DOI 10.1007/s11223-026-00920-7
Strength of Materials, Vol. 58, No. 3, May, 2026

EXPERIMENTAL EVALUATION OF TEST BATCH QUALITY OF METAL CYLINDERS USING FULL-SCALE HIGH-PRESSURE HYDRAULIC TESTS. PART 2. RESULTS OF FULL-SCALE CYCLIC TESTS

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UDC 539.4

The appearance of foreign-made high-pressure cylinders for technical gases on the Ukrainian market requires appropriate experimental testing of their quality to justify their purchase. Experimental studies of the strength parameters of a test batch of high-pressure cylinders (seven sizes, five cylinders of each type), manufactured from seamless tubes, established that two cylinder sizes failed to withstand standard cyclic pressure tests. The probable cause of failure is a manufacturing defect resulting from non-compliance with the regulatory document regarding the radius of curvature of the concave bottom and significant deformations under cyclic test pressure loading. Significant cyclic deformations of the cylinder bottoms contributed to the initiation and propagation of fatigue cracks in the stress concentration zone in the circumferential direction, which led to failure due to the action of longitudinal stresses. The following mechanisms of cylinder failure during standard testing have been experimentally established. Under static loading with increasing internal pressure, failure occurs due to the action of hoop stresses in the longitudinal direction of the cylinders via a quasi-static mechanism with signs of accumulated strain limitation. Under cyclic loading with test pressure, failure occurs due to longitudinal stresses via a low-cycle fatigue mechanism resulting from the initiation and growth of fatigue cracks to critical sizes in the stress concentration zone.

***Keywords*:** hydraulic testing, internal pressure, loading, stress, deformation.

Introduction. Due to the cessation of production in Ukraine of high-pressure cylinders for technical gases and a sharp increase in demand for them during the COVID-19 pandemic, it became necessary to procure them abroad. The economic justification for the mass procurement of high-pressure metal cylinders of various capacities and their certification in Ukraine requires the conduct of standard static and cyclic internal pressure hydraulic tests. Standard hydraulic test procedures involve loading with increasing pressure until failure; cyclic tests involve a test pressure based on 12,000 cycles. Detailed test procedures can be found in the relevant manufacturers' technical specifications (TS). For example, DSTU EN ISO 9809-2019, BS EN 12245-2009, GB 28053-2011, ISO 11119-2002, ISO 11439-2000. According to the rules for the manufacture and safe operation of pressure vessels, cylinders must be designed such that the wall stresses during a hydraulic test at the test pressure do not exceed 85–90% of the metal's yield strength at room temperature for this steel grade [1].

The test pressure is a design value and may exceed the working pressure by 1.25–3 times, depending on the material, cylinder design, and operating conditions. The test pressure should be less than one and a half times the working pressure, which, in turn, determines the gas pressure in the cylinder at 15°C. In the United States, the most common ratio is 5/3 (the ratio of test pressure to working pressure). In a number of regulatory documents, cyclic tests are conducted specifically at test pressure; apparently, in the absence of failure, this provides a greater guarantee of cyclic durability and reliability compared to those standards where tests are conducted at working pressure. It is also

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important that at higher loads, there is a high probability of detecting quasi-static failure due to rupture (tearing), which is not permitted for cylinders. Such failure can occur in the event of imperfections, such as the shape of concave bottoms, transition points from one geometric shape to another, and so on. Local inclusions, cavities, cracks, etc., can also lead to depressurization, but this occurs primarily through a fatigue mechanism involving the formation of a leak, which is less dangerous compared to a rupture.

According to [1], in serial production, cyclic tests are conducted on two or three cylinders, or one cylinder from the produced batch, or one cylinder from five batches, i.e., one per 1,000 produced cylinders. If the results are unsatisfactory, testing is conducted on five cylinders. Cylinders must withstand 12,000 cycles of test hydraulic pressure without depressurization (failure due to rupture or leakage resulting from a leak). For cylinders with a test pressure exceeding 450 bar, the upper cyclic pressure may be reduced to 2/3 of the test pressure, in which case the cylinders must withstand 80,000 cycles.

Detailed test procedures can be found in the relevant regulatory documents of cylinder manufacturers. The working pressure in a cylinder is determined by the gas pressure in the cylinder at a temperature of 15°C; according to some regulatory documents, this temperature is 20°C. It is clear that as the temperature rises, the gas pressure in the cylinder also increases. There are tables for filling cylinders with pressure depending on the gas temperature, and these vary slightly for different gases. At the maximum operating temperature of the cylinder, the pressure inside it must not exceed the test pressure. In some cases, brief exposure of the cylinder to high temperatures is permitted, provided that the steady-state temperature inside the cylinder does not exceed the maximum operating temperature.

During cyclic testing, ranges (tolerances) are specified for the maximum and minimum pressures in the cycle. For the maximum: from the nominal cyclic pressure to +3% or +10 bar, whichever is lower. For the minimum: from zero to 10% of the nominal cyclic pressure, but not exceeding 30 bar. Additionally, during testing, the frequency of pressure changes must not exceed 0.25 Hz (15 cycles/min), and the average temperature on the outer surface of the cylinders must not exceed 50°C.

