Transactions of the **13th International Conference on Structural Mechanics in Reactor Technology** (SMiRT 13), Escola de Engenharia - Universidade Federal do Rio Grande do Sul, Porto Alegre, Brazil, August 13-18, 1995

Cylindrical tubes under dynamic axial loading causing concertina type of deformations

Messemer, G., Göller, B., Hoffmann, G., Stratmanns, E., Müller, St.
Forschungszentrum Karlsruhe GmbH, Institut für Reaktorsicherheit, Karlsruhe, Germany

ABSTRACT: In case of severe reactor accidents a slug of molten fuel might be hurled against the head of the reactor pressure vessel. This impact problem is studied by small scale experiments at FZK (former KfK) in which a crucible containing liquid metal is shot against a model vessel head with velocities up to 150 m/s. After the crucible has reached its velocity it must be decelerated, while the movement of the liquid metal slug continues. During the deceleration the kinetic energy of the crucible of up to 300 kJ must be absorbed in a ring space having an inner diameter of 350 mm, an outer diameter of 500 mm and a length of 470 mm. Investigations have shown that cylindrical shells under axial loading causing concertina type of deformations may be used as energy absorbers. Static and dynamic experiments on this subject will be presented in this paper, including measured axial force versus axial compression. A comparison with results using corresponding formulae from the literature is included.

1. INTRODUCTION

It is assumed that during an in-vessel steam explosion a slug of molten core material is accelerated to high velocities. On its way upwards the slug will destroy the internal structures in the upper plenum of the vessel and will finally impact on the upper head [1 - 4]. In order to avoid the generation of missiles, which might endanger the containment integrity, it is required that the vessel head does not fail under this impact.

Based on data published in [1 - 4] the kinetic energy of molten fuel slugs is in the order of 1000 MJ, i.e., for a slug mass of 80 000 kg the velocity will be up to 150 m/s. These numbers should only allow for a rough orientation.

As the involved phenomena can hardly be described theoretically with a sufficient confidence, it is necessary to perform appropriate experiments. Therefore, at FZK the experimental facility BERDA is under construction, in which the impact of a liquid metal slug on the pressure vessel head including the internal structures can be investigated. For ease of feasibility, the experiments are performed in a reduced scale of 1:10, which means, that the slug mass is 80 kg with a velocity of up to 150 m/s.

It is necessary to model only the upper part of the reactor pressure vessel including the head with its bolts and the upper internal structures. In the experiments the molten fuel slug is simulated by a liquid metal with similar density at a temperature of 60 °C.

The liquid metal is contained in a crucible, which is accelerated upwards using a pneumatic drive mechanism. Pressures of up to 200 bar are applied in order to reach velocities of 150 m/s. As in BERDA only the impact of the liquid slug is of interest, the crucible must be stopped before it reaches the pressure vessel model. Investigations have shown that cylindrical shells under axial loading causing concertina type of deformations may be used as energy absorbers.

2. TEST FACILITY "VERDY" FOR DYNAMIC INVESTIGATION OF ENERGY DISSIPATING STRUCTURES

In order to develop appropriate mechanisms by which the high kinetic energy of the crucible can be surely dissipated without damaging the crucible itself, the experimental facility VERDY was built. Fig. 1 shows the design.

An aluminium piston with a mass of about 12 kg is loaded by pressurized gas up to 110 bar. After initiating rupture of the tension bar the piston is accelerated downwards along the guiding tube to velocities up to 150 m/s. At the lower end of the tube the piston velocity can be measured by a laser light barrier. Finally, the piston impacts against the specimen under investigation. The specimen is fixed at its lower end to a support structure, which contains a sensor for measurement of the dynamic impact forces. Specimens having a diameter of about 150 mm and a length up to 320 mm can be investigated in VERDY.

In order to reduce the dynamic loading of the surroundings, the whole setup is mounted on a concrete mass with a weight of 30 tons, which in turn is supported by several pneumatic springs.

The desired impact velocity is achieved by selecting an appropriate gas pressure. Assuming adiabatic expansion of the gas allows to calculate the final piston velocity with a good precision. For further post-test analyses the pressures in front and behind the piston and the accelerations of the set-up and of the basement are measured. All measured signals are digitized with a frequency of 500 kHz.

3. CRUSHING OF SMOOTH ALUMINIUM TUBES

It is known from literature [7 - 10] that metal tubes under axial plastic collapse are appropriate structures for dissipation of mechanical energy.

