Study of the tubesheet design of a multitubular reactor

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STUDY OF THE TUBESHEET DESIGN OF A MULTIPUDBULAR REACTOR.

Report of a training 1 August 1986 - 15 October 1986 at the SIPM, Den Haag

Coco Jongerius, Id.nr. 192260
WFW-rapport nr. WFW.87.034
STUDY OF THE TUBESHEET DESIGN OF A MULTITUBULAR REACTOR

Report of a training 1 august 1986 - 15 oktober 1986
at the SIPM, Den Haag

Student: Coco Jongerius
PREFACE

This study is performed within the framework of a training for me, being a post-candidate student at the mechanics department of the Technical University in Eindhoven. In this department, I belong to the group "Dynamics", which is under supervision of prof. dr. ir. D.H. van Campen.

The training is performed at the Central Office of the SIFM (Shell Internationale Petroleum Maatschappij) in Den Haag, at the department MFE/23. This is the mechanical engineering department.

I want to thank first mr. A.W.M. Zwetsloot and mr. B de Jong for their support during the performing of this training. Also I am grateful to mr. F.J. Deen and mr. C. Verheul, for making the training at their department possible.

Finally I want to thank everybody at MFE/23, for their help making my training to a succes. I have enjoyed working at this department very much.

Coco Jongerius

23 oktober 1986
CHAPTER 0: SUMMARY & CONCLUSIONS

This study is performed within the framework of a training for a student of the Technical University in Eindhoven.

It is based upon an existing design of a multitubular reactor, which is found to be too expensive. Its price can be decreased by reducing the dimensions of its components.

Since the shell, tubes, and spherical heads are simple parts, their dimensions can easily be optimally calculated. It is not expected that a new evaluation of these parts will result in a cheaper reactor.

The tubesheet however is a more complicated component: the interpretation of the stresses due to the various loads can be difficult. If this interpretation has not been carried out optimally, the tubesheets can become thinner.

Thereby also alternative constructions for the tubesheets are possible, which may reduce the required tubesheet material volume.

In this study possibilities to reduce the price of the reactor by reducing the required tubesheet volume are evaluated.

First the existing design is evaluated. It appeared, that within the stress limits based on the Codes (like ASME 8 [11], British Standard 5500 [2]), the achieved tubesheet thicknesses were optimal.

Secondly two alternative constructions are examined: a thick and a thin tubesheet, and dished tubesheets.

In the calculations for the first alternative unfortunately some mistakes have been made. Therefore it was impossible to draw any reliable conclusions for this alternative.

The second alternative, dished tubesheets can indeed give a low required tubesheet volume. However, this gain in kilo's is decreased because of a required reinforcement ring.

The use of dished sheets has also large consequences on other fields: the price per kilo will increase because of the more difficult manufacturing of dished plates with very many holes, and the price of the rest of the construction may increase, e.g. because of changes of the catalyst handling, and/or a higher required shell.

Therefore it is not expected that it is possible to achieve a price reduction that makes a further examination of this alternative interesting.

The calculations that are performed, are based on the stress limits given by the Codes. These stress limits for shake-down of secondary stresses to elastic behavior are based on elastic, ideally plastic material behavior. That means that no work hardening is included. It appeared that if work hardening is included, the allowable stresses for secondary stresses can be higher. Since the tubesheet thickness is determined by total (= primary plus secondary) stresses, then a less thick tubesheet can be used, which results in a cheaper reactor.
# Contents

<table>
<thead>
<tr>
<th>Chapter</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Preface</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Summary and conclusions</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Introduction</td>
<td></td>
</tr>
<tr>
<td>2.1</td>
<td>General Description of the reactor vessel</td>
<td>2</td>
</tr>
<tr>
<td>2.2</td>
<td>General Stress classification</td>
<td>4</td>
</tr>
<tr>
<td>2.3</td>
<td>General Description of the loads</td>
<td>4</td>
</tr>
<tr>
<td>2.4</td>
<td>General Determination of worst cases</td>
<td>7</td>
</tr>
<tr>
<td>3</td>
<td>Evaluation of the present design</td>
<td>10</td>
</tr>
<tr>
<td>3.1</td>
<td>Evaluation Introduction</td>
<td>10</td>
</tr>
<tr>
<td>3.2</td>
<td>Evaluation Division of the stresses according to §2.2</td>
<td>11</td>
</tr>
<tr>
<td>3.3</td>
<td>Evaluation Checking the calculated thicknesses</td>
<td>14</td>
</tr>
<tr>
<td>3.4</td>
<td>Evaluation Conclusion</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Different sized tubesheets</td>
<td>15</td>
</tr>
<tr>
<td>4.1</td>
<td>Different sized Introduction</td>
<td>15</td>
</tr>
<tr>
<td>5</td>
<td>Dished tubesheets</td>
<td>25</td>
</tr>
<tr>
<td>5.1</td>
<td>Dished Introduction</td>
<td>25</td>
</tr>
<tr>
<td>5.2</td>
<td>Dished Assumptions</td>
<td>26</td>
</tr>
<tr>
<td>5.3</td>
<td>Dished Required thickness of the tubesheet</td>
<td>28</td>
</tr>
<tr>
<td>5.4</td>
<td>Dished Required reinforcement ring</td>
<td>30</td>
</tr>
<tr>
<td>5.5</td>
<td>Dished Volumes of tubesheet and reinforcement ring</td>
<td>31</td>
</tr>
<tr>
<td>5.6</td>
<td>Dished Results</td>
<td>32</td>
</tr>
<tr>
<td>5.7</td>
<td>Dished Evaluation of the impact of the use of dished tubesheets</td>
<td>33</td>
</tr>
<tr>
<td>5.8</td>
<td>Dished Conclusion</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Evaluation of the basis of the stress limits</td>
<td>34</td>
</tr>
<tr>
<td>6.1</td>
<td>Evaluation Introduction</td>
<td>34</td>
</tr>
<tr>
<td>6.2</td>
<td>Evaluation Elastic, ideally plastic material behavior</td>
<td>36</td>
</tr>
<tr>
<td>6.3</td>
<td>Evaluation Including work hardening in the material behavior model</td>
<td>37</td>
</tr>
<tr>
<td>6.4</td>
<td>Evaluation Conclusion</td>
<td></td>
</tr>
</tbody>
</table>

# Literature list
APENDICES

1/ Temperature differences between tubes
2/ Calculations according to BS 5500 1985
3/ Calculations according to ASME Nuclear Code A 8000
4/ Evaluation of the groove in the tubesheet
5/ Temperature difference between tube bundle and shell
6/ Determination of equivalent pressure for dead weight
7/ Numerical values
The basis of this study is an existing design of a multitubular reactor, which is found to be too expensive. To achieve a reduction of the price, the dimensions of the components, that are not prescribed by the process that has to take place in the reactor, can be reduced when allowable.

The reactor can be seen as a large, fixed tubesheet heat exchanger. The global components are: shell, spherical heads, tubes, and tubesheets (see figure 2.1). Since shell, heads and tubes are simple construction parts, it can reasonably be assumed that they are optimally calculated in the present design. The tubesheet's dimensions however are more complicated to determine.

The evaluation of the stresses in the sheet, that means which criteria must be satisfied by which stress because of which load, is difficult to perform. If this has not been done optimally, the tubesheets may become thinner, and thus cheaper.

Also interesting is, that alternative constructions for the tubesheets are possible.

Therefore in this study the possibilities are evaluated to decrease the price of the reactor by decreasing the total needed tubesheet volume.

Chapter II gives the global dimensions of the reactor, the applied stress limits, and a description of the loads.

In chapter III, the evaluation of the stresses that lead to the present design are followed. If this evaluation has not optimally been performed, a first reduction of the tubesheet volume possibly can be achieved.

Next to the present design, that consists in two nearly equally thick tubesheets, other constructions are possible. Chapter IV handles the alternative that a thick and a thin tubesheet are used. The idea is that the power of a sheet to bear a moment increases quadratically with the thickness. If more load can be born by one tubesheet, this might give a tubesheet volume reduction.

In chapter V an evaluation of the use of dished tubesheets is given. This possibility is based on reduction of the bending stresses, so that the membrane stresses become determining. This may give thinner tubesheets.

These were only two constructional alternatives; many more are possible. E.g. a bellow in the attachment of tubesheet to shell, or a three dimensional tubesheet construction. It is very well possible that a tubesheet volume reduction can be achieved with these alternatives. However, within the time available for this study, these could not be evaluated.

The stress limits used in the previous chapters are based on an elastic, ideally plastic material behavior model. In chapter VI, the implementation of work hardening in this model is evaluated.

Beside the evaluation of the possibilities of tubesheet volume reduction, also is evaluated what the maximal allowable temperature difference is between tubebundle and shell, based on occurring temperature differences between tubes. This is done in Appendix 3.
CHAPTER II: GENERAL

At the back of the report, a fold out paper with the used variable names is attached.

§2.1 DESCRIPTION OF THE REACTOR VESSEL

The reactor can be seen as a large heat exchanger (See figure 2.1). It consists in a cylindrical shell, spherical heads, tube bundle and two fixed tubesheets.

The tubes are filled with catalyst. A gasflow goes downward through the tubes from the upper to the lower head. In the tubes, the catalyst and the gas react. The heat that results from this reaction is absorbed by the cooling fluid (water or oil) in the shell side of the reactor.

![Diagram of the reactor vessel](image)

figure 2.1
The total tubesheet construction can be seen as consisting in two parts: the tubesheet and the hammerhead (see figure 2.2).

![Figure 2.2](image)

The tubesheet itself is a circular plate, perforated for the tubes within the Outer Tube Limit Circle. The diameter will be called $D_e$. The hammerhead is the connection of tubesheet and shell. The groove is implemented to reduce the stresses in the tubesheet.

It is suggested that this construction is expensive, and another possibility, which would be cheaper, is proposed. In Appendix 4 is shown that the original construction is satisfactory.

**Data**

- **Materials**
  The materials that are used in the present design are:

  - Shell: SA 533 Gr.B
  - Heads: SA 533 Gr.B
  - Tubesheet: SA 162 F1
  - Tubes: not specified

  These materials are chosen by the designer. Evaluation of these choices does not fall within the scope of this study. It must be noted that the use of other materials will not influence the conclusions of this study, since it only influences the stress limits that are valid for all considered alternatives.

- **Parameter values for present design**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$d_o$</td>
<td>30 mm</td>
<td>Outside diameter of tube</td>
</tr>
<tr>
<td>$d_i$</td>
<td>26 mm</td>
<td>Inside diameter of shell</td>
</tr>
<tr>
<td>$d_h$</td>
<td>30 mm</td>
<td>Tube hole diameter</td>
</tr>
<tr>
<td>$D_e$</td>
<td>6866 mm</td>
<td>Outside diameter of shell</td>
</tr>
<tr>
<td>$D_o$</td>
<td>6456 mm</td>
<td>Diameter of outer tube limit circle</td>
</tr>
<tr>
<td>$D_1$</td>
<td>6730 mm</td>
<td>Diameter to which shell fluid pressure is exerted</td>
</tr>
<tr>
<td>$D_2$</td>
<td>6730 mm</td>
<td>Diameter to which tube fluid pressure is exerted</td>
</tr>
</tbody>
</table>
### 2.2 STRESS CLASSIFICATION

The stresses that are calculated must be evaluated against a criterion. To be able to do this, a classification of the stresses is necessary. Then the stresses in one class must satisfy the limit that belongs to that class.

