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# "A NOVEL PUSH-PULL MACHINE FOR TESTING HIGH CYCLE FATIGUE"

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## ABSTRACT

A novel machine designed and constructed to exert cyclic axial load for fatigue testing is herein reported. The main features of the test rig constitute the sinusoidal form of load wave and the stress - controlled system of loading.

Performance parameters comprise both static and alternating components of load as well as the frequency of load application. Typical results of tests run on steel C 60 and brass Ms 58 are presented.

The experimentally determined endurance values, show consistency and good agreement with published data, thus, inspiring confidence in the test rig.

The spectrum of material strength as influenced by cyclic loading, is herein displayed for the full range of the so-called "Dynamic Severity Criterion" first put forward by the author.

## NOTATION

**a,b** : Material-dependent Constants of the Stress Equation: $\sigma = b.N^{-a}$

**$N_f$** : Fatigue Life or Total Number of Load Cycles to Failure.

**R** : $\sigma_{min}/\sigma_{max} = (1-2\eta)$, sometimes called "Asymmetry Number".

**$S_n$** : Endurance Limit.

**$S_{np}$** : Endurance Limit under Pulsating (repeated) Load.

$S_{nr}$ : Endurance Limit under Fully Reversing (alternating) Load.
$S_u$ : Ultimate Static Strength of the Material under Axial Load ($=\sigma_B$ in German Standard Specifications).
$S_y$ : Yield Strength of the Material under Axial Load.
$\eta$ : Dynamic Severity Criterion $=\sigma_a/\sigma_{max} = (1-R)/2$.
$\sigma$ : Normal or Direct Stress.
$\sigma_a$ : Amplitude of Stress Wave.
$\sigma_m$ : Mean Stress in the Cycle $= (\sigma_{max} + \sigma_{min})/2$.
$\sigma_{max}$ : Maximum Stress in the Cycle $= (\sigma_m + \sigma_a)$.
$\sigma_{min}$ : Minimum Stress in the Cycle $= (\sigma_m - \sigma_a)$.

## 1. INTRODUCTION

Since the classical works of Wöhler, Bauschinger and others towards the turn of the last century [1-7], failure of metals under cyclic loading received ever increasing attention, e.g. [8-16]. Several specialized reference books and publications on fatigue also appeared in the last two decades [17-23]. These extensive investigations were conducted on a variety of test rigs, the particulars of which are critically reviewed in a separate paper [24].

The relationship between applied stress $\sigma$ and life in terms of the number of cycles N was first formulated by Basquin [6] who named it: "The Exponential Law of Endurance Tests" or more briefly: "The Power Function". This can be represented mathematically in the form:

$$\sigma = b.N^{-a} \tag{1}$$

in which a and b are constants which depend on the material.

The fatigue limit is defined as the stress at which the material can withstand an infinite number of load reversals without failure; it is herein denoted by $S_{nr}$.

In the long series of experiments on the fatigue behaviour of engineering metals, the present study is devoted to the investigation of the effect of cyclic axial loading on the strength of steel and brass specimens. Tests were so planned and conducted as to locate the fracture line for the full range of the dynamic severity criterion $\eta = \sigma_a / \sigma_{max}$ as put forward by the author [25-28].

## 2. SCOPE OF WORK

A push-pull fatigue testing machine is herein designed and constrcuted for determining the strength of materials subjected to fluctuating axial load, this being composed, in general, of an alternating component superposed on a steady load component. Maximum and minimum resulting stresses can thus be represented by the simple equation:

$$\sigma_{\substack{max \\ min}} = ( \sigma_m \pm \sigma_a) \qquad (2)$$

The novel machine is made to possess the following basic features:

1. The machine has a "stress-controlled" loading system, i.e. it operates with constant stress amplitude. The axial load is applied to the test specimen by means of a helical compression spring actuated by a rotating cam, Figure (1).

