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The equivalent amplitude stress as a solution of mean stress effect problem in fatigue analysis

Tomasz Bednarek[1], Izabela Marczewska[2], Artur Marczewski[3], Włodzimierz Sosnowski[4], Hieronim Jakubczak[5], Jerzy Rojek[6]

[1] Institute of Fundamental Technological Research, Polish Academy of Sciences, ul. Swietokrzyska, 21 00-049, Warsaw, Kazimierz Wielki University, Institute of Environmental Mechanics and Applied Computer Science, ul. Chodkiewicza, 30 85-064, Bydgoszcz, e-mail: bednarek@ippt.gov.pl
[2] Institute of Fundamental Technological Research, Polish Academy of Sciences, ul. Swietokrzyska, 21 00-049, Warsaw, e-mail: imar@ippt.gov.pl
[3] Institute of Fundamental Technological Research, Polish Academy of Sciences, ul. Swietokrzyska, 21 00-049, Warsaw, e-mail: asmar@ippt.gov.pl
[4] Institute of Fundamental Technological Research, Polish Academy of Sciences, ul. Swietokrzyska, 21 00-049, Warsaw, Kazimierz Wielki University, Institute of Environmental Mechanics and Applied Computer Science, ul. Chodkiewicza, 30 85-064, Bydgoszcz, e-mail: wsosn@ippt.gov.pl
[5] Warsaw University of Technology, Institute of Construction Machinery Engineering, ul. Narbutta, 84 02-524, Warsaw
[6] Institute of Fundamental Technological Research, Polish Academy of Sciences, ul. Swietokrzyska, 21 00-049, Warsaw, e-mail: jrojek@ippt.gov.pl

In this paper a numerical algorithm is presented to make possible adopting different loading schemes of specific structure at hand for instance hydraulic cylinders, to specific Wöhler curves characterizing fatigue resistance of given material. Hydraulic cylinders investigated under E. C. project «PROHIPP» are analyzed and some solutions of the crack problem in oil port area are proposed. Oil penetration in an oil port connection zone can be eliminated after some design modifications.

Key words: fatigue analysis, Wöhler curves, Goodman diagrams, hydraulic cylinders, endurance limit.

Nomenclature

ρ	density	S^{at}	threshold amplitude stress
E	Young modulus	S^{a}	stress amplitude
ν	Poisson ratio	S^{ae}	equivalent stress amplitude for $R = -1$
p	inner pressure	S^{m}	mean stress
$p_{\min}$	minimal values of inner pressure	$S^{\max}$	maximal stress
$p_{\max}$	maximal values of inner pressure	$S^{\min}$	minimal stress
R	stress ratio	n	exponent in Goodman relationship
N	number of cycles	f^{i}	factor of stress corresponding to initial stress
S_u	ultimate stress	f^{t}	factor of stress corresponding to threshold stress
S^{ai}	initial amplitude stress		

Introduction. There are two objectives of this paper. The first aim eies in presenting a numerical algorithm, which makes possible adopting different loading schemes of hydraulic cylinders to specific Wöhler curves characterizing fatigue resistance of given material. An uniaxial loading case is studied in order to confirm the relation between Goodman and Wöhler curves parameters. The second presenting calculations of hydraulic cylinders investigated under E. C. project «PROHIPP» and to propose some solution of the crack problem in oil port area, in order to eliminate oil penetration in an oil port connection zone.

It is well known that majority of fatigue tests is made for symmetric or nearly symmetric load when stress ratio is

$$R = \frac{p_{mix}}{p_{\max}} = \frac{S^{\min}}{S^{\max}} = -1 . \quad (1)$$

Number of codes accept input data only for such kind of symmetric load. In practice majority of structures for instance hydraulic cylinders, works under much more complex cyclic load characterized by different R values. In particular, some specific laboratory tests for hydraulic cylinders are made for $R = 0$ when the stresses changes from 0 to some specific, maximal value. In this case the external pressure is applied to the cylinder and then the pressure is removed. So the idea of equivalent amplitude stress is introduced in section 2 in order to find out number of cycles to failure for loads $R \neq -1$ in situations when test data are provided only for values $R = -1$.

Number of cycles in such tests depends on fatigue resistance of the weakest point of the cylinder. An example of such points is oil ports, where the welding residual stresses, local shear forces and forces due to oil penetration in the connection gaps lead to fatigue cracks causing final destruction.

Two kinds of numerical tests are performed. At the beginning the standard fatigue tests on workpiece shown in fig. 2 are simulated (Section 3). Goodman and Wöhler curves and idea of equivalent amplitude stress are used in order to calculate number of cycles to reach the fatigue limit of specimens with non-symmetric ($R \neq -1$) load.

Next in Section 4 the hydraulic cylinders fatigue problem is considered. Oil port typical deformations and design modifications possibility are shown in fig. 12. It can be observed that the gap between the oil port and cylinder in the oil port connection zone grows up with increasing oil pressure. Experiments (fig. 12) also confirm the crack sensitivity of the weld in this connection zone. Authors propose the improvement of this bad situation by using special washer or glue (as it is shown in fig. 15) in order to eliminate excessive gap between the oil port and cylinder surfaces. Such washer or glue prevents oil penetration thus eliminating the possibility of premature fatigue crack in neighboring welds.

The washer or glue material should be high temperature resistant — due to welding process of oil ports.

1. The idea of equivalent amplitude stress calculations for different stress ratio

The objective of this section is developing the algorithm of calculation of the amplitude stress $S^{ae}{}_{R=-1}$, which is *equivalent* to the given value $S^{a}{}_{R\neq-1}$. The assumption is made that such amplitude stress $S^{ae}{}_{R=-1}$ gives the same number of cycles to fatigue failure as it is for given $S^{a}{}_{R\neq-1}$. This assumption is based on Goodman and Wöhler curves dependency (fig. 1). Further the amplitude stress $S^{a}{}_{R\neq-1}$ can be obtainned from the finite element analysis of specific structure at hand loaded by arbitrary non-symmetric load, $R \neq -1$.