A common way to do this is given in the British Standard BS5500. (See figure 2.3.) The f in the figure is called the design stress, which is 2/3 of the yield stress.

In this report, except for chapter VI, this classification, with its stress limits, will be used.

### 2.3 DESCRIPTION OF THE LOADS

The loads that are exerted upon the tubesheets result from several causes. The stresses these loads induce, will be classified according to § 2.2, figure 2.3.

- **Channel side pressure**

  The channel side pressure is not equal for the whole channel side. The gasflow through the tubes finds resistance from the catalyst. Therefore a pressure difference between the upper and the lower head exists.

  We divide the pressure channel side in two parts:

  . Constant pressure channel side
    - This is the pressure in the lower head
  . Pressure drop
    - This is the pressure difference between upper and lower head, which results from the resistance of the catalyst.
### Stress Categories and Limits of Stress Intensity

<table>
<thead>
<tr>
<th>Category</th>
<th>Primary</th>
<th>Local</th>
<th>Secondary</th>
<th>Peak</th>
</tr>
</thead>
<tbody>
<tr>
<td>Symbol</td>
<td>$f_0$</td>
<td>$f_1$</td>
<td>$f_2$</td>
<td>$f_3$</td>
</tr>
</tbody>
</table>

**Combination of Stress Components and Allowable Limits of Stress Intensity**

NOTE 1: This limitation applies to the range of stress intensity. When the secondary stress is due to a temperature excursion at the point at which the stresses are being analysed, the value of $f_3$ is to be taken as the average of the $f$ values for the highest and the lowest temperatures of the metal during the transient. When part or all of the secondary stress is due to mechanical loads, the value of $f_3$ is to be taken as the $f$ value for the highest temperature of the metal during the transient.  

NOTE 2: The stresses in category $f_2$ are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc., and do not include primary stress which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of $f_0$, $f_1$, $f_2$ and not $f_3$ alone. Similarly, if the stress in category $f_2$ is produced by a stress concentration, the quantity $f_3$ is the additional stress produced by the notch, over and above the nominal stress. For instance, if a plate has a nominal stress intensity $f_1$, and has a notch with a stress concentration factor $K$, then $f_1 + f_2 + f_3 = f_0$, $f_1 + f_2 + f_3 = f_0$ and the peak stress intensity equals $f_0 + f_3 K (K - 1) = f_0 K$.  

NOTE 3: $f_0$ is obtained from the fatigue curves: the allowable stress intensity for the full range of fluctuation is $f_0$.  

**Figure A.3 Stress Categories and Limits of Stress Intensity**
The effect of the constant channel side pressure \( P_C \) is schematically given in figure 2.4(a). Maximum value is the tubeside design pressure.

Because of the large amount of tubes, most of the load will be carried by the tubes. The tubesheet will only deflect near the edge of the tubesheet, because of length difference between tubebundle and shell. This gives discontinuity stresses, which can be classified as secondary stresses (see figure 2.3).

The pressure drop \( \Delta P \) gives a load on the tubesheets like is drawn in figure 2.4(b). Now the tubes do not have a staying effect anymore; they only distribute the load over the two tubesheets.

Since this load has to be carried by the tubesheets alone, the resulting stresses are (mostly) load controlled. A part of the stresses, however, is self-relieving (e.g. discontinuity stresses). Because in a linear elastic Finite Element analysis it is not possible to determine which part of the stresses is self-relieving, as a conservative assumption all stresses are classified primary, (see figure 2.3).

The tubesheet will deflect under the load. Therefore it is expected that the largest part of the stresses will be primary bending stresses.

- **Shell side pressure**

  This loading cause gives the same load on the tubesheets as the constant channel side pressure, except for the value and the direction. See figure 2.3(c). Maximum value is the shell side design pressure.

  Therefore the same discussion is valid, and thus the occurring stresses due to this loading cause can be classified as secondary stresses.

- **Dead weight of tubes, tubesheets, catalyst and cooling fluid**

  Because all components, of which the dead weight has to be born by the tubesheets, are equally distributed over the reactor, the resulting load can be seen as a distributed load over the total tubesheet surface. For determination of the equivalent pressure, see Appendix 6.

  The tubes do not have a staying action, but distribute the load over the two tubesheets. Therefore the loading can schematically be represented as figure 2.4(b). The resulting stresses can, like those due to pressure drop, be classified as primary bending stresses.

- **Temperature and expansion differences**

  The reaction of the gases in the tubes with the catalyst produces heat. This heat is absorbed by the cooling fluid. Therefore the shell has a lower temperature than the tubebundle. (See data §2.1).
Secondly, the tube material has a thermal expansion coefficient, different from that of the shell material. Because of this, expansion differences between tubebundle and shell occur. This gives discontinuity stresses in the tubesheet: secondary stresses. The load can be seen as a non uniformly distributed load, exerted by the tubes on the tubesheet (see figure 2.4 (d))

- figure 2.4

Considering these five loadings, it appears that they can be divided in two classes: symmetrical and antisymmetrical.

• Symmetrical
  The loads in this class are: - channel side pressure
  - shell side pressure
  - temp./exp. differences
  See figure 2.4(a),(c),(d).
  These loads give only secondary stresses.

• Antisymmetrical
  These loads are: - pressure drop in tubes
  - dead weight
  See figure 2.4(b).
  These loads give primary and secondary stresses.

§2.4 DETERMINATION OF WORST CASES

The required thicknesses of the tubesheets must be calculated so that the stress limits will not be exceeded for any load that can occur within the design values. The worst situation is the situation that gives the largest stresses. If the stress limits are satisfied for this situation, they are satisfied for all occurring loads within the design values. Thus if the required thickness is to be determined, first the worst loading situation must be found. This will be done in this paragraph.

The largest stresses occur, when the load is maximal in one direction. Since the combinations of loading causes (see § 2.3) that give this maximum load differ for the two tubesheets, the worst case has to be determined for each tubesheet separately.
In paragraph 2.3 the stresses resulting from the different loading cases are divided in primary and secondary stresses. According to figure 2.3, different stress limits exist: limits for the primary stresses only, limits for the primary plus secondary stresses, and limits for primary, secondary plus peel stresses. Therefore worst cases must be found for only primary stresses, and for primary plus secondary stresses. The last limits are not included in this calculation, since no high cycle fatigue calculations are done.

- Upper tubesheet

* primary stresses

There are two loading causes, that give primary stresses: dead weight and pressure drop. Since the resulting load is pointed downward for both causes, the worst loading case is:

\[
\begin{align*}
\text{maximum pressure drop} \\
\text{dead weight}
\end{align*}
\]

* primary plus secondary stresses

The downward pointed loads are caused by: constant pressure channel side, dead weight and pressure drop. Shell side pressure and temperature difference give an upward load.

Now two possible worst loading cases are possible: maximum load upward, and maximum load downward:

**Upward:**
- minimum pressure channel side
- maximum pressure shell side
- no pressure drop
- dead weight
- maximum temperature difference

**Downward:**
- maximum pressure channel side
- minimum pressure shell side
- maximum pressure drop in tubes
- dead weight
- minimum temperature difference

Which load is worst can be found out when values for the pressures and temperature differences are substituted.

A temperature difference of 0° or 50°C gives for this reactor in magnitude nearly the same load, since the thermal expansion coefficients of shell and tube material differ (see appendix 5).

The downward load contains a constant channel side pressure of 4.4 N/mm². The upward load only gives 3.7 N/mm² pressure shell side. These pressures give the same sort of stresses in the tubesheets, but for the sign and the value.
Thus, since the constant pressure channel side is larger, the induced stresses from the downward load are higher than from the upward load. Because the downward load contains a load from the pressure drop over the tubes extra, the downward load is the worst load.

- **Lower tubesheet**
  
  - **primary stresses**
    
    Like for the upper tubesheet, the worst loading case is:
    1. maximum pressure drop
    2. dead weight

  - **primary plus secondary stresses**
    
    The downward pointed loads are caused by: pressure shell side, dead weight and pressure drop. Constant channel side pressure and temperature difference give an upward load.

    Now again two possible worst loading cases are possible: maximum load upward, and maximum load downward:

  Upward:  
  - maximum pressure channel side
  - minimum pressure shell side
  - no pressure drop
  - dead weight
  - minimum temperature difference

  Downward:  
  - minimum pressure channel side
  - maximum pressure shell side
  - maximum pressure drop in tubes
  - dead weight
  - maximum temperature difference

  Since a temperature difference of 0°C gives the same load as a difference of 50°C, this loading cause does not influence which loading combination gives the largest stresses.

  The upward load contains 4.4 N/mm² constant channel side pressure. The downward load contains 3.7 N/mm² shell side pressure, and 0.6 N/mm² pressure drop. Since the pressure drop must be carried by the tubesheets, and the pressures shell and channel side exert mostly on the tubes, the downward load gives the largest stresses in the lower tubesheet, and is thus the worst loading situation.
CHAPTER III: EVALUATION OF THE PRESENT DESIGN

§3.1 INTRODUCTION

The present design contains two nearly equally thick tubesheets. Normally, tubesheet thicknesses are determined using the Code rules. For large heat exchangers, a full stress analysis is recommended (British Standard 5500 1985, page 3/83), since the code rules might lead to excessive thick tubesheets.

In Appendix 2 a calculation according to British Standard 5500 is performed, Appendix 3 gives a calculation according to ASME Nuclear Code. Comparing these results with the FEM-analysis, performed by the designer of the present design, shows that the codes for this reactor indeed lead to excessive thick tubesheets.

In this chapter, the FEM-analysis is evaluated.

The perforated tubesheet is modeled as a plate with the same dimensions, but with equivalent elastic constants, according to ASME VIII Div.2 [3].

There are two possibilities that not the optimal thickness is reached:
Firstly, if the stresses are not optimally interpreted, too thick tubesheets might result. Secondly, it is possible that, when the determining stress limit is found, not the minimum possible tubesheet thickness is calculated. These two possibilities are considered in the next two paragraphs.

§3.2 DIVISION OF THE STRESSES ACCORDING TO § 2.2

Using elastic FEM-analysis it is possible to calculate the distribution of the stresses in a chosen cross section. From this distribution the stresses can be divided in membrane, bending, and peak stresses (Stoomwezen [5]). To be able to use the stress limits given by §2.2, figure 2.3, also a division must be made in primary and secondary stresses.

(Division in global and local stresses is not applicable here, since the occurring primary membrane stresses are expected to be very small, and thus not expected to have a large influence on the total primary stress).

It is not possible to determine whether a stress is self-relieving or not with elastic FEM-analysis, since then infinite elastic behavior is assumed. Therefore the loads have to be divided in loads that do, and loads that do not give primary stresses.

