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THE CITY UNIVERSITY

LONDON

THE DESIGN OF A 0.5 MN ROTARY FORGING MACHINE
AND EXPERIMENTAL ROTARY FORGING

by

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THE DESIGN AND CONSTRUCTION OF A 0.5 MN ROTARY FORGING MACHINE AND EXPERIMENTAL ROTARY FORGING

SUMMARY

This thesis describes research into the manufacturing process known as rotary forging. The scope of the work is concerned with the design and construction of a machine to be used for conducting experiments which relate directly to industrial applications of the rotary forging process. The experimental work which includes the forming of wrought materials and the compaction of powder metal, together with the subsequent plastic deformation of sintered powder metal preforms, was conducted primarily to assess performance of the machine. The thesis is presented in two parts. Part 1 relates to design and construction of the machine and Part 2 to experimental rotary forging.

Following a design synthesis formulation of the design requirements for the machine is discussed and the specification derived is stated. Design problems are analysed from fundamental considerations of the kinematics of the rotary forging process. Alternative solutions are discussed and the reasoning which leads to selection of the particular design is given.

The machine configuration selected is an inversion of the kinematic arrangement used by recent workers for research purposes. Features are incorporated in the design which form the subject of applications for patents to be applicable in the United Kingdom and certain foreign countries.

Provision is made in the design for the incorporation of instrumentation appropriate to the range of forging parameters. Description of tooling, ancillary equipment and installation are included.

Experiments to establish the performance of the machine as constructed and to form an initial assessment of the suitability for further development are described.

The objectives, procedures, results, discussion and conclusions are presented for the experiments listed as follows :

1. investigation into the upsetting of cylindrical workpieces,
2. initial investigation into upsetting cylindrical workpieces of low height/diameter ratio using dual conical dies,
3. evaluation of the machine when using contoured upper and lower dies to form a given axisymmetric component from a cylindrical billet,
4. the compaction and post sintered densification of cylindrical powder metal preforms, including development of a method devised to compact hollow cylinders.

It is concluded that the design concept forms a satisfactory basis for a cost effective range of machines applicable to manufacturing industry. Information is made available to aid the design and manufacture of machines, based upon a similar concept.

Modifications to the design are proposed following experience gained whilst conducting experiments. These are included in the suggestions for further work which also contains matters arising from the experimental work. A tentative method of predicting the torque required for platen rotation is presented in an appendix.

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NOTATION

A	total area of workpiece in a plane perpendicular to the direction of the applied force
A_H	predicted upper die-workpiece contact area using the method proposed by Hawkyard et al [24]
A_K	predicted upper die-workpiece contact area using the method proposed by Kubo et al [34]
C	constraint factor used in theory proposed by Kubo et al [34]
C_A	$\pi\sigma$
C_E	constant used for prediction of elastic deformation of rolls
C_H	coefficient determined from specific values of σ , θ , $f^{0.5}$
C_K	coefficient determined from specific values of C , θ , $\tan^{0.63}\theta$, $f^{0.63}$
C_{Z_1}	radial compliance of the Z_1 axis relative to the machine frame
C_{Z_2}	radial compliance of the Z_2 axis relative to the machine frame
C_1	constant which relates uniaxial compaction pressure to the remaining parameters for a given powder metal as proposed by Heckel [53]
C_2	constant applicable to a given powder metal in the theory proposed by Heckel [53] = $\ln\{1/[1 - (\rho_0/\rho_T)]\} + C_3$
C_3	constant applicable to a given powder metal to determine C_2 as proposed by Heckel [53]
D	current diameter of workpiece
D_0	initial diameter of workpiece
E	total energy required to cause a fractional reduction in height, R , by quasi-static, axisymmetric, homogeneous deformation
E_E	experimental value of energy to cause a fractional reduction in height $R = (H - f)/H$
E_f	predicted total energy required to cause a fractional reduction in height $R = (H - f)/H$
E_r	energy required for rotation of platen relative to the workpiece
E_t	energy required for translation of lower platen

F	force in direction of axis Z_1
F_A	force required to cause quasi-static, axisymmetric, homogeneous deformation
F_E	axial force determined by experiment
F_H	predicted axial force using method proposed by Hawkyard et al [24]
F'_H	predicted axial force using method proposed by Hawkyard et al [24] with allowance for elastic deformation of upper die
F_K	predicted axial force using method proposed by Kubo et al [34]
F'_K	predicted axial force using an empirical formula derived from a combination of the method of predicting axial force proposed by Kubo et al [34] and experimental results
F_P	predicted axial force assuming the existence of an instantaneous hinge
F'_P	predicted axial force assuming the existence of an instantaneous hinge and elastic deformation of the upper die
F_T	axial force predicted by applicable theory
F_{Z_e}	force in the direction Z_1 due to pressure on the element of radius r_e
F_{Z_1} ω_4	force resolved in direction parallel to Z_1 axis due to radial acceleration of mass, M , about the axis OA
F_r	force acting radially relative to the axis Z_1
$F_{r\omega_4}$	force resolved in direction perpendicular to the Z_1 axis due to radial acceleration of the mass, M , relative to the axis OA
F_{ω_4}	force due to radial acceleration of mass, M , acting radially relative to the axis OA
f	axial feed/revolution
H	current height of specimen
H_0	initial height of specimen
J_{Z_2}	second moment of inertia about the axis Z_2
k	current yield shear stress = $\sigma/2$
L	distance of centre of mass from the point O in the direction of the axis Z_2
ℓ	distance between instantaneous hinge, OA , and point at which force, F_{r_e} , due to element of torque dT_U , is applied
M	mass of upper platen assembly

M_A	moment of area
m_1	mass of dry compact in air
m_2	mass of oil impregnated compact including support wire
m_3	apparent mass of impregnated compact including support wire
P	normal pressure at die-workpiece interface
P_e	equivalent pressure, F_E/A
p	percentage porosity
R	fractional reduction in height
r	current radius of workpiece
r_e	radius of element
r_0	initial radius of workpiece
r_1	radius about axis Z_1
r_2	radius about axis Z_2
s	circumferential width of upper die-workpiece contact area
$\bar{s}$	mean circumferential width of upper die-workpiece contact area
$\bar{s}'$	mean circumferential width of upper die-workpiece contact area with allowance for elastic deformation of upper die
T	time
T_L	torque required, neglecting friction, to cause rotation of lower platen
T_T	total experimental torque required to rotate both upper and lower platens
T_{TP}	predicted total torque required to rotate both upper and lower platens
T_U	torque required, neglecting friction, to cause rotation of the upper platen
T_d	torque required to cause rotation of the plastically deforming zone relative to the workpiece
T_p	torque due to gyroscopic precession
T_μ	torque due to friction at workpiece - lower die interface
U	distance from centroid of predicted upper die-workpiece contact area to the instantaneous hinge OA
V	volume of workpiece
V_q	instantaneous velocity at point. Q, on upper die

V_i	relative velocity at upper die-workpiece interface
V_1	tangential velocity relative to axis Z_1
V_2	tangential velocity relative to axis Z_2
Y	yield stress
$\bar{Y}$	mean yield stress
z	distance in direction of Z_1 axis
β	angle between point, N, at the upper die-workpiece interface and the plane perpendicular to the axis Z_1 at the intersection of the platen axes
γ	$P/2k$ determined from $H/\bar{s}$
γ'	$P/2k$ determined from $H/\bar{s}'$
Δr	eccentricity
δr	relative radial displacement of axes
ϵ_z	natural strain in z direction
ζ_E	experimental <i>specific rotary forging force</i>
ζ_p	predicted <i>specific rotary forging force</i>
η	percentage process efficiency
θ	inclination of platen axes
ρ_E	average density determined by experiment
ρ_{E_f}	final average density determined by experiment
ρ_{E_i}	initial average density determined by experiment prior to post-sintered densification
ρ_0	apparent density of powder metal prior to compaction
ρ_T	theoretical density of metal
ρ_i	density of oil used for impregnation
ρ_w	density of water
σ	flow stress
χ	angle between the axis Z_1 and the generator, OA, workpiece interface
ω_p	angular velocity of gyroscopic precession

- ω_s angular velocity of spin about Z_2 axis
- $\omega_{sk_{Z_1}}$ angular velocity of skidding about the axis Z_1
- $\omega_{sk_{Z_2}}$ angular velocity of skidding about the axis Z_2
- ω_1 angular velocity about Z_1 axis
- ω_2 angular velocity about Z_2 axis
- ω_3 angular velocity about Z_3 axis
- ω_4 angular velocity about generator OA

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1.1 Basic principles and brief history of the rotary forging process

The requirement to change the shape of both pure metals and alloys of metals, has existed since they were first discovered by man.

If the desired shape cannot be achieved from the metal or alloy whilst in either a liquid state or a powder state then the options are either to remove material from a workpiece which is initially large enough to envelop the final shape, or to plastically deform material of initial volume at least equal to that of the final shape. The options may be stated as either the processes associated with material removal known generally as metal cutting, or processes associated with plastic deformation known as metal forming. Rotary forging is a metal forming process to be considered with the numerous others which are applicable to the latter of the options.

The forces required to cause a workpiece to plastically deform and thus change in shape, must be such that plastic yielding of the workpiece material occurs. The magnitude and direction of the stress to be applied to cause yield depends upon the material, its current state, the rate at which deformation takes place, and the remaining system of applied stresses. The stress-plastic strain characteristics of the material to be deformed may be modified to assist the process. For example, by increasing the

workpiece temperature for the duration of the forming process to reduce the yield stress, and with a view to increasing the ductility. This results in a reduction of the forces required to cause plastic deformation whilst increasing the amount of deformation which is practicable.

The magnitude and direction of the forces required to plastically deform a workpiece is therefore related to its size, shape, system of the remaining applied forces and the flow stress characteristics of the material.

From the earliest attempts by man to form metal it was realized that, with the forces available, the workpiece size and material properties could be such that the desired change in shape was only practicable by causing plastic deformation in a limited zone and progressing to the final desired form by a sequence of localised increments of deformation.

The progressive deformation processes, in current industrial applications, include hammer forging, swaging, strip rolling, rotary piercing, bar and tube drawing, extrusion, spinning, spin forging, etc.

In these processes, a plastically deforming zone is caused to move within the workpiece, in either a discrete or a continuous manner. The characteristics of those processes which operate in a continuous or steady state manner vary in the type of relative motion which occurs at the tooling-workpiece interface in that various combinations of sliding and rolling may occur.

Rotary forging is a process consisting predominantly of a rolling motion in which a plastically deforming zone is caused to rotate about an axis within the workpiece. The forging process proceeds in the direction of the axially applied force due to the plastically deforming zone being caused to move in a helical manner. Hence each revolution provides an increment of deformation in the axial direction. The process is therefore non-steady state and non homogeneous due to the localised nature of the plastically deforming zone together with frictional forces present at the die-workpiece interface. Clearly the rotary forging process was devised as a means of causing a permanent change in shape of a workpiece by applying the forces required to plastically deform a limited zone within the workpiece and thus be considerably less than that required to cause the total workpiece to plastically deform. The reduction in the required forces leads to reduction in size and cost of the total plant.

Many of the workpiece materials used in industry exhibit characteristics whereby the physical properties are modified by plastic strain. The plastic strain which occurs during the deformation associated with rotary forging can be resolved into radial, circumferential, and axial components. These phenomena may be exploited when the material properties, together with the resulting direction of plastic strains, can be arranged to enhance the physical properties of the final workpiece. The fundamentals of the process, illustrated in figure 1, are described in some detail in Chapter 4, and by Slater et al [1]. Figure 2 illustrates various states of a component being formed using an upper and lower die.

Typical metal forming operations which can be carried out are illustrated in figure 3 and figure 4 shows the types of relative motion which previous workers have found desirable to be induced at the upper die-workpiece interface. Since the process approximates to a rolling action at the upper die-workpiece interface, the requirement for lubrication is minimal and also results in low audio noise emission. The impact associated with many conventional forging processes is a source of both noise and high amplitude vibration levels which occur over a wide range of frequencies. These, together with the associated machine mass, are often such as to become problematic in building foundations and structures which are additional to difficulties which arise from both the working environment and factory siting.

Thus the advantages to manufacturing industry of rotary forging are related directly to both economic and environmental ramifications. The economic advantages arise from the ability to form components which conform with precision to the desired shape whilst using forces which are lower in magnitude, usually by at least a factor of five and at lower noise levels than those associated with conventional forging. Also for a given component the tooling is subjected to lower forces which results in a reduction in strength requirements together with reduced size and cost. Rotary forging machines when compared with conventional forging machines of equivalent workpiece capacity should therefore be smaller in size and weight, requiring less capital investment for machine procurement, installation, and tooling.

The methods of reducing the height of a cylindrical metal workpiece by causing a plastically deforming zone to move in a

helical manner with reference to its axis appears to have been applied successfully in the U.S.A. circa 1910 and is described in references [2] and [3]. Initially the purpose was the manufacture of wheels for railway wagons. Development of both the machine and the process appears to have continued to date and currently components up to approximately 1 metre diameter with height to diameter ratio usually $<1/10$ are manufactured.

A patent specification [4] published in Germany in 1920, naming a British inventor, refers to a method of manufacturing wheels for heavy vehicles including railway wagons. Investigation by the author revealed that United Kingdom patent applications numbers 19291 and 26837 were filed in August 1909 and November 1910, respectively by the same British inventor, namely Harry Samuel Bickton Brindley. The significance of this information is to indicate the date from which published information shows attempts being made to solve the manufacturing problems of forming large axisymmetric components of low height/diameter by using an axial force combined with a rolling action.

From information stated in a British Patent application filed in 1929 [5], Messrs. B & S Massey Limited in the U.K. had studied designs of machines based upon orbital motion of the upper die relative to the workpiece. It would appear that their work was discontinued until the period 1965-70, during which time they were engaged upon the design, construction and development of machines which are currently operated in manufacturing industry.

During the period 1966-70, experiments were conducted at the University of Manchester Institute of Science and Technology by

Slater et al [1]. Their work using model materials appears to be the earliest to demonstrate the advantages of the process in a quantitative form. Also during this period Marciniak, of the Warsaw Technical University, Poland, commenced work which included the introduction of planetary motion to the process as illustrated in figure 4. This resulted in patents being filed and subsequent licensing arrangements for machines to be manufactured in both the U.S.A. and Japan. Thus interest in the process may be stated to be world wide.

The quantity of powder metal used in manufacturing industry is increasing rapidly. The advantages offered by the appropriate techniques are various and relate to characteristics such as homogeneity of material structure with the possibility of producing alloys which are otherwise impracticable. Other advantages claimed for powder metal technology include, obviating the necessity to cut high strength alloys, the practicability of producing alloys where the temperature required is that for sintering as distinct from that required for smelting, economic use of costly raw materials, etc. The rotary forging process has been demonstrated to have industrial potential in the compaction of powder metal prior to sintering and also to post sintered densification. In addition the process can be applied to cause plastic deformation of sintered powder metal preforms for the purpose of achieving desired workpiece shapes.

By the end of the decade 1970-80 the various advantages and applications of the rotary forging process have been demonstrated, but for reasons which are not clear, a low growth rate in industrial applications of the process had occurred.

1.2 Current state of the art

Recent developments have led to current applications of the process to forming high strength alloys to produce large axisymmetric components of low height/diameter ratio, such as turbine discs and wheels.

Smaller components are currently being manufactured, albeit with a few exceptions, in pilot production projects. Reports indicate that the desired precision, surface properties, production rate and costs are currently being achieved. There is, however, some evidence, particularly in hot forming applications, that die wear remains a problem.

The rotary nature of the process, together with the prime force in the system being applied in an axial direction such as to cause compressive stress, results in rotary forging being inherently applicable to components of axisymmetric form with a final height to diameter ratio less than that of the initial workpiece. Some workers are currently engaged in the manufacture of components which are not axisymmetric and apply planetary motion at the upper die-workpiece interface to assist in inducing the desired workpiece material flow. There are also workers interested in examining processes which increase in height to diameter ratio such as forward and reverse extrusion.

Study of components which are currently manufactured successfully by rotary forging indicates that designers must acquire further knowledge and understanding of the process, before full

exploitation can be achieved. Absence of such knowledge leads perhaps to the situation where very few components are designed such that rotary forging is clearly the most economic method of manufacture or furthermore the only method which leads to the desired physical characteristics of the finished workpiece. Thus the process continues, in many cases, to be compared with conventional forging or metal removal techniques. Methods of manufacture employing metal removal remain with the fundamental disadvantage of material wastage. However, the metal cutting machine tools have become highly developed, with well proven equipment which is readily available over a wide range of capacity at competitive prices. This comment is of special significance as the range of both N.C. and C.N.C. machines are developed to become increasingly applicable to small batch production which continues to represent some 80% of the requirements in manufacturing industry.

For rotary forging, as with conventional forging, the workpiece is required to be available in a satisfactory form in terms of volume, shape, dimensional accuracy and acceptable surface finish. A facility for the efficient manufacture of billets, for example by cropping from bar, forms an essential part of the process and is clearly of paramount importance when considering the economics of the process. The current industrial significance of this with respect to the use of rotary forging may be that, although material wastage occurs during metal removal and parting-off operations when using bar fed metal cutting machines, the costs of both workpiece handling and additional plant costs are low. Current rotary forging machines are claimed to offer a reduction in size and cost when compared with conventional forging machines

of similar capacity, however one fact which may have influenced the present state of development, is that during the period 1970-80 there has been a surplus of forging press capacity in the U.K. A similar situation may also exist in both Western Europe and the U.S.A. From statistics published in the 1978 Annual Report of the National Association of Drop Forgers and Stampers [6] utilisation was some 70% of capacity with a decreasing trend. The annual report for 1979 [7] shows that whilst the downward trend may have ceased no upward trend exists. A further factor is that conventional forging machines tend to have a long service life evidenced by machines in current use, which, with appropriate overhaul and modification, have been in operation for over fifty years.

The author has been unsuccessful in discovering work published on the current application of rotary forging techniques to the compaction of powder metals in manufacturing industry and has also failed to determine whether or not the process has been used for post sintered densification. Development of rotary forging for this process may well have been restricted due to the capital cost of machines and the forms in which they have been available. Research has however been conducted in the academic field by workers in the United Kingdom, Poland and Japan.

1.3 Discussion on development of the process to date

Development work to date appears to have been motivated by widely differing interests, ranging from those driven to solve specific problems in manufacturing industry, to those seeking to establish and verify theories which are applicable to the process.

The rotary forging process has been in existence and available for development over a period during which rapid advances have been made in many areas of manufacturing engineering. Both experimental and theoretical work have shown the advantages which exist when comparison is made with conventional forging, yet progress in the development of industrial applications of the rotary forging process has remained relatively slow.

The design concept of recent rotary forging machines appears to be based mainly upon orbital or planetary motion in which the workpiece is either stationary, or moving in an axial direction only. This gives rise to considerable mechanical complexity due to the kinematic arrangement of the upper platen bearing system which is required to operate with orbital motion or even planetary motion at high axial loading.

Apart from machines designed for orbital riveting the author has found no evidence of development directed towards making machines available at low cost which utilise the principle but offer restricted capacity and performance. The objective being to minimise viable selling price in pursuance of a product and marketing policy to offer machines which provide an economic facility for relatively small batch manufacture of components with simple axisymmetric form. The machines to be suitable for workshops which are equipped for general purpose activities. For example, considerable demand may be envisaged for a machine requiring simple tooling which would be capable of forming components and could also be used for processes such as powder metal compaction and upsetting for fastening purposes. An analogy

of this is the highly successful type of moulding machine, designed for use with plastics, usually in the 10 to 20 MN range which has been manufactured in large numbers and is deemed to be an asset in numerous general purpose workshops.

The divergence in certain design characteristics and specifications of machines developed recently, together with the variances in operating parameters in current use, indicates that differing conclusions are being reached by various workers in the field, or alternatively, that widely differing objectives exist.

1.4 Establishing the need for the work

From appraisal of the current state of the art and present rate of progress, together with knowledge acquired during previous study and experience gained from experimental work on the process, the Head of the Manufacturing Engineering Section, in the Department of Mechanical Engineering, The City University, London, concluded that rotary forging should be the subject of further research. It had been observed that machines had become kinematically complex and costly to construct often requiring special purpose manufacture arising from the size, shape and desired precision of certain components. These factors could be construed as design weaknesses and he therefore concluded that a need existed for research into the basic design concept of machines. It was also reasoned that any experimental work, including the compaction of powder metals, should be conducted upon specimens which represent and relate directly to industrial applications of the process. To proceed with the work would necessitate the

availability of a rotary forging machine of suitable capacity, equipped with the appropriate tooling and instrumentation.

To procure a machine from the range of those available and carry out modifications in accordance with the outcome of research into design, would have resulted in expenditure which was prohibitive in terms of the financial resources available. The need was identified to conduct research into the design of machines which in the event of a successful outcome could be constructed within the constraints of both the financial resources and the manufacturing facilities available in the University Central Workshop and the Manufacturing Processes Laboratory at The City University, London.

With the need for the work established, the objective of the research can be defined as an attempt to advance knowledge of the rotary forging process by making available a machine of appropriate capacity with the design considered from the fundamental kinematics of the process and to be manufactured consistent with the above constraints. The machine to be proven subsequently as suitable for conducting appropriate experiments in metal forming and also compaction or post-sintered densification of powder metal specimens.

1.5 Scope of the work

The scope of the work stated in the flow diagram, figure 5, shows the various stages of the research work.

Published work from both academic and industrial sources was

studied and the characteristics of previous rotary forging machine designs were investigated.

The requirements for the machine were formulated and following a design synthesis alternative kinematic arrangements were investigated. Further study led to the preferred design solution and to the specification of the machine which was constructed. The instrumentation incorporated was calibrated and the machine performance assessed against the specification.

Experimental rotary forging was conducted on both metal forming and processes associated with powder metal technology. The main objective was to evaluate the suitability of the machine for current research purposes and to assess the industrial potential of the design. Investigations were made into limitations of the processes.

The scope of the research also included initial experiments with the machine in a configuration which employs conical platens in both the upper and lower platen positions. This work was based upon the proposal by the author that there would be special industrial significance in the availability of thin discs which exhibit anisotropy in elastic properties which are polar in characteristic. The characteristic would be distinct from the cartesian nature of the anisotropy which exists in discs blanked from rolled sheet. The symmetry of dual conical platen arrangement causes plastically deforming regions to rotate simultaneously in both the upper and lower surface of the workpiece. The initial investigation was concerned with the upsetting of

cylindrical specimens and methods of predicting axial force.

The research included initial experiments to determine the relationship between the rotary forging parameters and the resulting average density of sintered powder metal specimens of both solid cylindrical and hollow cylindrical form. A method was devised to produce hollow cylindrical compacts and appears to be novel. The average open porosity of specimens in the post-sintered state was also determined. The machine was used to investigate post-sintered densification.

1.6 Intention of the thesis

The thesis is in two parts and the intentions are to present the following :

Part 1. Design of a 0.5 MN rotary forging machine which includes :

- (a) a review of previous relevant published work in the field of rotary forging, together with reasoning and findings which confirm the original assessment of the need for the work,
- (b) the formulation of the design requirements, the design synthesis, the derivation of the final design specification for a 0.5 MN rotary forging machine, together with a description and illustration of design detail for the kinematic arrangement selected.

The drawings prepared for component manufacture and to guide assembly are listed, together with a description of the proprietary items procured. A schedule of the equipment selected and incorporated for measurement and control purposes is included,

- (c) a description of the methods used and the data obtained during calibration of instrumentation is stated together with the results obtained whilst verifying the performance of the machine against the design specification.

Part 2. Experimental rotary forging, including results and conclusions from experiments conducted primarily to assess the performance of the machine and associated tooling, consistent with the objectives stated in Section 4 of this Chapter, as follows :

- (a) upsetting of cylindrical specimens using a conical upper die and a flat lower die,
- (b) initial investigation into upsetting using conical upper and lower platens,
- (c) forming a component selected for both shape and material to represent directly an industrial application of the rotary forging process. The results include comparison of the forging parameters used with the dimensional accuracy and surface roughness obtained,

- (d) producing solid and hollow cylinders from iron powder metal with particular reference to the relation between average density, open porosity and axial force when using rotary forging for the compaction and post-sintered densification processes,
- (e) conclusions,
- (f) suggestions for further work.

A tentative method of predicting torque required which is based upon an instantaneous hinge concept is presented in an appendix.

PART 1. THE DESIGN AND CONSTRUCTION OF A 0.5 MN ROTARY FORGING MACHINE

CHAPTER 2

PREVIOUS RELEVANT PUBLISHED WORK

2.1 Previous work in the industrial field

As stated previously, the earliest report of the rotary forging process appears to be in references to the Slick Mill, used for rail car wheel manufacture, as described in a United States Patent filed in 1909 [2] and a British Patent filed in 1910 [3]. The machine is illustrated in figure 6. The Slick Wheel Mill was also reported in the publication *The Iron Age* during 1918 [8] and formed part of a plant designed to manufacture wheels, up to 965 mm diameter, for use on railway rolling stock. Other items, such as gear wheel blanks and flywheels for automobile engines, were also manufactured on the forging machine, which was some 12.2 m long, 5.5 m wide, and 4.3 m high, weighing 4.0 MN. The mill, complete with drive system, was some 21.3 m long.

Provision was made for an axial force of up to 13 MN and 1.5 MW power was available for rotational drive to the workpiece support platen. The platen axes, lying in the horizontal plane, were arranged to be at an angle of 10.66° . Separate hydraulic rams were incorporated for each of the specific functions to provide forging force, platen retraction and workpiece ejection.

Development of these machines has continued during the past 70 years and they are manufactured currently by The Bethlehem Steel Corporation with axial load capacity of 15 MN with up to 0.93 MW power available for rotational drive. In 1927, Hughes, of the Budd Wheel Corporation, Philadelphia, U.S.A., filed an application

for a patent [9] which is illustrated in figure 7. This is similar to the machine designed by Slick, but uses rotational drive to both platens.

During the latter part of the decade 1920-1930, rotary forging was being studied in the United Kingdom by B & S Massey Limited. The work appears to have ceased at an early stage of design, but resulted in United Kingdom patents being awarded in 1929 [5]. Work was resumed during the period 1960-1970 when an experimental model was built and followed by machines manufactured for use in industry. Further United Kingdom patents were awarded in 1969 [10] in which claims are related mainly to bearing arrangements and ring gears to provide positive drive to the upper platen as illustrated in figure 8. Hitherto, the necessary torque to overcome the upper platen bearing frictional torque was provided by friction present at the upper die-workpiece interface. Development of machines based upon the rocking die, or orbital concept, has continued in conjunction with equipment designed to mechanise the processes of billet shearing, workpiece heating, forging and the appropriate stages of workpiece transfer, including loading and unloading.

A great deal of the work by B & S Massey Limited, has been carried out on alloy steels and they claim to have optimised the process by operating with workpiece temperatures in the region of 600°C to 800°C. They state that operation at temperatures within this range gives an adequate ratio of tool strength to workpiece yield stress, thus providing satisfactory tool life.

They have adopted the trade name of 'Rotaform,' and three publications Reference [11], [12] & [13], describe the state to which the process has been developed. The information supplied is based upon experiments conducted and industrial experience, from which it is concluded that the process is applicable to a wide range of components. B & S Massey Limited also state that machines will be exhibited and offered for sale for the first time during 1980, and it is significant that the exhibition chosen is organised for the machine tool industry which hitherto has been concerned primarily with equipment for metal cutting. It is interesting to note that on a machine of 2.0 MN axial force capability and maximum workpiece diameter of 150 mm they choose to operate with a *wobble* angle, $\theta = 4^\circ$ and a 1:1 ratio ring gear system to provide drive between the upper platen and the workpiece. Thus they do not attempt to comply with the conditions which would provide a kinematic arrangement to maximise rolling and accordingly, they induce sliding motion at the workpiece upper platen interface. B & S Massey Limited also operate at 1000 rev/min, which is a higher angular velocity than that used currently by other workers. This, together with high axial feed rates, enables them to claim production rates equal to those of conventional forging. They also state that, when compared with conventional forging of steels, less decarbonisation occurs in rotary forging, due to the ability to operate at lower workpiece temperatures. A further claim is that the surface finish is superior, due to the rolling or ironing action. The machines manufactured to date have been operated by Stockton Precision Forge Limited, who provide a sub-contract service to industry, offering a facility for large batch production. Initially, manufacture appears to have been

concerned mainly with automobile components, but subsequently the process has been found to be particularly applicable to items such as high pressure pipe flanges in both alloy steels and stainless steels.

In 1968 a patent application was filed in the U.S.A. [14], which describes a machine designed by Ulrych in West Germany, for the Rheinstall Wagner Werkzeugmaschinenfabrik GmbH, and has some similarity to the machines described in [2] and [9].

With the exception of a reference by Williams [15], to time saving in the manufacture of clutch components by a Polish agriculture implement manufacturer, the author has been unsuccessful in obtaining information on the performance of the Polish designed units in industrial application. Limited information is, however, available from V.S.I. Automation Inc., U.S.A., who are licensed to manufacture the machines designed at the Warsaw Technical University.

In 1973, V.S.I. Automation Incorporated, Troy, Michigan, U.S.A., commenced to manufacture orbital forging machines using basic components imported from Poland. V.S.I. have apparently modified the original design to provide axial displacement of the lower platen thus obviating the necessity to move the complete upper platen assembly axially as in the Polish built machines. Maicki [16], described the various types of upper die motion which are available from a machine of 1.6 MN, axial force capacity operating at 150 rev/min, with a maximum workpiece diameter of 100 mm, and with the platen axes inclined between 1° and 2° . The machines

appear to be marketed as part of a range of both equipment and services offered for the solution of manufacturing problems identified as being of an unusual nature.

V.S.I. had previously been manufacturing rotary forging machines which were developed circa 1965. The development at that time was related mainly to fastening, and resulted in improvements to the upsetting process, where comparisons are made with conventional impact riveting. The market demand for the orbital riveting process was increased markedly in the U.S.A., by the introduction of legislation regarding noise level limitations applied to impact riveting. Maicki states in reference [16], that the V.S.I. orbital forging machine is particularly effective in forming parts which are relatively thin with respect to their diameters and in the summary of advantages claimed for the process the ability to forge intricate shapes is included. Maicki [16] also makes statements regarding tool life and reference is made to the use of carbide tools. He also indicates that the V.S.I. Inc., decision to proceed with development of rotary forging was based upon satisfactory experience over the previous ten years with orbital motion presses used for fastening.

Dredger [17], makes reference to rotary forging being applied in Japan to the manufacture of automobile components. Recent academic workers from Japan, however, make little reference to industrial applications.

The application of rotary forging techniques to the compaction of powder metal is mentioned briefly by Williams [15], when referring to the advantages and possible applications of rotary forging

machines. No published work has become available to the author which reports data acquired, or provides information to indicate experience of rotary forging in industrial applications related to powder metal technology.

2.2 Previous work in the academic field

(a) Metal forming

During 1967 Barooah reported in an M.Sc. dissertation [18] that under the supervision of Slater, and with the support of Johnson, an experimental rotary forging machine with 10 KN axial force capacity, based upon the orbital concept, had been designed and built at the University of Manchester Institute of Science & Technology. The results from experiments conducted on the machine were reported by Slater, et al [1].

Their work appears to provide the earliest information on the advantages of the process presented in a quantitative form together with identification of the parameters which influence the mushrooming phenomenon. They also determined a relation between the number of revolutions of the upper platen and the fractional reduction in workpiece height.

Slater and Appleton continued this work by conducting experiments with model materials [19], and also presented their findings from research upon the effects of upper platen configuration during rotary forging into a contoured lower platen [20].

Appleton and Slater [21] also conducted experiments on upsetting

both solid and hollow circular cylinders, with the objective of providing an aid to the design of machines.

Hawkyard continued the work at U.M.I.S.T. by designing and constructing a machine, of approximately 0.25 MN capacity, using the orbital or rocking die concept. The pressure distribution at the platen-workpiece interface was studied by Hawkyard, Gurani and Johnson [22]. Dimitrov and Hawkyard studied yielding of the workpiece during the rotary forging process by considering the indentation of thin discs [23]. This was followed by successful work on the prediction of forging force in the upsetting process by the application of a combination of plastic hinge theory and indentation theory [24]. The method of prediction was verified by experiment and is shown to be particularly applicable to workpieces of low height/diameter ratio. The subject of pressure distribution at the platen workpiece interface during upsetting in the rotary forging of a short billet is also dealt with by Johnson [25].

A machine has been designed and constructed at The University of Nottingham, which has similarities to the second machine constructed at U.M.I.S.T., but is equipped for automatic control. Standring and Appleton, at The University of Nottingham, have studied and reported extensively on the subject of rotary forging machine design [26]. Their investigations of the kinematic relationship between the angled die and workpiece in rotary forging [27] shows the problems of induced slipping and skidding in orbital type machines due to constraint of rotation of the upper conical die relative to the workpiece. They also show that

machines may be classified by the kinematic functioning of the die, which can be described in terms of the Euler angles of nutation, precession and spin. They are currently engaged upon work of great industrial significance in their study of the parameters which influence surface finish and die wear.

Marciniak, working at the Warsaw Technical University, Poland, circa 1965-70, was engaged in the design of rotary forging machines, and devised a method of providing planetary motion between the upper platen and the workpiece. The method illustrated in figure 10, is the subject of a U.K. patent [28]. The work carried out during this period is described in [29] which shows that the machines differed from those designed previously in industry by Messrs. B & S Massey Limited, in that the workpiece is stationary and the complete upper platen assembly moves in the direction of the lower platen axis in addition to rotating. In the machine described, rotation of the upper conical die relative to the workpiece is constrained to zero, using a sliding pin and slot mechanism which replaces the function of the ring gears in the 'Rotaform' machines.

The academic work in Poland appears to have been directed, with considerable success, towards designing and proving machines intended for industrial use. They have also been engaged upon the solution of tooling problems associated with manufacture of components of high precision and complex form such as the wedge-shaped workpiece described by Marciniak [30]. The success achieved to date is demonstrated by the range of components produced successfully by rotary forging which are described and

illustrated by Grzeskowiak [31]. This work concludes that certain cold forming of gears can only be achieved by employing planetary motion at the die-workpiece interface. The Polish PXW 100A *Rocking Die* machine uses a down stroking upper platen assembly capable of a force of 1.6 MN whilst rotating at 150-180 rev/min.

Rotary forging machines were built in Japan at the Government Industrial Research Institute and the work reported by Kubo et al [32] & [33], is mainly related to accuracy and defects of components. The rocking die type machine built in 1973 has 1.3 MN axial force capacity, 80-160 rev/min rotational drive, a facility for adjustment of *wobble* angle between 0 and 5°. It is suggested in reference [26] that the designs are similar to those produced earlier at U.M.I.S.T. Subsequent work by Kubo and Hirai [34], is concerned with the effect of the rotary forging process upon workpiece shape during upsetting operations. Comparisons are made between results from experiments using the rotary forging process and experiments upon similar specimens using conventional forging. They have also proposed a method of predicting the number of revolutions required to attain a given fractional reduction in height of cylindrical specimens. The method of predicting the axial force required is based upon the area of contact at the upper die-workpiece interface and material flow stress. The parameters used to estimate the area include the current workpiece radius, the axial feed per revolution and the angle of inclination of the platen axes. The conclusion from this work is that the values obtained from estimates agree fairly well with those obtained from experiments. It was also concluded that less inhomogeneous deformation occurs during upsetting by the

rotary forging process when compared with conventional forging.

The workpiece deformation behaviour during simultaneous extrusion and upsetting of cylindrical specimens by rotary forging has been studied at the Technological University of Nagaoka, by Kobayashi et al [35]. The conclusions from the experiments are, that both workpiece, initial height to diameter ratio, and lubrication of the workpiece-lower platen interface, are of significance. Also, that when comparisons are made with conventional forging, small differences result in the shape at the periphery of the flange of the final workpiece. The work in Japan since 1970 upon machine development, forging research and theoretical investigation is included in surveys by Standring and Appleton [36] & [37].

At the University of Michigan, U.S.A., work has been carried out by Little and Beyer [38] on a machine provided by V.S.I. Automation Inc. The work reported to date is concerned with the relationship between axial feed per revolution, the area of contact between the conical platen and workpiece, the natural strain in the direction of the axis of the specimen, and the true compressive stress-natural strain characteristics of the workpiece material. The agreement between theoretical and experimental results is stated to be adequate for the purpose of making predictions for most industrial requirements. The predictions are based upon upper die-workpiece contact area and workpiece flow stress which supports the findings reported in reference [34]. However, from a study of their published experimental results, it appears that large variances occur between predictions and experimental results when the natural strain is between approximately 0.1 and 0.2.

The method used to predict the upper die-workpiece control area and referred to as the *footprint* is not reported in [38].

(b) Powder metal technology

A number of workers in the academic field have applied rotary forging to powder metal technology. Evans et al [39] have reported their findings when producing specimens using rotary forging for the compaction of powder steel. They conclude that rotary forging techniques have proved suitable for the compaction of powder metals, but that unattractive features are exhibited by the existence of a low density zone at the centre of the compacting regions remote from the upper die. They also conclude that low rates of compaction give the highest component average density, and also assist in avoiding cracking of the *green* compact. Kotschy et al [40] have shown that the rocking-die method could be applied to the cold post-sintered densification of sintered powder metal preforms, and concluded that irrespective of the sintered preform density, the cold forming using the rocking-die method could increase the density to 98% of the theoretical value.

Kotschy et al [41] conducted experiments on the rotary forging of conventionally compacted and sintered specimens produced from powder iron. They showed that specimens of a complicated shape could be produced using the rocking-die process. They also found that when the process was used for cold post-sintered densification, specimens exhibited high wear resistance when subjected to a standard roller test.

From their work using rotary forging in closed dies for post-sintered densification of *cold* sintered specimens previously compacted by uniaxial compression, Ogrodnik and Karpinski [42], conclude that the high relative densities of 0.98 and low open porosities of 1.5% observed are only otherwise obtainable by using *hot* conventional forging. The experimental results also show the tensile strength which can be achieved. Their theoretical work is based upon relations between the axial force, the relative density and the number of revolutions to achieve a given density. This requires knowledge of the uniaxial force - density - height relation for a specimen of similar dimensions and powder metal, together with the yield stress at a relative density of unity.

Johnson in reference [43] draws attention to the significance of strain rates during compaction of powder metals. Dynamic compaction, as defined by the current volumetric rate of compaction quoted in the reference, would not appear to be practicable for rotary forging. However the question is raised of the possible effects of rotating the deforming zone at high angular velocity. The three stages of compaction described by Johnson [43] and respective transitions, could possibly be influenced in differing manners by strain rates resulting from high rotational speed. The work on powder compaction and sinter-forging in Japan since 1970 is summarised by Standring and Appleton [37].

2.3 Previous relevant work in powder metal technology unassociated with rotary forging

Previous relevant work in powder metal technology with specific reference to powder forging is summarised by Huppmann and

Hirschvogal [44]. The forging into closed dies with special reference to non-ferrous materials was described by Hirschorn and Bargainnier [45]. The economic aspects of powder metallurgy are dealt with by authors such as Jones [46], Anderson and Winqvist [47].

Rapid growth is taking place in the application of powder metal technology in manufacturing industry. The highest growth rates appear to exist in countries with the most advanced manufacturing engineering expertise.

The rate of development of powder metal technology, and in particular the range of materials available can be assessed by comparison of the alloys and components referred to by Marshall [48] in 1968 and those referred to by Brown in references [49], [50] and [51] approximately one decade later.

A theory applicable to uniaxial compression for compaction and post-sintered densification was proposed by Heckel [52] and [53] and was later investigated by Alexander and Quainton [54] for applicability to isostatic compaction.

Aketa, Tanaka and Tsuna [55] at Osaka University used fine lead balls and x-ray techniques to study density distribution in cylindrical compacts. From their findings they conclude that little or no radial and tangential flow of powder occurs when applying uniaxial compression in closed dies.

In their study of the deformation characteristics and the application of plasticity theory to sintered powder metals, Kuhn and Downey [56] show that both Poisson's ratio and the yield stress

characteristics are density dependent. A yield criterion is presented which contains a term for the current value of Poisson's ratio.

Hirschorn and Garey [57], describe the manner in which work hardening becomes problematic when attempting to achieve high relative densities with cold compaction, and shows correlation with the theory presented by Heckel in references [52] and [53].

The design of sintered powder metal preforms for forging was studied by Kuhn and Downey [58] and they describe how flow and fracture influence design. The results may be applied as an aid to design in a qualitative manner, or alternatively used in conjunction with plasticity theory.

From study of components manufactured using uniaxial compression it is evident that any method to be considered as an alternative process would need to be capable of producing hollow cylindrical forms.

2.4 Discussion of previous relevant work

It appears that most of the previous work in the industrial field has been carried out in the United States of America, the United Kingdom, Japan and Poland.

In the U.S.A. the Slick Mill concept continues to provide a solution to specific areas of manufacturing problems. However, in the U.K., the 'Rotaform' shows evidence of vigorous pursuit of a design concept based upon high production rates. The

development of rotary forging appears to form part of a policy to provide automated plant for use in large batch production of components using mainly *hot* and *warm* forging. This suggests that the process is seen as a module within a capital intensive unit capable of producing large quantities of components, at costs which are predicted to be advantageous when comparisons are made with alternative methods of manufacture.

Reports from industry, based upon direct operating experience, or the formal independent assessment of procured machines, have not become available to the author. Nevertheless, the author has held discussions with certain members of industry who have carried out preliminary investigations during consideration of the acquisition of rotary forging machines. They appear to have concluded that problems exist with machines using the orbital technique and make comment on matters such as inherent machine vibration, tool wear and unsatisfactory tool life.

There are divergences in both specification and operating parameters for machine designs arising from previous work. For example, machines designed and operated by B & S Massey Limited operate at approximately five times the angular velocity of those offered by V.S.I., for a similar range of axial force. Also B & S Massey Limited do not appear to be interested in the provision of a facility to adjust the *wobble* angle during the process or to cause planetary motion to occur at the upper die-workpiece interface. Axial feed rates or axial power available varies widely, as does the forging capacity in terms of axial force available compared with total machine weight.

It is also noted that both B & S Massey Limited in their 'Rotaform' machine and the Warsaw Technical University, Poland, constrain to zero the angular velocity of upper platen about the axis of the machine and workpiece. This constraint induces skidding at the upper die-workpiece interface as described by Standring and Appleton [27].

The magnitude of the forging parameters chosen by previous workers for both research and optimum operation appears to vary widely. For example, the workers at the Polish Technical University, Warsaw, V.S.I. Automation, U.S.A., Kubo et al, Government Industrial Research Institute, Japan, Little and Beyer, University of Michigan, U.S.A., and Kobayashi, et al, Technological University of Nagaoka, Japan, all appear to have conducted their experiments with the upper and lower platen axes inclined at an angle, $\theta = 2^\circ$. However, B & S Massey Limited appear to have selected 4° as the optimum angle at which to operate. However, Slick's wheel mill originally operated with, $\theta = 10.66^\circ$ and current machines manufactured to his basic design remain unchanged in this respect. They also continue to operate at relatively low axial feed/revolution which results in small percentage areas of contact and thus high ratios of advantage when the force required is compared with that for conventional forging.

It is not clear why industry has not been highly motivated to rapid development of a process which appears to have such obvious advantages. One reason may be the established structure of industry, whereby metal forming and the manufacture of components to high precision are deemed to be separate areas of expertise. Rotary forging has capabilities which are applicable to both areas

and therefore may be awaiting a revision of management attitudes in order to realise its full industrial potential.

Findings from work in the academic field seeking to establish relations between workpiece deformation and various rotary forging parameters are of particular interest when study is made of the values of various forging parameters selected for use in industry and for research purposes. Influence of the inclination of the platen axes, θ , upon the axial force required may be assessed from theories which have been demonstrated to be appropriate by experimental work. For example Kubo [34] concluded that satisfactory correlation was obtained between values of axial force, F_E , determined by experiment and the value predicted, F_K , which can be expressed as a function of die-workpiece contact area, A_K , a constraint factor, C , and the flow stress, σ , of the billet, such that

$$F_K \cong \sigma A_K C \quad (1)$$

also reference [34] shows that

$$A_K \cong \pi r^2 (0.48f/2r \tan \theta)^{0.63} \quad (2)$$

where

r = current workpiece radius

f = current axial feed/rev

θ = angle of inclination of the platen axes

Thus the current axial force, F_K , is approximately proportional to $r^{1.37}$ and $f^{0.63}$

From the complete procedure described ^{by} Hawkyard et al [24], the predicted axial force, F_H , may be considered to be approximately proportional to $\gamma, k, r^{1.5}, f^{0.5}/\theta^{0.5}$, where γ = a pressure

factor = $P/2k$, P = normal pressure at the die-workpiece interface
and k = current yield shear stress = $\sigma/2$.

For the purpose of approximation it should be noted that allowance for the effects of elastic deformation of the die as proposed in Reference [24] were omitted.

Results from experiments presented in Reference [24] include the axial forces, F_E , measured at specific values of natural compressive strain, using cylindrical specimens of various initial diameters. The specimens for the experiments were manufactured from two materials, namely aluminium alloy and commercially pure aluminium. The experiments were also conducted at various values of both axial feed per revolution, f , and angles, θ , ranging from 2° to 8° , and the results presented were obtained from specimens of various height/diameter ratio. Correlation is shown to exist between the predicted values of axial force and those obtained from experiments. By assuming a constant pressure factor, γ , and current yield shear stress the axial force may be stated to be approximately proportional to $r^{1.5}f^{0.5}/\theta^{0.5}$.

In an attempt to identify the influence of inclination of platen axes upon axial force required the above approximations from relevant theory, together with data from the experiments conducted with specimens of various initial diameter, height/diameter ratios. and values of axial feed/rev, were used to compare $F_E/r^{1.5}f^{0.5} - \theta$ and $F_E/r^{1.37}f^{0.63} - \theta$, relations.

The assumption is made that the true compressive stress-natural

strain characteristics are similar for all specimens of a given material, although of differing diameter and also that the strain rates were similar in all cases. The theory proposed by Hawkyard et al [24] is stated to be applicable to discs and therefore the results from experiments where the initial specimen height/diameter ratio, $H_0/D_0 > 0.20$ were excluded. The $F_E/r^{1.5}f^{0.5}$ - inclination of axes, θ , and $F_E/r^{1.37}f^{0.63}$ - inclination of axes, θ , relations are presented in figure 11 and figure 12 respectively together with the values $1/\theta^{0.5}$ and $\cot^{0.63}\theta$ as appropriate.

The experiments described by Hawkyard et al [24], using commercially pure aluminium specimens of differing initial radii, were conducted at similar feed rates. The angle, θ , was the forging parameter varied. The values of axial forces measured during the experiments are stated in Table 2, together with the values $F_E/r^{1.5}f^{0.5}$ for various values of θ , and the relation is presented in figure 13. Similar values were computed, but based on the method of estimating axial force proposed by Kubo et al [34], and the results are presented on the graph figure 14, together with $\cot^{0.63}$, plotted on an arbitrary scale.

Prabhu, et al [59], report their work on an orbital riveting attachment, and present results from experiments on the upsetting of cylindrical specimens which represent rivet heads.

Experiments were conducted at various rates of axial feed per revolution on mild steel specimens of similar initial radius, and the values of axial force measured are tabulated in Table 3. The theory proposed by Kubo et al [34] was then used to study the results for the relation between axial force, axial feed rate and

inclination of platen axes. The results are stated in Table 3 and presented graphically in figure 15.

It can be concluded from the relations shown in figures 11, 12, 13, 14 and 15 which are based upon experimental results and the appropriate theories, that when the criterion is to achieve deformation with minimum axial force, then for the range of inclination of platen axes investigated the advantage of rotary forging over conventional forging increases as the angle, θ , increases. This was also shown previously in the work with models by Slater et al [19] and is of particular interest when consideration is given to the value of the forging parameters selected by various workers in the field of rotary forging. For example, it is not clear why values of 2° or 4° are chosen by recent workers in both industrial and academic fields in preference to say, 8° or 10° . It can only be concluded that in recent designs the objectives are not related to minimising the axial force required.

For investigation of the influence of current workpiece radius upon axial force, the approximations may be expressed as follows :

- (a) From the method proposed by Hawkyard et al [24] and from theories based upon both a plastic hinge, together with indentation, and excluding allowances for elastic deformation, of the die, the axial force F_H , may be written approximately as

$$F_H \approx C_H r^{1.5} \quad (3)$$

where C_H = coefficient determined from specific values

of σ , θ , and f .

- (b) From the method proposed by Kubo, et al [34] based upon area of contact at upper platen workpiece interface, the axial force F_K may be written approximately as

$$F_K \cong C_K r^{1.37} \quad (4)$$

where C_K = coefficient determined from specific values of C , σ , θ , and f .

- (c) For a method based upon axisymmetric compression of a cylindrical specimen, of current radius, r , material flow stress, σ , subjected to quasi-static homogeneous deformation, the axial force, F_A , required to produce a reduction in height is given by

$$F_A = C_A r^2 \quad (5)$$

where $C_A = \pi \sigma$

If the same specific values of σ , f and θ , are substituted to evaluate equations (3), (4) and (5) then from equations (3) and (5)

$$F_H/F_A \cong C_H/r^{0.5}C_A \quad (6)$$

from equations (4) and (5)

$$F_K/F_A \cong C_K/r^{0.63}C_A \quad (7)$$

from equation (6)

$$\frac{d(F_H/F_A)}{dr} \cong -0.5C_H/r^{1.5}C_A \quad (8)$$

and from equation (7)

$$\frac{d(F_K/F_A)}{dr} \cong -0.63C_K/r^{1.63}C_A \quad (9)$$

From both the approximations (6) and (7) it can be seen that the ratio of the force required for rotary forging to the force required for conventional forging decreases as the value of r , increases. The two methods of analysis produce results which have a degree of similarity and the same general conclusion is apparently applicable.

Equations (8) and (9), are a convenient method of expressing the manner in which the advantage of the process may be predicted to increase as the workpiece radius increases. This reasoning goes far to explain the industrial significance of the early inventions, by those such as Slick [2] and Brindley [4] who were confronted with problems of forming large workpieces. An example of the influence of workpiece radius upon the advantages of the process can be shown by consideration of workpiece radii in the ratio of 1:10, say 0.05 m and 0.5 m respectively. The ratio of the workpiece areas is therefore 1:100 and would represent the ratio of axisymmetric compressive forces required to cause upsetting. However, from Kubo et al [34], the estimated ratio of the axial forces required, assuming a similar axial feed/rev, in each case would be $0.05^{1.37}:0.50^{1.37} = 1:23.44$. It should also be noted that the workpiece sizes with which Slick was concerned are large in radius when compared with those applicable to machines of more recent design.

From the study of previous relevant work it is considered by the author that due to the number of variables which influence the rotary forging process it would be advantageous for comparison purposes to express, in non dimensional form, the axial force

characteristics arising from specific rotary forging conditions, and for stated values of the forging parameters. The comparison to be made with the force to cause quasi-static axisymmetric homogeneous compression of a similar workpiece. It is proposed to introduce the use of a term to be known as the *specific rotary forging force*, denoted by, ζ , and to be referred to as either the experimental *specific rotary forging force*, ζ_E , determined by experiment or the predicted *specific rotary forging force*, ζ_P .

Thus the experimental *specific rotary forging force* is

$$\zeta_E = F_A/F_E \quad (10)$$

where F_A = the predicted axial force required to cause a reduction in height due to plastic deformation of a given workpiece by quasi-static, axisymmetric compression.

F_E = the axial force determined by experiment to cause a reduction in height by plastic deformation of the given workpiece by rotary forging.

Similarly the predicted *specific rotary forging force* is given by

$$\zeta_P = F_A/F_T \quad (11)$$

where F_A = the predicted axial force required to cause a reduction in height of a given workpiece by quasi-static, axisymmetric compression, and

F_T = the predicted axial force required to cause a reduction in height due to plastic deformation of the given workpiece by rotary forging.

The above could, of course, be expressed as the inverse. The method proposed is chosen for manufacturing engineering reasons

because increase in the value expressed indicates increase in advantage gained by using the process.

In figure 7(b) of Reference [38], Little & Beyer show the result of an experiment on upsetting specimens of steel to A.S.A. 1050, with $H_0/D_0 = 1.0$. Assuming the foot print techniques for predicted axial forging force to be applicable the results would give values :

$$F_E/F_T = 1.54 \text{ when } \epsilon = 0.15$$

and
$$F_E/F_T = 1.0 \text{ when } \epsilon = 0.83$$

The corresponding values of predicted *specific rotary forging force* ζ_p , are 1.6 and 2.0 respectively.

Kubo et al presents experimental results in figure 15 of Reference [34] which indicate that the work was carried out at typical values of experimental *specific rotary forging force*

$$\zeta_E = 1.8 \text{ and } \zeta_E = 2.25$$

at values of $\epsilon_z = 0.15$ and 0.50 respectively.

The corresponding values of F_E/F_K are calculated to be 0.72 and 0.75 respectively thus providing over estimates which would be satisfactory for most manufacturing engineering purposes.

From the above it is concluded that objectives of previous work have not been related to investigation of characteristics of the process at values of say $\zeta_E > 10$, thus representing values of *specific rotary forging force* which provide a marked advantage over conventional forging with regard to axial force required.

The work by Hawkyard et al [24] was carried out at experimental values of *specific rotary forging force* typically within the range 5.7 to 10.5.

In general terms, to operate experimentally at values of *specific rotary forging forces* $\zeta > 10$ requires typically $\theta \approx 10^\circ$ or $r/f > 250$.

The design features of a number of machines are listed for comparison purposes in Table 4. In an attempt to identify trends, the ratios between specific parameters are also stated. From characteristics of the machines listed it can be seen that large variances occur for ratios such as maximum axial force available/maximum workpiece area, etc. The large differences may be reconciled by consideration of the variations in operating parameters used, such as the angle of inclination of axes and the effect of maximum workpiece radius. This may be seen from the reduction in the variances which result from estimating the average induced stress at the upper die-workpiece interface, using the method of estimating axial force proposed by Kubo et al [34]. The large variances which occur in the ratios of maximum axial force available/machine weight are of interest, but the author is unable to suggest reasons why these have occurred, other than being associated with rigidity problems and operating speeds.

The variations in maximum workpiece area/power may arise from the rate of deformation required to achieve the desired production rates. The maximum rolling velocities which occur on both the Slick Mill and 'Rotaform' are similar, and are some six times greater than those specified on the PXW 100A and the OFP 100.

The effects of the skidding and slipping phenomena between the upper die and workpiece, as described by Standring and Appleton [27], are inherent in certain contoured upper die-workpiece interface

configurations. However, the induced precessional skidding which arises from the rotational constraint of the upper platen relative to the workpiece, is not inherent. The necessity to induce skid and slip arises only when the characteristics to be formed by the upper die in the workpiece have significance in terms of radial position. For example, gear teeth, radial grooves or keyways.

Appraisal of the foregoing confirms that in development of rotary forging to date, differences in objectives or opinions have arisen when optimising the process or the design features of rotary forging machines. These have led to differing design concepts and also wide variations in the range of values selected to be available as controllable forging parameters for use during the operation.

Factors relating to design of machines which have been identified from previous relevant published work may be summarised as follows :

- (a) Importance is attached by some designers to the feature of a stationary workpiece. The ensuing complexity of machines to achieve this objective indicates that the feature is assessed as being of great significance.
- (b) Certain design concepts accept that it is satisfactory to induce skidding and slipping at the upper die-workpiece interface of a magnitude greater than that which is inherent in the process. The total relative displacement at the die-workpiece interface may well exceed that

experienced in conventional forging. Although this feature clearly manifests itself in additional die wear and possible undesirable surface roughness phenomena, it is presumably deemed by some designers to form part of an economic design concept.

- (c) There would appear to be differing conclusions regarding the necessity to provide a facility to vary the magnitude of the angle of inclination of the platen axes, θ , during the forging process.
- (d) From the small value of, θ , chosen by designers for the operation of certain recent orbital type machines, presumably to minimise the resulting dynamic imbalance, there would appear to be limited motivation to minimise the axial force required to form a workpiece of a given size. This comment is of particular relevance, when consideration is given to the period of time during which empirical and theoretical design information has been available.
- (e) Angular velocities and, hence, average rolling velocities, together with axial feed per revolution, used by various workers vary widely.
- (f) With respect to axial force the advantage of the rotary forging process when compared with conventional forging increases as workpiece diameter increases.

From the foregoing review of previous relevant published work, the desirability for further study of rotary forging machine design becomes evident. However, to draw conclusions which are definitive with respect to the design specification and development objectives related to the work described in this thesis is apparently more difficult.

The previous work on rotary forging techniques applied to powder metal technology suggests that further investigation is required to improve knowledge of the process with respect to achieving high average relative density and reducing the variation of density which occurs during compaction. From discussion with industry, the failure to achieve uniform densities appears to be the most significant of current problems. Density variations manifest themselves in problems of both dimensional accuracy control and variation of physical properties of the workpiece. The achievement of desired density and dimensions, together with identification of limitations of the process are also objectives. The advantages of rotary forging when compared with uniaxial compression in the cold post-sintered densification process are of paramount importance.

The theory which has been shown to be applicable to both compaction and post-sintered densification when applying uniaxial compression should be investigated for possible application to compaction and post-sintered densification by rotary forging.

A satisfactory method for the manufacture of hollow cylindrical specimens can be stated as essential or alternatively a prerequisite

to any industrial potential of rotary forging associated with powder metal technology.

CHAPTER 3 FORMULATION AND STATEMENT OF THE DESIGN REQUIREMENTS

3.1 Determination of requirements

One of the objectives derived from the need for the work stated in Chapter 1, Section 4, was to study, from the fundamentals of the process, the design of a rotary forging machine to be constructed for the purpose of conducting experiments, which represent directly industrial applications of the process. The machine was to be designed, manufactured and installed in the Department of Mechanical Engineering, at The City University, London. For the design process to proceed, it was necessary to establish the range of forging parameters to be available, performance criteria, and the desired operating conditions.

When formulating the statement of requirements, it was necessary to recognise the constraints relating to limitations of resources in terms of both finance and manufacturing facilities.

3.2 Axial force required

Consistent with the stated objectives relating to both metal forming and powder metal technology, the first and most significant requirement determined was the magnitude of the axial force to be available.

From the financial constraints it was necessary to minimise construction costs which were clearly related to the workpiece dimensions and material properties. It therefore became necessary to determine the minimum acceptable final workpiece

diameter, height and physical properties whilst remaining consistent with the objective to make available a rotary forging machine suitable for research relating directly to industrial applications of the process. From consideration of requirements for powder metal compaction it became apparent from previous work, Reference [39], that a machine with adequate axial force capacity to plastically deform metal specimens of a given size would also be adequate to compact and cause post-sintered densification of equivalent specimens in powder metal. When attempting to establish machine capacity in terms of axial force requirements it was recognised that rotary forging is inherently suitable for the manufacture of axisymmetric components of small height to diameter ratio, and must commence with the workpiece in a billet or other preshaped form. The cost associated with workpiece preparation and workpiece handling tends to be lowest in those metal shaping processes in which the raw materials for the components are fed in bar, strip or wire form. For example in metal cutting machines, such as automatic or capstan lathes with bar feed, the workpiece is *parted-off* as an operation within the manufacturing cycle, often requiring to be re-set for a second operation. The cost of material wastage due to cutting to the desired shape and *parting-off* tends to be offset by the reduction in handling costs.

Metal cutting machines with bar feed tend to become problematic when diameters exceed 50 or 60 mm. The problems are associated with costs of both material wastage and handling difficulties arising from workpiece weight. Additional problems arise from the rotating mass of bar stock material which is required to overhang the headstock, and therefore require support and

suitable steadies. When diameters exceed those stated above the tendency is to use chucking machines which also require the workpiece to be in a billet or some other pre-shaped form. It was therefore proposed that at these and larger workpiece sizes rotary forging becomes competitive in industrial terms for component manufacture by metal forming. From this reasoning it was concluded that minimum acceptable machine capacity would be that associated with cold forming a low carbon steel workpiece to a final diameter of 50 mm.

Initial estimates of the axial force requirements for the machine were made by using the results from work by Appleton and Slater [21], the specific purpose of which was to aid design. They provide data obtained from experiments over a range of forging parameters with a model machine using specimens of true compressive stress-natural compressive strain characteristics which may be considered as model material. From the data presented in [21] and the assumption that the axial force required for the upsetting process is proportional to the flow stress, it was possible to estimate the axial force required to upset a workpiece of known true compressive stress-natural compressive strain characteristic of a given current radius.

From the work by Slater et al [19] the mean quasi-static yield stress, $\bar{Y}$, of the model material (lead) was found to be 2180 lbf/in² (15.03 MPa). The rotary forging experiments were carried out on cylinders of 25.4 mm diameter, $H_0/D_0 = 1.0$, with angle of inclination of platen axes, $\theta^\circ = 10^\circ$. From the graphs presented in Reference [19] showing the relationship between

fractional reduction and the number of revolutions of the upper conical platen, it can be seen that for values which are outside the threshold of the mushrooming phenomenon, after say 20 revolutions, with the application of a constant axial force of 1512 lbf (6.72 kN), the fractional reduction in height was approximately 0.5. Assuming the volume of the workpiece to be constant and deformation to be homogeneous, then at 0.5 fractional reduction in height the workpiece surface area is given by

$$\pi 25.4^2 / 4 \cdot 0.5 = 1013.4 \text{ mm}^2$$

or a current diameter $\approx 36 \text{ mm}$.

Consider a workpiece of annealed low carbon steel with a quasi-static flow stress of 550 MPa which is similar in size and shape to the lead specimens described and that similar rotary forging parameters are used to cause a fractional reduction in height equal to that caused in the experiments on model materials. Then, assuming the axial force required to be proportional to the workpiece quasi-static flow stress, the estimated axial force required to upset the low carbon steel specimen is estimated to be given by

$$6.72 \cdot 550 / 15.03 = 245.91 \text{ kN}$$

The machine is required to be capable of causing upsetting when the workpiece is 50 mm diameter (area = 1963.5 mm²).

Assuming the axial force required to be proportional to workpiece area, then the estimated axial force required for a low carbon steel workpiece, with a flow stress of 550 MPa and 50 mm final diameter is given by

$$245.91 \cdot 1963.5 / 1013.4 = 476.3 \text{ kN} \approx 0.476 \text{ MN}$$

The above result assumes the axial force required to be proportional to r^2 and represents an over estimate since it has

been shown by previous work that the axial force required is proportional to either :

$r^{1.5}$ from Reference [24]

or $r^{1.37}$ from Reference [34]

Alternative methods of estimating the maximum final workpiece radius may be made using the work on model materials in Reference [21] and applying the theories proposed in [24] and [32].

From the above reasoning and estimates, it is concluded that to make available a machine with an axial force capability less than 0.5 MN, would be unacceptable in terms of the objectives stated, and, also, that any greater capability would result in an unnecessary increase in the cost of construction.

The design requirement was therefore stated as a machine to be capable of producing an axial force of 0.5 MN, and adjustable to any desired value down to 25.0 kN.

3.3 Maximum workpiece diameter

From the previous estimate, an axial force of 0.245 MN is required to rotary forge a workpiece of annealed low carbon steel of 36.0 mm final diameter. This force is applicable when forging is carried out with platen axes inclined at $\theta = 10^\circ$, as used in the experiments by Slater et al [21]. From this data, together with the method of estimation of axial force proposed by Hawkyard et al [24] and Kubo et al [34], it is possible to estimate the maximum workpiece diameter for an axial force of 0.5 MN. From rearrangement of equation (3) the estimated

maximum final diameter may be calculated from :

$$(0.500/0.245)^{0.67} 36.0 = 57.89 \text{ mm}$$

From the ratio of the axial forces and by rearrangement of equation (4), and the ratio of the axial forces $0.500/0.245$, the estimated maximum final diameter may be calculated from

$$(0.500/0.245)^{0.73} 36.0 = 60.60 \text{ mm}$$

From the three methods of predicting the machine working capacity may be expressed in terms of maximum workpiece diameter for cold upsetting low carbon steel, and that 50 mm may be considered as an under estimate and 60.60 mm an over estimate. The workpiece capacity estimated is applicable to a machine with an axial force capability of 0.5 MN and angle of inclination of platen axes, $\theta = 10^\circ$. Since work may be carried out on materials of a flow stress much less than 550 MPa, it was decided to provide a facility for a final workpiece diameter not exceeding 150 mm.

3.4 Maximum workpiece height and axial displacement required

It was desirable for the machine to be loaded and unloaded manually, which limits the weight to be handled. Since the final workpiece diameter is limited to 150 mm the initial workpiece could be considered to be restricted to 100 mm diameter at $H_0/D_0 = 1.0$ and therefore 100 mm high. By allowing for a lower platen tooling height/workpiece height loading space ratio of 1:1.5, an axial stroke of 150 mm is required.

3.5 Power requirements for rectilinear drive

To estimate the power level of the axial force and displacement system, it was considered that a workpiece be reduced in height

at the rate of 10 mm/s at maximum axial force. The power required for a maximum force of 0.5 MN is therefore

$$5 \cdot 10^5 \cdot 10^{-2} = 5 \cdot 10^3 \text{ Joules/s} = 5 \text{ kW}$$

The requirement of both force and displacement adjustment leads rapidly to the conclusion that a hydraulic system is required. From consultations with hydraulic system suppliers, and from recognition of efficiency and the power requirements to control the velocity of axial displacement by flow restrictors, it was concluded that, from the available range, a standard hydraulic power unit rated at 7.5 kW was the most appropriate.

3.6 Power required for rotational drive

Difficulty was experienced using published data relating to previous forging machines for the purpose of estimating the power and range of velocities required for rotation of the die relative to the workpiece. The power available for rotation of the upper die relative to the workpiece in the model work by Appleton and Slater [20] was ≈ 0.20 kW. This information was used as a basis for estimating by assuming the power necessary to be proportional to both current workpiece radius and flow stress. From these assumptions, together with the workpiece radius and flow stress ratios from pages 48, 49, the estimate of power required for rotational drive is

$$0.20 \cdot 550.25 / 15 \cdot 03.18 \cdot 0 \approx 10.16 \text{ kW}$$

The estimate of power required for rotation of the platens may also be considered by extensions of the work by Appleton and Slater [21] where, E, the total energy required to cause a fractional reduction in height of the workpiece by upsetting,

where deformation is assumed to be homogeneous and the material to be rigid-perfectly plastic.

From reference [21]

$$E = V\bar{Y}\epsilon = V\bar{Y}.\ln[1/1(1 - R)] \quad (12)$$

where $\bar{Y}$ = mean quasi-static yield stress,

V = volume of the cylinder,

ϵ_z = compressive natural strain

R = fractional reduction in height

$$\text{and} \quad E = E_t + E_r \quad (13)$$

where E_t = translational energy supplied

and E_r = rotational energy supplied

The average power required can be expressed

$$E/T = (E_t/T) + (E_r/T) \quad (14)$$

where T is the time taken to cause the fractional reduction in height.

When determining the axial power requirements, values were assumed for translational velocity and mean yield stress of the workpiece to be 0.01 m/s and 550 MPa respectively.

For a workpiece, $r = 0.025$ m, and $H_0/D_0 = 1$

then $V = \pi \cdot 0.025^2 \cdot 0.05 = 9.81 \cdot 10^{-5} \text{ m}^3$

and for a fractional reduction of 0.5 the energy required is

$$\begin{aligned} E &= 9.81 \cdot 10^{-5} \cdot 550 \cdot 10^6 \cdot 0.693 \\ &= 3.74 \cdot 10^4 \text{ Joules} \end{aligned}$$

at 0.01 m/s the time, T , taken would be

$$T = 0.050 \cdot 0.5 / 0.10 = 2.5 \text{ s}$$

therefore total power required

$$= 3.74.10^4./2.5$$

$$= 14.96.10^3.\text{Joules/s say 15 kW}$$

the translational power is given by force x velocity

$$= 0.5 \text{ MN} \times 0.01 \text{ m/s}$$

$$= 0.5.10^6.0.010 = 5.10^3\text{J/s} = 5 \text{ kW}$$

The rotational power required is given by

$$E_r = E - E_t \quad (15)$$

$$= 15 - 5 = 10 \text{ kW}$$

The final diameter of 70.70 mm due to the fractional reduction of 0.5, exceeds the estimated maximum capacity for the proposed machine for workpiece material of 550 MPa yield stress. Since platen rotation would be gear or belt driven it is unnecessary to make large allowances for inefficiency. The rate of fractional reduction used for the estimate is very high compared with that used by workers to date and suggests the power requirement calculated may be considered as an over estimate.

A further method was to refer to specifications of machines of similar workpiece capacity intended for use in industry such as those in the V.S.I. range. This indicated that 10 kW was adequate power for rotational drive of the upper platen.

The requirement for rotational power was therefore concluded to be 10 kW and the nearest standard unit available is rated at 11 kW.

3.7 Speed of platen rotation

The work on model materials by Slater and Appleton [19], was

conducted at 80 rev/min on specimens up to 20 mm radius. In the absence of other design criteria, maximum rolling velocity was the parameter selected for comparison with machines of recent design. From Table 4 it can be seen that for the machines listed of a similar maximum workpiece radius capacity, the maximum rolling velocities range from 47.1 m/min to 471 m/min. Since this range is related to maximum speeds, it is clear that a requirement to cover the complete range would be problematic if an allowance is also made for a *turn down ratio* to be applied to the lower values resulting in speeds variable from, say 5 to 471 m/min. It was therefore concluded that the requirement was to design the machine to be capable of a rotational speed equivalent to 471 m/min maximum rolling velocity at the periphery of a 150 mm diameter workpiece. For the machine to be constructed initially it was decided to provide a maximum rolling velocity of approximately 100 m/min at a workpiece diameter of 100 mm, which is equivalent to a rotational speed of say 300 rev/min. Provision to be made in the design to operate at 1000 rev/min by an appropriate drive mechanism modification.

For the initial machine it was concluded that rotational speed should be adjustable over a range of at least 10:1, that is, 300 rev/min to 30 rev/min and to be infinitely variable. This is in excess of the rotational speed available on the machines type PX 100A and OFP 100, as stated in Table 4 but less than that used on the 'Rotaform'.

3.8 Workpiece ejection

A facility to eject the workpiece from a contoured lower die was considered to be essential. The ejection force available to be adjustable over the range 20.0 kN to 0.2 MN.

3.9 Inclination of platen axes

The range of adjustment required for inclination of the platen axes, θ , was decided to be 15° . It can be seen from Table 4 that this exceeds the range used by previous workers. The decision to make 15° available was subsequently found to be prudent following the decision to investigate the double conical platen concept of rotary forging. A further requirement was to provide a facility to vary the angle, θ , within the range available, during and without interruption of the forging process.

3.10 Manually operated controls

For a machine designed primarily to be used for research purposes, it was a requirement to control the magnitude of the axial force applied, and also the direction and velocity of the translational displacement. These parameters to be controllable over the widest practicable range. It was concluded that a hydraulic system provided the only feasible solution and was therefore specified as a design requirement. Stops are required to limit rectilinear displacement in both directions, under the conditions of maximum axial load. The upper and lower stops to be adjustable independently to any desired position within the

range of axial displacement.

Provision was required for manually operated controls for :

- (a) start and stop of hydraulic power unit,
- (b) magnitude of axial force to be applied in both directions,
- (c) velocity of axial displacement in both directions,
- (d) direction of axial displacement,
- (e) position of axial displacement limit stops,
- (f) start and stop of rotational drive,
- (g) inching of the rotational drive,
- (h) angular velocity of rotational drive,
- (j) adjustment of the angle at which the platen axes are inclined,
- (k) ejection mechanism,
- (l) emergency stop,
- (m) access door to forging zone.

3.11 Instrumentation

Instrumentation is required to provide facilities for visual monitoring and analogue recording of the following parameters :

- (a) axial force for both forging and ejection,
- (b) torque applied to cause rotation of the upper die in relation to the workpiece,
- (c) rectilinear displacement of lower platen,
- (d) inclination of platen axes,
- (e) angular displacement of upper platen about its axis,
- (f) angular displacement of the lower platen about its axis,

- (g) rotational velocity of upper platen,
- (h) rotational velocity of lower platen,
- (j) pressures of main hydraulic ram inlet port.

3.12 General requirements

The requirement was for the kinematics of the machine to be such that the relative motion and forces at the die-workpiece interface conform to the process known as rotary forging. The design concept to be such that operating configurations could be varied and the construction modular to permit extensive modifications to be practicable by sub-assembly replacement. The machine was required for research purposes, and therefore the forging zone should be both readily accessible and visible during operation. The machine was required to be suitable for cold forging, warm forging, hot forging and the compacting of powder metals. The machine was required to have facilities for mounting suitable tooling with interchangeability on upper and lower platen location and fixings.

The machine was required to be self contained and to operate when energised by a 440v, 50Hz, 3-phase power supply, and to stand on a concrete floor occupying the minimum total area. No specific constraints were applied to, or specified for, the overall dimensions and weight but both were required to be the minimum practicable consistent with meeting all other requirements. The design to be such that all main sub-assemblies could be removed without dismantling the complete machine. The machine was also required to be suitable for investigating the range of forging parameters specified without interaction between

individual adjustments and without interruption of the forging process.

Wherever practicable, proven proprietary components would be incorporated, consistent with minimising total cost and development time.

3.13 Operating environment

The machine was required to operate in a workshop environment which is protected from weather and at an ambient temperature in the range 15°C to 20°C. The machine should not be deranged when subjected to ambient temperatures in the range 0°C to +25°C.

3.14 Ergonomics

The tooling was to be capable of being man-handled into position and fastened by a single operator. The machine was to be manually operated including the loading and unloading of workpieces, and is therefore required to be acceptable from the ergonomic aspects. It is also essential for the operator of the machine to be able to observe the specimen, and the appropriate instrumentation provided for visual monitoring, whilst controlling both the forging and ejection phases of the process. The following requirements therefore arise :

- (a) The workpiece loading station to be between 1 m and 1.25 m from floor level.
- (b) Controls and visual monitoring equipment to be positioned such that the operator of the controls would also observe

the process throughout.

