3.48	in	installed length
L		Lcomp	2.48	in	compressed length
L		Lshut	2.33	in	shut height
		yinit	.53	in	inital deflection at assembly
		ymax	1.53	in	max working deflection
		yclash	.150	in	coil clash allowance
		yshut	1.68	in	deflection to shut height
					buckling parameters for Figure 13-14
		R1	2.77		ratio of free length /coil dia
		R2	.38		ratio of max deflect / free length
					surging calculations
L		Wactive	.40	lb	weight of active coils Eq 13-11b
L		Wtotal	.49	lb	weight of total coils Eq 13-11b
L		mass	1.05E-3	blob	mass of active coils
		nf	149.8	Hz	exact natural frequency in Hz
		nfapp	149.2	Hz	approximate natural frequency
		crpm	8,987.	rpm	natural frequency in rpm
L		FreqFac	9.0		ratio of nat'l freq to excitation freq
	31.00	Npts			number of points for Goodman plot list
		hole	1.73	in	will work in minimum hole diameter
		pin	1.17	in	will work over max pin diameter

3.6 Over Load Protection Spring Calculation

The overload protection spring calculation was also finished by using the TK 1.1 computer software. After design parameters had been designed, it was given to computer, then the results were obtained (See Table 3.4.).

Table 3.4. Over-Load Protection Spring

Sta	Input	Name	Output	Unit	Comment
					File: Over Load Bellevil.tk
					Belleville spring designer
					See plots - has nonlinear spring rate
					R. Norton 6/9/96
	3.0E7	E		psi	Young's modulus
	.3	v			Poisson's ratio
	'steel	matal			'steel or 'nonsteel
	'unset	set			'set or 'unset
	246,000.	Sut		psi	tensile strength from Table 13-5
		Sy	295,200.	psi	allowable stress from Table 13-13
		t	.090	in	thickness of washer
	1.414	hovert			ratio of h over t
		h	128.	in	inside height of spring
L	1.000	y		in	design deflection - must be input
		y%	783.6	%	% of h at deflection y
		yf	128.	in	deflection to flat condition
		yend	.255	in	end value for deflection list
	66.0	ymin%		%	% of h to minimum deflection
	Input	Name	Output	Unit	Comment
		ymin	.084		minimum deflection
	134.0	ymax%		%	% of h to maximum deflection
		ymax	.171	in	maximum deflection
	2.0	R			ratio of dout/din
	1.200	dout		in	outside diameter
		din	.600	in	inside diameter
L		F		lb	applied force at deflection y
	12,500.00	Flat		lb	force at flat condition

Table 3.4. Over-Load Protection Spring (continued)

L		Fnorm	31,651.95	%	% error in force from constant
L		sigto		psi	outer tensile stress at y
		sigtomi	2,063,906	psi	outer tensile stress at ymin
		sigtoma	3,098,112	psi	outer tensile stress at ymax
		sigtofl	2,719,577	psi	outer tensile stress at yflat
L		sigti		psi	inner tensile stress at y
		sigtimi	222,746.	psi	inner tensile stress at ymin
		sigtima	1,647,542	psi	inner tensile stress at ymax
		sigtifl	783,501.	psi	inner tensile stress at yflat
L		sigc		psi	compressive stress at y
		sigcmin		psi	compressive stress at ymin
		sigcmax		psi	compressive stress at ymax
		sigcfla		psi	compressive stress at yflat
		Nc	0.0		safety factors compression
		Ncflat	.1		
		Ncmax	.1		
		Nto	0.0		safety factors tension
		Ntoflat	.1		
		Ntomax	.1		
		K1	.689		parameter for stress calculations
		denom	.2		intermediate calculation
		K2	1.220		constant for compression stress
		K3	1.378		constant for compression stress
		K4	1.115		constant for tensile stress
		K5	1.000		constant for tensile stress
	51.0	N			no of data points for plots
		sigcCK			
		sigtiCK			
		sigtoCK			
					check against Wahl's equations
	1.441	Cwahl			from Wahl Fig 13-6 p159
		Rwahl	.6		
		sigcw			Wahl's Eq. 13-17 p171
		sigtiw			Wahl's Eq. 13-18 p171
		sigtow			Wahl's Eq. 13-19 p171

Chapter 4

DESIGN OF A ROTARY PRESS FOR COMPACTING 5.4-INCH DIAMETER LOGS

This chapter describes the preliminary design of a rotary press for compacting 5.4-inch diameter logs. This rotary press is intended to be a prototype machine for testing and demonstrating commercial operation of the CLP technology. This machine is designed to generate 20,000 psi compaction pressure and to produce 3600 logs per hour (one log per second). The type of compacting motion is double action by using two punches moving towards each other. In this design, The size of the compaction mold, the length-to-diameter ratio of compacts are similar to the previous design of the 250-ton hydraulic press. The 10-inch length of a compacted log, the 100 psi back pressure, and the three second compact dwell time were determined at a group meeting in February, 1999. The cam curves were designed by analyzing the compaction data of the 250-ton hydraulic press, and studying cam design books. The compaction by this press is a four-step cycle: 1) material feeding, 2) log compacting, 3) log ejecting, 4) log collecting. The detailed description of the design is given as follows.

4.1 Feeding System

The feed is supplied to the turret by a vertical screw feeder. The feeding rate can be adjusted for different kinds of compacting materials, the average feeding rate is 954 ft^3/hour. It was calculated from Equation 4.1.

$$Q = N \times V \times \frac{n}{12^3} \quad (ft^3 / hour) \tag{4.1}$$

where Q is the feeding rate, N is the number of mold = 30, V is the effective volume of a mold = $20\times 5.4^2\pi/4 = 458.0442$ in^3, and n = rotating speed of the turret = 120 revolution per hour.

The reason for choosing a screw feeder is that it can pre-compress the very loose material, and has the function of weight control. Without pre-compression, the thickness of the turret for the 5.4-inch diameter log compaction will be very large, which may increase the weight of the turret, power consumption, and costs. To ensure a constant weight of feed, the feeder covers several molds, allowing the molds to be completely filled with materials. The surplus is scraped away at the top of the turret by the edge of the cover, leaving the desired volume of materials to be compacted.

In order to ensure complete filling and avoid clogging, in addition to the vertical screw feeder (Fig. 4.1.), a powder feeding device is also designed based on the power feeder used on a modern high speed double-layer tablet machine (FETTE PT 3090, Wilhelm Fette GmbH, Schwarzenbek, FRG). The feeding device is shown in Fig. 4.2a. Two rotating wheels are used in this device to carry the coal materials delivered by the vertical screw feeder to the feeding location. The arrangement of this feeding device on the turret is given in Fig. 4.2b. A similar mechanism was chosen for the current feeder design.

Figure 4.1 Vertical Powder Feeder (Ridgeway-Watt, 1988)

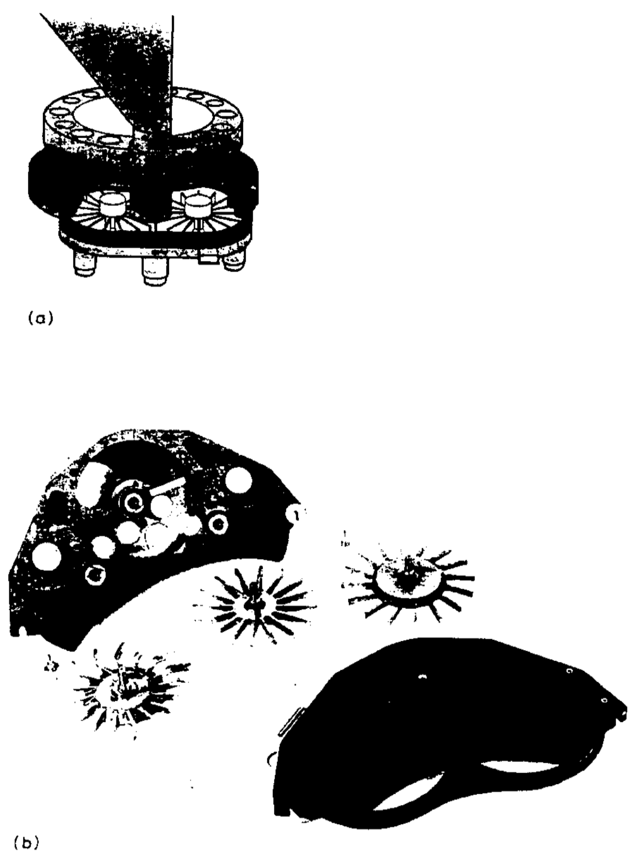


Figure 4.2 Feeder Components (Pietsch, 1991)

4.2 Compaction System

By analyzing the special production process requirements of making coal logs, the compaction system needs to have a three seconds of high pressure dwell time section when the punches travel along on the dwell cam with a constant linear speed of 9.424 inch/second. A conventional rotary press cannot provide such long compaction dwell time. Because the conventional rotary presses use cam to do precompaction and ejection, and use a pair of rollers by pressing a pair of punches to pursue compaction and dwell. The dwell time depends on the size of the compacting rollers and the size of the punch head. Both are limited by the size of the machine. For this reason, a complete new cam follower compacting system was designed for the 5.4-inch rotary press. In this design, a pair of rollers were fix on each punch's head (Figs. 4.3 and 4.4). Therefore, unlike most rotary presses which have sliding contact between the follower and the cam, this design has rolling contact between the follower and the cam. Cams were used through out the entire processing circle. Thus, the punches can travel along the cams with desired motion to make better logs. This new development is not only good for log quality but also good for greatly reducing the friction force, which translates to a lower power consumption and less wear. Although it may increase punch cost, it saves money on operating and machine maintenance in the long run. Good performance and low cost are the most important features for this machine. This arrangement increases the allowance for the wear of punches and cams, and reduces the downtime and cost for changing punches and cams. The compaction system was designed as a double action cam-follower system. It includes 30 pairs of upper and lower punches, upper track cam, lower track cam, and 30 molds in a turret. Once the feeding is completed, the material is compressed by the two punches passing between the two cams; both of them are loaded for applying the required pressure. This compacting system is substantially different from those used in the conventional tablet rotary presses.

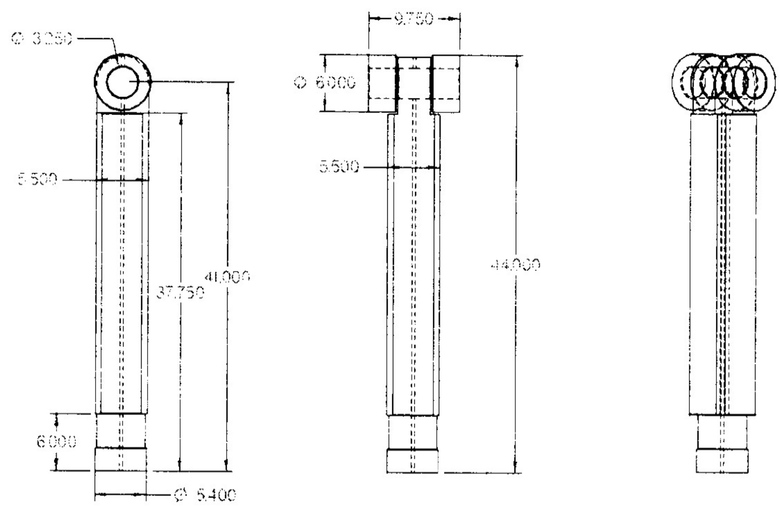


Figure 4.3 Upper Punch

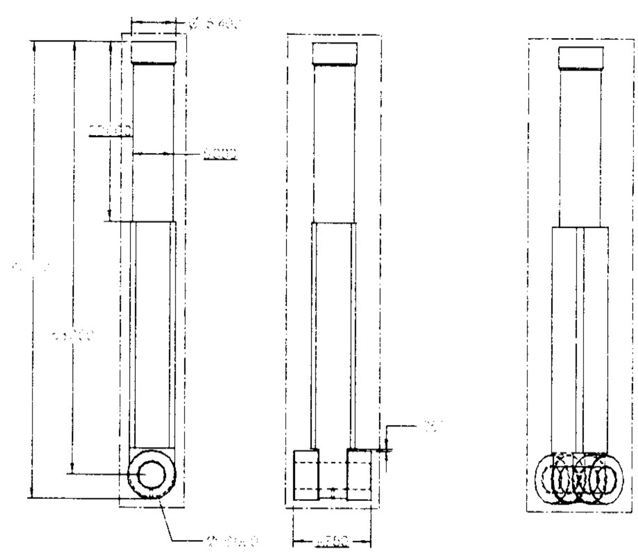


Figure 4.4 Lower Punch

4.2 The Mold

In this preliminary design, the turret has 30 molds, they are distributed on a 90 inch diameter circle which is around the rotational axis of the turret. In order to reduce the size of the turret and to lower the cost of the machine, mold inserts will not be used in this rotary press. Because it is a testing machine, it will not be used intensively in the future. Therefore, the mold surface wear out will be very small. In addition, since each mold insert includes two parts. One part is the mold body and the other is a carbide liner, its cost will be high for 30 molds. If the machine is designed for commercial applications, the mold needs the insert. Usually, mold inserts may be hold in place by clamping or by shrink fits. The later is preferred due to simplicity. It is recommended that (Powder Metallurgy Equipment Association, 1977) when production volume is high or abrasive conditions are encountered, molds should be made of tungsten carbide with low Cobalt (3-6%) for round molds. Therefore, it is decided not to use insert in the mold at this time.

The length of the mold is determent as follows:

If using log-diameter-to-pipe-diameter ratio of 0.9, for a 6 inch pipeline,

Log diameter: $D = 5.4$ in

Log aspect ratio: $L/D = 1.8$

Log length: $L = 9.72 \cong 10\text{in}$

Loading length: $L_0 = 2L = 20$

Bottom seal length: $L_s = 1$ in

Therefore, the length of the mold is designed as 21 inch for the 5.4-inch-diameter mold rotary press (see Fig. 4.5).

4.22 Turret Design

The complete design of the 5.4-inch rotary press starts from the turret design, because turret is the heart of a rotary press machine. Its dimensions and shape need to be determined first so that other calculations and geometric design for the rest of the components can be carried out based on its design.

1) Turret diameter

$$d_m = \frac{(d+2b+c)N}{\pi} = \frac{(5.4+2\times 0.5+3.024778)\times 30}{\pi} = 90'' \tag{4.2}$$

$$d_w = d_m + 2k = 90 + 2\times 6 = 102'' \tag{4.3}$$

where:

b --- the wall thickness of the mold

c --- the clearance between neighboring molds

d --- inner diameter of mold

d_m --- diameter of the circle where the 30 molds located

d_w --- outside diameter of the turret

k --- the rim of the turret

N --- the number of molds on the turret

Thus, the outside diameter of turret is 102 inches.

2) Turret height

The turret has three layers, the top layer, the middle layer, and the bottom layer. The total height of the turret is determined by analyzing the requirements of thickness of the three layers and the height of the three spaces between layers. All of the related factors need to be considered in the design. For example, the depth of the mold, the spaces of loading and unloading, the traveling height of the lower punch for log ejection, the spaces for punch installation and uninstall, etc., all need to be considered. Based on the maximum fill depth of the mold, the dimensions of the punches, and the arrangement of the punches on the turret to accomplish the required performance, the turret height is determined as follows.