A theoretical description of an axisymmetric crushing process is given in [10], which is based on moving plastic hinges. The half length λ of a folding wave is determined by an energy minimization procedure, which yields that the meridional bending energy and the circumferential stretching energy commit the same contribution. The following formulas are from this publication. The half length of one folding wave is:

$$\lambda = 1.88 \cdot \sqrt{R_o h_o} \, , \tag{1}$$

the mean crushing force is:

$$F_m = 12.45 \cdot \sigma_o \cdot h_o \cdot \sqrt{R_o h_o} \tag{2}$$

with initial radius R_o, initial wall thickness h_o and average flow stress σ_o.

For comparison the length of one folding half wave of a linear elastic infinitely long cylindrical shell is given according to classical shell theory [11] by:

$$\lambda_{el} = 1.73 \cdot \sqrt{R_o h_o} \, . \tag{3}$$

The design of a crushing tube for use in the BERDA facility based on the above formulas seemed to be rather questionable. Therefore, static and dynamic experiments were performed in VERDY. For the application in BERDA, two additional conditions have to be respected

- the motion of the slug through the crushing tube must not be hindered,
- the aluminium crucible should not be damaged by the tube.

In a first test series tubes of pure aluminium Al99.5F10 were used with minimum values for the ultimate stress and the yield stress of 115 MPa and 70 MPa, respectively, and an ultimate strain of about 6 % [12]. An average flow stress of $\sigma_o = 100$ MPa was assumed for estimates according to equation (1) and (2).

As tubes in the required dimensions were not available, they were rolled from plane sheets and then welded. The inner diameter of the tubes was 150 mm, the length was 285 - 320 mm and the wall thicknesses varied from 3 to 8 mm.

3.1 Static crushing of the tubes

The tubes were crushed axially on a machine used for standard uni-axial material tests. The prescribed velocity for compression was 2 mm/s, corresponding to an average strain rate of $7 \cdot 10^{-3}$/s. The dimensions of the tubes and characteristic results are given in Table 1, deformation modes and load deflection curves are given in Figs. 2 to 5. The theoretical values for the forces and folding lengths were computed from the formulas (1) and (2) with flow stress of $\sigma_o = 100$ MPa. The experimental half length of a folding wave was derived from the distance Δs of two neighboring force peaks in the force-displacement-diagram according to $\lambda = \Delta s/2 + h_o$.

The tubes No. 1 and 2 were welded by an electron beam. They were placed between two plane plates and crushed. Both tubes deformed in pure concertina modes, and during the experiment, a new folding wave occured only after completion of the old one.

In order to investigate a simpler welding method, the tubes No. 3 to 6 were arc welded by hand. The deformed tubes are shown in Fig. 2. The tubes 3, 4 and 6 deformed nearly regularly in a concertina mode, while tube No. 5 deformed in a diamond mode, which led to a reduced crushing force. In the neighborhood of the welding the symmetric deformation mode is disturbed. Near both ends of the tubes considerable deviation from the initial circular shape can be seen. This disturbance at the boundaries finally caused the diamond deformation mode of tube No. 5.

Tubes No. 7 to 9 were again welded by an electron beam. Additionally, the ends of these tubes were fixed by fitting them into circular grooves with depths of 5 mm (see sketch at Table 2 but without prebuckling). In order to investigate also an almost perfect tube, the tube No. 9 was machined down to 5 mm from a tube with 8 mm thickness. The experimental results are shown in Figs. 3a to 5a. Tube No. 7 deformed irregularly, starting at midheight of the tube. The two remaining tubes deformed regularly. The "perfect" tube No. 9 gave the most regular deformation mode. However, its mean crushing force was reduced considerably due to machining off the outer layers, thus changing the material properties.

3.2 Dynamic crushing of the tubes

The dynamic experiments were performed in the experimental facility VERDY described in chapter 2. The impact velocity was in the order of 80 m/s, the impact mass was 11.8 kg. The investigated tubes were rolled from plane sheets and welded by an electron beam, their ends were fixed in circular grooves. In order to promote deformation in the regular concertina mode, an axisymmetric imperfection with a shape of a local buckling wave, having an outward depth of 0.5 mm, was applied near the upper end of the tubes (see sketch at Table 2). This measure was also intended to reduce the initial rather high load peaks.

The data and results of four dynamic experiments are listed in Table 2. Deformed shapes and force histories are given in Figs. 3b to 5b. The tubes No. 10 to 12 correspond exactly to those with No. 7 to 9 from the static tests (Table 1).

All the tubes deformed in the regular concertina mode. The most regular deformation pattern occured at tube No. 12, which initially had a nearly perfect shape due to manufacturing it by machining from a thicker tube. In the lower part of tube No. 13 along half the circumference an irregular deformation pattern occured as shown in Fig. 3. Tubes No. 12 and 13 were completely compressed.