This division of the loads has already been done in §2.3:

- antisymmetrical loads give primary stresses:
  - pressure drop
  - dead weight

- symmetrical loads give no primary stresses:
  - constant pressure channel side
  - pressure shell side
  - temperature/expansion differences
- a calculation with loads that give primary stresses, to find an upper value for the maximal occurring primary stresses
- a calculation with all the loads, to find the total occurring stresses.

If the primary stresses appear to be limiting, a better interpretation of the stresses can lead to smaller primary stresses, thus to thinner tubesheets, thus to a cheaper reactor. If the total stresses are limiting, a better division of the total stresses in primary and secondary stresses has no use.

According to the designer, the total stresses appeared to be limiting, while the primary stresses remained far below their limits. Since only data are given for the total stresses, this statement can not be controlled, and has to be assumed.

Conclusion: No gain can be found in a better evaluation of the character of the stresses.

Remark
At this moment it could have been realized, that the incentive of reduction of the primary stresses, which lead to the alternative constructions of chapter 4 and 5, is in fact not important: it are the total stresses which are important, not the relatively small primary stresses. However, this has been realized after the training had been finished.

§3.3 CHECKING OF THE CALCULATED THICKNESS

It is possible that the thicknesses resulting from the total stresses are not optimally calculated. Therefore a control calculation will be done.

The only data available for this evaluation (see figure 3.1) were for a similar reactor vessel, with thicker tubesheets and a smaller diameter. These data give the stresses in the tubesheet in the (according to the designer) heaviest loaded zone: the Outer Tube Limit Circle.

That this is indeed the heaviest loaded zone is probable: Firstly, the moment per length in the tubesheet (in negative direction) increases with the radius r (Roark [4] page 362 case 10b):

\[ M = \frac{q \cdot ((1 + \nu) \cdot (4D)^2 - (3 + \nu) \cdot r^2)}{16} \]  

That means that the larger the radius r, the larger the moment and thus the larger the stresses.

Secondly, outside the OTLC, the tubesheet is not perforated anymore: the carrying surface increases, and the occurring stresses decrease. Therefore the OTLC can very well be the heaviest loaded zone.

Figure 3.1 gives the total radial, tangential, axial and shear stress in the small reactor through the thickness of the tubesheet, at the OTLC.
These stresses result from the following loads:
- pressure channel side: 41 bar
- pressure shell side: 35 bar
- pressure drop tubes: 6 bar
- dead weight
- temperature difference: 25°C (tube temperature = 270°C, shell temperature = 245°C)

Comparing this loading combination with the worst possible one, see § 2.4, it must be noted that this is not the worst situation.

From these stresses the stresses in the tubesheets of the reactor that is the subject of this study, will be estimated.

Estimation of the stresses in the "real" tubesheets

The maximal stresses that occur at the surface of the tubesheets of the small reactor are:
- radial stress = 72 N/mm²
- tangential stress = 33 N/mm²

This gives a total stress of \( \sqrt{72^2 + 33^2} = 80 \) N/mm²
To find an estimate for the real occurring stress in the tubesheets of the reactor considered in this study, three corrections must be done:

1/ Accounting for the perforation  
2/ Accounting for the larger diameter  
3/ Accounting for the thinner tubesheets.

Accounting for the perforation

This can be done by using the ligament efficiency \( \mu \). This is the ligament length between two closest holes, divided by the pitch of the holes (see figure 3.2)

\[ \mu = \frac{P - d_h}{P} \]

figure 3.2

The ligament efficiency is a measure for the decrease in surface that has to carry the load. Therefore the real occurring stresses are a factor \( 1/\mu \) higher than the calculated stresses.

With \( P = 37 \text{ mm} \), and \( d_h = 30 \text{ mm} \), the ligament efficiency is 0.188

Accounting for the larger diameter

A larger diameter results in a quadratic increase of the moment per length in the tubesheet:

\[ M = \frac{q \cdot ((1 + \psi)^2 - (3 + \psi)^2)}{16} \]

The stress is linearly related to the moment (Roark [4] page 332):

\[ \sigma = \frac{6 \cdot M}{e^2} \]

Therefore the stress will increase quadratically with the diameter.

With: \( D_o \) of small reactor = 6337 mm  
\( D_o \) of considered reactor = 6456 mm,

the real stresses become a factor \((6456/6337)^2 = 1.038\) larger because of the diameter difference.
• Accounting for thinner tubesheets

A thinner tubesheet gives a quadratic increase of the stress; see equation (3.2).

With: tubesheet thickness of small reactor = 280 mm
tubesheet thicknesses of considered reactor = 235 and 245 mm

The real stresses become a factor \((280/235)^2 = 1.42\) (lower tubesheet) and \((280/245)^2 = 1.31\) (upper tubesheet) larger because of difference in tubesheet thickness.

Then the total stresses in the tubesheets become:

- Lower tubesheet: \(\sigma = 80 / 0.189 \times 1.039 \times 1.42 = 625\) N/mm²
- Upper tubesheet: \(\sigma = 80 / 0.189 \times 1.025 \times 1.31 = 576\) N/mm²

The total stress must be limited to 3 times the design stress (see figure 2.3).

Therefore the limit for the calculated total stress is \(3 \times 200 = 600\) N/mm².

Because the estimated stresses reach their limits, while not even the worst loading case is applied, the thicknesses of the considered reactor can be taken to be optimally calculated.

Conclusion: The tubesheet thicknesses of the considered reactor are optimally calculated considering only total (primary plus secondary) stresses.

§3.4 CONCLUSION

For the design with two nearly equal tubesheets, the tubesheet thicknesses are optimally calculated within the stress limits as given in § 2.2, figure 2.3.
CHAPTER IV: DIFFERENT SIZED TUBESHEETS

§4.1 INTRODUCTION

In this chapter, the idea is treated to reduce the required tube-sheet material volume by using two tubesheets of different thickness. The philosophy behind this idea is given in § 4.2. The performed calculations are not shown in this report, since it unfortunately appeared they were not carried out properly. Therefore it was not possible to draw reliable conclusions.

§4.2 OPTIMAL CONFIGURATIONS

First the optimal configurations for symmetrical and anti-symmetrical loads are determined. Then some remarks are made on the optimal configuration for a combination of both load-types.

Symmetrical loads

These loads are mostly born by the tubes (see figure 2.4(a), (c), (d), and figure 4.1)

![Figure 4.1](image)

The load is performed by temperature/expansion difference between tube bundle and shell, and pressures at channel- and shell side.

Suppose the present configuration consists in two equally thick tubesheets, which are as thin as allowable. If we make one of the sheets thicker, the load on the other sheet will not change. That means that the thickening of one tubesheet will not lead to becoming thinner of the other, and thus not lead to reduction of the total required tubesheet material volume.

Conclusion: The optimal configuration for symmetric loads consists in two equally thick tubesheets.

Antisymmetrical loads

The tubes now do not have a staying effect, but they keep a nearly constant distance between the tubesheets: they distribute the load over the two sheets (see figures 2.4(b) and 4.2).
Suppose again that the present configuration is one with two equally thick tubesheets with minimum thickness. Under a load the total construction will deflect. (see figure 4.3)

If one of the tubesheets is made thicker, the whole construction will deflect less. This means that the other sheet has to carry a smaller part of the load $q$, and therefore can become thinner.

In the limit situation, the total load is carried by one tubesheet. Since the power to bear a load perpendicular to the tubesheet increases quadratically with the thickness:

$$\sigma = \frac{6 \cdot M}{e \cdot t^2} \quad (4.1, \text{ see } (3.2))$$

a tubesheet has to grow a factor $\sqrt{\frac{e}{t}}$ thicker to bear a factor $e$ more load.

This means for the limit situation, in which one tubesheet carries all the load, one sheet has to grow a factor $\sqrt{\frac{e}{t}}$ thicker, while the other can disappear.

Conclusion: the optimal situation for antisymmetric loads is one tubesheet that carries the total load.

Combination of the two load-types

Now it is less easy to determine the optimal configuration. It depends on which load-type is dominant: the symmetrical or the antisymmetrical. If the first is determining, two equally thick tubesheets will give the least required tubesheet volume. If the other type is dominant, one thick tubesheet will result as optimal configuration. It is however very well possible that both types are important. In that case a compromise may result: a thin and a thick tubesheet.

Calculations have to be made to find out which of the above mentioned alternatives is the optimal one for the considered reactor. Due to some errors, the performed calculations can give reliable conclusions.
CHAPTER V: DISHED TUBESHEETS

§5.1 INTRODUCTION

The highest occurring stresses in flat tubesheets are bending stresses. If we use dished tubesheets, the bending stresses due to antisymmetrical loading will decrease, and in an optimal tubesheet form be neglectable. Then, the tubesheet thickness is limited by the occurring membrane stress.

![Diagram of a dished tubesheet](image)

To be able to calculate an estimate for the required tubesheet thickness, some assumptions will be made. With these, the required thickness is determined.

Because the resulting tubesheets are very thin, they only can bear load in the plane of the plate. This implies, that a reinforcement ring is necessary to avoid buckling of the shell. The volumes of plate and ring are calculated, to determine whether a reduction of the required tubesheet volume can be achieved with using dished tubesheets.

Finally the implications of this alternative on the existing design are evaluated.

§5.2 ASSUMPTIONS

In the calculation, some assumptions are made:

1. The form of the dished sheet is assumed spherical.
   The form where the least bending stresses occur, will not be spherical, since that is the optimal form for load perpendicular on the surface. Here the load is vertical.
   Determination of the exact form that has the least bending stresses has not been performed yet. If the alternative of dished tubesheets is useful, it can be done later.
This assumption has no influence on the calculation of the tubesheet thickness, only on the determination of the volume. It is however not expected that this volume will differ very much from the exact bending less tubesheet's volume.

2/ The weakening of the sheet due to the holes is said to be accounted for with the ligament efficiency $\mu$. Is this assumption conservative?

The cross section of the plate changes with the angle $\alpha$ (see figure 5.3) with the horizontal of the tubesheet: see figure 5.2.

\[ \alpha = 0^\circ \quad 0^\circ < \alpha < 90^\circ \quad \alpha = 90^\circ \]

figure 5.2

Thus: the larger the angle $\alpha$, the larger the carrying surface. Therefore, the weakening effect accounted for by the ligament efficiency, is too large for an angle $\alpha$ unequal to 0. Therefore the assumption is conservative.

3/ The minimum required tubesheet thicknesses are determined based on primary stresses. It is very well possible that not the primary, but the secondary stresses are limiting. This idea is supported by the knowledge, that, when the equally thick, flat tubesheets were calculated, also the secondary stresses were determining.

§5.3 REQUIRED THICKNESS OF THE DISHED PLATE

The loading of the plate is shown in figure 5.3.
Because the tubesheet does (can) not bear bending-moments, the uniform distributed boundary force \( f_{\text{ct}} \) works under an angle of \( \alpha' \).

