DESIGN, CONSTRUCTION, CALIBRATION AND TESTING OF A PUNCH PRESS DYNAMOMETER

by

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INTRODUCTION

From our vantage point if we turn back and have a look at our great great grand ancestor - the primitive man - we find a huge gap in between. He was inhabiting caves and had very few means of subsistence. He had not reached the stage called civilization, in the strict sense of the term, at all. On the other hand, we are exploring the far reaching depths of oceans, atoms and outer space. Moreover, our means of subsistence are plenty and our civilization is far more superior to any one that flourished on this planet.

But the whole of the modern civilization, of which we are a part, has not been achieved overnight. It is the product of the intelligence, labor - mental as well as physical - and perseverence of thousands of individuals for thousands of years. The primitive man has slowly, steadily and continuously travelled on the path of progress to come to the stage of the modern man. We have not reached the dead end on this path of progress, and therefore we must march ahead.

In this connection new devices are being designed and developed every day. Tremendous amount of research work is being done to seek new information to improve these devices. Moreover, these devices have to be tested to ascertain whether they function according to their expectations.

While testing these devices (power producing or power consuming) the most vital factors, about the information is generally sought, are power and force.

WHAT IS A DYNAMOMETER?

The devices used for measuring power and force are called dynamometers. These dynamometers are broadly divided into two classes, viz;

1. Power dynamometers
2. Force dynamometers

Power dynamometers. These dynamometers essentially measure the power produced or consumed by machines. These dynamometers are further subdivided into three classes as under.

Brake or Absorption Dynamometers. These are the dynamometers in which the power produced by a rotating shaft or wheel is measured by absorbing it or by converting into heat by the friction of the brake.

Transmission Dynamometers. In these types of dynamometers the power by a rotating shaft or wheel is measured by transmitting the power, through a belt or other transmitting mechanism, to another shaft where it is productively used. Theoretically, the transmitting mechanism does not absorb any power. In this case the power produced by machine under test is not wasted during the period of testing.

Traction Dynamometers. These are the dynamometers which are generally used for determining the power produced by a machine by utilizing this power to accelerate a car, wagon or other load, either by pulling or pushing.

Force Dynamometers. These dynamometers, as the name suggests, are used generally for measuring force or forces (three dimensionally). These dynamometers are especially used for the measurement of forces acting on machine tools. In general these dynamometers can be called tool force measuring devices. They put the analysis of metal cutting operations on quantitative basis.

There is not a single dynamometer which can be used for the measurement of all forces on all machines (lathes, milling machines, punch presses, grinding machines, etc.), as the forces to be measured in different machines vary in their nature and direction from machine to machine. It is the machine tools which play an important part as the dynamometers are designed to suit the given specific conditions to be encountered while measuring the forces on a given machine. It may happen that a dynamometer which is suitable for one milling machine may not be suitable for another milling machine. A dynamometer suitable for horizontal milling machine may not be suitable at all for a vertical milling machine. Hence, each dynamometer has to be dealt with as a unit in itself while designing. Therefore the dynamometers are classified as punch press dynamometers, lathe dynamometers, milling dynamometers, etc. depending upon the machine tool.

PROJECT AND ITS PURPOSE

In this project it is proposed to design a punch press dynamometer for a punch press of the following specifications:

Specifications:

Vertical punch press with tilting bed.

Maximum punching force is 10 tons.

Variable stroke from 1/4" to 1 1/2".

R.P.M. 150

This dynamometer will be able to measure the punching forces at various thicknesses of the stock during the punching operation. The dynamometer shall be designed for a maximum force of 10,000 lbs. The thickness of the stock and the size and shape of the hole to be punched can be varied at will so long as the maximum force required to punch that hole does not exceed 10,000 lbs.

The main purpose of embarking upon this project is to have one punch press dynamometer in the department so that more research work in the field of punching operations can be undertaken.

REVIEW OF LITERATURE

Extensive work seems to have been done so far as the field of dynamometry in general is concerned. But in the field of punch press dynamometers not much work seems to have been done.

At Syracuse University, Graves (5) and Jenson (6) have done some work in which they analyzed a punch press force transducer. They have also selected instruments which can be used for force-stroke-time data.

Howard and Tisley (11) have done interesting work in the field of punching of sheet materials. They have found that when shearing mild steel with a flat punch and sharp-edged die, the maximum shearing force occurs when the punch has travelled approximately one-third of the distance through the material. They conclude in their work that the maximum punching force when the tools are provided with 'shear' equal to the thickness of the metal being punched, (n 1), is equal to the average punching force obtained when the tools have no 'shear'. In this connection they have derived the following general formula:

$$E_{max} = \frac{\bar{F}}{n} \qquad 1$$

where E_{max} = Maximum punching force

$\bar{F}$ = Average punching force

n = Fraction metal thickness when computing 'shear'

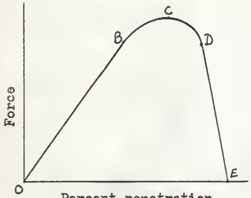
DYNAMOMETER DESIGN AND ANALYSIS

In the case of a punch press dynamometer, it is supposed to measure the punching force. Therefore before we proceed with the design of the dynamometer as such we must analyze the nature, magnitude and direction of the forces to be measured by the dynamometer. Therefore first we shall analyse the theory of failure during punching operation.

Theory of Failure During Punching Operation. Fremont (15) was the first to attribute the failures in punching operations to both a tensile and shear type failure, rather than a pure shear type failure. As pointed out in Fremont's work, the destorted area the punch and the die takes on approximately the form of a parellogram, with fracture ensuing along diagonal of the parellogram from the edge of the punch to the edge of the die. As is presently believed today, there is a trixial state of stress in the material while punching and that tensile, compressive and shear stresses occur in the material.

Basically, the punching operation can be broken down into a number of different phases. The first is from the beginning of the contact of the punch and work material until the time when yielding occurs. Very often this phase of the force penetration curve closely approximates the slope of the curve found on a stress-strain diagram. The time elapsed for this to occur is very small, being only a few percent of the total time elapsed. The second phase is from the beginning of yielding until maximum load is reached. This phase of the curve represents the time during which the material is undergoing plastic deformation. The third phase begins at the ultimate load. From the various papers presented on this subject, it has been concluded that fracture begins when the ultimate load is reached.

If the clearance has been selected to give an optimum break, the cracks from the punch side and the die side propagate and meet in a straight line.



AB - Material strained elastically. Slope corresponds to that on stress-strain diagram.

BC - Material strained plastically.

C - Point of maximum load and the beginning of fracture.

CD - Propagation of cracks.

D - Point where complete severance has occured.

Fig. 1.

The rate of propagation of crack progresses faster than the punch in most cases. The force, shortly after reaching the ultimate load, drops off to zero for optimum conditions. This situation is illustrated in Fig. 1. Even though a triaxial state of stress exists in the material while punching, and that tensile, compressive and shear stresses occur in the material, the parent force causing all the stresses is purely compressive one. Thus the dynamometer will be subjected to compressive force. Of course, there will be very negligible tensile force (stripping force) when the punch is returning.

Dynamometer as an Instrument. Technically dynamometer is simply an instrument. Therefore it is simply a device used for determining the value of a quantity or a condition. In this particular case the quantities to be measured are 'force' and 'distance'.

Dynamometer is necessarily an indirect measuring device as the quantities to be measured - force and distance - are measured indirectly by measuring the effect of force (strain) and distance (electrical signal).

All indirect measuring devices necessarily consist of four elements. The four elements are:

1. Sensing element (Primary element)
2. Converting element (Secondary element)
3. Correcting element (Operating element)
4. Functioning element

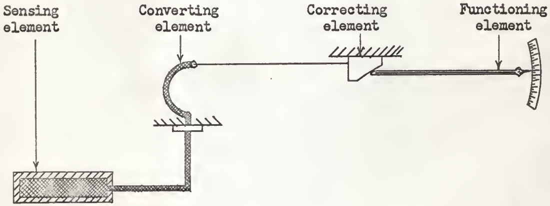


Fig. 2.

All these four elements, put together in the order shown above constitute an instrument.

Design of Sensing Element (Load Column)

Here, the sensing element is called the load column. The load column, first of all, senses the quantity to be measured. Thus the design of the load column depends upon the force to be measured.

Design Data

Design load 10,000 lbs. = P lbs.

Material shall be hot rolled 1020 steel

Maximum permissible stress in compression for 1020 steel

30,000 lbs/sq. in. = f_c lbs/sq. in.

The column shall be designed as a pure compression member. Moreover, the column shall be kept hollow to increase the stability and resistance to bending.

