

**EXPERIMENTAL STRAIN ANALYSIS OF THE HIGH PRESSURE
STRAIN GAUGE PRESSURE TRANSDUCER AND VERIFICATION
BY USING FINITE ELEMENT METHOD**

**T.C. YÜKSEKÖĞRETİM KURULU
DOKÜMANTASYON MERKEZİ**

**Ph. D. Thesis by
M. Haluk ORHAN, M.Sc.
(509952026)**

104250

Date of submission : 31 January 2001

Date of defence examination : 22 May 2001

Supervisor (Chairman) : Prof. Dr. Galip TEPEHAN

Members of the Examining Committee Prof. Dr. Fikret HACIYEV (TÜBİTAK-UEKAE)

Prof. Dr. Hüseyin GÜVEN (İ.T.Ü.)

Assoc. Prof. Dr. Nazmi POSTACIOĞLU (İ.T.Ü.)

Prof. Dr. Sevim AKYÜZ (İ.Ü.)

MAY 2001

**GERİNİM ÖLÇERLİ YÜKSEK BASINÇ DÖNÜŞTÜRÜCÜSÜNÜN
DENEYSEL GERİNİM ANALİZİ VE SONLU ELEMANLAR
METODU İLE DOĞRULANMASI**

DOKTORA TEZİ
Y. Müh. M. Haluk ORHAN
(509952026)

Tezin Enstitüye Verildiği Tarih : 31 Ocak 2001
Tezin Savunulduğu Tarih : 22 Mayıs 2001

Tez Danışmanı : Prof. Dr. Galip Tepahan

Diğer Jüri Üyeleri Prof. Dr. Fikret HACIYEV (TÜBİTAK-UEKAE)

Prof. Dr. Hüseyin GÜVEN (İ.T.Ü.)

Doç. Dr. Nazmi POSTACIOĞLU (İ.T.Ü.)

Prof. Dr. Sevim AKYÜZ (İ.Ü.)

MAYIS 2001

PREFACE

The scope of this study is a review of the studies on the pressure transducers based on the cylindrical vessel theory and extension of these studies by utilizing Finite Element Method to optimize the design parameters prior to the construction of the pressure transducer prototypes. Hopefully, using FEM the results for the recently designed and constructed prototypes presented in this thesis will be helpful for both practical static pressure measurements and appropriate design.

I am pleased to acknowledge my debt to several persons and wish to express my gratitude to them for their valuable cooperation and help; to my research group colleagues Çetin Doğan, Sinan Fank, Savaş Acak and to my pressure laboratory team at the TÜBİTAK-Ulusal Metroloji Enstitüsü, without whom the knowledge and the experience I built up during many years could not have been acquired, and to Dr. Gianfranco Molinar, Prof. Roland Wisniewski for revisions of and various suggestions for my paper printed in the journal "Measurement Science and Technology".

I would like to thank my academic advisor Prof. Dr. Galip Tepehan who always encouraged and supported me for experimental studies and to Dr. Hikmet Kocabaş whom assisted me to use Finite Element Method software at the Mechanical Engineering Department of Istanbul Technical University.

I express my very special thanks to Dr. Tarık Öğüt, Dr. Zeki Erman and Halil Bilal at Figes Ltd. for their great help by making ANSYS available for me, also to Reinhard Kaufmann at Measurements Group of Vishay for his great support about strain gauge technology, making their facility and equipment available to me.

Last but not least special thanks are due to my lovely wife Denizhan for her indispensable help in preparation of every stage of this document and to my mother, father and sister to whom this study is dedicated.

May, 2001

M. Haluk ORHAN

CONTENTS

ACRONYMS	v
LIST OF TABLES	vi
LIST OF FIGURES	vii
LIST OF SYMBOLS	ix
SUMMARY	xi
ÖZET.....	xiii
1. INTRODUCTION.....	1
2. PRESSURE METROLOGY	3
2.1. Realization of the Pressure Unit	3
2.1.1. Pressure units and conversion factors.....	4
2.1.2. Pressure modes	5
2.2. Primary Pressure Standards	7
2.2.1. Pressure balances (dead weight testers)	7
2.3. Pressure Measuring Instruments.....	10
2.3.1. Mechanical deformation instruments.....	13
2.3.2. Strain gauge pressure transducers.....	13
2.3.3. Calibration method of the pressure transducers.....	14
2.3.4. The determination of the calibration uncertainty.....	16
2.3.4.1. Repeatability.....	17
2.3.4.2. Hysteresis	17
2.3.4.3. Linearity	17
2.3.4.4. Zero drift error.....	18
2.3.4.5. Resolution.....	18
3. THE DESIGN AND THE CONSTRUCTION OF THE PRESSURE	
TRANSDUCER PROTOTYPES	20
3.1. Investigation of the Design by the Measurement Method	20
3.1.1. An introduction to strain measurement.....	20
3.1.2. An introduction to strain gauge technology	21
3.1.3. The relation between the change in resistance and unit deformation	22
3.1.4. The basics of Wheatstone bridge used in strain gauge applications.....	24
3.2. Construction of the Pressure Transducer Prototypes	29
3.2.1. The Operating Principle of the Transducer.....	29
3.2.1.1. Thick wall cylindrical vessel theory.....	29
3.2.2. The design parameters of the transducer prototype	33
3.2.3. The selection and the properties of the materials	35

3.2.4. The selection criteria of the strain gauges	36
3.2.4.1. The strain gauges used for the investigated pressure transducers ..	37
3.2.5. The bonding procedure of the strain gauges	38
3.2.5.1. Surface preparation for strain gauge bonding	38
3.2.5.2. Gauge alignment.....	39
3.2.5.3. Gauge preparation	40
3.2.5.4. Adhesives and adhesive application.....	41
3.2.5.5. Clamping	43
3.2.5.6. Curing and postcuring	44
3.2.5.7. Inspection of bonded strain gauges	45
3.2.5.8. Soldering and removal of flux.....	47
3.2.5.9. Protective coatings	49
3.3. Three Dimensional Strain Analysis of the Prototype by the Finite	
Element Method (FEM).....	49
3.3.1. The basics of the FEM	50
3.3.2. The model used for the FEM analysis	51
3.3.3. The element used for the FEM analysis.....	53
3.3.4. The FEM analysis results.....	53
4. EXPERIMENTAL STRAIN GAUGE MEASUREMENTS	57
4.1. Measurement Setup.....	57
4.2. Other Details of the Measurement.....	59
4.3. The Metrological Characterization Results of the Manufactured Pressure	
Transducers	59
5. CONCLUSIONS AND DISCUSSION.....	65
REFERENCES.....	69
BIOGRAPHY OF THE AUTHOR	71

ACRONYMS

AISI	: American Iron and Steel Institute
CGPM	: Conférence Générale des Poids et Mesures
FEM	: Finite Element Method
FS	: Full Scale
PT	: Pressure Transducer
SI	: Système International d'unités
UME	: Ulusal Metroloji Enstitüsü



LIST OF TABLES

	<u>Page No</u>
Table 2.1. Pressure units and conversion factors	4
Table 2.2. Proposed probability distribution of the uncertainty components.....	17
Table 3.1. Properties of the materials used for the active cylinder and the coupling cup	36
Table 3.2. The dimensional properties of the selected strain gauge.....	38
Table 4.1. Calculated average hysteresis error values of UMEPTIn-01	60
Table 4.2. Calculated repeatability error values of UMEPT4340-01	61
Table 5.1. Comparison of the average measured output values with the average of derived FEM results of the UMEPT4340-01	65

LIST OF FIGURES

	<u>Page No</u>
Figure 2.1 : Pressure modes	6
Figure 2.2 : Pressure Balance. C: Cylinder, M: Masses, P: Piston	8
Figure 2.3 : Classification of pressure measurement techniques	12
Figure 2.4 : Pressure range and common instruments	12
Figure 2.5 : Strain gauge pressure transducer	14
Figure 2.6 : Bonded foil strain gauge pressure transducer (Courtesy Sensotech Inc.)	14
Figure 2.7 : Calibration setup of the pressure transducer.....	16
Figure 2.8 : a-Terminal linearity, b-Zero-based linearity, c-Best straight line linearity	18
Figure 3.1 : Typical stress-strain diagram for a metal.....	21
Figure 3.2 : Strain gauge sensing	23
Figure 3.3 : Principle of the voltage-fed Wheatstone bridge circuit	24
Figure 3.4 : Forms of the Wheatstone bridge circuit used in strain gauge techniques a) quarter bridge b) half bridge c) double quarter or diagonal bridge d) full bridge.....	27
Figure 3.5 : Characteristic of the Wheatstone bridge circuit in the “quarter bridge” operating mode according to equation (3.11) and the comparison characteristic based on the approximation formula (3.12)	29
Figure 3.6 : Section of thick wall cylindrical vessel with boundary conditions	31
Figure 3.7 : Schematic cross-section diagrams of the pressure transducer: 1. Pressure plug, 2. Outer cylinder, 3. Coupling cup, 4. Active element, 5. O-ring, 6. Strain gauge, 7. Blind plug, 8. Flattened part of the active element	34
Figure 3.8 : Placing a Mylar tape on the strain gauge.....	42
Figure 3.9 : Application of the adhesive	42
Figure 3.10 : Clamping components	43
Figure 3.11 : Spring clamp.....	44
Figure 3.12 : Protection of the strain gauge before the soldering process	47
Figure 3.13 : Comparison of flux removal solvents (drift versus time).....	49
Figure 3.14 : FEM model of the active cylinder of the investigated pressure transducer with the applied boundary conditions.....	52
Figure 3.15 : Solid 45, 3-D, 8-Node structural solid FEM element.....	53
Figure 3.16 : FEM strain results in x-direction for pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p = 150$ MPa.....	54

Figure 3.17 :	FEM strain results in y-direction for pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p = 150$ MPa.....	55
Figure 3.18 :	FEM strain results in z-direction for pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p=150$ MPa.....	55
Figure 3.19 :	FEM strain results in z-direction for non-pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p=150$ MPa ...	56
Figure 4.1 :	Photograph of the AISI 4340 Steel made active cylinder inside the coupling cup	57
Figure 4.2 :	Measurement setup	58
Figure 4.3 :	Photograph of the measurement setup.....	58
Figure 4.4 :	The hysteresis error graph of the UMEPT4340-01	60
Figure 4.5 :	The repeatability error graph of the UMEPTIn-01	62
Figure 4.6 :	Linearity behavior of the UMEPT4340-01	63
Figure 4.7 :	Linearity behavior of the UMEPTIn-01	64
Figure 5.1 :	Comparison of the hysteresis error results of the transducer prototypes	67
Figure 5.2 :	Comparison of the repeatability error results of the transducer prototypes	67



LIST OF SYMBOLS

M	: Mass
L	: Length
T	: Time
ρ	: Density of the fluid
g	: Acceleration due to the gravity
h	: Height of the liquid column
p	: Pressure
A	: Area
P	: Piston
C	: Cylinder
p_e	: Gauge pressure measured at the bottom of the piston
m_i	: Individual mass value of each weight applied on the piston
ρ_a	: Density of air
ρ_{mi}	: Density of each weight
σ_l	: Surface tension of the liquid
c	: Circumference of the piston or its extension in the level where it emerges from oil
A_0	: Effective area of the piston cylinder assembly at null pressure and at 20°C temperature
λ	: Pressure distortion coefficient of the piston cylinder assembly
α_p, α_c	: Linear thermal expansion coefficients of the piston and the cylinder, respectively
t	: Measured temperature of the piston cylinder assembly during its use
$u_c(\text{tra})$	: Relative combined standard uncertainty
$U(\text{tra})$	: Relative expanded uncertainty
k_c	: Coverage factor
$u(\text{rep})$	: Uncertainty due to repeatability error
$u(\text{hys})$	: Uncertainty due to hysteresis error
$u(\text{int})$	: Uncertainty due to interpolation
$u(\text{zero})$	: Uncertainty due to zero drift error
$u(\text{res})$	: Uncertainty due to resolution
$U(\text{ref})$	: Uncertainty of the dead weight tester
U	: Uncertainty of the calibration
a_w	: Half-width of the input quantity
E	: Elasticity (Young) Modulus
σ	: Stress
ε	: Strain
ΔR	: Change in the resistance value
ΔL	: Change in the length of the resistor wire
R_0	: Initial resistance of the wire
L_0	: Initial length of the wire

k	: Gauge factor
V_s	: Supply voltage
R	: Resistance
R_1, R_2, R_3, R_4	: Wheatstone bridge resistors
R_6	: Voltage supply resistance
v_1, v_4	: Partial voltages
V_0	: Bridge output voltage
$\epsilon_1, \epsilon_2, \epsilon_3, \epsilon_4$	: Strains
r, θ	: Polar coordinates
ϕ	: Stress function as a function of r and θ
A_1, B_1, C_1, D_1	: Integration constants
σ_r	: Stress in radial direction (in polar coordinates)
σ_θ	: Stress in angular direction (in polar coordinates)
ϵ_θ	: Angular strain
a, b	: Inner and outer radii of the cylinder
p_i, p_o	: Uniform internal and external pressures
u	: Radial displacement
$\sigma_{\theta\max}$	: Maximum stress in angular direction (in polar coordinates)
e	: Distance between the centers of the bore and the external surface
W_{ap}	: The approximate ratio of the outer flattened surface of the cylinder to the inner cylindrical diameter
ν	: Poisson's ratio
α	: Linear thermal expansion coefficient ($^{\circ}\text{C}^{-1}$)
x, y, z	: Cartesian coordinates
S	: Surface
DMX	: Maximum displacement
SMN	: Minimum value of the plotted item
SMX	: Maximum value of the plotted item
EPTOX	: The plotted strain values in x-direction
EPTOY	: The plotted strain values in y-direction
EPTOZ	: The plotted strain values in z-direction
ϵ_a	: Axial strain
ϵ_c	: Circumferential strain
$pV_{\max}, pV_{\min}$	: Maximum and minimum electrical output values of the transducer

EXPERIMENTAL STRAIN ANALYSIS OF THE HIGH PRESSURE STRAIN GAUGE PRESSURE TRANSDUCER AND VERIFICATION BY USING A FINITE ELEMENT METHOD

SUMMARY

Broadly speaking, a transducer is a device, which transforms one type of energy into another. In this study, specific class of transducers was considered: devices, which translate an input of mechanical energy into equivalent electrical signals for measuring and/or controlling the input phenomena. Within the scope of this study, a high pressure strain gauge pressure transducer was designed and constructed.

At present, Finite Element Method analysis is the most powerful tool to evaluate and optimize stress distribution problems, especially in complicated designs. Considering the design of the constructed pressure transducer prototypes in this study, that the thick wall cylindrical vessel was modified by flattening the two surfaces of the active cylinder, the analytical approach was needed some assumptions to find the exact strain values on the area where the strain gauges were bonded. It was therefore decided that an existing design of a pressure transducer should be investigated using finite elements. Based on the findings of this investigation an improved version was designed. A total of eight specimens were constructed and two kinds of measuring elements were made of two different materials: AISI 4340 steel and Invar steel which work in the hydraulic gauge pressure range of up to 150 MPa and 100 MPa respectively. Prior to its use, AISI 4340 Steel was hardened for improving the hysteresis and creep effects.

As it was explained in the previous paragraphs, the Finite Element Method (FEM) was used in this study for the analysis of the strain distribution of a strain gauge pressure transducer for hydrostatic pressure measurements up to 150 MPa. The investigated pressure transducer, based on the “thick-wall cylindrical vessel” theory has a free steel active element. Pressure is applied to the inside and both open ends of this active element. The symmetric shape of the transducer and all the design parameters of the active element were selected to assure a symmetrical stress and strain distribution even at the maximum working pressure of the transducer. Thus, the FEM analysis was conducted by investigating one half of the element in three dimensions. This study presents the FEM output strain values on the area where the strain gauges were bonded. The validity of those values was established comparing them with the results obtained from the strain gauge measurements. The relative difference between the two value sets was determined as being lower than 13% of the full scale.

Although the relation between the output of the transducer and the applied pressure is often calculable, as a rule it can not be estimated with sufficient accuracy. Consequently, transducers need calibration against a primary or secondary standard, appropriately selected as to accuracy and the pressure range. The transducers were

calibrated using piston pressure balance working in the pressure range from 10 MPa to 150 MPa. Calibration is necessary for the determination not only of the basic relations between the output signal of the transducer and pressure, but also of all the metrological characteristics of a transducer such as; sensitivity, repeatability, linearity, hysteresis and zero drift. Using the calibration (measurement) results, the metrological specifications of total of eight specimens were evaluated and compared with each other. All these experimental results make evident that hardened AISI 4340 steel is an appropriate material for the pressure transducer application to measure gauge pressure of up to 150 MPa with an uncertainty of 0.1 % of the full scale. But this uncertainty value can be improved by a small modification in design or by a selection of more appropriate material and heat treatment, to reduce the repeatability and hysteresis errors of the device, which are the main parameters in the uncertainty evaluation. The results presented in this thesis will be helpful for practical static pressure measurements as well as for the appropriate design of this kind of pressure transducer using FEM.



GERİNİM ÖLÇERLİ YÜKSEK BASINÇ DÖNÜŞTÜRÜCÜSÜNÜN DENEYSEL GERİNİM ANALİZİ VE SONLU ELEMANLAR METODU KULLANILARAK SONUÇLARIN DOĞRULANMASI

ÖZET

Genel olarak söylenebilir ki, dönüştürücü bir tip enerjiyi başka bir enerjiye çeviren cihazdır. Bu çalışmada özel sınıfta bulunan dönüştürücüler göz önünde bulundurulmuştur. Bu cihazlar girdi büyüklüğünün ölçülmesi ve/veya kontrolü için, uygulanan mekanik enerjiyi, eşdeğer elektriksel enerjiye (sinyallere) dönüştürürler. Bu tez çalışması kapsamında, yüksek basınç ölçümünde kullanılacak olan bir basınç dönüştürücüsü tasarlanmış ve imal edilmiştir.

Günümüzde Sonlu Elemanlar Metodu analizi, özellikle karmaşık tasarımlarda stres dağılımı problemlerinin çözülmesi ve optimize edilmesinde kullanılan en güçlü araçtır. Kalın cidarlı silindirik tüpün, aktif elemanın iki yüzeyinin düzleştirilmesi ile modifiye edildiği bu çalışmadaki imal edilen basınç dönüştürücü prototiplerinin tasarımı göz önünde bulundurulduğunda, analitik yaklaşım, gerinim ölçerlerin yapılandırıldığı bölgedeki gerinim değerlerinin bulunması için bazı varsayımlara ihtiyaç duyar. Bu sebeple, basınç dönüştürücüsünün mevcut tasarımının sonlu elemanlarla incelenmesi gerektiğine karar verilmiştir. Elde edilen sonuçlar doğrultusunda tasarım iyileştirilmiştir. Toplam sekiz adet numune imal edilmiştir. İki farklı ölçme elemanı, iki farklı malzeme kullanılarak imal edilmiştir. Kullanılan malzemeler; 150 MPa bağıl basınç değerine kadar çalışan dönüştürücü için AISI 4340 çeliği ve 100 MPa bağıl basınç değerine kadar çalışan dönüştürücü için Invar çeliğidir. Kullanımdan önce AISI 4340 çeliğine histeresiz ve sürünme etkilerinin iyileştirilmesi için ısıt

Dönüştürücünün çıkış değerleri ile uygulanan basınç arasındaki ilişkinin çoğu zaman hesaplanabilir olmasına rağmen, bu ilişki yeterli doğrulukla belirlenemez. Bu nedenle, dönüştürücülerin, uygun doğrulukta ve ölçüm aralığında çalışan primer (birincil) veya sekonder (ikincil) standartlar kullanılarak kalibre edilmeleri gerekir. İmal edilen basınç dönüştürücüleri de 10 MPa ile 150 MPa basınç aralığında çalışan pistonlu basınç standardı kullanılarak kalibre edilmiştir. Kalibrasyon işlemi, elektriksel çıkış sinyali ile basınç arasındaki temel ilişkinin bulunması için gerekli olmasının yanı sıra, dönüştürücünün, hassasiyet, tekrarlanabilirlik, lineerite (doğrusallık), histerisiz ve sıfıra dönüş hatası gibi metrolojik karakteristiklerinin belirlenmesine de olanak sağlar. Elde edilen kalibrasyon (ölçüm) sonuçları kullanılarak, toplam sekiz adet numunenin metrolojik özellikleri belirlenmiş ve bu özellikler numuneler arasında karşılaştırılmıştır. Bütün deneysel sonuçlar göstermiştir ki, sertleştirilmiş AISI 4340 çeliği, 150 MPa hidrolik basınç değerine kadar çalışan ve, maksimum gösterge değeri cinsinden % 0.1 ölçüm belirsizliğine sahip bir basınç dönüştürücüsünün imalinde kullanılabilecek bir malzemedir. Bu belirsizlik değeri, tasarımda yapılacak küçük değişiklik ile veya daha uygun malzemenin ve ısıtma işleminin seçilmesi ile iyileştirilebilecektir. Bu çalışmada sunulan sonuçlar, statik basınç ölçümleri ile birlikte bu tip basınç dönüştürücüsünün tasarımının SEM yardımıyla gerçekleştirilmesi konusunda yardımcı olacaktır.

