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Manufacture of Cylinders With Axial Imperfections

Final Report

By
Rd. Khairilhijra KHIROTDIN

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Supervisor
Dr David N. Moreton

SUMMARY

The report outlines the methodology and procedures used to manufacture the thin walled cylinder with its axial imperfections using a NUM 1060 CNC Lathe machine. By introducing the known geometrics imperfections, the thin cylinder manufactured could be analyzed under different load conditions and with different type of applications. The cylinders manufactured were to have minimum thickness set up with their flat end were machined to follow a periodic sinusoidal wave's function of specified amplitude and frequency. In addition, the flat end cylinder circumference surface was also required to have a constant chamfer angle either from the outside or inside cylinder surface.

A number of experiments were attempted to determine the minimum thickness, the sinusoidal wave's function and the constant chamfer angles required. At the end of the project, the machining techniques and procedures used to manufacture the cylinder were successfully established as well as the CNC part programs which played a major role in producing the imperfections required. A result of a satisfactory final thin cylinder was manufactured even though some difficulties had occurred throughout the whole project. Finally, all the information contained were vital for understanding the concept of manufacturing the cylinders with axial imperfections. It indicates some essential findings for the clear description of the project purposes and the necessary works that undertaken through the completion of the project

ACKNOWLEDGEMENT

Grateful thanks are due especially to my supervisor, Dr. David N. Moreton for his help, guidance and assistance in completion of this project as well as Dr. Jan T. Blachut for his help whenever needed.

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NOTATION *(define all symbols used (together with the units for each quantity)*

P	Load	Newton (N)
P_{cr}	Critical Load	Newton (N)
Δ	Displacement	Millimetre (mm)
Δ_{cr}	Critical Displacement	Millimetre (mm)
R	Radius	Millimetre (mm)
t	Thickness	Millimetre (mm)
t_c	Chamfer thickness	Millimetre (mm)
σ_{cr}	Critical compressive axial normal stress	Newton (N)
E	Modulus of Elasticity	GPa
ν	Poisson's Ratio	-
L	Length	Millimetre (mm)
γ	"Knock-down" factor	-
ϕ_{OD}	Nominal outside diameter	Millimetre (mm)
ϕ_{ID}	Nominal inside diameter	Millimetre (mm)
A	Amplitude	Millimetre (mm)
n	Cycle	-
θ	Theta (Angle)	Degree

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1. INTRODUCTION

Manufacturing thin cylinders with axial imperfections is part of a continuous research programme undertaken by the University of Liverpool in order to investigate their behaviour under different load conditions with different applications. The errors that are going to be detected and ways on how to minimise such errors were also being analysed. The cylinders manufactured are considered to be a simulation of large cylinders used in different applications. Hence, all the dimensions used were carefully selected to simulate the larger ones. In general, cylinders sometimes become unbalanced as a result of imperfect machining. Unbalanced cylinders create excessive vibrations, which leads to problems in the cylinders and thus severe deformation and failure occurred as shown in Figure 1 below;

Figure 1 – Example of cylinder undergone severe deformation and failure

Many studies and works have been carried out to analyze the influence of geometric imperfection on the buckling behaviour of cylinders. The influence that imperfections in the geometry or in the load have on some bifurcation buckling problems has been acknowledged since the pioneering work of Koiter in the context of a general theory of structural stability. Shells and other slender structures with unstable post-buckling behaviour are sensitive to the presence of small imperfections, and bifurcation behaviour typical of such structural forms in their idealized or perfect configurations is lost when imperfections are considered. The response of some structures depends strongly on the imperfections in the original geometry, particularly if the buckling modes interact after buckling occurs. Imperfection sensitivity has long been recognized as the main factor for discrepancies between experimental buckling loads and analytical predictions for shell structures in general, and for cylindrical shells subject to axial compression in particular. Some very important findings undertaken by the Imperial College of Science for the behaviour of cylinders and the precautions that has to be taken to avoid imperfections while manufacturing [8-3]. In the review of N. Panzeri and C. Poggi works [8-5], the effects of the geometric imperfections and other defects due to the manufacturing process, like the thickness variations or possible misalignments are analysed with the objective of producing relevant suggestions for the designers. In addition, Minjie, J. Mark and J. Micheael [8-6] have identified two possible and distinct buckling phenomena for a wide range of shell geometric imperfections. It was also found that the influence of imperfections have a moderate effect on the buckling strength of the shell examined. Moreover, a study by Charles D. Babcock, Jr. [8-7] on the shells fabricated by a copper electroforming process in which the imperfection would take the form of axially symmetric shells shape and had the form of a half sine wave in the length direction was carried out to examine the imperfection effect on the buckling load of a circular shell under axial compression loading.