Purpose and Objectives of the Research. The appearance on the Ukrainian market of foreign-made cylinders requires appropriate experimental testing of their quality and reliability to justify their purchase. An experimental quality check of the delivered sample batch of cylinders was conducted by testing for static and cyclic strength at a specialized certified laboratory of the Paton Electric Welding Institute of the National Academy of Sciences of Ukraine, with the participation of specialists from the Pysarenko Institute for Problems of Strength of the National Academy of Sciences of Ukraine.

Methodology for Testing the Cyclic Strength of Cylinders. The study involved 35 high-pressure cylinders manufactured from seamless tubes (seven sizes, with five cylinders of each type). For each size, testing was conducted using increasing internal hydraulic pressure until two cylinders failed, and cyclic loading with test pressure on three cylinders. Standard tests involving cyclic hydraulic internal test pressure on the test batch of cylinders were conducted in accordance with the requirements of DSTU EN ISO 9809-3:2019 on a specially designed test bench [2]. The test pressure (P_t) reached 5/3 of the working pressure (P_w). The latter was 150 bar, and the test pressure was 250 bar. According to the technical specifications, the cycle count for the hydraulic test pressure cyclic tests was 12,000 cycles. It should be noted that it is precisely in this range of load cycle numbers that the transition boundary lies from low-cycle quasi-static to low-cycle fatigue failure of carbon steels.

Each of the 21 cylinders selected for cyclic hydraulic testing was subjected to a test pressure P_t in a 'water jacket' prior to the start of testing, with the residual expansion coefficient determined. The cylinders had not previously been pressurized. They were assembled in a series. The cylinders (21 units) were subjected to cyclically varying internal hydraulic test pressure. The total volume of the cylinders was 273.8 L.

Figure 1 shows the cylinders assembled in a string for cyclic testing with a test pressure of 250 bar.

The actual load parameters are as follows: maximum cycle pressure – 249.5…255.4 bar (average ~252.4 bar), minimum cycle pressure – 15.2…17.7 bar (average ~16.3 bar), cycle range – 231.8…240.2 bar (average ~236.1 bar), temperature of water exiting the system – 25.5…41 °C, cycle duration – 0.3969 min (23.82 s), loading frequency – 2.52 cycles/min (151.16 cycles/h), loading time – approximately 16 s, pressure decay time – approximately 7 s.

Fig. 1. Cylinders assembled in a garland for cyclic testing with test pressure.

Figure 2 shows a screenshot of the time-pressure diagram at the end of the cyclic tests – 12,044 load cycles at a test pressure of 250 bar.

A GN 600-800 pump was used to generate the cyclic pressure. Maximum pressure: 60 MPa; flow rate: 800 l/h.

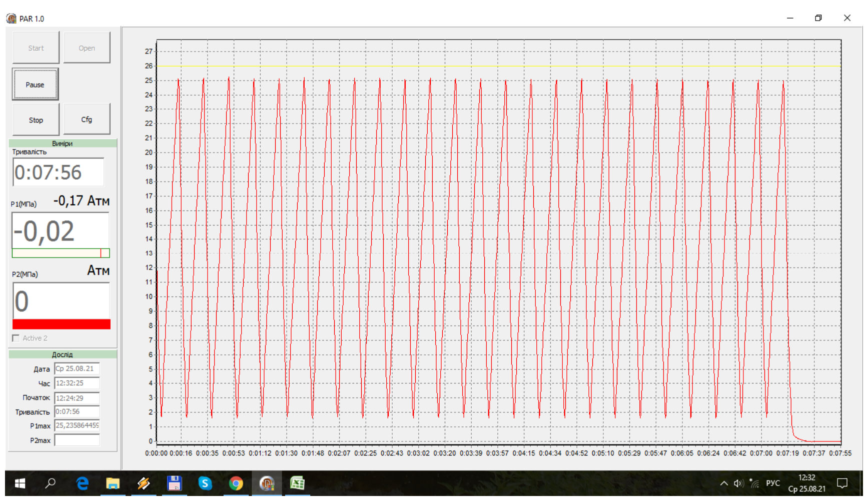


Fig. 2. Screenshot of the time-pressure diagram at the end of cyclic testing.

Results of Experimental Studies. The standard procedure for loading with increasing internal hydraulic pressure until failure and with cyclic test pressure involves preloading the cylinders with test pressure in a water jacket. Table 1 presents data on cylinder parameters during loading with test pressure in a water jacket.

Cyclic tests conducted at test pressure on the experimental batch of cylinders showed that cylinders of sizes 2 (Nos. 2.1…2.3) and 5 (Nos. 5.1…5.3) did not withstand the base (12,000) number of loading cycles, and their failure occurred near the bottom. The other cylinders withstood 12,044 cycles without failure or leakage. Fatigue failure occurred in the transverse direction of the bottom section of the cylinder (Fig. 3) due to the action of longitudinal stresses in the stress concentration zone. Under cyclic loading with test pressure, the ratio of the test pressure to the yield pressure of the cylinder metal during tests with increasing internal pressure to failure ranged from 0.59 to 0.86.

Table 2 presents data on the circumferences/perimeters (PER) of cylinders 5.1, 5.2, and 5.3 before and after the first loading with test pressure in a water jacket during cyclic tests at cross-sections of the cylindrical part: PER1 – near the neck, PER3 – midpoint, PER5 – bottom of the cylinder.