The top layer of the turret is designed for guiding the upper punch up and down. There are 30 square holes for guiding the upper punches, the depth of the holes should be more than two times of one side length of the square. Therefore, the thickness of the upper layer of the turret is determinate to be 12 inches. The square shape of the hole was designed to keep the punch from rotating and to reinforce the bending strength of the punches. Circular shape can be used too (Fig 4.25), if longitudinal key is installed on turret.

The fist space between the top layer and the middle layer is used for the feeding device, log ejection, and the upper punch length adjustment. The height of the space was selected to be 20 inches. The middle layer contains 30 molds. The thickness of the layer is the same as the length of the mold: 21 inches. The second space, which is between the middle layer and the bottom layer, is designed to give a suitable distance for the bottom punches. In order to pursue compacting and ejecting logs, the stroke of the lower punch is 20 inches. Thus, the minimum height of the second space is 20 inches. The bottom layer has the same function as the top layer, and the driving gear is bolted on it. Because the head of the lower punch is always in the mold, the guiding length of the bottom square hole dose not needs to be as long as the ones on the top layer. The bottom layer was determined to be 8 inches thick. The third space is from the bottom layer to the bottom of the turret. It was designed to make room for the lower cams. The height of this space is 22 inches.

By adding up all of the dimensions, the height of the turret (Fig. 4.5) is

$$12 + 20 + 21 + 20 + 22 = 95 \text{ inches.}$$

Since this turret is rather large, it was designed as two pieces for casting and machining (Figs 4.6 and 4.7).

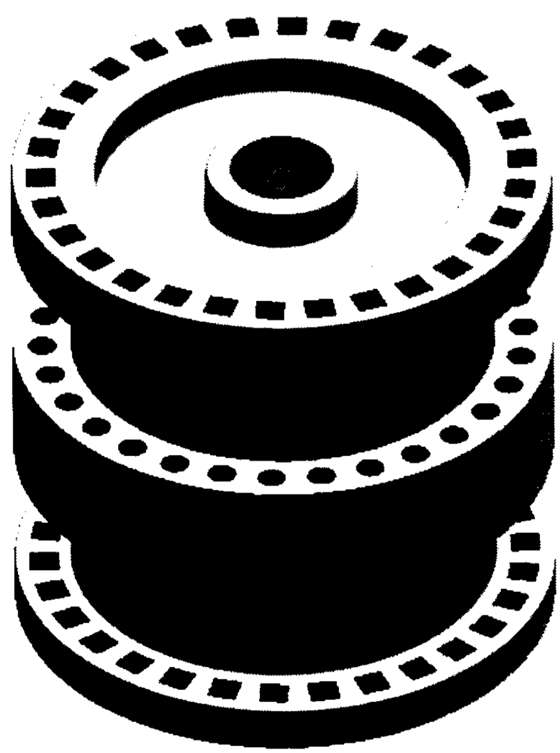


Figure 4.5 The Turret Designed for 5.4-Inch Rotary Press

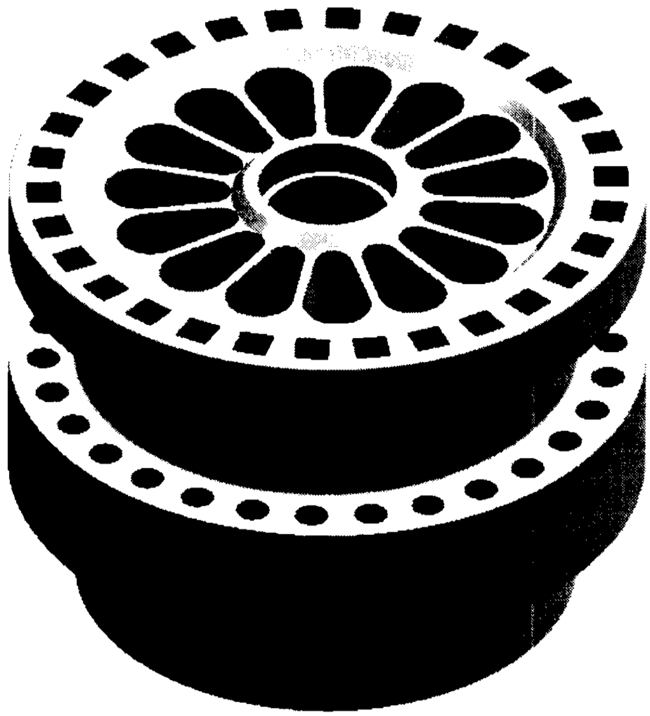


Figure 4.6 Top Part of the Turret

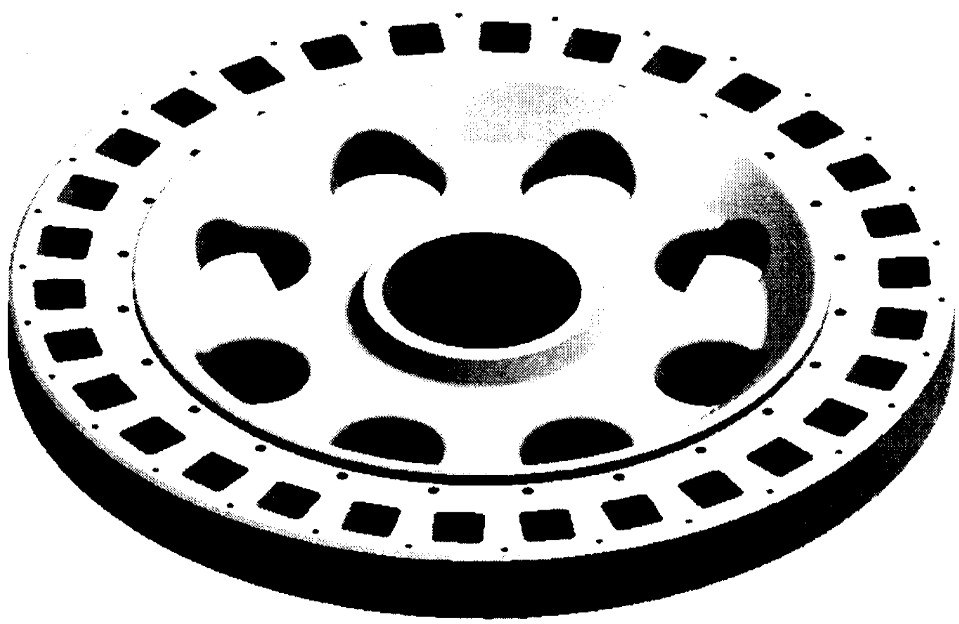


Figure 4.7 Bottom Part of the Turret

4.23 Punch Design

The upper and lower punches are designed based on the coal-log rotary press special requirements. It is different from the punches of the tablet rotary presses published in several references (Swartz et al., 1962; Swartz, 1969). Each punch has three sections, the punch head, the guide, and the working head. In order to reduce friction force, two rollers were added to each punch head. The guide section is a square bar to maintain the roller's orientation. The guide section also can be designed as a round bar with a key on it, it may or may not save manufacturing cost. Considering the bending strength of the punches, the punch guide section was chosen to be square shape. In order to compact different lengths of logs for different materials, the working head was designed to be replaceable. Thus, the length of the punch can be adjusted by changing the length of the working head (Fig 4.9). The stroke of the punches was fixed, the stroke of the upper punch is 19 inches, and the stroke of the lower punch is 20 inches. The upper and the lower punch are shown in Fig. 4.8. The detailed dimensions of the upper and lower punches used for the coal log rotary press are given in Figs 4.3 and 4.4. The upper and the lower punches weigh about 356 lb and 413 lb, respectively.

According to *Powder Metallurgy Equipment Manual* (Powder Metallurgy Equipment Association, 1977), punch steel requirements are very different from die steel requirements. For punch steel, toughness is an important factor. High carbon, high chrome steels are too brittle in most cases. In our case, the AISI 320 standard stainless steel containing 1.75% nickel was used for the working head, and SAE 4140 alloy steel was selected for the guide section.

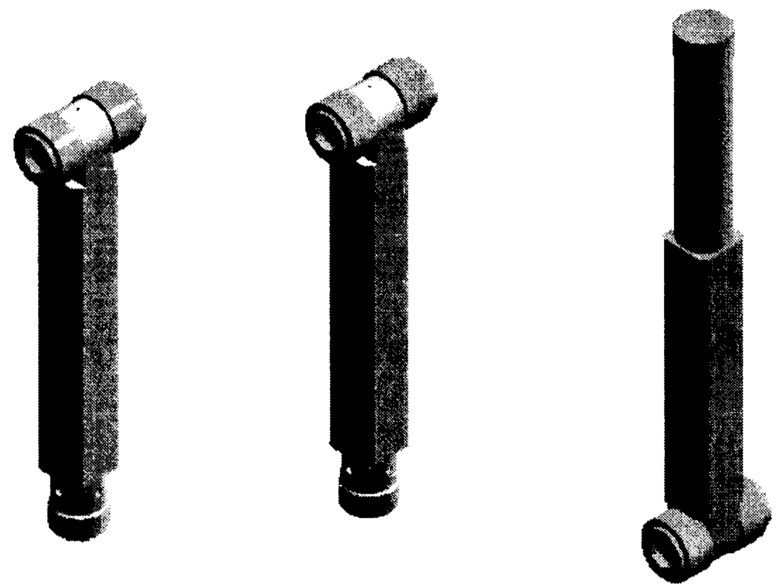


Figure 4.8 Upper and Lower Punches Designed for the 5.4-inch Rotary Press

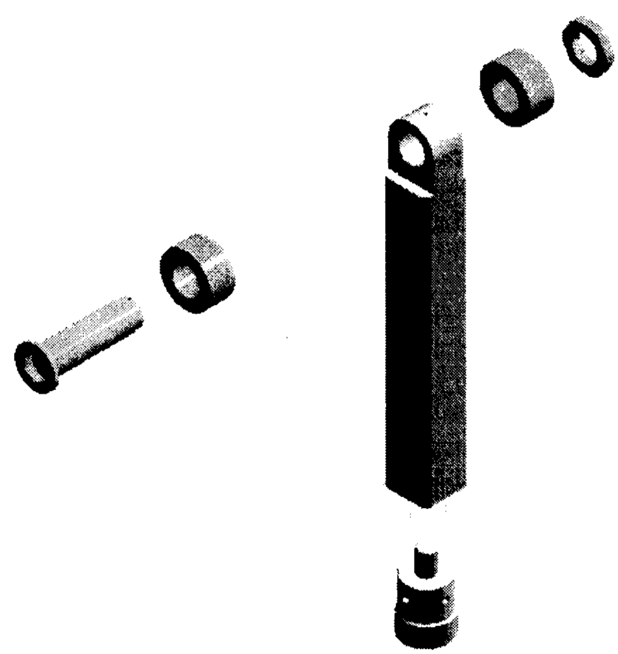


Figure 4.9 Exploded View of Upper Punch Designed for the 5.4-inch Rotary Press

4.24 Cam Design

Cam drive is commonly used for rotary die presses that feature a series of punches and dies arranged in a common, rotating turret. The stationary axis around which the turret rotates provides a fixed reference point for mounting the cams. Cam drives are used to move punches. Pressing speed, timing, and motion are adjustable by changing the contours of the cams or cam inserts. Cams are needed to guide both the upper punch and the lower punch of each working position to perform filling, weighting, compaction, and ejection. The design of the back pressure cam and the ejection cam depends on knowing the relationship between the compacting force and displacement. A plot of compaction force vs displacement was obtained from existing laboratory tests (Fig. 3.2). The cam calculation and the cam curve design were carried out in Chapter 3. The cam parts design will be shown in this chapter.

The lower cam group design is shown in Fig. 4.10. The group is installed on a

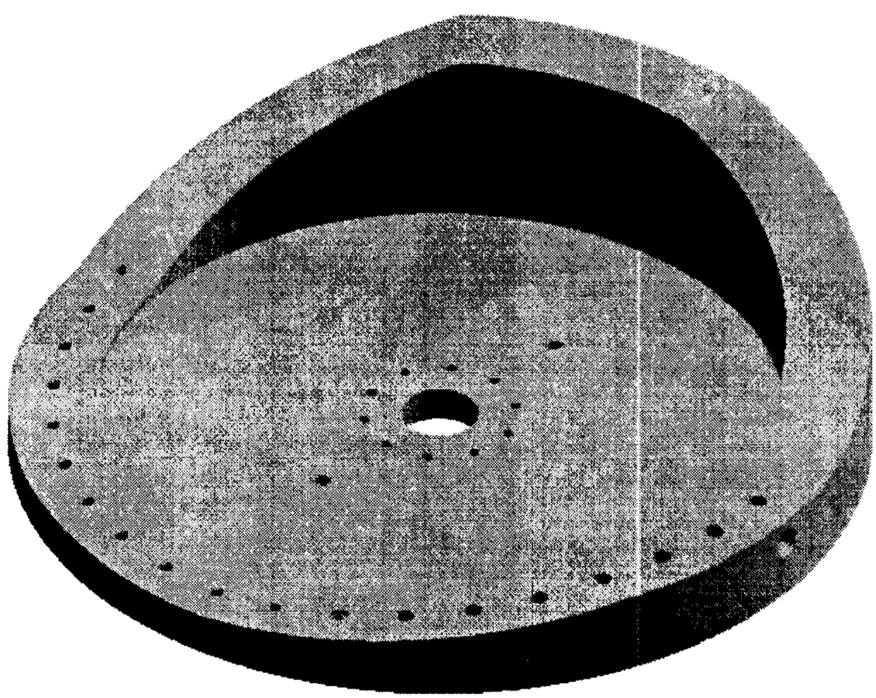


Figure 4.10 Lower Cam Group

three-inch chick steel board by bolts. There are four pieces of cams in the group: The first piece in the group is the feeding cam, which is made of one-inch-thick PVC board to reduce noise. The second piece is the compaction cam (Fig 4.13), which is made of steel. Its one end is sitting on a disc spring, and it is connected to the base board by a pair of slot joints and bolts. The half hole on its back is for over-load disk spring (Fig 4.15). The third cam is the compacting dwell cam (Fig 4.14); it is supported by three groups of over-load springs, and the holes on its back (Fig 4.16). It has the same function as the compacting cam. The fourth cam is the ejection cam; it is a steel welded structure with a height of 21 inches. This cam has three functions: 1) the rising section is for ejecting logs, 2) the leveled section is for unloading logs, and 3) the down-hill section is for loading material (see Fig 4.17).