3.15 Safety

The combination of large forces and a powerful mechanism, with both rotational and translational displacement, together with rapid response, renders a rotary forging machine potentially dangerous to operators and observers. In accordance with the Health and Safety at Work Act 1974, it was required that all reasonable measures would be taken to ensure that the machine would not be a hazard to persons, nor cause an adverse effect upon the environment in which they work. It is clear that since the machine was required to be manually loaded, unloaded, and operated, then all possible precautions must be taken to protect operators against human error.

The design requirement was therefore that all power operated moving mechanism would be suitably guarded. Access to the working zone to be possible only through a gate provided at the workpiece loading and unloading station with interlock of controls to be such that all power is removed from the machine operating mechanism when the access gate is in any position other than fully closed. Isolators, fuses, alternative emergency stop switches and a key operated electrical switch lock to be provided. All pressures in the hydraulic system to be released immediately as a result of power failure or emergency stop.

It was specified that any component of the machine with a weight greater than 300 N would be provided with suitable eyebolt fixing for lifting purposes.

4.1 Fundamental kinematics of the rotary forging process

Consider a conical upper platen of semi-angle $(\pi/2) - \theta$ about an axis Z_2 , which is inclined at an angle, θ to the axis Z_1 as illustrated in figure 16.

The axes Z_1 and Z_2 intersect at the point 0. Let the cylindrical workpiece be plastically deformed due to indentation by the upper conical platen which is caused by the application of a force, F , in the direction of the axis Z_1 as illustrated. The lower platen and workpiece are caused to rotate about the axis Z_1 which induces the conical surface to roll on the workpiece upper surface, thus causing the upper conical platen to rotate about the axis Z_2 . The plastically deforming region will be caused to rotate relative to the workpiece as illustrated in figure 16. If the lower platen is also subject to rectilinear displacement in the direction of the force, F , towards the plane of the generator, Or , then, as the process proceeds, the cylindrical workpiece will be reduced in height, and assuming the workpiece volume to remain constant, the diameter will be increased.

Consider a point, P , in a plastically deforming region which is at a radius r_1 and rotating about the vertical axis Z_1 , of the workpiece in the plane Or , at an angular velocity ω_1 as illustrated in figure 16. The instantaneous tangential velocity v_1 , of a point, P , is given by

$$v_1 = \omega_1 r_1 \quad (16)$$

Let the point, Q, on the surface of the upper conical platen be at a distance r_2 from the axis Z_2 and rotating at an angular velocity ω_2 about the Z_2 axis. The instantaneous tangential velocity v_2 , of the point, Q, is given by

$$v_2 = \omega_2 r_2 \quad (17)$$

Assuming that perfect rolling occurs at the interface between the workpiece and that the points, P, and Q, become coincidental then

$$v_1 = v_2$$

$$\text{or} \quad \omega_1 r_1 = \omega_2 r_2 \quad (18)$$

$$\text{but} \quad r_2 = r_1 \cos \theta \quad (19)$$

From equations (18) and (19)

$$\omega_1 / \omega_2 = \cos \theta \quad (20)$$

Thus, the plastically deforming region may be caused to rotate about the axis of the workpiece, with no slip occurring in the plane Or, by any combination of angular velocities which satisfy equation (20).

4.2 Kinematics at upper die-workpiece interface

When perfect rolling ceases to occur, then relative motion in the form of sliding takes place at the upper die-workpiece interface. Frictional forces at the interface between the elastically and plastically deforming regions of the workpiece and the elastically deforming regions of the upper die, together with the directions of workpiece material flow, gives rise to deformation which is inhomogeneous and non-steady state. The complexity of deformation is such that present knowledge does not permit rigorous analysis of the interface relative velocities. However, the analysis which follows, is intended to assist the design

synthesis by considering the kinematics at the upper die-workpiece interface in the plane containing the axes Z_1 , Z_2 , and the generator Or . The assumptions are made that plastic deformation of the workpiece material does not influence interface relative velocities, and also that interface relative velocities due to elastic deformation are negligible.

With reference to figure 17, consider the generator OA , which is at an angle, χ , to the axis Z_1 . The axis Z_2 , is inclined to the axis Z_1 , at an angle, θ , and the axes intersect at the point O . The point, P , is at a radius, r_1 , from the axis, Z_1 and the point, Q , is at a radius, r_2 , from the axis Z_2 .

From figure 17, it can be seen that when the points, P , and Q , are considered to be coincident at a point on the generator OA

$$\text{then } r_2 = r_1 \cos[(\pi/2) - \chi + \theta] / \cos[\chi - (\pi/2)] \quad (21)$$

Let the point, P , rotate about the axis Z_1 , at an angular velocity of ω_2 . Let v_p and v_q be the instantaneous velocities of the points, P , and Q , in a direction perpendicular to the plane containing the axes Z_1 , Z_2 and the generator OA .

Then for no slipping or skidding, that is when Or is a neutral

$$\text{plane } v_p = v_q = \omega_1 r_1 = \omega_2 r_2 \quad (22)$$

and from equations (21) and (22)

$$\omega_1 / \omega_2 = \cos[(\pi/2) - \chi + \theta] / \cos[\chi - (\pi/2)] \quad (23)$$

Equation (23) may also be determined from consideration of both the physical diagram and the vector diagram of angular velocities illustrated in figure 18. From this, the neutral plane on which no difference in interface velocities occur, is assumed to be

generated by OA which is perpendicular to the axis Z_3 , and is inclined at an angle $\chi - (\pi/2)$ to the axis Z_1 . The axis Z_2 is inclined at an angle, θ , to the axis Z_1 and the axes Z_1 , Z_2 , and Z_3 , intersect at the point, O. From the physical diagram figure 18, it can be seen that Z_3 must be inclined at an angle $(\pi/2) - \chi + \theta$ to the axis Z_2 .

Let the lower platen axis, Z_1 , be caused to rotate about the axis of the generator OA, Z_3 , at an angular velocity ω_3 , thus causing the lower platen to rotate about its own axis at an angular velocity ω_1 . Simultaneously the upper platen axis Z_2 , is caused to rotate about the axis of the generator OA, Z_3 at an angular velocity ω_3 , thus causing the upper platen to rotate about its axis, Z_2 , at an angular velocity ω_2 .

From the vector diagram, illustrated in figure 18, the vector $\vec{oa}$ is given by :

$$\omega_3 = \omega_1 \cos[\chi - (\pi/2)] \quad (24)$$

the vector $\vec{oa'}$ is given by :

$$\omega_3 = \omega_2 \cos[(\pi/2) - \chi + \theta] \quad (25)$$

For no difference in interface velocities on the generator OA, the vectors $\vec{oa}$ and $\vec{oa'}$ must be equal in magnitude and direction, therefore from equations (24) and (25)

$$\begin{aligned} \omega_1 \cos[\chi - (\pi/2)] &= \omega_2 \cos[(\pi/2) - \chi + \theta] \\ \omega_1/\omega_2 &= \cos[(\pi/2) - \chi + \theta]/\cos[\chi - (\pi/2)] \end{aligned} \quad (26)$$

which is the result obtained in equation (23) and determines the relation between the parameters ω_1 , ω_2 , θ , and χ , for no difference in instantaneous velocities at any interface on the line OA. It should be noted that when

$$\omega_1 = \omega_2$$

$$\chi - (\pi/2) = (\pi/2) - \chi + \theta$$

$$\chi = (\pi/2) + (\theta/2) \quad (27)$$

In figure 17 let the point, P', be a point on the surface of the workpiece and point, Q', be a point on the surface of the upper die and moving at velocities v'_p and v'_q respectively in a direction perpendicular to a plane containing the axes Z_1 , Z_2 and the point N. When the points, P', and Q', are in contact, they form an interface at point N at which the relative instantaneous velocity, v_i , perpendicular to the plane containing the axes Z_1 , and Z_2 , is given by :

$$v_i = v'_p - v'_q \quad (28)$$

The condition when $v'_p > v'_q$, or v_i is + ve, is defined by previous workers, Standring et al [27] as skidding. When $v'_q > v'_p$, or v_i is -ve, the condition is defined as slipping. They also refer to the plane upon which no slipping or skidding takes place as the neutral plane.

The relative interface velocity at a point N, figure 17, may be predicted by considering a point, P', on the surface of the workpiece at a distance, r_1 , from the axis Z_1 , and a distance, z , from the point O, in the direction of the Z_1 axis. A point Q' on the surface of an upper conical die is at a distance r'_2 from the axes Z_2 . The axes Z_1 , Z_2 and the points N, P', Q', lie in the same plane.

From the diagram figure 17

$$r'_2 = r_1 \cos(\beta + \theta) / \cos \beta \quad (29)$$

where

$$\tan \beta = z/r_2$$

and

β is the angle N-O-r

Let the points, P' on the workpiece and Q' on the upper die form an interface at the point N. The instantaneous velocity difference v_i , at the point N on the interface is given by

$$v_i = v_p' - v_q' \quad (30)$$

where

$$v_p' = \omega_1 r_1$$

and

$$v_q' = \omega_2 r_2'$$

therefore

$$v_i = \omega_1 r_1 - \omega_2 r_2' \quad (31)$$

but

$$r_2' = r_1 \cos(\beta + \theta) / \cos \beta \quad (32)$$

and from equation (26)

$$\omega_2 = \omega_1 \cos[\chi - (\pi/2)] / \cos[(\pi/2) - \chi + \theta]$$

then combining equations (26), (31) and (32) gives

$$\begin{aligned} v_i &= \omega_1 r_1 - \omega_1 r_1 \cos[\chi - (\pi/2)] \cos(\beta + \theta) / \cos[(\pi/2) - \chi + \theta] \cos \beta \\ &= \omega_1 r_1 \{1 - \cos[\chi - (\pi/2)] \cos(\beta + \theta) / \cos[(\pi/2) - \chi + \theta] \cos \beta\} \quad (33) \end{aligned}$$

From this it is evident that for $\omega_1 < \omega_2$ skidding occurs when

$(\pi - \beta) < \chi$. This can be stated alternatively as at any

interface which occurs between the upper die and workpiece in the

zone defined by the sector from Z_2 to OA. Similarly slipping

occurs when $\omega_1 < \omega_2$ and $(\pi/2) - \beta < \chi$. It should be noted

that when z is negative the angle β is considered negative.

Due to the frictional forces which occur at the upper die-workpiece interface, the presence of either the slipping or skidding phenomena will cause energy to be dissipated, giving rise to wear of the upper die, and may influence workpiece surface roughness.

Figure 19, illustrates a typical upper die-workpiece interface configuration, and the graphs show the manner in which the angle χ , influences the slipping and skidding velocities for a given value of θ . The curves also indicate the significance of skidding, or slipping, which occurs at the vertical upper die-workpiece interfaces formed at A_1-A_2 , B_1-B_2 , C_1-C_2 . It should also be noted that a change of direction of relative velocity occurs at interfaces C_1-C_2 , when $\chi = 95.71^\circ$ and at interfaces A_1-A_2 , and B_1-B_2 , when $\chi = 84.29^\circ$. The calculations and results are shown in Table 5 and illustrate the basis of a method which may be used to examine a component design and subsequent die configuration to optimise the value of, χ , by control of ω_1 , ω_2 and θ . For purposes of comparison, the relation between ω_1/ω_2 , θ , and χ are shown on a graph in figure 20 from which it becomes clear that these parameters together with the values of z and r should be given consideration during design of components to be manufactured by rotary forging using contoured dies.

The pin and slot mechanism as described by Standring and Appleton in [26], produces cyclic variations in relative velocities at the die-workpiece interface, whereas ring gears produce constant relative interface velocities.

Consider the arrangement shown in the physical diagram figure 21(a) where Z_2 is inclined to the axis Z_1 , at an angle, θ , and is caused to rotate about Z_1 , at an angular velocity of ω_1 . The upper conical platen is also constrained to rotate about Z_2 , at ω_1 . This is the arrangement used in the design by Massey [10] which constrains the rotation of the upper die relative to the workpiece by means of a ring gear.

Let the generator, OA, be at an angle, $(\pi/2)$ to the axis Z_1 and form a line of contact between the upper die and the workpiece.

From the vector diagram in figure 21(b) it can be seen that due to constraint the vector $\vec{oa}$ is equal in magnitude to the vector $\vec{oc}$. The vector $\vec{ob}$ represents the angular velocity at the interface between the upper die and the workpiece with perfect rolling at OA. The direction of rotation of the upper die about the axis Z_2 , is opposite to the direction of rotation of Z_2 , about Z_1 . Thus from the vectors $\vec{ob}$ and $\vec{cd}$ respectively, the velocities of skid, ω_{skZ_1} about the Z_1 axis and ω_{skZ_2} about the Z_2 axis, are given by

$$\omega_{skZ_2} = \omega_1(1 - \cos\theta) \quad (34)$$

$$\omega_{skZ_1} = \omega_1(\sec\theta - 1) \quad (35)$$

The vector diagram figure 21(b) also shows the existence of a small angular velocity of skidding, ω_{skOA} , about the axis OA.

From the above considerations, it is deduced that, to achieve a desired value of, χ , it is necessary to control the three parameters, ω_1 , ω_2 , and θ . It can also be seen from the example that certain upper die shapes could have the total interface relative velocities minimised, by modulation of the ratio ω_1/ω_2 , and θ during the forging process. Consideration was also given by the author to optimising the interface relative velocities by moving the point, O, with respect to the upper die. This concept was not pursued because of complexity of the kinematics. The advantages to be gained from reduction of interface velocities could not be quantified in a manner which would permit cost effectiveness of increased complexity to be assessed.

It is however concluded that to minimise the slipping and skidding phenomena a fundamental difference in design concept to that pursued by previous workers is required. Previous machines based upon orbital, planetary, or fixed inclined axes concepts have provided the facility to control combinations of either ω_1 , and θ , or ω_2 , and θ . It is predicted that the ability to control ω_1 , ω_2 , and θ , as three independent parameters during the process, will assist in minimising die wear, and perhaps assist with control of final workpiece characteristics.

4.3 Alternative configurations which provide the desired kinematics at the upper platen-workpiece interface

Some alternative configurations which are illustrated in figures 22 and 23 are described as follows :

- (a) Figure 22(a) illustrates an arrangement whereby the lower platen together with the workpiece is constrained from rotation relative to the stationary axis Z_1 . The axis Z_2 , is inclined to the axis Z_1 , at an angle θ , and is driven at an angular velocity ω_2 , about the axis Z_1 . The upper conical platen is caused to rotate at an angular velocity ω_2 , about the axis Z_2 . This relative motion is referred to as *wobbling*. The resulting process is known as planetary forging when the angle, θ , is modulated during the process. The axial displacement is provided by the lower platen assembly moving in the direction of the Z_1 , axis and caused by the force F . This arrangement is used in the 'Rotaform' machine Reference [12].

- (b) An alternative arrangement is illustrated in figure 22(b) which differs from the above in the method used to achieve axial displacement by applying a force F , to cause the complete upper platen assembly to move in the direction of the Z_1 , axis, whilst the base of the workpiece remains stationary. This is the basis of the arrangement described by Marciniak [29], with an additional drive applied to cause modulation of the angle θ , and hence provide planetary motion as illustrated in figure 10.
- (c) Figure 23(a) illustrates an arrangement in which the axis Z_2 , is inclined at an angle θ , to the axis Z_1 , and both axes remain stationary. The lower platen together with the workpiece is driven at an angular velocity, ω_1 , about axis Z_1 . Rotation of the upper conical platen about the axis Z_2 , at an angular velocity, ω_2 , is caused by frictional forces at the upper die-workpiece interface. The interface conditions are maintained as deformation proceeds by displacement of the upper platen assembly in the direction of the axis Z_1 . This is the arrangement used by Slick [2]. It can be seen that rotational drive could also be applied simultaneously to the upper platen at an angular velocity ω_2 .
- (d) An alternative arrangement illustrated in figure 23(b) is similar to that described in 4.3(c), and provides both rotational drive and axial displacement to the lower platen relative to the Z_1 , axis. The upper platen is

caused to rotate at an angular velocity, ω_2 , by frictional forces at the upper die-workpiece interface due to the force, F , and when workpiece deformation occurs, causes displacement of the lower platen assembly in the Z_1 , direction. Simultaneous rotational drive to both the upper and lower platens is also practicable with this arrangement.

- (e) Figure 23(c) illustrates an arrangement whereby the upper platen is maintained stationary relative to the axis Z_2 , and the axis Z_1 , is inclined to the axis Z_2 , at an angle θ , and driven about the axis Z_2 , at an angular velocity ω_2 . The lower platen, together with the workpiece, is caused to rotate at an angular velocity ω_1 about the axis Z_1 . Thus the workpiece and lower platen are *wobbling* about the upper conical platen. Axial force and displacement would be required in the Z_1 , direction.
- (f) By the introduction of a further axis say Z_3 , as illustrated in figure 23(d) which shows the axes Z_1 , and Z_2 , to be inclined to the axis Z_3 at angles θ_1 and θ_2 , respectively. In addition, all three axes are arranged to intersect at a point O . It would then be possible to have both the upper conical platen and workpiece, together with the lower platen, *wobbling* about the axis Z_3 . This arrangement would be capable of numerous variations in relative motion at the upper die-workpiece interface by variation of both rotational drive velocities, and control of the phase relationship of angular displacement about the axis Z_1 , and Z_2 . It is clear, however, that the kinematic

complexity of such a system for rotary forging leads to practical problems of such magnitude that, in the absence of very large advantages being foreseen, the configuration is not considered further.

4.4 Inertia forces due to radial acceleration

From a study of the kinematics of the process it becomes clear that the desired relative angular velocities about the axes Z_1 , Z_2 , and the generator OA, can be achieved by numerous solutions with respect to the manner in which various members of the system change their positions in space. Figures 22 and 23 illustrate that the desired relative motion at the upper die-workpiece interface, can be provided by various kinematically acceptable arrangements. However they may give rise to widely differing outcomes with respect to the inertia forces which would arise within the system. It is clearly highly undesirable for forces to exist which result from dynamic imbalance or to induce couples which are not static with respect to the machine.

The various applicable kinematic arrangements were searched for the possible existence of inertial forces arising from the mass and acceleration of components within the machine which cannot be dynamically balanced. From study of the various designs it was concluded that further detailed examination was required in arrangements where relative motion of the axes Z_1 and Z_2 occurs.

Consider an upper conical platen in contact with the conical face of a workpiece, such that the apices of the cones are

coincident at the point, 0, as illustrated in the physical diagram figure 24(a). The axis Z_2 , of the upper platen of semi-conical angle, $\chi - \theta$, is at an angle, θ , to the axis of the workpiece, Z_1 , and the upper surface of the workpiece has a semi-conical angle, $\pi - \chi$. The workpiece is stationary relative to the axis Z_1 , about which the axis Z_2 is caused to rotate at a uniform angular velocity of ω_1 . Perfect rolling is assumed to occur at the upper conical platen-workpiece interface on the generator OA, which is at an angle, χ , to the axis Z_1 .

From the vector diagram of angular velocities figure 24(b)

$$\omega_1 / \sin(\chi - \theta) = \omega_2 / \sin(\pi - \chi) \quad (36)$$

Note that when

$$\chi = \pi/2$$

$$\text{then } \omega_1 / \omega_2 = \sin[(\pi/2) - \theta] = \cos\theta$$

Also from the vector diagram of angular velocities

$$\omega_1 / \sin(\chi - \theta) = \omega_4 / \sin\theta \quad (37)$$

$$\text{or } \omega_4 = \omega_1 \sin\theta / \sin(\chi - \theta) \quad (38)$$

where ω_4 = angular velocity about the generator OA.

Consider an upper platen assembly, including bearings and appropriate bearing housings, to be of total mass, M, and the centre of mass of the assembly to be at the point, B, on the axis Z_2 , at a distance L, from the point, 0.

The force F_{ω_4} due to radial acceleration of the mass, M, in the direction perpendicular to the generator OA and in the plane containing the axes Z_1 and Z_2 , is given by :

$$F_{\omega_4} = \omega_4^2 ML \cos[(\pi/2 - \chi + \theta)] \quad (39)$$

The force is acting in the plane which contains the axes Z_1 and Z_2 , and is therefore rotating about the axis Z_1 at an angular velocity ω_1 . From the vector diagram of forces figure 24(c) the component, $F_{r_{\omega_4}}$ due to the force F_{ω_4} , resolved in the direction perpendicular to the Z_1 axis is given by :

$$F_{r_{\omega_4}} = F_{\omega_4} \sin[\chi - (\pi/2)] \quad (40)$$

The force $F_{z_{\omega_4}}$ due to F_{ω_4} in the plane of the axes Z_1 and Z_2 , resolved in the direction of the Z_1 axis is given by :

$$F_{z_{\omega_4}} = F_{\omega_4} \cos[\chi - (\pi/2)] \quad (41)$$

The mass, M , at point B, is at a distance $L \sin \theta$ from the axis Z_1 about which it is rotating at an angular velocity ω_1 . It is therefore practicable to achieve dynamic balance about the Z_1 axis.

However, it is not practicable to achieve dynamic balance of the rotating mass about the generator OA, and furthermore, the out of balance forces and couples which arise are rotating at an angular velocity ω_1 , about the axis Z_1 . Therefore, these forces and couples are sinusoidal with respect to time, when referred to any plane within a machine based upon this design concept, and accordingly become problematic as a source of forced vibration. It should be noted that these forces are zero when the angular velocities of both the axis Z_2 , and the line OA, about the axis Z_1 , are zero.

To examine the practical significance of such forces, the magnitude of typical forces are estimated in an example as follows:

An orbital type rotary forging machine capable of 0.5 MN axial force is fitted with an upper platen spindle assembly complete with upper die of total mass, $M = 150$ kg. The centre of mass of the assembly is situated on the axis, Z_2 , at a distance $L = 0.3$ m from the point of intersection of the platen axes. The axis, Z_2 , is inclined to the axis Z_1 , at an angle $\theta = 10^\circ$, and the neutral plane is at an angle to the Z_1 axis $\chi = 95^\circ$. The Z_2 axis is caused to rotate about the Z_1 axis at 1000 rev/min. By substitution of the appropriate values in equations (38) and (39)

$$\omega_4 = 18.25 \text{ rads/s}$$

$$\text{and } F_{\omega_4} = 14.93 \text{ kN respectively.}$$

The forces resolved in the r and Z_1 , directions respectively as illustrated in figure 24(c) are

$$F_{r_{\omega_4}} = 1.30 \text{ kN and } F_{Z_{\omega_4}} = 14.87 \text{ kN}$$

The force $F_{Z_{\omega_4}} = 14.87$ kN is acting in the same direction as the maximum forging force 0.5 MN. The additional force applied to the machine is therefore only 3% of the maximum forging force and is consequently assessed as not being very significant although it represents an additional load on the upper platen thrust bearing. The value determined for $F_{r_{\omega_4}} = 1.30$ kN, is the amplitude of a force which acts in a sinusoidal manner at a frequency of 16.7 Hz and occurs in any plane within the machine which contains the Z_1 axis. An excitation force would therefore occur of 2.60 kN peak-to-peak magnitude, the origin of which would be located at 0.3 m above the working zone height. This would be typically at least 1.55 m from floor level.

It is assessed that forces of such magnitude and frequency would be unacceptable due to possible resonances of the

machine structure and also that high costs would be incurred to achieve adequate structural stiffness.

4.5 Inertia torque due to gyroscopic precession

Consider the arrangement for orbital forging illustrated by the physical diagram of figure 25(a), where the axis Z_2 of the upper conical platen is caused to rotate about the axis, Z_1 , of a cylindrical workpiece which is stationary. Assuming that no relative velocity occurs at the line of contact between the conical surface of the upper platen and the workpiece shown as the generator OA, using the right hand screw rule for determining senses the vector diagram of angular velocities is as shown in figure 25(b).

Also, from the vector diagram figure 25(b) the instantaneous angular velocity of spin in space about the axis Z_2 , ω_s , is given by

$$\begin{aligned}\omega_s &= \omega_2 - \omega_1 \cos \theta \\ &= (\omega_1 / \cos \theta) - \omega_1 \cos \theta\end{aligned}\quad (42)$$

where $\omega_2 = \omega_1 / \cos \theta$

The instantaneous angular velocity of precession ω_p , is given by

$$\omega_p = \omega_1 \sin \theta \quad (43)$$

Let J_{Z_2} be the second moment of inertia of the conical upper platen and spindle assembly about the axis Z_2 . Then the instantaneous precessional torque T_p , about the axis Z_p , is given by

$$T_p = J_{Z_2} \omega_s \omega_p \quad (44)$$

By combining equations (42), (43) and (44) yields

$$\begin{aligned} T_p &= J_{Z_2} \{(\omega_1/\cos\theta) - \omega_1\cos\theta\}\omega_1\sin\theta \\ &= J_{Z_2} \omega_1^2 (\tan\theta - \sin\theta\cos\theta) \end{aligned} \quad (45)$$

The torque due to precession, T_p , is acting about the axis Z_p , at an angle of $(\pi/2) - \theta$ to the axis Z_1 , and in a plane which is rotating at, ω_1 , about the axis Z_1 .

The practical significance of this result is examined in an example as follows :

An orbital type rotary forging machine capable of 0.5 MN axial force has an upper platen spindle and die assembly, of mass 150 kg and a radius of gyration about the axis Z_2 , of 0.075 m. The axis Z_2 , is caused to rotate about the axis Z_1 , at 1000 rev/min. The axis Z_1 , is inclined to the axis Z_2 , at an angle $\theta = 10^\circ$, and the neutral plane is at an angle $\chi = \pi/2$ to the Z_1 axis.

The second moment of inertia about the Z_2 axis is given by

$$\begin{aligned} J_{Z_2} &= 150 \cdot 0.075^2 \text{ kg.m}^2 \\ &= 0.8437 \text{ kg.m}^2 \end{aligned}$$

and $\omega_1 = 104.7 \text{ rad/s}$

By substitution of the appropriate values in equation (45)

$$\begin{aligned} T_p &= 0.8437 \cdot 104.7^2 \cdot 0.0053 \\ &= 49.02 \text{ Nm.} \end{aligned}$$

The axis of the torque T_p , will rotate about the vertical axis of the machine Z_1 . Thus the direction of the torque of magnitude T_p will reverse once during each revolution of the platen when referred to any plane which contains the vertical axis of the machine Z_1 .

A precessional torque of the magnitude shown in the example would clearly be undesirable, since the axis of the applied torque is rotating about the Z_1 axis at 1000 rev/min, thus providing an unbalanced couple at a frequency of 16.7 Hz, having a sinusoidal characteristic with reference to any approximately vertical plane within the machine.

4.6 Facility to provide the required relative force and axial displacement between the upper platen and lower platen

To achieve rotary forging with each of the configurations described previously in section 4.3, it is necessary to provide both force and displacement between the upper and lower platens in the direction of the axis Z_1 . This can be achieved by maintaining either the upper platen or lower platen stationary, with respect to displacement in the direction of the axis Z_1 , and then displacing the remaining member by the application of an appropriate force. The desired relative axial displacement could also be achieved by displacing both the upper and lower platen simultaneously by the application of forces of the required magnitude in opposite directions. There appears, however, to be no practical advantage in this arrangement, except to provide a kinematic system which conforms in a precise manner to that required when double conical platens are used. A number of possible arrangements are illustrated in figures 22 and 23.

Feasible methods of applying the force, F , in the direction of the axis, Z_1 , could range from screw jacks to hydraulic devices. A method of control which permits both force and velocity to be

controlled independently, is considered essential in a machine to be used for research purposes, and may also prove to be applicable to machines for use in manufacturing industry.

4.7 Rotational drive to the conical upper platen-workpiece interface in the alternative kinematic arrangements

It was shown by Slater et al [1] that the process requires energy to be supplied for both rectilinear displacement and rotational displacement of the platens. Rotational energy is dissipated in work done to overcome the frictional forces in the platen bearing systems, and frictional forces at the upper die-workpiece interface, which is in addition to that required to cause plastic deformation of the workpiece. The magnitude and direction of the frictional forces depends upon the rotational drive arrangement. Consider the arrangement illustrated in figure 26(a) and let T_d be the torque necessary to cause rotation of the plastically deforming zone relative to the workpiece axis Z_1 , also let E_f , be the energy required to cause one revolution of the plastically deforming region relative to the workpiece and thus cause a reduction in height, f , the energy ^{being} supplied by a drive torque, T_L , applied to the lower platen bearing spindle. Neglecting frictional torques at both the upper and lower platen bearing systems, the energy per revolution is then given by :

$$E_f = 2\pi T_L = 2\pi T_d < 2\pi T_\mu \quad (46)$$

where T_μ = frictional torque at the lower die-workpiece interface. When $T_d < T_\mu$ rotational drive of the workpiece will occur without slip relative to the lower die. Similarly when the arrangement is as illustrated in figure 26(b) and rotational drive is

achieved by the torque T_U applied to the upper platen, then,

$$E_f = 2\pi T_U / \cos\theta = 2\pi T_d < 2\pi T_\mu \quad (47)$$

Consider the arrangement in figure 26(c) where drive torques T_U and T_L are applied simultaneously to the upper and lower platen spindles respectively. Then, neglecting frictional torques due to bearings, the relationship between the energy and the applied torque may be expressed :

$$E_f = 2\pi T_L + 2\pi T_U / \cos\theta \approx 2\pi T_d \quad (48)$$

Assuming that the energy required to rotate the plastically deforming region is not influenced by the method of rotational drive, then it becomes clear that when $\cos\theta \approx 1$, and $T_U \approx T_L$, the frictional torque, T_μ , required to cause rotational drive at the lower die-workpiece interface, is approximately half the value required when $T_U = 0$. The torque transmitted by the workpiece is also approximately halved, and may be of significance in processes such as compaction of powder metals, or where the frictional torque at the lower die-workpiece interface forms a process limitation.

The rolling nature of the relative motion which occurs at the upper die-workpiece interface forms an important function in the process.

From the description and analysis of the rolling process by Johnson et al [60], it is possible to make qualitative considerations of the influence of various rotational drive configurations in rotary forging by analogy. The analysis based upon plane strain considerations, shows the influence of forward

and back tension upon the normal roll pressure and torques required to produce a given reduction in workpiece thickness. The rolling analogy is made by considering an annular element of a small length of arc on the upper surface of the workpiece. An element on the conical surface of the upper die may be assumed to be a roll being acted upon by a normal pressure, due to an element of the total axial force, F , in the direction Z_1 . The analogy may be extended by the lower die being considered as a roll of very large radius, and the process to be that of asymmetric rolling.

Figure 17(a), represents an arrangement where rotational drive is applied to the upper platen only. The lower platen may be considered as a roll of very large radius driven by friction at the lower platen workpiece interface. The direction of the frictional forces at the lower platen workpiece interface is therefore such as to oppose the force causing entry and exit of material from the neutral zone as described by Slater [61].

Figure 27(b), shows an arrangement where rotational drive is applied to both upper and lower platens. The direction of the upper and lower frictional forces is such that both assist in causing entry and exit of material from the neutral zone. It should also be noted that if the rotational drive is applied to the lower platen, or assumed lower roll only, then the upper platen or upper roll would be driven against the friction of the bearing system. The frictional forces at the upper platen-workpiece interface would be reversed in direction, thus opposing the forces causing material to enter and exit the neutral zone. In the

design of early rotary forging machines based upon the orbital concept, the practice was to drive the upper die about its axis by friction at the upper die-workpiece interface. Also, in the machine invented by Slick, the equivalent of the upper roll was driven by friction at the upper die-workpiece interface.

From these considerations it becomes clear that to investigate the influence of various drive configurations a rotary forging machine designed for research purposes should have the facility to select rotational drive to either, or both, the upper and lower platens.

4.8 Review of designs to date

Recent machines constructed for either research or industrial purposes, appear to be based upon the *wobble* concept. Machines built in the United Kingdom have utilised a rotating upper platen assembly using both conventional rolling and hydrostatic bearings, with the workpiece constrained in terms of rotation about the vertical axis of the machine. The lower platen, together with the workpiece, is moved in the direction of the applied force relative to both the vertical axis of the machine and the upper platen. The kinematic arrangement is illustrated in figure 8, and forms the subject of a British patent [10].

Provision is made in the machines for the adjustment of the angle of inclination of platen axes, θ , but the adjustment can only be made with the machine stationary. In the machines designed and developed for industry [12], facilities are incorporated for high production rates of hot forged components, and include

features such as automatic workpiece feed, ejection and cooling. Control equipment is provided to modulate the axial feed/rev during the forging process. High speed retraction and closing is applied to minimise workpiece ejection and reloading times. The model machine designed by Slater et al [18], has a facility for driving the upper platen spindle using Hooke's joints. This method of drive was not used in the experiments described in References [1], [19], [20], [21]. From illustrations it would appear that non-uniform angular velocity would arise due to the effect of the joints and angles between drive mechanism shaft axes.

For machines based upon the orbital concept the torque required to rotate the upper platen spindle assembly relative to the bearing support, is provided by either friction between the upper platen and workpiece, or for example in the case of the machine described in Reference [8], by the novel feature of a ring type bevel gear assembly. The configuration of the Slick Mill, illustrated in figure 6, has rotational drive applied to the equivalent of an upper platen system, which is displaced axially, but arranged with the plane containing the axes to be horizontal. During loading and after completion of the forging process the workpiece cannot be retained in position by forces due to gravity. The machine was designed for workpieces of such size that lifting equipment is required, and therefore a vertical *throat* is advantageous.

Machines manufactured in Poland and the U.S.A. maintain the workpiece completely stationary. The design by Marciniak is the subject of a British Patent [28] which describes a method of

modulating the angle of inclination of the platen axes during the forging process. Spin of the upper platen assembly relative to the Z_1 axis is prevented by a pin and slot mechanism. The design concept includes a mechanism whereby displacement of the complete upper platen assembly is caused by application of the required axial force in the direction of the machine axis. In the particular design illustrated in figure 10 this is achieved by the use of a hydraulically operated arrangement to apply both axial force and translational displacement, whilst also providing fluid pressure for the hydrostatic bearing system including the system which induces planetary motion.

Thus all the moving members required in the system are incorporated in the upper platen assembly and a detailed illustration of the machine drive kinematics is included in Reference [16] with a description of the manner in which planetary motion is controlled by adjustment of the magnitude of eccentricity. From study of this design it becomes apparent that the components required for the hemispherical hydrostatic bearings, and the eccentric bearings, are of complex shape, and must also be manufactured with high precision.

With the exception of the model machine Reference [1], all machines built previously use hydraulic systems for applying and controlling both axial force and displacement. Most machines with a means of workpiece ejection use an additional hydraulic ram for the purpose. The problem of making this provision is simplified in the design concept where all axial displacement required for the forging process is contained in the upper platen

mechanism.

During the rotary forging process forces between the upper die and workpiece occur in the radial direction. Manufacturing applications can arise where the tools are designed specifically to cause radial deformation of the workpiece in order to achieve a desired final form. Relative radial displacement of the platen axes will cause errors in geometry of the workpiece, and influence surface roughness due to angular velocity relationships which do not comply with the kinematic requirements identified in Section 4.1. Relative radial movement of the axes of the upper platen and workpiece will be an additional source of tool wear. Stiffness of the members locating the die-workpiece axes system is therefore required to ensure that the forces which arise cause the minimum practicable relative radial displacement.

Consider the arrangement illustrated in figure 28(a) where a radial force, F_r , acts on both the upper conical platen and the workpiece. Let the bearing system between the upper platen spindle and bearing housing, have a radial compliance C_{Z_2} and the radial compliance between the bearing housing and the frame be C_{Z_1} . Assuming the radial compliance of the workpiece relative to the frame to be zero, and that θ , is small and $\cos\theta \approx 1.0$, then the total radial displacement, δ_r , of the axis Z_2 relative to the axis Z_1 , at the point, 0, due to the force F_r , is given approximately by :

$$\delta_r \approx F_r(C_{Z_1} + C_{Z_2}) \quad (49)$$

Consider the arrangement illustrated in figure 28(b) and let the bearing system have radial compliance C_{Z_2} between the upper

conical platen spindle and the machine frame. Also, let the bearing system between the lower platen spindle and the machine have a radial compliance C_{Z_1} with respect to the machine frame. Then, assuming that the radial compliance of the workpiece relative to the lower platen spindle is zero, and the angle, θ , to be small, the total radial displacement δ_r , of the upper platen axis Z_2 , relative to the workpiece axis Z_1 , is given approximately by :

$$\delta_r \cong F_r(C_{Z_1} - C_{Z_2}) \quad (50)$$

Thus the kinematic arrangements can be such as to achieve either the sum or the difference of the respective radial compliances of the rotating components in the machine. The arrangement illustrated in figure 28(b) has obvious advantages since increasing radial stiffness of any member invariably increases cost.

The arrangement by Slick is an example of a design in which relative displacement of axes due to the difference in radial compliance is applicable, whilst the invention by B & S Massey Limited, illustrated in figure 8, would be affected by the sum of the radial compliances. The machines currently manufactured by B & S Massey Limited are fitted with hemispherical hydrostatic bearings to provide the necessary degrees of freedom, and have similarities in this respect with the design by Marciniak. The radial compliance problems are reduced by providing the two degrees of freedom using a single bearing interface but are costly to manufacture. The radial compliance considerations applicable to the design by Marciniak illustrated in figure 10, are more complex, due to the added radial compliance of the sliding of the upper platen assembly.

4.9 Summary from design synthesis

The following conclusions were reached from the design synthesis :

- (a) orbital arrangements have the advantage of a stationary workpiece, but lead to kinematic complexity, and also the oscillatory nature of the inertia forces are problematic due to inherent imbalance. The concept leads to designs which are costly to manufacture in order to achieve the desired radial stiffness. The special purpose bearing assemblies and components used in certain previous designs could result in high cost of ownership arising from maintenance requirements,
- (b) in order to achieve inherent dynamic balance and to avoid undesirable inertia forces the platen axes should be stationary in space and require an inversion of the mechanisms which produce orbital motion,
- (c) an inversion of the mechanism capable of planetary motion is necessary if the findings from all previous work are recognised,
- (d) for research purposes the facility to control the angular velocities ω_1 and ω_2 , about the axes Z_1 and Z_2 respectively, and the inclination of the platen axes, θ , as three independent variables without interruption of the process is a requirement to be appended to those stated in Chapter 3, section 7,

- (e) the design should make provision for various upper die and workpiece rotational drive configurations,
- (f) induced slipping or skidding as part of a design concept should be avoided,
- (g) the kinematic arrangement should be such that relative radial compliance of platen axis is minimised.

5.1 The preferred kinematic arrangement

From the requirements stated in Chapter 3, and with due consideration given to the design synthesis in Chapter 4, the design solution preferred from those considered is based upon the kinematic arrangement illustrated in figure 23(b) with the addition of a facility to adjust the inclination of the platen axes, as desired, during the forging process.

Adjustment of the inclination of the platen axes, prior to, and during the process, is achieved by a trunnion bearing system, the axis of which passes through the point of intersection of the geometric axes about which the upper and lower platens rotate. The trunnion axis is perpendicular to the plane which contains the platen axes. The kinematic arrangement is illustrated in figure 29.

Individual control of the upper and lower platen rotational speeds are required to permit the ratio of angular velocities, ω_1/ω_2 to be adjusted to satisfy the fundamental kinematic requirements as determined from given values of, θ , and χ , as specified in equation (23). Individual rotational drive to both the upper and lower platens can be provided from a single rotational drive the output axis of which is coincident with the trunnion axis. This permits adjustment of, θ , with no interruption to the forging process and the method of drive is applicable to all the arrangements illustrated in figure 30. It should be noted that the introduction of a differential gear system at the drive input

as illustrated in figure 30(c) would be advantageous for the purpose of maintaining the desired angular velocity relation of the upper and lower platens when, θ , is modulated in a cyclic manner.

The kinematic arrangement of a facility to provide ejection of the workpiece is illustrated in figure 31, which also depicts the operating cycle, and the manner in which a force from the down stroke of the operating ram is used in conjunction with an interrupter mechanism to effect control of the operation.

It is concluded that the preferred design has advantageous characteristics as follows :

- (a) proven components of well defined performance are available as proprietary items for a large number of the parts required. This leads to advantages related to a high degree of confidence in both performance and cost predictions,
- (b) manufacturing costs may be minimised since components required to construct the machine are all of simple geometric form, requiring no machining at compound angles, no forms comprised of other than cylindrical or plane surfaces, and all raw materials are available as stock items,
- (c) simplicity of the rotational drive system permits alternative methods of supplying rotational energy for

the purpose of investigating the effect of various drive configurations upon both the upper and lower die,

- (d) the facility for rotational drive to both upper and lower platens permits double conical dies to be investigated,
- (e) adjustment of, θ , can be made in clockwise and anti-clockwise directions with respect to the vertical axis of the machine, without interruption of either the rotational or translational drives,
- (f) the plastically deforming zone is stationary relative to the machine base and this feature permits the deforming zone to be observed more readily than if it were rotating relative to the machine base,
- (g) differences in upper and lower platen bearing compliances will be applicable to forces causing relative radial displacement of the platen axes,
- (h) dynamic balance is inherent with the exception of any workpiece, fixture or upper platen which may be used which is not axisymmetric. Increase in the inclination of the platen axis to reduce the axial force required does not give rise to an increase in inertia forces.

Features of this design solution are the subject of applications for United Kingdom and foreign patents as described in References [62], [63], [64], [65] and [66].

The preferred design is identified to have the following disadvantages :

- (a) the workpiece rotates and is therefore not readily observed in total during the deformation process,
- (b) the arrangement is unsuitable for applications of the rotary forging process to assembly operations. For example, when using the heading process for fastening, the components to be fastened would be caused to rotate about each fastener axis,
- (c) time will elapse and energy will be dissipated, in accelerating to a given angular velocity, and bringing to rest the rotating components of the lower platen. This feature could be problematic in a machine being considered for very high production rates. It is pointed out, however, that these operations could take place during the time to raise and lower the lower platen assembly. The raising and lowering operations are essential to provide the clearance necessary for loading and unloading the workpiece,
- (d) in hot forging, the largest proportion of the heat given up by the workpiece goes to the lower die due to period of contact time. To cool a rotating platen is clearly a more complex matter than cooling a stationary platen or a platen subjected to translational displacement only.

5.2 The preferred design

(a) Basic configuration

From a number of design layouts evolved from the illustrations in figures 29, 30 and 31, the design preferred for manufacture is illustrated in the orthographic views of figure 32. Consistent with the requirements stated in Chapter 3, section 12, the layout is based upon the use of rolling bearings selected from a range of proven proprietary products capable of satisfactory operation when subjected to axial loads of up to 0.5 MN under either static or dynamic conditions.

(b) Platen bearings

The platen bearing arrangement is such that the effect of radial compliance is minimised by ensuring that equation (50) is applicable. The pillar and plate type construction used provides adequate axial stiffness and surfaces which are convenient for attachment to the main frame. It should also be noted that torsional stiffness of the frame increases as axial load increases.

Radial loads of 0.15 MN were also taken into consideration during selection of the bearing arrangement. Bearings of adequate performance were found to be available from a number of sources and vendor selection was made on a price basis. The design layout of the bearing and spindle arrangement, together with application details, were approved by the Engineering Department of the bearing manufacturer.

The spindle and housing diameter was determined largely by the main thrust bearing dimensions. Figure 33 is a typical layout and assembly drawing produced for design and manufacturing purposes. This illustrates the sub-assemblies which form the spindles and housings, for both the lower platen, figure 32, item 1, and the upper platen, figure 32, item 2. The platen bearing assembly contains a spherical roller thrust bearing, figure 33, item 1, which has static and dynamic load ratings of 1.2 MN and 0.63 MN respectively. The cylindrical roller bearing, figure 33, item 2, has both static and dynamic load ratings of 0.17 MN. The bearing positioned adjacent to the 150 mm diameter working face of the spindle to react the radial loads. A taper roller bearing, figure 33, item 3, with both static and dynamic load ratings of 0.14 MN, is used in conjunction with a screw adjustment on the spindle to provide axial pre-load to the thrust bearing, in addition to providing a journal bearing. The thrust bearing is located by shoulders on the spindle and bearing housing, and the cylindrical roller bearing is retained axially by D.I.N. standard circlips, figure 33, items 4 and 5. It should be noted that the spherical thrust bearing is located radially on the spindle only. Hence all radial loads are reacted by the cylindrical roller bearing at the drive gear end. This is the correct method of installation for spherical thrust bearings which the author has noted is not always adhered to by designers.

The spindles, figure 33, item 6, are hollow and fitted

with parallel keyways to suit the drive gear.

Seal rings, figure 33, item 7, as supplied and recommended by the bearing manufacturers, are fitted in the end caps to both retain the lubricant and protect the bearings against the ingress of foreign bodies.

The bearings are lubricated by a grease suitable for operation at temperatures up to $+180^{\circ}\text{C}$. The bearings are packed with lubricant and suitable for a life of 3,000 operating hours. Lubrication and replacement of seals can be carried out readily by removal of the spigot located end plates.

The large diameter face of the spindle provides spigot location and fastenings for the die adaptor plates, figure 32, item 3.

Provision is made on the adaptor plates for spigot location and fastening of the upper and lower dies. A typical conical upper die and flat lower die for upsetting operation are illustrated in position in figure 34.

It should be noted that with the exception of a thrust shoulder on the outer diameter of the upper platen bearing housing, lower platen and upper platen assemblies are identical.

(c) Adjustment of inclination of platen axes

To provide adjustment of the inclination of platen axes,

the upper platen system is mounted on a support plate, figure 32, item 4, which is stiffened in bending by members, illustrated in figure 32, item 5. The support plate is fastened by four high tensile bolts to each of the two plummer blocks, figure 32, item 6, which contain double taper roller bearings providing a total safe working load of 0.72 MN. The fastening bolts are arranged to pass through the support plate, stiffening members and plummer block bases. The bearings provide an axis, about which the complete upper platen assembly can rotate through 15° from either side of the vertical axis of the machine. The bearing housings are selected from a range of proprietary products, manufactured from spheroidal graphite iron, specifically for the purpose of operating in tension with respect to the plummer block base. The plummer block housings selected are rated for loads up to 0.50 MN each and the spherical double roller bearings are each rated for operation with 0.36 MN static or dynamic loads. The rolling bearings were selected to limit the frictional torque required to modulate the angle of inclination of platen axes whilst the machine is under operating load. The self-aligning characteristics of the bearings obviates alignment problems during construction. The upper platen support plate, figure 32, item 4, is fitted with extension arms, figure 32, item 7, to provide the upper attachment points for the mechanism used to adjust the angle, θ .

(d) Upper platen rotation drive

The drive to the upper platen spindle is achieved from a

gear train, figure 32, item 8, which is also illustrated in figure 35, and is driven from the output shaft of a 29:6 ratio reduction gearbox, figure 32, item 9. The input shaft of the gearbox is driven by a three groove pulley, figure 32, item 10, which is in turn driven at a nominal 2:1 reduction from three V-belts, engaged in three grooves of a six groove pulley, on the output shaft of an 11 kW motor and variator, figure 32, item 11.

The gear train for the upper platen drive, illustrated in figure 35, is located in a pillar and plate assembly with spigot location on the upper platen bearing housing. The pillars are from centreless ground stock with turned spigot diameters and screw threads. The plates are bored to suit pillar spigots and gear spindle bearings.

The upper platen rotational drive gears illustrated in figure 32, item 8, are 1:1 ratio, 48 teeth, 6 DP.En.9 steel, spur gears located on their respective spindles by standard taper lock attachments and with parallel keyways where applicable. The spindles are manufactured from centreless ground stock and a standard size is used where practicable throughout the machine.

The upper platen reduction gearbox is supported by channel section members fastened to the upper platen support plate and there is provision for belt tensioning of the upper platen drive as illustrated in figure 36.

In the machine constructed, provision has not been made to make the ratios of the reduction gearboxes infinitely variable in order to obtain the desired value ω_1/ω_2 .

The fitting of various pitch diameter pulleys at the input drive shaft to the upper platen reduction gearbox proved to be the most economic method of providing a range of values of ω_1/ω_2 and was therefore considered appropriate to the current stage of development. Space is available for a variable speed drive to be fitted retrospectively.

The axis of the six groove drive pulley on the output of the motor and variator, figure 32, item 11, is coaxial with the plummer block bearings, figure 32, item 6, thus belt tensioning requirements do not change with adjustment of inclination of axes, θ .

(e) Lower platen rotational drive

The lower platen spindle bearing housing and rotational drive system is mounted on a pillar and plate assembly, figure 32, item 12. Rotational drive is achieved by a spur gear train of 6 DP.En.9 steel gears. The assembly is illustrated in a photograph taken during manufacture, figure 37. The gears are all 48 teeth with the exception of the 72 teeth gear, which is necessary to provide a centre distance adequate to obtain clearance of the lower platen support from the long faced gear, figure 32, item 13, driven from the output shaft of a 29:4 ratio reduction gearbox, figure 32, item 14. The input shaft of the gearbox is driven by a three groove pulley from three V-belts engaged in three grooves of the six groove pulley on the variator

output shaft. The method of belt tensioning is similar to that used on the upper platen drive.

Economy in manufacturing costs were effected by designing such that seven of the nine gears manufactured for the machine are identical in all respects and the long face gear figure 32, item 13, is of the same pitch circle diameter, diametral pitch, and material. The gear strength is such that full power can be transmitted at 30 rev/min through either of the gear trains which are designed for lubrication by grease. The input shaft of the reduction gearbox is driven by a three groove pulley, figure 32, item 15, with 2:1 ratio reduction from the six groove pulley on the output of the variator.

(f) Translational displacement of lower platen

Lower platen translational drive is achieved by power from a 7.5 kW hydraulic unit, figure 32, item 16. The pressure output which is adjustable from 100 to 3000 lbf/in², (0.68 to 20.4 MPa) operates the 152.4 mm (6.0 inch) stroke 177.8 mm (7.0 inch) diameter standard industrial ram, figure 32, item 17, which is connected by a 76.2 mm (3.0 inch) diameter threaded output shaft to the lower plate of the lower platen assembly. The ram has hydraulic cushion stops to provide protection against suddenly applied loads due to striking the end stops at speed. The hydraulic power unit, including drive motor, pump, tank, filters, flow and pressure controls form a complete sub-assembly which is mounted in the support frame using vibration

isolators. The hydraulic system was designed to meet the requirements specified by the author, and is illustrated diagrammatically in figure 38.

(g) Main frame

The main machine frame was constructed with circular upper and lower plates, figure 32, items 18 and 19 respectively, which are separated by four 76.20 mm diameter, threaded and spigotted pillars, figure 32, item 20, fastened by nuts. The faces of the plates were machined parallel by facing on a lathe. The holes bored in the centres are concentric with the outside diameters and all pillar spigot location holes in the upper plate were bored in conjunction with the corresponding holes in the lower plate. The pillar spigot diameters and shoulders were turned between centres thus the plates when assembled are concentric and the pillars are orthogonal to the plate faces.

The lower plate, figure 32, item 19, forms the mounting for the hydraulic ram and also reacts the translational stop forces. The translational displacement stops are provided to limit extension and retraction of the ram by adjustment of nuts on the four screwed rods, figure 32, item 21, which extend from the lower platen support pillars through clearance holes in the lower plate of the main frame. A spigot and tapped holes on the end face of the lower platen bearing housing provides location and fixing to the lower platen pillar and plate assembly, figure 32,

item 12. The lower platen assembly is guided in the axial direction by the ram bearings and the outside diameter of the lower platen bearing housing sliding in a cast iron bearing ring, figure 32, item 22, in the main frame upper plate.

The two main frame pillars adjacent to the lower platen rotational drive provide the surfaces against which the lower platen drive mechanism bearing pads react the forces due to the drive torque. The four bearing pads, illustrated in figure 39, are of phosphor bronze and cylindrical in form but the mounting holes in the axial direction are eccentric thus providing adjustment of clearance by angular rotation prior to clamping.

The spigot diameters of the main frame pillars extend through the upper plate of the main frame and pass through location holes in rectangular spigotted separator blocks, figure 32, item 23, which form part of the trunnion bearing support plate assemblies. The blocks are provided with spigots and screwed ends for both location and fixing to the trunnion bearing support plates. Spigots on each end of the trunnion bearing spindles are located in the support plates which are spaced by the rectangular spigotted separator blocks. Figure 40 illustrates this assembly during construction. The inner pair of trunnion bearing support plates, figure 32, item 24, are extended to form a structure for the lower mounting of the screw jack mechanism. The handwheel, figure 32, item 25, is provided for manual

adjustment of the angle of inclination of platen axes, θ . The mechanical advantage of the handwheel and screw jack are such that manual adjustment may be made during the forging process. This mechanism is also illustrated in figure 41.

(h) Workpiece ejector mechanism

The workpiece ejector pin, figure 32, item 26, passes through the hollow lower platen spindle. The two interrupters used in the ejection mechanism are positioned by 180° rotation of pillars which are located by self lubricating sintered bronze bearings in the upper and lower main frame plates. The mechanism is illustrated in the disengaged and engaged positions in figures 42(a) and 42(b) respectively. The interrupter column forces are reacted directly to the lower main frame plate by compression pads. The rotational drive for the pillars is provided by pneumatic actuators powered by the workshop air supply and controlled by transverse operation of the main control lever. Electro-pneumatic control valves are provided with a switch which is operated by a cam on the translational displacement system. This provides an electrical interlock to prevent damage due to incorrect ejector operation. Figure 43 illustrates the ejector pin extended with a typical workpiece in position. It should be noted that the hollow upper platen spindle permits ejector pins of various end configuration to be interchanged without dismantling the machine.

(j) Support frame

The main frame is mounted in a support frame, figure 32,

item 27, which is constructed from rolled steel channel and provides location for the rotational drive motor, together with a fixing for the lower platen drive reduction gearbox. The main frame is constructed using M10 fasteners and standard gusset plates throughout, also provides a mounting for the hydraulic power unit and tank. The main support frame is used to provide a fixing for the weld mesh safety guard which covers the machine completely and also provides guides for the sliding access door to the working area. A view of the working area with the door in the open position is illustrated in figure 34.

Main frame members, which form the base and floor attachment points, also provide support for a 7 mm thick steel chequer plate, figure 32, item 28, to form a platform for the operator.

(k) Instrumentation

A piezo electric type force transducer, figure 32, item 29, in a thrust washer configuration, is interposed between the output shaft of the hydraulic ram, figure 32, item 17, and the lower platen support plate figure 32, item 12. There is provision for pre-loading the transducer such that forces may be measured in both the forging and ejection directions of the ram. The transducer output is supplied to a charge amplifier which provides a D.C. voltage proportional to force. The output voltage is made available for both analogue recording and visual monitoring.

The torque to the upper and lower platen rotational drives are measured by the forces reacted at the penultimate drive gear spindles. A typical arrangement is illustrated in figure 44. Self-aligning ball bearings are placed in housings which are free to slide in a direction tangential to the pitch circles of the gears at the point of meshing. The bearing housings are constrained from other degrees of freedom. Piezo electric type force transducers in the form of thrust washers are interposed between the sliding member and fixed member of the bearing housing support plates. The outputs from the transducer at each end of the spindle are summed and connected to a charge amplifier to provide a D.C. voltage proportional to torque. The voltage is used for both monitoring and analogue recording. The measuring systems are identical for both upper and lower platen drives.

Proximity detectors are fitted, with suitable clearance, on the outside diameter of the spur gears at the final drive stage of both the upper and lower platen drive shafts. The proximity detector on the lower platen drive gear is illustrated in figure 42. The square wave output of D.C. voltage, as each tooth transits the sensing face of the detector, is fed to digital display meters which are provided with mode selectors for pulse count, period, or frequency with additional controls for start, hold and reset.

Translational displacement in the direction of the axis of the lower platen is transduced by a rectilinear potentiometer using a hybrid technology of wound wire and conductive plastics. This is attached to the lower main plate and

provides a resolution better than 1 part in 10,000, together with satisfactory voltage output law conformity over the operating range. The four digit monitor permits a readability of 0.1 mm. The system is energised from a 0.1% stabilised 5 V.D.C. power supply and the scaling potentiometers are adjusted to provide direct digital readouts in units of millimetres.

The angle of inclination of the platen axes is transduced using a similar type of rectilinear potentiometer connected between the fixed and moving members of the angle adjustment mechanism. This provides a D.C. voltage which is approximately proportional to θ , and the four digit monitor permits a readability equivalent to 0.01 degree. A repeater monitor is provided adjacent to the control wheel since the operator cannot observe the control panel whilst making an adjustment. The scaling potentiometers are adjusted to provide direct readout of angle in degrees.

A pressure transducer is connected to the input port of the hydraulic ram, and although intended primarily for recording purposes, terminals are provided for monitoring purposes.

Two pressure gauges mounted on the control panel are connected to the hydraulic system to provide visual monitoring of ram pressure for both forging and ejection.

(1) Operators controls

The operator's main control panel, illustrated in figure 45, forms a mounting for all the appropriate switching and visual monitors including warning lights, together with the terminals or connectors for analogue output from the instrumentation. It also provides a mounting for the key operated main switch and a two axis lever for the control of both displacement and ejection mechanism. Switches to operate the hydraulic power unit and rotational drive including an *inching* facility are also mounted on the control panel together with an emergency stop switch.

Two additional emergency stop switches are placed on the machine frame at the corners most remote from the control panel. The operation of any emergency stop button removes power from the rotational drive and hydraulic pump, in addition all hydraulic pressures in the system are relieved. All emergency switches are of the type which remain in the locked off position once operated and require specific reset action before operation can be resumed.

The 440V 50 Hz, 3-phase electrical supply is connected to a single box on the rear of the machine and houses fuses, current trips and the main isolator switch. A pressure connection is available for workshop air supply.

Warning lamps and total elapsed running time indicators are fitted and connected to both the hydraulic power unit

and the rotational drive motor. The control panel forms a door to the box which contains all the instrumentation circuit boards, with trim controls for all circuits including the 5 V.D.C. and 10 V.D.C. stabilised power supplies. Figure 46 is a wiring diagram of the D.C. energised instrumentation which is incorporated. The output from the piezo electric force transducers are connected direct to the co-axial sockets on the control panel and are not included in the diagram.

(m) General information

Figure 47 illustrates the machine viewed from the front and side during construction and figure 48 illustrates the machine in the present stage of development. A schedule of component drawings and assembly drawings is presented in Table 14. The proprietary items incorporated are listed in Table 15.

5.3 Specification for the 0.5 MN rotary forging machine

The specification for the machine designed is as follows :

1. Axial force applied to lower platen is adjustable over the range 25.0 kN to 0.5 MN.
2. Rotational velocities of the upper and lower platen and workpiece is adjustable over the range 30 to 300 rev/min. The design is such that modification of the speed range to provide 120 to 1200 rev/min would not entail replacement of the platen bearing system.
3. Translational displacement of the lower platen is 150 mm

with stops provided to limit displacement in either direction within the stroke to any desired value with maximum axial force applied.

4. Angular adjustment of inclination of the upper platen geometric axis relative to the vertical axis of the machine is 0° to 15° . The machine to be constructed with a facility for manual adjustment but provision to be made for fitment of equipment for powered operation.
5. Space is provided for workpiece dimensions as follows :

<u>Initial</u>	<u>Final</u>
Maximum diameter 100 mm	Maximum diameter 150 mm
Maximum height 100 mm	Minimum height 0.0 mm

6. Power available for rotating platens relative to workpiece is 11.0 kW.
7. Power for translational displacement is provided by 3000 lbf/in² (20.69 MPa) hydraulic unit with a pump driven by a 7.5 kW motor.
8. Ejection forces adjustable to a maximum of 100 kN with a stroke of 50 mm may be applied axisymmetrically to a workpiece.
9. Instrumentation is incorporated to measure :
 - (a) axial force applied to the lower platen for both forging and ejection,

- (b) torque applied to the upper platen rotational drive,
- (c) torque applied to the lower platen rotational drive,
- (d) translational displacement of lower platen,
- (e) rotational velocity of lower platen about Z_1 axis,
- (f) rotational velocity of upper platen about Z_2 axis,
- (g) total angular displacement of the upper platen about the axis Z_2 during deformation of workpiece,
- (h) total angular displacement of lower platen about the axis Z_1 during deformation of workpiece,
- (j) inclination of the geometric axis of the upper platen relative to the vertical axis of the machine,
- (k) outputs to recorders are provided at terminals and sockets on front face of the control panel. Signal conditioning units are mounted as *plug in* circuit boards at the rear of the hinged control panel.

9. Controls are available for the following :

- (a) key operated main switch with warning lamp,

- (b) start and stop of hydraulic power unit,
- (c) translational displacement of the lower platen assembly,
- (d) independent adjustment of pressures applied to the ram in both the extension (forging) and retraction (ejection) directions,
- (e) independent adjustment for translational velocity of the lower platen in both the ram extension (forging) and ram retraction (ejection) directions,
- (f) adjustment of rotational velocities of the upper and lower platens,
- (g) engagement and disengagement of ejector mechanism,
- (h) start, stop and inching rotational drive motor,
- (j) adjustment of angle between axes of platens,
- (k) single action operated controls are provided which stop both the rotation of the platens and remove all hydraulic ram forces simultaneously.

10. Power supplies required are :

- (a) 440 V, 3-phase, 50 Hz, 30 kW,

(b) 5 V D.C. 5 W ,

(c) 10 V D.C. 5 W,

(d) Air supply - workshop 80 lbf/in² (0.54 MPa).

11. General characteristics of the machine are :

(a) The machine is capable of operating in the platen rotational drive configurations as follows :

- (i) Rotational drive to the upper platen only,
- (ii) Rotational drive to the lower platen only,
- (iii) Rotational drive to both upper and lower platens simultaneously.

(b) A facility to vary the angle of inclination between axes of platens during the forging process is incorporated.

(c) The machine is designed such that the workpiece height is between 1 m and 1.25 m from the operator's platform.

(d) Initially the angle of inclination of platen axes is variable by means of manual adjustment. Provision is made in the design for the incorporation of powered adjustment. This facility is available with any of the configurations in 11(a) above.

- (e) The machine base is provided with suitable fittings for attachment to the floor.
- (f) The machine is fitted with guards to provide operator protection and conforms in all aspects to the requirements of Section VI of the Health and Safety Act 1974. It is possible for the operator to view the workpiece and upper and lower platens during the forging process when all necessary guards are in position.
- (g) Eyebolts or facilities to attach eyebolts are provided to lift the complete machine or any sub-assembly or components of weight ≥ 300 N.
- (h) The total weight of the machine is 35 kN.

6.1 Performance testing

Salient items from the specification, Chapter 5.3 were checked on the completed machine as follows :

Specification Item 1.

Pressure from the hydraulic system available at the ram input port was found to be adjustable to any value in the range from 100 lbf/in² (0.69 MPa) to 3000 lbf/in² (20.7 MPa) giving estimated forces from 17.1 kN to 513 kN.

Specification Item 2.

The rotational speed of the platens was found to be adjustable to any speed within the range 35.0 rev/min to 308.8 rev/min. This does not conform to the design specification for the minimum rotational speed of 30 rev/min and is probably due to incorrect stop settings in the variator mechanism.

Specification Item 3.

The translational displacement range of the lower platen assembly was found to be 152.4 mm. and the displacement stops were found to be capable of adjustment to any required position, in either direction, within the displacement range. The stops are also capable of operation at maximum axial load.

Specification Item 4.

The inclination of the platen axes was found to be adjustable over the range of 15° from the vertical.

6.2 Compliance

This performance characteristic of the machine is unspecified. It was, however, decided to assess compliance between various parts of the structure including the upper platen, figure 32, item 2, the trunnion bearing system, figure 32, items 6 and 23. and the upper platen support plate figure 32, item 4.

The method of test was to place dial test indicators at various points as illustrated in figure 49 and then measure the relative displacement - load relation. The results are expressed as compliance, $\mu\text{m/kN}$. In the absence of published information on this aspect of previous rotary forging machines the design objectives with respect to compliance were established on an arbitrary basis. Assessment that $1.0 \mu\text{m/kN}$ appeared to be both applicable and readily achievable from the current state of the art in rolling bearing technology. This value was used as a criterion by the author.

The desired loads were applied by adjusting the pressure control valve in the hydraulic system whilst maintaining the lower platen translational control locked in the *raise* position. The pressure was adjusted and measured using the pressure gauge on the control panel which had been calibrated previously using a sub-standard gauge. This test revealed a problem with the hydraulic system

as designed and installed. In the system specified by the author it was considered desirable to apply a hydraulic lock concept in the ram extension circuit. This feature was incorporated to prevent cyclic variation in translational displacement at the upper die-workpiece interface causing perturbations in the pressure system. Hence the check valve is positioned as illustrated in figure 38, item 11. This means that pressure in the ram extension volume of the cylinder cannot be reduced without changing the translational direction selected by the appropriate control. This would clearly invalidate any test for hysteresis. This problem could be obviated by temporary removal of the check valve but this is not readily accessible since it is situated physically in a sub-assembly which comprises a sandwich of items, 9, 10, 11 and 16 in figure 38. It was therefore decided to accept this shortcoming of the design since it is considered to be an advantageous feature when forging. The fitting of a manually operated by-pass valve could be considered at the first major overhaul of the machine. During the compliance tests it was, of course, practicable to observe the hysteresis in the return to zero load condition and in all cases was found to be less than 2.0% of maximum relative displacement measured.

The initial tests were carried out with the dial indicators positioned as shown in figure 49.  The results are stated in Tables 6(a) and 6(b) respectively. From this it was concluded that the compliance in bending about the base fixings of the spheroidal iron plummer block was excessive and did not match the trunnion bearing compliance predicted. The results were discussed with engineering representatives

of the vendor who informed the author that problems had been encountered with similar components and that compliance of the complete bearing assembly when fastened in position could be decreased by using the cap bolt fastenings to provide a reduction in the bending moment applied to the base of the plummer block. This was carried out by drilling clearance holes in the tapped holes provided for the cap bolts in the base member of the bearing housings. Two additional high tensile steel studs were passed through each of the bearing caps, the plummer block base and holes in the upper platen mounting plate, figure 32, item 4. Following the modification, tests were repeated and the results obtained are listed in Table 6(c). During this test the compliance was also assessed between the upper and lower plates of the main frame and found to be $1.0 \mu\text{m/kN}$.

The results indicate that a marked decrease in compliance had been achieved albeit not in accordance with the performance criterion established by the author. Because of the arbitrary nature of the criterion it was decided to operate the machine as modified in an attempt to identify any shortcomings in performance which may arise with the compliance as measured. To decrease compliance, for example, by using higher load rated proprietary components, or the manufacture of special purpose components, is not technically difficult but will result in increased cost of machines. The subject of axial compliance is therefore referred to in recommendations for further work.

It was, however, decided to design and manufacture an alternative type of bearing housing to replace the proprietary unit. These are now available but will not be fitted until specific

inadequacies of the existing units have been identified by experiment. Shortcomings resulting from greater axial compliance than that specified arbitrarily do not appear to have occurred during experiments within the scope of the work described in this thesis.

A further axial compliance test was carried out after some 50 hours of operation and the results obtained are listed in Table 6(d). From these test results there was no evidence of an increase in compliance. An additional test was conducted with the magnetic base of a dial indicator clamped to the lower die mounting plate, and the measuring probe in contact with the cylindrical surface which forms part of the upper conical die of semi-angle 80° . The inclination of platen axes, θ , was 10° . The dial indicator was positioned to measure relative displacement in a direction perpendicular to the lower platen axis and in a plane which contains both the upper and lower platen axes. When axial loads were applied over the range 0 to 500 kN the relative displacement measured did not exceed 0.02 mm. A cylindrical workpiece of mild steel 76 mm diameter was used. The assessment of radial compliance with the platens rotating was not attempted due to the complexity of the instrumentation required.

6.3 Vibration

The design specification does not state in a quantitative form machine performance requirements with respect to acceptable levels of vibration.

The machine was tested during construction for evidence of resonance over the rotational speed range of the upper and lower platens and throughout the range of translational displacement of the lower platen but without axial force applied. During initial

tests the amplitude of vibration was detected by tactile methods and was therefore subjective. No evidence of resonance was detected by this method and following some 15 hours total rotational operation it was decided to conduct tests using an accelerometer in an attempt to quantify the levels of vibration for reference purposes.

The method of test was to fix, by bolting, a piezo electric acceleration transducer Endevco Type 213E Serial No. DN.03 to the underside of the upper platen support plate, figure 32, item 4, and adjacent to the rotating upper die adaptor plate. Tests were conducted with the sensing axis of the accelerometer in each of three positions as follows :

Test No.1 parallel to the platen axis Z_2 ,

Test No.2 perpendicular to the platen axis Z_2 and
parallel to the trunnion bearing axis,

Test No.3 perpendicular to both the platen axis Z_2 and
the trunnion bearing axis.

The output of the acceleration transducer was connected to a charge amplifier and indicator Endevco Type 6634 Serial No. 2735. The acceleration levels were read in units of acceleration due to gravity, where $1g = 9.81 \text{ m/s}^2$. The frequency was measured using an oscilloscope with an internal time base selected to sweep at 5 ms/cm. Measurements of both acceleration and frequency were made at various rotational speeds of the platens. The angle of

inclination, θ , of the platen axis was 4° . Tests were conducted with the machine in the loaded and unloaded condition. A forging force of 0.25 MN was applied for testing in the loaded condition using a conical upper platen of semi-angle 82° and a workpiece of mild steel 76.2 mm diameter, $H_0/D_0 \approx 0.2$. The test results are stated in Table 7.

The waveform of the acceleration - time relation was not sinusoidal. However, with maximum allowance for this it was estimated from the results that the amplitude of the vibration did not exceed 0.0025mm in the worst case.

From study of the test results the source of vibration at frequencies ranging from 0.8 to 2.1 kHz is not clear. From discussion with the technical department of the bearing suppliers, measured amplitude together with the approximate proportionality in the vibration frequency - rotational speed relation is characteristic of the vibration generated by the number of rolling elements in the bearing system. Further investigation was deemed not to be relevant to the current objectives.

The conclusion from the test was that vibration performance would appear to be satisfactory and also that the outcome of the subjective tests was confirmed. The data obtained is considered to be of use for future reference and comparison purposes.

6.4 Calibration of the axial force transducer

The method of test was to place a sub-standard piezo electric force

transducer in a thrust washer configuration, between flat, hardened and ground, parallel plates which were then placed between the upper and lower platens. The inclination of the platen axes was adjusted to zero.

The sub-standard transducer, Type 9091, Serial number 117555, was provided by Kistler Instruments A.G. together with a charge amplifier Type 5005, Serial number 3016, with a system calibration referenced to the Swiss National Bureau of Standards, Berne. The output signal was measured using a precision digital voltmeter Phillips Type 2422, Serial number D.71015. The 1.0 MN force transducer, Kistler Type 9091, Serial number 107730, fitted to the rotary forging machine is connected, through a 10:1 attenuator, to channel No.1 of a 3-channel charge amplifier Kistler Type 5801, Serial number 17702, and a galvanometer Type B100 in the U.V. recorder. The magnitude of the load applied to the sub-standard transducer was adjusted by means of the hydraulic pressure control to provide 50 kN increments over the range 0 to 500 kN and monitored by the precision digital voltmeter. The displacement of the galvanometer was recorded at each increment of load. The test results are stated in Table 8 and the axial force - U.V. trace displacement relation is presented in figure 50. The problem of pressure adjustment discussed in Chapter 6, section 2, prevented assessment of hysteresis at any point other than zero load.

6.5 Calibration of upper platen and lower platen torque transducers

Two force transducers, Kistler Type 9031, mounted one at each of the spindle bearing mounting plates for the final drive gear on

both the upper and lower platen drives, are connected to channels 2 and 3 of 3-channel charge amplifiers Kistler Type 5801, serial number 17702 and a galvanometer Type B100 in the U.V. recorder. The transducers are selected in pairs to exhibit similar force/charge coefficients such that the outputs can be summed directly by parallel connection. Calibration was carried out by applying known tangential forces to a radial arm attached to the die adaptor plate which provided torques of various magnitude to each of the platens in turn. The input shaft of the respective reduction gearboxes were locked against rotation thus making it possible to apply a static torque to the gear train. The tangential force applied at a distance of 0.685 m from the platen axis was measured using a 0 to 200 lbf (0 to 0.89 kN) Salter spring balance, Serial number 050829, which was connected to the main frame of the machine by a length of screwed rod. Nuts at the fixing point on the screwed rod were used to adjust the force to the value required within the range 0 to 200 lbf (0 to 0.89 kN). The U.V. trace displacements were recorded for various values of applied torque. The results are stated in Table 9, and the applied torque - U.V. recorder trace displacement for upper and lower platen are presented in figures 51 and 52 respectively.

6.6 Calibration of the lower platen translational displacement transducer

The transducer incorporated is a 200 mm stroke rectilinear conductive plastics-wire wound hybrid potentiometer, Penny & Giles Limited, Type HLP 190, Serial number 9080056.

The output is connected to a ± 199.9 m.v. L.E.D. digital panel meter, R.S. Limited, Type 258-11 and a galvanometer Type B100 in the U.V. recorder.

The translational displacement of the lower platen was measured with reference to the upper plate of the main frame using a 0.01 mm sensitivity dial indicator and a vernier height gauge.

The translational displacement was adjusted to various values and the digital voltmeter readings, together with galvanometer deflections were recorded. The results are stated in Table 10 and presented in the graph, figure 53, which illustrates the relation between translational displacement of the lower platen and U.V. recorder trace displacement.

The resistor in series with the galvanometer may be shunted if so desired to increase the recording sensitivity from 1.07 mm to 4.12 mm galvanometer deflection per mm translational displacement.

6.7 Calibration of the system for indication of angular velocity and displacement of the upper and lower platens about the axes Z_2 and Z_1 respectively.

On each measuring system a proximity detector, Setpoint Limited, Type I.E.3001, is operated from the outside diameter of the platen spindle 48 teeth drive gears attached to the platen spindle. Hence 1 rev/sec of the gear is equivalent to 0.048 kHz.

For each of the upper and lower platens the 2.5 V.D.C. square

wave pulses from the proximity detector are connected to a four digit panel mounted combined frequency meter and counter, R.S. Limited, Type 527-824.