By introducing the known geometric imperfections required, the buckling behaviour of the cylinders could be analyzed with different type of applications under different loads. In this project, the used of Computer Numerical Control (CNC) machining and programming as well as Pro-Engineer software were implemented in order to manufacture the thin cylinder with their specific imperfections required. Most of the experimental works used a NUM 1060 CNC lathe machine. At the end of the project, the cylinder manufactured had satisfying the characteristics required even though the number of cylinder manufactured was less than expected. However, the complete manufacturing procedures and processes were successfully explained and established in the report as well as the part programs to manufacture the cylinder. The reasons for the small amount of cylinder manufactured were also indicated but they were mostly depended on the difficulty of the project, the material available, the availability of the lathe machine and the CNC programming that consumed most of the time throughout the project.

2. PROJECT OBJECTIVE/SPECIFICATIONS

The main objective of this project is to manufacture thin walled cylinders having axial imperfections. By introducing geometric imperfection in the cylinder, the buckling load of the cylinder can be found under different load conditions with different applications respectively. The manufacturing concept and procedures used to machine the cylinders could also be established. There were several tasks set up in order to achieve the aim targeted. The mentioned tasks are;

i) The dimensions of the cylinders including the diameter, the thickness and the length are needed to be carefully selected since small imperfections will induced a different buckling load in the cylinders
ii) The coordinate points on the periodic wave profile have to be obtained accordingly
iii) The programming codes to machine the cylinder with its imperfection profile has to be generated
iv) All the drawings prepared in designing the cylinder will use Pro/Engineer software
v) CNC part programming and CNC lathe machine (Churchill/NUM 1060) will be used as a machining method to manufacture the cylinders
vi) The dimensional measurement will use a specific measuring device in order to gain an accurate and precise assessment of the machining accuracy

Some specifications were required in order to meet the tasks mentioned above. These specifications are described below;

i) The cylinders manufactured must be constant in thickness, diameter and length
ii) The targeted minimum thickness of the thinnest cylinder is 0.20 mm
iii) The imperfection would take the form of sinusoidal function periodic waves
iv) The amplitude of the waves would be varying but in between specified limit
v) The cylinders thin section would have a constant chamfer angle on its flat end either from the inside or outside surface of the cylinder
vi) The number of cylinders manufactured will depend on the material and the time available

3. THEORETICAL BASIS

3.1 The Buckling Phenomenon

Thin-walled cylindrical shells are commonly used in aerospace, civil, mechanical engineering and others. In particular for cylindrical shells under axial compression, the buckling behaviour is an important design factor. *Buckling* of a structure means failure due to excessive displacements (loss of structural stiffness), and/or loss of stability of an equilibrium configuration of the structure. The rule of thumb is that buckling is considered a mode of failure for slender members in compression, or for thin panels in compression or shear. Stability of equilibrium means that the response of the structure due to small disturbance from its equilibrium configuration remains small; the smaller the disturbance the smaller the resulting magnitude of the displacement in the response. If a small disturbance causes large displacement, perhaps even theoretically infinite, then the equilibrium state is unstable.

Practical structures are stable at no load. Now consider increased the load slowly. We are interested in the value of the load, called the critical load, at which buckling occurs. That is, we are interested in when a sequence of equilibrium states as a function of the load, one state for each value of the load, ceases to be stable (Figure 3.1.1).

