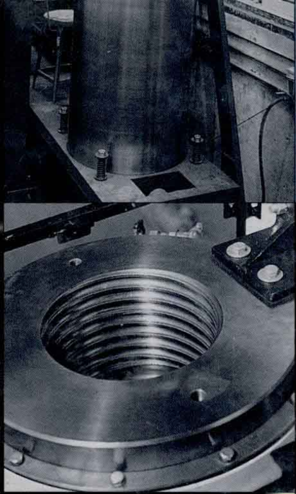
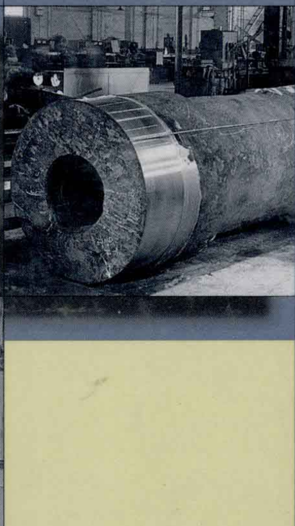


High Pressure Vessels



Donald M. Fryer • John F. Harvey

High Pressure Vessels

Donald M. Fryer
Consultant
Fairview, PA

John F. Harvey
Consultant
Akron, OH

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Preface

The pronounced increase in the petrochemical, isostatic pressing, deep sea drilling, deep diving submersible, and cultured crystal industries with their requirements for extremely high pressure vessels has led to numerous attempts to foster their design and safety by regulatory safety codes and standards. The latest of these have been those of the American Society of Mechanical Engineers, namely the Boiler and Pressure Vessels Code, Section VIII, Division 3, and the High Pressure Systems Standard, HPS-1994.

It is the purpose of this text to treat the unique prerequisites of these vessels from a stress, economic, safety, and hazard viewpoint which may demand specific design and construction features. The text is intended for use by engineering students interested in pressure technology and also by practicing engineers who may be new to the field of high pressure and its wide variety of applications. The book introduces technology, and presents some application concepts. It is not, however, intended to be a design recipe book.

Accordingly, Chapter 1 introduces the economics of design by way of safety tradeoffs, as well as their significance and function in establishing material property, stress, and quality assurance requirements.

Chapter 2 establishes the basic stress analysis of thin-walled vessels such as cylinders, spheres, ellipsoids, cones, and tori. Likewise, the analysis of thick-walled cylinders, including compression fit construction, autofrettage and other techniques are developed which have made possible designs for higher pressures. In the thicker proportions, the importance of residual stress and the Bauschinger material effect are paramount.

Chapter 3 gives an overview of material properties and their physical behavior. Here the significance of various theories of failure under tensile, shear, and fatigue conditions as applicable to high pressure vessels is examined.

Chapter 4 examines in more detail the subject of fatigue and the potential disastrous effects of cyclic loading, as opposed to simple static safety factor analysis. Subjects such as classical endurance analysis, material toughness, crack initiation, crack growth, crack arrest, and cumulative damage are included.

Chapter 5 discusses special considerations associated with specific techniques of design and construction of high pressure vessels.

Chapter 6 reviews hazards and safety considerations related to installing, operating, and maintaining high pressure vessels. This is presented from the perspective of the energy confined in a vessel which might be released, possibly catastrophically, if a vessel should fail suddenly.

Chapter 7 discusses several methods of analysis which may be used during the design of a high pressure vessel and also for any subsequent appraisal which may be associated with a periodic in-service examination.

John F. Harvey is a consulting engineer in pressure vessel design. In contributing to the preparation of this book, he has drawn from his previous publications, lecture notes used in teaching at the University of Akron, engineering experience with the Babcock & Wilcox Company, and as a member of numerous committees of the American Society of Mechanical Engineers, and as chairman of the ASME Pressure Vessels Research Committee. He is a Fellow of the ASME.

Donald M. Fryer is a consultant dealing with high-pressure vessel engineering and safety. His background for contributing to this book includes employment at E.I. du Pont de Nemours & Co., Inc., and Autoclave Engineers, Inc.; conducting seminars on high pressure equipment; and extensively participating in High Pressure Technology, and Codes and Standards committees of the American Society of Mechanical Engineers. He also is a Fellow of the ASME.