The calculation is based upon primary stresses. Therefore the load is:

- Pressure drop over tubes due to catalyst (\( \Delta p \))
- Dead weight, accounted for by equivalent pressure (\( p_g \))

The determination of the required tubesheet thickness is done with the following steps: first, using the equilibrium of forces in vertical direction, an expression is derived for the vertical distributed boundary force \( f_v \).

Then with expressions for total boundary force \( f_{\text{ct}} \) as function of \( f_v \), and for the stress \( \sigma \) as a function of the tubesheet thickness \( \sigma \) and the boundary force \( f_{\text{ct}} \), an expression for the stress in the tubesheet is determined.

Implementing the weakening effect of the perforation of the tubesheet, and the limit for the primary membrane stresses according to §2.2, figure 2.3, an expression is derived for the minimum required tubesheet thickness.

- Equilibrium of forces in vertical direction gives:

resulting force from bound. forces = resulting force from distr. loads

\[
\langle = \rangle \quad f \cdot p_l \cdot D = \frac{1}{2} \Delta p \cdot (\frac{1}{2} D)^2 + p_g \cdot (\frac{1}{2} D)^2
\]

\[
\langle = \rangle \quad f = \Delta p \cdot D/8 + p_g \cdot D/4
\]

With for boundary force \( f \) and for membrane stress \( \sigma \) :

\[
f_{\text{ct}} = f_v \sin \alpha \quad \text{and} \quad \sigma = f/e,
\]

the stress in the tubesheet is:

\[
\sigma = \frac{(\Delta p \cdot D/8 + p_g \cdot D/4)}{e \cdot \sin \alpha}
\]

However, no account is taken for the weakening effect of the holes. This can be done by using the ligament efficiency \( \mu \). (See assumption 3 and §3.3)

Implementing the weakening effect, the occurring stress increases with a factor \( 1/\mu \). Thus:

\[
\sigma = \frac{(\Delta p \cdot D/8 + p_g \cdot D/4)}{\mu \cdot e \cdot \sin \alpha}
\]
With allowable stress $\varepsilon$ according to the British Standard (see §2.2) the minimal required tubesheet thickness is:

$$e_{\min} = \frac{(\alpha \rho \cdot D/8 + \pi \rho \cdot D/4)}{\mu \cdot \varepsilon \cdot \sin \alpha} \quad (5.1)$$

The resulting thicknesses are very low (see §5.5, table 5.1, row 2). Therefore no bending stresses can be carried by the tubesheets; a reinforcement ring is necessary to avoid buckling of the shell.

### §5.4 REQUIRED REINFORCEMENT RING

In the previous part, the vertical equilibrium of forces is worked out. The uniform distributed membrane force also has a horizontal component. In order to avoid buckling of the shell due to this force, a reinforcement ring is necessary.

Assumption: The cross section of the ring is T-formed. This is a rather common choice, since it combined high stiffness with low material volume. To be able to make some calculations, the dimensions of the ring are taken as in figure 5.4. That means that only one variable remains to be determined. It may be possible to find a better configuration, but this configuration gives a good estimate for the required dimensions and the volume of the required reinforcement ring.

![Diagram of a T-formed reinforcement ring](image)

**figure 5.4**

There are two criteria on which the minimal required cross sectional area of the ring is to be determined.

- **Strength**: The stresses in the ring may not exceed the maximal allowable membrane stress (see § 2.2, figure 2.3)
- **Stability**: Buckling of the ring has to be avoided.
1/ Strength.

The uniform distributed load $f_h$ causes a hoop stress in the ring:

$$
\sigma_h = \frac{f_h \cdot D}{A_r}
$$

With $A_r = \text{cross sectional surface of the ring} = 2 \cdot h \cdot t$

and the relation:

$$
f_h = f_v \cdot \tan \alpha
$$

we find:

$$
\sigma = \frac{f_v \cdot D}{2 \cdot h \cdot t \cdot \tan \alpha}
$$

This stress must be limited to the design stress, since it is a primary membrane stress (see §2.2)

2/ Stability

According to Dubbel's Taschenbuch [6], the critical boundary force can be calculated. If this is done with the dimensions from figure 5.4, we find an expression for the minimum height $h$ of the reinforcement ring:

$$
h > \left( \frac{f_v}{0.031 \cdot \tan \alpha} \right)^{\frac{1}{4}}
$$

The limits then become:

1/ Strength: $h > \left( \frac{f_v \cdot D}{\tan \alpha \cdot t} \right)^{\frac{1}{4}}$

2/ Stability: $h > \left( \frac{f_v}{0.031 \cdot \tan \alpha} \right)^{\frac{1}{4}}$
§5.5 VOLUMES OF TUBESHEET AND REINFORCEMENT RING

**Tubesheet**

The volume of a part of a sphere (see figure 5.5) is [7]:

$$V = \pi j^2 (r - j/3)$$

The volume of a dished tubesheet is the difference between the volumes of two sphere parts: respectively a part with height \( j+e \) from a sphere with radius \( r+e \), and a part with height \( j \) from a sphere with radius \( r \).

This gives for the volume of the dished tubesheet:

$$V = \pi j+e^2 (r+e-(j+e)/3) - \pi j^2 (r-j/3)$$

**Reinforcement ring**

The ring can be split in two parts: a horizontal part (the "leg" of the T) and a vertical part. See figure 5.4.

The volume of the horizontal part is:

$$V = \frac{\pi}{4} \left[ (D + 2(h-h/5))^2 - D^2 \right] \cdot \frac{h}{5}$$

$$= \frac{\pi}{20} \left[ (D + 8/5h)^2 - D^2 \right] \cdot h$$

The volume of the vertical part is:

$$V = \frac{\pi}{4} \left[ (D + 2h)^2 - (D + 8/5h) \right] \cdot h$$

The total volume can be found by adding these volumes.
Calculations are performed with the following parameter values, which represent as good as possible the situation of the considered reactor:

- Pressure drop $\Delta p = 0.6 \text{ N/mm}^2$
- Diameter tubesheet $D = 5730 \text{ mm}$
- Equivalent pressure for dead weight $p_d = 0.265 \text{ N/mm}^2$
- Allowable membrane stress $S = 200 \text{ N/mm}^2$
- Ligament efficiency $\mu = 0.189$

The angle $\alpha$ is varied between $10^\circ$ and $89.9^\circ$.

This gives for minimal required tubesheet thickness $e$, volume tubesheet $V_e$, height of reinforcement ring $h$, volume reinforcement ring $V_r$, and total volume $V_e + V_r$, the values given in table 5.1

<table>
<thead>
<tr>
<th>$\alpha$</th>
<th>$e$ mm</th>
<th>$V_e$ $\text{m}^3$</th>
<th>$h$ mm</th>
<th>$V_r$ $\text{m}^3$</th>
<th>$V_e + V_r$ $\text{m}^3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>145</td>
<td>5.49</td>
<td>426</td>
<td>1.56</td>
<td>7.05</td>
</tr>
<tr>
<td>20</td>
<td>74</td>
<td>2.88</td>
<td>286</td>
<td>0.71</td>
<td>3.59</td>
</tr>
<tr>
<td>30</td>
<td>50</td>
<td>1.98</td>
<td>235</td>
<td>0.44</td>
<td>2.42</td>
</tr>
<tr>
<td>40</td>
<td>39</td>
<td>1.61</td>
<td>156</td>
<td>0.31</td>
<td>1.91</td>
</tr>
<tr>
<td>50</td>
<td>33</td>
<td>1.44</td>
<td>163</td>
<td>0.21</td>
<td>1.65</td>
</tr>
<tr>
<td>60</td>
<td>29</td>
<td>1.39</td>
<td>136</td>
<td>0.14</td>
<td>1.54</td>
</tr>
<tr>
<td>70</td>
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<td>1.43</td>
<td>107</td>
<td>0.09</td>
<td>1.52</td>
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<td>26</td>
<td>1.55</td>
<td>75</td>
<td>0.05</td>
<td>1.60</td>
</tr>
<tr>
<td>89.9</td>
<td>25</td>
<td>1.80</td>
<td>7</td>
<td>0.004</td>
<td>1.80</td>
</tr>
</tbody>
</table>

**table 5.1**

Graphical representation of the result is given in figure 5.6. The total required tubesheet volume is given as a function of the angle $\alpha$. 

![Graphical representation](image-url)
As a reference, the volume of the equal tubesheet is represented by the horizontal dotted line in the graphic:

\[ V = \pi \times \frac{1}{4} \times D^2 \times \text{tot.thickn.}/2 \]

\[ = \pi \times 6.75 \times 0.468 \times \frac{4 \times 2}{4} \]

\[ = 8.4 \, \text{m}^3 \]

**Interpretation**

According to the results, a large tubesheet volume reduction can be achieved for large angles of attachment.

These results, however, are determined considering only primary stresses. It is very well possible (see Assumption 3) that the total (primary plus secondary) stresses are determining. This would mean a larger required tubesheet volume, but it can still be a volume reduction compared with the present required volume.

Conclusion: It is possible that a tubesheet volume reduction can be achieved by using dished tubesheets.

Whether this leads to a cheaper reactor, will be discussed in the next paragraph.

§5.7 EVALUATION OF THE IMPACT OF THE USE OF DISHED TUBESHEETS

The use of dished tubesheets has impact on various areas: strength, manufacturing, total construction. These impacts are mentioned below.

**Strength**
- To avoid buckling of the shell, a reinforcement ring must be implemented. This has already been accounted for.

**Manufacturing**
- Holes must be drilled in spherical plates, under angles less than 90°.
- The dished sheets are more difficult to manufacture than flat sheets.
- Machines are needed, than can handle objects, not only with a diameter of 7 meters, but also a height of several meters.

These points result in an increase in the price per kilo of the tubesheet material.
Construction
- The total construction may have to become higher, in order to make place for the extra room, needed for the dished sheets
- The catalyst handling must be adapted to dished sheets
- Problems can occur with the attachment of the support grids, when the sheets are very dished. Some of these must then be attached to the tubesheets

This gives a higher price for the rest of the construction.

All these aspects are disadvantageous, when dished tubesheets are used:

The strength aspect reduces the gain in kilo's, manufacturing aspects increase the price per kilo, and the constructional aspects reduce the achieved financial gain.

5.8 CONCLUSION

Based on only primary stresses, a considerable reduction of the required tubesheet volume is possible.

However, it is expected, that the total (primary plus secondary) stresses are determining. Since the secondary stresses (due to asymmetrical loads) are not reduced, the achieved gain in tubesheet volume will most probably be much less.

Secondly, the remaining tubesheet volume will cost more per kilo, due to manufacturing aspects.

In third place, the use of dished tubesheets may have large impacts on the total construction, like shell height, catalyst handling. These increase the price of the rest of the construction.

It is therefore not expected, that a price reduction can be achieved, which makes a further examination of the alternative of dished tubesheets interesting.
CHAPTER VII: EVALUATION OF THE BASIC OF THE STRESS LIMITS

§6.1 INTRODUCTION

The stress limits that are used up till now, are based upon elastic, ideally plastic material behavior. This means, that no work hardening is encountered. This chapter will evaluate the result of including work hardening in the material behavior model on the calculation of the tubesheet thicknesses.

