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INVESTIGATION OF LIQUEFIED PETROLEUM GAS CYLINDERS IN JORDAN

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List of Contents.

List of Figures

Nomenclature

DR Radial deformation.

d_{st} maximum deflection caused by dynamic load.

d_{max} maximum deflection caused by dynamic load.

E modules of elasticity.

e_t unit elongation in tangential direction.

f shear deformation factor.

h height of dropping.

i^,j^,k^ unit vectors.

J Jacobian.

LPG Liquefied Petroleum Gas .

N_i shape function.

n^ normal vector

P internal pressure.

P_{dy} dynamic loading.

r,s,t element coordinate.

R radius of cylinder .

R_t tangential radius of the surface

R_L lougituinal radius of the surface.

S ratio of the slope of the stress strain in the plastic range to the slope in the elastic range.

S_{st} static deflection under weight load only .

S_{dy} dynamic deflection under dynamic loading.

t thickness of cylinder wall

u,v,w nodal displacement in x,y,z direction.

u*,v*,w* global displacement caused by rotation of the normal.

v Poison ratio.

W weight.

x*,y*,z* global coordinate of mid surface node.

x,y,z general global coordinate.

σ_t tangential (governing) stress

σ_L lougituinal stress

Abstract

INVESTIGATION OF LIQUEFIED PETROLEUM GAS CYLINDERS IN JORDAN

BY

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The main objective for this study is to investigate Liquefied Petroleum Gas (LPG) Cylinders under normal conditions of usage. Cylinder under high and low pressure, static loading, dynamic loading and corrosion effect.

Finite element model with 5180 D.O.F and 468 shell elements permitting elastic-plastic analysis of the thin shell structure of LPG cylinder was used and compared with some experimental and analytical results.

chapter 1

Introduction

1.1 General

In Jordan, Transposable Liquefied Petroleum Gas Cylinders are widely used for domestic and some industrial applications. More than 2 million cylinders are available, Due to mishandling 40 000 repaired annually, also 6000 are put out of use.

1.2 Objective of research.

The need for this kind of research is obviously due to the huge number of loses in gas cylinders due to mishandling. So cylinder have been studied under normal condition of usage: pressurizing, static loading, dynamic loading and corrosion to see the effect of each on cylinder strength.

1.3 Scope of work .

In analyzing LPG cylinder the principal loads to be considered are internal pressure, static load, dynamic load and corrosion.

i. *Internal pressure* : cylinder under low pressure (0-10 bar) and high pressure (10-100 bar) was studied using finite element, analytical relations and experimental result obtained from bursting tests performed by Oil Refinery, then results are compared.

In the static structural design of ductile internally pressurized pressure cylinder closed by thin torispherical heads , Generally there are two failure modes which have to be taken into account. One of these is nonsymmetric buckling, in which waves or "wrinkles" from around the circumference in the knuckle, region, this mode has been considered in a fair amount of detail in previous papers.

The other possible failure mode is plastic collapse which is an axisymmetric mode, in which large plastic deformations occur; this mode is similar to the limit pressure modes, in which yield circles occur at several meridian positions. In this study one of the interests will be calculating pressure effect on maximum principle stresses in liquefied petroleum gas cylinders, Also interest in the calculation of the plastic collapse pressures of torispherical shells.

ii. *Static load :* During storage , cylinders are arranged one over one , thus lowest one is subjected to highest static load which will increase the stresses in the cylinder membrane.

When cylinders are layout static load is considered to be a distributed line force a long the cylinder.

iii *Dynamic load:*

Dynamic loading in LPG cylinder refers to force suddenly applied to cylinder, which may be shock or impact loads ,shock loading is usually produced by sudden application of force which may be result of hitting cylinder by means of metallic bodies, whereas impact loading result from the collision of bodies and that is exactly what happen when dropping cylinder and cylinders collision.

A finite element modeling for LPG cylinder was introduced with 471 elements and 5180 degree of freedom. Nine nodes isoparametric Lagradian elements with 5 degree of freedom per element as shown in fig. (1.1) which simulates LPG cylinder under different using and handling conditions,that is cylinder under internal pressure, cylinder under static and dynamic loading, and corrosion.

1.4 Literature Survey:

There are rather extensive literature on subject of investigation of pressurized vessel or cylinder using a finite element modeling.

Sing *et al* (1990) use finite element modeling with eight-node axisymmetric solid of revolution to simulate stresses in thick pressure vessel heads. Numerical values of circamferenctial and meridian stresses from that analysis shows excellent agreement with experimental dada from the literature.

Trinh *et al* (1989) study the nonlinear dynamic response of pressurized composite bottle subjected to shock loading, where a compared experimental and analytical program has been developed using finite element analysis to examine the structural dynamic response of pressurized composite bottle subject to spatially localized loading ,impulsive, exponentially decaying of such load where applied using plast tabe technique .

Also Radhamohan and Galletlu *et al* (1979) study the plastic collapse of thin internally pressurized Torispherical shells , they try to calculate the plastic collapse pressure of a range of torispherial shells, the influence of both geometry parameters (i.e., r/D , Rs/d and D/t and material properties (yield stress and strain-hardening coefficient) on the plastic collapse pressure were investigated. Both steel 207 GPa , and aluminum with E = 69 GPa shells where analyzed, a proximate design equation for calculating the plastic collapse pressure are suggested , The axisymmetric buckling pressure for torispherical shells (obtained form a companion paper) are also compared with the plastic collapse pressure , to which are the lower and thus control the mode of failure , a good agreement between theory and test was obtained .

On the other hand Bashnell et al (1977) study the other failure mode of torispherical dished end that is nonsymmetric Buckling of internally Pressurized ellipsoidal and torispherical elastic- plastic pressure vessel heads in which Bifuraction buckling are calculated with BOSOR5 computer program based on Rayleigh - Ritz method, where predicted behavior is found to be sensitive to pre-buckling geometric nonlineanity as well as material nonlinearily. The former effect increase the critical pressure and the latter decrease it. He found that in one case use flow theory leads to a predication that no buckling will occur whereas use the deformation theory leads to opposite conclusion . A change in the slope of the post yield hardening rate of the material dramatically affects the pre.buckling behavior.Also Bushel states in this paper that Buckling occurs only for very thin specimens for example with 2:1 heads non symmetric buckling occurs only if the diameter -to- thickness ratio greater than about 500 .

Soric and Zahlten *et al* (1995) make an Elastic- plastic analysis of internally pressurized torispherical shells ,where constitutive equation for elastic - plastic material model was derived using the VonMises yield critierion and assuming isotropic , strain hardening , Alagered finite element permitting geometrically linear and geometrically nonlinear elastic - plastic analysis of thin shell structures is presented, Also they study the effect of linear strain hardening on the size of plastic region and the distribution of internal forces in an internally pressurized, concluding that increase in the size of the plastic region was found to produce greater differences in the compilation of meridian bending moment than the compilation of hoop stresses resultants .

Also Part , Chi-Mo yim and Sang - Jeon *et al* (1993) make an ultimate strength analysis of ring - stiffened cylinders under hydrostatic pressure where nonlinear elastic- plastic large deformation, finite element analysis is developed to predict the collapse load of ring -stiffened circular cylinder subjected to compound axial compression and circumferential pressure loading by introducing anew type of axisymmetic shell element which can take into account the plastically effect , present approach is compared with the test results of previous work .

Another application of finite element analysis to pressure vessels during pressurization was carried by Tsybenko *et al* (1989) where a finite element molding with triangular simplex elements was used to examine the state of stresses and strain of thick- walled pressure vessels during pressurization , Result of this study show that stress intensity factor change with increasing pressure and distribution patterns of elastic - plastic in the meridian section of thick walled pressure vessel .

Chang , Kutlu and Zafer *et al* (1995) study the crushing repose of cylindrical composite shells subject to transverse loading , where investigation was performed to study the load- carrying capacity of fiber- reinforced composite shells subjected to quasi - static transverse crushing loads where both experimental and analytical work have been performed. an analytical model was developed which was based on progressive damage analysis , their analysis consists of a stress analysis, a failure analysis , deformation , stresses and strains in a composite cylinder where analysis based on a large deformation theory , damage accumulation and the residual stiffness and strength of tubes were evaluated, a nonlinear finite element program was developed

Chapter two

Problem Statement

2.1 General

LPG cylinders under conditions of usage are subjected to a high temperature which increase the internal pressure to dangerous level, Also cylinders are subjected to static loads during storage and transporting, Also impact loads during handling leave its effect on the cylinder strength, In addition LPG cylinders are subjected to atmospheric corrosion.

Jordanian Oil Refinery CO. is responsible for providing the local market with these cylinders ,also responsible for repairing and testing LPG cylinders, Cylinders are locally manufactured in the oil refinery or imported from outside ;in both cases all cylinders must pass the tests of Jordanian standard.

A finite element model shown in fig. (1.1) with nine nodded shell element is used to simulate the cylinder under three main loads, high pressure, static and dynamic loads. A FORTRAN computer program with fife input files uses finite element formulation mentioned in chapter two to calculate maximum principle stress and maximum deflection in cylinder body. To take elastic plastic behavior into account,moduls of elasticity taken as 170E+3 N/mm^2 if stress still in elastic range (below yield), after yielding a factor S was introduced with value of 0.1, where S = Slope of stress-strain in plastic range to the slope in the elastic range as shown in fig. 1.3.

2.2 Description of LPG cylinder

Transposable LPG cylinder is a steel cylinder with capacity of 26.2 liters which used to transpose the commercial LPG with maximum quantity 12.5 kg of gas ,which have a pressure of 0.6 N/mm^2 at temperature=50°c.

Capacity of LPG cylinder is the volume which defined by the amount at water which fills the interior of the cylinder at a temperature of 15.5°C, the cylinder must be designed in a manner to take up 26.2 liters but not to exceed 27.0 liters with allowable tolerance of ±0.25 liter of the adopted capacity. Also Jordan standard specifies the minimum requirement of strength and stability , and no leakage is must. (for more information about Jordan standard see appendix.)

Cylinder parts :

The cylinder consists of the following parts

1. *Body* : which is fabricated from two cold deep draw halves having an overlap joint assembled by submerged arc welding, material is constructed from killed steed plates prepared by one of the following processes:

(a) open hearth (b) basic oxygen (c) electric furnace.

2. *Foot ring:* To ensure stability when resting on a horizontal plane the cylinder is welded to a co-axial footing to the cylinder bottom which have vent holes at the top and along the circumference.

3. *Valve and valve boss*: A brass valve is fitted at the top of LPG cylinder with a thread at the inlet - outlet.

Valve is fitted through a valve boss which is a steel fabricated, machined and threaded to hit the valve. Valve boss is welded in its appropriate location by submerged are welding.

4. *Valve Guard:* The main requirement of valve guard is to protect valve during handling and transportation ,valve guard is a cylindrical shape with side openings it welds to the body of the cylinder.

Valve material is fabricated from a steel having the same properties as that of the body ,the thickness of valve guard is not as body thickness

Also a valve handle is formed as an integral part of valve guard or in a form of a round steel bar electrically welded and designed to ensure comfortable handling.

In addition, the cylinder is painted to protect it from corrosion or rust.

Heat Treatment of LPG cylinder:

cylinder in fabrication (before painting, after all welding operations completed) should be heat treated at temperature equal to 900 °C with four minutes holding time to remove residual stresses in the cylinder.

Tests performed on LPG cylinder:

Types of tests:

1. Mechanical tests.
2. Hydraustatic test.
3. Leak test.
4. Bursting test.
5. Dimension test.
6. Chemical test.

from the weld or the body. Also this test performed to the repaired cylinders.

4- Bursting test :

Bursting test should be carried out on specimens of manufactured or imported cylinders by exposing cylinders to a gradually increasing hydraulic pressure until the cylinders burst. No cracks should occur before a pressure of 45 kg/cm^2. Failure, when occurring should be at least 6 mm. away from the weld region.

5- Dimensions test:

i. Cylinder thickness test

Cylinder steel thickness should be measured in various position. The thickness should not be less than 3.0 mm. All dimension of the cylinder should comply with Jordan standard.

ii. Threard test :

Cylinder inlet valve internal thread should be measured by thread standard gauges especially for this purpose. Threads should comply with Jordan standard. All cylinders with threads that do not comply with Jordan standard should be rejected.

6- Chemical test:

Chemical analysis carried out on the steel in various parts of the cylinder and of the valve flange,Table below shows a chemical composition test for a cylinder tested in Royal Scientific Society.

C	S	Mn	Si	P
.150	.009	.75	.22	.02

*Royal Sientific Society, LPG Cylinder body chemical test, Ref. (3)148/55/1/20270

Chapter three

Theory

3.1 Membrane Stress Analysis Of Cylinder Shell Components:-

In structural analysis , all structures with shapes resembling curved plates, closed or open , are referred to as shells. Obviously, in pressure vessel design the shells are always closed. Most pressure vessels in industrial practice basically consist of few shapes: spherical or cylindrical with hemispherical, ellipsoidal , conical, toriconical , torispherical, or flat end closures. The shell components are welded together, forming a shell with a common rotational axis.

Generally, the shell elements used are axisymmetrical surfaces of revolution, formed by rotation of a plane curve or a sample straight line, called a meridian or generator, about an axis of rotation in the plane of the meridian . The plane is called meridian plane and contains the principal meridian radius of curvature .

In engineering strength of materials a shell is treated as thin if the wall thickness is quite small in comparison with the other two dimensions and the ratio of the wall thickness to the minimum principal radius of curvature is R_L / t > 20 or R_L / t > 20.This also means that the tensile , compressive , or shear stresses produced by the external loads in the shell wall can be assumed to be equally distributed over the wall thickness.

Further , most shells used in vessel construction are non shallow thin (membrane) shells in the range 1/500 < t / R < 1/20 , whose important characteristic is that bending stresses due to concentrated external loads are of high intensity only in closed proximity to the area where the loads are applied .The attenuation length or the decay length, the distance from the load where the stresses almost die out , is limited and quite short- e.g. , for a cylinder, $\sqrt{Rt}$.

The radial deformation DR of the shell under a load is assumed to be small .

Fortunately , most vessel problems occurring in practice can be solved with satisfactory results using a simplified approach . The main reason for this is that under certain loading conditions which occur in practice either shells of revolution ,some stress resultants are very small and can be disregarded or , because of axial symmetry , are equal to zero.

- *Membrane shell theory* solves shell problems where the internal stresses are due only to membrane stress resultants . The shear stress resultants for axisymmetrical loads such as internal pressure are equal to zero , which further simplifies the solution . The membrane stress resultants can be computed from basic static equilibrium equations

- *Bending shell theory,* in addition to membrane stresses, includes bending stress resultants and transverse shear forces . Here the number o unknowns exceeds the number of the static equilibrium conditions and additional differential equations have to be derived from the deformation relations .

The main conditions for a membrane analysis to be valid can be summarized s follows .

* All external loads must be applied in such a way that the internal stress reactions are produced in the plane of the shell only . Membrane stress analysis assumes the basic shell resistance forces are tension , compression , and shear in the shell plane and that a thin shell cannot respond with bending of transverse shear forces.

* The change in meridian curve is slow and without cusps or sharp bends. Otherwise bending and transverse shear stresses will be included at such gross geometrical discontinuities .

* The membrane stress resultants are assumed uniformly distributed a cross the wall thickness. This can be assumed if the ratio of the radius of curvature to the wall thickness is about $R/t<20$ and the change in the wall thickness if any is very gradual .

* The radial stress is small and can be neglected . A plane state of stress is assumed .

3.2 Membrane Stresses and Radial Deformation Produced by Internal Pressure

In this section analytical method for stress and radial deflection calculations are described, from which analytical results obtained.

It is important to mention that the most important case in the vessel design is a thin- shell surface of revolution , subjected to internal pressure of intensity P (internal pressure) is an axisymmetrical load; it is a uniform gas pressure . In the latter case usually two calculation are performed to find the stresses due to equivalent gas pressure .

Due to uniform pressure and axisymmetry there are no shear stresses on the boundaries of the element . The longitudinal and tangential stresses are the principal stresses and they remain constant cross the element.

Bedner *et al* 1981 states that the first equilibrium equation in the direction normal to the shell element is:

$$P[2R_t \sin(d\theta/2)][2R_L \sin(d\phi/2)] = 2\ \sigma\ ds_1 t \sin(d\theta/2) + 2\ _L ds_2 t\sin(d\phi/2)$$

substituting $\sin(d\,\theta/2) = (ds_2/2)/R_t$ and $\sin(d\,\phi/2) = (ds_t/2)/R_L$

$$P\,ds_1 ds_2 = (\sigma_t/R_t)\,t\,ds_1 ds_2 + (\sigma_{lt}/R_t)\,t\,ds_1 ds_2$$

or

$$P/t = (\sigma_t/R_t) + (\sigma_l/R_L)$$

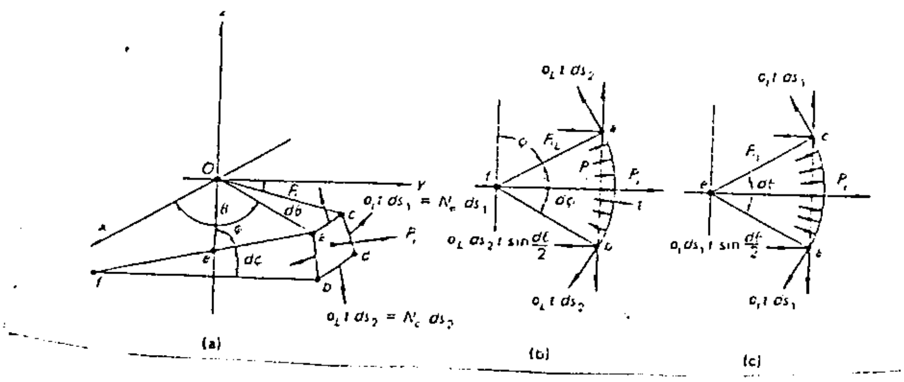


Fig. 1.5 Shell element of axisymmetrical shell subjected to internal pressure P, Henry Bednar,1981.

This last equation (the first equilibrium equation) is of fundamental importance for stress analysis of axisymmetrical membrane shells subjects to loads symmetrical with respect to the axis of rotation . The principal radius of curvature R can be positive (if it points toward the vessel axis), be negative (if it points away from the vessel axis) , or become infinite (at an inflection point). In the above analysis the radial stress was assumed to be negligible (average radial stress = p/2) .

The second equation of equilibrium required to solve for σ_t, and can be obtained by summation of the forces and stresses in the direction of the axis of rotation . Since the shell is axisymmetrical the entire finite shell section can be used at once as follows :

$$2\pi R(t\,\sigma \sin\phi) = P\pi R^2$$
$$.\sigma_l = PR/2t\sin\phi \quad = PR_t/2t$$

substituting this value in to the first equilibrium equation yields

$$\sigma t = (PR_t/t)[1-(R_t/2R_L)]$$

If $e_t = (1/E)(\sigma_t - \upsilon\,\sigma_L)$ the unit elongation in the tangential direction . the radial growth can be derived as follows .

$$2\pi(R/DR) = 2\pi R + 2\pi Re_t$$
$$DR = Re_t = (R/E)(\sigma_t - \upsilon\,\sigma_l)$$

The third equation required for static equilibrium of a biaxial state of stress is automatically satisfied, since load and the resultant stress in the tangential direction re defined as symmetrical with respect to the axis of rotation .

At this point it would seem important to point out that change in the radius of curvature will introduce in the shell bending stresses, that the membrane stress analysis assumes negligible .

The cylindrical shell is the most frequently used geometrical shape in pressure vessel design. It is developed by rotating a straight line parallel with the axis of rotation..The meridian radius of curvature R_L= infinity, and the second minimum radius of curvature is the radius of the formed cylinder $R_t = R$.

The stresses in a closed-end cylindrical shell under internal pressure P can be computed from the conditions of static equilibrium in the longitudinal direction.

$$2\pi R \sigma_L t = P \pi R^2 \quad \text{or} \quad \sigma_L = PR/2t$$

so that

$$(\sigma_L / \text{infinity}) + (\sigma_t / R) = P/t$$
$$\sigma_t = PR/t$$

The radial growth of the shell is

$$DR = R e_t = (R/E)(\sigma_t - \upsilon \sigma_L)$$
$$= (PR^2 / E t)(1 - \upsilon / 2)$$

There is no end rotation of a cylindrical shell under internal pressure.

The tangential (governing) stress can be expresses in terms of the inside radius Rt:

$$\sigma_t = PR/t = P(R_i + 0.5t)/t$$

3.3 IMPACT OR DYNAMIC LOADS :-

Forces suddenly applied to structures are turned shock or impact loads and result in dynamic loading.

A dynamic force acts to modify the static stress and strain fields as well the resistance properties of a material . Shock loading is usually produced by a sudden application of force or motion to a member , whereas impact loading results from the collision of bodies . When the time of application of a load is equal to or smaller than the largest natural period of vibration of the structural element, shock or impact loading is produced.

In this study we will try to develop a formula for dynamic load due to dropping cylinder from height h.

Although following a shock or impact loading , vibrations commence, our concern here will be only with the influence of impact forces on the maximum stress and deformation of the body,

The impact problem will be analyzed using the elementary theory together with the following assumptions :

1. The displacement is proportional to the applied forces , static and dynamic .
2. The inertia of a member subjected to impact loading may be neglected .
3. The material behaves elastic /. in addition, it is assumed that there is no energy loss associated with the local inelastic deformation occurring at the point of impact or at the supports. Energy is thus conserved within the system.

To idealize an elastic system subjected to an impact force , consider weight W . which falls through a distance h , striking the end of a free standing spring . As the velocity of the weight is zero initially and is again zero at the instant of maximum deflection of the spring (d_{max}) ,the change in kinetic energy of the system is zero , and likewise the work done on system. The total work consists of the work done by gravity on the mass as it falls and the resisting work done by the spring :

$$W(h+d_{max}) - \frac{1}{2}kd_{max}^2 = 0$$

K is known as the spring constant.

Note that the weight is assumed to remain in contact with the spring.

The deflection corresponding to a static force equal to the weight of the body is simply W/K .This is termed the static deflection ,(d_{st}).

Then the general expression of maximum dynamic deflection is

$$d_{max} = d_{st} + \sqrt{(d_{st})^2 + 2d_{st}h}$$

or ,by rearrangement,

$$d_{max} = d_{st}(1 + \sqrt{1 + \frac{2h}{d_{st}}})$$

The impact factor , the ratio of the maximum dynamic deflection to the static deflection ,is given by

$$\frac{d_{max}}{d_{st}} = 1 + \sqrt{1 + \frac{2h}{d_{st}}}$$

Multiplication of the impact factor by W yield an equivalent static or dynamic load

$$P_{dyn} = W(1 + \sqrt{1 + \frac{2h}{d_{st}}})$$

To compute the maximum stress and deflection resulting from impact loading , the preceding load may be used in the relationships.

3.4 Residual Stress and Deflection :-

Any structure will deform under load application, this deformation may be instantaneous, and the structure will turn back to its original shape after load removing, this will happen only if load affect the structure elastically; in other word every point in the structure is subjected to stress that does not exceed the yield stress (Stress less than 30. N/mm^2 in fig. 1.3)

Increasing load after that point will allow for presence of higher stresses that exceed the yield, so that point will enter the plastic range and will subject to plastic deformation which is permanent deformation.