1. INTRODUCTION

On-site measurement and control of pressure, the strain gauge pressure transducer is a usual selection, due to its easy to use operation and relatively low cost. For a long time, pressure transducers with cylindrical elastic elements have been widely used in many applications.

Generally these transducers use an elastic element realized as a hollow cylinder. They are made of different materials (hard tool steel, tungsten carbide and ceramics) depending on the pressure range of their use. They have a blind end on one side while on the opposite side, the cylinder is rigidly fixed (through an appropriate mechanical fitting) to the apparatus where pressure is to be measured. In this condition it is difficult to achieve a very high accuracy of pressure measurements because of the indefinite state of stress at both ends. Consequently, analytical calculations can only provide an approximative prediction of the effects of deformation versus applied pressure. To improve the overall metrological characteristics (especially to obtain reduced hysteresis and optimum repeatability), many studies were carried out for the pressure transducers, which have cylindrical, free active elastic element. Some of these studies have been recently reviewed in [1].

The following approach would have to be adopted, in order to optimise the design by improving the stress distribution: Whilst maintaining the average strain level in the gauge area at a given pressure, the maximum stress in zones of stress concentrations would be reduced, and also the size of these zones if possible.

In addition to the analytical information given in [2] and to extend the scope of the previous studies, in this study, finite element method (FEM) was utilized to analyse the strain distribution on the similar type pressure transducer. For symmetry reasons, only one half of the sensor was modelled and meshed. These measures were necessary in order to simplify and speed up the process of meshing and furthermore to reduce the amount of memory needed to store and handle the information. This analysis is also very useful for evaluating or verifying a design by modelling its

behaviour prior to manufacture. The strain values obtained from the FEM analysis were compared with those obtained from the strain gauge experiments. To verify and optimise the presented model, a number of further models were generated in two and three dimensions, which included the analysis of the full part active element.

Material selection of the transducer generally depends on the capacity, design and environmental conditions. The magnitude of the hysteresis is mainly affected by the material and heat treatment. Creep like hysteresis, depends on material, heat treatment and some design aspects, but also very much on the type of the strain gauge and bonding details. This dependence allows one to compensate for creep by selecting a strain gauge layout corresponding to the specific needs of the sensor in question [3]. Considering all these facts, two different materials were used for the cylindrical active element and its coupling cups. First material was the AISI 4340 steel, which is the best steel alloy for the transducer applications [4]. It was hardened by quenching from 845°C in room temperature oil and heated at 425°C and then cooled at room temperature [5] for improving the hysteresis and creep effects. After the heat treatment the hardness of the material was measured and found to be 45-50 HRC. Invar steel was selected as the second material aiming to reduce the temperature effects due to the high pressurization by taking advantage of its very low thermal expansion coefficient that is almost one order of magnitude lower than the structural steel. Since Invar steel has lower hardness than AISI 4340 steel, the focus of this study was expanded to examine whether the hardness and the other elastic properties really affected the experimental data concerning hysteresis. The hardness of the Invar steel was also measured and found to be 125 HB.

Compared to previous designs, another difference was placing the O-rings on the active element instead of inside the coupling cup. Also, the analysis of the effects of the change in position of the O-ring on the strain and stress distribution on the active element was realized by using the FEM.

2. PRESSURE METROLOGY

The measurement of pressure and vacuum plays an extensive and important role in the modern world. The pressure generated by transforming water into steam largely powered the Industrial Revolution. The need to measure pressure, over wider ranges with increasing accuracy, has expanded ever since. Applications are found in industries as various as nuclear, power, gas, petrochemical, biological, pharmaceutical, meteorological, automotive, environmental, semi-conductor, optical, aerospace, defense, ventilation, filtration and process control in general. The validity of the measurements is essential for trade, efficiency, quality and safety [6].

2.1. Realization of the Pressure Unit

The International System of Units, known as the SI system, is the system of units adopted and recommended by the General Conference on Weights and Measures (CGPM). It is based on seven base quantities: Length, mass, time, electric current, thermodynamic temperature, amount of substance and luminous intensity.

Pressure is not a base quantity of SI system but a derived quantity, with dimensions of length, mass and time. This can be demonstrated by considering the two fundamental ways of measuring pressure: directly in terms of force-per-unit-area measurements and with liquid columns.

(i) Pressure is defined as force per unit area. This gives pressure the dimensions of $\text{mass} \times \text{length}/(\text{time}^2 \times \text{length}^2)$, which simplifies to $\text{mass}/(\text{length} \times \text{time}^2)$ or $\text{M.L}^{-1}.\text{T}^{-2}$.

(ii) The pressure at the bottom of a fluid column is calculated by multiplying together the density of the fluid, the acceleration due to gravity and the height of the column ($\rho \times g \times h$). Since density is mass/volume it has dimensions $\text{mass}/\text{length}^3$. Acceleration has dimensions $\text{length}/\text{time}^2$, so the product $\rho \times g \times h$ simplifies to $\text{M.L}^{-1}.\text{T}^{-2}$ and is dimensionally identical to the force/area calculation as in (i) above.

The SI unit of pressure is the Pascal, abbreviated to Pa, the name given to a pressure

of one Newton per square metre (N/m^2). In relation to everyday life, one pascal is a very small quantity, atmospheric pressure being roughly 100 000 Pa.

To avoid the use of cumbersome numbers, multiples of 10^3 and 0.001 are assigned prefixes so that, for example, 100 000 Pa (10^5 Pa) can be written as, 100 kPa or 0.1 MPa.

2.1.1. Pressure Units and Conversion Factors

The relationships between the Pascal and some other pressure units are shown in Table 2.1, but note that not all are, or can be, expressed exactly. The superscript roman numerals in the table refer to the notes, which follow it. Note also that the term standard atmosphere is not a pressure unit.

Table 2.1. Pressure units and conversion factors [6]

Unit	Symbol	Number of pascals
Pascal	Pa	1
bar	bar	1×10^5 (exactly)
millibar ⁽ⁱ⁾	mbar	100 (exactly)
hectopascal ⁽ⁱ⁾	hPa	100 (exactly)
conventional millimetre of mercury ^(ii,iii)	mmHg	133.322...
conventional inch of mercury ^(ii,iii)	inHg	3 386.39...
inch of water ^(iii,iv)	inH ₂ O	248.6... to 249.1...
torr ^(v)	torr	101 325/760 (exactly)
kilogram-force per square centimeter	kgf/cm ²	98 066.5 (exactly)
pound-force per square inch ^(vi)	lbf/in ²	6 894.76...

(i) **millibar and hectopascal:** Following the 8th Congress of the World Meteorological Organization, from 1 January 1986 the term hectopascal (hPa) is preferred to the numerically identical millibar (mbar) for meteorological purposes. This choice was made, despite the fact that hecto ($\times 100$) is not a preferred multiple in the SI system, to avoid having to change the numerical values on barometer scales.

(ii) **millimetres and inches of mercury:** The conventional millimetre of mercury (mmHg) and the conventional inch of mercury (inHg) are defined in terms of the pressure generated by a mercury column of unit length and of assigned density 13595.1 kg/in³ at 0 °C under standard gravity of 9.80665 m/s².

(iii) **manometric units:** The so-called ‘manometric’ unit definitions such as millimetres of mercury and inches of mercury depend on an assumed liquid density and acceleration due to gravity, assumptions which inherently limit knowledge of their relationship with the pascal. In order to encourage the demise of non-SI units,

whose definitions are becoming inadequate for the most precise measurement of pressure, there is international effort to exclude them from conversion tables or, in the meantime, restrict the precision of newly published conversion factors. It is strongly recommended that, wherever possible, all new applications of pressure measurement use the pascal, with multiples or sub-multiples as appropriate to the magnitude of the pressure values.

(iv) **inch of water:** The conventional inch of water (inH₂O) is defined in terms of the pressure generated by a water column of unit length and of assigned density 1000 kg/in³ whilst subject to standard gravity of 9.806 65 m/s². As with other 'manometric' unit definitions, this definition inherently limits knowledge of its relationship with the pascal. Furthermore, there are contradictory definitions in use, which lead to pressure values differing by up to 0.2%. This can cause serious errors and continued use of the unit is firmly discouraged.

(v) **torr:** The torr is defined as exactly 101 325/760 Pa - the '760' coming from the original and arbitrary definition of standard atmosphere. Its value differs from the conventional millimetre of mercury by about 1 part in 7 million.

(vi) **pound-force per square inch:** The correct abbreviation for pound-force per square inch is lbf/in². Many instruments using this unit are labelled psi, however, and the label is incorrectly described as meaning pounds-per-square inch - with the word force missing. This is a significant conceptual error as a pound is a mass, not a force.

(vii) **standard atmosphere:** The current definition of standard atmosphere (atm) is 101325.0 Pa exactly. It is still occasionally used in defining a reference environment, i.e. for specifying gas density, but it is not a pressure unit and should not be used to express pressure values.

2.1.2. Pressure Modes

If a vessel were to contain no molecules at all, the pressure would be zero. Pressures measured on the scale, which uses this zero value as its reference point, are said to be absolute pressures. Atmospheric pressure at the surface of the earth varies but is approximately 10⁵ Pa (1 000 mbar); this is 10⁵ Pa absolute pressure because it is expressed with respect to zero pressure - that is no molecules at all.

In everyday life, however, many applications of pressure are not so much dependent

on the absolute value of a pressure as the difference between it and the pressure of the atmosphere. A punctured car tire is said to have ‘no air in it’ and a connected pressure gauge would read zero whilst obviously still containing atmospheric air. Such a gauge is designed to measure pressure values expressed with respect to atmospheric pressure and thus indicates zero when its measurement port ‘merely’ contains molecules at atmospheric pressure. These measurements are commonly known as gauge-mode pressure measurements. Thus the difference between an absolute pressure value and a gauge pressure value is the variable value of atmospheric pressure:

$$\text{absolute pressure} = \text{gauge pressure} + \text{atmospheric pressure} \quad (2.1)$$

In some cases - such as engine manifold pressure measurements - pressure excursions below atmospheric pressure are required. This is sometimes known as a negative gauge pressure but it should be appreciated that the concept of a negative absolute pressure is meaningless.

In other applications, where knowledge of the pressure difference between two systems is needed, the reference pressure may not necessarily be either zero or atmospheric pressure but some other value. These are known as differential pressures. For example, the flow of gas along a pipeline depends on the pressure difference between the ends of the pipe and in practice both ends are usually at comparatively high pressures.

If serious errors are to be avoided, it is important when making pressure measurements to be clear which mode of measurement is being employed: absolute, gauge (positive or negative) or differential.

Pressure modes are illustrated in Figure 2.1; note that the reference line for gauge-mode measurements is not straight, illustrating the changeable nature of atmospheric pressure [6].

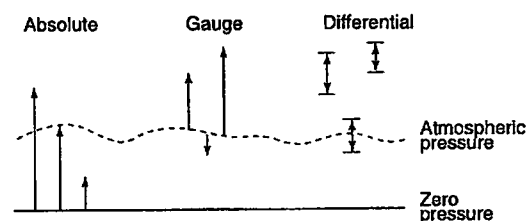


Figure 2.1. Pressure modes

2.2. Primary Pressure Standards

The primary standards used for pressure measurements are measuring systems that can be metrologically characterized in a complete and independent way with reference only to the basic units of the SI system. When pressure is defined as force per unit area (pressure balances) or as the height of a liquid column (interferometric manometers), pressure is, dimensionally, $p = M.L^{-1}.T^{-2}$; a primary pressure standard will thus involve the measurement of mass (M), length (L) and time (T).

The ideal primary standard for the pressure measurements operates correctly only when pressure is homogeneous in space, hydrostatic and time independent.

In this study, the primary pressure standard, pressure balance was used to test and calibrate the constructed transducer prototypes.

2.2.1. Pressure Balances (Dead Weight Testers)

Pressure balances are widely used for maintaining calculable pressures in a range extending from about 3 kPa (gas media, absolute- or gauge-mode) to 1 GPa (hydraulic, gauge-mode). Consisting essentially of finely machined pistons mounted vertically in very close-fitting cylinders, the internal pressure required to support the weight of the rotating piston and associated masses is calculated from the fundamental relationship between three quantities; mass, length and time:

$$\text{pressure} = \frac{\text{force}}{\text{area}} = \frac{m \times g}{A} \quad (2.2)$$

Where m is the mass of the piston and associated masses, g is the local value of acceleration due to gravity and A is the effective area of the piston-cylinder combination, taken to be the area bounded by the neutral surface in the fluid between the piston and the cylinder. Strictly the equation gives the pressure value above that of the air surrounding the top of the piston and masses. Thus if the apparatus is surrounded by the atmosphere, the pressure value calculated is a gauge pressure; if mounted in a vacuum chamber it is an absolute pressure.

There is a small gap between the piston and the cylinder and when the piston rotates in the cylinder. It is centralized by lateral forces in the pressure medium, thus avoiding contact between the piston and cylinder. If the gap between the piston and

cylinder is too small the piston will not spin freely and frictional forces will introduce significant errors. If the gap is too large the pressure fluid will leak away rapidly causing the piston to ‘fall’ rapidly within the cylinder. Note that it is not necessary for the piston to rotate; in some designs it is the cylinder that rotates around the piston.

Masses are generally loaded either directly on top of the piston or via an overhanging weight carrier; the latter lowers the center of gravity and can improve pressure stability. Non-magnetic stainless steel is the preferred material for masses and weight carriers as it is more stable than other materials, such as brass or cast iron.

Pressure balances are also known as piston gauges. When fitted with a means of pressure control, additional pressure ports and masses etc, the complete system is sometimes known as a dead-weight tester (see Figure 2.2). Not all practitioners differentiate between these terms and dead-weight tester is often used as a multi-purpose default description [6].

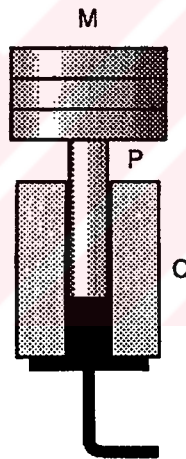


Figure 2.2. Pressure Balance. C: Cylinder, M: Masses, P: Piston

Pressure balances are amongst the most reproducible of pressure instruments and are used for calibrating a wide range of mechanical and electrical pressure gauges. Strictly they maintain a calculable pressure rather than measure it and hence cannot be used for most on-line measurement applications. When loaded with specific masses, they maintain just one calculable pressure. At low pressures relatively large diameter pistons are used but as the pressure increases smaller diameter pistons are used to prevent the number of masses from becoming unmanageably large.

Equation 2.2 is simplified; in practice it has to be expanded to take account of other

factors. The general definition of the pressure measured by the balance is obtained by analyzing the different components of the forces applied to the system. For the liquid operated pressure balance in gauge mode, the pressure definition is as follows [7]:

$$p_e = \frac{\sum m_i g \times \left(1 - \frac{\rho_a}{\rho_m}\right) + \sigma_l \times c}{A_0 \times (1 + \lambda p) \times (1 + \alpha_p + \alpha_c)(t - 20^\circ \text{C})} \quad (2.3)$$

where:

p_e is the gauge pressure measured at the bottom of the piston,

m_i is the individual mass value of each weight applied on the piston,

ρ_a is the density of air,

ρ_{mi} is the density of each weight,

σ_l is the surface tension of the liquid,

c is the circumference of the piston or its extension in the level where it emerges from oil,

A_0 is the effective area of the piston cylinder assembly at null pressure and at 20°C temperature,

λ is the pressure distortion coefficient of the piston cylinder assembly,

p is an approximate value of the measured pressure p_e . It can be the nominal value,

α_p, α_c are the linear thermal expansion coefficients of the piston and the cylinder,

t is the measured temperature of the piston cylinder assembly during its use.

The effective area changes with temperature and also with pressure - as the pressure increases the piston tends to taper inwards at its base whilst the cylinder tends to flare out although re-entrant designs reduce the effect to some extent by applying pressure to the outside of the cylinder, thus squeezing it inwards as pressure is increased. This helps to prevent the clearance between the piston and cylinder increasing with pressure, but the design can introduce other problems and it should not be assumed that re-entrant devices are necessarily metrologically superior. The effect of temperature changes can be corrected by knowing the expansion coefficient of the material from which the piston and cylinder are made but the distortion must be

determined by calibration. At very high pressures (100s of MPa) the uncertainty in measuring the distortion can dominate performance.

The total downward force acting on the piston has to take into account buoyancy effects of the piston and masses and also allowance has to be made for height differences between the bottom of the piston and the point at which the pressure is applied to a test artifact. The method used to make this allowance can depend on whether the pressure medium is a gas or a liquid. If it is a liquid, a hydrostatic head correction is applied. The same method may also be applied when working with a gas. With gas-operated dead-weight testers, however, it is possible to calculate a modified value of area (sometimes called a conventional area) that assumes the datum level to be somewhere more convenient than the bottom of the piston. This is possible because the value of the hydrostatic head correction is proportional to pressure, but it should be noted that the modified value is only valid for gases of a specified density - if other gas species or temperatures are used the correction will be invalid.

Most conventional styles of pressure balance use pistons and cylinders made of hardened and stabilized tool steels or tungsten carbide, which are relatively wear-resistant, as are some newer ceramic components.

2.3. Pressure Measuring Instruments

A number of quite different principles are utilized in pressure measuring instruments. Some of these are fundamental in character such as measuring the height of a liquid column of known density. One such example is a mercury barometer where the atmospheric pressure can be balanced against the column of mercury. An extension to this idea for use at higher pressures is the use of metal weights acting over a known area to provide the force rather than the weight of the liquid alone.

Often the pressure may be determined by measurement of the mechanical deformation of a sensing element that undergoes elastic deformation as the pressure difference across its surfaces changes. The mechanical deflection can be both implemented and sensed in a number of ways. One of the commonest types of moving mechanical elements is an elastic diaphragm. Another example is a Bourdon tube where the internal pressure forces the straightening of a curved tube.

Such mechanical deformation may be sensed in a number of ways: a series of mechanical levers to give a direct display of the deformation, resistance measurement in a strain gauge, capacitance measurement, change in frequency of a resonating element under tension or compression and so on.

When the pressure is very low and the mechanical deflection is therefore too small to be measured, indirect means are used that measure a physical property such as thermal conductivity, ionization or viscosity that is dependent on the number density of molecules.

Figure 2.3 shows one possible classification of some of the methods of measurement. The chart omits techniques that are employed only rarely. It should be noted that the chart shows only one possible representation of the wide range of pressure and vacuum measuring instruments and it would be equally valid to group techniques in other ways. For example, sensing techniques shown associated with diaphragms could also be used with other elastic deformation components [6].

Figure 2.4 shows the approximate pressure ranges of some common pressure and vacuum measuring devices. Once again, it should be noted that this representation is meant as a general guide and is not a rigorous classification [6].