TABLE 1. Data on the Value of H1 in the Initial State, H1WJ in a Water Jacket under Test Pressure, and H1k after the Water Jacket (WJ) in the Absence of Pressure

V, l	D, mm	Cylinder size number	Cylinder No.	Test type	H1 in the final state, mm	0.12×D, mm	H1WJ at $P_t \neq 0$, mm	H1k after WJ at $P_t = 0$, mm
2	108	1	1.1	CT	18.1	12.96		
			1.2	CT	18.0			
			1.3	CT	18.1			
			1.4	ST	18.0		17.6	15.6
			1.5	ST	17.9		(Δ2 mm)	15.7
3.2	10	2	2.1	CT	19.5	14.40		
			2.2	CT	19.3			
			2.3	CT	19.2			
			2.4	ST	19.0		18.0	15.0
			2.5	ST	19.3		(Δ3mm)	15.8
5	140	3	3.1	CT	22.1	16.80		
			3.2	CT	22.3			
			3.3	CT	22.3			
			3.4	ST	21.8		20.6	19.4
			3.5	ST	22.0		(Δ1.2 mm)	20.4
10	140	4	4.1	CT	21.7	16.80		
			4.2	CT	22.0			
			4.3	CT	20.3			
			4.4	ST	22.6		21.3	20.0
			4.5	ST	21.9		(Δ1.4 mm)	19.9
10	159	5	5.1	CT	25.9	19.08		
			5.2	CT	26.3			
			5.3	CT	26.0			
			5.4	ST	26.4		24.6	20.5
			5.5	ST	26.4		(Δ4.4 mm)	20.2
20	180	6	6.1	CT	27.3	21.60		
			6.2	CT	26.9			
			6.3	CT	27.4			
			6.4	ST	27.0		25.7	23.2
			6.5	ST	26.6		(Δ2.6 mm)	23.1
40	219	7	7.1	CT	27.2	26.28		
			7.2	CT	26.7			
			7.3	CT	26.8			
			7.4	ST	26.9			25.5
			7.5	ST	27.5			25.5

Note: V is cylinder volume, D is cylinder diameter, CT means cyclic tests, ST means static tests, H1≥0.12×D.

Table 3 presents the results of cyclic testing of cylinders at a test pressure of 250 bar based on 12,000 cycles.

Fatigue failure occurred in the most heavily loaded (b) bottom part of the cylinder without signs of plastic deformation in the stress concentration zone due to a design flaw in the shape of the concave bottom (Fig. 4).

Cylinder failure occurred during cyclic loading within the range of 2,942 to 7,584 cycles (against a standard of 12,000) in the stress concentration zone located on the inner side of the cylinders (Fig. 4a and b). According to the calculation data (Fig. 4b), the maximum stress intensity corresponds to the bottom part of the cylinder in the stress concentration zone. The small radius of curvature at the transition from the conical part of the bottom to the lower part is the cause of fatigue failure. This sharp transition of small radius in the form of a ring could be seen through the cylinder neck. It should be noted that when measuring the wall thickness of the cylinders with a TUZ-2 thickness gauge using the ultrasonic testing method from the outside in the zone of this radius (stress concentrator), the signal was lost. This stress concentrator is associated with the non-compliance of the fillet radius (0.05…0.057 of its nominal outer diameter) of the transition from the conical part of the cylinder to the lower part of the bottom with the requirements of ISO 9809-3. According to this standard, the transition radius on the inside of the cylinder must be no less than 0.075 of its nominal outer diameter, and the thickness of the cylindrical section approaching the transition to the bottom must increase. The concentrator was likely formed during hot ‘seaming’ of the bottom due to insufficient heating of the metal along the length of the pipe end section. The small radius of the transition from the conical part of the bottom to the lower section may have been caused by the small radius of the punch forming the inner surface of the bottom during hot ‘seaming’. This defect is also present on Type 1 cylinders, but failure did not occur due to the wall thickness margin. For Type 1–5 cylinders, the minimum allowable wall thickness is 4 mm, with the actual thickness being approximately 4.5 mm. Cylinders of Types 3, 4, 6, and 7 do not have this defect; there is a smooth transition in these cases. After pressurizing the cylinders with test pressure in the water jacket, deformation of the bottom section was detected (changes in dimension H1, Table 1, Fig. 4a). At the same time, failure occurred only in cylinders 2.1…2.3 and 5.1…5.3, for which the maximum deformation of the bottom section was 3.0 and 4.4 mm, respectively (difference between H1WJ and H1k, Table 1, Fig. 4a) after the first loading in the water jacket.

Fig. 3. Photographs of failed cylinders after cyclic hydraulic testing in their initial state and after failure.