§6.2 ELASTIC, IDEALLY PLASTIC MATERIAL BEHAVIOR

This behavior can be represented by figure 6.1. The material is linearly elastic, up till the yield stress \( \sigma_y \). When the yield stress is reached, the material will keep yielding.

![figure 6.1](image)

Primary stresses

These stresses may never exceed the yield stress, since then the material would keep yielding, which would give unallowable distortions.

Total stresses

The total stresses, that are the primary plus the (self relieving) secondary stresses, may not exceed twice the yield stress. This will be explained, under the assumption that the primary stresses do not exceed the yield stress.
Let the calculated stress become higher than the yield stress (point 1). In reality, stresses cannot exceed the yield stress since yielding will occur. Therefore, the material will yield, while the secondary (self relieving) stresses will decrease. This takes place until the total stress has reached the yield stress (point 2). Then the situation is stable.

When now the load is released, pressure stresses will occur (point 3). As long as this load is lower than the yield stress, applying and releasing of the load results in elastic deformations between points 2 and 3.

Suppose now that the total stress exceeds two times the yield stress (figure 6.3, point 1).

Again stress releasing will occur, until the yield stress is reached (point 2). If the load is released, pressure stresses will occur. But now they will exceed the yield stress (point 3), and the material will yield again (point 4). Applying the load (point 5) means again yielding (point 2): every load cycle will mean two times yielding (see figure 6.4).
Since this can cause low cycle fatigue, the total stress must remain below twice the yield stress.

§6.3 INCLUDING WORK HARDENING IN THE MATERIAL BEHAVIOR MODEL

The stress-strain curve will look now as figure 6.4. When yielding takes place, the yield stress increases with the strain.

figure 6.4

**Primary stresses**

If now the primary stress exceeds the yield stress (see figure 6.5, point 1), yielding will stop when the strain is reached, where the yield stress is equal to the primary stress (point 2). This means that higher primary stresses can be allowed, depending on the allowed strain.

figure 6.5

However, when higher primary stresses are allowed, the occurring stress values can come nearer to the real rupture stress. This means that the safety of the construction towards rupture is decreased, and thus less than the Codes prescribe. Therefore it is not recommended to include work hardening in the material behaviour model when primary stresses are evaluated.
Total stresses

Let now the total stress exceed twice the yield stress (figure 6.6, point 1).

\[\text{stress} \quad \text{stress} \quad \text{stress} \quad \text{stress} \quad \text{stress}
\]

\[\text{strain} \quad \text{strain} \quad \text{strain} \quad \text{strain} \quad \text{strain}
\]

Figure 6.6

In reality, the material will yield until the yield stress for this strain is reached (point 2). Note that this is a higher yield stress. When the load is taken away, the pressure stress may increase as long as it is lower than the new yield stress (point 3). Then the deformation remains elastic.

If after unloading the pressure yield stress is exceeded, yielding will occur again, but also the yield stress will increase. So if the total stress is not too high, after a few cycles the loading cycle will give only elastic, thus allowable deformations.

When evaluating total stresses, shakedown to elastic behaviour is required. Including work hardening in the material behaviour model does not change this requirement, it only allows more yield. The safety to rupture remains guarded by the limits for primary stresses.

66.4 Conclusions

Including work hardening in the model of the material behaviour increases the limits for secondary stresses, without loss of safety to rupture. Since the considered reactor tubesheet thicknesses are determined by total stresses, including work hardening will lead to thinner tubesheets, and thus to a cheaper reactor.
APPENDIX 1: TEMPERATURE DIFFERENCES BETWEEN TUBES

Introduction

It is possible, that one or more tubes become (partly) blocked, due to fouling, or faults.

In the tubes, a reaction takes place, by which heat is produced. When now a tube gets (partly) blocked, less gas can flow through, so less reaction will take place. Because less heat is produced, the tube will be colder than the other tubes. Because the cold tube will expand less, an expansion difference occurs between the tubes. This causes discontinuity stresses in the tubesheet, since all the tubes are connected to it. Also the cold tube is stretched, thus an axial stress will occur in the tube.

Assuming the cold tube will be totally blocked, and become as cold as the shell, what is the maximum allowable temperature difference between tubebundle and shell?

Remark: In the calculations, it is assumed that buckling of the tubes is prevented by the support grids.

Modelling

In order to be sure, that the derived limits are conservative, two models will be used:

- One for determining the maximum allowable temperature difference for axial tension in the tube, and shear-stress in the tubesheet.

- A second to find the maximum allowable temperature difference for radial and tangential stresses in the tubesheet.

Model 1

The worst situation for axial stress in the cold tube, and shear stress in the tubesheet, is, when only one tube remains cold. If more tubes were cold, the tubesheet would deflect more, so that the cold tube would be stretched less.

• Assumptions

- No deflection of the tubesheet. This is a conservative assumption, since the calculated stress values will be higher than in practice.

- The place where one cold tube will induce the largest stresses is exactly in the center of the plate. This assumption is based upon the fact, that in the center of the tubesheet, the deflection due to temperature difference between tubebundle and shell is the largest.

- The deflection at the center of the tubesheet due to temperature difference between tubebundle and shell is equal to the expansion difference between these two.

Therefore: The elongation of the cold tube is equal to the expansion difference between tubebundle and shell.
Two types of stresses are considered: axial stress and shear stress.

- Axial stress in the tube.

The lengthening is equal to the temperature difference between tubebundle and stress:

\[ \Delta l = l \cdot \alpha_e \cdot \Delta T \]

Hooke's law:

\[ \sigma = E \cdot \epsilon \]

\[ = \Delta l / l \cdot E_{t} \]

(A1.1) and (A1.2) give:

\[ \sigma = \alpha_e \cdot E_{t} \cdot \Delta T \]

The stress in the tube is thus linearly dependent on the temperature difference.

With the parameter values:

- Temperature expansion coefficient \( \alpha_e = 1.265 \times 10^{-5} \) 1/°C
- Youngs modulus for tube material \( E_t = 184113 \) N/mm²
- Maximum allowable axial stress \( \sigma = 600 \) N/mm² (3*design stress),

the maximum allowable temperature difference is:

\[ T_{(\text{max})} = \frac{600}{1.265 \times 10^{-5} \times 184113} \]

\[ = 257 \degree C \]

- Shear stress \( \tau \) in tubesheet around the tube

The force that must be born by a cylindrical surface around the tube in the tubesheet, is the same as the force that strains the tube.

Thus:

\[ \tau \cdot \text{cross.sect.surf} = \tau \cdot \text{carrying surf. tubesh.} \quad (A1.4) \]

cross sectional surface tube = \( \pi / 4 \cdot (d_i^2 - d_e^2) \)
carrying surface tubesheet = \( 6 \cdot k \cdot e \)
\( k = \) ligament length (see figure A1.1)
Rewriting equation (A1.4), and substituting expressions for the carrying surfaces, the expression for the shear stress $\tau$ becomes:

$$\tau = \frac{k\cdot E\cdot \Delta T\cdot \pi / 4 \cdot (d_c^2 - d_i^2)}{6 \cdot h \cdot e} \quad (A1.5)$$

With parameter values as given above, and

- Maximal allowable shear stress $\tau = \frac{1}{2}$ times maximal allowable membrane stress
  - Inner diameter tube $d_i = 26$ mm
  - Outer diameter tube $d_c = 30$ mm
  - Tubesheet thickness $e = 200$ mm
  - Ligament length $k = 7$ mm,

the maximum allowable temperature difference is:

$$T = \frac{6 \cdot 7 \cdot 200 \cdot 300}{1.265 \cdot 5 \cdot 164113 \cdot (30^2 - 26^2) \cdot \pi}$$

$$= 1538 \, ^\circ C$$

Since we already have an upper limit of $257 \, ^\circ C$, the shear stress in the tubesheet is not determining.

**Model 2**

The maximum allowable temperature difference for radial and tangential stress will now be determined.

- **Assumptions:**
  - The staying action of the tubes is neglected. This is a conservative assumption, since the tubesheet is now calculated to deflect more than in reality, so also the calculated radial and tangential stresses are too high.
  - The cold tube(s) is(are) positioned in the middle of the tubebundle. This is the worst place, since there the deflection of the tubesheet due to temperature difference between tubebundle and shell, and thus the expansion difference, is maximal.
The load, performed by the cold tubes, is thought to be equally distributed over the area in which it is connected to the tubesheet.

This gives the next model (see figure A1.2):

A circular, massive plate, simply supported at the edge, loaded with a uniformly distributed load over a circular area positioned at the center of the plate.

![Figure A1.2](image)

The load on the tubesheet depends on the elongation of the tubes: the more the tubes are stretched, the heavier the load. This lengthening \( \Delta l \) depends on the expansion difference \( \alpha \cdot l/2 \cdot \Delta T \) and the deflection of the tubesheet \( y \):

\[
(A1.6) \quad 1 = \alpha \cdot l/2 \cdot \Delta T - y
\]

with:
- \( \Delta l \) = elongation of tubes
- \( l \) = length of tubes
- \( \alpha \) = therm. exp. coeff.
- \( \Delta T \) = temp. difference
- \( y \) = deflection tubesheet

The central deflection of the tubesheet due to a circular load at the center of the plate is (Roark [4], page 366, case 16):

\[
(A1.7) \quad y = \frac{q \cdot r_c^2 \cdot 12 \cdot (1 - \nu^2) \cdot (3 + \nu)}{15 \cdot E \cdot e \cdot \nu^3} \cdot \left( \frac{2 \cdot \pi^2 \cdot (a^2 - r_c^2) - 2 \cdot r_c \cdot \ln(a/r_c)}{1 + \nu} \right)
\]

with:
- \( q \) = distr. load
- \( E \) = stiffness tubesheet
- \( e \) = thickness tubesheet
- \( \nu \) = poisson ratio tubesheet
- \( a \) = radius tubesheet
- \( r_c \) = load radius

The load \( q \) is the equivalent uniform distributed load, which represents the forces from the cold tubes.

Thus: Force on the tube = force on the tubesheet

\[
(A1.8) \quad N \cdot \Delta l/(1/2) \cdot E \cdot \pi l/4 \cdot (d_i^2 - d_c^2) = q \cdot \pi l \cdot r_c^2
\]

\[
(A1.9) \quad q = \frac{N \cdot \Delta l \cdot E \cdot (d_i^2 - d_c^2)}{4 \cdot 1 \cdot r_c^2}
\]

with:
- \( E \) = stiffness tube mat.
- \( d_i \) = inner tube diameter
- \( d_c \) = outer tube diameter
- \( N \) = amount of cold tubes
These three equations form a set of three equations with three unknowns.

