Resource A

PAPER 5

Catastrophic failure of a polypropylene tank Part II: comparison of the DVS 2205 code of practice and the design of the failed tank

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Catastrophic failure of a polypropylene tank Part II: comparison of the DVS 2205 code of practice and the design of the failed tank

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Abstract

The design of a failed, large (20 m^3) polypropylene storage tank is compared with the recommendations of the German Code of Practice, DVS 2205, to which it allegedly conformed. It is shown that the tank was seriously under-designed, and that the situation was exacerbated by the introduction of residual tensile stresses in its walls during its manufacture.

Keywords: Code of practice; Design; Failure; Polypropylene; Standard; Tank; Weld

1. Introduction

The problem of designing load-bearing structures in plastics differs from that of designing comparable structures in metals such as steels in several important ways, particularly if the design life of the structure is intended to be a long one (20 or 30 years, say). These differences arise because the behaviour of plastics under load is not only time-dependent but also non-linear, because their range of recoverable strains is typically some ten times larger than in metals, because plastics can often be more sensitive to stress concentrations than metals, and because plastics react in a different way to environmental agents than metals. Failure to appreciate these differences has led (and unfortunately still does lead) to premature failure of plastics products, and to their acquiring an early reputation for being 'cheap and nasty'.

The basis of much rational design with plastics is the so-called 'pseudo-elastic design method' proposed initially by Baer et al. [1]. In this, the appropriate time- and temperature-dependent values of modulus and Poisson's ratio are substituted for the elastic ones in the standard stress strain solutions for a given loading configuration and part geometry. Initially, before sufficiently

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comprehensive data on the creep and creep rupture behaviour of specific plastics became available, this approach was limited to strains small enough that an assumption of linear viscoelastic behaviour was a good approximation. Nowadays, this restriction does not apply since copious data are available on all the commoner thermoplastics largely generated from investigations into the long-term behaviour of materials for pressurised pipes.

One of the few, perhaps the only, report of a combined theoretical and experimental investigation into the design against failure of large plastics tanks is that of Forbes et al. [2]. They applied the pseudo-elastic design method to polypropylene tanks with capacities up to 9100 gallons (41 m^3). The design was based on a stress analysis solution of a fourth order linear differential equation as given by Timoshenko and Woinowsky-Kreiger [3] which takes into account the effects of the transition from horizontal base to vertical wall and of transitions in wall thickness. These effects are manifested as increases in the radial expansion of the tank walls just above the transition points, but they can also be thought of as kinds of stress concentrating features. Using a limiting hoop strain of 1%, the results of this analysis produced a design chart for the wall thickness of tanks of increasing capacity up to 10,000 gallons (45 m^3). Their results were validated by full-scale tests on two large tanks.

The failure of a 20 m^3 polypropylene storage tank and the ensuing investigation were described in Part I of this work [4]. The tank was constructed to a design which was verified by the calculations of a consultant engineer and allegedly conformed to the design code DVS 2205 [5], the German Code of Practice for the design of free-standing thermoplastics containers (there is no corresponding British Standard, although there is one for GRP tanks, BS4994: 1987). This code of practice provides a guide to the determination of the maximum permissible stresses that will avoid different modes of failure in thermoplastics containers over specified lifetimes. It takes into account, interalia, the type of thermoplastic, its chemical interaction, if any, with the contents of the container, the operating temperature, and effects arising from changes of wall section and method of manufacture. This paper reviews the design methodology of DVS 2205, and compares the design of the failed tank with the detailed recommendations that result from DVS 2205. Figure 1 shows the dimensions of the tank as designed (taken from the design sketch), together with the wall thicknesses, in mm, at different heights.

2. Design methodology of DVS 2205

The following translated extract from DVS 2205, Part 1 [5] outlines the essentials of the design methodology.

3. Strength parameters

3.1. General

The fundamental bases of the design calculations are the long-term values of materials parameters. In general, depending on the type of loading, three limiting criteria are possible:

(1) stress or strain
(2) deformation (e.g. excessive bending)
(3) stability (e.g. kinking or buckling)

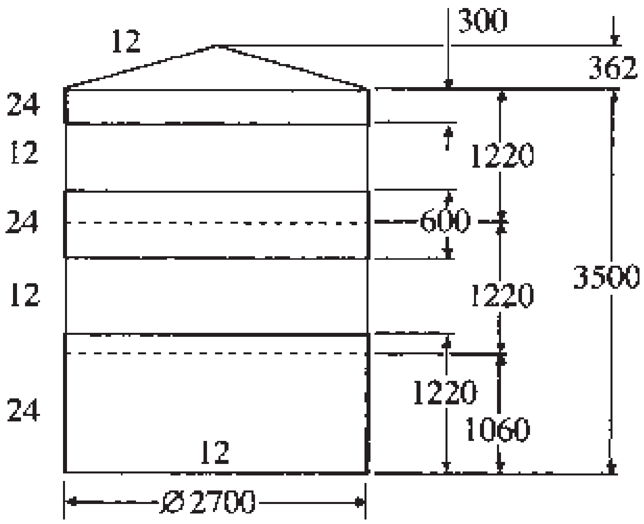

Fig. 1. Sketch showing wall thicknesses and dimensions (in mm) of the failed tank.