TABLE 2. Perimeters of Cylinders 5.1, 5.2, and 5.3 before (above the Line) and after (below the Line) the First Test Pressure Loading During Cyclic Testing at Cross-Sections PER1, PER3, and PER5

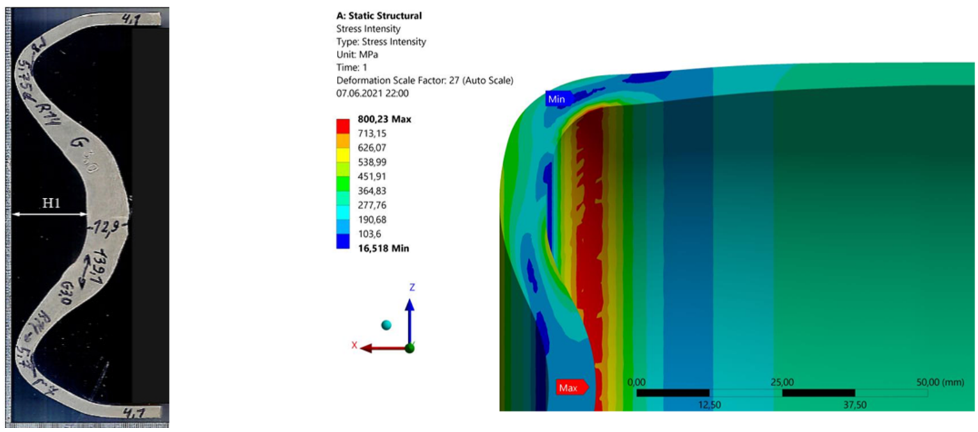
Cross-section No.	Cross-section location	Perimeter, mm			Deformation, %		
		Cylinder number					
		5.1	5.2	5.3	5.1	5.2	5.3
PER1	At the top, mm	499.0 500.0	498.5 500.0	495.5 498.0	0.0	0.30	0.10
PER3	Middle, mm	501.0 501.5	501.5 501.5…502.0	500.0 500.5	0.10	0.10	0.10
PER5	Near the bottom, mm	501.0 501.5	501.0 501.5	499.0 500.0	0.10	0.10	0.20

Under cyclic loading with a test pressure, failure occurred due to the initiation and propagation of a fatigue crack to critical dimensions in the circumferential direction (low-cycle fatigue failure, Fig. 3) due to the action of longitudinal (axial) stresses, in contrast to static loading, where failure occurred in the longitudinal direction of the cylinder due to hoop stresses. The change in cylinder volume after the first test pressure loading in the water jacket was likely due to irreversible deformation of the bottoms (change in dimension H1, Fig. 4a).

TABLE 3. Results of Cyclic Testing of Cylinders at a Test Pressure of 250 bar

Cylinder size number	Cylinder number	N, cycles	ΔN	Cylinder parameters
5	5.3	2942	0	V=10 l D=159 mm
	5.2	3144	202	
	5.1	3382	238	
2	2.1	5576	0	V=3.2 l D=120 mm
	2.3	6953	1377	
	2.2	7584	631	
	Base number of loading cycles	12044		

Note: N is the number of cycles withstood by the cylinder; ΔN is the difference in the number of cycles compared to the previous cylinder of the same type.



a b

Fig. 4. Cross-section of the bottom part of the cylinder (a) and stress intensity distribution (b) under test pressure loading.

Since cylinders 2.1…2.3 and 5.1…5.3 did not withstand the base number of cycles to failure (12,000 cycles) under a test pressure of 250 bar, additional cyclic hydraulic tests were conducted on another batch of cylinders ($V = 10$ l, $D = 159$ mm and $V = 3.2$ l, $D = 120$ mm) at a slightly lower internal pressure of 225 bar.

Table 4 presents data comparing the effect of internal pressures of 225 and 250 bar on the cyclic endurance of cylinders ($V = 10$ l, $D = 159$ mm and $V = 3.2$ l, $D = 120$ mm).

TABLE 4. Comparison of the Effect of Internal Pressures of 225 and 250 bar on the Cyclic Endurance of Cylinders

No.	Cylinder parameters	P_t, bar	Fail/No	n_{f1}	n_{f2}	n_{f3}	M_{nf}	σ_{nf}	ν_{nf}	P_y, bar	P_b, bar
1	V=10 l D=159 mm-	225	fail	4085	4214	4504	4268	215	0.05	290.4	425.3
2		250	fail	2942	3144	3382	3156	220	0.07	299.4	436.2
3	V=10 l D=159 mm*	225	fail	5686	6197	7062	6315	696	0.11	332	474.4
										330	481.6
4	V=3.2 l D=120 mm*	225	no	12089	12089	12089	-	-	-	366.4	514.9
5	V=3.2 l D=120 mm	250	fail	5576	6953	7584	6704	1027	0.15	374.4	532.7

Note: n_{fi} is the number of cycles the cylinder withstood; fail/no indicates whether the cylinder's failure occurred or not; an asterisk indicates a new, higher-quality batch; P_y and P_b are yield and burst pressures, respectively, which are determined by applying increasing internal pressure to other cylinders of the same type.

Cyclic hydraulic tests of cylinders ($V = 10$ l, $D = 159$ mm) at test pressures of 225 and 250 bar showed that a 10% pressure reduction increases the cyclic endurance of the tested batch of cylinders with a manufacturing defect by 33.0…38%. For a batch of higher-quality cylinders ($V = 10$ l, $D = 159$ mm), a 10% pressure reduction nearly doubles the cyclic endurance compared to the previously tested batch. Only the cylinders with $V = 3.2$ l, $D = 120$ mm withstood the cyclic hydraulic tests at a lower test pressure of 225 bar, despite the presence of a manufacturing defect.