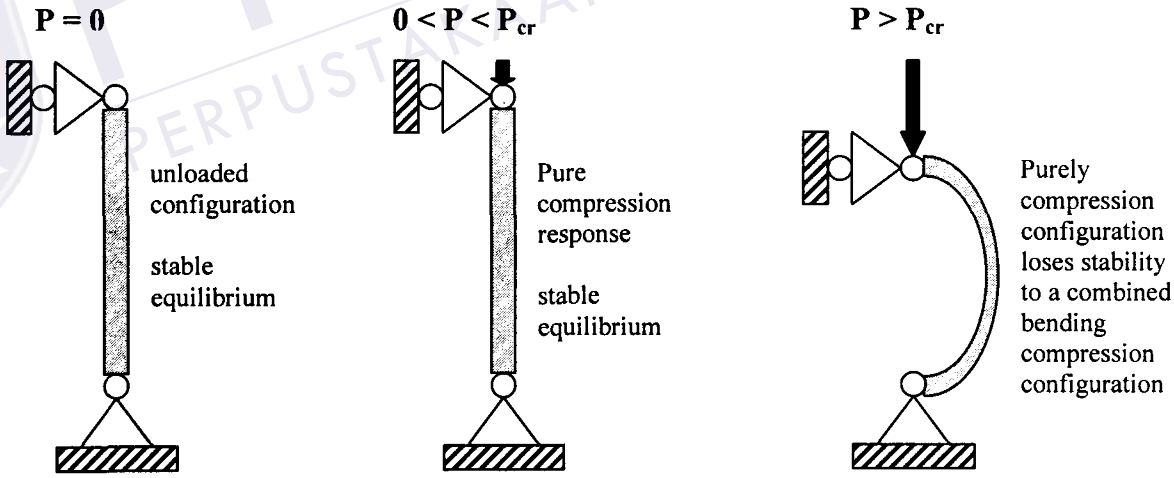


Figure 3.1.1 – Stability of equilibrium with respect to load

By observing on the load-shortening curve (Figure 3.1.2) for a circular cylindrical shell, it was found that the shell cannot resist increased load after buckling. The load and displacement decrease on the initial, unstable post buckling equilibrium path. The shell has no post buckling strength. Thus, designers have to "knockdown" the value of P_{cr} obtained from the theory of the perfect shell by a substantial amount.

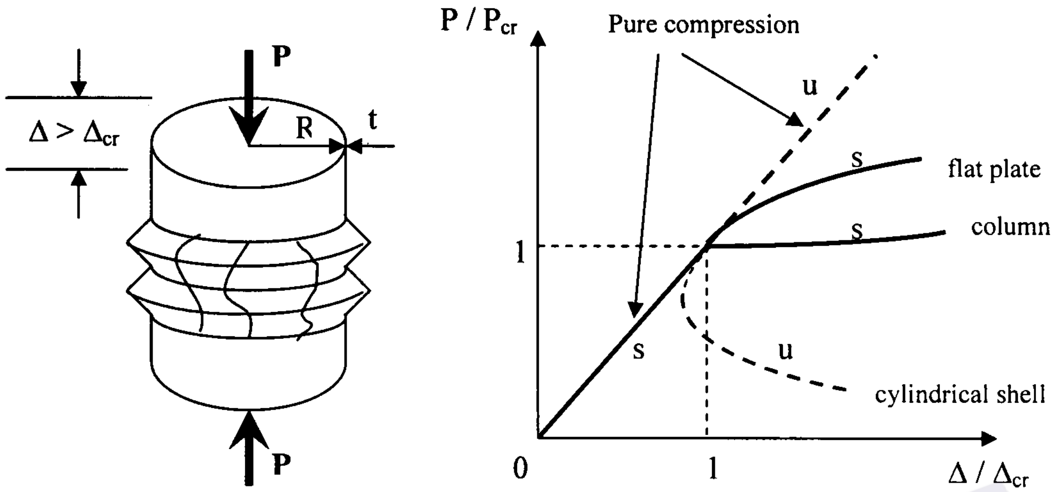


Figure 3.1.2 – Load-shortening curves

Buckling is associated not just to a structure, but also to the whole structural system. To visualize a buckling process it is necessary to consider the load-deflection diagram as shown in Figure 3.1.3. The sequence of equilibrium points in this diagram is known as an *equilibrium path*. The equilibrium path emerging from the unloaded configuration is called the *fundamental* or *primary path*, also the *pre-buckling path*. This path may be linear (or almost linear) or maybe nonlinear. The load level at which there is a change in the shape is called *buckling load* and the emerging geometry is called the *buckling mode*.