> For (1). The calculation can be based either on the creep rupture strength or on a limiting value of creep strain. In most cases there will be multiaxial stress states. Here it is the largest stresses or the largest strains in the principal stress directions that are to be compared with the permissible stress and permissible strain respectively.
>
> The permissible values are obtained by modifying the materials parameters through reduction factors (Section 4), a joint factor (Section 5) and safety factors (Section 6). The factors given in Sections 4 and 5 should only be applied to the stresses. The same applies to the safety factors in Table 4 of Section 6.
>
> For (2) and (3). The determining strength parameter here is the creep modulus. This can be obtained from the creep modulus diagrams, which show its dependence on time, temperature and stress. For criteria based on stability, there is a corresponding safety factor (Section 6) to be taken into account.

The tank failed at a welded joint under the action of a hoop stress (i.e. a stress acting circumferentially). Therefore excessive deformation and stability can be discounted, and the appropriate limiting criteria to explore are those of stress or strain. DVS 2205, Part 1, Section 3.3, provides a way of deciding on which of these criteria the design calculations should be based.

> Where not all the strains are known (for example, strains associated with residual or internal stresses in weld beads . . . or notches), which would necessitate extra safety factors to compensate for this uncertainty, the design calculations should follow the stress-based route (see Section 3.2.1.).

Since the failed tank was of welded construction and, indeed, failed at a welded joint, the above suggests that the limiting stress criterion should be the one adopted, as offering the more conservative approach.

2.1. Calculation of the limiting stress, σ_{zul}

Graphical creep rupture data provided for different thermoplastics materials (Figs 5–10 in DVS 2205, Part 1) allow the corresponding creep rupture stress, K, to be evaluated at the design lifetime and the intended service temperature. A maximum permissible stress, σ_{zul} ('zul' is the abbreviation of 'zulässig', the German for 'permissible') is then calculated by multiplying K by a series of factors which take into account the effects of type of welded joint, any chemical interaction between the container and its contents, the specific strength of the container material, any fluctuating loading and the degree of hazard of the contents.

Details of the calculation of the limiting stress for the failed tank are set out in Appendix 1. From this we get that the maximum permissible stress level, σ_{zul}, for a 25-year life of polypropylene copolymer similar to that used in the failed tank at 20°C is

$$\sigma_{zul} = 2.54\ \text{N mm}^{-2} \tag{1}$$

2.2. Calculation of wall thickness

The required wall thickness, s, of the container at different depths, h, from the surface of the contents in the full container can now be determined from the standard equation for hoop stress, σ_θ, as a function of the static head pressure, p, exerted by the contents at those depths. The basic equation for the wall thickness, s, is derived in Appendix 2 and is

$$s = \frac{dh\rho g}{2\sigma_\theta} \tag{2}$$

where d is the container diameter and g is the acceleration due to gravity. In DVS 2205, Part 2 [5], by putting $\sigma_{zul} = \sigma_\theta$ in eqn (2), three cases are considered. These are:

(i) for containers with constant wall thickness

$$s = C\frac{dh\rho g}{2\sigma_{zul}} \tag{2a}$$

(ii) for containers with graded wall thickness, s_n at depth h_n (e.g. Fig. 2, which approximates to the dam wall type of structure referred to in Part I of this work [4])

$$s_n = C\frac{dh_n\rho g}{2\sigma_{zul}} \tag{2b}$$

where $(h_n - h_{n-1}) \geqslant 500$ mm.

The factor C in (i) and (ii) takes into account the constraining effect of the joint with the base of the container in case (i) and the similar effect of change of wall thickness in case (ii). The value of C varies between $C = 1$ and $C = 1.82$. For a flexible base and/or a gradual change in wall thickness, $C = 1$ can be used. For a rigid base and/or large and abrupt changes in wall thickness, the value of $C = 1.82$ should be applied (this is the equivalent of the corrections to the radial expansion arising from the solution of Timoshenko and Woinowsky-Kreiger [3] discussed earlier).

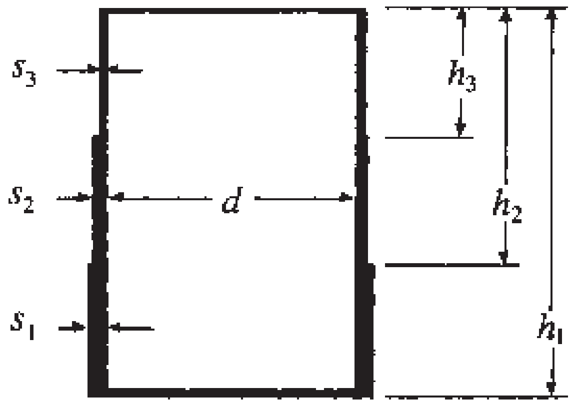

Fig. 2. Tank with graded wall thickness.

(iii) for containers with vertical welds, the increase in wall thickness has already been taken into account in the joint factor, f_v, so that

$$s = \frac{dh\rho g}{2\sigma_{all}} \tag{2c}$$

The largest value of wall thickness obtained from eqns 2(a)–(c) is the definitive value to be used.

3. Application to the failed tank

3.1. Equation for the wall thickness of the failed tank

The failed tank had four abrupt and large changes in wall thickness (a factor of two) between its base and its top (see Fig. 1). Taken together with the constraining effect of the base, this suggests that a value of $C = 1.82$ should be used in eqns 2(a) and (b) to calculate the wall thickness. Then, with

$d = 2700\,\text{mm} = 2.7\,\text{m}$

$\rho = 1540\,\text{kg m}^{-3}$

$g = 9.81\,\text{m s}^{-2}$

$\sigma_{all} = 2.54\,\text{N mm}^{-2} = 2.54 \times 10^6\,\text{N m}^{-2}$

The equation for s becomes

$$s = 1.82 \times \frac{2.7 \times 1540 \times 9.81 \times h}{2 \times 2.54 \times 106}$$

$$= 1.46 \times 10^{-2} h \tag{3}$$

If h is expressed in mm, eqn (3) gives s also in mm. This eqn allows the minimum wall thickness at any vertical position on the container to be calculated.