The method of calibration was to use a Dawe Instruments Limited Strobflash Type 1200E, Serial number 34389. The Strobflash is fitted with a vibrating reed frequency standard operating at 50 Hz. The flash frequency was adjusted to a zero beat frequency with the 50 Hz reed and the flashing light source was then directed onto the teeth of the upper and lower platen drive gears in turn. The speed control on the variator was used to adjust the angular velocity of the respective platen such that stationary images of the teeth occurred at 50 Hz, 100 Hz, 150 Hz and 200 Hz. The digital readings of the frequency meter were noted and the test results are stated in Table 11(a).

A further test was conducted using the calibrated rev/min scale on the Strobflash and the results stated in Table 11(b).

The digital frequency indicator is based upon counting circuits driven from an oscillator and can with confidence be expected to perform to similar accuracies at speeds in the range from 60 to 30 rev/min, which is not within the scope of the calibration equipment used. Some difficulties were experienced with readability of the scale when using the rev/min speed indication on the Strobflash. The digital frequency indicators can be used with confidence in the pulse count mode since checks at various frequencies showed stable readings with no evidence of pulse *drop out*. One pulse is equivalent to $1/48$ of a degree of arc. The results indicate that the pulse count system can

be used for measuring small differences in velocities of the upper and lower platens.

It was concluded that the digital meters could be used with no correction applied when used in either the pulse count or frequency modes.

6.8 Calibration of the system to measure pressure at the ram inlets

The oil damped, 3000 lbf/in² (20.7 MPa) range pressure gauges, fitted to both the extension and retraction inlet ports of the hydraulic ram, were calibrated at 500 lbf/in² (3.45 MPa) increments over the range 0 to 3000 lbf/in² (0 to 20.7 MPa), using a substandard pressure gauge Budenburg Limited, Serial number 10280537 tapped into the appropriate pressure line. The gauges tested were found to be within $\pm 1\%$ full scale.

A diffused strain gauge pressure transducer, Druck Limited Type PDCR1D, Serial number 22204, is fitted to the pressure line adjacent to the ram extension inlet port. The 0 to 5 mV. D.C. output from the transducer is connected in parallel to a galvanometer in the U.V. recorder and monitor terminals on the main control panel.

The calibration of the system was carried out by setting the translational displacement limit stops in both directions at approximately 75 mm and adjusting the pressures to the ram in 500 lbf/in² (3.45 MPa) increments over the range 0 to 3000 lbf/in² (0 to 20.7 MPa) using the hydraulic pressure control valve and

monitoring the applied pressure using the substandard pressure gauge.

The results obtained are stated in Table 12, and the graph figure 54, illustrates the relation between ram pressure and U.V. recorder trace displacement.

The comments made in Chapter 6, Section 2 are the facility to decrease pressure.

6.9 Calibration of the inclination of platen axes transducer

The transducer is a 200 mm stroke, Penny & Giles Limited rectilinear conductive plastics wire wound hybrid potentiometer Type HLP 190 Serial number 9080068. The output of the transducer is fed to 199.9 mV. digital panel meter, R.S. Limited, Type 258-11, and a Type B100 galvanometer in the U.V. recorder.

The angle of inclination of the axes of the platens was measured using a clinometer Hilger and Watts Limited, Type FF 2-3, Serial number 142/43/45. The datum used was obtained from the face of the die adaptor plate of the lower platen. The clinometer was then placed on the die adaptor plate of the upper platen and adjusted by means of the handwheel at one degree increments over the range. The scaling potentiometers in the indicator circuit were adjusted to read direct one millivolts equivalent to one degree to a tolerance of $\pm 0.05^\circ$. The inclination of the platen was adjusted in increments of one degree using the clinometer as a monitor and the U.V. recorder trace displacement was determined.

The results are stated in Table 13 and the graph, figure 55 illustrates the relation between inclination of platen axes and U.V. recorder trace displacement. Hysteresis in the system was found to be $\pm 0.02^\circ$.

6.10 Summary of initial assessment of performance

Subject to the various points made regarding compliance, speed range and hydraulic pressure adjustment, the machine appeared to be satisfactory. The torque traces from the recorder exhibit a high frequency content. Low pass filters were designed and built but it was considered prudent to leave the signal raw and take the mean of the analogue trace pending further investigation.

7.1 Cold upsetting of cylindrical workpieces of initial height to diameter ratio $H_0/D_0 \approx 0.2$ using a conical upper platen and flat lower platen

(a) Introduction

The rotary forging process is considered to be particularly suitable for forming axisymmetric components of low final height to diameter ratio. Previous workers have developed satisfactory methods for prediction of the axial force required during upsetting using machines with orbital motion to conduct experiments.

It was considered necessary to verify that the methods of predicting axial force are equally applicable where the machine used is an inversion of orbital motion.

(b) The objects of the experiment were to :

- (i) assess the 0.5 MN rotary forging machine, designed and constructed at The City University for suitability to conduct experiments on the upsetting process,
- (ii) identify any findings which differed from those of previous workers using machines of orbital configuration,
- (iii) compare the values of axial force obtained by

experiment with those predicted using the methods proposed by Hawkyard et al [24], Kubo et al [34], and a method proposed by the author,

(iv) determine from experiment the process efficiency,

(v) to conduct initial investigation to identify characteristics or limitations applicable to the upsetting process.

(c) Applicable theories :

(i) Hawkyard et al [24], proposed that the axial force, F_H' , with an allowance for elastic deformation of the upper die may be predicted by the following:

$$F_H' = F_H(\gamma'/\gamma)(\bar{s}'/\bar{s}) \quad (51)$$

where F_H = predicted axial force with no allowance for elastic deformation of the upper platen

$$= 2.k.\gamma.\bar{s}.r \quad (52)$$

The method is based upon the product of normal pressure and area of the upper die-workpiece interface. It is assumed that a plastic hinge exists and causes a compressive stress, σ , to act in a circumferential direction with respect to the workpiece and perpendicular to the surface upon which the normal pressure, P , is acting. The value, $\gamma = P/2k$ is derived from the indentation model and theory proposed by Hill [67].

The relation between $H/\bar{s}$ and $\gamma = P/2k$ given by Hawkyard et al [24], is illustrated in figure 56, which shows that the flat lower platen is assumed to be an indenter.

H = current workpiece height

$$\bar{s} = A_H/r$$

= mean circumferential width of area of contact at the upper conical die-workpiece interface

A_H = estimated current area of contact at upper conical die-workpiece interface

$$= (2.2^{1/3}/3)(fr/\theta)^{1/2}r \quad (53)$$

$\bar{s}'$ = mean circumferential width of area of contact with allowance for elastic deformation of the upper die

P = die pressure

$k = \sigma/2$ = current yield shear stress

σ = current flow stress

r = current workpiece radius

f = axial feed per revolution

θ = angle of inclination of the platen axes

$$\bar{s}'/\bar{s} = [1 + (C_E F_H / fr)]^{1/2} \quad (54)$$

where C_E = constant for steel rolls from Johnson [60] and Slater [61].

The value used for all predictions associated with this experiment was $0.0216 \text{ mm}^2/\text{kN}$.

(ii) Kubo et al [34] proposed that the axial force, F_K , may be predicted by the following :

$$F_K = C\sigma A_K \quad (55)$$

where the area of contact between upper die and workpiece A_K is given by :

$$A_K = 1.278r^{1.37}f^{0.63}/\tan^{0.63}\theta \quad (56)$$

C = constraint factor and is assumed to be unity for the purpose of this experiment

r = current workpiece radius

f = axial feed/revolution

θ = angle of inclination of platen axes

σ = current flow stress

(iii) From theory applicable to rolling and analogy to rotary forging, Hawkyard et al [24] make an allowance for roll flattening by considering the roll force/unit width to be given by F_H/r . The force, F_p , figure 57, which produces the equivalent of roll pressure may therefore be considered to be acting at the centre of mean circumferential width of contact given either by $\bar{s}$ or $\bar{s}'$. It is suggested that, when attempting to predict the axial force to cause upsetting, the upper die could be considered as a rigid surface of length, r , and mean circumferential width $\bar{s}$ hinged instantaneously about the generator OA of the conical die, illustrated in figure 57.

However if F_p is considered to be the force required at the centre of the rigid surface this is not coincident with the centroid of the plastically

deforming region where the force F_H^1 may be assumed to be acting. An allowance may be made, based upon these considerations, when applying the method proposed by Hawkyard et al [24] by considering an element of the conical die-workpiece contact area of radius r_e , width dr_e , and length of arc, s , from OA, as illustrated in figure 57, where

$$s = 2.2^{\frac{1}{2}}(2fr/\theta)^{\frac{1}{2}}/3 \quad (57)$$

where r = current workpiece radius.

If the radius of roll is considered to be large compared with the average circumferential width of the area of the interface, and if perfect rolling is assumed to be taking place on the line OA which forms a frictionless hinge, then the force F_p , on the roll may be assumed to be acting at a distance $\bar{s}/2$ from OA as illustrated in figure 57 and given by :

$$\bar{s}/2 \approx 2^{\frac{1}{2}}(f/\theta)^{\frac{1}{2}}r^{\frac{1}{2}}/3 \quad (58)$$

The moment of area, dM_A , of an element of radius r_e , width dr_e and length of arc, s , about the generator OA is given approximately by :

$$\begin{aligned} dM_A &\approx (s/2)sdr_e \\ &\approx [(2fr_e/\theta)^{\frac{1}{2}}(2fr_e/\theta)^{\frac{1}{2}}/2]dr_e \\ &\approx (fr_e/\theta)dr_e \end{aligned}$$

the total moment of area, M_A , is given by :

$$\begin{aligned} M_A &\approx \int_0^r (fr_e/\theta)dr_e \\ &\approx fr^2/2\theta \end{aligned} \quad (59)$$

The centroid of the area of contact between the upper die and workpiece is a distance, U , from the generator OA as shown in figure 57 and from equations (57) and (59)

$$U \approx 3(fr/\theta)^{\frac{1}{2}}/4.2^{\frac{1}{2}} \quad (60)$$

It will be assumed that F_H is uniformly distributed over the upper die-workpiece contact area and can therefore be considered to act at the centroid a distance, U , from the generator OA. Then considering moments of the forces F_H and F_P about OA and using equations (58) and (60)

$$\begin{aligned} F_P \bar{s}/2 &= F_H U \\ F_P 2^{\frac{1}{2}}(fr/\theta)^{\frac{1}{2}}/3 &\approx F_H(fr/\theta)^{\frac{1}{2}}3/4.2^{\frac{1}{2}} \end{aligned} \quad (61)$$

$$\text{or} \quad F_P = 9F_H/8 \quad (62)$$

and when allowance is made for elastic deformation of the roll from $s'/s = (1 + C_E F_P/fr)^{\frac{1}{2}}$

$$F'_P = F_P(\gamma'/\gamma)(\bar{s}'/\bar{s}) \quad (63)$$

From the reasoning based upon the various assumptions stated it follows that for equilibrium in the axial direction an additional force must be assumed to exist which is given by $F'_P - F'_H$, and acts on the line of the hinge, OA, in the direction of the axis Z_1 .

This concept is subsequently shown to be of no significant advantage when predicting axial force but does however lead to a tentative method of estimating the torque required to cause the conical upper die to rotate as described in Appendix 1.

(iv) For the purpose of this experiment the process efficiency is defined as the energy required to cause a fractional reduction in height by quasi-static axisymmetric homogeneous deformation expressed as a percentage of the energy required by experiment to cause the same reduction in height by rotary forging. To evaluate the process efficiency it is convenient to consider the energy required to cause a reduction in height equal to, f , the axial feed for one revolution.

The energy, E_f , required to cause a fractional reduction in height, $R = (H - f)/H$, of a cylinder of current height, H , and current diameter, r , by quasi-static, axisymmetric, homogeneous deformation is given by :

$$E_H = \sigma \cdot \pi \cdot r^2 \cdot H \cdot \ln[H/(H - f)] \quad (64)$$

where σ = flow stress

The energy, E_E , required by experiment to cause a fractional reduction in height $R = (H - f)/H$, equivalent to one revolution of the plastically deforming region relative to the workpiece, may be stated as :

$$E_E = F_E \cdot f + 2\pi T_T \quad (65)$$

where T_T = total experimental torque applied to the platens to cause rotation.

The process efficiency, η , expressed as a percentage, is then given by :

$$= 100 \cdot \sigma \cdot \pi \cdot r^2 \cdot H \cdot \ln[H/(H - f)] / (F_E \cdot f + 2\pi T_T) \quad (66)$$

(d) The equipment used :

(i) 0.5 MN rotary forging machine complete with instrumentation as specified in Chapter 4.

(ii) 150 mm diameter upper dies of semi-conical angle 88° , 86° , 84° , 82° and 80° , together with a 150 mm diameter flat lower platen, having a workpiece location recess of 12.95 mm diameter and 1.0 mm deep positioned axisymmetrically in the upper surface. The upper dies and lower platen were manufactured from air hardening tool steel to BS.4659 H.13, hardened and tempered to 35 Rockwell C scale and polished to a surface roughness $\leq 0.15 \mu\text{m C.L.A.}$

(iii) U.V. recorder

(iv) high pressure lubricating paste, Dow Corning Limited, type Molykote D

(v) cleaning and degreasing fluid

(vi) cylindrical specimens $H_0/D_0 \approx 0.20$ of the following materials and diameters :

(1) Aluminium Alloy BS.1476 HE.30 TF, (fully heat treated condition).

Diameters of 37.5 mm, 56.95 mm and 76.25 mm.

The aluminium alloy with low strain hardening characteristics is in accordance with the material specification of the specimens used by

Hawkyard et al [24] and selected to permit comparison of results.

(2) Alloy steel BS970 En.18SS 37.5 mm diameter.

This alloy steel was selected to provide workpieces of high flow stress with marked strain hardening characteristics.

The true compressive stress-natural compressive strain characteristics for specimens from each bar of material were determined by quasi-static axisymmetric compression tests. The results are presented in figure 58.

All specimens were from bar stock and machined on end faces only with a 12.95 mm diameter 1.0 mm axial length location spigot on one face.

(e) Procedures :

The upper conical die selected together with the recessed flat lower platen were fitted in position. The appropriate pulley wheel was fitted to simulate a relative motion at the upper die-workpiece interface similar to that produced in orbital type machines with induced skidding.

That is $\omega_1 = \omega_2$

The rotational speed was adjusted to the value desired and selected from the range 60 rev/min to 240 rev/min. The angle of inclination of the platen axes, θ , was adjusted to give a value of $\chi = \pi/2$. Values of θ used were 2°, 4°, 6°, 8° and 10° for aluminium alloy specimens and 4°, 6°, 8° and 10° for alloy steel specimens.

The lower stops of the translational displacement system were adjusted to provide clearance for the largest specimens to be loaded. The hydraulic pressure regulator and flow control valve were adjusted to the desired values.

Specimens were cleaned, degreased and then lubricated on all surfaces, using a brush to apply the paste. The initial diameter D_0 and height H_0 were measured and noted for each specimen.

Each test commenced with the lower platen assembly in a position determined by the ram being fully retracted against the lower translational stops. Following location of the specimen on the lower platen, rotational drive was commenced. By operation of the translational displacement control the upsetting process was caused to proceed to approximately 0.5 fractional reduction in height. The lower platen was retracted, platen rotation stopped and the specimen removed.

For the test used to assess the methods of predicting axial force the axial feed/rev and axial force were determined from the U.V. recorder trace at the position on the time base where reduction in height of the specimen was equivalent to a natural compressive strain $\epsilon_z = 0.15$. The axial feed/rev, f , was determined using the one second time base and the analogue record of translational displacement, together with the measured rotational speed of the platens.

At a point on the time base equivalent to $\epsilon_z = 0.15$ the total

torque T_T , was determined from the recorded values of upper and lower platen torque. An allowance was made for frictional torque in the platen bearing system based upon data obtained from the bearing manufacturer. The sum of the upper and lower platen frictional torque - axial force relation is shown in figure 59. The current radius, r , at $\epsilon_z = 0.15$, was estimated assuming that each specimen remained cylindrical. Using the appropriate theories and the measured values together with the value of flow stress, σ , determined from figure 58, the values of F'_H , F_K , F'_P , were calculated using the equations stated in 7.1(c). The values F_A/F_E , F_K/F_E , F'_H/F_E , F'_P/F_E and n , were then determined for each test.

The above procedure was then repeated for 48 specimens of aluminium alloy and 9 specimens of alloy steel.

To conduct initial investigations into the upper limits of *specific rotary forging force* applicable to specimens of aluminium alloy H.30 TF. $D_0 = 76.24$ mm, $H_0/D_0 \approx 0.2$, tests were conducted with platens of various conical angles and the platen axes adjusted to the appropriate angles, θ . With the pressure in the hydraulic system adjusted to give the minimum axial force ≈ 20 kN and the platens rotating at 120 rev/min the lower platen was raised such that the surface defined by the generator of the upper conical die was in contact with the upper surface of the cylindrical specimen.

The pressure was increased by an increment of approximately 10 kN and the process permitted to continue for approximately

100 revolutions. The lower platen was then retracted, rotation was stopped and the distance between the upper surface of the specimen and the face of the flat lower platen was measured using a depth micrometer. The process was repeated with increments in axial force until a reduction of approximately 0.1 mm from the initial height was noted. This is equivalent to an axial feed of approximately 0.001 mm per rev. The values of axial force and the translational displacement of the lower platen were recorded on the U.V. recorder paper.

An alternative procedure was an uninterrupted method to determine the commencement of the forging process by slow and continuous increase of the hydraulic pressure with the platens rotating, thus causing the axial force to be increased to such a value that axial feed per revolution could be detected from the U.V. recorder trace.

To determine the lower limit of *specific rotary forging force* the axial force, and hence axial feed/rev, were increased on a number of tests up to the values at which workpiece fracture occurred.

One specimen of alloy steel En.18, 38.86 mm diameter

$H_0/D_0 \approx 0.2$, $\theta = 10^\circ$, was also used to determine the maximum value of F_E for $f \approx 0$.

(f) Results

The results stated in Tables 16 and 17 are from tests on

specimens of aluminium alloy and alloy steel respectively, the values of both axial force, F_E , and total torque T_T , obtained are stated for tests using specimens of specified values of D_0 at various values of, f , θ , and rotational speeds.

The mean values of F_E/F_H' and F_E/F_P' , together with the respective values of standard deviation for both aluminium alloy and alloy steel specimens are presented in Table 18.

Typical surface roughness values on the specimen faces after upsetting were $0.20 \mu\text{m}$ C.L.A. and the final thickness variations were $<0.25 \text{ mm}$. The photographs presented in figure 60(a) show undeformed specimens and figure 60(b) illustrates the small departure from cylindrical form of typical aluminium alloy and alloy steel specimens following a fractional reduction in height of $R \approx 0.42$, $\epsilon_z = 0.55$ and $R = 0.47$, $\epsilon_z = 0.60$ respectively.

The photograph in figure 60(c) shows a specimen fractured at a fractional reduction in height, $R = 0.15$, $\epsilon_z = 0.16$. $\theta = 10^\circ$, $F_E = 273 \text{ kN}$. Figure 60(d) illustrates the upper surface of an alloy steel specimen showing the spiral pattern referred to in [24] which is also evident in figure 60(c). Figure 60(d) shows the initial stage of the departure from cylindrical form which occurred in the tests used to establish the axial force - axial feed/rev relation at very low feed rates to determine the upper limit of *specific rotary forging force* at $\theta = 10^\circ$. The relation between *specific rotary forging force*, F_A/F_E and F_E/F_K for both aluminium alloy

and alloy steel specimens is presented in the graph of figure 61. The results of tests on specimens of aluminium alloy HE.30 TF. $D_0 = 76.24$ mm, $H_0/D_0 \approx 0.2$, to determine the relation between axial feed/rev and axial force for various values of, θ , including the attempts to measure the force which causes fracture are stated in Table 19. A maximum force of 61 kN for $f \approx 0$ and a value of $F_A = 830$ kN was obtained for the alloy steel specimen. This gave $F_A/F_E = 13.6$ for $\theta = 10^\circ$.

Figure 62, illustrates the relation between F_A/F_E , θ , and the upper and lower limits of upsetting by the rotary forging process. The values of F_A/F_E obtained at $D_0 = 76.24$ mm and $\epsilon_z = 0.15$ during the upsetting tests as stated in Table 16 are also plotted in figure 62.

The maximum values of F_E for $f = 0$, obtained from experiment were used to determine the upper limits of the *specific rotary forging force* applicable to the specimens over the range $\theta = 2^\circ$ to $\theta = 10^\circ$. Figure 63 shows the relation between axial force F_E , and axial feed/rev obtained from data in Tables 16 and 19 for workpiece $D_0 = 76.24$ mm and $H_0/D_0 = 0.2$ and values of $\theta = 2^\circ$ to $\theta = 10^\circ$. The scatter of results for values of $\theta = 8^\circ$ and 10° was such that a separate curve for 8° could not be drawn through the points determined from the experimental results.

The logarithmic relation between the value of torque and axial force obtained from tests on specimens $D_0 = 76.24$ mm,

$H_0/D_0 \approx 0.2$, is shown in figure 64. Figure 65 shows the relation between process efficiency and axial feed/rev for aluminium alloy specimens $D_0 = 76.24$ mm, $H_0/D_0 = 0.2$ and $\theta = 10^\circ$. The specific rotary forging force limit is also shown.

The temperature of the specimens did not exceed $+55^\circ\text{C}$ at completion of any of the upsetting processes.

(g) Discussion of results :

The values of F_E/F'_H obtained and stated in Table 16 confirm the method of prediction to be satisfactory and show a similarity to the results presented by Hawkyard et al [24] operating with orbital motion using specimens of similar material, dimensions and platens of similar geometry. The results stated in Table 18 show the method of predicting axial force proposed by Hawkyard et al [24] to be satisfactory when applied to alloy steel specimens. From the results stated in Table 18, it is apparent that the modified method of predicting axial force proposed by the author give values of F_E/F'_p marginally closer to unity with a reduction in standard deviation when compared with the values of F_E/F'_H . Also from Table 18 the test on alloy steel using the modified method of predicting axial force gives reduced under estimates and a marginal reduction in standard deviation. The concept of the existence of an instantaneous hinge offers no marked advantage. However, as proposed in Appendix 1, the concept may be extended to provide a tentative method of predicting torque required to cause rotation of the platens.

The method of predicting axial force proposed by Kubo et al [34] gave values of F_E/F_K ranging from approximately 1.40 to 4.75. The data obtained from the experiment was used to test for correlation between F_A/F_E and F_E/F_K .

The application of linear regression theory gave

$$F_A/F_E = 2.66 F_E/F_K + 0.88 \quad (67)$$

With a correlation coefficient of 0.77 and applying linear regression to $\log (F_A/F_K)$ and $\log (F_E/F_K)$ gave

$$F_A/F_E = 2.86(F_E/F_K)^{1.04} \quad (68)$$

with a correlation coefficient of 0.79.

The graph of the experimental results presented by Kubo et al [34] for steel specimens of $H_0/D_0 = 1.0$ was used to obtain data to calculate values of F_E/F_K which were also plotted on the graph in figure 61. The result suggests that an empirical method of estimating the axial force F , for specimens of $H_0/D_0 \approx 0.2$ given by :

$$F'_K = 0.60.F_A^{0.49}F_K^{0.51} \quad (69)$$

and substituting equations (55) and (56) in equation (69) gives :

$$F'_K = 1.19.\sigma.r^{1.68}.f^{0.32}/\tan^{0.32}\theta \quad (70)$$

The method is relatively simple to apply and may be satisfactory for many manufacturing engineering applications. From the results stated in Table 19 and presented in figure 62, it becomes clear that a zone exists within which satisfactory rotary forging proceeds and can be expressed in terms of upper and lower limiting values of *specific rotary forging force* for various values of θ . The test

procedures to determine the upper limits of F_A/F_E entailed direct measurement of the specimen height following a minimum of 100 revolutions and therefore the fundamental nature of the test does not lead to questions on validity of the results. The findings, however, indicate that a variance exists between the results from experiments and the theories used in this work to predict axial force since the applicable theories predict that when $f = 0$, the axial force $F_K = 0$, $F_H' = 0$ or $F_P' = 0$. The experimental results show that for $f = 0$, F_E may be much greater than 0. The test on the alloy steel specimen also gave an upper limit of *specific rotary forging force* of 13.6 for $\theta = 10^\circ$ which is remarkably similar to that obtained for the aluminium alloy.

The values of F_E to cause fracture were used to determine the lower limit of *specific rotary forging force* over the range 4° to 10° . The point was measured by stopping the process when fracture was either heard or seen to have occurred and the forces measured can therefore only be considered to be approximate. From the U.V. records it was observed that as the axial feed/rev at which fracture occurred, was approached, cyclic variations became apparent in both the axial force and torque traces. Presumably this was due to circumferential variation in workpiece characteristics. It should be noted that fracture was not attempted for $\theta = 2^\circ$. This was due to the hazardous nature of the experiment and the characteristics of specimen fracture. The machine is fitted with external guards but

modifications are required to seal the working zone from the main drive mechanism in a manner which would protect the moving parts from flying debris.

The procedure used to detect fracture was considered unsatisfactory for further investigation and an improved experimental technique should be devised. The outcome of the experiment has, however, led the author to the opinion that further work leading to an understanding of the fracture mechanism, and an attempt to establish the forging parameters which could be stated in the form of various fracture criteria, would make a valuable contribution to knowledge of the process.

From the photograph in figure 60(c) fracture appears to be due to a tensile stress occurring on a plane which is located with respect to the cylindrical workpiece both approximately diametrically, and approximately parallel to the axis. The theory proposed in [24] which was developed from a plastic hinge concept, assumes the stress to be compressive.

From the two pieces of the specimen illustrated in figure 60(c) there is some evidence to support the hinge concept albeit the direction of the applied stress had reversed and increased to such magnitude that instability in tension and subsequent fracture was caused. It therefore appears necessary to consider the conditions which lead to transition of stress, σ_H , from compressive to tensile. The cases of indenting and cutting with two equal dies or

indenters, described by Johnson and Mellor [60] Section 14.3 to 14.5 and by Slater [61] Section 7.4.2 to 7.4.7 could form the basis from which to commence future investigation.

It should be noted that the procedures used and described in section 7.1(e) were such that initial contact between the upper die and workpiece was made with a rolling motion. Approximate symmetry of shape of the contact area at the die-workpiece interface would only occur during elastic deformation and therefore would not conform to the condition at the plastic indentation phase as described by Standring, Moon and Appleton [68].

Kuhn, Lee and Erturk [69], in their study with the objective of formulating fracture criteria for cold forming, describe and illustrate what they refer to as vertical cracking during upsetting of cylindrical specimens. They conclude that plots of tensile versus compressive strain are valuable for the study of fracture in processes such as *heading*. Samanta [70] also used natural strain relations expressed by a system of cylindrical co-ordinates for his work upon the effect of friction and specimen geometry on fracture in upsetting.

The manner in which the fracture phenomenon limits the lower value of *specific rotary forging force* is associated with axial feed/rev and is clearly of industrial significance. From experiment it becomes clear that when $H/D < 0.2$ with given material characteristics and forging conditions, the

process has limitations which can be stated in terms of *specific rotary forging force*. It is also interesting to note that the value of upper limits of *specific rotary forging force* determined for the alloy steel specimens $\theta = 10^\circ$ is similar to the values obtained for aluminium alloy albeit the values of D_0 and material flow stress differ widely.

From the values of process efficiency obtained it is difficult to reach conclusions with the exception that from figure 65 the relation shows that for specimens of aluminium alloy $D_0 = 76.24$ mm, $H_0/D_0 = 0.2$ and $\theta = 10^\circ$, the efficiency of the process increases as axial feed/rev increases but is limited by *specific rotary forging force* constraints as illustrated. Perusal of the results indicates that as workpiece initial diameter increases the process efficiency tends to increase and also that when forging the alloy steel specimens the efficiency appears to be greater than when forging aluminium alloy. Also for given axial forces when $f = 0$, energy is required for rotation and hence the efficiency, $\eta = 0$.

Little can be concluded from the values of torque measured for specimens of aluminium alloy $D_0 = 76.24$ mm, $H_0/D_0 = 0.2$ for $\theta = 10^\circ$, with the exception that torque is apparently approximately proportional to axial force. It could also be stated that for θ , over the range 2° to 10° with upper dies of the appropriate semi-conical angle, and for a given axial feed/rev the inclination of the platen axes had apparently no pronounced influence upon the constant of proportionality.

From the experimental values of torque the power made available for rotational drive in the design would be inadequate if the full translational power could be applied. For design purposes the torque requirements were derived from an energy concept with no allowance for inefficiency due to power dissipated to cause the rotation which was not causing plastic deformation.

The energy which is required to cause rotation when the axial feed/rev ≈ 0 , is presumably due to friction at the die-workpiece interface together with hysteresis or inelastic deformation processes which occur and are not manifest in a reduction in height. The elastically deforming zones are caused to rotate in both the workpiece and dies. The relatively small temperature rise in the workpieces indicated that the amount of mechanical energy converted to heat energy, was not of significance. Since an elastically deforming zone is caused to rotate when a plastically deforming zone is rotating then presumably it is necessary for an allowance to be made when attempting to predict the rotational drive torque.

It is also indicated that when predicting axial force, consideration should be given to making an allowance for the area of contact between upper die and workpiece where elastic deformation only is occurring. In particular the zone in which elastic recovery is taking place immediately following the generator OA shown in figure 57.

Study of rolling bearing engineering and in particular the Herzian pressure relationship could be of assistance.

7.2 Cold upsetting of cylindrical workpieces of initial height to diameter ratio $H_0/D_0 \approx 0.2$ using upper and lower dies of equal conical angle

(a) Introduction :

The purpose of conical dies on both the upper and lower platens is to cause plastically deforming regions to rotate simultaneously in both the upper and lower surfaces of the workpiece. The advantage as a manufacturing process is associated with symmetry of plastic deformation which can be caused about a plane which is perpendicular to the workpiece axis. The possibility of using dual rolling contoured dies provides greater scope when designing tools. The process is also predicted to provide a means of obtaining a significant increase in the axial feed/rev for a given *specific rotary forging force*.

To achieve an ideal kinematic arrangement as illustrated in figure 66(a), would require the upper and lower die-workpiece interfaces to be moving during the forging process such that they remain equidistant from the intersection of the Z_1 and Z_2 axes.

When axial displacement of the lower platen only occurs the dimensions of the upper die can be so arranged that the ideal kinematic condition may be approached as the workpiece reaches its final form. No attempt was made in this experiment to optimise the arrangement by die modification since the upper die dimensions are correct when used for

upsetting with a single conical die. The arrangement used for the experiment illustrated in figure 66(b), shows that eccentricity occurs in the relative motion at the upper die-workpiece interface. The magnitude of the eccentricity Δr , is given approximately by :

$$\Delta r \approx H \tan \theta \quad (71)$$

and clearly eccentricity Δr decreases as the current workpiece height, H , decreases.

For the experiments conducted the maximum initial workpiece height is 15.50 mm, and the maximum value of $\theta = 8^\circ$.

Assuming the apex of the upper conical die to be coincident with the axis Z_1 of the lower platen the maximum value of $\Delta r = 2.14$ mm.

For values of $\theta \leq 8^\circ$, the axial feed/rev at each of the upper and lower die-workpiece interfaces in the direction of the workpiece axis is given approximately by $f/2$, where f , is the total axial feed/rev of the lower platen in the direction of the axis Z_1 .

(b) Objectives :

The objectives of the experiments on the cold upsetting of cylindrical workpieces of $H_0/D_0 \approx 0.2$, using two conical dies of equal conical angle were to :

- (i) assess the machine and tooling, including the method of workpiece location, for suitability to carry out the upsetting process,

(ii) compare predicted values of axial force with those obtained from experiment,

(iii) make comparisons with the upsetting process using a single conical die,

(iv) determine the efficiency of the process.

(c) Applicable theory

From the assumption that the value of axial feed/rev at the upper and lower surfaces are equal and given by $f/2$ where $f \cong$ the total axial feed/rev, then the theory proposed by Hawkyard et al [24], may be considered to be applicable by the following reasoning.

The upper and lower dies are of equal conical angle and for the generators of the two dies in contact with the workpiece to be parallel, then the semi-conical angles of each die is $(\pi - \theta)/2$ and each platen axis is therefore at an angle of $\theta/2$ to the axis of the workpiece as illustrated in figure 66(b).

When determining the value of $\gamma = P/2k$ the conical upper and lower platens may be considered as opposed indenters with two equal dies and therefore no assumptions are required as for the flat lower die in reference [24].

The average width $\bar{s}$ of the die contact areas is then given approximately by :

$$\bar{s} \approx 2.2^{\frac{1}{2}}(2fr/2\theta)^{\frac{1}{2}}/3 \quad (72)$$

$$\approx 2.2^{\frac{1}{2}}(fr/\theta)^{\frac{1}{2}}/3 \quad (73)$$

The conical dies are assumed to be acting as opposed rolls, and therefore when making an allowance for elastic deformation of the rolls in order to predict the mean circumferential width of contact $\bar{s}$, the condition may be described as being identical to that illustrated in [60] figure 11.49 on page 339 and [61] figure 7.42 page 251. The assumptions made in [24] related to the flat lower platen acting as an indenter are unnecessary and therefore

$$\bar{s}'/\bar{s} = [1 + (0.0216 F_H/fr)]^{\frac{1}{2}} \quad (74)$$

The predicted value of axial force for dual dies of equal conical angle is therefore similar to that obtained in [24] for a single die using half the angle of inclination of platen axes and half the axial feed/rev.

When allowance is made for elastic deformation of the rolls the axial force F_H' , for dual platens of equal conical angle, is given by equation (51) and γ' is obtained from the graph in figure 56 and from value $H/\bar{s}$. From reasoning similar to the above and the method proposed in 7.1(c)(iii), the axial force F_p' , may also be predicted from equation (63).

(d) Equipment :

The upper dies used in the experiment, of semi-conical angle 88° and 86° , were those as used and described in 7.1(d). The lower conical dies were 150 mm diameter with semi-conical angles of 88° and 86° , each provided with a workpiece radial location recess 12.95 mm diameter and

1.0 mm deep positioned at the apex as illustrated in figure 66(b). The dies were constructed from tool steel to BS.4659 BH.13 hardened and tempered to 35 Rockwell C scale. The operating faces were polished to produce a surface roughness $\leq 0.15 \mu\text{m C.L.A.}$

(e) Procedures :

The procedures adopted were similar to those described in 7.1(c). Fracture of specimens was not attempted for the reasons stated in 7.1(g). Two values of θ were used namely 4° and 8° , and all specimens were of aluminium alloy H.30 TF. Tests were conducted on 25 specimens of various initial diameters and were also carried out on 3 additional specimens to determine the value of axial force at which reduction in height commenced.

(f) Results :

The results of the tests are presented in Tables 20 and 21. From Table 20 the mean value of F_E/F_H' was found to be 1.07 and standard deviation 0.11. The mean value of F_E/F_P' was found to be 0.94 and the standard deviation 0.08. The results stated in Table 21 were obtained from tests using specimens of initial diameter $D_0 = 76.24 \text{ mm}$, $H_0/D_0 \approx 0.20$. For comparison purposes the relation between axial force and axial feed/rev with $\theta = 4^\circ$, is presented in figure 67, together with the results obtained from the experiment described in 7.1, using similar specimens and a conical upper die of 88° semi-conical angle, $\theta = 2^\circ$, and a flat lower die. Typical surface roughness on the upper and

lower faces of the final specimen was $\pm 0.25 \mu\text{m C.L.A.}$ The departure from cylindrical form appears to be less pronounced at equivalent axial feed/rev than with similar forging conditions when using a single conical die and flat lower die.

(g) Discussion :

The machine is apparently satisfactory for conducting the experiment and the tooling, including the method of locating the workpiece, was found to be adequate. The advantages of the design concept of the machine which permits the workpiece axis to remain stationary when using dual opposed platens of equal conical angle are demonstrated.

The results indicate that for the range of specimens tested the methods used to predict axial force were satisfactory for manufacturing engineering purposes.

The curves presented in figure 67 show clearly the advantages in terms of axial feed/rev - axial forces relation when the dual conical die arrangement is compared with that using a single conical die. The experimental results from the two configurations expressed as a ratio, show similarity to the ratios obtained from predicted values. For example experimental results presented in the graph figure 67, at $f^{\frac{1}{2}} = 0.30$ ($f = 0.09 \text{ mm/rev}$) the axial forces are 248 kN and 336 kN for dual and single conical dies respectively. Since the platen axes are inclined to the

workpiece axis at 2° in both cases the axial feed/rev is equivalent to 0.045 mm/rev at each surface. The predicted forces F_H^1 are calculated to be in the ratio $229/288 = 0.80$. The values from experiment are $248/336 = 0.74$. For a given axial force, 300 kN, the values of $f^{\frac{1}{2}}$ are in the ratio $0.50/0.22 = 2.27$. This is greater than the predicted value since for a given axial force a similar value of f , should occur at each die-workpiece interface and the ratio should therefore be 2.0. The difference in the ratios is apparently associated with the maximum axial force measured for axial feed/rev ≈ 0 , using two conical dies being some 15% less than the value obtained with a single conical die operating under similar conditions.

The angle of inclination of each platen axis to the workpiece axis in this experiment is $\theta/2$ and therefore $\chi = (\pi - \theta)/2$. The angle, χ , is the parameter which should remain constant when making comparisons between the results from the experiments reported in sections 7.1 and 7.2. Apparently a difference in conditions exists which causes a greater value of *specific rotary forging force* to be obtained for a given value of χ , when using dual conical platens and a flat lower platen. Clearly the frictional conditions and forces at the lower die differ widely and accordingly could account for much of the difference.

The initial investigation shows the advantages and hence industrial significance, of the dual conical platen arrangement which may be summarised as the ability to attain

increased axial feed/rev for a given axial force, and subject to findings from further investigation, a possible increase in the upper limit of *specific rotary forging force* for a given value of χ . This suggests that for a given axial force capacity a machine with the capability to operate with dual conical platens has advantage with respect to workpiece size and *floor to floor* times.

The less marked departure from cylindrical form at very low axial feed/rev is presumably due to the plastically deforming zones meeting. The condition in the deforming zones may be expected to be similar to that described by Johnson & Mellor [60] figure 14.24 and 14.26 which assumes the existence of a perfectly smooth plane which is parallel to the workpiece surface at a distance $H/2$ and located between two opposed indenters of similar form.

No conclusions could be reached regarding process efficiency with the exception that the values obtained were generally similar to those of the experiment reported in section 7.1.

7.3 Initial experiment to study the cold forming of an axisymmetric component by rotary forging using contoured upper and lower dies

(a) Introduction :

Rotary forging is particularly applicable to the manufacture of axisymmetric components, of low height to diameter ratio,

using contoured upper and lower dies. Satisfactory results have been reported by a number of previous workers in, for example, references [13], [16] and [20].

It is therefore necessary to make an initial assessment of the 0.5 MN rotary forging machine which was constructed when used for the above purpose. Assessment was carried out by comparison of dimensions of manufactured components with the appropriate dimensions of the dies. This was based upon the assumption that shortcomings in the machine and tooling would give rise to variation in final form within a batch of workpieces. Since the machine is designed to carry out work which represents use in manufacturing industry, it was considered appropriate to evaluate performance in terms of measured *floor to floor* times.

A subject of interest to manufacturing industry, is the achievement of satisfactory *die filling* at the minimum cost. One of the elements of cost arises from the magnitude of capital investment which in general terms can be related to axial force capacity in the case of rotary forging machines. Other elements of cost are of course labour and energy.

This initial experiment is intended to assess the suitability of the machine for investigation of the relation between axial force, workpiece initial and final geometry of the workpiece, together with the influence of workpiece

material characteristics,

As an initial experiment the work was restricted to attempting to form a satisfactory component which is typical in shape to those produced by previous workers, using contoured upper and lower dies. No study was made of axial feed/rev, rotary drive torque, efficiency, etc.

(b) Objectives of the experiment were to :

- (i) assess the suitability of the machine to conduct experimental work upon forging with contoured upper and lower dies,
- (ii) establish the manufacturing precision applicable to various dimensions of a component formed from a cylindrical billet of commercially pure aluminium,
- (iii) determine the average *floor to floor* time during manufacture of a small batch of components,
- (iv) carry out an initial investigation into the relation between axial force and inclination of platen axes, θ , when forming a component of the shape and material described.

(c) Applicable theory :

No theory was considered applicable with the exception that to predict the axial force required to form the component when $\theta = 4^\circ$ from the value determined by experiment when

$\theta = 2^\circ$ then the process was considered to consist mainly of upsetting and the theory from section 7.1(c) was applied. This gave :

$$(\text{axial force for } \theta = 2^\circ)/(\text{axial force for } \theta = 4^\circ) \approx (4/2)^{\frac{1}{2}}$$

(d) Equipment :

Upper and lower contoured dies together with a lower form tool figure 68(a), manufactured from steel to BS.4659 BH.13 hardened and tempered to 35 Rockwell C scale, designed to form the component as illustrated in figure 68(b). No draft angles were used on the lower die. The operating surfaces of the dies were polished to a surface roughness $\frac{1}{2}$ 0.20 μm C.L.A. Upper dies were manufactured to operate at angles of $\theta = 2^\circ$ and $\theta = 4^\circ$. Billets were of commercially pure aluminium 31.75 mm diameter and 40.00 mm length. For initial location purposes the diameter at one end of the billet was reduced by machining to 30.00 mm for a length of 4.0 mm. The billets were otherwise as parted from bar stock using a lathe with a consequent surface roughness $\frac{1}{2}$ 25 μm C.L.A. on the faces.

A Surtronic Roughness Meter Type 112/1100 S/N 319251 was used to measure surface roughness of the formed components and a Mercer Type 215 Comparator was used, together with other standard metrology equipment to determine the various dimensions. Dow Corning Molykote paste was used as a lubricant.

(e) Procedure :

The contoured upper and lower dies were fitted in position as illustrated in figure 68(a), with the appropriate upper die in position the angle of inclination of the axis was adjusted to 2° .

The speed of platen rotation was adjusted to 180 rev/min and with the flow control adjusted to give an initial feed rate of approximately 2 mm/rev. With a billet in position tests were made to establish the minimum hydraulic pressure required in the system to achieve filling of the die. This axial force associated with the value of pressure determined was found to be 0.28 MN.

The upper axial displacement stops were adjusted to provide the desired overall length of the component. The lower stops were positioned to provide adequate die clearance for unloading the ejected specimen. The hydraulic pressure in the down stroke direction was set to provide a maximum ejection force of 0.10 MN. A workpiece is illustrated in the *ejected* position in figure 43.

A batch of 24 billets was cleaned and then lubricated on all surfaces by applying the paste with a brush. The operating cycle consisted of placing a billet, as illustrated in figure 69(a), in position on the contoured lower die, closing the guard gate, starting the platen rotational drive motor, forging component by raising lower platen to upper stop,

engaging ejection mechanism, operating hydraulic ram to lower stop position, stopping rotational drive, opening guard gate, removing component, disengaging the ejection mechanism and returning the lower die to the loading position. Prior to reloading the upper and lower dies were subjected to an air blast for cleaning purposes. The elapsed time from commencement to completion of the batch of 14 components was measured using a stop watch.

The upper die was then replaced with the die suitable for operating at $\theta = 4^\circ$ and the hydraulic pressure adjusted to give a maximum axial force of 0.20 MN. A further batch of 10 components was then produced using the above procedure.

The formed components were cleaned and degreased prior to measurement of diameters, lengths and surface roughness.

(f) Results :

The dimensions of formed components are stated in Table 22 and typical values of surface roughness measured on various faces indicated in figure 68(b) are also included.

The diameter D_2 , figure 68(b), was formed satisfactorily when $\theta = 2^\circ$. However, when $\theta = 4^\circ$ the diameter was fully formed for a length of approximately 2 mm only from surface referred to as item 7 figure 68(b). This is illustrated in the photograph figure 69(b) and shows the surface with a surface roughness of $15.0 \mu\text{m}$.

Typical *flash* dimensions were approximately 0.5 mm high, 0.5 mm thick for specimens formed at $\theta = 2^\circ$ and 2.0 mm high, 0.3 mm thick for those formed at $\theta = 4^\circ$. Typical sections of specimens are illustrated in the photograph figure 69(c).

For the 24 components the average *floor to floor* time was found to be 28 secs. of which approximately 7 secs. was the average time required to form the metal.

It should also be noted that during the process of establishing the axial forces and the stop position it was found that continued application of the force required for *die filling* without stops caused continuous reverse extrusion of the *flash*.

(g) Discussion :

For conditions appertaining to the experiment, the mean range and standard deviation of the diameters and lengths determined indicate that diameters had a closer relationship to tool dimensions than lengths. The range obtained for diameters was marginally less than those stated by Grzeskowiak [31] when cold forming using orbital motion on the PXW 100 machine.

The extruded dimension, L_3 , is less than the tool dimension and magnitude of the difference increased when θ was increased from 2° to 4° and the axial force was decreased. L_5 was the dimension most influenced by the

change in forging parameters.

The negative values obtained for the tool dimension minus the mean dimension of the finished specimens suggests that recovery from elastic deformation of the workpiece occurs. Presumably in the case of the dimensions L_2 , L_4 , L_5 , L_6 , and L_7 , elastic deformation, and hence elastic recovery, increases as the value of θ , was increased from 2° to 4° . This together with the increase in *flash* height which could be considered as a reverse extrusion phenomenon, suggests that although the axial force had been reduced from 0.28 MN to 0.20 MN, the increase of θ from 2° to 4° was presumably causing higher pressure over the area of contact between the upper die and workpiece. Variation and differences between tool and workpiece dimensions were not consistent with compliance of the machine or the axial force applied. The value of predicted relation between axial forces required for satisfactory *die filling* based upon theories applicable to the upsetting process, results in under estimation of the force required when increasing inclination of the platen axes from 2° to 4° .

When θ is increased from 2° to 4° , and the axial force is reduced from 0.28 MN to 0.20 MN, the shortcomings in *die filling* associated with dimension D_2 only, might be obviated by modulation of θ over the range 2° to 4° during the forging process.

The marked difference between tool surface roughness and

typical surface roughness of the components stated for surface reference No.6, specimens B, Table 22, indicates that the surface roughness of the die wall at this diameter is not determining the surface roughness of the formed component.

The surface roughness values stated in Table 22 also indicate that the surface upon which no rolling is taking place exhibit the lowest values of surface roughness. A shortcoming in the test procedure is identified from study of surface roughness in that the method of cleaning the dies between operations is inadequate. The workshop air blast did not remove metal particles which had been transferred from the workpiece to the upper die. An improved method of die cleaning is required to be interposed in the cycle of operation.

The results of the experiment draw attention to the significance of study of *flash* dimensions. They are of particular interest to manufacturing industry since if removal is necessary then additional costs are incurred.

Not more than 25% of the *floor to floor* time is required for the actual forming of metal. Thus to reduce *floor to floor* times improvements are required in the areas of loading, ejection and unloading. This was apparently the finding of the previous workers at B & S Massey [11] and [12].

A disadvantage of the 0.5 MN machine as designed and constructed is the absence of a clutch in the rotational drive to the platens.

7.4 Compaction of solid and hollow cylindrical powder metal preforms of various height/diameter ratio including methods of both warm and cold post-sintered densification

(a) Introduction :

As stated in Chapter 2, previous workers have reported findings on the application of rotary forging to the processes of both compaction and post-sintered densification of powder metal specimens, using machines with orbital motion. In these machines the compact is stationary relative to the machine axis and the energy to cause rotation of the zone undergoing compaction causing the torque effect referred to by Davies, Evans and Hawkyard [39] is supplied to the upper die only.

The economic reasons for applying powder metallurgy technology in manufacturing engineering remain substantially as stated a decade previous in the references [46] and [47]. Likewise the technical reasons for applying the process appear to remain as requirements to control characteristics such as strength, dimensional accuracy, surface roughness, bearing surface properties and other physical properties such as magnetic. Almost without exception this leads to the necessity to control either or both of the parameters of density and open porosity.

It is therefore necessary to assess the 0.5 MN machine for suitability to conduct research on the powder metal technology processes of both compaction and post-sintered

densification. It was concluded from Chapter 2, section 3, that it is desirable to develop a method of using rotary forging for powder metal compaction which is applicable to the manufacture of hollow cylinders. A method devised by the author was assessed during this experiment.

Complete evaluation of manufactured components would require metallurgical and physical examination. However, the scope of this work is confined to an initial study of average density, open porosity, and dimensional characteristics of specimens manufactured from iron powder metal. The specimen sizes were chosen to permit direct comparison with results from experiments by previous workers.

(b) Objectives :

The objects of the experiment were to :

- (i) assess suitability of the machine to conduct research on the application of rotary forging to the manufacturing processes of both compaction and post-sintered densification of powder metal,
- (ii) investigate the relation between the axial force, average density and open porosity of solid and hollow cylindrical post-sintered iron powder metal compacts,
- (iii) determine the variation in dimensions of sintered hollow cylindrical compacts of various density, internal diameter and height/diameter ratio.

(c) Applicable theory :

Suitable theory applicable to compaction and post-sintered densification of powder metal using rotary forging does not appear to be available from previous work. Theories proposed by Heckel [53] have been demonstrated to be applicable to uniaxial compaction and experiments using closed dies show that a compaction pressure-relative density relation exists which can be expressed in the form

$$\ln\{1/[1 - (\rho_E/\rho_T)]\} = C_1 P + C_2 \quad (75)$$

where ρ_E = density of compact from experiment
 ρ_T = theoretical density
 P = normal pressure at upper die
 workpiece interface
 C_1 and C_2 are constants

The constant C_2 is shown to be given by :

$$C_2 = \ln\{1/[1 - (\rho_0/\rho_T)]\} + C_3 \quad (76)$$

where C_3 = a constant
 ρ_0 = apparent density of the powder metal
 prior to compaction

In [53] it is proposed that the constants can be related to the three stages in powder consolidation described by Johnson [43].

Therefore from equation (75) and (76)

$$\ln\{1/[1 - (\rho_E/\rho_T)]\} = C_1 P + \ln\{1/[1 - (\rho_0/\rho_T)]\} + C_3 \quad (77)$$

In the absence of a suitable applicable theory the author proposes that for initial investigation purposes the axial force applied in rotary forging is considered to be causing

an equivalent pressure, P_e , given by :

$$P_e = F_E/A$$

Where A is the total area of the compact projected in the direction of the applied force. This consideration is applied to the experiment described and is intended to form an initial investigation into methods which could be used for manufacturing engineering purposes to relate the data obtained from uniaxial compaction of powder metal to rotary forging methods.

(d) Equipment :

To carry out compaction and post-sintered densification of solid cylindrical specimens the tooling was arranged as illustrated in figure 70(a) with upper dies available both of 86° and 88° semi-conical angle. The lower workpiece support mandrel has a radial line scribed on the upper surface which if formed satisfactorily on the compact is a witness to the absence of relative rotational motion.

The hollow cylindrical die of 30 mm nominal diameter is provided with mandrels, located radially by appropriate workpiece support sleeves, to provide either 14 mm or 20 mm internal diameters. The tool arrangement is illustrated in figure 70(b) and the photograph figure 71. With the exception of the springs all items of tooling were manufactured from hot forging steel to BS.4659 BH.13 hardened and tempered to 35 Rockwell C scale. Following grinding all working surfaces were polished to surface roughness $\leq 0.20 \mu\text{m C.L.A.}$

A vacuum furnace with inner controlled temperature zone 100 mm diameter, 300 mm long, Carbolite Type 9/79/445 was used for sintering.

Iron powder metal, produced by the atomization process, Hoganas Type ASC.100.29 100 mesh, 2.98 g/cm^3 apparent density, with 0.8% by weight, admixed zinc stearate, was used throughout.

A sealed chamber, 8 litre capacity, and vacuum pump was used for impregnation of sintered compacts with oil, type Shell Voluta 27, density 0.873 g/cm^3 at 20°C .

Griffin & Tatlocks 250g chain dial balance No.830303 and Oertling single pan balance type R.20 No.18614, were used to determine mass in air and apparent mass in water. Tinned copper wire for specimen support 0.075 mm diameter.

Lubrication pastes used for post-sintered densification.

Dow Corning, Molykote: 0°C to 250°C .

Dow Corning, High temperature Lubricant: Type D -30°C to 1100°C .

A chromel/alumel thermocouple probe was used for specimen temperature measurement.

(e) Procedures :

With the appropriate tools in position the inclination of platen axes was adjusted to 2° and the speed of rotation of the platens set at 120 rev/min. The hydraulic pressure

in the system was adjusted to produce the desired value of maximum axial force and the flow control adjusted to give an initial off-load axial feed/rev ≈ 1.0 mm/rev. A quantity of powder metal determined by weight was placed in the lower die. Rotation was commenced and by subsequent axial displacement the compaction process was permitted to proceed until the axial feed/rev was observed to be ≈ 0 . The total number of revolutions required was dependent upon both axial force and the mass of powder metal. Typically, the number ranged from 15 to 40 revolutions.

The lower platen was retracted and the *green* compact was ejected from the lower die. The dies were cleaned by an air blast and the procedure was repeated for 46 specimens of various mass and with various values of axial force. No lubricant was used other than the zinc stearate admixed with the powder metal. The specimens were weighed prior to sintering at 1035°C for 40 minutes in the vacuum furnace at a pressure $\frac{1}{2}$ 2 mb. and then permitted to cool with vacuum applied over a period of at least 12 hours. The procedure was repeated with appropriate tools to manufacture 25 hollow cylindrical specimens.

The density and open porosity of sintered specimens were then determined in accordance with International Standard No. ISO.2738 1973 [70]. This standard is approved by representation from countries including U.S.A., West Germany, Japan and Sweden. The standard is not considered

acceptable by the representatives of the United Kingdom, and the specification is endorsed accordingly. The procedure, which is known to be in current use by workers in manufacturing industry within the United Kingdom, was adopted in the absence of a practicable alternative.

The average density determined from experiment, ρ_E , in accordance with ISO.2738, is given by :

$$\rho_E = m_1 \rho_W / (m_2 - m_3) \quad (77)$$

where m_1 = mass of the dry compact in air
 m_2 = mass of the oil impregnated compact including support wire
 m_3 = apparent mass of the impregnated compact including the support wire when immersed in water
 ρ_W = density of water assumed to be 0.997 g/cm³

The open porosity, p , expressed as a percentage by volume in accordance with ISO 2738 is given by :

$$p = (m_2 - m_1) \rho_W / (m_2 - m_3) \rho_i \quad (78)$$

where ρ_i = density of the impregnating oil

The relative average density = ρ_E / ρ_T , where for this experiment ρ_T = theoretical density of iron and assumed to be 7.873 g/cm³. The oil impregnation procedures using vacuum were also in accordance with ISO 2738. A typical mass in air of support wire used during the experiment was 0.005 g.

The procedure for *warm* and *cold* post-sintered densification using closed dies was similar to that used for compaction described above with the exception that the lower die was loaded with a sintered preform and the semi-conical angle of the upper die was 86° with θ adjusted to 4° . The specimens were loaded in the die with the upper and lower surfaces in the same relative position as that during compaction. The procedure was adopted to ensure uniformity and the most disadvantageous conditions in effecting increase in relative average density. For cold forging the lubricant was as described in 7.1(d)(iv) and applied to the cold compact. For warm forging the sintered compact was heated approximately to 800°C and the dies were lubricated with the paste suitable for a temperature of 1000°C . Forging commenced with the specimens as close to 550°C as practicable. To reduce heat loss from the specimen in the cases where *warm* procedures were applied the upper and lower platens were heated to $+300^\circ\text{C}$ prior to forging.

With the exception of lubricants, the procedures for the attempt to cause post-sintered densification by *warm* and *cold* upsetting were similar to the procedures described in section 7.1(d), using an upper die of 86° semi-conical angle for all tests.

(e) Results :

The result of the initial tests to prove the tooling indicated that severe cracking and in some cases segregation

of the compact occurred during ejection. The photograph figure 72(a) illustrates a typical specimen. This was presumed to be due to the *spring back* phenomenon referred to by the powder manufacturer Reference [72], taking place as the compact passed through the square cornered die exit and causing steep pressure gradients to occur within the compact. To obviate this the parallel internal bore of the lower die was modified to provide a conical exit. The bore of the die was tapered uniformly over a length of 10mm from 30.00 mm to 30.75 mm at the die exit.

Following the modification it was discovered that by increasing the axial force to a given level, cracking or segregation was caused in a zone adjacent to the upper surface, as illustrated in figure 72(b), thus limiting the average density which could be achieved. It was also found that by permitting the process to proceed the segment above the surface at which segregation occurred appeared to remain constant in shape in the form of a dead metal zone and a reverse extrusion of compacted powder could be observed to take place in what appeared to be a continuous manner. The surface of a typical crater formed is of such shape that relative rotary motion is inconceivable. Detection of the onset is difficult by external visual examination of the *green* compact but can however be observed readily in the post-sintered state. The phenomenon is also applicable to hollow cylinders as illustrated in figure 72(c).

Typical satisfactory solid and hollow cylindrical specimens are illustrated in the photograph, figure 73(a). The line which is witness to no rotational displacement between the compact and the lower die is illustrated in figure 73(b). Figure 73(c) shows satisfactory solid cylindrical specimens. Results from tests, including the mass loss during sintering solid cylindrical specimens, are stated in Table 23. The axial force - relative density relation is presented in figure 74 and shows that due to scatter in the results it was practicable to draw curves for specimens of H/D values 0.8, 0.6 and 0.4 only. For specimens of $H/D < 0.3$ the points are plotted for information. A similar comment regarding scatter is also applicable to the post-sintered relative density $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ - equivalent pressure and percentage open porosity - axial force relations presented in figures 75 and 76 respectively.

The results from compaction of hollow cylindrical specimens are stated in Table 24 and the dimensions of external and internal diameters, together with typical values of surface roughness are presented in Table 25. The relative average density - axial force relation and percentage open porosity - axial force relation are presented in figures 77 and 78. The $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ equivalent pressure relation is shown in figure 79. The point shown joined by the dash line requires investigation which is not within the scope of this work.

The results of tests to cause post-sintered densification by

warm and *cold* rotary forging in closed dies at a constant axial force, are stated in Table 26. The relation between initial and final density are presented in figure 80.

Twelve solid cylindrical specimens were manufactured each of 150 g mass, 30.00 mm diameter and of height to diameter ratio ≈ 0.75 with post-sintered average relative densities ranging from 7.05 to 7.24. Nine of the specimens were then machined to a height of 20 mm with a location spigot on one face of dimension 12.95 mm diameter and 1 mm high. Figure 81(a) illustrates the cracking and departure from cylindrical form which occurred when a 0.15 fractional reduction in height was caused by cold upsetting with a maximum axial force of 100 kN.

Figure 81(b) illustrates a similar result from solid cylindrical specimens when fractional reductions in height $R \approx 0.50$ were caused by *warm* upsetting using a maximum axial force of 75 kN. The condition of the specimens was such that the method of density and open porosity measurement used in this experiment would be inappropriate.

Hollow cylindrical specimens Nos. 4 and 13, Table 24, were subjected to *cold* upsetting to cause a fractional reduction in height of approximately 0.5. The relative average densities were found to have increased from 0.87 to 0.97 and from 0.87 to 0.96 respectively. The values of open porosity decreased from 6.5% to 2% and from 9.0% to 2%

respectively. Two hollow cylindrical specimens numbers 7 and 11, Table 24 were subjected to fractional reductions in height of approximately 0.5 by *warm* upsetting and the relative average densities were found to have increased from 0.92 to 0.97 and from 0.90 to 0.96. The values of open porosity reduced from 6% to 2% and from 11% to 2% respectively. The photographs of figures 82(a) and 82(b) illustrate that cracking at the upper surface is not apparent following either *warm* and *cold* upsetting of hollow cylinders.

Three specimens were each machined to form four discs 30.03 mm diameter and 4.3 mm thick. The density of each disc was measured and the results are stated in Table 27. The four discs from one specimen were each trepanned to provide a further disc of 12.25 diameter leaving annular rings of 30.03 external diameter and 16.90 mm internal diameter. The average density of the rings and discs were measured and the results are shown in Table 27.

(f) Discussion :

From the phenomenon described in section 7.4(e) which occurs in a zone adjacent to the upper die-compact interface, together with the trends of the curves given in figure 74 and 75, it is apparent that, for the particular powder metal with lubricant, and forging parameters applied, it is not practicable to exceed a post-sintered relative average density ≈ 0.95 . Investigation of the phenomenon which appears to form a limitation to the relative average density which can be achieved is beyond the scope of this

work and is clearly a subject for further investigation.

The onset of the phenomenon which limits *green* density is applicable to both solid and hollow cylindrical forms and is presumably indicated by both relative average density - axial force relation and percentage open porosity - axial force relation as illustrated in the figures 74 and 76 respectively. The onset is also apparently indicated by the departure from linearity of the equivalent pressure - $\frac{1}{1 - (\rho_E/\rho_T)}$ relation.

It is not clear from previous relevant published work using orbital type machines whether or not the segregation phenomenon described which is limiting the *green* density is related to the configuration or performance particular to the 0.5 MN machine. This may be the reason for the use of a disc interposed between the upper die and powder metal described in figures 15 and 16, reference [37]. However, it should be noted that the average densities achieved are much lower than those achieved in the experiment described here. The mass loss during sintering indicates the total admixed lubricant is diffused during the process.

Comparison of results with data published by the powder metal manufacturer [72] for uniaxial compression confirm the statement by Johnson and Hawkyard [73] that approximately a tenth of the axial force is required when using rotary forging. The relative average sintered density which can be achieved at a single *pressing* is also equalled.

Table 24 and figures 77 and 78 indicate that, subject to metallurgical examination of the specimens, the method of compacting hollow cylindrical specimens is generally satisfactory. The method of removing the mandrel is satisfactory for research purposes but would be unsatisfactory for high production rates. The machine configuration permits an ejector pin to be arranged to operate through the axis of the upper die in the $\theta = 0^\circ$ position and could be synchronised with the lower platen ejection cycle. The hollow upper platen spindle is therefore readily adaptable to a design suitable for fast operating times. It is of interest to note the similarity of equivalent pressure - $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ relation which exists for solid and hollow specimens when using powder metal from the same batch. There is also a similarity between percentage open porosity results of hollow and solid cylindrical specimens when comparisons of average relative density are made for given values of H/D ratio.

The tolerances determined on bore and external diameter are in the range IT.8 to IT.10 and surface roughness $\pm 1.0 \mu\text{m}$ are similar to those stated by Huppmann and Hirschevel [44] for conventional methods of compaction.

The results from post-sintered densification at a force of $F_E = 75 \text{ kN}$ (equivalent pressure F_E/A of 100 kN/mm^2) show the manner in which the average density may be increased and open porosity reduced using axial forces which in some cases were less than those used to achieve compaction of the *green*

compact with an increase from 2° to 4° of θ . This is also presumably due to the force being applied over a smaller area due to the modification of material characteristics following the sintering process. The large *flash* created at the upper surface illustrated in figure 3 of reference [40] did not occur in this experiment. It would be of interest to determine the effect of inverting the position of the specimen when loaded into the lower die such that the lowest density zone is adjacent to the upper die.

The *warm* post-sintered densification produces higher values of average relative density for a given axial force at a given value of H/D . The difficulties in controlling the experiment due to variation of heat loss of the workpiece, and hence differing and non-uniform yield stress, are presumably the reason for much of the scatter which occurred in the results. This also indicates the problem of controlling such a process in manufacturing industry. Significant improvement in procedures are required for any future experiments.

The relation between the post-sintered relative average density ρ_i/ρ_T , and that obtained following post-sintered densification ρ_f/ρ_T , is based upon the method proposed by Heckel [53] and the experiment was conducted with equivalent pressure, P_e , at a constant value. All other material characteristics are assumed to be those applicable to the third or final stage of compaction which consists mainly of plastic deformation. Therefore when studying

the results consideration should be given to the strain hardening which may be presumed to be occurring in zones adjacent to the surface which form porosity. The forging parameters are sensibly uniform therefore C_1 , C_2 and C_3 from equation (77) may be considered to have values applicable to a given H/D ratio. From figure 75, for values $H/D \approx 0.60$ and 0.80 a degree of correlation with the theory is evident. It can also be concluded that subject to improvement in the design of the experiment further work would be of value.

The results from the sliced specimens Table 27 confirm the existence of density variation similar to that reported by previous workers and that the density is lowest in the zone most remote from the upper surface. The result from the single specimen cut to annular rings and discs does not exhibit the radial variations in density shown by previous workers. The results from the measurement of average density of the discs also served to show the practical problems of density measurement on specimens of the size of the smallest discs. It also indicates that future study of density variations using the method of measurement described in section 7.3(c) should be conducted with specimens of larger size. This comment is also applicable to the tests on *cold* and *warm* post-sintered densification where, for example, with the existing machine capacity specimens could be increased in area by say a factor of 4. This comment may also be applicable to

the scatter in the results of height to diameter ratio less than ≈ 0.3 as mentioned above. From this it is also evident that a subject for further research should be a machine configuration and tooling which would permit conical dies to be applied simultaneously to the upper and lower surfaces of a workpiece constrained in a hollow die.

The cracking which occurred in specimens during attempts to cause post-sintered deformation by upsetting is in accordance with the findings of Kuhn and Downey [58] when applying uniaxial compression. Cracking and fracture during upsetting was also found by workers in Japan as reported by Standring and Appleton [37]. The findings on the *cold* and *warm* upsetting of hollow cylinders are of interest since they do not appear to crack at the low values of fractional reduction in height as indicated by the solid cylinders and both post-sintered densification and reduction in open porosity can be achieved.

Kuhn and Downey [73] in their work on forming hollow powder metal preforms using uniaxial compression made similar findings. Further investigation is beyond the scope of this work but is clearly a subject for future study.

From this investigation one of the differences between characteristics of compaction by uniaxial compression and compaction by rotary forging can be identified from plasticity theory consideration.

For example, from the axisymmetric nature of uniaxial compression in a cylindrical closed die, the vectors of deviatoric stresses and strains during compaction must lie in planes which are radial and parallel to the axis. This is not necessarily so when applying rotary forging because the respective vectors would include a tangential component. This may develop material characteristics which could possibly be an advantage that would be additional to achievement of higher relative average densities and a reduction in axial force required.

There is also an indication that when using rotary forging, less or no lubricant is required, other than that necessary to lubricate the die walls. This would provide further advantages in terms of increased *green* strength, increased pre-sintered and post-sintered relative average density together with reduced problems in the sintering process.

It is also interesting to consider investigation of the manner in which reversal of the direction of rotation would influence the processes of compaction and post-sintered densification using upsetting or closed dies, bearing in mind the twist phenomenon [34], tangential strain, and strain hardening which presumably takes place.

Further investigation was not within the scope of the work reported in this thesis. However, the tests did indicate that the machine and tooling are suitable for conducting research on compaction and post-sintered densification which could include a study of an arrangement to apply to dual conical die configurations.

8.1 Part 1 Design and construction of a 0.5 MN rotary forging machine

Conclusions from work on the design and construction of the 0.5 MN rotary forging machine are as follows :

- (a) The machine as designed and constructed conforms to the specification with advantages of a correct kinematic arrangement, inherent dynamic balance and a capability to perform rotary forging with dual conical platens.
- (b) The range of proven proprietary components offered would permit machines from approximately one tenth to ten times the capacity to be constructed without resort to investment for special purpose manufacture or expenditure on development.
- (c) Performance within the capacity of the machine stated is satisfactory for conducting experiments which relate directly to industrial applications of the rotary forging process.
- (d) The design, in both total concept and certain detail features which include minimising the influence of bearing compliance, forms a suitable basis for a range of machines to be constructed at relatively low cost for use in industry.
- (e) Features which include a method of workpiece ejection form the basis of claims in the relevant

applications are fundamental to achieving the machine facilities, operating capacity and performance, within the financial constraints of this project.

8.2 Part 2 Experimental rotary forging

Conclusions from experiments conducted using the machine are as follows :

- (a) Upsetting specimens of initial height to diameter ratio $H_0/D_0 \approx 0.2$ using a conical upper platen :
 - (i) the machine is satisfactory for conducting experiments on the upsetting process,
 - (ii) results obtained are similar to those obtained by previous workers who have used specimens similar in both dimensions and material but utilised a machine based upon the orbital concept,
 - (iii) the method of predicting axial force proposed by Hawkyard et al [24] was confirmed to be satisfactory for most manufacturing engineering purposes,
 - (iv) values of axial force predicted by the method presented by Kubo et al [34], assuming a constraint factor of unity, were found to be substantially less than those obtained from experiment and the magnitude of the difference between predicted and experimental

values tends to increase as the *specific rotary forging force* increases,

- (v) the ratio of the axial force obtained from experiment to those obtained from prediction show the method proposed by the author to give a mean value marginally closer to unity and a reduced value of standard deviation when compared with the method proposed by Hawkyard et al [24],
- (vi) the methods of predicting axial force used in this experiment cannot be used to predict the upper and lower limits of *specific rotary forging force*,
- (vii) the magnitude of the predicted value of axial force used to make an allowance for elastic deformation of the upper die is apparently always less than the value obtained from experiment and the magnitude of the difference appears to increase as both the axial force per unit radius and the workpiece yield stress increases,
- (viii) with specific reference to prediction of axial force the process does not appear to be influenced by rotational speed within the range tested,
- (ix) efficiency apparently increases as axial feed/rev increases,

(x) the rotational torque required is approximately proportional to axial force.

(b) Upsetting specimens of $H_0/D_0 \approx 0.2$ using conical upper and lower platens of equal conical angle :

(i) the machine is suitable for operating with upper and lower conical platens on workpieces ranging from initial diameter of 37.5 mm to 76.2 mm of H_0/D_0 ratio ≈ 0.2 , platens of conical angle 178° and 176° with values of $\theta = 4^\circ$ and 8° respectively,

(ii) for the range of specimens used in the experiment and by considering the process as two simultaneous upsetting processes with platen axes inclined at $\theta/2$ and half the total axial feed/rev, $f/2$, predictions of axial force may be made which would be satisfactory for most manufacturing engineering purposes,

(iii) the dimensions of the workpiece location recess at the apex of the lower conical platen were found to be satisfactory,

(iv) for given values of axial force and semi-conical angle of the die, a pronounced increase in axial feed/rev can be achieved by using dual conical dies,

(v) the values obtained for process efficiency and *specific rotary forging force* using dual conical dies is generally similar to that obtained using a single

conical dies of the same semi-angle. However, when using dual dies the upper limit of the *specific rotary forging force* may be marginally increased.

(c) The manufacture of a component to conform to the drawing in figure 68, using contoured upper and lower dies from a billet of $H_0/D_0 \cong 2.0$

(i) the machine is satisfactory for the manufacture of components to the requirements on the drawing from raw material specified in cylindrical billet form,

(ii) diameters can be controlled to greater precision than axial lengths,

(iii) filling of the lower die with specific reference to the diameter D_2 as shown in figure 68(b) is apparently influenced by the values of angle θ and the axial force, F_E , used in the experiment whilst the diameters D_1 , and D_3 appear not to be influenced,

(iv) the *flash* or reverse extrusion phenomenon is apparently influenced by the values of θ and F_E used in the experiment,

(v) forging time is less than a quarter of the *floor to floor* time hence development of mechanical methods for workpiece transfer, together with lower platen

rotational braking and clutch operation for rotational start-stop, are required to effect a significant increase in production rate,

- (vi) the lubricating paste used provides a satisfactory lubricant but is problematic when attempting to maintain cleanliness of the upper and lower die surfaces due to the tendency to retain debris against a cleaning process such as air blasting.
- (d) Use of the machine for compaction and post-sintered densification of both solid and hollow cylindrical powder metal preforms :
 - (i) the machine and tooling are satisfactory for compaction of iron powder to produce both solid and hollow cylindrical preforms of 30 mm diameter and $H_0/D_0 \cong 0.25$ to 1.0,
 - (ii) for an upper die of semi-conical angle 88° , the relative average density increases with axial force until the cracking or segregation phenomena occur at a zone adjacent to the upper die workpiece interface,
 - (iii) the method devised to produce hollow cylindrical compacts is satisfactory but requires innovation with respect to a rapid method of removing the mandrel as part of the ejection operation,

- (iv) the relative average density of a solid cylindrical compact decreases with distance from the upper die-workpiece interface,
- (v) for a given axial force and conical die of semi-angle 88° the average density of the compact increases as the height to diameter ratio H_0/D_0 decreases,
- (vi) post-sintered densification by *cold* and *warm* forging into a closed die can be achieved and the methods of lubrication were satisfactory,
- (vii) the dimensional accuracy of post-sintered hollow cylindrical preforms is similar to that achieved when using uniaxial methods of compaction,
- (viii) *cold* and *warm* post-sintered densification by upsetting solid cylindrical specimens was found to be impracticable due to radial cracking on the upper surface,
- (ix) *cold* and *warm* post-sintered densification by upsetting hollow cylindrical specimens is apparently practicable.

8.3 Suggestions for further work

- (a) Improvements to the machine :

- (i) modify the hydraulic system to provide control of axial feed/rev using a translational velocity transducer and appropriate electro-hydraulic control to obviate the performance limitations of pressure compensated flow control valves,
 - (ii) provide a clutch and braking system preferably electrically operated, in the rotational drive and verify that the design concept of the working zone within the machine is compatible with proprietary mechanical aids appropriate to workpiece feed and removal,
 - (iii) design and manufacture a power operated system to modulate the angle, θ , during the forging process,
 - (iv) design and manufacture upper and lower contoured dies suitable for use with the dual conical platen arrangement,
 - (v) design and manufacture tooling to provide mandrel ejection following compaction of hollow cylindrical preforms,
 - (vi) study the influence of machine compliance on the finished workpiece for future design purposes.
- (b) Further work arising from the experiments :

- (i) examine methods of predicting axial force applicable to the upsetting process over a range of materials of widely varying yield stress and diameters with $H_0/D_0 \cong 0.01$ to 1.0 ,
- (ii) establish a method of predicting the upper and lower limits of *specific rotary forging force* applicable to the upsetting process,
- (iii) investigate the relationship between torque and process efficiency during upsetting,
- (iv) investigate the parameters which influence die filling and optimising .. of billet size when forging into closed dies using upper and lower contoured platens,
- (v) identify the phenomenon which limits density of powder metal compaction when using a conical upper die,
- (vi) determine the relation between relative average density, equivalent pressure and post-sintered densification of powder metal specimens when forging into closed dies,
- (vii) study a design to use conical upper and lower dies when forging into a hollow die during the powder metal compaction and post-sintered densification process,

(viii) conduct experiments upon compaction with no admixed lubricant in the powder metal and lubricate the die walls only,

(ix) study the effect of reversing the direction of rotation of the deforming region relative to the axis of the workpiece.