Figure 4.11 Feeding Cam

Figure 4.12 Ejection Cam

Figure 4.13 Compacting Cam

Figure 4.14 Dwell Cam

Figure 4.15 The Bottom View of Compaction Cam

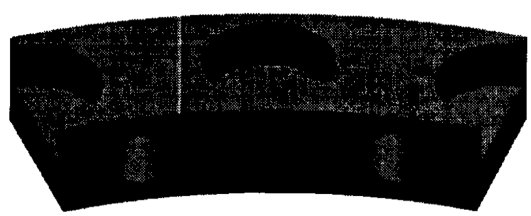


Figure 4.16 The Bottom View of Dwell Came

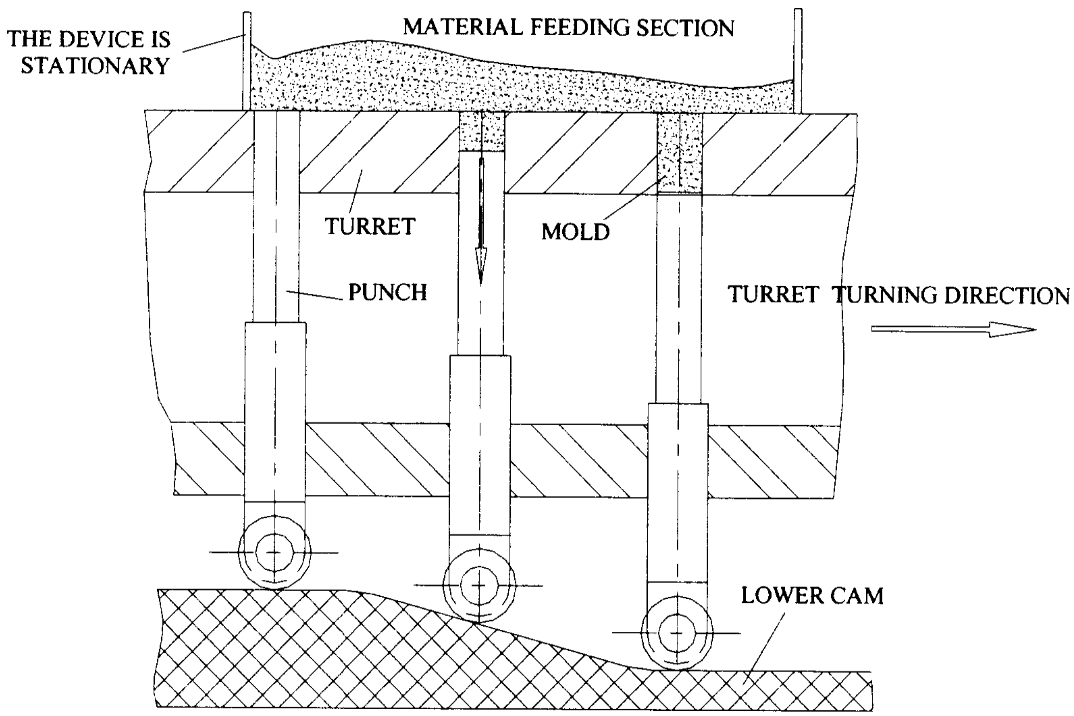


Figure 4.17 Lower Cam in Feeding Section

In the material feeding section, the upper cam is designed to have a simple leveled shape. After ejection section, the cam contacts only one roller with each punch, see Fig 4.18. To keep the punch at the highest level until the material filling is over, and then the cam let the punches down on a 40-degree slope to reach the mold for log compaction. The upper cam group design is shown in Fig 4.19, which includes 5 pieces of cams and a cam frame. The cam frame is fixed on the top of the machine frame. The cams that are in the compaction section, dwell section, and ejection section are fastened to both frames.

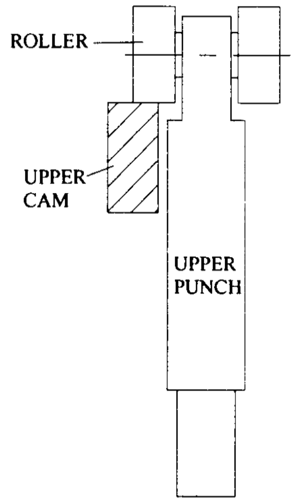


Figure 4.18 Upper Cam in Feeding Section

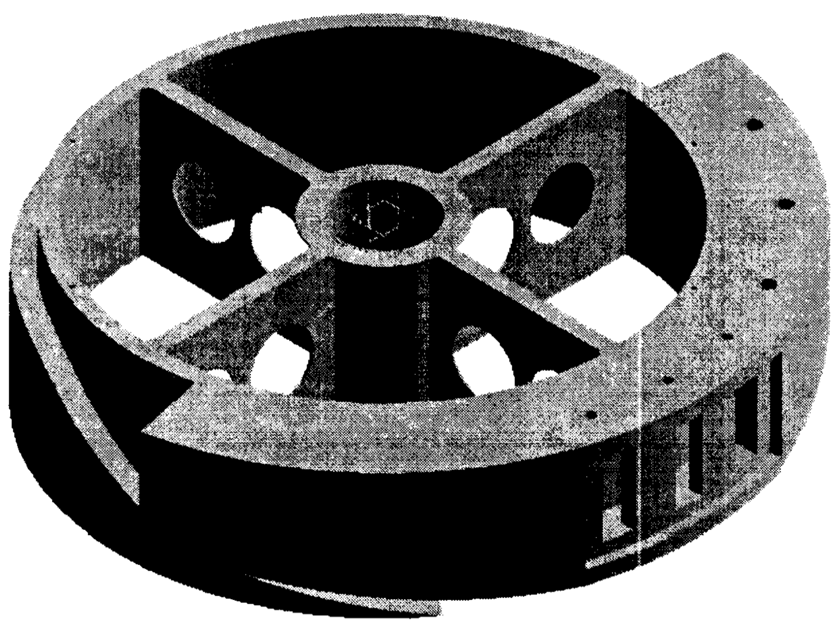


Figure 4.19 The Upper Cam Group of the 5.4-inch Diameter Mold Rotary Press

In compaction section, the slopes of upper cam curve and the lower cam curve are the same which is 5 degree. The horizontal arc length of the compaction cams is63.5 inch, the angle of the arc is 80.8, the length of stroke is 5 inch. The material of the cams is 1020 carburized steel, the surface hardness is 50-58 Rc.

The cam are shown in Figure 4.20.

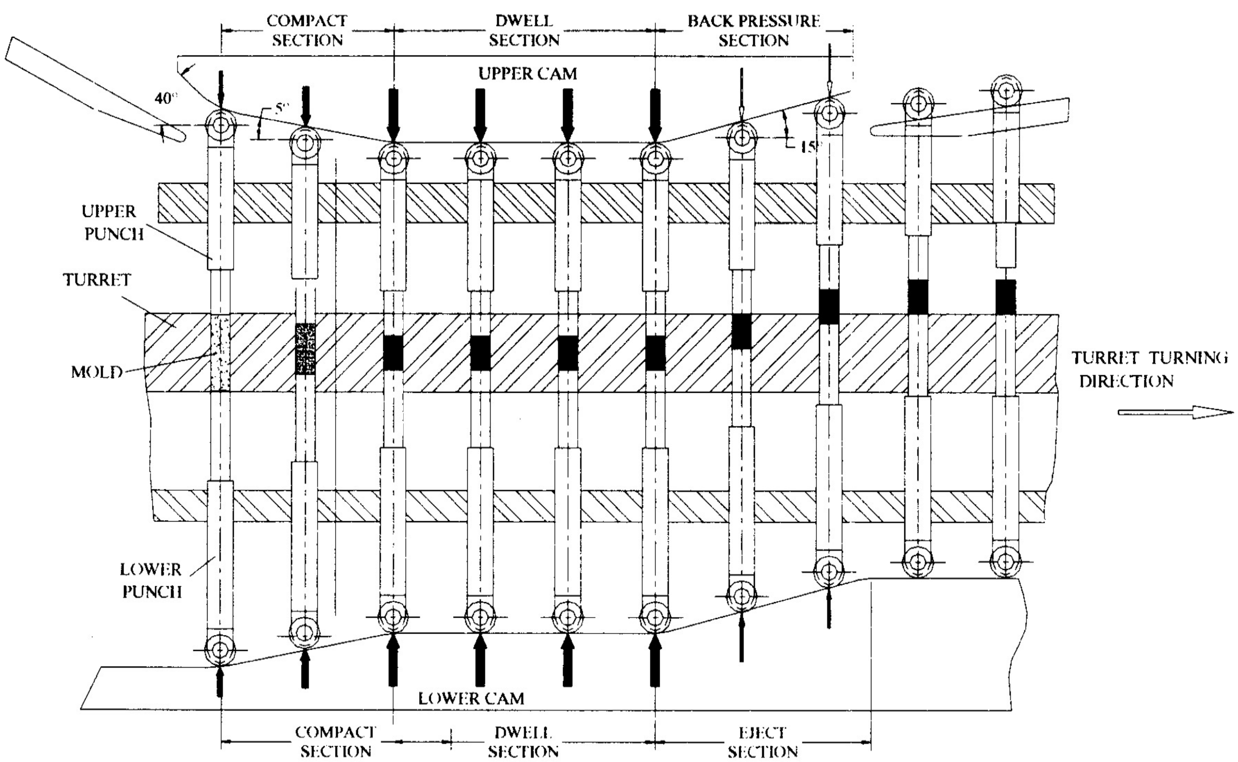


Figure 4.20 Cams

After compaction, the coal log will bounce back a small amount. Thus, the ejection cam starts working at a height of 6 inch, where the upper punches and the coal log are still in contact with each other at the end of the compaction. In order to push the coal log out of the mold smoothly with low resistance, the ejection cam rising angle was designed to be 15 degree. The punch moves up for 0.01 inch above the turret surface and stays at this position for about 5 sec; then it goes back to the initial position. The reason upward ejection is to reduce the overall height of the machine and thickness of the turret. Because the spacing designed for the feeding system can be used for ejecting coal logs as well.

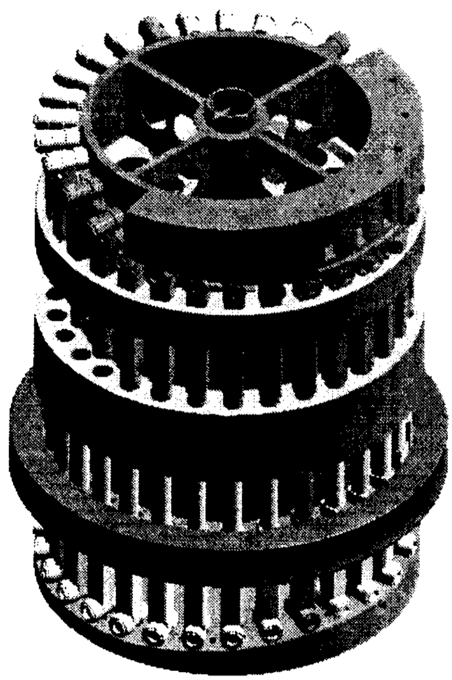


Figure 4.21 Compaction System Designed for the 5.4-inch Rotary Press

4.3 Product Collection System

After ejection, the logs are collected by the log collection system. This system has an orientation device and a belt conveyer. The orientation device is a screw shaped wall of sheet metal. It has two functions. One is to take the logs out of the rotary press, and the other is to turn the logs from vertical direction to horizontal direction on the belt conveyer.

4.4 Overload Protection Design

The overload protection system (see Figure 4.22) protects the machine from damage due to overloading the dies. This is done by dropping the lower compacting cam or dwell cam. There are three kinds of overload protection devices -- spring, air and hydraulic -- used in rotary press machines. In the present case, we chose spring overload protection device and used Belleville springs in order to save space and material. The Belleville springs directly supports the lower cams and permits it to move down to release the load when the machine is overloaded.

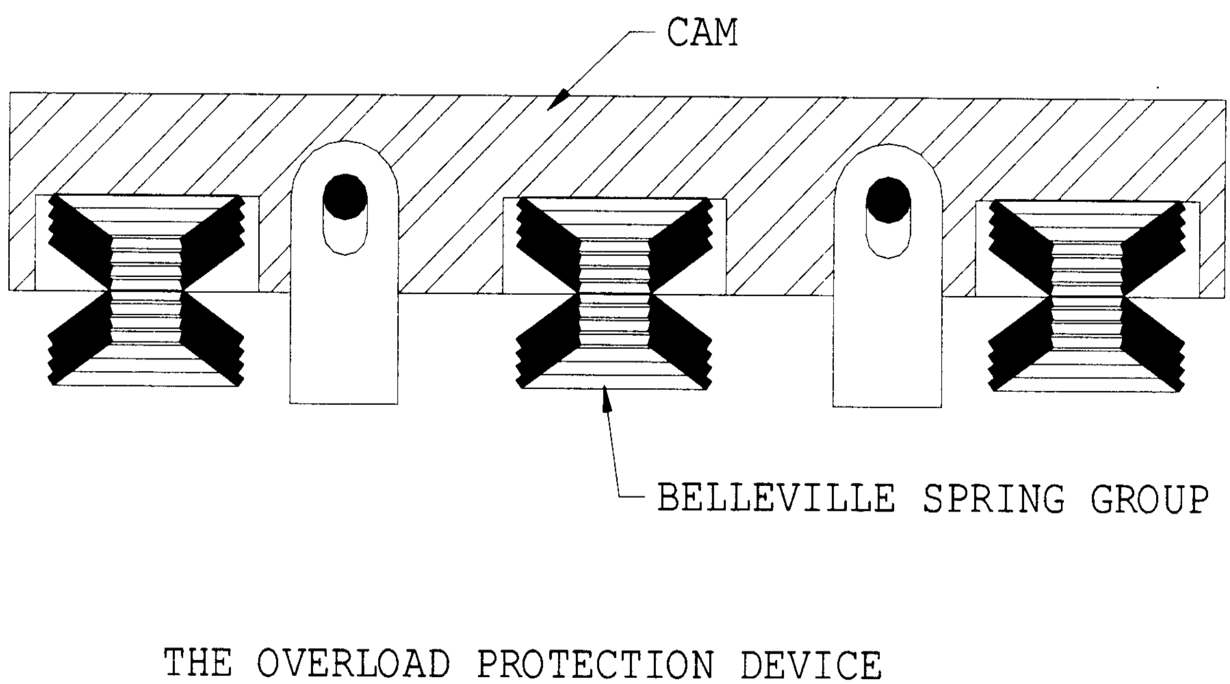


Figure 4.22 Over Load Protection Device

The supporting pressure exerted by the spring can be adjusted by a hand-wheel, which has a capacity of about $4 \times 250 = 1000$ tons of the total load. The lower dwell cam is guided by two joints (Joints 1 and 2). They are welded on the front right and left side of the base.

4.5 Frame Design

The frame design for this rotary press was complicated. The frame design must be: 1) strong enough to withstand the huge bending and tension forces, 2) properly to fit the machine operating components, connection, and parts replacing requirements, 3) simple, 4) easy to be manufactured and assembled. Therefore, the frame chosen has been designed very carefully in each step. After several design cycles, the frame was a box structure with bolted bridge (Fig. 4.23). It is a group of three steel structures: the base, the column, and the cap. The driving motor, speed reducer, overload protection device, lower cam, and shaft bearing is mounted on the base. The upper cams and the upper shaft bearing are fixed on the cap. The feeding device and the log collection device are installed on the columns. Finally, the base and the cap are connected by the columns. After assembly, the 5.4-inch diameter mold rotary log press is shown in Fig. 4.24.

