PRESSURE VESSEL HANDBOOK

FIFTH EDITION

PRESSURE VESSEL
HANDBOOK

PRESSURE VESSEL HANDBOOK

with foreword by

PAUL BUTHOD

Professor of Chemical Engineering
University of Tulsa

by

EUGENE F. MEGYESY
Designer

PRESSURE VESSEL HANDBOOK
PUBLISHING INC.
P. O. Box 35365, Tulsa, OK 74135

Library of Congress Catalog Card Number
79-84601
ISBN 0-914458-08-6

First Edition October 1972
Second Edition July 1973
Third Printing June 1974
Third Edition March 1975
Fourth Edition October 1977
Fifth Printing May 1979
Fifth Edition January 1981

Printed in the United States of America

FOREWORD

Engineers who design equipment for the chemical process industry are sooner or later confronted with the design of pressure vessels and mounting requirements for them. This is very often a frustrating experience for anyone who has not kept up with current literature in the field of code requirements and design equations.

First he must familiarize himself with the latest version of the applicable code. Then he must search the literature for techniques used in design to meet these codes. Finally he must select material properties and dimensional data from various handbooks and company catalogs for use in the design equations.

Mr. Megyesy has recognized this problem. For several years he has been accumulating data on code requirements and calculational methods. He has been presenting this information first in the form of his "Calculation Form Sheets" and now has put it all together in one place in the Pressure Vessel Handbook.

I believe that this fills a real need in the pressure vessel industry and that readers will find it extremely useful.

Paul Buthod

PREFACE

This reference book is prepared for the purpose of making formulas, technical data, design and construction methods readily available for the designer, detailer, layoutmen and others dealing with pressure vessels. Practical men in this industry often have difficulty finding the required data and solutions, these being scattered throughout extensive literature or advanced studies. The author's aim was to bring together all of the above material under one cover and present it in a convenient form.

The design procedures and formulas of the ASME Code for Pressure Vessels, Section VIII Division 1 have been utilized as well as those generally accepted sources which are not covered by this Code. From among the alternative construction methods described by the Code the author has selected those which are most frequently used in practice.

In order to provide the greatest serviceability with this Handbook, rarely occurring loadings, special construction methods or materials have been excluded from its scope. Due to the same reason this Handbook deals only with vessels constructed from ferrous material by welding, since the vast majority of the pressure vessels are in this category.

A large part of this book was taken from the works of others, with some of the material placed in different arrangement, and some unchanged.

The author wishes to acknowledge his indebtedness to Miss Christiane Fries, Mr. Arthur L. Wade and Mr. Glenn Warren for their assistance in preparing the manuscript, to the American Society of Mechanical Engineers and to the publishers, who generously permitted the author to include material from their publications.

Suggestions and criticism concerning some errors which may remain in spite of all precautions shall be greatly appreciated. They will contribute to the further improvement of this Handbook.

Eugene F. Megyesy

CONTENTS

PART I.

DESIGN AND CONSTRUCTION OF PRESSURE VESSELS

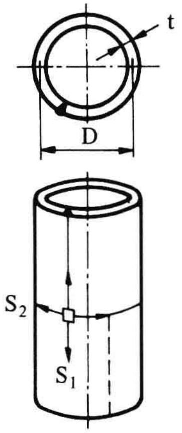
IN REFERENCES THROUGHOUT THIS BOOK "CODE" STANDS FOR ASME (AMERICAN SOCIETY OF MECHANICAL ENGINEERS) BOILER AND PRESSURE VESSEL CODE SECTION VIII RULES FOR CONSTRUCTION OF PRESSURE VESSELS, DIVISION 1 – AN AMERICAN STANDARD.

STRESSES IN CYLINDRICAL SHELL

Uniform internal or external pressure induces in the longitudinal seam a unit stress two times larger than in the circumferential seam because of the geometry of the cylinder.

A vessel under external pressure, when other forces (wind, earthquake, etc.) are not factors, must be designed to resist the circumferential buckling only. The Code provides the method of design to meet this requirement. When other loadings are present, these combined loadings may govern and heavier plate will be required than the plate which was satisfactory to resist the circumferential buckling only.

The formulas below give the compression stress due to external pressure and tension stress due to internal pressure.



FORMULAS

CIRCUMFERENTIAL JOINT	LONGITUDINAL JOINT
$S_1 = \dfrac{PD}{4t}$	$S_2 = \dfrac{PD}{2t}$

NOTATION

D = Mean diameter of vessel, inches
P = Internal or external pressure, psi
S_1 = Longitudinal stress, psi
S_2 = Circumferential (hoop) stress, psi
t = Thickness of shell, corrosion allowance excluded, inches

EXAMPLE

Given D = 96 inches
P = 15 psi
t = 0.25 inches

$$S_1 = \frac{PD}{4t} = \frac{15 \times 96}{4 \times 0.25} = 1440 \text{ psi}$$

$$S_2 = \frac{PD}{2t} = \frac{15 \times 96}{2 \times 0.25} = 2880 \text{ psi}$$

INTERNAL PRESSURE

1. OPERATING PRESSURE

The pressure which is required for the process, served by the vessel, at which the vessel is normally operates.

2. DESIGN PRESSURE

The pressure used in the design of a vessel. It is recommended to design a vessel and its parts for a higher pressure than the operating pressure. A design pressure higher than the operating pressure with 30 psi or 10 percent, whichever is greater, will satisfy this requirement. The pressure of the fluid and other contents of the vessel should also be taken into consideration. See tables on page 27 for pressure of fluid.