If we rewrite the equations (A1.6), (A1.7) and (A1.8), we find:

(A1.9) \[ \Delta_1 = C1 \cdot \Delta T - y \]
(A1.10) \[ y = C2 \cdot q \]
(A1.11) \[ q = C3 \cdot \Delta_1 \]

To calculate the occurring stresses, the moments must be known. In Roark [4], page 366, case 16, we find:

- radial

(A1.12) \[ M = \frac{a \cdot r_o \cdot e^2}{16} \cdot \left( 4 \cdot (1 + \nu) \cdot \ln(a/r_o) + (1-\nu) \cdot (a^2 - r_o^2) \cdot \frac{r_e \cdot e^2}{a^2} \right) \]

with \( r_e' = \sqrt{1.6 \cdot r_o \cdot e^2 + e^{-2}} \) \(- 0.675 \cdot e \)

- tangential

(A1.13) \[ M = \frac{a \cdot r_o \cdot e^2}{16} \cdot \left( 4 \cdot (1 - \nu) \cdot \ln(a/r_o) + (1-\nu) \cdot (4 - r_o \cdot e^2) \right) \]

The stresses can be calculated from (Roark [4], page 332):

(A1.14) \[ \sigma = \frac{6 \cdot M}{e^2} \]

The moments are linearly dependent on the distributed load \( q \), and thus (see equation (A1.5 and 11)) linearly dependent on the temperature difference.

Because (see equation (A1.14)) the stresses are linearly dependent on the moments, the stresses are also linearly dependent on the temperature difference.

Rewriting equations (A1.12), (A1.13) and (A1.14) gives

(A1.15) \[ M = C4 \cdot q \]
(A1.16) \[ M = C5 \cdot q \]
(A1.17) \[ \sigma = C6 \cdot M \]
With this, we find:

- radial: \( \sigma_r = \frac{C_1 \cdot C_2 \cdot C_4 \cdot C_6}{C_2 \cdot C_3 + 1} \cdot \Delta T \)

- tangential: \( \sigma_\theta = \frac{C_1 \cdot C_3 \cdot C_5 \cdot C_6}{C_2 \cdot C_3 + 1} \cdot \Delta T \)

Thus, the maximal allowable temperature difference is:

\[
\Delta T_{\text{max}} = \frac{C_2 \cdot C_3 + 1}{C_1 \cdot C_3 \cdot C_4 \cdot C_6} \cdot S
\]

\[
\Delta T_{\text{max}} = \frac{C_2 \cdot C_3 + 1}{C_1 \cdot C_3 \cdot C_5 \cdot C_6} \cdot S
\]

These constants can be calculated for different values of the amount of cold tubes.

The results are given in figure A1.3 where the maximal allowable temperature difference is given for radial (and largest) stress, as a function of the amount of cold tubes.

The limit will always be higher than 5000°C temperature difference.

Since a limit of 257°C is already given, the radial and tangential stresses in the tubesheet are not determining.

**Conclusion**

The maximal allowable temperature difference between tubebundle and shell is 257°C. Because in practice, this temperature difference will not exceed 100°C, the occurrence of cold tubes in the tubebundle will never be limiting for the allowable temperature difference between tubebundle and shell.
APPENDIX 2: CALCULATIONS ACCORDING TO BS 5500 1965

Introduction

The British Standard 5500 gives rules to determine the dimensions of parts of pressure vessels.
In the rules for the tubesheet thickness a remark is made for large vessels:

When the shell has a diameter greater than 1500 mm, the simplified equations that are given in the BS, may lead to excessive tubesheet thicknesses. For these exchangers a full tubesheet stress analysis is recommended.

Since the vessel under consideration has a shell diameter of 6700 mm, this recommendation must be followed. This has been done: a FEM-analysis is performed.
This calculation is done in order to control how much the difference is between the results of the FEM-analysis and of the BS.

Calculations

The rules of the BS are based on symmetrical loads: this means that loads like dead weight and pressure drop in the tubes due to catalyst filling are not included.
Therefore we first calculate the minimal required thickness for symmetrical loads.
Later some remarks are made for antisymmetric loads.

Symmetrical loads

- Characteristics of the tubesheet:
  - Ligament efficiency \( \mu \)
    Because the tubes are welded, the ligament efficiency becomes:
    \[ \mu = \frac{P - dh}{P} = \frac{37 - 30}{37} = 0.188 \]

    The reactor is designed with two fixed tubesheets. Therefore it belongs to the exchangers with unconfined heads.

- Required constants:
  - Mean strain ratio tube bundle/shell \( K \)
    \[ K = \frac{E_t \cdot e_t \cdot (D - e_s)}{E_s \cdot e_s \cdot N \cdot (d - e_d)} \]
    \[ = 0.332 \]
- Tubesheet support factor $F$

The support factor depends on the ratio shell wall thickness/shell inner diameter.
From figure 3.9(1) then follows:

$$F = 1.0$$

- Tubesheet factor $F_p$

$$F_p = 0.25 + (F-0.6) \times \left( \frac{300+e_s^{-2}E_s}{K+L+E} \right) \times (D_t)^{0.25}$$

$$= 0.25 + 0.4 \times \left( \frac{1.961 \times 10^6}{(12000-2*e)\times 10^3} \right)$$

or 1.0, whichever is greater.

- Factor $f$

$$f_s = 1 - N \times (d/D_1)^2$$

$$= 0.479$$

- Factor $f$

$$f_s = 1 - N \times \left( \frac{2*e_s}{D_2} \right)^2$$

$$= 0.608$$

* Equivalent pressures

- Pressure due to thermal expansion

$$p_s = \frac{4 \times E_s \times e_s \times (\alpha_s \times T_s - \alpha_e \times T_e)}{(D-3*e) \times (1+K \times F_p)}$$

$$= 7930 \times \left( 1.379 \times 10^4 - 1.26 \times 10^4 \right)$$

$\left( 1 + 0.332 \times F_p \right)$

- Pressure shell side

$$p_s' = \frac{0.4 \times (1.5 + K \times (1.5 + e_s)) \times p_s}{(1 + 0.332 \times F_p)}$$

$$= 3.192$$

$\left( 1 + 0.332 \times F_p \right)$

The equivalent pressure shell side $p_s'$ is the larger of $p_s'$ and $(p_s' + p_e)/2$
- Pressure channel side

\[ p_{ch}' = \frac{1 + 0.33 \times (1.5 + \rho) \times p_{ch}}{1 + 0.33 \times F} \]

\[ = \frac{5.632}{1 + 0.33 \times F} \]

The equivalent pressure channel side \( p_{ch}' \) is the larger of \( p_{ch}' \) and \( (p_{ch}' + p_{ch})/2 \)

* Minimum required tubesheet thicknesses:

The required tubesheet thickness is the larger of:

- Bending shell side

\[ e = \frac{F \times D_1}{2} \sqrt{\frac{p_{ch}'}{f}} \]

- Bending channel side

\[ e = \frac{F \times D_2}{2} \sqrt{\frac{p_{ch}'}{f}} \]

- Shear shell side

\[ e = 0.155 \times D_1 \times \rho_{ch} \]

\[ \text{lambda} \times \text{tau} \]

- Shear channel side

\[ e = 0.155 \times D_2 \times \rho_{ch} \]

\[ \text{lambda} \times \text{tau} \]

with \( \text{lambda} = \mu \) and \( \text{tau} = f/2 \)

**Results**

Regarding only symmetrical loads (constant pressure channel side, pressure shell side, and temperature difference), two possible worst cases for which the tubesheet thickness has to be determined are present:

I Maximal load inward:
- No temperature difference
- No shell side pressure
- Maximum constant channel side pressure

II Maximal load outward:
- Maximum temperature difference
- Maximum shell side pressure
- No constant channel side pressure

See figure 2.4 (a), (c) and (d).
For these situations, the occurring stresses are calculated; the largest resulting value for the tubesheet thickness \( e \) is the required value.

<table>
<thead>
<tr>
<th>Case</th>
<th>( p )</th>
<th>( p' )</th>
<th>( t )</th>
<th>( e )</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>0.0</td>
<td>4.4</td>
<td>0</td>
<td>322</td>
</tr>
<tr>
<td>II</td>
<td>3.7</td>
<td>0.0</td>
<td>50</td>
<td>220</td>
</tr>
</tbody>
</table>

Thus, for symmetrical loads, the minimal required tubesheet thickness according to the British Standard 5500 1985 is 322 mm exclusive corrosion allowance per tubesheet.

The fact that case I is determining is easy to understand: the temperature difference of 0 °C or 50 °C gives about the same load (see Appendix 5). The other difference in loading are pressures channel side and shell side. Since pressure channel side is larger, case I will be determining.

The required tubesheet thickness according to BS 5500 is already higher than the thickness derived with FEM-analysis, while not even the antisymmetrical loads are accounted for. Therefore, the rules of the BS indeed lead to excessive tubesheet thicknesses.

**Antisymmetrical loads**

A calculation that accounts for antisymmetrical loads is not directly provided in the BS. The reason for it is, that the BS gives rules for small (neglectable dead weight) heat exchangers (no pressure drop since no catalyst in the tubes). Therefore the antisymmetrical loads that occur in the considered large reactor vessel can not be accounted for in the BS.

**Conclusion:**

The rules given by the British Standard 5500 1985 do for this reactor, for part of the load, lead to much thicker tubesheets than achieved with the FEM-analysis for the total load!
APPENDIX 2: CALCULATIONS ACCORDING TO ASME NUCLEAR CODE A 8000

Introduction

The ASME NUCLEAR CODE A 8000 gives expressions to determine the stresses in perforated plates under several loads. For determination of the tubesheet thickness, this code is followed.

Question is, like in appendix 2, whether this code leads to excessive thicknesses.

The required thickness due to antisymmetric load is easy to calculate with these rules. However, for the thickness for bearing symmetric load, some problems occurred:

1/ It was mentioned that the staying action of the tubes could be accounted for, but no indication was given how.

2/ The calculation of the effect of temperature difference also was only mentioned, but no guidelines were included.

To show that the Nuclear Code leads to thicker tubesheets than achieved with the FEM-analysis, only a calculation with antisymmetrical loads is satisfactory.

Load description

Only antisymmetrical loads are implemented. This means, that the uniform distributed load consists in:

- pressure drop over the tubes due to catalyst \( \Delta p = 0.6 \text{ N/mm}^2 \)
- equivalent pressure for dead weight \( p_q = 0.2 \text{ N/mm}^2 \)

Assuming that the load is equally distributed over the two tubesheets, the uniform distributed load per sheet is:

\[
q = \frac{0.6 + 0.2}{2} = 0.4 \text{ N/mm}^2
\]

The tubesheet is loaded with this uniform distributed load \( q \), a boundary load to make equilibrium with the load \( q \), and a moment \( M \) at the boundary due to the attachment to the shell (see figure A3.1).

![Figure A3.1](image-url)
According to the examples in the code, we divide this load in two distinct cases:

I Pressure on a circular plate, simply supported at the edge.
II Moment on the outer boundary of a circular plate.