The widely used Bourdon tube dial gauge has a Bourdon tube of elliptical cross-section that is bent to form a circular arc. When pressure is applied to the inside of the tube the tube tends to straighten out. This is amplified mechanically using gears and levers to operate a pointer. Bourdon tube dial gauges operate at pressures up to about 1.5 GPa.

The diaphragm dial gauge is similar to a Bourdon tube dial gauge except that the moving element is a diaphragm. Its movement is transmitted through a connecting rod to an amplifying lever and gears, which rotate a mechanical pointer.

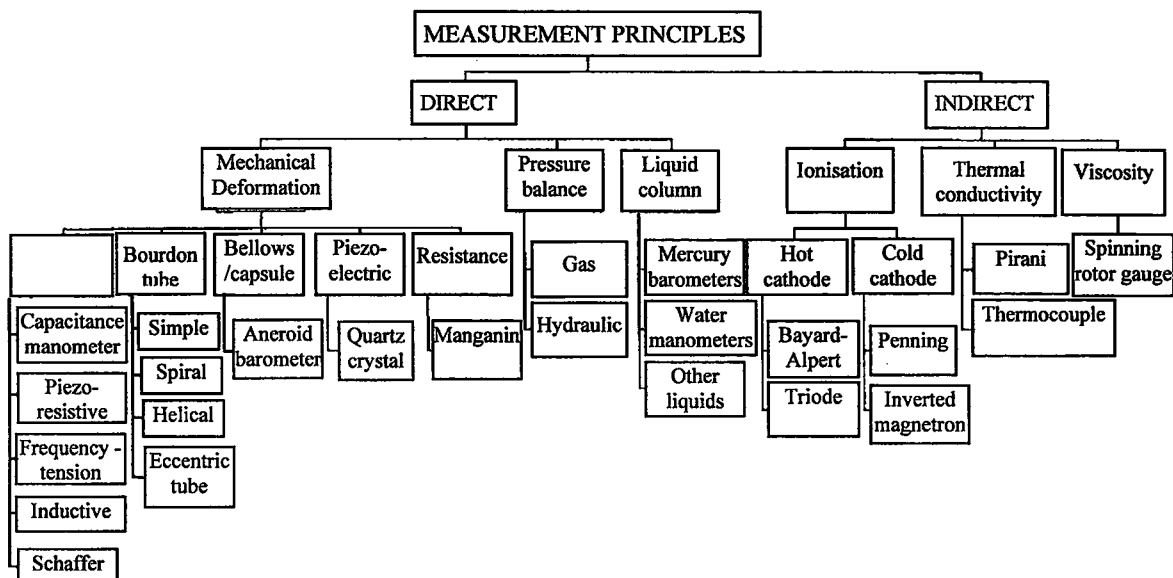


Figure 2.3. Classification of pressure measurement techniques

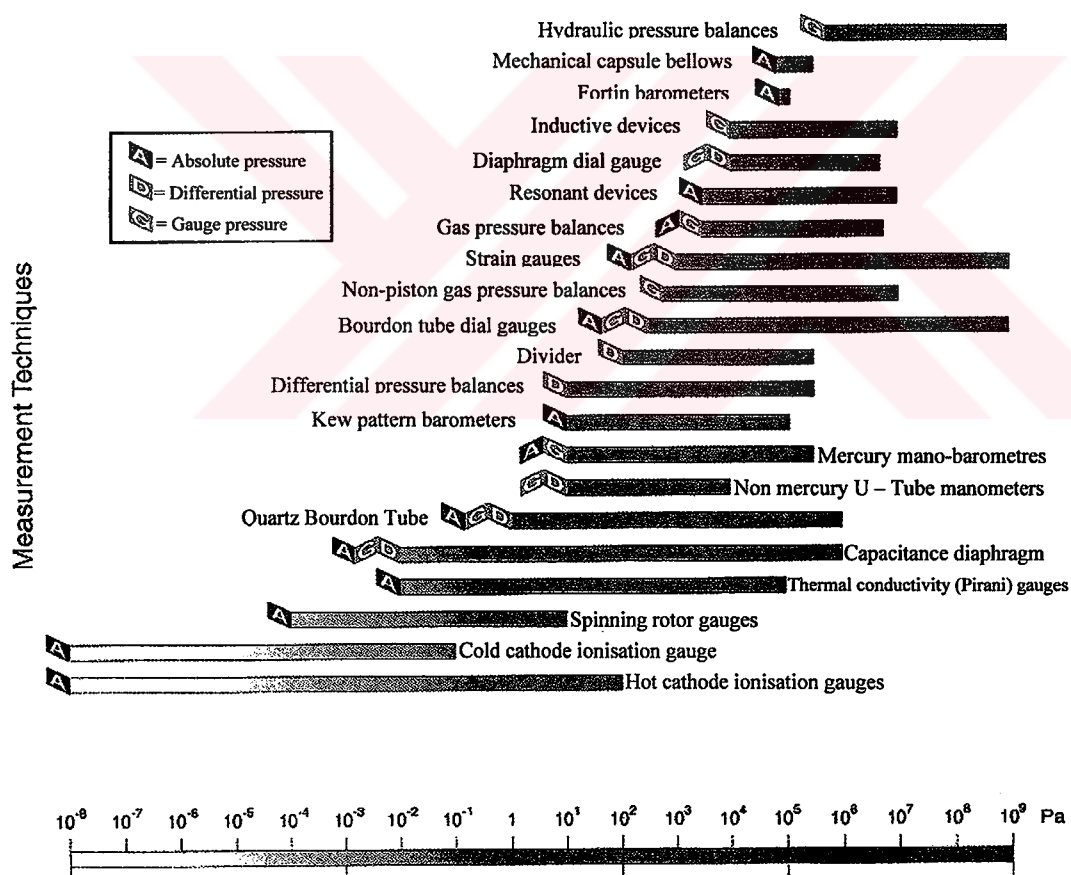


Figure 2.4. Pressure range and common instruments

2.3.1. Mechanical Deformation Instruments

When pressure is applied to a deforming element it will move. To produce a useful pressure sensor, the movement must be small enough to remain within the elastic limit of the material but large enough to be detected with sufficient resolution. Hence thin flexible components are used at lower pressure and thicker stiffer ones at higher pressures. There are several techniques in use to determine the extent of the deflection. These range from mechanical amplification producing a visible deflection of a pointer or light beam to electronic detection methods.

Cylinders are one of the mechanical deformation elements and their operation principle can be briefly explained as the following: When a cylindrical tube is pressurized from the inside, a hoop stress is imposed in the wall which gives rise to a strain at the outer surface of the tube. This can be measured by attaching resistive strain gauges to the outer surface of the tube. Commercial instruments of this kind are available up to 1 GPa. Instead of using a separate cylinder, a common practice is to fit strain gauges to the outer wall of a pressure vessel so that the vessel itself can be used as the deforming element.

Another form of cylinder-type pressure gauge suitable for use at high pressures can be made from a rod of circular cross-section that has a hole drilled down its length parallel to, but not on, the symmetry axis of the rod. When pressure is applied internally a movement of the end of the cylinder similar to that of the Bourdon tube is observed and it is detected in a similar way to Bourdon tube dial gauges.

2.3.2. Strain Gauge Pressure Transducers

Broadly speaking, a transducer is a device, which transforms one type of energy into another. In this study, specific class of transducers was considered: devices, which translate an input of mechanical energy into equivalent electrical signals for measuring and/or controlling the input phenomena.

A very common and relatively inexpensive device used to measure fluid pressure is the diaphragm strain gauge pressure transducer, sketched in Figure 2.5. The test pressure is applied to one side of the diaphragm, a reference pressure to the other side, and the deflection of the diaphragm is sensed with strain gauges. In the most common design, the reference pressure is atmospheric, so that the transducer measures gage pressure. In some cases the reference side of the transducer is sealed

and evacuated so that the transducer measures absolute pressure. Finally, both sides of the transducer can be connected to different test pressures so that the measurement is of differential pressure. Transducers for these three applications have slightly different construction details [8].

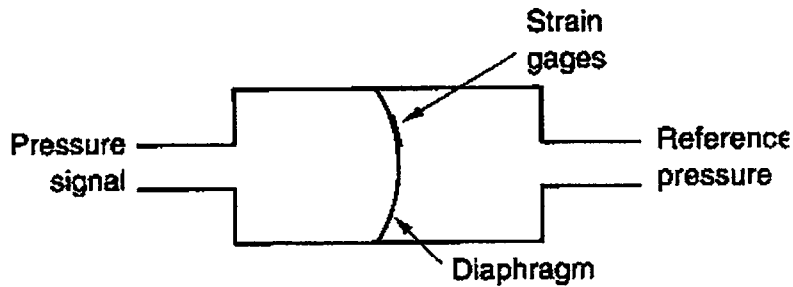


Figure 2.5. Strain gauge pressure transducer

Figure 2.6 shows a commercial type strain gauge pressure transducer.

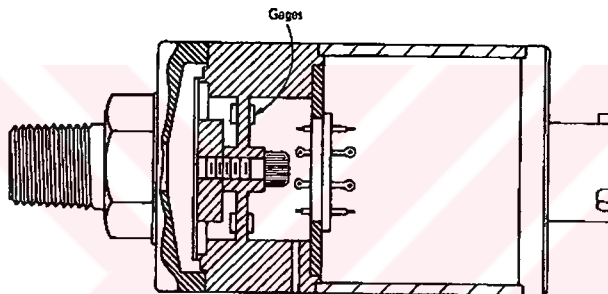


Figure 2.6. Bonded foil strain gauge pressure transducer (Courtesy Sensotech Inc.)

2.3.3. Calibration Method of the Pressure Transducers

In common with all measuring devices, pressure measuring instrumentation needs periodic calibration, primarily to monitor changes in performance. Formally, calibration is said to be “a set of operations that establish, under specified conditions, the relationship between the values of quantities indicated by a measuring instrument and the corresponding values realized by standards”. Thus a certificate of calibration commonly contains a list of pressure values determined by a pressure standard, a corresponding list of pressure values indicated by the instrument under test and a list of the differences between these values at each pressure.

Sometimes the word calibration is misused to describe the process of altering the performance of an instrument to ensure that the values it indicates are correct within specified limits. This is adjustment, not calibration, although the nature and

magnitude of the adjustment is often determined by a pre-adjustment calibration, sometimes known as an 'as found' calibration. When adjustment is undertaken, the full procedure is thus first calibration followed by adjustment and then second calibration. Results from the first calibration will also be needed if the device's reproducibility is to be estimated (essentially its change in characteristics over a long time interval) [6].

In performing a calibration, the following steps are necessary [9]:

1. Examine the construction of the instrument, and identify and list all the possible inputs.
2. Decide, as best you can, which of the inputs will be significant in the application for which the instrument is to be calibrated.
3. Procure apparatus that will allow you to vary all significant inputs over the ranges considered necessary.
4. By holding some inputs constant, varying others, and recording the output(s), develop the desired static input-output relations.

Generally, the calibration of the pressure transducers is realized by using the dead weight tester. A diagram of a dead-weight tester being used to calibrate a pressure transducer is shown in Figure 2.7.

After the first initial pressure loading test cycles, the calibration pressure points are being generated by using the dead weight tester (the calibration pressure points must be selected within the full range of the transducer and equally spaced). The pressure read-out values of the transducer are recorded at each generated pressure point. The calibration is always performed in increasing and decreasing pressure steps to calculate the hysteresis error. And the complete cycle must be repeated at least three times to determine the repeatability error with having a great amount of data for establishing a statistical base for the determination of uncertainty of the calibration and of the transducer. The transducer must be calibrated with its electronics including its display and its power supply as a complete set for the sake of the reliability of the calibration. The recommended calibration period for the transducers varies within 6 to 12 months. The calibration certificate should be supplied with an uncertainty statement corresponding to 95 % confidence level that is traceable to national primary standards.

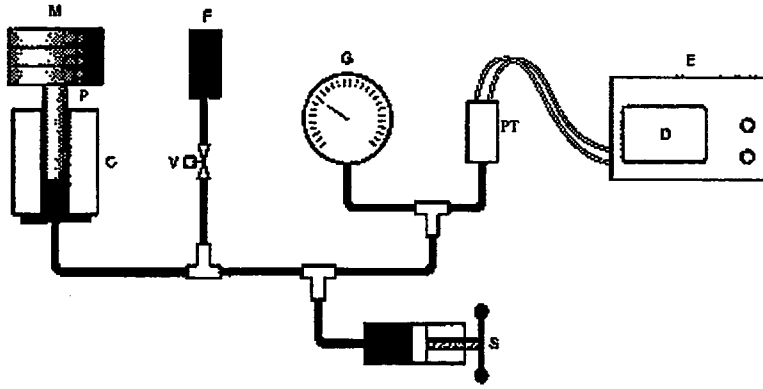


Figure 2.7. Calibration setup of the pressure transducer [5]

In Figure 2.7, the keys are: C, cylinder; D, display; E, electronics; F, fluid reservoir; G, gauge; M, masses; P, piston; S, screw-press; PT, pressure transducer; V, valve

2.3.4. The Determination of the Calibration Uncertainty

The calibration uncertainty always evaluated by using the calibration results. The uncertainty value of the transducer is generally given in terms of percentage of the full scale (calibrated span). After the relative variance of each pressure step, the relative combined standard uncertainty $u_c(\text{tra})$ and the relative expanded uncertainty $U(\text{tra})$ for $k_c = 2$ (% 95 confidence level) will be calculated by the following equations (2.4) and (2.5) for each pressure step [10].

$$u_c(\text{tra}) = \sqrt{u^2(\text{rep}) + u^2(\text{hys}) + u^2(\text{int}) + u^2(\text{zero}) + u^2(\text{res})} \quad (2.4)$$

$$U(\text{tra}) = k_c \times u_c(\text{tra}) \quad (2.5)$$

The relative uncertainty of the calibration (U) will be determined by considering the uncertainty of the dead weight tester $U(\text{ref})$ as follows:

$$U = \sqrt{U^2(\text{tra}) + U^2(\text{ref})} \quad (2.6)$$

The uncertainty value of the dead weight tester for the generated reference pressure must be taken from its latest certificate, which should be traceable to national standards.

Table 2.2 shows the proposed probability distribution of the uncertainty components.

Table 2.2. Proposed probability distribution of the uncertainty components (a_w =half-width of the input quantity)

Uncertainty components (Input quantities)	Probability distribution	Estimated relative variance
Repeatability	Rectangular	$u^2(\text{rep}) = a_w^2/3$
Hysteresis	Rectangular	$u^2(\text{hys}) = a_w^2/3$
Linearity (Interpolation) error	Rectangular	$u^2(\text{int}) = a_w^2/3$
Zero drift error	Rectangular	$u^2(\text{zero}) = a_w^2/3$
Resolution	Rectangular	$u^2(\text{res}) = a_w^2/3$

2.3.4.1. Repeatability

Repeatability is defined as the “closeness of the agreement between the results of successive measurements of the pressure carried out under the same conditions of measurement” where the conditions include the procedure, observer, instrument, conditions, location and a short period of time. It is normally quantified in terms of the standard deviation of a number of readings.

2.3.4.2. Hysteresis

Hysteresis is defined as the “property of a measuring instrument whereby its response to a given stimulus depends of the sequence of preceding stimuli”. For example, calibration corrections determined in a rising-pressures sequence may differ, in a reproducible fashion, from those determined in a falling-pressure sequence. Without knowledge of the pressure sequence, hysteresis enlarges the repeatability contribution to the uncertainty.

2.3.4.3. Linearity

The mathematical relationship between the pressure applied to a sensor and its ‘raw’ output is often not linear; and this can make it difficult to estimate pressure values from sensor output values which are somewhere between earlier calibration points. Most sensors employ signal conditioning in one form or another in an attempt to produce a linear relationship between applied pressure and sensor output.

Whatever signal conditioning is used, it will only approximate to the required pressure/output relationship and will probably change with time. Thus a graph of sensor output versus applied pressure will not be perfectly straight (or logarithmic). The differences between the measured pressure/output function and a straight line give a measure of non-linearity, sometimes expressed in the form of a worst-case difference such as, 'non-linearity = $\pm 0.1\%$ of calibrated span'. Three ways are commonly used to define the co-ordinates of the straight line, against which the sensors 'real' characteristic are compared (see Figure 2.8). Terminal linearity draws the straight line from the lower range value to the upper range value; zero-based linearity draws it from the lower range value with a slope chosen to ensure that the maximum positive and negative deviations are equal; best-straight-line linearity draws the line from a least-squares linear regression fit and does not necessarily intersect either end point.

The likely degree of non-linearity should be considered when considering the number and spacing of calibration points [6].

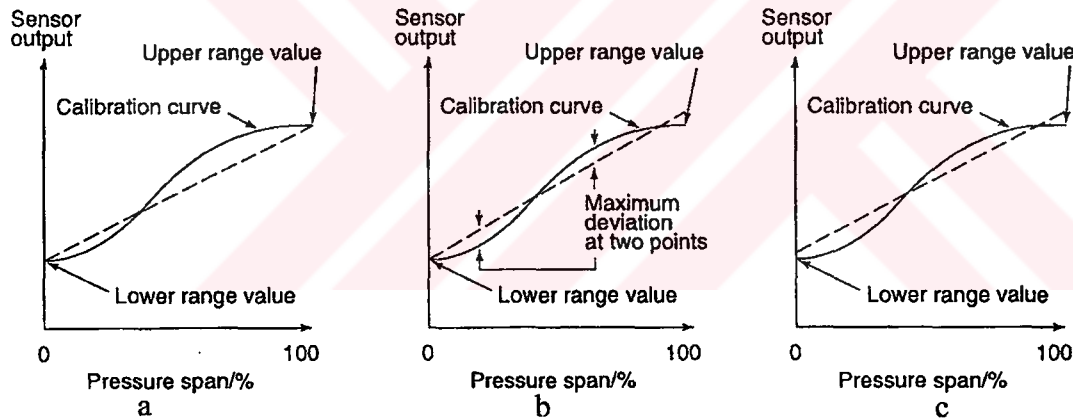


Figure 2.8. a-Terminal linearity, b-Zero-based linearity, c-Best straight line linearity

2.3.4.4. Zero Drift Error

Quantification of zero drift error normally involves comparing a set of before and after measurements and expressing the change in the zero value of the transducer. Drift is often expressed in terms of a change in pressure characteristic per unit time.

2.3.4.5. Resolution

Resolution is defined as the "smallest difference between indications of a displaying device that can be meaningfully distinguished". It is important not to confuse the resolution of a display alone with the resolution of a pressure measuring system,

which incorporates a display; the system will have less (poorer) resolution than the display alone. Also, resolution is not synonymous with uncertainty - if the uncertainty of a pressure measuring system has been properly evaluated it will include a component for resolution but there will be other factors, which will increase the uncertainty (reduce the accuracy) beyond that simply due to resolution.



3. THE DESIGN AND THE CONSTRUCTION OF THE PRESSURE TRANSDUCER PROTOTYPES

3.1. Investigation of the Design by the Measurement Method

3.1.1. An Introduction to Strain Measurement

Strain is a fundamental engineering phenomenon. It exists in all matter at all times, due either to external loads or to the weight of the matter itself. Strains vary in magnitude from atomic dimensions to distances easily discernible by the naked eye, depending upon the materials and loads involved. Scientists and engineers have worked for rise in the attempt to measure strain accurately, but only the last few decades have seen outstanding advancement in the art of strain measurement [11].

The measurement of strain has been of importance ever since the seventeenth century, when Hooke pointed out that for many common materials there is a constant ratio between stress and strain. The constant of proportionality between stress and strain implied in Hooke's law is known as the modulus of elasticity of the material, or, after the man who is credited with defining it, Young's Modulus. Mathematically, this can be expressed as:

$$E = \frac{\sigma}{\epsilon} \quad (3.1)$$

where E , ϵ and σ , are the modulus of elasticity, strain and stress, respectively. In a graphical representation given in Figure 3.1, E is the slope of the stress-strain curve.