In Table 2 the mean crushing force was determined from the measured time-history. The half lengths λ of the folding waves were calculated from the initial tube length L_o and the number of waves n observed after the experiment: $\lambda = L_o/2n$.

The average axial compressions $\varepsilon = \Delta L/L_o$ are rather high with values of 70 % to 80 %. With total deformation time T read from the force time-histories the average strain rate was calculated according to: $\dot{\varepsilon} = \varepsilon/T$. The values are between 100/s and 223/s. Table 2 gives also the ratio of the measured mean dynamic force to the measured mean static force. This ratio increases considerably with increasing strain rate (except for tube No. 13). The highest increase of 24 % agrees well with values derived from [13] for the actual conditions.

In all experiments the crushing of the tubes changed the inner diameter D_{in} and the outer D_{out}. The difference between both values corresponded to approximately one length of a folding wave ($D_{out} - D_{in} \approx 2\lambda$). The deformation, however, was not symmetric: $D_{in} \approx 2R_o - 2\lambda/3$, $D_{out} \approx 2R_o + 4\lambda/3$.

4. CRUSHING OF TUBES WITH STIFFENING RINGS

The presented experiments showed that the reduction of the inner diameter was too large and the movement of the slug through the crushing tube would be disturbed in BERDA. A corresponding increase in tube diameter was not possible.

In order to prevent the reduction of the inner diameter, tubes with stiffening rings were designed. To ensure that higher buckling modes between two neighboring rings do not appear, their distance was selected to be equal to the half length λ of one folding wave from eq. (1). This value is only slightly larger than that for linear buckling of long tubes, see eq. (3).

For every axial segment of the tube a different wall thickness can be selected, thus obtaining a progressing crushing force with increasing deformation. A crushing tube was designed with five segments with wall thicknesses increasing from 3 to 7 mm (Fig. 6 and Table 3a). The flange at the top contained a ring of wood to test a special adaptation problem in BERDA.

Two tubes were manufactured from solid bars of aluminium AlMg3F23 with ultimate stress and yield stress of 230 and 170 MPa respectively, and with ultimate strain of 9 % [12]. Here, an average flow stress of $\sigma_o = 230$ MPa was used for eq. (2).

4.1 Static crushing

Fig. 7a shows the tube before and after crushing. Additionally, the measured force versus deformation diagram is given. For each segment of the tube the increase in average crushing force can be seen. At 640 kN the capacity of the used machine was reached, thus terminating the test. The first three segments with wall thicknesses of 3,4 and 5 mm are completely compressed.

Table 3a lists the measured mean crushing force for each segment of the tube. The theoretical values from equation (2) are also given. However, the theoretical model for equation (2) is based upon smooth, long shells, and thus is slightly out of bounds here.

4.2 Dynamic crushing

The facility VERDY was used to investigate the second tube dynamically with an impacting mass of 11.4 kg and an impact velocity of 115 m/s.

Table 3b lists the significant data for this experiment, Fig. 7b shows the deformed tube and the measured force-time history. As expected, the mean force increases gradually with time. Again, a correlation with the crushing of individual axial segments of the tube is not possible. The sharp increase of the force near the end indicates that maximal compression of the tube was reached.

All axial segments show regular deformations. The axial deformation was $\Delta L = 132$ mm, corresponding to an average compression of 73 %, taking into account only the deformable sections with a total length of 182 mm. With the deformation time of $T = 1{,}6$ ms an average strain rate of $\dot{\varepsilon} = 456/s$ turned out.

The comparison of static and dynamic forces is only possible up to a deformation of the tube of about 80 mm due to the early termination of the static test. The corresponding point is labeled in the dynamic force diagram. It was gained by integration of the deceleration of the impact mass, which can be calculated from the force. The mean static crush force is about 325 kN, the mean dynamic force in this segment is 471 kN, which leads to a ratio of 1.45.

The force-time history and the calculated velocity were used to determine the power and finally the dissipated energy for which a value of $E_d = 69$ kJ was obtained. The kinetic energy of the impact mass was calculated as $E_{kin} = 75$ kJ.

5. COMPARISON OF THEORETICAL VALUES AND EXPERIMENTAL RESULTS

For a better comparison of the numerous theoretical and measured values given in the tables, two comprehensive diagrams are presented.

Fig. 8 shows the comparison of the measured folding lengths with the theoretical values calculated from eq. (1) and eq. (2). For an increasing ratio of $2R_o/h_o$, i.e. for thinner shells, the deviations increase. This tendency was also discussed in [10].

The deviations of the folding lengths and of the observed forces, which are normalised to the ultimate force F_u, are rather large. Presumably, deviating material properties give the highest contribution. Unfortunately, plastic material behavior is never guaranteed, so that for any new series of crushing tubes the real stress-strain behavior should be measured. Here, however, tabulated values from [12] were used.