Let

Outer diameter of the column be	D_o inches
Inner diameter of the column be	D_i inches
Cross sectional area of the column be	A sq. in.
Net length of the column be	l inches
Outer diameter of the upper flange be	D_1 inches
Thickness of the upper flange be	t_1 inches
Outer diameter of the lower flange be	D_2 inches
Thickness of the lower flange be	t_2 inches
Over all length of the column be	L inches

$$\text{Force} = \text{Stress} \times \text{Area}$$

$$P = f \times A$$

$$10{,}000 = 30{,}000 \times \pi/4\,(D_o^2 - D_i^2)$$

$$(D_o^2 - D_i^2) = \frac{10{,}000 \times 4}{30{,}000 \times \pi}$$

$$0.4245 \text{ sq. inches.}$$

Here it is assumed that the diameter of the hole to be punched will not increase 1 inch. Therefore for the ease of the removal of blanks D_i = 1.25 inches

$$\therefore (D_o^2 - D_i^2) = 0.4245$$

$$\therefore (D_o^2 - 1.25^2) = 0.4245$$

$$\therefore D_o^2 = 1.9870$$

$$\therefore D_o = 1.41 \text{ inches}$$

Therefore let $D_o = 1.5"$ for the ease of construction and for adequate thickness of the column.

$$D_o = 1.5" \qquad D_i = 1.25"$$

A length of 2" shall be provided to provide sufficient space for the installation of strain gages. To increase the stability of the load column a flange of 4" diameter at the bottom of the plate shall be provided. Moreover, a flange of 3" diameter at the top of the column shall also be provided. There shall also be provided, on the lower flange, 4 3/16" diameter, equally spaced, clear holes on a 3.5" diameter circle. The thickness of both the flanges shall be kept 1/4". Moreover, all the corners, where different sections meet, shall be rounded to the radius of 1/8" to avoid stress concentration. The final dimensions of the load column are as under.

Outer diameter of the column	1.50"
Inner diameter of the column	1.25"
Length of the column	2.00"
Diameter of the top flange	3.00"
Thickness of the top flange	0.25"
Diameter of the bottom flange	4.00"
Thickness of the bottom flange	0.25"
Overall length of the column	2.50"
Diameter of four equally spaced clear holes on a 3.5" diameter circle	0.1875"

The drawing of the load column is shown in Fig. 3 with all the dimensions. The scale adopted is full size.

Design of Converting Element

The basic purpose of designing the dynamometer is to measure the force. To measure the force, the load column has been designed. The load column senses the force.

The effect of sensing the force is revealed in the form strain produced in the column. So to know the force, strain must be measured - directly or indirectly. The strain produced in the column is to be measured directly. Therefore, the device which converts strain into a quantity which can be easily and accurately measured is called the converting element.

Of all the devices available for the measurement of strains, strain gages are very popular and very widely used. The strain gages work on the principle that the resistance of any wire changes as the cross-sectional area and the length of the wire change.

Let R be the resistance of the wire in ohms whose length is inches, cross-sectional area is A sq. inches and the resistivity is ohms inch. In that case

$$R = \sigma . l/A \text{ ohms} \qquad 2$$

The resistivity of any wire remains constant for a given material. Therefore when length or cross-sectional area of the wire changes, the resistance of the wire also changes. The change in the resistance of the wire is indicated by the change in current passing through the wire provided the voltage applied across the wire remains constant. The change

Scale: Full size

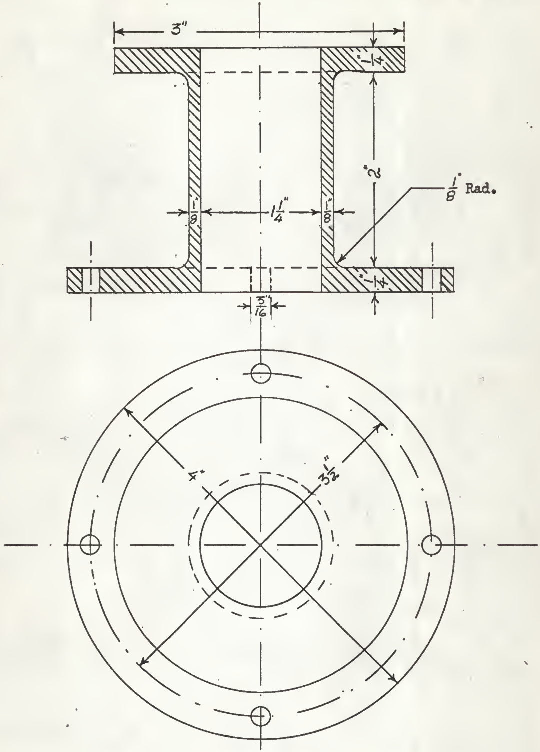


Fig. 3. Load column.

in current appears in the form of change in voltage across the wire. The change in voltage can be magnified or multiplied for the purpose of measurement.

Therefore the strain gages are essentially made of very fine filaments of wires or foils. The strain gages are specified by their resistances and gage factors - 'G'. The gage factor 'G' of a strain gage is defined as the proportional change in resistance for a proportional change in length. It is mathematically expressed as

$$G = \frac{\Delta R/R}{\Delta L/L} \qquad 3$$

where G = Gage factor

ΔR = Change in resistance of the filament of wire because of the strain in the wire in ohms.

ΔL = Change in length of the filament of wire because of the strain in the wire in inches.

R = Initial resistance of the strain gage filament in ohms.

L = Initial length of the strain gage filament in inches.

The gage factor of a strain gage is, therefore, thus an index of the strain sensitivity of the gage. The higher the gage factor, the more sensitive the gage and the greater the electrical output for indication or recording purposes, other variables remaining the same.

Moreover for the effective and convenient use of the strain gages, they should be

1. Very small in size
2. Very insignificant in weight
3. Relatively simple to attach to the test piece

4. Fairly sensitive to strain (high gage factor)

5. Unaffected by the ambient variables

6. Able to indicate both - static as well as dynamic - strains with equal ease

7. Convenient for remote indication or recording

8. Inexpensive

Today there is not a single strain gage that fully meets all the above requirements. As one authority in the strain gage field puts it:

"While it is theoretically possible to develop a strain gage which is infinitesimal in size and weight, has an infinite sensitivity to strain and an infinitesimal sensitivity to all other variables, the cost of such a strain gage would be .... infinite."

Of all the strain gages available on the market, the bonded wire strain gages affect the greatest compromise among all the requirements enumerated above. The bonded wire strain gages are available in two types - advance and isoelastic. The advance type gages are used for static strains while the isoelastic (a severely cold worked elin bar) type gages are used for the dynamic strains.

Scale: Double size

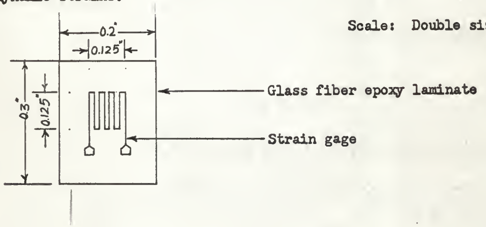


Fig. 4.
Fig. 4.

As we are going to encounter dynamic strains during the punching operations, we shall use the isoelastic type bonded wire strain gages. Therefore, we have decided to make use of the "EA-XX-125AA-120" strain gages manufactured by the Micro-Measurements, Inc. of Romulus, Michigan. Their specifications are as follows:

Gage Specifications

Resistance in ohms	120 ohms
Gage factor	2.05
Gage length	0.125 inch
Over all length	0.175 inch
Grid width	0.125 inch
Over all width	0.200 inch

Design of the Correcting Element

It is quite possible that some undesirable effects might be present with the two elements designed so far. Moreover sometimes these undesirable effects are not possible of elimination at earlier stages. It may also be possible that the response of the converting element may not be linear. For accurate and easy measurements it is necessary that all these undesirable effects should be taken care of.

This objective is accomplished by the element that we are going to design now. Therefore it is called correcting element. Sometimes the sensitivity of an instrument can also be improved with the help of this correcting element. It manipulates the output from the converting element in such a way that it is easier to read and more accurate.

Now before we can correct any error, it is necessary to know the errors and their nature. Therefore let us study and correct the errors side by side.

Error Due to Non-Linearity. This error is of great importance as it interferes with the final reading or recording. Therefore this error should always be taken care of depending upon its nature (quadratic, cubic, etc.).

In our case, we are not going to load our column beyond elastic limit. The stress-strain curve is linear within the elastic limit. Therefore we do not encounter this error.

Error Due to Eccentricity of Load. This type of error should always be taken care of because the chances of removing this error by designing are very dim.

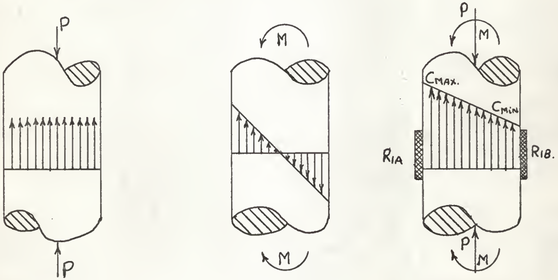
Even if our design is perfect by the way of turning and shaping the load column, the material may not be uniform in density. In such cases eccentricity is inevitable as geometric center of gravity and physical center of gravity will not be coincident.

Moreover the punch may not apply the force exactly at the center of gravity of the load column. In that case also some eccentricity will always creep in. This eccentricity may vary (within a very small range) during different punching operations.

Because of all these factors which are beyond our control, we must assume some eccentricity. Therefore, when eccentric loading is there, some amount of bending stress will always exist in the load column. These stresses will unduly affect the strains of strain gages and will give us erroneous results.