The failed tank had a design capacity of 20 m^3, so that its fill level, corresponding to the maximum of the hydrostatic head h_{max}, was

$$\begin{aligned} h_{max} &= \frac{\text{volume}}{\text{base area}} \\ &= \frac{20 \times 4}{\pi d^2}\,\text{m} \\ &= \frac{20 \times 4}{\pi \times 2.72}\,\text{m} \\ &= 3.5\,\text{m or } 3500\,\text{mm} \end{aligned}$$

Then, from eqn (3), the minimum wall thickness just above the base should have been 51.1 mm. In fact it was 24 mm—just over a factor of 2 less.

3.2. Comparison between the failed tank and DVS 2205

The line marked DVS 2205 in Fig. 3 shows eqn (3) plotted in terms of height from the base (i.e. $(h_{max} - h)$, rather than hydrostatic head h. Also shown is the outline of the wall thickness variation as shown in Fig. 1 for the failed tank. The shading indicates the regions of the tank wall where the thickness is less than that obtained from the DVS 2205 design code. It is clear that there are serious discrepancies between the thicknesses of the failed tank wall and those derived from the design code.

The extent of the discrepancy between the design of the failed tank and the DVS 2205 require-

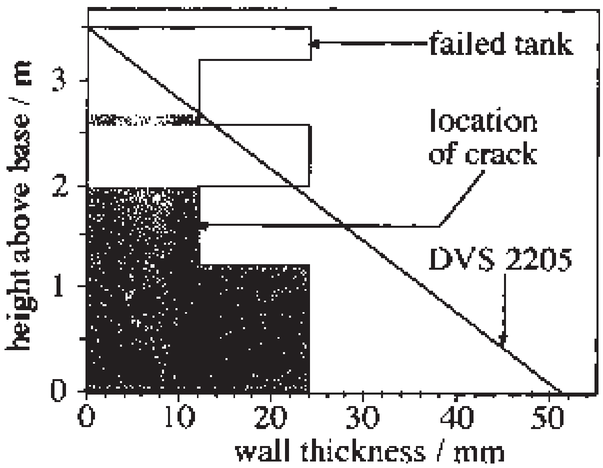

Fig. 3. Variation of wall thickness, s, of failed tank with height, $h_{max} - h$, from its base compared with that specified by DVS 2205 design code (shaded areas are less than the code's thicknesses).

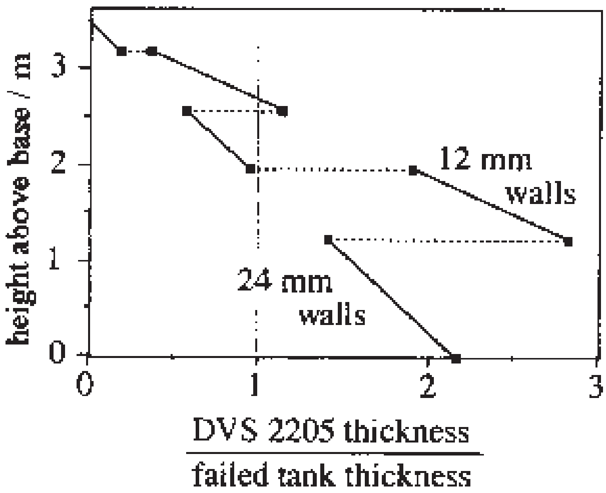

Fig. 4. Ratio of DVS 2205 wall thickness to that used in the failed tank and its variation with height above the base of the tank.

ments is highlighted in Fig. 4. This plots the ratio of the wall thickness from eqn (3) to that of the failed tank as a function of height from the base. The largest discrepancy is found in the lower 12 mm thick section—just the section where the failure originated.

4. DVS 2205 and consultant engineer's calculations

From the consultant engineer's calculations that were made available to us, it is apparent that he worked with a limiting strain criterion—a creep strain of 2% after 25 years. From this he obtained a value of the corresponding stress as 3.95 N mm^{-2} by iteration and interpolation on the appropriate creep modulus vs time curve (Fig. 26 of DVS 2205, Part 1—Appendix 3). For some reason he did not use the recommended procedure of obtaining the value directly from the appropriate isochronous stress-strain curve (Fig. 15 of DVS 2205, Part 1 - Appendix 3), though this would not have affected his result significantly. His value of 3.95 N mm^{-2} was used as σ_{zul} in his design calculations. What was ignored was that a similar safety factor to that used for the stress-based calculation should have been applied to the limiting strain *before* determining the corresponding stress level (see eqn (11) in DVS 2205, Part 1). This is important because, as mentioned earlier, plastics exhibit non-linear stress–strain behaviour, so that stress cannot be assumed to be proportional to strain in a thermoplastic such as the polypropylene copolymer in this case. Had he applied a value of $S = 2.0$, he would have obtained a stress level of about 2.4 N mm^{-2}—a value much closer to the one derived here. Also ignored was the factor C (see eqns 2(a) and (b) above), which takes into account the constraints due to the base joint and the changes in wall thickness. The net result is the discrepancies in thickness shown in Figs 3 and 4, which translate into a maximum hoop stress in the tank walls which is almost a factor of *three times greater* than would have arisen under the recommendations of DVS 2205.