The deviations for the ring stiffend tubes are rather large, indicating that the theoretical model of [10] should not be applied in this case.

6. CONCLUSIONS

In order to facilitate the design of a shock absorbing element for the dissipation of high kinetic energy, the axial crushing of tubes was investigated.

Experiments were performed with smooth aluminium tubes under static and dynamic loading. In the dynamic experiments impact velocities of 80 m/s were chosen, leading to average strain rates up to 200/s. This caused an increase of the dynamic crushing force by 24 %. Deformation in the optimal concertina mode occured, if the ends of the tubes were fixed in circular grooves and had an axisymmetric pre-buckle near one tube end.

In order to avoid the reduction of the inner diameter, ring stiffened tubes were investigated statically and dynamically. Now a reduction of the inner diameter of 3 % was obtained which is acceptable. The distance of the stiffeners was chosen to be one half length of the theoretical folding length of a smooth tube. Different wall thicknesses in the individual segments of the tube allow for an optimal adjustment of the force-displacement characteristics. In the dynamic experiment an average strain rate of 456/s was reached, leading to increased mean crushing force by 45 %.

The obtained results allow for an appropriate design of the crushing tube for the BERDA facility.

7. REFERENCES

[1] R. Krieg et al: Extreme Loadings of Inner Structures of Next Generation PWR-Containments, Fifth Workshop on Containment Integrity, Washington, DC, May 1992

[2] R. Krieg, G. Hailfinger, T. Malmberg, G. Messemer: Steam Explosion Driven Impact of Molten Core Material on the Upper Pressure Vessel Structures of a PWR, 12th Int. Conf. on Structural Mechanics in Reactor Technology, Div. U, Stuttgart, Aug. 1993
[3] R. Krieg, T. Malmberg, G. Messemer, T. Stach, E. Stratmanns: Slug impact Loading on the Vesel Head during a Postulaed In-vessel Steam Explosion in Pressurized Water Reactors, Nuclear Technology, to be published
[4] R. Krieg, G. Messemer, T. Stach, E. Stratmanns: Experiments on Slug Impact Loading of the Reactor Vessel Head during a Postulated Steam Explosion, Kerntechnik, Vol. 59, No. 4-5, 1994
[5] G.E. Lucas, W.H. Amarasooriya, T.G. Theofanous: An Assessment of Steam Explosion Induced Containment Failure, Part. IV: Impact Mechanics, Dissipation and Vessel Head Failure, Nucl. Sc. Eng. 97, pp 316-326, 1987
[6] T. Malmberg: Abschätzungen von Kollapslasten und plastischen Deformationsenergien der oberen RDB-Einbauten, Kernforschungszentrum Karlsruhe, Internal Report 1993
[7] J.M. Alexander: An Approximate Analysis of the Collapse of Thin Cylindrical Shells under Axial Loading, Quart. J. Mech. and Appl. Math. 13,1, pp 10-15, 1960
[8] W. Abramowicz: The Effective Crushing Distance in Axially Compressed Thin-walled Metal Columns, Int. J. Impact Engng. 1, pp. 309-317, 1983
[9] W. Abramowicz and N. Jones: Dynamic Axial Crushing of Circular Tubes, Int. J. Impact Engng. 2, pp. 263-281, 1984
[10] T. Wierzbicki, S.U. Bhat: A Moving Hinge Solution for Axisymmetric Crushing of Tubes, Int. J. Mech. Sci. Vol. 28, No. 3, pp. 135-151, 1986
[11] K. Girkmann: Flächentragwerke, Wien, 1963
[12] E. Doege et al.: Fließkurvenatlas metallischer Werkstoffe, München, 1986
[13] T. Nicholas: Material Behavior at High Strain Rates, In: J.A. Zukas et al., editors, Impact Dynamics, New York, pp. 277-332, 1982