2. The variation of the rate of load application (i.e. load frequency) is obtained through the inclusion of a gear box in the driving system.

3. The test rig is provided with a device for applying the static component of load in either direction (i.e. tension or compression). This device caters for the steady or mean stress component of load $\sigma_m$.

4. The loading system is made with a natural frequency much higher than that of the machine. This is accomplished by the use of a heavy spring together with a light leverage system.

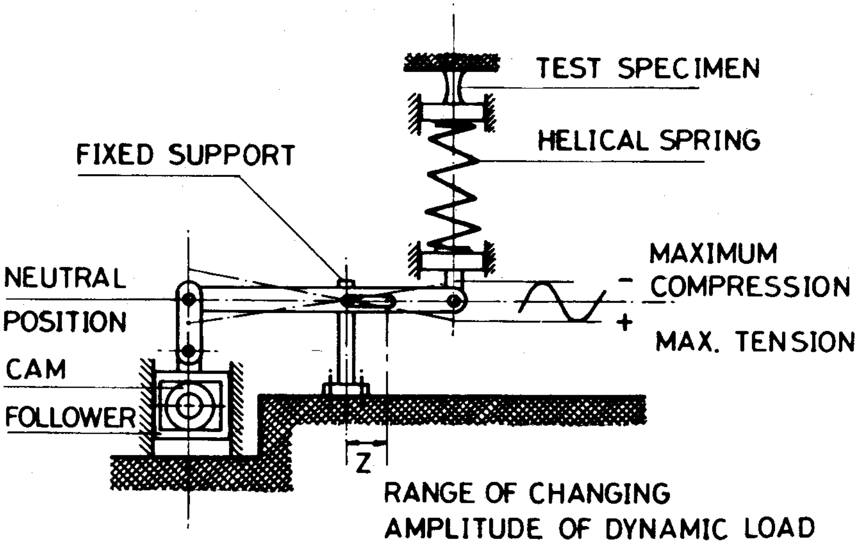


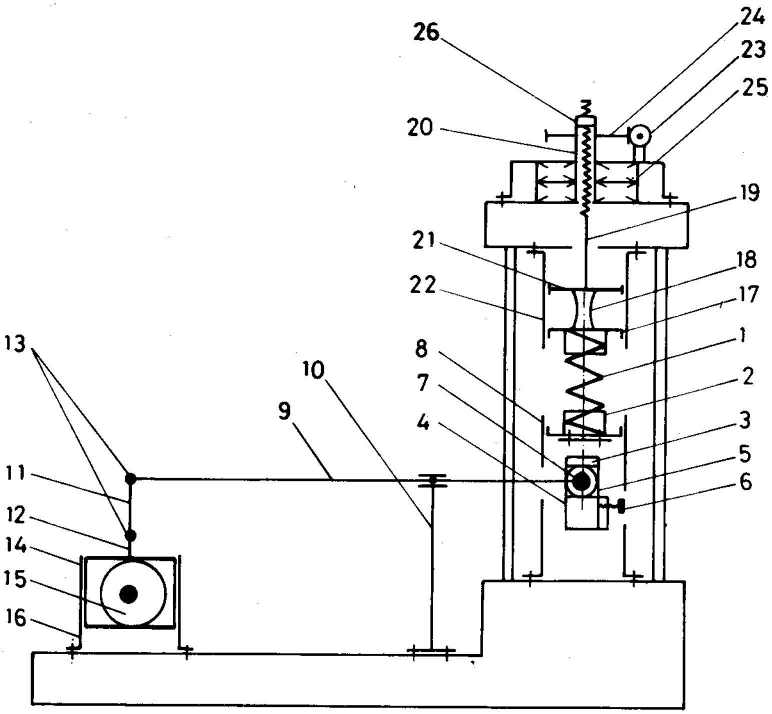
Figure 1: Conceptual Arrangement of Loading System.

## 3. DESCRIPTION OF TESTING MACHINE

A schematic layout of the machine is shown in Figure (2) and a constructional drawing of the complete rig in Figure (3).