REFERENCES

- 1 SLATER, R.A.C.
BAROOAH, N.K.
APPLETON, E. and
JOHNSON, W. The rotary forging concept and initial
work with an experimental machine.
Proc. Instn.Mech.Engrs. 184 Part 1.
(1969-70)
- 2 SLICK, E.E. United States Patent Office
Specification Number 965036. Method
of, and apparatus for, forging metal.
(1910)
- 3 SLICK, E.E. British Patent Specification Number
16039. Improvements in or relating
to car wheels. (1910)
- 4 BRINDLEY, H.S.B. Deutsches Reich Patentschrift Number
371749, Walzanlago zur Herstellung von
Radern fur Eisenbahnwagon und andere
schwere Fahrzeuge. (1920)
- 5 MASSEY, H.F. British Patent Specification Number
319065. Improvements in upsetting and
forging machines. (1929)
- 6 NATIONAL ASSOCIATION OF
DROP FORGERS & STAMPERS Annual Report (1978). Published by
N.A.D.F.S. Birmingham.
- 7 NATIONAL ASSOCIATION OF
DROP FORGERS & STAMPERS Annual Report (1979). Published by
N.A.D.F.S. Birmingham.
- 8 ANON The Slick wheel mill. The Iron Age
Vol. 102 No.9 page 491. Published
New York. August (1918)
- 9 HUGHES, J.W. United States Patent Office Number
1855499. Method and machine for
working metal. (1932)
- 10 B & S MASSEY LIMITED British Patent Specification Numbers
1224259 and 1224260. Improvements in
upsetting and forging machines. (1969)
- 11 NUNNINGTON, F.T.
WILSON, G. Recent developments in Rotaform orbital
forging, B & S Massey Ltd., Openshaw,
Manchester. (1979)
- 12 NUNNINGTON, F.T. Rotaform orbital forging, 1st
International conference. Rotary
metal working processes, London.
November (1979)
- 13 NUNNINGTON, F.T. Rotaform operating experience. 1st
International conference on rotary
metal working processes, London (1979)

- 14 ULRYCH, O. United States Patent Office, Number 3611771. Method for rolling disks and a disk rolling mill for the practice of the method. (1971)
- 15 WILLIAMS, V. Breakthrough in cold forming. The Magazine of Manufacturing Production. August (1974)
- 16 MAICKI, J.R. Orbital forging. Presented at the annual meeting of the Drop Forging Research Association, Leicester. May (1977) and published in Metallurgia and Metal Forming. June (1977)
- 17 DREDGER, D.R. Noiseless cold forming. Machine Design. January (1974)
- 18 BAROOAH, N.K. An experimental rotary forging machine. M.Sc. dissertation, Department of Mechanical Engineering, University of Manchester Institute of Science and Technology. May (1967)
- 19 SLATER, R.A.C. and APPLETON, E. Some experiments with model materials to simulate the rotary forging of hot steels. Advances in Machine Tool Design and Research. Pergamon Press (1971)
- 20 APPLETON, E. and SLATER, R.A.C. Effects of upper platen configuration in the rotary forging process and rotary forging into a contoured lower platen. Proceedings of the Thirteenth International Machine Tool Design and Research Conference. Pergamon Press (1973)
- 21 APPLETON, E. and SLATER, R.A.C. The analysis of rotary forging investigations as an aid to design. Proceedings of the Twelfth International Machine Tool Design and Research Conference. Edited by F. Koenigsberger and S.A. Tobias. Macmillan Press Ltd. (1972)
- 22 HAWKYARD, J.B. GURNANI, C.K.S. and JOHNSON, W. Pressure distribution measurements in rotary forging. Journal Mechanical Engineering Science Vol.19 No.4 135-142. August (1977)
- 23 DIMITROV, D. and HAWKYARD, J.B. Investigation of yield in rotary forging by the indentation of thin discs. Proceedings of the Eighteenth Machine Tool Design and Research Conference, London. (1977)

- 24 HAWKYARD, J.B.
YUNUS, N.M. and
GURNANI, C.K.S. Prediction of force in the rotary forging of short circular cylinders. 1st International Conference on Rotary Metal-Working Processes. London. Nov.(1979)
- 25 JOHNSON, W. Continuum mechanics and deformation processing. Proceedings of 21st Sagamore Army Research Conference, U.S.A. Plenum Press August (1974)
- 26 STANDRING, P.M. and
APPLETON, E. The development of rotary forging machine design. The Institution of Mechanical Engineers Vol.26 No.4 April (1979)
- 27 STANDRING, P.M. and
APPLETON, E. The kinematic relationship between angled die and workpiece in rotary forging. 1st International Conference on Rotary Metal-Working Processes. London November (1979)
- 28 POLITECHNIKA, WARSAW British Patent application Number 1205171 (1968)
- 29 MARCINIAK, Z. A rocking die technique for cold forming operations. Machinery and Production Engineering. November (1970)
- 30 MARCINIAK, Z. Cold forming of wedge-shaped workpieces. 1st International Conference on Rotary Metal-Working Processes, London.(1979)
- 31 GRZESKOWIAK, J. Possibility on cold and warm rotary forging of gears. 1st International Conference on Rotary Metal-Working Processes, London. (1979)
- 32 KUBO, K. HIRAI, Y.
OGISO, S. and
ITO, S. Preliminary work on rotary forging with an experimental press. Journal of Japan Society for Tech. of Plasticity Vol.14 No.151. August (1973)
- 33 KUBO, K. and
HIRAI, Y. Accuracy and defects of rotary forged parts. Journal of Japan Society for Tech. of Plasticity Vol. 17 No.187 August (1976)
- 34 KUBO, K. and
HIRAI, Y. Deformation characteristics of cylindrical billet in upsetting by a rotary forging machine. Proceedings of 1st International Conference on Rotary Metal-Working Processes, London. November (1979)
- 35 KOBAYASHI, M.
NAKANCE, T.
KAMADA, A. and
NAKAMURA, K. Deformation behaviour in simultaneous extrusion-upsetting by rotary forging. Proceedings of 1st International Conference on Rotary Metal-Working Processes. London November (1979)

- 36 STANDRING, P.M. and APPLETON, E. Rotary forging developments in Japan Part 1. Machine Development and Forging Research Journal of Mechanical Working Technology Vol. 3, (1980)
- 37 STANDRING, P.M. and APPLETON, E. Rotary forging developments in Japan Part II. Theoretical investigations, powder metal compaction and sinter-forging. Journal of Mechanical Working Technology Vol.4 No.1. (1980)
- 38 LITTLE, R.E. and BEYER, R. Load-deformation relationships during upsetting by rotary forging. Proceedings of 1st International Conference on Rotary Metal-Working Processes, London. November (1979)
- 39 EVANS, D. HAWKYARD, J. and DAVIES, T.J. The rotary forging of steel powder. Powder Metallurgy International. Vol.9 No.4. (1977)
- 40 KOTSCHY, J. and MARCINIAK, Z. Cold forming of P/M preforms by the rocking-die method. The International Journal of Powder Technology Vol.9 No.4. October (1973)
- 41 KOTSCHY, J. MARCINIAK, Z. and SOBCZYK, M. Cold forming of P/M preforms by the rocking-die method. The International Journal of Powder Metallurgy and Technology, Vol.12 No.1 January (1976)
- 42 OGRODNIK, J. and KAPPINSKI, T. Cold forming of sintered materials with self-aligning dies. Powder Metallurgy International Vol.9 No.3 (1977)
- 43 JOHNSON, W. Application processes involving high strain rates. Inst. Phys. Conf. Ser. No.47 Chapter 4. (1979)
- 44 HUPPMANN, W.J. and HIRSCHVOGEL, M. Powder forging. International Metals Reviews No.5. (1978)
- 45 HIRSCHORN, J.S. and BARGAINNIER, R.B. The forging of powder metallurgy preforms Second inter-American Conference on Materials Technology, Mexico City. August (1970)
- 46 JONES, P.K. The technical and economic advantages of powder-forged products. Powder Metallurgy. Vol.13 No.86 (1970)
- 47 ANDERSON, D.S. and WINQUIST, L.A. The economics of powder metallurgy. International Journal of Powder Metallurgy. Vol.9 No.1 (1973)
- 48 MARSHALL, P.R. The production of powder metallurgy parts. Metallurgical Review No.123 (1968)

- 49 BROWN, G.T. Development of alloy systems for powder forging. Metal Technology (1976)
- 50 BROWN, G.T. Alloy systems for powder forging. Metallurgia and Metal Forming March (1976)
- 51 BROWN, G.T. High duty cost conscious parts by powder metallurgy techniques. Chartered Mechanical Engineer Vol.27 No.7 July (1980)
- 52 HECKEL, R.W. The determination of density - pressure relationships. Transactions of the Metallurgical Society of A.I.M.E. No.221 (1961)
- 53 HECKEL, R.W. A new approach to the study of powder compaction. Progress in Powder Metallurgy Vol.17 (1971)
- 54 ALEXANDER, J.M. and QUANTON, D.R. The isostatic compaction and hydrostatic extrusion of iron powder. 12th International Machine Tool Design and Research Conference. Manchester Edited by Koenigsherger and Tobias (1971)
- 55 AKETA, Y
TANAKA, Y. and
TSUNA, H. Studies on compacting metal powder Technology Report No.15 Osaka University (1965)
- 56 KUHN, H.H. and DOWNEY, C.L. Deformation characteristics and plasticity theory of sintered powder metals. International Journal of Powder Metallurgy Vol.7 No.1 (1971)
- 57 HIRSCHORN, J.S. and GAREY, M.W. Plastic deformation of compacted iron powders. International Journal of Powder Metallurgy Vol.5 No.1 (1969)
- 58 KUHN, H.A. and DOWNEY, C.L. How flow and fracture affect design of preforms for powder forging. International Journal of Powder Metallurgy Vol.10 No.4 (1974)
- 59 PRABHU, V.
ARAVIND, B.M. and
JAGARAMAN, G. Orbital riveting attachment. Proceedings of 1st International Conference on Rotary Metal-Working Processes. London November (1979)
- 60 JOHNSON, W. and MELLOR, P.B. Engineering Plasticity. Van Nostrand Reinhold, Chapters 11 and 14. (1973)
- 61 SLATER, R.A.C. Engineering Plasticity - Theory and application to metal forming processes. Macmillan Press. Chapter 7. (1977)

- 62 PENNY, W.A. and
SLATER, R.A.C. Rotary forging machine. United
Kingdom Patent Application No.7903561
Pub. No. 2041268 Feb.(1979)
- 63 PENNY, W.A. and
SLATER, R.A.C. Rotary forging machine. European
Patent Application No.80300290.6 Pub.
No.0014570 Jan,(1980)
- 64 PENNY, W.A. and
SLATER, R.A.C. Rotary forging machine. Japan Patent
Application No. 10835/80 Pub. No.
130353/80. Jan.(1980)
- 65 PENNY, W.A. and
SLATER, R.A.C. Rotary forging machine. Poland
Patent Application No. P221742
Feb.(1980)
- 66 PENNY, W.A. and
SLATER, R.A.C. Rotary forging machine. U.S.A. Patent
Application No. 116542
Jan.(1980)
- 67 HILL, R. The mathematical theory of plasticity.
Oxford University Press Chapter 9 (1950)
- 68 STANDRING, P.M.
MOON, J.R. and
APPLETON, E. Plastic deformation produced during the
indentation phase of rotary forging.
Metals Technology April (1980)
- 69 KUHN, H.A.
LEE, P.W. and
ERTURK, T. A fracture criterion for cold forming.
Journal of Engineering Materials and
Technology. Oct. (1973)
- 70 SAMANTA, S.K. Effect of friction and specimen
geometry on the ductile fracture in
upset forging. Transactions A.S.M.E.
Jan.(1975)
- 71 INTERNATIONAL STANDARDS
ORGANISATION Permeable sintered metal materials
determination of density and open
porosity. I.S.O. No.2738(E). (1973)
- 72 HOGANAS A.B. Hoganas iron powder. ASC.100.29
Hoganas A.B. Sweden (1980)
- 73 JOHNSON, W. and
HAWKYARD, J. Recent developments in ring rolling
and rotary forging. Metallurgia
and Metal Forming. Jan. (1976)
- 74 DOWNEY, C.L. and
KUHN, H.A. Applications of a forming limit design
concept to powder preforms for
forging. Journal of Engineering
Materials and Technology. April (1975)

APPENDIX 1 A TENTATIVE METHOD OF PREDICTING THE TORQUE REQUIRED TO
CAUSE ROTATION OF THE UPPER DIE RELATIVE TO THE WORKPIECE
DURING ROTARY FORGING OF SPECIMENS OF $H_0/D_0 \cong 0.2$

It is proposed by the author that the instantaneous hinge concept described in Chapter 7, section 1(c), may be used to provide a method of predicting the torque required to cause rotation of the upper die relative to the workpiece during the upsetting process.

Figure 83, illustrates an element in a plastically deforming zone of radius, r_e , width dr_e , and circumferential length of arc, s . The generator of the conical upper platen, OA, is perpendicular to the workpiece axis, Z_1 , and since the angle of inclination of the platen axes is assumed to be $\leq 10^\circ$ then, for the purpose of the reasoning which follows, the axes Z_1 and Z_2 are considered to be coincident. The generator OA is assumed to be in contact with the workpiece in a plane where plastic deformation due to rolling is complete for the given point in the cycle of upper platen revolution.

The pressure, P , at the upper die-workpiece contact area is assumed to be uniform and when acting upon the element will cause a resultant force F_{Z_e} to act at the point, B, in a direction parallel to the axis, Z_1 . The point, B, is at the centre of area of the element at a distance approximately $s/2$ from the instantaneous hinge which is considered to exist at OA.

A rigid cranked lever is assumed to exist between the points B, D and C, such that a tangential force F_{r_e} , acting at the point, C, causes a force to act at the point, B, due to the moment about the hinge OA.

The point, C, at which the force F_{r_e} , is assumed to be acting is at a distance, λ , from the point, D, on the hinge OA and at a distance r_e , from the axis Z_1 .

At the point, C, an instantaneous, frictionless pivot is considered to exist between the end of the lever of length, λ , and the radial line O'C.

The distance CD is considered to be the radius, λ , of the upper roll of width dr_e , and the force F_{r_e} , may therefore be considered to act at the axis of the roll and in a direction tangential to the axis Z_1 .

The tangential force, F_{r_e} , acting at a distance, r_e , from the axis is considered to be due to an element, dT_U , of the torque T_U , required to cause rotation of the upper platen relative to the workpiece by rolling action, and hence cause the plastically deforming zone to rotate relative to the workpiece axis.

Assuming quasi-static and frictionless deformation the total torque may be predicted by the following reasoning :

From Hawkyard et al [24]

$$s \cong (2fr_e/\theta)^{\frac{1}{2}} \quad (A1)$$

the force F_{z_e} , due to the normal pressure, P , on the element of radius r_e , and width dr_e , is given by :

$$F_{z_e} \cong P \cdot (2fr_e/\theta)^{\frac{1}{2}} dr_e \quad (A2)$$

Considering moments about OA

$$F_{r_e} \lambda \cong F_{z_e} s/2 \quad (A3)$$

$$\text{but } \lambda \cong s^2/2f \cong (2fr_e)/(2\theta f) \quad (A4)$$

The element of torque, dT_U , about the axis Z_1 , is given by :

$$dT_U \approx F_{r_e} r_e \quad (A5)$$

or
$$F_{r_e} \approx dT_U / r_e \quad (A6)$$

From equations (A2), (A3), (A4) and (A6)

$$(dT_U / r_e) (r_e / \theta) \approx P (2fr_e / 2\theta) de$$

hence
$$dT_U \approx P \cdot f r_e \cdot dr_e \quad \text{and} \quad (A7)$$

the torque T_U about the axis Z_1 , is given by

$$T_U \approx P \cdot f \cdot \int_0^r r_e \cdot dr_e$$

$$\approx P f r^2 / 2 \quad (A8)$$

An alternative method used to derive this equation is shown on page 202.

From equation (51) P may be expressed approximately by:

$$P \approx F_H' / r\bar{s} \quad (A9)$$

If rolling is assumed to be taking place at the upper and lower platens simultaneously as proposed by Hawkyard et al [24] then the predicted total torque, T_{Tp} , can be obtained from equations (A8) and (A9)

$$T_{Tp} \approx F_H' f r / \bar{s} \quad (A10)$$

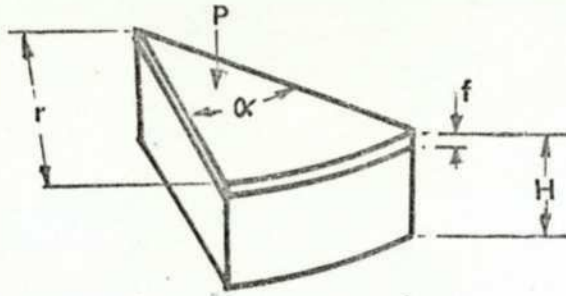
Using equation (A10) and the results from experimental upsetting stated in Tables 16, 17 and 18 both the mean and standard deviation of the values (experimental value of total torque)/(total torque predicted), T_T / T_{Tp} , were as follows :

Specimens of height/diameter ratio $H_0/D_0 \approx 0.2$	Mean value T_T / T_{Tp}	Standard Deviation
Aluminium Alloy HE.30 conical upper platen, flat lower platen	1.35	0.36
Aluminium Alloy HE.30 dual conical platens	1.02	0.32
Alloy Steel En.18 conical upper platen, flat lower platen	0.84	0.17

From the results the value of the standard deviation may be considered to be high. However, the method of predicting total torque does not produce variances as large as the method based upon energy considerations. It is suggested that the proposed method of prediction would be useful for design purposes until an improved method can be devised.

An alternative method of deriving the relationship $T_U = Pfr^2/2$

Consider a segment of a cylindrical workpiece of rigid perfectly plastic material of current radius, r , axial height, H , of angle α radians, as illustrated below.



A normal pressure, P , at the interface of the conical upper die-workpiece interface will cause plastic deformation. Application of a torque, T_U , will cause the upper conical platen, and hence the plastically deforming region, to rotate relative to the workpiece. Let the application of the torque, T_U , through an angle, α , cause the workpiece to be reduced in height by, f , the axial feed/rev. An element of volume of the segment of the workpiece, dV , will therefore be displaced and assuming, $f \ll H$, then

$$dV = r^2 \alpha f / 2 \quad (A.11)$$

Hence energy dissipated in causing this displacement = PdV .

Equating the energy dissipated to the energy supplied then

$$PdV = T_U \alpha \quad (A.12)$$

Combining equations A.11 and A.12 gives

$$T_U = Pfr^2/2$$

which is the same as given in equation A.8.

TABLE NO. 1 RESULTS FROM EXPERIMENTS BY HAWKYARD ET AL [24] STANDARDISED FOR WORKPIECE RADIUS AND AXIAL FEED/REV.

Based upon the methods of estimating the axial force proposed by Hawkyard et al [24] and Kubo et al [34]
Material for specimens: Aluminium alloy Specification BS.1474 HE.30 TF. fully heat treated.

θ Degrees	r_o mm	H_o/D_o	ϵ	r mm	f mm/rev	F_E kN	$f^{0.5}$	$f^{0.63}$	$r^{1.5}$	$r^{1.37}$	$F_E/r^{1.5} f^{0.5}$	$F_E/r^{1.37} f^{0.63}$
2	19.0	0.12	0.14	20.37	0.20	132	0.45	0.36	91.94	62.13	3.21	5.85
2	25.4	0.12	0.05	26.00	0.04	135	0.20	0.13	132.57	86.80	5.09	11.78
2	25.4	0.19	0.07	26.30	0.06	135	0.24	0.17	134.88	88.17	4.08	9.01
4	19.25	0.12	0.16	20.63	0.09	67	0.30	0.22	93.70	63.22	2.38	4.84
4	19.05	0.12	0.29	22.02	0.15	81	0.39	0.30	103.33	69.13	2.03	3.87
4	19.05	0.12	0.29	22.02	0.25	122	0.50	0.42	103.33	69.13	2.36	4.23
4	25.4	0.12	0.22	28.35	0.09	113	0.30	0.22	150.95	97.72	2.50	5.28
4	25.4	0.12	0.27	29.07	0.17	135	0.42	0.33	156.74	101.14	2.06	4.08
4	31.75	0.12	0.22	35.44	0.16	230	0.40	0.31	210.98	132.68	2.73	5.50
8	25.4	0.12	0.18	27.80	0.30	115	0.55	0.52	146.58	95.13	1.51	2.33
8	25.4	0.19	0.25	28.80	0.20	107	0.45	0.45	154.56	99.85	1.55	2.40

TABLE NO. 2 RESULTS FROM EXPERIMENTS BY HAWKYARD ET AL [24] STANDARDISED FOR WORKPIECE RADIUS AND AXIAL FEED/REV.

Based upon the method of estimating axial force proposed by Kubo et al [34]

Material for specimens : Commercially pure aluminium

θ Degrees	r_o mm	H_o/D_o	ϵ	r mm	f mm/rev	F_E kN	$r^{1.37}$	$f^{0.63}$	$f^{0.5}$	$F_E/r^{1.37}f^{0.63}$
2	38.1	0.12	0.16	41.27	0.12	50	163.46	0.26	.35	1.16
2	50.8	0.25	0.16	55.03	0.12	66	242.45	0.26	.35	1.03
4	38.1	0.12	0.16	41.27	0.12	35	163.46	0.26	.35	0.81
4	50.8	0.12	0.16	55.03	0.12	58	242.45	0.26	.35	0.91
8	38.1	0.12	0.16	41.27	0.12	32	163.46	0.26	.35	0.75
8	50.8	0.12	0.16	41.27	0.12	50	242.45	0.26	.35	0.78

TABLE NO. 3 RESULTS FROM EXPERIMENTS BY PRABHU ET AL [59] ON AN ORBITAL RIVETING ATTACHMENT STANDARDISED FOR WORKPIECE RADIUS AND AXIAL FEED/REV.
Based upon the method of estimating axial force proposed by Kubo et al [24]
Material for specimens : Mild steel

θ Degrees	r_o mm	H_o/D_o	f mm/rev	F_E kN	$f^{0.63}$	$r^{1.37}$	$F_E/r^{1.37}f^{0.63}$
2	2.35	0.8	0.21	11.71	0.38	3.22	9.41
2	2.35	0.8	0.17	10.10	0.33	3.22	9.58
2	2.35	0.8	0.08	8.33	0.20	3.22	12.67
4	2.35	0.8	0.21	9.02	0.38	3.22	7.44
4	2.35	0.8	0.17	7.45	0.33	3.22	7.07
4	2.35	0.8	0.08	6.37	0.20	3.22	9.69
6	2.35	0.8	0.21	6.96	0.38	3.22	5.74
6	2.35	0.8	0.17	6.67	0.33	3.22	6.33
6	2.35	0.8	0.08	5.98	0.20	3.22	4.93
8	2.35	0.8	0.21	6.18	0.38	3.22	5.09
8	2.35	0.8	0.17	5.20	0.33	3.22	4.93
8	2.35	0.8	0.08	4.41	0.20	3.22	6.71

TABLE NO. 4 SUMMARY OF DESIGN AND PERFORMANCE OF CERTAIN ROTARY FORGING MACHINES.

Description of Machine	Slick Mill U.S.A.	Rotoform U.K.	PXW 100 A Poland	OFP 100 U.S.A.	U.M.I.S.T. U.K.	University of Nottingham	G.I.R.I. Japan	The City University London
Maximum axial force, F , max.	13.0 MN	2.0 MN	1.6 MN	1.5 MN	0.25 MN	0.30 MN	1.20 MN	0.5 MN
Maximum workpiece radius, r , max.	0.482 m	0.075 m	0.050 m	0.050 m	0.050 m	0.045 m	N.A.	0.075 m
Maximum workpiece area, A , max.	0.731 m ²	0.018 m ²	0.0078 m ²	0.0078 m ²	0.0078 m ²	0.0064 m ²	N.A.	0.010 m ²
Axial displacement	0.50 m	0.30 m	0.14 m	0.14 m	0.165 m	0.20 m	0.20 m	0.15 m
Inclination of axes	10.66°	4°	1°- 2°	1°- 2°	0°- 8°	0°- 10°	0°- 5°	0°- 10°
Weight	3.98 MN	0.24 MN	49 kN	66.7 kN	12.26 kN	14.7 kN	N.A.	34.3 kN
Total Power available	1.5 MW	180 kW	30 kW	30 kW	9.6 kW	25 kW	13 kW	18.5 kW
*Estimated average induced stress at upper die-workpiece interface at maximum workpiece radius for axial feed 0.002 m/rev.	416 MPa	433 MPa	390 MPa	365 MPa	151 MPa	234 MPa	N.A.	193 MPa
*Estimated average induced stress at upper die-workpiece interface for axial feed 0.001 at workpiece radius	644 MPa	670 MPa	603 MPa	565 MPa	234 MPa	362 MPa	N.A.	300 MPa
$F_{\max}/A_{\max}$	18 MPa	111 MPa	205 MPa	192 MPa	32 MPa	47 MPa	N.A.	28 MPa
Maximum axial force/weight	3.3	8.3	32.6	22.5	20.4	20.4	N.A.	14.6
Total power available/maximum workpiece area	2.1 MW/m ²	10.0 MW/m ²	3.8 MW/m ²	3.8 MW/m ²	1.2 MW/m ²	3.9 MW/m ²	N.A.	2.4 MW/m ²
Oscillations per Min. (Maximum)	100	1000	180	150	360	400	160	1200
Maximum rolling velocity	303 m/min	471 m/min	56 m/min	47 m/min	113 m/min	113 m/min	94 m/min	400 m/min

* Based upon method of estimating axial force proposed by Kubo et al [34]

TABLE NO. 5 ESTIMATE OF UPPER DIE-WORKPIECE INTERFACE RELATIVE VELOCITIES FOR THE CONTOURED UPPER DIE ILLUSTRATED IN FIGURE 19

x	Location	r_1	z	β	$x - (\pi/2)$	$\beta + \theta$	$(\pi/2) - x + \theta$	$V_{1\omega} r_1$	$V_{1\omega_1}$
95.71	0	0	0	0.00	5.71	8.00	2.29	0.000	0.00
95.71	r_{1A_1}	10	0	0.00	5.71	8.00	2.29	0.014	0.14
95.71	r_{aA_2}	10	+5	+26.56	5.71	34.56	2.29	0.083	0.83
95.71	r_{bB_1}	20	0	0.00	5.71	8.00	2.29	0.014	0.28
95.71	r_{bB_2}	20	+5	+14.04	5.71	22.04	2.29	0.048	0.96
95.71	r_{cC_1}	30	0	0.00	5.71	8.00	2.29	0.004	0.12
95.71	r_{cC_2}	30	-5	-9.46	5.71	-1.46	2.29	-0.009	-0.27
95.71	r_{dD_1}	50	-5	-5.71	5.71	-2.29	2.29	0.000	0.00
90.00	0	0	0	0.00	0.00	8.00	8.00	0.000	0.00
90.00	r_{aA_1}	10	0	0.00	0.00	8.00	8.00	0.000	0.00
90.00	r_{aA_2}	10	+5	+26.56	0.00	34.56	8.00	0.070	0.70
90.00	r_{aB_1}	20	+0	0.00	0.00	8.00	8.00	0.000	0.00
90.00	r_{bB_2}	20	+5	+14.04	0.00	22.04	8.00	0.035	0.70
90.00	r_{cC_1}	30	+0	0.00	0.00	8.00	8.00	0.000	0.00
90.00	r_{cC_2}	30	-5	-9.46	0.00	-1.46	8.00	-0.023	-0.69
90.00	r_{dD_1}	50	-5	-5.71	0.00	-2.29	8.00	-0.014	-0.70
84.29	0	0	0	0.00	-5.71	8.00	13.71	0.000	0.00
84.29	r_{aA_1}	10	0	0.00	-5.71	8.00	13.71	-0.014	-0.14
84.29	r_{aA_2}	10	+5	+26.56	-5.71	34.56	13.71	0.057	+0.57
84.29	r_{bB_1}	20	0	0.00	-5.71	8.00	13.71	-0.014	-0.28
84.29	r_{bB_2}	20	+5	+14.04	-5.71	22.04	13.71	-0.021	-0.41
84.29	r_{cC_1}	30	0	0.00	-5.71	8.00	13.71	-0.014	-0.42
84.29	r_{cC_2}	30	-5	-9.46	-5.71	-1.46	13.71	-0.038	-1.14
84.29	r_{dD_1}	5	-5	-5.71	-5.71	-2.29	13.71	-0.028	-1.40

TABLE NO. 6 RESULTS OBTAINED DURING AXIAL COMPLIANCE TESTS

(a)

Pressure applied to ram MPa (lbf/in ²)	Estimated load kN	Compliance $\mu\text{m}/\text{kN}$ at test points specified in Figure 49			
		①	②	③	④
6.90 (1000)	171	2.8	Displacement not measurable with dial gauge	4.0	Displacement not measurable with dial gauge
10.34 (1500)	256	2.4		3.8	
13.79 (2000)	342	2.6		3.8	

(b)

Pressure applied to ram MPa (lbf/in ²)	Estimated load kN	Compliance $\mu\text{m}/\text{kN}$ at test points specified in Figure 49			
		⑥	⑤	①	⑦
6.90 (1000)	171	3.2	1.2	2.9	2.9
10.34 (1500)	256	2.2	1.0	2.6	2.8
13.79 (2000)	342	2.9	1.1	2.9	2.9

(c)

Pressure applied to ram MPa (lbf/in ²)	Estimated load kN	Compliance $\mu\text{m}/\text{kN}$ at test point specified in Figure 49			
		①	②	③	⑩
3.45 (500)	86	0.6	1.5	0.3	0.58
6.90 (1000)	171	0.7	2.0	0.7	0.70
10.34 (1500)	257	0.9	1.8	0.9	0.74
13.79 (2000)	342	1.1	2.0	1.1	0.76
17.24 (2500)	428	1.3	2.3	1.2	0.79
20.69 (3000)	514	1.3	2.1	1.3	0.78

(d)

Pressure applied to ram MPa (lbf/in ²)	Estimated load kN	Compliance $\mu\text{m}/\text{kN}$ at test points specified in Figure 49			
		①	④	③	⑧
3.45 (500)	86	1.7	1.0	1.5	1.7
6.90 (1000)	171	1.3	1.0	0.7	1.3
10.34 (1500)	257	1.0	1.6	1.4	1.5
13.79 (2000)	342	1.3	1.6	1.4	1.8
20.69 (3000)	514	1.43	1.5	1.3	1.4

TABLE NO. 7 RESULTS OBTAINED DURING VIBRATION TESTS

Rotational velocity of upper platen rev/sec		TEST 1		TEST 2		TEST 3	
		Axial load off	Axial load on	Axial load off	Axial load on	Axial load off	Axial load on
1	Acceleration g	0.05	0.07	0.06	0.06	0.04	0.04
	Frequency kHz	0.4	0.4	0.4	0.4	0.4	0.4
2	Acceleration g	0.08	0.13	0.10	0.12	0.06	0.08
	Frequency kHz	0.9	0.9	0.9	0.9	0.9	0.9
3	Acceleration g	0.15	0.26	0.11	0.18	0.11	0.16
	Frequency kHz	1.2	1.2	1.2	1.2	1.2	1.2
4	Acceleration g	0.19	0.40	0.14	0.28	0.16	0.41
	Frequency kHz	1.7	1.7	1.7	1.7	1.7	1.7
5	Acceleration g	0.23	0.56	0.20	0.36	0.23	0.56
	Frequency kHz	2.1	2.1	2.1	2.1	2.1	2.1

TABLE No. 8 AXIAL FORCE TRANSDUCER CALIBRATION TEST RESULTS

Axial Force kN	U.V. Recorder Trace displacement 
0	0
50	8.5
100	18.0
150	27.5
200	37.5
250	47.5
300	58.5
350	68.0
400	78.0
450	87.5
500	97.0

10:1 ATTENUATOR Kistler Type 5361

Charge amplifier sensitivity setting 5 kN/cm

TABLE NO. 9 RESULTS OBTAINED DURING CALIBRATION OF UPPER AND LOWER
PLATEN TORQUE TRANSDUCERS

Applied force		Torque	U.V. Recorder trace displacement	
lbf	(N)		Upper platen mm	Lower platen mm
0	0.0	0.0	0.0	0.0
20	89.0	60.9	10.5	10.0
40	177.9	121.9	20.0	19.5
60	266.9	182.8	30.0	30.0
80	355.9	243.8	41.5	40.0
100	444.5	304.7	52.5	50.5
120	533.8	365.6	61.5	60.5
140	622.7	426.6	73.0	71.5
160	711.7	487.5	86.0	82.0
180	800.7	548.5	98.0	94.0
200	889.6	609.4	110.0	104.0
180	800.7	548.5	100.0	94.0
160	711.7	487.5	88.0	84.5
140	622.7	426.6	76.5	74.5
120	533.8	365.6	64.5	63.0
100	444.5	304.7	55.0	55.0
80	355.9	243.8	44.0	43.5
60	266.9	182.8	34.0	32.0
40	177.9	121.9	23.0	22.0
20	89.0	60.9	11.0	13.0
0	0.0	0.0	0.0	0.0

Charge amplifier sensitivity setting 1 kN/cm

Position	Transducer Serial Nos.	Transducer Serial Nos.
Upper end spindle	104839	104384
Lower end spindle	104921	104873

Transducers Kistler Type 9031

TABLE NO. 10 RESULTS FROM CALIBRATION OF THE LOWER PLATEN
TRANSLATIONAL DISPLACEMENT TRANSDUCER.

Vernier Height gauge reading mm	Digital Voltmeter mV. D.C.	U.V. Recorder trace displacement mm
0	0.0	0.0
15.0	15.3	0.8
30.1	30.8	35.5
45.6	45.2	51.5
59.9	60.3	66.0
75.0	75.4	82.0
75.3	75.7	83.5
90.7	90.8	100.5
105.4	105.5	115.5
121.1	120.7	132.0
135.9	135.1	147.0
146.0	145.2	157.0

Galvanometer Type B.100 Serial No.8-7121

TABLE NO. 11 CALIBRATION RESULTS, UPPER AND LOWER PLATEN ANGULAR VELOCITY MEASURING SYSTEMS

Tests using Dawe Instruments Limited Strobeflash Type 1200E
Serial No. 34389.

(a) Test at fixed frequencies using vibrating reed incorporated in Strobeflash

Strobeflash frequency Hz	Panel mounted digital frequency meters	
	Upper platen Hz	Lower platen Hz
50	0.050	0.050
100	0.100	0.100
150	0.149	0.150
200	0.200	0.200

(b) Test using indicating rev/min scale on Strobeflash

Strobeflash rev/min	Panel mounted digital frequency meters			
	Upper platen kHz rev/min		Lower platen kHz rev/min	
60	0.048	60.0	0.048	60.0
120	0.096	120.0	0.096	120.0
180	0.145	181.2	0.144	180.0
240	0.191	238.7	0.192	240.0
300	0.237	296.2	0.237	296.2

TABLE NO. 12 PRESSURE TRANSDUCER CALIBRATION

Sub-standard Pressure gauge MPa (lbf/in ²)	mV	U.V. Recorder trace displacement mm
0.0 0	0	0.0
3.45 (500)	0.85	40.0
6.90 (1000)	1.67	79.5
10.34 (1500)	2.55	116.0
13.79 (2000)	3.40	156.5
17.24 (2500)	4.24	192.0
20.69 (3000)	5.16	228.0

Budenburg sub-standard gauge Serial No.10280537

Druck Ltd., pressure transducer Type PDCR1D
Serial No. 22204

Galvanometer Type B.100 Serial No. 8-7286

TABLE NO. 13 RESULTS FROM CALIBRATION OF TRANSDUCER USED
FOR INCLINATION OF THE GEOMETRIC AXIS OF THE
UPPER PLATEN RELATIVE TO THE VERTICAL AXIS OF
THE MACHINE.

Inclination of platen axes θ Degrees	U.V. Recorder trace displacement mm
-5.0	-66.5
-4.0	-53.5
-3.0	-40.5
-2.0	-27.0
-1.0	-13.5
0.0	0.0
+1.0	+13.5
+2.0	+27.5
+3.0	+41.0
+4.0	+54.5
+5.0	+69.0
+6.0	+83.5
+7.8	+98.0
+8.0	+113.0

Galvanometer Type B.100 Serial No. 8-7214

TABLE 14 LIST OF DRAWINGS

DRAWING NO.	DESCRIPTION	No. OFF	REMARKS
1001	Plate, bearing support, outer	2	
1002	Plate, bearing support, inner	2	
1003	Blocks, bearing plate	4	
1004	Shaft, trunnion bearing	2	
1005	Spacer ring	2	
1006	Bearing support plate	1	
1007	Hydraulic cylinder support plate	1	
1008	Bearing ring	1	
1009	Upper platen support plate	1	
1010	Spindle	2	
1011	Bearing housing	2	
1012	Bearing housing	2	
1013	Bearing cap, small	2	
1014	Bearing cap, large	2	
1015	Sub assembly drg		
1016	Pillar main support	4	
1017	Pillar sub-frame	4	
1018	Lower platen drive plate, upper	1	
1019	Lower platen drive plate, lower	1	
1020	Nut, special	4	
1021	Torque transducer, bearing housing	4	
1022	Location pin, torque transducer	4	
1023	Torque transducer support plate	4	
1024	Thrust plate	1	
1025	Thrust washer	1	
1026	Shaft, gear mounting, torque transducer, lower platen	1	
1027	Shaft, gear, speed reduction, Lower platen	1	
1028	Slide bearing adjustable	4	
1029	Top bearing plate upper platen drive	1	
Sheet 1 of 3			THE CITY UNIVERSITY LONDON.
			0.5 MN ROTARY FORGING MACHINE

TABLE 14 CONTINUED

DRAWING NO.	DESCRIPTION	No. OFF	REMARKS
1030	Lower bearing plate, upper platen drive	1	
1031	Pillar, upper platen drive plates	6	
1032	Pillar, extension plate support	2	
1033	Gear spindle, upper platen drive	1	
1034	Gear spindle, upper plate drive	1	
1035	Spacer, upper platen drive	2	
1036	Thrust pin	1	
1037	Extension plate, upper platen drive	1	
1038	Mounting plate	1	
1039	Gusset plate		
1040	Nut trunnion	1	
1041	Main frame	1	
1042	Lead screw	1	
1043	Stop collar	1	
1044	Lead screw	1	
1045	Gear spindle	1	
1046	Coupling sleeve	1	
1047	Clamp ring	1	
1048	Eccentric bush	8	
1049 ^A _B	Lead screw support ^A - Lower ^B - Upper	2	1-off each type
1050	Bearing support	4	
1051	Upper platen beam	2	
1052 ^A _B	Upper platen, gearbox support	2	1-off each type
1053	Belt tensioning support	1	
1054 ^A _B	Upper platen, gearbox mounting beam	2	1-off each type
1055	Ejector pin	1	
1056	Workpiece holder Number 1	1	
1057 ^A _B	Workpiece support, P.M. compacting	1	
1058	Conical upper platen	5	1-off each conical angle 186° 172° 168° 164° 160°
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TABLE 14 CONTINUED

[illegible]

TABLE 15 SCHEDULE OF PROPRIETARY ITEMS

DESCRIPTION	SUPPLIER	Qty	REMARKS
Thrust bearing	S.K.F. Ltd.	2	SKF Ltd. No.29420B
Cylindrical roller bearing	S.K.F. Ltd.	2	SKF Ltd. No.NU1030
Taper roller bearing	S.K.F. Ltd.	2	SKF Ltd. No.32019
Circlips external 150 mm	Anderton Ltd.	4	
Seals	S.K.F. Ltd.	4	
Plummer Blocks	S.K.F. Ltd.	2	Spheroidal graphite iron
Bearings 90 mm	S.K.F. Ltd.	2	
Deep groove ball bearings	S.K.F. Ltd.	4	30 mm inside diameter
Circlips internal 72 mm	Anderton Ltd.	8	
Circlips external 30 mm	Anderton Ltd.	8	
48T 6 DP spur gear 40F	David Brown Gears Ltd.	7	Bored to suit taper lock bushes
72T 6 DP spur gear 40F	David Brown Gears Ltd.	1	" "
48T 6 DP spur gear 225F	David Brown Gears Ltd.	1	Parallel keyway for 40 mm shaft
Taper lock bush 3030	Fenner Limited	4	40 mm diameter bore
Taper lock bush 3030	Fenner Limited	2	75 mm diameter bore
Taper lock bush 3030	Fenner Limited	2	1.75 in. diameter bore
Variator 9:1 11 kW	Allspeeds Limited	1	
Hydraulic cylinder	Parker Hannifin Ltd.	1	7 in. diameter bore
Hydraulic pressure supply 7.5 kW	Parker Hannifin Ltd.	1	Complete with controls & gauges
Gearbox, reduction 29:4	Fenner Limited	1	
Gearbox, reduction 29:6	Fenner Limited	1	
Bushes, flanged, bearing	Manganese Bronze Ltd.	8	40 mm O.D. Self lubricating 30 mm
Pulley, 6 groove 125 mm	Fenner Limited	1	
Pulley, 3 groove 250 mm	Fenner Limited	2	
Pulley, 3 groove 125 mm	Fenner Limited	2	Used for belt tensioning
Lead Screw	Kingston Eng. Limited	1	
Nut for lead screw	Kingston Eng. Limited	1	
Handwheel 300 mm diameter	Clayton Limited	1	
Belt tension pulley bearing	Fenner Limited	4	Type 140L.0525 25 mm dia. ball bearing
Sheet 1 of 2			THE CITY UNIVERSITY LONDON.
			0.5 MN ROTARY FORGING MACHINE

TABLE NO. 16 RESULTS FROM EXPERIMENTS ON COLD UPSETTING SOLID CYLINDRICAL SPECIMENS INITIAL HEIGHT TO DIAMETER $H_0/D_0 \approx 0.2$ USING A CONICAL UPPER DIE AND FLAT LOWER DIE

Specimen Material: Aluminium Alloy BS.1474 HE.30 TF.

* Specific Rotary Forging Force

Test No.	θ Degrees	D_0 mm	H_0/D_0	ϵ_z	rev / min	f mm/rev	F_E kN	F_A/F_E *	T_T Nm	F_E/F_K	F_E/F_H	F_E/F_P	n %
1A	2	37.50	0.19	0.15	60	0.054	78	6.17	27	2.01	1.04	0.90	15
2A	2	37.50	0.19	0.15	120	0.050	77	6.25	30	2.09	1.05	0.91	12
3A	2	37.50	0.19	0.15	120	0.040	81	5.94	60	2.53	1.19	0.99	5
4A	2	37.50	0.19	0.15	180	0.023	70	6.87	47	3.09	1.18	1.01	4
5A	2	37.50	0.19	0.15	240	0.034	70	6.87	59	2.59	1.07	0.93	4
6A	2	56.94	0.20	0.15	60	0.112	240	4.99	155	2.09	1.16	1.00	13
7A	2	56.94	0.20	0.15	120	0.075	230	5.20	148	2.58	1.24	1.06	10
8A	2	56.94	0.20	0.15	120	0.091	235	5.09	148	2.28	1.19	1.02	12
9A	2	56.94	0.20	0.15	180	0.073	215	5.57	154	2.40	1.16	1.00	9
10A	2	56.94	0.20	0.15	180	0.096	212	11.26	157	1.99	1.05	0.90	11
11A	2	56.94	0.20	0.15	120	0.214	290	10.20	178	1.64	1.07	0.92	21
12A	2	76.24	0.18	0.15	180	0.122	355	5.90	253	1.98	1.04	0.90	16
13A	2	76.24	0.18	0.15	240	0.094	335	6.25	216	2.19	1.07	0.92	14
14A	2	76.24	0.18	0.15	120	0.146	373	5.61	240	1.84	1.02	0.88	20
15A	2	56.94	0.18	0.15	180	0.232	309	3.88	133	1.71	1.10	0.96	31
16A	2	56.94	0.18	0.15	180	0.211	280	4.28	126	1.60	1.03	0.90	30
17A	8	76.24	0.18	0.15	120	0.170	211	9.92	144	2.28	1.09	0.97	38
18A	8	76.24	0.18	0.15	120	0.165	221	9.47	141	2.48	1.17	1.03	38
19A	8	56.94	0.18	0.15	120	0.266	153	7.83	106	1.84	1.26	1.11	45
20A	8	37.50	0.18	0.15	120	0.289	67	7.18	46	1.44	1.16	1.01	46
21A	6	56.94	0.20	0.15	120	0.167	154	7.78	101	2.04	1.11	0.97	30
22A	6	37.50	0.18	0.15	120	0.177	67	7.18	58	1.64	1.06	0.93	22

TABLE NO. 16 CONTINUED

* Specific Rotary Forging Force

Test No.	θ Degrees	D_0 mm	H_0/D_0	ϵ_z	rev/ min	f mm/rev	F_E kN	F_A/F_E *	T_T Nm	F_E/F_K	F_E/F_H	F_E/F_P	η %
23A	6	76.24	0.18	0.15	120	0.232	278	7.53	173	2.06	1.16	1.04	42
24A	6	37.50	0.18	0.15	180	0.289	88	5.47	82	1.58	1.27	0.99	26
25A	6	76.24	0.18	0.15	180	0.068	216	9.70	126	3.64	1.26	1.10	17
26A	6	76.24	0.18	0.15	120	0.131	262	8.00	74	2.78	1.28	1.11	55
27A	4	76.24	0.18	0.15	120	0.116	262	8.00	81	2.32	1.11	0.96	45
28A	6	56.94	0.18	0.15	180	0.190	139	8.61	64	1.70	1.01	0.87	53
29A	4	56.94	0.20	0.15	120	0.192	196	6.11	92	1.84	1.12	0.98	38
30A	4	37.50	0.19	0.15	120	0.268	97	4.96	56	1.41	1.03	0.90	35
31A	10	37.50	0.18	0.15	60	0.414	78	6.17	*	1.54	1.12	0.99	
32A	10	37.50	0.18	0.15	60	0.390	76	6.33	*	1.57	1.12	0.99	
33A	10	56.95	0.20	0.15	60	0.420	190	6.30	*	1.95	1.26	1.10	
34A	10	56.95	0.20	0.15	120	0.228	154	7.78	*	2.32	1.26	1.10	
35A	10	56.94	0.20	0.15	120	0.122	118	10.15	*	2.64	1.16	1.01	
36A	10	76.24	0.18	0.15	120	0.109	201	10.42	*	3.32	1.29	1.14	
37A	10	76.24	0.18	0.15	120	0.268	227	9.22	*	2.12	1.13	0.99	
38A	10	76.24	0.18	0.15	120	0.087	191	10.97	*	3.65	1.31	1.15	
39A	10	56.95	0.20	0.15	120	0.100	120	9.98	*	3.05	1.18	1.03	
40A	10	37.50	0.18	0.15	120	0.081	52	9.25	*	2.73	1.31	1.16	
41A	10	76.24	0.18	0.15	120	0.076	178	11.76	*	3.69	1.26	1.11	
52A	10	76.24	0.20	0.15	120	0.075	185	11.32	124	4.60	1.27	1.11	19
53A	10	38.10	0.20	0.15	120	0.094	46	10.79	44	2.27	1.05	0.93	16
54A	10	56.95	0.20	0.15	120	0.156	124	9.66	90	2.37	1.14	1.02	32
55A	10	56.95	0.20	0.15	120	0.125	116	10.33	78	2.55	1.13	0.99	29
66A	10	76.24	0.18	0.15	120	0.604	316	6.63	342	1.79	1.10	.97	38
74A	10	76.24	0.20	0.15	120	0.054	180	11.63	133	4.62	1.17	1.10	9
75A	10	76.24	0.20	0.15	120	0.059	180	11.63	111	4.38	1.28	1.11	12

* Values not recorded.

TABLE NO. 17 RESULTS FROM EXPERIMENTS ON COLD UPSETTING SOLID CYLINDRICAL SPECIMENS INITIAL HEIGHT TO DIAMETER $H_0/D_0 \approx 0.2$ USING A CONICAL UPPER DIE AND FLAT LOWER DIE
Specimen Material: Alloy Steel En.18 BS.970

* Specific Rotary Forging Force

Test No.	θ Degrees	D_0 mm	H_0/D_0	ϵ_z	rev / min	f mm/rev	F_E kN	F_A/F_E *	T_T Nm	F_E/F_K	F_E/F_H	F_E/F_P	η %
1S	10	38.86	0.20	0.15	120	0.304	185	6.29	N.R.	1.88	1.13	1.00	
2S	10	38.86	0.18	0.15	120	0.375	227	5.13	187	2.03	1.30	1.13	35
3S	10	38.86	0.18	0.15	120	0.225	165	7.05	90	1.96	1.12	1.00	44
4S	8	38.86	0.18	0.15	120	0.375	242	4.81	158	1.87	1.18	1.03	41
5S	8	38.86	0.18	0.15	120	0.150	165	7.05	87	2.27	1.15	1.00	31
6S	6	38.86	0.18	0.15	120	0.262	252	4.62	141	2.02	1.25	1.09	32
7S	4	38.86	0.18	0.15	120	0.354	340	3.42	216	1.75	1.17	1.03	28
8S	4	38.86	0.18	0.15	120	0.250	320	3.64	184	2.06	1.28	1.11	24
9S	4	38.86	0.18	0.15	120	0.625	386	3.01	214	1.40	1.06	0.92	48

TABLE NO. 18 MEAN AND STANDARD DEVIATION OF THE VALUES F_E/F_H AND F_E/F_P FOR COLD UPSETTING SOLID CYLINDRICAL SPECIMENS OF ALUMINIUM ALLOY AND ALLOY STEEL INITIAL HEIGHT TO DIAMETER $H_0/D_0 \approx 0.2$ USING A CONICAL UPPER DIE AND FLAT LOWER DIE

	Aluminium alloy HE.30 TF		Alloy steel En.18	
	F_E/F_H	F_E/F_P	F_E/F_H	F_E/F_P
Mean Value	1.15	1.00	1.18	1.03
Standard deviation	0.09	0.08	0.10	0.07

TABLE NO. 19 RESULTS FROM EXPERIMENT TO DETERMINE THE MINIMUM AND MAXIMUM VALUES OF F_A/F_E FOR VARIOUS VALUES OF, θ , DURING COLD UPSETTING OF SOLID CYLINDRICAL SPECIMENS OF INITIAL HEIGHT TO DIAMETER $H_0/D_0 \approx 0.2$ USING A CONICAL UPPER DIE AND FLAT LOWER DIE

Specimen Material: Aluminium Alloy BS.1474 HE.30 TF.

Specimen No.	θ degrees	D_0 mm	H_0/D_0	rev/min	F_E kN	f mm/rev	T_T Nm	Specific rotary forging force F_A/F_E	n %	Remarks
52A	10	76.30	0.20	120	105	0	114	14.1	0	
					150	0.029	125	9.9	5	
					185	0.075	156	8.0	11	
67A	10	76.24	0.20	120	103	0	97	14.4	0	
					129	0.014	102	11.5	3	
					154	0.036	136	9.6	6	
					180	0.113	162	8.2	16	
69A	10	76.24	0.20	120	28	0	31	52.4	0	
					41	0	41	36.0	0	
					46	0	49	32.0	0	
					56	0	61	26.2	0	
					72	0	68	20.6	0	
					85	0	73	17.4	0	
					103	0	99	14.4	0	
					118	0.005	109	12.5	1	
					144	0.010	120	10.3	2	
					165	0.087	207	8.9	10	
					216	0.125	229	6.8	12	Fracture occurred
70A	8	76.24	0.20	120	129	0	94	11.5	0	
					160	0.007	104	9.3	2	
					247	0.437	250	6.0	38	Fracture occurred
71A	6	76.24	0.20	120	144	0	99	10.3	0	
					160	0.007	115	9.3	1	
					167	0.008	137	9.0	1	
					252	0.250	145	6.0	30	
72A	4	76.24	0.20	120	288	0.360	251	5.1	38	Fracture occurred
					149	0	96	9.9	0	
					169	0.009	118	8.7	2	
					191	0.010	145	7.7	2	
					335	0.350	284	4.4	27	Fracture occurred
73A	2	76.24	0.20	120	128	0	61	11.6	0	
					149	0	112	9.9	0	
					169	0	112	8.7	0	
					195	0	119	7.3	0	
					206	0	132	7.2	0	
					216	0.003	153	6.8	0.5	
					242	0.004	185	6.1	0.5	
					260	0.012	191	5.7	1	
					273	0.014	204	5.4	2	

TABLE NO. 20 RESULTS FROM EXPERIMENTSON COLD UPSETTING SOLID CYLINDRICAL SPECIMENS INITIAL HEIGHT TO DIAMETER $H_0/D_0 \approx 0.2$ USING CONICAL UPPER AND LOWER DIES OF EQUAL CONICAL ANGLE

Specimen Material: Aluminium Alloy BS.1474 HE.30 TF.

* Specific Rotary Forging Force

Test No.	θ Degrees	D_0 mm	H_0/D_0	ϵ_z	rev / min	f mm/rev	F_E kN	* F_A/F_E	T_T Nm	F_E/F_K	F_E/F_H	F_E/F_P	η %
43A	4	56.95	0.20	0.15	120	0.232	190	6.30	127	1.59	1.02	0.82	33
44A	4	56.95	0.20	0.15	120	0.378	245	4.89	150	1.61	1.13	0.98	45
45A	4	56.94	0.20	0.15	120	0.142	180	6.65	84	2.04	1.13	0.98	31
46A	4	76.45	0.20	0.15	120	0.142	247	8.53	115	1.92	0.98	0.89	39
47A	4	76.45	0.20	0.15	120	0.250	306	6.38	121	1.66	1.03	0.90	63
48A	4	76.45	0.20	0.15	120	0.034	221	9.53	113	4.29	1.28	1.06	10
49A	4	38.10	0.20	0.15	120	0.375	134	3.70	100	1.54	1.20	1.05	28
50A	4	38.10	0.20	0.15	120	0.357	108	4.60	76	0.29	1.00	0.87	35
51A/1	4	76.45	0.20	0.15	120	0.076	247	8.53	146	2.85	1.23	1.00	17
51A/2	4	76.45	0.20	0.37	120	0.088	297	8.95	182	2.65	1.18	1.02	20
56A	8	76.24	0.20	0.15	120	0.050	154	13.60	109	3.62	0.91	0.89	15
57A/1	8	76.24	0.20	0.15	120	0.064	190	11.02	124	3.80	1.22	1.06	16
57A/2	8	76.24	0.20	0.20	120	0.102	211	10.57	124	3.04	1.06	0.92	28
58A	8	38.10	0.20	0.15	120	0.260	72	6.89	62	1.62	1.08	0.95	32
59A	8	56.95	0.20	0.15	120	0.243	149	8.04	112	1.87	1.08	0.97	39
60A	8	56.95	0.20	0.15	120	0.028	77	15.56	62	3.70	0.93	0.84	8
61A	8	76.24	0.20	0.15	120	0.062	131	16.00	162	2.66	0.87	0.82	12
62A	8	56.95	0.20	0.15	120	0.075	103	11.63	67	2.71	1.06	0.93	21
63A	8	37.55	0.20	0.05	120	0.100	44	10.20	33	2.00	0.98	0.87	21
64A	8	37.50	0.20	0.15	120	0.250	64	7.51	48	1.50	1.00	0.88	38
65A	8	56.95	0.20	0.15	120	0.094	124	9.66	95	2.81	1.19	1.05	19

TABLE NO. 21 RESULTS FROM EXPERIMENT TO DETERMINE THE VALUES OF AXIAL FORCE TO CAUSE AXIAL FEED/REV WHEN USING UPPER AND LOWER PLATENS OF EQUAL CONICAL ANGLE DURING UPSETTING ALUMINIUM ALLOY SPECIMENS
 $D_0 = 76.24 \text{ mm}$, $H_0/D_0 \approx 0.2$

Specimen material: Aluminium Alloy BS.1474 HE.30 TF.

* Specific Rotary Forging Force

Specimen No.	θ Degrees	Semi conical angle ($\pi - \theta$)/2 Degrees	D_0 mm	H_0/D_0	F_E kN	f mm/rev	* F_A/F_E
48A	4	88	76.24	0.2	165 219 235	0 0.028 0.055	8.98
51A 50A	4	88	76.24	0.2	126 167 183 247 296	0 0 0.020 0.063 0.088	8.88
56A	8	86	76.24	0.2	128 141 155 311	0 0.013 0.056 0.200	11.58

TABLE 22 RESULTS FROM EXPERIMENT ON SMALL BATCH MANUFACTURE OF COMPONENTS USING CONTOURED UPPER AND LOWER DIES.

BILLET : Commercially pure aluminium
 Diameter : 31.75 mm
 Height : 40.00 mm
 Component dimensions : Figure 68
 Rotational speed : 180 rev/min.

Specimen No.	θ degrees	F_E MN	DIMENSION mm									
			D1	D2	D3	L1	L2	L3	L4	L5	L6	L7
A1	2	0.28	56.67	42.77	29.96	7.49	4.11	11.93	6.41	7.37	2.20	25.71
A2	2	0.28	56.67	42.77	30.00	7.57	4.19	11.86	6.36	7.35	2.23	25.80
A3	2	0.28	56.67	42.77	30.00	7.42	4.42	11.89	6.39	7.29	2.24	25.79
A4	2	0.28	56.67	42.77	29.96	7.48	4.15	11.94	6.39	7.30	2.21	25.78
A5	2	0.28	56.67	42.77	30.00	7.45	4.17	11.86	6.34	7.33	2.24	25.72
A6	2	0.28	56.67	42.77	30.02	7.51	4.20	11.92	6.39	7.31	2.18	25.81
A7	2	0.28	56.67	42.77	29.96	7.40	4.25	11.81	6.30	7.34	2.18	25.73
A8	2	0.28	56.69	42.79	30.02	7.45	4.37	11.94	6.67	7.42	2.21	25.97
A9	2	0.28	56.67	42.79	29.92	7.38	4.37	11.94	6.42	7.69	2.13	25.82
A10	2	0.28	56.69	42.80	29.97	7.45	4.24	11.86	6.39	7.49	2.21	25.76
A11	2	0.28	56.67	42.77	29.88	7.42	4.32	11.96	6.30	7.45	2.21	25.78
A12	2	0.28	56.69	42.80	30.00	7.42	4.19	11.96	6.41	7.43	2.21	25.71
A13	2	0.28	56.67	42.77	30.01	7.42	4.19	11.87	6.46	7.38	2.23	25.76
A14	2	0.28	56.67	42.80	30.02	7.49	4.13	11.96	6.37	7.54	2.18	25.76
B1	4	0.20	56.69	42.77	30.00	7.29	4.17	11.80	7.04	6.31	2.16	25.75
B2	4	0.20	56.67	42.77	30.00	7.19	4.17	12.04	7.05	6.35	2.26	25.67
B3	4	0.20	56.67	42.77	30.02	7.32	4.22	11.86	6.96	6.37	2.40	25.78
B4	4	0.20	56.69	42.74	30.02	7.47	4.17	11.56	7.04	6.10	2.29	25.59
B5	4	0.20	56.69	42.74	30.02	7.34	4.25	12.01	7.09	6.35	2.34	25.77
B6	4	0.20	56.69	42.74	30.02	7.47	4.17	11.91	7.05	6.35	2.33	25.87
B7	4	0.20	56.69	42.74	30.02	7.32	4.11	11.86	7.00	6.37	2.36	25.74
B8	4	0.20	56.69	42.74	30.02	7.32	4.11	11.66	7.00	6.37	2.36	25.74
B9	4	0.20	56.69	42.74	30.02	7.39	4.19	11.81	7.04	6.22	2.34	25.73
B10	4	0.20	56.67	42.75	30.02	7.34	4.14	11.84	7.05	6.46	2.31	25.88

TABLE 22 CONTINUED : RESULTS FROM EXPERIMENT ON SMALL BATCH
MANUFACTURE OF COMPONENTS USING CONTOURED
UPPER AND LOWER DIES.

	D1	D2	D3	DIMENSION mm						
				L1	L2	L3	L4	L5	L6	L7
Range Specimen A	0.02	0.03	0.04	0.15	0.26	0.10	0.37	0.25	0.11	0.26
B	0.02	0.04	0.06	0.18	0.11	0.48	0.66	1.30	0.20	0.28
Mean Specimen A	56.67	42.78	29.98	7.46	4.22	11.91	6.40	7.40	2.21	25.78
B	56.68	42.75	30.00	7.34	4.18	11.86	7.03	6.32	2.32	25.75
Standard Deviation Specimen A	0.01	0.01	0.04	0.05	0.08	0.05	0.09	0.12	0.05	0.07
B	0.01	0.02	0.03	0.08	0.05	0.13	0.04	0.10	0.07	0.10
I.S.O. Tolerance Specimen A	IT.6	IT.7	IT.9	IT.12	IT.14	IT.11	IT.14	IT.13	IT.12	IT.13
B	IT.6	IT.8	IT.9	IT.12	IT.12	IT.14	IT.15	IT.16	IT.14	IT.13
Tool Dimensions	56.69	42.78	30.00	7.50	4.00	12.00	7.00	7.00	2.20	25.70
Tool Dimension minus mean dimension of specimens										
Specimen A	0.02	0.00	0.02	0.04	-0.22	0.09	0.06	-0.40	-0.01	-0.08
B	0.01	0.03	0.00	0.16	-0.18	0.14	-0.03	-0.68	-0.12	-0.05

The maximum eccentricity measured between any two diameters was 0.04 mm.

Typical values of surface roughness
measured after completion of the
forming process : μm C.L.A.

		Specimens A	Specimens B
Surface reference (see figure 68)	No.1	1.0	1.5
	No.2	1.0	1.0
	No.3	0.5	6.5
	No.4	3.0	6.0
	No.5	0.8	0.8
	No.6	3.0	15.0
	No.7	0.4	0.6

TABLE NO. 23 RESULTS FROM TESTS USING ROTARY FORGING FOR THE COMPACTION OF IRON METAL IN CLOSED DIES
TO PRODUCE SOLID CYLINDERS

Powder Metal: Hognas Type ASC.100.29

Nominal Diameter of Billet: 30.00 mm

Angle of Inclination of Platen Axes: 2°

Test No.	Mass of green compact g	Axial Force F_E kN	Mass of sintered compact g	Percentage mass loss during sintering %	Mass after oil impregnation g	Volume cm ³	Average density ρ_E g/cm ³	Average open porosity %	Diameter D mm	Height/Diameter H/D	Relative Density ρ_E/ρ_T	$\ln(1/[1 - (\rho_E/\rho_T)])$
1	86.76	36	86.03	0.84	88.72	13.94	6.16	22.00	30.08	0.66	0.78	1.51
2	87.98	62	87.25	0.84	89.02	13.10	6.66	15.46	30.10	0.62	0.85	1.90
3*	85.50	93	84.85	0.79	85.92	12.05	7.04	10.26	30.10	0.57	0.89	2.21
4*	85.21	121	84.52	0.82	85.30	11.71	7.21	7.62	30.09	0.55	0.92	2.53
5	85.64	160	84.89	0.86	85.60	11.60	7.32	6.93	30.10	0.55	0.93	2.66
6	84.42	198	83.71	0.84	84.38	11.37	7.36	6.70	30.10	0.54	0.94	2.81
7*	84.11	224	83.40	0.84	84.05	11.31	7.37	6.50	30.10	0.53	0.94	2.81
8	85.01	245	84.28	0.86	84.93	11.47	7.35	2.07	30.09	0.54	0.93	2.66
9	124.31	59	123.28	0.84	126.93	19.81	6.22	21.07	30.09	0.93	0.79	1.56
10	121.70	90	120.70	0.82	122.88	17.83	6.77	13.93	30.10	0.84	0.86	1.97
11*	123.08	121	122.11	0.80	123.69	17.39	7.02	10.35	30.06	0.82	0.89	2.21
12*	123.19	160	122.19	0.81	123.40	17.05	7.17	8.12	30.09	0.80	0.91	2.41
13	121.29	203	120.30	0.81	121.41	16.53	7.28	7.63	30.05	0.78	0.93	2.66
14	118.97	245	118.12	0.71	118.76	16.29	7.34	4.45	30.06	0.76	0.93	2.66
15	60.10	46	59.40	0.83	60.38	9.11	6.52	9.39	30.10	0.42	0.85	1.90
16	59.79	98	59.40	0.81	59.57	8.24	7.21	2.28	30.09	0.38	0.92	2.53
17	59.89	157	59.45	0.77	59.63	8.10	7.34	2.53	30.07	0.37	0.93	2.66
18	59.85	201	59.45	0.80	59.60	8.08	7.35	2.11	30.07	0.37	0.93	2.66
19	59.69	247	59.29	0.79	59.40	8.03	7.38	1.65	30.08	0.37	0.94	2.81
20	40.01	36	39.74	0.67	40.44	5.85	6.79	13.69	30.07	0.29	0.86	1.97
21	39.92	85	39.63	0.72	39.86	5.47	7.24	4.69	30.06	0.27	0.92	2.53

TABLE NO. 23 CONTINUED:

Test No.	Mass of green compact g	Axial Force F_E kN	Mass of sintered compact g	Percentage mass loss during sintering %	Mass after oil impregnation g	Volume cm ³	Average density ρ_E g/cm ³	Average open porosity %	Diameter D mm	Height/Diameter H/D	Relative Density ρ_E/ρ_T	$\ln[1/[1 - (\rho_E/\rho_T)]]$
22*	39.92	124	39.58	0.84	39.90	5.41	7.30	6.70	30.08	0.26	0.93	2.66
23*	39.92	160	39.65	0.69	39.95	5.43	7.29	6.40	30.07	0.27	0.92	2.53
24*	40.00	201	39.72	0.99	40.00	5.43	7.31	5.90	30.08	0.27	0.93	2.66
25	40.06	15	39.75	0.78	40.97	6.42	6.19	21.69	30.07	0.30	0.78	1.51
26	40.01	41	39.69	0.81	40.26	5.75	6.89	13.26	30.08	0.27	0.87	2.04
27	39.61	70	39.29	0.82	39.47	5.35	7.33	3.82	30.08	0.25	0.93	2.66
28	39.95	88	39.62	0.82	39.71	5.29	7.48	2.07	30.08	0.25	0.95	3.00
29	39.86	116	39.53	0.83	39.66	5.25	7.53	2.76	30.08	0.25	0.95	3.00
30	59.76	18	59.20	0.94	61.42	10.16	5.82	24.95	30.10	0.48	0.74	1.35
31	59.83	44	59.35	0.81	60.57	9.06	6.54	15.36	30.12	0.42	0.83	1.77
32	59.99	67	59.49	0.83	60.13	8.48	7.01	8.61	30.12	0.40	0.89	2.21
33	59.92	95	59.43	0.81	59.75	8.16	7.28	4.55	30.10	0.38	0.92	2.53
34	59.67	121	59.18	0.82	59.27	7.95	7.43	1.30	30.10	0.37	0.94	2.81
35	30.00	41	29.72	0.91	30.11	4.23	7.02	10.05	30.08	0.20	0.89	2.21
36	29.98	72	29.74	0.83	29.91	4.02	7.39	4.54	30.07	0.19	0.93	2.66
37	29.90	95	29.62	0.91	29.79	3.94	7.51	6.01	30.07	0.18	0.95	3.00
38*	29.97	118	29.71	0.89	30.35	4.50	6.59	16.33	30.08	0.21	0.83	1.77
39	29.98	61	29.61	0.20	29.86	4.03	7.33	6.99	30.04	0.19	0.93	2.66
40	20.00	36	19.80	1.00	20.18	2.91	6.80	14.70	30.04	0.14	0.86	1.97
41	20.01	46	19.85	0.87	20.08	2.78	7.13	9.61	30.04	0.13	0.90	2.30
42	20.00	59	19.83	0.82	20.00	2.68	7.38	7.02	30.05	0.13	0.93	2.66
43	19.94	72	19.76	0.88	19.91	2.63	7.49	6.24	30.60	0.12	0.95	3.00
44*	19.99	98	19.79	0.98	20.02	2.69	7.34	9.40	30.07	0.13	0.93	2.66
45*	19.74	118	19.43	1.50	19.73	2.73	7.09	12.26	30.07	0.13	0.91	2.41
46*	19.64	124	19.35	1.47	19.69	2.83	6.82	13.89	30.08	0.13	0.86	1.97

* From external visual examination a possibility that cracking or segregation occurred.