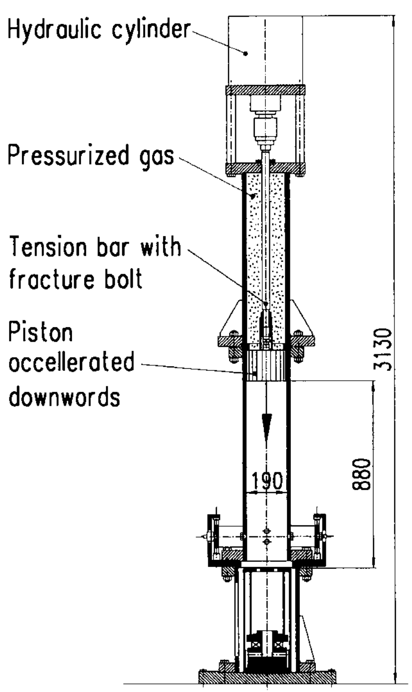
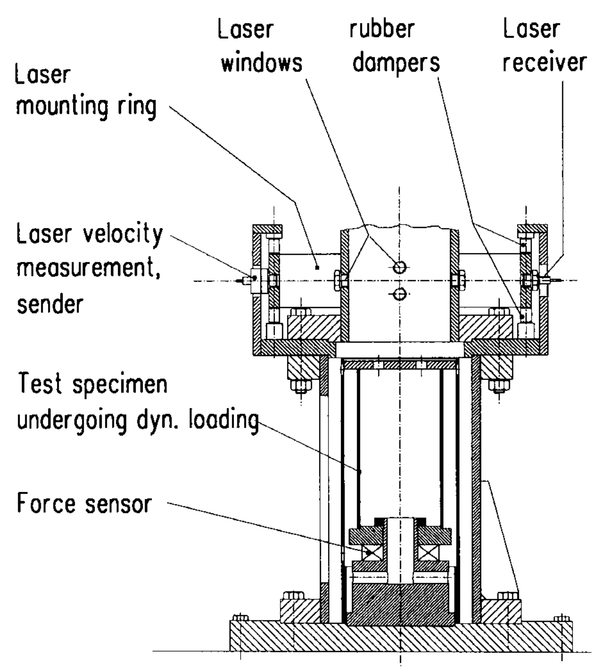


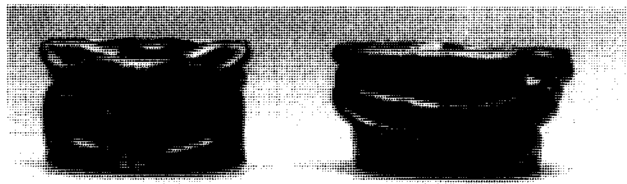
Fig. 1: Test device VERDY



Lower Part of VERDY

Tube No. 3
ho = 3mm

Tube No. 4
ho = 4mm

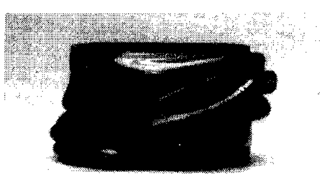


Undeformed tube

Tube No. 5
ho = 5mm

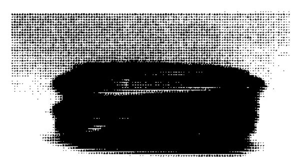
Tube No. 6
ho = 6mm

Fig. 2: Crush tubes of aluminium Al 99.5 under static loading (all tubes rolled from plane sheets and arc-welded manually)

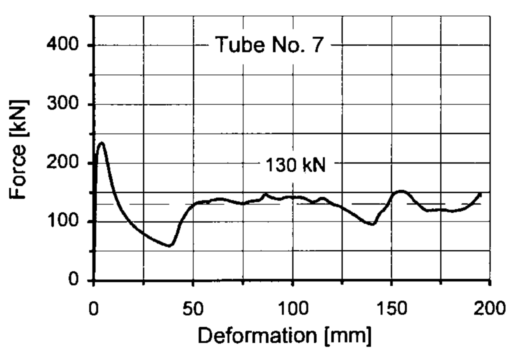


Tube No. 7

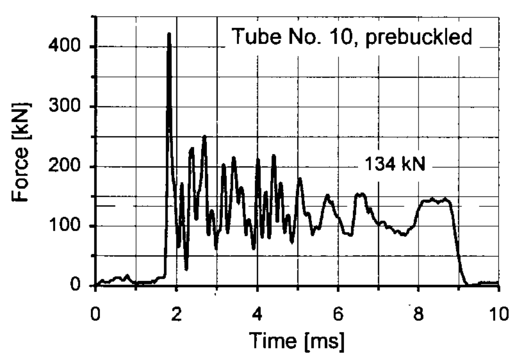
Tube No. 10



Tube No. 13

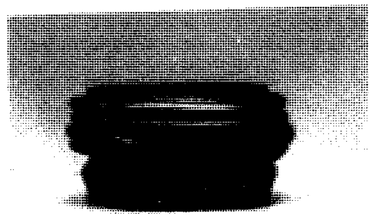


a) Static loading

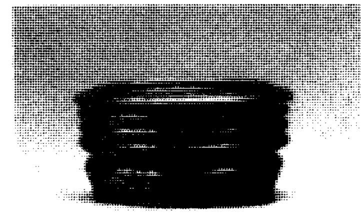


b) Dynamic loading

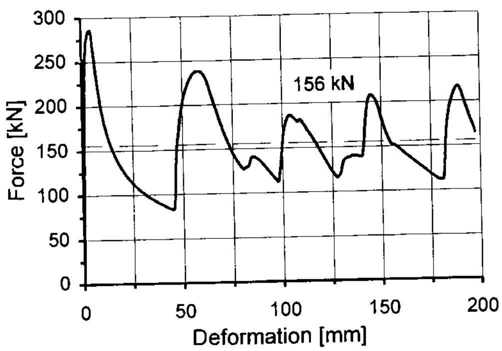
Fig. 3: Rolled tubes of aluminium Al 99.5, wallthicknes 4mm, EB-welded, the ends guided in circular grooves