The consultant engineer later claimed that he used the joint factor, f_s, despite its non-appearance in his original calculations, and its non-applicability to a calculation based on a limiting strain. Even then, he took a value of f_s of 0.8, corresponding to that for heated-tool, butt-welding in DVS 2205. The design code is quite clear that where hot-gas, extrusion is used, as it was for the horizontal welds in the failed tank, the lower value of $f_s = 0.6$ should have been used.

5. Extra bending strains

In Part I of this work, it was noted that extra strains were introduced in one of the stages of fabricating the tank. The tank was built up of rectangular, flat, polypropylene panels, the edge of which were butt welded together in a machine to produce a flat strip whose length equalled the circumference of the tank. This strip was then bent round into a circular hoop and its ends welded together to form a section of the tank. The bending was done mechanically with no assistance from elevated temperatures such as would have been used in thermoforming, and with no subsequent annealing. Thus the strains associated with the bending were permanent and contributed to the overall strain in the tank walls.

5.1. *Determining the extra strains and resulting stresses*

The magnitude of these extra strains can be estimated from simple bending theory. The elastic strain ε in a member bent to a radius R is given by

$$\varepsilon = \frac{y}{R} \tag{4}$$

where y is the distance from the central plane of the member's thickness (the neutral axis, Fig. 5). Applying eqn (4) to the failed tank gives

$$\varepsilon = \frac{6}{1350} = 0.44\% \text{ for 12 mm thick material}$$

Since the 24 mm material was fabricated by adding an extra 12 mm thickness band to the tank after the horizontal welds had been formed, the bending strains in the thicker section would have been virtually the same as in the 12 mm material.

In polymeric materials subjected to a constant strain, the associated stress falls with time due to viscoelastic stress relaxation (this is analogous to the creep that occurs under constant stress

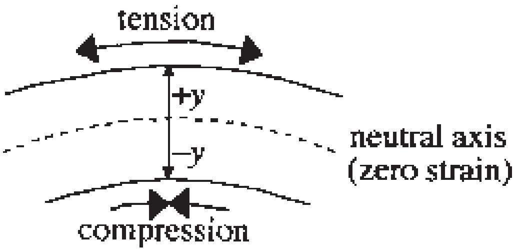

Fig. 5. Schematic of bending.

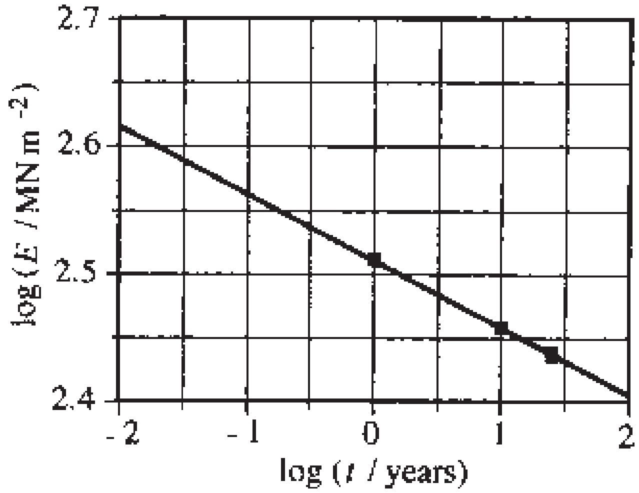

Fig. 6. Creep modulus vs time for polypropylene copolymer (data from Figs 24–26 of DVS 2205 Part 1).

conditions). Strictly speaking, the appropriate values of the stress relaxation modulus should be used to determine the levels of stress at different times. However, in the absence of data on the stress relaxation modulus, the creep modulus can be used as a reasonable approximation. Figure 6 shows the variation of creep modulus with time (logarithmic plot) for polypropylene copolymer using data from Figs 24–26 of DVS 2205 Part 1 (see Appendix 3) at a stress level of 2 MN m^{-2}. From this, between 0.01 years (4 days) and 0.5 years (6 months), corresponding to the life of the failed tank, the creep modulus, E, varies from about 410 MN m^{-2} to about 335 MN m^{-2}. At three months, corresponding to the mean lifetime of the tank, E is about 350 MN m^{-2} (all these values of E are approximate because of the assumed validity of a linear extrapolation on a log–log plot). From the relation that stress = modulus × strain, these values of modulus suggest a maximum tensile bending stress (at the outer surface of the tank) varying between about 1.8 MN m^{-2} at short times and about 1.5 MN m^{-2} at six months ('about' because of the use of creep modulus instead of stress relaxation modulus).

5.2. *Effect of the additional bending stresses*

The additional bending stresses add to the hoop stress resulting from the hydrostatic pressure exerted by the contents of the tank. The addition of the two stress distributions is shown schematically in Fig. 7, with a hoop stress value of 3.2 MN m^{-2}, corresponding to that acting at the level of the crack with a full tank, and a bending stress of 1.6 MN m^{-2} corresponding to three months' stress relaxation (i.e. an approximate mean time between fabrication and failure). It can be seen that the resultant stress distribution ranges from a maximum tensile stress on the outer surface of the tank of 4.8 MN m^{-2} to a minimum, but still tensile, stress of 1.6 MN m^{-2} on the inner surface. In other words, the maximum tensile stress has been increased by about 50% by the presence of the residual bending stresses. However, the mean stress remains at 3.2 MN m^{-2} at the mid-plane position.