The history of standard fatigue test goes back to Wöhler who designed and built the first rotating that is beam test machine that produced fluctuating stress of a constant amplitude in test specimens [1, 2]. In tests Wöhler established a material property, known today as the fatigue limit. When specific fatigue data are missing, for example, number cycles to failure for the given stress ratio, one can use, in some range, empirical relation between amplitude stress and fatigue life N as a linear approximation of the $S-N$ curve in log-log coordinates. This range is described by two points: initial amplitude stress (S^{ai}) at about 1000 cycles and threshold stress (S^{at}) at approximately 2 millions cycles. Such linear Wöhler *S-N* curve for steel is shown on the left side of fig. 1 and can be represented (for $R = -1$) by the equation

$$S^{a}(N) = 10^{\frac{\log\left[S^{at}_{(R=-1)}\right]-\log\left[S^{ai}_{(R=-1)}\right]}{\log(N_t)-\log(N_i)}\left[\log(N)-\log(N_i)\right]+\log\left[S^{ai}_{(R=-1)}\right]}, \tag{2}$$

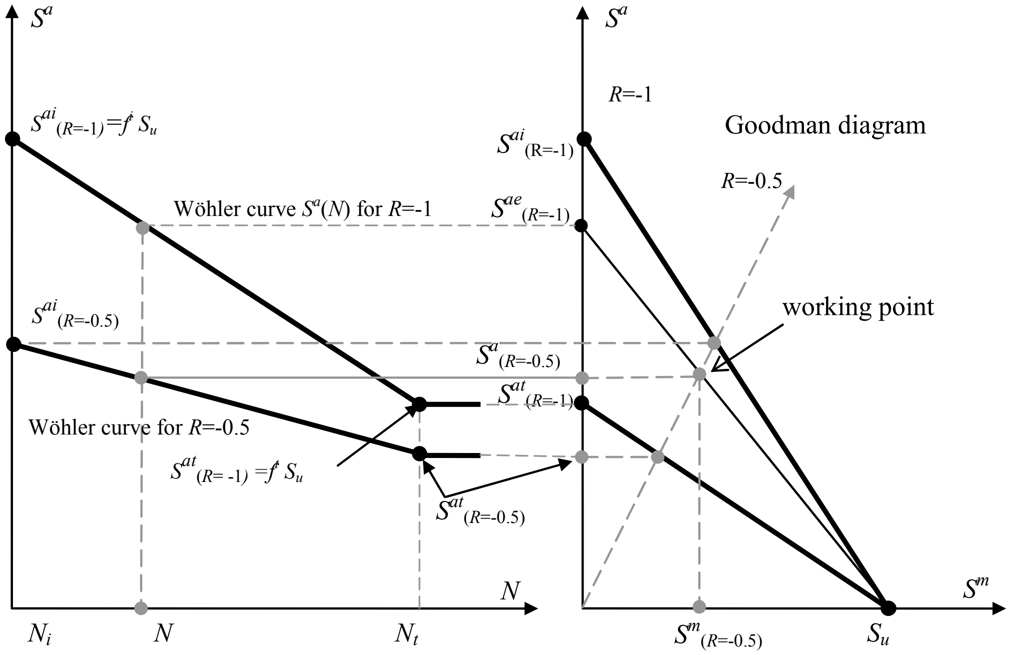

Fig. 1. Wöhler and Goodman diagrams dependency

where $N_i = 1e3$ and $N_t = 2e6$. Initial amplitude stress $S^{ai}_{(R=-1)}$ for $N_i = 1e3$ number of cycles is smaller than the ultimate stress S_u

$$S^{ai}_{(R=-1)} = S_u f^i . \tag{3}$$

Value of f^i can be found in literature [1, 3, 4]. In this paper f^i equals 0,9. The value of threshold amplitude stress $S^{at}_{(R=-1)}$ for $N_t = 2e6$ number of cycles is obtained by equation

$$S^{at}_{(R=-1)} = S_u f^t , \tag{4}$$

where f^t is the decreasing factor of stress corresponding to the stress threshold value. Value of f^t changes between 0,05 and 0,5 and depends on material properties and stress concentration factor [1, 3, 4]. In this paper f^t is equal 0,5. In following text simplified notation $S^{ai}_{(R=-1)} = S^{ai}$ and $S^{at}_{(R=-1)} = S^{at}$ will be used.

Substituting the amplitude stress in the considered point of the structure by working point $S^a(N) = S^a_{(R=-1)}$ and modifying the equation (2) we obtain the number of cycles to failure N

$$N = 10^{\frac{\log\left[S^a_{(R=-1)}\right]\log(N_i)-\log\left[S^a_{(R=-1)}\right]\log(N_t)+\log(S^{ai})\log(N_t)-\log(S^{at})\log(N_i)}{\log(S^{ai})-\log(S^{at})}} . \tag{5}$$

In this case $S^a_{(R=-1)} = S^{\max}$ due to the assumption of $R = -1$ load scheme.

Wöhler curves are obtained for the load scheme $R = -1$ by bending fatigue tests and rarely the axial-load tests ($R = 0$). However thisese zero or -1 mean stress ratio is not typical for real industrial components working under cyclic load.

On the basis of a on value of ultimate stress S_u of the material and a value of the amplitude stress for different $R \neq -1$ we can obtain the *equivalent* value of the amplitude stress for $R = -1$ from Goodman diagram (see fig. 1 right). Then this value can be applied in the equation (5) in order to calculate number of cycles to failure N.

Goodman diagram (fig. 1) is represented by the line between amplitude stress $S^a_{(R=-1)}$ (S^m equals 0) and ultimate stress of the material on S^m axis and the equation (6) takes plase

$$S^a_{(R\neq-1)} = S^a_{(R=-1)}\left[1-\left(\frac{S^m}{S_u}\right)^n\right] \tag{6}$$

When $n = 1$ the equation (6) is Goodman equation, at $n = 2$ we obtain Gerber equation. Here we use safe Goodman case $n = 1$ or $n = 1{,}2$. This relationship permits to obtain the equivalent amplitude stress S^{ae} for any load scheme ($R = -1$).

From the typical finite element analysis we can get maximal stress at the most stressed point of the structure.