In fig. 1.3, elastic deformation occur if load make stresses less than 30. N/mm^2, any point in the structure described in fig 1.3 subjected to stress equal to b will make a strain c during load application, and load removing will leave a permanent strain d such that line oa is parallel to line bd.

Note that for small deformation stress-strain diagram can be exchanged by force-deflection diagram.

3.5 Corrosion:-

Corrosion is defined as a destruction process of a metal by a chemical and/or electrochemical reaction with its environment a familiar example is the rusting of iron.

In LPG cylinders the most common types of corrosion that cylinders are subjected to are:

1. Atmospheric corrosion.
2. LPG corrosion.

The first type of corrosion is present mainly on the outer surface of cylinder body which depend upon :

i. the length of time that moisture is in contact with the surface.

ii. the extent of pollution of the atmosphere.

iii. the chemical composition of the gas cylinder material.

The table below shows the loss in weight (gm) and the corrosponding reduction in thickness (mm) for a low carbon steel specimen (152 x 102 x 3) mm which exposed to industrial atmosphere for a period of time (year) , see fig. 6.1.

Time [year]	loss of weight [gram]	thickness [mm]
0	0	3.000
1	12	2.900
2	15	2.876
3	16	2.867
4	17	2.859
5	18	2.851
6	19	2.842
7	20	2.834
8	21	2.826

(Herbert, 1948)

The second type of corrosion referred to chemical reaction between LPG and cylinder material which is low carbon steel, this reaction depends mainly upon presence of sulfur in LPG, this type of corrosion is not considerable because its magnitude is small compared with the first type.

3.6 FINITE ELEMENT FORMULATION:

3.6.1 NINE NODED SHELL ELEMENT

The nine noded curved shell element based on the assumptions that normals to the modsurfae remain straight after deformation and the normal stress is zero . This shell element is based on the degeneration concept and numerical integration technique is adopted for evaluation of the integral across the thickness. The basic expressions to define the shell characteristics and necessary equation leading to the formulation of the stiffness matrix and element load vector are explained in this section .

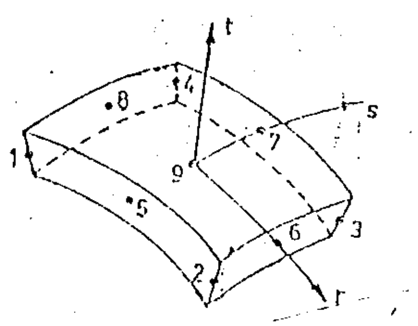


Fig 1.6 local coordinate of nine noded shell element,
Jahnw Bull 1988.

3.6.2 Shape Function for Nine Node Shell Element

A typical curved shell element , with eight nodes on the mid-surface and an additional node at the center of the midsurface along with the global coordinate system (x,y,z) . The shape functions treating it as a two dimensional isoparametric element in (r,s) coordinates , are given below .

for nodes number 1, 2, 3 and 4

$$N_i = \frac{1}{4}[(1+rr_i)(1+ss_i)(rr_i+ss_i-1)]$$

also for nodes number 5 and 7

$$Ni = (1/2)(1-r^2)(1+ss_i)r_i$$

and nodes 6 and 8

$$N_i = \frac{1}{2}(1-s^2)(1+rr_i)$$

finally, node 9

$$N_i = (1-r^2)(1-s^2)$$

where r_i and s_i are the coordinates r and s of the node i of the mid-surface. It may be noted that shape functions for all the nodes expect the central node are the same as those used for eight noded plates.

Assuming the lines joining the top and bottom nodes to be straight, the shape of the element is defined by the eight nodal values , as ,

$$\begin{Bmatrix} x \\ y \\ z \end{Bmatrix} = \sum_{i=1}^{8} N_i \left\{ \begin{Bmatrix} x_i \\ y_i \\ z_i \end{Bmatrix} + t \begin{Bmatrix} x_i^* \\ y_i^* \\ z_i^* \end{Bmatrix} \right\}$$

where x^*_i, y^*_i, z^*_i are the global coordinates of the midsurface node i which is computed by taking the average of the top and bottom node coordinates and x^*_i, y^*_i, z^*_i are the global coordinates of the point (r_i ,s_i, 1) with respect to nodes (r_i ,s_i, 1) and obtained by dividing the differences between the top and bottom node coordinates by two.

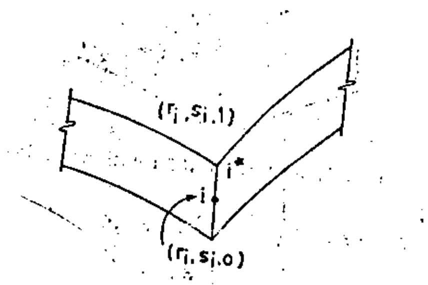


Fig. 1.7 Local coordinate of top and mid surface, Jahnw Bull 1988.

3.6.3 Displacement Field :

Analytical investigations have been reported that adding an internal node improved the performance of the element . Thus , using the shape functions. the displacement field within the element is given by:

$$\begin{Bmatrix} u \\ v \\ w \end{Bmatrix} = \sum_{i=1}^{8} N_i \left\{ \begin{Bmatrix} u_i \\ v_i \\ w_i \end{Bmatrix} + t \begin{Bmatrix} u_i^* \\ v_i^* \\ w_i^* \end{Bmatrix} \right\}$$

where

u_i, v_i, w_i : are the displacements of the node i in the x,y,z direction respectively , and

u_i^*, v_i^*, w_i^* : are the relative global displacements at point i caused by the rotations of the normal . As N_9 vanishes at r, s=+1, inter element compatibility is unaffected .

Now the relative displacements u_i^*, v_i^*, w_i^* should be expressed in terms of the rotations *p* and *q* at the node i, in order to specify the displacements in terms of the nodal displacements and rotations .

Let *alpha* and *beta* be defined as the rotations of the normal about the axes a^ and b^ which lie in the midsuface . The two vectors a^ and b^ are obtained as follows .

Let i^ , j^ , k^ be the unit vector in the x,y and z directions respectively. Vector n^ is defined by input data so that it is normal to the mid-surface ,i.e., the top and bottom nodes ,whose coordinates are given as input should lie on the normal to the mid-surface . Now x*, y*,z* are the coordinates of a vector along n^ or.

$$\bar{n} = x^* \bar{i} + y^* \bar{j} + z^* \bar{k}$$

Now a^ is defined as a vector perpendicular to n^ and also to vertical k^

$$\begin{aligned} \bar{a} &= \bar{k} \times \bar{n} \\ &= \bar{k} \times (x^* \bar{i} + y^* \bar{j} + z^* \bar{k}) \\ &= -y^* \bar{i} + x^* \bar{j} \end{aligned}$$

We now define b^ as a vector perpendicular to both n^ and a^.

$$\bar{b} = \bar{n} x \bar{a}$$

$$= (x^* \bar{i} + y^* \bar{j} + z^* \bar{k}) x (-y^* \bar{i} + x^* \bar{j})$$

$$= -x^* z^* \bar{i} - x^* z^* \bar{j} + (y^* y^* + x^*) \bar{k}$$

It is seen that when k^ and n^ coincides, vectors a^ and b^ are not specific and in that case they are defined as

$\bar{a} = \bar{j}$ **and** $\bar{b} = \bar{i}$

Now by normalizing the vector a^ and b^ we can get the direction cosines of these vectors with respect to x,y and z directions.

Let a1 ,a2 and a3 be the direction consines of b^ with respect to x ,y and z axis respectively, and b1,b2 and b3 be the direction consines of(b) with respect to x,y and z axis respectively .

as α and β are the rotation with respect to a^ and b^ respectively ,the displacement at

$$t = 1 \quad \text{is} \quad \frac{h}{2} \; (-\bar{b}\alpha + \bar{a}\beta)$$

where h is the thickness at point .

Hence ,the relative displacement at i in the global directions are given by

$$[u_i^* \;\; v_i^* \;\; w_i^*] = -\frac{1}{2} h_i \beta_i [b_1 \;\; b_2 \;\; b_3]_i + \frac{1}{2} h_i \alpha_i [a_1 \;\; a_2 \;\; a_3]_i$$

Substituting in the above equation we get

$$\begin{Bmatrix} u \\ v \\ w \end{Bmatrix} = \sum_{i=1}^{9} N_i \left\{ \begin{Bmatrix} u_i \\ v_i \\ w_i \end{Bmatrix} - \frac{t h_i \alpha_i}{2} \begin{Bmatrix} b_1 \\ b_2 \\ b_3 \end{Bmatrix} + \frac{t h_i \beta_i}{2} \begin{Bmatrix} a_1 \\ a_2 \\ a_3 \end{Bmatrix} \right\}$$

3.4.4 Jacobian Matrix:-

Noting that the shape functions are functions of r and s alone ,the Jacobian matrix may be computed as shown below.

$$[j] = \begin{vmatrix} \sum_{i=1}^{8}(x_i + tx_i^*) * \frac{\partial N_i}{\partial_r} & \sum_{i=1}^{8}(y_i + ty_i^*)\frac{\partial N_i}{\partial_r} & \sum_{i=1}^{8}(z_i + tz_i^*)\frac{\partial N_i}{\partial_r} \\ \sum_{i=1}^{8}(x_i + tx_i^*)\frac{\partial N_i}{\partial_r} & \sum_{i=1}^{8}(y_i + ty_i^*)\frac{\partial N_i}{\partial_r} & \sum_{i=1}^{8}(z_i + tz_i^*)\frac{\partial N_i}{\partial_r} \\ \sum_{i=1}^{8} N_i x_i^* & \sum_{i=1}^{8} N_i y_i^* & \sum_{i=1}^{8} N_i z_i^* \end{vmatrix}$$

By inverting the matrix [J] we can get the derivatives with respect to global coordinates.

3.6.5 Strain Displacement Matrix.:-

The element strains and the nodal displacement are related as.

$$\{e\} = [B]\{d\}$$

where $\{d\}^T = [u_1\ v_1\ w_1\ |\alpha_1 \beta_1\ u_2\ v_2\ w_2\ \alpha_2\ \beta_2| \ldots |u_9\ v_9\ w_9\ \alpha_9\ \beta_9]$

In order to establish the strain displacement with relation we have to first obtain the derivatives of the displacements with respect to local coordinates and they are obtained by differentiating the equation . with respect to r,s and t. Thus,

$$\begin{vmatrix} \frac{\partial u}{\partial r} & \frac{\partial v}{\partial r} & \frac{\partial w}{\partial r} \\ \frac{\partial u}{\partial s} & \frac{\partial v}{\partial s} & \frac{\partial w}{\partial s} \\ \frac{\partial u}{\partial t} & \frac{\partial v}{\partial t} & \frac{\partial w}{\partial t} \end{vmatrix} = \sum_{i=1}^{9} \begin{vmatrix} \frac{\partial N_i}{\partial r} \\ \frac{\partial N_i}{\partial s} \\ 0 \end{vmatrix} [u_i v_i w_i] - \frac{1}{2} \sum_{i=1}^{9} \begin{vmatrix} th_i \frac{\partial N_i}{\partial r} \\ h_i \frac{\partial N_i}{\partial s} \\ h_i N_i \end{vmatrix} \alpha_i [b_i b_2 b_3]_i$$

$$+\frac{1}{2} \sum_{i=1}^{9} \begin{vmatrix} t\,h_i \frac{\partial N_i}{\partial r} \\ t\,h_i \frac{\partial N_i}{\partial s} \\ h_i N_i \end{vmatrix} \beta_i [a_1\ a_2\ a_3]_i$$

Now using the Jacobian inverse, we can find the derivatives with respect to global coordinates and obtain strain displacement matrix[B].

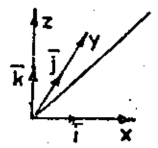
3.6.6 Strain Transformation Matrix:-

As explained earlier, the stresses normal to the mid surface is negligible and is taken as zero .Since the expressions for stresses obtained are in the global directions, we must rotate the axes from x,y,z system to a set x^ ,y^ z^ as shown in figure below .This set of mutually perpendicular axes at the point being considered ,is determined from the local axes r,,s,t as follows.

x' is identical to r
z' is identical to r and s (r,s,z' forming a right handed set)
y' is prependicular to x' and z' (x',y', z' forming a right handed set)

Let (e^1,e^2,e^3) be unit vectors along r,s and t directions respectively and (i^, j^, k^) be unit vectors along x,y and z directions. These vectors are related by the Jacobian matrix [J] as given below .

$$\begin{Bmatrix} \bar{e}_1 \\ \bar{e}_2 \\ \bar{e}_3 \end{Bmatrix} = [J] \begin{Bmatrix} \bar{i} \\ \bar{j} \\ \bar{k} \end{Bmatrix}$$



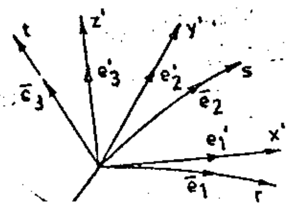


Fig 1.8 Global and local axis,
Jahnw Bull 1988.

We have to define three vectors e^1, e^2, e3^ along x^,y^ and z^ respectively. From the definition of x^ ,y^ , z^ system ,we have

e'1 = e^1

e'3 : perpendicular to e^1 and e^2

e'2 : perpendicular to e^1 and e^3

where

$$\begin{Bmatrix} \bar{e}_1 \\ \bar{e}_2 \\ \bar{e}_3 \end{Bmatrix} = \begin{vmatrix} J_{11} & J_{12} & J_{13} \\ J_{21} & J_{22} & J_{23} \\ J_{31} & J_{32} & J_{33} \end{vmatrix} \begin{Bmatrix} \bar{i} \\ \bar{j} \\ \bar{k} \end{Bmatrix}$$

Now,

$$e_1' = \bar{e}_1 = J_{11}\bar{i} + J_{12}\bar{J} + J_{13}\bar{k} = J_{11}'\bar{i} + J_{12}'\bar{j} + J'_{13}k$$

$$e_3' = \bar{e}_1 * \bar{e}_2 = (J_{12}J_{23} - J_{13}J_{22})\bar{i} + (J_{13}J_{23} - J_{11}J_{22})\bar{J} + (J_{11}J_{22} - J_{12}J_{21})\bar{k}$$
$$= J'_{31}\ \bar{i} + J'_{32}\,j + J'_{33}\,k$$

$$e_2' = \bar{e}_3 * \bar{e}_1 = (J'_{32}J'_{13} - J'_{313}J'_{12})\bar{i} + (J'_{11}J'_{33} - J'_{13}J'_{31})\bar{J} + (J'_{31}J'_{12} - J'_{32}J'_{11})\bar{k}$$
$$= J'_{21}\ \bar{i} + J'_{22}\,j + J'_{23}\,k$$

Thus we have

$$\begin{Bmatrix} \bar{e}_1 \\ \bar{e}_2 \\ \bar{e}_3 \end{Bmatrix} = \begin{vmatrix} J'_{11} & J'_{12} & J'_{13} \\ J'_{21} & J'_{22} & J'_{23} \\ J'_{31} & J'_{32} & J'_{33} \end{vmatrix} \begin{Bmatrix} \bar{i} \\ \bar{j} \\ \bar{k} \end{Bmatrix}$$

If we normalize the vectors [j'_{11} j_{12} j'_{13}], [j'_{21} j'_{22} j'_{23}] , and [j'_{31} j_{32} j'_{33}] we get the [D] matrix representing direction cosines of the new axes direction e'_1,e'_2 and e'_3 .

The strain $\{\varepsilon'\}$ with respect to x^,y^,z^ system is related to $\{\varepsilon'\}$ in the global x,y,z system by the matrix $[T\varepsilon]$

$$\{\varepsilon'\} = [T]\{\varepsilon\}$$

also
$$\{\varepsilon'\} = [T][B]\{d\}$$
$$= [B']\{d\}$$
$$[B'] = [T][B]$$

3.6.7 Stress Displacement matrics:-

Stress are related to displacement as

$$\{\sigma\} = [C][B]\{d\}$$

$$= [\,C\,B\,]\{d\}$$

Considering the case of isotropic material, and imposing the vondition that ($\sigma_z = 0$) the following relation is obtained for stress - strain relation in x^,y^,z^ coordinates,

$$\{\sigma'\} = [\,C\,]\{\,\varepsilon'\,\}$$

or

$$\begin{Bmatrix} \sigma_{x'} \\ \sigma_{Y'} \\ \tau_{x'y'} \\ \tau_{y'z'} \\ \tau_{z'x'} \end{Bmatrix} = \frac{E}{1-\eta} * \begin{bmatrix} 1 & \eta & 0 & 0 & 0 \\ \eta & 1 & 0 & 0 & 0 \\ 0 & 0 & \frac{1-\eta}{2} & 0 & 0 \\ 0 & 0 & 0 & \frac{1-\eta}{2}\alpha & 0 \\ 0 & 0 & 0 & 0 & \frac{1-\eta}{2}\alpha \end{bmatrix}$$

Where α is a factor used to account for a better representation of shear deformation when a constant strain is assumed across the thickness, rather than the correct quadratic and a value of $\alpha = 5/6$ has been taken.

Substituting for $\{\varepsilon\}'=[B'\{d\}$ we get

$$\{\sigma'\} = [\,C\,][\,B'\,]\{d\}$$

$$= [\,C\,B'\,]\{d\}$$

The stress thus obtained will be in the local coordinated system x^,y^,z^ which is convenient in the case of curved surfaces.

3.6.8 Element Stiffness Matrix

As we have obtained the [B]matrix and [CB] matrix in local coordinates x^,y^,z^ coordinates we should use the same matrices for the computation of element stiffness matrix. Here the transformation is derived based on the concept that during any virtual displacement ,the resulting increment in strain energy density must be the same regardless of the coordinate system in which it is computed. Hence the matrix is given by

$$[k] = \begin{array}{l} = \iiint [B]^T [C][B] dV \\ = \iiint [B'']^T [C][B''] dV \\ = \iiint [B'']^T [CB''] dV \end{array}$$

Where [B'] and [CB'] matrices are computed as explained in the previous section.

The integration is carried out numerically resulting in an element stiffness matrix of size 45×45 .

The integration scheme ,i.e. ,choice of Gauss points to calculate the stiffness matrix requires careful study .

It has been noted that a 3×3×2 integration gives good results for thick shells, whereas a reduced integration scheme of 2×2×2 is economical and reasonably accreted for thin shells.

chapter four

results and discussion

4.1 Cylinder Under Internal Pressure

LPG Cylinder is investigated here under low and high internal pressure

1- Cylinder under low internal pressure:-

Pressure varying between 0.1 N/mm² to 1.0 N/mm² (1.0 to 10 bar) where a finite element modeling was used to study LPG cylinder under low pressure, and compared with results of pressurized thin shell surface of revulsion (analytical results). Finite element simulation shows that cylinder material behaves completely elastic, fig(2.1) shows maximum principal stress variation with internal pressure for cylinder under low internal pressure, a finite element results for maximum stress and exact result, shows linear relation so depending on finite element results we can say

max. principal stress = 60.8 x P (within elastic range)

2. Cylinder under high internal pressure:-

With pressure variation between 1.0 N/mm^2 to 10 N/mm^2 (10 to 100 bar), a finite element modeling was used to study LPG cylinder under high internal pressure. Results was compared with oil Refinery experimental data; where deflection of the cylinder is predicted for several locations in the cylinder during bursting test.

Experimentally the cylinder under bursting test shows that plastic deformation begin at pressure = 50 bar. This is clear in fig 2.3, where we see how deflection vary with pressure, in which *plastic collapse* begin at pressure = 50 bar.

On the other hand, fig 2.2 shows the finite element results, where maximum principal stress in pressurized cylinder versus heigh pressure was obtained, in which plastic collapse can be predicted. The curve shows how an elastic behavior is obtained for a cylinder under internal pressure less than 50.bar, this is clearly noted from the linear relation between maximum stress and pressure; where more increasing in pressure result in a heigher stress that exeed the yield, after which the material begin to behave a plastic behavior, this phenomena is called *plastic collapse of internally pressurized cylinder*. On the other hand fig. (2.3) and fig. (2.4) shows how pressure vary with maximum deflection based on finite element results. the curve is coincide with stress-pressure curve and both are coincide with stress strain curve.

Fig. (2.2) shows that ultimate stress happen after P=90. bar and that is coincide with the experimental bursting test at which bursting occur at P=100 bar.

This is summary of plastic collapse failure mode, The other failure mode which is axisymmetric deformation or buckling is not occur neither in experiment nor in finite element results i.e. no buckling occure in LPG cylinder even at high pressure, that is mainly due to that R/t is so small (R / t < 500).

4.2 Cylinder under static loading

LPG Cylinder under static loading is also studied using the same finite element modeling with a line distributed force along cylinder body with range of 2000 N to 10 000 N , fig. (3.1) shows maximum principal stress with static force for empty and full cylinder, also fig. (3.2) shows maximum radial deflection with static force for empty and full cylinder, empty cylinder have less deflection than full cylinder, that is mainly due to presence of internal pressure inside the full cylinder which givesan opposite radial deflection.

4.3 CYLINDER UNDER IMPACT LOADING:-

Impact loading of cylinder is mainly due dropping cylinder from hights into edge point, a different cases is studied here. First impact factor has been calculated from d_{st} (See theory section 3.3).Fig(4.1)and fig(4.2)shows how impact factor vary with height of dropping for three locations in the cylinder, increasing height of dropping gives higher impact factor, also note that for upper and lower location the impact factor is so closed and less than that of mid point which represent impact on weld area; this area is double in thickness which give more strength to that area, so lower d_{st}, causing higher impact factor.

Dynamic loading ($P_{dynamic}$) also calculated, fig(4.4) and fig (4.5) shows dynamic forces for full cylinder and empty one, for the three locations of impact ; higher dynamic force is obtained for full cylinder despite of lower impact factor, that is mainly due to weight of full cylinder which is more than empty cylinder.

For upper and lower impact location, dynamic force is higher in value than mid-side location (weld area), that is due to higher impact factor mentioned above, also dynamic force for both locations are so closed, that is mainly due to aproximate symmetry about the weld circle. Higher dynamic force and impact factor on the upper part over the lower part is mainly due presence of base and guard of the cylinder.

Fig (4.7) and Fig(4.8) shows maximum radial deflection occur in cylinder(for full and empty cylinder). Results shows a weak areas in upper and lower location comparing with weld region that is mainly to doubling in thickness on the weld area mentioned previously, it is worth to mention here that for a cylinder under impact, maximum principle stress occur on the point of load application only for upper and lower impact, but for impact cylinder on weld region, maximum principle stress will occur away from point of force application, maximum principle stress will not occur (in any case) in weld region.

4.4 Residual deflection and stresses :

cylinder subjected to load P will deform under effect of loading, combining the dynamic load fig. 4.3 & fig. 4.4 and radial deflection fig. 4.7 & fig. 4.8 ; Deflection in cylinder is presented in fig. 5.* , which shows radial deflection of cylinder d versus dynamic load P.