John F. Harvey
Donald M. Fryer

Nomenclature

A	Cross section, moment arm
A_c	Critical crack depth
a	Crack depth
b	Subscript for bending
a, b, c, d	Distances
BEF	Bauschinger effect factor
C	Coefficient, factor
c	subscript for *critical*, and *compression*
CVN	Charpy V-notch energy
d	Diameter
E	Modulus of elasticity
e	Unit strain, thickness
e_x, e_y, e_z	Unit strain in x, y, and z directions
$e_{Y.P.}$	Yield point strain
F	Force
h	Thickness, height
h_t	Thread tooth height
i	Subscript for *inside*
ID	Inside diameter
IR	Inside radius
J	J-integral test
K	Stress intensity factor, ratio of either OD/ID or OR/IR
k	Ratio of specific heats
K_t	Stress concentration factor
K_{1c}	Fracture toughness
ΔK	Stress intensity factor range
ℓ	Length, span, subscript for *longitudinal*
M	Bending moment

m	exponent
N	Number of design cycles
N_R	Neutral radius
n	Strain hardening exponent, number of threads, number of secondary design operating cycles
o	Subscript for *outside*
OD	Outside diameter
OR	Outside radius
P	Pressure release
p	Pressure
p_t	Thread pitch
R	Radius
R_k	Ratio of residual K's or ΔK's
r	Radius
S	Stress intensity
T	Twisting moment, temperature
ΔT	Temperature difference
t	Thickness, and subscript for *tension*
U	Strain energy
u	Rate of strain, displacement in x direction
V	Volume
W	Weight
Y	Ratio of either OD/ID or OR/IR
x, y, z	Rectangular coordinates
ksi	Kips per square inch (1 kip = 1,000 lb)
psi	Pounds per square inch
α	Angle, coefficient of thermal expansion
β	Angle, damping factor
γ	Shearing strain, liquid compressibility
Δ	Distance, deflection, difference
δ	Total distance, total deflection, total elongation
θ	Angle
μ	Poisson's ratio
ρ	Distance, radius
σ	Unit normal stress
$\sigma_1, \sigma_2, \sigma_3$	Principal stresses

σ_x, σ_y, σ_z	Unit normal stresses on planes perpendicular to the *x*, *y*, and *z* axes
σ_E	Unit stress at the endurance limit
σ_{ult}	Ultimate stress
$\sigma_{Y.P.}$	Yield point stress
τ	Unit shear stress
τ_{xy}, τ_{yz}, τ_{zx}	Unit shear stresses on planes perpendicular to the *x*, *y*, and *z* axes, or parallel to the *z* and *x* axes
$\tau_{Y.P.}$	Yield point stress in shear
Φ	Angle, coefficient
Σ	Sum of stress intensities

Si Conversions

Quantity	To Convert From	To	Multiply by	
Pressure	psi	MPa	6.894 757	E-03
Temperature	°F	°C	(°F − 32)/1.8	
Temp. difference	°F	°C	5.555 555	E-01
Linear dimension	in.	m	2.54	E-02
	ft	m	3.048	E-01
Area	in.^{2}	m^2	6.451 6	E-04
	ft^2	m^2	9.290 304	E-02
Volume	ft^3	m^3	2.8317	E-02
Modulus	psi	MPa	6.894 757	E-03
Weight	lb	kg	4.535 924	E-01
	ton	metric ton	9.07 847	E-01
Force	lb	N	4.448 222	
Moment	lb ft	Nm	1.355 818	
	lb in	Nm	1.129 848	E-01
Stress	psi	MPa	6.894 757	E-03
Toughness	ksi $\sqrt{\text{in}}$	MPa $\sqrt{m}$	1.098 843	
Energy	ft lb	J	1.355 818	
Coeff. of expansion	$\frac{\text{in.}}{\text{in.°F}}$	$\frac{\text{m}}{\text{m°C}}$	1.8	
Enthalpy	$\frac{\text{Btu}}{\text{lb}}$	$\frac{\text{J}}{\text{kg}}$	2.326	E+03
Specific volume	$\frac{\text{ft}^3}{\text{lb}}$	$\frac{\text{m}^3}{\text{kg}}$	6.242 8	E-02
Angle	degrees	radians	1.745 329	E-02