Given that it has been established that the hoop stresses in the tank were almost three times

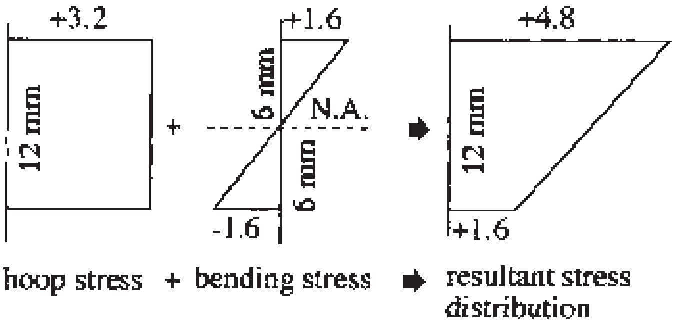

Fig. 7. Schematic addition of stress distributions through the thickness of the tank wall (stresses are shown in MN m^{-2}; tensile stresses are taken to be positive).

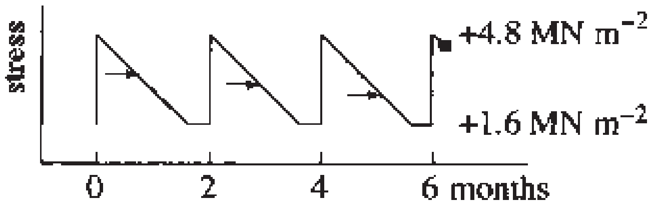

Fig. 8. Schematic of stress variation at outer surface in response to filling cycles of the tank: the arrows indicate schematically the critical stress for slow crack growth.

greater than those derived from DVS 2205 (see Section 4), the effect of an extra 50% stress cannot be anything but serious. Cracks tend to start at surface flaws, which is just where the tensile stresses are highest, and in Part 1 of this work it was found that this was indeed how the crack in the failed tank started (see Figs 3 and 6 in Part I). Two mitigating features, that delayed the failure of the failed tank, were, firstly, that there was a stress gradient through the thickness of wall material, so that any crack starting at the outer surface would propagate into a decreasing stress field, and, secondly, the loading of the tank was periodic. This meant that the maximum surface tensile stress at the site of failure varied between about 4.8 MN m^{-2} when the tank was full, to about 1.6 MN m^{-2} when the liquid level had sunk below the failure height. This is sketched schematically in Fig. 8 for the four loadings of the tank in its six or so months of service. (Bear in mind that this is a simplified schematic –the maximum stress should fall off slightly as the bending stresses relax. In addition, there is no information on the exact form of the loading cycle.) Also indicated (by the arrows) is the schematic behaviour of the critical stress for slow crack growth, showing how this stress would have fallen at each loading cycle, reflecting the crack growth in the preceding cycle.

6. Conclusions

In our opinion, the under-dimensioning of the wall thickness of the failed tank, leading to hoop stresses in the tank walls up to nearly three times higher (about two and a half times higher at the site of the failure) than the maximum values permitted by the DVS 2205 design code was the most significant factor in causing the failure. The fact that one, or perhaps two, of the vertical welds, in particular, were of poorer quality than the others (see Part 1) would have been accommodated by the reduction and safety factors enshrined in the code. Reinforcing the horizontal welds in the tank walls, but not the vertical ones, reflects a basic lack of comprehension of the stresses involved. The largest stresses are those acting horizontally (i.e. circumferentially) and are tensile, whilst the vertical stresses are much smaller and, in the absence of local deformation, are more likely to be compressive. Finally, the stage of manufacture that involved inducing permanent bending strains into the tank walls made what was already a high risk of failure even higher by adding up to 50% to the maximum stress arising from the hydrostatic pressure exerted by the tank's contents. Taken together, these factors made the premature failure of the tank inevitable.

Acknowledgements

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Appendix 1

Calculation of the limiting stress, σ_{zul}

Equation (1) of DVS 2205, Part 1 [5] gives σ_{zul} as

$$\sigma_{zul} = \frac{K_{(A_1,A_3)} f_s}{A_2 A_4 S} \text{ N mm}^{-2} \tag{A1}$$

where

$K_{(A_1,A_3)}$ is the creep rupture stress in N mm^{-2} at the appropriate time and temperature,
A_1–A_4 are the reduction factors,
f_s is the joint factor (if joints have to be taken into account), and
S is the safety factor.

The reduction factors are material-specific, and take the following into account:

A_1 dependence of the strength on the duration of loading
A_2 effect of the surrounding medium (reciprocal resistance factor)
A_3 dependence of the strength on temperature over the load duration
A_4 effect of specific toughness.

The values of A_1 and A_3 are implicit in the accompanying creep rupture curves. The value of the strength parameter $K_{(A_1,A_3)}$ required for the calculation is obtained from the diagrams in Section 10 for a specified service life and service temperature.

The failed tank was made from a polypropylene copolymer, and was to have a design life of 25 years at 20°C when containing caustic soda with a specific gravity of 1.54. Although the vertical seams were machine welded, the horizontal ones were formed by hot gas extrusion welding. From DVS 2205, Part 1 we get the following values for the above factors for the copolymer:

$K = 9.3$ N mm^{-2} (from Fig. 7 for 2.2×10^5 h at 20°; see Appendix 3)
$f_s = 0.6$ (for hot gas extrusion welding)
$A_2 = 1.0$ (i.e. no chemical interaction)
$A_4 = 1.1$ for polypropylene copolymer
$S = 2.0$ (caustic soda *is* hazardous).