3. MAXIMUM ALLOWABLE WORKING PRESSURE

The internal pressure at which the weakest element of the vessel is loaded to the ultimate permissible point, when the vessel is assumed to be:

- (a) in corroded condition
- (b) under the effect of a designated temperature
- (c) in normal operating position
- (d) under the effect of other loadings (wind load, external pressure, hydrostatic pressure, etc.) which are additive to the internal pressure

A common practice followed by many users and manufacturers of pressure vessels is to limit the maximum allowable working pressure by the head or shell, not by small elements as flanges, openings, etc.
See tables on page 26 for maximum allowable pressure for flanges.
See tables on page 106 for maximum allowable pressure for pipes.

The term, maximum allowable pressure, new and cold, is used very often. It means the pressure at which the weakest element of the vessel is loaded to the ultimate permissible point, when the vessel:

- (a) is not corroded (new)
- (b) the temperature does not affect its strength (room temperature) (cold)

and the other conditions (c and d above) also need not to be taken into consideration.

4. HYDROSTATIC TEST PRESSURE

One and one-half times the maximum allowable working pressure or the design pressure when calculations are not made to determine the maximum allowable working pressure.

If the stress value of the vessel material at the design temperature is less than at the test temperature, the hydrostatic test pressure should be increased proportionally.

In this case, the test pressure shall be:

$$1.5 \times \begin{matrix}\text{Max. Allow. W. Press.}\\ \text{(Or Design Press.)}\end{matrix} \times \frac{\text{Stress Value S At Test Temperature}}{\text{Stress Value S At Design Temperature}}$$

Vessels where the maximum allowable working pressure limited by the flanges, shall be tested at a pressure shown in the table:

Primary Service Pressure Rating	150 lb	300 lb	400 lb	600 lb	900 lb	1500 lb	2500 lb
Hydrostatic Shell Test Pressure	425	1100	1450	2175	3250	5400	9000

Hydrostatic test of multi-chamber vessels: Code UG-99 (e)

A Pneumatic test may be used in lieu of a hydrostatic test per Code UG-100

Proof tests to establish maximum allowable working pressure when the strength of any part of the vessel cannot be computed with satisfactory assurance of safety, prescribed in Code UG-101.

5. MAXIMUM ALLOWABLE STRESS VALUES

The maximum allowable tensile stress values permitted for different materials are given in table on page 135. The maximum allowable compressive stress to be used in the design of cylindrical shells subjected to loading that produce longitudinal compressive stress in the shell shall be determined according to Code par. UG-23 b.

6. JOINT EFFICIENCY

The efficiency of different types of welded joints are given in table on page 122. The efficiency of seamless heads is tabulated on page 124.

The following pages contain formulas used to compute the required wall thickness and the maximum allowable working pressure for the most frequently used types of shell and head. The formulas of cylindrical shell are given for the longitudinal seam, since usually this governs.

The stress in the girth seam will govern only when the circumferential joint efficiency is less than one-half the longitudinal joint efficiency, or when besides the internal pressure additional loadings (wind load, reaction of saddles) are causing longitudinal bending or tension. The reason for it is that the stress arising in the girth seam pound per square inch is one-half of the stress in the longitudinal seam.

The formulas for the girth seam accordingly:

$$t = \frac{PR}{2SE + 0.4P} \qquad\qquad P = \frac{2SEt}{R - 0.4t}$$

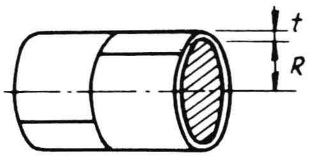
See notation on the following page.

INTERNAL PRESSURE

FORMULAS IN TERMS OF INSIDE DIMENSIONS

P = DESIGN PRESSURE OR MAX. ALLOWABLE WORKING PRESSURE PSI
S = STRESS VALUE OF MATERIAL, PSI., PAGE 135.
E = JOINT EFFICIENCY, PAGE 122
R = INSIDE RADIUS, INCHES
D = INSIDE DIAMETER, INCHES
t = WALL THICKNESS, INCHES
C.A. = CORROSION ALLOWANCE, INCHES

A



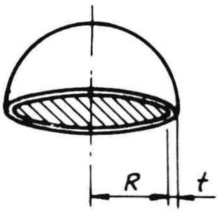
CYLINDRICAL SHELL (LONG SEAM)[1]

$$t=\frac{PR}{SE-0.6P}$$

$$P=\frac{SE\,t}{R+0.6t}$$

1. Usually the stress in the long seam is governing. See preceding page.
2. When the wall thickness exceeds one half of the inside radius or P exceeds 0.385 SE, the formulas given in the Code UA 2 shall be applied.

B



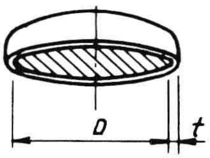
SPHERE and HEMISPHERICAL HEAD

$$t=\frac{PR}{2SE-0.2P}$$

$$P=\frac{2SE\,t}{R+0.2t}$$

1. For heads without a straight flange, use the efficiency of the head to shell joint if it is less than the efficiency of the seams in the head.
2. When the wall thickness exceeds 0.356 R or P exceeds 0.665 SE, the formulas given in the Code UA. 3. shall be applied.

C



2:1 ELLIPSOIDAL HEAD

$$t=\frac{PD}{2SE-0.2P}$$

$$P=\frac{2SEt}{D+0.2t}$$

1. For ellipsoidal heads, where the ratio of the major and minor axis is other than 2:1, see Code UA.4(C).