Figure 5 shows the dependence of cyclic endurance on minimum wall thickness; Fig. 6 shows the dependence of cyclic endurance of cylinders 2.1…2.3 and 5.1…5.3 on stress intensity σ_i during tests at test pressures of 250 and 225 MPa. It should be noted that the calculated stress intensity values obtained during tests at test pressures of 250 and 225 MPa are significantly lower than the corresponding stress intensity values determined by the finite element method (FEM) in the bottom part of the cylinders, which amounted to 800 MPa, as shown in Fig. 4b. Here, the stress intensity $\sigma_i = \frac{\sqrt{3}}{2}\sigma_t = \frac{\sqrt{3}Pr_0}{2s_o}$ was determined as the equivalent stress according to the von Mises yield criterion. For the calculations of the test pressure of cylinders 3.5 and 7.5, the metal's strength characteristics were adopted, which, according to the certificate, were $\sigma_{0.2} \geq 505$ MPa and $\sigma_b \geq 715$ MPa, respectively.

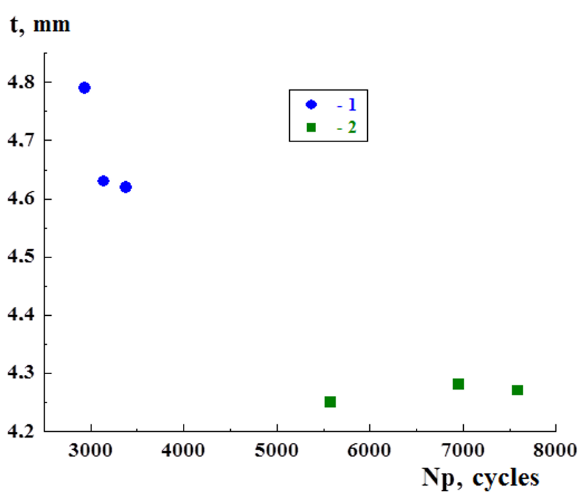


Fig. 5.

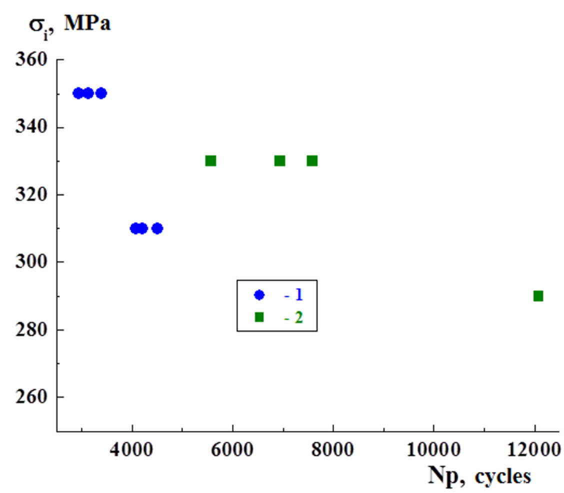


Fig. 6.

Fig. 5. Dependence of the cyclic fatigue life of cylinders on their minimum wall thickness. Here and in Fig. 6: (*1*) corresponds to cylinder size 5; (*2*) to size 2.
Fig. 6. Dependence of the cyclic durability of cylinders on the design stress intensity.

Operational experience with structures and specialized studies have shown that high local stresses arise at locations with abrupt changes in element dimensions, sharp corners, notches, holes, scratches, etc., and such locations are referred to as stress concentration zones. Under cyclic loading, fatigue cracks typically form at these locations. Local stresses in stress concentration zones σ_{max} are significantly higher than similar stresses in the same cross-section of parts σ_n without a stress concentrator. A measure of stress concentrators is the theoretical stress concentration factor $\alpha_\sigma = \frac{\sigma_{max}}{\sigma_{н}}$, which is determined theoretically based on approaches from elasticity theory under the assumption of an 'elastic' stress distribution [3]. Values of the theoretical stress concentration factor α_σ for the most common cases are given in reference books.

In stress concentration zones, a complex stress state (plane or volumetric) typically arises. The magnitude of the stress concentration factor is determined by the degree of abrupt change in the object's shape. If the transition from one shape to another is abrupt and at a right angle, the factor α_σ takes on the highest values. If the cross-section changes gradually, the value of α_σ decreases sharply and may be equal to one. Stress concentration is less dangerous for ductile materials than for brittle ones. In practice, under static loading of products made of ductile materials, stress concentration does not affect their strength; that is, ductile materials are not very sensitive to stress concentration. Under cyclic loading, the reduction in the fatigue limit is evaluated using the effective stress concentration factor $k_\sigma = \frac{\sigma_{-1}}{\sigma_{-1k}}$ [4] as the ratio of the fatigue limit of a smooth specimen without stress concentration σ_{-1} to the fatigue limit of a specimen with a stress concentrator σ_{-1k}, which has the same absolute cross-sectional dimensions.