Now this error can be eliminated by properly fixing the strain gages on the load column. If the strain gages are fixed properly in different positions on the load column and if properly put in adequate arms of the Wheatstone bridge, they together act in such a way that the effects due to eccentric loading are mutually cancelled and we get the true reading of pure compressive strain due to load only.

In our case we want to see that the strain gage is not reading any strain due to eccentricity of the load. If there is no eccentricity, the stresses on the column will be as shown in Fig. 5a and the strain gage will read these stresses.



Compression

a

Bending

b

Compression and Bending Combined

c

Fig. 5.

But when some eccentricity due to load is there, the column will try to bend as shown in Fig. 5b. Because of this bending, the column will develop tensile and compressive stresses as shown in Fig. 5b. Therefore, our strain gage will pick up these stresses also and our results will be erroneous. The stress in the column will be as shown in Fig. 5c, due to combined effect of compression and bending. Now to correct this error, we shall use two instead of one strain gage. Let us call them R_{1A} and R_{2B}. Let us fix these strain gages on two points diametrically opposite to each other on the surface of the column as shown in Fig. 5c. Now connect these strain gages in the arms AB and CD of the Wheatstone bridge of Fig. 6. For the sake of simplicity in analysis let us assume that

$$R_{1A} = R_2 = R_{1B} = R_4 \qquad 4$$

Now when no current is passing through the galvanometer

$$\frac{R_{1A}}{R_4} = \frac{R_2}{R_{1B}} \qquad 5$$

Now let us say there is some eccentricity. Because of this the column will try to bend as shown in Fig. 5b. Therefore R_{1A} will undergo compressive strain and R_{1B} will undergo tensile strain. Thus R_{1A} will change to $(R_{1A} - \Delta R_{1A})$ and R_{1B} will change to $(R_{1B} + \Delta R_{1B})$. Thus when new condition of equilibrium is established and R_g does not pass any current, we will get

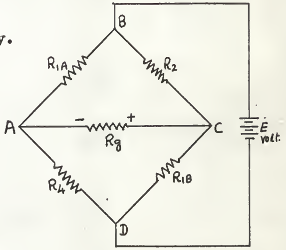


Fig. 6.

$$\frac{R_{1A} - \Delta R_{1A}}{R_4} = \frac{R_2}{R_{1B} + \Delta R_{1B}} \qquad 6$$

where R_2 and R_4 do not necessarily have the sames values as before.

Now let us multiply the numerators and denominators of left hand and right hand side of equation 6 by R_{1A} and R_{2B} respectively.

$$\therefore \frac{R_{1A}}{R_{1A}} \frac{(R_{1A} - \Delta R_{1A}}{R_4} = \frac{R_2}{(R_{1B} + \Delta R_{1B})} \frac{R_{1B}}{R_{1B}}$$

Cross-multiplying the above equation we get

$$\frac{\frac{R_{1A}(R_{1A}^2 - \Delta R_{1A}^2)}{R_{1A}^2}}{R_4} = \frac{R_2}{R_{1B}}$$

$R_{1A} = R_{1B}$ and as compressive and tensile stresses due to bending are equal $R_{1A} =$ R_{1B} and therefore R_{1A} and R_{1B} will be equal in magnitude but opposite in sign.

$$\therefore \frac{R_{1A} - \frac{\Delta R_{1A}^2}{R_{1A}}}{R_4} = \frac{R_2}{R_{1B}}$$

$\therefore \frac{R_{1A}}{R_4} = \frac{R_2}{R_{1B}}$ $\because$ R_{1A} is very small

$\therefore$ R_A is still smaller and R_{1A}^2/R_{1A} will still be smaller and therefore neglected.

$$\therefore \frac{R_{1A}}{R_4} = \frac{R_2}{R_{1B}} \qquad 7$$

Equations 5 and 7 are exactly the same. Thus by putting the strain gages as shown in Figs. 5c and 6 the error due to eccentricity can be completely eliminated.

Now when only compressive stress is to be measured, the resistance of both R_{1A} and R_{1B} will decrease by, say, ΔR_{1A} and ΔR_{1B} amounts respectively.

$\Delta R_{1A} = \Delta R_{1B}$ $\because R_{1A} = R_{1B}$

Now when new equilibrium is established and R_g does not pass any current

$$\frac{R_{1A} - \Delta R_{1A}}{R_4} = \frac{R_2}{R_{1B} - \Delta R_{1B}} \qquad 8$$

Here R_4 and R_2 do not necessarily have the same value as before. Now let us multiply the numerators and denominators of left hand and right hand sides of equation 8 by R_{1A} and R_{2B} respectively.

$$\therefore \frac{R_{1A}}{R_{1A}} \; \frac{R_{1A} - \Delta R_{1A}}{R_4} = \frac{R_2}{R_{1B} - \Delta R_{1B}} \; \frac{R_{1B}}{R_{1B}}$$

Cross-multiplying the above equation we get

$$\frac{R_{1A}\left(\frac{R_{1A} - \Delta R_{1A}}{R_{1A}}\right)^2}{R_4} = \frac{R_2}{R_{1B}}$$

$\because R_{1A} = R_{1B}$
and $R_{1A} = R_{1B}$

$$\therefore \frac{(R_{1A}^2 - 2R_{1A} \times \Delta R_{1A} + \Delta R_{1A}^2)}{R_{1A} \times R_4} = \frac{R_2}{R_{1B}}$$

$$\therefore \frac{R_{1A} - 2\,\Delta R_{1A} + \Delta R_{1A}^2/R_{1A}}{R_4} = \frac{R_2}{R_{1B}}$$

$$\therefore \frac{R_{1A} - 2\Delta R_{1A}}{R_4} = \frac{R_2}{R_{1B}}$$

$\because$ R_{1A} is very small
$\therefore$ R_{1A}^2 is still smaller
and R_{1A}^2/R_{1A} will still be smaller $\therefore$ neglected

$$\therefore \frac{R_{1A} - 2\,\Delta R_{1A}}{R_4} = \frac{R_2}{R_{1B}} \qquad 9$$

Comparing equations 4 and 8 we find that by using two strain gages as shown in Figs. 5 and 6 we have doubled the sensitivity. The signal is $2\Delta R_{1A}$ instead of ΔR_{1A}.

Thus by using two strain gages as shown in Figs. 4 and 5 we derive two advantages:

1. We eliminate the error due to eccentricity.

2. We double the sensitivity.

Error Due to Temperature Changes. In very rare cases the temperature remains constant during the measurement of strain by strain gages. These temperature changes occur due to change in ambient temperature. Sometimes temperature changes occur due to the application of load also. Therefore adequate temperature compensation is an absolute necessity for accurate measurement of strains with all modern, bonded wire strain gages.

The need for temperature compensation of strain gages arises from two factors. First, there is the fact that the resistance of most of the wires change with the temperature. The second factor is the fact that the thermal coefficient of expansion of the strain gage wire is different from that of material of the wire to which it has been bonded. Thus even if the strain gage wire had a zero thermal coefficient of expansion, it will still be subject to false strain indications with temperature changes unless it had some proper compensation for temperature errors. If such a gage were constructed so that it was completely free of temperature errors when bonded to steel, it would be greatly in error if bonded to aluminium or some other metal with a different thermal coefficient of expansion from steel.

In our case the column is subjected to compressive stresses only. In such case the poisson arrangement of strain gages is the only manner in which complete temperature compensation can be accomplished without the use of an unstrained dummy. In poisson arrangement two strain gages of identical type are mounted at right angles to each other as shown in Fig. 7. The strain gage R_{1A} measures the compressive strain while the

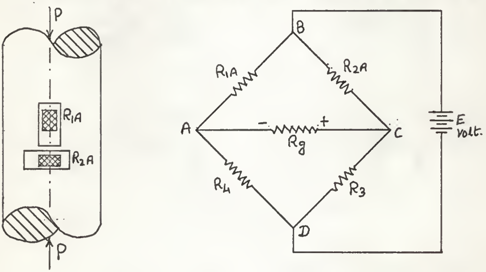


Fig. 7.

strain gage R_{2A} compensates for temperature changes. For the cancellation of error due to temperature change, the strain gages are connected in the electrical circuit as shown in Fig. 7.

So let us say that before the temperature changes, the network is in equilibrium and no current passes through R_g. In that case

$$\frac{R_{1A}}{R_4} = \frac{R_{2A}}{R_3} \qquad 10$$

Now when the temperature changes, the temperatures of R_{1A} and R_{2A} will change by the same amount as R_{1A} and R_{2A} are two identical strain gages. Therefore the equilibrium will be disturbed. Let us say that the resistances of R_{1A} and R_{2A} change by ΔR_{1A} and ΔR_{2A} respectively.

$$\therefore \Delta R_{1A} = \Delta R_{2A}$$

$$\therefore R_{1A} + \Delta R_{1A} = R_{2A} + \Delta R_{2A}$$

$$\therefore \frac{R_{1A} + \Delta R_{1A}}{R_{1A}} = \frac{R_{2A} + \Delta R_{2A}}{R_{2A}} \qquad 11$$

Now by manipulating the resistances R_2 and R_4, we establish new equilibrium. When new equilibrium is established and R_g does not pass any current

$$\frac{R_{1A} + \Delta R_{1A}}{R_4} = \frac{R_{2A} + \Delta R_{2A}}{R_3} \qquad 12$$

Here R_4 and R_3 do not necessarily have the same value as before. Now by dividing equation 12 by equation 11 we get

$$\frac{R_{1A}}{R_4} = \frac{R_{2A}}{R_3} \qquad 13$$

This is the same as equation 9. Thus our original equilibrium is not disturbed at all even if the temperature changes. Therefore even if the temperature changes, the galvanometer will not show any deflection and thus our strain gages are immune to the temperature changes.