Now removing load P will leave a residual deflection d_r, one can determine d_r by drawing a line in fig. 5.* from the curve to deflection axis parallel to elastic line as seen in fig. 5.1.

after determining d_r one can find the maximum principal residual stress by determine the correspond dropping high from fig. 4.7 and fig. 4.8 then find the correspond stress from fig. 4.5 and fig. 4.6 which is the maximum residual stress, this is shown in this example.

Example :

A LPG cylinder full with gas with pressure 0.6 N/mm^2 is dropping from hight equal to 1.5 m to an edge, find the permanent deflection and the corresponding residual stress.

a. First we must determine the dynamic force resulting from dropping the cylinder from height = 1.5 m, From fig. 4.3 , for height equal to 1.5 m we have a dynamic load P = 28.0 KN.

b. Now, From fig. 5.1 for load = 28.0 KN, radial deflection d= 2.2 mm, and permanent radial deflection (from the parallel line) d_r= 1.2 mm.

c. from fig. 4.7, for radial deflection equal to 1.2 mm, we have height = 1.0 m

d. from fig. 4.5, the residual stress = 360 N/mm^2

4.5 Effect of corrosion:

The main effect of atmospheric corrosion on LPG cylinder is reduction in thickness of cylinder body, this reduction will weaken the structure of cylinder body causing an increase in stresses the cylinder subjected to.

Fig. 6.1 shows reduction in thickness with time (in years) for a 3.0 mm plate subjected to atmospheric corrosion (see section 3.5), noting quick reduction in thickness in the first year.

For a cylinder subjected to an internal pressure equal to 0.6 N/mm², finite element results shows that the maximum principle stress equal to 36.17 N/mm², The same cylinder next year (assuming atmospherical corrosion) will have a thickness equal to 2.9 mm in stead of 3.0 mm, So applying finite element analysis again for the new thickness (2.9 mm) shows an increasing from 36.17 [N/mm²] to 37.41 N/mm² in the first year, Performing the same calculations for the next 20 years to obtain fig 6.2., From which we see the rate of increasing in stress due to corrosion, this rate increases sharply in the first year, before it comes to be constant rate after ten years, that is :

max. principal stress [N/mm²] = *38.74 + 0.14 x time [year]*

where time > 10 years.

Chapter five

conclusions and Recommendations

5.1 Conclusions

Several important points have emerged from this work,which can be summarized as follow :

* For low pressure (up to 50. bar) maximum stress occur in LPG cylinder can be calculated from the relation

$$\text{maximum stress} = 60.8 \times P$$

* A plastic deformation in LPG cylinder occur after pressure equal to 50. bar, which indicate that a cylinder with a plastic collapse have been subjected to pressure greater than 50 bar.

* For a cylinder free from residual stresses falling on an edge point, the dangerous height -after which fracture occure- is greater than 2.5 m for full cylinder and 3.0 m for empty one.

* Any failure due to pressure or impact, if occur, will never be in the weld rejoin, but will transfer to cylinder body.

* Predict a method to calculate the permanent radial deflection and residual stresses in cylinder.

* Calculating the effect of atmospheric corrosion in increasing cylinder stresses.

5.2 Recommendations

the present work suggests that the following points need to be investigated:

* Residual stresses in LPG cylinder under successive impact loading.

* Studying the temperature effect on LPG pressure, and take account this study to investigate temp effect on stresses affecting the cylinder.

* Examine the effect of long term service periods on the mechanical and chemical properties of cylinder material

* Using finite element analysis to study the effect of thickness on factor of safety and effect of corrosion which reduces structural thickness.

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APPENDICIES

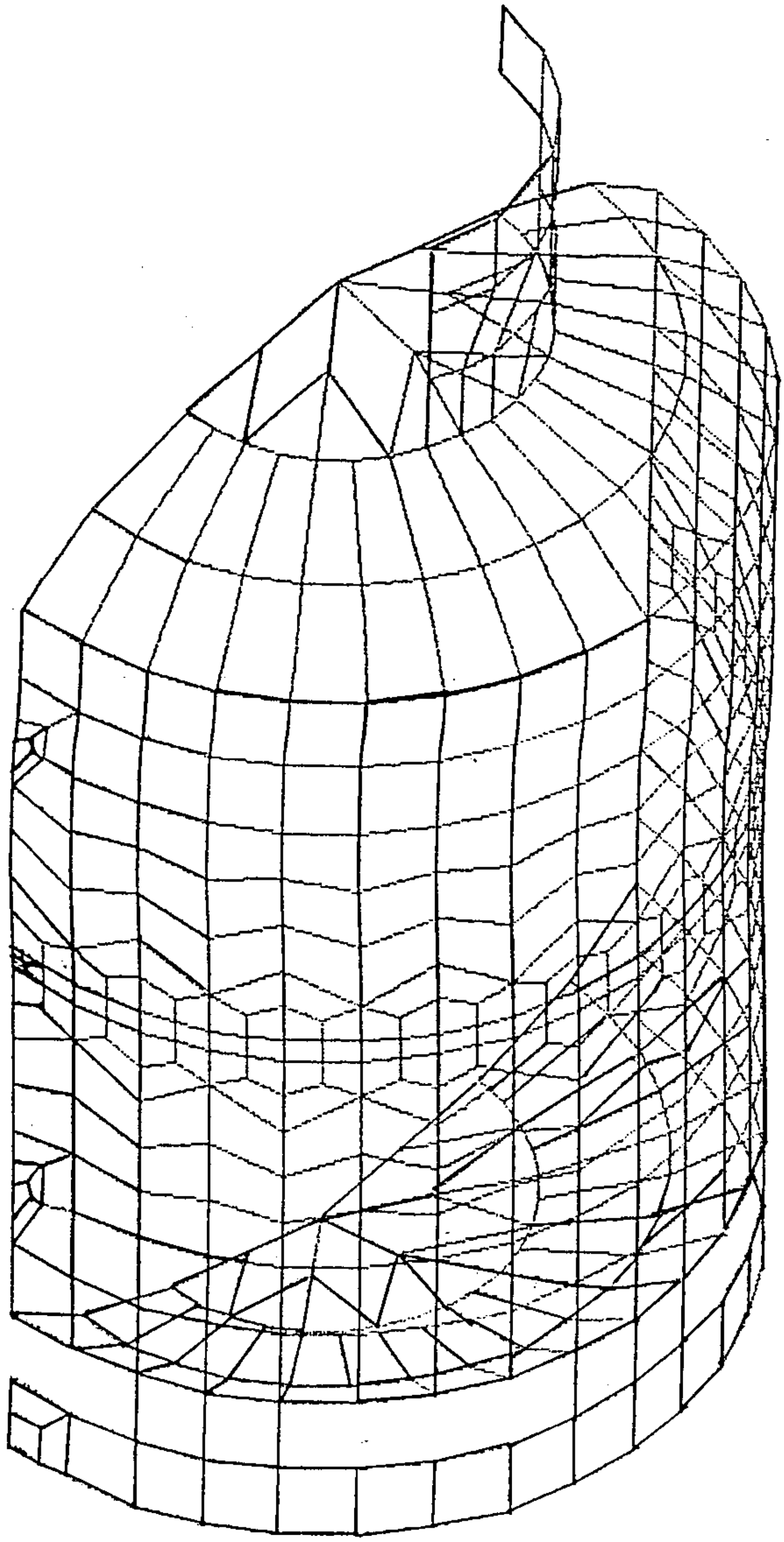


Figure (1.1) LPG Cylinder descitizing into 9 noded finite shell elements

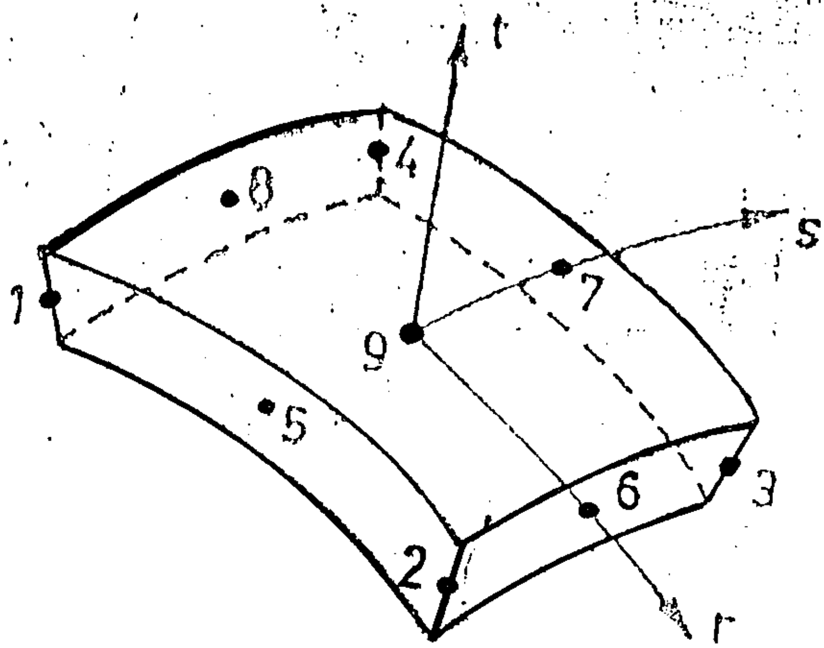
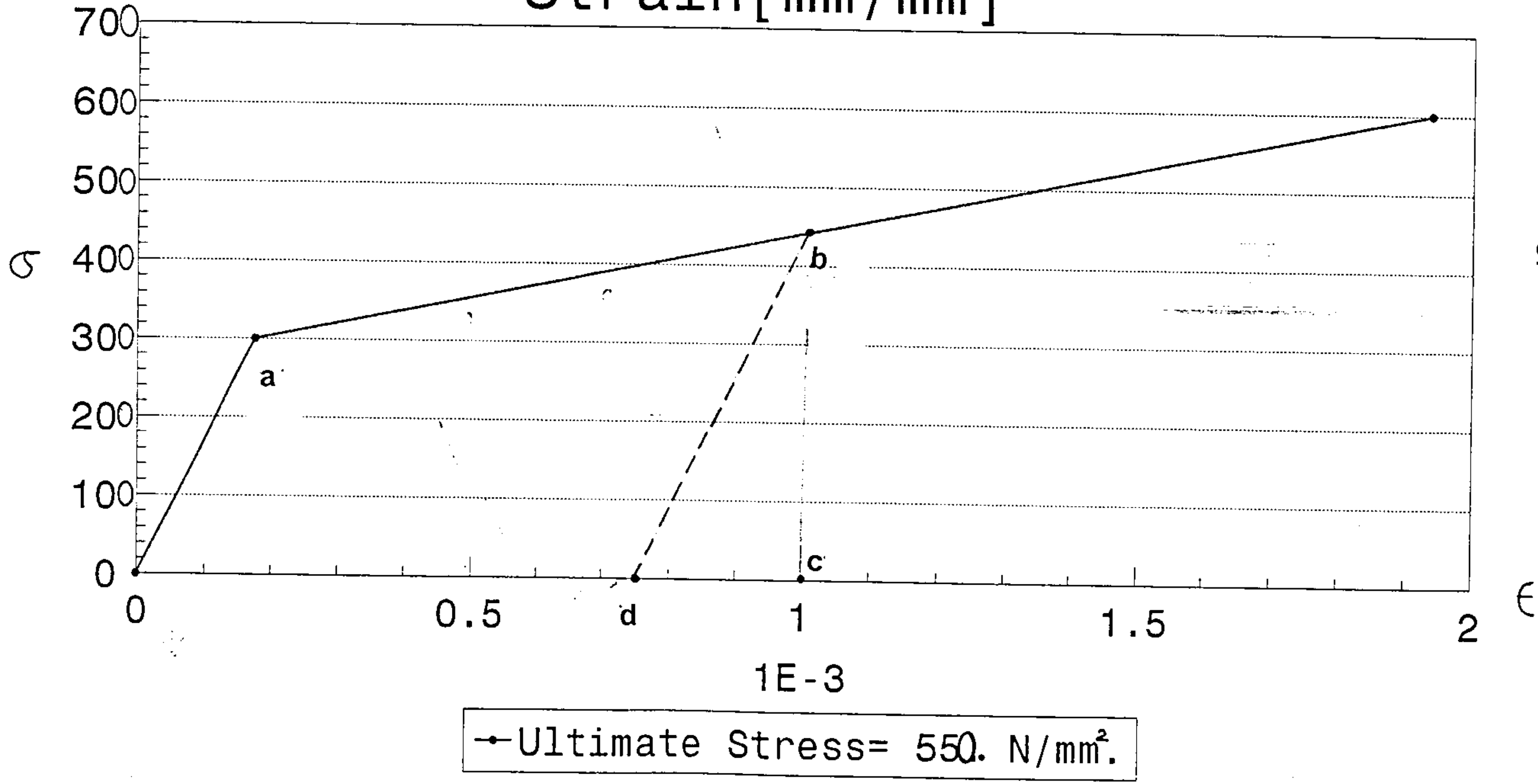


Figure (1.2) Nine noded element

Fig.(1.3) Stress[N/mm^2]verus Strain[mm/mm]



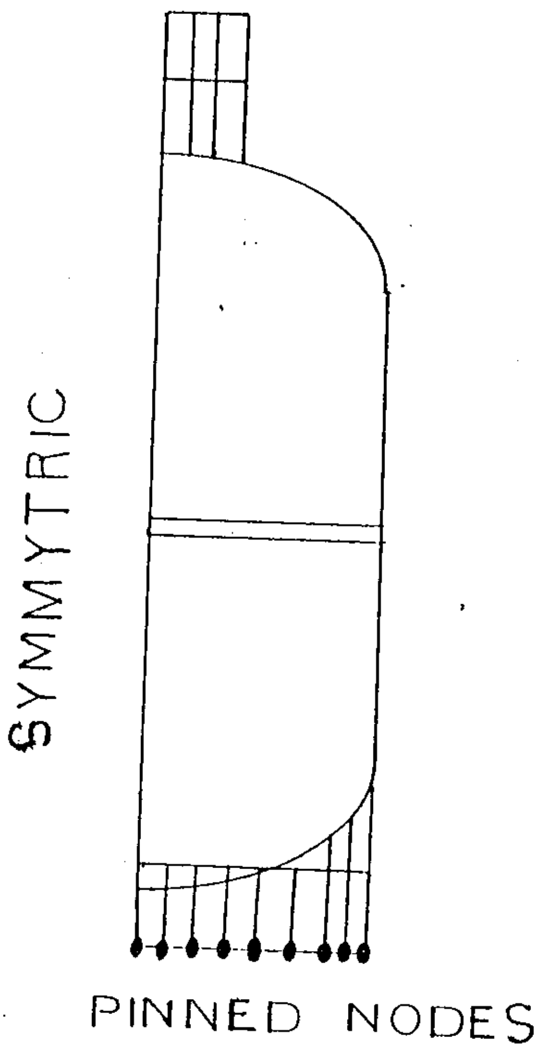


Figure **(1.4)** Boundary conditions

Fig. (2.1)Max. principle Stress [N/mm^2] verus cylinder internal Pressure[N/mm^2]

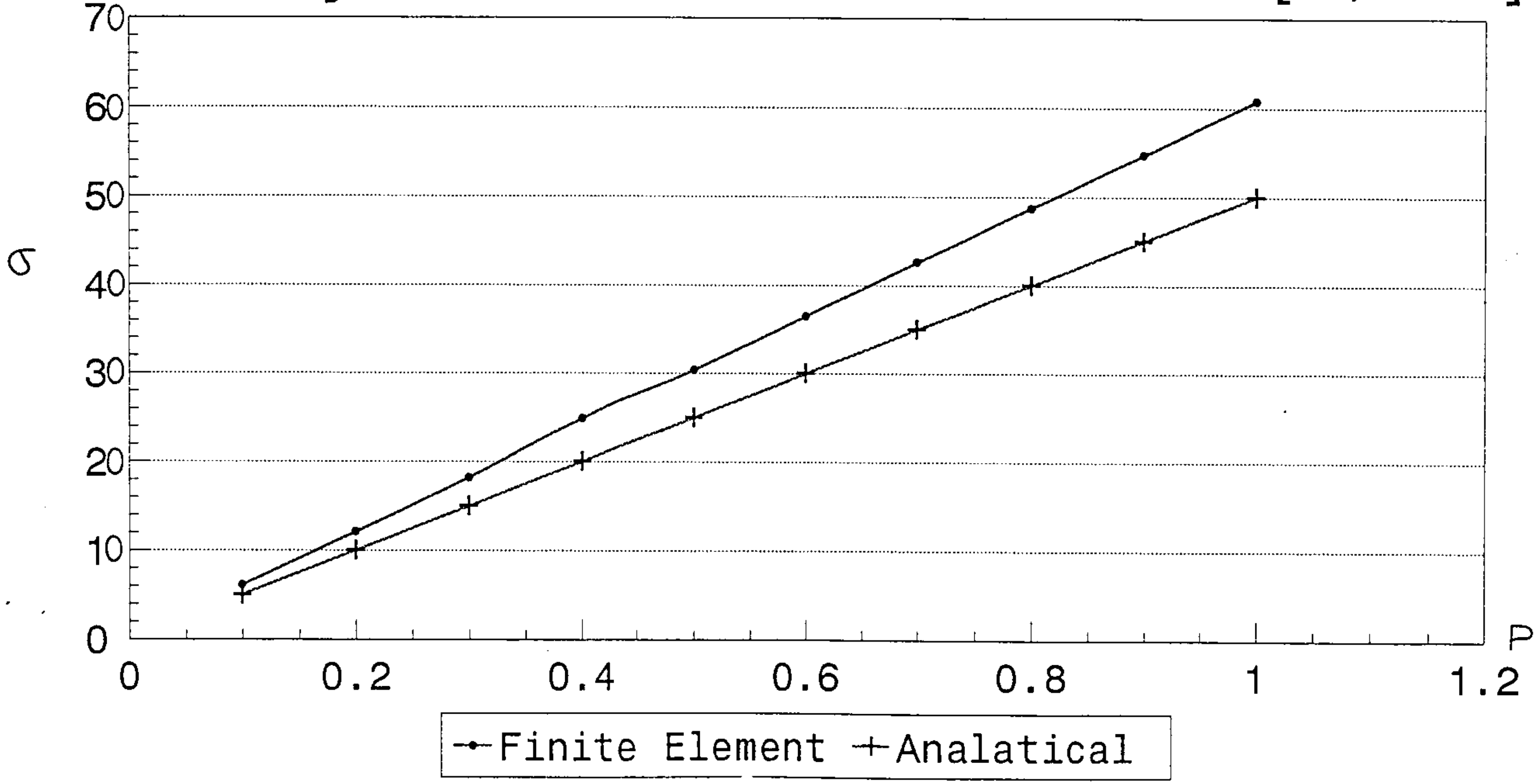


Fig. (2.2) Max. Principal Stress[N/mm^2] verus Cylinder Internal Pressure[N/mm^2]

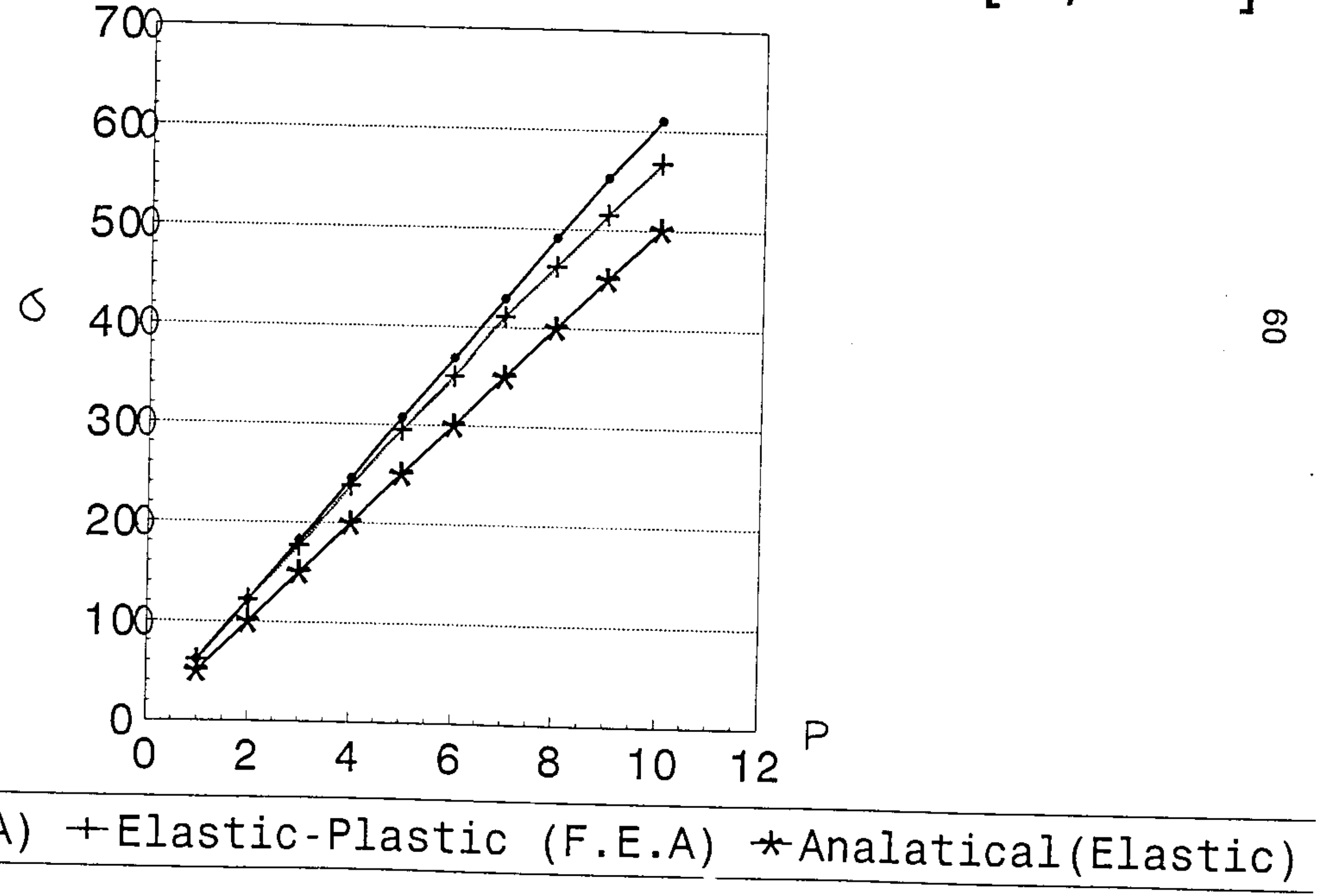


Fig. (2.3) Radial Deflection[mm] verus Internal Pressure[N/mm^2],Finite Element

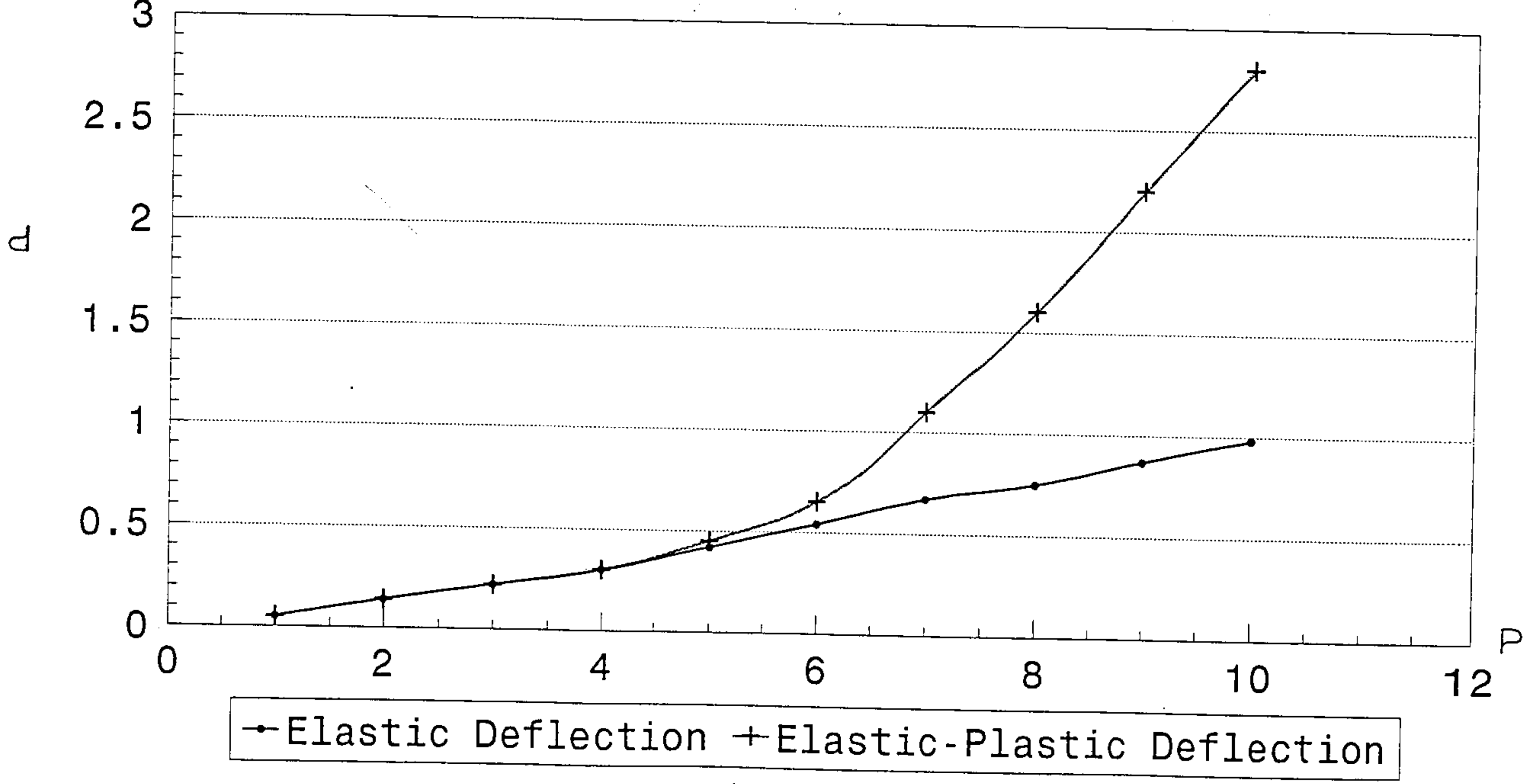


Fig. (2.4) Radial Deflection[mm] verus Cylinder Internal Pressure[N/mm^2]