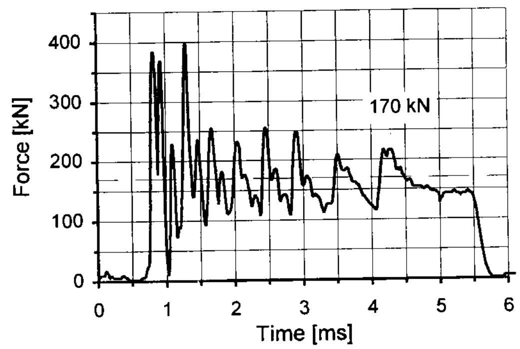
Tube No. 8



Tube No. 11

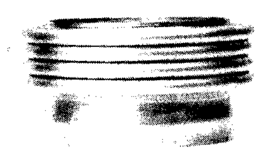


a) Static loading

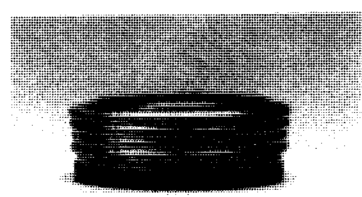


b) Dynamic loading with prebuckling

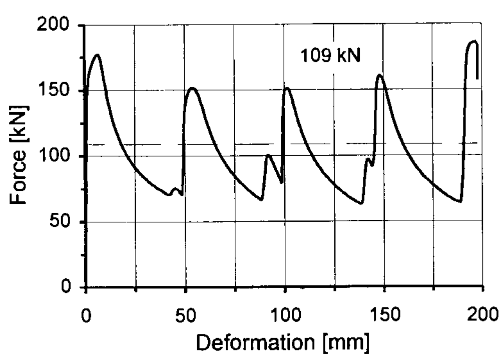
Fig. 4: Rolled tubes of aluminium Al 99.5, wallthickness 5mm, EB-welded, the ends guided in circular grooves



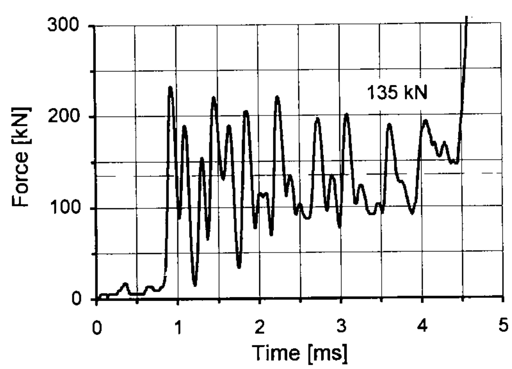
Tube No. 9



Tube No. 12



a) Static loading



b) Dynamic loading with prebuckling

Fig. 5: Rolled tubes of aluminium Al 99.5, wallthickness machined from 8mm to 5mm, EB-welded, the ends guided in circular grooves

Tab. 1: Static loaded crush tubes of pure aluminium Al 99.5 F10

tube No.	inner diam-eter [mm]	thick-ness h_o [mm]	free tube length L_o [mm]	2Ro/h	welding method	crushing force theoret. Fm [kN]	crushing force measured Fm [kN]	folding length theoret. λ [mm]	folding length measured λ [mm]	folding mode	Fig.No.
1	150	5	320	31	EB	123	139	37	28	concert.	-
2	150	8	320	19.75	EB	250	240	48	44	concert.	-
3	150	3	300	51	manually	57	59	29	23	concert.	2
4	150	4	300	38.5	manually	87	92	33	26	concert.	2
5	150	5	300	31	manually	123	95	37	28	diam.	2
6	150	6	300	26	manually	162	135	41	40	concert.	2
7	150	4	285	38.5	EB	87	130	33	-	irreg.	3a
8	150	5	285	31	EB	123	156	37	28	concert.	4a
9*	150	5	285	31	EB	123	109	37	29	concert.	5a