Thus we can eliminate all the possible errors that we may come across. Therefore finally we conclude that we will install two strain gages in opposite arms along with poisson arrangement of two other strain gages to eliminate the error due to temperature. This is shown in Fig. 8. By doing so we will have three advantages as under.

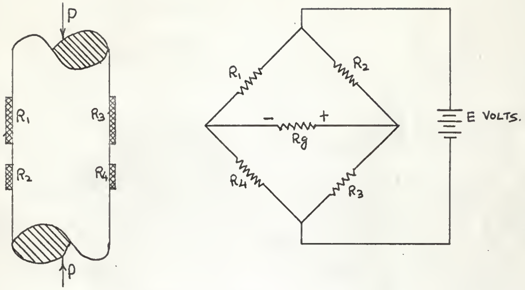


Fig. 8.

1. The error due to eccentricity is removed.
2. The error due to temperature changes is removed.
3. The sensitivity is doubled.

Therefore our final design for strain gages is as under.

Gage Specifications

Resistance in ohms	120
Gage factor	2.05"
Gage length	0.125"
Over all length	0.175"
Grid width	0.125"
Over all width	0.200"
Number required	4

It will be highly desirable to have 4 strain gages from the same lot. They shall be arranged as shown in Fig. 8.

Design of the Functioning Element

The corrected output from the correcting element is of no use unless we can read it in some way. Therefore we must have some element which will either record or indicate the signal coming out from the correcting element. This purpose is accomplished by the functioning element and hence the name.

Selection of the Equipment. In the design of this element, there is practically nothing to be designed. Actually we have to select the instrument (recording or indicating) properly such that it will satisfactorily read the variables under consideration.

In the Department of I.E. there is one Twin-viso Sanborn Recorder manufactured by the Sanborn Company of Waltham, Massachusetts. This seems to be most suitable for our purpose as it has a complete set of power supply for strain gages and amplifier in itself. But we are going to measure dynamic strains. Moreover our press is working at 150 R.P.M. Therefore the punching operation will last for 0.2 second. The pointer of the recorder is too sluggish to react so fast. Therefore we shall couple an oscilloscope to the recorder. Thus by using amplifying system of recorder coupled with an oscilloscope will be quite adequate to indicate our variable under consideration. The oscilloscope used will be the Heathkit oscilloscope.

To attain proper coupling between the recorder and the oscilloscope a coupling circuit is designed as indicated by power box in Fig. 24. The oscilloscope and the recorder will be connected through this circuit.

For recording purposes an oscilloscope camera operating on the Polaroid Land principle shall be used.

Design for the Measurement of Stock Thickness

The purpose of this project is to measure the punching force at different thicknesses of the stock during a punching operation. Therefore now we proceed ahead with the design for the measurement of stock thickness. For this purpose we shall make use of the Linear Variable Differential Transformer (LVDT) manufactured by Schaevitz Engineering of Camden, New Jersey.

What is a LVDT?

The Linear Variable Differential Transformer is an electromechanical transducer which produces an electrical output. Proportional to the displacement of a separate movable core. As shown in Fig. 9 three coils are equally spaced on a cylindrical coil form. A rod shaped magnetic core positioned axially inside this coil assembly provides a path for magnetic flux linking the coils.

When the primary is energized with alternating current, voltages are induced in the two outer coils. In the transformer installation the secondary coils are connected in series opposition so that the two voltages in the secondary circuit are opposite in phase, the net output of the transformer being the difference of these voltages. For one central position of the core this output voltage will be zero. This is called the balance point or null position.

When the core is moved from this balance point, the voltage induced in the coil toward which the core is moved increases, while the voltage induced in the opposite coil decreases. This produces a differential voltage output from the transformer which with proper design varies

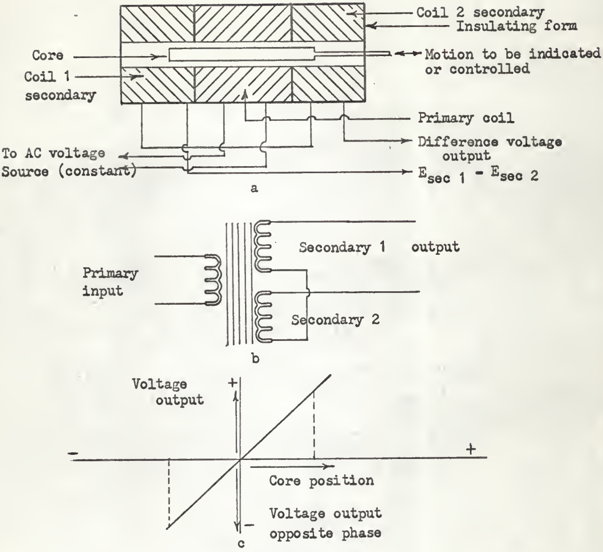


Fig. 9.

linearly with change in core position. Motion of the core in the opposite direction beyond the null position produces a similar linear voltage characteristic, but with the phase shift of 180°. A continuous plot of voltage output versus core position (within the linear range limits) appears as a straight line through the origin if opposite algebraic signs are used to indicate opposite phases as shown in Fig. 9c.

From this analysis we see that this LVDT is perfectly suitable for our purpose. Therefore we shall use on LVDT of Schaevitz Engineering with the following specifications:

Type	E 300 D
Linear range	± 0.300 in.
Linearity	within ± 1%
Excitation	6V. 60 cps
Input power	0.57 watts
Resistance	
Primary ohms	55
Total secondary ohms	220
Impedance	
Primary ohms	62
Total secondary ohms	230
Sensitivity (mv/.001"/volt input) 500 k ohm load	0.7
Transformer length	2 3/4 in.
Transformer OD	3/4 in.
Temperature range	65°F to 180°F

From the above specifications we see that the excitation voltage is 6V, 60 cps. Therefore a circuit as indicated by power box in Fig. 24 shall be used to have the 6V, 60 cps excitation voltage.

Recording the Variables

For this purpose the output from the LVDT shall be connected to the X-axis of the oscilloscope. For coupling LVDT to the oscilloscope, it is necessary that the AC output from transformer by converted into DC one. This conversion shall be achieved with the help of the coupling circuit as indicated by J-box in Fig. 24. The recording of the variable will be done with the help of oscilloscope camera operating on the Polaroid Land principle.

Design of the Accessory Parts

The dynamometer will be used on the Bliss Punch Press in the Department of I.E. Therefore it is necessary that the dynamometer should be adaptable to the physical configuration of the Bliss Punch Press. The design of the following accessory parts has been undertaken with this purpose in mind.

a. Base Plate

The function of this base plate is to receive the load column as well as LVDT and to transmit the load on the column safely to the bolster plate of the punching press. The plate has to be of non-magnetic material because of the LVDT. Therefore we will use Brass plate 3/4" x 6" x 10" as the base plate. The area of 6" x 10" is quite sufficient to transmit and distribute safely the load of 10,000 lbs. to the bolster plate of the punching press. The design and dimensions of the plate are as shown in Fig. 10.

b. Die

While using the dynamometer a die is needed. Moreover it is necessary to design a die which will fit adequately on the load column. For our purpose we shall punch 1 inch diameter hole in 1/16 inch sheet steel. Now let

P_{max} = Maximum force required to punch a hole of 1" diameter in 1/16" sheet steel.

d = Diameter of hole - 1"

L = Length of cut

t_1 = Stock thickness - 1/16"

f_s = Resistance to shear of the material - 10,000 lbs/sq. in.

Now $P_{max} = Ltf_s$

$= \pi d t_1 f_s$

$= \pi \times 1 \times 1/16 \times 10{,}000$

$= 1963$ lbs. force

Just to be on the safe side the design force for die is kept at 5,000 lbs. Material for die will be 1095 steel for which f_t = 30,000 lbs/sq. in. (tensile). The die can be regarded as a flat plate with thickness "t" inches and a central hole of 1" diameter. It will be subjectted to a force of 5,000 lbs. acting all around the periphery of 1" diameter hole.

Let f_{tmax} be the maximum tensile stress developed in the material under these conditions then $f_{tmax} = KP/t^2$ where K = constant = 0.194

$$\therefore 30{,}000 = \frac{0.194 \times 5{,}000}{t^2}$$

$$\therefore t = .03235 = 0.18" \quad 3/16"$$

$\therefore$ a thickness of, say, 1/4" shall be provided. Now clearance = 1/6 stock thickness = 1/6 x 1/16 = .01041

$\therefore$ Diameter of the hole in the die = 1.01041"

A straight = 1/8" shall be provided. An angular clearance of 15° shall be provided.