Then

$$\sigma_{zul} = \frac{9.3 \times 0.6}{1.0 \times 1.1 \times 2.0} \text{ N mm}^{-2}$$
$$= 2.54 \text{ N mm}^{-2}. \tag{A2}$$

Appendix 2

Equation for wall thickness

The standard expression for the hydrostatic head pressure, p at a depth h in a liquid of density ρ is

$$p = h\rho g \tag{A3}$$

where g is the acceleration due to gravity.

The hoop stress, σ_θ, in a circular vessel of diameter d and wall thickness s due to an internal pressure p is

$$\sigma_\theta = \frac{pd}{2s}, \tag{A4}$$

so that the wall thickness is

$$s = \frac{pd}{2\sigma_\theta}. \tag{A5}$$

Combining eqns (A3) and (A5) gives the equation for the wall thickness, s

$$s = \frac{dh\rho g}{2\sigma_\theta}. \tag{A6}$$

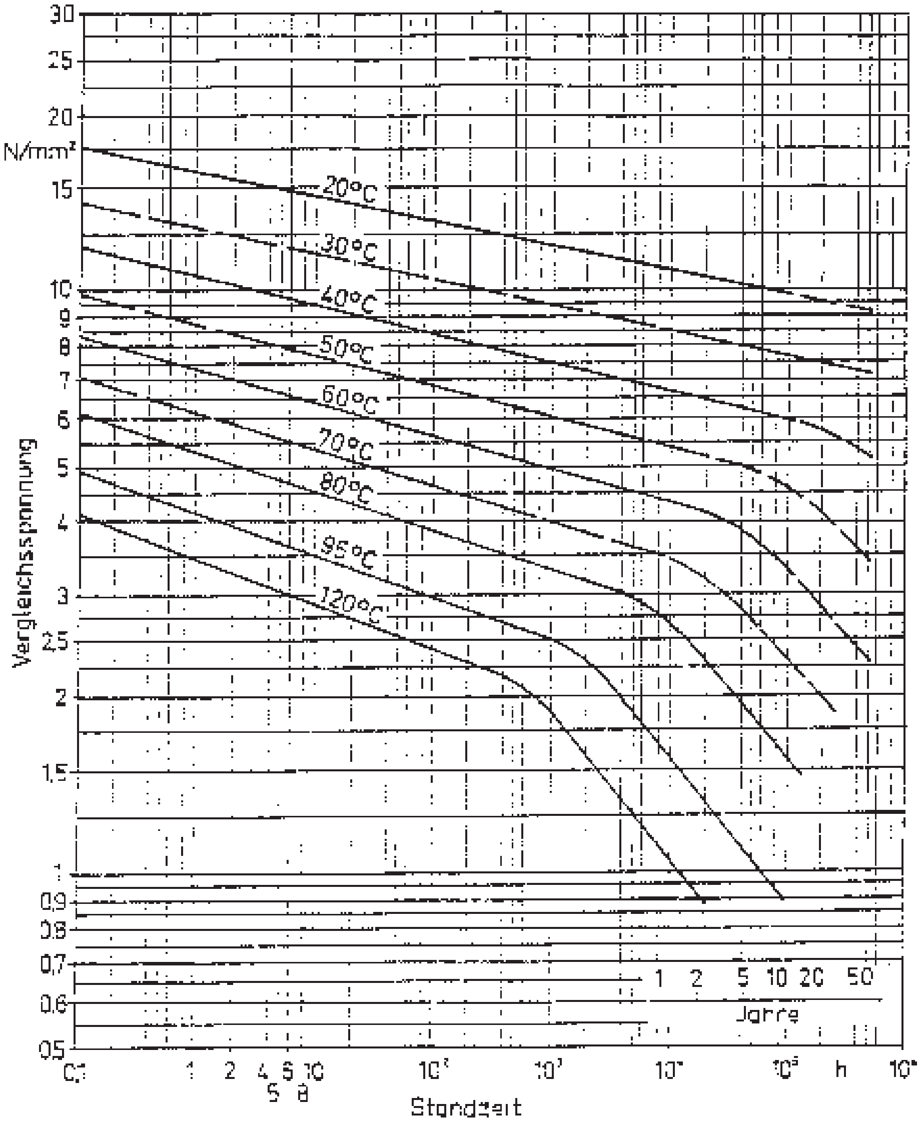

Fig. A1. Creep rupture curves for polypropylene (PP) Type 2 pipes conforming to DIN 8078 (Fig. 7 from DVS 2205 Part 1).

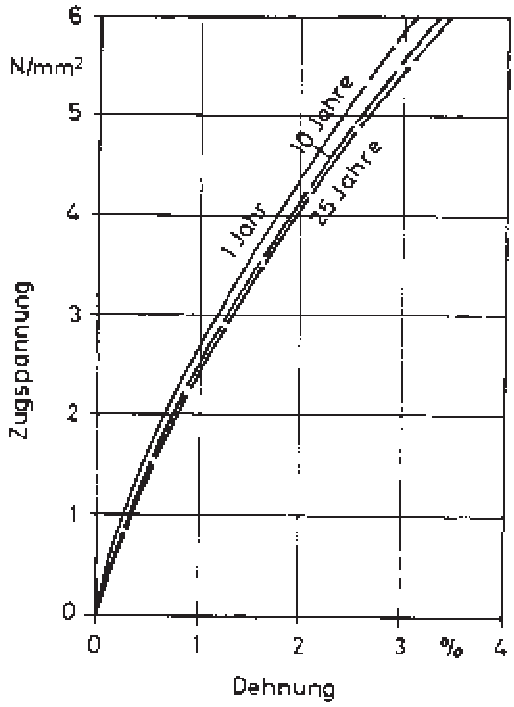

Bild 15. Isochrones Spannung-Dehnung-Diagramm von Polypropylen (PP), Typ 2, für 20°C.