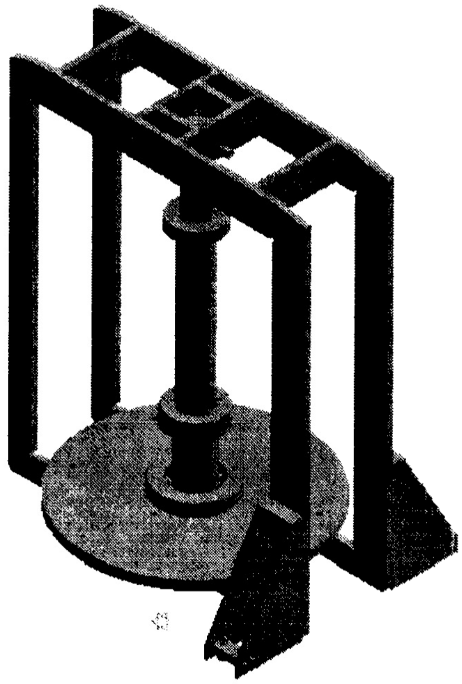


Figure 4.23 Frame of the 5.4-inch Diameter Mold Rotary Press

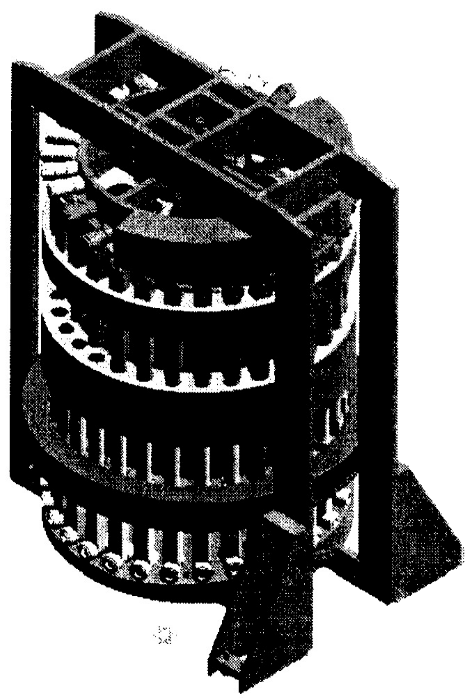


Figure 4.24 Design of the 5.4-inch Diameter Mold Rotary Press

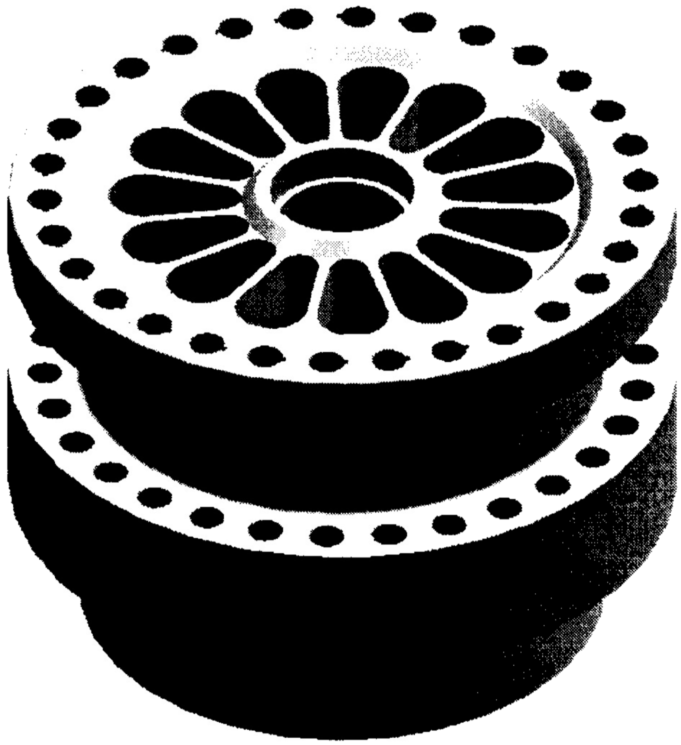


Figure 4.25 Turret with Key Way Guidance for Cylinder Shape Punches Design

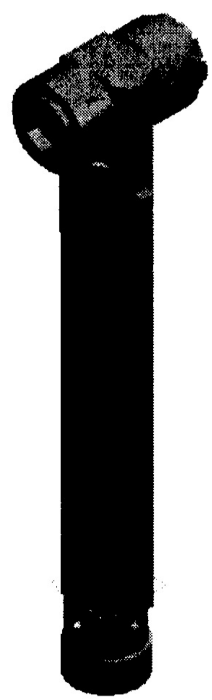


Figure 4.26 Cylinder Shape Punch Design

Chapter 5

COST ESTIMATE OF THE ROTARY PRESS

After the preliminary design of the 5.4-inch-disameter mold rotary press has been completed, cost estimate is the next step. The cost estimate for machine designed is important because it impacts critically the coal log production cost. The purpose of cost estimating in general includes (Winchell, 1989):

1. Establish the bid price of a product for a quotation or contract.
2. Verify quotations submitted by suppliers.
3. Ascertain whether a proposed product can be manufactured and marketed profitably.
4. Provide data for make-versus-buy decisions.
5. Help determine the most economical method, process, or material for manufacturing a product.
6. Costs at the beginning of a project.
7. Help in evaluating the design.

There are three basic elements in product cost estimate:

a) Purchased parts and components
b) Manufactured parts and components
c) Assembly

These three elements of the product cost estimate for the rotary press designed here were carried out as follows:

5.1 Purchased Parts and Components

About 20% of the components (base on the cost)in this design are purchased parts. Their prices are listed in Table 5.1. The price in Table 5.1 were obtained from Internet, product catalogs, and in certain cases, directly from venders.

Table 5.1 Prices of Purchased Parts for the 5.4" Rotary Press (April 1999 Prices)

Item	Description	Qty	Unit Price ($)	Price ($)
1	RBY8 Roller Bearing d/D=3.25/8 (BEARINGS, INC.)	120	300	36,000
2	TP Bearing 160 TP165 (TORRINGTON COMPANY)	1	450	450
3	Spherical Roller Bearing 23052 YM (TORRINGTON COMPANY)	2	200	200
4	Vertical Screw Feeder	1	3,500	3500
5	AC Electric Motor 300 hp	2	9,650	19,300
6	Speed Reducer (JONES:330/DH)	2	8,900	17,800
7	Back Pressure Spring	4	5	20
8	Over Load Spring	4	45	180
9	Lubrication Component	2	75	150
10	Fasteners			1,000
Total				$78,600

5.2 Manufactured Parts and Components

Because no commercial rotary press exists for the size needed for coal log compaction, many parts and components, such as the turret, cams, punches and frame, must be fabricated in a suitable machine shop that can handle such large components. The greatest cost of the rotary press exists here. There are three factors in the manufactured part cost estimation, 1) the price of raw material, 2) the labor cost, and 3) the machining cost. Fabrication of these parts are expensive because of the large size and small quantity.

The cost estimating procedure is as follows:

1) Prepare detail part drawings with types of materials and weights listed on the drawing

2) Gathering prices of the raw materials used.

3) Analyze and calculate the time needed for setting up each machining step.

5) Add total machining time needed for each part.

6) Calculate labor cost for each part.

7) Add the costs and list the results in a table

Based on the foregoing procedure, the results of estimated costs for the 5.4" rotary press is shown in Table 5.2. the estimates were made by Mr. Orral (Rex) Gish, Supervison, Mechanical & Aerospace Engineering Shop, UMC.

Table 5.2 Cost of Machined Parts for the 5.4" Rotary Press

#	Description	Qty	Weight (lb.)	Mater -ial	Unit Price of Mater -ial ($/lb.)	Machin- ing (hour)	Cost of Labor+ mach- ining ($/h)	Sub-total ($)
1	Upper Pistons	30	356	1030	1.00	10.2	45	19,251*
2	Lower Pistons	30	413	1030	1.00	12	45	25,507
3	Turret	1	39,429	Cast iron	0.30	163+50	65	25,674
4	B-turret	1	14,139		0.30	63+40**	65	10,937
5	Frames	2	5,802	1020	0.80	12	65	5,422
6	Shaft	1	5,872	1020	0.80	65	65	8,923
7	UC-cam	1	550	1040	1.20	10	65	1,310
8	UD-cam	1	600	1040	1.2	6	65	1,110
9	UB-cam	1	367	PVC	1.3	5	65	802
10	UE-cam	1	280	PVC	1.3	5	65	689
11	UF-cam	1	277	PVC	1.3	5	65	685
12	LF-cam	1	39.5	PVC	1.3	2.5	45	164
13	LC-cam	1	498	1040	1.2	5.4	65	948
14	LD-cam	1	432	1040	1.2	3	65	713
15	LE-cam	1	1,000	1020	0.8	5	65	1,080
16	Base plate	1	8,822	1020	0.8	7	65	7,512
17	Shoulder	1	182	1020	0.8	5	65	470
18	Supporters	2	1,006	1020	0.8	8	65	2,650
19	Frame spacers	4	295	1020	0.8	3	65	1,724
20	Shift spacer	1	98	1020	0.8	2	65	208
21	UC-cam frame	1	1300	1020	0.8	20	65	2,340
22	Worm gear	1	7028	Bronz	4	24	65	29,672
23	Worm	1	2450	3310	2.5	10	65	6,775
24	Cap	1	500	1020	0.8	6	65	790

* This number is for 30 pieces.

** The 40 hour is the time estimated for casting.

The machining and lab hour in Table 5.2 are for the first work pieces. If the quantity of the same part is more than one, the Learning Curve Formula (Eq.(5.3)

Winchell, 1989) should be used for direct labor cost estimate. The 30 upper piston cost estimate calculation is shown as follows:

$$C_N = C_1 * N^{3.322*\log P} \tag{5.3}$$

Where C_N = the cost of unit x,

C_1 = the cost of the first unit produced,

N = the number that unit N is in the sequence, and

P = a fixed proportion of direct labor that each doubled quantity requires.

For part #1: $C_1 = 10.2$, $N = 30$, $P = 0.95$

The value of P is different for each company and product. A commonly used value of P is 95 percent (Winchell, 1989).

Thus, $C_N = C_{30} = 10.2 \times 30^{3.322*\mathrm{Log}0.95}$, and the total cost of 30 upper punches is

$$30 \times (356 \times 1.00 + 10.2 \times 30^{3.322*\mathrm{Log}0.95}) = 19{,}251\ (\$).$$

5.3 Assembly

The final stage of product fabrication is assembly and test. Because the 5.4" rotary press is a large machine, "once it is tested at the factory, it will be disassembled and shipped in several pieces to the user"(Liu, 1999). After delivery, it needs reassembling and testing again. Therefore, this assembly cost estimate should include assembly, disassembly, delivery, and installation.

5.3.1 Assembly and Disassembly

The cost estimate for assembling and disassembling is not accurate number at this stage. It is based on a 160 hours of labor for assembly, and 80 hours of labor for disassembly, using a crew of two skilled workers. The cost of assembly and disassembly is (160+80)*65 = 15,600 ($).

5.3.2 Delivery and Installation

The delivery and installation cost is estimated to be 5% of total cost of purchased parts and components, manufactured parts and (Liu, on group meeting. April 1999) components and the assembly cost. For the 5.4" rotary press it is

$$(78{,}600 + 156{,}256 + 15{,}600) \times 5\% = 12{,}523 \ (\$)$$

5.4 Result

The total product cost estimation for the 5.4" rotary press is

$$78{,}600+156{,}256+15{,}600+12{,}523=262{,}979 \ (\$).$$

Chapter 6

CONCLUSIONS

The well-known Coal Log Pipeline (CLP) technology requires innovative compaction machine to make strong coal logs. Therefore, the research and development on coal log fabrication machine is an important part of the CLP technology. After the successful field tests of the CLP technology in a 5-mile commercial underground pipeline in Conway, Kansas, a second generation coal log and biomass compaction machine was conceived based on the rotary press concept. The advantage of using rotary press design for coal log compaction machine is to increase the output rate without reducing the compaction time for each coal log.

In this thesis, the preliminary design of a commercial scale rotary press for the coal log and biomass compaction is completed. In this design, the compaction system is a double action cam-follower compaction system, which includes 30 pairs of upper and lower punches, 30 molds and a turret. The critical components included in this design are the cams, the punches, the turret, the feeding system, and the power train. To reduce noise, vibration and wear, the cam profiles between linear sections were designed based on a cycloidal motion curve, which does not introduce abrupt changes in acceleration. There are three features which the conventional rotary press does not has: 1) 3 second maximum pressure dwell time, 2) 100 psi ejection back pressure, 3) the energy save compaction system.

The time required to fill the mold was calculated based on the actual experimental data of coal flow properties. The compaction time was determined based on the pressure-displacement curve during compaction of 5.4" coal log using the 250 ton hydraulic press. To handle heavy load, long stroke, and slow rotation, the *Parabolic Motion* cam curve was chosen to be used at the start and the end of compact section and eject section.

The turret is 102 inches in diameter and 21 inches in height. The maximum turret driving power was calculated to be 584 HP. A mechanical engineering design software, TK solver, was used for the worm gear calculation and optimization. To protect the machine from damage due to overloading the dies, a spring overload protection device was also designed. The cost estimation was conducted, and the total production cost for this rotary press is $262,979. This cost includes manufacturing, assembling, disassembling, delivery, and installation.

REFERENCES

Alec, S. 1992. *Gear Handbook Design and Calculation*, Butterworth-Heinemann.

Armstrong, N. A. and Palfrey, L. P. 1987. Punch velocities during the compaction process. *J. Pharm. Pharmacol.*, 39: 497-501.

Charlton, B. and Newton, J. M. 1984. Theoretical estimation of punch velocities and displacements of single-punch and rotary tablet machines. J. *Pharm. Pharmacol.*, 36: 645-651.

Cherry -Burrell Corporation, 1969. *Cherry-Burrell Tablet Presses*

Cooper, J. and Rees, J. E. 1972. Tableting research and technology. *Journal of Pharmaceutical Sciences,* 61: 1511-1555.

Crawford, R. J., Cumberland, D. J., and Keating, T. G. 1989. Design of a multi-spring floating die compaction system for plastic powders, *Journal of Testing and Evaluation,* 17: 344-356.

Green, R. E., and McCauley, C. J. 1996. *Machinery's Handbook*, 25th ed., Industrial Press Inc., New York.

Kuhn, H. H. 1978. Optimum die design for powder compaction. *International Journal of Powder Metallurgy & Powder Technology*, 14: 259-275.

Lin, Y., Wen, G., Li, H., and Xue, K. 1996. *The design of a 300-ton hydraulic press for high speed compaction of coal logs*. Presented at the 1996 ASME International Mechanical Engineering Congress & Exhibition, Atlanta, GA. November 17-22.

Lin, Y., Wen, G., Li, H., and Xue, K. 1997a. Compacting coal into logs. *Mechanical Engineering*, November, pp. 84-86.

Lin, Y., Marrero, T., Xue, K., and Zhang, M. 1997b. Research and design of a rotary press for the fast compaction of powdered materials. Presented at 15th Int'l Conf. On Automation in Warehousing (ICAW) & 2nd Int'l Conf. On Material Handling (ICMH), ??.

Liu, H. March, 1999, US Patent 5,879,421.

Liu, H. April, 1999, speech on machine design group meeting of CPRC at UMC.

Liu, H. 1997. CPRC Quarterly Report.

Maekawa, H., Sakamoto, T., and Shimada, Y. 1971. U. S. pat.3,555,608.

Marshall, K. and Rudnic, E. M. 1990. Tablet dosage forms. *Modern Pharmaceutics*. ed. by G. S. Banker and C. T. Rhodes. Marcel Dekker Inc., New York.

Orr, C. Jr. 1966. *Particulate Technology*. Macmillan Co., New York.

Pietsch, W. 1991. *Size Enlargement by Agglomeration.* John Wiley & Sons, New York.

Powder Metallurgy Equipment Association. 1977. *Powder Metallurgy Equipment Manual.*

Ridgeway-Watt, P. 1988. *Tablet Machine Instrumentation in Pharmaceutics.* Wiley, New York.