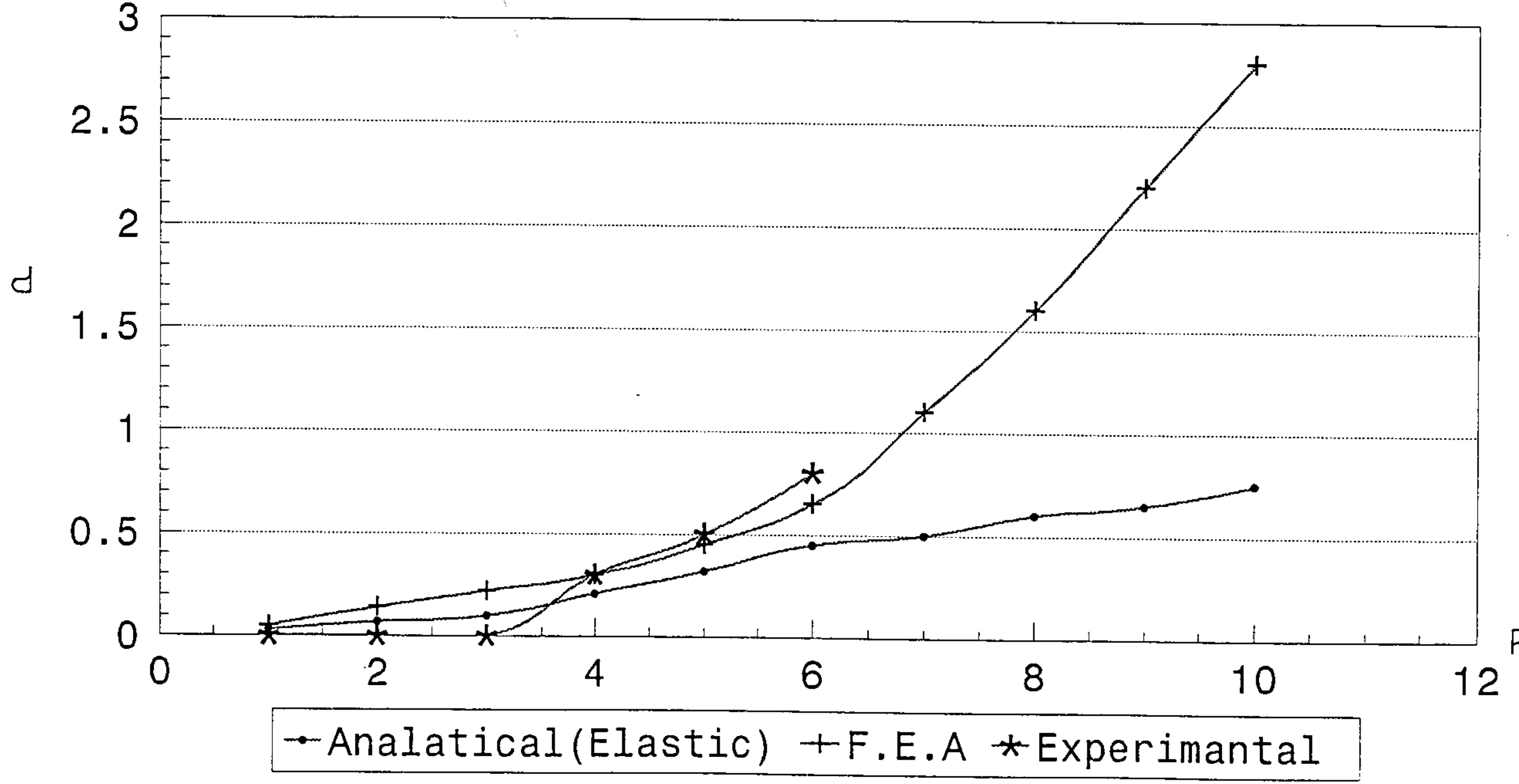


Fig. (3.1) Max. Principle Stress[N/mm^2] verus Static force Along cylinder[N]

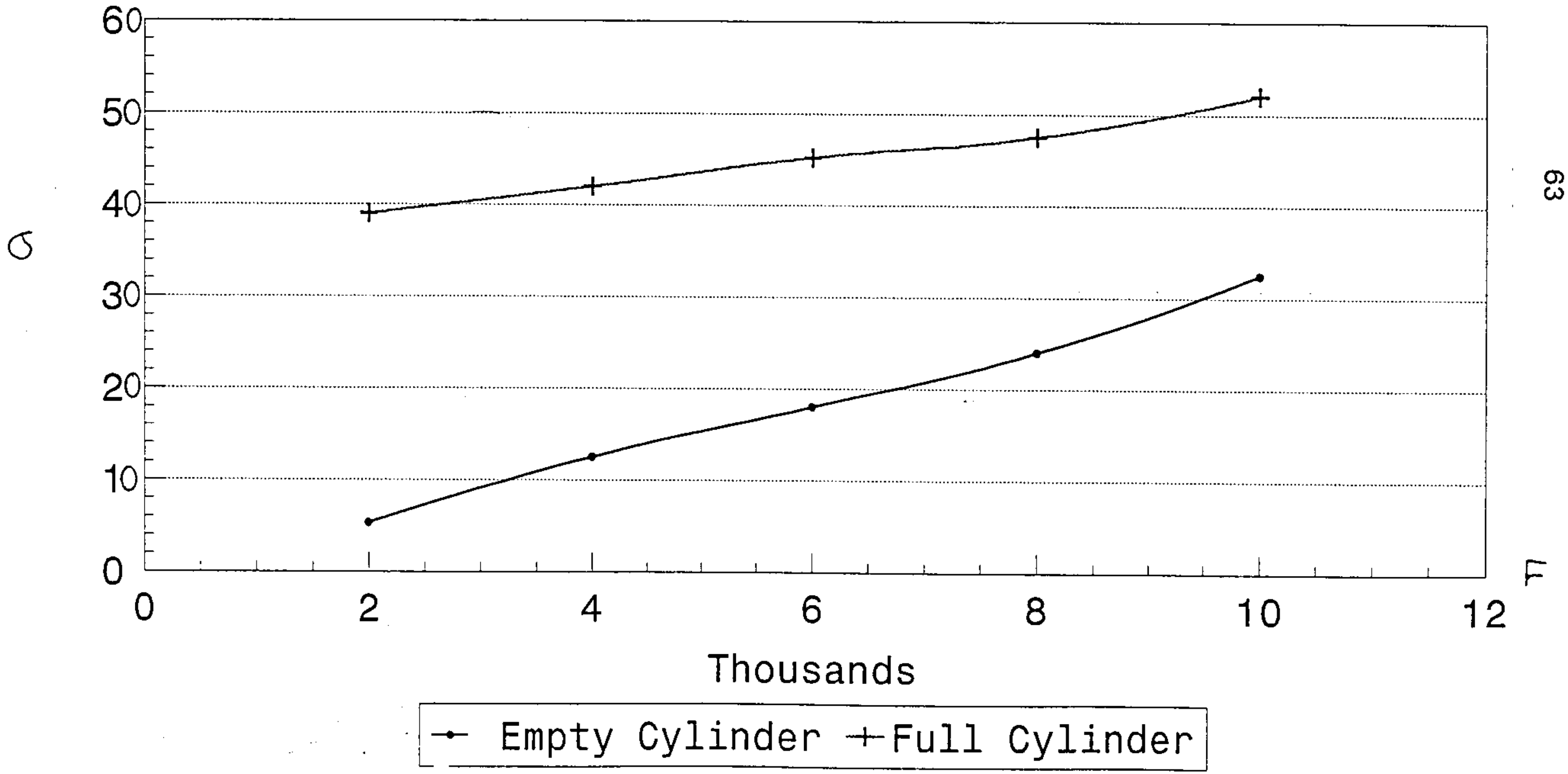


Fig. (3.2) Max. Radial Deflection[mm] verus Static force along cylinder [N]

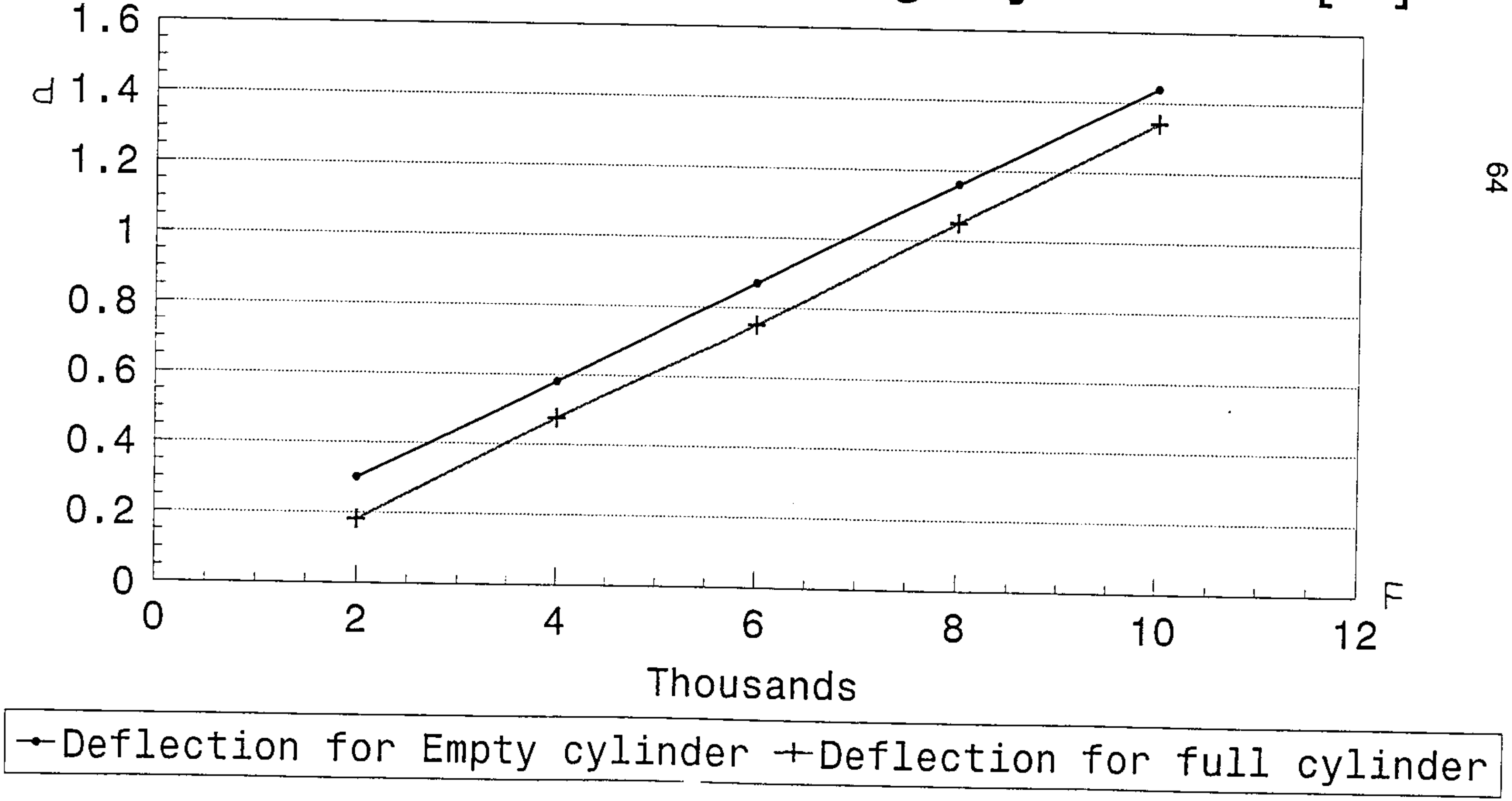


Fig. (4.1) Impact factor verus Hight [m] for full cylinder

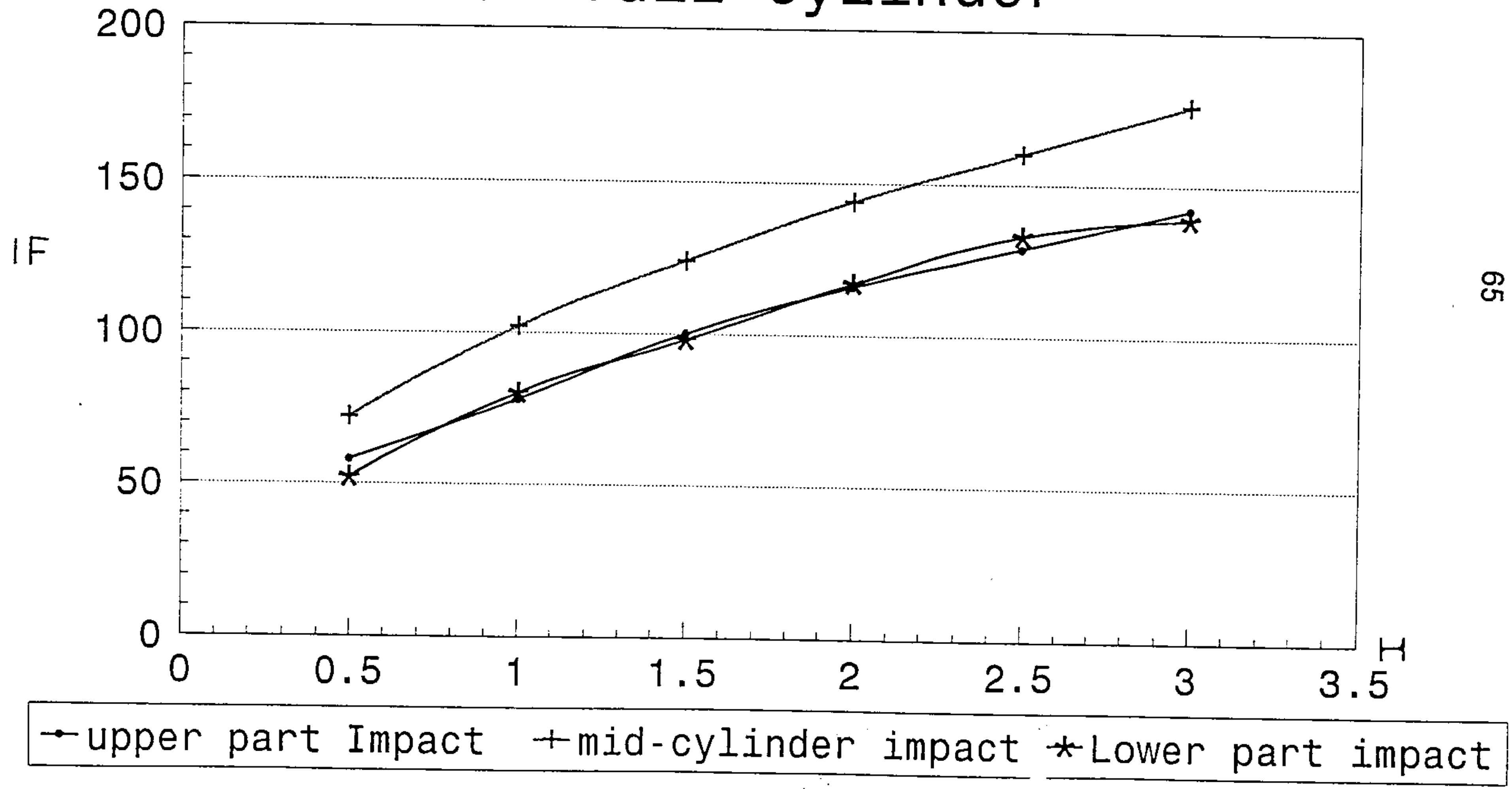


Fig. (4.4) Dynamic load [KN] verus Hight [m] for empty cylinder impact

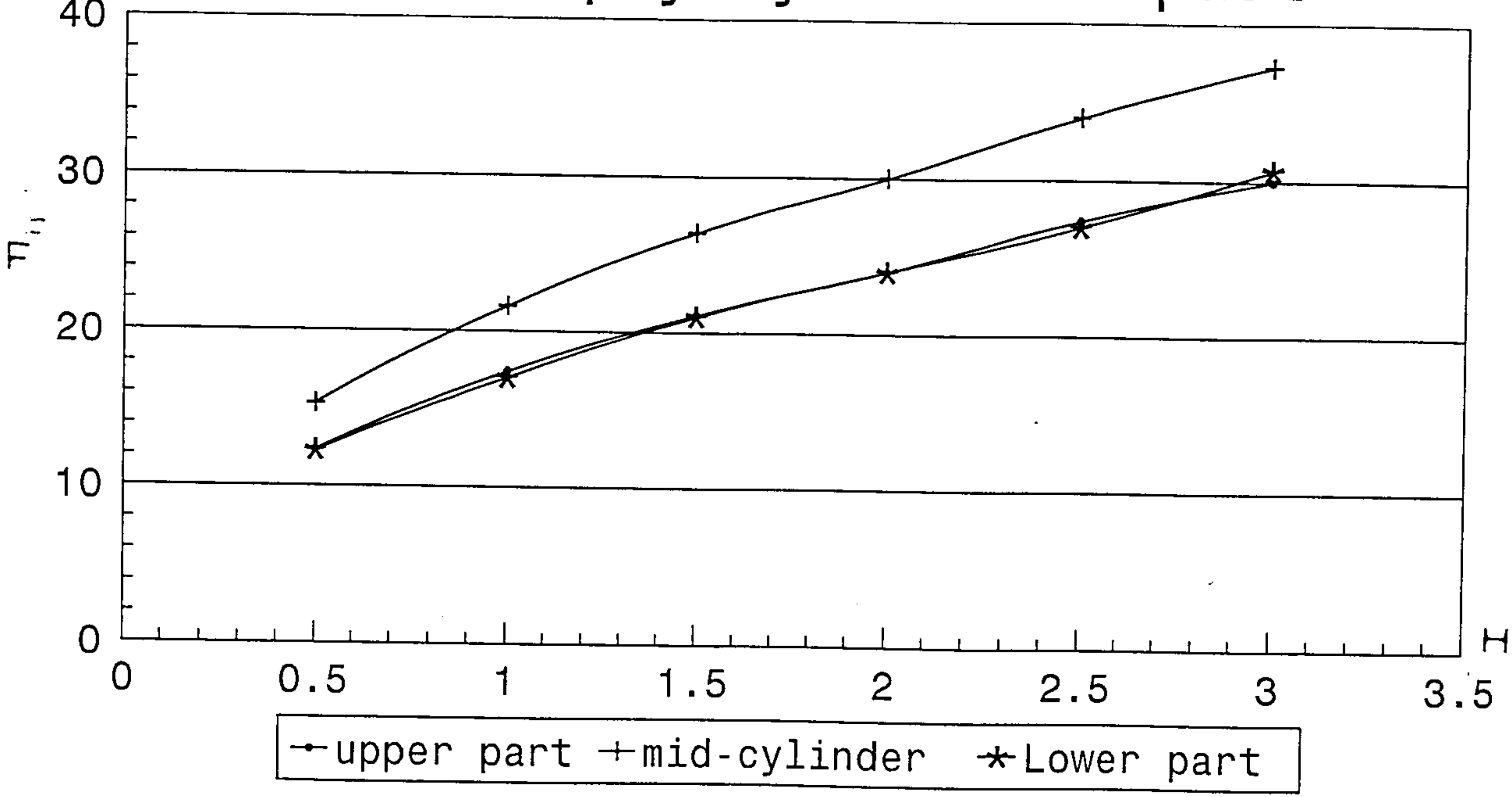


Fig. (4.5) Max. principle stress [N/mm^2] verus Hight [m] for full cylinder impact.