* rolled from a sheet of 8 mm thickness and machined down to 5 mm thickness

EB = Electron Beam; concert. = concertina; diam. = diamond

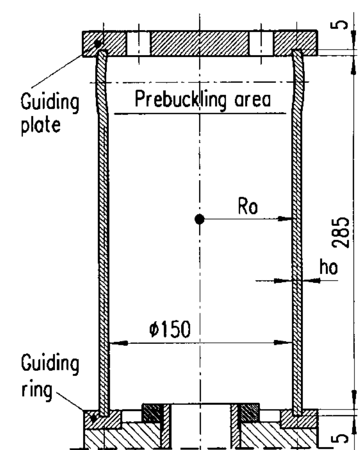
Tab. 2: Dynamic loaded crush tubes of pure aluminium Al 99.5 F10

tube No.	inner diam-eter [mm]	thick-ness h_o [mm]	free tube length L_o [mm]	welding method	dyn. mass [kg]	VERDY pressure [bar]	velocity measured [m/s]	kinetic energy [kJ]	axial defor-mation ΔL [mm]	defor-mation time T [ms]	strain rate $\dot{\varepsilon}$ [1/s]	folding length measured λ [mm]	folding mode
10	150	4	285	EB	11.8	25	75	31	215.5	7.2	105	24	concert.
11	150	5	285	EB	11.8	30	78	34	195	4.3	159	29	concert.
12*	150	5	285	EB	11.8	39	86	44	235	3.7	223	29	concert.
13	150	4	285	EB	11.6	30	83	35	230.3	4.8	168	24	concert.

tube No.	inner diameter D_{in} [mm]	outer diameter D_{out} [mm]	reduced inner diameter [%]	axial compress. $\Delta L/L_o$ [%]	totally com-press.	crush force measured Fm [kN]	crush force ratio dyn/stat	crush energy [kJ]	Fig. No.
10	134	183	89	76	no	134	1.05	29	3b
11	136	196	91	68	no	170	1.11	33	4b
12*	132	196	88	82	yes	135	1.24	32	5b
13	irreg.	irreg.	-	81	yes	130	1.00	30	3c

* rolled from a sheet of 8 mm thickness and machined down to 5 mm thickness

EB = Electron Beam; concert. = concertina; irreg. = irregular



Clamping at the ends of the tube

Tab. 3: Ring-stiffened tube of aluminium alloy AlMg3F23

a) Results of static test (Fig. 7a)

segment No.	inner diameter [mm]	thickness h_o [mm]	segment length [mm]	crush force Fm theoret. [kN]	crush force Fm measured [kN]	crush force Fm mean value [kN]
1	150	3	28	130	159	325
2		4	32	201	299	
3		5	37	282	450	
4		6	40	372	-	
5		7	45	470	-	

free tube length L_o [mm]	axial deformation ΔL [mm]	axial compress. [%]	reduced inner diameter [%]
182	85	47	96

b) Results of dynamic test (Fig. 7b)

VERDY pressure [bar]	dynamic mass [kg]	velocity [m/s]	kinetic energy E [kJ]	mean crush force Fm measured [kN]	crush force ratio dyn/stat	defor-mation time T [ms]
70	11.4	115	75	600	1.45	1.6

strain rate $\dot{\varepsilon}$ [1/s]	axial deformation ΔL [mm]	axial compress. [%]	dissip. energy E_d [kJ]	reduced inner diameter [%]
453	132	73	69	97