Snyder, G. 1967. High volume production of P/M parts. *International Journal of Powder Metallurgy*. 3:33-43.

Swartz, C. J. 1969. Compressing machinery and tooling technology. *Dug Cosmet. Ind.,* 105 (3): 51-56,160.

Swartz, C. J. and Anschel, J. 1968. Evaluation of tableting tool life records. *Journal of Phamaceutical Sciences*, 57: 1779-1782.

Swartz, C. J., Weinstein, S., Windheuser, J., and Cooper, J. 1962. A punch and die control program and its contribution to tableting technology. *Journal of Pharmaceutical Sciences,* 51:1181-1186.

Winchell, W.1989. *Realistic Cost Estimation for Manufacturing*, 2nd edition, Society of Manufacturing Engineers, Michigan.

Xue, K. 1995a. Conceptual design of a 30 ton rotary press for fast compaction of 1.9 inch coal logs.

Xue, K., July, 1995b. CPRC internal report.

APPANDEX

Table 3.5 Laboratory Data from Testing of 250-Ton Hydraulic Press (May 28,1998. By Guoping Wen)

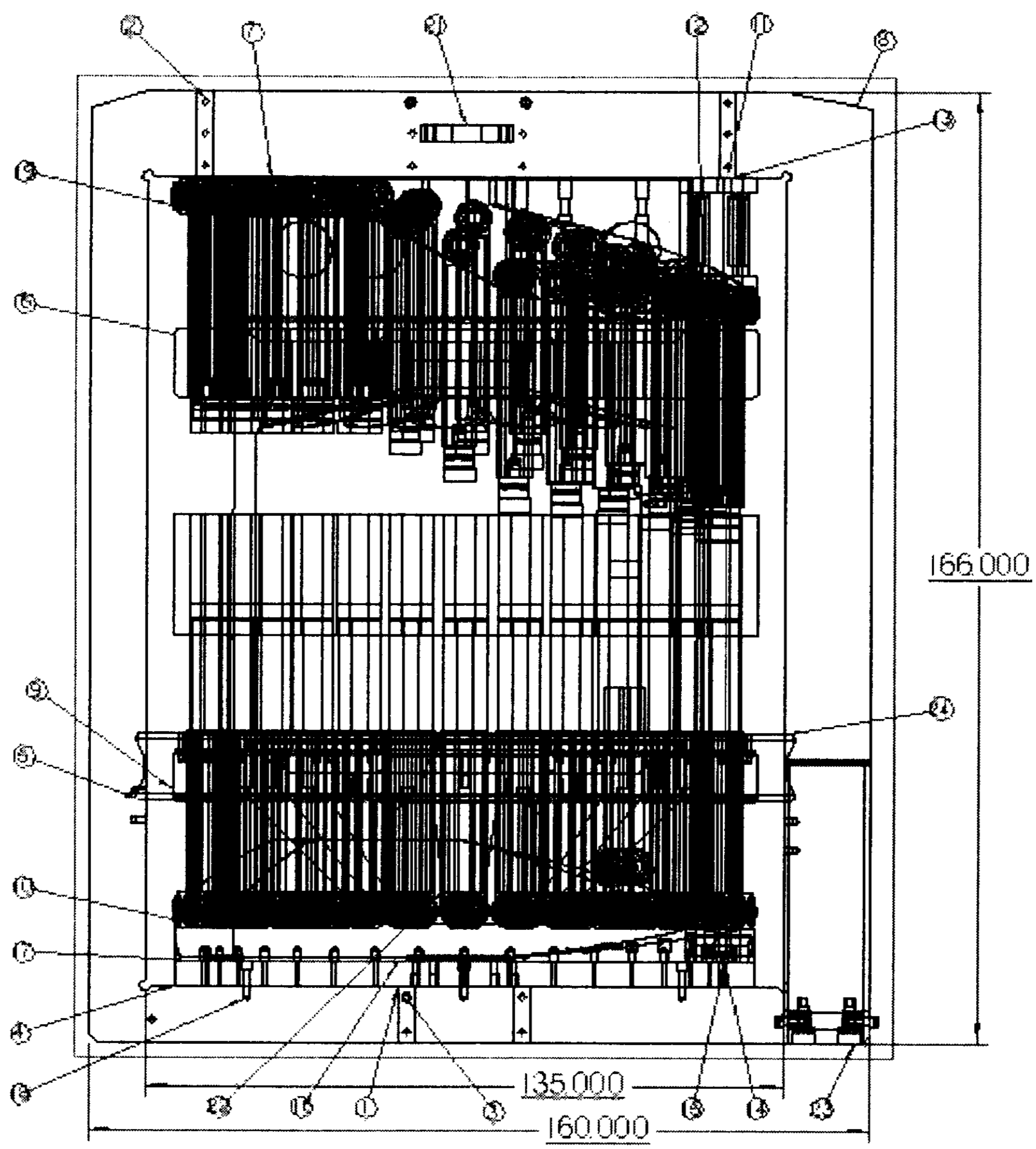
Bottom cylinder pressure (Psi)	Bottom piston displace- -ment(in)	Top piston displace- -ment(in)	Top cylinder pressure (Psi)	Bottom piston speed (in/s)	Top piston speed (in/s)
78.96	4.18	1.35	25.58	0.02	0.04
78.79	4.18	5.7	49.98	0.02	15.68
78.79	4.18	5.7	49.98	0.02	15.68
78.96	4.18	9.02	81.31	0.02	15.73
78.96	4.18	9.02	81.31	0.02	15.73
78.96	4.18	11.62	111.97	0.02	15.73
78.96	4.18	11.62	111.97	0.02	15.73
77.77	4.18	15.56	88.73	0.02	9.29
77.77	4.18	15.56	88.73	0.02	9.29
77.6	4.18	17.06	68.78	0.02	9.13
77.6	4.18	17.06	68.78	0.02	9.13
83.03	4.36	19.03	58.72	2.01	6.28
83.03	4.36	19.03	58.72	2.01	6.28
85.74	4.86	20.21	48.67	2.39	5.39
85.74	4.86	20.21	48.67	2.39	5.39
89.3	5.39	21.35	42.89	2.39	5.31
89.3	5.39	21.35	42.89	2.39	5.31
94.9	6.01	22.55	41.41	2.47	3.81
94.9	6.01	22.55	41.41	2.47	3.81
106.6	6.49	22.95	47.35	2.72	0.95
106.6	6.49	22.95	47.35	2.72	0.95
132.73	6.96	22.99	59.55	2.62	0.22
132.73	6.96	22.99	59.55	2.62	0.22
255.03	7.6	23.02	101.26	2.08	0.12
375.8	8	23.02	166.87	1.68	0.08
375.8	8	23.02	166.87	1.68	0.08
461.8	8.23	23.02	219.3	1.41	0.08
543.05	8.41	23.02	261.34	1.17	0.09
543.05	8.41	23.02	261.34	1.17	0.09
643.81	8.61	23.02	313.93	0.8	0.11
643.81	8.61	23.02	313.93	0.8	0.11
724.55	8.73	23.02	368.82	0.46	0.08
724.55	8.73	23.02	368.82	0.46	0.08
766.11	8.8	23.02	393.39	0.38	0.08
766.11	8.8	23.02	393.39	0.38	0.08
833.45	8.88	23.02	445.98	0.33	0.12
991.03	8.96	23.22	725.91	0.55	3.2
991.03	8.96	23.22	725.91	0.55	3.2
1427.6	9.08	23.51	1423.6	1.5	2.13
2188.7	9.25	23.71	2253.3	0.42	0.63
2188.7	9.25	23.71	2253.3	0.42	0.63

2976.5	9.32	23.81	3123.1	0.33	0.46
2976.5	9.32	23.81	3123.1	0.33	0.46
3382.9	9.36	23.86	3570.4	0.16	0.28
3382.9	9.36	23.86	3570.4	0.16	0.28
3425	9.37	23.87	3615.7	0.02	0.1
3425	9.37	23.87	3615.7	0.02	0.1
3400.9	9.37	23.87	3590.2	0.02	0.09
3400.9	9.37	23.87	3590.2	0.02	0.09
3385.9	9.37	23.87	3573.8	0.02	0.1
3385.9	9.37	23.87	3573.8	0.02	0.1
3369.5	9.37	23.87	3560.6	0.02	0.08
3369.5	9.37	23.87	3560.6	0.02	0.08
3359.3	9.37	23.88	3550.4	0.02	0.09
3359.3	9.37	23.88	3550.4	0.02	0.09
3350.7	9.37	23.88	3541.2	0.02	0.08
3350.7	9.37	23.88	3541.2	0.02	0.08
3342	9.38	23.87	3531.6	0.02	0.08
3342	9.38	23.87	3531.6	0.02	0.08
3335.6	9.38	23.88	3524.7	0.02	0.08
3335.6	9.38	23.88	3524.7	0.02	0.08
3329.5	9.38	23.88	3517.8	0.02	0.08
3329.5	9.38	23.88	3517.8	0.02	0.08
3324.5	9.37	23.88	3511.5	0.02	0.08
3324.5	9.37	23.88	3511.5	0.02	0.08
3319.4	9.38	23.88	3505.1	0.02	0.08
3319.4	9.38	23.88	3505.1	0.02	0.08
3314.9	9.38	23.88	3498.5	0.02	0.08
3314.9	9.38	23.88	3498.5	0.02	0.08
3310.6	9.38	23.88	3492.6	0.03	0.09
3310.6	9.38	23.88	3492.6	0.03	0.09
3306.7	9.38	23.88	3487.8	0.02	0.08
3306.7	9.38	23.88	3487.8	0.02	0.08
3306.7	9.38	23.88	3487.8	0.02	0.08
3302.7	9.38	23.88	3482	0.02	0.08
3302.7	9.38	23.88	3482	0.02	0.08
3302.7	9.38	23.88	3482	0.02	0.08
3294.2	9.38	23.88	3474.9	0.02	0.08
3294.2	9.38	23.88	3474.9	0.02	0.08
3288.6	9.38	23.88	3469.1	0.02	0.08
3288.6	9.38	23.88	3469.1	0.02	0.08
3284.3	9.38	23.88	3463.9	0.02	0.08
3284.3	9.38	23.88	3463.9	0.02	0.08
3281.1	9.38	23.88	3460.7	0.02	0.08
3281.1	9.38	23.88	3460.7	0.02	0.08
3261.3	9.38	23.88	3448.4	0.02	0.08
3261.3	9.38	23.88	3448.4	0.02	0.08
3007.7	9.37	23.88	3191.4	0.09	0.15
3007.7	9.37	23.88	3191.4	0.09	0.15
1206.4	9.31	23.84	1548	0.38	0.38
1206.4	9.31	23.84	1548	0.38	0.38
258.08	9.24	23.77	368.82	0.21	0.4
258.08	9.24	23.77	368.82	0.21	0.4
134.59	9.21	23.71	123.68	0.09	0.23

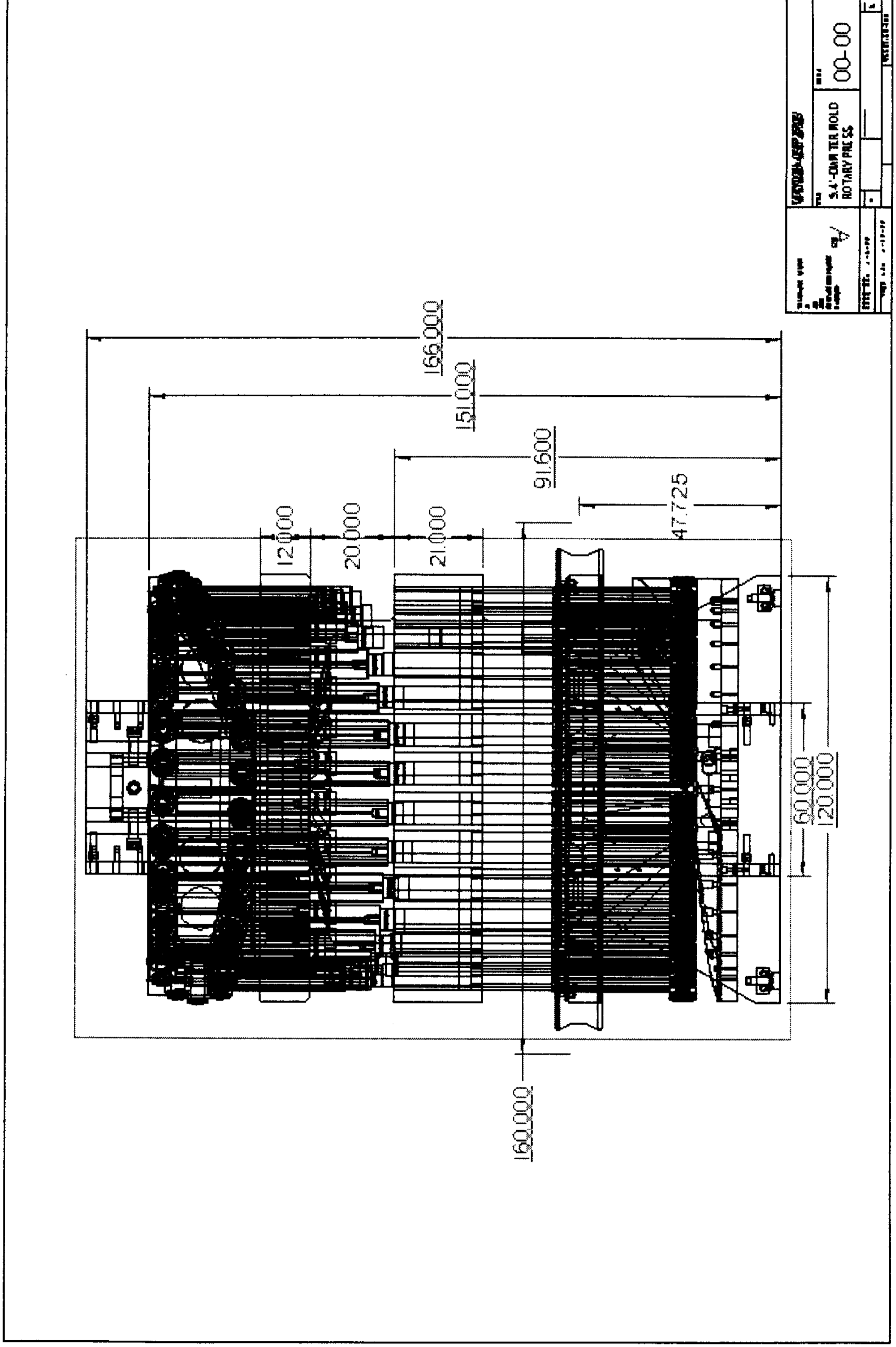
134.59	9.21	23.71	123.68	0.09	0.23
134.59	9.21	23.71	123.68	0.09	0.23
668.24	9.3	23.56	170.33	1.7	1.91
668.24	9.3	23.56	170.33	1.7	1.91
702.67	10.12	22.76	918.31	2.49	2.74
702.67	10.12	22.76	918.31	2.49	2.74
669.59	10.72	22.04	456.7	2.61	3.77
669.59	10.72	22.04	456.7	2.61	3.77
682.82	11.26	21.19	360.58	2.61	4.29
682.82	11.26	21.19	360.58	2.61	4.29
710.98	11.82	20.29	257.21	2.53	4.21
710.98	11.82	20.29	257.21	2.53	4.21
749.66	12.41	19.36	214.51	2.53	4.17
749.66	12.41	19.36	214.51	2.53	4.17
772.22	12.96	18.48	191.1	2.56	4.17
772.22	12.96	18.48	191.1	2.56	4.17
792.57	13.39	17.81	177.42	2.5	4.16
792.57	13.39	17.81	177.42	2.5	4.16
819.88	14.1	16.7	162.58	2.56	4.15
845.33	14.63	15.86	155.17	2.49	4.16
845.33	14.63	15.86	155.17	2.49	4.16
839.9	15.2	14.99	147.75	2.58	4.14
839.9	15.2	14.99	147.75	2.58	4.14
839.05	15.57	14.4	144.12	2.55	4.14
839.05	15.57	14.4	144.12	2.55	4.14
835.83	16.06	13.65	139.83	2.59	4.13
835.83	16.06	13.65	139.83	2.59	4.13
841.25	16.63	12.75	136.7	2.5	4.15
841.25	16.63	12.75	136.7	2.5	4.15
824.46	17.18	11.89	133.24	2.58	4.15
824.46	17.18	11.89	133.24	2.58	4.15
821.41	17.62	11.22	130.93	2.49	4.13
821.41	17.62	11.22	130.93	2.49	4.13
756.27	18.33	10.12	127.96	2.58	4.13
672.31	18.89	9.27	125.49	2.6	4.12
672.31	18.89	9.27	125.49	2.6	4.12
615.99	19.27	8.68	123.68	2.56	4.15
549.5	19.65	8.11	122.85	2.61	4.14
549.5	19.65	8.11	122.85	2.61	4.14
477.24	20.18	7.3	121.86	2.69	4.14
422.11	20.61	6.64	121.04	2.7	4.13
422.11	20.61	6.64	121.04	2.7	4.13
377.33	21.04	5.99	119.72	2.72	4.13
341.54	21.44	5.37	119.39	2.62	4.13
341.54	21.44	5.37	119.39	2.62	4.13
314.4	21.79	4.84	118.24	2.58	4.13
314.4	21.79	4.84	118.24	2.58	4.13
284.2	22.36	3.97	117.25	2.6	4.15
284.2	22.36	3.97	117.25	2.6	4.15
241.97	23.02	2.96	115.27	2.61	4.15
241.97	23.02	2.96	115.27	2.61	4.15
211.77	23.55	2.14	116.42	2.6	4.17
211.77	23.55	2.14	116.42	2.6	4.17