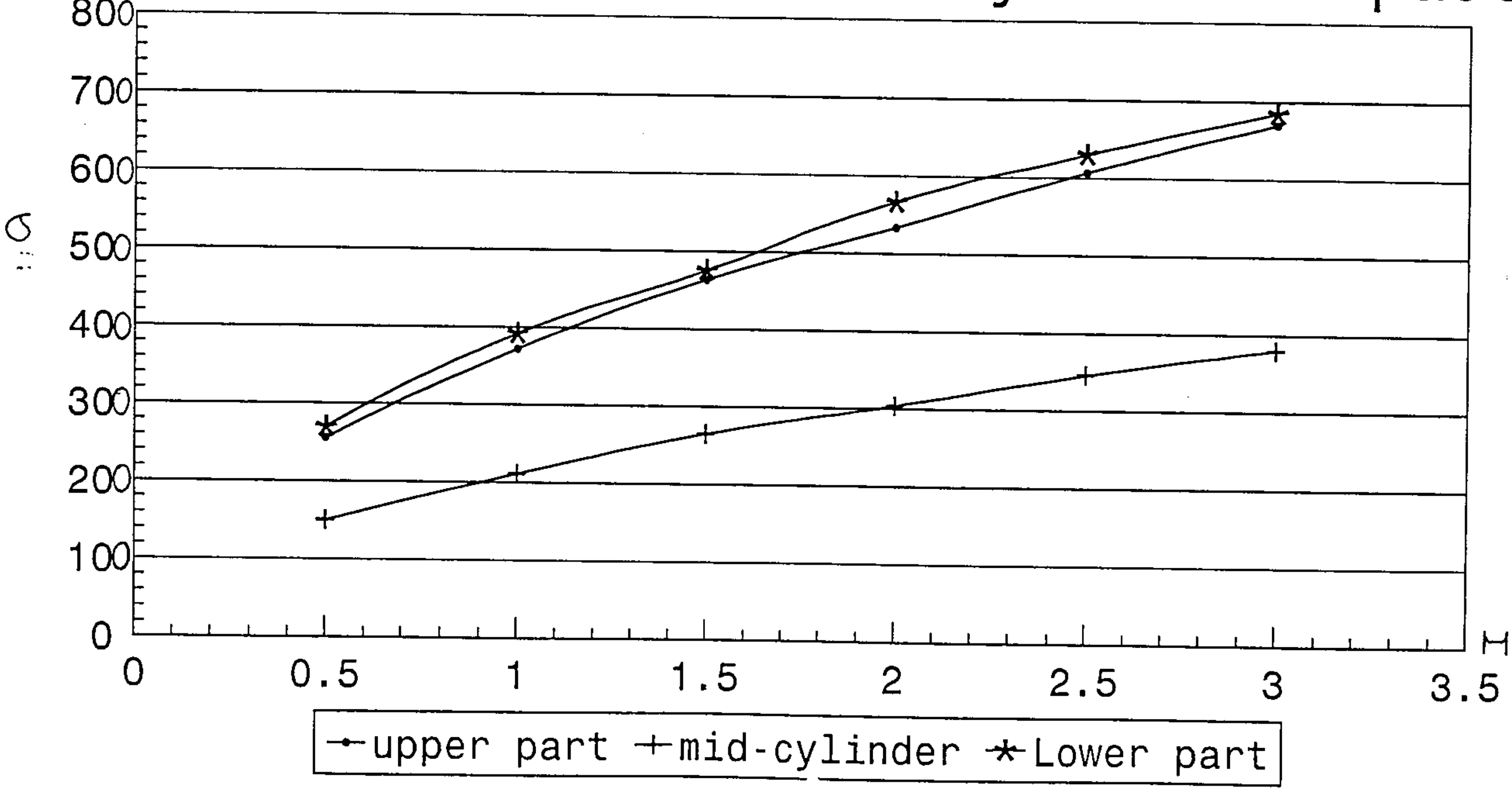


Fig. (4.6) Max. principle stress [N/mm²] verus Hight [m] for empty cylinder

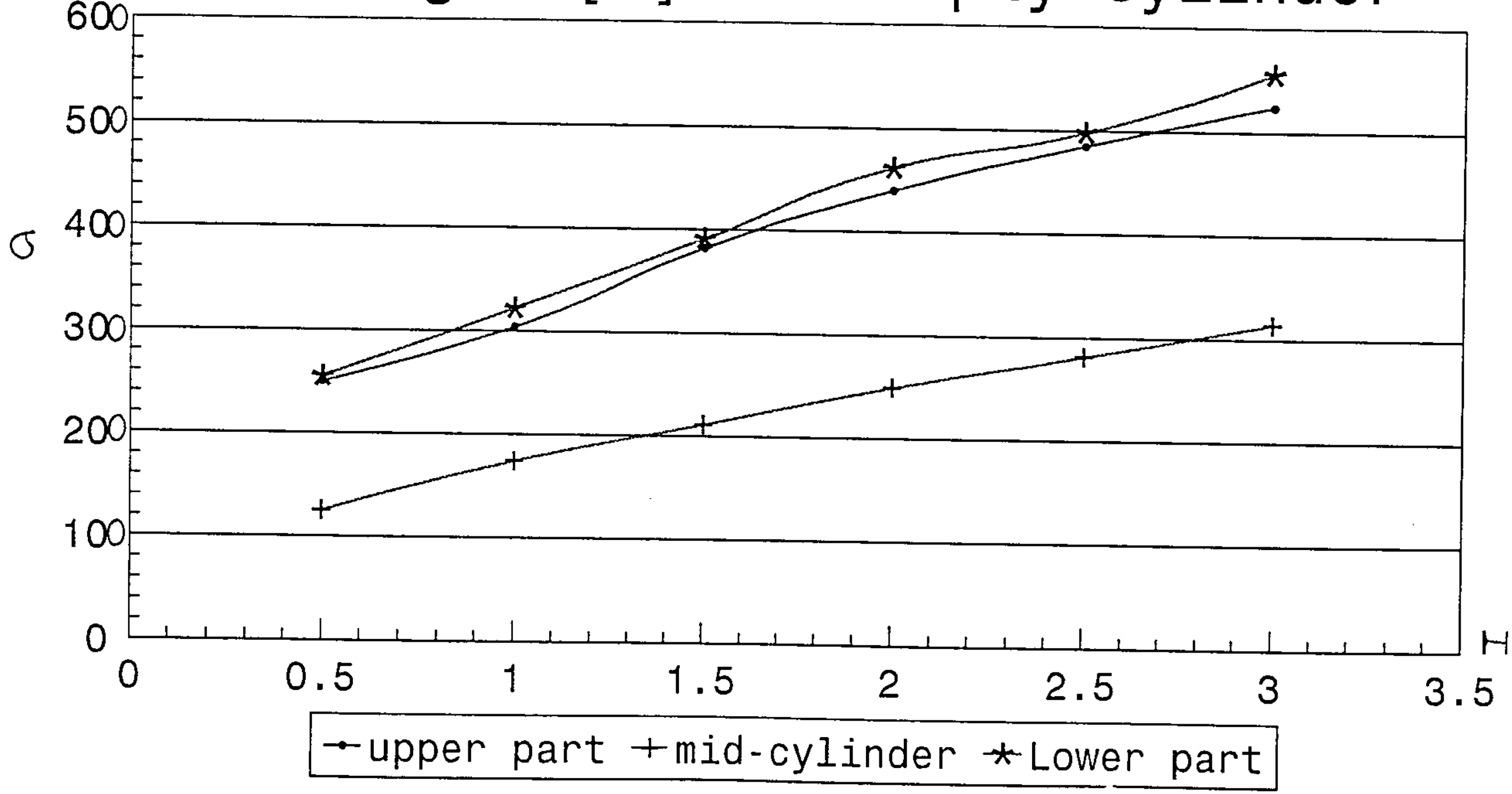


Fig. (4.7) Max. deflection [mm] verus Hight [m) for full cylinder impact.

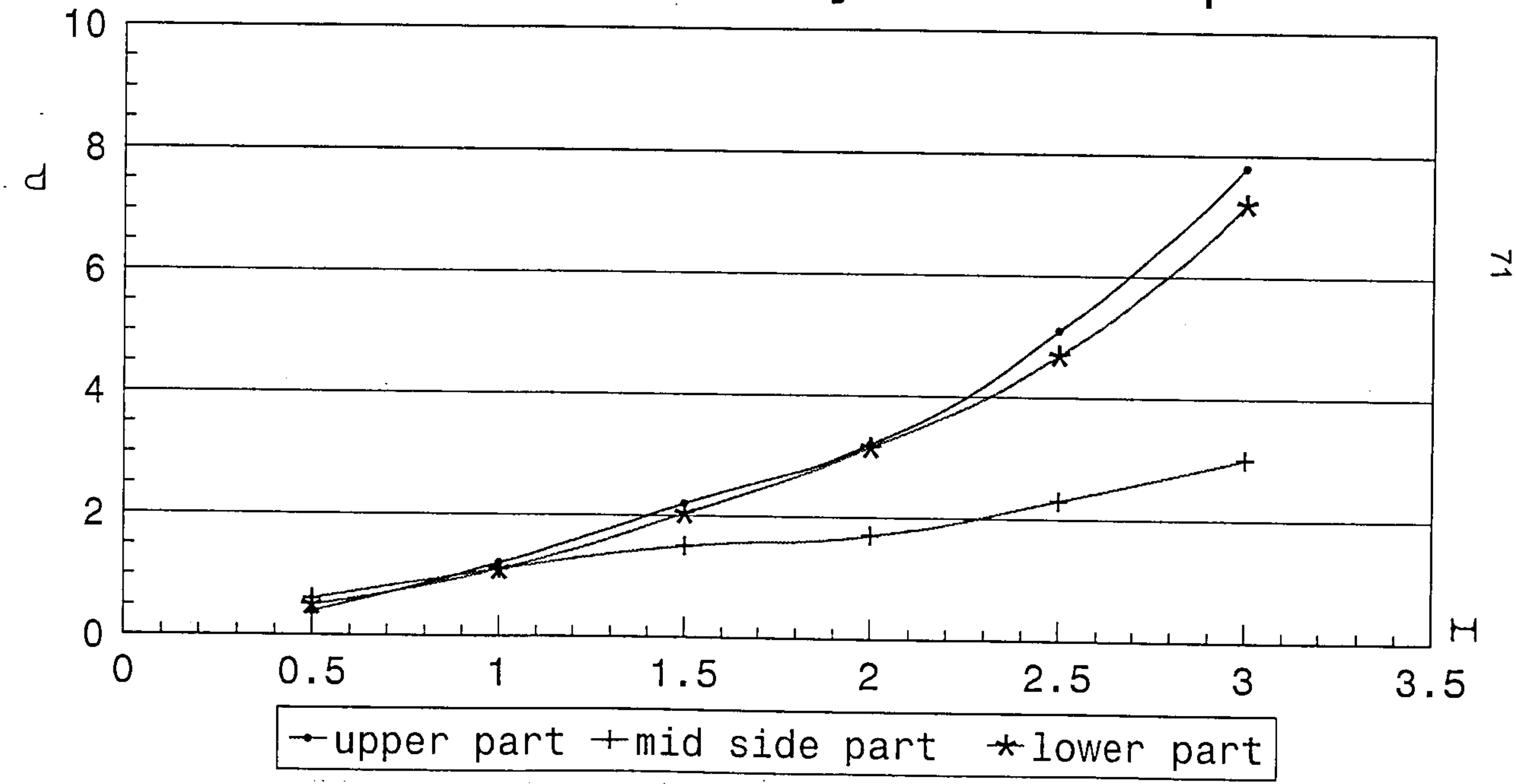


Fig. (4. 8) Max. deflection[mm] verus Hight[m) for empty cylinder impact.

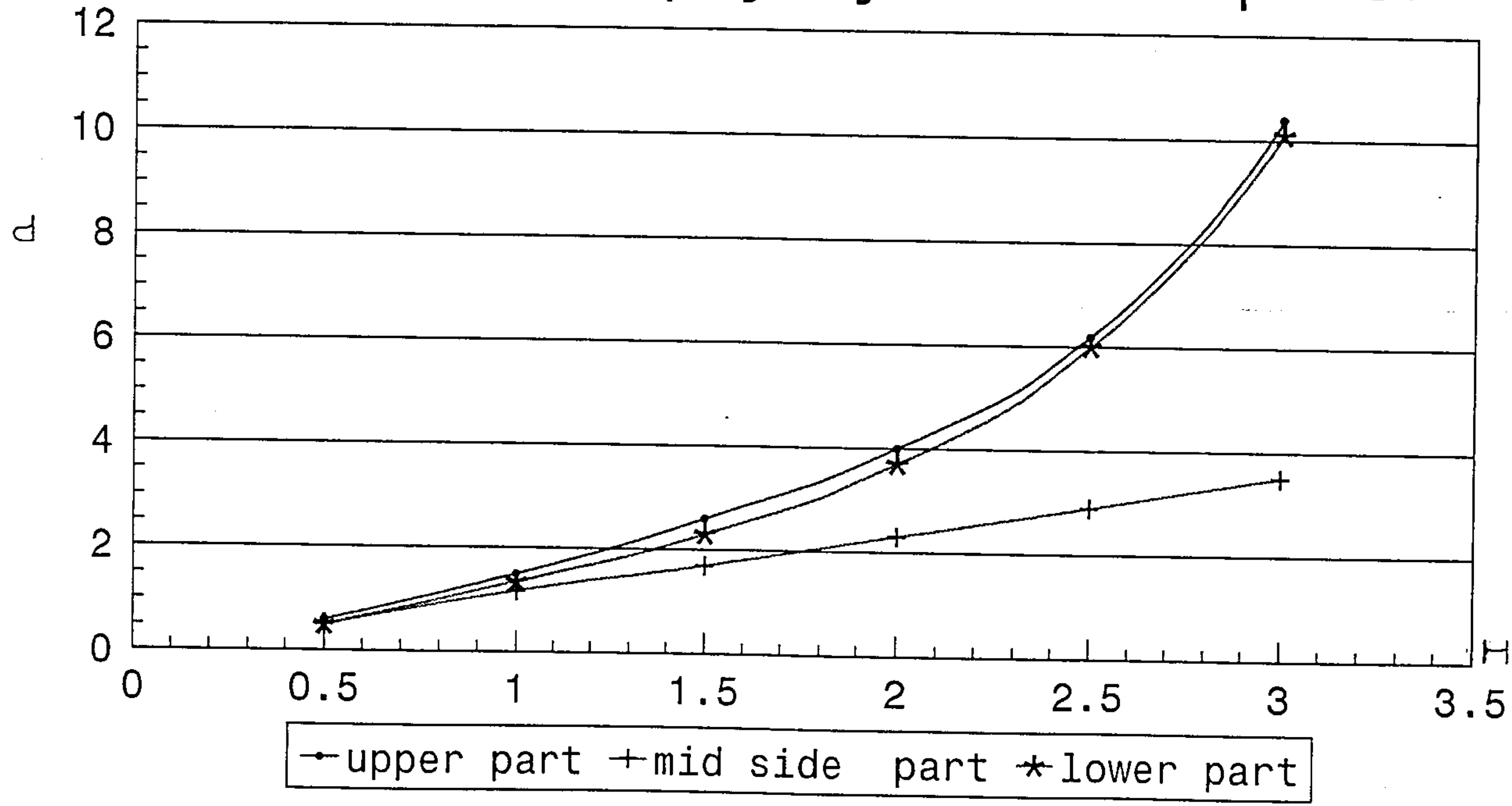


Fig.(5.1) Dynamic load [KN] verus radial deflection[mm] for full cylinder.

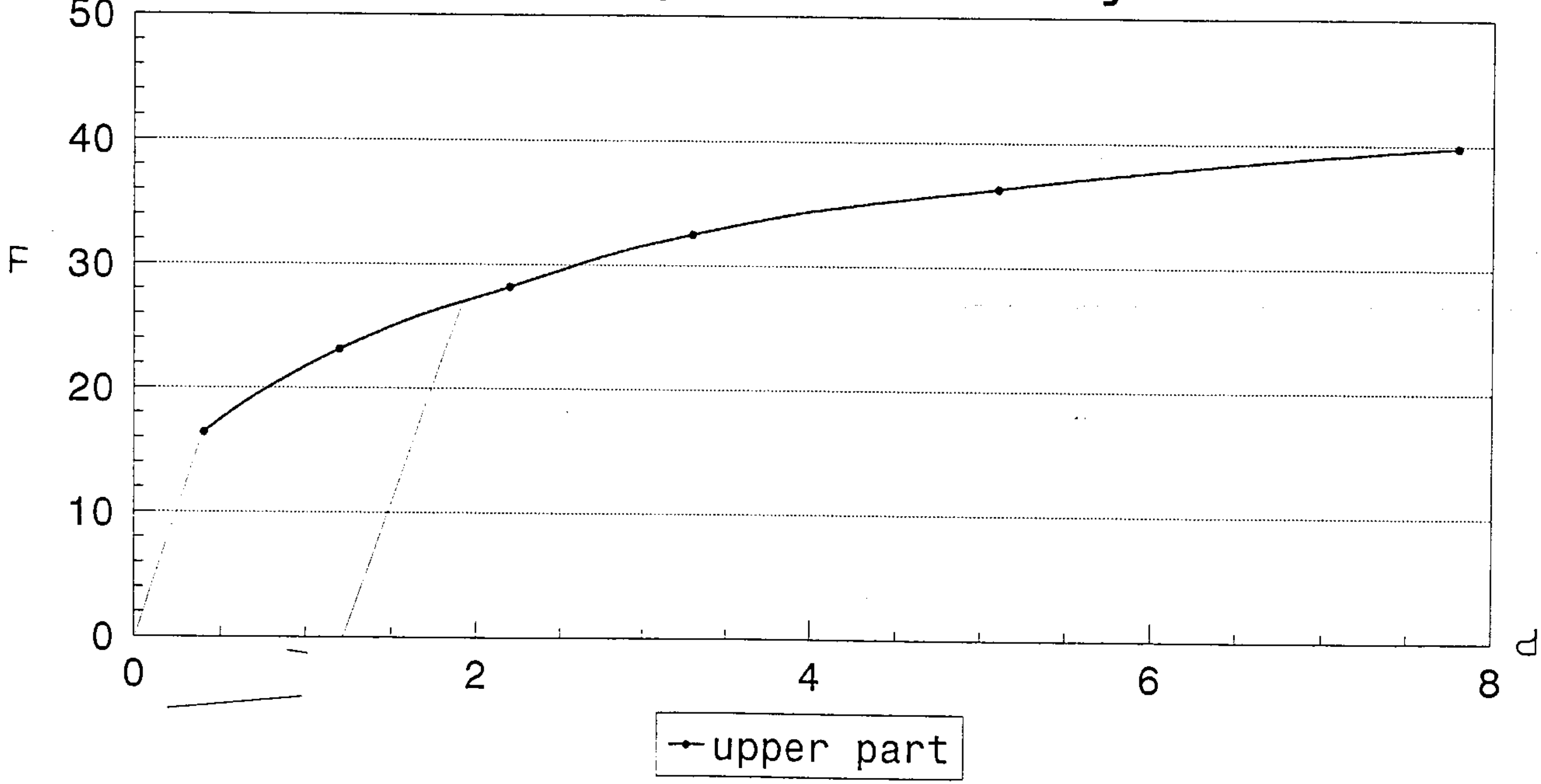


Fig.(5.2) Dynamic load [KN] verus radial deflection[mm] for full cylinder.

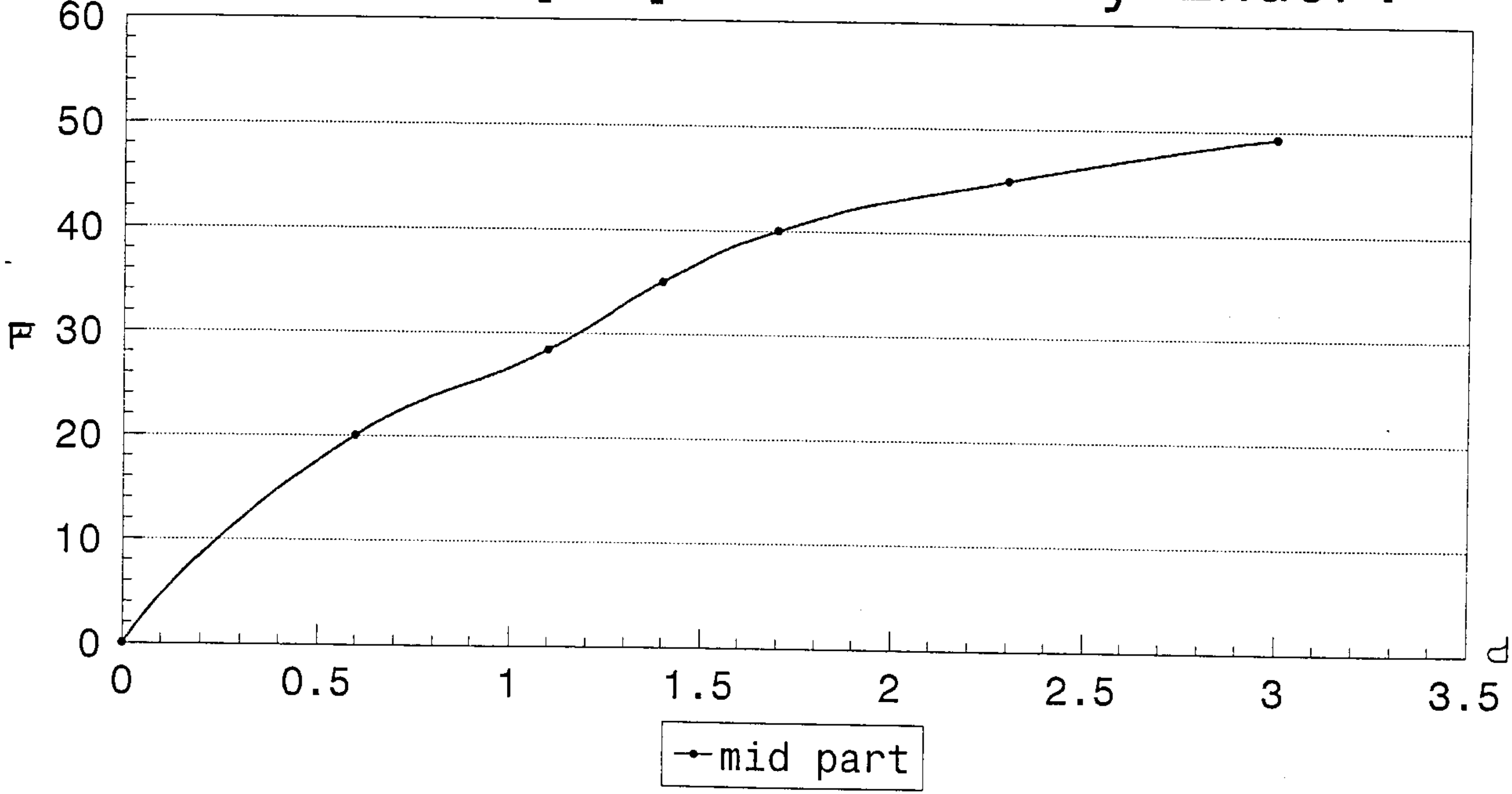


Fig.(5.3) Dynamic load [KN] verus radial deflection[mm] for full cylinder.

