

Explanatory supplement to BS 5500:1988 “Specification for unfired fusion welded pressure vessels”, section 3 “Design” —

Part 4: Heat exchanger tubesheets

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Foreword

This Part of PD 6550 was prepared under the direction of the Pressure Vessel Standards Policy Committee.

This explanatory supplement to section 3 “Design” of BS 5500 “*Specification for unfired fusion welded pressure vessels*”, has been written to assist both the newcomer to pressure vessel design and the experienced vessel designer. It describes the background to BS 5500 and its theoretical basis in design. Answers are given to many of the questions that are asked about the detailed and complex rules and requirements concerning the design of pressure vessels to BS 5500.

PD 6550 is published in separate Parts.

Each Part of the supplement is current with the 1988 edition of BS 5500, the nomenclature of which has been followed throughout. It is assumed that the reader is reasonably familiar with the terms used in BS 5500.

The committee wishes to acknowledge the fundamental contribution of Mr R A McFarlane of British Petroleum International Limited, as the primary author of the text of this Part of the supplement.

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Summary of pages

This document comprises a front cover, an inside front cover, pages i and ii, pages 1 to 24, an inside back cover and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

1 Introduction

In BS 5500, heat exchangers are divided into those with floating heads or U-tubes and those with fixed tubesheets. The main difference between the two groups, as regards mechanical design, is that in the former the tube bundle is free to expand without restraint from the shell whereas in the latter there can be substantial restraint on the bundle from the shell.

The type of exchanger chosen for a duty depends on the process requirements and on the limitations imposed by materials, mechanical design and cost. In all exchangers, the tubesheet forms an important structural component. It is an essential part of the barrier between shellside and tubeside fluids. It must withstand the pressures of these fluids and must be strong enough to ensure that the numerous tube to tubesheet attachments are reliable in service and remain free from leaks.

A floating head exchanger is shown in Figure 1(a). The floating head, at the left hand end in the diagram, is totally immersed (or confined) in the shellside fluid, hence the term confined floating head. Figure 1(b) shows a U-tube exchanger. It has only one tubesheet and the tubes are bent through 180°. The U-tube exchanger provides an economic unit but one where access to the bore of the tubes for cleaning or inspection is limited.

Fixed tubesheet exchangers, as shown in Figure 1(c), have two tubesheets which are attached to the shell, so the bundle cannot move independently of the shell. The tubesheets must be designed for each of the shellside and tubeside pressures, and for the load due to the thermal expansion of the bundle relative to the shell. The loads due to thermal expansion can be large and it is sometimes necessary to fit a flexible bellows to the shell.

The design of tubesheets presents three particular problems. First, a tubesheet is a highly perforated component and methods for analysing the stresses in such components are complex. Originally, the deflection performance of perforated plates could only be obtained from measurements on test samples but in recent years analytical methods for predicting stresses and deflections have become available.

Second, in floating head and fixed tubesheet exchangers, the tube bundle, tubesheets and shell together form a statically indeterminate structure. To calculate the stresses in such a structure, an analysis must be made of the interaction of all the component parts. Tubesheets in U-tube exchangers are not statically indeterminate but for reasons which are explained, their design is treated in BS 5500 as a special case of floating head exchanger design.