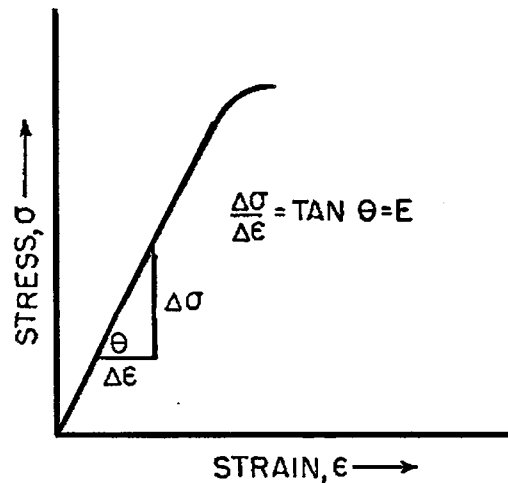


Figure 3.1. Typical stress-strain diagram for a metal

3.1.2. An Introduction to Strain Gauge Technology

In 1856 Lord Kelvin reported that the resistance of copper and iron wires increased when subjected to a tensile strain. This basic discovery has ultimately led to the development of the modern resistance foil strain gauge. The 'birth date' of the strain gauge is considered to be 1938, when Ruge and Simmons, working independently, each measured strains using bonded-wire strain gauges. The underlying concept of resistance strain gauges is very simple. In essence, an electrically conductive wire is securely bonded to a structure of interest, and the resistance of the wire is measured before and after the structure is loaded. Since the wire is firmly bonded to the structure strains induced in the structure are also induced in the wire. This results in a change in wire resistance, which serves as an indirect measure of the strain induced in the structure.

Although the underlying concept is simple, there are several sources of error, which may lead to erroneous strain measurements if not properly accounted for. These sources of error can be loosely grouped into the following six categories:

- (a) The wire must be firmly bonded to the structure, so that the deformation of the wire is an accurate reflection of the deformation of the structure.
- (b) The wire must not locally reinforce the structure, otherwise the strain field in the vicinity of the wire will be disturbed and an inaccurate measure of strain will be obtained.
- (c) The wire must be electrically insulated from the structure.

(d) The change in wire resistance per unit microstrain is very small; for a $350\ \Omega$ gauge the change in resistance is on the order of $0.0007\ \Omega/\mu\epsilon$. This small change in resistance must nevertheless be measured accurately if an accurate measure of strain is to be obtained.

(e) The structure (as well as the wire) may be deformed by mechanisms other than an applied load. The most common example is a change in temperature. If a structure composed of a homogeneous material is free to expand or contract, then a uniform change in temperature will cause the structure to uniformly expand or contract. This gives rise to 'apparent thermal strains', which are not associated with stresses induced within the structure and are generally not of interest.

(f) The resistance of the wire may be changed by mechanisms other than physical deformation. For example, an aggressive environment may cause oxidation of the wire, causing a change in wire resistance and hence an erroneous strain reading.

The original bonded strain gauges developed by both Ruge and Simmons consisted of several 'loops' of a continuous wire; several loops were used to increase the sensitivity of the gauge. However, the wire strain gauge has been largely displaced by the metal-foil strain gauge, which was introduced in the mid-1950s. Foil strain gauges are produced from a parent metal foil using a photoetch process. The foil gauge is normally bonded to a thin polymeric backing material, and the entire assembly (i.e., the strain gauge) is adhesively bonded to the test structure. Foil gauges are generally preferred over wire gauges due to improvements in gauge resistance and sensitivity tolerances, a decrease in gauge thickness, and because of the wide variety of grid shapes that can easily be produced using the photoetch process. Although the vast majority of strain gauges in use today are metal-foil gauges, wire gauges are still used for measurements at high temperatures ($\approx 400\ ^\circ\text{C}$ - $1100\ ^\circ\text{C}$) and other special-purpose applications [12].

3.1.3. The Relation Between the Change in Resistance and Unit Deformation

As it was previously mentioned, strain gauges are essentially devices whose electrical resistance changes when they are strained, by extending or compressing them. When bonded to, or embodied in, a diaphragm they can be used to measure the pressure-induced movement of the diaphragm (or other moving element) of a pressure sensor. The technique is very commonly used in pressure sensors and four

such gauges are normally connected in a Wheatstone bridge circuit as shown in Figure 3.2. The phenomenon of a change in resistivity due to strain, induced by a mechanical force, is known as piezo-resistivity and is exhibited by most conductors and semiconductors [6].

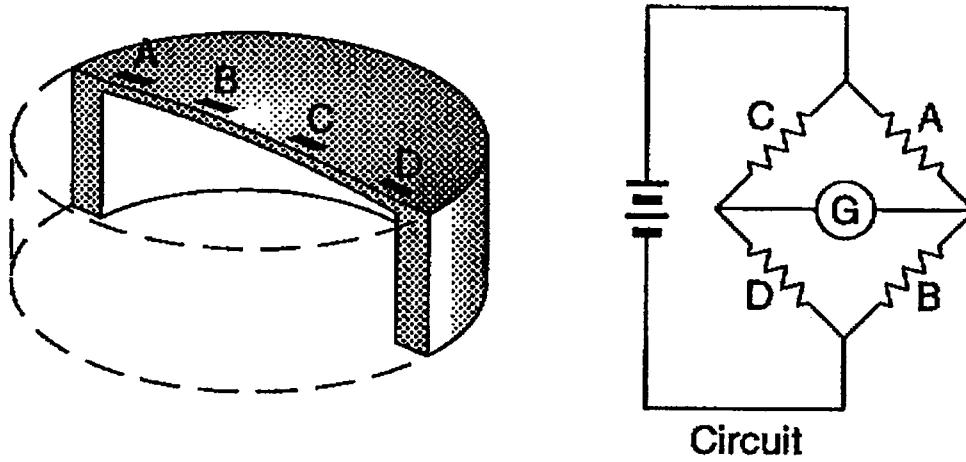


Figure 3.2. Strain gauge sensing

When a metal wire is stretched (strained) it becomes longer and thinner, and its resistance will increase by an amount related to geometry and piezo-resistivity.

A strain gauge's sensitivity (gauge factor) is expressed by the ratio of the relative change of resistance to the strain and it is represented by the symbol k :

$$k = \frac{\Delta R / R_0}{\Delta L / L_0} = \frac{\Delta R / R_0}{\epsilon} \quad (3.2)$$

Where, R_0 is initial resistance of the wire, L_0 is initial length of the wire, ϵ is the strain in mm/m.

The gauge factor is very much greater in semiconductors than in metals - typically about 50 times greater - because the piezo-resistive contribution to the gauge factor in semiconductors is very large. This makes them much more sensitive and suitable for use as strain gauges. Indeed, in the context of pressure measurement, the term piezo-resistive sensor is normally assumed to refer exclusively to semiconductor devices.

Bonded strain gauges are so-called because they are attached to the surface of pressure sensing elements, usually diaphragms. They may be metal foil, silicon, thin

film or thick film. The performance of the instrument will depend not only on the strain gauge material but also on the quality of the adhesion and of the diaphragm.

Comparable performances can be achieved with many different designs and matching the device's characteristics to the user's application makes the best choice.

Strain gauges can also be attached to a strain member, which is mounted rigidly at one end and connected at the other, via rod, to the pressure-sensing diaphragm.

3.1.4. The Basics of Wheatstone Bridge Used in Strain Gauge Applications

Sir Charles Wheatstone (1802 -1875), an English scientist, reported in 1843 on a circuit, which made the accurate measurement of electrical resistances possible. This circuit has become known as the "Wheatstone bridge circuit".

The Wheatstone bridge can be used in various ways to measure electrical resistances:

- For the determination of the absolute value of a resistance by comparison with a known resistance.
- For the determination of relative changes in resistance.

The latter method is used in strain-gauge techniques. It enables the relative changes of resistance in the strain gauge, which are usually around the order of 10^{-4} to 10^{-2} Ω/Ω to be measured with great accuracy.

If a supply voltage V_s , is applied to the two bridge supply points 2 and 3 then this is divided up in the two halves of the bridge R_1 , R_2 and R_4 , R_3 as a ratio of the corresponding bridge resistances, i.e. each half of the bridge forms a voltage divider, (Figure 3.3) [13].

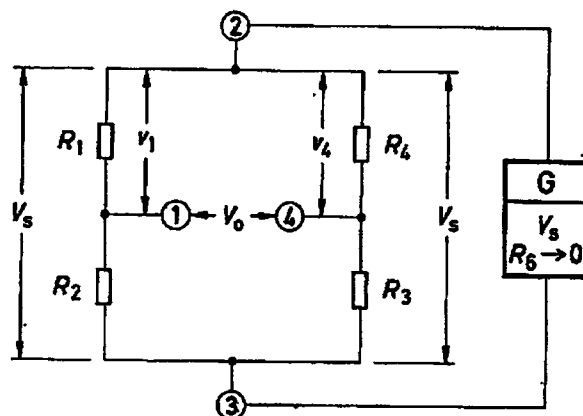


Figure 3.3. Principle of the voltage-fed Wheatstone bridge circuit

The following treatment of the bridge circuit assumes that the source resistance R_s of the voltage supply is negligibly small and that the internal resistance of the instrument for measuring the bridge output voltage is very high and does not cause any load on the bridge circuit. This method of treatment is acceptable, because the instruments used in strain gauge techniques fulfill these requirements to a large extent.

The partial voltage v_1 on bridge node 1 can be calculated as

$$v_1 = \frac{R_1}{R_1 + R_2} \cdot V_s \quad (3.3)$$

and the partial voltage v_4 on bridge node 4 is

$$v_4 = \frac{R_4}{R_3 + R_4} \cdot V_s \quad (3.4)$$

The difference between the two partial voltages is the bridge output voltage V_0 :

$$V_0 = V_s \left(\frac{R_1}{R_1 + R_2} - \frac{R_4}{R_3 + R_4} \right) = (v_1 - v_4) \quad (3.5)$$

If the unbalance in the bridge is defined as the relative output voltage V_0/V_s then equation (3.5) appears in the form

$$\frac{V_0}{V_s} = \frac{R_1}{R_1 + R_2} - \frac{R_4}{R_3 + R_4} \quad (3.6.a)$$

or

$$\frac{V_0}{V_s} = \frac{R_1 \cdot R_3 - R_2 \cdot R_4}{(R_1 + R_2) \cdot (R_3 + R_4)} \quad (3.6.b)$$

There are two conditions where $V_0 = 0$;

a) When all bridge resistors are equal value.

$$R_1 = R_2 = R_3 = R_4 = R,$$

b) When the resistor ratios in the two halves of the bridge are the same,

$$\frac{R_1}{R_2} = \frac{R_4}{R_3}$$

In these cases

$$\frac{V_0}{V_s} = 0$$

and the bridge circuit is “balanced”.

If the bridge resistors R_1 to R_4 change their value by an amount ΔR , then the bridge circuit becomes “unbalanced” and an output voltage V_0 is present between points 1 and 4. Equation (3.6.a) becomes

$$\frac{V_0}{V_s} = \frac{R_1 + \Delta R_1}{R_1 + \Delta R_1 + R_2 + \Delta R_2} - \frac{R_4 + \Delta R_4}{R_3 + \Delta R_3 + R_4 + \Delta R_4} \quad (3.7)$$

All bridge conditions can be calculated with equation (3.7), irrespective of the basic resistance R that the individual arms of the bridge possess. However, in strain gauge techniques all the arms of the bridge should have the same resistance; at least the two bakes of the bridge R_1 , R_2 and R_3 , R_4 must have the same resistances. Variations due to the tolerance on the strain gauge resistance do not affect the measurement accuracy. Even differences of 5% in the resistances of R_1 and R_2 produce errors of less than 0.1 %.

In strain-gauge techniques the amounts by which the resistance changes in the metal strain gauges are very small and of the order of about 10^{-3} . It is therefore usual to use the approximation below, which provides sufficiently accurate results for practical requirements, instead of equation (3.7):

$$\frac{V_0}{V_s} = \frac{1}{4} \left(\frac{\Delta R_1}{R_1} - \frac{\Delta R_2}{R_2} + \frac{\Delta R_3}{R_3} - \frac{\Delta R_4}{R_4} \right) \quad (3.8)$$

The approximate formula also shows that the relative change of resistance of each arm of the bridge is the governing factor in balancing the bridge and not the absolute change of resistance. Equation (3.2) can be rewritten thus:

$$\frac{\Delta R_i}{R_i} = k \cdot \epsilon_i \quad (3.9)$$

where equation (3.8) in the form

$$\frac{V_0}{V_s} = \frac{k}{4} (\epsilon_1 - \epsilon_2 + \epsilon_3 - \epsilon_4) \quad (3.10)$$

is substituted (the indices correspond to the designations for the bridge arms).

The equations (3.7) and (3.8) assume that all the resistances in the bridge change. This situation occurs for example in transducers or with test objects performing a similar function. In experimental stress analysis this is hardly ever the case and usually only some of the arms of the bridge contain active strain gauges, the remainder being made up of bridge completion resistors. Designations for the various forms such as quarter bridge, half bridge, double quarter or diagonal bridge and full bridge are commonplace. Figure 3.4 illustrates the different forms.

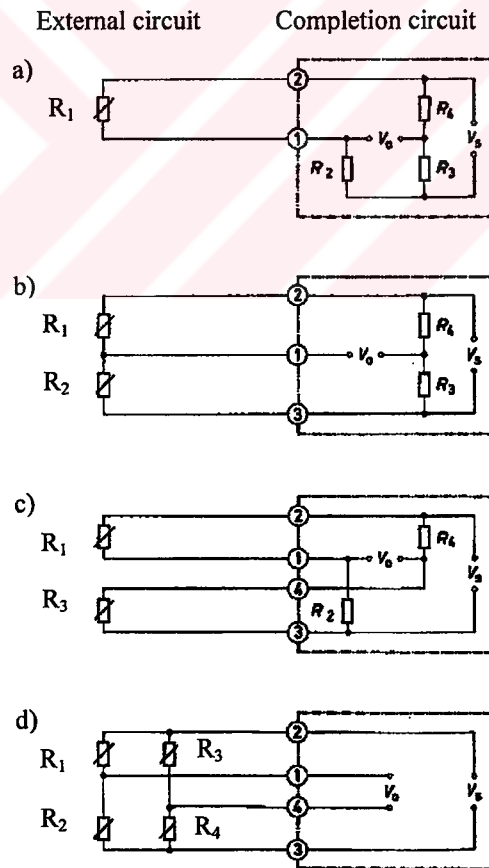


Figure 3.4. Forms of the Wheatstone bridge circuit used in strain gauge techniques
a) quarter bridge b) half bridge c) double quarter or diagonal bridge d) full bridge

From equations (3.8 and 3.10) it can be seen that the resistance changes and the strains which cause them contribute to the unbalance in the bridge, and hence to the measurement signal, with different signs. On connecting a measuring instrument designed for taking strain measurements the following effects take place:

Positive indication if $\varepsilon_1 > \varepsilon_2$ and/or $\varepsilon_3 > \varepsilon_4$

Negative indication if $\varepsilon_1 < \varepsilon_2$ and/or $\varepsilon_3 < \varepsilon_4$

Please note that: The signs “greater than” ($>$) and “smaller than” ($<$) apply in the algebraic sense and not to the magnitudes!

A markedly linear characteristic arises if the sum of the resistance changes in both halves of the bridge $\Delta R_1 + \Delta R_2$ and $\Delta R_4 + \Delta R_3$ is zero.

Maximum linear deviation occurs with the quarter bridge circuit and to the same extent with the double or diagonal bridge circuit. Figure 3.5 shows the dependence of the bridge unbalance V_0/V_s to the relative change of resistance $\Delta R/R$ in bridge arm 1, calculated according to the equation

$$\frac{V_0}{V_s} = \frac{\Delta R}{2(2R_0 + \Delta R)} \quad (3.11)$$

Assuming $R_1 = R_2 = R_3 = R_4 = R = R_0$. For comparison the linear characteristic

$$\frac{V_0}{V_s} = \frac{\Delta R}{4R_0} \quad (3.12)$$

Which is calculated from the approximate formula is shown as a broken line (see Figure 3.5) [13].

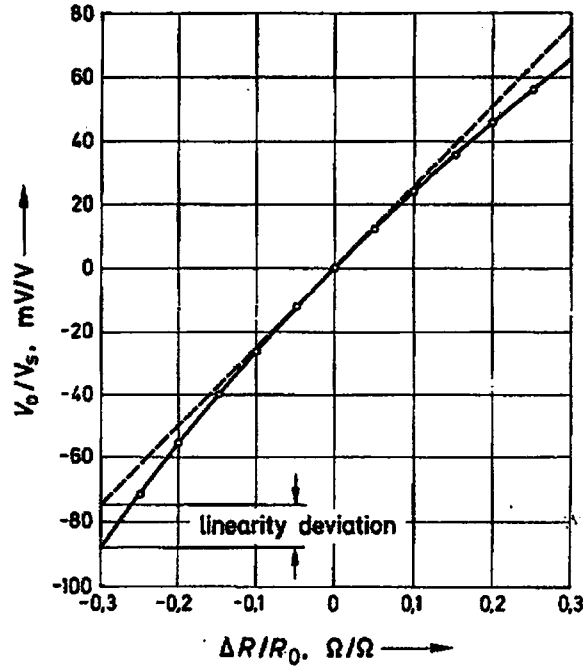


Figure 3.5 Characteristic of the Wheatstone bridge circuit in the “quarter bridge” operating mode according to equation (3.11) and the comparison characteristic based on the approximation formula (3.12)

3.2. Construction of the Pressure Transducer Prototypes

3.2.1. The Operating Principle of the Transducer

The operating principle of the constructed transducer was based on the thick wall cylindrical vessel theory.

3.2.1.1. Thick Wall Cylindrical Vessel Theory

Stress distribution symmetrical about an axis in polar coordinates is given by [14]:

$$\left(\frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} \right) \left(\frac{\partial^2 \phi}{\partial r^2} + \frac{1}{r} \frac{\partial \phi}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \phi}{\partial \theta^2} \right) = 0 \quad (3.13)$$

Where ϕ is the stress function as a function of r and θ .

When the stress function depends on r only, the equation of compatibility (3.13) becomes [14]

$$\left(\frac{d^2}{dr^2} + \frac{1}{r} \frac{d}{dr} \right) \left(\frac{d^2 \phi}{dr^2} + \frac{1}{r} \frac{d\phi}{dr} \right) = \frac{d^4 \phi}{dr^4} + \frac{2}{r} \frac{d^3 \phi}{dr^3} - \frac{1}{r^2} \frac{d^2 \phi}{dr^2} + \frac{1}{r^3} \frac{d\phi}{dr} = 0 \quad (3.14)$$

This is an ordinary differential equation, which can be reduced to a linear differential equation with constant coefficients by introducing a new variable t such that $r = e^t$. In this manner the general solution of Equation (3.14) can be easily obtained. This solution has four constants of integration, which must be determined from the boundary conditions. By substitution it can be checked that [14]

$$\phi = A_1 \log r + B_1 r^2 \log r + C_1 r^2 + D_1 \quad (3.15)$$

is the general solution.