Fig. A2. Isochronous stress–strain curves for polypropylene (PP) Type 2 at 20°C (Fig. 15 from DVS 2205 Part 1).

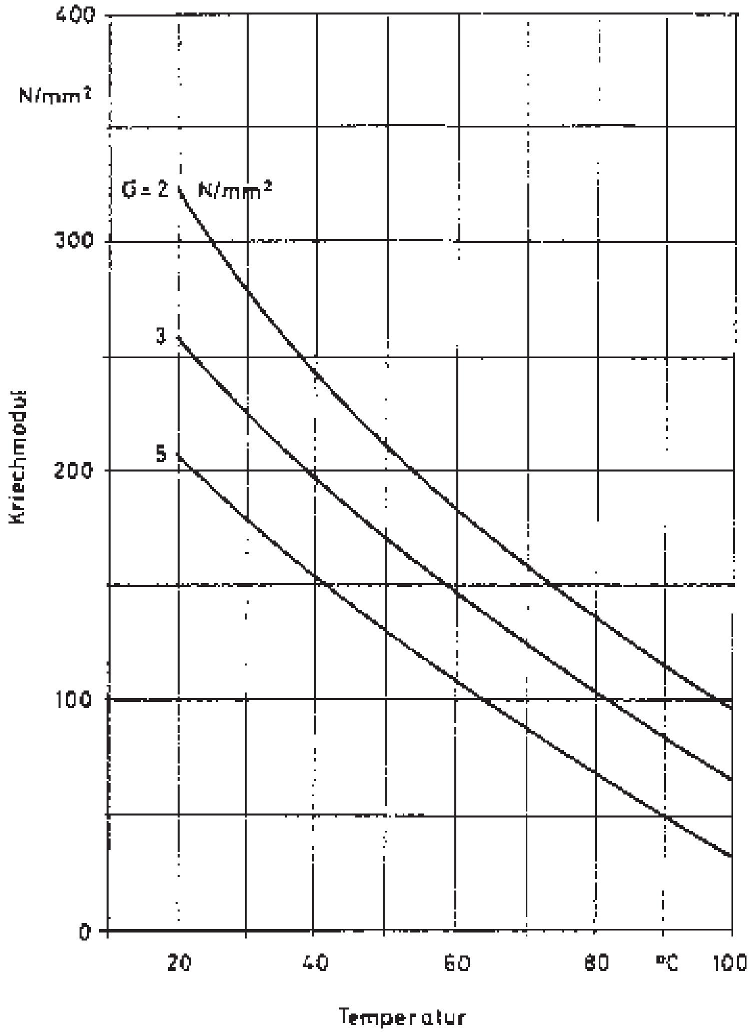

Fig. A3. Creep modulus of polypropylene (PP) Type 2 at 1 year (Fig. 24 from DVS 2205 Part 1).