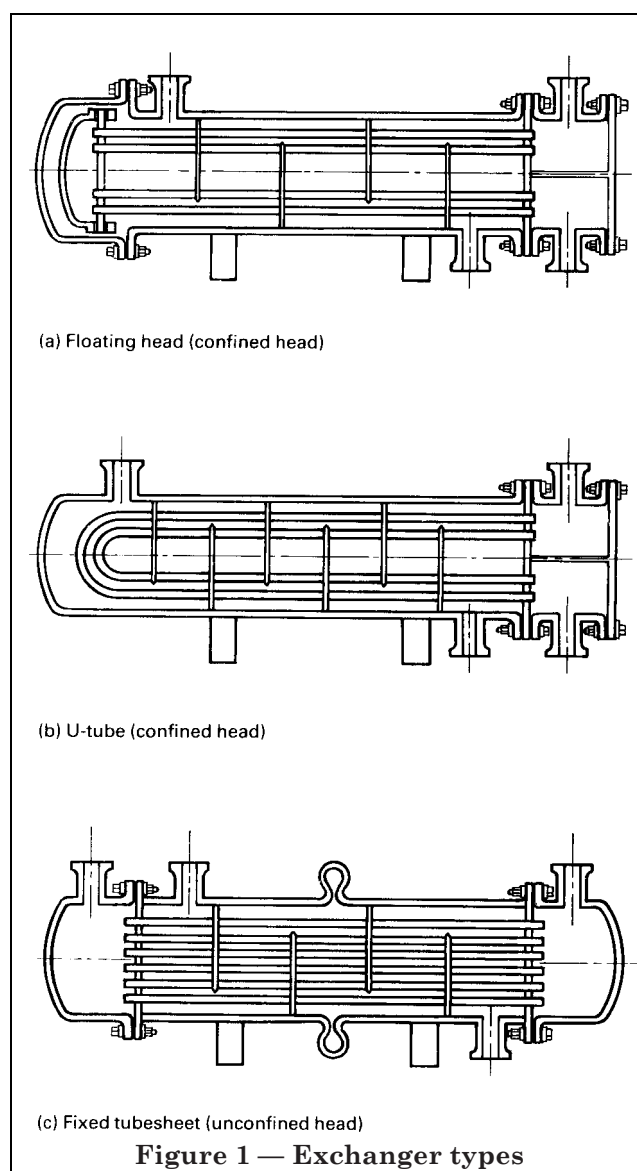


Figure 1 — Exchanger types

Third, tubesheets experience different combinations of pressure and thermal load under different phases of operation. The BS 5500 design method ensures that the most onerous combination is selected and that the stresses are correctly assessed for the type of loading being applied.

NOTE The title of the BSI publication referred to in this Part of PD 6550 is listed on the inside back cover. References to publications (other than BSI publications) are identified in the text by numbers in square brackets and are listed in clause 7.

2 A brief history and a comparison of design methods

Although shear must also be checked, it is usually the radial bending stresses which determine tubesheet thickness. The equation for the maximum radial bending stress $(\sigma_r)_{\max}$ in a flat solid circular plate laterally loaded by a uniform pressure is derived by Timoshenko and Woinowsky-Krieger [1]. Solved for thickness, it has the following form:

$$e = \text{constant } D \sqrt{\frac{p}{(\sigma_r)_{\max}}} \quad (1)$$

For this simple case of plate bending, the constant depends solely on the restraints applying at the plate edge. It has the value 0.566 when simply supported and 0.433 when clamped.

The first rules for the design of tubesheets in shell and tube heat exchangers appeared in 1941. They were presented using the same form of equation as equation (1) with allowable values being given for $(\sigma_r)_{\max}$. The formulae were empirical and contained single values for the constants.

In 1948, an analysis of the stresses in tubesheets in floating head exchangers was published in the USA by K.A. Gardner [2]. To take account of the perforations in the tubesheet, Gardner introduced the concept of the equivalent solid plate. The perforated tubesheet was considered as a solid plate of the same dimensions but with modified properties. The analysis treated the tubesheet as a plate on an elastic foundation, as shown in Figure 2. The coupling action of the tube bundle and its effect on tubesheet stresses was quantified. The paper was a major advance in the understanding of the subject and, most importantly, was presented in the form of a practical design method.