When the two-dimensional problems will be solved by means of polar coordinates [14].

$$\sigma_r = \frac{1}{r} \frac{\partial \phi}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \phi}{\partial \theta^2} \quad (3.16a)$$

$$\sigma_\theta = \frac{\partial^2 \phi}{\partial r^2} \quad (3.16b)$$

$$\tau_{r\theta} = \frac{1}{r^2} \frac{\partial \phi}{\partial \theta} - \frac{1}{r} \frac{\partial^2 \phi}{\partial r \partial \theta} = -\frac{\partial}{\partial r} \left(\frac{1}{r} \frac{\partial \phi}{\partial \theta} \right) \quad (3.16c)$$

The solutions of a group of problems of symmetrical stress distribution with no body forces can be obtained from this. The corresponding stress components from equations (3.16) are [14]

$$\sigma_r = \frac{1}{r} \frac{\partial \phi}{\partial r} = \frac{A_1}{r^2} + B_1(1 + 2 \log r) + 2C_1 \quad (3.17a)$$

$$\sigma_\theta = \frac{\partial^2 \phi}{\partial r^2} = -\frac{A_1}{r^2} + B_1(3 + 2 \log r) + 2C_1 \quad (3.17b)$$

$$\tau_{r\theta} = 0 \quad (3.17c)$$

If there is no hole at the origin of coordinates, constants A_1 and B_1 vanish, since otherwise the stress components (3.17) become infinite when $r = 0$. Hence, for a plate without a hole at the origin and with no body forces, one case of stress distribution symmetrical with respect to the axis exist, namely, that when $\sigma_r = \sigma_\theta = \text{constant}$ and

the plate is in a condition of uniform tension or uniform compression in all directions in its plane [14].

If there is a hole at the origin, other solutions than uniform tension or compression can be derived from expressions (3.17). Taking B_1 as zero, for instance, Equations (3.17) become [14]

$$\sigma_r = \frac{A_1}{r^2} + 2C_1 \quad (3.18a)$$

$$\sigma_\theta = -\frac{A_1}{r^2} + 2C_1 \quad (3.18b)$$

This solution may be adapted to represent the stress distribution in a hollow cylinder submitted to uniform pressure on the inner and outer surfaces (Fig. 3.6). Let a and b denote the inner and outer radii of the cylinder, and p_i and p_o the uniform internal and external pressures. Then the boundary conditions are [14]:

$$(\sigma_r)_{r=a} = -p_i$$

$$(\sigma_r)_{r=b} = -p_o$$

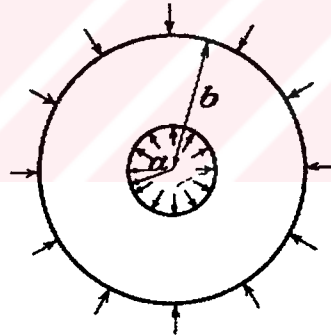


Figure 3.6. Section of thick wall cylindrical vessel with boundary conditions

Substituting in the first of equations (3.18), we obtain the following equations to determine A_1 and C_1 : [14]

$$\frac{A_1}{a^2} + 2C_1 = -p_i$$

$$\frac{A_1}{b^2} + 2C_1 = -p_o$$

from which

$$A_1 = \frac{a^2 b^2 (p_o - p_i)}{b^2 - a^2}$$

$$2C_1 = \frac{p_i a^2 - p_o b^2}{b^2 - a^2}$$

Substituting these in equations (3.18) the following expressions for the stress components are obtained: [14]

$$\sigma_r = \frac{a^2 b^2 (p_o - p_i)}{b^2 - a^2} \frac{1}{r^2} + \frac{p_i a^2 - p_o b^2}{b^2 - a^2} \quad (3.19a)$$

$$\sigma_\theta = \frac{a^2 b^2 (p_o - p_i)}{b^2 - a^2} \frac{1}{r^2} + \frac{p_i a^2 - p_o b^2}{b^2 - a^2} \quad (3.19b)$$

The radial displacement u is easily found since here $\epsilon_\theta = u/r$, and for plane stress [14]

$$E\epsilon_\theta = \sigma_\theta - \nu\sigma_r$$

It is interesting to note that the sum $\sigma_r + \sigma_\theta$ constant through the thickness of the wall of the cylinder. Hence the stresses σ_r and σ_θ produce uniform extension or contraction in the direction of the axis of the cylinder and cross sections perpendicular to this axis remain plane. Hence the deformation produced by the stresses (3.19) in an element of the cylinder cut out by two adjacent cross sections does not interfere with the deformation of the neighboring elements, and it is justifiable to consider the element in the condition of plane stress as, we did in the above discussion [14].

In the particular case when $p_o = 0$ and the cylinder is submitted to internal pressure only, equations (3.19) give

$$\sigma_r = \frac{a^2 p_i}{b^2 - a^2} \left(1 - \frac{b^2}{r^2} \right) \quad (3.20a)$$

$$\sigma_\theta = \frac{a^2 p_i}{b^2 - a^2} \left(1 + \frac{b^2}{r^2} \right) \quad (3.20b)$$

These equations show that σ_r is always a compressive stress and σ_θ is a tensile stress. The latter is greatest at the inner surface of the cylinder, where [14]

$$(\sigma_\theta)_{\max} = \frac{p_i(a^2 + b^2)}{b^2 - a^2} \quad (3.21)$$

$(\sigma_\theta)_{\max}$ is always numerically greater than the internal pressure and approaches this quantity as b increases, so that it can never be reduced below p_i , however much material is added on the outside. Various applications of equations (3.20) and (3.21) in machine design are usually discussed in elementary books on the strength of materials [14].

The corresponding problem for a cylinder with an eccentric bore was solved by G. B. Jeffery [14]. If the radius of the bore is “ a ” and that of the external surface “ b ”, and if the distance between their centers is “ e ”, the maximum stress, when the cylinder is under an internal pressure p_i , is the tangential stress at the internal surface at the thinnest part, if “ $e < (1/2)a$ ”, and is of the magnitude

$$\sigma = p_i \left[\frac{2b^2(b^2 + a^2 - 2ae - e^2)}{(a^2 + b^2)(b^2 - a^2 - 2ae - e^2)} - 1 \right]$$

if $e = 0$, this coincides with equation (3.21) [14].

3.2.2. The Design Parameters of the Transducer Prototype

As it was mentioned in the previous sections, the theory of thick-wall cylinder was applied to the design of active elastics element in the transducer within the scope of this study. The elastic thick-wall cylinder is the situation covered by “Lamé equations” (G. Lamé and B.P.E. Clapeyron, Mem.Acad.Sci.Paris 4, 465 (1883)) in most theoretical assessments of the strength. Here it is assumed that when the cylinder expands under the influence of high-internal pressure, it remains cylindrical. Equations relate the radial, tangential (hoop) and axial stress components to the cylinder radii and the pressure contained. It is shown that all the stresses and strains except in axial direction are the function of radius and it is also shown that axial stress and strain are independent of radius. These equations refer to the pressure range within which cylinder can be used completely elastic and they are valid only for significantly distant from both ends of the active element [15].

Figure 3.7 shows the cross-sections of the pressure transducer and the active cylinder.

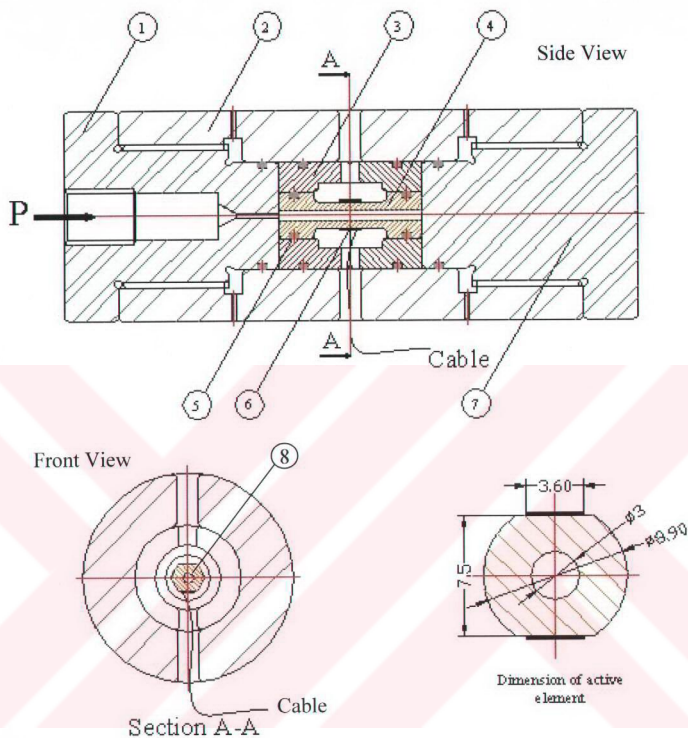


Figure 3.7. Schematic cross-section diagrams of the pressure transducer: 1. Pressure plug, 2. Outer cylinder, 3. Coupling cup, 4. Active element, 5. O-ring, 6. Strain gauge, 7. Blind plug, 8. Flattened part of the active element

The centre of the active cylinder is basically a 8.9 mm diameter cylinder, but only two opposite surfaces were flattened in order that strain gauges could be bonded on a flat surface. The radial distance between the two flat surfaces is 7.5 mm. The length of the active cylinder is only 40 mm with an internal diameter of 3 mm. The ratio (W) of the diameter of the outer cylindrical surface of the active element to diameter of the inner cylinder is approximately 3. The ratio (W_{ap}) of the outer flattened surface of the cylinder to the inner cylindrical diameter was determined as

approximately 2.5, in order to have a 2 mV/V output reading at the maximum applied pressure, both for hardened AISI 4340 steel and Invar steel made sensors.

Finite element analysis was also applied to evaluate the displacement and deformations when the active element is pressurized internally and at its open ends. This was done for determining the mechanical tolerances in order not to have contact between the active cylinder and the coupling cup even at the applied highest pressure. Since the deformations are not larger than 5 to 25 μm , the mechanical tolerances were kept at maximum 30 to 40 μm .

The pressure sealing of the transducer was obtained by means of three pairs of O-ring for sealing between the active cylinder and the coupling cups, between the coupling cups and the outer cylinder and between the plug and the outer cylinder. Since the applied pressure is not greater than 150 MPa, only Viton O-rings were preferred for pressure sealing. All the other parts of the transducer were made up with stainless steel.

3.2.3. The Selection and the Properties of the Materials

The principal attributes which need to be considered in the process of selecting a transducer spring material - namely, mechanical properties, thermal properties, and a group of additional factors classified as "manufacturing considerations".

Linear-elastic load response, with minimal hysteresis, was identified as one of the most desirable mechanical properties for a spring material. Among thermal properties, the thermal conductivity was highlighted as a key factor affecting transducer accuracy and stability. Because practicality in fabrication is always important, machinability and distortion-free hardenability ranked high among the manufacturing considerations.

The elastic modulus of the material was selected with special attention because of its important role in determining spring element section dimensions with respect to load rating. While steel alloys make excellent materials for high-capacity spring elements, they are often unsatisfactory for low-capacity units. In the latter case, to achieve suitable strain levels in reasonably thick and easy-to-machine sections, it is generally necessary to use low-modulus materials such as the aluminum alloys.

Due to the facts described above after a literature survey, as it was mentioned in the Introduction section, AISI 4340 and Invar steels were selected as the spring material.

The properties of the selected materials used for the manufacture of the active cylinder and the coupling cup are given in Table 3.1. [4], [16]

Table 3.1. The properties of the materials used for the active cylinder and the coupling cup

	AISI 4340 Steel	Invar Steel
E, Young's modulus (GPa)	207	141
ν , Poisson's ratio	0.29	0.29
Ultimate Tensile Strength (MPa)	1500	448
Yield Strength (MPa)	1365	276
α , Linear thermal expansion coefficient ($^{\circ}\text{C}^{-1}$)	11×10^{-6}	1.2×10^{-6}
Thermal conductivity (W/m $^{\circ}\text{C}$)	37.4	10.5

3.2.4. The Selection Criteria of the Strain Gauges

The initial step in preparing for any strain gauge installation is the selection of the appropriate gauge for the task. It might at first appear that gauge selection is a simple exercise, of no great consequence to the stress analysis; but quite the opposite is true. Careful, rational selection of gauge characteristics and parameters can be very important in: (1) optimizing the gauge performance for specified environmental and operating conditions, (2) obtaining accurate and reliable strain measurements, (3) contributing to the ease of installation, and (4) minimizing the total cost of the gauge installation.

Gauge Parameters

The installation and operating characteristics of a strain gauge are affected by the following parameters, which are selectable in varying degrees:

- strain-sensitive alloy
- self-temperature-compensation number
- backing materials (carrier)
- grid resistance
- gauge length
- options

- gauge pattern

Test Parameters

Basically, the gauge selection process consists of determining the particular available combination of parameters, which is most compatible with the environmental and other operating conditions. At the same time satisfies the best installation and operating constraints. These constraints are generally expressed in the form of requirements such as:

- Accuracy
- Test duration
- Stability
- Cyclic endurance
- Temperature
- Ease of installation
- Elongation
- Environment

The cost of the strain gauge itself is not ordinarily a prime consideration in gauge selection, since the significant economic measure is the total cost of the complete installation, of which the gauge cost is generally a small fraction [17].

3.2.4.1. The Strain Gauges Used for the Investigated Pressure Transducers

The strain gauges, model N2A-06-S063Q-350 Ω (gauge factor $k = 2.09$), manufactured by Micromasurements, were used. The strain gauges were 9.0 mm in length and 3.6 mm in width. They were bonded to the centre of the two opposing flat surfaces of the flattened active cylinder. As a bi-axial Rosette pattern was used, the strain gauges were capable of sensing both circumferential and axial strains. Therefore, it was possible to form the full Wheatstone bridge with one pair of strain gauges and the output signal was read in mV/V unit using the Hottinger Baldwin Messtechnik DMP 40 digital precision measuring instrument.

As it was shown in Table 3.2, this strain gauge was recommended by the manufacturer for the small column type transducers [18].

Table 3.2. The dimensional properties of the selected strain gauge

GAUGE PATTERN	ACTUAL MATRIX SIZE	GAUGE DESIGNATION	RES. IN OHMS	STD. CREEP CODE	DIMENSIONS (¹ inches and ² millimeters)			
					GAUGE LENGTH	OVERALL LENGTH	GRID WIDTH	OVERALL WIDTH
		N2A-06-S063Q-350	350±0.2%	Q	0.062 ¹	0.295 ¹	0.080 ¹	0.080 ¹
					1.57 ²	7.49 ²	2.03 ²	2.03 ²
					Often used on small column transducers			
	0.35 ¹ L x 0.14 ¹ W							
	9.0 ² L x 3.6 ² W							

3.2.5. The Bonding Procedure of the Strain Gauge

3.2.5.1. Surface Preparation for Strain Gauge Bonding

The followed surface preparation procedure is as follows: [19]

1. After machining and heat-treatment, the elements should be placed in a solvent degreasing solution. The most effective method is a hot vapor degreasing with 1-1-1 Trichloroethane or an equivalent substitute.
2. Following degreasing and drying, the gauge bonding areas should be abraded to produce a 0.4 to 1.6 micrometer RMS finish. This abrasion can be done with silicon-carbide paper, but forced-air grit blasting is preferred because it produces cleaner and more uniform abrasion at a much faster rate.

Grit blasting should be done using filtered, dry air. Tanks of compressed carbon dioxide or nitrogen may be used in place of air. Shop air may also be used, providing proper filters are employed to remove oil and water contamination. Filters should be inspected regularly.

Alumina powder abrasive grit in 25-30 micron particle size is recommended, and produces approximately a 25 micrometer RMS finish. Do not use silica (sand).

Grit should not be recycled. Air pressure is typically regulated to 2.7 to 4 bar depending on blast unit type, to produce approximately 300 m/second air stream at the nozzle.

Abrasion is accomplished by using even strokes, similar to spray painting. The nozzle should be kept 50 to 75 mm away from the surface, and always at a 45° to 60° angle from the surface.

3. After abrasion, units should be placed in an ultrasonic bath for final cleaning. The ultrasonic agitation helps remove lodged abrasive grit. Since this is the last step in surface preparation, the solvent should be changed regularly to prevent a buildup of contaminants. Isopropyl alcohol is a common solvent; others may be used with due consideration for toxicity and flammability.
4. Slight warming of the elements during drying is recommended to prevent moisture condensation.
5. After cleaning, the units should be handled with lint-free gloves and placed in a closed container to await gauge bonding.
6. Just prior to adhesive application, scrub the gauge bonding areas with a cotton applicator that has been moistened with M-Prep Neutralizer 5A. Wipe dry.

After surface preparation, all element materials will begin to oxidize in the presence of air; therefore, elements should be cleaned in batch sizes, which will allow gauge bonding on all elements within a specific period of time before oxidation becomes excessive. The time period between surface preparation and bonding differs for element material types. For AISI 4340 Steel and Invar Steel the time should not exceed 2 hours

3.2.5.2. Gauge Alignment

Accurately locating the gauge on the element usually requires the use of layout lines. These lines can be positioned in several ways: [19]

1. The easiest alignment method is to scribe lines directly on the element with a pointed tool such as a dental probe. This method produces a clear, sharp line, but can result in unwanted side effects:
 - a. Scribe lines produce stress concentrations, particularly in thin-beam elements. Premature gauge failures can result from fatigue cracks that grow out of such sharp-edged concentrations.
 - b. Heavy scribe lines can trap contamination possibly resulting in bond losses.

Burnished alignment marks are an alternative to scribe lines. These are “polished” lines that do not damage the gauging surface. A ballpoint pen (no ink) is preferred for burnishing on steel alloys. A 4-H drafting pencil will burnish aluminum, but the marks must be scrubbed clean with Neutralizer 5A and a cotton swab.

2. Optical projection of alignment lines is the most desirable gauge locating method. The element is placed in a fixture that has a prealigned stereomicroscope attached. When viewed through the microscope, alignment lines etched into the microscope lens appear to project to the element surface. Gauge-to-gauge spacing can be set by appropriately indexing the fixture.

3. Transducer-Class[®] strain gauges have their backing trim size held to very close tolerances typically ± 0.13 mm from grid centerlines to any backing edge. On some transducers, this will allow the gauge backing itself to be used for gauge alignment rather than the small marks provided on the gauge.

3.2.5.3. Gauge Preparation

Inspection: Gauges supplied by Micro-Measurements are subjected to rigorous 100% quality inspection by trained specialists. An untrained eye will not be able to detect gauge defects by optical inspection. Actually, many apparent optical imperfections visible under 40X magnification may not be functional defects, and the gauge could be rejected in error.

Without training and experience in inspection procedures, quality control is difficult to administer on incoming gauges. An accepted procedure is to make only a cursory check of the gauge before bonding and leave the close inspection of the assembly until after the gauge has been installed.

It has to be noted that gauges should never be handled with fingers; but rather with clean, blunt-nosed tweezers.

Gauge Cleaning: Micro-Measurements gauges are prepared for bonding when they are packaged and do not require cleaning before bonding unless they subsequently become contaminated. If fingerprints or dust particles are visible on the gauge backing, they should be removed by wiping the gauge bonding surface with a cotton applicator slightly moistened with Neutralizer 5A. Cleaning is best accomplished on

a glass plate kept clean by frequent washings with Neutralizer 5A. Allow sufficient time for any residual liquid to evaporate before bonding [19].

3.2.5.4. Adhesives and Adhesive Application

Micro-Measurements adhesives are generally recommended for transducer applications. M-Bond 610 was used which the detailed information given below.

M-Bond 610: Two-Component Epoxy-Phenolic. M-Bond 610 is the most popular transducer adhesive, and produces the thinnest, most creep-free glueline. It is recommended for all gauge types and transducers where cure and post-cure temperatures of up to 205 °C can be tolerated. When used on aluminum alloy elements, it may be necessary to alter the cure cycle.

Cleanliness is of primary importance in a adhesive application. Laminar-flow workbenches are recommended in this area of assembly. Working on easily cleaned surfaces, such as glass, is also advisable.

The 610 adhesives are supplied with nylon brush caps for adhesive application.

The preferred method for installing N2A-Series gauges entails coating the bonding surface of the gauge, the gauge edges, and the open grid area. This results in a bonded gauge with a very thin layer of a adhesive protection over the sensing grid. The possibility of damage - such as tool marks, solder splash, or fingerprints - occurring in later operations is greatly reduced.

Because 610 adhesives become fluid when heated to their cure temperature, a large amount of flow out is expected. Without adequate protection for the gauge tabs, adhesive will flow over them and inhibit soldering processes. This adhesive layer can be removed with a scalpel or with careful abrasion using alumina powder, but this usually proves too time consuming for production methods. A belier approach is to mask adhesive flow by covering the gauge tabs with a thin-film tape before bonding. This tape has a dual purpose in that it will also hold the gauge in place during alignment, clamping, and curing [19].