The heavy helical spring (1), Fig. (2), has its driving end coupled to a totally encased lower seat (2), this being tied directly to the loading joint (3) which is rectangular in shape. A tapered liner (4) is used to grip the spherical bearing (5) so as to take up any misalignment by means of an adjusting screw (6). An axle (7) keeps the bearing on the central plane of the loading system. The lower seat (2) is guided by the cylindrical guide (8).

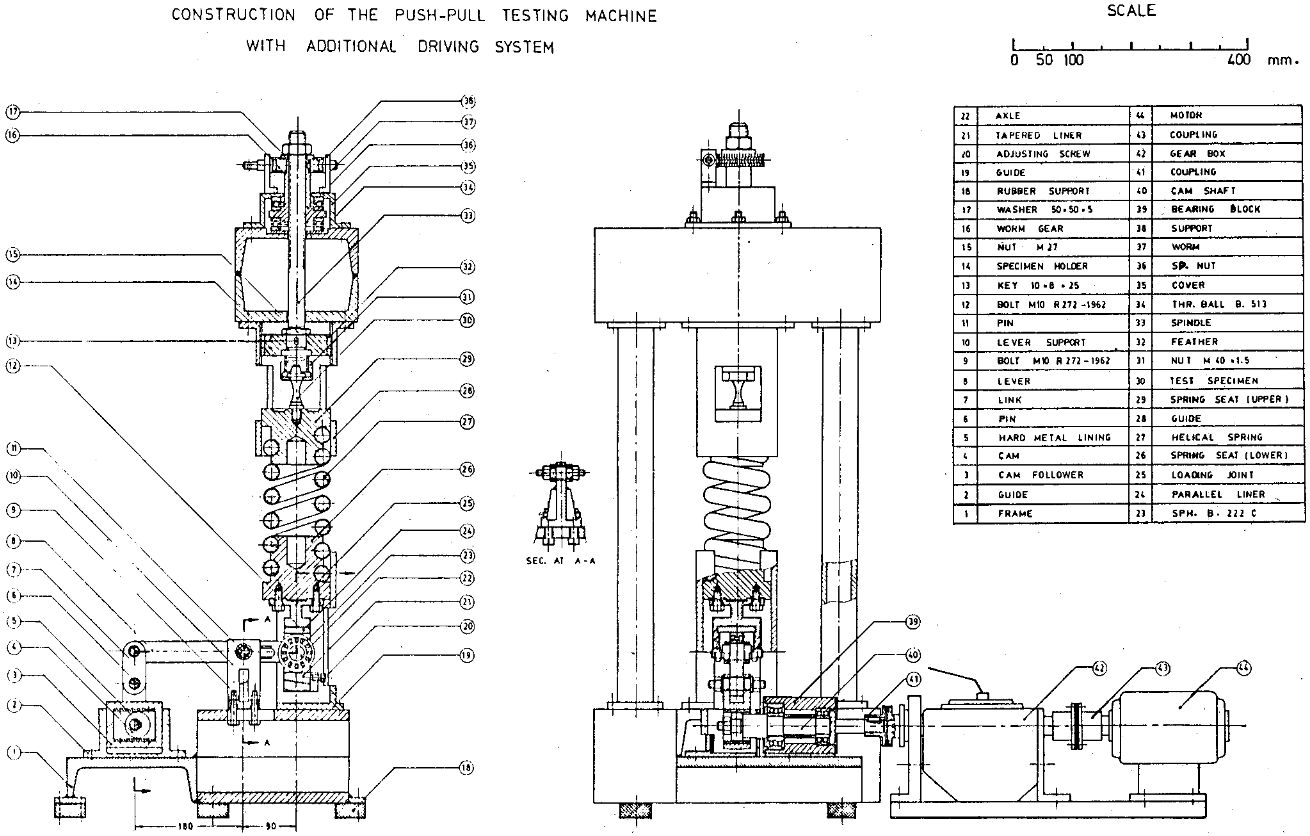
Two horizontal and parallel levers (9), pivoted at the lower support (10), enable the leverage ratio of the loading system to be varied, thus changing the magnitude of the variable stress component $\sigma_a$. The levers (9) are connected to the vertical link (11) and the follower (12) through two pivoted pins (13). The follower is provided with two hard metal linings (14) which come in contact lwith the cam (15). The cam follower assumes the form of a box travelling within the guide (16) so that contact between cam and follower is maintained at all times. The cam surface is heat treated for increased hardness.