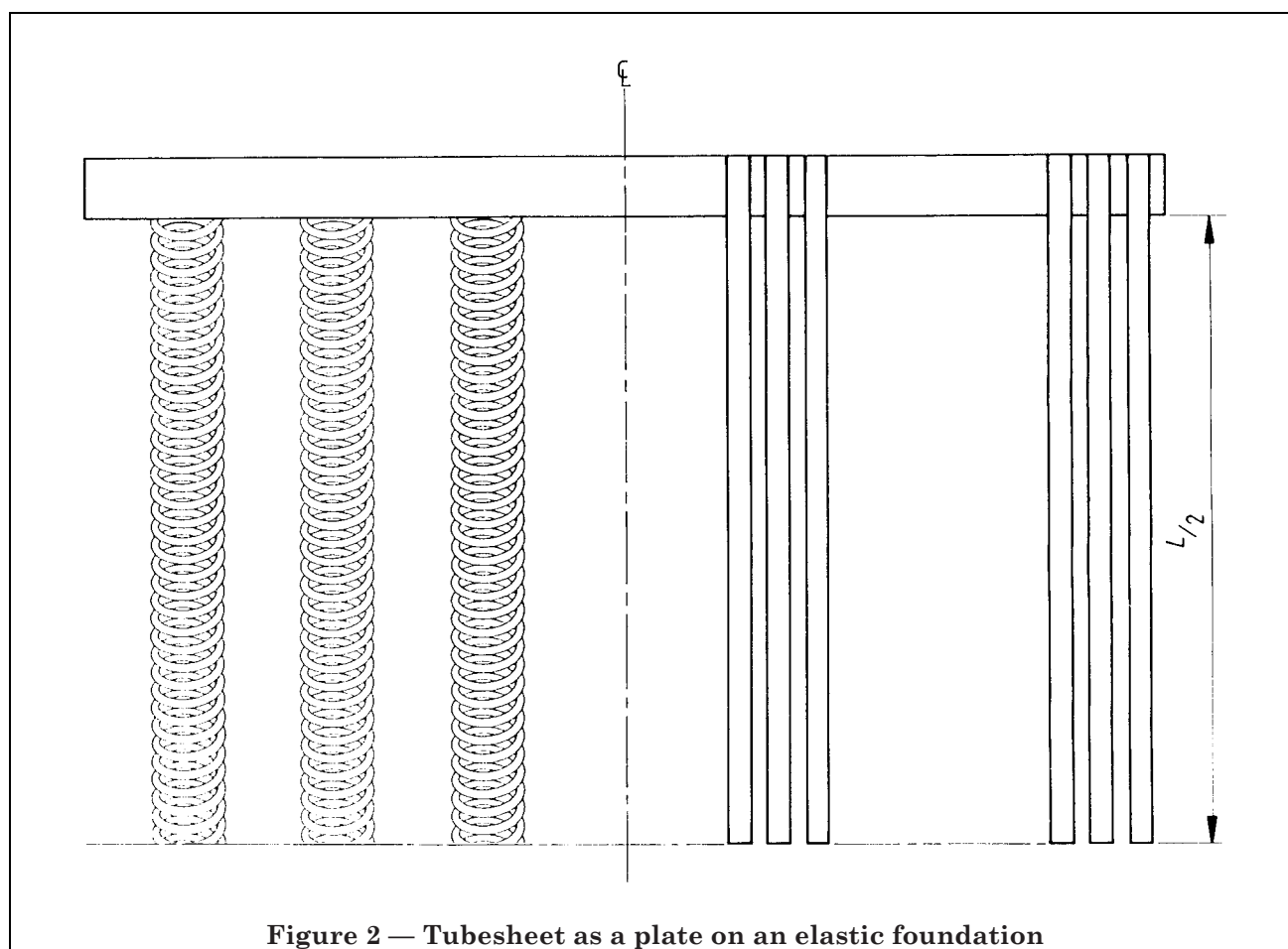


Figure 2 — Tubesheet as a plate on an elastic foundation

A second paper by Gardner, on the design of tubesheets in fixed tubesheet exchangers, appeared in 1952 [3]. This was a development of the 1948 paper and contained the extra condition, essential in a fixed tubesheet exchanger, that there is at the tubesheet edge compatibility of the axial deflection of the shell and the tube bundle. The method was modified by the Tubular Exchanger Manufacturers' Association (TEMA) and entered their standard. The method currently in 3.9.4 of BS 5500 is essentially that published in the sixth edition of TEMA [4]. The bending equation as shown in Table 1 has the same form as that of equation (1). Gardner was aware that there were many factors which affected the stresses in tubesheets. Indeed, in reference to the early TEMA formulae where the constant in the equation for tubesheet thickness was given by single values, he suggested that the constant "must represent a really superb compound of lumped parameters". He continued to develop the stress equations and with the advent of computers was able to study the effect of different parameters on tubesheet stresses using generalised stress equations in a computer model. He identified many factors and in 1969 published his findings for floating head and U-tube exchangers [5]. These too were presented as a design method with the final equation taking the same form as equation (1). The subclause in BS 5500 on the design of tubesheets for floating head and U-tube exchangers, 3.9.3, is almost entirely taken from this paper. Gardner intended to make a similar parametric study of fixed tubesheet exchangers but never completed it.

So, there are in BS 5500 two design methods separated by about twenty years. They originate from the same source and have final design equations of the same form but they also have many differences.

In Table 1, some of the main features of the two methods are compared. For example, with floating head exchangers the "constant" is determined using a complex graph and a calculation, whereas with fixed tube exchangers it is obtained much more simply. With floating head exchangers, the pressure term is simply the hydrostatic pressure but with fixed tubesheet exchangers it is an effective pressure which takes into account many of the complex geometric and loading factors.

In Table 2, the constants from equation 1, for the bending of a plate with simply supported or clamped edge conditions, are compared with those which appear in BS 5500 for the different exchanger types. There is some agreement between the values shown, although with floating head exchangers C_o varies over a wide range.