The effective stress concentration factor is usually less than the theoretical value, and only for brittle homogeneous materials is it very close to the theoretical value $k_\sigma \leq \alpha_\sigma$. For ductile materials, the effective stress concentration factor is close to unity. In stress concentration zones, local stresses arise that reach the yield point and cause plastic deformation of the metal. As a result, the ability of local stresses to propagate in a product made of ductile materials is sharply reduced. For brittle materials, stress concentration leads to a decrease in strength, since there is no factor that mitigates the effect of concentration, namely the material's yield property. Under impact and cyclic loads, if deformations and stresses change rapidly over time, the stresses do not have time to equalize, and the harmful effect of stress concentration persists. For homogeneous brittle materials, the non-uniformity of stress distribution due to concentration persists at all stages of loading, including the static stage. Material failure (crack formation) begins at points of maximum stress. Hardened steel is particularly sensitive to stress concentrators, and this sensitivity increases with higher strength properties. The theoretical stress concentration factor depends on the geometry of the stress concentrator and does not reflect the properties of the actual material. The combined effect of the stress concentrator's geometry and the material's properties is determined by testing specimens of the material until failure. In such tests, the effective (actual) stress concentration factor is determined, which reflects the influence of the stress concentrator's geometry and the material's mechanical properties. Under static loading, the effective stress concentration factor ($k = \frac{P_I}{P_{II}}$) is defined as the ratio of the breaking load of the specimen without a stress concentrator (P_I) to the breaking load of the specimen with a stress concentrator (P_{II}) [4].

In the range of multi-cycle fatigue, stress concentration leads to a decrease in the endurance limit of products made from both ductile and brittle materials [5]. In the absence of experimental data for the effective stress concentration factor, empirical formulas are sometimes used that relate it to the material's tensile strength.

For the range of multi-cycle fatigue, the endurance limits of specimens with a stress concentrator are lower than the corresponding values for a smooth specimen without a concentrator [6], i.e., $k_\sigma = \frac{\sigma_{-1}}{\sigma_{-1k}} > 1$. It should be noted that the value of α_σ is also greater than one. In real materials, due to plastic deformations in the microzone, stress concentrations are somewhat redistributed and smoothed out. For example, Fig. 7 shows fatigue curves for smooth cylindrical specimens and those with a stress concentrator (α_σ=1.8, k=0.65) made of Grade 20 steel with the same minimum cross-section during low-cycle fatigue testing with control of maximum and minimum stresses and a cycle asymmetry coefficient of $R_\sigma = -0.5$. For a specimen with a stress concentrator in the form of a groove ($\frac{r}{d} = 0.25$), the theoretical stress concentration factor according to Table 11 [3] is 1.8. Under low-cycle loading, the effective stress concentration factor k_σ based on 5,000-cycle tests is 0.73…0.77. It should be noted that for the high-cycle fatigue range, we have $k_\sigma = \frac{\sigma_{-1}}{\sigma_{-1k}} > 1$, whereas for the low-cycle fatigue range, $k_\sigma = \frac{\sigma_{max}}{\sigma_{max\,k}}$. According to the data presented (Fig. 7), in the stress concentration zone, stresses exceed the conventional yield strength of steel ($\sigma_{0.2}$), whereas the specimen without a stress concentrator deforms elastically (maximum stresses do not reach the values of $\sigma_{0.2}$).

Figure 8 shows a schematic of a smooth specimens and those with a stress concentrator.

The considered ratios of maximum stresses for a smooth specimen and a specimen with a stress concentrator are valid under uniaxial loading conditions, where the applied stresses in the stress concentration zone exceed the yield strength of the steel.

In the case of a plane stress state of the cylinders, the values of the longitudinal stresses σ_z are half those of the hoop stresses σ_t. Under static loading with increasing internal pressure, failure occurred in the longitudinal direction of the cylinders due to the action of hoop stresses σ_t, the maximum values of which practically coincided with the strength characteristics of laboratory specimens in tensile tests (see Part 1 of this study [7]). For example, the failure of cylinder 3.5 occurred at a pressure of 441.1 bar, while the internal pressure at which metal yielding began was 324.4 bar; accordingly, the calculated hoop failure stresses were 651.39 MPa. The calculated hoop yield stress of the metal reached 479.05 MPa. Cylinder 7.5 failed at a pressure of 444.6 bar, with the internal pressure at which metal yielding began being 299.4 bar; accordingly, the calculated hoop failure stress was 720.05 MPa. The calculated hoop yield stresses of the metal were 484.89 MPa. It should be noted that the strength characteristics of the cylinders obtained during the burst test are in good agreement with the results of tests on laboratory specimens made from cylinders 3.4 and 7.4, where the tensile strength of the metal in cylinders 3.4 and 7.4 was 712.62 MPa and 729.25 MPa, and the yield strength was 461.6 MPa and 483.21 MPa, respectively.