Case I

A drawing for the loading situation is given in figure A3.2

The radial stress at the upper surface of the plate is given in equation (A3.1):

\[
G_r = -\frac{\pi}{8} \cdot \left( \frac{R}{e} \right)^2 \cdot \left( 3 + \nu \right) \cdot \left( 1 - \left( \frac{r}{R} \right)^2 \right) \cdot q
\]

Substituting the values of the parameters that are known:
- radius of outer tube limit circle \( R = 3228 \text{ mm} \)
- Poisson ratio \( \nu = 0.49 \) (from fig A-6131-1, with ligament efficiency = 0.189)
- uniform distributed load \( q = 0.4 \text{ N/mm}^2 \)
- tubesheet thickness \( e \) is constant

This gives:

\[
G_r = -5.4568 + 0.523r^2
\]

\[
G_r = 0.523r^2
\]
The tangential stress at the upper surface of the plate is given in equation (A3.3) as a function of the distance \( r \) to the center of the plate:

\[
\sigma_t = -\frac{3}{6} \times (\frac{R}{r})^2 \times \left(3 + \frac{D}{R}\right) - (1 - 3\times D) \times (\frac{r}{R})^2 + q
\]

Substituting the values of the parameters that are known gives:

\[
\sigma_t = -6.45E6 + 0.371\times r^2
\]

Graphically represented:

![Graph of radial and tangential stress](image)

**Figure A3.3**

**Case II**

The moment at the edge gives a constant stress at the upper surface of the tubesheet:

\[
\sigma = 6 \times \frac{M}{e^2}
\]

It is not given what the value of the moment is; if the plate is simply supported, the moment is zero.

The maximum moment occurs when the plate is fixed; then the moment is:

\[
M_\text{t} = M_\text{r} = \frac{r \times (D/2)^2}{8}
\]

Since the shell has a finite stiffness, the real moment will be between these moments.
The total stress at the surface is the sum of the stress due to the pressure and the stress due to the moment. (see Figure A3.4)

Radial and tangential stresses are shown in Figure A3.4.

The total stress is calculated from:

\[ \sigma_t = \sqrt{\sigma_r^2 + \sigma_\theta^2} \]

Graphically represented:

The stress distribution is optimal, when the stress in the center is equal to the stress at the edge. The then resulting maximal stress is:

\[ \sigma_m = \frac{1}{2} \cdot \sqrt{\sigma_r(0)^2 + \sigma_\theta(0)^2} \]

\[ = \frac{3.85 \cdot 10^6}{\text{E}^{-2}} \]

Remark: Attention must be paid to the fact, that this is the best possible situation, not the worst!
Stress intensity:

The stress intensity is:

\[ S = K \cdot (F/h) \cdot \sigma \]

Determination of stress concentration factor \( K \):

The stress ratio is the smallest when \( r = R \):

\[ \beta = \frac{\sigma_y(R)}{\sigma_y(R)} = \frac{1.135E6}{2.725E6} = 0.42 \]

From fig A-B142-1 follows: \( K = 1.04 \)

Thus the stress intensity is:

\[ (A3.7) \quad S = \frac{2.12E7}{e^{-2}} \]

Minimal required thickness:

The allowable stress \( S \) is 300 N/mm\(^2\), since the occurring stresses are primary bending stresses.

After rewriting equation (A3.7) and substituting the parameter values, the required tubesheet thickness can be found:

\[ e \ (\text{min}) = 266 \ \text{mm} \]

Conclusion

With very unconservative assumptions, and only part of the total load, the required tubesheet thickness according to the Nuclear Vessel Code is larger than the thickness following from the FEM-analysis.

Therefore the conclusion can be drawn, that this code leads to excessive thick tubesheets.
APPENDIX A: EVALUATION OF THE GROOVE IN THE TUBESHEET

Introduction

In the present design, a groove is made in the unperforated part of the tubesheet (see figure A4.1)

![Figure A4.1](image1)

![Figure A4.2](image2)

The load that induces primary stress in the shell is the channel side pressure. A hoop stress is induced, and that stress must remain below the design stress, since it is a primary membrane stress.

Equilibrium of forces gives (see figure A4.3):

\[ p \times 2R = 2 \times \sigma \times e \]

With \( S \) is the design stress follows for the minimal required shell thickness:

\[ \text{minimal } e = \frac{p \times R}{S} \]

With parameter values:
- channel side pressure \( p = 4.4 \text{ N/mm}^2 \)
- radius shell \( R = 3399 \text{ mm} \)
- allowable hoop stress \( S = 200 \text{ N/mm}^2 \)

the required shell thickness is \( 74 \text{ mm} \).
Between the two tubesheets, the pressure is 3.7 N/mm². Then the required thickness is 65 mm.

• Secondary stresses

These stresses occur mostly around the attachment of tubesheet to shell, because of discontinuities.

Therefore we first cut the construction of figure A4.2 in three pieces: upper shell (a), tubesheet (b), and lower shell (c).

For these parts, we derive formulae for the deformations because of forces and moments.

The deformations of adjacent parts have to be equal. With this, we can calculate the distortions, and the stresses that occur.

The total (primary plus secondary) stresses must then be limited to three times the design stress (see §2.2).

Derivation of the formulae

a. Upper shell

The shell is loaded with the channel side pressure, and forces and moments at the attachment edge. See figure A4.3:

\[ \text{figure A4.3} \]

--The radial displacement due to the pressure:

The hoop stress is \( \sigma_h = \frac{p \cdot R}{t} \)

With Hooke's law: \( \sigma = \epsilon \cdot E \) follows:

\[ \epsilon = \frac{p \cdot R}{t \cdot E} \]
The radius then becomes:

\[ R + \Delta R = \frac{\text{stretched circumference}}{2 \cdot \pi} = \frac{(1 + \varepsilon) \cdot 2 \cdot \pi \cdot R}{2 \cdot \pi} = (1 + \varepsilon) \cdot R \]

Thus the radial displacement is:

\[ \Delta R = \varepsilon \cdot R = \frac{e \cdot R - \delta}{e \cdot \varepsilon} \]

Due to the constant pressure channel side, no deflection is induced.

The deformations due to a force \( V_A \) are (Roark [4], page 461 case 8):\[
\begin{align*}
\Delta y &= -\frac{V_A}{2 \cdot D \cdot A^{-3}} \\
\Delta \varphi &= \frac{V_A}{2 \cdot D \cdot A^{-2}}
\end{align*}
\]

The deformations due to a moment \( M \) are (Roark [4], page 461 case 8):\[
\begin{align*}
\Delta y &= \frac{M}{2 \cdot D \cdot A^{-2}} \\
\Delta \varphi &= -\frac{M}{D \cdot A}
\end{align*}
\]

Total:\[
\begin{align*}
\Delta y &= \frac{D \cdot R \cdot 2}{e \cdot \varepsilon} + \frac{M}{2 \cdot D \cdot A^{-2}} - \frac{V_A}{2 \cdot D \cdot A^{-3}} \\
\Delta \varphi &= -\frac{M}{D \cdot A} + \frac{V_A}{2 \cdot D \cdot A^{-2}}
\end{align*}
\]

**Tubesheet**

The tubesheet is loaded with (see figure A4.4):
Deformations due to the symmetrical pressure $p_s$:

$$y = \frac{p_s + p_e}{E}$$

$$\varphi = 0$$

Deformations due to the antisymmetrical pressure $p_e$:

upper surface: $y = \frac{1}{2}e \times \sin (\varphi_t)$

lower surface: $y = \frac{1}{2}e \times \sin (\varphi_t)$

$$\varphi_t = \frac{-p_e \times R^3 \times 12 \times (1 - \nu)}{8 \times E \times e^{-3}}$$

Remark: in these calculations the stepping effect of the tubes is omitted. If this effect is included, the tubesheet deflection is reduced remarkably, which may influence the conclusion of this appendix rigorously!

Deformations due to the forces $V_A$ and $V_B$:

$$y = \frac{R}{E \times e} \times (1 - \nu) \times (V_A - V_B)$$

$$\varphi_t = 0$$

Deformations due to the moments $M_A$ and $M_B$:

upper surface: $y = \frac{1}{2}e \times \sin (\varphi_t)$

lower surface: $y = -\frac{1}{2}e \times \sin (\varphi_t)$

$$\varphi_t = \frac{12R \times (1 - \nu) \times (M_A + M_B)}{E \times e^{-3}}$$

Thus:

$$\varphi_t = \frac{12 \times (1 - \nu) \times R}{E \times e^{-3}} \times \left( \frac{-p_e \times R^2 + M_A + M_B}{8} \right)$$
Lower shell

This part is loaded with (see figure A4.5)

- a force \( V_b \)
- a moment \( M_\alpha \)

\[
\begin{align*}
\gamma_A &= \frac{P \cdot \nu \cdot \alpha}{E} + \frac{1}{E} \sin \left( \frac{\varphi}{\nu} \right) + \frac{R \cdot (1 - \nu) \cdot (\nu \cdot \alpha - \alpha \cdot \beta)}{E \cdot e} \\
\gamma_B &= \frac{P \cdot \nu \cdot \alpha}{E} - \frac{1}{E} \sin \left( \frac{\varphi}{\nu} \right) + \frac{R \cdot (1 - \nu) \cdot (\nu \cdot \alpha - \alpha \cdot \beta)}{E \cdot e}
\end{align*}
\]

figure A4.5

The deformations due to the force \( V_b \):

\[
\begin{align*}
\gamma &= -\frac{V_b}{2D \cdot \alpha^3} \\
\varphi &= \frac{V_b}{2D \cdot \alpha^2}
\end{align*}
\]

The deformations due to the moment \( M_\alpha \):

\[
\begin{align*}
\gamma &= \frac{M_\alpha}{2D \cdot \alpha^3} \\
\varphi &= -\frac{M_\alpha}{D \cdot \alpha^2}
\end{align*}
\]

Thus the total displacements and deflections are:

\[
\begin{align*}
\gamma &= -\frac{M_\alpha}{2D \cdot \alpha^3} + \frac{V_b}{2D \cdot \alpha^3} \\
\varphi &= \frac{M_\alpha}{D \cdot \alpha^2} + \frac{V_b}{2D \cdot \alpha^2}
\end{align*}
\]
Combination

Equating the formulae for the displacements and deflections of the adjacent surfaces of the different parts gives four linear equations with four unknowns.

This set of equations is solved with the next parameter values:

- Equivalent tubesheet stiffness (from FEM-analysis): 24164 N/mm²
- Thickness tubesheet = 200 mm (arbitrarily)
- Equivalent Poisson ratio tubesheet material = 0.492
- Stiffness shell material = 194220 N/mm²
- Poisson ratio shell material = 0.3
- Symmetric pressure = 4.4 N/mm²
- Antisymmetric pressure = 0.3 N/mm²
- Design stress = 200 N/mm²

The solution is found using a personal computer. The results are:

<table>
<thead>
<tr>
<th>PARAMETER VALUES</th>
<th>SOLUTION</th>
<th>ABSOLUTE ERROR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equivalent tubesheet stiffness</td>
<td>24164</td>
<td>-5.0E-06</td>
</tr>
<tr>
<td>Thickness tubesheet</td>
<td>200</td>
<td>-2.0E-06</td>
</tr>
<tr>
<td>Equivalent Poisson ratio</td>
<td>0.492</td>
<td>-5.0E-06</td>
</tr>
<tr>
<td>Stiffness shell material</td>
<td>194220</td>
<td>-2.0E-06</td>
</tr>
<tr>
<td>Thickness shell</td>
<td>54</td>
<td>-3.0E-06</td>
</tr>
<tr>
<td>Poisson ratio shell material</td>
<td>0.3</td>
<td>-4.0E-06</td>
</tr>
<tr>
<td>Symmetric pressure</td>
<td>4.4</td>
<td>-3.0E-06</td>
</tr>
<tr>
<td>Antisymmetric pressure</td>
<td>0.3</td>
<td>-5.0E-06</td>
</tr>
<tr>
<td>Design stress</td>
<td>200</td>
<td>-6.0E-06</td>
</tr>
</tbody>
</table>

Thus, for a shell thickness of 54 mm, the stresses in shell at the connecting point A reaches the limit for total stresses of 600 N/mm².