TABLE 24 RESULTS FROM TESTS ON ROTARY FORGING FOR THE COMPACTION OF IRON POWDER METAL IN CLOSED DIES WITH CO-AXIAL MANDREL TO PRODUCE HOLLOW CYLINDERS

Test No.	Mass of green compact g	Axial Force F_E kN	Mass of sintered compact g	Percentage mass loss during sintering %	Mass after oil impregnation g	Volume cm ³	Average density ρ_E g/cm ³	Average open porosity %	Height/Diameter H/D	Relative Density ρ_E/ρ_T	$\ln(1/[1 - (\rho_E/\rho_T)])$
1	39.83	41	39.50	0.84	40.21	5.83	6.76	13.99	13.16	0.86	1.97
2	39.86	69	39.53	0.83	49.90	5.45	7.24	8.35	13.79	0.92	2.53
3	39.68	93	39.32	0.89	39.59	5.28	7.43	5.72	13.46	0.94	2.81
4	39.85	121	39.50	0.87	39.81	5.34	7.39	6.54	13.60	0.93	2.66
5	49.76	38	49.32	0.88	50.42	7.51	6.56	16.78	19.05	0.83	1.77
6	49.75	67	49.33	0.84	49.87	6.89	7.15	8.95	17.40	0.89	2.21
7	49.71	95	49.28	0.86	49.64	6.74	7.30	6.13	17.07	0.92	2.53
8	49.55	111	49.13	0.84	49.50	6.67	7.36	6.34	16.90	0.93	2.66
9	49.82	41	49.41	0.82	50.48	7.58	6.51	16.12	12.40	0.82	1.71
10	49.88	70	49.47	0.82	49.97	6.86	7.20	11.85	11.77	0.91	2.41
11	49.90	70	49.48	0.83	50.03	6.92	7.14	9.09	11.86	0.90	2.30
12	49.92	40	49.51	0.82	50.46	7.29	6.78	16.00	13.48	0.86	1.97
13	49.95	72	49.52	0.87	50.06	6.89	7.18	9.01	12.34	0.91	2.41
14	49.23	45	44.82	0.90	49.72	6.73	6.66	15.32	11.96	0.84	1.83
15	49.95	70	49.55	0.80	50.01	6.84	7.24	7.65	12.21	0.92	2.53
16	49.99	93	49.59	0.81	50.01	6.79	7.29	7.08	12.12	0.92	2.53
17	49.85	100	49.45	0.80	49.88	6.79	7.28	5.57	12.14	0.92	2.53
18	49.92	122	49.48	0.87	49.88	6.75	7.32	6.68	12.14	0.93	3.00
19	49.93	120	49.52	0.81	49.88	6.74	7.34	6.11	12.12	0.93	2.66
20	49.98	124	49.55	0.86	49.90	6.74	7.34	6.33	12.09	0.93	2.66
21	60.02	40	59.48	0.90	60.78	9.00	6.60	16.55	16.15	0.83	1.77
22	59.94	71	59.45	0.82	60.05	8.29	7.16	8.32	14.55	0.91	2.41
23	59.91	103	59.39	0.87	59.90	8.11	7.31	7.20	14.42	0.92	2.53
24	59.87	122	59.33	0.90	59.87	8.12	7.30	7.67	14.55	0.92	2.53
25	53.78	125	53.38	0.73	53.76	7.25	7.36	5.95	5.92	0.93	2.66

TABLE NO.25 DIMENSIONS OF EXTERNAL AND INTERNAL DIAMETERS
DETERMINED FROM HOLLOW CYLINDRICAL SPECIMENS

Test No.	Relative density ρ_E/ρ_T	Mean diameter of Hollow die mm	Mean diameter of Mandrel mm	External diameter		Bore diameter	
				Maximum mm	Minimum mm	Maximum mm	Minimum mm
1	0.86	30.01	19.93	30.08	30.06	19.96	19.95
2	0.92	30.01	19.93	30.07	30.06	19.94	19.99
3	0.94	30.01	19.93	30.07	30.05	19.96	19.95
4	0.93	30.01	19.93	30.06	30.05	19.95	19.93
5	0.83	30.01	19.93	30.12	30.09	19.94	19.92
6	0.89	30.01	19.93	30.08	30.07	19.95	19.94
7	0.92	30.01	19.93	30.06	30.04	19.94	19.94
8	0.93	30.01	19.93	30.08	30.05	19.94	19.93
9	0.82	30.01	13.97	30.07	30.04	13.94	13.94
10	0.91	30.01	13.97	30.08	30.04	13.92	13.91
11	0.90	30.01	13.97	30.07	30.04	13.94	13.94
12	0.86	30.01	13.97	30.07	30.04	13.94	13.94
13	0.91	30.01	13.97	30.06	30.05	13.96	13.96
14	0.84	30.01	13.97	30.04	30.03	13.97	13.97
15	0.92	30.01	13.97	30.07	30.06	13.97	13.97
16	0.92	30.01	13.97	30.06	30.05	13.97	13.97
17	0.92	30.01	13.97	30.07	30.05	13.97	13.97
18	0.93	30.01	13.97	30.06	30.05	13.96	13.96
19	0.93	30.01	13.97	30.06	30.05	13.96	13.96
20	0.93	30.01	13.97	30.05	30.05	13.97	13.97
21	0.83	30.01	13.97	30.06	30.05	13.97	13.97
22	0.91	30.01	13.97	30.07	30.07	13.97	13.97
23	0.92	30.01	13.97	30.06	30.06	13.96	13.96
24	0.92	30.01	13.97	30.06	30.05	13.97	13.98
25	0.93	30.01	13.97	30.06	30.04	13.97	13.97

Nominal bore diameter	20 mm		14 mm	
	Mean External diameter	Mean Bore diameter	Mean External diameter	Mean Bore diameter
RANGE mm	00.040	00.030	00.030	00.050
MEAN mm	30.060	19.950	30.060	13.960
STANDARD DEVIATION mm	00.020	00.020	00.007	00.016
INTERNATIONAL STANDARD TOLERANCE	IT.8	IT.8	IT.8	IT.10
TYPICAL SURFACE ROUGHNESS μm C.L.A.	1.000	1.000	1.000	1.000

Eccentricity: The eccentricity of internal diameter to external diameter for all specimens was found to be ≥ 0.03 mm, ≤ 0.06

TABLE NO. 26 RESULTS FROM TESTS TO CAUSE POST SINTERED DENSIFICATION OF SOLID CYLINDRICAL IRON COMPACTS
BY BOTH WARM AND COLD FORGING IN CLOSED DIES *Results from warm forging shown in italics*

Specimen from Test No.	Mass g	Height/ Diameter H/D	Average density g/cm ³		Average porosity %		Relative density ρ_E/ρ_T		$\ln(1/[1 - (\rho_E/\rho_T)])$	
			Sintered	After Forging	Sintered	After Forging	Sintered	After Forging	Sintered	After Forging
1	86.03	0.66	6.16	7.21	22.00	0.77	0.78	0.92	1.51	2.53
2	87.25	0.62	6.66	7.34	15.46	0.54	0.85	0.93	1.90	2.66
3	84.85	0.57	7.04	7.44	10.26	0.52	0.89	0.95	2.21	3.00
4	84.52	0.55	7.21	7.54	7.62	0.20	0.92	0.96	2.53	3.22
5	84.89	0.55	7.32	7.59	6.93	0.15	0.93	0.96	2.66	3.22
6	83.71	0.54	7.36	7.62	6.70	0.97	0.94	0.97	2.81	3.51
7	83.40	0.53	7.37	7.68	6.50	0.98	0.94	0.97	2.81	3.51
10	120.70	0.84	6.77	7.19	13.93	0.52	0.86	0.91	1.97	2.41
12	122.19	0.80	7.17	7.40	8.12	0.50	0.91	0.93	2.41	2.66
13	120.30	0.78	7.28	7.47	7.63	0.36	0.93	0.95	2.66	3.00
15	59.40	0.42	6.52	7.60	9.39	1.60	0.85	0.96	1.90	3.22
16	59.40	0.38	7.21	7.63	2.28	2.30	0.92	0.97	2.53	3.51
17	59.45	0.37	7.34	7.70	2.53	1.30	0.93	0.98	2.66	3.91
18	59.45	0.37	7.35	7.73	2.11	1.70	0.93	0.98	2.66	3.91
19	59.29	0.37	7.38	7.80	1.65	1.00	0.94	0.99	2.81	4.61
20	39.74	0.28	6.79	7.40	13.69	2.20	0.86	0.94	1.97	2.81
21	39.86	0.27	7.24	7.49	4.69	4.60	0.92	0.95	2.53	3.00
22	39.58	0.26	7.30	7.75	6.70	2.50	0.93	0.97	2.66	3.51
23	39.65	0.27	7.29	7.69	6.40	2.40	0.92	0.97	2.53	3.51
24	39.72	0.27	7.31	7.78	5.90	2.90	0.93	0.99	2.66	4.61
31	59.35	0.42	6.54	7.42	15.36	1.86	0.83	0.94	1.77	2.81
32	59.49	0.40	7.01	7.48	8.61	1.07	0.89	0.95	2.21	3.00
33	59.43	0.38	7.28	7.59	4.55	0.30	0.92	0.96	2.53	3.22
34	59.18	0.37	7.43	7.76	1.30	0.30	0.94	0.99	2.81	4.61
35	29.72	0.20	7.02	7.63	10.05	0.90	0.89	0.97	2.21	3.51
37	29.62	0.18	7.51	7.63	6.01	1.50	0.96	0.97	3.22	3.51
38	29.71	0.21	6.59	7.49	16.33	2.30	0.83	0.95	1.77	3.00
39	29.61	0.19	7.33	7.68	6.99	1.50	0.93	0.97	2.66	3.51
40	19.80	0.14	6.80	7.69	14.70	2.10	0.86	0.97	1.97	3.51
41	19.85	0.13	7.13	7.60	9.61	2.10	0.90	0.96	2.30	3.22
42	19.83	0.13	7.32	7.38	7.02	3.60	0.93	0.94	2.66	2.81
43	19.76	0.12	7.49	7.71	6.24	2.20	0.95	0.98	3.00	3.91
44	19.79	0.13	7.34	7.64	9.40	3.10	0.93	0.97	2.66	3.51

TABLE NO.27 THE RELATIVE AVERAGE DENSITIES OF SOLID CYLINDRICAL SPECIMENS AND ELEMENTS CUT FROM THE SPECIMENS

Specimen details : Sintered iron powder metal HOGANAS ASC.100/29 0.8% zinc stearate
30.00 mm diameter H/D ratio 0.66, $F_E = 250$ kN, $\theta = 2^\circ$, 120 rev/min.

Post sintered cylindrical
specimens before slicing
H/D = 0.66

Specimen No.48

0.92

Specimen No.49

0.92

Specimen No.50

0.92

Post sintered specimens
sliced into four discs
H/D = 0.14

0.94
0.94
0.90
0.85

0.92
0.93
0.91
0.88

0.93
0.94
0.90
0.85

Discs and annular rings
cut from discs above

	0.93	0.93
	0.94	0.93
	0.90	0.90
	0.87	0.84

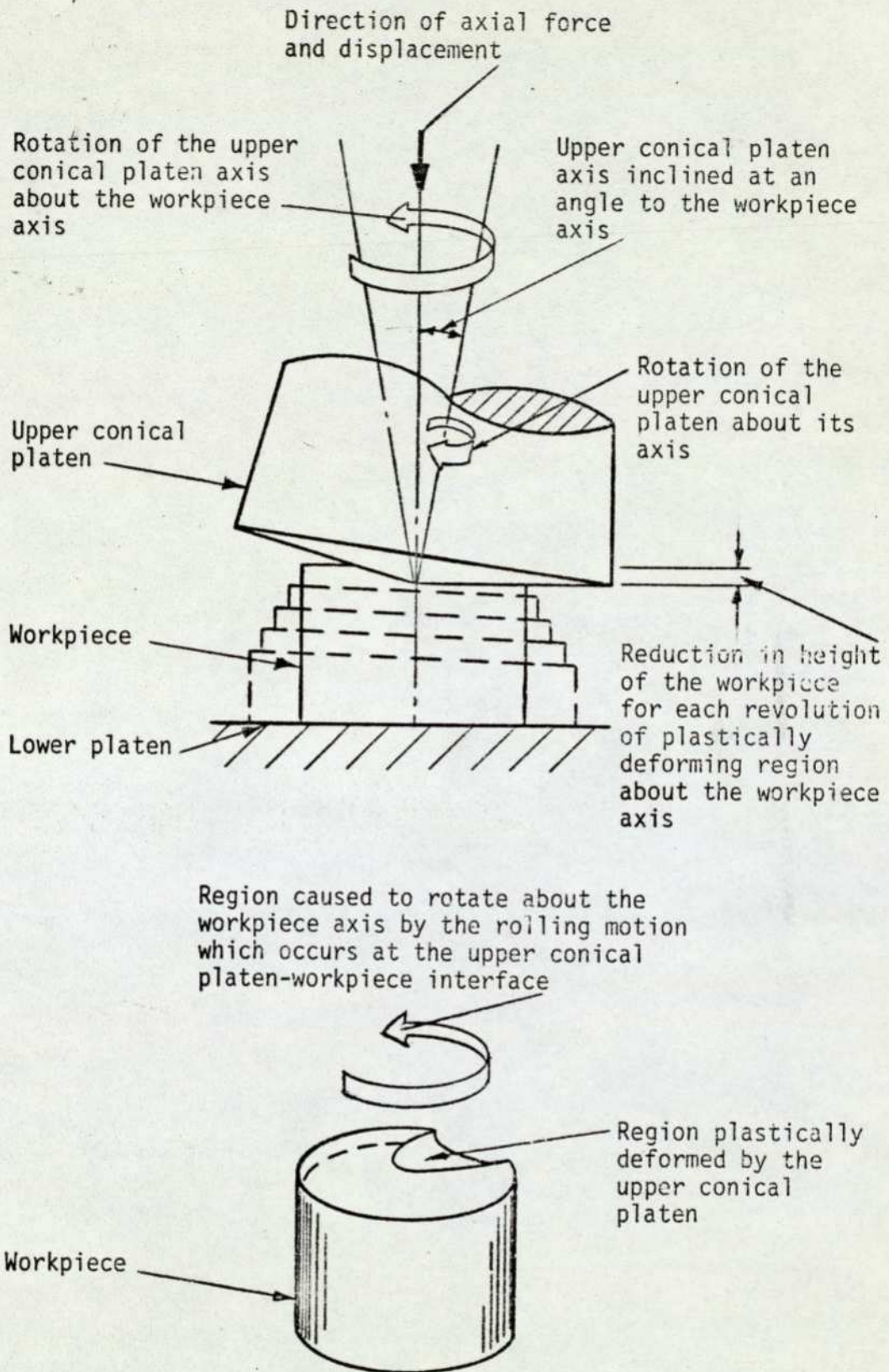
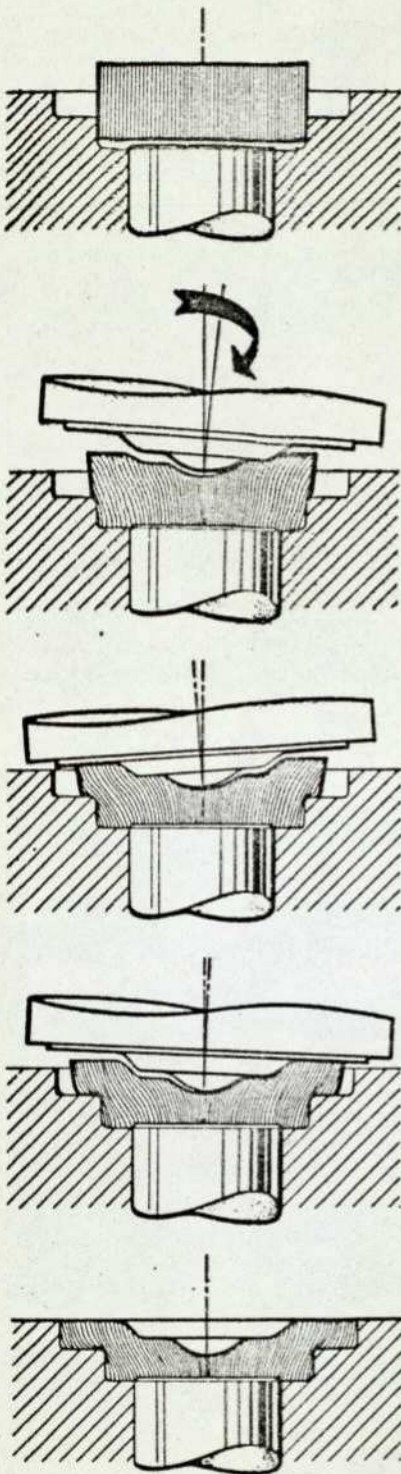
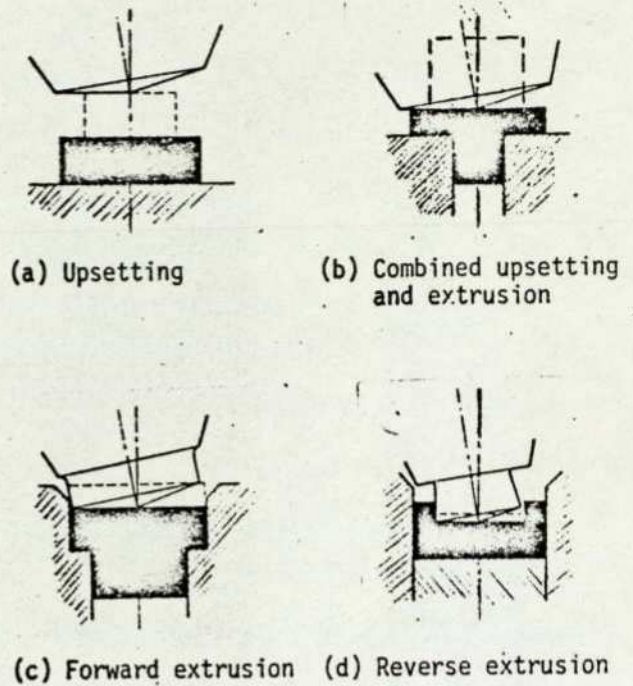


FIGURE 1 THE BASIC PRINCIPLES OF THE ROTARY FORGING PROCESS



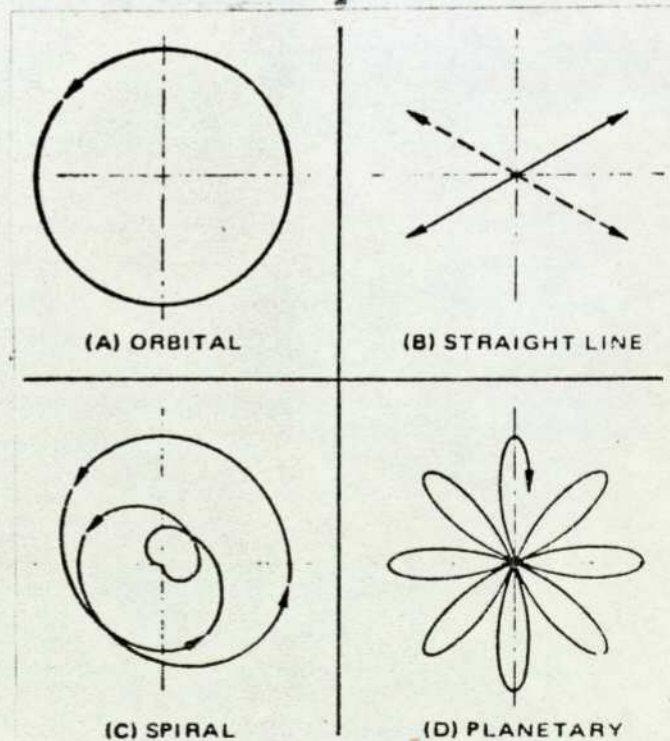
After Williams [15]

FIGURE 2 TYPICAL STAGES OF THE ROTARY FORGING PROCESS USING CONTOURED UPPER AND LOWER DIES.



After Dredger [17]

FIGURE 3 TYPICAL OPERATIONS WHICH CAN BE CARRIED OUT BY ROTARY FORGING.



After Dredger [17]

FIGURE 4 DIAGRAM OF RELATIVE MOTIONS GENERATED AT THE UPPER DIE-WORKPIECE INTERFACE.

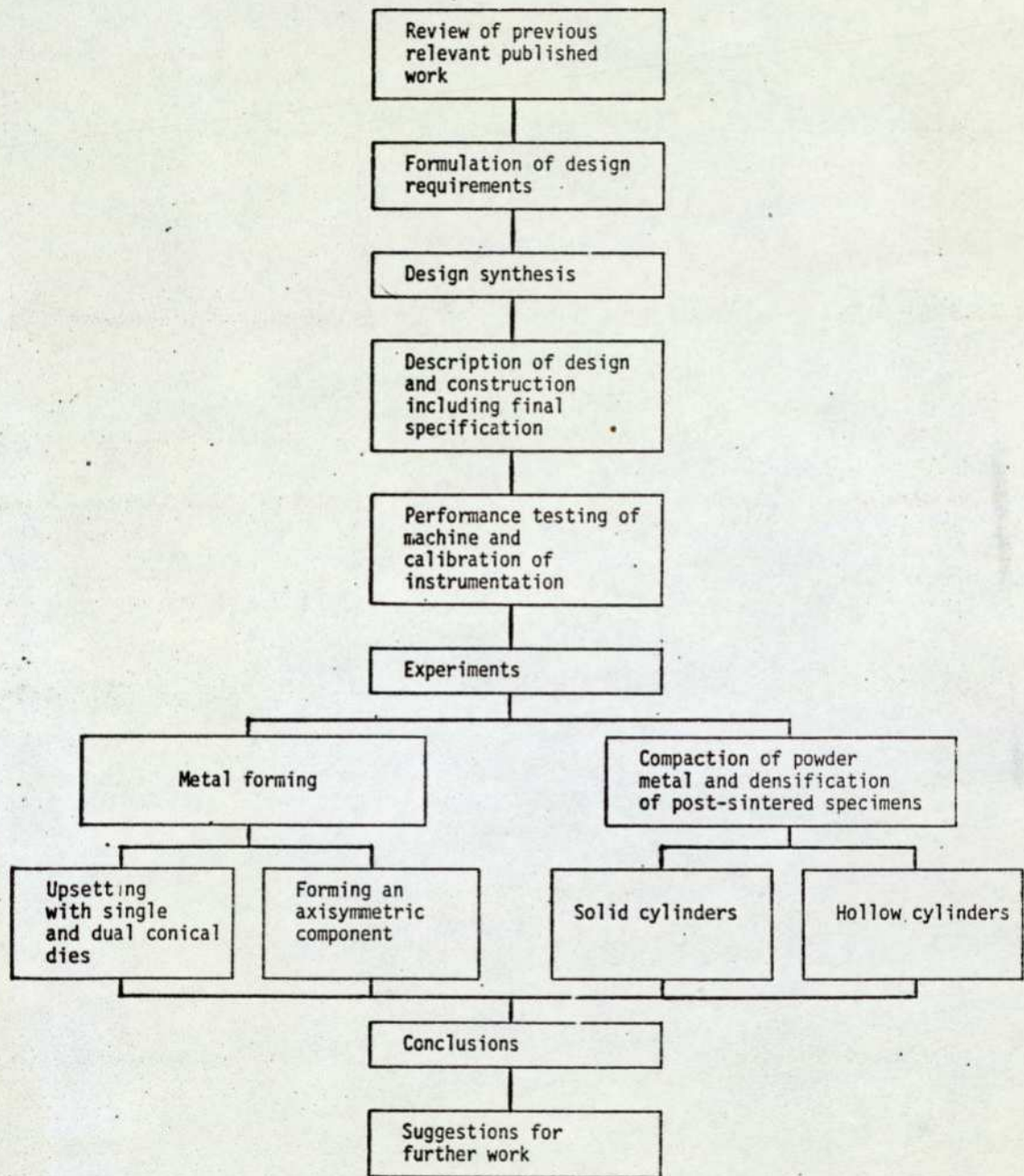


FIGURE 5 FLOW DIAGRAM OF RESEARCH WORK

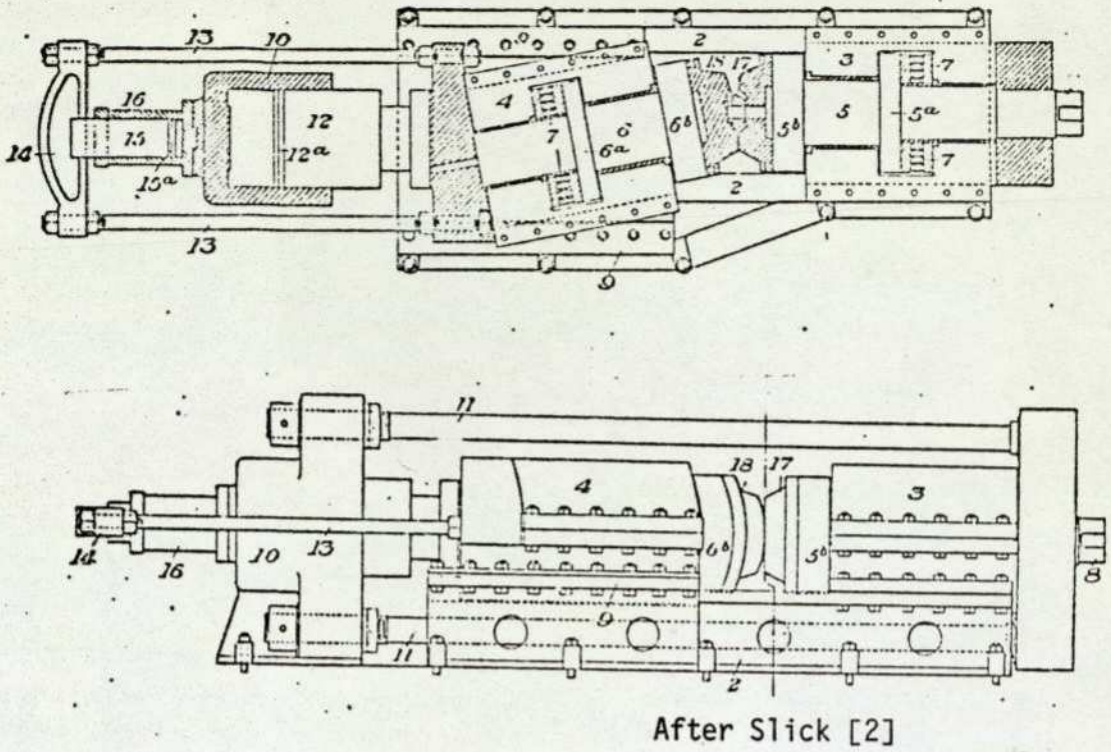


FIGURE 6 THE SLICK MILL

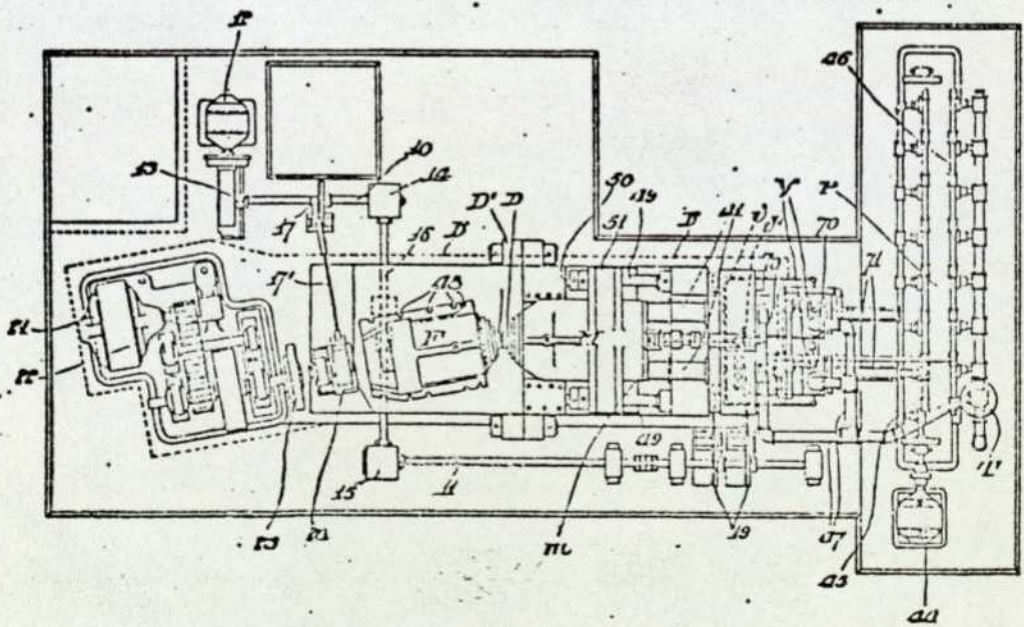
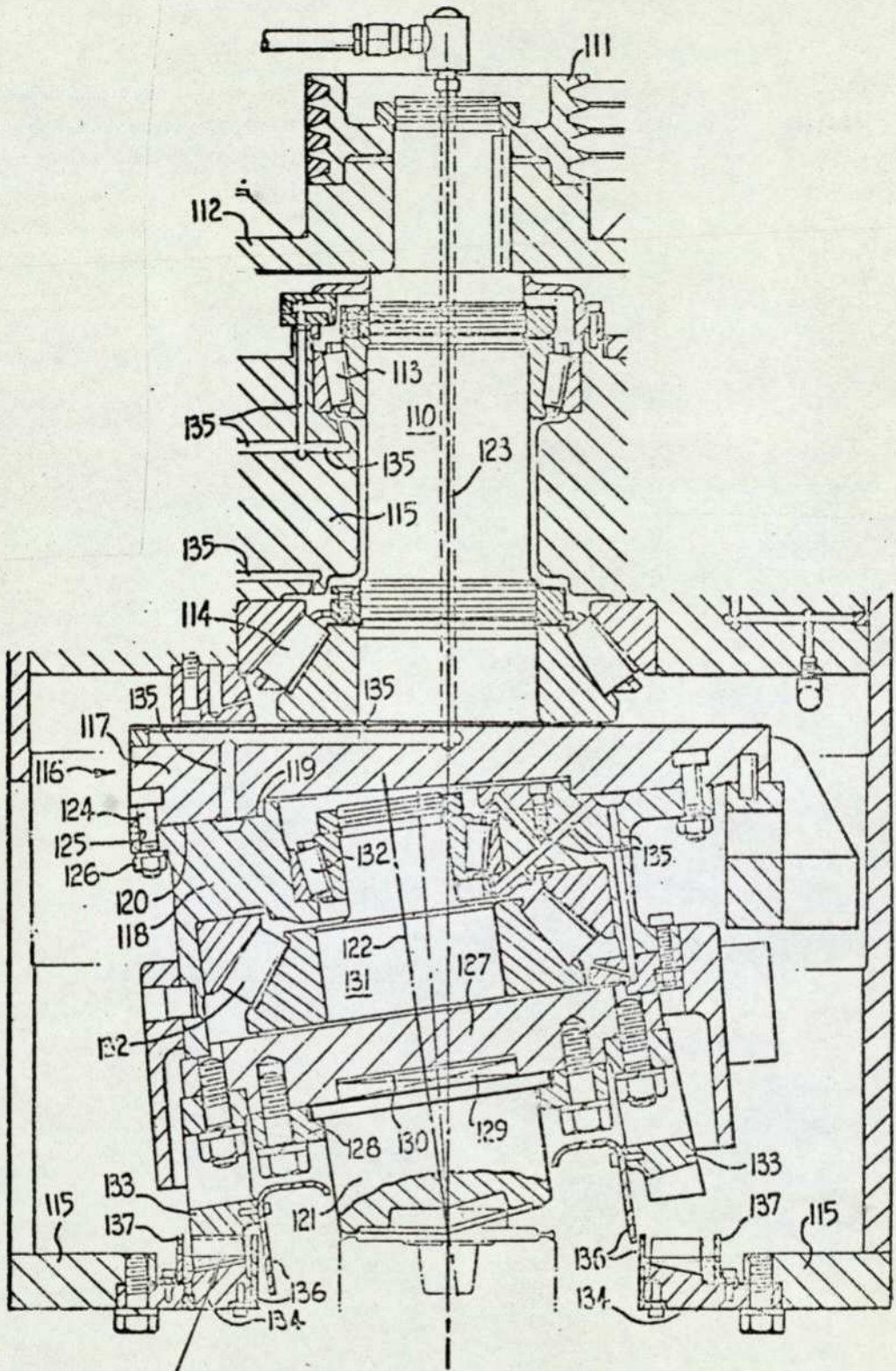


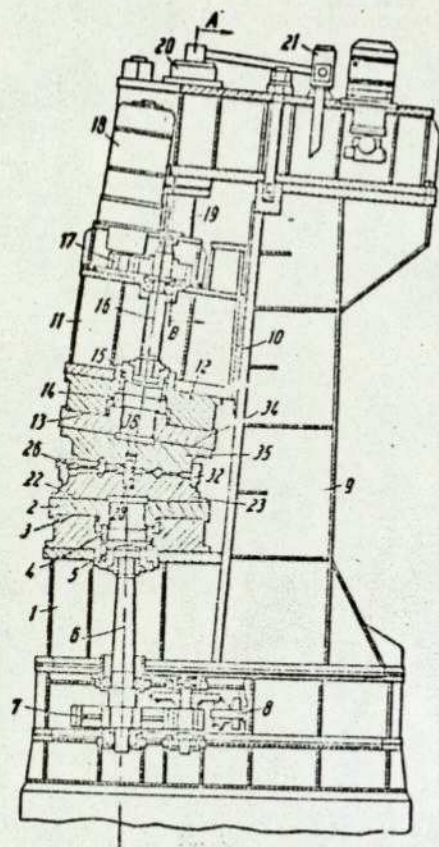
FIGURE 7 THE MACHINE DESIGNED BY HUGHES [4]



Bevel gear

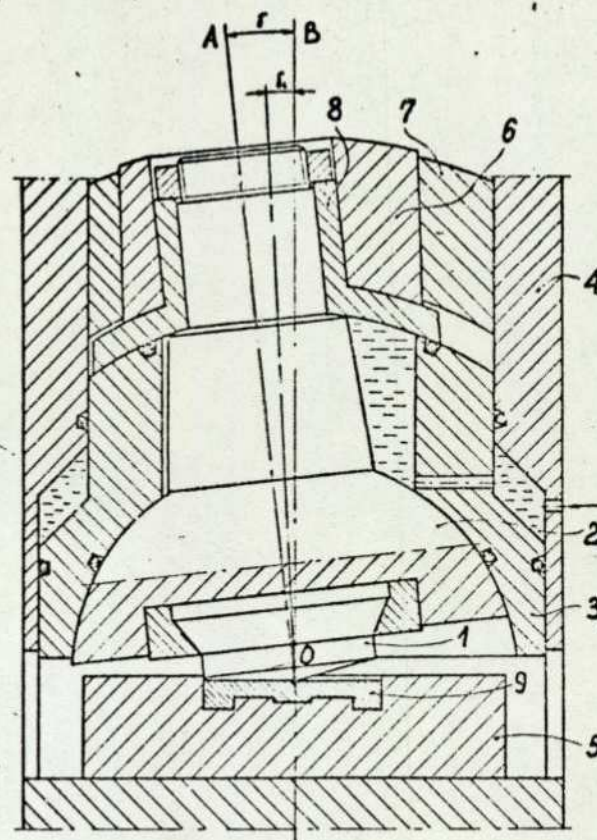
After B & S Massey Ltd. [10]

FIGURE 8 ARRANGEMENT OF THE UPPER PLATEN ASSEMBLY OF A MACHINE DESIGNED BY B & S MASSEY LIMITED SHOWING THE BEVEL GEAR DRIVE.



After Ulrych [14]

FIGURE 9 MACHINE INVENTED BY ULRYCH



After Marciniak [28]

FIGURE 10 KINEMATIC ARRANGEMENT OF THE MACHINE INVENTED BY MARCINIAK TO ACHIEVE PLANETARY MOTION.

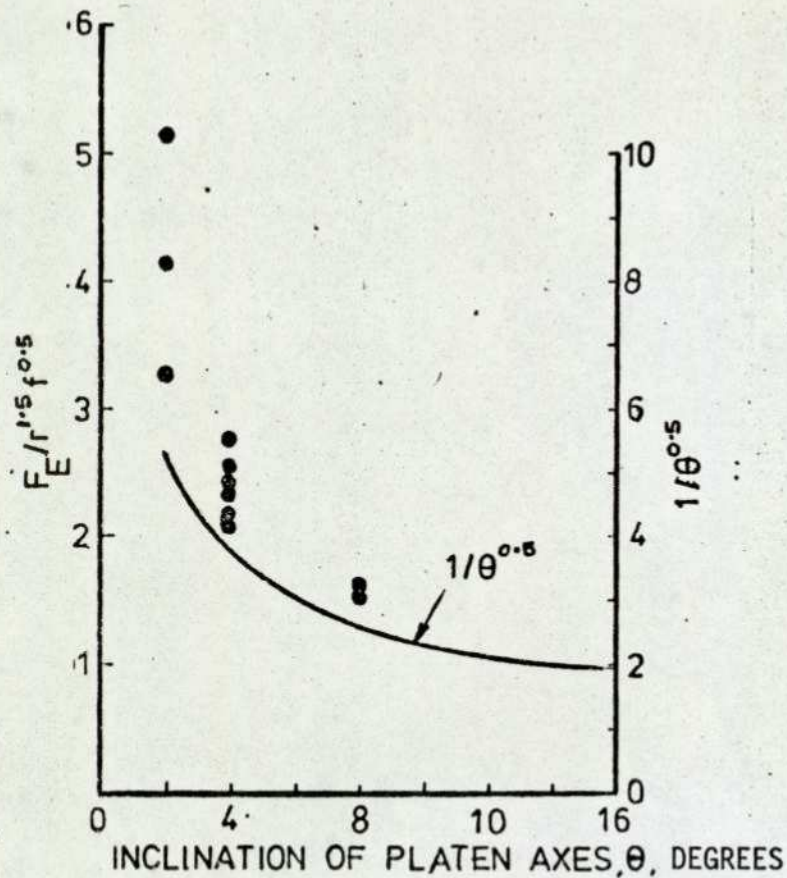


FIGURE 11 $F_E / r^{1.5} f^{0.5}$ - INCLINATION OF PLATEN AXES, θ , .
RELATION CALCULATED FROM DATA ACQUIRED BY
HAWKYARD ET AL [24] DURING EXPERIMENTS ON
UPSETTING CYLINDRICAL ALUMINIUM ALLOY
SPECIMENS.

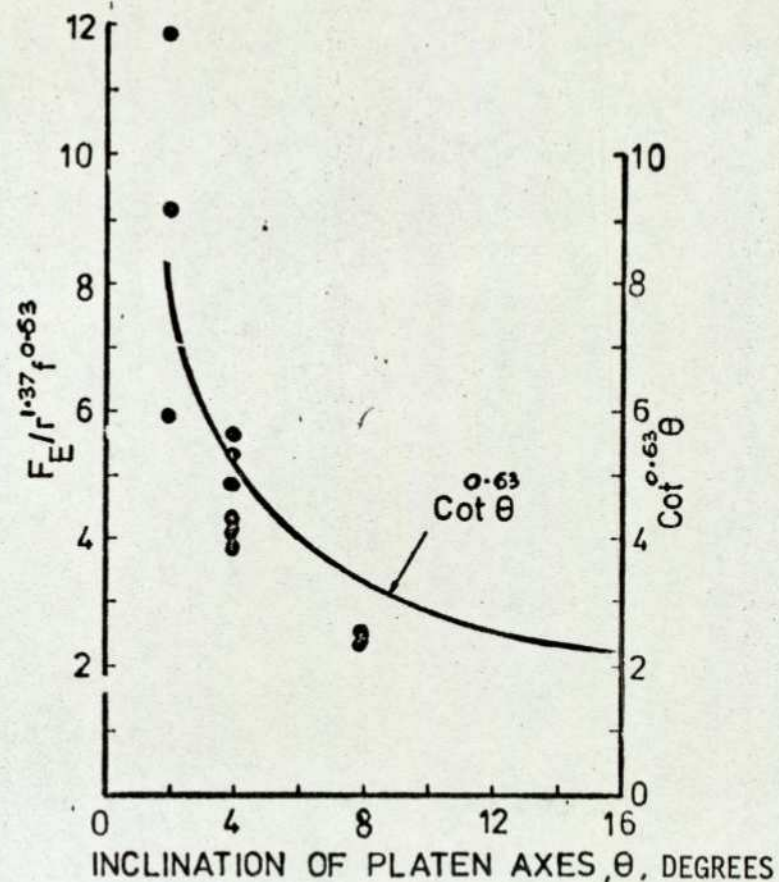


FIGURE 12 $F_E / r^{1.37} f^{0.63}$ - INCLINATION OF PLATEN AXES, θ
RELATION CALCULATED FROM DATA ACQUIRED BY
HAWKYARD ET AL [24] DURING EXPERIMENTS ON
UPSETTING CYLINDRICAL ALUMINIUM ALLOY
SPECIMENS.

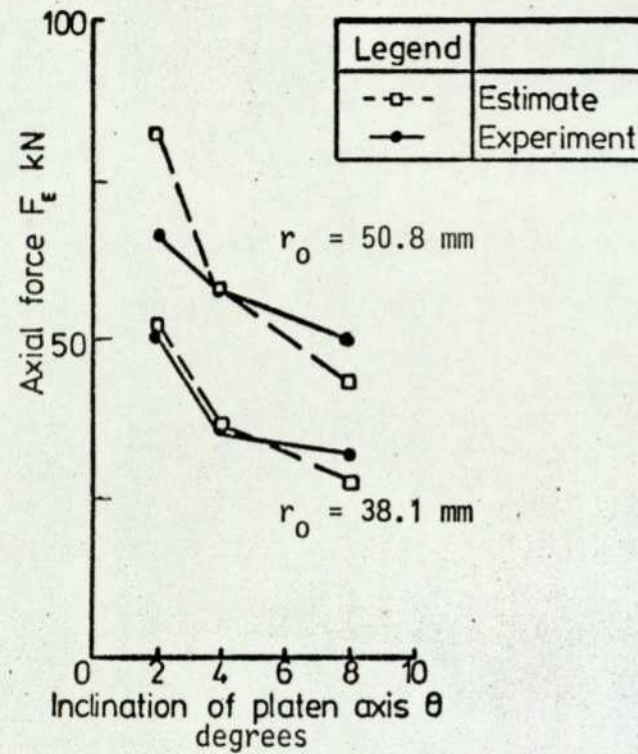


FIGURE 13 PREDICTED AND EXPERIMENTAL VALUES OF AXIAL FORCE - INCLINATION OF PLATEN AXES, θ , RELATION PRESENTED BY HAWKYARD ET AL [24].

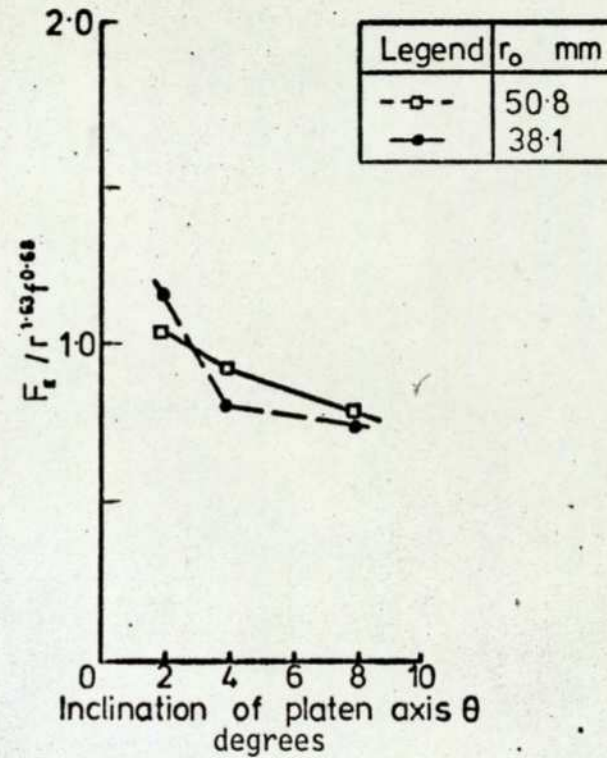


FIGURE 14 $F_E / r^{1.37} f^{0.63}$ - INCLINATION OF PLATEN AXES, θ , RELATION CALCULATED FROM DATA ACQUIRED BY HAWKYARD ET AL [24] DURING EXPERIMENTS ON UPSETTING CYLINDRICAL COMMERCIAL PURE ALUMINIUM SPECIMENS.

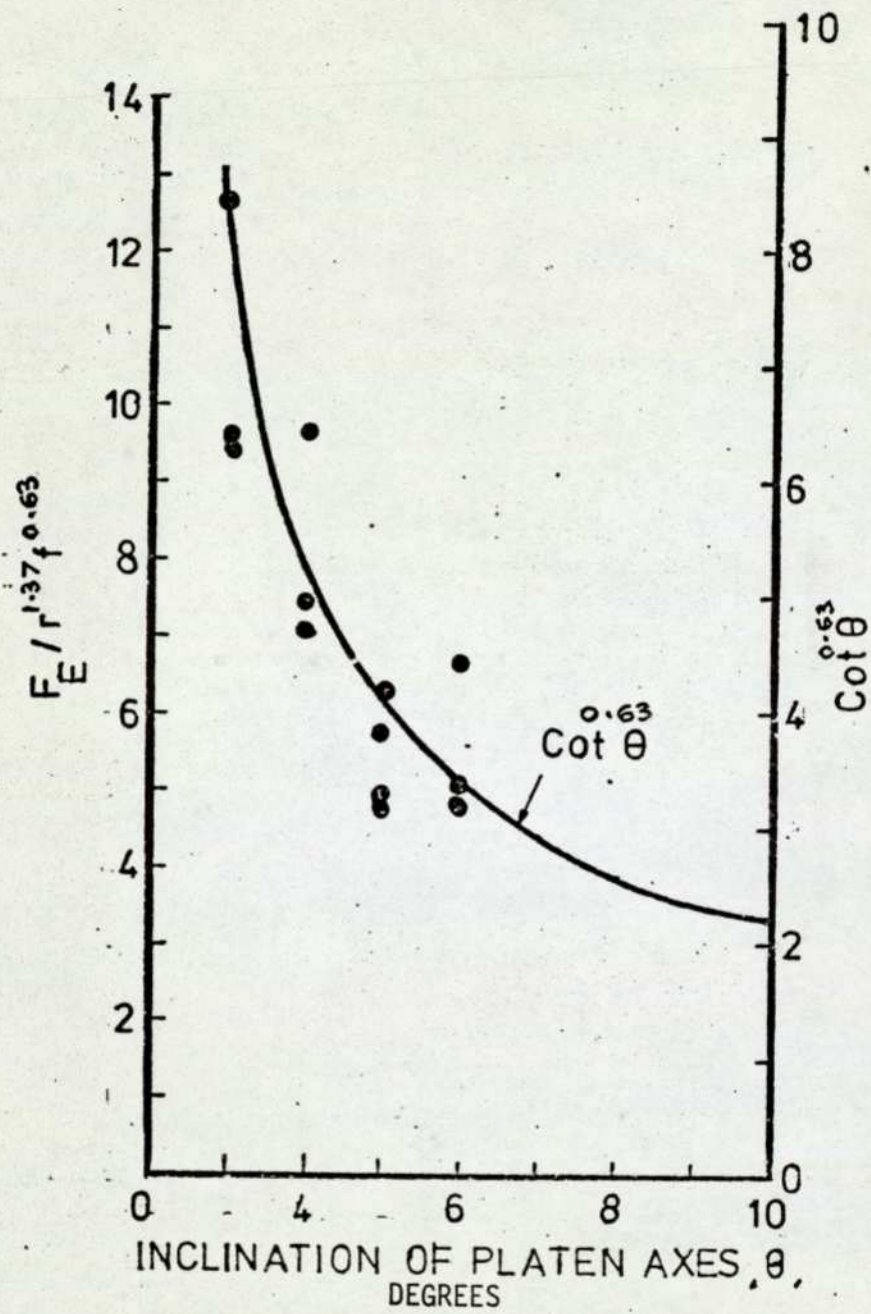


FIGURE 15 $F_E / r^{1.37} f^{0.63}$ - INCLINATION OF PLATEN AXES, θ , RELATION CALCULATED FROM DATA ACQUIRED BY PRABHU ET AL [59] DURING EXPERIMENTS ON ORBITAL RIVETTING.

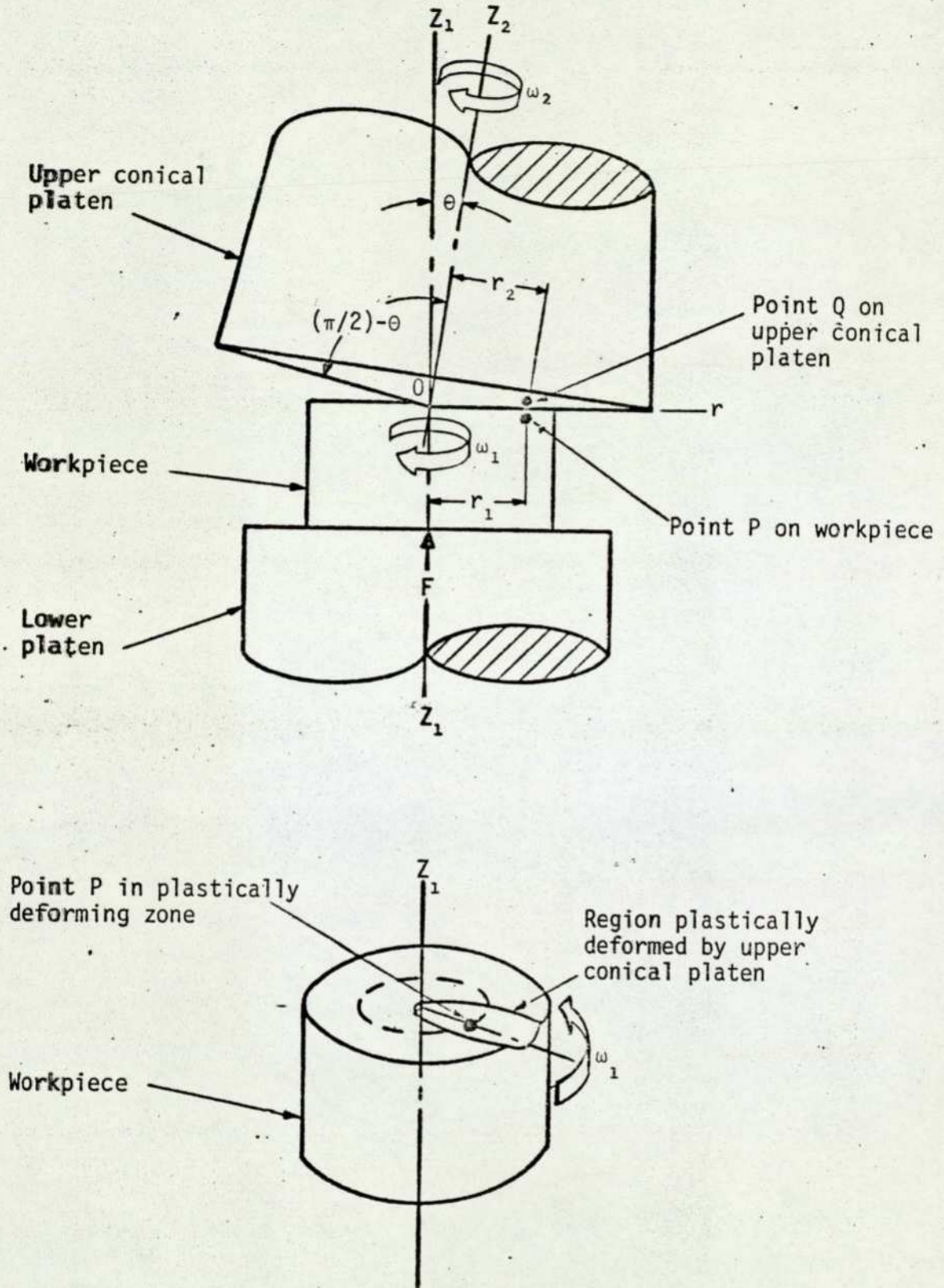


FIGURE 16 FUNDAMENTAL KINEMATICS OF THE ROTARY FORGING PROCESS

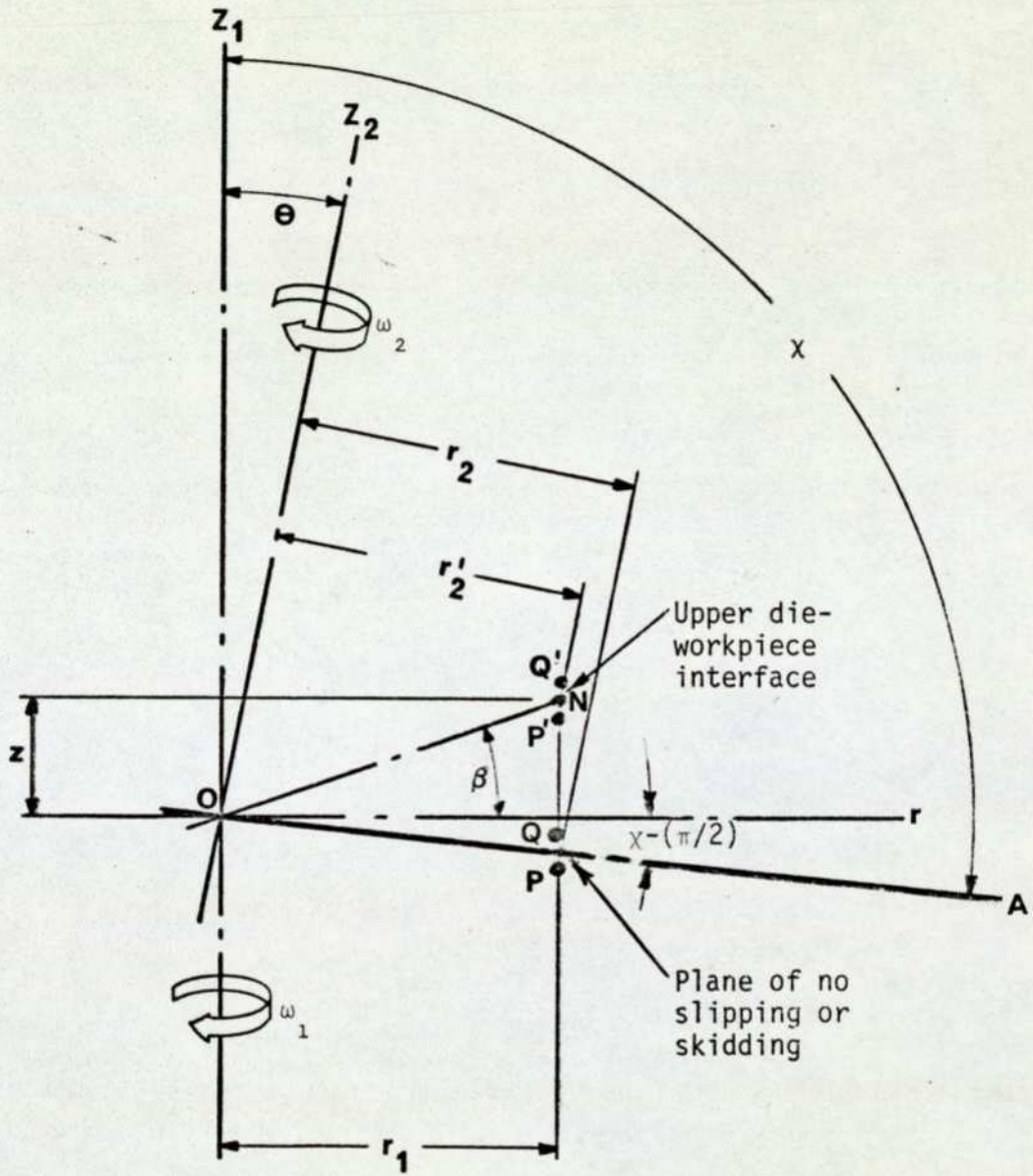
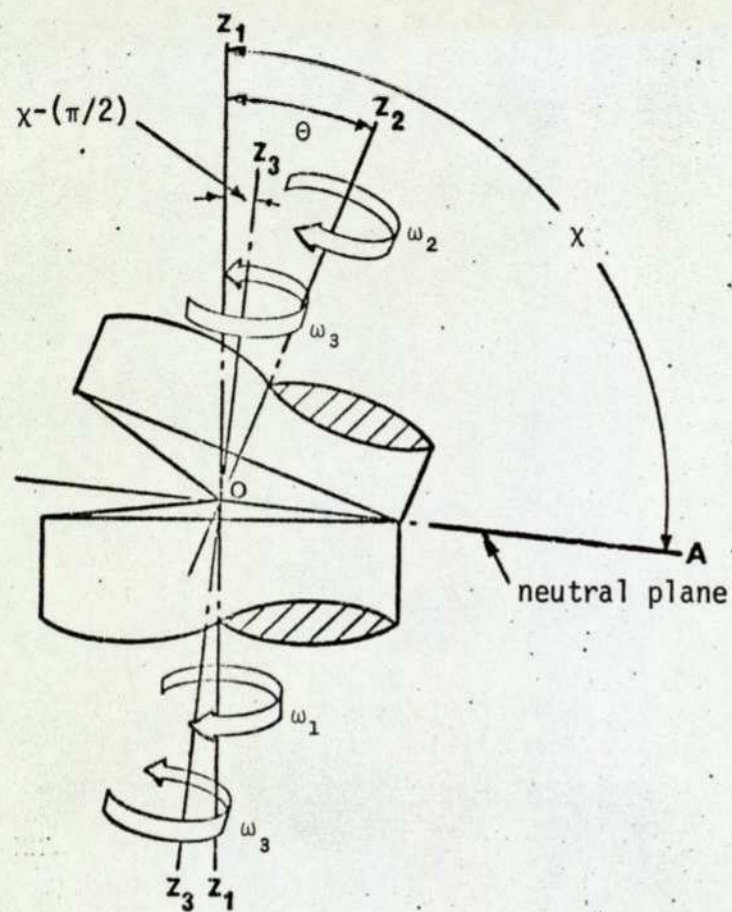
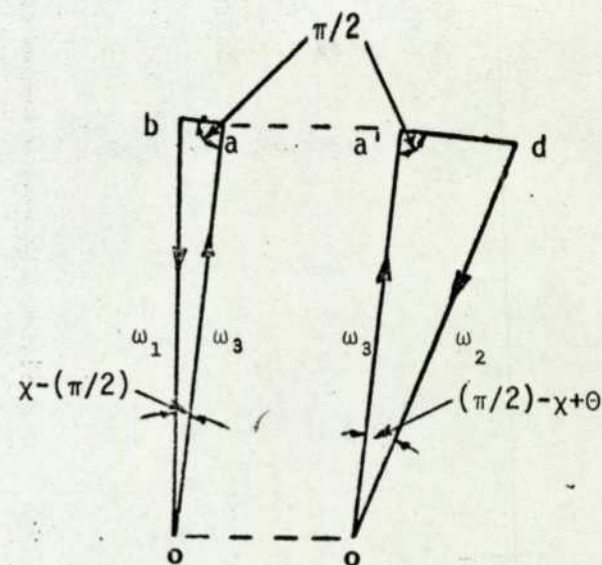


FIGURE 17 DIAGRAM OF UPPER DIE-WORKPIECE INTERFACE



(a) Physical diagram



(b) Vector diagram of angular velocities

FIGURE 18 DIAGRAMS OF METHOD TO DETERMINE THE RESPECTIVE ANGULAR VELOCITIES FOR A GIVEN POSITION OF THE NEUTRAL PLANE AND INCLINATION OF PLATEN AXES.

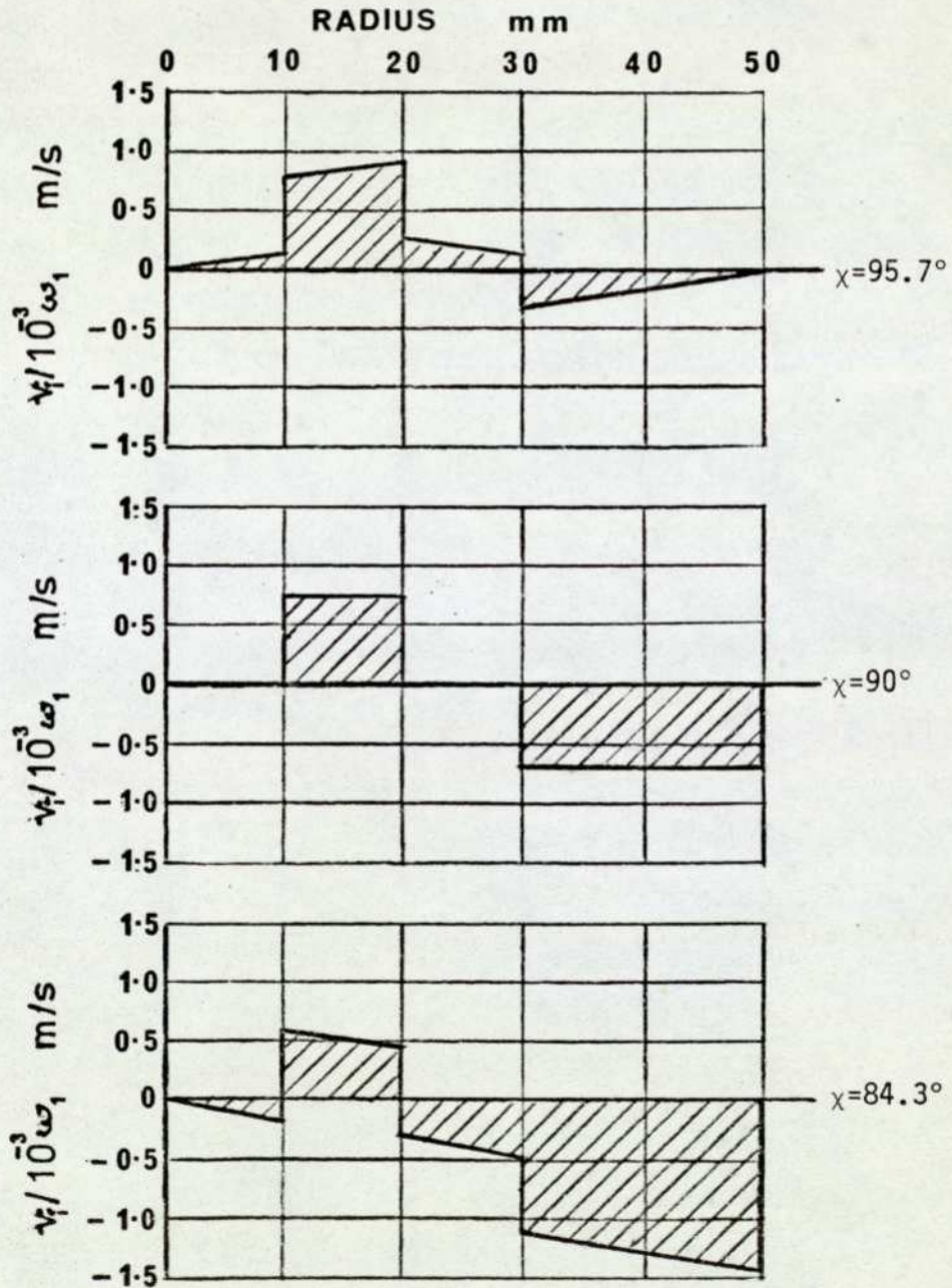
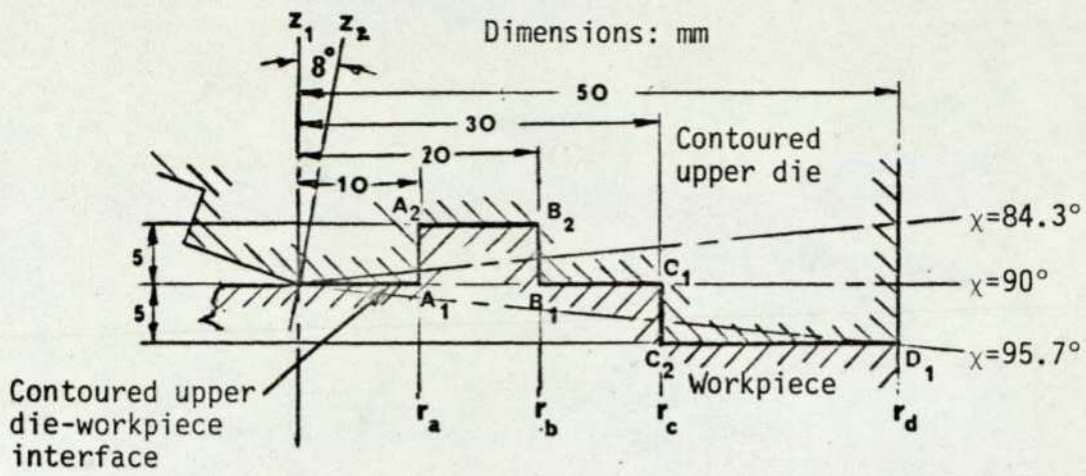


FIGURE 19 TYPICAL CONTOURED UPPER DIE-WORKPIECE INTERFACE VELOCITY RELATION FOR $\theta = 8^\circ$ AND VARIOUS VALUES OF χ .

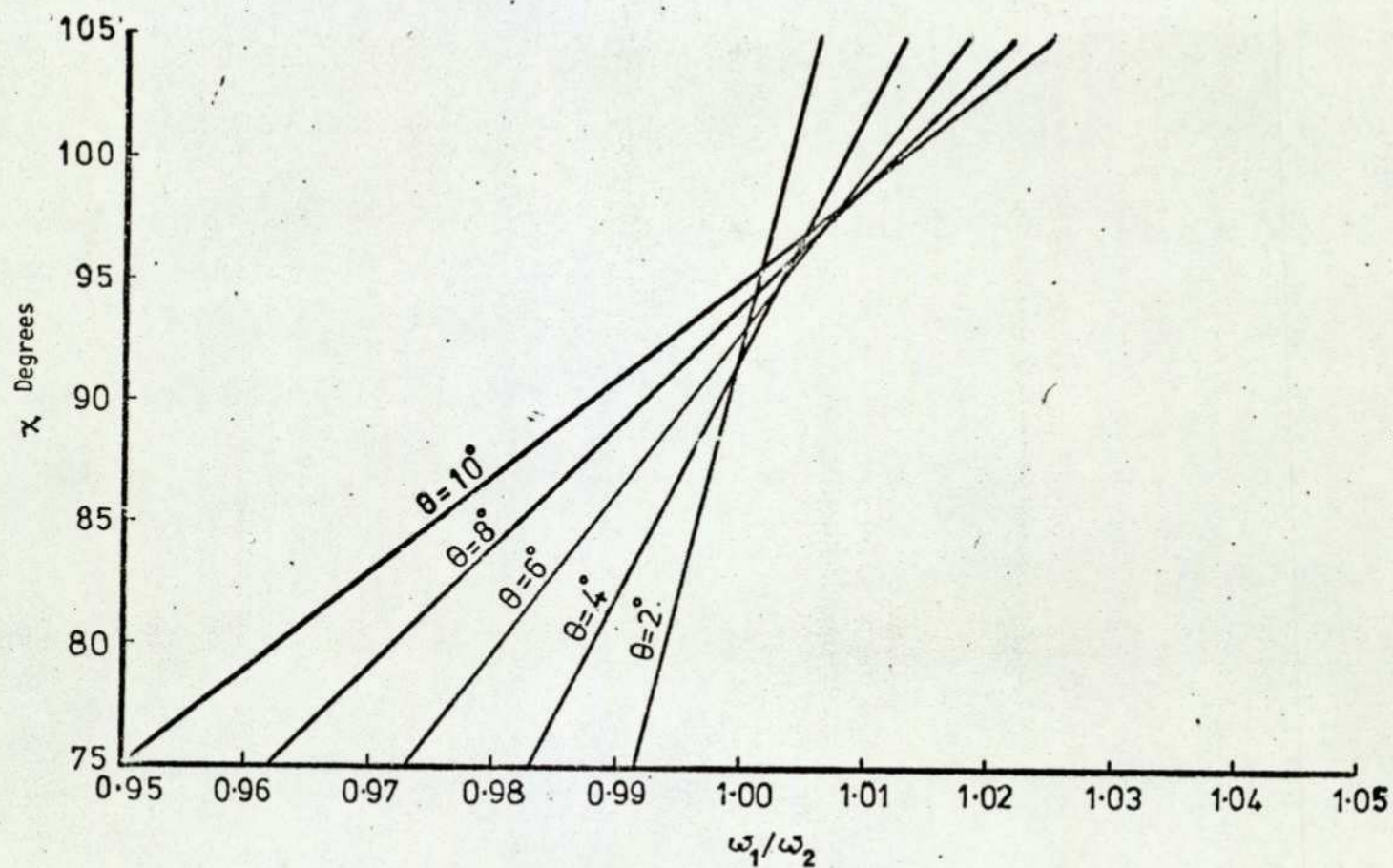
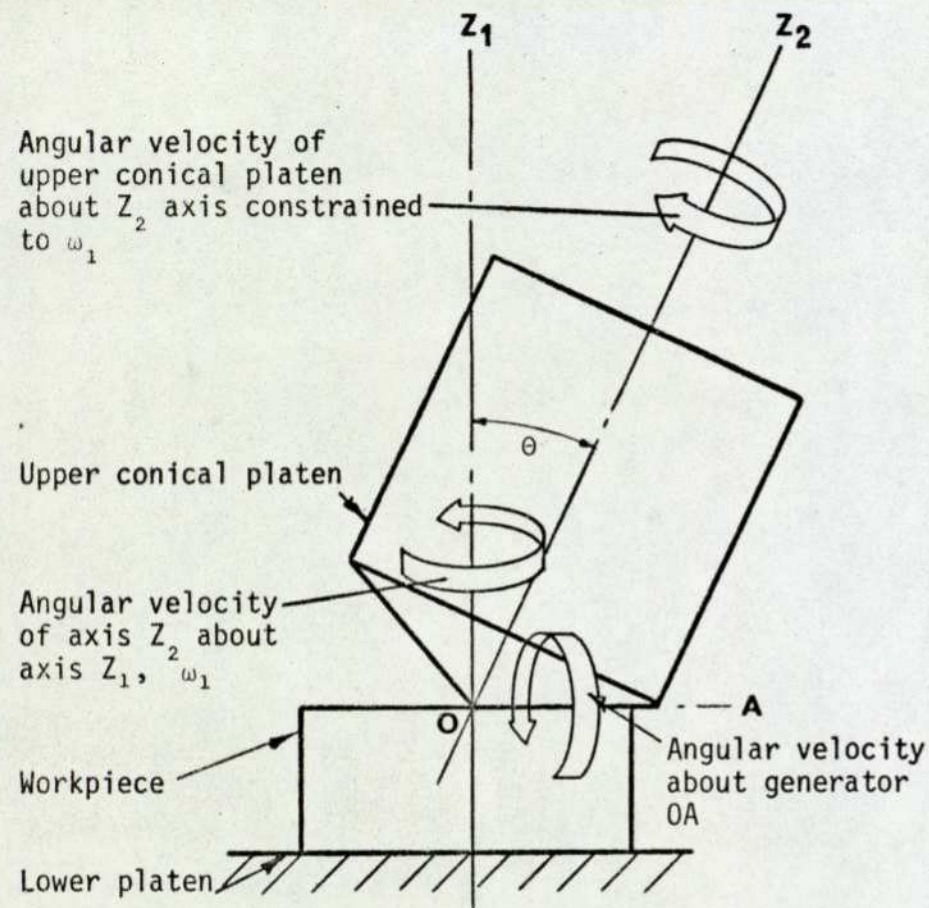
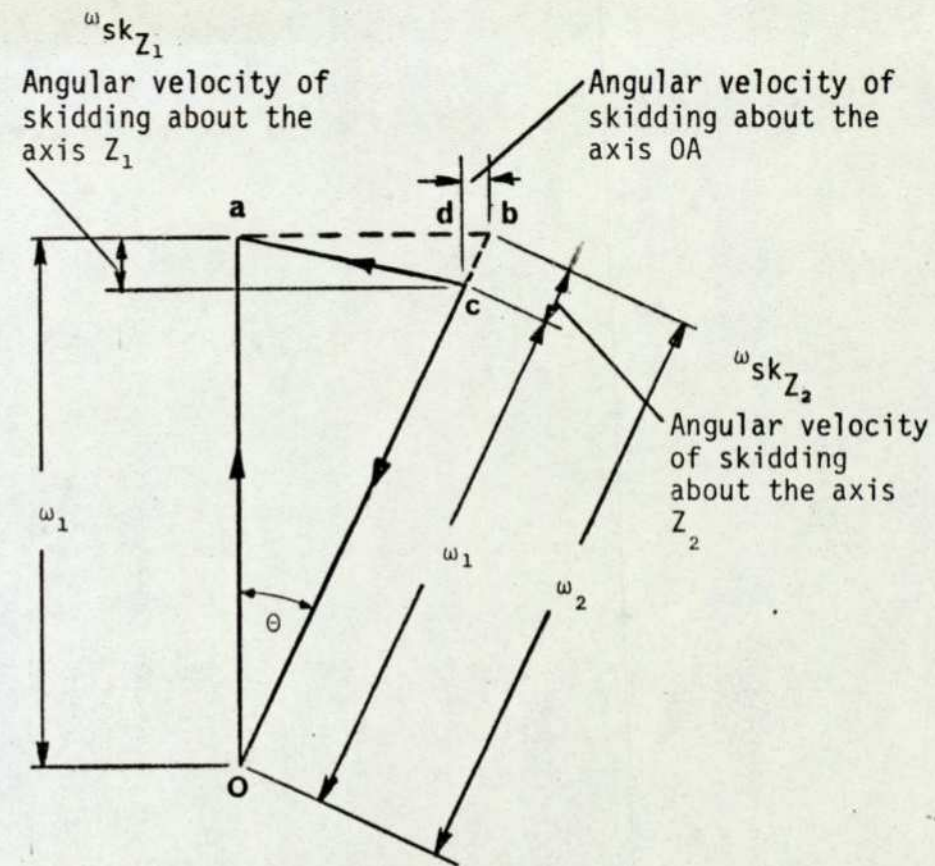


FIGURE 20 RELATION BETWEEN ANGLE OF NEUTRAL PLANE, χ , INCLINATION OF PLATEN AXIS, θ , AND PLATEN ROTATIONAL SPEED RATIO, ω_1/ω_2



(a) Physical diagram



(b) Vector diagram of angular velocities

FIGURE 21 ILLUSTRATION OF INDUCED SKIDDING AT UPPER DIE-WORKPIECE INTERFACE

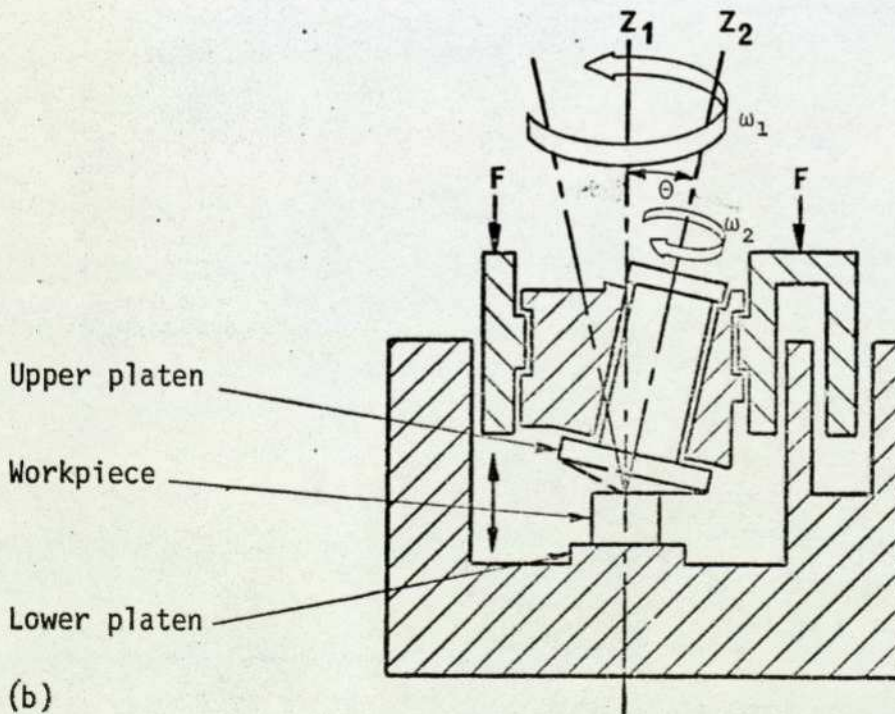
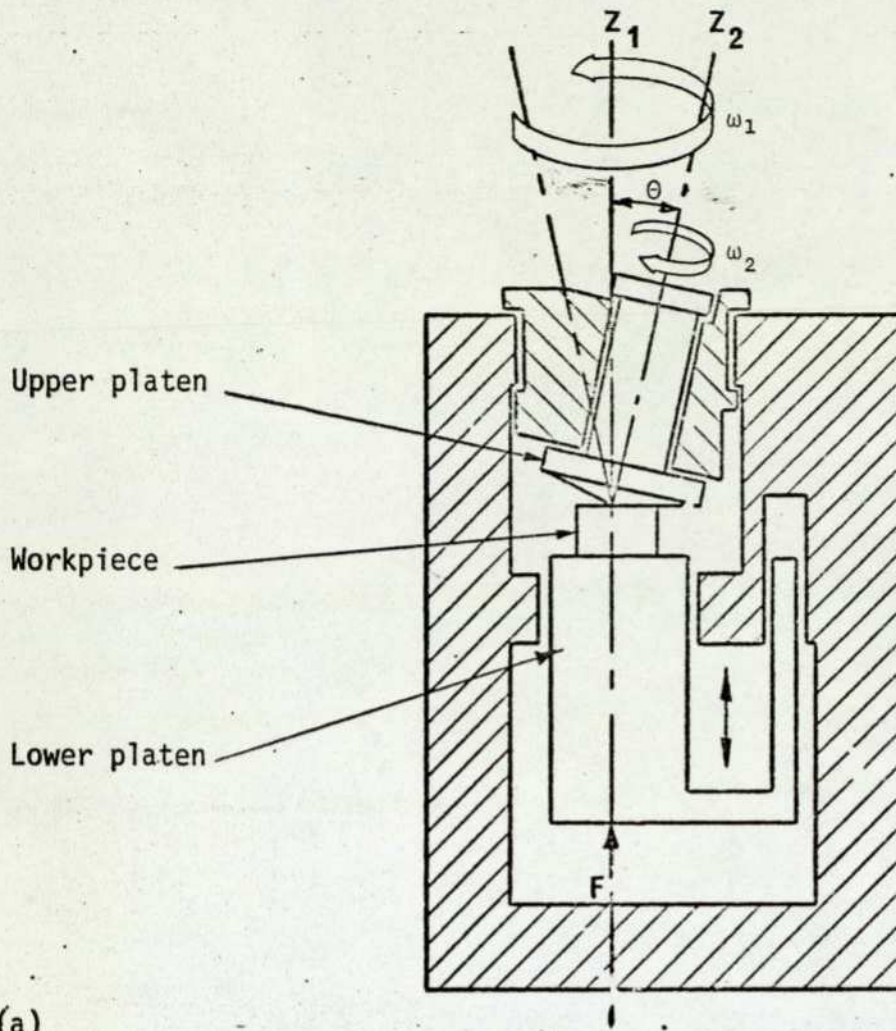


FIGURE 22 KINEMATIC ARRANGEMENT USED IN MACHINES OF RECENT DESIGN

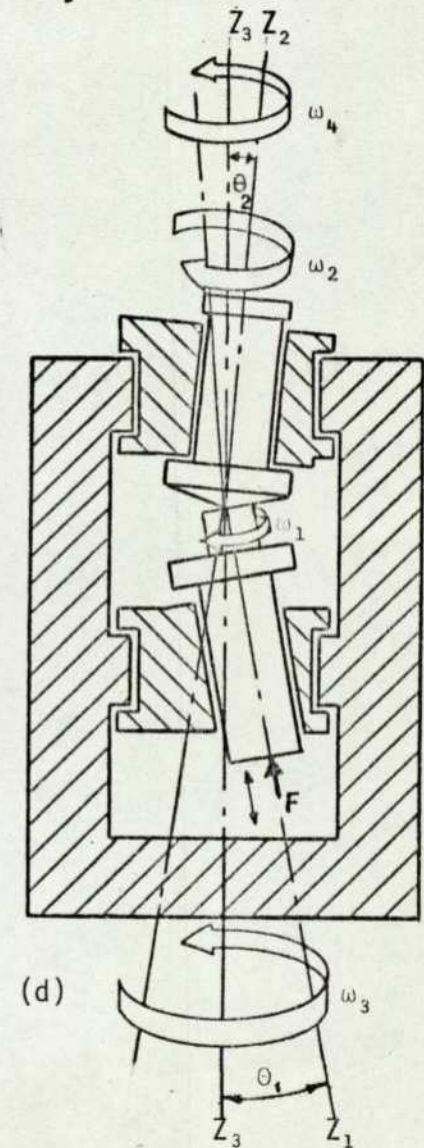
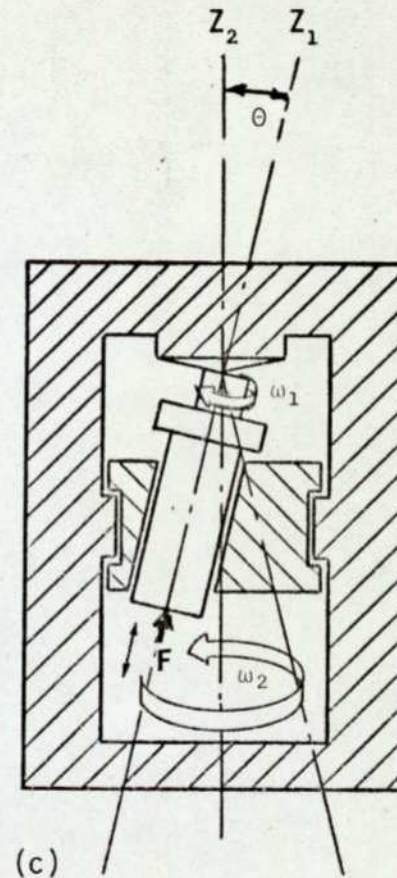
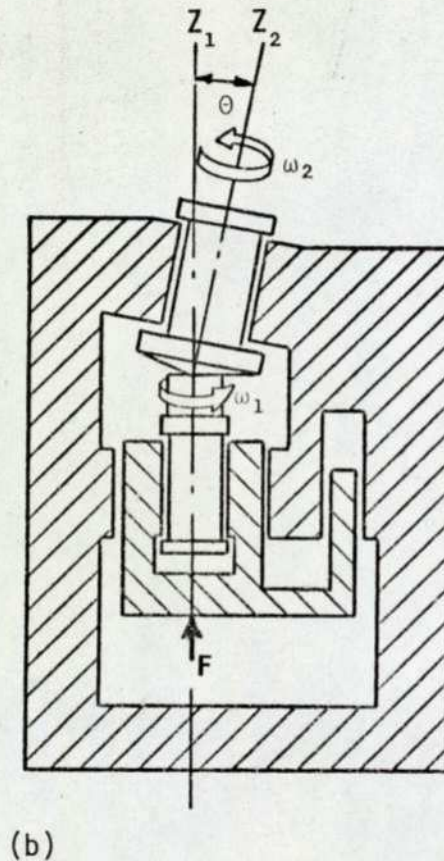
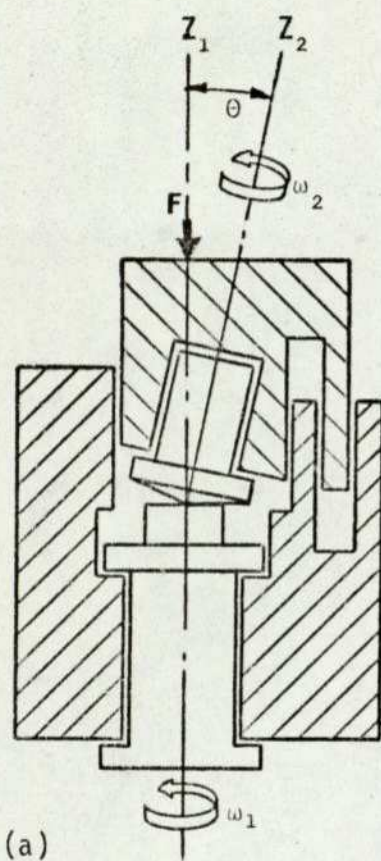


FIGURE 23 ALTERNATIVE KINEMATIC ARRANGEMENTS.