The value of the mean stress S^{m} in the considered structure point can be obtained from the equation

$$S^{m} = \frac{S^{\max}(1+R)}{2} \tag{7}$$

where $S^{\max}$ is a maximal stress value obtained for any R from FEM analysis. Then $S^{a}_{(R\neq -1)}$ can be calculated from the following relations

$$S^{a}_{(R\neq -1)} = S^{\max} - S^{m}$$

or

$$S^{a}_{(R\neq -1)} = -\frac{S^{\max}(1+R)}{2} + S^{\max} . \tag{8}$$

These values can be used in the modified equation (6)

$$S^{ae}_{(R=-1)} = \frac{S^{a}_{(R\neq -1)}}{1-\left(S^{m}/S_{u}\right)^{n}} \tag{9}$$

After substituting the values from the equations (7) and (8) into the equation (9) we obtain the equivalent amplitude stress as a function of a maximal stress value from FE analysis for any stress ratio R

$$S^{ae}_{(R=-1)} = \frac{-0{,}5\,S^{\max}_{(R\neq -1)}(R+1) + S^{\max}_{(R\neq -1)}}{1-\left[\frac{1}{2}(R+1)\frac{S^{\max}_{(R\neq -1)}}{S_{u}}\right]^{n}} . \tag{10}$$

When we use Goodman relationship (n = 1) the equation (10) takes more simpler

$$S^{ae}_{(R=-1)} = \frac{S^{\max}_{(R\neq -1)}(1-R)S_{u}}{2S_{u} - S^{\max}_{(R\neq -1)}(R-1)} . \tag{11}$$

Then the value of the equivalent amplitude stress calculated from the equations (10) or (11) can be substituted into the equation (5), where $S^{a}_{(R=-1)} \equiv S^{ae}_{(R=-1)}$, in order to calculate the number of cycles to failure of the analyzed structure loaded by arbitrary non-symmetric load with any stress ratio $R \neq -1$.

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2. Numerical examples. Damage analysis of uniaxially loaded specimen

The following example shows the capability of the algorithm in comparison with some experimental results available [5].

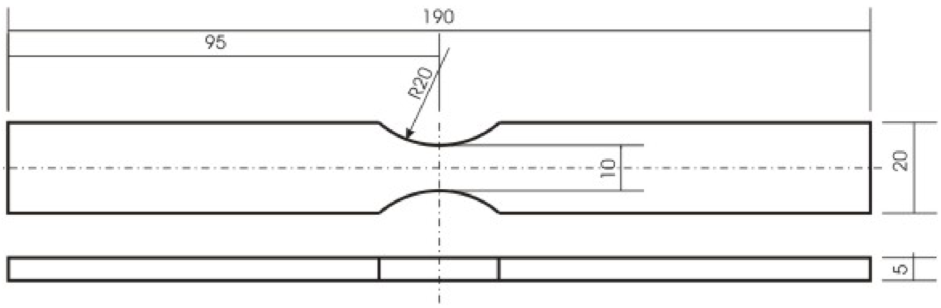

Fig. 2. The considered specimen

Tab. 1. Number of cycles to failure calculated and obtained by experiment [5]

Lp.	S^m [MPa]	S^a [MPa]	R	Number of cycles to faliure (experiment)	Number of cycles to faliure (computation)
1	75	250	-0,538	439300; 402500; no faliure	no failure
2	75	270	-0,565	358200; 854700; 318700	857 294
3	75	290	-0,589	252300; 376300; 379700	340 262
4	75	310	-0,610	54800; 123400; 45000	143 639
5	150	270	-0,286	172100; 121500; 233100	143 963
6	150	290	-0,318	124300; 41900; 60500	57 139
7	225	230	-0,011	413900; 204500; 545200	125 280

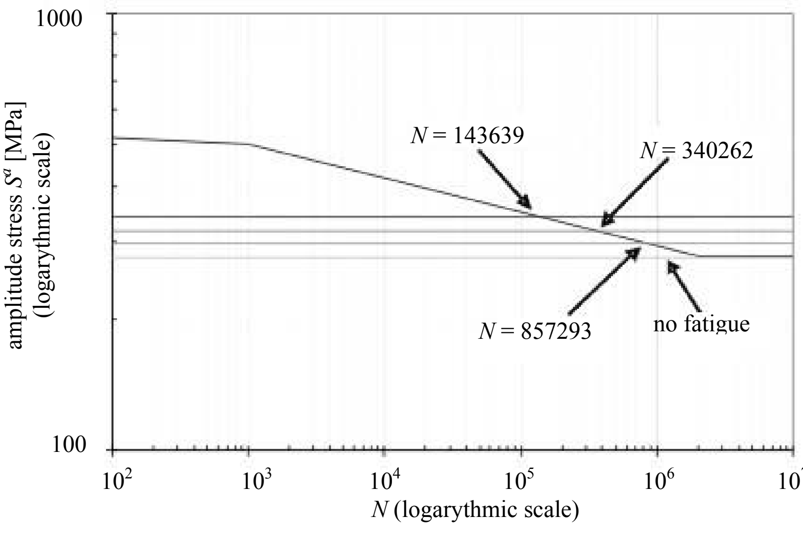

Fig. 3. The S-N curve and equivalent stress amplitudes $S^{ae}_{(R=-1)}$ for S^m = 75 [MPa]

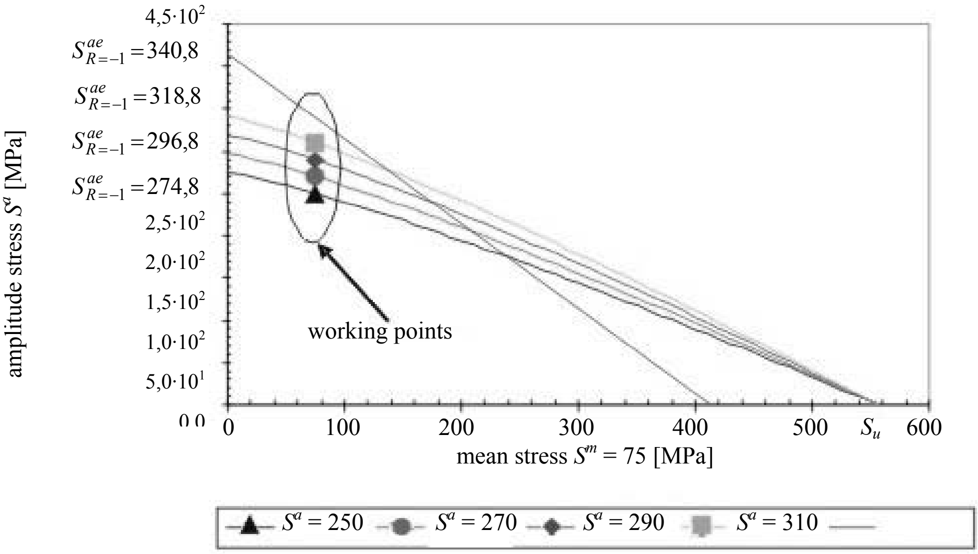

Fig. 4. Goodman diagram for load schemes with S^m = 75 [MPa]