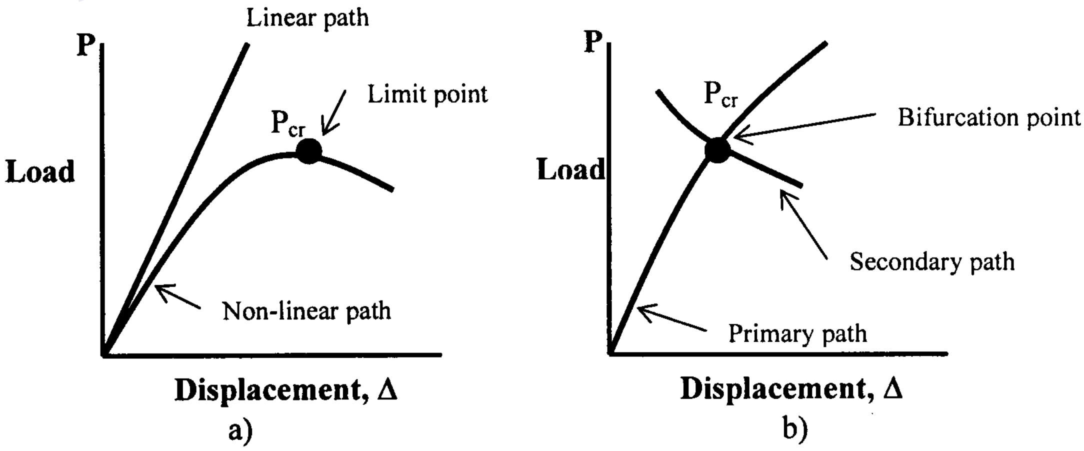


Figure 3.1.3 – Load deflection diagrams showing equilibrium paths a) limit point, and b) bifurcation point

There are several ways in which this process may happen; in *snap buckling* where the fundamental path is nonlinear and reaches a maximum load, at which the tangent to the path is horizontal. This state is called *limit point* (Figure 3.1.3 a)). The change in the shape occurs in a violent way. In *bifurcation buckling*, the fundamental path may be linear and it crosses another equilibrium path, which was not present at the beginning of the loading process (Figure 3.1.3 b)). The state at which both path across is called a *bifurcation point*. Both limit and bifurcation points are called *critical points* or critical states.

3.2 General Consideration

When a cylindrical shell is subjected to uniform axial compression (*Figure 3.2.1*), buckling can occur in two possible modes;

i) overall column buckling which does not involves a local deformation of the cross-section

ii) shell buckling which involves local deformation of the cross-section and can be either;

   a) axisymmetric, in which the displacements are constant around any circumferential section

   b) asymmetric, in which waves are formed in both axial and circumferential directions

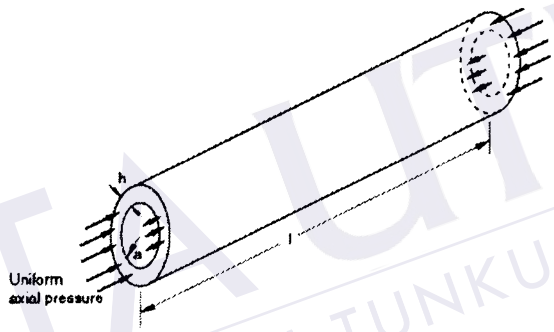


Figure 3.2.1 – Cylindrical shell with uniform axial loading

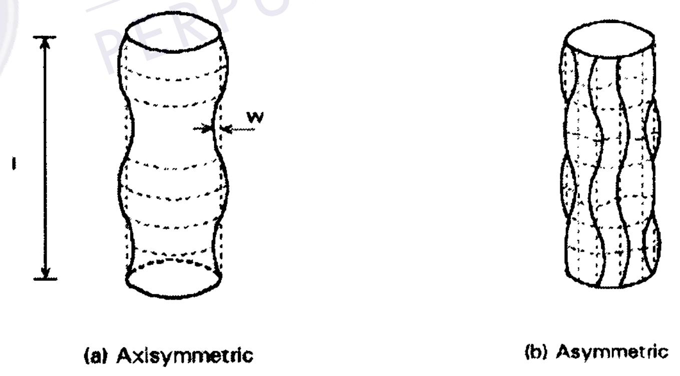
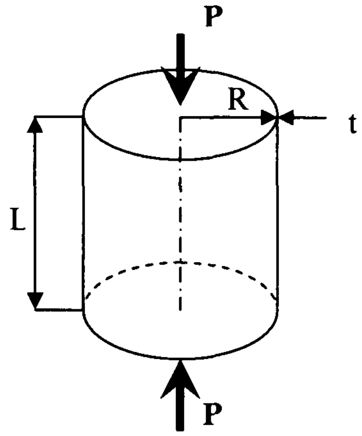


Figure 3.2.2 – Buckling modes for cylinders under axial compression

Axisymmetric buckling is more often encountered in short and/or relatively thick cylinders and asymmetric buckling is more common in thin and/or relatively long cylinder.