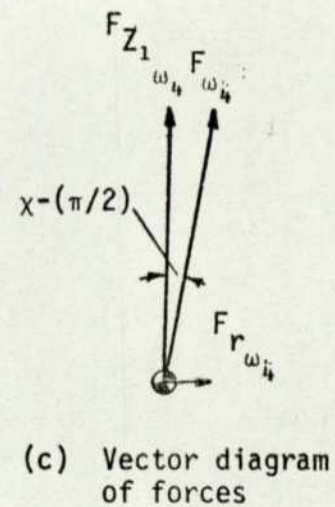
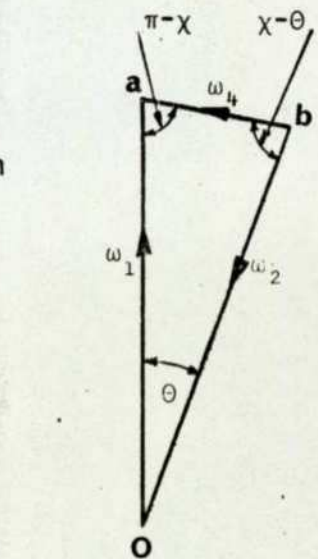
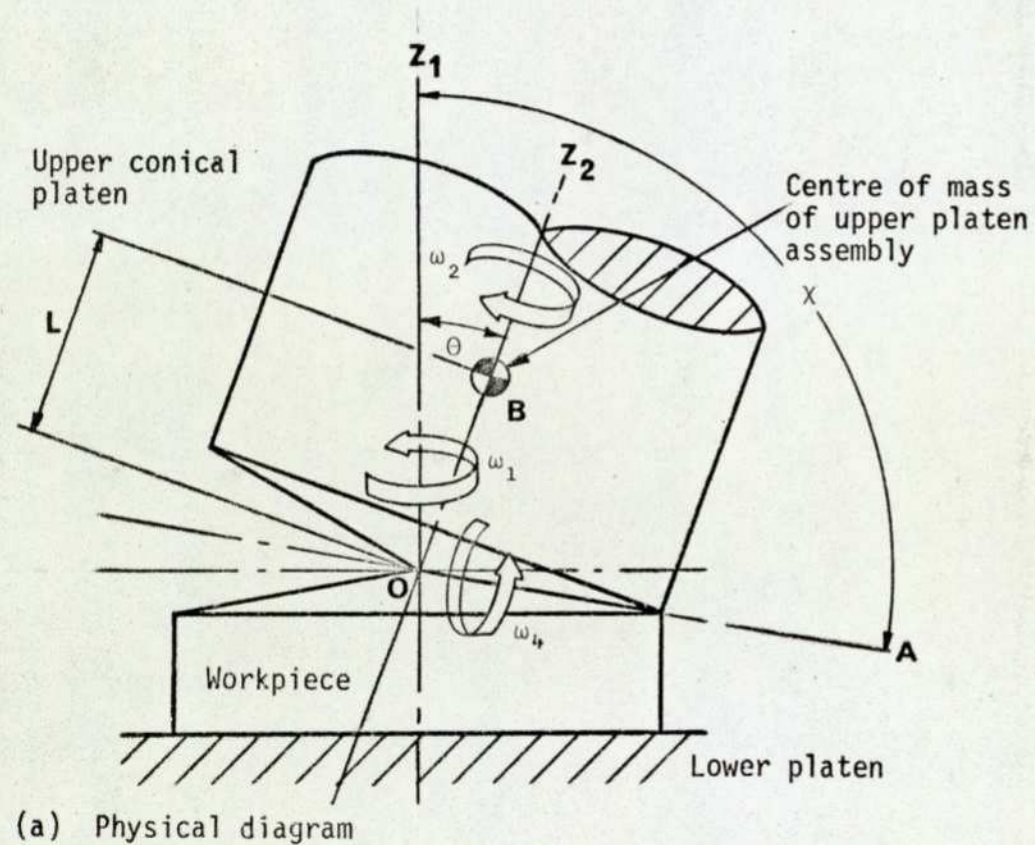
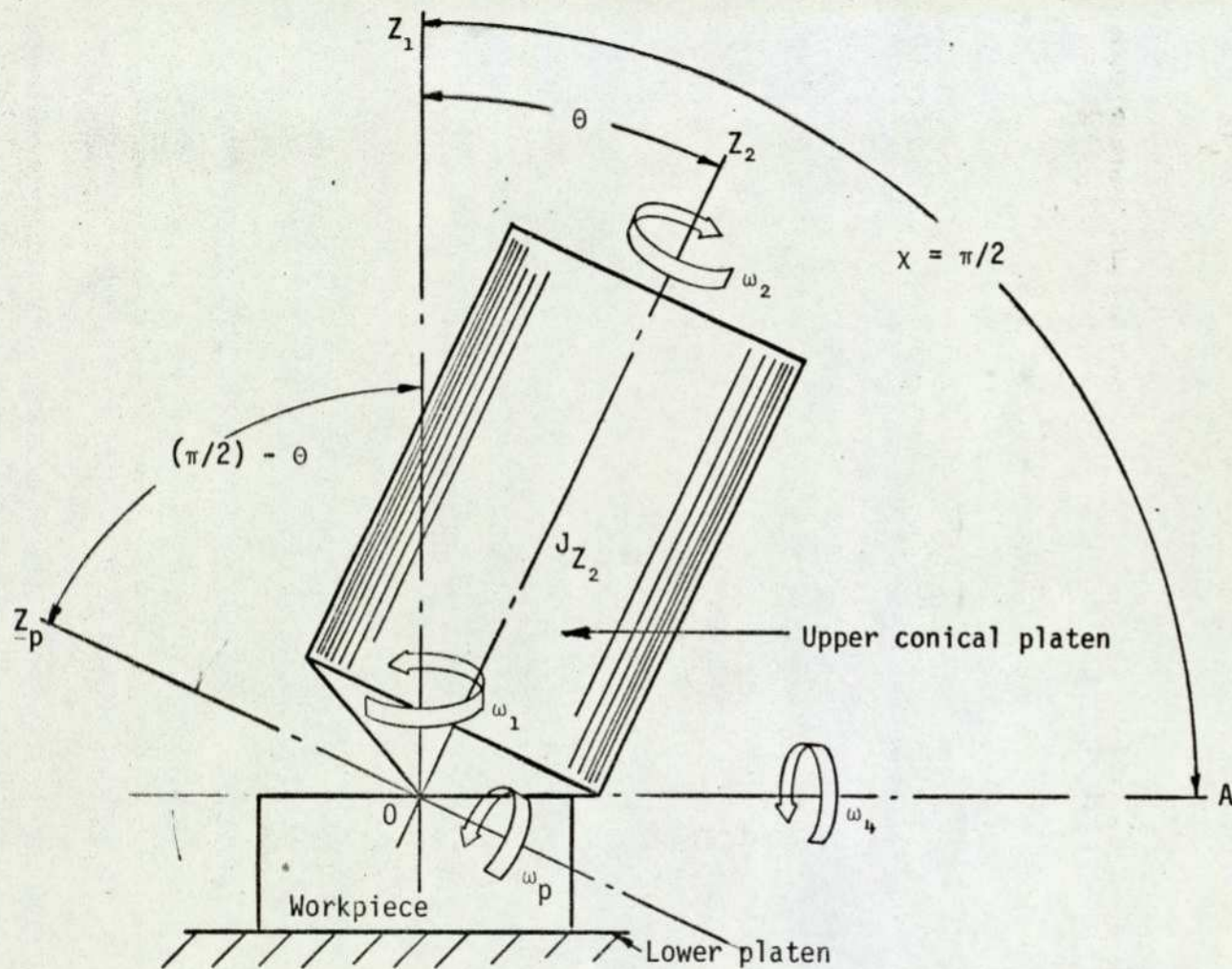
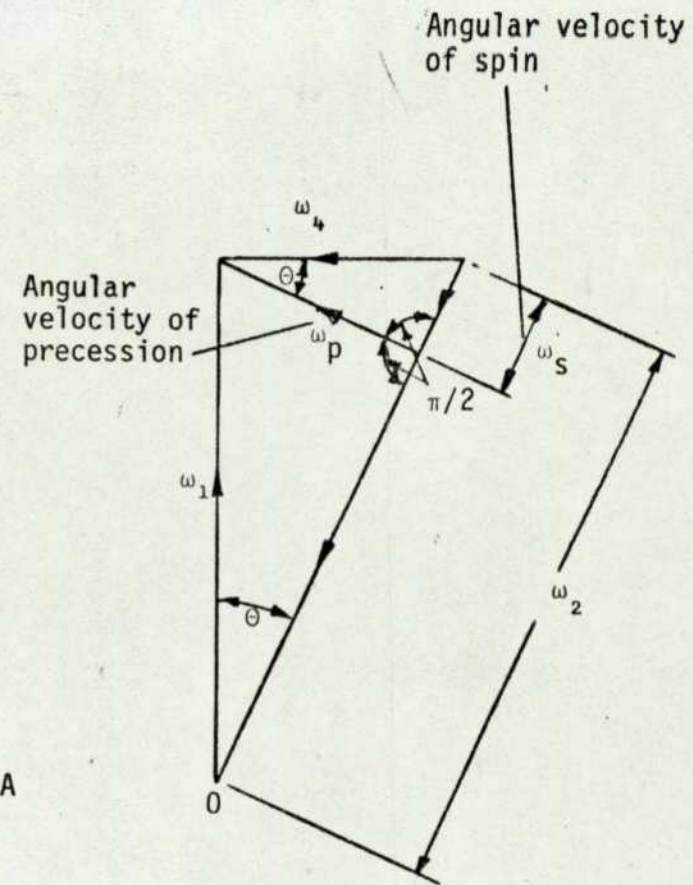


FIGURE 24 ILLUSTRATION OF UNBALANCED INERTIA FORCES ARISING DUE TO RADIAL ACCELERATION IN AN ORBITAL FORGING ARRANGEMENT.



(a) Physical diagram



(b) Vector diagram of angular velocities

FIGURE 25 ILLUSTRATION OF GYROSCOPIC COUPLE ARISING FROM ORBITAL MOTION.

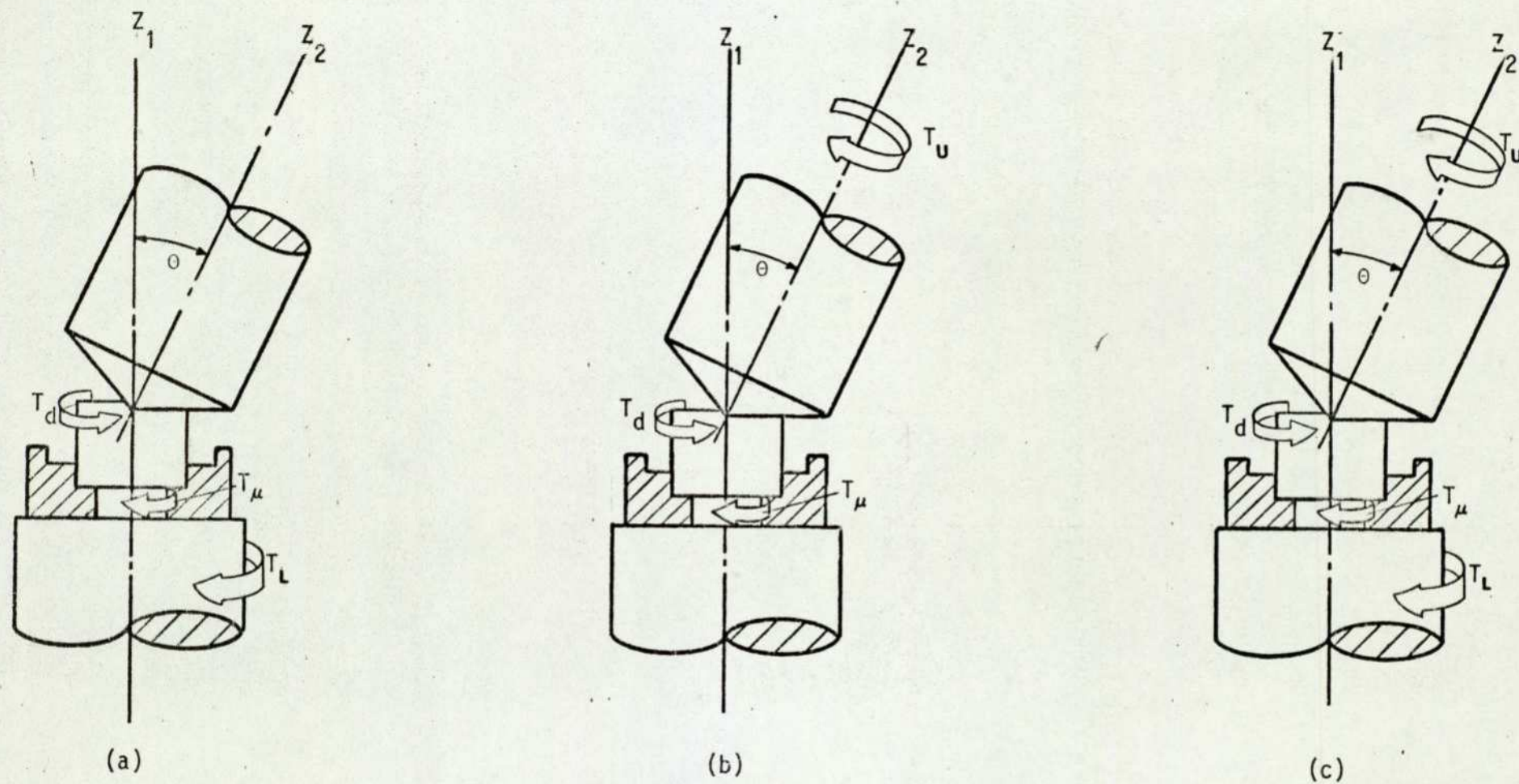


FIGURE 26 ILLUSTRATION OF TORQUE RELATION FOR VARIOUS ROTATIONAL DRIVE ARRANGEMENTS.

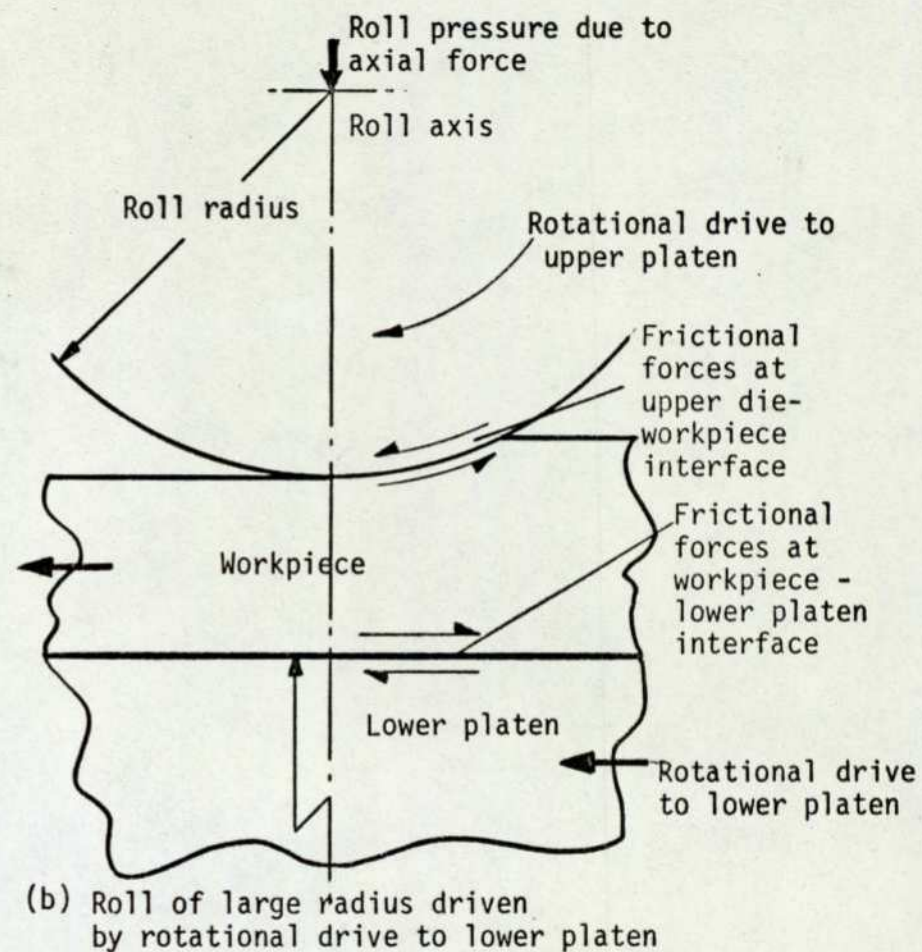
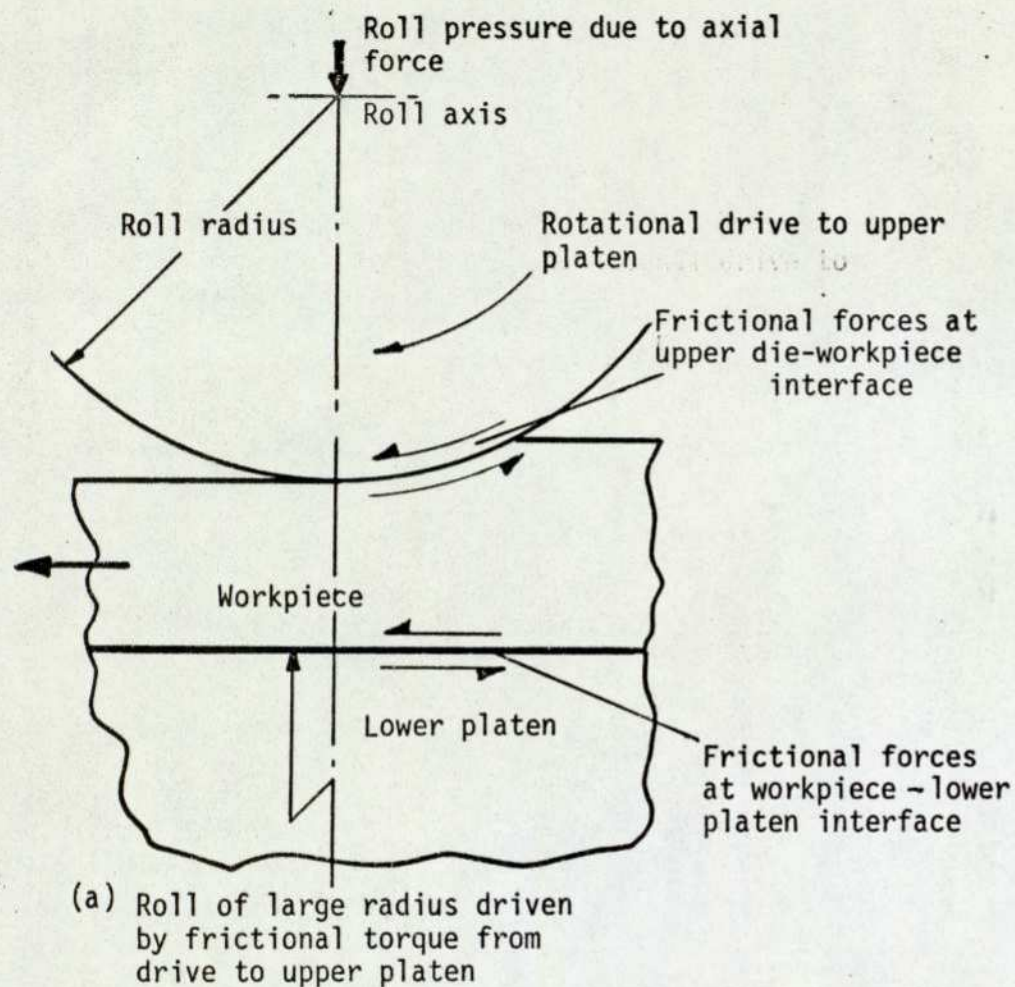


FIGURE 27 ILLUSTRATION OF THE PLANE STRAIN ROLLING ANALOGY APPLIED TO ROTARY FORGING.

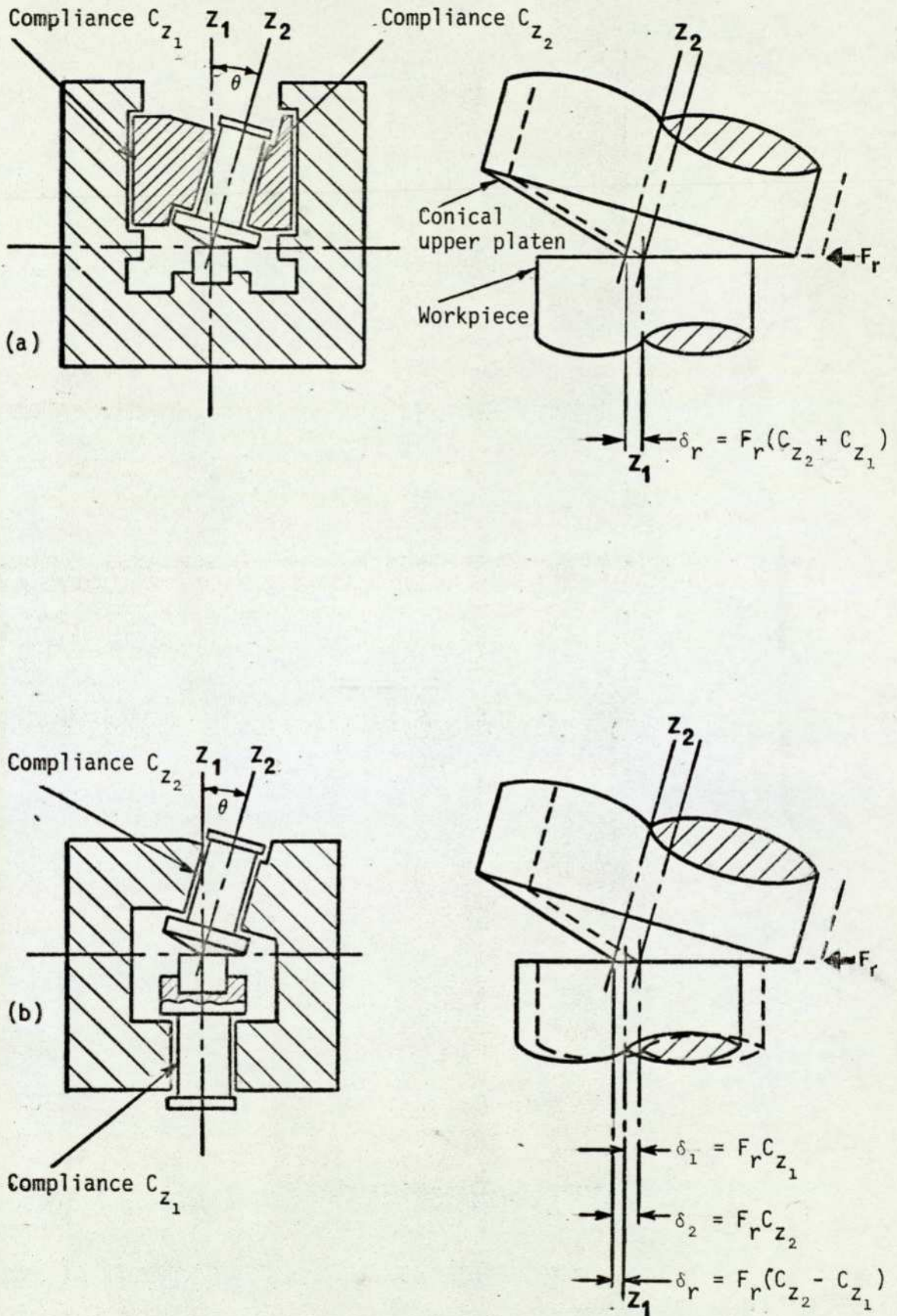


FIGURE 28 ALTERNATIVE KINEMATIC ARRANGEMENTS ILLUSTRATING THE INFLUENCE OF BEARING COMPLIANCE UPON RELATIVE DISPLACEMENT OF AXES Z_1 AND Z_2 :

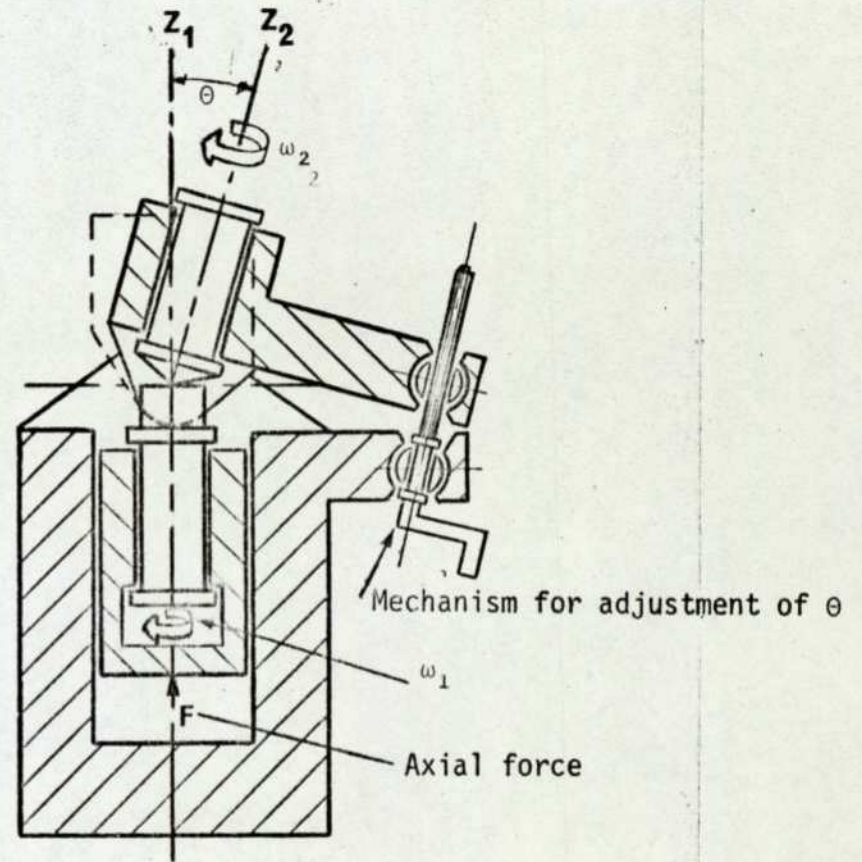
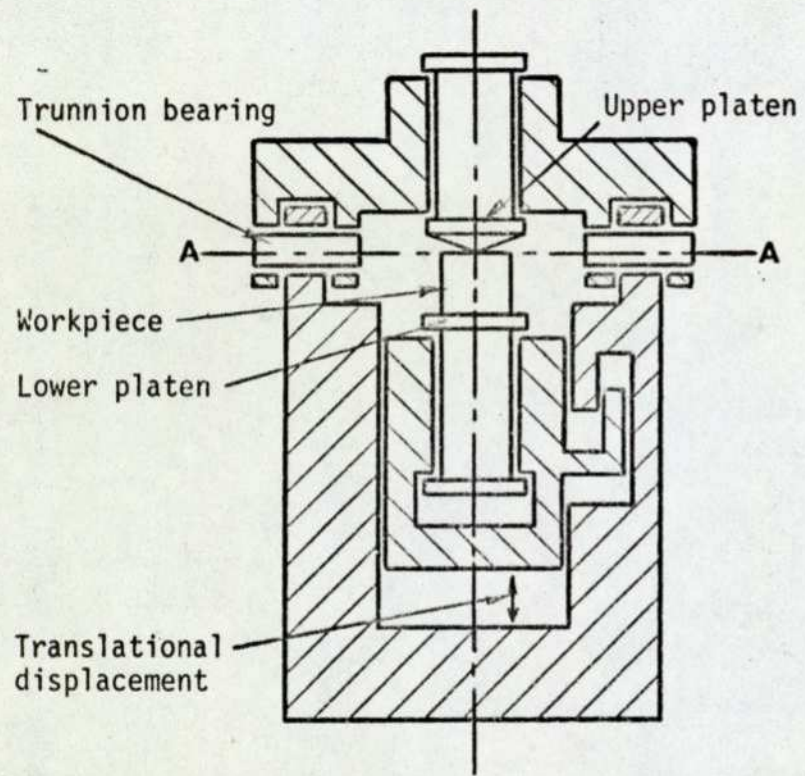
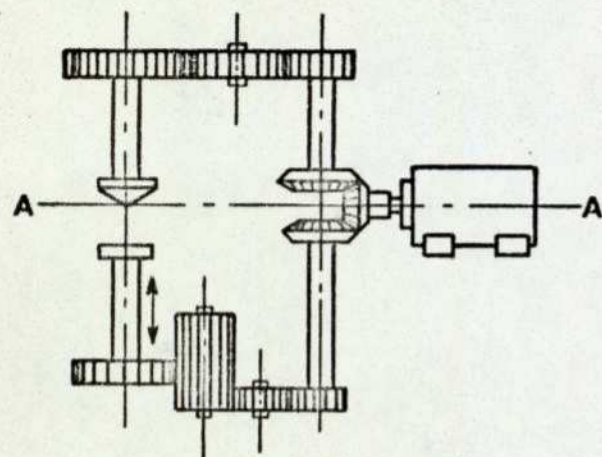
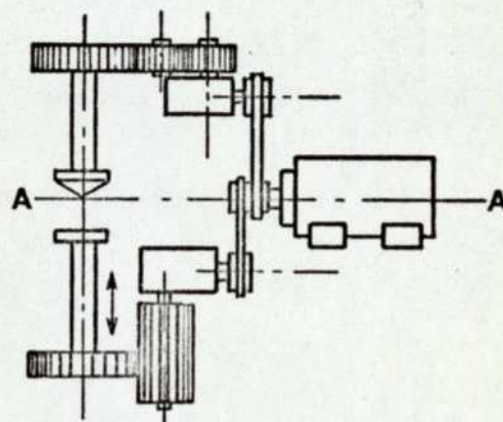


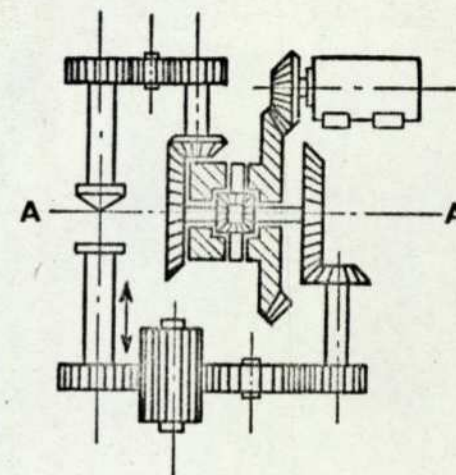
FIGURE 29 KINEMATIC ARRANGEMENT OF PROPOSED DESIGN



(a) Gear drive

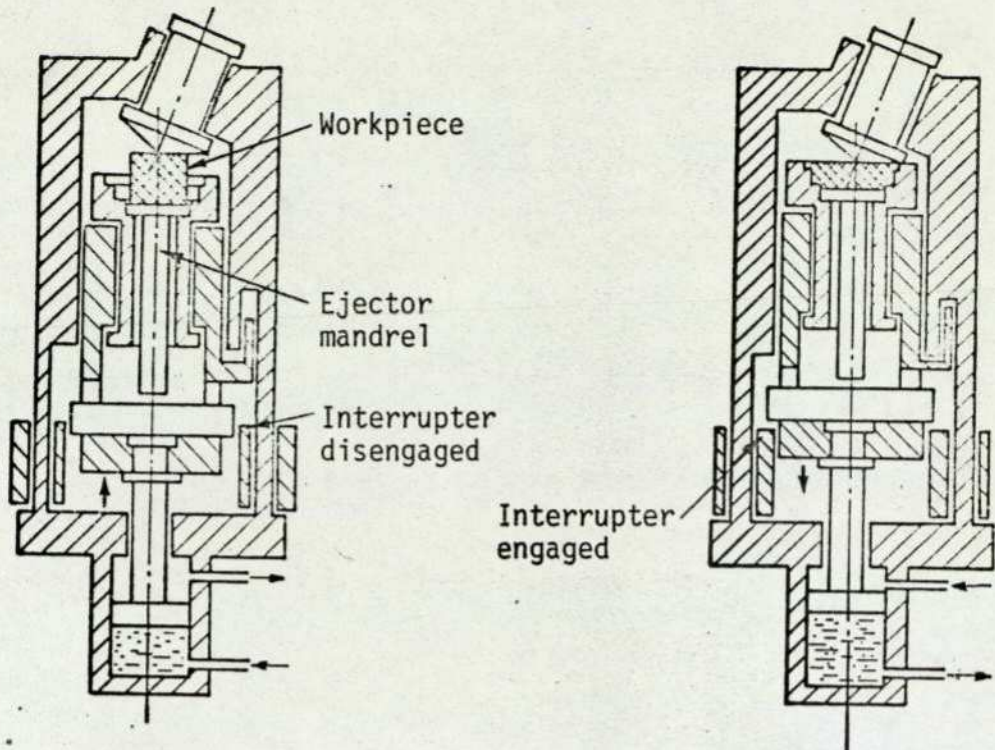


(b) Belt drive



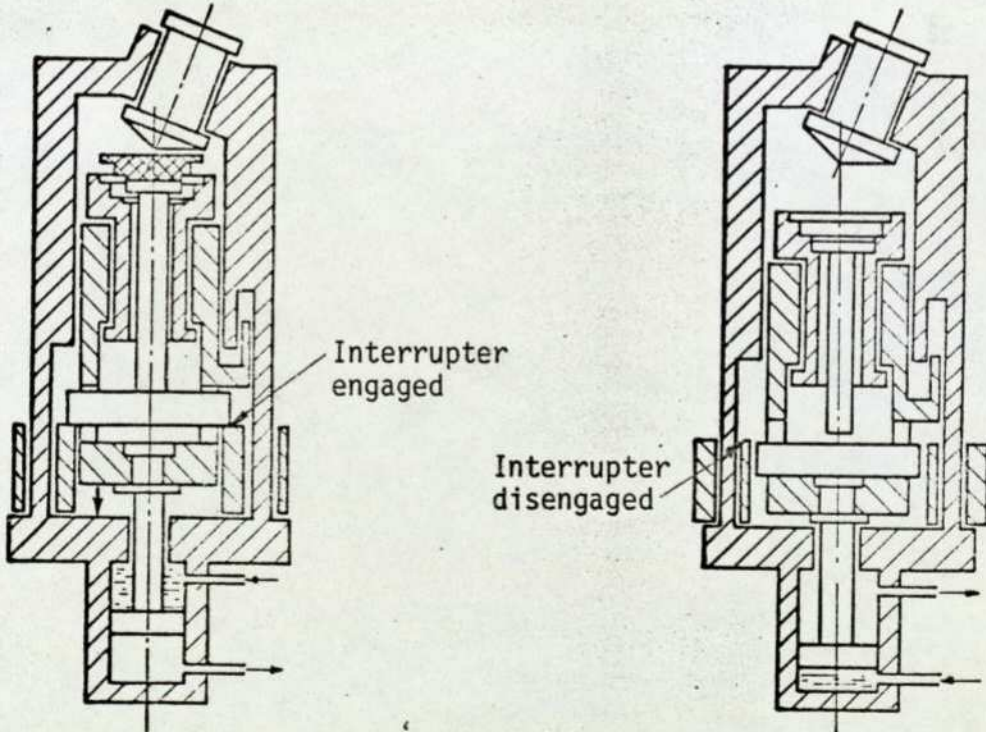
(c) Differential gear drive

FIGURE 30 ALTERNATIVE ROTATIONAL DRIVE ARRANGEMENTS.



(a) Workpiece loaded and deformation in progress.

(b) Deformation process complete, commencement of ejection process.



(c) Ejection process complete

(d) Machine ready for loading

FIGURE 31 OPERATING CYCLE OF THE WORKPIECE EJECTION SYSTEM.

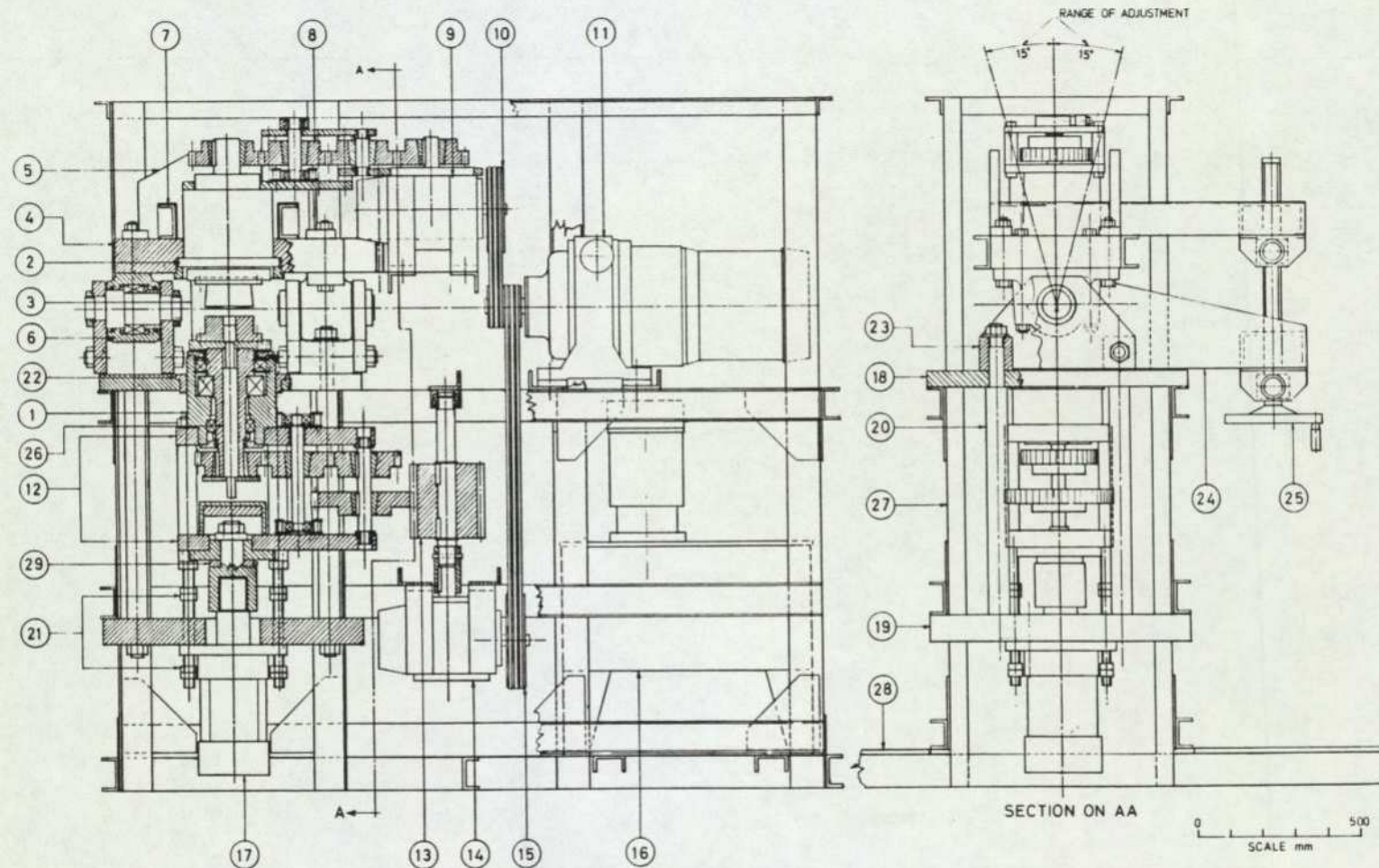


FIGURE 32 ORTHOGRAPHIC VIEWS OF 0.5 MN ROTARY FORGING MACHINE

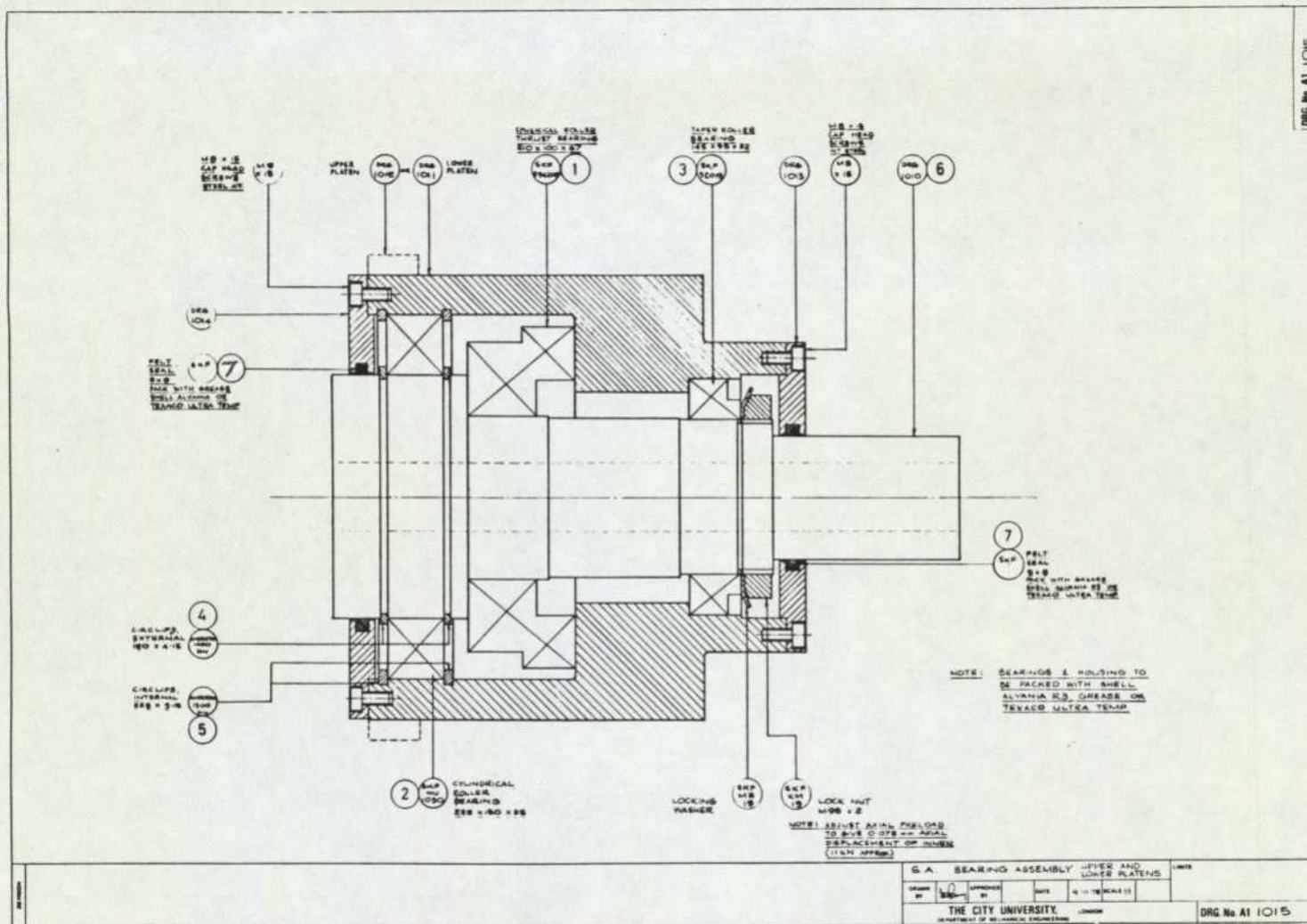


FIGURE 33 SUB-ASSEMBLY OF PLATEN BEARINGS

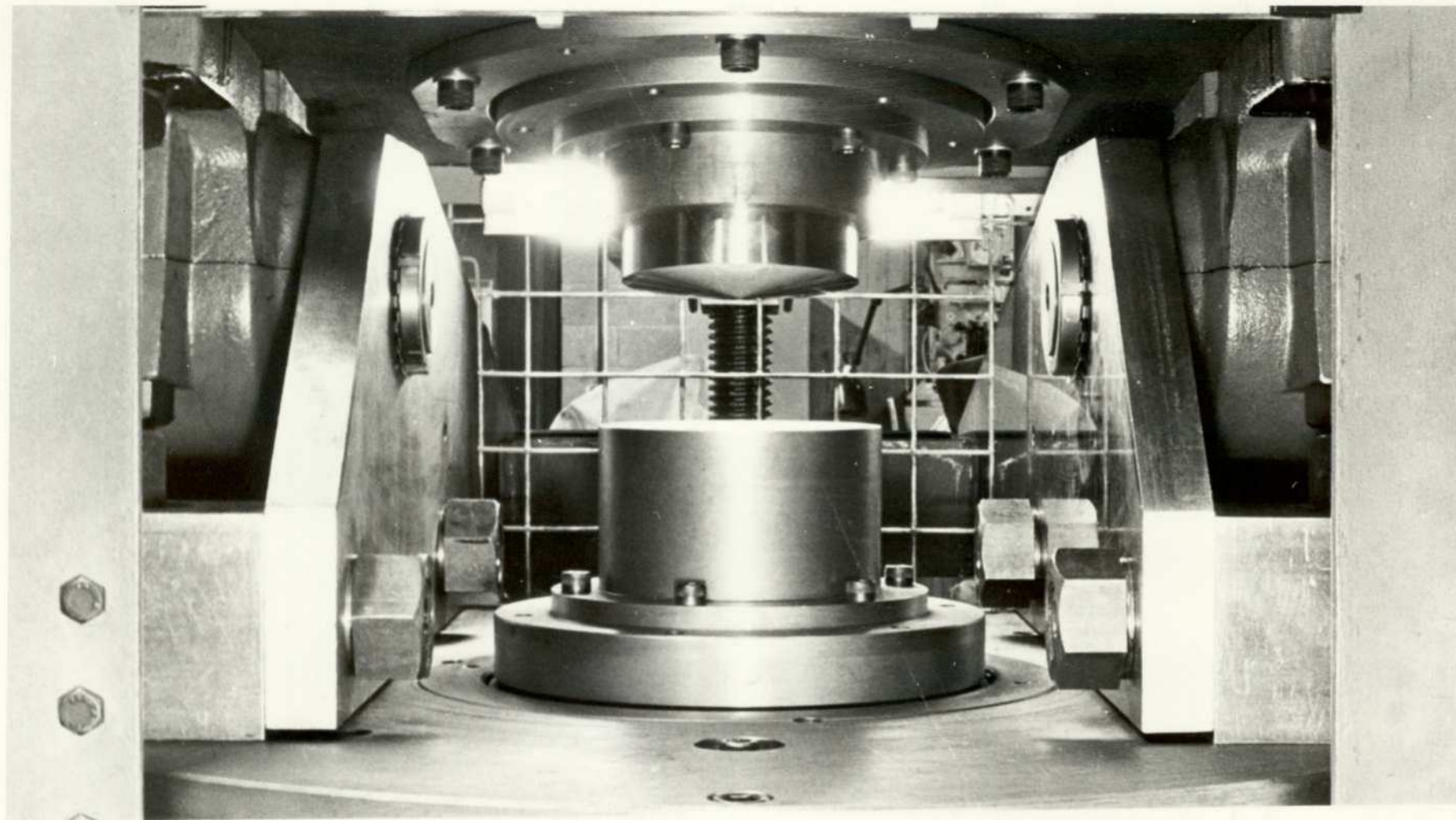


FIGURE 34 TYPICAL CONICAL UPPER DIE AND FLAT LOWER DIE IN POSITION

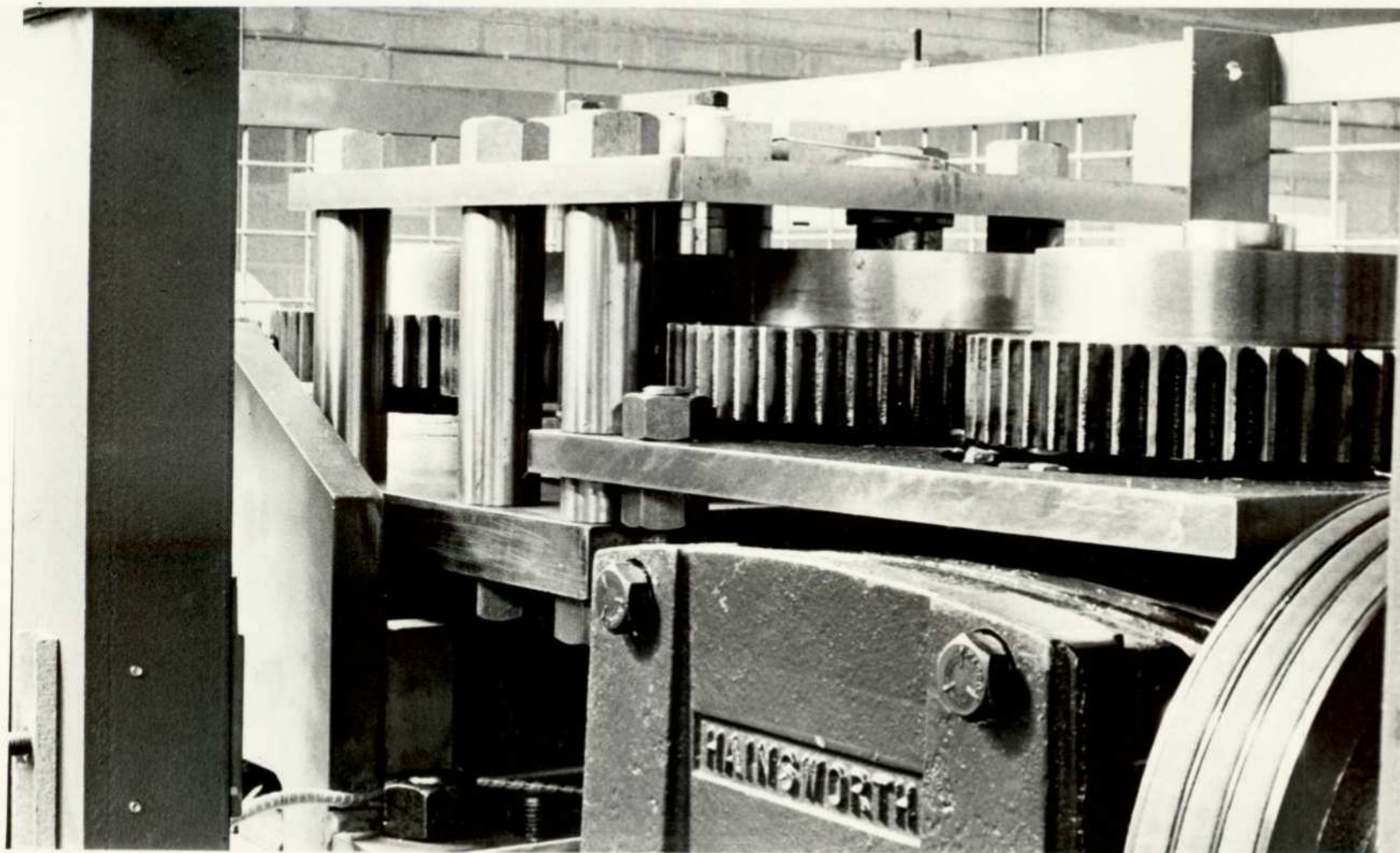


FIGURE 35 GEAR TRAIN FOR UPPER PLATEN ROTATIONAL DRIVE

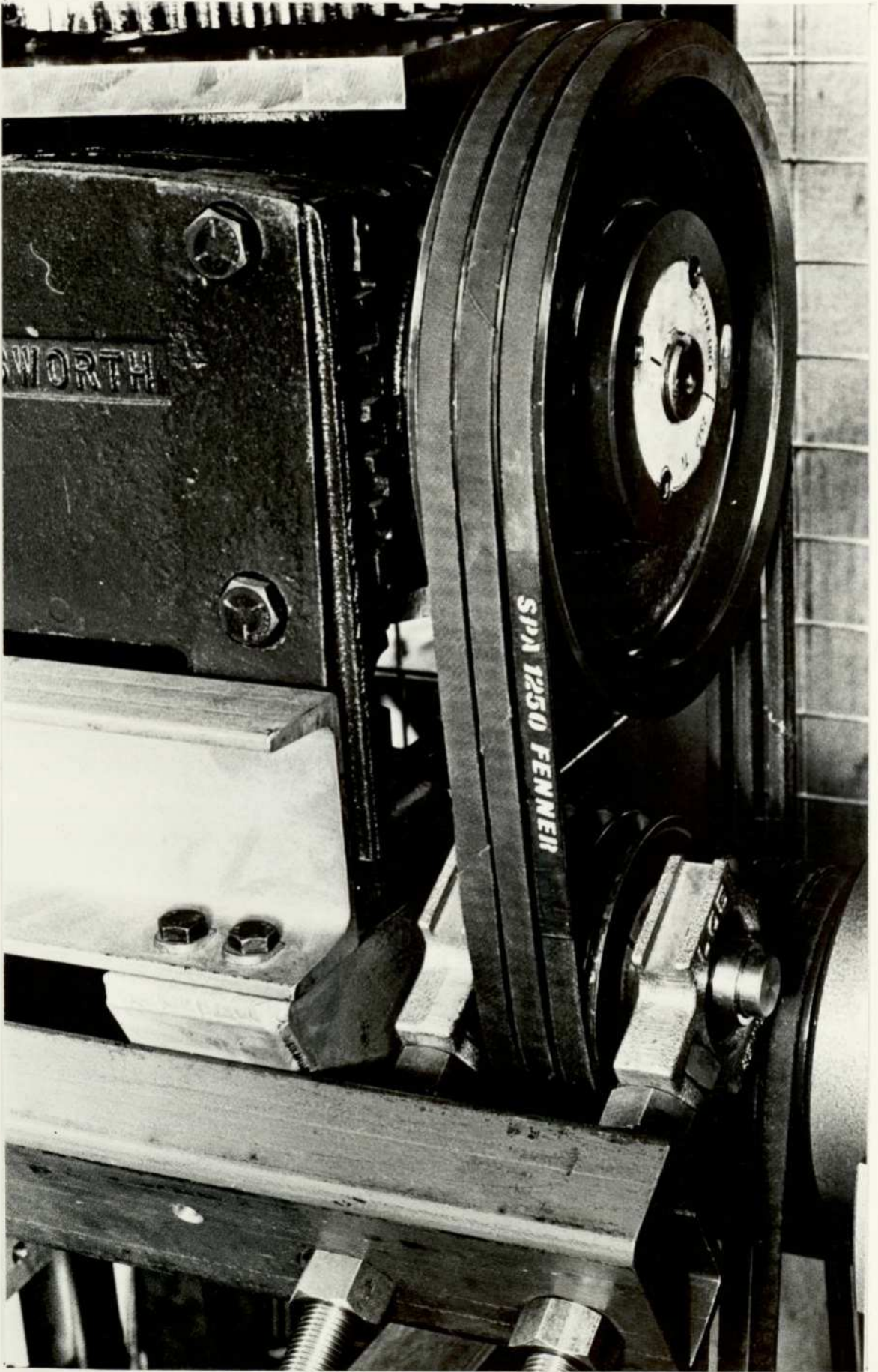


FIGURE 36 BELT TENSIONING TO ACCOMMODATE VARIOUS UPPER DRIVE PULLEY DIAMETERS

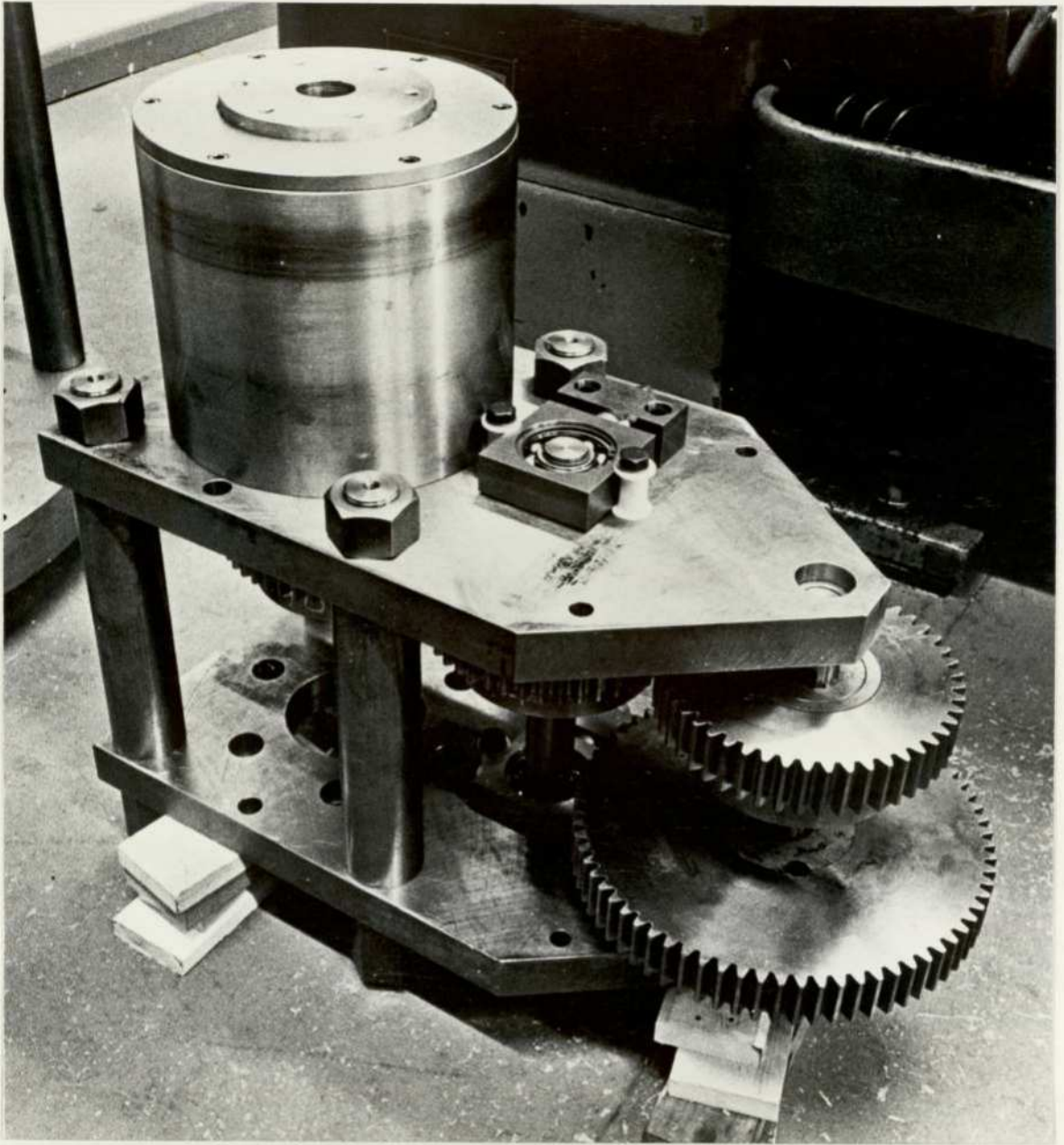


FIGURE 37 LOWER PLATEN ASSEMBLY DURING CONSTRUCTION

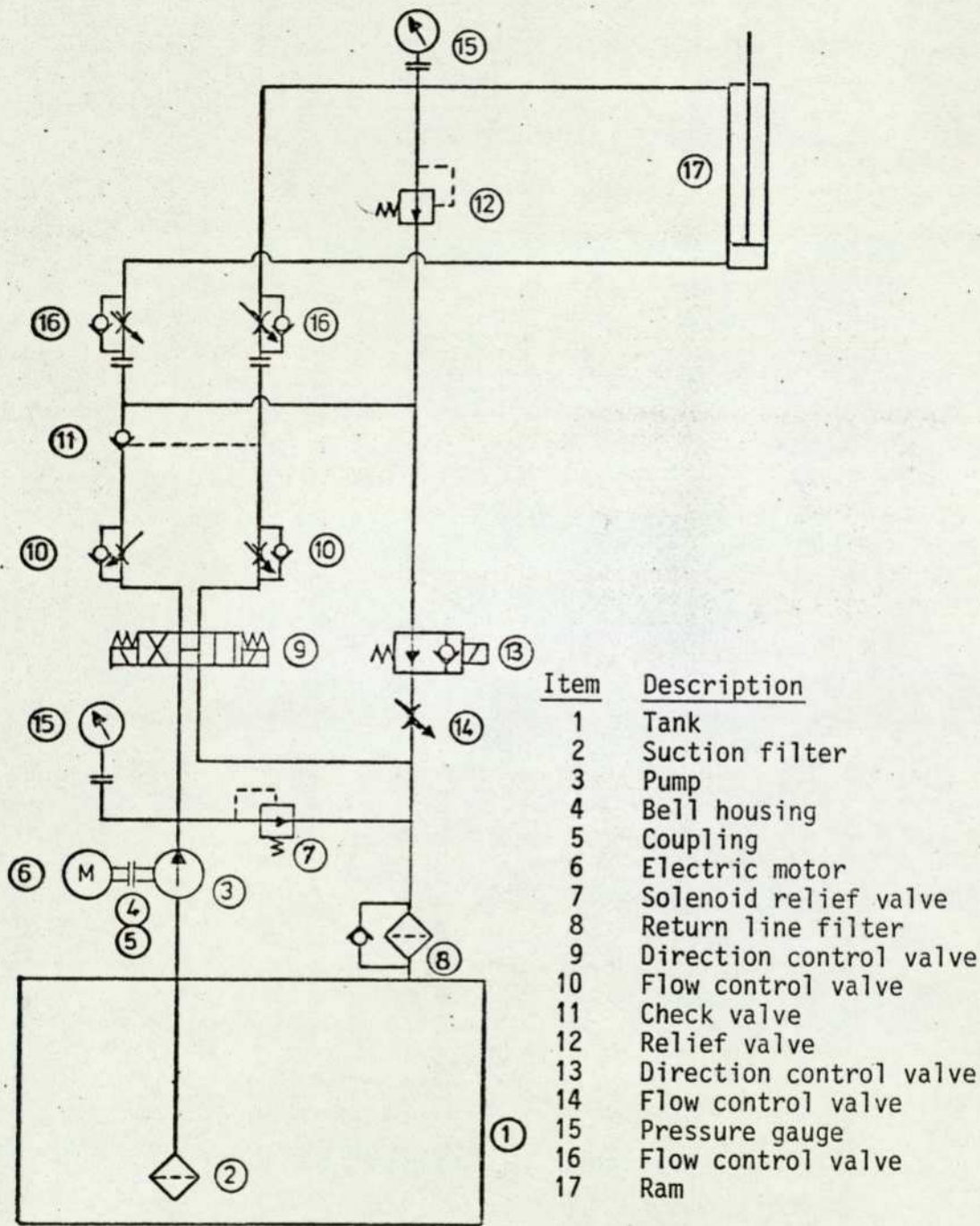


FIGURE 38 HYDRAULIC SYSTEM

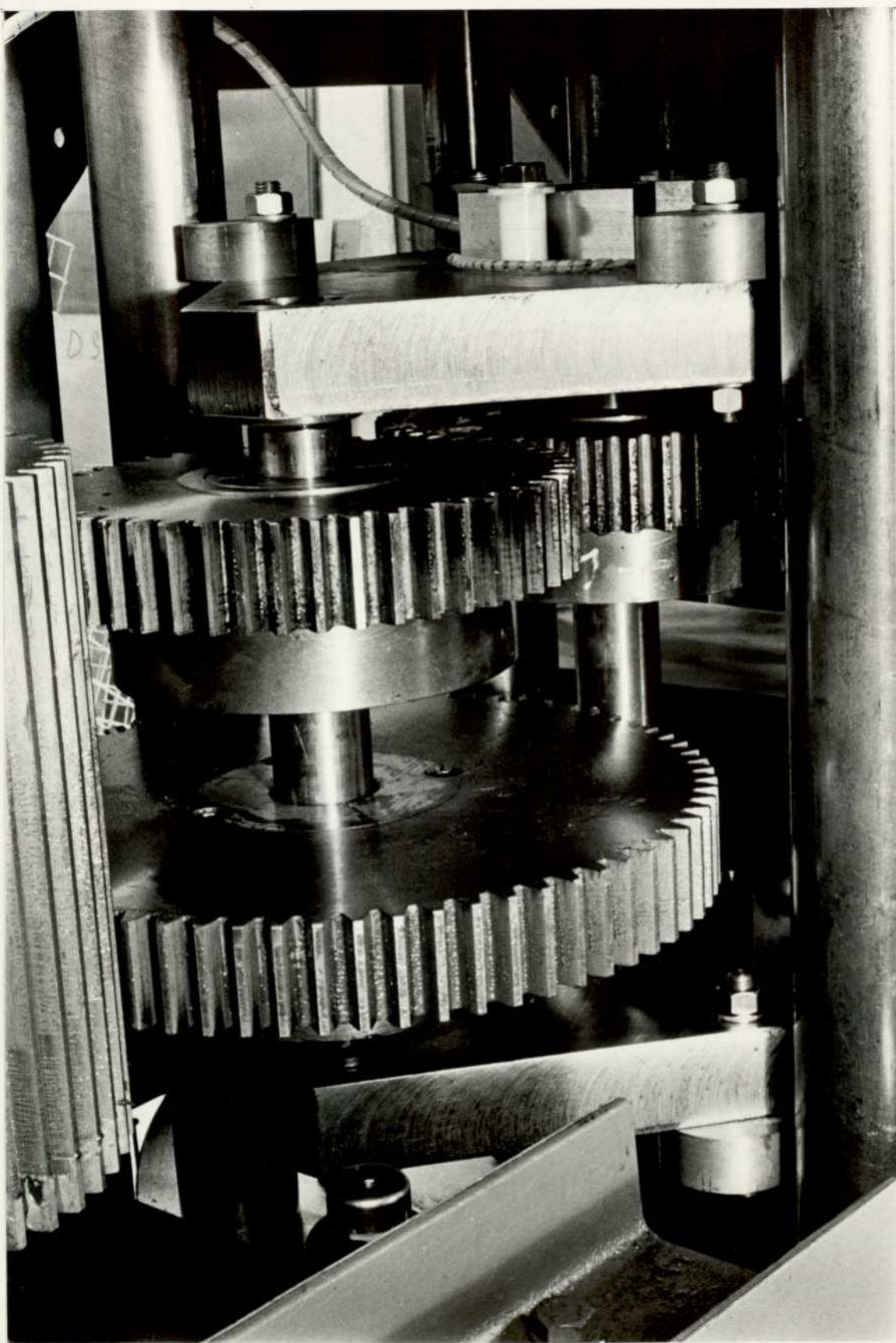


FIGURE 39 LOWER PLATEN ROTATIONAL DRIVE ILLUSTRATING LONG FACE GEAR BEARING PADS AND MAIN FRAME PILLARS.