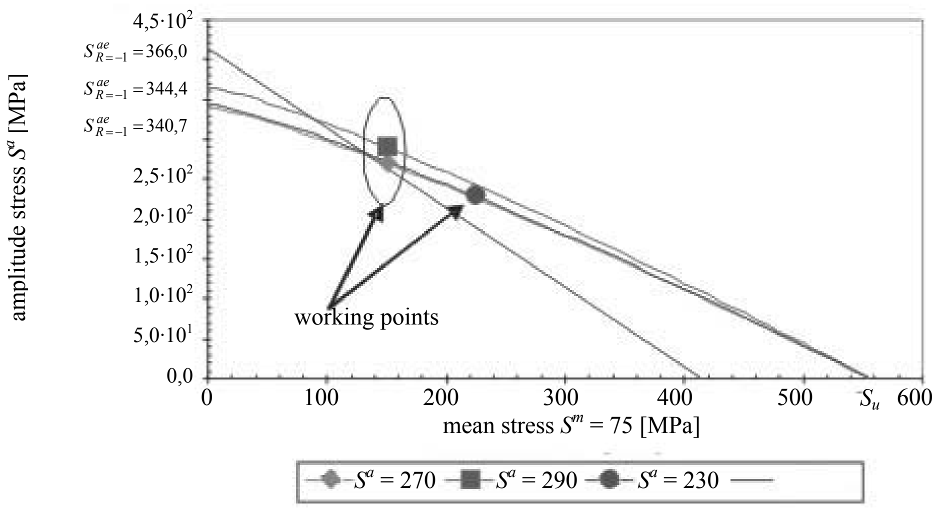

Fig. 5. Goodman diagram for load schemes with S^m = 150 [MPa] and S^m = 225 [MPa]

Uneasily loaded specimen (fig. 2) was analyzed experimentally in [5]. The geometrical siges are shown in fig. 2. The length of the specimen is 0,019 [m]. Material of the specimen is 10HNAP steel, parameters are as follows: density ρ = 7800 [kg/m^3], Young modulus E = 2,10e+11 [Pa], Poisson ratio ν = 0,3, yield stress R_e = 414 [MPa] and ultimate stress S_u = 556 [MPa]. Exponent factor n in the equation (6) is assumed as 1,2. Three different value of mean stress are considered: 75, 150 and 225 [MPa]. In the case of S^m = 75 [MPa] four different stress amplitudes are assumed: S^a = 250, 270, 290 and 310 [MPa], in case of S^m = 150 [MPa] stress amplitudes are S^a = 270 and 290 [MPa] and in the case of S^m = 225 [MPa] stress amplitude equals 230 [MPa] [5].

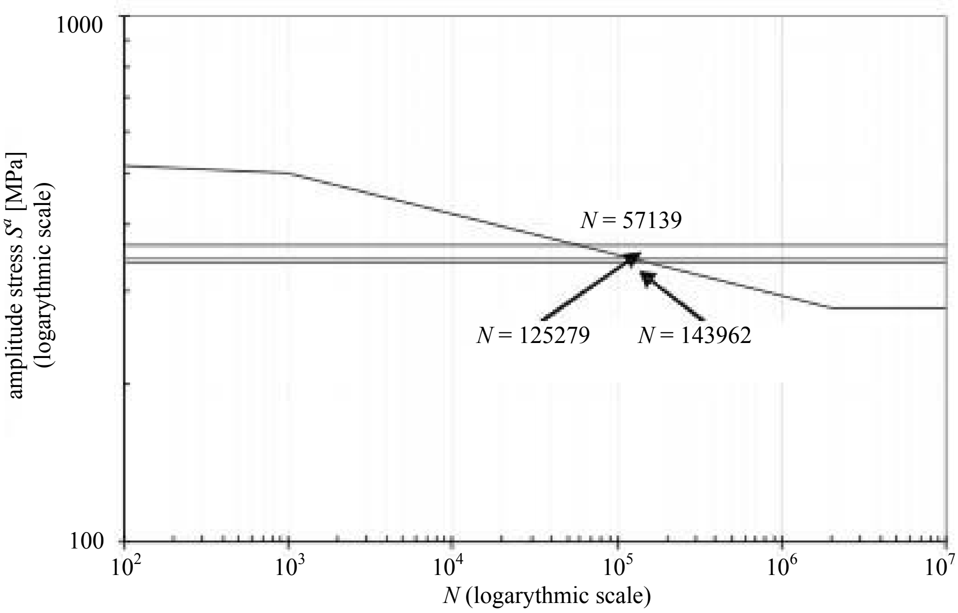

Fig. 6. Wöhler curve and equivalent stress amplitudes $S^{ae}_{(R=-1)}$ for S^m = 150 [MPa] and S^m = 225 [MPa]

The first case: S^m = 75 [MPa].

Four values of the amplitude stress S^a are analyzed: S^a = 250, 270, 290 and 310 [MPa]. Maximal and minimal stresses $(S^{\max}, S^{\min})$ are: (325,–175); (345,–195); (365,–215); (385,–235) respectively. Working points and equivalent amplitude stresses for $R = -1$ are presented in fig. 4. Number of cycles to failure (obtained from the equation (5)) are in table 1 (see fig. 3). Taking into consideration very large range of experimental life of specimen, obtained predicted life of specimens are mostly located in experiment range.

The second case: S^m = 150 [MPa] and S^m = 225 [MPa].