The simplest case to analyse the buckling load for a axially compressed circular cylindrical shells is by implementing the formula for the critical compressive axial normal stress, σ_{cr} that is as follows;

$$\sigma_{cr} = \frac{1}{\sqrt{3(1-\nu^2)}}\left(\frac{Et}{R}\right) \qquad (1)$$



where E is the Modulus of Elasticity, t is the cylinder wall thickness, R is the cylinder radius and ν is the Poisson's Ratio. The corresponding compressive axial normal force, P_{cr} at the buckling is obtained from

$$P_{cr} = \sigma_{cr}\,(2\pi Rt) \qquad (2)$$

The elastic critical buckling stress equation (Equation (1)) could not be used directly for design because cylindrical shells are extremely sensitive to imperfections under axial compression. To account for imperfections, design rules traditionally use a "knock-down" factor, γ, which accounts for the fact that experimental values of the buckling load of axially compressed circular cylinder shells are substantially less than the theoretical prediction. That is, the design buckling load is related to the theoretical value by;

$$\sigma_{cr}\Big|_{design} = \gamma\sigma_{cr}\Big|_{theory} \qquad (3)$$

The product of $\gamma\sigma_{cr}$ represents the buckling load of the imperfect shell. In addition, the plasticity effects, which are important for a certain range of cylinder geometries, must also be taken into consideration. The "knock-down" factor is in general a function of shell geometry, loading conditions, initial imperfection amplitude and other factors and is normally evaluated from comparison with experimental results. The "knock-down" factor is selected so that a high percentage of experimental results (for example, 95%) should have buckling loads higher than the corresponding loads predicted by the design method.

Due to the high sensitivity to imperfections, the design method should specify the maximum allowable level of imperfections. These tolerances are related to the imperfection amplitudes measured in the tests used in determining the appropriate "knock-down" factors. Clearly, the tolerances should not be so strict that they cannot be achieved using normal manufacturing processes. It should be noted that the use of experimental databases containing a large number of test specimens which are not representative of full-scale manufacturing, might lead to inaccurate "knock-down" factors. Ideally, the design method should also enable a designer to evaluate the buckling load of a cylinder with imperfections, which exceed the allowable limits. Currently, very few design methods are valid for larger imperfections and the importance of adhering to the stated tolerances cannot be over-emphasised. The experimental databases used by various codes in estimating "knock-down" factors can vary substantially. It is true to say that some design proposals are based on old, limited or inappropriate data. For this reason, design predictions for the buckling load of nominally identical cylinder geometries can vary substantially.

By the use of a "knock-down" factor, the strong buckling sensitivity of this class of structures to geometric imperfections is usually taking into account the estimated load carrying capability and this is thereby reduced to a level deemed appropriate for shells of a given radius to thickness ratio. "Knock-down" factor has been established empirically using a large but diverse set of experimental buckling loads obtained for cylinders, which are designed to buckle in the elastic range.

A structure is imperfection sensitive if small changes in an imperfection change the buckling load significantly. The buckling behaviour of cylinders is highly influenced by the presence of geometric imperfections and that can cause large discrepancies between experimental loads and analytical values. By introducing a geometric imperfection in the cylinder, the buckling load of the cylinder can be analyzed. The sensitivity to imperfection depends primarily on the type of cylinder and type of loading and to some extent on the boundary conditions. It may vary from moderate to extreme, even for the same cylinder geometry under different loading or boundary conditions. An imperfection destroys a bifurcation point, and a new equilibrium path is obtained. As the amplitude of the imperfection increases, the paths deviate more from the path of the perfect system. Some of the general considerations mentioned above were taken from the previous researcher [8-4] which is vital to the clear understanding of the concept and purposes of the project.

3.3 Background on the Machining

Numerical control of machine tools has evolved steadily over the past forty or so years. Numerical Control (NC) is the term used to describe the control of machine movements and various other functions by instructions expressed as a series of numbers and initiated via an electronic control system. Early control systems were capable of only numerical control of a machine's axes. With the increase in performance and availability of microprocessors the manufacturers of control systems were able to increase the flexibility of their systems. Such control systems, having computing capabilities, have become known as Computer Numerical Controlled (CNC) Systems. CNC is the term used when the control system includes a computer. CNC has been used since the early 1970s.