FIGURE 40 MAIN FRAME ASSEMBLY DURING CONSTRUCTION

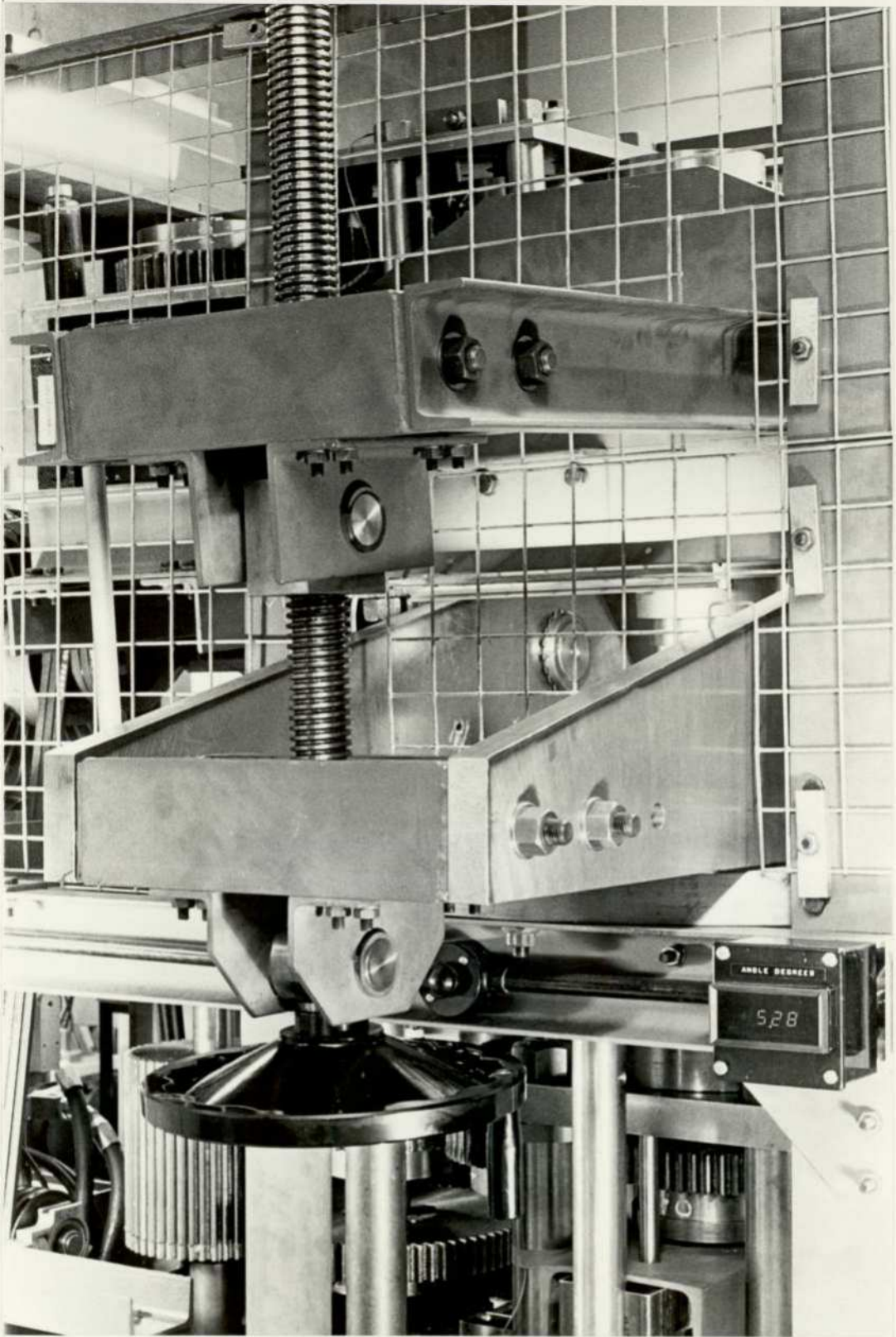
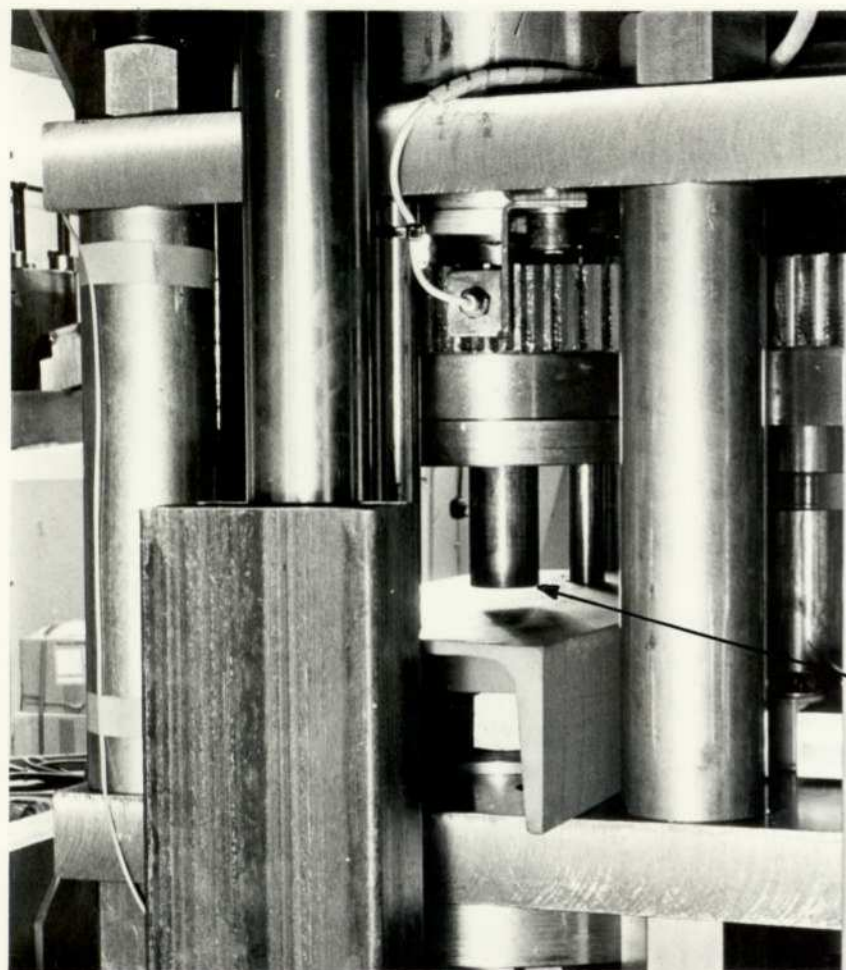
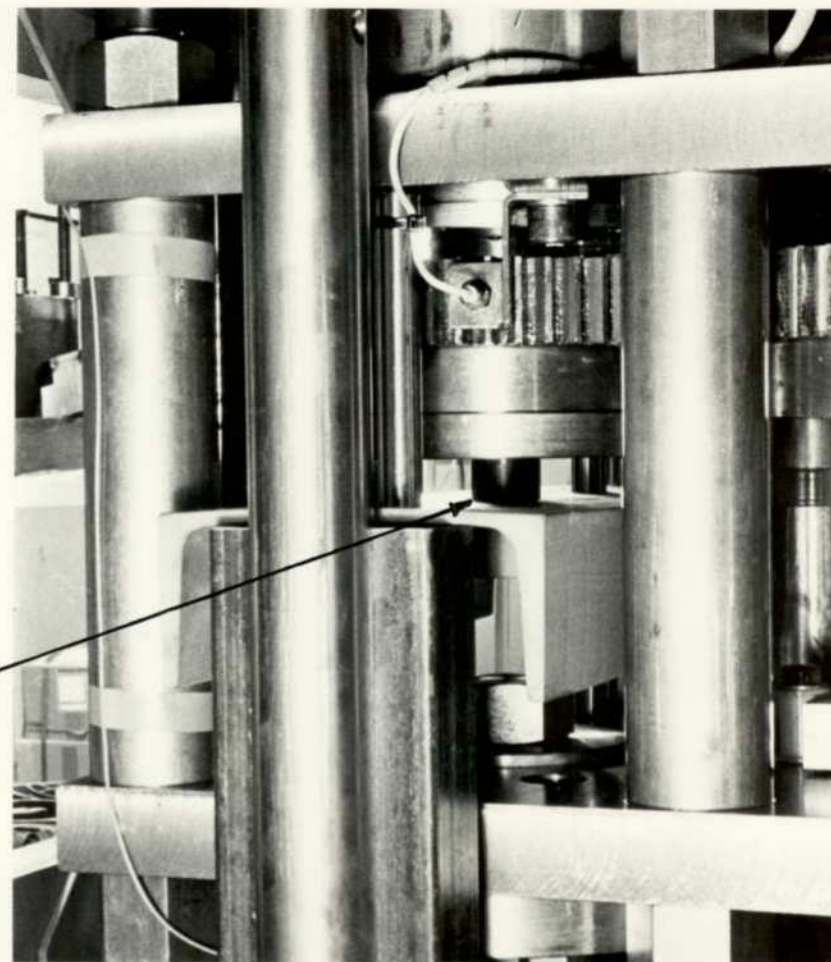


FIGURE 41 FACILITY FOR MANUAL ADJUSTMENT OF INCLINATION OF PLATEN AXES, θ .



(a) Ejector disengaged



(b) Ejector engaged

Ejector
mandrel

FIGURE 42 WORKPIECE EJECTION MECHANISM

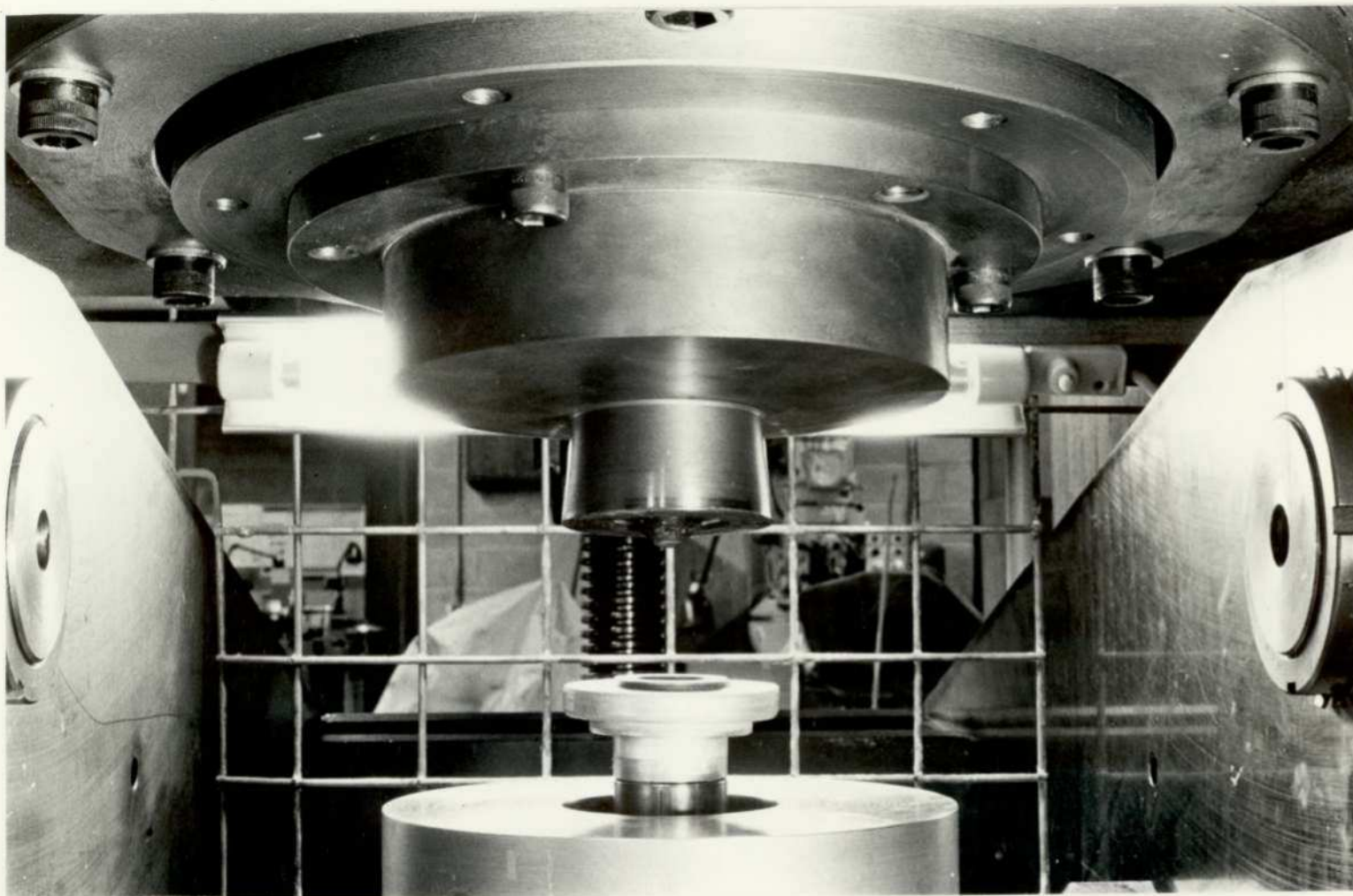
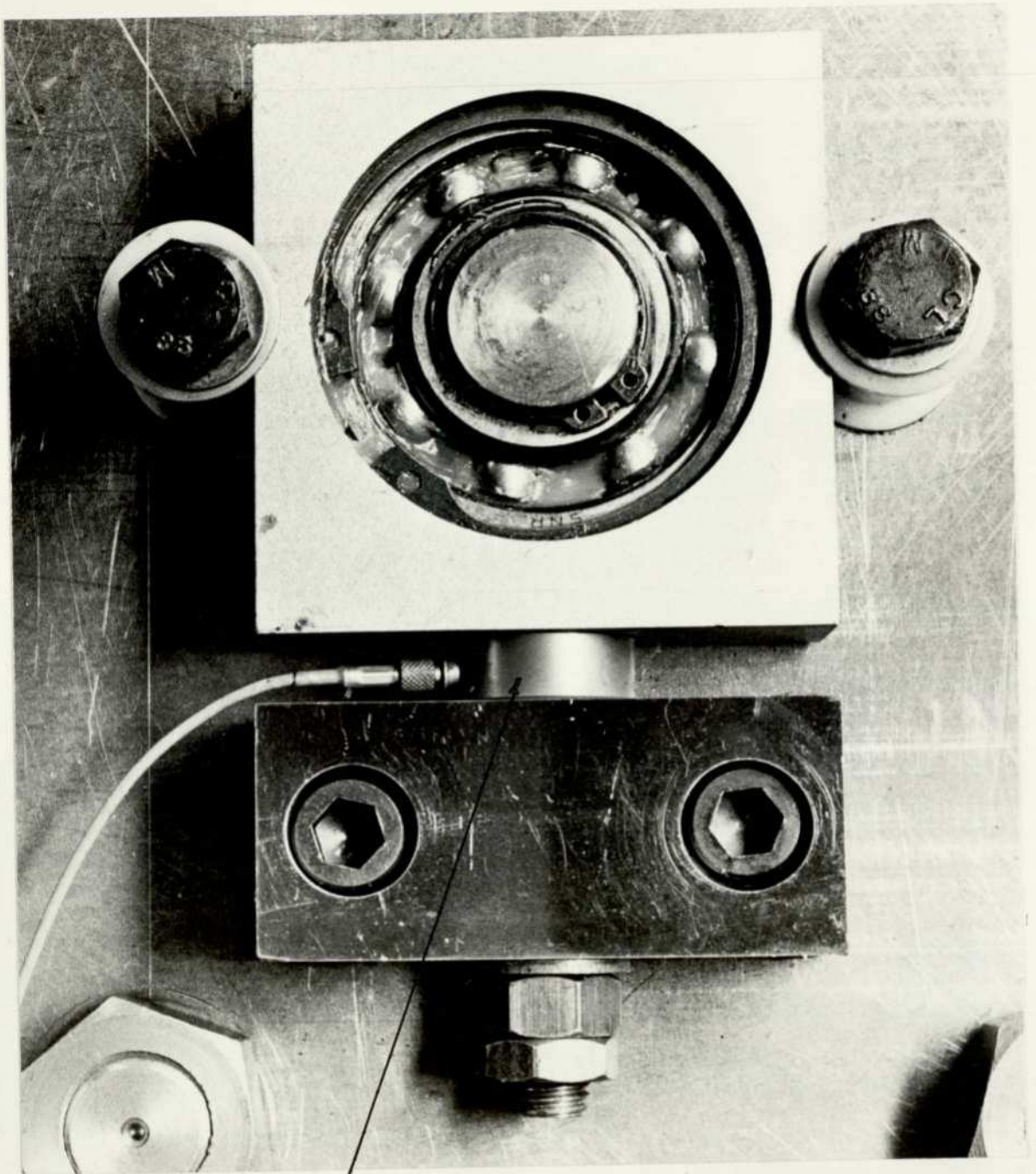


FIGURE 43 TYPICAL WORKPIECE IN POSITION ON THE EJECTION PIN AFTER FORMING WITH CONTOURED UPPER AND LOWER DIES.



Force transducer

FIGURE 44 TYPICAL ROTATIONAL DRIVE TORQUE TRANSDUCER SHOWN WITH BEARING COVER PLATE REMOVED.

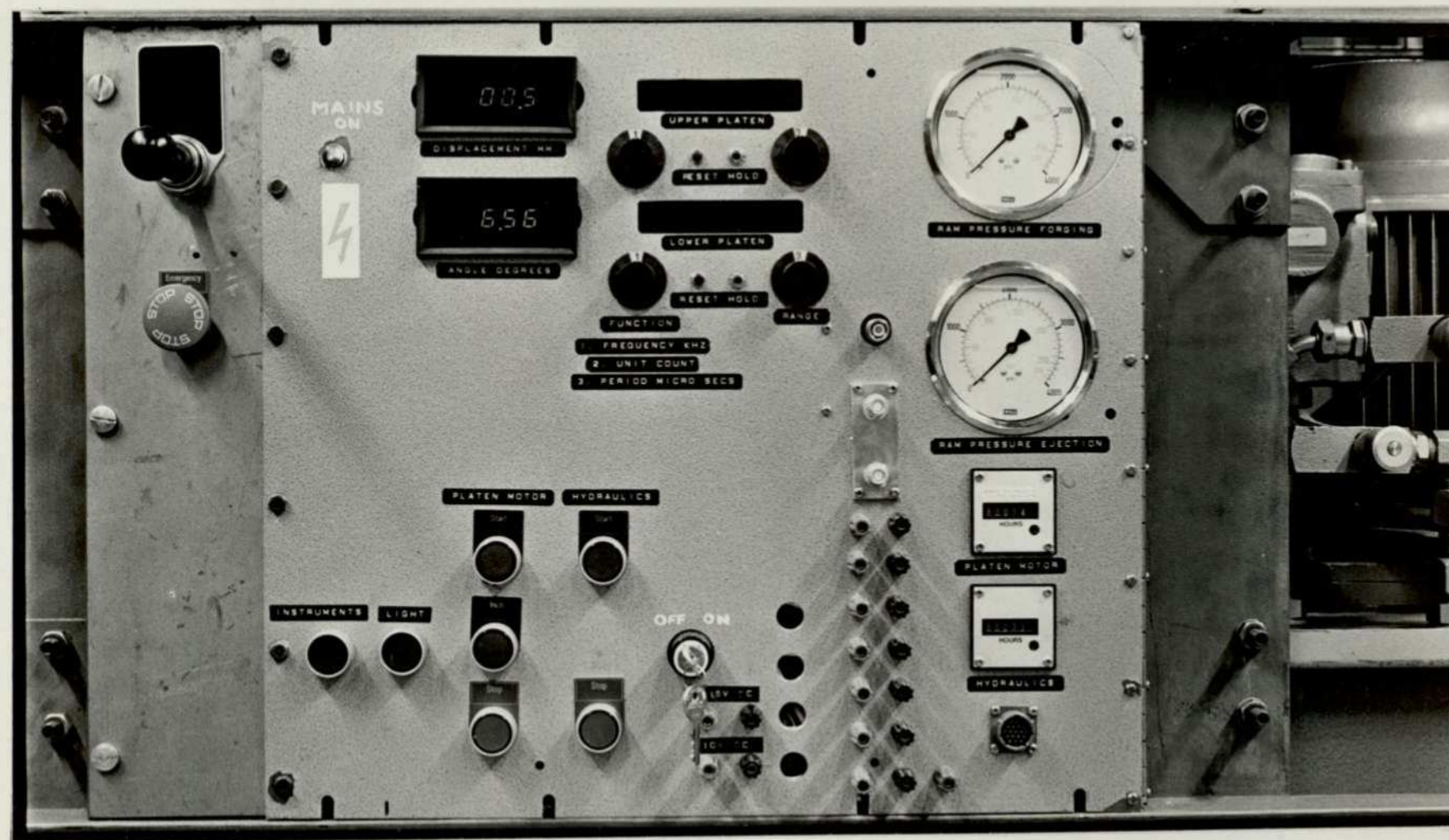
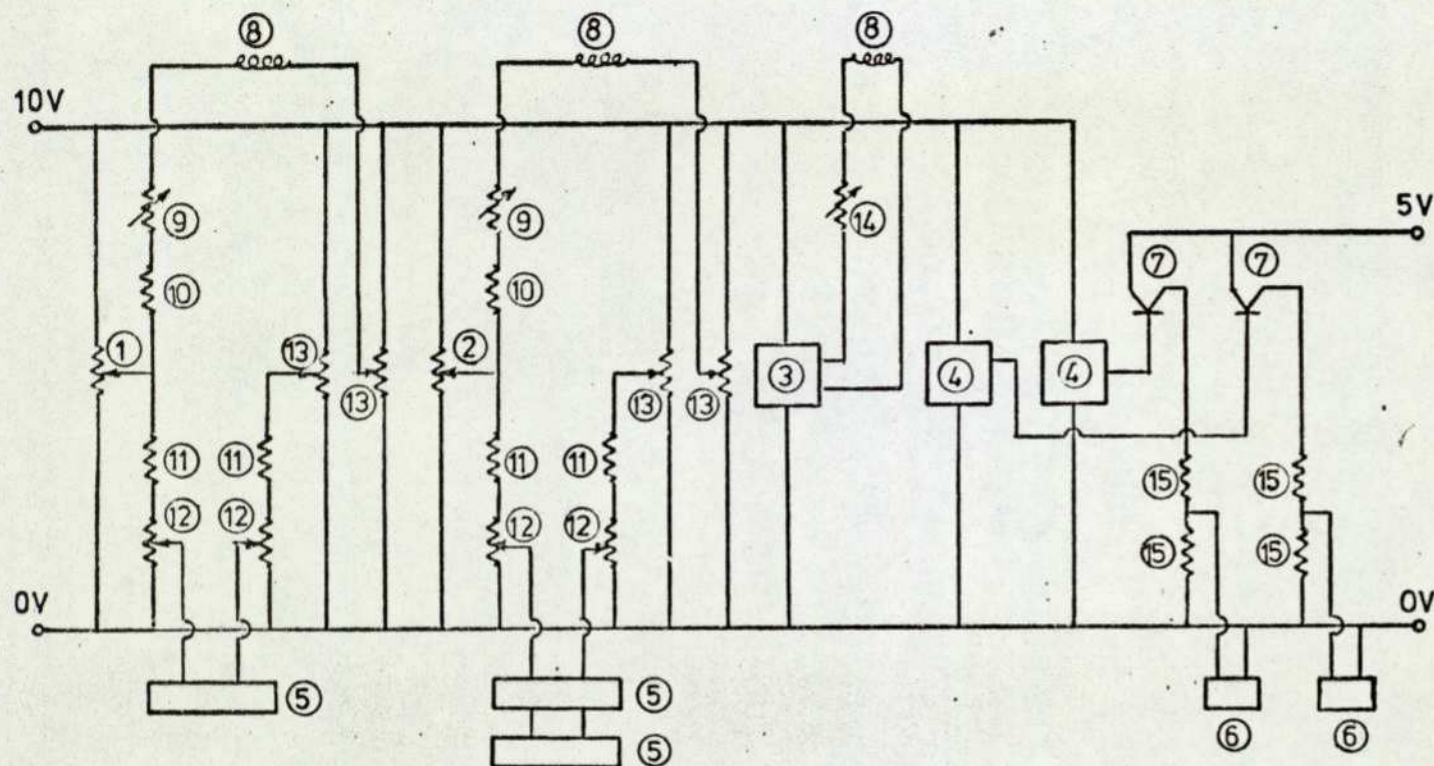
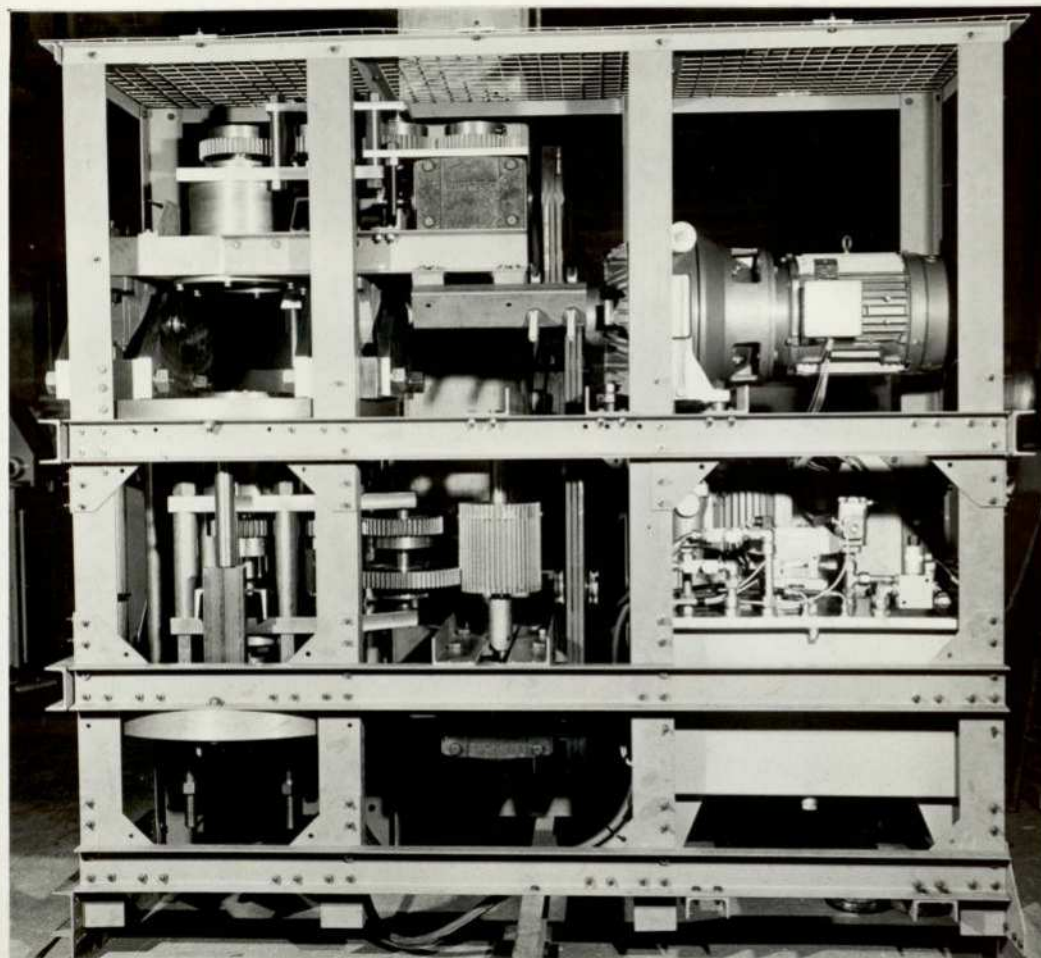


FIGURE 45 MAIN CONTROL PANEL

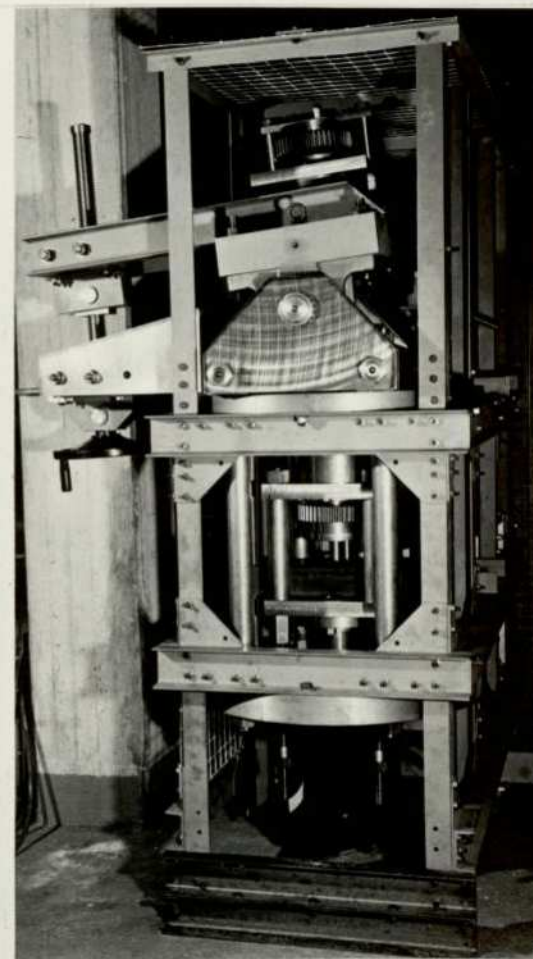


Item	Description
1	Translational displacement transducer
2	Inclination of axes transducer
3	Pressure transducer
4	Proximity transducers - upper and lower
5	Platen rotational velocity
6	Digital frequency meters and pulse counters
7	Transistors Type BCY72 PNP
8	Galvanometer Type B.100
9	50 kΩ potentiometer
10	150 kΩ resistor
11	220 kΩ potentiometer
12	50 kΩ potentiometer
13	10 kΩ potentiometer
14	2 kΩ potentiometer
15	1 kΩ resistor

FIGURE 46 INSTRUMENTATION WIRING DIAGRAM



(a) View from front showing general layout



(b) View from end showing general layout

FIGURE 47 THE MACHINE DURING CONSTRUCTION VIEWED FROM THE FRONT AND THE SIDE

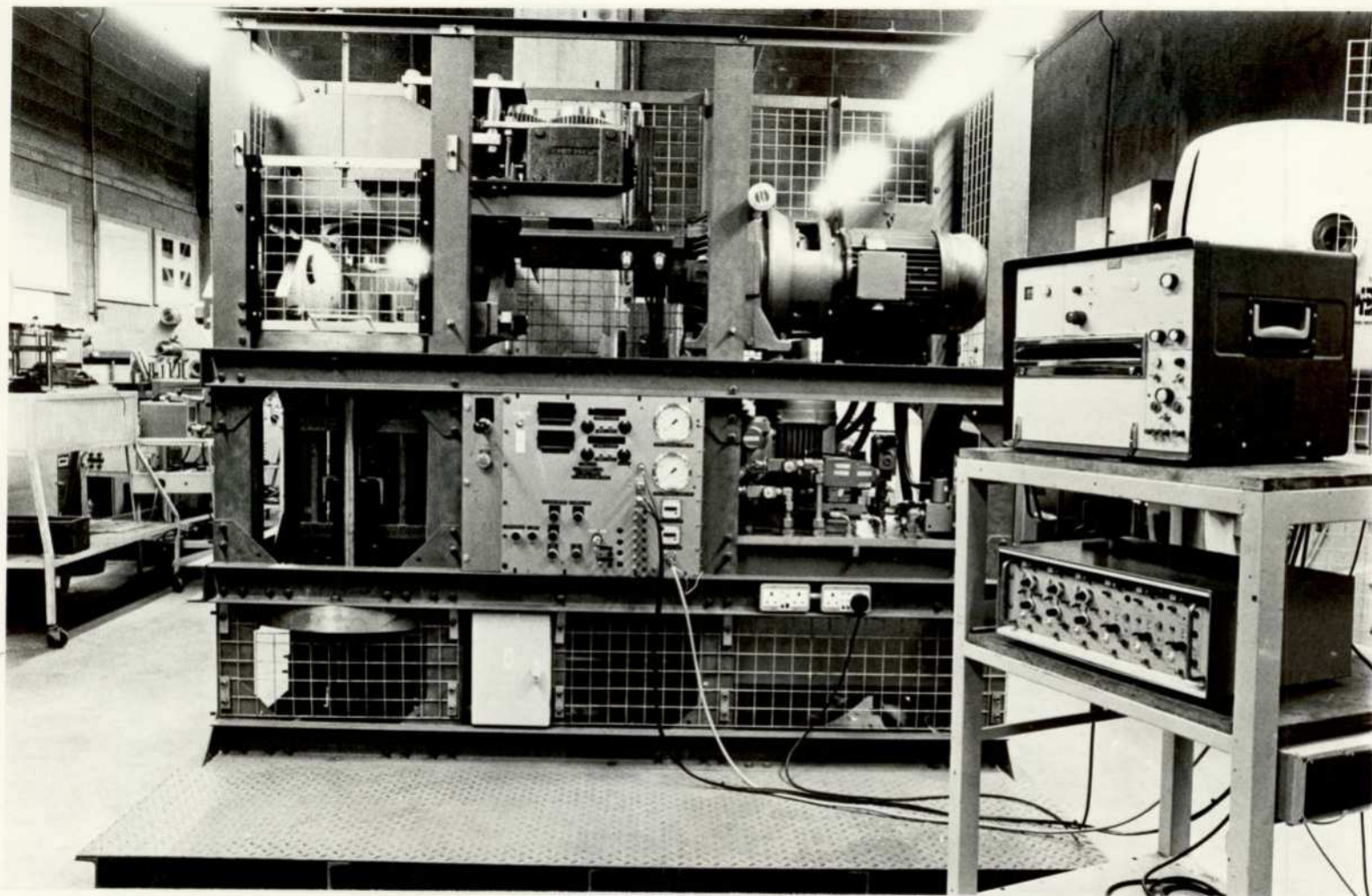


FIGURE 48 THE MACHINE AT THE PRESENT STAGE OF CONSTRUCTION WITH FRONT AND TOP GUARDS REMOVED

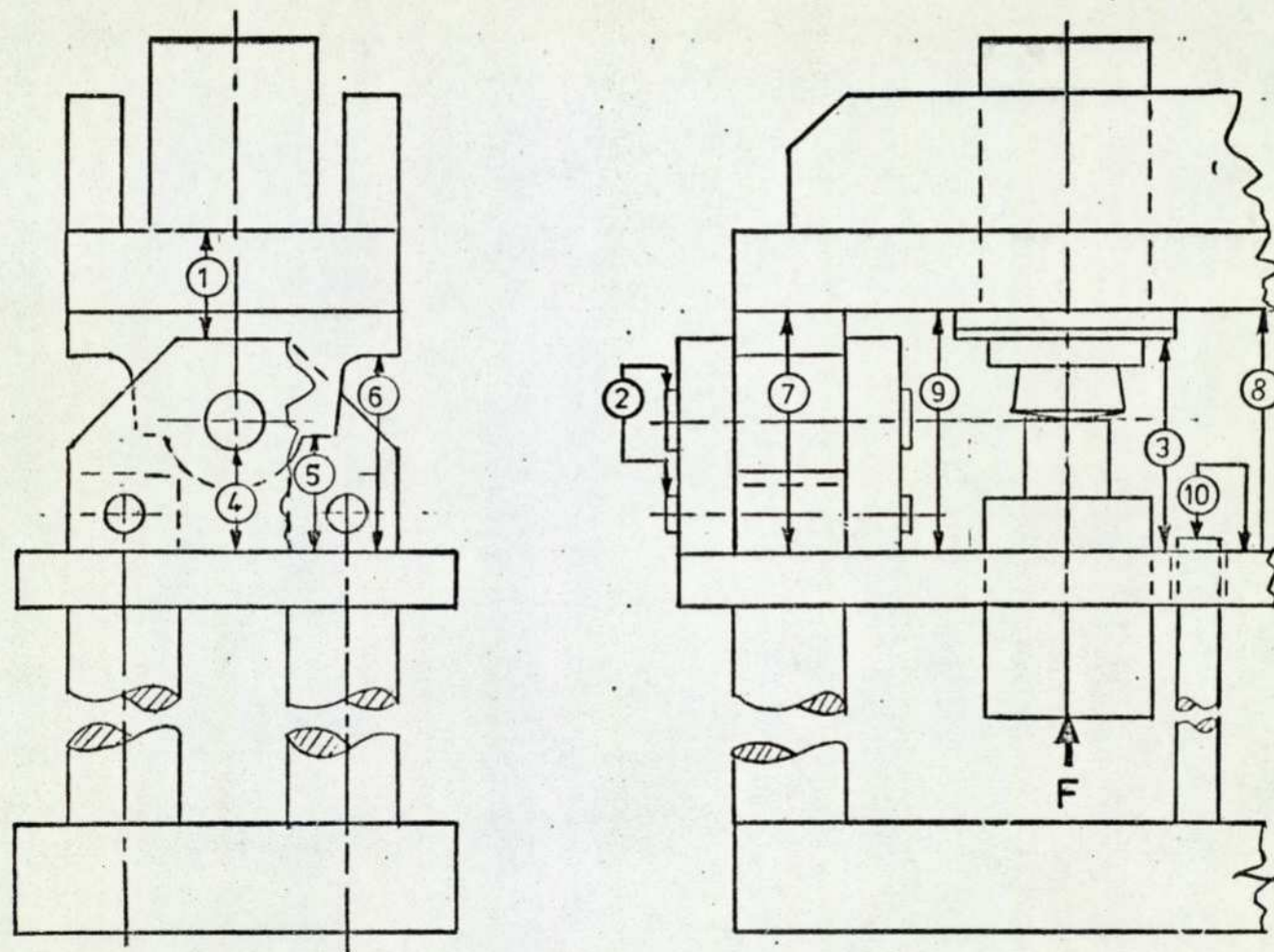


FIGURE 49 POSITIONS OF THE DIAL INDICATORS USED DURING THE COMPLIANCE TESTS

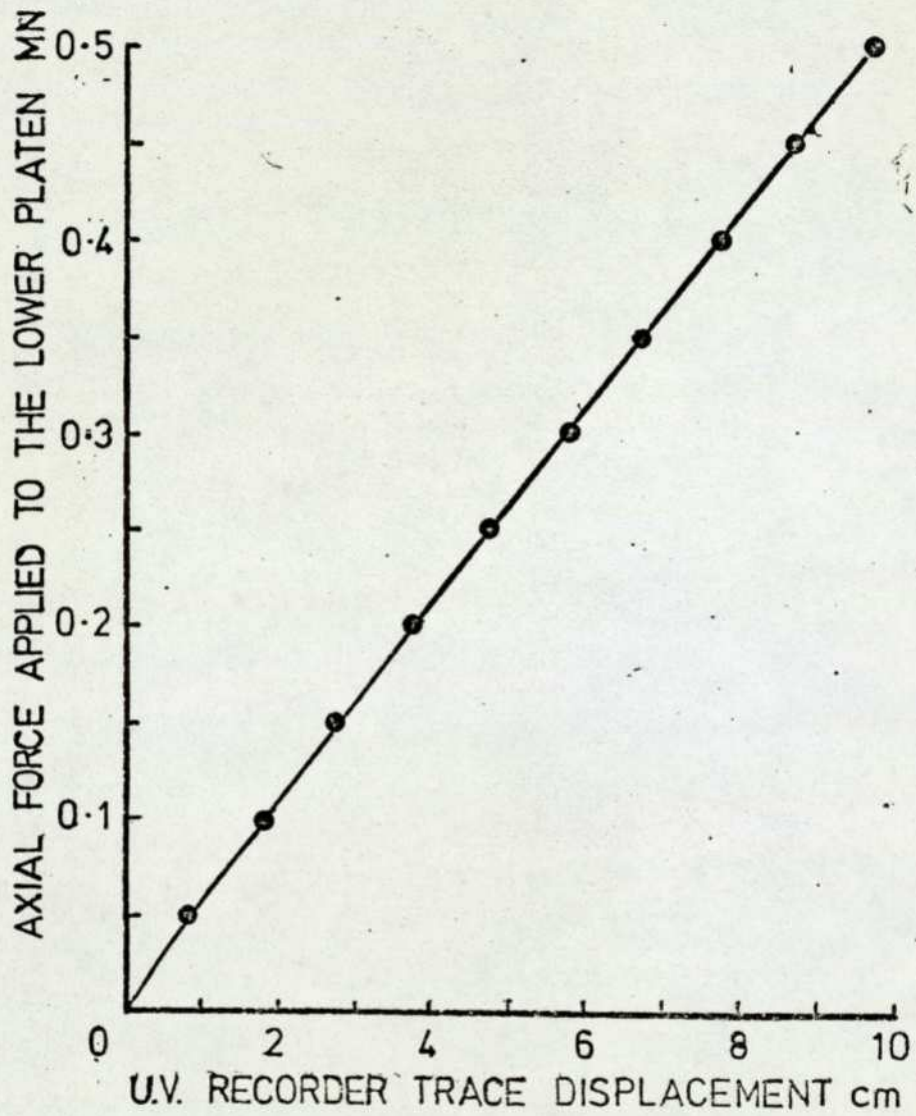


FIGURE 50 RELATION BETWEEN AXIAL FORCE APPLIED TO THE LOWER PLATEN AND U.V. RECORDER TRACE DISPLACEMENT.

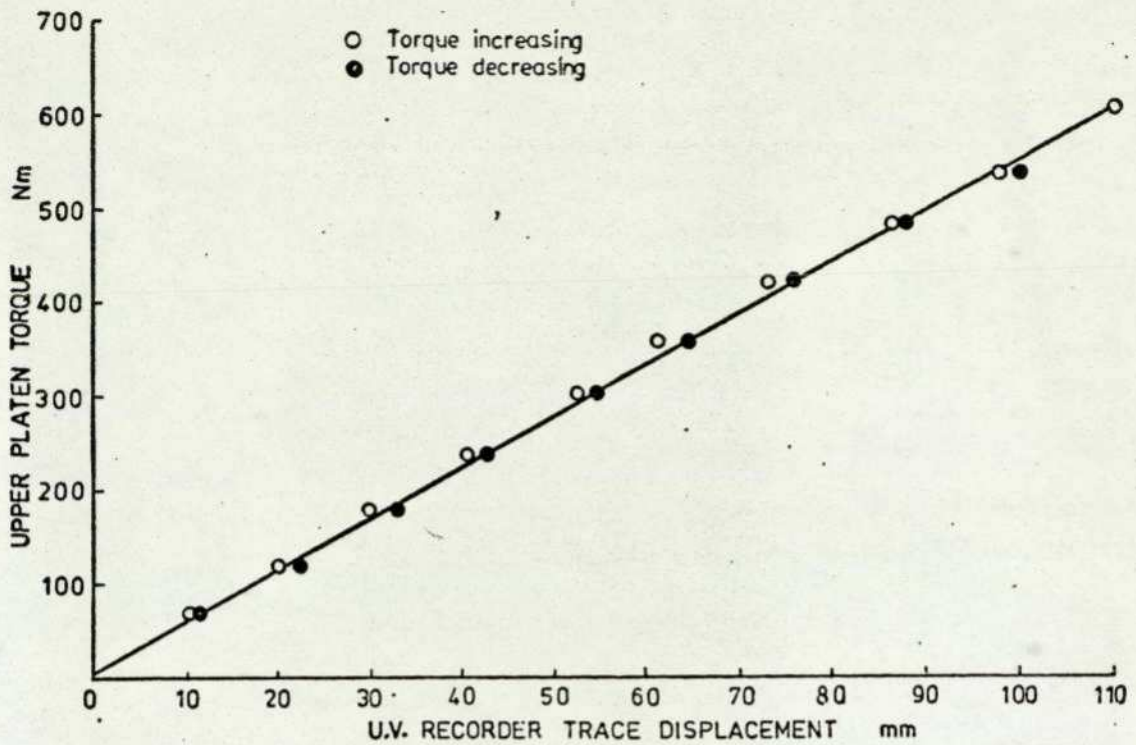


FIGURE 51 RELATION BETWEEN TORQUE APPLIED TO THE UPPER PLATEN AND U.V. RECORDER TRACE DISPLACEMENT.

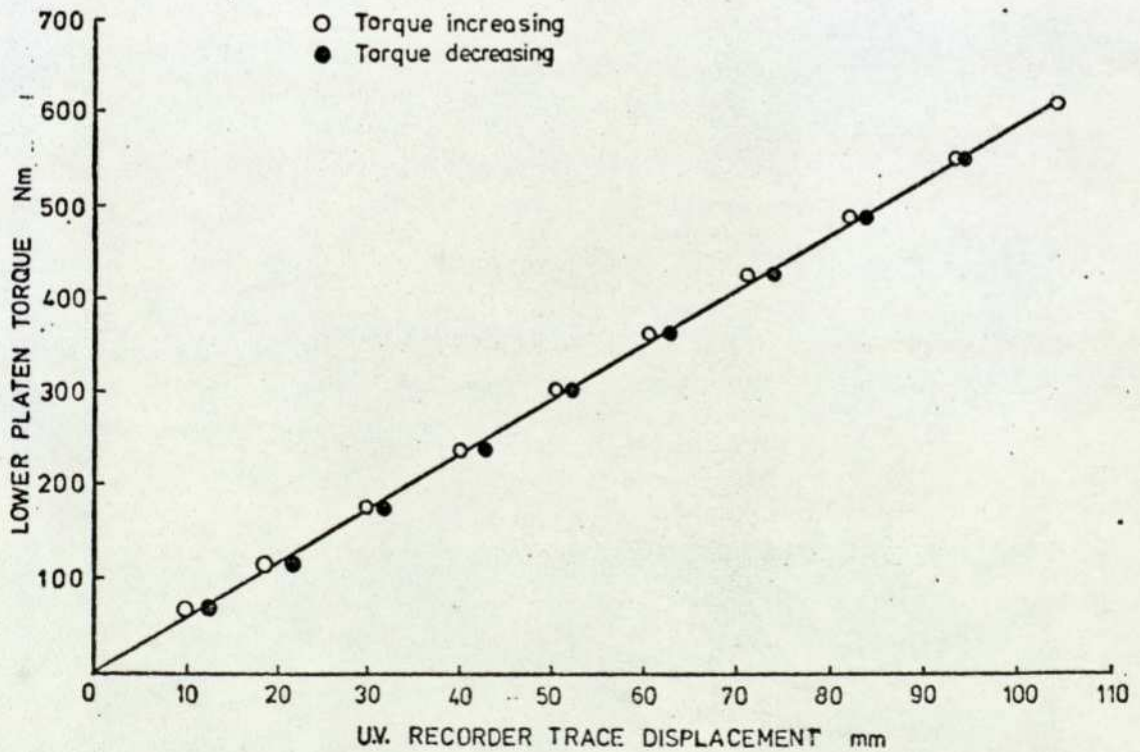


FIGURE 52 RELATION BETWEEN TORQUE APPLIED TO THE LOWER PLATEN AND U.V. RECORDER TRACE DISPLACEMENT.

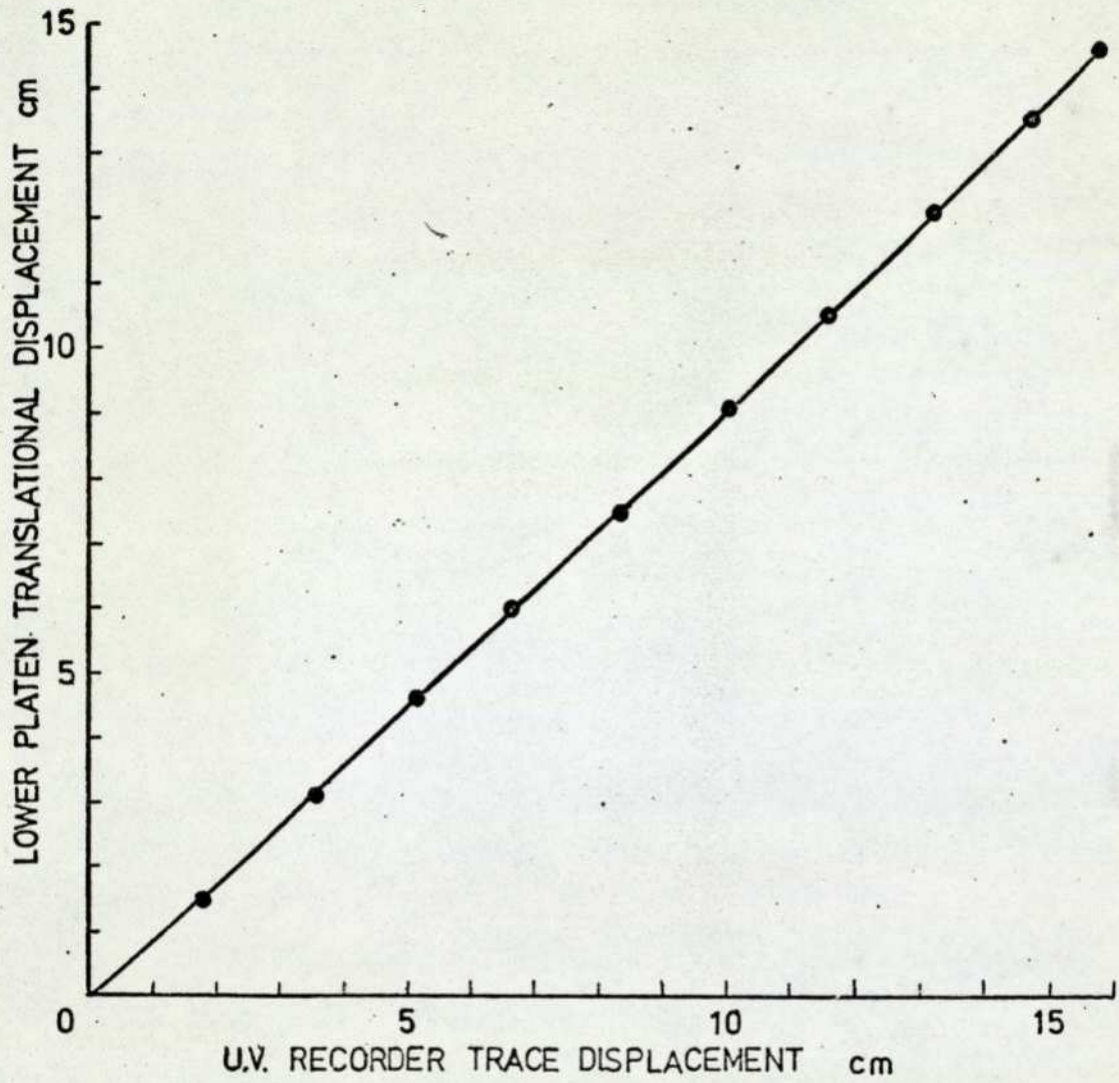


FIGURE 53 RELATION BETWEEN TRANSLATIONAL DISPLACEMENT OF THE LOWER PLATEN AND U.V. RECORDER TRACE DISPLACEMENT.

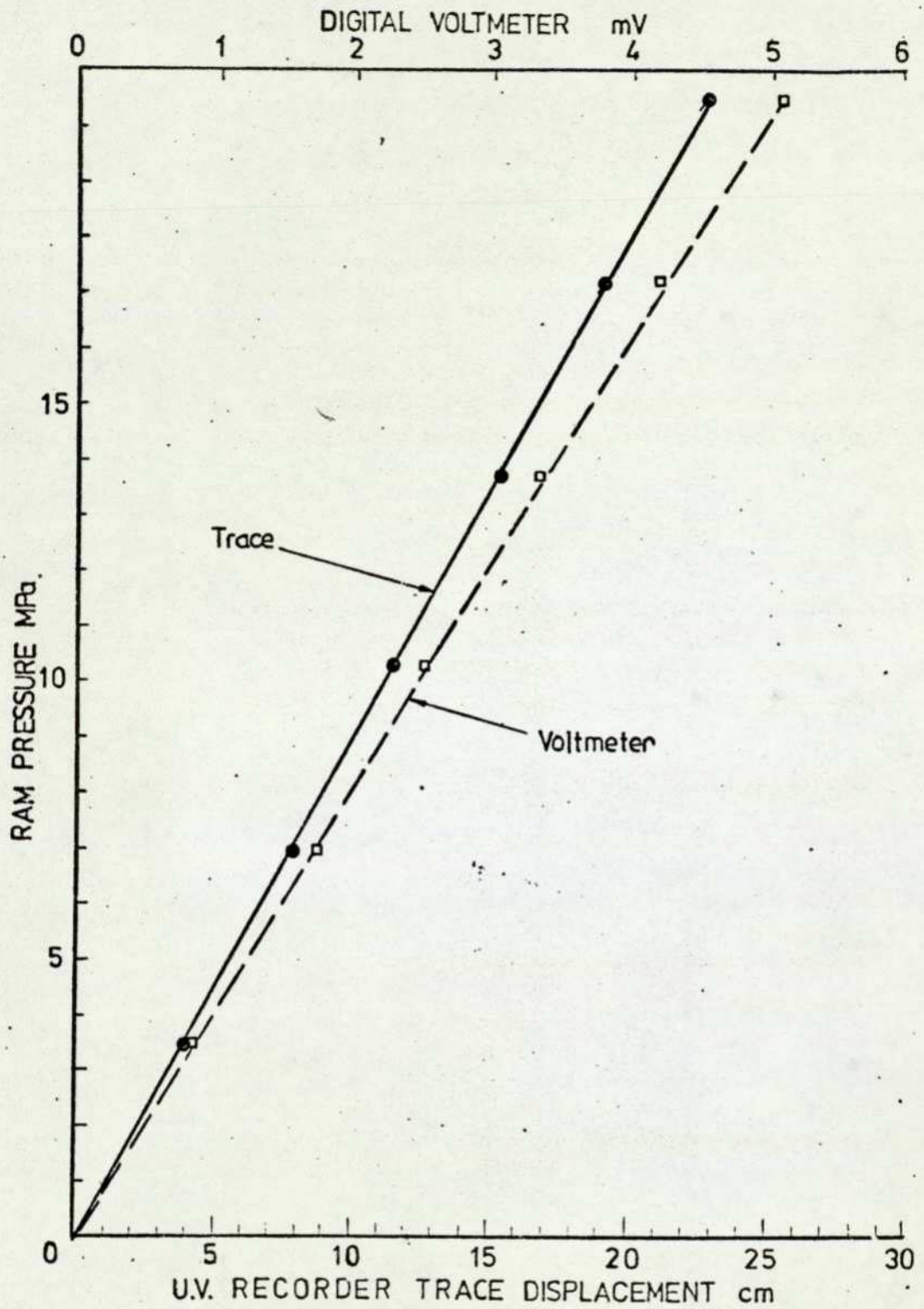


FIGURE 54 RELATION BETWEEN RAM PRESSURE AND U.V. RECORDER TRACE DISPLACEMENT.

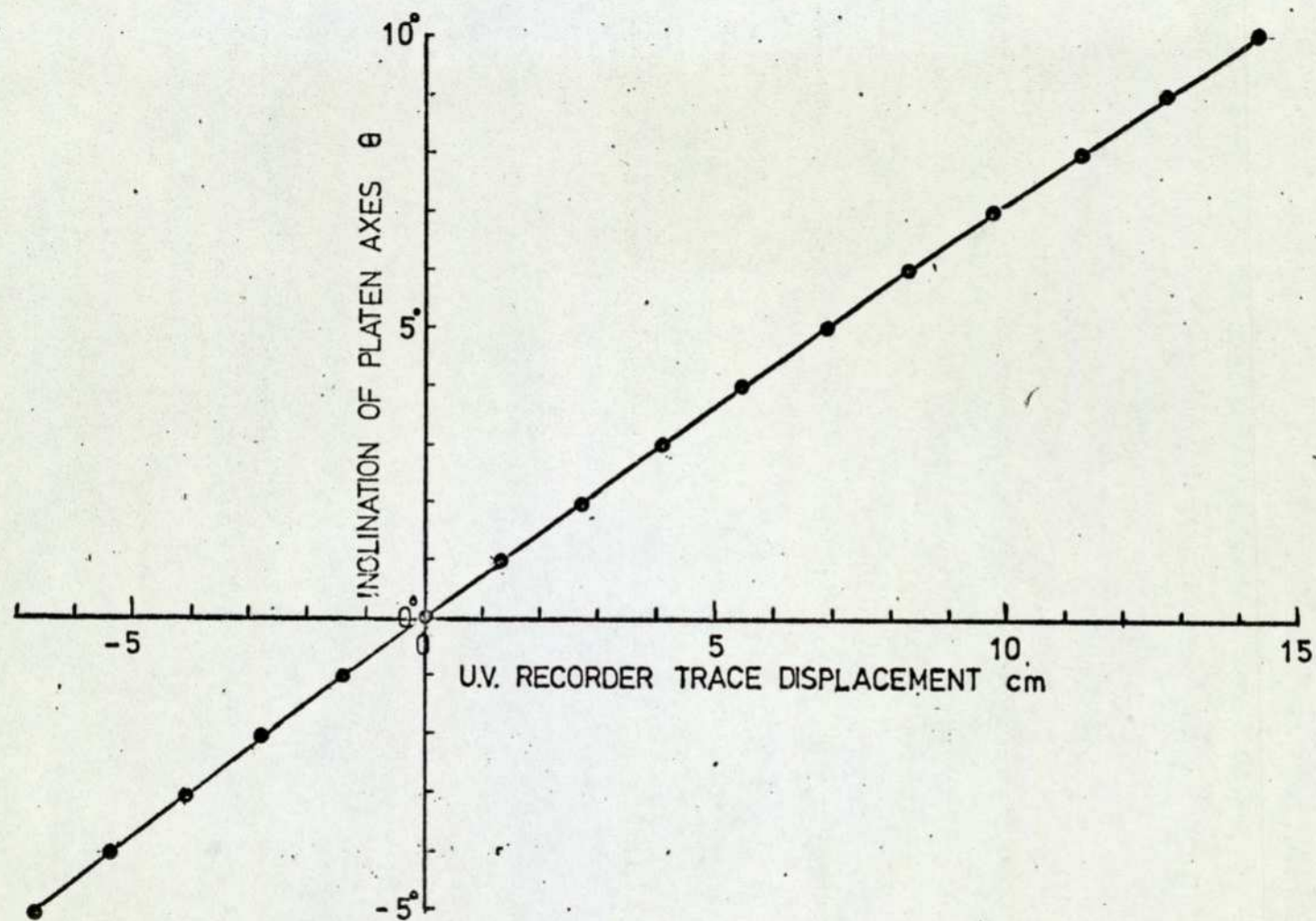


FIGURE 55 RELATION BETWEEN INCLINATION OF PLATEN AXES AND U.V. RECORDER TRACE DISPLACEMENT.

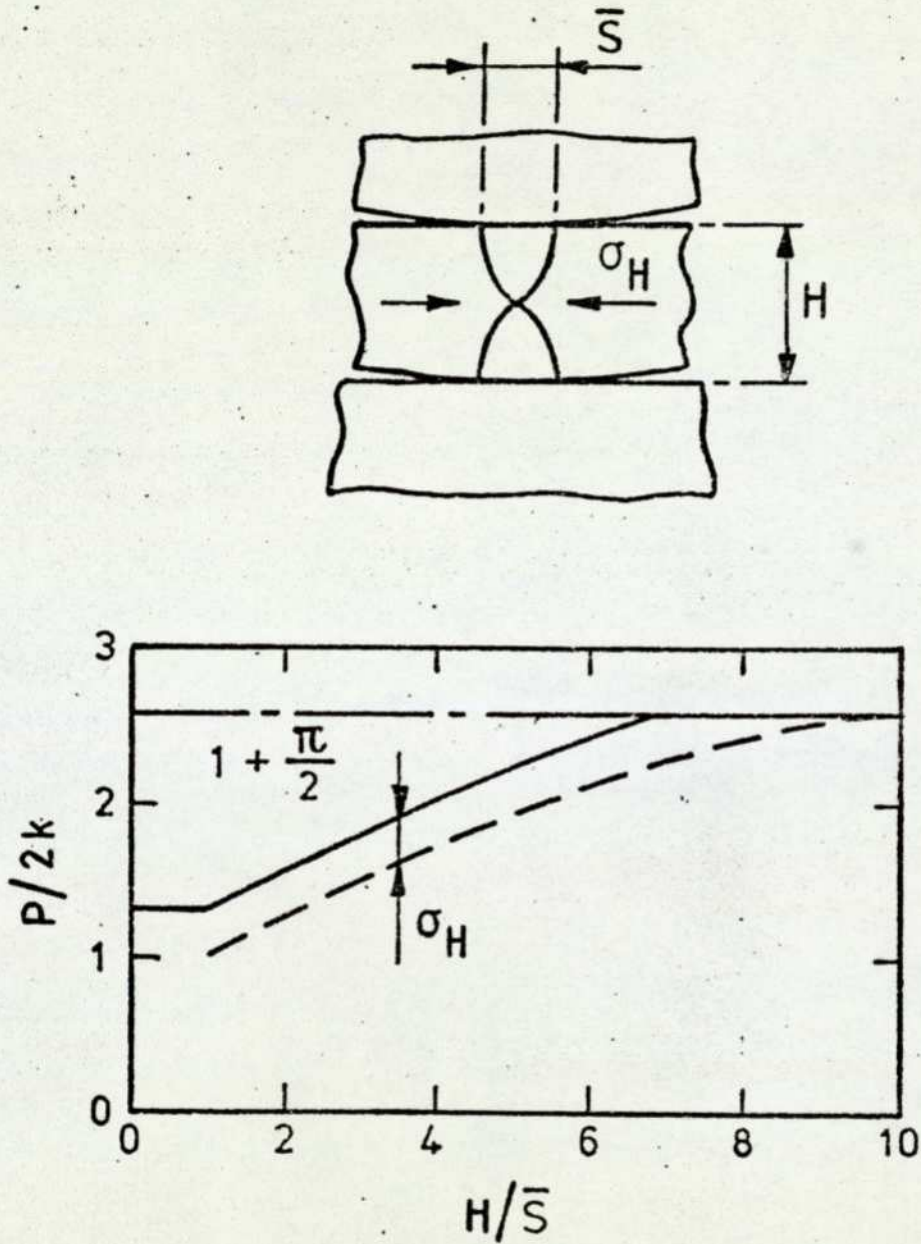


FIGURE 56 DETERMINATION OF DIE PRESSURE, P , USING INDENTATION MODEL ASSUMING THE EXISTENCE OF A PLASTIC HINGE. AFTER HAWKYARD ET AL [24]

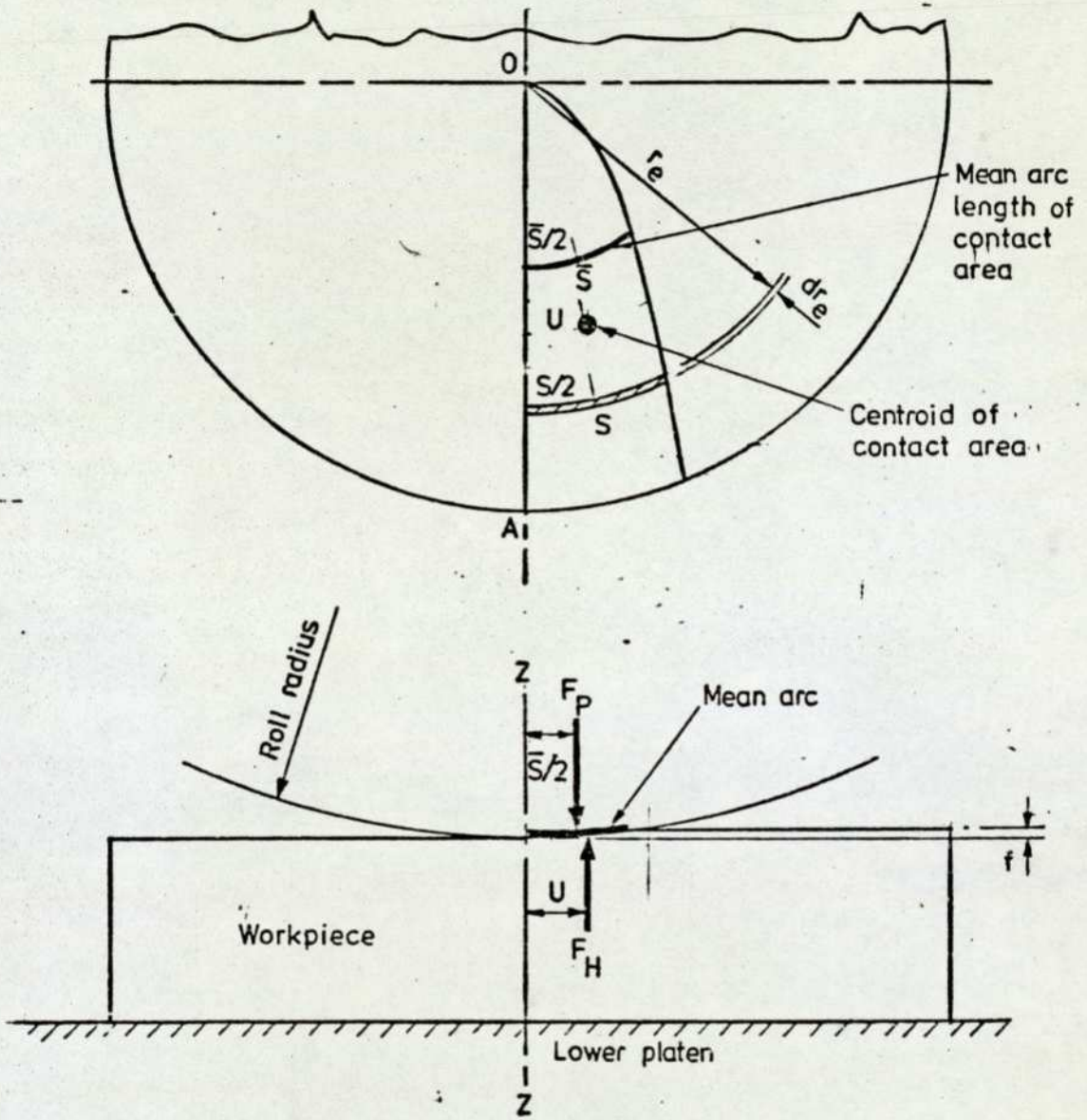


FIGURE 57. DIAGRAM OF FORCES CONSIDERED TO BE ACTING AT UPPER DIE-WORKPIECE INTERFACE DUE TO EXISTENCE OF AN INSTANTANEOUS HINGE.

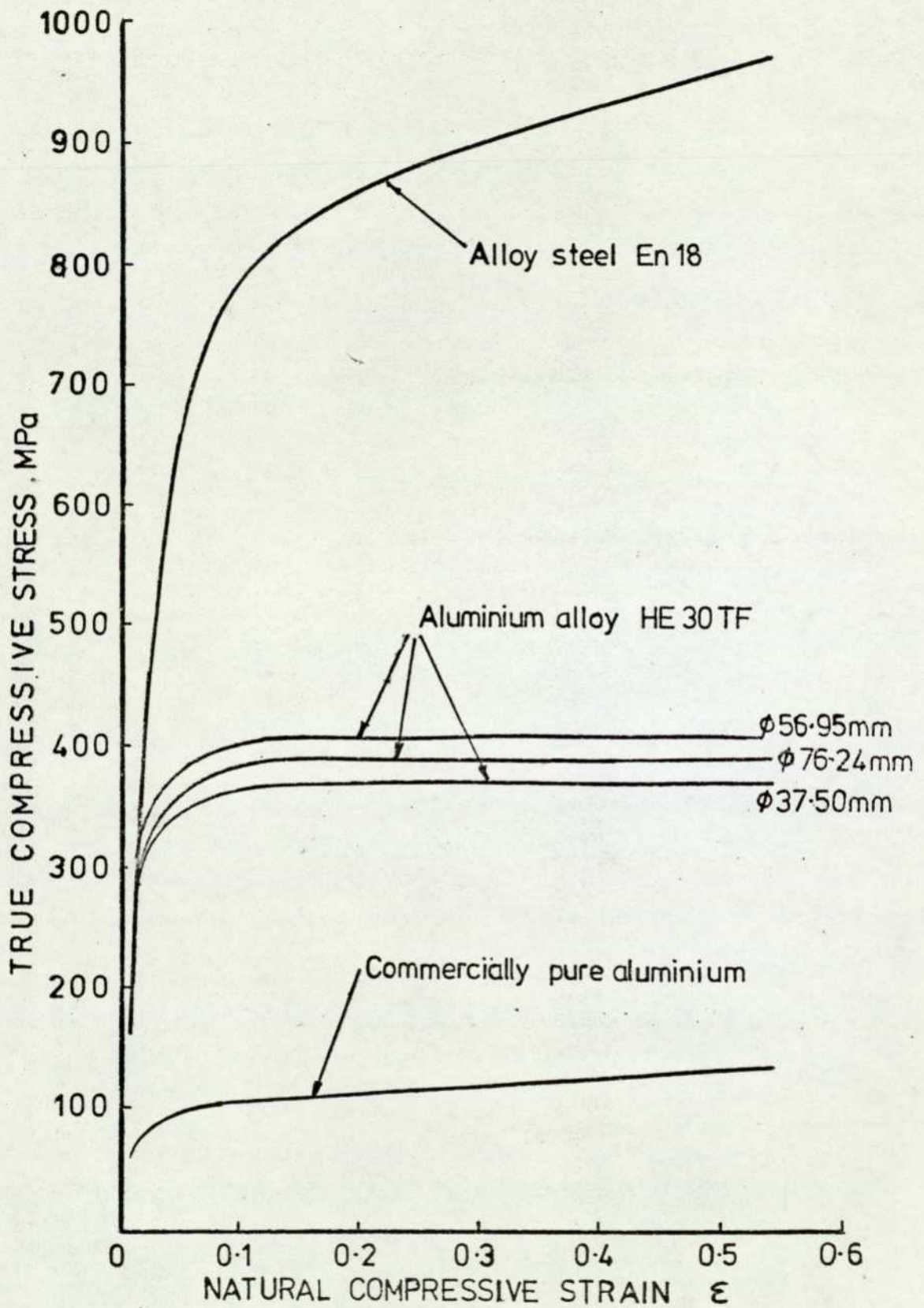


FIGURE 58 TRUE COMPRESSIVE STRESS-NATURAL COMPRESSIVE STRAIN RELATION FOR SPECIMENS USED.

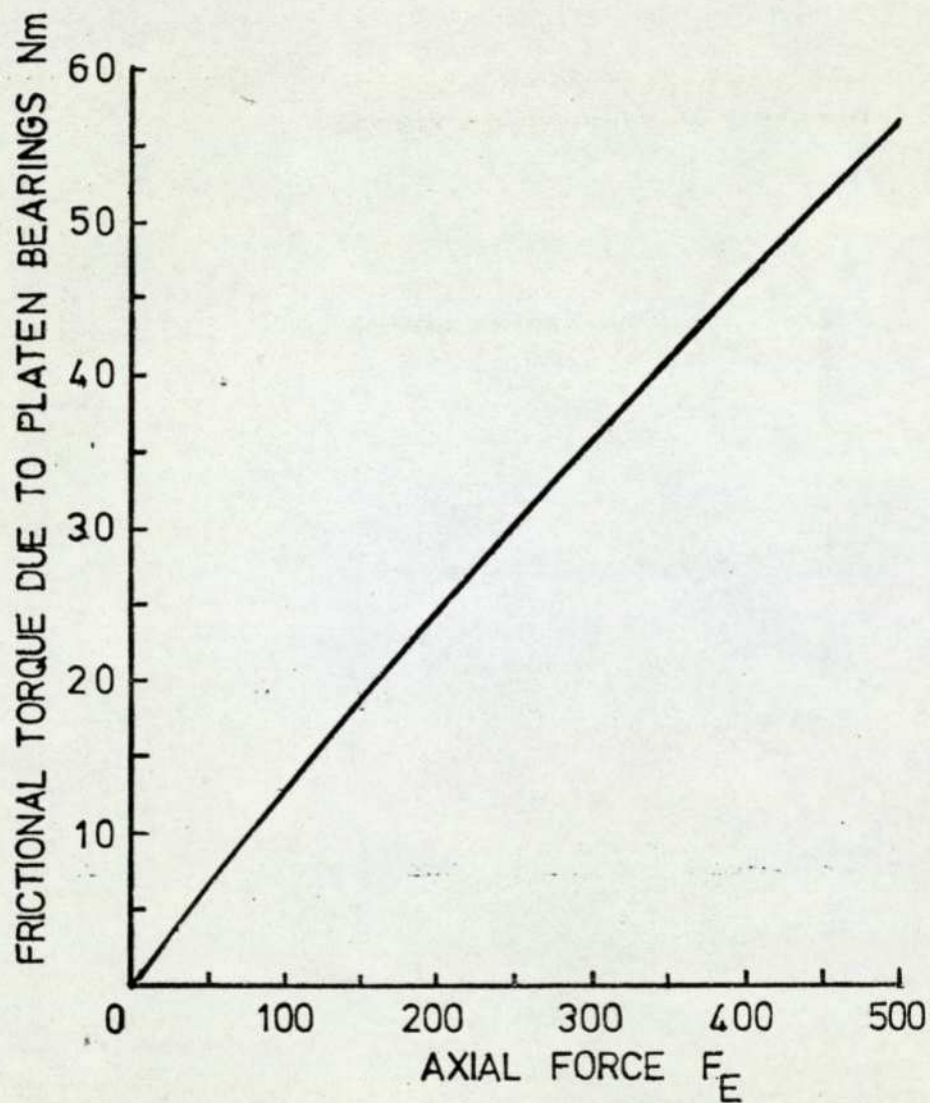
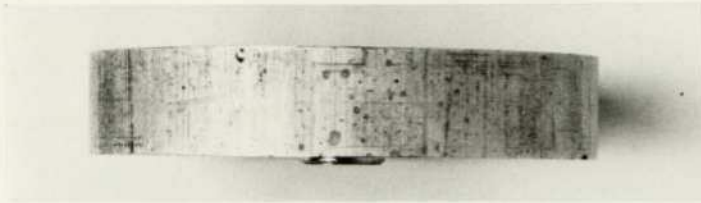


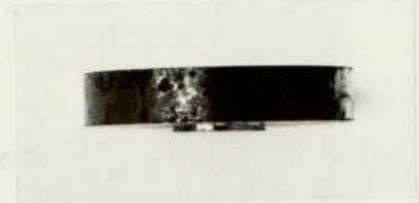
FIGURE 59 FRICTIONAL TORQUE DUE TO PLATEN BEARINGS - AXIAL FORCE
RELATION

Aluminium Alloy HE.30 TF.