Three values of the amplitude stress S^a are analyzed: S^a = 270 and 290 [MPa] which corresponds with S^m = 150 [MPa] and S^a = 230 [MPa] as well as with S^m = 225 [MPa]. Maximal and minimal stresses $(S^{\max}, S^{\min})$ are (420,–120); (440,–140); (455,–5) respectively. Working points and equivalent amplitude stresses for $R = -1$ are presented in fig. 5. Number of cycles to failure (obtained from the equation (5)) are in table 1 (see fig. 6). In this case range of experimental data is very large.

The calculated values of number of cycles to failure are close to experimental data, except $S^m = 225$ [MPa]. Here a larger value of exponent n in the equation (6) might be used.

Fig. **7.** Geometrical shape of the cylinder 1

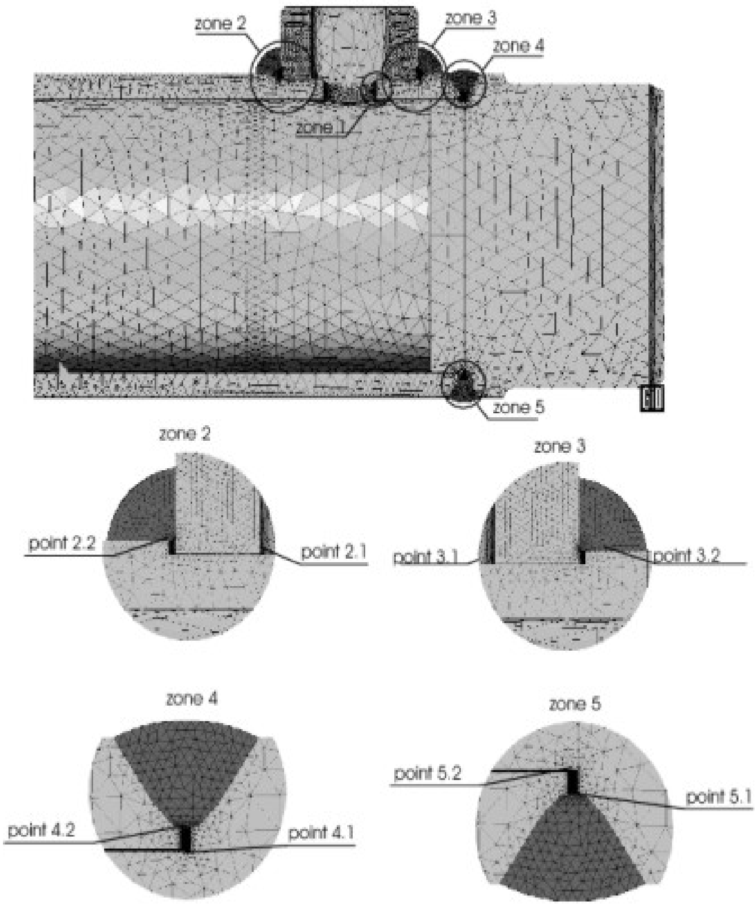

Fig. 8. Critical zones and critical points in the cylinder 1

3. Hydraulic cylinder design, oil ports fatigue and possible design modifications

3.1. Fatigue analysis of the hydraulic cylinder 1. The cylinder shown in fig. 7 was tested.

The cylinder has two oil ports. The material of the specimen is steel St52. The material data are: density $\rho = 7800$ [kg/m3], Young modulus $E = 2{,}10e{+}11$ [Pa], Poisson ratio $\nu = 0{,}3$, initial flow stress $Re = 350$ [MPa], saturation flow stress $S_u = 520$ [MPa]. Inner pressure is $30 \pm 0{,}08$ [MPa]. Stress ratio $R = p_{\min} / p_{\max}$ equals 0.

Initial and threshold amplitude stress for steel St52 and $R = -1$ equals: $S^{at}_{(R=-1)} = 0{,}9 S_u = 468\,[\text{MPa}]$ and $S^{at}_{(R=-1)} = 0{,}45 S_u = 234\,[\text{MPa}]$.

Five critical zones determined in the numerical experiment are shown in fig. 8. Number of cycles to failure (see fig. 9) is obtained as intersection of adopted to fully symmetrical load scheme $(R = -1)$ stress amplitude with classical Wöhler curve. The values of maximal, amplitude and mean stress in each of the zones and number of cycles to failure are shown in table 2.

Tab. 2. Number of cycles to failure in critical zones of the cylinder 1

	zone 1	zone 2	zone 3	zone 4	zone 5
$S^{\max}{}_{(R=0)}$ [Mpa]	399,13	367	352,10	357,38	376,76
$S^{a}{}_{(R=0)}$ [Mpa]	199,57	183,50	176,05	178,69	188,38
$S^{m}{}_{(R=0)}$ [Mpa]	199,57	183,50	176,05	178,69	188,38
$S^{ae}{}_{(R=-1)}$ [Mpa]	323,85	283,57	266,16	272,24	295,39
number of cycles to failure	$5{,}67 \cdot 10^4$	$2{,}43 \cdot 10^5$	$4{,}8 \cdot 10^5$	$3{,}8 \cdot 10^5$	$1{,}55 \cdot 10^5$

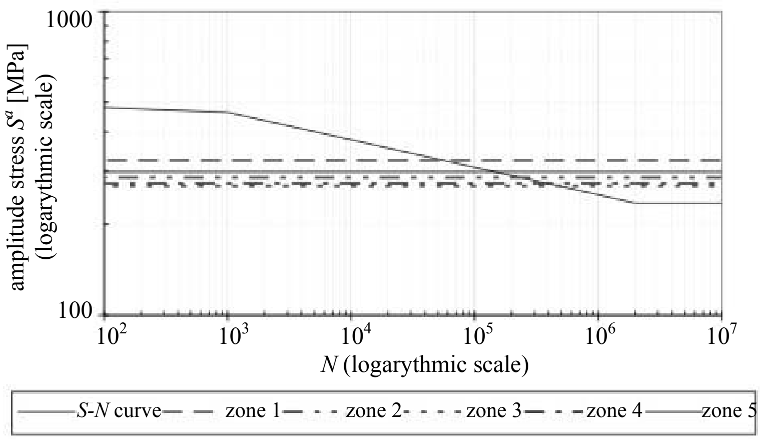

Fig. 9. Behavior of the Wöhler curve and values of amplitude stress S^{ae} in critical zones for the cylinder 1

The largest value of stress appears in the zone 1, then next in the zone 5 (point 5.1) and successively in the zone 2 (point 2.1), and in the zones 4 and 3 (points 4.1 and 3.1) (see fig. 8). Location, where fatigue crack appears, first are points 5.1 and 2.1. That fact was confirmed in real experiment where the weakest point was in the zone 5.