Contents

1

High Pressure Vessels

1.1 Introduction

Ultra high strength vessels first came on the scene in the thirteenth century with the advent of the military cannon. Little was known of their stress analysis and design, and sizing was on a trial-and-error or proof test basis. They were usually made of bronze or iron castings with their associated brittle structure and defects. Hence, explosions were frequent and casualties common. While efforts were continually made to increase their safety and reliability, there was no understanding of the stresses in their barrels until 1833, when Lamé established the elastic stress distribution in a thick-walled cylinder, which made it possible to understand the strength of high pressure vessels. This led to the analysis and advantages of shrink fits, autofrettage, wire or strap windings, banded, and other types of vessels. Today ballistic requirements have been joined by those of industry for vessels in the chemical, petrochemical, isostatic pressing, undersea mining, deep-diving submersibles, down well and deep ocean simulation, waterjet cutting, cultured crystal production, supercritical extraction, and food sterilizing industries in the 10,000-psi to approximately 200,000 psi range.

While it may seem adequate to simply define *pressure vessel* as a container for confining a fluid having an internal energy greater than in its free state, one must recognize that in high pressure applications, the sources of added energy also are subject to the same kinds of consideration as this book discusses for vessels.

Further, whether the added fluid energy is from any, or any combination, of compression, temperature, or reaction; once at high pressure, the fluid physical behavior may be quite different from that of low pressure fluids. For example, as pressure increases, liquids become significantly compressible, whereas gases become less compressible than the so-called "perfect gas laws" will predict. As a result, proper sizing of equipment becomes equally important as providing adequate structural integrity. This book, however, deals only with the structural integrity aspect of these considerations.

It is necessary to understand the phenomena of pressure vessel failure in order to prevent adopting of requirements which can be self-defeating, such as high factors of safety. This is particularly important with the use of high strength materials of low toughness, which have much less forgiveness for the presence of defects and crack propagation than do those of lower strengths. We now have the knowledge in stress analysis, metallurgy, and inspection techniques to rationalize their significance, rather than blindly complying with an arbitrary absolute code, standard, or existing rule requirement. For instance, surface discontinuities in the forms of mechanical damage, material defects, weld shrinkage, and machined undercuts may become sites of high stress concentration, and subsequently main sources of cyclic loading fatigue failure. Grinding such areas to restore a smooth blended surface, even though it creates a minor reduction of structural thickness, will significantly restore much of the lost fatigue strength. Further, such local reduction in thickness will have a negligible effect on static bursting strength of the vessel.

Likewise, if there are longitudinal defects or scratches on the inside surfaces, pressure acting on the flanks tends to open them up and this will increase the hoop tensile stress at the notch tip by an amount equal to the acting pressure, or twice the nominal shear stress acting at the bore. It may be difficult to grind out internal surface defects, so it may be necessary, when practical, to ream or bore them out. This increases the internal diameter slightly but, again, it may be preferable in light of its significance to other alternatives. Other methods involving coatings with elastomers or burnishing may not be compatible with temperature or other environmental conditions. Figure 1.1.1 illustrates the above discussion about surface discontinuities.

1.2 Economic Considerations and Cost Reduction

The economic trend in all industries is toward large size equipment in order to reduce capital investment and operating cost by eliminating the duplicity of ser-

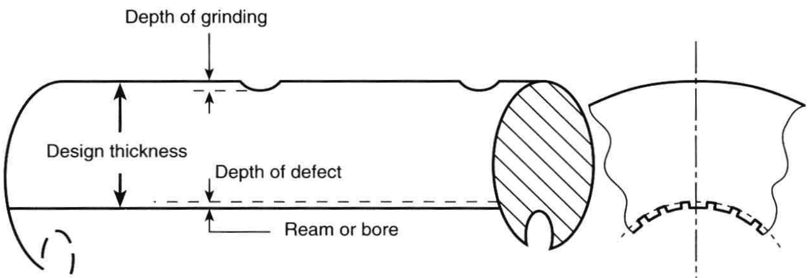

Figure 1.1.1. Pressure Vessel Grinding and Boring to Remove Defects

vicing appurtenances, controls, and so forth. Pressure vessels follow this same economic trend. There are exceptions, of course, when engineering and material limitations, as well as economic ones, still may dictate designing multiple smaller vessels rather than one large one. Construction must be sufficiently strong and reliable, yet embody the maximum saving in materials. This, together with present-day commercial competition, presents a formidable economic factor in high pressure vessel design. Reduction in weight involves an increase in allowable design stresses of materials which can be permitted with safety only on the basis of a thorough stress analysis of the vessel, and after careful experimental investigation of the properties of the material in its projected environment.