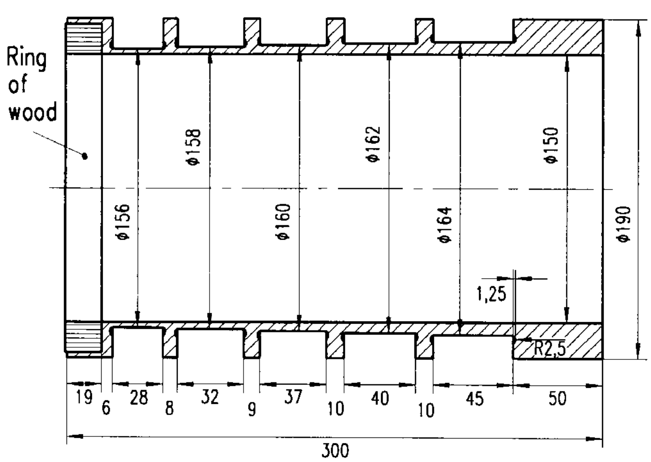


Fig. 6:
Ring-stiffened tube
of aluminium AlMg3 F23

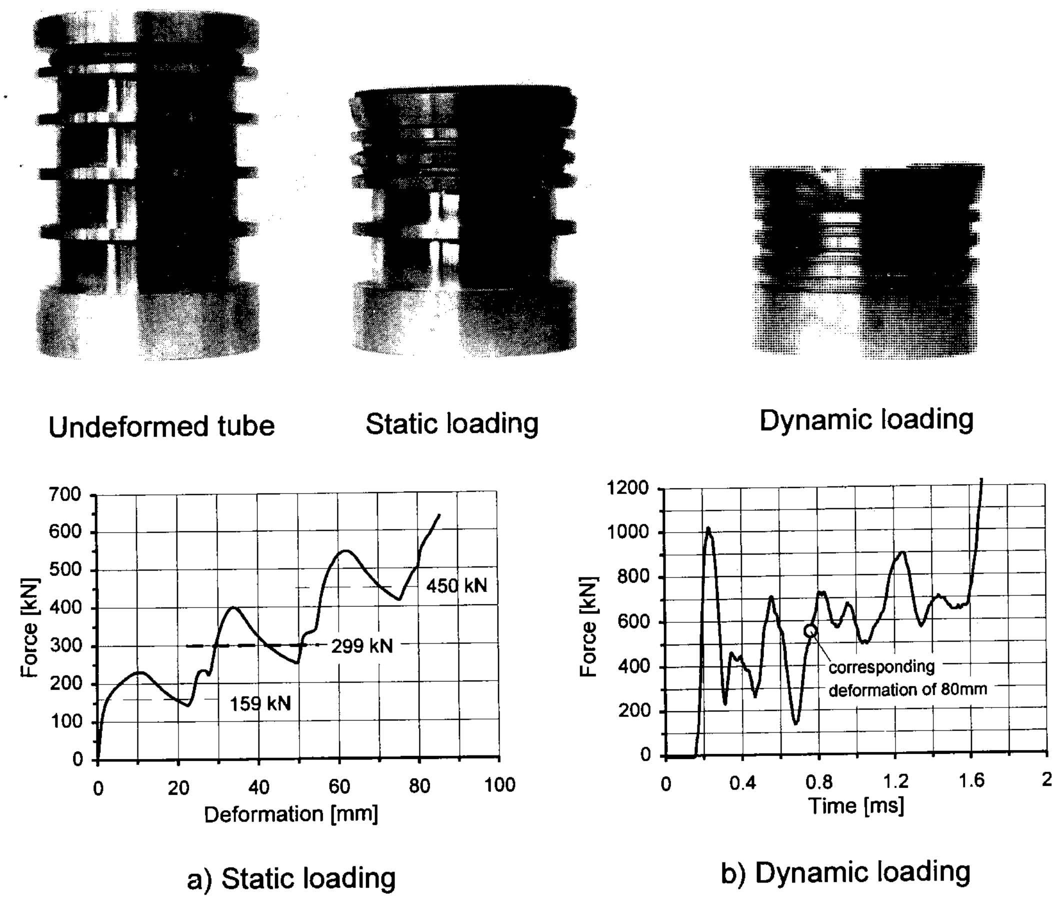


Fig. 7: Ring-stiffened tubes of aluminium AlMg3 with segments of different wall thicknesses

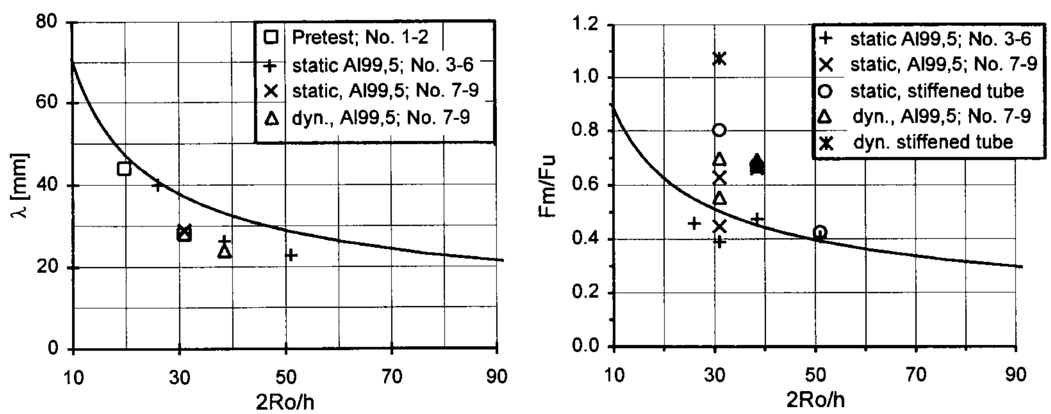


Fig. 8: Comparison of static and dynamic results for the folding length λ and the crushing force Fm with theorie from [10] (the crushing force is normalised to the ultimate force $F_u = \sigma_o 2R_o \pi h_o$)