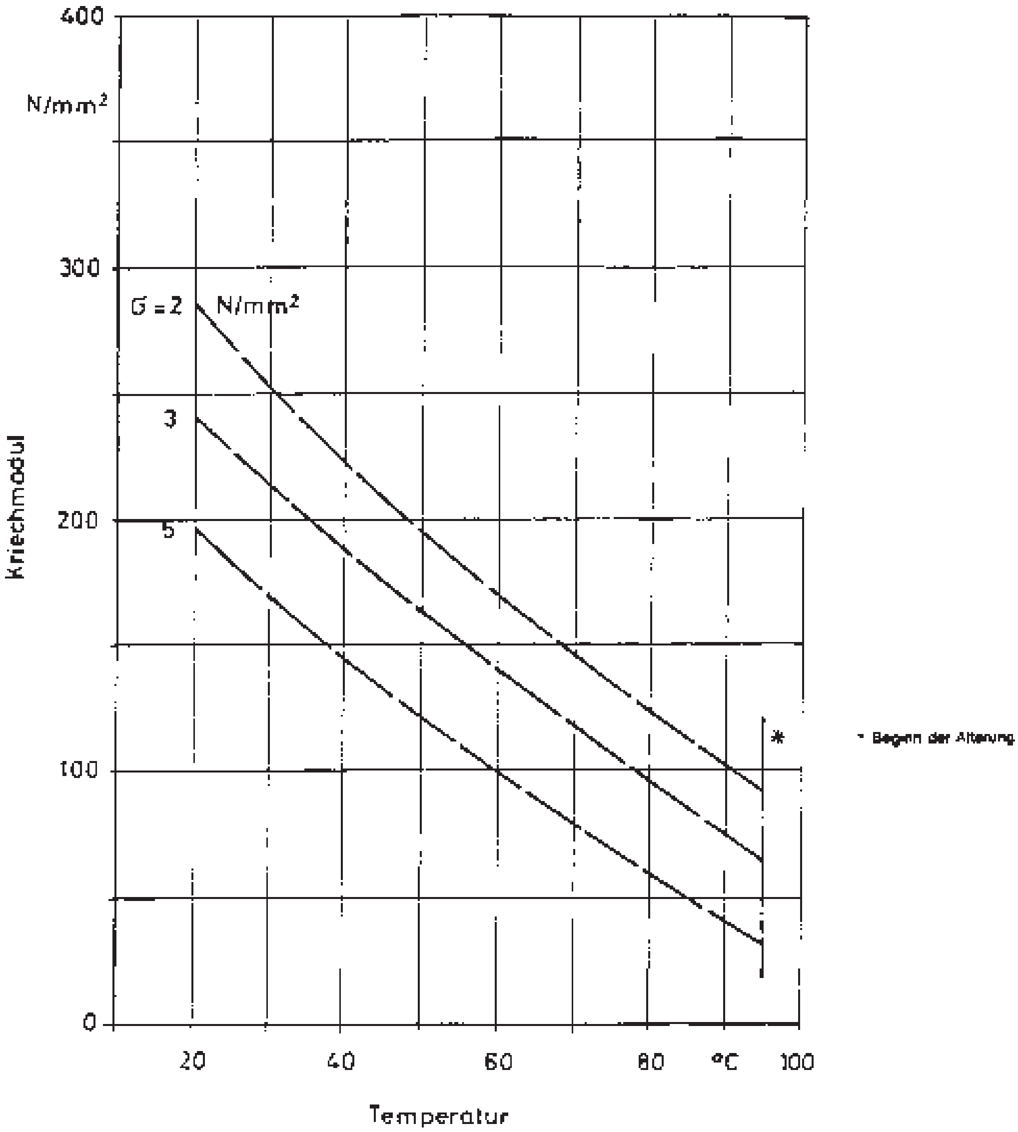

Fig. A4. Creep modulus of polypropylene (PP) Type 2 at 10 years (Fig. 25 from DVS 2205 Part 1).

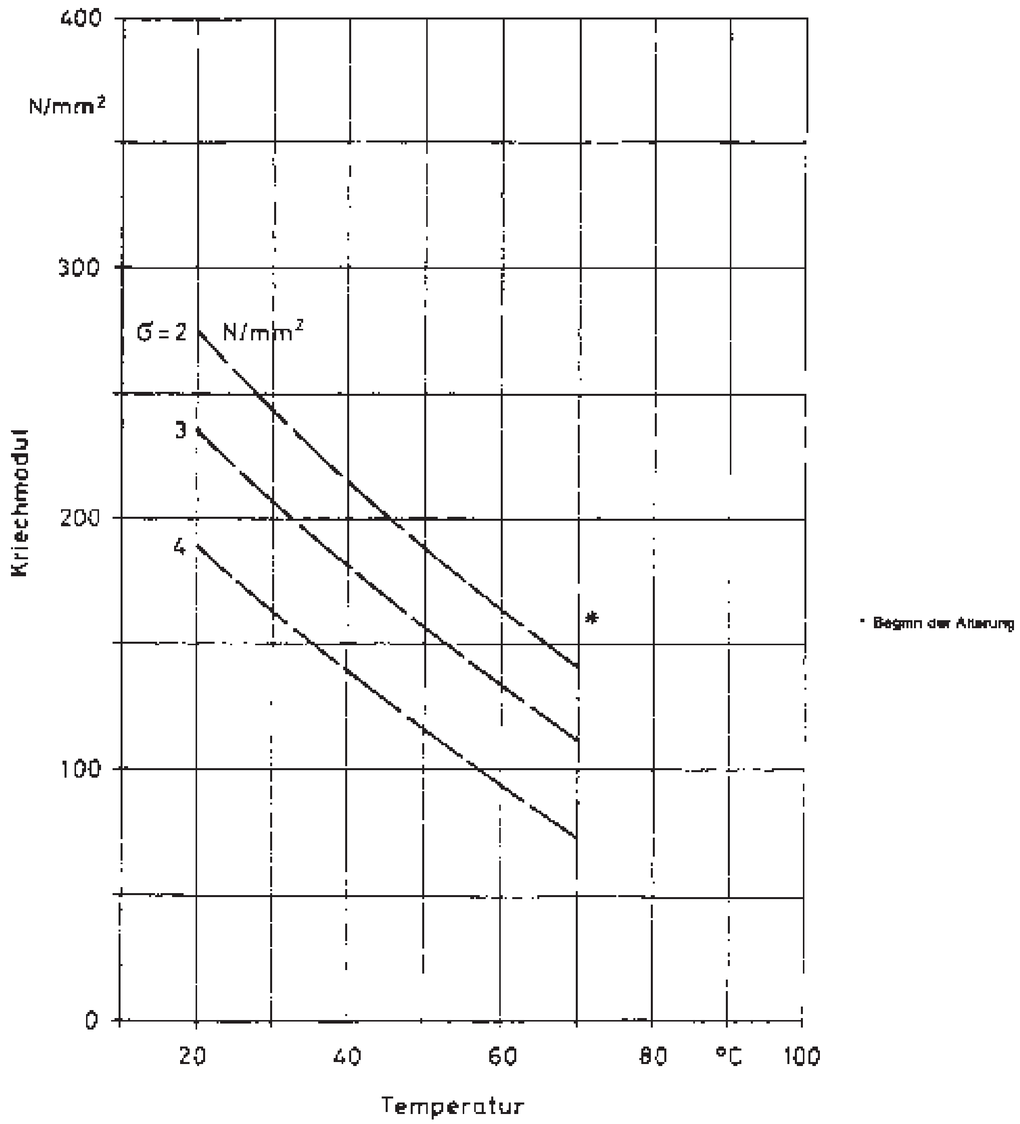

Fig. A5. Creep modulus of polypropylene (PP) Type 2 at 25 years (Fig. 26 from DVS 2205 Part 1).

Appendix 3

Figures 7, 15 and 24–26 from DVS 2205 Part 1

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