Procedure

1. Using tweezers, the gauge was removed from its carrier and it was placed bond-side-down onto a clean glass plate. A short length of Mylar tape (Micro-

Measurements MJG-2) was placed over half of the gauge tabs as shown in Figure 3.8.

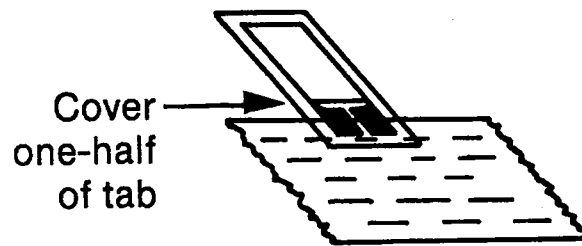


Figure 3.8 Placing a Mylar tape on the strain gauge

2. The gauge/tape assembly was lifted from the surface by peeling the tape back at a shallow angle until the gauge was free of the stir-face. The gauge was transferred to the proper position on the transducer, and the tape was pressed in place.
3. Lifting again at a shallow angle, one end of the taped assembly was peeled back to raise the gauge from the surface and exposed the bonding side. By curling this Mylar tape back upon itself, it remains in position and becomes ready for application of adhesive.
4. The gauge backing, foil, and edges were coated with a thin layer of adhesive. The specimen in the gauge-mounting area was also coated (Figure 3.9).

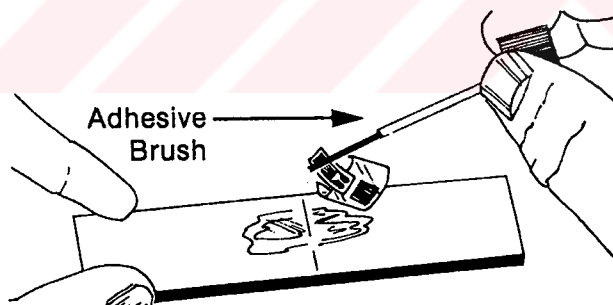


Figure 3.9. Application of the adhesive

5. The adhesive was permitted to air-dry for 5 to 30 minutes at +25 °C and 50 % relative humidity. Longer air-drying times are required at lower temperatures and higher humidities.
6. The tape was rotated back to its original position (gauge-bonding- side down), and preceded with clamping.

3.2.5.5. Clamping

Adhesives 610 require clamping to provide uniform pressure on the gauge before cure can be started. Final transducer performance characteristics, particularly uniformity from unit to unit, are greatly dependent upon the application of a uniform and repeatable clamping pressure on the gauges. Clamping pressure controls glue-line thickness and state of stress in the installed gauge - both major factors in performance.

Clamping force can be applied through several methods, but all use the same components over the gauge, as shown in Figure 3.10.

Teflon A thin TFE Teflon® film release sheet is required to keep adhesive from sticking to pressure pads. The type of film used should be approved by a strain gauge manufacturer, as some Teflons are unsatisfactory. Teflon tape applied to the pressure pad can often be used instead of film to reduce clamping assembly steps.

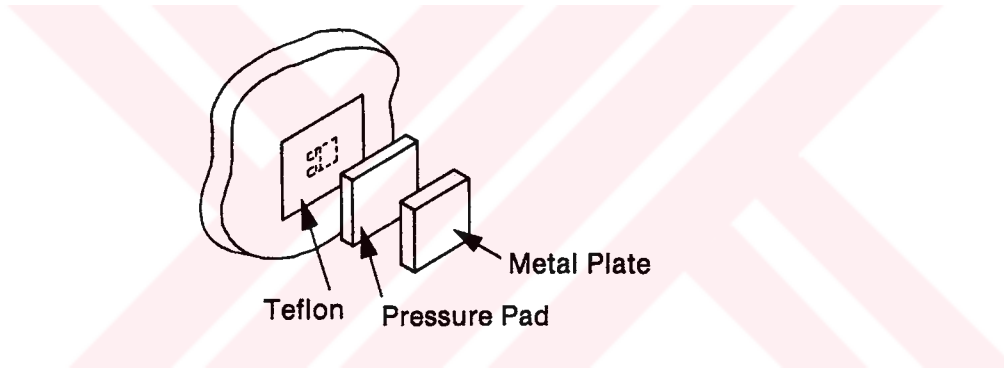


Figure 3.10. Clamping components

Pressure Pad: A soft rubber pad, usually silicone rubber, is used to convert the applied force to a uniform pressure on the gauge surface. Pads are reusable, but they should be checked periodically for deformation and hardness, as many types will change with time and number of temperature excursions. Recommended hardness: durometer A40-60; recommended thickness: 2.5 mm [19].

Metal Plate: Metal back-up plates are used to apply uniform force to the rubber pad; they are often permanently attached to the clamping device. The metal should match the contour of the transducer.

Clamping Pressure: A clamping pressure of 3 to 4 bar is recommended for 610 adhesives. This should be carefully computed and monitored for previously mentioned reasons. Clamping pressure is easily computed by dividing clamping

force by area. If the rubber pad is smaller in area than the metal plate, the area of the pad should be used for computing clamping force. If the rubber pad is larger than the metal plate (not normally recommended), the area of the metal plate should be used for calculations [19].

Many forms of clamping devices can be adapted to strain-gauge bonding fixtures. For this application spring clamps were used (Figure 3.11).

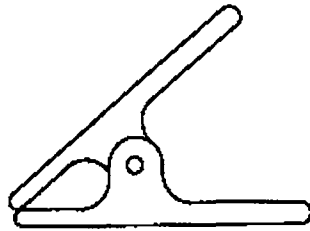


Figure 3.11. Spring clamp

Spring Clamps: Spring clamps are the most desirable form of clamping due to the relatively high forces available from small fixtures. Simple clip-on type clamps can be used for short-run production, but the absence of locating pins often results in slippage and loss of uniform pressure. Application and removal can also become tiring to personnel.

3.2.5.6. Curing and Postcuring

The time period between the clamping and the curing processes is limited to four hours because of the properties of the M-Bond 610 adhesive. This time period does allow large batches to be placed in the oven together. For proper cure, adhesive glue-line temperature must be raised at a rate between 3° to 11°C per minute from ambient. Failure to do so will result in uneven thickness (rate too slow) or bubble formation (rate too fast). Unless the oven has cooled from the preceding cure cycle, small transducer elements (low thermal mass) could possibly undergo a temperature rise in excess of the recommended 11°C per minute.

Large transducers (large thermal mass) can often be placed in preheated ovens for curing because their size automatically slows the temperature rise at the gauge locations. Slow moving conveyor-line ovens also can be continuously heated since the temperature rise is regulated by the speed of the conveyor.

Post-curing of strain gauge installations is often misunderstood but an essential step in assuring stable transducer performance. During initial cure, the adhesive will polymerize (set) at an elevated temperature. Due to the high forces applied to the gauge during clamping, residual stresses will be locked-in to the gauge when polymerization occurs. Proper post-curing steps (with clamps removed) will greatly reduce these residuals and their pronounced effects on backing and adhesive stability [17].

Post-Cure Procedure

1. After reaching the required cure temperature and holding for the recommended time, the elements should be allowed to cool down at least 55 °C before clamps are removed.
2. With clamps off, the element temperature should be raised to the required level, and allowed to post-cure for the recommended period. Elements may be placed in a hot oven for post-curing if no additional over coating adhesive has been applied.
- 3 After post-cure, the elements are removed from oven and allowed cooling to room temperature.

For this study, the active elements were cured at 175 °C for 1 hour and post-cured at 200 °C for 2 hours.

3.2.5.7. Inspection of Bonded Strain Gauges

Inspection of the bonded gauge is a very important step in the quality-assurance procedure. Early identification of bonding rejects will prevent lost effort in wiring and testing. Simple transducer elements are often discarded, rather than being reworked, when gauge bonding defects are found. This practice should be carefully considered, as removing a defectively bonded gauge is not an easy process. Virtually no commercial stripper will remove the bonded gauge without damaging the transducer element. When necessary, gauges can usually be removed by careful scraping with a razor blade followed by grit blasting.

Inspection is best accomplished under a stereomicroscope with 10x and 40x zoom capability. Direct lighting should be augmented with an additional spotlight set at a 45° to 60° angle from the surface. Controlled by a foot switch, the angled light can

be turned on (at the same time direct lighting is turned off) to help determine depth of surface irregularities.

Bonding faults are generally easy to see and can take any of three forms: [19]

1. Voids are areas not adhesively bonded. They appear as light-colored sections against a darker background. Sizes range from spots smaller than the gag tab width, to half of the gauge surface area and more. Voids are easily confused with adhesive that has assumed a non-uniform covering of the gauge face. The alternate use of direct and oblique lighting can help differentiate between the two. Placing a drop of solvent, such as isopropyl alcohol, over the surface of the gauge has also been found to help identify whether a void is under the grid. Careful probing of the area in question is also possible. When touched, voids cause sufficient movement of the backing to be clearly identifiable. Any gauge that shows a void in the grid area should be rejected. Small voids in areas outside the grid can be accepted, but their presence indicates a problem in clamping or adhesive application and should be investigated.

2. Bubbles are similar to voids in that the areas are not bonded, but they differ in appearance and cause. Bubbles are always round and small, and usually occur in great number. They are caused by solvent entrapment in the adhesive layer during cure. When bubbles are present, several areas should be investigated to determine the cause. These include:

- a. Air-drying time
- c. Clamping pressure
- b. Oven and specimen temperature rise time
- d. Expiration date of adhesive

A bubble under the grid area is cause for rejection.

3. Uneven Adhesive Thickness is almost always caused by improper clamping application, and is one of the most difficult problems to detect. One way is to compare the adhesive flow-out at each edge of the gauge.

3.2.5.8. Soldering and Removal of Flux

Proper soldering and fluxing is essential in the application of gauge bonding. After the selection of the soldering iron, solder type and the flux the following procedure should be applied [19]:

1. The element is placed under suitable magnification.
2. If the gauge face was not protected with a layer of bonding adhesive during assembly, it will have to be protected against flux and solder splash. An adequate amount of paper drafting tape is placed over the gauge face, leaving approximately half of the gauge tab exposed, as shown in Figure 3.12:

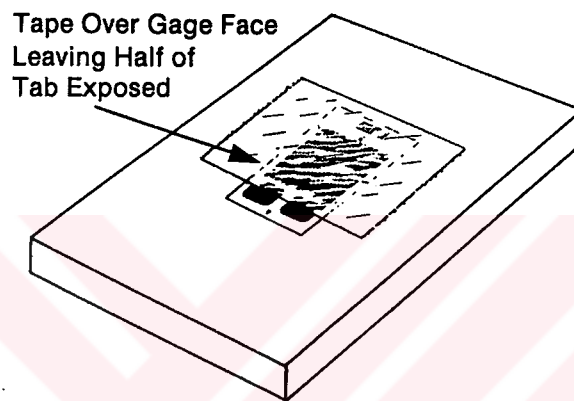


Figure 3.12. Protection of the strain gauge before the soldering process

If the gauge has been protected with a layer of bonding adhesive, tape is not necessary

3. Solder wetting is improved by cleaning the exposed gauge tabs prior to tinning. A soft 'pink' pencil eraser, lightly rubbed over the tabs, will clean solder areas without damage to the gauge foil. The eraser should never be allowed to contact the gauge grid area.
4. The soldering-iron tip is cleaned and tinned.
5. The end of the wire solder is rested on the exposed gauge tab (if a separate flux is being used, it should be applied to the gauge at this time).
6. The solder and tab is pressed down with the flat part of the soldering tip and fed in a small length of solder. If using flux-cored solder, this will provide additional flux.

7. The soldering iron is quickly and simultaneously lifted and soldered. Step 6 should be accomplished in 1 to 2 seconds. If tinning is incomplete, cooling should be allowed for 1 to 2 seconds and Steps 4 through 6 should be repeated.

8. The operation should be repeated on all remaining tabs.

9. The solder areas are inspected to be smooth, shiny, and “wetted” to tabs

10. The tinned wire is held in place on the gauge tab. Orientation of the wire should be in the direction of minimum strain, if possible. Residual flux from the tinning operation may be adequate for leadwire soldering.

11. The soldering iron is pressed onto the connection to re-flow the solder and lifted after one second.

12. Wire should be inspected not to extend out of the solder pillow; it should lie flat on the gauge tab, with no peaks of solder present.

13. If the solder connections are not adequate, the steps 10 through 12 should be repeated with the addition of small, separate amounts of rosin flux (Micro-Measurements M-Flux AR).

Removal of all soldering flux residue is essential for stable transducer performance. Rosin fluxes contain small amounts of activator. Usually these are weak acids. They must be removed by a thorough cleaning with a rosin-flux solvent. Some types that can be used are:

1. M-LINE Rosin Solvent (RSK-1)

2. 1-1-1 trichloroethane (CSM-1)

3. Axarel 32

Any solvent selected should be applied in ample quantities to thoroughly flush all connections. Agitation with a soft brush is recommended to loosen heavy deposits.

Considering the comparison results of flux removal solvents given in Figure 3.13, RSK-1 was used for the removal of flux.

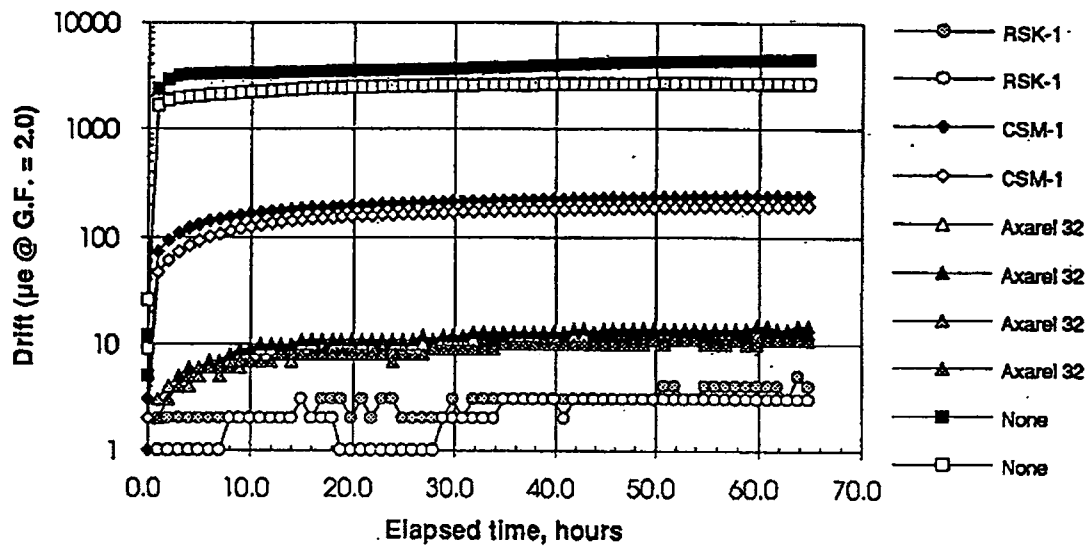


Figure 3.13. Comparison of flux removal solvents (drift versus time)

3.2.5.9. Protective Coatings

Sensitive strain gauge circuits require protection against most transducer operating environments. Even ambient indoor conditions can result in enough variation in temperature and humidity to affect long-term zero stability. Considering that the strain gauge is a resistor that must maintain stability equal to or better than hermetically sealed resistors, the need for protection is clear.

Selection of the best protection method is complicated, because the protection used must not affect transducer performance. Often, organic coatings (i.e., epoxies), which may provide an adequate barrier against moisture, stiffen the transducer flexure and cause time-dependent creep effects. In addition, many transducers must be certified to withstand intermittent water splash or even short-term immersion without damage. Most easily applied coatings will not tolerate this treatment over long periods [19].

For this study, M-Coat A was used that is air-drying solvent-thinned (xylene) polyurethane. It is transparent, has good flexibility and can be removed with M-LINE Rosin Solvent. Film thickness is 0.1 to 0.25 mm per coat.

3.3. Three Dimensional Strain Analysis of the Prototype by the Finite Element Method (FEM)

At present, Finite Element Method analysis is the most powerful tool to evaluate and optimize stress distribution problems, especially in complicated designs. Considering

the design, that the thick wall cylindrical vessel was modified by flattening the two surfaces of the active cylinder, the analytical approach was needed some assumptions to find the exact strain values on the area where the strain gauges were bonded. It was therefore decided that an existing design of a pressure transducer should be investigated using finite elements. Based on the findings of this investigation an improved version was designed.

3.3.1. The Basics of the FEM

The basic philosophy of the finite element method is discretization and approximation. Simply, the finite element method is a numerical approximation technique that divides a component or structure into discrete regions (the finite elements) and the response is described by a set of functions that represent the displacements or the stresses in that region [20]. The overall process is comprised of 3 steps:

- Creating the model (also called finite element modelling). This consists in modelling the geometry, generating the elements, selecting the material and defining the environment
- Analysing the model and assessing the results
- Optimising the design as necessary to achieve better results

In finite element modelling, a structure or component is analysed by being meshed into an assemblage of small, discrete elements whose vertices are called grid points or nodes. Creating an accurate geometric representation of the physical object is the first step in creating the model. Both Computer Aided Design (CAD) and finite modelling software were used for creating the finite element model.

The most important advantage of the FEM emerges in the design stage of a transducer, when output and maximum stress concentrations can be simulated and evaluated quickly, without going through the process of machining, bonding and testing.

In this study all the steps mentioned above were carried out to analyse the stress and strain distribution on the active cylinder of the pressure transducer. FEM software ANSYS version 5.6.1 was used to generate the geometric model and to evaluate the strain values on the area where the strain gauges were bonded. These values were

compared with those obtained from the strain gauge measurements to see the compatibility between the numerical method and the experiments.

3.3.2. The Model Used for the FEM Analysis

In order to optimise the design by improving the stress distribution, the following approach would have to be adopted: Whilst maintaining the average strain level in the gauge area, at a given pressure, the maximum stresses in zones of stress concentrations would be reduced, and also the size of these zones, if possible.

The symmetric shape of the transducer and all the design parameters of the active element were selected to assure a symmetrical stress and strain distribution even at the maximum working pressure of the transducer. Thus, the geometric model used for FEM analysis was generated for the one half of the active element in three dimensions on the Cartesian coordinate system.

Figure 3.14 shows the half part of the whole active cylinder with the boundary conditions and FEM meshed regions.

The boundary conditions to evaluate the strains on the active cylinder, in relation to the geometric domain of Figure 3.14 are:

For the applied pressure on the surface (diameter = 13 mm) of the active cylinder:

$$p(x,y,z) = p \quad 1.5 \text{ mm} \leq (x^2 + y^2)^{1/2} \leq 6.5 \text{ mm} \quad z = 0$$

For the applied pressure inside the active cylinder (internal diameter = 3 mm):

$$p(x,y,z) = p \quad (x^2 + y^2)^{1/2} = 1.5 \text{ mm} \quad 0 \text{ mm} \leq z \leq 20 \text{ mm}$$

Symmetric constraint has been applied to surface S.

In addition to the applied pressures defined above, due to the use of the O-ring for the pressure sealing, the upper part of the cylinder receives pressure till the end of the O-ring groove. In order to find out whether or not the strain values on the area where the strain gauges were bonded are affected by this pressure, the model was analysed with and without applying pressure (p) on the active element, up to the end of the O-ring groove on both sides. It was only an assumption that the pressure inside the O-ring groove was equal to the applied pressure (p). Therefore, another study is required to find out the actual pressure distribution inside the O-ring groove.