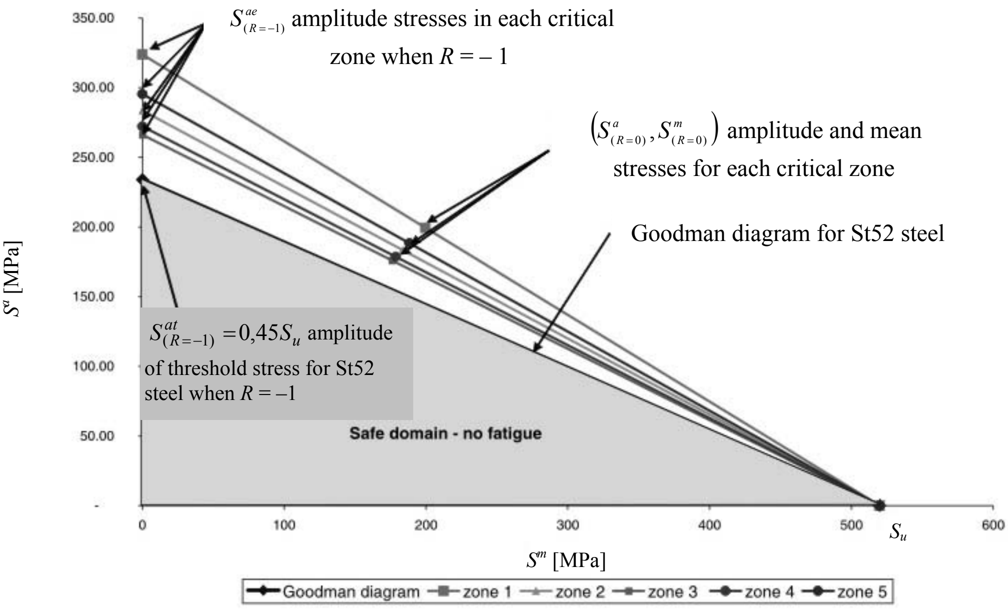

Fig. 10. Goodman diagram and working points for cylinder 1
for different critical zones

Number of cycles to failure in the zone 5 (see fig. 8) equals $N_{zone5} = 155416$. The difference between results obtained by experiment and numerical test do not exceed 9%.

3.2. Fatigue analysis of the **hydraulic cylinder 2.** The part of hydraulic cylinder is analyzed. The geometrical shape is shown in fig. 11. The cylinder has two oil ports.

The material of the cylinder is steel St52. Material data are the same as in previous example. Inner pressure equals 10 [MPa]. Stress ratio R equals 0. The initial and threshold stress for steel St52 are taken from previous example.

Deformations and stress redistribution in the connection zone caused by oil penetration and shear forces inside anoil tube are shown in fig. 12.

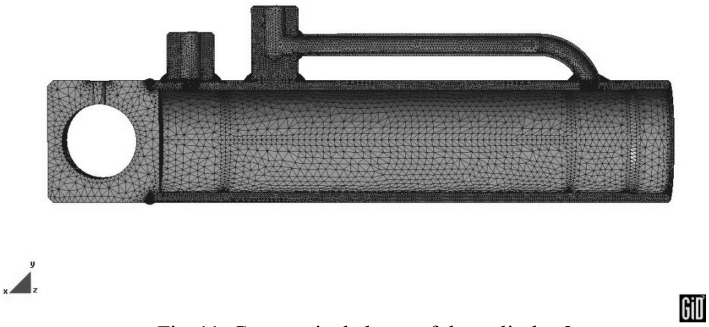

Fig. 11. Geometrical shape of the cylinder 2

Tab. 3. Number of cycles to failure in critical zones of the cylinder 2 calculated using FEM

	zone 1	zone 2	zone 3	zone 4	zone 5
$S^{\max}_{(R=0)}$ [MPa]	350,00	219,00	255,00	130,00	352,00
$S^{a}_{(R=0)}$ [MPa]	175,19	109,34	127,25	65,25	175,84
$S^{m}_{(R=0)}$ [MPa]	175,19	109,34	127,25	65,25	175,84
$S^{ae}_{(R=-1)}$ [MPa]	254,20	138,45	168,48	74,61	265,68
number of cycles to failure	$5{,}28 \cdot 10^5$	no fatigue	no fatigue	no fatigue	$4{,}97 \cdot 10^5$

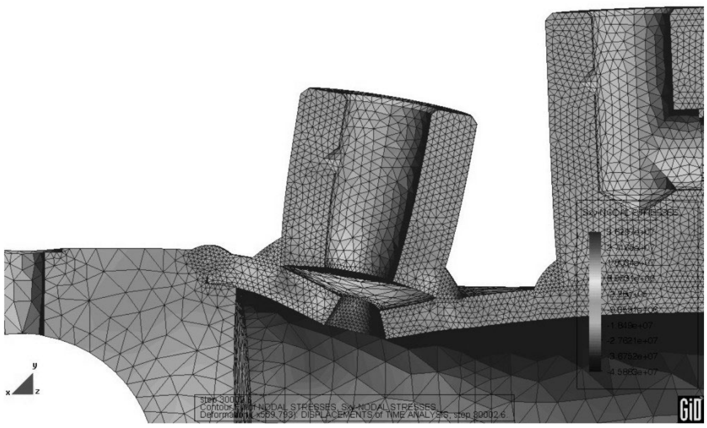

Fig. 12. Deformations and stress redistribution in connection zone caused by oil penetration and shear forces inside oil tube of the cylinder 2