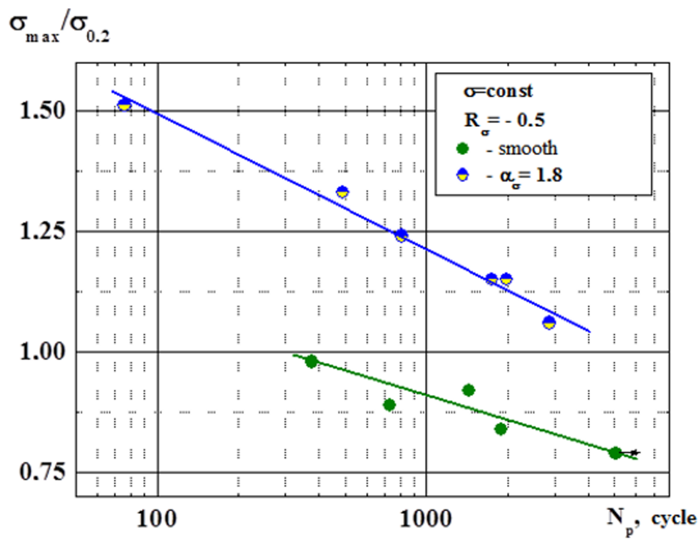


Fig. 7. Fatigue curves for smooth cylindrical specimens and specimens with a stress concentrator made of Grade 20 steel with the same minimum cross-section.

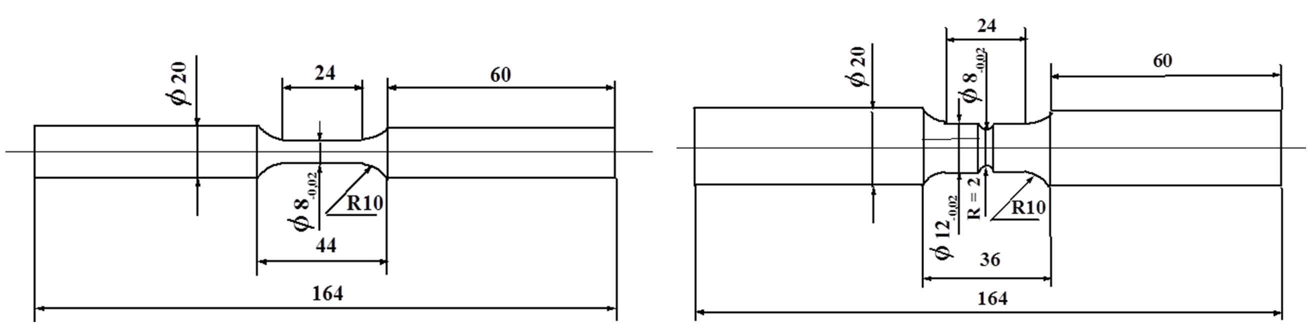


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Fig. 8. Geometry of specimens without a stress concentrator (a) and with a stress concentrator (b).

Under internal pressure loading, the hoop stresses were determined using the relationship $\sigma_t = \frac{Pr_0}{s_0}$, and the longitudinal (axial) stresses using $\sigma_z = \frac{\sigma_t}{2}$, where $r_0 = \frac{D_{av} - s_0}{2}$ is the radius of the mid-surface, and s_0 is the average wall thickness of the cylinders.

Under cyclic loading conditions for cylinder 2.4, the calculated hoop stresses σ_t from the test pressure (250 bar) were 336.8 MPa ($0.73\sigma_{0.2}$). The calculated longitudinal stresses σ_z from the test pressure of 250 bar were 168.4 MPa ($0.36\sigma_{0.2}$). For cylinder 5.4, the corresponding hoop stresses from the test pressure of 250 bar were 410.4 MPa (i.e., $0.89\sigma_{0.2}$). The calculated longitudinal stresses from the test pressure of 250 bar were 205.18 MPa ($0.44\sigma_{0.2}$).

The obtained data agree satisfactorily with similar calculations of longitudinal stresses determined as the effect of an internal test pressure of 250 bar on the bottom surface area. The calculated longitudinal stresses for cylinder 2.4 determined in this way were 155.9 MPa ($0.34\sigma_{0.2}$). For cylinder 5.4, the corresponding longitudinal stresses were 193.0 MPa ($0.42\sigma_{0.2}$). The error in determining these stresses using the two methods described is 6.0% and 7.5%, respectively, for cylinders 5.4 and 2.4.

As noted above for cylinders 2.1…2.3, the small radius of curvature near the bottom – which was 0.05 times the outer diameter of 120 mm, or 6.0 mm – acts as a stress concentrator. For cylinders 5.1…5.3, a similar fillet radius was 0.057 times the outer diameter of 159 mm, i.e., 9.0 mm. According to Table 11 [3], the ratio of the fillet radius to the mandrel diameter is 0.0625, which is very close to the cylinders under consideration, where the corresponding ratio is 0.05…0.057, and the coefficient (α_σ) according to the same table is 1.75. For the cylinders under consideration, the radius of curvature at the bottom of the cylinders is less than the standard specified in ISO 9809-3 and must be ≥0.075 of the nominal outer diameter, i.e., 12 mm. Thus, taking into account the theoretical stress concentration factor, the value of the maximum longitudinal stresses due to the test pressure in the cylinder neck zone 2.1…2.3 should be 294.7 MPa ($0.64\sigma_{0.2}$), and for cylinders 5.1…5.3 – 359.06 MPa ($0.78\sigma_{0.2}$). That is, the longitudinal stresses in the stress concentrator zone did not reach the conventional yield strength of the cylinder metal.