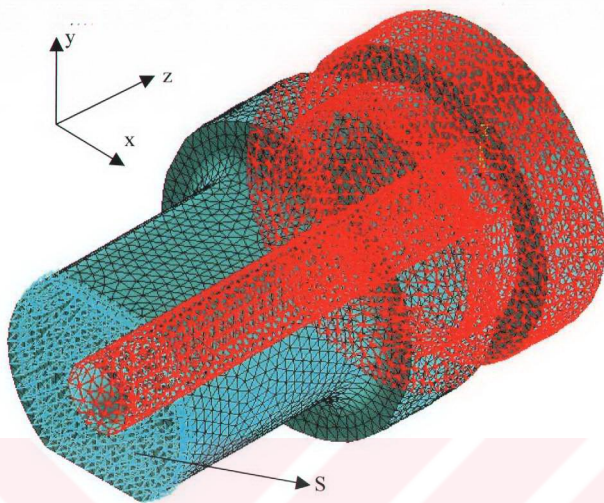


Figure 3.14 FEM model of the active cylinder of the investigated pressure transducer with the applied boundary conditions

For the applied pressure till the end of the O-ring groove on the surface of the active cylinder:

$p(x,y,z) = p$	$(x^2 + y^2)^{1/2} = 6.5 \text{ mm}$	$0 \text{ mm} \leq z \leq 3.0 \text{ mm}$
$p(x,y,z) = p$	$5 \text{ mm} \leq (x^2 + y^2)^{1/2} \leq 6.5 \text{ mm}$	$z = 3.0 \text{ mm}$
$p(x,y,z) = p$	$(x^2 + y^2)^{1/2} = 5.0 \text{ mm}$	$3.0 \text{ mm} \leq z \leq 5.6 \text{ mm}$
$p(x,y,z) = p$	$5 \text{ mm} \leq (x^2 + y^2)^{1/2} \leq 6.5 \text{ mm}$	$z = 5.6 \text{ mm}$

The analysis was realized at three different pressure (p) values of 60, 100 and 150 MPa and for modelling the active elements, following assumptions were made:

- Elastic properties of the active element were the same in all direction (isotropic material)
- There were no body forces on the active element such as gravitational forces, magnetic forces, inertia forces, etc.
- Pressure distributions were uniform at inner and outer surfaces of the active element.

- Hook law was obeyed.
- Tabulated values for modulus of elasticity's and Poisson's ratios supplied by their own manufacturers were true values.

Since the reported metrological specifications in Section 4.3 were better than the invar steel, the material for the FEM analysis was selected as AISI 4340 steel.

3.3.3. The Element Used for the FEM Analysis

“Solid 45” was used for the three-dimensional modelling of solid structure. The element, defined by eight nodes having three degrees of freedom each, has translations in the nodal x, y, and z directions and exhibits orthotropic material properties. The geometry, node locations, and the coordinate system for this element are shown in Figure 3.15. Pressure can be applied as surface loads on the element faces shown by the circled numbers [21].

A total of 83928 elements and 16415 nodes were used for the complete FEM stress and strain analysis of the active cylinder. Grid size was selected as 0.5 mm.

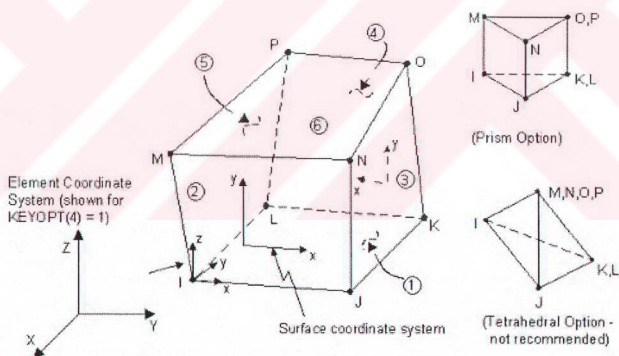


Figure 3.15. Solid 45, 3-D, 8-Node structural solid FEM element

3.3.4. The FEM Analysis Results

Figures 3.16, 3.17 and 3.18 show the FEM analysis strain results for the x (radial), y (angular) and z (axial) directions and for the applied pressure of 150 MPa. The strain values on the area where the strain gauges were bonded were plotted with arrows on the coloured fringe (mapped) graphs. The strain values are shown in mm/m. Figure 3.20 illustrates the results for z (axial) direction where pressure does not exist on the

O-ring groove side till the end of the O-ring groove for the pressure value of 150 MPa. For figures 3.16, 3.17, 3.18 and 3.19, the explanation of the important legends are: DMX = Maximum displacement; SMN = Minimum value of the plotted item; SMX = Maximum value of the plotted item; the plotted strain values EPTOX, EPTOY, EPTOZ were equal to the total sum of the elastic strain component, plastic strain component and creep strain component in X, Y and Z direction, respectively.

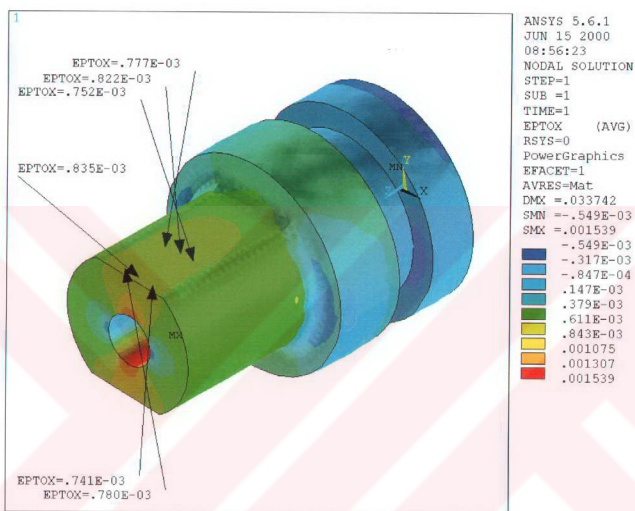


Figure 3.16. FEM strain results in x-direction for pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p = 150$ MPa

The results given in figures between 3.16 and 3.18 clearly show that the strain values in the y-direction are not negligible. Consequently, the circumferential strain measurements are not directly equal to the strains in x-direction. Due to this fact, for the FEM results the circumferential strain was assumed as the square root of the sum of squares of the strains in x and y direction.

The results illustrated with graphics in Figures 3.18 and 3.19 show that the pressure applied on the O-ring groove side till the end of the O-ring groove affected the strain values on the strain gauge area by only few microstrains. Yet, the stress distribution around the O-ring groove was taken into account during the design of the O-ring groove on the active cylinder.

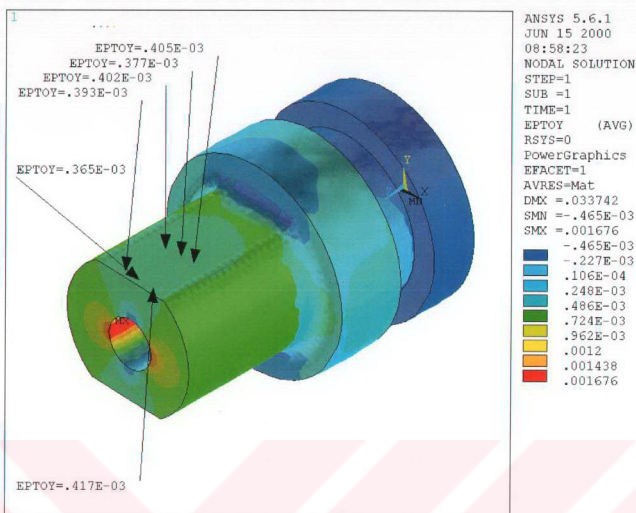


Figure 3.17. FEM strain results in y-direction for pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p = 150$ MPa

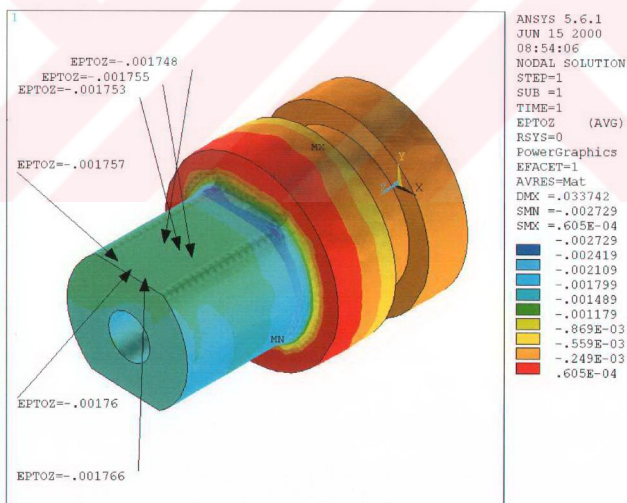


Figure 3.18. FEM strain results in z-direction for pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p = 150$ MPa

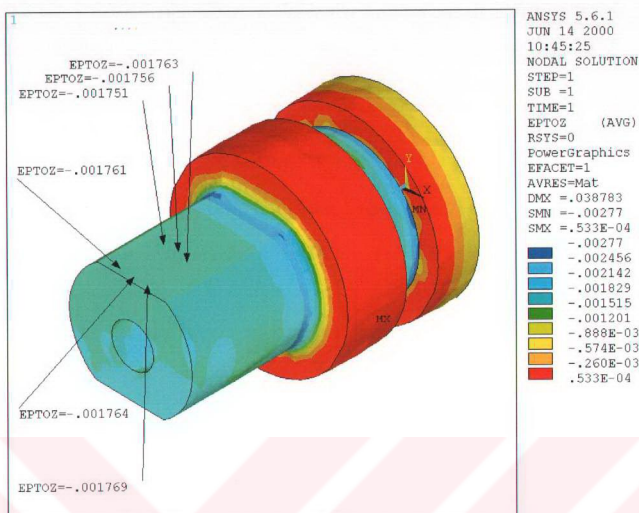


Figure 3.19 FEM strain results in z-direction for non-pressurized O-ring groove side of the AISI 4340 Steel made pressure transducer at $p = 150$ MPa

The position of the O-ring on the active cylinder was also analyzed by FEM to understand how it affects the strain and stress distribution on the active element. The 2.6 mm wide O-ring groove was moved ± 2 mm (in z-direction) from its original position ($z = 3$ mm). At the end of this analysis it has been seen that the strain and stress distribution changed in the area close to the O-ring groove, but it did not really affect the strain in the area where the strain gauges were bonded. The difference in the strain was less than 2 %.

In Section 5, the FEM output results were compared with those derived by the experimental measurements to see the compatibility.

The assumptions listed at the end of the section 3.3.2, were accepted in general for all scientists and due to their contributions to errors in any analysis were stated as they result an error less than 10 % of the full scale together with discretization error in 3-D FEM.

4. EXPERIMENTAL STRAIN GAUGE MEASUREMENTS

4.1. Measurement Setup

The photograph of the active cylinder inside the coupling cup was shown in Figure 4.1. Experiments were realized by calibrating the manufactured transducers with a primary pressure balance.

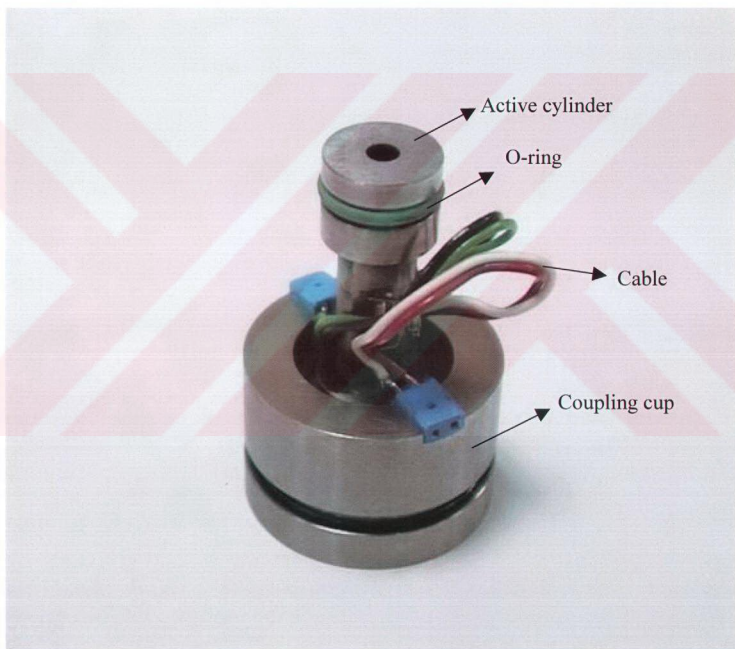


Figure 4.1. Photograph of the AISI 4340 Steel made active cylinder inside the coupling cup.

The measurement setup used during the experiments was shown schematically in Figure 4.2 and as a photograph in Figure 4.3.

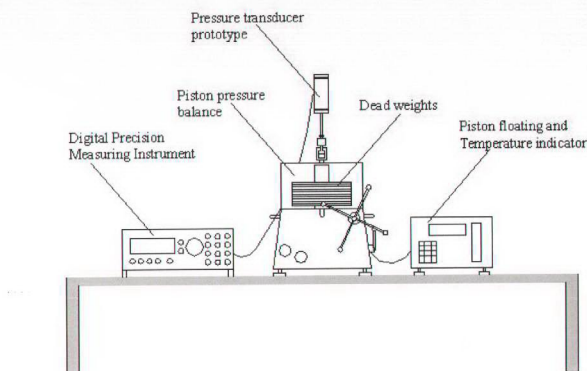


Figure 4.2. Measurement setup



Figure 4.3. Photograph of the measurement setup

The reference piston pressure balance standard was an oil-operated dead weight tester equipped with a piston cylinder assembly. The effective area of the piston cylinder unit was $4.902\,463\text{ mm}^2$ with an uncertainty of $0.000\,122\text{ mm}^2$ at the 95 % confidence level ($k_c = 2$). The uncertainty on the generated effective pressure was $3.1 \times 10^{-5} \times p$ at the 95 % confidence level ($k_c = 2$). During the measurements the temperature of the assembly was kept at $(20 \pm 1)^\circ\text{C}$. Also, a thermocouple sensor was placed inside the transducer on the active cylinder to measure the temperature of the transducer. The temperature variations of the transducer were inside the stability

of (20 ± 0.2) °C, for a single calibration cycle. The pressure medium was the di-ethyl-hexyl-sebacate. The output readings of the transducers were obtained by using a digital precision measuring instrument after thirty seconds of stabilization period. The resolution of this instrument is 1×10^{-6} mV/V which corresponds to 75 Pa for the 150 MPa full-scale range and 50 Pa for the 100 MPa full-scale range. The 5V source voltage of the Wheatstone Bridge is also supplied with the same instrument. All of the transducers were calibrated in vertical mounting position.

4.2. Other Details of the Measurement

A total of eight specimens were calibrated, five made up with hardened AISI 4340 steel and three made up with invar steel. For each specimen four measurement cycles consisting of increasing and decreasing pressure steps were run. For the hardened AISI 4340 transducer the pressure steps were at the values of: 0, 20, 40, 60, 80, 100, 120, 140, 150 MPa and for the invar steel transducer the pressure steps were at the values of: 0, 10, 20, 40, 50, 60, 80, 100 MPa.

Additional measurements were performed to compare the FEM analysis strain results with those obtained from the experimental strain measurements in terms of strain ($\mu\text{m}/\text{m} = \mu\epsilon$). For this purpose the measurement circuit was modified to half Wheatstone bridge. One of the strain gauges located in the axial direction was used with a dummy strain gauge that was bonded on the non-pressurized AISI 4340 steel to measure the axial strain (ϵ_a). Then, to measure the circumferential strain (ϵ_c), the other strain gauge located in the radial direction was used with the same dummy gauge. Hottinger Baldwin Messtechnik MGC Amplifier System was used to perform the half-Wheatstone bridge strain measurements.

4.3. The Metrological Characterization Results of the Manufactured Pressure Transducers

The average hysteresis error values calculated from the output values of one of the hardened AISI 4340 Steel pressure transducer (UMEPT4340-01) in terms of percentage of full scale (2 mV/V) are shown in Figure 4.4.

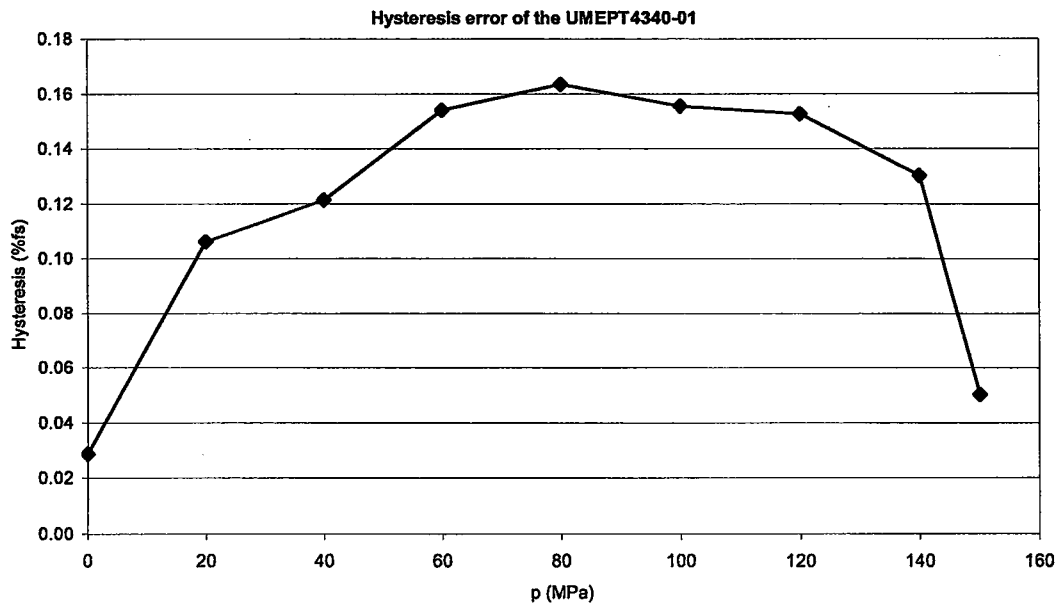


Figure 4.4. The hysteresis error graph of the UMEPT4340-01

The average hysteresis error values calculated from the output values of one of the Invar steel transducers (UMEPTIn-01) in terms of percentage of full scale (2 mV/V) are given in Table 4.1.

Table 4.1. Calculated average hysteresis error values of UMEPTIn-01

p (/ MPa)	Hysteresis error (/ % fs)
0	0.22
10	1.09
20	1.83
40	2.64
50	2.90
60	2.99
80	2.06
100	0.20

This large hysteresis was assumed to be due to the elastic properties and behaviour of the Invar steel.

The short-term repeatability error values of the investigated pressure sensors were calculated using equation (4.1) for each applied pressure in terms of percentage of full scale (% fs).

$$\text{Rep. error \% fs} = \frac{p_{V_{\max}} - p_{V_{\min}}}{2 \text{ mV/V}} \times 100 \quad (4.1)$$

In equation (4.1) $p_{V_{\max}}$ and $p_{V_{\min}}$ are the maximum and the minimum electrical output values of the transducer, respectively for each pressure point in the four measurement cycles.

The errors found for UMEPT4340-01 are given in Table 4.2 and errors found for UMEPTIn-01 illustrated with graphics in Figure 4.5 (within four cycles, all measured in one day). The wide variation of repeatability errors, seen in Figure 4.5, could be attributed to the properties of the material, such as strain/stress relaxation and hysteresis, which is also suggested by the large figures in Table 4.2.

Table 4.2 Calculated repeatability error values of UMEPT4340-01

p (/ MPa)	Repeatability error (/ % fs)
0	0.02
20	0.03
40	0.04
60	0.04
80	0.04
100	0.05
120	0.03
140	0.03
150	0.03

It follows from these considerations that Invar steel is not a good choice as a high-pressure transducer material.