179.54	24.21	1.14	116.42	2.61	4.12
179.54	24.21	1.14	116.42	2.61	4.12
179.54	24.21	1.14	116.42	2.61	4.12
149.18	24.86	0.16	112.3	2.58	3.88
149.18	24.86	0.16	112.3	2.58	3.88
1078.3	25.97	0.12	53.94	4.28	0.08
1078.3	25.97	0.12	53.94	4.28	0.08
1078.3	25.97	0.12	53.94	4.28	0.08
2973.6	26.01	0.12	41.08	0.03	0.08
2973.6	26.01	0.12	41.08	0.03	0.08
3001.9	26.01	0.12	33.17	0.02	0.08
3001.9	26.01	0.12	33.17	0.02	0.08
3001.4	26.01	0.12	28.22	0.02	0.08
3001.4	26.01	0.12	28.22	0.02	0.08
3001.4	26.01	0.12	28.22	0.02	0.08
3001.2	26.01	0.12	24.6	0.02	0.08
3001.2	26.01	0.12	24.6	0.02	0.08
3000.9	26.01	0.12	21.96	0.02	0.08
3000.9	26.01	0.12	21.96	0.02	0.08
3000.7	26.01	0.12	19.98	0.02	0.08
3000.7	26.01	0.12	18.83	0.02	0.08
3000.7	26.01	0.12	18.83	0.02	0.08
3000.2	26.01	0.12	17.18	0.02	0.08
3000	26.01	0.12	16.52	0.02	0.08
3000	26.01	0.12	16.52	0.02	0.08
2999.7	26.01	0.12	15.2	0.02	0.08
2999.7	26.01	0.12	15.2	0.02	0.08
2999.7	26.01	0.12	15.2	0.02	0.08
2999.7	26.01	0.12	13.88	0.02	0.08
2999.5	26.01	0.12	13.55	0.02	0.08
2999.5	26.01	0.12	13.55	0.02	0.08
2999.2	26.01	0.12	12.56	0.02	0.08
2999.2	26.01	0.12	11.9	0.02	0.08
2999.2	26.01	0.12	11.9	0.02	0.08
2998.9	26.01	0.12	11.24	0.02	0.08
2998.7	26.01	0.12	10.75	0.02	0.08
2998.7	26.01	0.12	10.75	0.02	0.08
2998.7	26.01	0.12	10.25	0.02	0.08
2998.5	26.01	0.12	9.76	0.02	0.08
2998.5	26.01	0.12	9.76	0.02	0.08
2998.3	26.01	0.12	9.26	0.02	0.08
2908.4	26.01	0.12	-6.56	0.02	0.08
2908.4	26.01	0.12	-6.56	0.02	0.08
2998.2	26.01	0.12	8.77	0.02	0.08
2998	26.01	0.12	8.27	0.02	0.08
2998	26.01	0.12	8.27	0.02	0.08
2998	26.01	0.12	7.78	0.02	0.08
2997.8	26.01	0.12	7.78	0.02	0.08
2997.8	26.01	0.12	7.78	0.02	0.08
2997.2	26.01	0.12	7.29	0.02	0.08
2997.3	26.01	0.12	7.12	0.02	0.08
2997.3	26.01	0.12	7.12	0.02	0.08

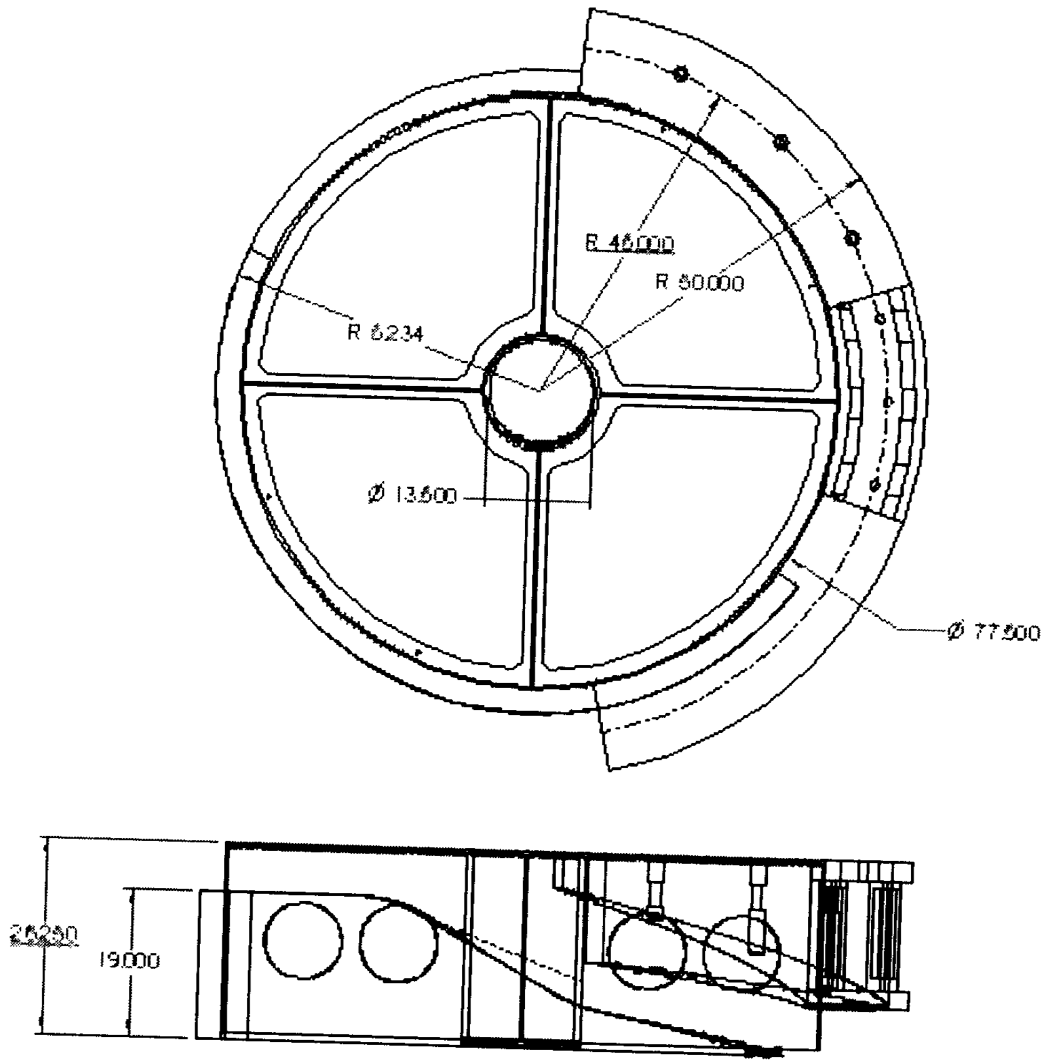
2997.3	26.01	0.12	6.96	0.02	0.08
2997.2	26.01	0.12	6.79	0.02	0.08
2997.2	26.01	0.12	6.79	0.02	0.08
2997	26.01	0.12	6.46	0.02	0.08
2996.6	26.01	0.12	6.13	0.02	0.08
2996.6	26.01	0.12	6.13	0.02	0.08
2996.5	26.01	0.12	5.8	0.02	0.08
2996.5	26.01	0.12	5.8	0.02	0.08
2996.8	26.01	0.12	5.64	0.02	0.08
2996.8	26.01	0.12	5.64	0.02	0.08
2996.1	26.01	0.12	5.31	0.02	0.08
2996.1	26.01	0.12	5.31	0.02	0.08
2996.3	26.01	0.12	5.14	0.02	0.08
2996.3	26.01	0.12	4.81	0.02	0.08
2996.3	26.01	0.12	4.81	0.02	0.08
2995.8	26.01	0.12	4.65	0.02	0.08



Item Number	Document Number	Title	Quantity
1	00-01	Bottom Spacer	2
2	00-02	Top Spacer	2
3	B18.3-1986	1.25"x6" Bolt	4
4	00-03	Base Plate	1
5	00-04	Worm Gear	1
6	00-05	Turret-T	1
7	00-06	Shaft	1
8	00-07	Frame	2
9	00-08	Tuuret-B	1
10	01-00	Lower Punch	30
11	02-00	Upper Cam	1
12	00-09	Cylinder	1
13	23060 YMB	Spherical Bearing	1
14	00-10	Disc Spring	1
15	03-00	Lower Cams	1
16	200 TP 173	Thrust Bearing	1
17	360 RU 30	Cylindrical Bearing	1
18	B18.3-1986	1"x4.5" Cap Screw	1
19	04-00	Upper Punch	30
20			
21	05-00	Locater	1
22	00-11	Shoulder	1
23	06-00	Supporter	1
24	00-12	worm	1