Table 1 — BS 5500 formulae for tubesheet bending

	BS 5500 3.9.3 Floating head or U-tube	BS 5500 3.9.4 Fixed tubesheet
Bending formula	$e = C D_o \sqrt{\frac{\rho}{\Omega \mu f}}$	$e = \frac{F}{2} D_1 \sqrt{\frac{\rho_1}{f}}$
The "constant"	C: edge support, solidity, width unpierced ring	$\frac{F}{2}$: edge support
Diameter	D_o : outer tube limit	D_1 (or D_2): pressurised diameter
Pressure	p : hydrostatic pressure	p'_1 (or p'_2): design effective pressure
Allowable stress	$\Omega \mu f$: bending stress, ligament weakening	f : no allowance for bending, implicit ligament efficiency

Table 2 — Comparison of the constants

	Theory	U-tube C_o	Floating Head C_o	Fixed Tubesheet $F/2$
Simply supported	0.566	0.560 to 0.660	0.200 to 0.650	0.5
Clamped	0.433	0.433	0.200 to 0.560	0.4

3 Perforated plates

3.1 The variables used in BS 5500

In the BS 5500 design methods, a number of variables are used to take account of the properties of perforated plates.

They are as follows:

- η the flexural efficiency of the tubesheet
- λ the ligament efficiency in shear of the tubesheet
- μ the ligament efficiency in bending of the tubesheet and tube walls
- ν^* the value of Poisson's ratio for the perforated plate

The ligament efficiency in shear (λ) appears in all the shear equations. Values of v^* are given in BS 5500 Figure 3.9(8) and Figure 3.9(9) but only for the purpose of a full stress analysis of fixed tubesheet exchangers; v^* is not used in the code method as currently formulated. The other variables appear explicitly in BS 5500 for floating head exchanger design. For fixed tubesheet exchangers, they do not appear explicitly but they have been included, and clause 5 of this Part of PD 6550 shows how this is done.

3.2 Flexural efficiency

A solid circular plate subject to a uniform moment M along its circumference, as shown in Figure 3, will deflect to a spherical radius of curvature (r) given by:

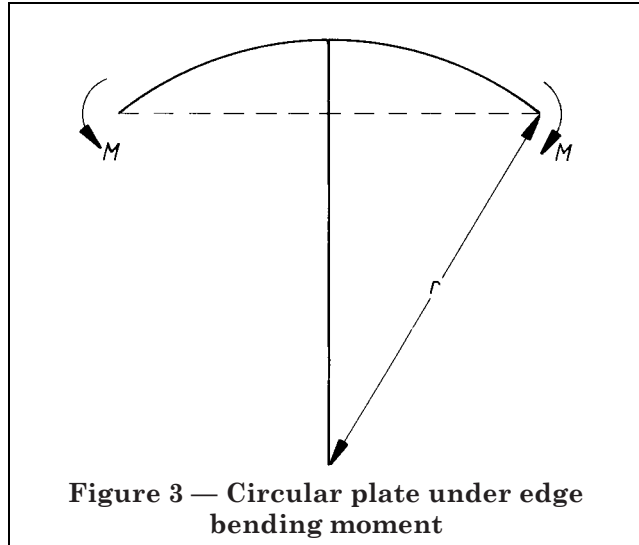
$$\frac{1}{r} = \frac{M}{D(1 + \nu)}$$

This equation is analogous to the familiar one for the bending of a beam, $1/r = M/EI$, except that for plate bending the E/I term is replaced by $D/(1 + \nu)$. D is known as the flexural rigidity of the plate and is given by:

$$D = \frac{Ee^3}{12(1 - \nu^2)}$$

Holes in a plate create local stress concentrations and reduce the structural strength of the plate. When the holes are arranged in arrays, as in a tubesheet, the stress distribution has a regular doubly-periodic pattern. For heat exchanger design, the elastic characteristics of the perforated tubesheet may be expressed in terms of a solid plate of the same dimensions but with modified values for Young's modulus and Poisson's ratio. The latter are known as E^* and ν^* respectively. They can be combined into a single quantity flexural efficiency (η) – as follows:

$$\begin{aligned} \eta &= \frac{D^*}{D} = \frac{\text{Flexural rigidity of perforated plate}}{\text{Flexural rigidity of solid plate of same dimensions}} \\ &= \frac{E^*(1 - \nu^{*2})}{E(1 - \nu^2)} \end{aligned} \quad (2)$$



Flexural efficiency, sometimes known as deflection efficiency, provides the deflection characteristic of the perforated plate. It enables the designer to analyze the interaction of the tubesheet with the other components in the exchanger and calculate the junction forces and moments. There is no advantage to be gained in specifying conservative values of flexural efficiency. To be conservative would create an unreal set of forces and moments between components and possibly lead to an underestimate of the stresses in components other than the tubesheet. Values of flexural efficiency must therefore be realistic.