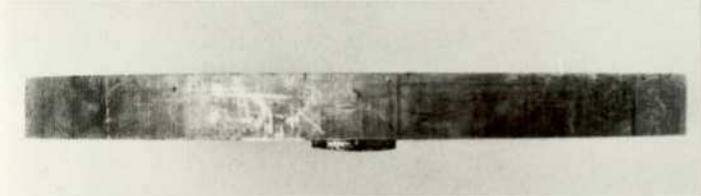
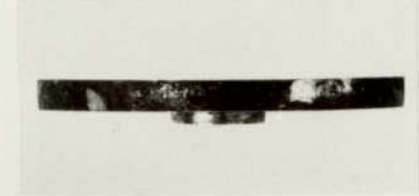
$$D_o = 76.24 \text{ mm} \quad H_o/D_o = 0.2$$

Alloy Steel En.18

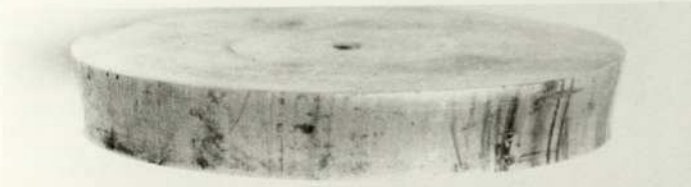
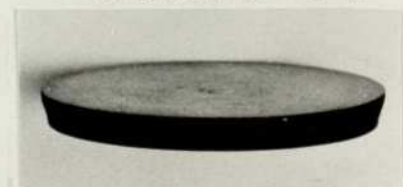


$$D_o = 37.50 \text{ mm} \quad H_o/D_o = 0.2$$

(a) UNDEFORMED SPECIMENS

Upset specimen $R = 0.42$ Upset specimen $R = 0.47$

(b) DEFORMED SPECIMENS

(c) FRACTURED ALUMINIUM ALLOY SPECIMEN
 $R = 0.15$, $F_E = 273 \text{ kN}$, $\theta = 10^\circ$ (d) UPPER SURFACE OF
DEFORMED ALLOY STEEL
SPECIMEN $R = 0.47$ (e) DEPARTURE FROM CYLINDRICAL FORM AT LOW AXIAL FEED/REV DURING
TESTS TO ESTABLISH UPPER LIMIT OF SPECIFIC ROTARY FORGING FORCEFIGURE 60 TYPICAL ALUMINIUM ALLOY AND ALLOY STEEL SPECIMENS
SUBJECTED TO COLD UPSETTING

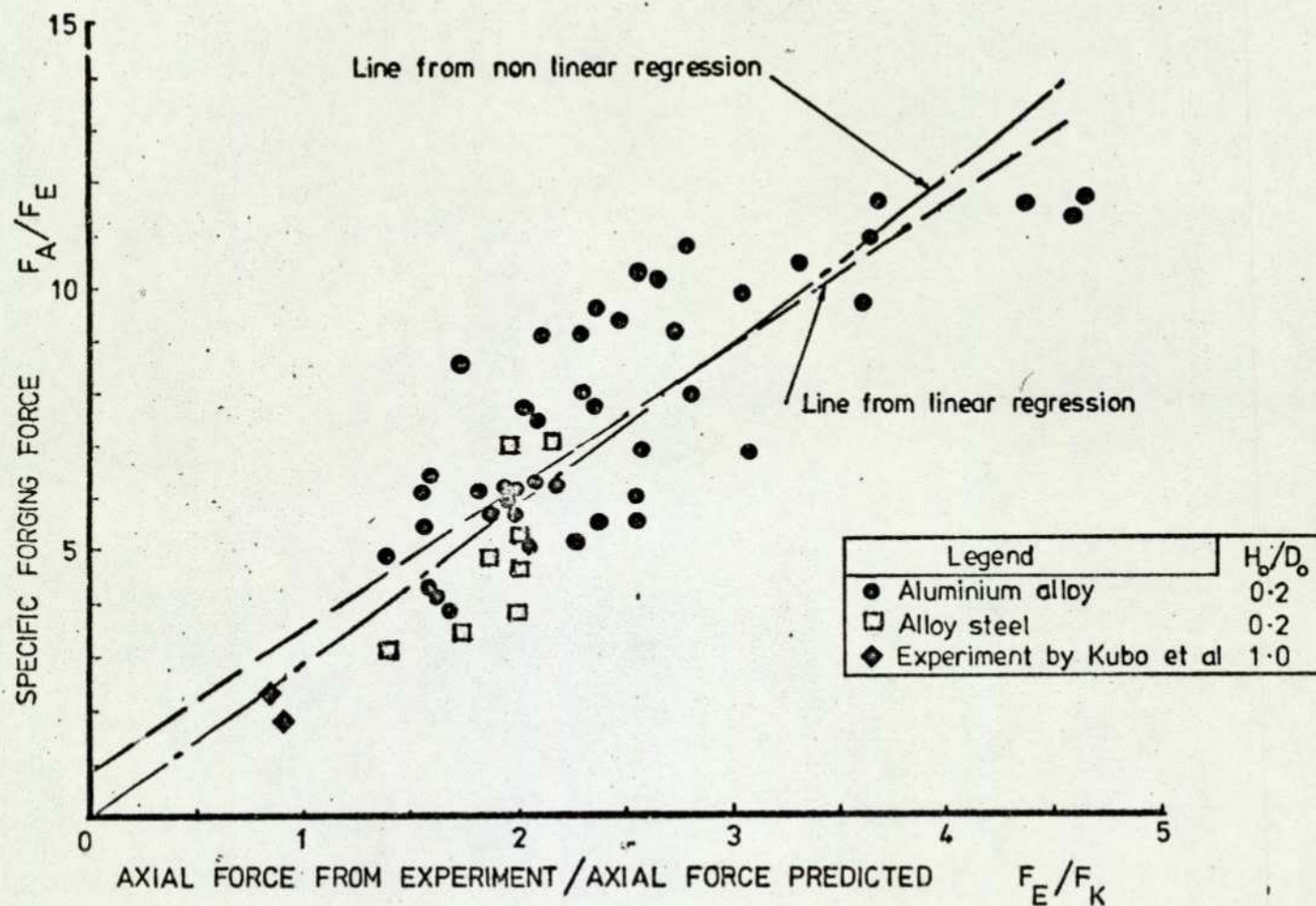


FIGURE 61. SPECIFIC FORGING FORCE - F_E/F_K RELATION DURING UPSETTING OF CYLINDRICAL SPECIMENS, $H_0/D_0 \approx 0.2$, OF VARIOUS DIAMETERS, MATERIAL AND INCLINATION OF PLATEN AXES, θ .

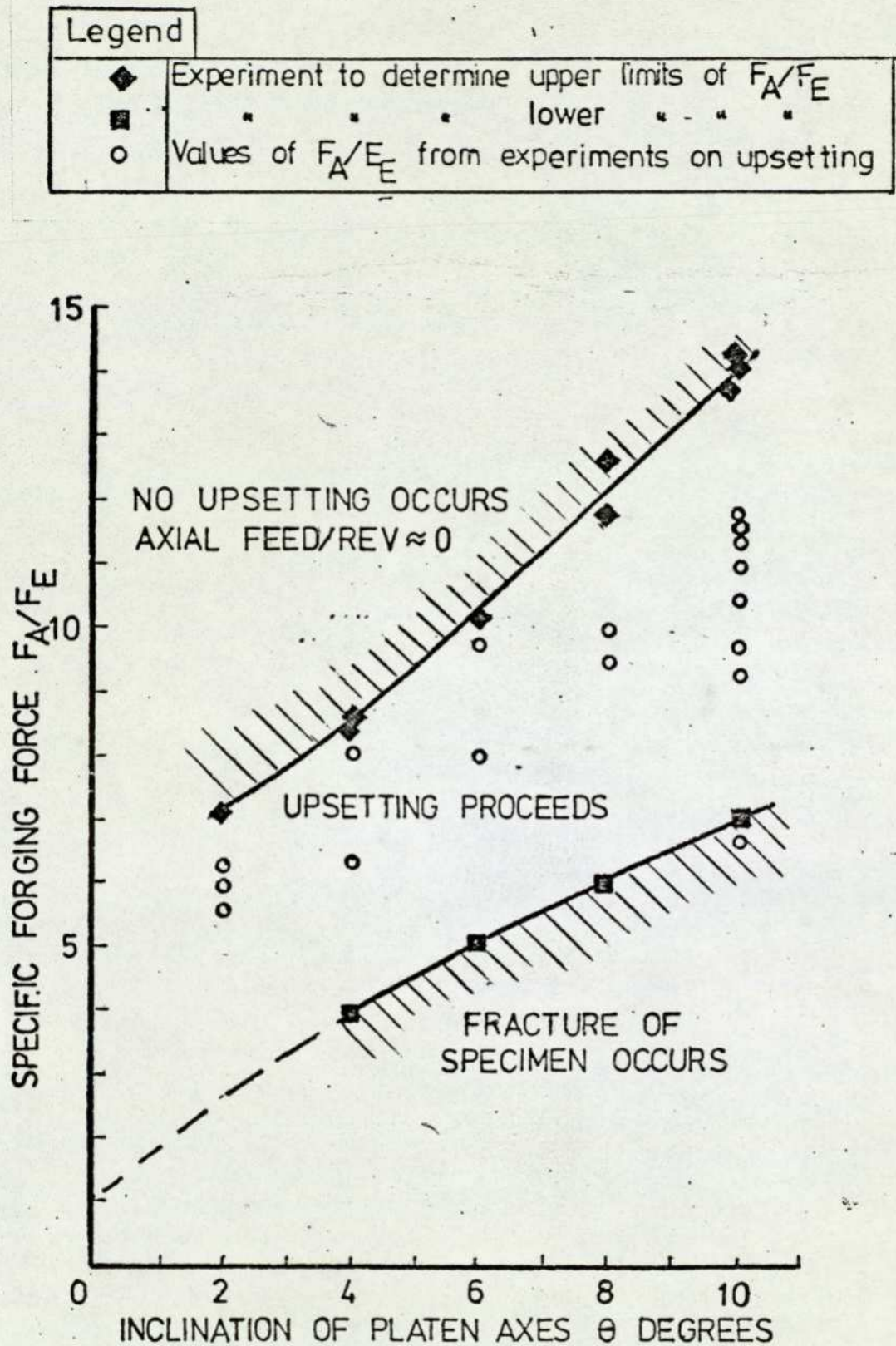


FIGURE 62. RELATION BETWEEN SPECIFIC ROTARY FORGING FORCE F_A/F_E AND INCLINATION OF PLATEN AXES, θ , DURING UPSETTING OF ALUMINIUM ALLOY SPECIMENS $D_0 = 76.24$, $H_0/D_0 \approx 0.2$.

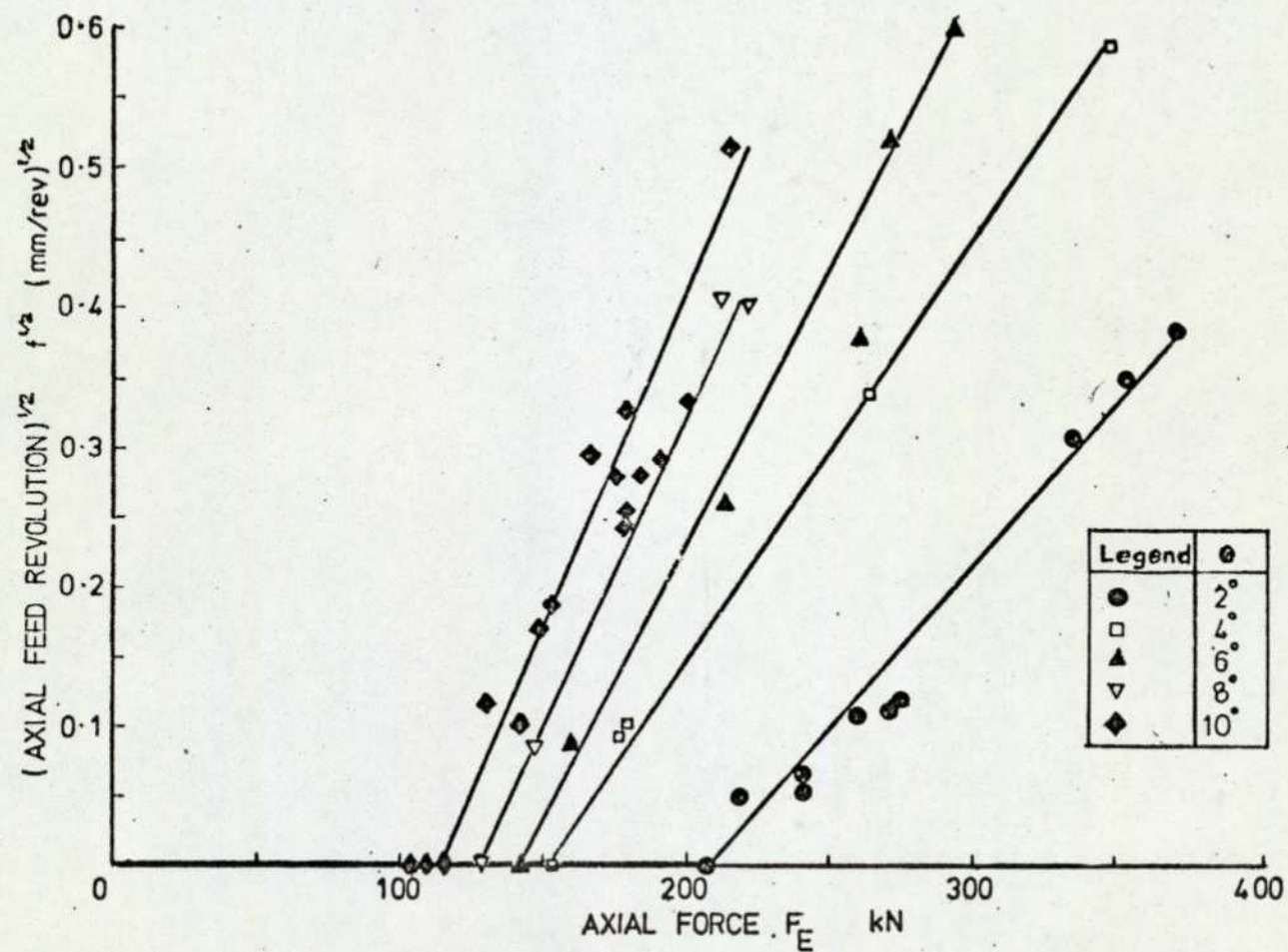


FIGURE 63 AXIAL FEED/REV, f , AXIAL FORCE, F_E ; RELATION FOR COLD UPSETTING CYLINDRICAL ALUMINIUM ALLOY SPECIMENS $D_0 = 76.24$ mm, $H_0/D_0 \cong 0.2$ AND VARIOUS VALUES OF INCLINATION OF PLATEN AXES.

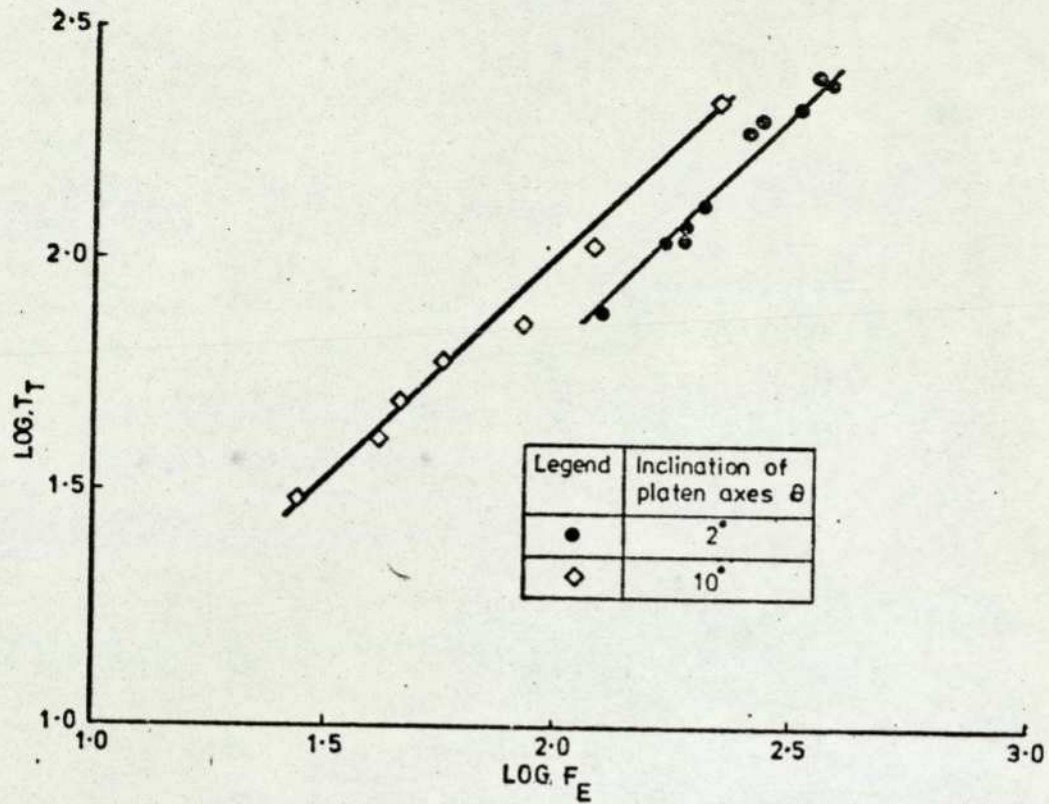


FIGURE 64 TOTAL ROTATIONAL DRIVE TORQUE, T_T - AXIAL FORCE, F_E , RELATION FOR COLD UPSETTING CYLINDRICAL ALUMINIUM ALLOY SPECIMENS $D_0 = 76.24$ mm, $H_0/D_0 \cong 0.2$, $\theta = 2^\circ$ and 10° .

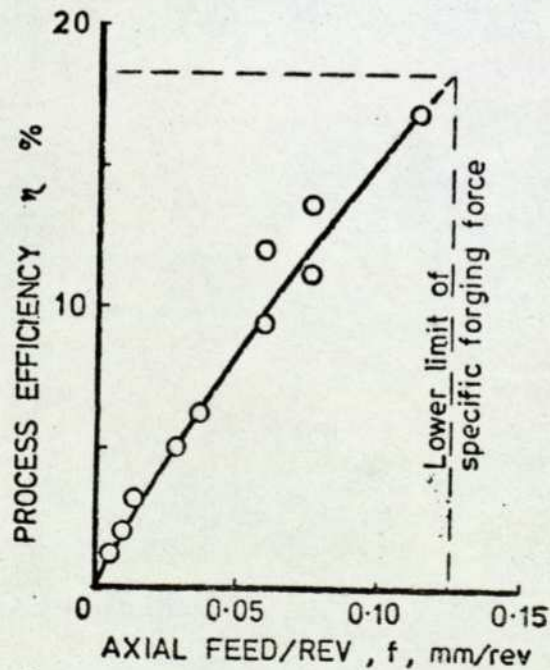


FIGURE 65 PROCESS EFFICIENCY, η - AXIAL/REV, f , RELATION FOR COLD UPSETTING OF CYLINDRICAL ALUMINIUM ALLOY SPECIMENS $D_0 = 76.24$ mm, $H_0/D_0 \cong 0.2$, $\theta = 10^\circ$

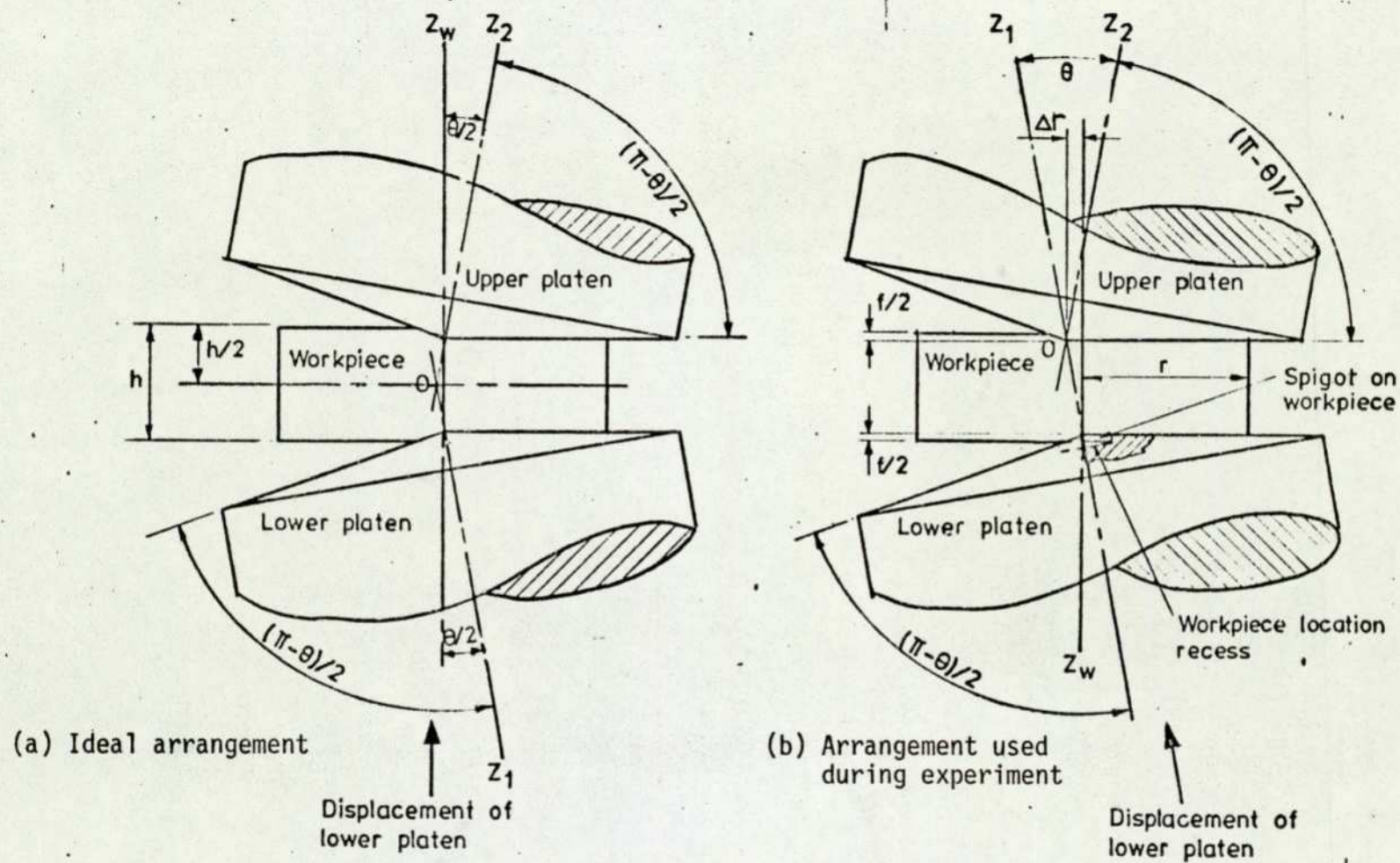


FIGURE 66 DIAGRAM OF KINEMATIC ARRANGEMENTS FOR UPSETTING USING CONICAL UPPER AND LOWER DIES.

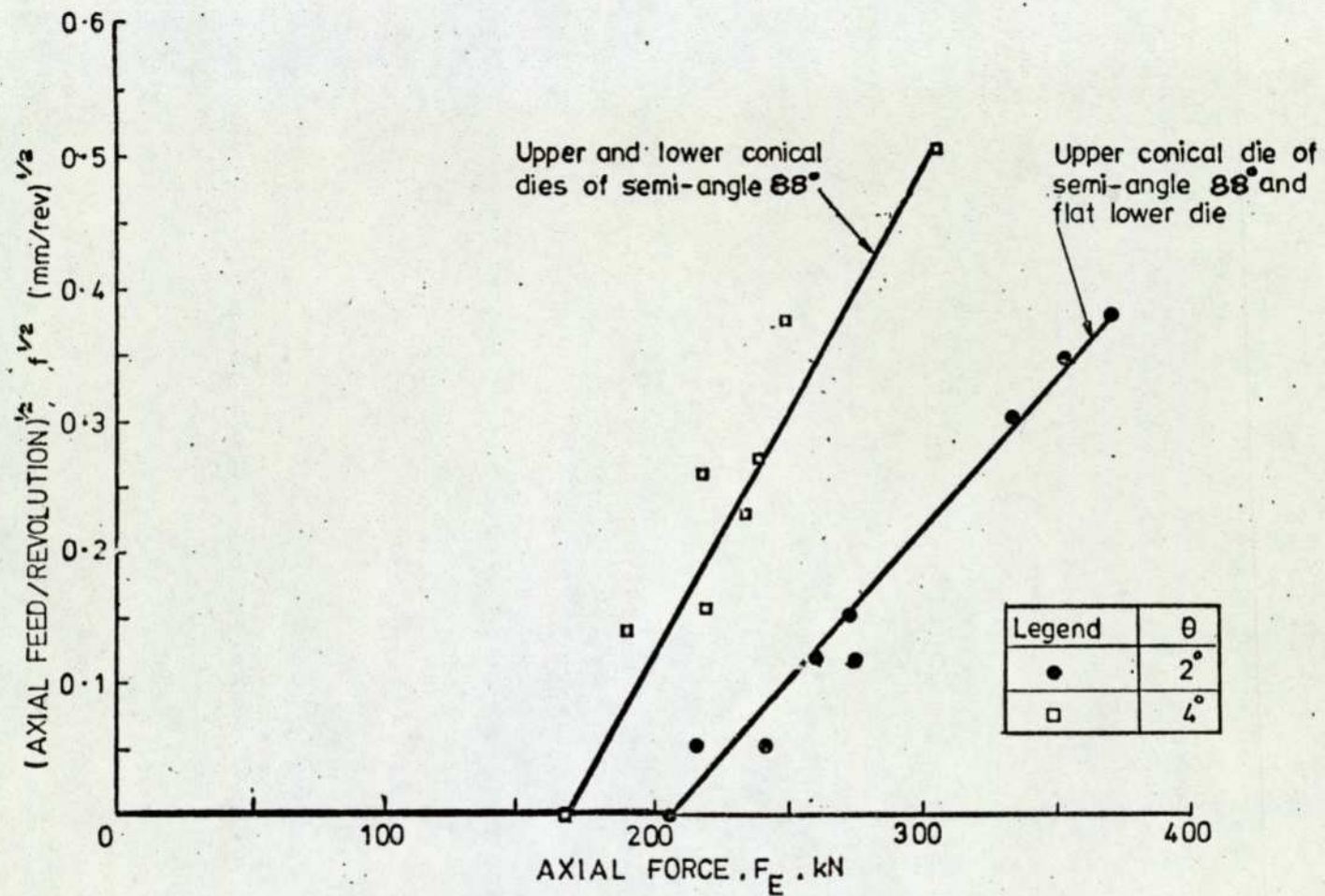
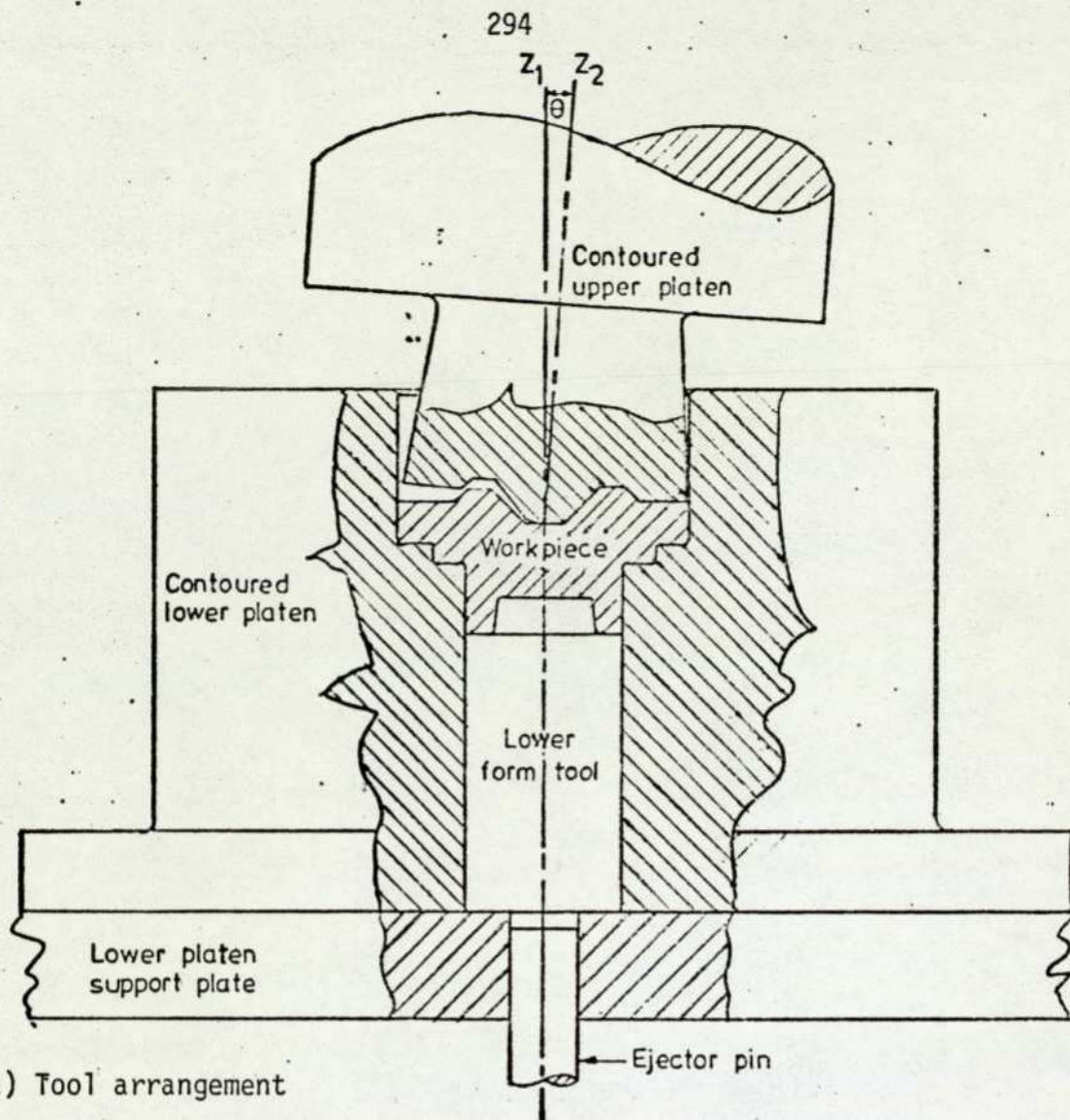
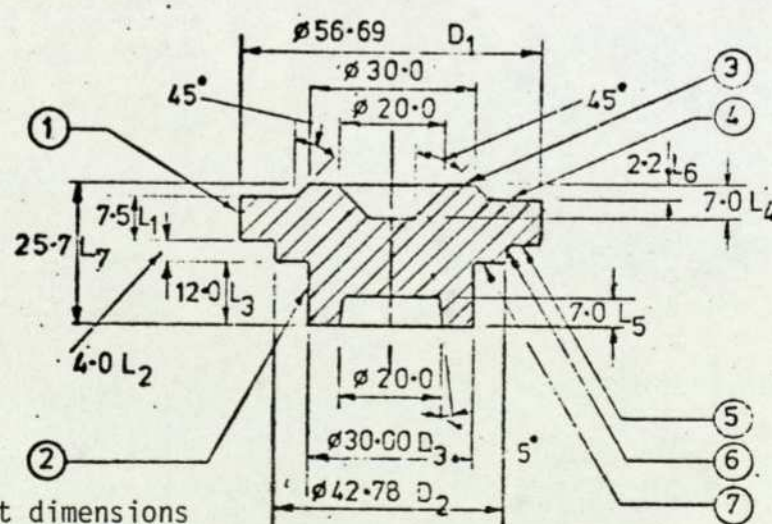


FIGURE 67 AXIAL FEED/REV, f , -AXIAL FORCE, F_E , RELATION FOR UPSETTING CYLINDRICAL ALUMINIUM ALLOY SPECIMENS
 $D_0 = 76.24$ mm DIAMETER, $H_0/D_0 = 0.2$, USING SINGLE AND DUAL DIES OF SEMI-CONICAL ANGLE 88°



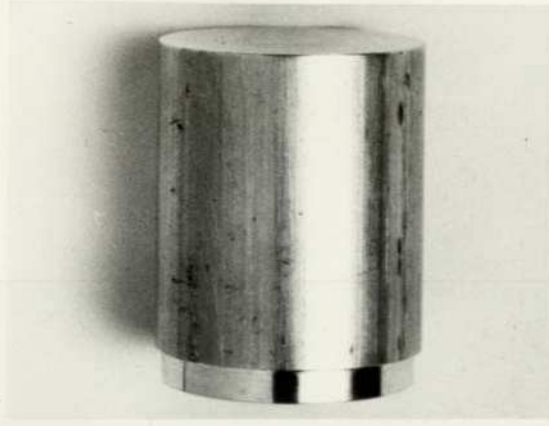
(a) Tool arrangement



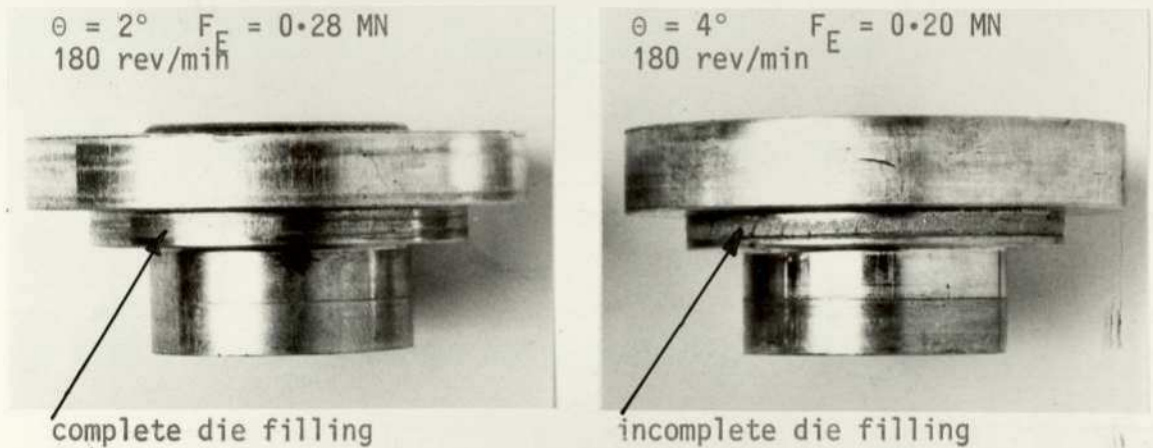
(b) Component dimensions

Dimensions: mm

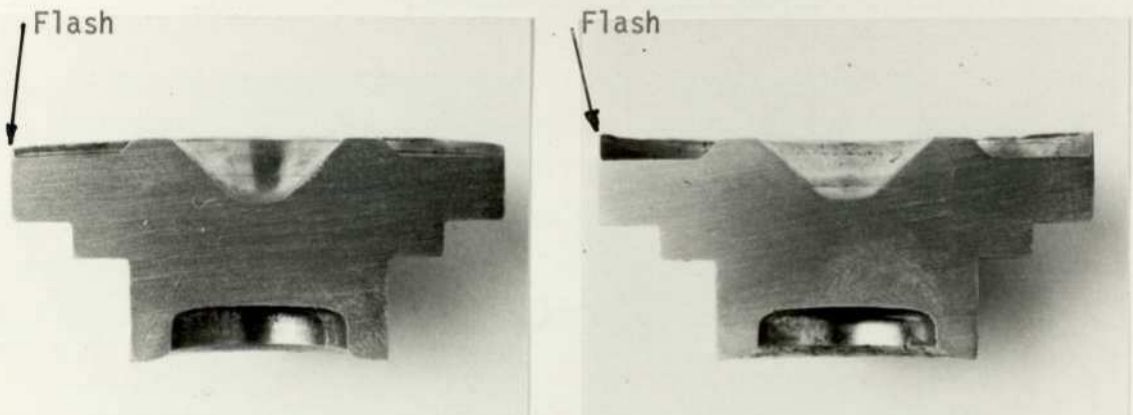
FIGURE 68 DIAGRAM OF THE TOOL ARRANGEMENT AND DIMENSIONS OF THE AXISYMMETRIC COMPONENT FORMED USING CONTOURED UPPER AND LOWER DIES



(a) Undeformed commercially pure aluminium billet

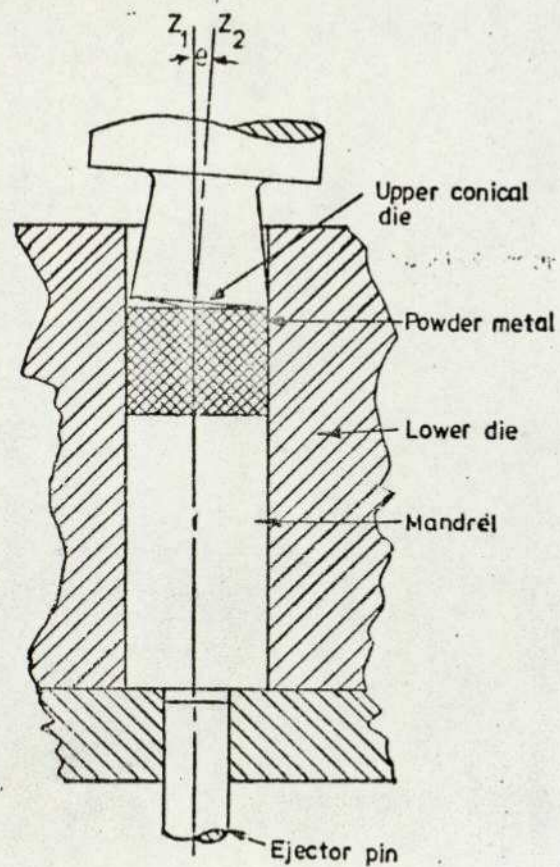


(b) Typical formed components showing differing surface roughness characteristics on diameter reference D_2 Figure 68 resulting from the forging parameters stated.

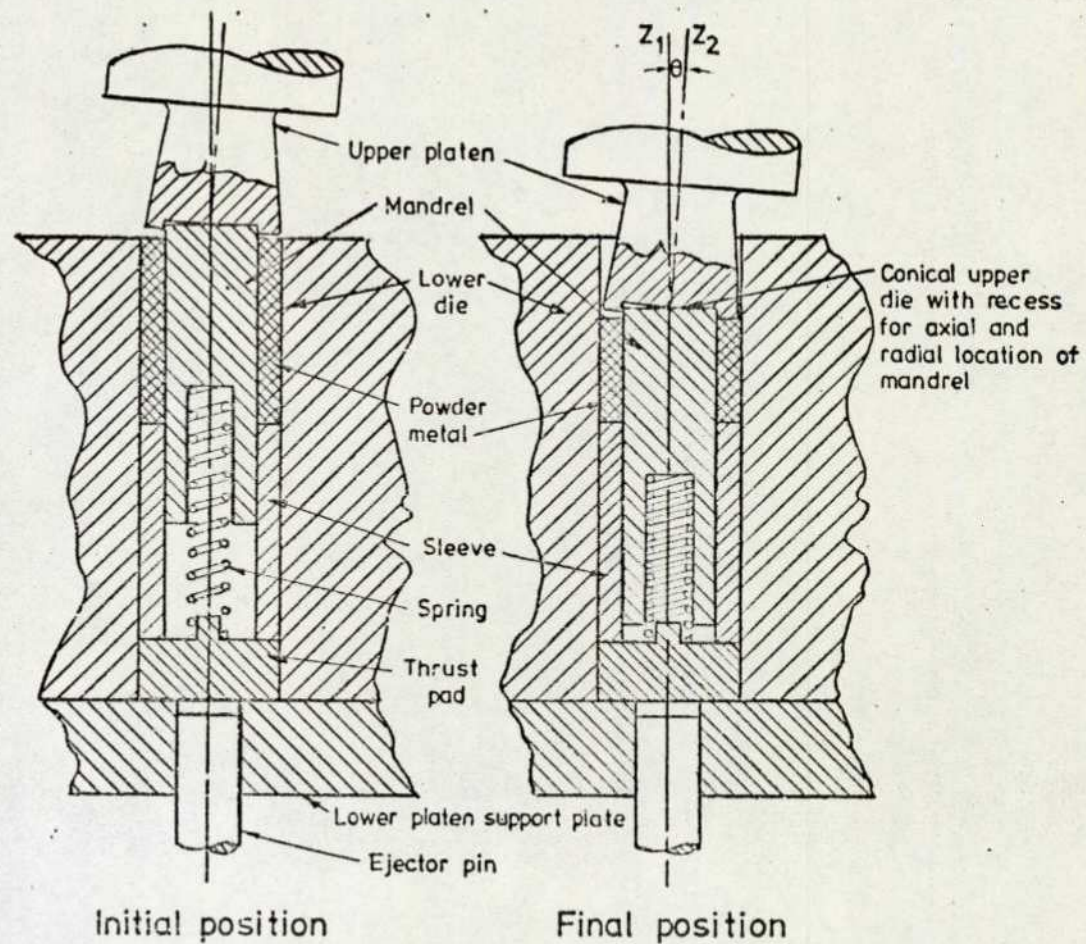


(c) Typical formed components showing differing *flash* heights resulting from the forging parameters stated.

FIGURE 69 TYPICAL SPECIMENS FORMED BY CONTOURED UPPER AND LOWER CLOSED DIES WITH DIFFERING FORGING PARAMETERS.



(a) Die arrangement for solid cylinders.



(b) Die arrangement for hollow cylinders.

FIGURE 70 UPPER AND LOWER DIE ARRANGEMENTS USED FOR COMPACTION AND POST SINTERED DENSIFICATION.

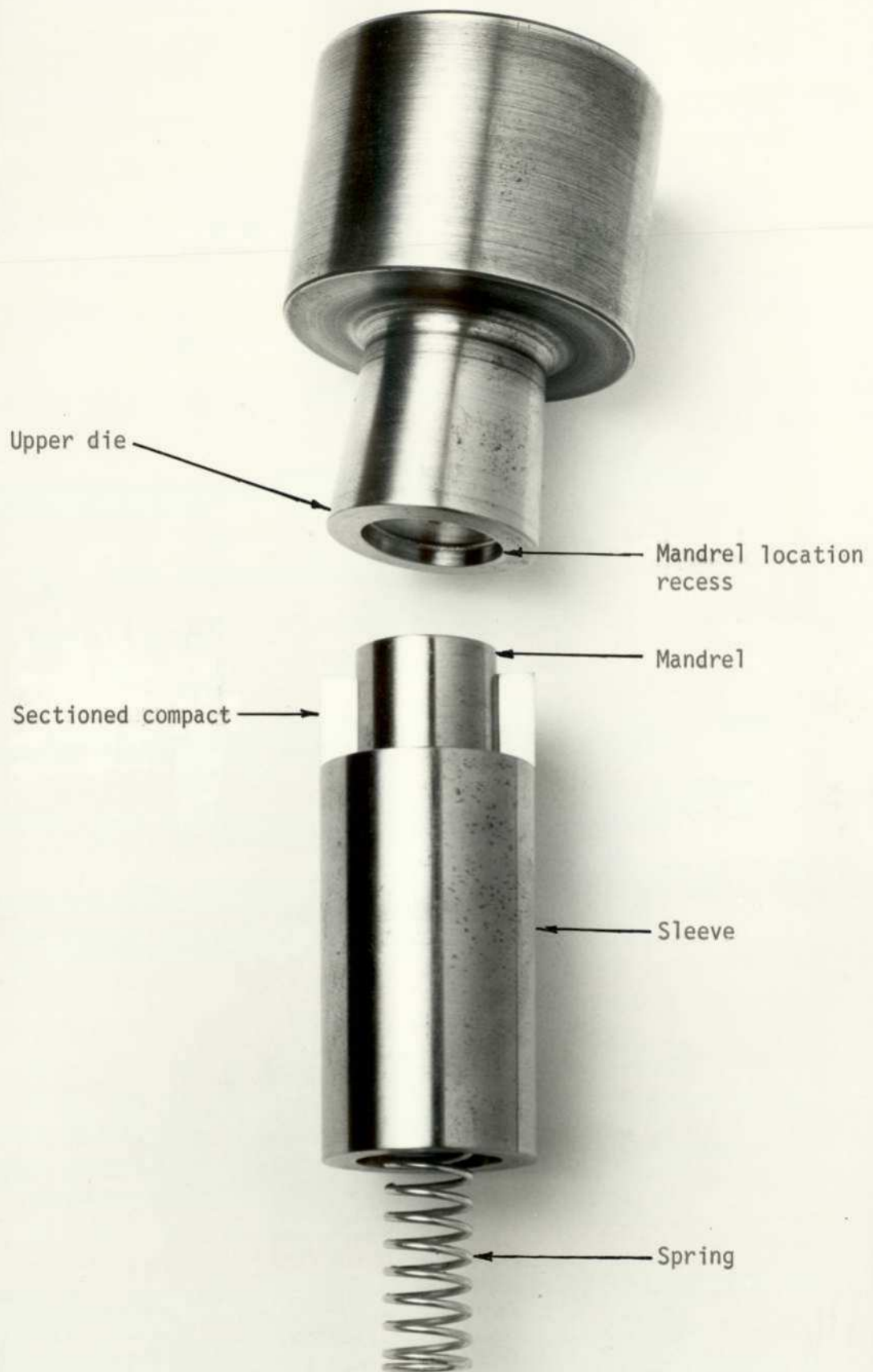
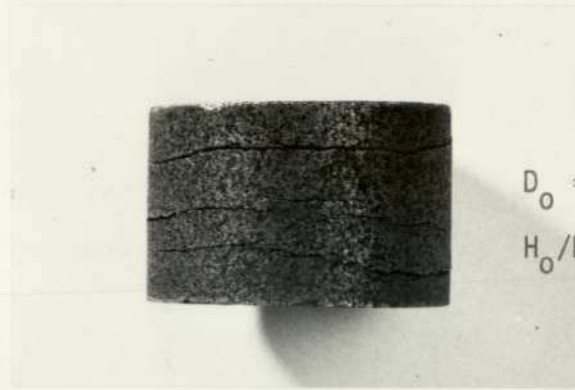


FIGURE 71 ARRANGEMENT OF TOOLS USED IN CONJUNCTION WITH A HOLLOW CYLINDRICAL DIE TO PRODUCE IRON POWDER METAL HOLLOW CYLINDRICAL COMPACTS 30 mm EXTERNAL DIAMETER AND 20 mm INTERNAL DIAMETER.



$$D_o = 30 \text{ mm}$$

$$H_o/D_o = 0.62$$

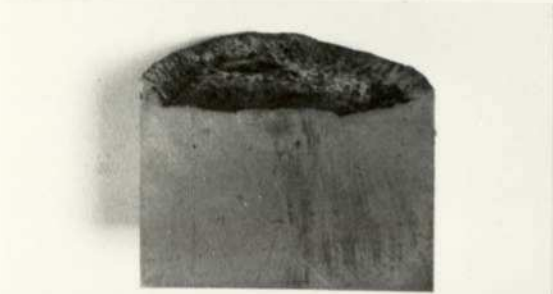
- (a) Typical post-sintered specimen with cracking and segregation compacted prior to die exit modification.



View showing upper surface of segment



View showing lower surface of segment



- (b) Typical post-sintered solid cylindrical specimens exhibiting segregation phenomenon when attempting to achieve relative average density > 0.96 during compaction.



Internal diameter 14 mm
External diameter 30 mm

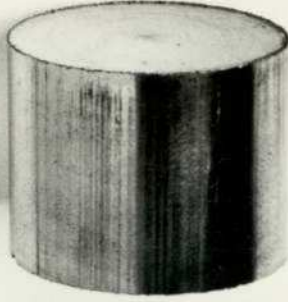


Internal diameter 20 mm
External diameter 30 mm

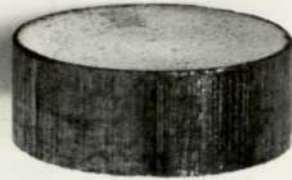
- (c) Typical hollow cylindrical specimens exhibiting segregation phenomenon when attempting to achieve relative average density > 0.96 during compaction

FIGURE 72 TYPICAL UNSATISFACTORY POST-SINTERED SPECIMENS

$D_o = 30 \text{ mm}$ $H/D = 0.80$



$D_o = 30 \text{ mm}$ $H/D = 0.40$



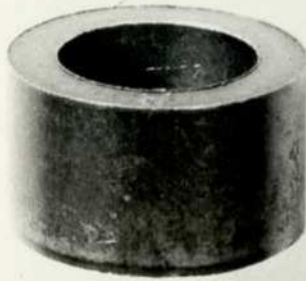
(a) Typical satisfactory post-sintered iron powder metal solid cylindrical specimens.

$H/D = 0.8$

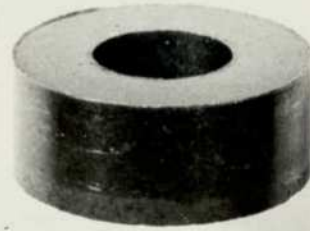


Witness line

(b) Typical *green* solid cylindrical compact showing witness line formed by radial line scribed on lower die.



Internal diameter 20 mm



Internal diameter 14 mm



Internal diameter 20 mm sectioned

(c) Typical satisfactory post-sintered iron powder metal hollow cylindrical specimens of 30 mm external diameter.

FIGURE 73 TYPICAL SATISFACTORY SOLID AND HOLLOW CYLINDRICAL SPECIMENS.

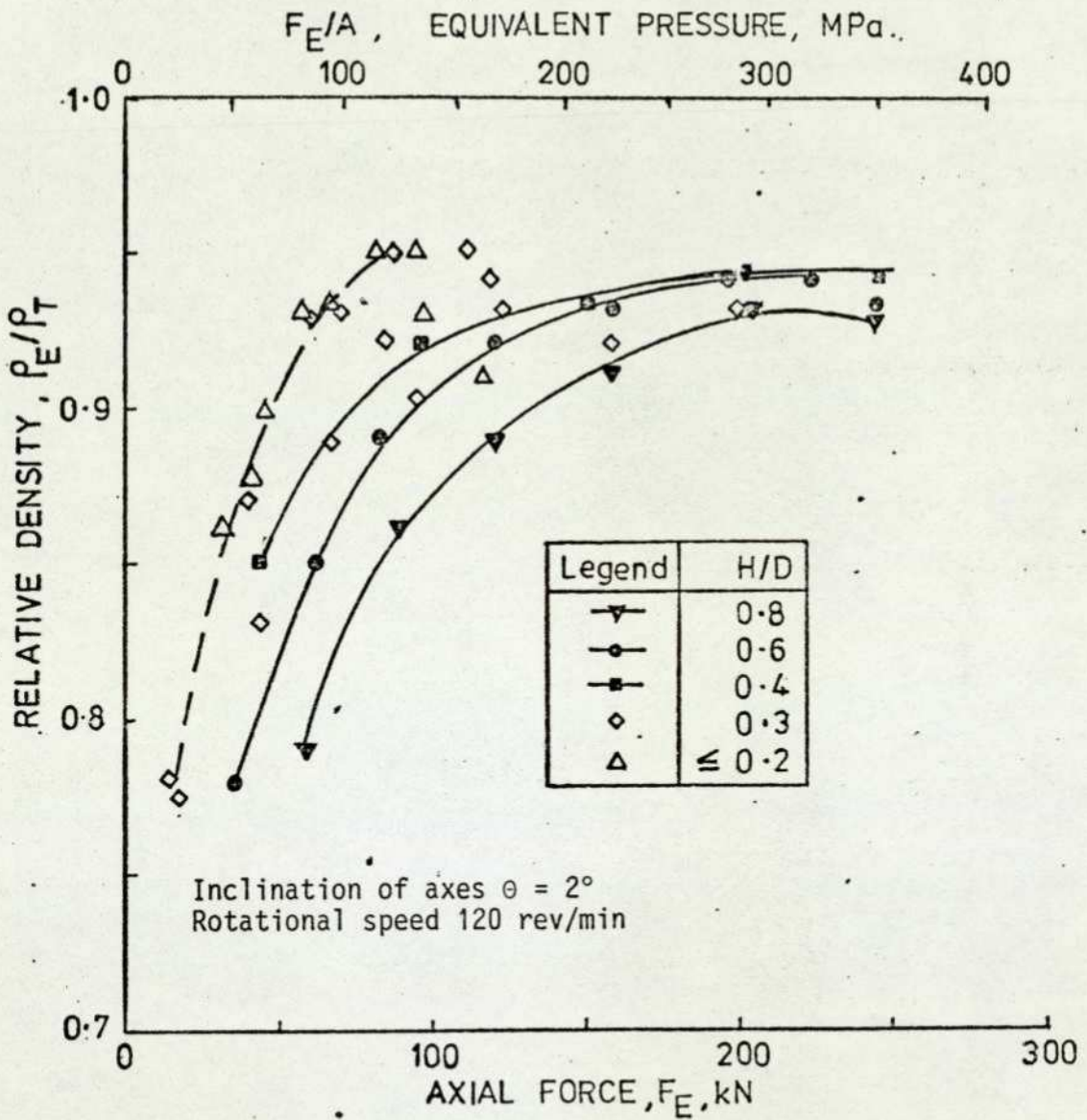


FIGURE 74 RELATIVE DENSITY, ρ_E/ρ_T - AXIAL FORCE, F_E , - EQUIVALENT PRESSURE, F_E/A RELATION FOR SINTERED IRON POWDER METAL SOLID CYLINDRICAL SPECIMENS OF VARIOUS HEIGHT/DIAMETER RATIO.

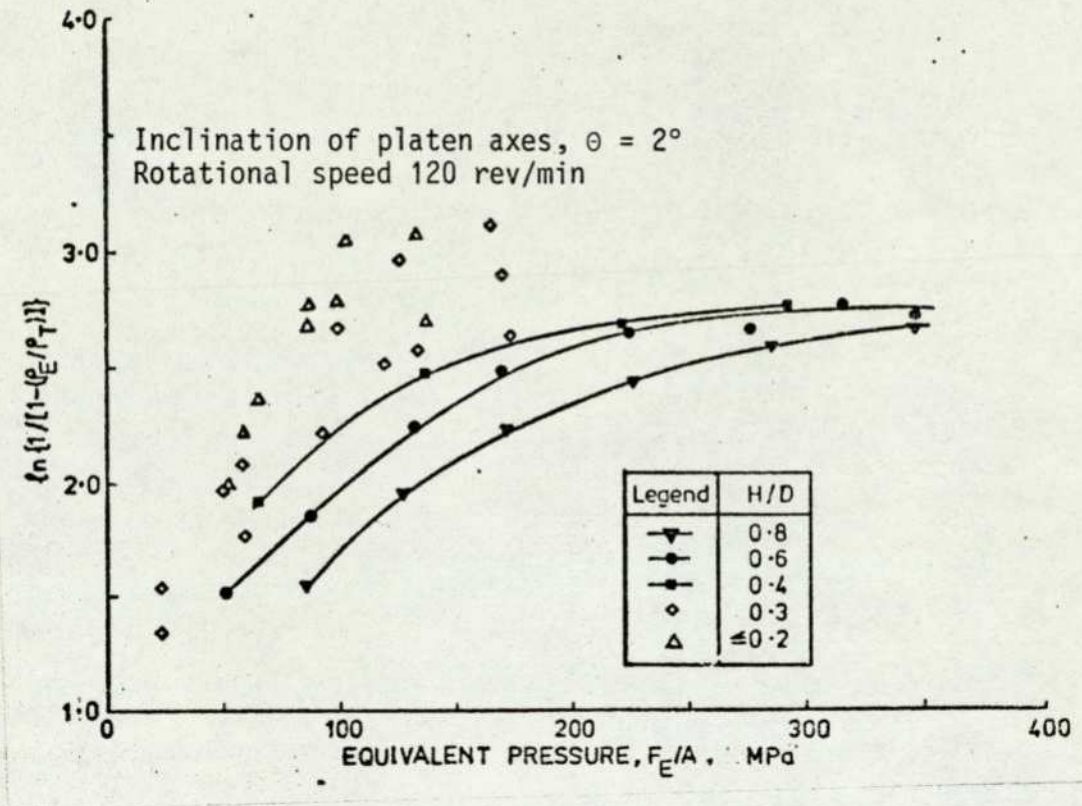


FIGURE 75 $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ - EQUIVALENT PRESSURE, F_E/A , RELATION FOR POST SINTERED, IRON POWDER METAL, SOLID CYLINDRICAL COMPACTS OF VARIOUS HEIGHT/DIAMETER RATIO.

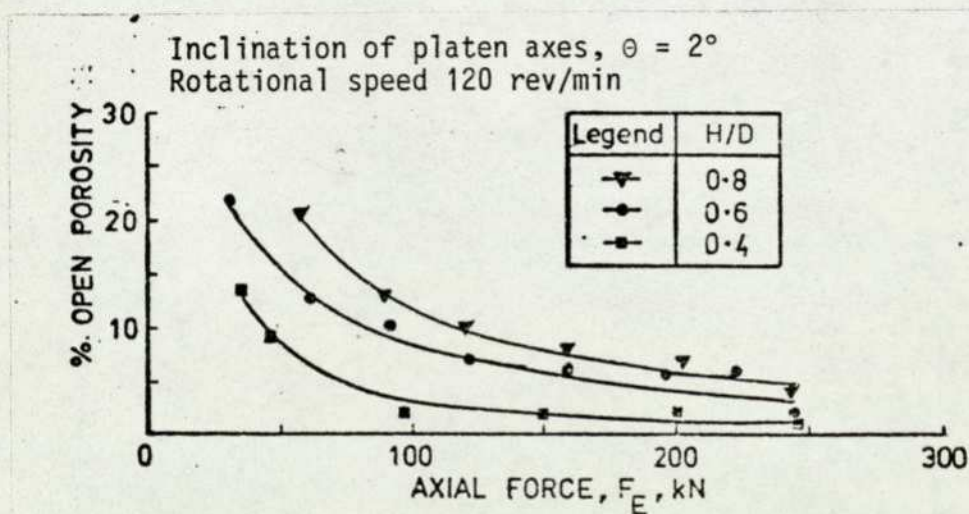


FIGURE 76 OPEN POROSITY - AXIAL FORCE RELATION FOR POST SINTERED IRON POWDER METAL, SOLID CYLINDRICAL COMPACTS OF VARIOUS HEIGHT/DIAMETER RATIO.

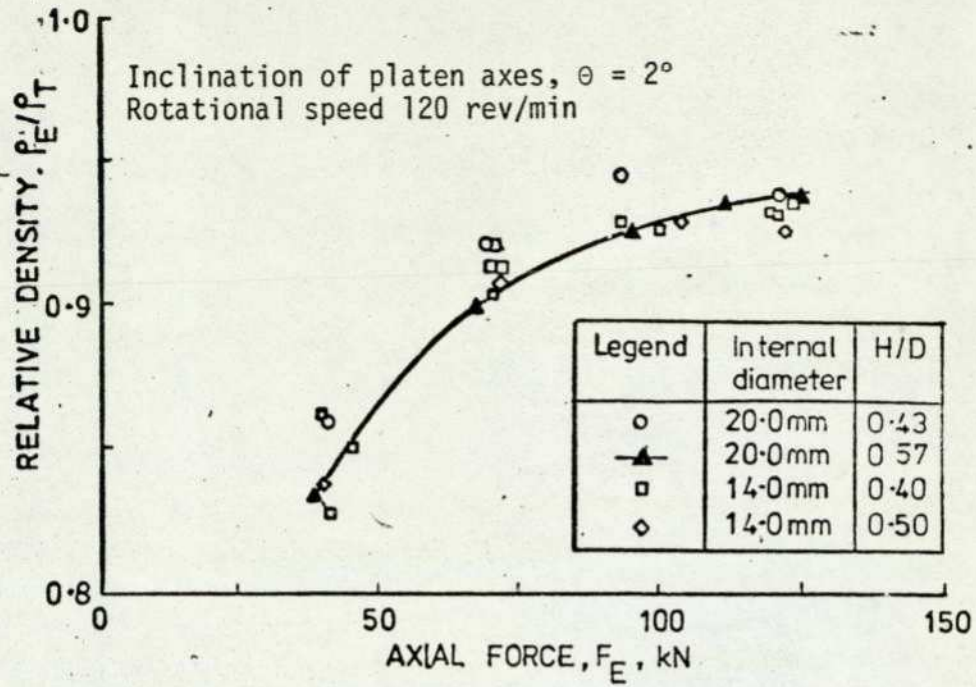


FIGURE 77 RELATIVE DENSITY, ρ_E/ρ_T , - AXIAL FORCE, F_E , RELATION FOR POST SINTERED IRON POWDER METAL HOLLOW CYLINDRICAL SPECIMENS OF VARIOUS HEIGHT/DIAMETER RATIOS.

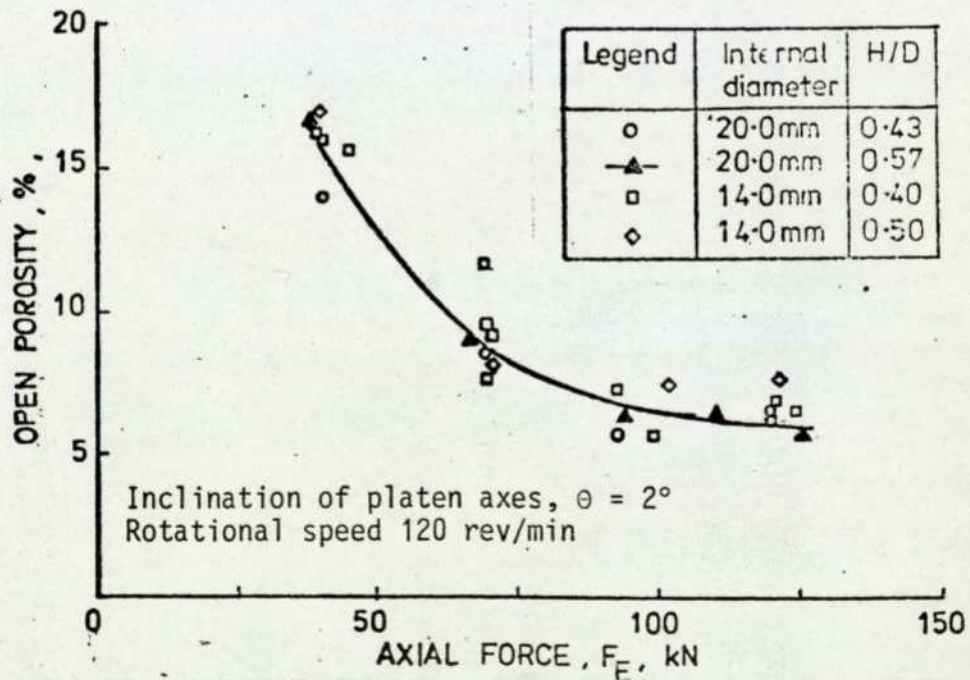


FIGURE 78 OPEN POROSITY, %, - AXIAL FORCE, F_E , RELATION FOR POST SINTERED IRON POWDER METAL HOLLOW CYLINDRICAL SPECIMENS OF VARIOUS HEIGHT/DIAMETER RATIOS.

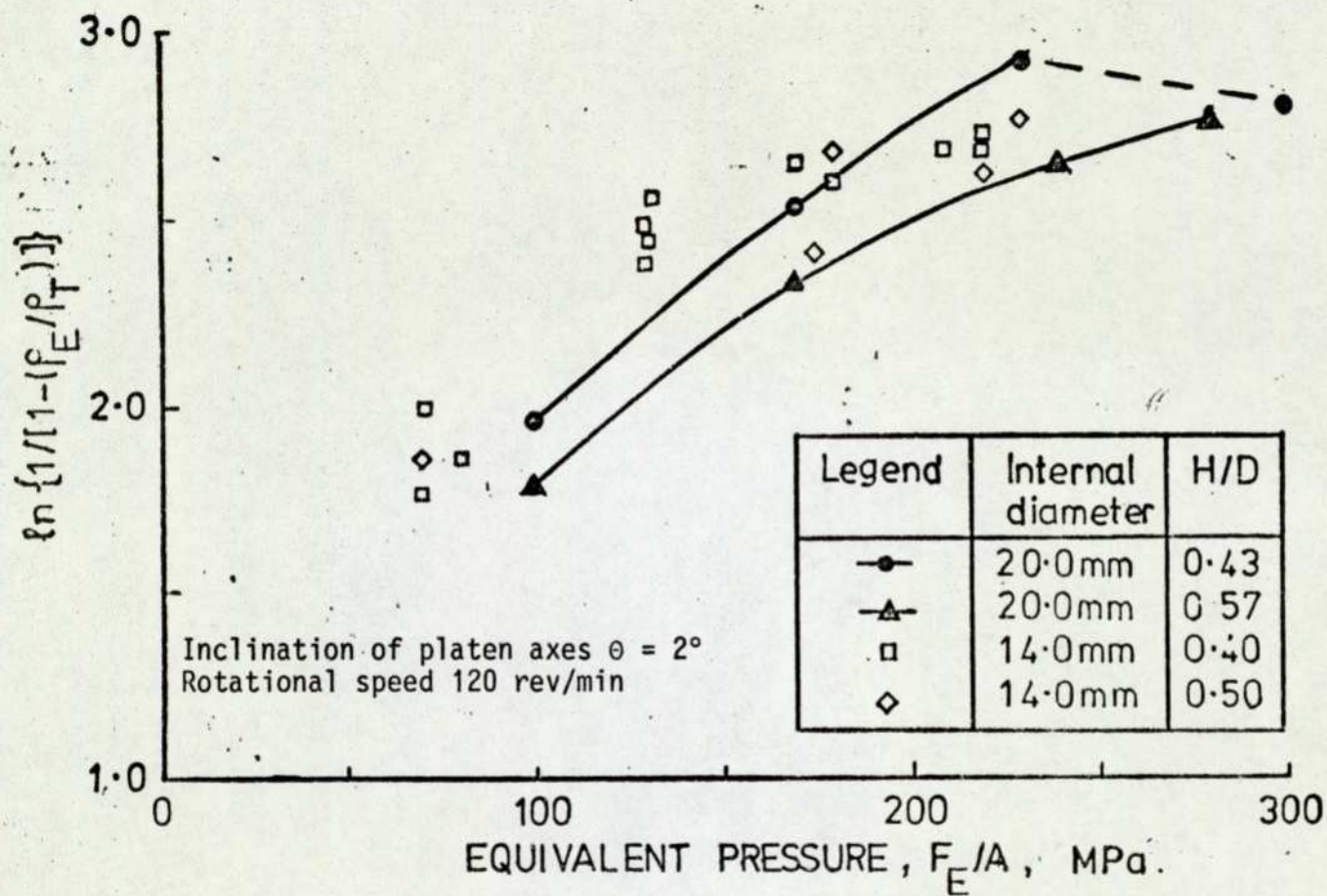


FIGURE 79 POST SINTERED DENSITY $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ - EQUIVALENT PRESSURE, F_E/A , RELATION FOR IRON POWDER METAL HOLLOW CYLINDRICAL SPECIMENS OF VARIOUS HEIGHT/DIAMETER.

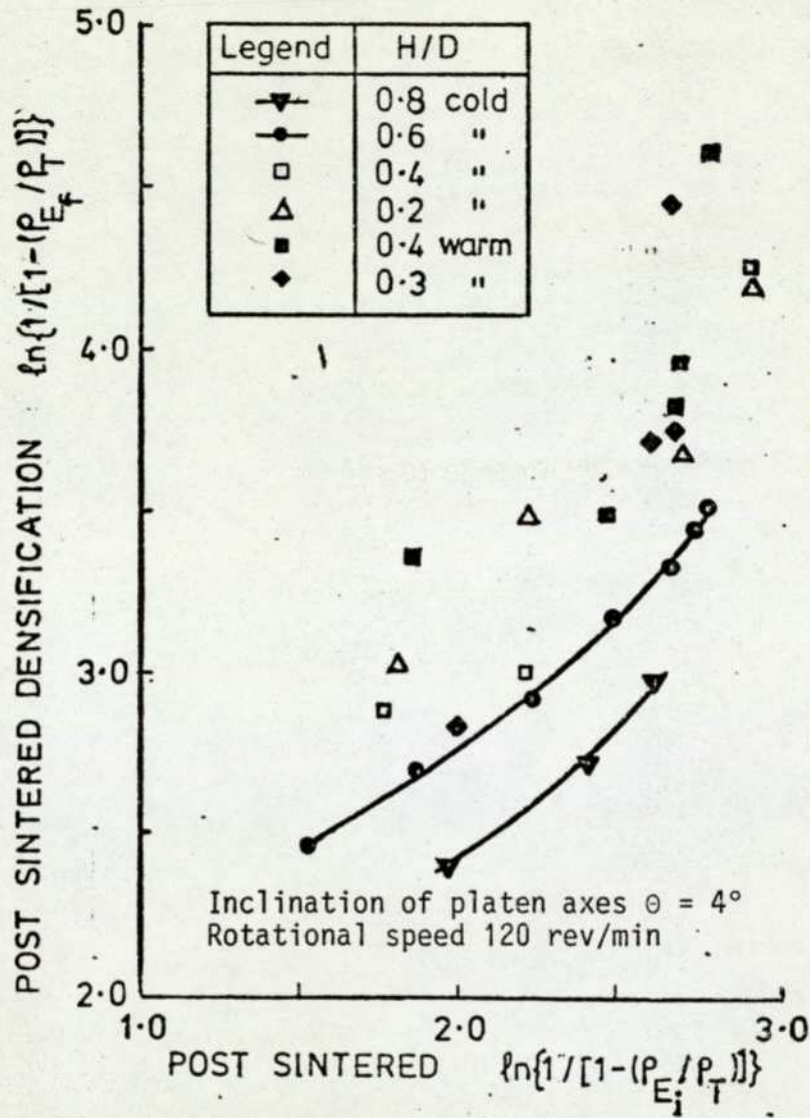


FIGURE 80 POST SINTERED DENSIFIED $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ -
POST SINTERED $\ln\{1/[1 - (\rho_E/\rho_T)]\}$ RELATION FOR
SINTERED IRON POWDER METAL ⁱSOLID CYLINDRICAL SPECIMENS
OF VARIOUS HEIGHT/DIAMETER ROTARY FORGED IN A CLOSED
DIE WITH A CONSTANT AXIAL FORCE OF 75 kN.

$R = 0.15$

 $R = 0.15$


(a) Typical post-sintered iron powder metal specimens $D_0 = 30$ mm, $H_0/D_0 = 0.66$ after fractional reduction in height caused by *cold* upsetting


 $R = 0.51$

 $R = 0.56$


(b) Typical post-sintered iron powder metal specimens $D_0 = 30$ mm, $H_0/D_0 = 0.66$ after fractional reduction in height caused by *warm* upsetting

FIGURE 81 TYPICAL IRON POWDER METAL SOLID CYLINDRICAL SPECIMENS EXHIBITING CRACKING WHEN SUBJECTED TO *COLD* AND *WARM* UPSETTING IN AN ATTEMPT TO CAUSE POST-SINTERED DENSIFICATION

Initial internal diameter 20 mm



Initial internal diameter 14 mm



- (a) Upper surfaces of hollow cylindrical specimens after fractional reduction in height $R = 0.5$ was caused by *cold* upsetting

Initial internal diameter 20 mm



Initial internal diameter 14 mm



- (b) Upper surfaces of hollow cylindrical specimens after fractional reduction in height $R = 0.5$ was caused by *warm* upsetting

FIGURE 82 TYPICAL SINTERED IRON POWDER METAL HOLLOW CYLINDRICAL SPECIMENS OF INITIAL EXTERNAL DIAMETER 30 mm SUBJECTED TO *COLD* AND *WARM* UPSETTING TO CAUSE POST-SINTERED DENSIFICATION

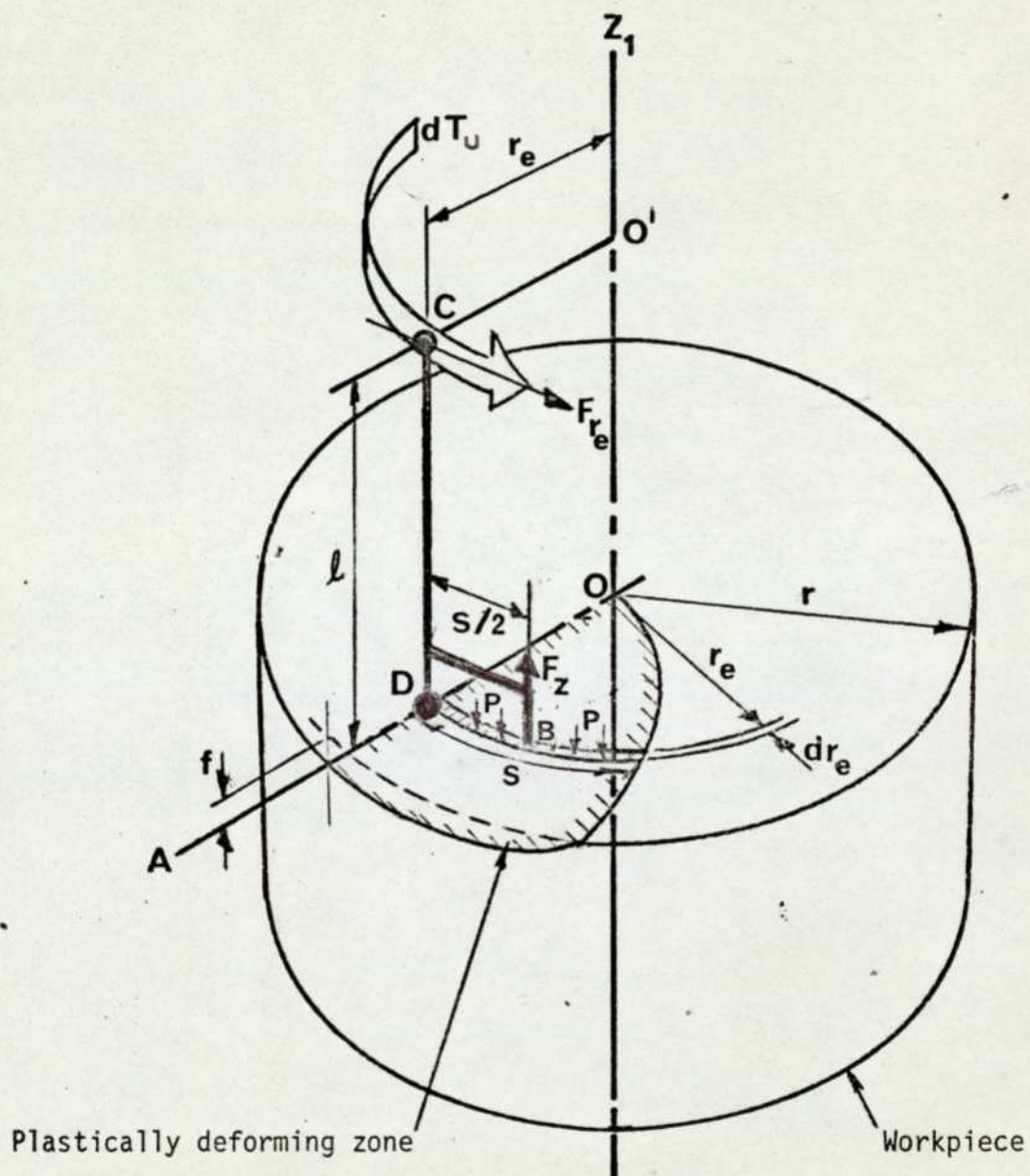


FIGURE 83 DIAGRAM OF INSTANTANEOUS HINGES ASSUMED TO EXIST FOR PREDICTION OF TORQUE REQUIRED TO ROTATE THE UPPER PLATEN RELATIVE TO THE WORKPIECE.