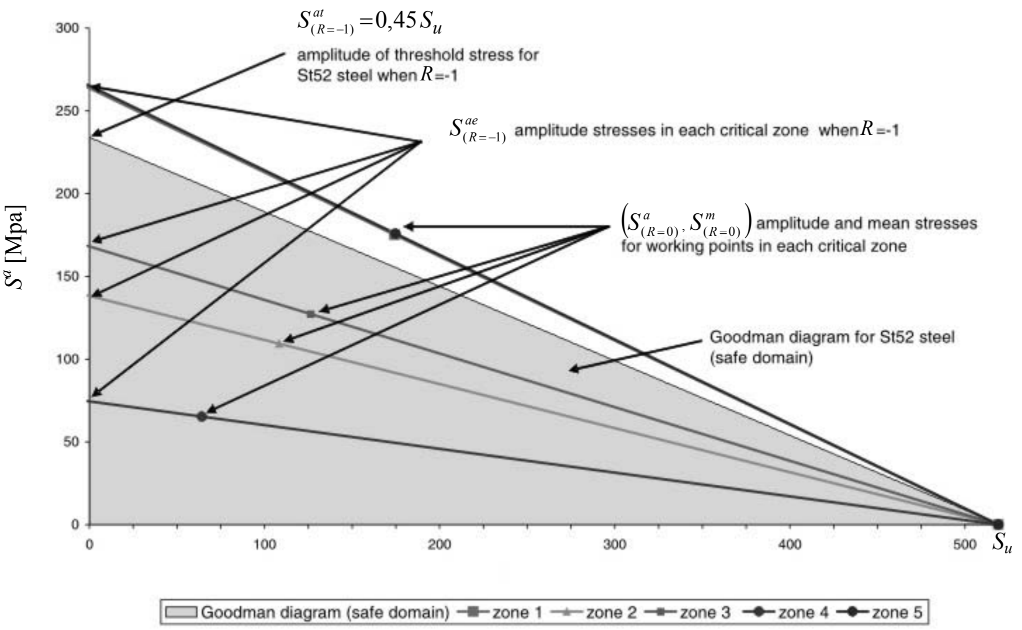

Fig. 13. Goodman diagram and working points in critical zones for the cylinder 2

The values of the maximal stresses, amplitude and mean stresses for $R = 0$ and $R = -1$ and number cycles to failure calculated using FEM in each zones are shown in table 3. The values of the maximal stresses in the zone 1 and zone 5 are almost the same.

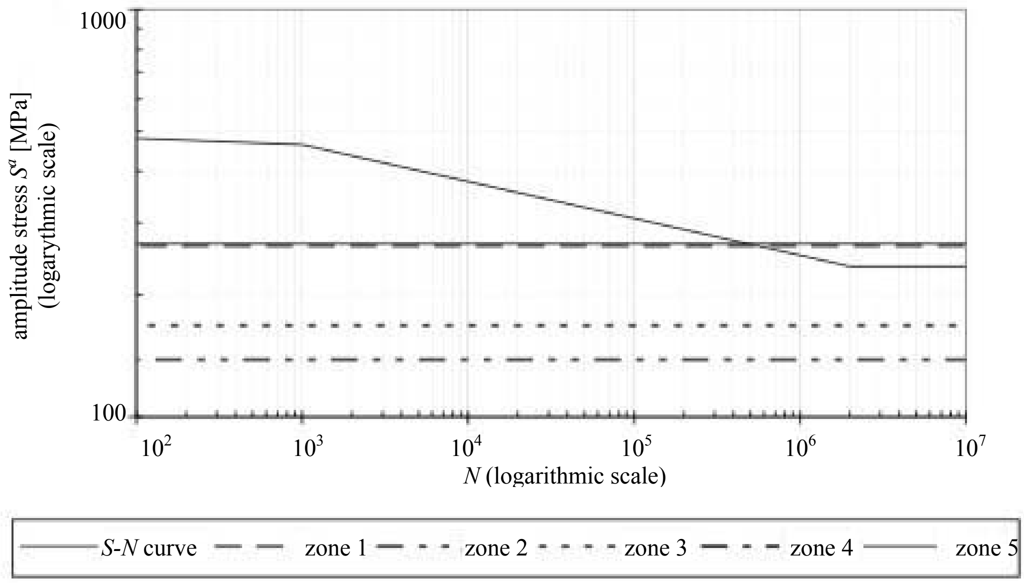

Fig. 14. Behavior of the Wöhler curve and values of equivalent amplitude stress S^{ae} in critical zones for the cylinder 2

These two working points (zone 1 and zone 5) are located above the Goodman diagram, other points are lying below the Goodman diagram in the safe area (see fig. 13). Fatigue crack appears in the zones 1 and 5. Cracks do not appear in other zones. Number of cycles to failure in the zone 1 and zone 5 obtained as intersection of the equivalent amplitude stress for $R = -1$ with Wöhler curve is shown in table 3 and in fig. 14.

3.3. Fatigue analysis of the hydraulic cylinder 1 with resistant material in oil ports and cup. The part of hydraulic cylinder is analyzed. The geometrical shape and material data are the same as in the example from subsection 4.1. Authors propose modification of oil ports area by introducing the washer or glue as it is shown in fig. 15. The cylinder has two oil ports with washers.

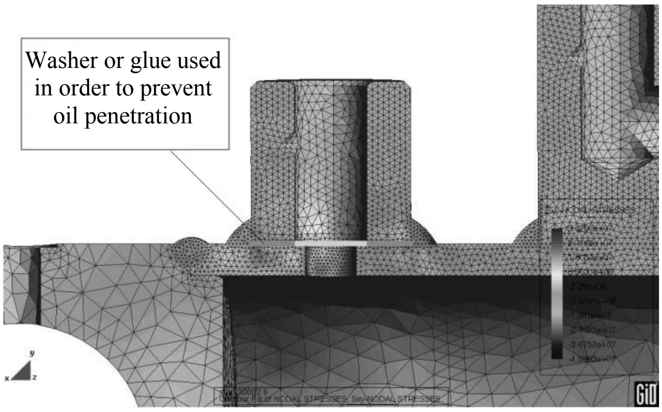

Fig. 15. Washer or used in order to prevent oil penetration

Tab. 4. Number of cycles to failure in critical zones of the cylinder 1 with washer

	zone 1	zone 2	zone 3	zone 4	zone 5
$S^{\max}_{(R=0)}$ [MPa]	398,17	365,66	360,42	359,14	357,11
$S^{a}_{(R=0)}$ [MPa]	199,08	182,83	180,21	179,57	178,55
$S^{m}_{(R=0)}$ [MPa]	199,08	182,83	180,21	179,57	178,55
$S^{ae}_{(R=-1)}$ [MPa]	322,59	281,97	275,78	274,29	271,93
number of cycles to failure	$5{,}92\cdot 10^4$	$2{,}59\cdot 10^5$	$3{,}30\cdot 10^5$	$3{,}50\cdot 10^5$	$3{,}85\cdot 10^5$

Number of cycles to failure in each zones obtained as intersection of the amplitude stress with Wöhler curve is shown in table 4.