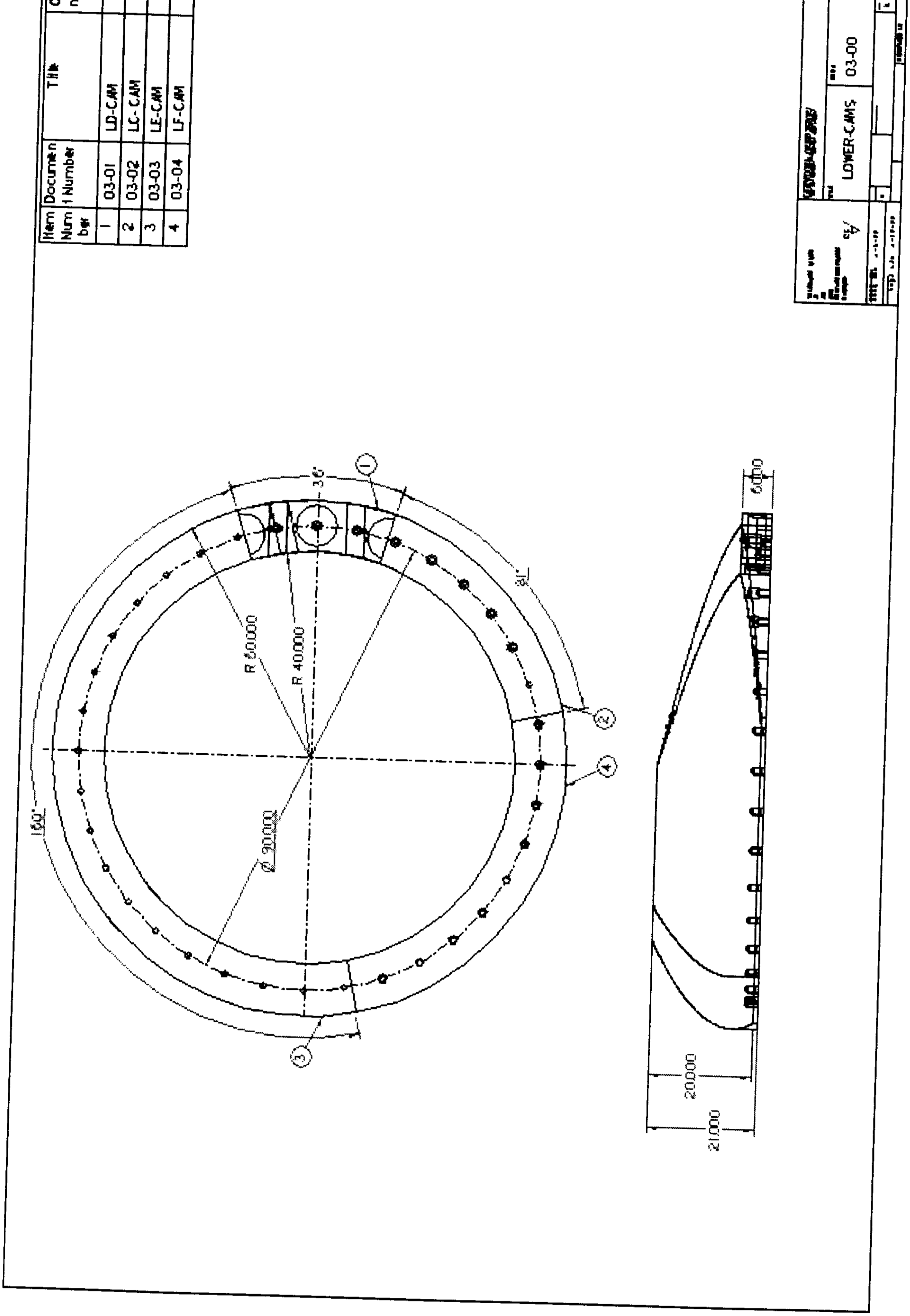

166.000
151.000
91.600
47.725
12.000
20.000
21.000
60.000
120.000
160.000
00-00

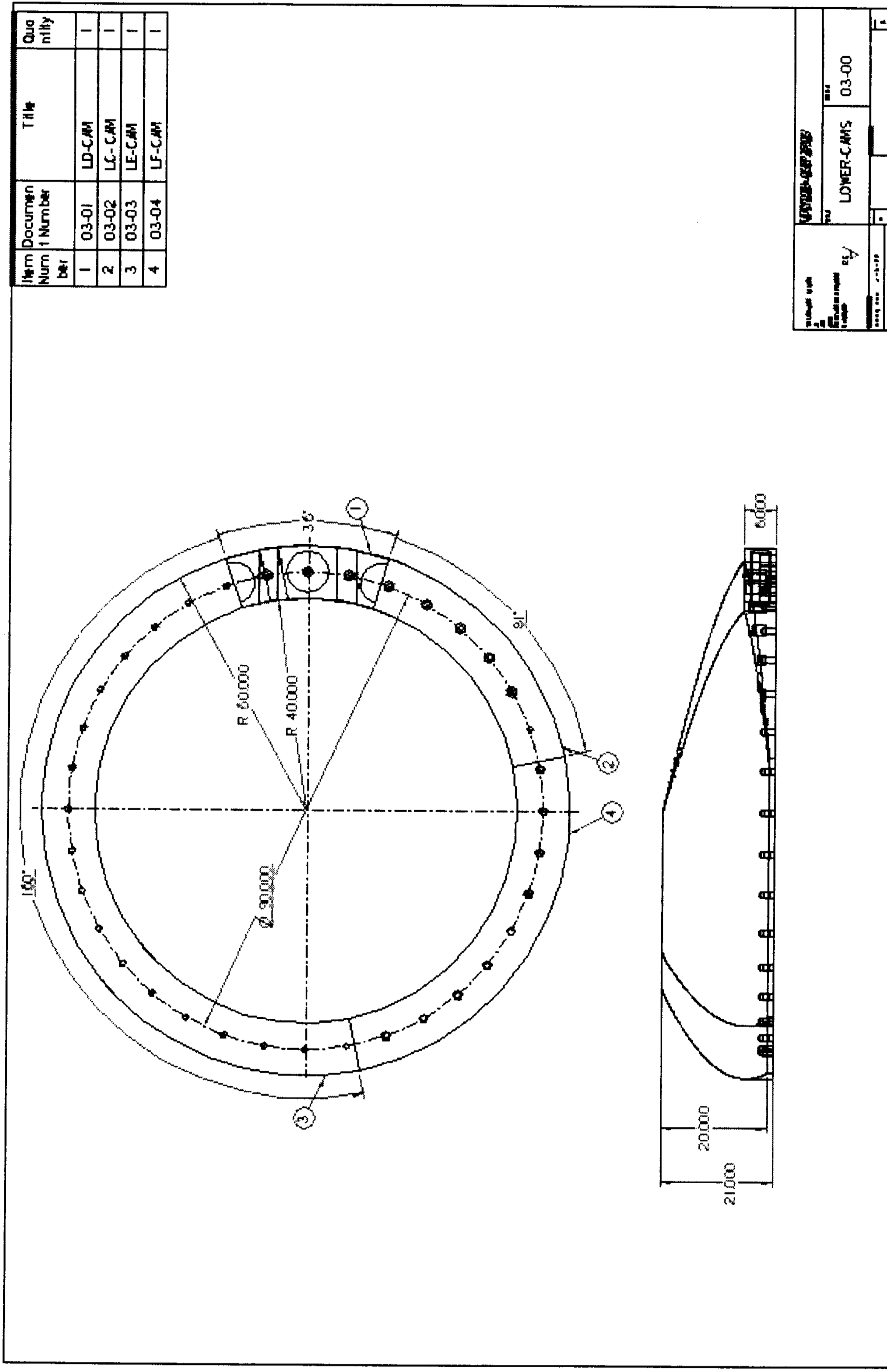
Item Number	Document Number	Title
1	02-10	UC-FRAME
2	02-01	UC-DWELL CAM
3	02-02	UC-BP-CAM
4	02-03	UC-CP-CAM
5	02-04	UC-FD-CAM



UPPER CAM

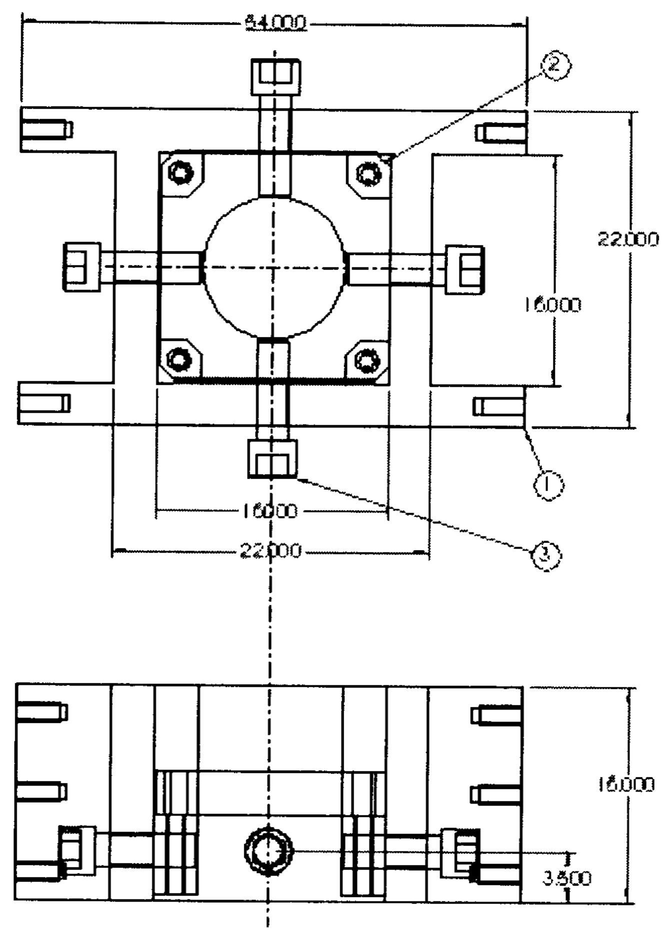
02-00


Item Number
Document Number
Title
1
03-01
LD-CAM
2
03-02
LC-CAM
3
03-03
LE-CAM
4
03-04
LF-CAM
LOWER-CAMS
03-00
R 60.000
R 40.000
Ø 90.000
36°
81°
180°
6.000
20.000
21.000



Item Number	Document Number	Title	Quantity
1	03-01	LD-CAM	1
2	03-02	LC-CAM	1
3	03-03	LE-CAM	1
4	03-04	LF-CAM	1

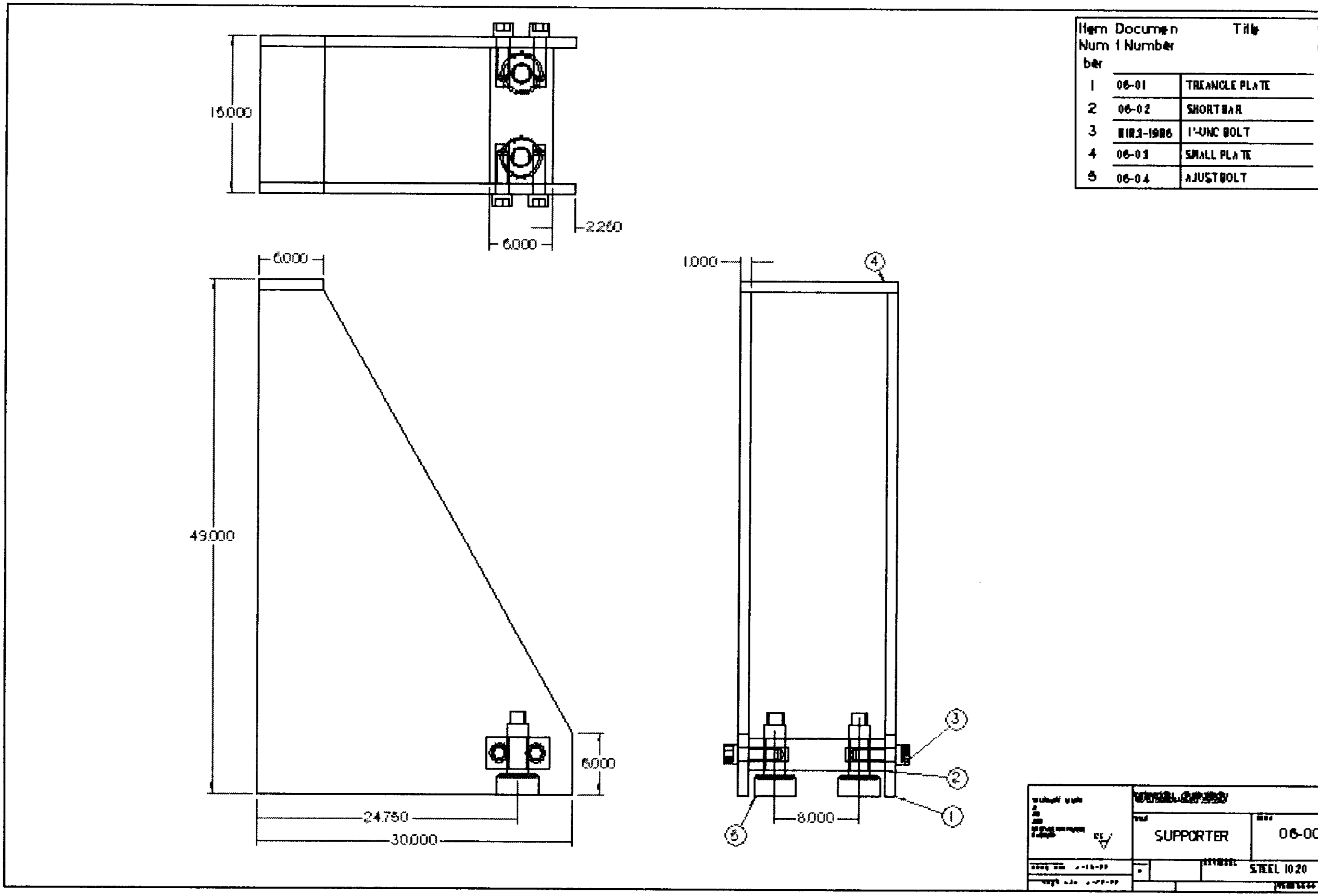
Item Number	Document Number	Title	Quantity
1	05-01	L-FRAME	1
2	05-02	L-SHOULDER	1
3	9183-1986	2 1/4 'BOLT	4