.·. Lateral clearance

during a length of $1/8" = 1/8 \times \tan 15^{o}$

$= 1/8 \times .2685 = .0335"$

.·. Diameter due to angular clearance of 15^{o}

during a length of $1/8" = 1.01041 + 2 \times .0335$

$= 1.07741"$

We shall keep the diameter of die = 4". A shoulder of 1/8" thickness shall be provided with internal diameter of 3" to provide a proper seat on the load column.

.·. Overall thickness of die = t + 1/8 = 3/8". 4 - 3/16" diameter equally spaced clear holes on a 3.5" diameter circle shall be provided. This die will be heat treated to the hardness of C60. The diagram and dimension of the die are shown in Fig. 11

c. Holder

This die holder has been provided to see that the die and the stripper plate are not lifted up from the column by the punch during its return stroke. The diagram and dimensions are as shown in Fig. 12.

d. Transformer holders

As the name suggests, these are provided to protect the transformer. They shall be made of brass. They will be clamped together with the help of clamps after having been put around the transformer. The diagram and dimensions are as shown in Fig. 13.

e. Protective cylinder.

This protective cylinder, covering the entire load column, has been provided to protect the load column and especially the strain gages on it from any external influence. The material of cylinder is 1020 steel. The design and dimensions of the cylinder are as shown in Fig. 14.

f. Stripper plate

This is an absolutely necessary part. A taper of 15° has been provided to guide the punch. Moreover provision has been made to feed the stock. The diagram and dimensions are as shown in Fig. 15.

g. Top plate

The main function of this plate is to prevent the transformer from any kind of motion once it has been fixed. It also provides a lateral protection to the die. It is made of brass. The dimensions and diagram are as shown in Fig. 16.

h. Punch

The punch is made of 4345 steel. It will be heat treated to the hardness of C57. The diagram and dimensions of the punch are as shown in Fig. 17.

j. Extension plate

This plate has been provided to have an extension from the ram of the punch press. This facilitates the connection of core of the transformer with the ram of the punch press. The dimensions and diagram are shown in Fig. 18.

k. Connecting rod　　Scale: Full size

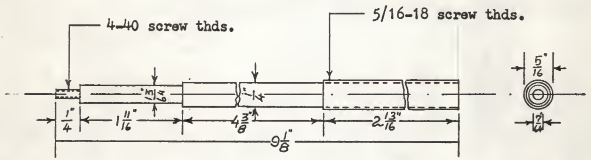


Fig. 19. Connecting rod.

This rod connects rigidly the extension plate and the core of the

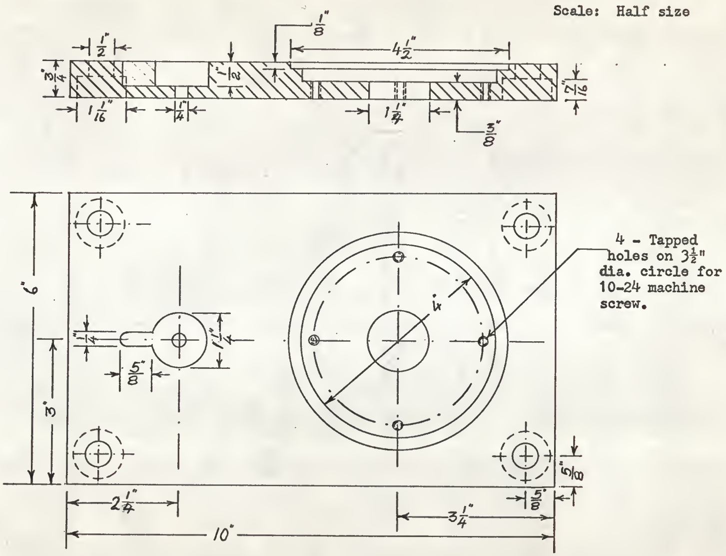


Fig. 10. Base plate.

Scale: Full size

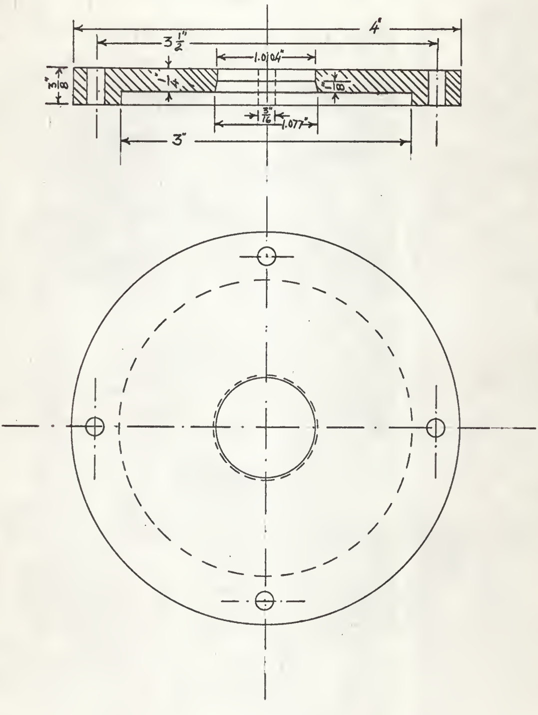


Fig. 11. Die.

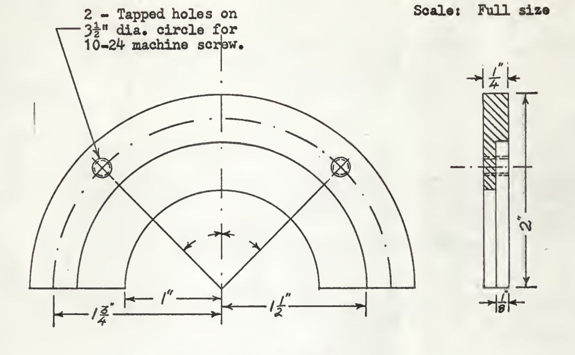


Fig. 12. Die holder.

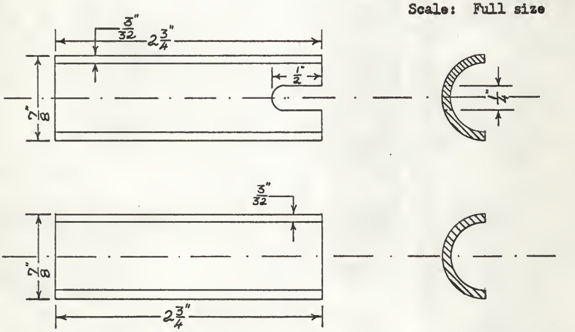


Fig. 13. Transformer holders.

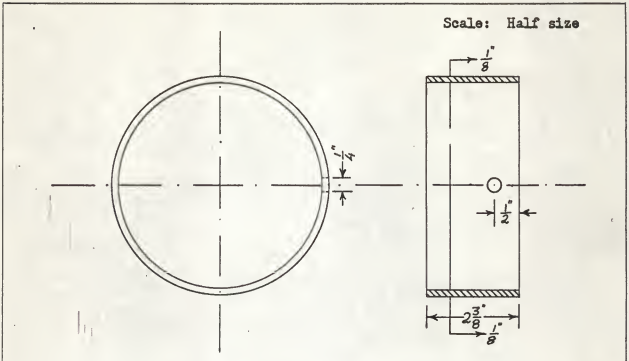


Fig. 14. Protective cylinder.

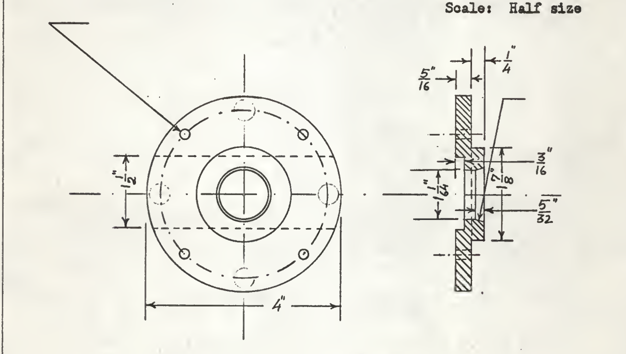


Fig. 15. Stripper plate.

Scale: Half size

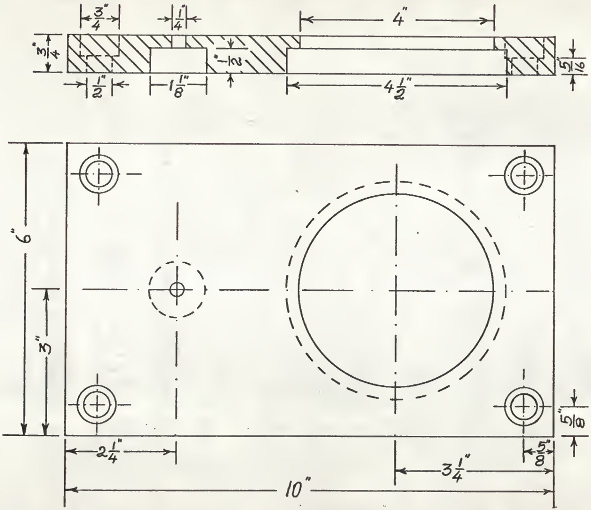


Fig. 16. Top plate.