Under axial loading conditions, the fatigue curve of the specimen without a stress concentrator lies below the corresponding curve of the specimen with a stress concentrator (Fig. 7), for which the stresses exceed the metal's yield strength. The stress values for the fatigue curve of the specimen without a stress concentrator are significantly lower than the metal's yield strength. Unlike under axial loading, the metal of cylinders subjected to internal pressure is in a plane stress state. In this case, the longitudinal stresses are half the magnitude of the hoop stresses. If we follow the analogy with the patterns observed under axial loading of laboratory specimens with and without stress concentrators, the maximum values of longitudinal stresses in the cylinder metal within the concentration zones should exceed the material's yield strength. The calculated longitudinal stresses from the test pressure in the stress concentration zone for cylinders 2.1–2.3 and 5.1–5.3 were $0.64\sigma_{0.2}$ and $0.78\sigma_{0.2}$. Similar calculated stresses without accounting for stress concentration for cylinders 2.1–2.3 and 5.1–5.3 were $0.36\sigma_{0.2}$ and $0.44\sigma_{0.2}$, respectively. Since the failure of the cylinders occurred in the transverse direction due to the action of longitudinal stresses, which had very low values, the discrepancy between the failure patterns of laboratory specimens under uniaxial loading and those under biaxial loading during cyclic testing of the cylinders is likely related to the specific effects of biaxiality and the stress concentrator in their bottom section.

Conclusions. To justify the purchase of high-pressure cylinders of various capacities from abroad and their certification in Ukraine, standard static and cyclic hydraulic internal pressure tests were conducted at the certified laboratory of the Paton Electric Welding Institute of the National Academy of Sciences of Ukraine in accordance with the requirements OF DSTU EN ISO 9809-3:2019 on a pilot batch of high-pressure cylinders (seven sizes, with five cylinders of each type) manufactured from seamless tubes. For each size, testing was conducted with increasing internal hydraulic pressure until two cylinders failed and with cyclic loading at test pressure on three cylinders.

Experimental studies of the strength parameters of the test batch of high-pressure cylinders established that two cylinder sizes failed to withstand standard cyclic testing at a test pressure of 250 bar. Tests of an additional improved batch of cylinders of this type also failed to withstand the specified number of loading cycles. Only after reducing the test pressure by 10% was a positive result obtained for one type of cylinder.

During cyclic loading of the cylinders with internal hydraulic test pressure, failure occurred in the bottom section at the stress concentration zone where the cylindrical part transitions to the concave bottom. The probable cause of failure for the two cylinder sizes is a manufacturing defect involving non-compliance with the standard regarding the radius of curvature in the bottom section of the cylinders, which caused significant deformation of the bottoms under test pressure loading. Significant cyclic deformations of the cylinder bottoms contributed to the initiation and propagation of fatigue cracks in the stress concentration zone in the circumferential direction, which led to failure due to the action of longitudinal stresses.

Fatigue failure under cyclic test pressure loading due to the action of longitudinal stresses is likely related to the peculiarities of the stress-strain state in the stress concentration zone under conditions of a plane stress state and the influence of the bidirectional nature of the load.

The failure mechanisms of cylinders under standard testing conditions have been experimentally established: under static loading with increasing internal pressure, failure occurs due to the action of hoop stresses in the longitudinal direction of the cylinders via a quasi-static mechanism with signs of accumulated strain limitation; under cyclic loading with test pressure, failure occurs via a low-cycle fatigue mechanism due to the initiation and growth of fatigue cracks to critical dimensions in the stress concentration zone in the form of a manufacturing defect—a deviation of the radius of curvature of the concave bottom from the standard document.

Funding. This study received no external funding.

Authors Contribution. All authors equally contributed to writing this paper.

Conflict of Interests. The authors declare that they have no conflicts of interest.

Data Availability. Not applicable.

REFERENCES

1. *DSTU EN ISO 9809-3:2019. Gas Cylinders. Refillable Seamless Steel Gas Cylinders. Design, Construction, and Testing. Part 3. Normalized Steel Cylinders (EN ISO 9809-3:2010, IDT; ISO 9809-3:2010, IDT)*, Effective from January 1, 2020.
2. R. I. Dmitrienko, E. F. Garf, and V. P. Chizhichenko, Final expansion of cylinders (brief overview), *Technical Diagnostics and Non-Destructive Testing*, No. 1, 23–28 (2014).
3. G. S. Pisarenko, *Strength of Materials* [in Russian], Vyshcha Shkola, Kyiv (1973).
4. G. S. Pysarenko, O. L. Kvitka, and E. S. Umansky, *Strength of Materials* [in Ukrainian], Vyshcha Shkola, Kyiv (2004).
5. V. I. Shvabyuk, *Strength of Materials* [in Ukrainian], Znannya, Kyiv (2016).
6. L. E. Matokhnyuk, A. V. Byalonovych, A. A. Kotlyarenko, et al., Influence of loading frequency on fatigue resistance of D16T Alloy, *Strength Mater*, **54**, 407–414 (2022). https://doi.org/10.1007/s11223-022-00416-0
7. O. P. Gopkalo, R. I. Dmytrienko, O. L. Palienko, et al., Experimental evaluation of test batch quality of metal cylinders using full-scale high-pressure hydraulic tests. Part 1. Results of full-scale static tests, *Strength Mater*, **58**, 470–481 (2026). https://doi.org/10.1007/s11223-026-00902-9