Measurements in different plates with varying types of perforation were the first source of information on what the values of flexural efficiency were likely to be. Subsequently, an analytical method was developed which predicted the flexural efficiency for most types of perforation over a wide range of plate thicknesses.

Flexural efficiency is a function of ligament efficiency, the minimum distance between tube holes divided by the pitch. It is given in BS 5500 in two graphs, Figure 3.9(8) and Figure 3.9(9), depending on whether the tubesheet thickness is greater than or less than twice the tube pitch. The graphs are compared in Figure 4 where it may be seen that for a given ligament efficiency the flexural efficiency of a thin plate is greater than that of a thick one.

The graphs come from the analytical work of Meijers [6] but the demarcation between thick and thin plates is the result of experimental work by Sampson and presented by O'Donnell and Langer [7]. The experimental work showed that there was a substantial change in the values of E^* and ν^* (the variables in the calculation of η) when plate thickness exceeds twice tube pitch. Sampson's results are shown in Figure 5(a) and Figure 5(b) and the transition at $H/R = 4$ is indicated. The tests were made on plastic plates with a material Poisson's ratio (ν) of 0.5 but the results are relevant for materials such as steel with other values of Poisson's ratio.

Meijers used continuum mechanics to obtain the doubly-periodic stress distribution in a perforated plate. His calculations were for both plane stress and bending. Figure 6(a) shows the geometry as defined by Meijers. The origin is at the centre of one of the holes and the region inside the hole is designated S_o – with contour L_o . The doubly-periodic set of non-intersecting contours congruent to L_o is denoted by L . The hole diameter is taken to be unity. The geometry can be fully described by two basic parameters which are the ratio of the two basic periods $2\omega_1$ and $2\omega_2$, and the ratio of the hole diameter to one of the basic periods. The loading in plane stress comprises mean normal stresses S_x and S_y together with a mean shear stress T_{xy} . The loading under bending comprises mean couples per unit length, M_x^* and M_y^* , and the mean torsion couple M_{xy}^* . Figure 6(b) shows the loading and the cartesian coordinate system xyz^* which is orientated such that the x - y plane is in the middle surface of the plate. An important assumption made in the analysis is that the hole boundaries are unloaded.

Meijers' solutions for plane stress loading and for bending use common equations. The solution for bending is based on the Kirchhoff equations and so is only valid for plates which are relatively thin compared with the pitch of the holes. A set of Meijers' curves is shown in Figure 7(a) and Figure 7(b). They are for a plate in bending with holes on a square pitch. The curve in BS 5500 Figure 3.9(8) is directly derived from these. The curve for triangular pitch in BS 5500 Figure 3.9(8) is also directly derived from other curves given in [6].

For thick plates, curves in [6] for plane stress loading are applicable and Figure 3.9(9) is based on these. The curve for triangular pitch is directly derived from [6]; that for square pitch is derived from [6] but a correction factor has been applied.

It has been found that the presence of unpierced sections of tubesheet beneath partition plates reduces deflections and stresses. It is therefore stated in the assumptions at the beginning of 3.9 of BS 5500 that untubed partition lanes are acceptable. Also stated is the assumption that the tubed area is circular and uniformly perforated.

3.3 Ligament efficiency

The tubesheet ligaments (i.e. the metal between the holes) have to withstand the stresses set up by the loads and moments acting on the tubesheet. Early theoretical work and tests showed that the stresses in the ligaments were well predicted when the stresses in the equivalent solid plate were divided by the ligament efficiency. The calculation of ligament efficiency using the linear dimensions ($\mu = (P - d_h)/P$) was more satisfactory than one based on cross-section areas.

Tubes which have been fully expanded into the tubesheet also strengthen it. The extent of this strengthening has been the subject of testing in the USA. Good correlation was found in the tests between the measured stresses and those calculated using a ligament efficiency based on the inside diameter of the tube rather than the outside diameter. BS 5500 specifies that for tubes which are expanded for the full depth of the tubesheet or are explosion bonded, the ligament efficiency may be calculated using the tube mean diameter rather than the hole diameter.

Ligament efficiency in shear (λ) may differ from that in bending (μ), but in BS 5500 the two are taken to be equal.