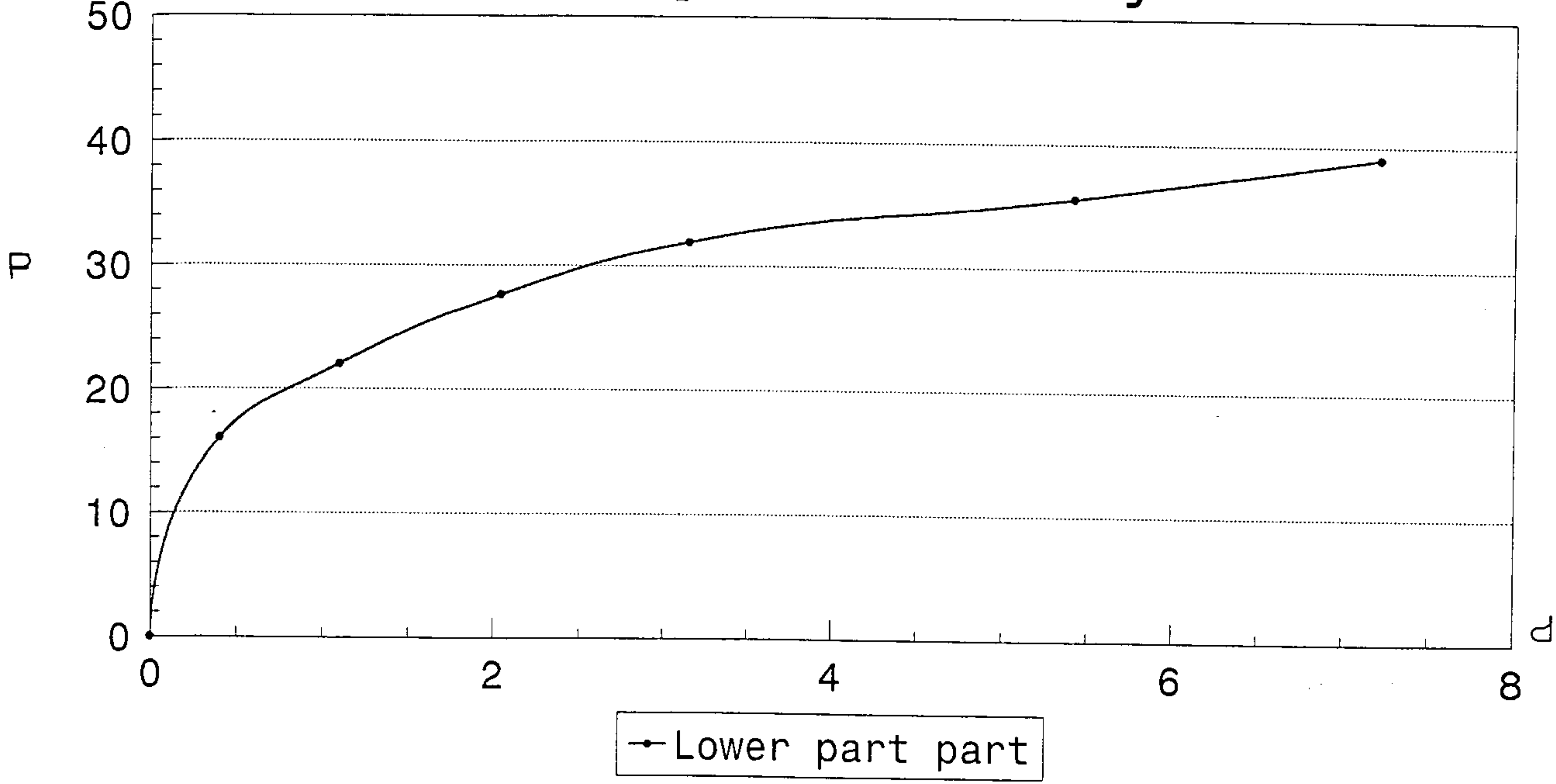


Fig.(5.4) Dynamic load [KN] verus radial deflection[mm] for empty cylinder.

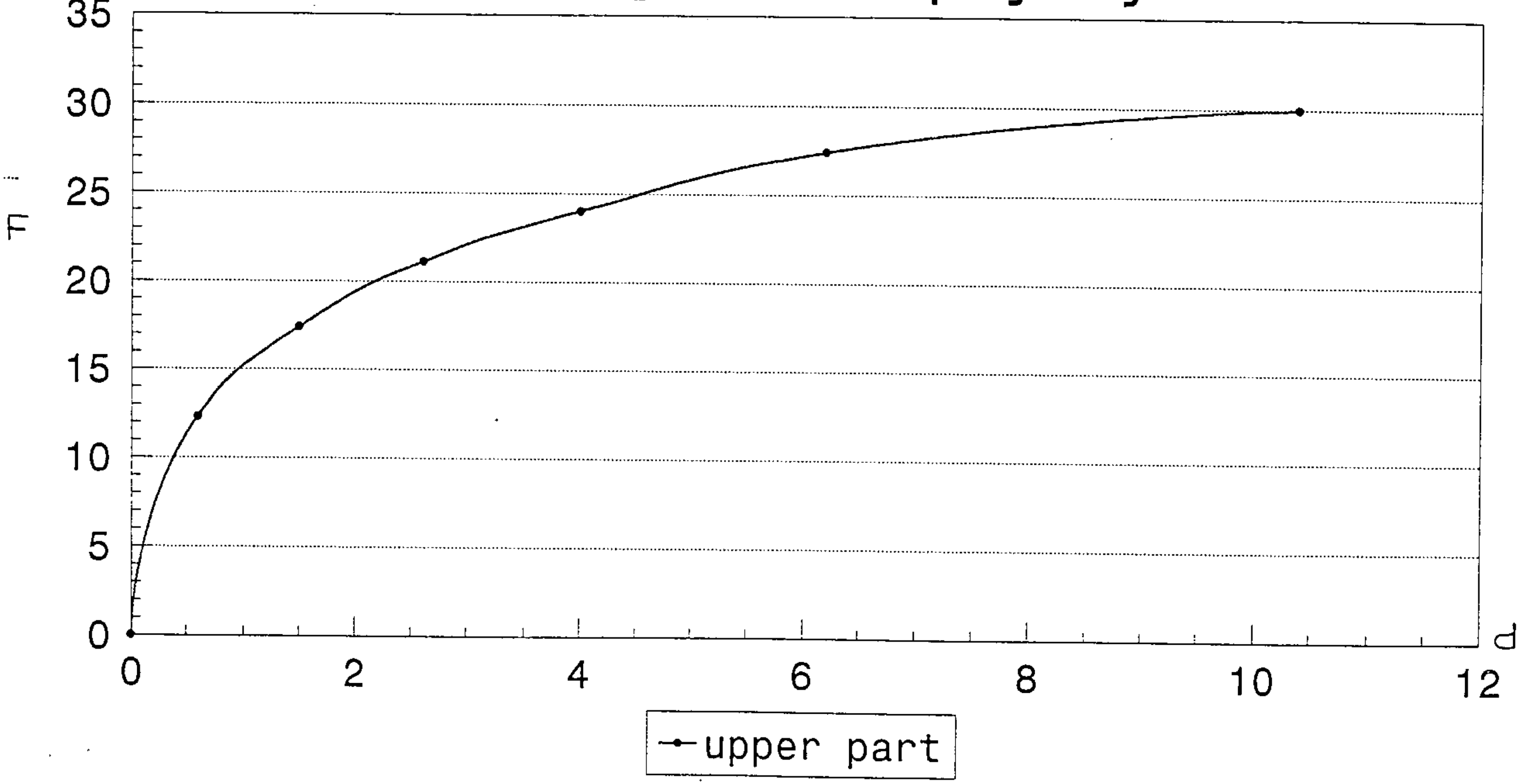


Fig.(5.5) Dynamic load [KN] verus radial deflection[mm] for empty cylinder.

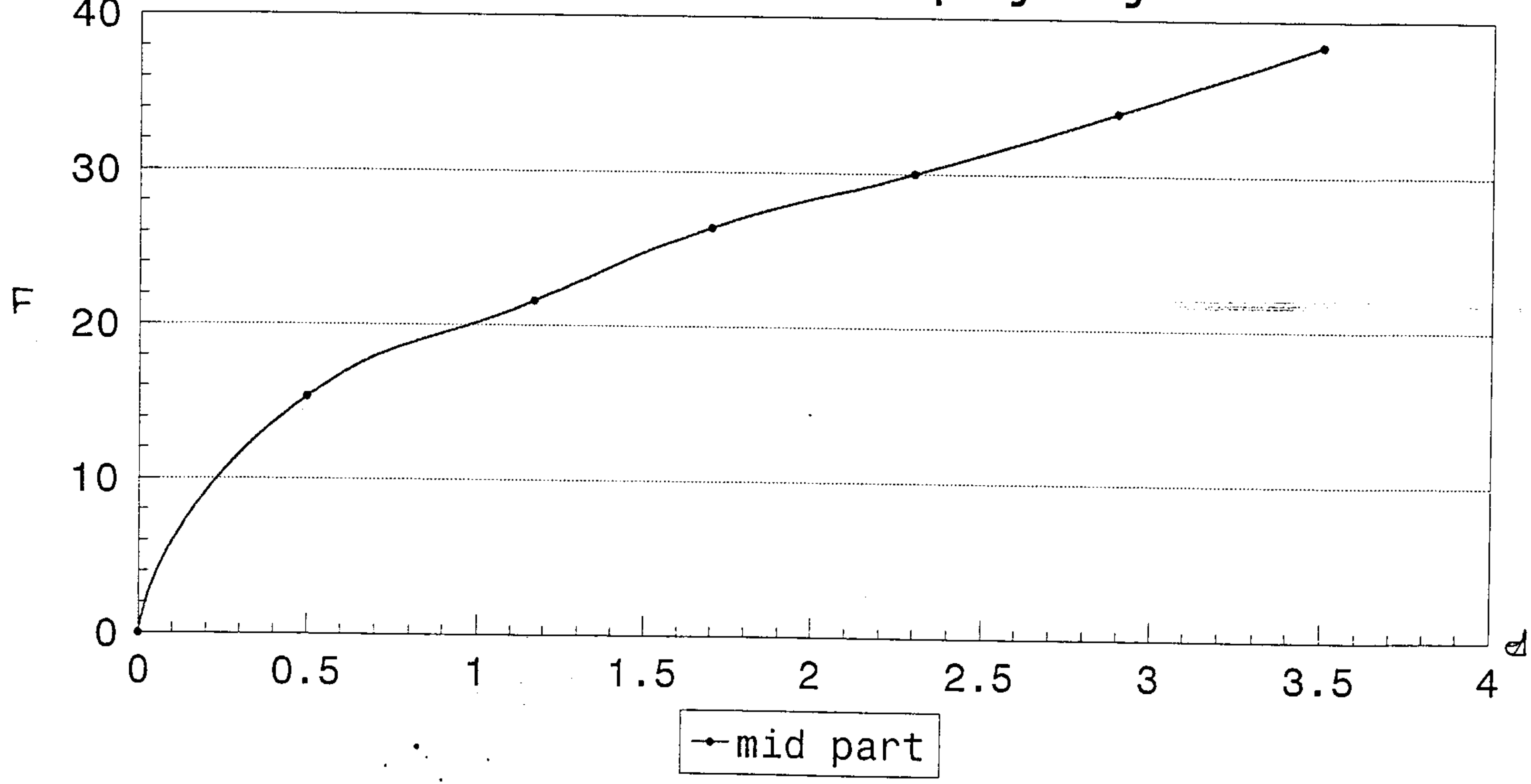


Fig.(5.6) Dynamic load [KN] verus radial deflection[mm] for empty cylinder.

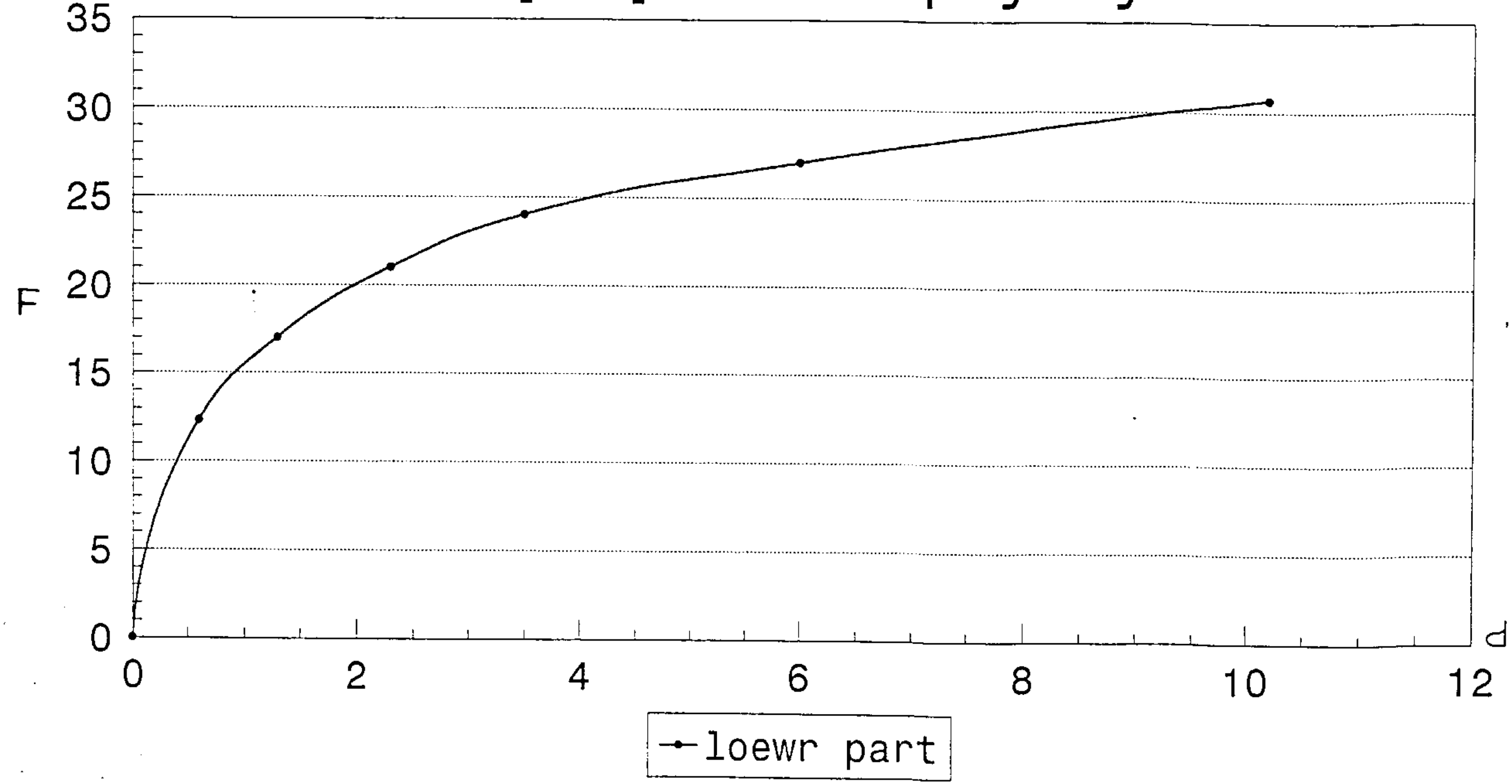


Fig.(6.1) Corrosion rate, Thickness [mm] verus time[year]

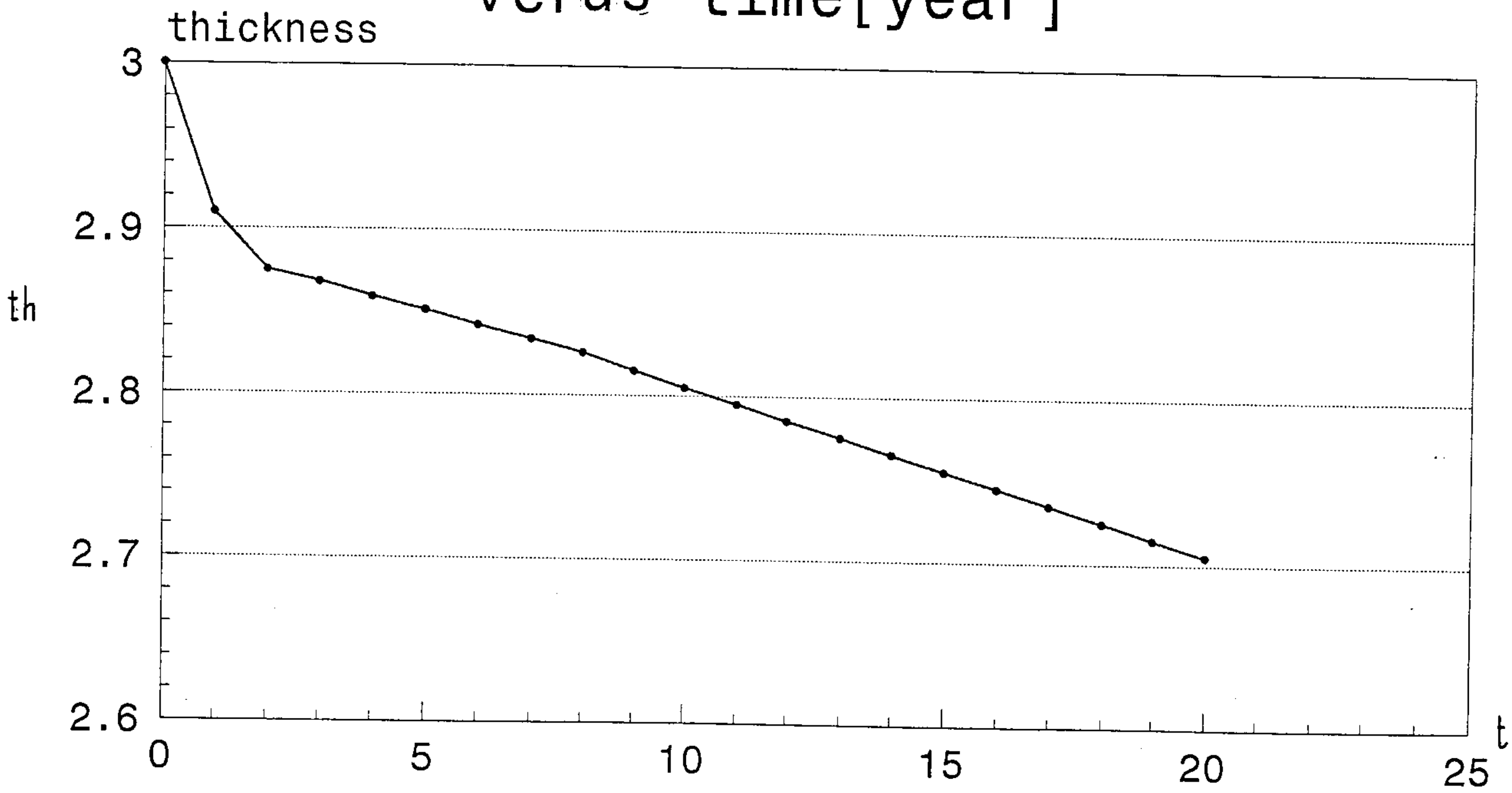
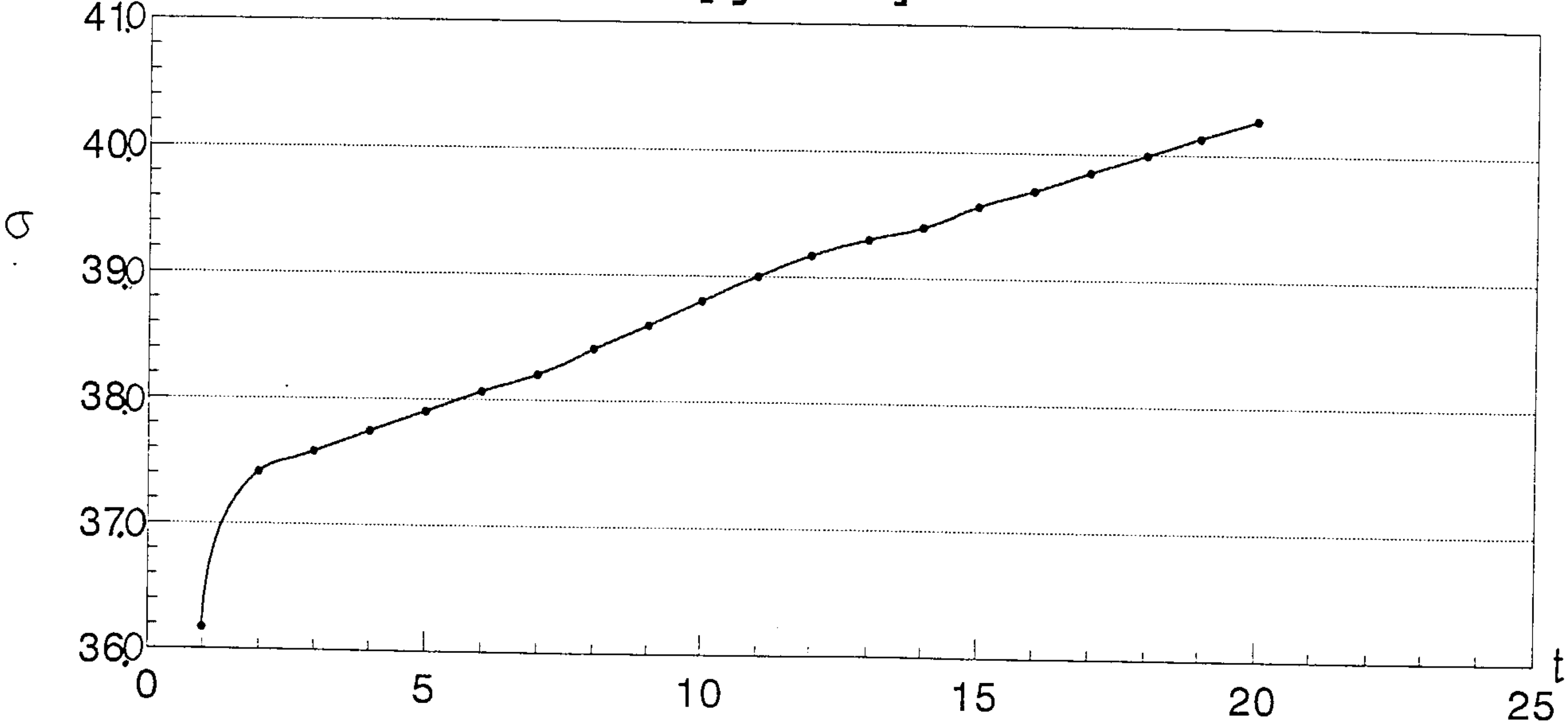