Scale: Half size

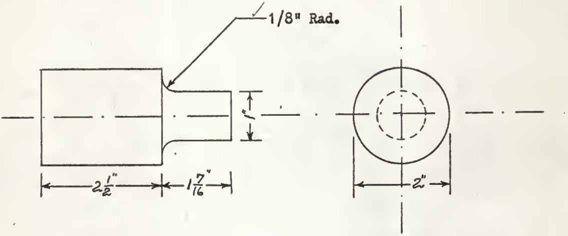


Fig. 17. Punch.

Scale: Full size

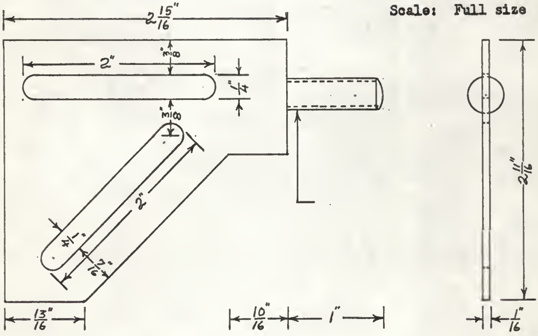


Fig. 18. Extension plate.

transformer. Its function is to provide the motion of the ram of the punch press to the core of the transformer. The dimensions and diagram are as shown in Fig. 19.

Design Criteria for the Dynamometer

In this section we propose to analyze our dynamometer for various design criteria which it must fulfill. We shall deal with them one by one.

1. Cross-sensitivity

Cross-sensitivity is defined as the influence of one force on another one. This does not give the correct value of the force being measured. And therefore this should always be avoided. This effect is there only when more than one force is being measured by the dynamometer. But in our case we are measuring only one force. Therefore we do not face this cross-sensitivity at all.

2. Slenderness-ratio

While designing our load column we have assumed that it works as a member under pure compression. So we will see whether our assumption is correct.

Radius of gyration (least) of the hollow circular section is given by

$$K^2 = \frac{D_o^2 + D_i^2}{16}$$

where K = Least radius of gyration

D_o = Outer diameter of section

D_i = Inner diameter of section

Substituting these values for our load column

$$K^2 = \frac{(1.5^2 + 1.25^2)}{16}$$

$$\therefore K = \left(\frac{2.25 + 1.5625}{16}\right)^{1/2}$$

$$= (0.2385)^{1/2} = 0.568"$$

Slenderness ratio = L/K where L = length of section

$$= \frac{2.5}{0.568}$$

$$= 4.4$$

Here $L/K = 4.4 << 120$

Therefore the assumption that load column works as a member under pure compression stands valid.

3. Sensitivity

The sensitivity of a good dynamometer should be such that the measurements are accurate to within $\pm$ 1%.

4. Linear calibration

It is convenient to use a system having a linear calibration. In such a case the measurement of the variables concerned is more easy and precise.

In this case the load column will have linear calibration as the stress and strain are linearly related within the elastic limits. Moreover for the load of 10,000 lbs. force, for which we have designed our dynamometer, the stress in the load column will never exceed the elastic limit. The LVDT will also give linear calibration for $\pm$ 0.3 inches. Therefore criterian of linear calibration is fulfilled.

5. Stability

It is necessary for the dynamometer to be stable with respect to time and temperature. Once calibration has been made, it should only have to be checked occasionally.

This dynamometer will be quite stable, because for the force of 10,000 lbs. the stress in the column will never exceed the elastic limit. Therefore, the moment the force is removed, the column will regain its original dimensions. But the calibration of this dynamometer should be occasionally checked. Because a little permanent deformation might arise because of repeated and prolonged use. The condition of repeated use is further severed by the repetitive nature of the punching action.

6. Natural frequency

All machine tools operate with some vibrations and in certain cutting operations these vibrations may have large amplitudes. In order that the recorded force may not be affected by the exciting force, it natural frequency must be large (at least 4 times as large) compared to the frequency of exciting force.

In our case the flywheel of the punch press is rotating at 150 R.P.M. Therefore the frequency of the exciting force, say, $_e$ will be

$$\omega_e = \frac{150}{60} = 2.5 \text{ cps} \qquad 14$$

Therefore the natural frequency of vibration of the dynamometer must be greater than or equal to 10 cps. For purpose of analysis any dynamometer can be reduced to a mass supported by a spring. The natural frequency $_n$ of such a system is given by

$$\omega_n = 1/2\pi \times \sqrt{K/m} \text{ cps} \qquad 15$$

where ω_n = Natural frequency of vibration in cps

K = Spring constant in lbs/inch

m = Mass in lbs. sec^2/inch

In terms of the supported weight of the dynamometer, say, W lbs.

$$\omega_n = 1/2\pi\sqrt{\frac{K \times 32.2 \times 12}{W}}$$

$$= 1/2\pi\sqrt{\frac{386.4 \times K}{W}} \text{ cps} \qquad 16$$

Where W = weight of the dynamometer in lbs. For the ease of analysis we shall consider the load column only.

Volume of the load column

$$= \frac{\pi}{4}\ (3^2 - 1.25^2)1/4 + (4^2 - 1.25^2)1/4 + (1.5^2 - 1.25^2)2$$

$$= \frac{\pi}{4}\quad 7.44 \times 1/4 + 14.44 \times 1/4 + 0.6875 \times 2$$

$$= \frac{\pi}{4} \times 6.845$$

$$= 5.37 \text{ in}^3 \qquad 17$$

Now the density of 1020 steel is approximately 0.284 lbs/in^3.

∴Weight of the load column = 0.284 x 5.37

= 1.525 lbs. 18

For analysis we shall consider the load column in three parts.

1. Column number one with outer and inner diameters as 3" and 1.25" with length of 1/4". Let the deflection of this column under the design load of 10,000 lbs be δ_1 inch. Now the stress developed due to load of 10,000 lbs. is

$$f_1 = \frac{10,000}{\pi/4(3^2 - 1.25^2)} = \frac{10,000}{.785 \times 7.44} = 1,750 \text{ lbs/in}^2$$

$$\therefore\ \delta_1 = \frac{f_1 \times 1/4}{E} = \frac{1,750 \times 1}{4 \times 30 \times 10^6} = .0000146" \qquad 19$$

2. Column number two with outer and inner diameters as 1.5 inch and 1.25 inch respectively with the length of 2 inches. Let the deflection of this column under the load of 10,000 lbs. be δ_2. Then the stress developed due to load of 10,000 lbs. is

$$f_2 = \frac{10,000}{\pi/4(1.5^2 - 1.25^2)} = \frac{10,000}{.785 \times .6875} = 18,500 \text{ lbs/in}^2$$

$$\delta_2 = \frac{18,500 \times 2}{30 \times 10^6} = .001232'' \qquad 20$$

3. Column number three with outer and inner diameters as 4 inches and 1.25 inch respectively with the length of 1/4 inch. Let the deflection of this column under the load of 10,000 lbs. be δ_3. Then the stress developed due to load of 10,000 lbs. is

$$f_3 = \frac{10,000}{.785(4^2 - 1.25^2)} = \frac{10,000}{.785 \times 14.44} = 882 \text{ lbs/in}^2$$

$$\delta_3 = \frac{882 \times 1}{4 \times 30 \times 10^6} = .00000735'' \qquad 21$$

Let the total deflection of load column be . Then

$$\delta = \delta_1 + \delta_2 + \delta_3$$
$$= .0000146 + .001232 + .00000735$$
$$= .001254'' \qquad 22$$

Now K is defined as load required for unit deflection. In our case the load column deflects through .001254" under load of 10,000 lbs.

$$\therefore K = \frac{10,000}{.001254}$$
$$= 7975 \times 10^3 \text{ lbs/inch} \qquad 23$$

Substituting the values of K and W from equations 23 and 18 respectively into equation 16 we get

$$\omega_n = 1/2\pi \sqrt{\frac{386.4 \times 7975 \times 10^3}{1.525}}$$

$$= 1/2\pi \sqrt{\frac{30.83 \times 10^8}{1.525}}$$

$$= 1/2\pi \sqrt{20.22 \times 10^8}$$

$$= \frac{4.48 \times 10^4}{2}$$

$$= 7150 \text{ cps} \qquad 24$$

From equations 23 and 13 it can be seen that natural frequency of vibration of the load column (7,150 cps) is far greater than the exciting frequency (2.5 cps).

Construction of the Dynamometer

All the parts designed so far - base plate, load column, die, die holder, stripper plate, protective cylinder, punch, transformer holders, top plate, extension plate, and connecting rod - were made by the author according to the design in the workshop of Industrial Engineering Department by using proper machine tools. The heat treatment of the punch and the die was done as follows:

The furnace used for this purpose was Waltz Electric furnace with salt bath. Both the parts were heated into this furnace in neutral salt bath at the temperature of 1475°F for 45 minutes. The neutral salt bath prevents the oxidation of the metal. After 45 minutes the die was quenched in brine solution and then tempered at 350°F for one hour in a dry furnace. Similarly after 45 minutes the punch was quenched in fish oil and then tempered at 350°F for one hour in a dry furnace. The hardness obtained with the die was found to be C61 Rickwell, while that with the punch was found to be C57 Rockwell.