1. Helical Spring (for dynamic loading)
2. Lower Seat for Loading Spring
3. Loading Joint
4. Tapered Liner
5. Spherical Bearing
6. Adjusting Screw
7. Axle
8. Cylindrical Guide
9. Horizontal Levers
10. Lever Support
11. Vertical Link
12. Cam Follower
13. Pivoted Pins
14. Hard Metal Linings
15. Cam (Eccentric)
16. Follower Guide
17. Upper Seat for Loading Spring
18. Test Specimen
19. Spindle
20. Special Nut
21. Specimen Holder
22. Cylindrical Guide
23. Worm
24. Worm Wheel
25. Thrust Bearings
26. Locking Nut

**Figure 2: Schematic Layout of Push-Pull Fatigue Testing Machine.**

On the other hand, the upper end of the spring is mounted in a totally encased upper seat (17) to which the lower threaded end of the test specimen (18) is screwed. The upper conical end of the specimen is rapidly fixed to the spindle (19) by means of a special nut (20). Both specimen holder (21) and upper seat (17) are guided by the cylindrical guide (22) so as to ensure axiality of load application.



| | | | |
|---|---|---|---|
| 22 | AXLE | 44 | MOTOR |
| 21 | TAPERED LINER | 43 | COUPLING |
| 20 | ADJUSTING SCREW | 42 | GEAR BOX |
| 19 | GUIDE | 41 | COUPLING |
| 18 | RUBBER SUPPORT | 40 | CAM SHAFT |
| 17 | WASHER 50×50×5 | 39 | BEARING BLOCK |
| 16 | WORM GEAR | 38 | SUPPORT |
| 15 | NUT M 27 | 37 | WORM |
| 14 | SPECIMEN HOLDER | 36 | SP. NUT |
| 13 | KEY 10×8×25 | 35 | COVER |
| 12 | BOLT M10 R 272-1962 | 34 | THR. BALL B. 513 |
| 11 | PIN | 33 | SPINDLE |
| 10 | LEVER SUPPORT | 32 | FEATHER |
| 9 | BOLT M10 R 272-1962 | 31 | NUT M 40×1.5 |
| 8 | LEVER | 30 | TEST SPECIMEN |
| 7 | LINK | 29 | SPRING SEAT (UPPER) |
| 6 | PIN | 28 | GUIDE |
| 5 | HARD METAL LINING | 27 | HELICAL SPRING |
| 4 | CAM | 26 | SPRING SEAT (LOWER) |
| 3 | CAM FOLLOWER | 25 | LOADING JOINT |
| 2 | GUIDE | 24 | PARALLEL LINER |
| 1 | FRAME | 23 | SPH. B. 222 C |

**Figure 3: Construction of the Push-Pull Testing Machine with Additional Driving System.**

**The static component of load is applied through a worm (23) and a worm wheel (24), these being operated manually to the vertical displacement of the power screw (19).**