High pressure vessel construction has three major requirements:

1. Engineering design knowledge
2. High strength materials of suitable toughness and fatigue resistance
3. Quality assurance to confirm that 1 and 2 are pursued (Fig. 1.2.1)

While maintaining present costs or reducing costs modestly is a routine operation, slashing them radically to revive a dwindling market may require a new solution to the cost compatibility equation:

$$\text{Simplicity} = \text{Strength} = \text{Economy}$$

employing creative engineering, new materials, and fabrication methods as input data. It must be remembered that all the data to this equation are time dependent, as is the answer.

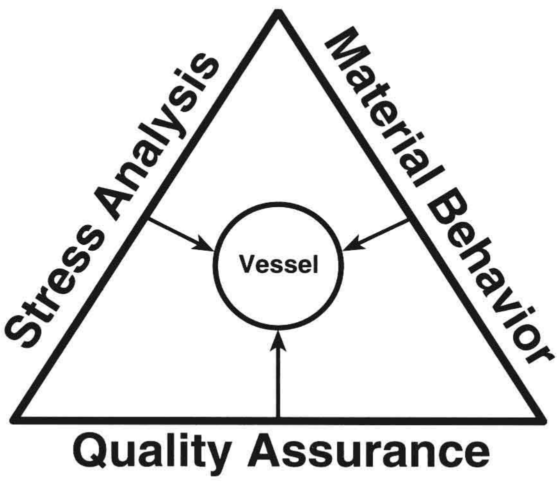

Figure 1.2.1. Requirements for High Pressure Vessels

1.3 Stress Significance

It is important not only to determine the value of a stress, but to interpret its meaning or significance as well—the two go hand-in-glove. For instance, stress concentrations are of little importance in low strength steels with their high toughness when only a static load is involved; whereas, the same material may have severe limitations if the loading is cyclical. This condition is known as *fatigue*. Likewise, high strength materials, with their usually low toughness, is most susceptible to stress concentrations both under static and cyclical loading because the plastic property of the material affords little internal strain redistribution or gross distortion to an external shape readjustment to present a more favorable shape. Hence, crack growth associated with fatigue is the result of high local stresses occurring at the tip of a notch or crack seeking equilibrium with resistance offered by the material toughness property. The stress analysis of cracks is called *fracture mechanics*.

The basic equations for determining stresses are based on the assumption that they are caused only by external loads; and residual stresses set up in the fabrication or construction processes, such as weld shrinkage, casting cooling, and metal heat treatment, are not considered. Although these stresses are secondary ones, since their magnitude is self-limiting (they are not produced by unrelenting loads), they may be of great significance in high strength brittle materials and even in ductile material when the material is subject to cyclical loading. Equally important is the danger of creating, in conjunction with those from the applied loading stresses, a three-dimensional stress pattern in thick sections that are restrictive to the redistribution of high localized peak stresses through yielding. It is for this reason that stress relieving of welded thick vessels, usually required by construction codes, is more important than in thin ones in which the state of stress is essentially two-dimensional.

1.4 Factors of Safety and Their Optimum Value

A single parameter to describe pressure vessel design progress is best measured by the total advancement in knowledge of the three basic elements; namely, engineering, materials, and fabrication that establish safety and costs. Such a single parameter is usually called a factor of safety and is used to indicate proximity to failure or economic survival. It is well to assign this factor to a physical property of the structure, such as the material, in order to permit a measure of time and environmental effects on its degradation (both original, in fabrication and in-service). A material strength factor such as ultimate strength, yield strength, toughness, or fatigue strength commensurate with the mode of failure is a good selection.

Brittle materials may be defined as those in which the yield and ultimate strengths are the same. High strength materials approach this, and are most meaningfully characterized by their yield strength. This is the one material property