**Evaluation**

The calculations are only done with pressure loads. The required thicknesses for primary stresses are 74 mm and 65 mm.

Total (primary plus secondary) stresses only due to pressure loads do not require a reinforcement of the shell. If however temperature/expansion differences are taken into account, it is very well possible that the total stresses will give reason for implementing this reinforcement. To find this out, a calculation including the temperature/expansion differences must be performed.
APPENDIX 5: TEMPERATURE DIFFERENCE BETWEEN TUBE Bundle AND SHELL

Introduction

In this appendix, the question is handled what is the influence of the temperature difference between tube bundle and shell on the tubesheet thickness. Several ways are possible to determine this influence: the codes can be examined, or a FEM-analysis can be performed.

The first possibility will be executed: a calculation of the required tubesheet thickness according to the British Standard 5500 1985, paragraph 3.9, due to only temperature difference. The result is given for a scope of temperature differences.

Calculation

The equivalent pressure for temperature difference is (BS §§3.9.4.3.3):

\[ p_e = \frac{4 \cdot E \cdot e \cdot ((\kappa_s \cdot T_s - \kappa_t \cdot T_t))}{(D - 3 \cdot e) \cdot (1 + K \cdot F)} \]

Parameter values:
- stiffness shell material \( E = 194220 \, \text{N/mm}^2 \)
- thickness shell \( e = 66 \, \text{mm} \)
- therm. exp. coeff. shell material \( \kappa_s = 1.379 \times 10^{-5} \, \text{1/°C} \)
- therm. exp. coeff. tube material \( \kappa_t = 1.265 \times 10^{-5} \, \text{1/°C} \)
- outside diameter shell \( D = 6666 \, \text{mm} \)
- mean strain ratio \( K = 0.329 \) (See appendix 2)
- design stress \( f = 200 \, \text{N/mm}^2 \)
- tubesheet support factor \( F = 1.0 \) (See appendix 2)

Then follows:

\[ p_e = 7930 \cdot \frac{(1.379 \times 10^{-5} \cdot T_s - 1.265 \times 10^{-5} \cdot T_t)}{(1 + K \cdot F)^2} \]

The required thickness is:

\[ e = \frac{F \cdot D}{2} \cdot \sqrt{\frac{p_e}{f}} \]

\[ e = 168.25 \cdot \sqrt{p_e} \]  \hspace{1cm} (A5.2)

\( F \) (see equation A5.1) depends on the tubesheet thickness \( e \):

\[ F \approx 0.25 + 4744 \cdot \frac{(12000-2\cdot e)^{3/2}}{e} \]  \hspace{1cm} (A5.3)

For chosen values of the temperatures shell and channel side, the minimum required tubesheet thickness has the value, that satisfies both equation (A5.1) and (A5.2).
These values are determined for a constant shell side temperature = 245 °C, and a temperature difference which is positive when the channel side is hotter.

<table>
<thead>
<tr>
<th>T °C</th>
<th>e mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>-70</td>
<td>221</td>
</tr>
<tr>
<td>-50</td>
<td>185</td>
</tr>
<tr>
<td>-30</td>
<td>145</td>
</tr>
<tr>
<td>-10</td>
<td>100</td>
</tr>
<tr>
<td>0</td>
<td>74</td>
</tr>
<tr>
<td>10</td>
<td>44</td>
</tr>
<tr>
<td>15</td>
<td>27</td>
</tr>
<tr>
<td>20</td>
<td>3</td>
</tr>
<tr>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>30</td>
<td>43</td>
</tr>
<tr>
<td>50</td>
<td>99</td>
</tr>
<tr>
<td>70</td>
<td>144</td>
</tr>
<tr>
<td>90</td>
<td>184</td>
</tr>
<tr>
<td>110</td>
<td>220</td>
</tr>
</tbody>
</table>

Graphically:

A minimum required thickness occurs when the temperature difference is 20 °C. Then the expansion of the shell and the tube bundle is equal: the being hotter of the tube bundle is compensated by the higher thermal expansion coefficient of the shell material.

Conclusion.

The construction materials are chosen in a way, that the operation temperatures give the least expansion difference between tube bundle and shell. Therefore, the influence of the temperature difference is minimalized.
APPENDIX B: DETERMINATION OF EQUIVALENT PRESSURE FOR DEAD WEIGHT

One of the loads that is carried by the tubesheets is the dead weight of: tubes, cooling fluid, catalyst, and tubesheets. Since these components are uniformly distributed over the reactor, this load can be accounted for by an equivalent pressure or uniformly distributed load.

Therefore first the weight of the distinct parts will be calculated. Then the equivalent load will be determined.

DETERMINATION OF THE LOADS.

Tubes

Data:
- amount of tubes: 26220
- length of tubes: 12 m
- inner diameter: 0.026 m
- outer diameter: 0.030 m
- specific weight: 9872 kg/m³

Then the weight is:

\[ \text{weight} = 26220 \times 9872 \times \pi/4 \times (0.030^2 - 0.026^2) \times 12 \times 9.81 \]

\[ = 5360755 \text{ N} \]

Cooling fluid

Data:
- medium: water
- length of volume: 12 m
- cross sectional diameter: 6.73 m
- reduction: volume of 26220 tubes
- specific weight: 1000 kg/m³

This gives for the weight:

\[ \text{weight} = \pi/4 \times (6.73^2 - 26220 \times 0.030^2) \times 12 \times 1000 \times 9.81 \]

\[ = 2005647 \text{ N} \]

Catalyst

Data:
- inner diameter tubes: 0.026 m
- length tubes: 12 m
- specific weight catalyst = 850 kg/m³
- amount of tubes = 26220
This gives for the weight:

\[ \pi/4 \cdot 0.005^2 \cdot 12 \cdot 25220 \cdot 850 \cdot 9.81 \]

\[ = 1392959 \text{ N} \]

**Tubesheets:**

Data:
- amount: 2
- diameter 6.73 m
- thickness (estimated) 0.250 m
- amount of holes: 26 220
- diameter holes: 0.030 mm
- specific weight tubesheet material: 7850 kg/m³

This gives for the weight:

\[ \pi/4 \cdot (6.73^2 - 26220 \cdot 0.03^2) \cdot 7850 \cdot 0.250 \cdot 2 \cdot 9.81 \]

\[ = 656079 \text{ N} \]

**Total weight**

The total weight becomes:

5360755 + 2005847 + 1392959 + 656079

\[ = 9415640 \text{ N} \]

**EQUIVALENT PRESSURE**

The pressure must be exerted on the total tubesheet surface

Data
- weight = 9415640 N
- diameter tubesheet 6730 mm

The equivalent pressure becomes:

\[ 9415640 \div (\pi/4 \cdot 6730^2) \]

\[ = 0.265 \text{ N/mm}^2 \]
### Appendix 7: Numerical Values for Graphics

#### Figure 4.7

**Thickness ratio = a [-]**

Minimal required thickness for symmetrical loads = e [mm]

<table>
<thead>
<tr>
<th>a</th>
<th>e</th>
<th>a</th>
<th>e</th>
<th>a</th>
<th>e</th>
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#### Figure 4.6

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### Parameter List

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<tr>
<th>Parameter</th>
<th>Description</th>
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<tr>
<td>a</td>
<td>Thickness ratio thick/thin tubesheet</td>
</tr>
<tr>
<td>d₀</td>
<td>Outside diameter of tube</td>
</tr>
<tr>
<td>dᵢ</td>
<td>Inside diameter of shell</td>
</tr>
<tr>
<td>dₜ</td>
<td>Tube hole diameter</td>
</tr>
<tr>
<td>D₀</td>
<td>Outside diameter of shell</td>
</tr>
<tr>
<td>D₁</td>
<td>Diameter of outer tube limit circle</td>
</tr>
<tr>
<td>D₂</td>
<td>Diameter to which shell fluid pressure is exerted</td>
</tr>
<tr>
<td>Dₚ</td>
<td>Diameter to which tube fluid pressure is exerted</td>
</tr>
<tr>
<td>e₀</td>
<td>Tubesheet thickness exclusive corrosion allowance and partition groove</td>
</tr>
<tr>
<td>e₁</td>
<td>Tubesheet thickness for symmetrical loads</td>
</tr>
<tr>
<td>e₂</td>
<td>Additional tubesheet thickness for antisymmetrical loads</td>
</tr>
<tr>
<td>e₆</td>
<td>Shell thickness including corrosion allowance</td>
</tr>
<tr>
<td>E</td>
<td>Nominal tube thickness</td>
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<tr>
<td>Eₜ</td>
<td>Elastic modulus of tube sheet material</td>
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<tr>
<td>Eₛ</td>
<td>Elastic modulus of shell material</td>
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<td>Elastic modulus of tube material</td>
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<tr>
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<td>Nominal design strength of tubesheet material</td>
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<tr>
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<td>Total distributed boundary force</td>
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<td>fᵥ</td>
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<td>h</td>
<td>Height of reinforcement ring</td>
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<td>j</td>
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<td>k</td>
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<td>L</td>
<td>Tube length between inner faces of tubesheets</td>
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<tr>
<td>m</td>
<td>Moment per length</td>
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<td>Number of tube holes in tubesheet</td>
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<tr>
<td>N₂</td>
<td>Tube pitch</td>
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<tr>
<td>p₁</td>
<td>Shell side design pressure</td>
</tr>
<tr>
<td>p₂</td>
<td>Tube side (channel side) pressure</td>
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<tr>
<td>P</td>
<td>Equivalent pressure for dead weight</td>
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<td>P₀</td>
<td>Pressure drop over the tubes due to catalyst in the tubes</td>
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<tr>
<td>q₀</td>
<td>(Uniformly) distributed load</td>
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<td>Part of antisym. load q that is carried by the thin tubesheet</td>
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<td>Total load carried by the thick tubesheet</td>
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<tr>
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<td>radius of concentrated uniformly distributed load</td>
</tr>
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<td>Maximal allowable stress</td>
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<td>Maximal allowable value for stresses due to antisym. loads</td>
</tr>
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<td>sₜ</td>
<td>Maximal allowable value for stresses due to symmetrical loads</td>
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<tr>
<td>Tₛ</td>
<td>Mean shell metal temperature lower 10°C</td>
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<td>Tₖ</td>
<td>Mean tube wall metal temperature less 10°C</td>
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<td>Thermal expansion coefficient of shell material</td>
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