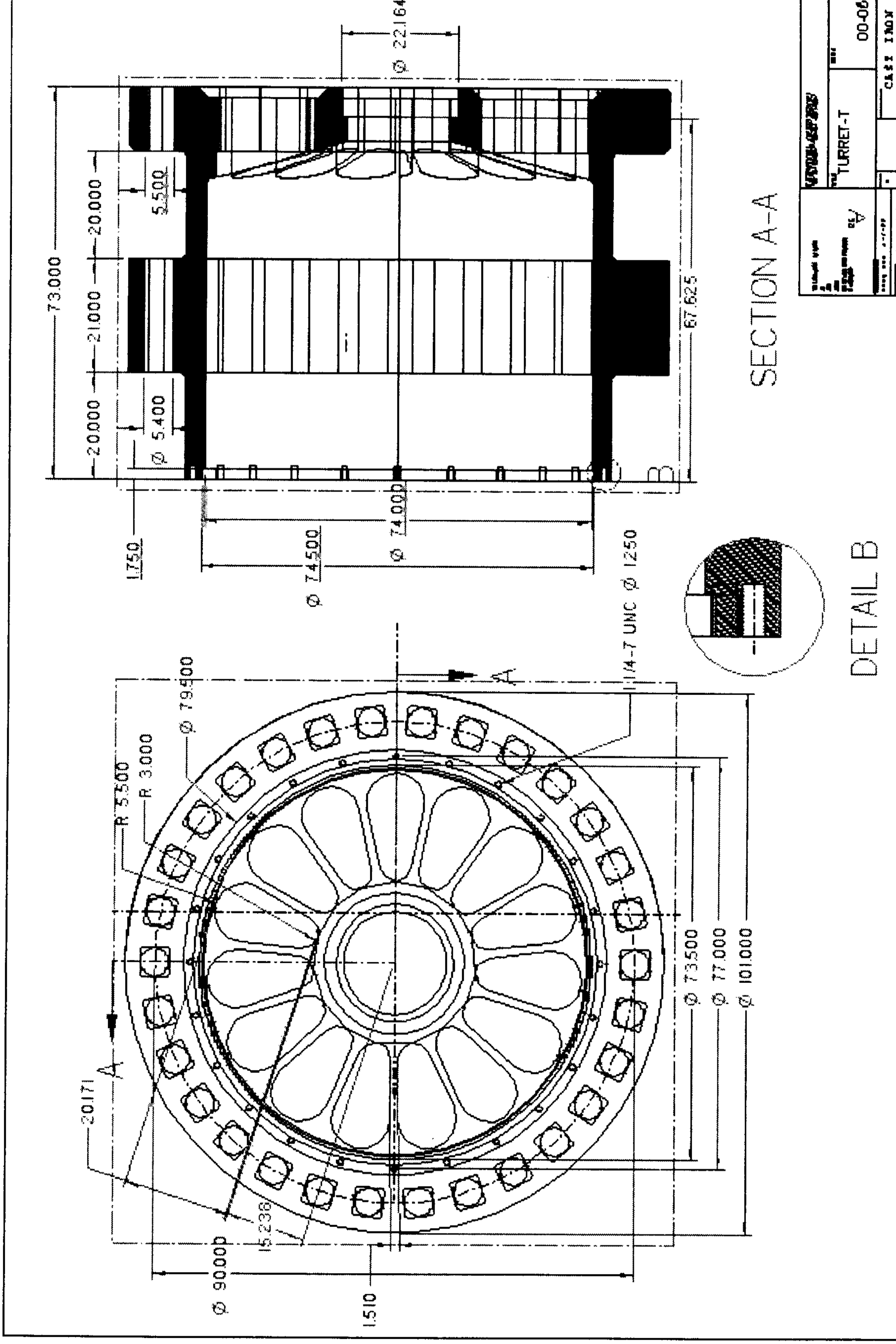
LOCATER

05-00

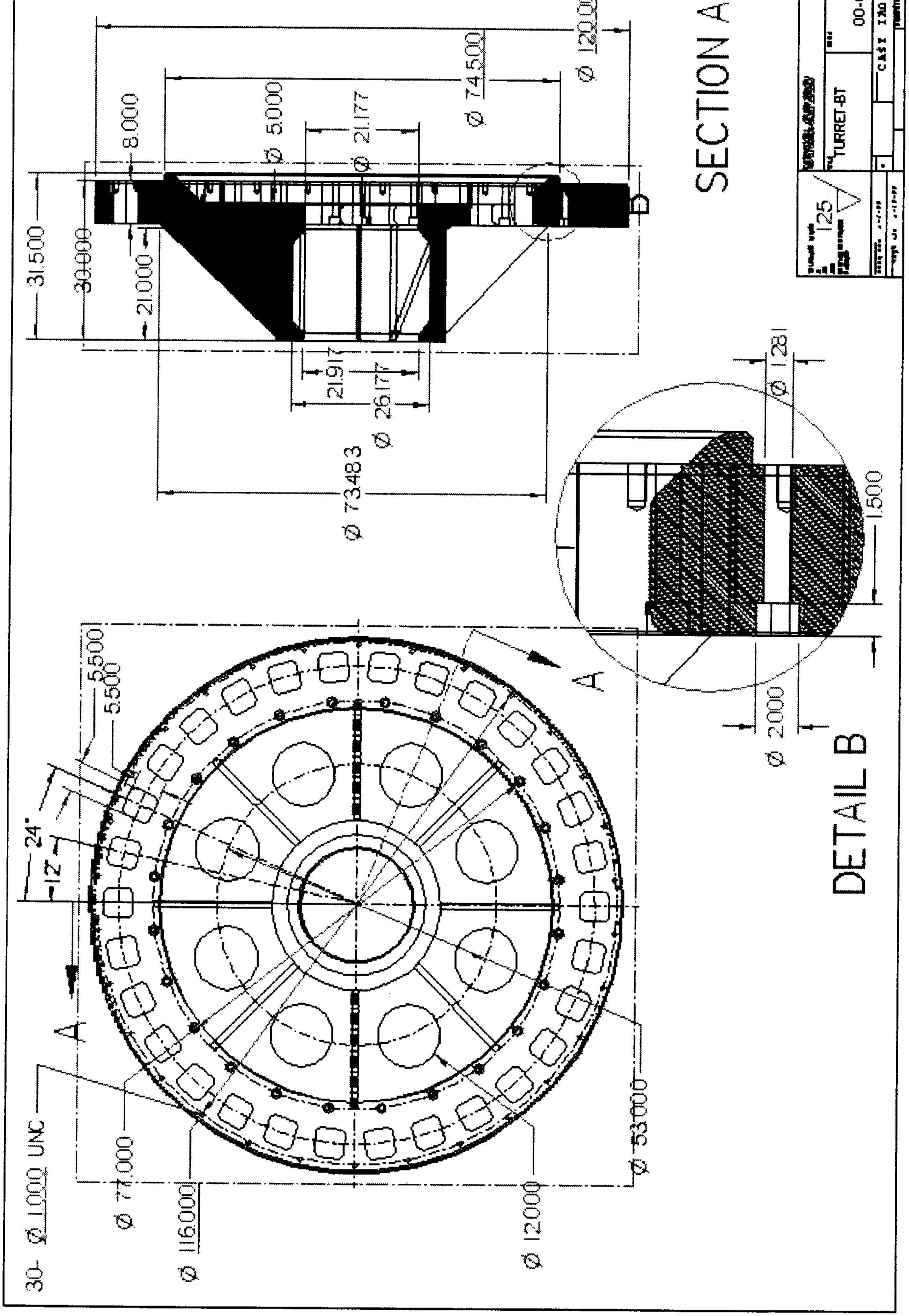
STEEL 1020

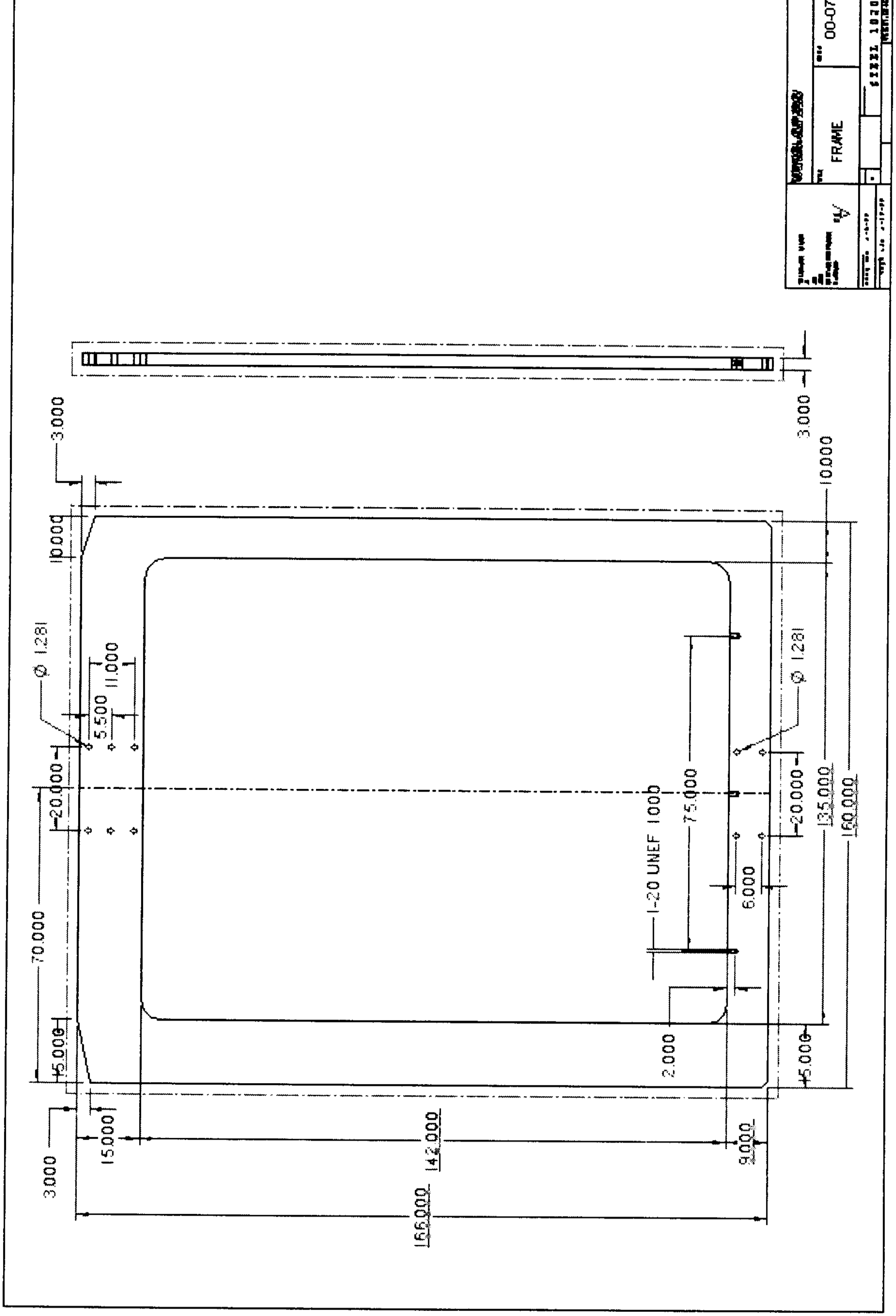


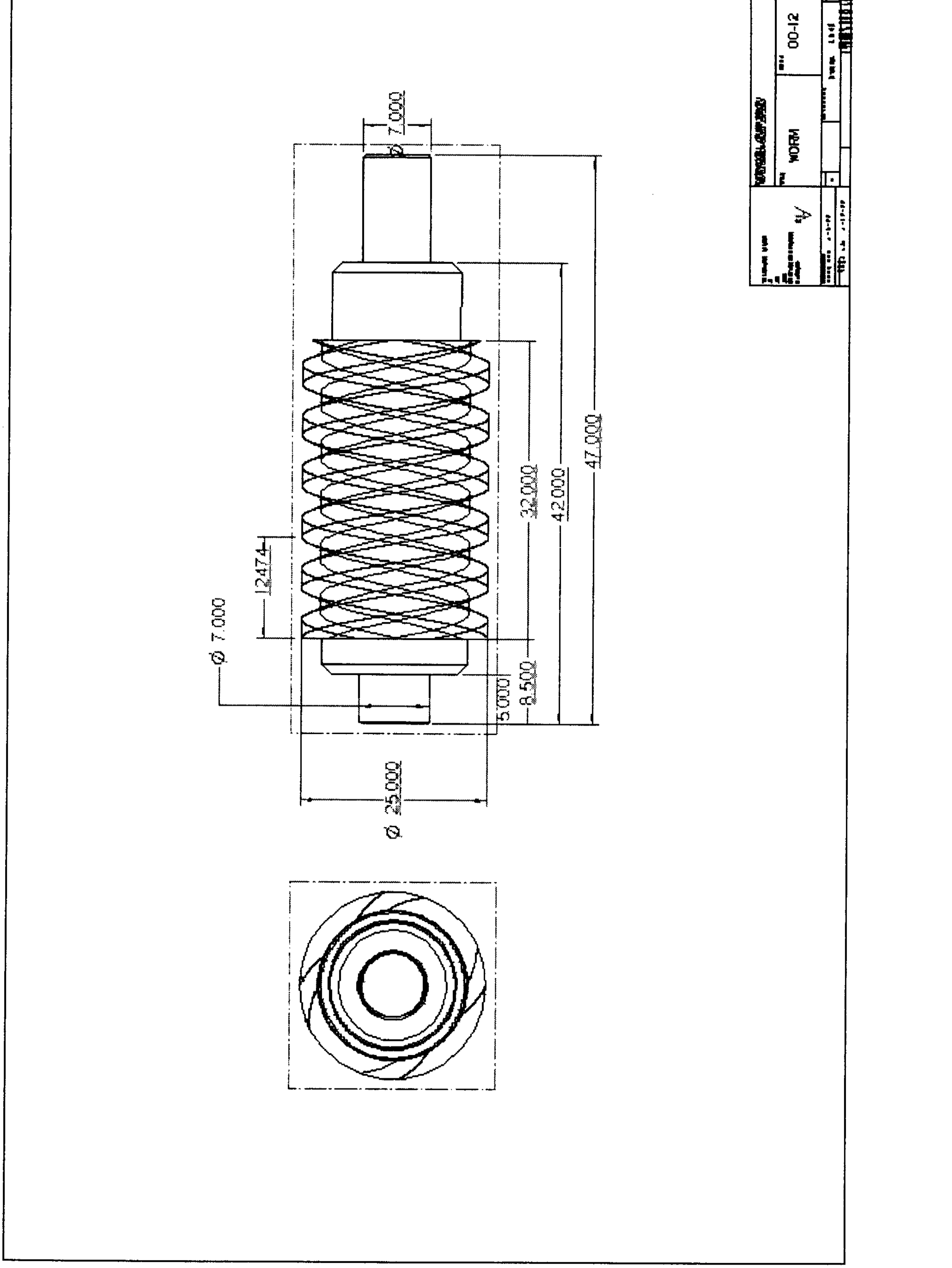
Item Number	Document Number	Title
1	06-01	TRIANGLE PLATE
2	06-02	SHORT BAR
3	BIR3-1986	1"-UNC BOLT
4	06-03	SMALL PLATE
5	06-04	AJUSTBOLT



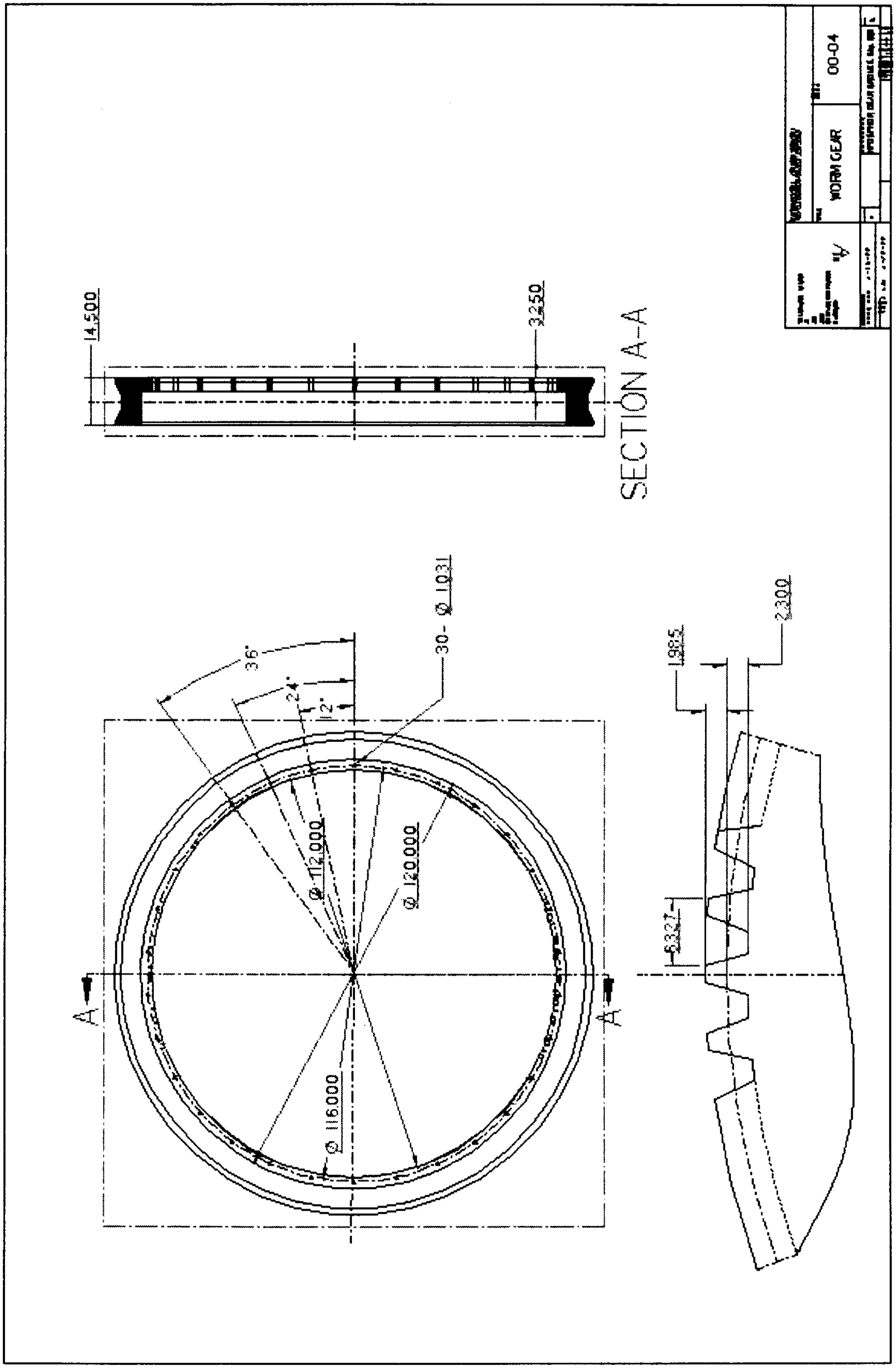
SECTION A-A
DETAIL B
TURRET-T
73.000
20.000
21.000
20.000
5.500
Ø 5.400
67.625
1.750
Ø 74.000
Ø 74.500
1 1/4-7 UNC Ø 1.250
Ø 22.164
Ø 79.500
R 5.500
R 3.000
20.171
15.236
Ø 90.000
1.510
Ø 73.500
Ø 77.000
Ø 101.000
A
A
B


SECTION A
DETAIL B
Ø 120.000
Ø 74.500
Ø 5.000
Ø 21.177
8.000
31.500
30.000
21.000
21.917
Ø 26.177
Ø 73.483
Ø 1.281
1.500
Ø 2.000
TURRET-BT
125
5.500
5.500
24°
12°
A
A
30- Ø 1.000 UNC
Ø 77.000
Ø 116.000
Ø 12.000
Ø 53.000


FRAME
00-07
3.000
10.000
Ø 1.281
5.500
11.000
20.000
70.000
5.000
1-20 UNEF 1.000
75.000
6.000
2.000
135.000
160.000
15.000
142.000
9.000
166.000



Ø 7.000
47.000
42.000
32.000
12.474
Ø 7.000
8.500
5.000
Ø 25.000
WORM
00-12



SECTION A-A
A
A
14.500
3.250
30- Ø 1.031
36°
24°
12°
Ø 112.000
Ø 120.000
Ø 116.000
1.985
2.300
WORM GEAR
00-04