Fig. [6.2] Corrosion ffect on stress; Max.principle stress [N/mm^2] verus time [year]



ii. Finte element program

```
C ^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
c
C              BASEM AL-OMARI
C
C              MASTER THESIS
C
C
C ^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
C
C                   MAIN PROGRAM
C ^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
C
C        INVESTIGATION OF LPG CYLINDERS IN JORDAN
C
C         M,L SUPERVISOR: Dr. SA AD AL_HAEALI
C                UNIVERSITY OF JORDAN
C         CO-SUPERVISOR : Dr. AZZAM ODEH
C           ROYAL SIENTEFIC SOCIETY
C ^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^

      DIMENSION  BB( 100000)
      OPEN (1,FILE=' BASEM\o8.1')
      OPEN (2,FILE=' BASEM\o8.2')
      OPEN (3,FILE=' BASEM\o8.3')
      OPEN (4,FILE=' BASEM\o8.4')
      OPEN (5,FILE=' BASEM\o8.5')
      OPEN (6,FILE=' BASEM\o8.6')
      OPEN (7,FILE=' BASEM\o8.7')
      OPEN (8,FILE=' BASEM\o8.8')
   OPEN (10,FILE=' BASEM\o8.10')
      COMMON NO,IE,N5,IEq,MIN,NL,NII,NJJ,NKK,NVM
      NIN = 9
      INP =4
     PRINT*,' DATA INTRY FROM FILES LPG.1......LPG.5 '
      READ (1,*)  NO , IE  , JBC , JFO
      NL = IE * 9
      NII = 1
      MM = 2
      NJJ=MM
      NVM=NII*NJJ
      N5=  5
      IEq=NO* N5
       KJp=1+NL
       KIW=KJp+IEq +1
       KIM=KIW+NO
       KJ9=KIM+IE
       KIAP=KJ9+IE
       KIBC=KIAP+IE
       KIFO=KIBC+JBC
```

```
          KX=KIFO+JFO
          KY=KX+NO*2
          KZ=KY+NO*2
          KXX= KZ+NO*2
          KYY= KXX+NO*2
          KZZ= KYY+NO*2
          KTH=KZ+NO*2
          KD=KTH+NO*2
          KBCS=KD+IEq*2
          KFrc=KBCS+JBC*2
          KVM=KFrc+JFO*2
          KFg=KVM+NVM*2
          KDg=KFg+IEq*2
          KA=KDg+IEq*2
          MIN= (100000 - KA+1) / 2
        CALL  MAIN  (BB,BB(KJp),BB(KIW) , BB(KIM),BB(KJ9),BB(KIAP),
     -  BB (KIBC), BB(KIFO), BB(KX), BB( KY),BB(KZ),BB(KXX),BB(KYY),
     -  BB(KZZ) , BB(KTH),  BB(KD),
     -  BB(KBCS) ,BB(KFrc), BB(KVM),BB(KFg) , BB(KDg) ,BB(KA),NIN,INP)
        STOP
      END

          SUBROUTINE MAIN ( IA,Jp,IW,IM,J9,IAP,IBC,IFO,X,Y,Z,
     -       DXX,DYY,DZZ,TH,D,BCS,VM,Frc,Fg,Dg, A,NIN , INP  )
C

        IMPLICIT DOUBLE PRECISION  (A-H,O-Z)
        DIMENSION  IA(*) , Jp(*),IW(*),IM(*),IAP(*),VM(*),
     - IBC(*),IFO(*) , X(*) , Y(*) , Z(*) , TH (*) , D (*) , BCS(*),
     - Frc (*) , Fg (*) , Dg(*)  ,A(*), SK(45 , 45) ,bc(45,45)
     -  DXX(*),DYY(*),DZZ(*)

        COMMON  NO , IE , N5 , IEq , MIN , NL , NII,NJJ ,NKK,NVM
        COMMON / EMO/C1 (2)
        COMMON / POS/C2 (6)
        DATA HUGE/1.D16/
        J9  = 9
        DO 77 IK= 1,IE
 77      IAP(IK) = (IK-1) * 9 + 1

        READ (5,*)   ( IA(I) ,I=1,NL)

        READ (4,*)( X(I),Y(I) ,Z(I)   ,i = 1,NO)
 78       NNB=5*NO - 5
       Do 92 ith =  1, NO
 92    th(ith) = 3.0


       CALL BASE  (IE , IAP ,  IA  , NIN, X , Y , Z , Fg , Dg)
```

```
DO 54 II= 1,IE
  54      IM(iI) = 1
        READ   (2,*)    (VM(I) , I= 1,NVM)
        CALL CALCULATE   (N5, IE, IAP, IA , NIN, NO, Jp, IW)
        NA=Jp(IEq)
        WRITE  (7,10)    NA
  10     FORMAT  (/ ' PROFILE OF THE GLOBAL STIFFNESS MATRIX  =',I8)
        DO  66  I=1 ,IE
        WRITE(*,*)"CALCULATION FOR  ELEMENT NUMBER ",I
        IT=J9
        IC=IM(I)
        INT=IAP(I)
        NI9=NIN
        NP=INP
        NEF=NI9*N5
        K=NJJ*(IC-1)
        DO  51  J=1,NJJ
 51      C1(J)=VM(K+J)

        IZZ = 1
        CALL STIFNES  (IZZ,INT,IA,NI9,NP,X,Y,Z,TH,NEF,SK,bc,Fg,Dg)

        CALL ASEMBLY  (INT,IA,NI9,N5,NEF,SK,Jp,A,D)
        CALL ASEMBLY  (INT,IA,NI9,N5,NEF,bc,Jp,A1,D1)
 66      LT=IT
        PRINT*,' MAKING ASSEMBLY   '
PRINT*,' MAKING ASSEMBLY FOR LOAD VECTOR   '
        PRINT*,' TAKING B.C s INTO ACCOUNT  '
READ  (3,*)  JBC,JFO
        READ  (6,*) (IBC(I) , BCS(I) ,  I=1,JBC)
        DO    33   I=1,IEq
 33      DG(I)=   0.D0
        DO  11  I=1,JBC
        D(IBC(I))=HUGE
 11      Dg(IBC(I))=HUGE*BCS(I)
        CALL DECOMP   (IEq,Jp,D,A)
      CALL DECOMP   (IEq,Jp,D1,A1)
        READ  (3,*)   (IFO(I),Frc(IFO(I)),I=1,JFO)
C       READING PRESSURE VALUES OF ELEMENTS
     CALL PRESS (IE,IA,X,Y,Z,Frc)
     DO 73 I=1, IEq
 73      DG(I)= Dg(I) + Frc(i)

        CALL  SOLVING (IEq,Jp,D,A,Dg)
CALL  SOLVING (IEq,Jp,D1,A1,Dg1)
      WRITE(8,30)   (Dg(I) ,I =  1,IEq)
      WRITE(8,31)   (Dg1(I) ,I =  1,IEq)
31      FORMAT   (5F17.5)
30      FORMAT   (5F17.5)
```

```
       JJ = 0
       DO  89 IJ = 1,NO
       DXX(IJ) = Dg(1 + 5* JJ)
       DYY(IJ) = Dg(2 + 5* JJ)
       DZZ(IJ) = Dg(3 + 5* JJ)
 89     JJ = JJ + 1
       RETURN
       END
 C

       SUBROUTINE  DECOMP(IEq, Jp , D , A)
       IMPLICIT  DOUBLE  PRECISION  (A - H , O - Z)
       DIMENSION  Jp (*) , D(*) , A (*)
       DO  11  K=2 , IEq
       K1=K-1
       LK = Jp(k) -K1
       KH=Jp(K1) -LK+1
       S=D(K)
       DO  73  J=KH+ 1 , K1
       J1=J-1
       LJ=Jp(J) -J1
       JH=MAX(Jp(J1) - LJ + 1 , KH)
       T=A(LK+J)
       DO  33  M=JH,J1
 33     T=T -A(LJ+M) * A(LK+M)
 73     A (LK+J)=T
       DO   51   J=KH,K1
       L=LK+J
T=A(L)
       A(L)=T/D(J)
 51     S = S - T * A(L)
 11     D(K)=S
       RETURN
       END


       SUBROUTINE SOLVING (IEq,Jp,D,A,B)
        IMPLICIT DOUBLE PRECISION (A -H,O-Z)
       DIMENSION Jp (*),D(*),A(*),B(*)
       DO  11  J=2,IEq
       J1=J-1
       Lj=Jp(J) -J1
       JH=Jp(J1)- LJ  +1
       T=B(J)
       DO  73  M=JH,J1
 73     T=T-A(LJ+M) *B(M)
 11     B(J) = T
       DO   33    K=1 , IEq
      if( d(k) .EQ. 0.0)D(k)=1.0D-60
```

```
33     B(K) = B (K) /D(K)
C BACKWARD SUBSTITUTION
       DO   51  K=IEq,2,-1
        K1=K-1
       LK=Jp(K) -K1
       KH=Jp(K1) - LK+1
       T = B(K)
       DO    51    J=KH,K1
51       B(j) = B(j) -T *A(Lk+J)
       RETURN
       END
```

```
       SUBROUTINE PRESS (IE,IA,X,Y,Z,Frc)

       IMPLICIT DOUBLE PRECISION (A-H, O-Z)
       DIMENSION NT1(8), X(*),Y(*),Z(*),AREA(8),
     -          NT2(8),NT3(8),Frc(*),P(1100)
     -        ,IA(*) ,ANO1(8),ANO2(8),ANO3(8)

C      NT1 ,NT2 ,NT3  ( ELEMENT NUMBER  , 1,2,...,8 )

       DO 101   I = 1,IE
   C   READING THE VALUE OF PRESSURE P AT ELEMENT # I (i.e ....
    P(I)
       P(I) = 0.D0
       CALL TRI(I,IA ,X,Y,Z,NT1,NT2,NT3,AREA,ANO1,ANO2,ANO3)

       DO 101 J = 1,8
       Frc(5*NT1(j)-4)=Frc(5*NT1(j)-4)+ANO1(J) *AREA(j) *P(i)/3.0D0
       Frc(5*NT1(j)-3)=Frc(5*NT1(j)-3)+ANO2(J) *AREA(j) *P(i)/3.0D0
       Frc(5*NT1(J)-2)=Frc(5*NT1(J)-2)+ANO3(J) *AREA(j) *P(i)/3.0D0

       Frc(5*NT2(J)-4)=Frc(5*NT2(J)-4)+ANO1(J) *AREA(j) *P(i)/3.0D0
       Frc(5*NT2(J)-3)=Frc(5*NT2(J)-3)+ANO2(J) *AREA(j) *P(i)/3.0D0
       Frc(5*NT2(J)-2)=Frc(5*NT2(J)-2)+ANO3(J) *AREA(j) *P(i)/3.0D0

       Frc(5*NT3(J)-4)=Frc(5*NT3(J)-4)+ANO1(J) *AREA(j) *P(i)/3.0D0
       Frc(5*NT3(J)-3)=Frc(5*NT3(J)-3)+ANO2(J) *AREA(j) *P(i)/3.0D0
101     Frc(5*NT3(J)-2)=Frc(5*NT3(J)-2)+ANO3(J) *AREA(j) *P(i)/3.0D0
        RETURN
        END
```

```
      SUBROUTINE    STIFNES  ( INT, IA, NI9, NP, X, Y,
     -        Z, TH, NEF, SK,bc, P, Q)
      IMPLICIT DOUBLE PRECISION   (A - H, O - Z)
      DIMENSION  IA(*) ,X(*) ,Y(*) , Z(*) , TH(*) , P(*) , Q(*)
     -      , SK (NEF ,* )  , DET (18),bc(nef,*)

      COMMON   / ANNN/WT(14) , ANN (280) , dNX1(280) ,dNX2(280)
C
C  CALCULATE ELEMENT STIFFNESS MATRIX FOR VARIOUS
PROBLEMS AND ELEMENT TYPES
C       SHELL  ELEMENT
      CALL   INTERPOL   (NP ,WT ,ANN ,dNX1 ,dNX2)

5     CALL STIFLOC (INT ,IA ,NI9 ,X ,Y,Z ,TH ,NP ,ANN ,dNX1 ,dNX2 ,WT
     -           ,DET ,P ,Q ,SK,skk)
      RETURN
      END
      SUBROUTINE  ASEMBLY  (INT ,IA ,NI9 ,N5 ,NEF ,SK  ,Jp ,A ,D)
      IMPLICIT DOUBLE PRECISION  ( A- H , O - Z)
      DIMENSION  IA(*) , SK(NEF,*) , Jp (*) , A(*) , D(*)

      KK=INT-1
      DO  11  J1=1,NI9
      J2=IA(KK+J1)
      DO  73   I1=1,NI9
      I2 =IA(KK+I1)
      IF  (I2.GT.J2)      GOTO   73
      IF    (I2.EQ.J2)      THEN



      DO  55  J=1,N5
      L1=N5*(J1-1) +J
      L2=N5*(J2-1) +J
      DO   33  I=1 , J-1
      K1=N5*(I1-1) +I
      L=Jp (L2) +I - J + 1
33    A(L)=A(L) +SK(K1,L1)
55    D(L2)=D(L2) +SK(L1,L1)
      ELSE

      DO   51   J=1,N5
```

```
      L1=N5* (J1-1) +J
      L2=N5* (J2-1) +J
      DO   51   I=1 , N5
      K1=N5* (I1-1) +I
      K2=N5* (I2-1) +I
      L=Jp(L2) +k2-L2+1
 51    A(L)=A(L) +SK (K1 , L1)
      ENDIF
 73    CONTINUE
 11    CONTINUE
      RETURN
      END



      SUBROUTINE BASE  (IE, IAP, IA, NIN, X, Y, Z, P, Q)
      IMPLICIT  DOUBLE  PRECISION  (A - H , O -Z)
      DIMENSION IA(*) , IAP (*) ,X(*),Y(*),Z (*)
     -              , P (*) , Q(*)
      COMMON   / ANNN /  WT(14)  , ANN (280) , dNX1 (280) , dNX2(280)
      NP=0
      CALL  INTERPOL   (NP, WT, ANN, dNX1, dNX2)
      DO 11 I = 1,IE
      NI9 = NIN
      INT = IAP(I) -1
      L = 0

      DO  73 J=1,NI9
      K=IA(INT+J)
      KK=5*K-5
      U1=0.D0
      U2=0.D0
      U3=0.D0
      V1=0.D0
      V2=0.D0
      V3=0.D0
C  derivaives of  (x , y , z) TO element coordinates
      DO  33  JJ=1,NI9
      II=IA(INT+JJ)
      L=L+1
      U1=U1 +dNX1 (L) *X(II)
      U2=U2 +dNX1 (L) *Y(II)
      U3=U3 +dNX1 (L) *Z(II)
      V1=V1 +dNX2 (L) *X(II)
      V2=V2 +dNX2 (L) *Y(II)
 33    V3=V3 +dNX2 (L) *Z(II)

      U=DSQRT(U1*U1+ U2*U2+ U3*U3)
      U1=U1/U
```

```
      U2=U2/U
      U3=U3/U
      W1=U2*V3-U3*V2
      W2=U3*V1-U1*V3
      W3=U1*V2-U2*V1
      W=DSQRT(W1 *W1 +W2*W2 +W3*W3)
      W1=W1/W
      W3=W3/W
      W2=W2/W
      W=W/DSQRT(V1*V1+V2*V2+V3*V3)
      IF  (W.LT.0.5)  WRITE  (7,20)  I,J,K,W
20     FORMAT  ( / ' ELEMENT' , I4, '    VERTICE, ', I2,
   -         '        NODE.' , I4, '    W = ' , F9 . 5)
       ENDIF
      ENDIF
      IF (W. GT . Q(KK+5))  THEN
      P (KK+1) =U1
      P (KK+2) =U2
      P(KK+3) =U3
      P(KK+4) =W2*U3-W3*U2
      P(KK+5) =W3*U1-W1*U3
      Q(KK+1) =W1*U2-W2*U1
      Q(KK+2) =W1
      Q(KK+3) =W2
      Q(KK+4) =W3
      Q(KK+5) =W
      ENDIF
73     CONTINUE
11     CONTINUE
      RETURN
      END

      SUBROUTINE STIFLOC ( INT, Id,NI9,X,Y,Z,TH
   -   ,NP,ANN,dNX1,dNX2
   -          ,WT,DET,P,Q,ZK,bc)
      IMPLICIT DOUBLE PRECISION (A-H, O-Z)
      DIMENSION Id(*),X(*),Y(*),Z(*),TH(*),ANN(*)
   -          ,dNX1(*),dNX2(*),DET(*)
   -      , P(*),Q(*),WT(*),ZK(45,*),XX(9),YY(9),ZZ(9),TT(9),
   -          U1(9), U2(9), U3(9), V1(9), V2(9), V3(9),
   -           W1(9), W2(9), W3(9) ,bc(*,*)
   - dNXX(162),dNXY(162),dNXZ(162) , dNIXX(162),dNIXY(162)
   -    ,dNIXZ(162) ,EdS11(18),
   -EdS12(18),EdS13(18),EdS21(18),EdS22(18),EdS23(18),EdS31(18)
   -   ,EdS32(18),EdS33(18)

      COMMON /EMO/ELAS,POIS

C   CALCULATE THE LOCAL STIFFNESS MATRIX
```

```
      K = INT - 1
      DO 51 i= 1, NI9
        NY = Id(K+i)
        XX(I)= X(NY)
        YY(I)= Y(NY)
        ZZ(I)= Z(NY)
        TT(I)=TH(NY)
        NYJ= 5*NY- 5
        U1(I) = P(NYJ+1)
        U2(I) = P(NYJ+2)
        U3(I) = P(NYJ+3)
        V1(I) = P(NYJ+4)
        V2(I) = P(NYJ+5)
        V3(I) = Q(NYJ+1)
        W1(I) = Q(NYJ+2)
        W2(I) = Q(NYJ+3)
51      W3(I) = Q(NYJ+4)

        NP2 = NP *2
        KK= 0
C     NP2 :  # OF INTEG. POINTS
       DO 33 M = 1,2
        ZETA = (2 * M -3) / DSQRT(3.D0 )
        NY= 0
        DO 11 K = 1, NP
        KK = KK+ 1

        F11 = 0.D0
        F12 = 0.D0
        F13 = 0.D0
        F21 = 0.D0
        F22 = 0.D0
        F23 = 0.D0
        F31 = 0.D0
        F32 = 0.D0
        F33 = 0.D0
        DO 73 I= 1,NI9
        L= NY+I
        T= TT(i) / 2.D0
        ZT = ZETA * T

        F11 = F11 + dNX1(L) * (XX(I) + ZT * W1(I))
        F12 = F12 + dNX1(L) * (YY(I) + ZT * W2(I) )
        F13 = F13 + dNX1(L) * (ZZ(I) + ZT * W3(I) )
        F21 = F21 + dNX2(L) * (XX(I) + ZT * W1(I) )
        F22 = F22 + dNX2(L) * (YY(I) + ZT * W2(I) )
        F23 = F23 + dNX2(L) * (ZZ(I) + ZT * W3(I) )
```

```
      F31 = F31 +  ANN(L) *         T * W1(I)
      F32 = F32 +  ANN(L) *         T * W2(I)
     F33 = F33 +  ANN(L) *         T * W3(I)
73    CONTINUE
C
          F1 = F12*F23 - F22*F13
          F2 = F13*F21 - F23*F11
          F3 = F11*F22 - F21*F12
          F = DSQRT(F1*F1 +F2*F2 +F3*F3)

          F1 = F1/F
          F2 = F2/F
          F3 = F3/F

          F = DSQRT(F11*F11 +F12*F12+ F13*F13 )
C         IF(F.EQ.0.0)F=1.0D-60
          d1 = F11 / F
          d2 = F12 / F
          d3 = F13 / F

         E1 = F2 *d3 - d2*F3
         E2 = F3 *d1 - d3*F1
         E3 = F1 *d2 - d1*F2

         EdS11(KK) = d1
         EdS12(KK) = d2
         EdS13(KK) = d3
         EdS21(KK) = E1
         EdS22(KK) = E2
         EdS23(KK) = E3
         EdS31(KK) = F1
         EdS32(KK) = F2
         EdS33(KK) = F3


         D11 = d1 * F11 +d2*F12 +d3 *F13
      D12 = E1 * F11 +E2*F12 +E3 *F13
         D13 = F1 * F11 +F2*F12 +F3 *F13
         D21 = d1 * F21 +d2*F22 +d3 *F23
         D22 = E1 * F21 +E2*F22 +E3 *F23
         D23 = F1 * F21 +F2*F22 +F3 *F23
         D31 = d1 * F31 +d2*F32 +d3 *F33
         D32 = E1 * F31 +E2*F32 +E3 *F33
         D33 = F1 * F31 +F2*F32 +F3 *F33

          DDD1 =   D11* (D22*D33 -D23*D32  )
          DDD2 =   D12* (D23*D31 -D21*D33 )
          DDD3 =   D13* (D21*D32 -D22*D31 )
          DT =    DDD1+DDD2+DDD3
```

```
      F11= (D22 *D33 - D32*D23 ) / DT
      F12= (D32 *D13 - D12*D33 ) / DT
      F13= (D12 *D23 - D22*D13 ) / DT
      F21= (D23 *D31 - D33*D21 ) / DT
      F22= (D33 *D11 - D13*D31 ) / DT
      F23= (D13 *D21 - D23*D11 ) / DT
      F31= (D21 *D32 - D31*D22 ) / DT
      F32= (D31 *D12 - D11*D32 ) / DT
      F33= (D11 *D22 - D21*D12 ) / DT

      DET(KK) = DT * WT(K)


C
      DO 55   I=1,NI9
      L= NY+I
      LL = NI9 * (KK-1) +I

      dNXX(LL)  = F11 *dNX1(L)+F12 * dNX2(L)
      dNXY(LL)  = F21 *dNX1(L)+F22 * dNX2(L)
      dNXZ(LL)  = F31 *dNX1(L)+F32 * dNX2(L)

      dNIXX(LL) =ZETA *dNXX(L)+ F13 * ANN(L)
      dNIXY(LL) =ZETA *dNXY(L)+ F23 * ANN(L)
      dNIXZ(LL) =ZETA *dNXZ(L)+ F33 * ANN(L)
55    CONTINUE
11    NY=NY+NI9
33    CONTINUE
C
      f1F = ELAS/(1.-POIS*POIS)
      F3F = (1.-POIS)/2.D0
      F4F = 5.*F3/6.

      DO 732 Kd = 1,NI9
      K1 = (Kd-1) *5
      T = TT(Kd)/2.D0

      V1d = -T*V1(Kd)
      V2d = -T*V2(Kd)
      V3d = -T*V3(Kd)


      U1d =  T*U1(Kd)
      U2d =  T*U2(Kd)
      U3d =  T*U3(Kd)


      DO 732 KE =Kd,NI9
```

```
K2 = (KE-1) *5
T = TT(KE)/2.D0

V1E = -T*V1(KE)
V2E = -T*V2(KE)
V3E = -T*V3(KE)

U1E =  T*U1(KE)
U2E =  T*U2(KE)
U3E =  T*U3(KE)

Z11=  0.D0
Z12 = 0.D0
Z13 = 0.D0
Z14 = 0.D0
Z15 = 0.D0

Z21 = 0.D0
Z22 = 0.D0
Z23 = 0.D0
Z24 = 0.D0
Z25 = 0.D0

Z31 = 0.D0
Z32 = 0.D0
Z33 = 0.D0
Z34 = 0.D0
Z35 = 0.D0

Z41 = 0.D0
Z42 = 0.D0
Z43 = 0.D0
Z44 = 0.D0
Z45 = 0.D0

Z51 = 0.D0
Z52 = 0.D0
Z53 = 0.D0
Z54 = 0.D0
Z55 = 0.D0


DO 333 K = 1,NP2
 F= DET(K)* F1F
d1 = EdS11(K)
d2 = EdS12(K)
d3 = EdS13(K)
```

```
      dNX1 (MM+2) = DF3 * G1
      dNX1 (MM+3) = DF3 * G3
      dNX1 (MM+4) = DF1 * G3
      dNX1 (MM+5) = DF2 * G1
      dNX1 (MM+6) = DF3 * G2
      dNX1 (MM+7) = DF2 * G3
      dNX1 (MM+8) = DF1 * G2
      dNX1 (MM+9) = DF2 * G2

      dNX2 (MM+1) = F1 * DG1
      dNX2 (MM+2) = F3 * DG1
      dNX2 (MM+3) = F3 * DG3
      dNX2 (MM+4) = F1 * DG3
      dNX2 (MM+5) = F2 * DG1
      dNX2 (MM+6) = F3 * DG2
      dNX2 (MM+7) = F2 * DG3
      dNX2 (MM+8) = F1 * DG2
      dNX2 (MM+9) = F2 * DG2
 11    MM = MM + 9

      RETURN
      END


       SUBROUTINE DET3 (W, S, X, E, D, c, R, f, V, AA)
      IMPLICIT DOUBLE PRECISION (A-H, O-Z)

       AA = W * (D * V - f * c) + E *(f*X-S*V)+R*(S*c- X * D )
       RETURN
      END



      SUBROUTINE TRI(I,IA
     ,X,Y,Z,NT1,NT2,NT3,AREA,ANO1,ANO2,ANO3)

      IMPLICIT DOUBLE PRECISION (A-H, O-Z)
      DIMENSION NT1(8), X(*),Y(*),Z(*),AREA(8),
     -         NT2(8),NT3(8)
     -        ,IA(*) ,ANO1(8),ANO2(8),ANO3(8)
C     NT1 ,NT2 ,NT3  ( ELEMENT NUMBER  , 1,2,...,8 )

      NT1(1)=  IA(1+9*(I-1))
      NT2(1)=  IA(5+9*(I-1))
      NT3(1)=  IA(9+9*(I-1))

      NT1(2)=  IA(5+9*(I-1))
      NT2(2)=  IA(2+9*(I-1))
      NT3(2)=  IA(9+9*(I-1))
```

```
      NT1(3)=  IA(2+9*(I-1))
      NT2(3)=  IA(6+9*(I-1))
      NT3(3)=  IA(9+9*(I-1))

      NT1(4)=  IA(6+9*(I-1))
      NT2(4)=  IA(3+9*(I-1))
      NT3(4)=  IA(9+9*(I-1))

      NT1(5)=  IA(3+9*(I-1))
      NT2(5)=  IA(7+9*(I-1))
      NT3(5)=  IA(9+9*(I-1))


      NT1(6)=  IA(7+9*(I-1))
      NT2(6)=  IA(4+9*(I-1))
      NT3(6)=  IA(9+9*(I-1))


      NT1(7)=  IA(4+9*(I-1))
      NT2(7)=  IA(8+9*(I-1))
      NT3(7)=  IA(9+9*(I-1))

      NT1(8)=  IA(8+9*(I-1))
      NT2(8)=  IA(1+9*(I-1))
      NT3(8)=  IA(9+9*(I-1))
C       DO 23 II = 1, 8


      DO 12 J = 1,8

      CALL DET3(X(NT1(J)),X(NT2(J)),X(NT3(J))
     -        ,Y(NT1(J)),Y(NT2(J)),Y(NT3(J))
     -        , 1.D0 , 1.D0  ,    1.D0,AA3 )

      CALL DET3(Y(NT1(J)),Y(NT2(J)),Y(NT3(J))
     -        ,Z(NT1(J)),Z(NT2(J)),Z(NT3(J)) ,
     -          1.D0, 1.D0 ,1.D0, AA1 )


      CALL DET3(Z(NT1(J)),Z(NT2(J)),Z(NT3(J) )
     -        , X(NT1(J)),X(NT2(J)),X(NT3(J) )
     -        , 1.D0, 1.D0 ,1.D0, AA2 )

       AA1= AA1/2.D0
       AA2= AA2/2.D0
       AA3= AA3/2.D0
```

```
      AREA(J)= DSQRT(AA1**2 + AA2**2 +AA3**2 )
C       WRITE (10,*)'no',I,J,AA1,AA2,AA3,AREA(J)
       ANO1(J)= (AA1)/AREA(J)
      ANO2(J)= (AA2)/AREA(J)
      ANO3(J)= (AA3)/AREA(J)
12     CONTINUE
C      AN  ....... NORMAL X Y Z ( ELEMENT # , TRIANGLE # )
      RETURN
      END
```