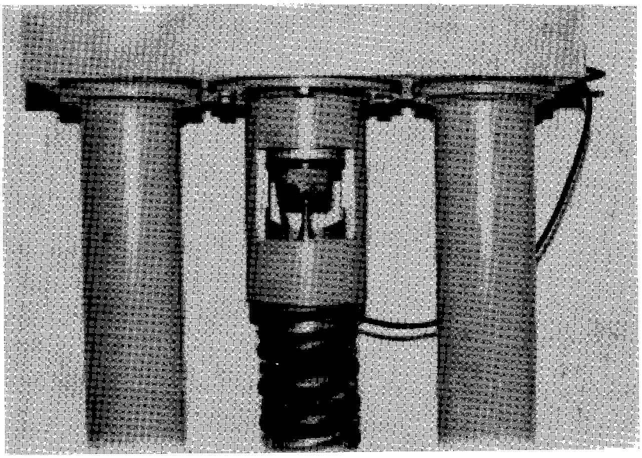


**Figure 4: Specimen Mounted in Test Rig.**

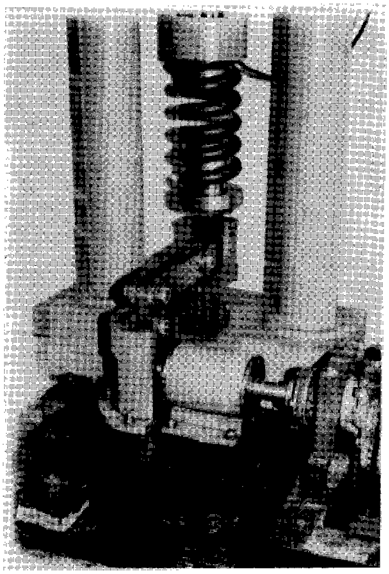
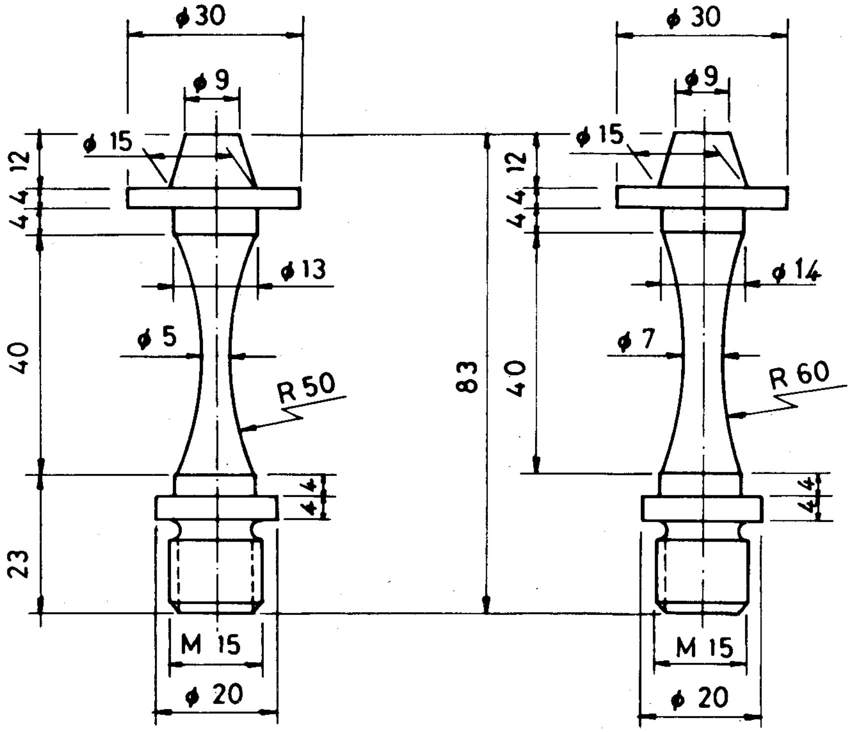


**Figure 5: General View of Loading System.**

The spindle (19) has a square power thread (with a small pitch of 4 mm) which can be operated by the special nut (20). Both static and alternating load components are carried by two thrust bearings (25). A locking nut (26) is used to prevent any backlash between the power screw (19) and its nut (20).

While Figure (4) shows the test specimen mounted in position ready for testing, Figure (5) displays a general view of the loading system. Specimens used for fatigue testing are shown in Figure (6).