While the specific intention and application for CNC machines vary from one machine to the other, all forms of CNC have their common benefits. The first benefit offered is automation. The operator intervention related to produce work pieces could be reduced or eliminated. Most of the CNC machine nowadays runs unattended during the entire machining process, freeing the operator to do other tasks. This gives the CNC user several side benefits including reduced operator fatigue; reduced mistakes by human error, consistent and predictable machining time for each work piece. The second benefit is consistent and accurate work piece produced. Today CNC machines produced almost unbelievable accuracy and repeatability specifications. Multiple identical work pieces can be easily produced with precision and consistency. A third benefit offered by most forms of CNC machine tools is flexibility. Since these machines are run from programs, running a different work piece is almost as easy as loading a different program. Once a program has been verified and executed for one production run, it can be easily recalled the next time the work piece is to be run. This leads to yet another benefit, fast changeovers. Since these machines are very easy to setup and run, and since programs can be easily loaded, they allow very short setup time. This is imperative with today's Just-In-Time product requirements.

The most basic function of any CNC machine is automatic, precise, and consistent motion control. Rather than applying completely mechanical devices to cause motion as is required on most conventional machine tools, CNC machines allow motion control in a revolutionary manner. Instead of causing motion by turning cranks and hand wheels as is required on conventional machine tools, CNC machines allow motions to be commanded through programmed commands. The preparation of numerical data prior to input of the machine control unit is referred to as programming. In numerical terms, part programming refers to complete programming. The extent of the programming preparation is depended on the complexity of the components that is going to be manufactured. Generally speaking, the motion type (rapid, linear, and circular), the axes to move, the amount of motion and the motion rate (feed rate) are programmable with almost all CNC machine tools.

All forms of CNC equipment have two or more directions of motion, called axes. These axes can be precisely and automatically positioned along their lengths of travel. The two most common axis types are linear (driven along a straight path) and rotary (driven along a circular path). All CNC controls allow axis motion to be commanded in a much simpler and more logical way by utilizing some form of coordinate system. The two most popular coordinate systems used with CNC machines are the rectangular coordinate system and the polar coordinate system. By far, the more popular of these two is the rectangular coordinate system.

3.4 CNC Lathe Machine Overview

The mentioned cylinders will be manufactured using "Churchill" lathe machine located in the laboratory in the basement of the Harrison Hughes building. The picture of the machine is shown below in *Figure 3.4.1*;

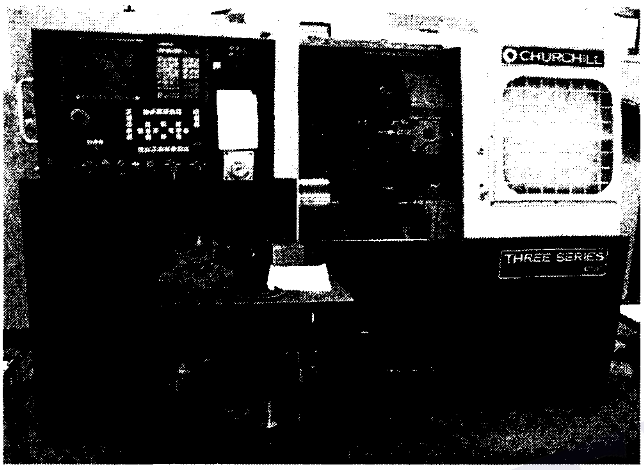


Figure 3.4.1 – Picture of Churchill Lathe Machine

The standard ISO 841 defines the coordinate system of this machine in accordance to its manufacturer. The X, Y and Z-axes, parallel to the machine slide ways; form a right-handed rectangular Cartesian coordinate system. This coordinate system measures tool movements with respect to the part to be machined, assumed fixed. The direction of the axis of a machine depends on the type of machine and the layout of its components. In general, a rectangular Cartesian coordinate system is a direct three axes system of three linear axes, X, Y and Z with which are associated three rotary axes, A, B and C as shown below in *Figure 3.4.2*;