COMMERCIAL LIQUIFIED PETROLEUM GAS CYLINDERS

CAPACITY 26.2 LITRES

1) SCOPE

This standard specifies the minimum requirements of production, performance, and safety for valves and cylinders (Capacity 26.2 Litres). Cylinders are to be filled with maximum quantity of 12.5 Kg. of commercial liquified petroleum gases (L.P.G.) having a maximum pressure of 6 Kg/cm^2 at 50 C^o. Cylinders are to be used for domestic purposes and should be adequate for handling and refilling.

This standard covers the required testing procedures to realize above requirements.

2) TERMINOLIGY

2.1- Capacity

The volume is identified by the amount of water which fills the interior of the cylinder at a temperature of 15.5^{o}C. The cylinder is to be designed in such a manner to take up 26.2 Litres, but not to exceed 27.0 Litres with allowable tolarance of + 0.25 Litre of the adopted capacity. In no case shall the capacity be less than 26.2 Litres.

3) CONSTRUCTION

3.1- Finishing

The finishing is to be at such a degree of perfection, so that the surface may be smooth and clear from visual voids such as roughness and unpainted parts which may accelerate the breakage of paints. Scratches, cracks and laminations should not be concealed by paste or any such like material.

2/...

(The internal surface must be perfect and free from visual defects such as roughness and flakes).

3.2- Strength and stability

The cylinder should be well constructed, and stable when placed on a vertical axis. The cylinder should stand breakage, cracking, or permanent deformation if subjected to conditions mentioned below (Para 8.0).

3.3- Gas leakage

Gas should not leak out of the cylinder under any circumstances. The cylinder should pass all the tests mentioned in Para 7. of this standard.

3.4- Cylinder Parts

The cylinder consists of the following main parts: (See Fig. 1)

a) Cylinder valve.
b) The body.
c) Valve boss.
d) Foot ring.
e) Valve gaurd.

3.5- Cylinder Valve

3.5.1-Valve Material

The valve should be fabricated from hot pressed brass with valve rating of 67 Kg/cm^2. DIN 47

3.5.2- The valve inlet and outlet should be threaded as shown in Fig. (2).

3.5.3-General requirements for cylinder valves:

a) Each valve should be provided with a suitable closure nut to fit the outlet.
b) The valve hand wheel shall be embossed with the words OPEN and CLOSE and suitable direction arrows.
c) The valve should withstand a differantial filling pressure of 6 Kg/cm^2.
d) The valve should withstand a boxed-in-line pressure of 16 Kg/cm^2.

e) The valve should allow the entry of 12.5 Kg. of (L.P.G.) in a time limit of 45 seconds.

f) The valve construction should be of repairable type.

3.6- The Body

3.6.1-Construction of The Body

The cylinder body should be fabricated from two cold deep drawn halves, having an overlap joint assembled by submerged arc welding, The welding should be uniform. free from cracks, voids, slag inclusion, burnt oxides, under cuts or over layers, or any other defects. The welding should pass the tests mentioned in this standard, (See Fig. 5).

3.6.2- Body Material

The body should be fabricated from killed steel plates prepared by one of the following processes:

a) Open hearth.

b) Basic oxygen.

c) Electric furnace.

The plates should be hot drawn,free from defects such as cracks and laminations, plates should pass all mechanical and chemical tests mentioned in this standard.

3.6.3-Body Thickness

The thickness of the body, after drawing, at any point, should not be less than 3mm. (See Fig. 1).

3.7- Valve Boss

3.7.1-Fabrication of Valve Boss

The valve boss should be machined and threaded to fit the valve, as indicated in (Fig. 2). The valve boss should be welded in its appropriate location by submerged arc welding.

4/...

The welding should conform with related standard specifications.

3.7.2-Valve Boss Material

The valve boss should be fabricated from steel having same properties as that of the body.

3.8- Footring

3.8.1-Fabrication of Footring

The cylinder should be fitted with a footring to insure stability when resting on a horizontal plain. The footring should be co-axially arc welded to the cylinder bottom. The footring should have vent holes at the top and along the circumferance. The bottom edge of the footring should be rounded inwards so that it does not have a sharp edge. The footring should have drain holes as shown in (Fig. 3).

3.8.2-Footring Material

The footring should be fabricated from steel having the same properties as that of the body. The thickness of the footring should not be less than that of the body.

3.9- Valve Gaurd

3.9.1-Valve Gaurd Fabrication

The valve gaurd should have a cylindrical shape. It should protect the valve during transportation and handling. It should have side openings. It should be arc welded on the body as shown in (Fig. 4a & b).

3.9.2-Valve Gaurd Material

The valve gaurd should be fabricated from steel having the same properties as that of the body. The thickness of the valve gaurd should not be less than that of the body.

5/...

3.9.3-Valve Gaurd Handle

The handle should be formed as an integral part of the valve gaurd, or in a form of a round steel bar electrically welded and designed to insure comfortable handling.

3.10- After all welding operations have been completed, each cylinder should be properly heat treated to conform with all mechanical properties indicated in para 3.12.

3.11- Allawable tolerances of cylinder dimensions

3.11.1-The cylinder should be uniform in shape, free from cracks, irregularities or welding defects. The vertical axis of the footring and that of the body should not deviate more than one degree when placed on a horizontal plane.

3.11.2-Precision should be considered when joining the two halves of the cylinder body, such that the axis of the two halves coincide, with a maximum allowable tolerance of 10% of body thickness.

3.11.3-The horizontal cross-section of the cylinder should be circular and uniform with a maximum allowable tolerance of 1% of the outside diameter.

3.12- Mechanical Properties And Chemical Composition of Cylinder Material.

Mechanical properties and chemical composition of the cylinder material should conform with the following table.

6/...

MECHANICAL PROPERTIES			% OF CHEMICAL COMPOSITION					
Tensile Stress Kg/mm^2	Yield Stress Kg/mm^2	% Elongation* (Min)	S (Max)	P (Max)	S + P (Max)	C. (Max)	Si. (Max)	Mn. (Max)
35 to 45	22 to 26	25	0.045	0.045	0.09	0.20	0.30	0.75

* Gauge Length = 5.65 $\sqrt{\text{Cross Section}}$

4) PAINT

4.1- The cylinder should be painted with a suitable paint to protect it from corrosion or rust.

4.2- Preparing the cylinder surface for paint, all flakes and any signs of rust or soil should be removed before painting by any one of the following methods:

4.2.1- Treatment in Acidic bath (Pickling) according to any recognized standard specifications.

4.2.2- Sand blasting according to any recognized standard specifications. ?

4.3- Type of Paint

After cleaning, each cylinder should be painted with a suitable paint that fulfil the following requirements:

a) The paint should not change in colour in any part of the cylinder, sticks, indicate signs of deterioration or smells.

b) The final finish of the paint should be bright in colour and reflects heat.

c) The paint should not be affected by light petroleum liquids.

7/...

5) INFORMATION AND MARKING

5.1- On The Footring

1. Manufacturer name and trade mark.
2. Serial number of the cylinder.
3. Tare weight in Kg.
4. Volumatric capacity in litre.
5. Date of manufacture.
6. Test pressure (Kg/cm^2) and test date.
7. Maximum working pressure (Kg/cm^2).
8. Inspection authority stamp.
9. Type and weight of liquified gas.

5.2- On The Valve Flange

1. Serial number of cylinder if possible.

6) STORAGE

6.1- The Empty Cylinders should be stored after fitting the valve and closing it, and screwing the valve closure nut. In case the cylinder is stored without a valve, the inlet should be closed by a suitable valve closure nut so as to prevent humidity effects. The cylinder should not be subjected to any stresses during storage or transport.

6.2- The cylinder should be designed to with-stand pressure resulting from the vapour pressure of the contents when stored in the sun at a temperature of 70°C.

7) GENERAL REQUIREMENTS

7.1- This Part Explains the general requirements for the raw materials and construction of cylinders. That is to insure safety, strength, and function under normal operating conditions.

7.2- Cylinder parts should be inspected visually after carrying out the tests mentioned in this standard to insure safe handling.

8/...

7.3- Additional Tests

In case of doubt in raw materials quality, construction, or if the recognized authority in the Ministry of National Economy suspects changes in the cylinders quality, its strength, or its safe use in handling; the additional necessary tests may be conducted to insure that cylinders are free from mentioned defects.

7.4- Sampling For Destructive Tests

One cylinder should be taken at random from each lot of 200 or less. The destructive tests which are mentioned in this standard, should be carried out on this sample.

7.5- Cylinders Volumetric Capacity, surface conditions, up rightness when placed on a horizontal plane, and welding factor, should conform with this standard.

7.6- The mentioned tests in (para. 8) should be carried out after complete construction and before painting cylinders.

7.7- Acceptance and Rejection

7.7.1- In case any cylinder of the sample fails any of the tests, the tests should be carried out on twice the number of the initial sample. The lot will be accepted if the second sample fulfils conditions indicated in these tests. The lot will be returned to the producer or importer for repairs if any cylinder of the second sample does not fulfil any of the conditions indicated in these repeated tests.

9/...

7.7.2- In case the lot of cylinders are returned to the producer or importer for repairs, the tests should now be carried out on three times the number of initial sample. If any cylinder of this sample fails one of the conditions indicated in these tests, the lot is to be finally rejected

7.7.3- The lot should be finally rejected in case any of the samples fails one of the following tests:

7.7.3.1- Raw materials in para. 3.12

7.7.3.2- Hydrostatic test according to para. 8.2

7.7.3.3- Bursting test according to para. 8.4

8) TESTS

8.1- Mechanical Tests

8.1.1- Tensile and Bending Tests

Tensile and bending tests should be carried out on eight test specimens to be taken from each half of the cylinder as shown in Fig. 6. Test specimens should be prepared according to relevant standards.

The tested specimens should satisfy conditions indicated in para. 3.12

10/...

welding - single

8.1.2- Test for Welded Joints of cylinder

Tensile test, bending test and nick break test are to be conducted for two test specimens cut out from the circumferential cylinders welds as illustrated in Fig. 6. Test specimens are to be prepared in accordance with standard specifications of mechanical testing of welded joints.

The results of these tests should conform with the standard specifications of welded joints.

8.2- Hydrostatic Test

All cylinders should be hydrostatically tested under a pressure of 30 Kg/cm^2. for a minimum time of 30 Sec. The cylinders should withstand this pressure without indication of any cracks. After testing, cylinders should be dried by compressed air.

8.3- Leakage Tests

After attaching the cylinder valve, each cylinder should be subjected to an air test of 8 Kg/cm^2 during which it should be submerged in water for one minute or spraying the cylinder with soap foam, especially the welded areas. No leakage should be observed from the welds or the body.

8.4- Bursting Test

Bursting test should be carried out on manufactured or imported cylinders by exposing cylinders to a gradually increasing hydraulic pressure until the cylinders burst. No cracks should occur before a pressure of 45 Kg/cm.. Failure, when occuring should be at least 6mm. away from the weld region.

8.5- Cylinder Dimensions Test

8.5.1- Cylinder thickness test

Cylinder steel thickness should be measured in various positions. The thickness should not be less than 3mm.

All dimensions of the cylinder should comply with this standard.

8.5.2- Thread test
The cylinder inlet valve internal thread should be measured by thread standard gauges especially for this purpose. Threads should comply with this standard. See Fig. (2). All cylinders with threads that do not comply with this standard should be rejected.

8.6- Chemical Test
Chemical analysis should be carried out on the steel in various parts of the cylinder and of the valve flange. The percentages of sulfur and phosphorous or their combinations should not exceed those indicated in para. 3. 12.

9) INSPECTION, ACCEPTANCE, AND REJECTION

9.1- The manufacturer or importers should provide all the facilities to the representative of the Ministry of National Economy/Directorate of Standards & Specifications, or to a recognized authority which is authorized to test or inspect the various testing procedures to insure that raw materials or manufactured products comply with this standard.

9.2- Inspecting authority certificate
Every lot of cylinders should be accompanied with a certificate from a recognized inspecting authority indicating that the lot has been inspected by the authority or under authority supervision. The certificate should include the following:

9.2.1- A mill certificate: For every cast of steel indicating its chemical composition and its mechanical properties.

9.2.2- Drawings showing the dimensions of the cylinder, valve gaurd, valve, and valve flange.

12/...

9.2.3- Weight of valve, and the tare weight of the cylinder with the valve.

9.2.4- The volumatric capacity of the cylinder in litres and variation in capacity between cylinders for each batch.

9.2.5- Cylinders manufacturing procedures.

9.2.6- The chemical analysis of the steel plates supported by certificates specifying at least composition of Carbon, Manganese, Silicon, Phosphorous and Sulphur.

9.2.7- Certificate indicating results of tests for mechanical properties.

9.2.8- Thickness of the steel plate used in manufacture.

9.2.9- Description of welding methods used in manufacture.

9.2.10- Methods of welding inspection and weld tensile and bend test results.

9.2.11- Normalizing temperature and duration.

9.2.12- Thickness of base coat and final coating.

9.2.13- Results of hydraulic pressure testing of each cylinder.

9.2.14- Results of Bursting tests.

9.2.15- Cylinder valve specifications.

9.2.16- Serial number of the cylinders.

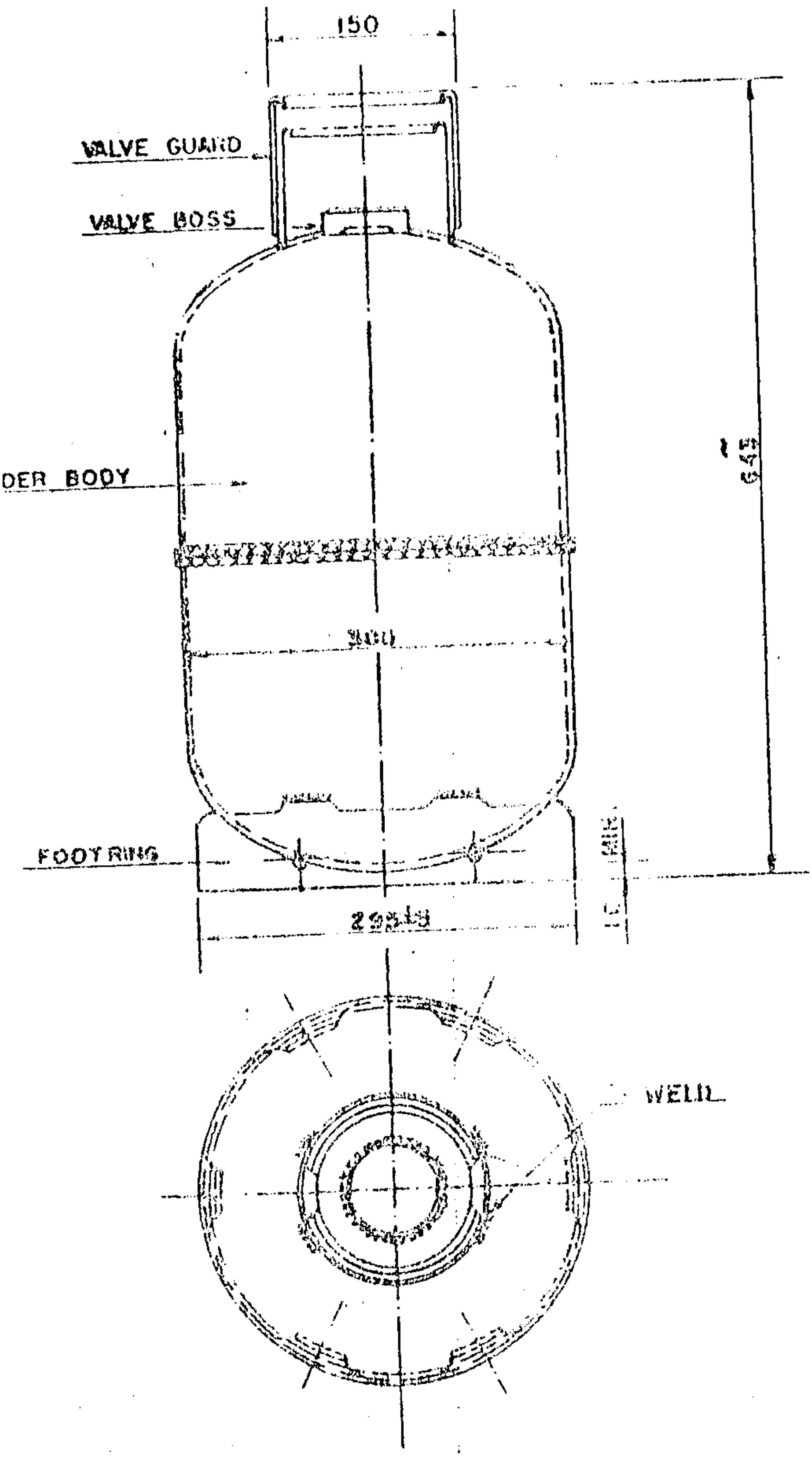


FIG 1

ملخص

فحص اسطوانات الغاز المسال في الاردن

اعداد

باسم محمد العمري

المشرف

الدكتور سعد الحبالي

المشرف المشارك

الدكتور عزام عوده

تهدف هذه الدراسه بشكل رئيس الى فحص اسطوانات الغاز المسال في الاردن في ظروف الاستعمالات العادية , بحيث درست ثلاث حالات رئيسية : اسطوانة تحت ضغط منخفض وضغط مرتفع، واسطوانة تحت حمل ستاتيكي واسطوانة تحت حمل ديناميكي ناتج عن ضرب الاسطوانة.

درست الاسطوانات باستخدام تحليل العنصر المحدود باســتعمال 5180 درجة تحرر و 468 عنصرا قشريا بحيث يستخدم تحليل ذو مرونة ولدونة للــهيكل ذو القشرة الرقيقة للاسطوانة و بعدها استعرض وقورن ببعض النتــائج العمليــة والتحليلية.

راينا بعدها كيف يتغير الاجهاد والانحراف القطري مع الضغط والحمل الستاتيكي ، كما تم احتســاب معــامل الاصطــدام والقيــم القصــوى للاجــهاد والانحراف القطري لاسطوانه تحت حمل ديناميكي ، ووضعت طريقــه لاحتســاب الاجهاد و الانحراف القطري المتبقي. بالاضافه لذلك تمت دراسه تاثير التــاكل تحت الظروف الجويه على الاجهادات في الاسطوانه.