STEEL SPECIMEN

BRASS SPECIMEN

Figure 6: Test Specimens for Steel (C60) and Brass (Ms 58).

The main particulars of the machine are given hereunder.

## 4. TECHNICAL SPECIFICATIONS OF TESTING MACHINE

| | | |
|---|---|---|
| Maximum Load of Spring | : | 2.4 kN |
| Corresponding Spring Deflection | : | 2.5 mm |
| Maximum Static Load Component | : | 1 kN |
| Height of Machine Columns | : | 550 mm |
| Total Height of Loading Lever | : | 270 mm |
| Distance from Lever Pivot to Cetre Line of Cam | : | 180 mm |
| Leverage Ratio of Loading System (Normal Position) | : | 1:2 |
| Eccentricity of Cam | : | 5 mm |
| Speed Reduction Ratios of Gear Box | : | 1, 1.8, 3 & 4 |
| Rated Speed of Driving Motor | : | 2900 r.p.m. |
| Rotational Speeds of Cam | : | 2900 r.p.m.<br>1611 r.p.m.<br>967 r.p.m.<br>725 r.p.m. |
| Power Rating of Driving Motor | : | 3.8 kW |
| Reduction Ratio of Worm Drive | : | 1:31 |

## 5. INSTRUMENTATION AND MEASUREMENTS

The operating variables measured in the present investigations include the following:

1. Static component of load,
2. Alternating component of load,
3. Frequency of load appliction,
4. Number of load cycles to failure.

While applied loads were measured by means of highly sensitive strain gauges, the power screw was calibrated so as to determine its load-deflection characteristic under static load.

The frequency of load application was obtained by knowledge of motor speed and gear ratio used. The number of load cycles to failure was determined by a special cycle counter.

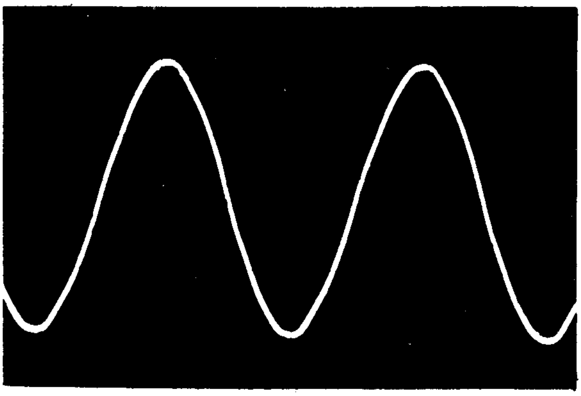
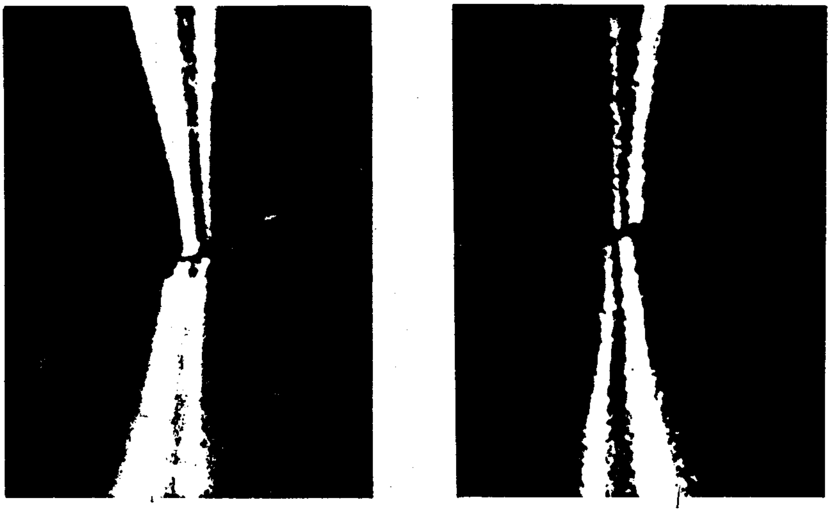


**Figure 7: Typical Recording of Axial Load Wave.**

Figure (7) displays a typical recording of load variation, this approaching a sinusoidal wave form.