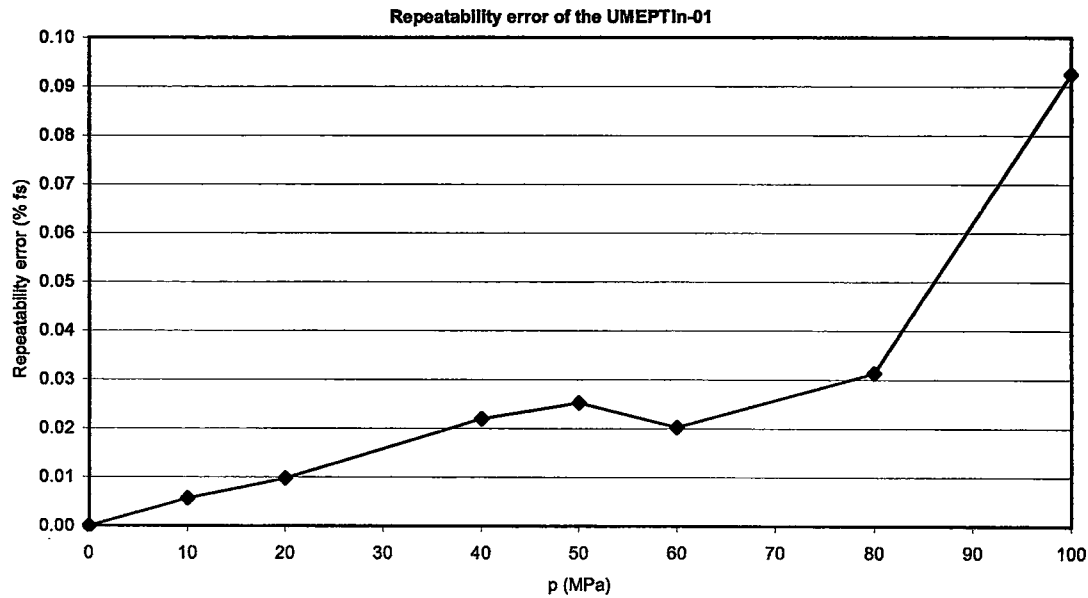


Figure 4.5. The repeatability error graph of the UMEPTIn-01

Having considered the best result in each case, the results for the metrological characterization of the hardened AISI 4340 steel made pressure transducers can be summarized as below:

- Transducer zero return shift (after one cycle of complete calibration period) 0.02 MPa
- Transducer zero drift -on no load (for 96 hours): 0.01 MPa
- Transducer creep at 150 MPa for twenty minutes: 0.08 % - 0.09 % full scale
- Short-term repeatability: 0.02 % full scale
- Long term repeatability (for 10 days and only for UMEPT4340-05): 0.1 % full scale
- Hysteresis: 0.1 % full scale

Having considered the best result in each case, the results for the metrological characterization of the invar steel made pressure transducers can be summarized as below:

- Transducer zero return shift (after one cycle of complete calibration period): 0.17 MPa
- Transducer zero drift -on no load (for 96 hours): 0.01 MPa
- Transducer creep at 100 MPa for twenty minutes: 0.52 % - 0.55 % full scale

-Short-term repeatability: 0.09 % full scale

-Hysteresis: 3 % full scale

The linearity behavior of the transducers were shown in Figures 4.6 and 4.7 for AISI 4340 steel and Invar steel pressure transducers respectively which were derived directly from the experimental measurements.

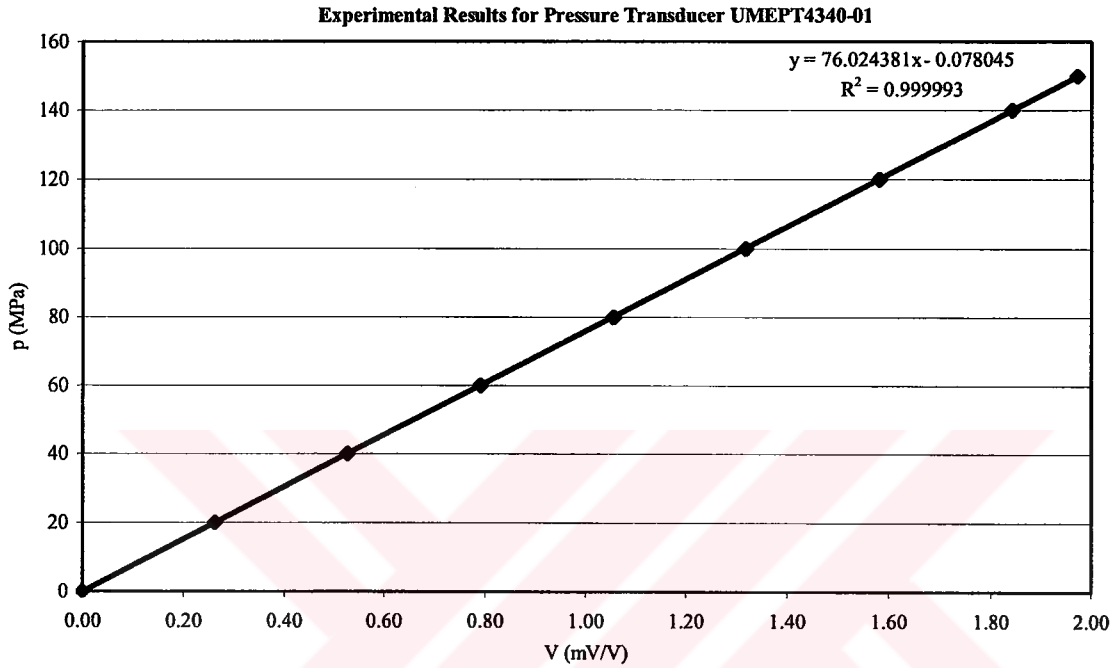


Figure 4.6. Linearity behavior of the UMEPT4340-01

As it was explained in Section 2.3.4, the measurement uncertainty of the investigated pressure transducers was individually determined by using equations (4.2), (4.3) and (4.4):

$$u_c(\text{tra}) = \sqrt{u^2(\text{rep}) + u^2(\text{hys}) + u^2(\text{int}) + u^2(\text{zero}) + u^2(\text{res})} \quad (4.2)$$

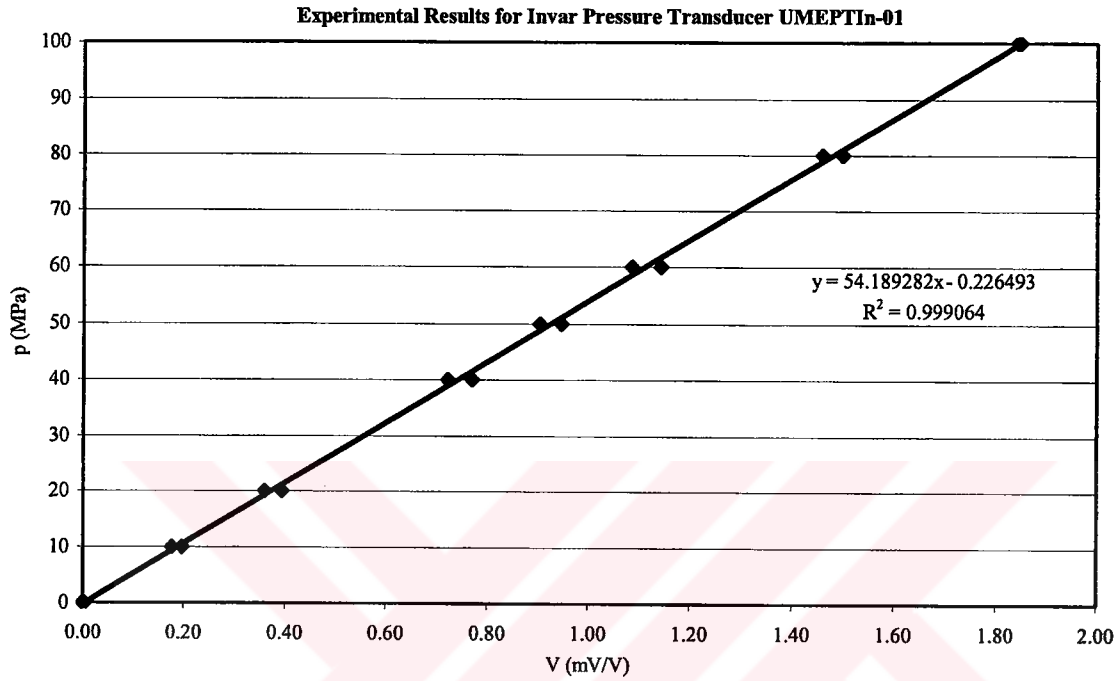
Respective components for uncertainty calculation are uncertainties due to the repeatability error (u_{rep}), hysteresis error (u_{hys}), linearity (interpolation) error (u_{int}), maximum zero drift (u_{zero}) and resolution error (u_{res}).

$$U(\text{tra}) = k_c \times u_c(\text{tra}) \quad (4.3)$$

The relative uncertainty of the calibration (U) will be determined by considering the uncertainty of the dead weight tester (U_{ref}) as follows:

$$U = \sqrt{U^2(\text{tra}) + U^2(\text{ref})} \quad (4.4)$$

Resultant calculated uncertainty values with respect to full scale at 95 % confidence level ($k_c = 2$) for investigated pressure transducers are 0.1 % for active element with AISI 4340 steel and 2 % for active element with Invar steel respectively.



5. CONCLUSIONS AND DISCUSSION

The average of the measured output strain values ($\mu\text{m}/\text{m} = \mu\epsilon$) and the average strain values on the area where the strain gauges were bonded, the FEM results were compared with each other. Table 5.1 shows this comparison as a relative difference in full scale (1850 microstrain for ϵ_c , circumferential strain and 1000 microstrain for ϵ_a , axial strain) percentage (% fs) for one of the hardened AISI 4340 steel made pressure transducer (UMEPT4340-01).

Table 5.1. Comparison of the average measured output values with the average of derived FEM strain results of the UMEPT4340-01

p (/ MPa)	Measured output (/ $\mu\epsilon$)		FEM output (/ $\mu\epsilon$)		Rel. difference (/ % fs)	
	ϵ_c	ϵ_a	ϵ_c	ϵ_a	$\Delta\epsilon_c$	$\Delta\epsilon_a$
60	390.9	-693.8	347.0	-704.0	4.4	0.6
100	664.4	-1229.9	566.0	-1174.0	9.9	-3.0
150	1006.3	-1846.5	880.0	-1757.0	12.6	-4.9

The reasons for the greater difference between the measured and the FEM output for the circumferential strain values are:

- Errors due to the misalignment of the strain gauge during the bonding process and the thickness of the bonding material.
- Errors due to the measuring principle of the strain gauge. The strain in y-direction affects the strain in x-direction and it is not possible to detect the strains both in x and y-direction with the strain gauge at the same time.
- The error limit of the numerical model was estimated as between 1 and 5 % of the full scale when the results were compared with the analytical solution of the model.
- The strain gauges were bonded on a flattened surface. If it were bonded on a cylindrical surface, then the FEM analysis could have been realized in cylindrical

coordinate system. Hence, there would have been the possibility of increasing the compatibility of the numerical and experimental results.

To improve the repeatability of the transducer, the prototype has been modified with an aim to reduce the number of mechanical contact inside the transducer. For this purpose, the plugs used on both sides were processed in a lathe to form a groove on the surface with a diameter of 13.1 mm and a depth of 1.5 mm. With this modification, the contact between the plug and the active element could be avoided at zero pressure. Some additional test measurements were performed with the UMEPT4340-05. After the measurements it was seen that this modification reduced the short-term repeatability error from 0.02 % fs to 0.01 % fs and the stabilization period of the indicated electrical output value from 30 seconds to around 15 seconds. All these further investigations have demonstrated that the prototype can be improved with better design considerations and better material selection and maybe with better built-in electronics for initial zero balance, zero and span shift with temperature change.

The design, modelling, stress and the strain analysis of the manufactured high-pressure strain gauge transducer prototypes, by using the FEM, have been described. The metrological characteristics have also been evaluated. The comparison values in Table 5.1 demonstrate that the numerical and experimental output values are in good agreement (the relative difference is less than 13 % of the full scale) with each other. Therefore FEM can be used efficiently for designing purposes.

The FEM results illustrated with the graphics in Figures 3.18 and 3.19 show that the pressure applied on the O-ring side up to the end of the O-ring groove affected the strain values on the area where the strain gauges were bonded by only a few microstrains.

When we moved the O-ring position 2 mm backwards and forwards from its original position, using the FEM, it was also seen that the stress and strain distribution on the active element changed in the area closer to the O-ring groove but it does not significantly affect the strain values on the area where the strain gauges were bonded.

Figures 5.1 and 5.2 shows the comparison of the repeatability and hysteresis errors of the constructed pressure transducer prototypes. Especially considering the compared hysteresis results it can be easily said that Invar steel is not an appropriate material for high pressure transducer applications.

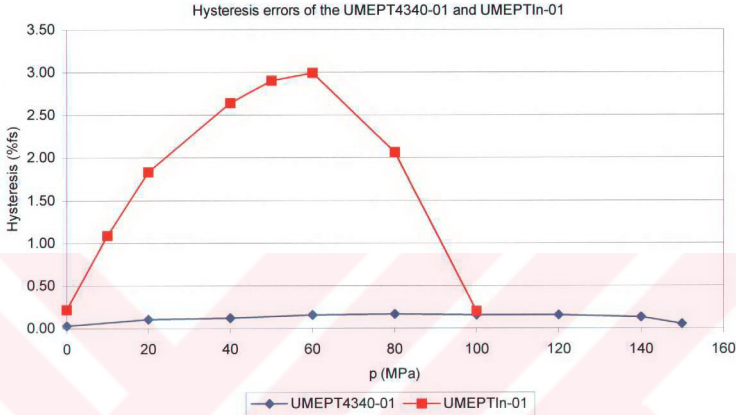


Figure 5.1. Comparison of the hysteresis error results of the transducer prototypes

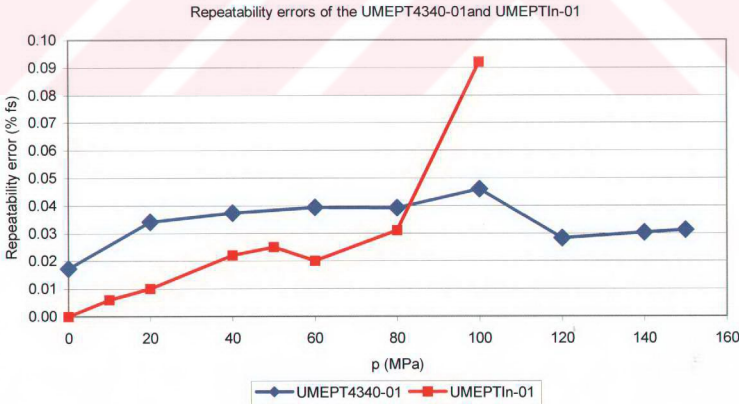


Figure 5.2. Comparison of the repeatability error results of the transducer prototypes

The experiments show a repeatability error of less than 0.02 % fs for hardened AISI 4340 steel made and less than 0.1 % fs for invar steel made pressure transducers. The

hysteresis error is less than 0.1 % fs for hardened AISI 4340 steel made and 3 % fs for invar steel made pressure transducers. All these experimental results make evident that hardened AISI 4340 steel is an appropriate material for the pressure transducer application to measure gauge pressure of up to 150 MPa with an uncertainty of 0.1 % of the full scale. The 0.05% uncertainty of commercially available transducers working in the pressure range from 1 MPa to 50 MPa [22] is better than the manufactured prototypes. On the other hand, this prototype has the advantage of maintaining the 0.1 % fs uncertainty for even higher working pressures, which can be extended as high as 500 MPa [1], [2] with appropriate material.

The experiments also showed that temperature is not an influence parameter, since the full Wheatstone bridge was built with low temperature coefficient (6 ppm °C⁻¹) strain gauges. The thermal variations in the active element were within ± 0.2 °C for each calibration cycle.

Some of these studies written in this document also have recently been reviewed in [23].

Further studies can be made for the hysteresis and creep compensation to improve the performance and metrological characteristics of these transducers. Such transducers can be easily designed and analysed by the FEM.

REFERENCES

- [1] Molinar, G., Maghenzani, R. and Wisniewski, R., 1999. Free-cylinder strain gauge, pressure transducers up to 2 GPa, *Physica B Condensed Matter* **265**, 239-245.
- [2] Molinar, G., Bray, A., Wisniewski, R., Maghenzani, R. and Nespoli L., 1998. Pressure transducer for pressure measurements up to 0.5 GPa, *Measurement* **24**, 161-172.
- [3] Allgeier, T., 1994. Factors influencing the mechanical hysteresis in stainless steel load cells, *PhD Thesis*, University of Glamorgan, UK.
- [4] The Technical Staff of Measurements Group, Inc., 1998. Strain Gage Based Transducers, 2nd Edition, pp. 37-38, USA.
- [5] ASM International, 1998. Heat Treater's Guide, 2nd Edition, pp. 690-692, USA.
- [6] National Physical Laboratory, 1998. Guide to the Measurement of Pressure and Vacuum, The Institute of Measurement and Control, London.
- [7] EAL-G26, 1997. Calibration of pressure balances, 1st Edition, *European cooperation for Accreditation of Laboratories*, Paris.
- [8] Wheeler, J.A. and Ganji, R.A., 1996. Introduction to Engineering Experimentation, pp. 233-234, Prentice Hall, Inc., New York.
- [9] Doebelin, E.O., 1990. Measurement Systems Application and Design, pp. 38-40, Mc Graw-Hill Inc., New York.
- [10] Sawla, A., 1994. PTB-Mitteilungen 104, *Guidance for the determination of the best measurement capability of force calibration machines and uncertainty of calibration results of force measuring devices*, pp. 237-242, Braunschweig.
- [11] Perry, C.C. and Lissner, H.R., 1962. The Strain Gauge Primer, 2nd edition, pp. 1-3, Mc Graw-Hill Inc., New York.
- [12] Dally, J.W. and Riley, W.F., 1991. Experimental Stress Analysis, 3rd edition, pp. 223-224, Mc Graw-Hill Inc., New York.
- [13] Hoffmann, K., 1989. An Introduction to Measurements Using Strain Gauges, pp. 145-153, Hottinger Baldwin Messtechnik GmbH, Druckerei Druck, Alsbach.
- [14] Timoshenko, S.P. and Goodier J.N., 1970. Theory of Elasticity, 3rd edition, pp. 65-71, Mc Graw-Hill International Editions, Singapore.
- [15] Sherman, W.F. and Stadtmuller A.A., 1987. Experimental Techniques in High Pressure Research, pp. 7-14, John Wiley and Sons Ltd., Great Britain.
- [16] Ed Fagan Inc., 1999. Invar Steel Product Sheet, USA.

- [17] **Measurements Group Inc.**, 1989. Strain Gauge Selection, Technical Note TN 505-4, USA.
- [18] **Micro-Measurements Division**, 1992. Transducer Class®, Strain Gages Bondable Resistors Installation Accessories, TC 116-4, Measurements Group, USA
- [19] **Micro-Measurements Division**, 1992. Strain Gage Installation Procedures for Transducers, Measurements Group, USA
- [20] **John Wiley and Sons**, 1998. A Concise Guide for the Professional Engineer, 2nd edition, pp. 145-146, John Wiley and Sons, USA
- [21] **ANSYS Inc.**, 1998. ANSYS version 5.6 Software Help, Solid 45, USA.
- [22] **Paul, H.**, 1993/94. New high-precision reference pressure transducers from 1 MPa to 50 MPa, *Metrologia*, **30**, 677-681.
- [23] **Orhan, M.H., Doğan, Ç., Kocabaş, H., Tepehan, G.**, 2001. Experimental strain analysis of the high pressure strain gauge pressure transducer and verification by using a finite element method, *Measurement Science and Technology*, **12**, 335-344.



BIOGRAPHY OF THE AUTHOR

M. Haluk Orhan was born in Ankara, in 1971. He graduated from Istanbul Technical University in 1992, as a Physics Engineer, and earned his Master of Science degree at the same university in 1995. He worked as an expert researcher at the National Pressure Metrology Laboratory of the TÜBİTAK-Ulusal Metroloji Enstitüsü (UME) between 1993 and 2000. During his professional career at UME, he had 5 articles published in international journals and presented 9 research projects at the national and international metrology conferences in the field of pressure metrology. He got married in 1999. He received his title of Microsoft® Certified System Engineer in 2000.