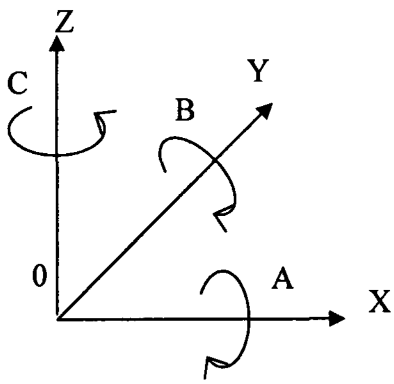


Figure 3.4.2 – Diagram of X, Y and Z-axes associated with three rotary axes, A, B, C

The relationship of the X, Y and Z-axes is easily remembered by the right-hand rule as shown below in *Figure 3.4.3*;

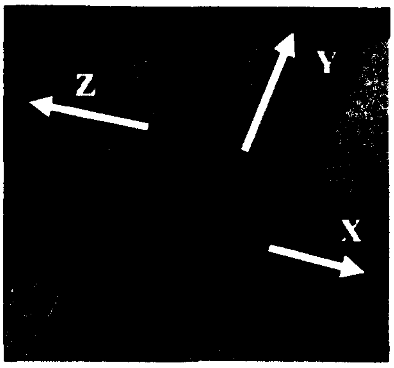


Figure 3.4.3 – Diagram of X, Y and Z using the right-hand rule

The positive direction of the rotation of a rotary axis corresponded to the direction of screwing of a right hand screw on the associated axis. For a lathe machine;

a) the Z-axis is the same as the spindle axis

ii) the X-axis is perpendicular to the Z axis and corresponds to radial movement of the tool holder which is located in the turret

iii) the Y-axis is generally a dummy axis which forms a right-handed coordinate system with the X and Z-axes

The positive movement along the Z or X-axes increases the distance between the part and the tool. The rotary axes A, B and C defines rotations around axes parallel to X, Y and Z and the secondary linear axes U, V and W may or may not be parallel to primary axes X, Y and Z. The *Figure 3.4.4* shows a typical arrangement of a lathe machine and the axes in order to manufacture the cylinder required.

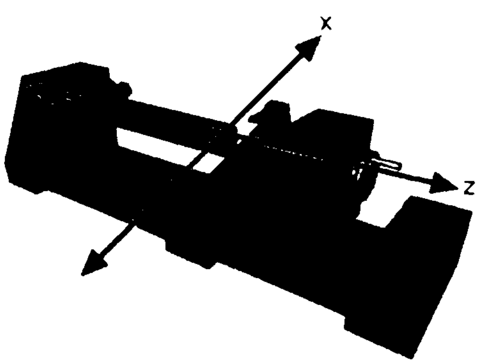


Figure 3.4.4 – Typical arrangement of a lathe machine

The programming of the components could be done in cylindrical coordinates. The NUM 1060 system used here performs cylindrical/polar coordinate conversion (Y-Z converted to Z-C). Interpolation of the C-axis allows milling on the developed cylinder with radius X. An auxiliary spindle drove the tool as shown in *Figure 3.4.5* below;

Figure 3.4.5 – C-axis interpolation

4. DESCRIPTION OF EXPERIMENTS

4.1 The Material Used

The specification of the material used to manufacture the cylinder is shown below in Table 4.1.1;

Material used : Carbon Steel / ASTM A106 Grade B Tube

Table 4.1.1 – Material dimensions

No.	Parameter	Hand Measurements	Manufacturer Reference Book
1	Nominal Outside Diameter, ϕ_{OD}	170.00 mm	168.30 mm
2	Nominal Inside Diameter, ϕ_{ID}	146.00 mm	146.36 mm
3	Nominal Thickness, t	12.00 mm	10.97 mm
4	Total Length, L	229.00 mm	-

Figure 4.1.2 shows a photograph of the material used to manufacture the thin-wall cylinder.

Figure 4.1.2 – Carbon Steel/ASTM A 106 Grade B Tube

Some of the typical mechanical properties for the Carbon Steel were given below in Table 4.1.3, which is useful for testing purposes.

Table 4.1.3 – Mechanical properties

No.	Properties	Value	No.	Properties	Value
1	Young's Modulus	207 GPa	5	Fracture Toughness	87.4 MPa √ m
2	Shear Modulus	83 GPa	6	Tensile Strength	552 MPa
3	Poisson's Ratio	0.30	7	Density	7.85 g/cm3
4	Yield Strength	448 MPa	8	Ductility	25% EL

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