Deformations and stress redistribution in the connection zone are illustrated in fig. 16.

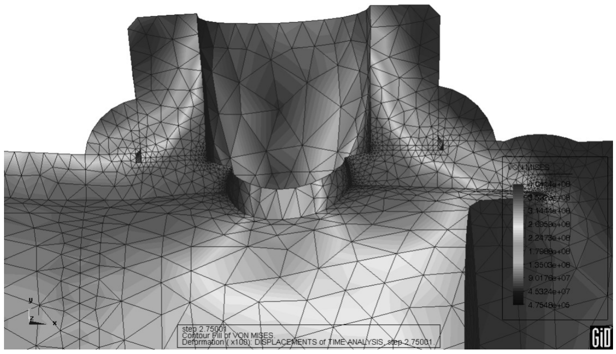

Fig. 16. Deformations and stress redistribution in the connection zone with washer used in order to prevent oil penetration in the cylinder 1

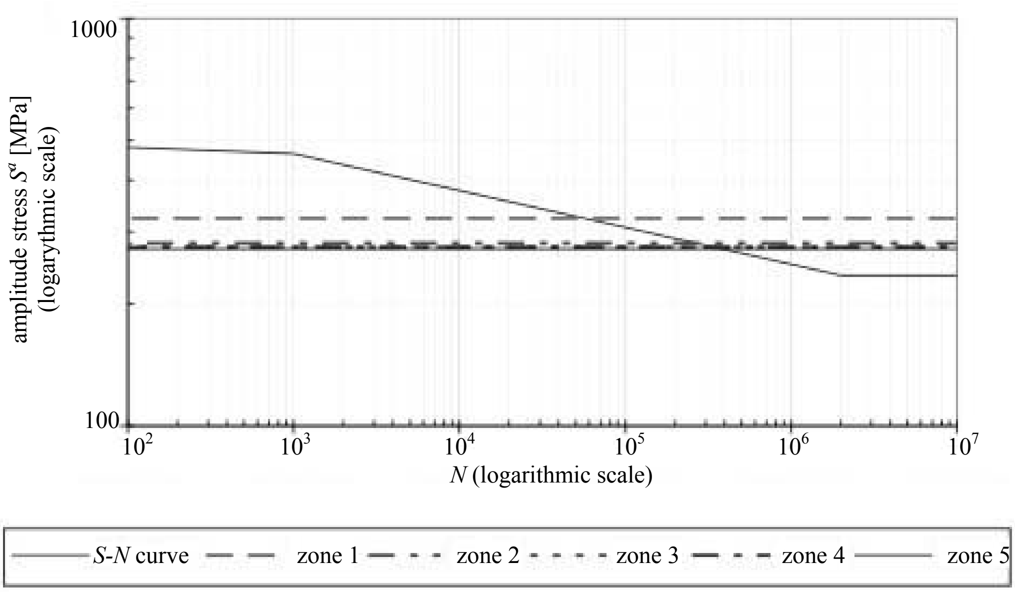

Fig. 17. Behavior of the Wöhler curve and values of amplitude stress S^{ae} in the critical zones for the cylinder 1 with resistant material in the port zone

Behavior of the Wöhler curve and values of the equivalent amplitude stress $S^{ae}{}_{(R=-1)}$ in the critical zones are shown in fig. 17. Significant improvement is archived in the zone 5 (the most endanger to fatigue zone) where fatigue strength increased and number of cycles was bigger then about 100000 cycles. In the zones 2, 3 and 4 there was a slight improvement in the value of fatigue strength.

The fatigue in the zone 2 appears later if we use washer between the oil port and cylinder (see fig. 15). The value of maximal stress is lower then in the case when there is no modification between the oil port and cylinder surface.

Conclusions. The idea of equivalent amplitude stress is introduced in order to find out number of cycles to failure for non-symmetric loads in situations when material data are provided only for stress ratio $R = -1$.

In order to solve the crack problem in an oil port area we propose to use the washer made from temperature resistant material or glue in order to fill up the gap between oil port — cylinder surface (see fig. 15). Such washer or glue will prevent oil penetration into the above mentioned gap thus eliminating the possibility of promotive fatigue crack in neighboring welds.

Life expectancy are values of the moment absolutely theoretical and pending of the adjustment of the model and experimental validation.

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Использование эквивалентных амплитудных напряжений для учета средних напряжений в задачах усталостной прочности

Томаш Беднарек, Изабелла Марчевска, Артур Марчевски, Влодзимеж Сосновски, Хероним Якубчак, Ежы Роек

В работе представлен численный алгоритм, позволяющий выбирать различные схемы нагружения конструкции, например, гидроциллиндра, в соответствии с заданными кривыми Вёлера, характеризующими сопротивление материала усталостному разрушению. Проанализированы гидроциллиндры, рассматриваемые в проекте E. C. «PROHIPP», и предложены *некоторые решения задачи о трещине в области штуцера. Показано, что просачивание масла в зоне соединения штуцера может быть устранено после некоторых модификаций конструкции.*

Використання еквівалентних амплітудних напружень для врахування середніх напружень у задачах на втомну міцність

Томаш Беднарек, Ізабела Марчевська, Артур Марчевські, Влодзімеж Сосновскі, Гєронім Якубчак, Єжи Роєк

У роботі представлено числовий алгоритм, який дозволяє вибирати різні схеми навантаження конструкції, наприклад, гідроциліндра, відповідно до заданих кривих Велера, що характеризують опір матеріалу на втомне руйнування. Проаналізовано гідроциліндри, які розглянуті в проекті Е. С. «PROHIPP», і запропоновано деякі розв'язки задачі про тріщину в області штуцера. Показано, що просочування олії в зоні з'єднання штуцера може бути усунуто після деяких модифікацій конструкції.

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