Fixing the Strain Gages

The strain gages were placed at the upper end of the column. The gage R_1 was put with its axis coinciding with that of the column. The gage R_2 was placed in poisson orientation with the gage R_1. Similarly the gage R_3 was placed with its axis coinciding with that of the column. But it was placed diametrically opposite to the gage R_1. The gage R_4 was placed in poisson orientation with the gage R_3.

The fixing of the strain gages was done in the following steps:

1. Cleaning the surface
   a. All foreign matter from the surface was removed with the help of fine emery-paper.
   b. The surface was cleaned with gauze saturated with alcohol.
   c. The gage location was indicated with pencil.
   d. The surface was again cleaned with alcohol.
2. We used the epoxy cement. It was prepared as follows:
   a. Twelve drops of resin "A" were squeezed into one plastic cup. To this 9 drops of activator "B" were added.
   b. This mixture was gently stirred for 2 to 3 minutes, using a glass rod.
3. The gage was placed face up on clean surface and a piece of scotch tape was attached to lead wire end.
4. The gage assembly was lifted and terminal strip at the end of gage was located.
5. The back of the gage was cleansed with the cotton swab moistened with alcohol and then it was air dried.

6. Thin coat of cement was applied to the back of the gage and terminal strip, using glass stirring rod.
7. The gage was placed in position on speciman.
8. The gage installation was covered with sheet teflon and held in place with scotch tape.
9. The gage was clamped to the surface (1 to 15 psi) utilizing silicone gum pad.
10. It was allowed to dry for 24 hours.
11. Pad, teflon and tape were removed. Tape residue from specimen was cleaned and gage tabs were cleaned with alcohol again.
12. Lead-in wires were installed. Solder flux was removed with alcohol. Protective coating was applied.

After this the gage resistance test and bend damage test were conducted and the strain gages were found to be in good condition.

After this, all the elements (parts) were assembled together. Thus the dynamometer was made ready for calibration. Figure 20 shows the load column with strain gages fixed on it. Figure 21 shows the dynamometer in the assembled form.

Force Calibration

In order to properly calibrate the dynamometer, the strain gages, Sanborn recorder, J-box, power box and oscilloscope were connected as shown in Fig. 24 and the whole system was switched on. Then, in order to compensate for the residual unbalance of the bridge circuit and its cabling, the unit was balanced in the following manner:

1. Connect the amplifier and warm up for 30 minutes. Set the panel controls:

Fig. 20. Load column.

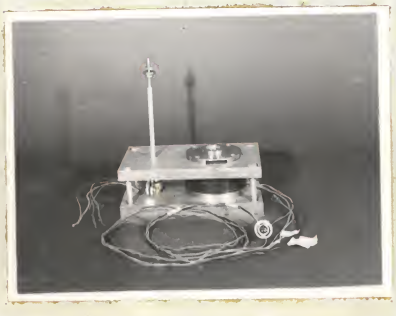


Fig. 21. Dynamometer.

R/T ..... T ATTENUATOR ..... OFF GAIN ..... FULL RIGHT

2. Set the FINE/COARSE switch to FINE. Center the stylus with the zero control, then set the FINE/COARSE switch to COARSE.

3. Remove all strain from the bridge (if there is any). Turn the ATTENUATOR to the right for a stylus deflection. Bring the stylus to its null position with the RES BAL and CAP BAL controls. Continue advancing the ATTENUATOR and bringing the stylus back to its null position until the ATTENUATOR is at XI position.

4. Set the FINE/COARSE switch to FINE. Now make a final adjustment of the RES BAL control so the stylus does not move when turning the ATTENUATOR between XI and OFF. Then return the ATTENUATOR to OFF.

5. The full bridge is now balanced, and the system is ready for calibration.

The method employed to calibrate the unit was to provide a know varying force on the load column and to measure the deflection on the oscilloscope screen at the desired force intervals, so that graph can be drawn with the force on Y-axis and the corresponding deflection on the X-axis.

The following steps were observed for the calibration:

1. The dynamometer was fixed to the base plate of a Universal (hydraulic) testing machine.

2. In order to simulate the actual conditions for the acting force, the die was placed on the load column.

3. The ATTENUATOR was set on X2 and GAIN was set to 10 lines of calibration under no load condition. This was measured by the CAL button.

4. The known force was applied at fixed intervals of 500 pounds. The deflection was noted on the oscilloscope screen.

5. The force was applied up to 7,500 lbs. and then was decreased by the same interval as before. The average of these two readings gave a deflection on the oscilloscope screen for a particular force. The readings are shown in Table 1. From this data a calibration curve, as shown in Fig. 22 was drawn. It was found that 10 mms. on oscilloscope screen amounted to 1,000 lbs. of punching force.

Table 1. Readings for the calibration of force.

Obs. No.	Punching force in lbs.	Oscilloscope signal traverse in mms. Increasing	Decreasing	Mean
1	0	0.00	0.00	0.00
2	500	5.00	5.00	5.00
3	1,000	10.00	10.00	10.00
4	1,500	15.00	15.00	15.00
5	2,000	20.00	20.00	20.00
6	2,500	25.00	25.00	25.00
7	3,000	30.00	30.00	30.00
8	3,500	35.00	35.00	35.00
9	4,000	40.00	40.00	40.00
10	4,500	45.00	45.00	45.00
11	5,000	50.00	50.00	50.00
12	5,500	55.00	55.00	55.00
13	6,000	60.00	60.00	60.00
14	6,500	65.00	65.00	65.00
15	7,000	70.00	70.00	70.00
16	7,500	75.00	75.00	75.00

Explanation of Fig. 22.

Calibration curve for punching force.

ATTENUATOR - X2		CAL - 10 lines
Scale:	X-axis	10 mms. = 3/4 inch
	Y-axis	10,000 lbs. = 3/4 inch

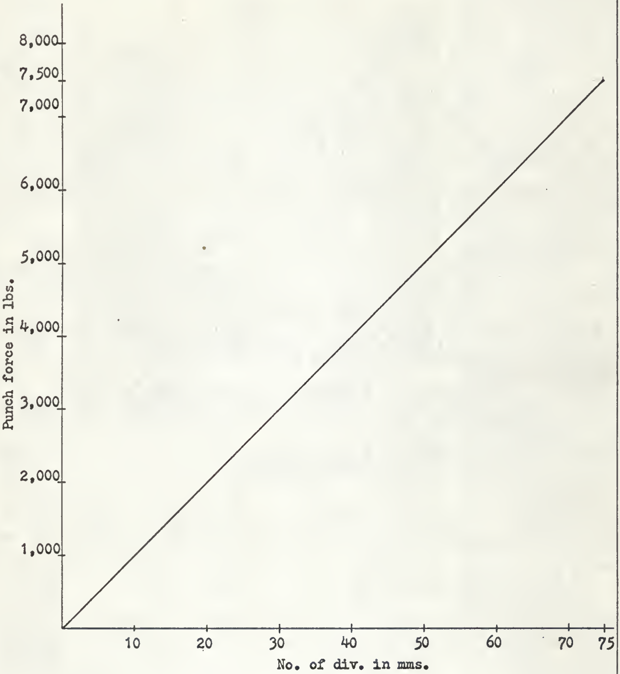


Fig. 22. Calibration curve for force.

Calibration for Stock Thickness

First of all the dynamometer was mounted on the punch press. Then the LVDT, J-box, power box and oscilloscope were connected as shown in Fig. 21. Then the whole system was switched on. Several pieces of 1/32" thick metal were taken and their thickness was measured at several different points. The thickness was found to be 0.033". These plates were inserted over the die and the ram was lowered down such that it just touched these plates. The point was noted on the oscilloscope. Then these plates were pulled out one by one and the ram was lowered by the corresponding amount. The traverse or movement on the oscilloscope signal was noted for every position of ram. The readings are shown in Table 2. From this data calibration curve, as shown in Fig. 23, was drawn. It was found that 10 mms. on oscilloscope represented .033" of the ram movement.

Table 2. Readings for the calibration of stock thickness.

Obs. No.	Thickness in thousands of an inch	Oscilloscope signal traverse in mms.
1	0	0
2	33	10
3	66	20
4	99	30
5	132	40
6	165	50
7	198	60

Explanation of Fig. 23.

Calibration curve for stock thickness

Scale:	X-axis	1 inch = 10 mms.
	Y-axis	1 inch = .025 inch

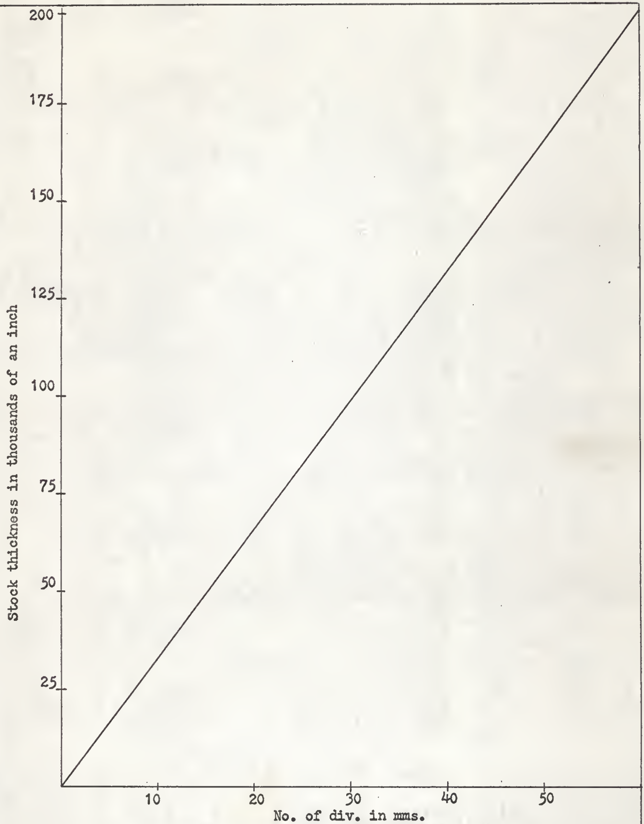


Fig. 23. Calibration curve for stock thickness.

Testing of the Dynamometer

First of all the dynamometer was rigidly fixed on the base plate of the Bliss punch press. Then the punch was inserted in the ram and alligned with the hole in the die. The stroke of the ram was adjusted to 1/4 of an inch. Then the strain gages and LVDT were connected to J-box, power box and oscilloscope as shown in Fig. 24. The entire system was then electrically switched on. The recorder was allowed to warm up for 30 minutes. The recorder was balanced and attenuator was set on X2. The gain was set to 10 lines with the help of CAL button. Then the origin was fixed on the screen of the oscilloscope. The punch press was switched on. About 10 runs of punching 1/16" mild steel plate were made to ascertain that entire curve appears on the oscilloscope screen. The oscilloscope camera was fixed on the oscilloscope screen and the system was then ready to run the experiment.

The experiment was conducted by punching 1 inch diameter holes in 1/16 inch thick mild steel plate with 1/4 inch stroke. The force versus stroke of the ram curve was obtained as shown in Fig. 26.

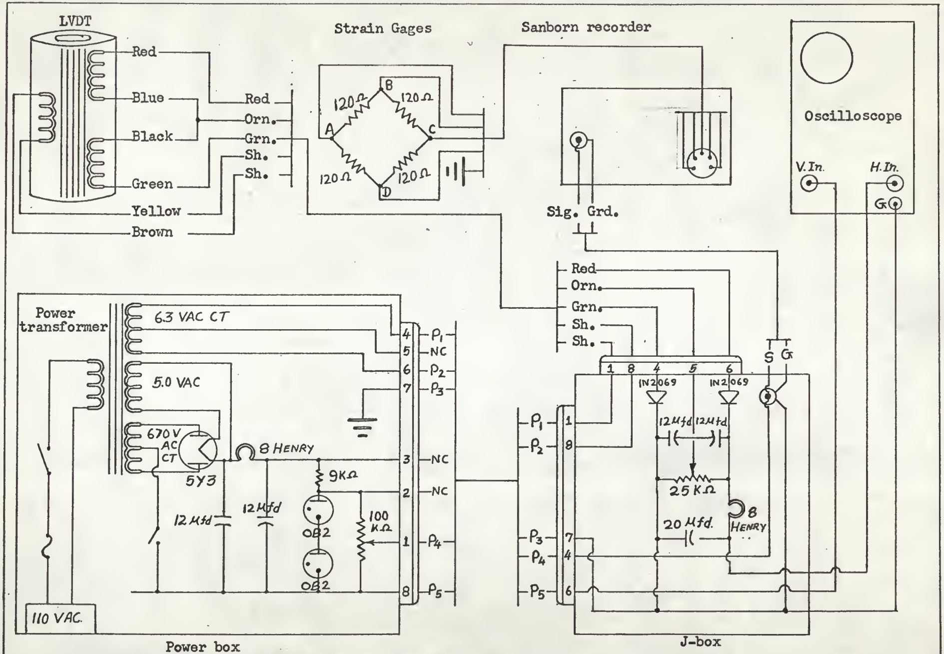


Fig. 24. Instrumentation system for plotting punch force versus position.

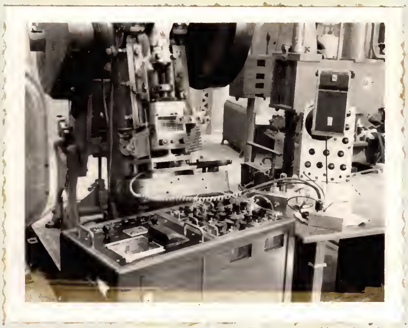


Fig. 25. Instrumentation set up for the test.

Diameter of hole punched = 1 inch.

Scale for force on Y-axis 1 cm. = 1,000 lbs.

Scale for stock thickness on X-axis 1 cm. = .033 inch.

Fig. 26. Force versus stock thickness curve for 1/16" thick mild steel plate.

RESULTS AND CONCLUSION

The maximum force was found to be 4,000 lbs. when the length of the stroke was one-quarter of an inch. At this time the maximum force was found to be acting at approximately one-third of the stock thickness of the metal.

The theoretical force required to punch a 1 inch diameter hole in 1/16 inch thick mild steel plate was:

$$P_{max} = Ltf_s \qquad \text{where } L = \odot^{ce} \text{ of the hole}$$

$$= \pi d\ t\ t_s \qquad t = \text{thickness of stock}$$

$$= \pi \times 1 \times 1/16 \times 10{,}000$$

$$= 1963 \text{ lbs. force.} \qquad f_s = \text{shear strength of material}$$

25

Thus the force recorded during the test was found to be double the theoretical force. This was believed to be due to the impact loading of the load column.

During the above test, it was observed that the stroke was too small to strip off the stock from the punch. Therefore one more test was conducted by increasing the stroke to one-half inch. During this test the force was found to be 2,000 lbs. which is very closely equal to the theoretical force of 1963 lbs. This time also the maximum force was found to be acting at approximately one-third of the stock thickness of the metal.

Thus it was found that the force recorded by the dynamometer concurred with the theoretical force when the stroke was one-half inch. On the other hand, the dynamometer recorded double the theoretical force when the stroke was one-quarter inch.

The punching force should remain constant irrespective of the length of the stroke. Therefore more work needs to be done to account for these peculiar results that were obtained. But the work of this project was concluded at this stage as the further investigation into this problem was beyond the scope of this project.

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DESIGN, CONSTRUCTION, CALIBRATION AND TESTING OF A PUNCH PRESS DYNAMOMETER

by

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The need of this project was felt to further the research work in the field of metal stamping at Industrial Engineering Department. The purpose of this project was to design and construct a punch press dynamometer that will corelate the punching force with the stock thickness of the material being punched. Thus the variables to be measured were the force and the distance simultaneously.

The force was measured indirectly. For this purpose, a hollow cylinder of steel, called load column, was made. The load column was designed in such a way that, for the design load of 10,000 lbs., the stress in the column never exceeded the elastic limit. Its cylinder ratio was 4.4. The die was placed on the load column. Thus during any punching operation, the punching force was ultimately transferred to the load column. Because of this punching force the load column underwent longitudional deflection. This deflection was sensed by the four strain gages which were fixed on the load column. The strain gages were fixed on the load column in a pair of two on two diametrically opposite sides of the column. The two strain gages of each pair were poisson oriented with each other. The two strain gages forming a pair were series connected. The poisson oriented strain gage of one pair was connected with the non-poisson oriented strain gage of the other pair. Thus the four strain gages formed the circuit of the Wheatstone's bridge. These strain gages were connected with the Sanborn recorder. But the stylus of the recorder was too sluggish to react to the dynamic punching forces. Therefore, the recorder was further connected, by the proper coupling circuit, to the Y-axis of the oscilloscope. Thus the punching force and deflection of the beam on Y-axis were linearly related with each other. We got a linear relationship because of the linear stress-strain curve within the elastic limit.

The distance was also measured indirectly. For this purpose a linear variable differential transformer (LVDT) was used. The linear range of the transformer was ± 0.3 inch. The core of the LVDT was rigidly connected with the ram of the punch press by proper links. When the primary circuit of the LVDT was supplied with an excitation of 6V 60 cps., the LVDT secondary circuit gave out a signal depending upon the position of the ram. The signal given out by the secondary was linearly related with the displacement of the core. The signal out-put of the LVDT secondary was supplied to the X-axis of the oscilloscope by proper coupling circuit. Thus the deflection of the beam was linearly related with the movement of the ram.

Thus the beam of the oscilloscope was subjected to two simultaneous motions proportional to the punching force and the position of the ram.

The dynamometer was calibrated for force as well as distance. The scale was found to be 1 cm. equal to 1,000 lbs. for force and 1 cm. equal to .033" for distance.

The instrument was tested by punching 1 inch diameter hole in a 1/16 inch thick mild steel plate. On the oscilloscope screen a curve was obtained indicating the forces at different thicknesses of the stock. This was recorded with the help of an oscilloscope camera working on the Polaroid Land principle.

The maximum force was found to be 4,000 lbs. acting at approximately one third the thickness of the stock from the top, when the stroke was one quarter of an inch. The maximum force fell down to 2,000 lbs., but still acting at one third the thickness of the stock, when stroke was increased to half an inch.

The punching force should remain constant irrespective of the stroke length. Therefore, more further work was needed to account for these peculiar results that were obtained. But the work of this project was concluded at this stage as further investigation into this problem is beyond the scope of this project.