Figure (8) exhibits fatigue fractures for steel and brass specimens.



Steel C60
$N_f = 1.2 \times 10^6$

Brass SCB. 4-C
$N_f = 8 \times 10^7$

**Figure 8: Typical Fatigue Fracture of Steel and Brass Specimens.**

## 6. MECHANICAL PROPERTIES OF TESTED MATERIALS

| **Material Property** | **Steel C60(1)** | **Brass Ms 58(2)** |
|---|---|---|
| | MPa | MPa |
| Ultimate Tensile Strength $S_u$ | 750 | 380 |
| Yield Strength $S_y$ | 480 | — |
| Endurance Limit for Repeated Load $S_{np}$ | 430 | 280 |
| Endurance Limit for Fully Reversing Load $S_{nr}$ | 270 | 190 |
| Brinell Hardness Number HB 30 | 238 | 89 |
| Elongation $\delta_5$% | 14 | 35 |

(1) According to DIN 17006 (2) According to DIN 1709

## 7. RESULTS AND CONCLUSIONS

Typical results as obtained under fully reversing conditions of loading, at a frequency of 12 Hz, are reproduced in Figures (9) and (10) in the form of the well known S-N (or $\sigma$-N) curves. For both steel and brass specimens, test points herein recorded represent average values of at least two test runs under same conditions.

The so-called S-N or Wöhler's [1] curve can be mathematically represented in an exponential form as first proposed by Basquin [6], Equation (1).

Test results herein obtained, Figures (9) and (10), show that for steel C60:

$$\sigma = 700\, N^{-0.067} \text{ MPa} \qquad (3)$$

and for brass Ms 58:

$$\sigma = 366\, N^{-0.0063} \text{ MPa} \qquad (4)$$

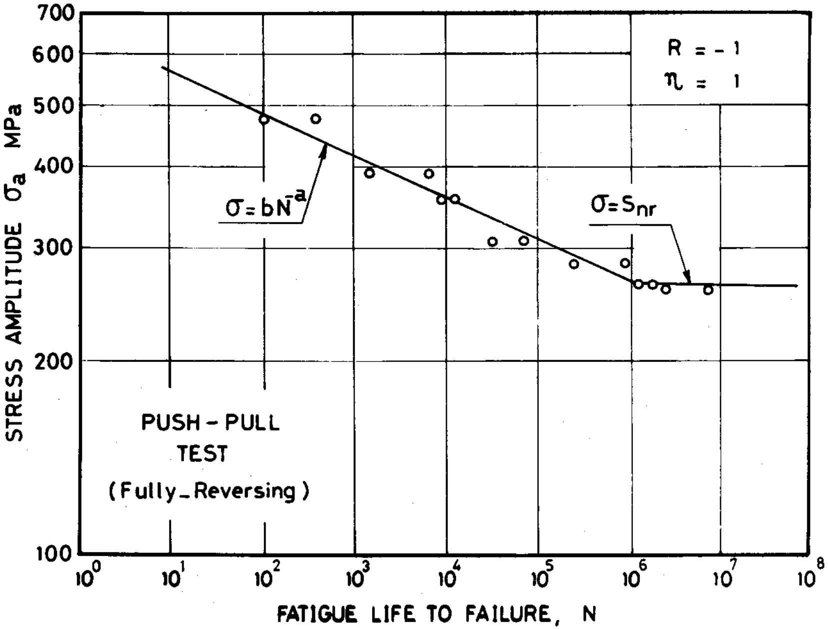


**Figure 9: Fatigue Test Data for Steel C 60 (DIN 17200) Under Stress-Controlled Conditions.**

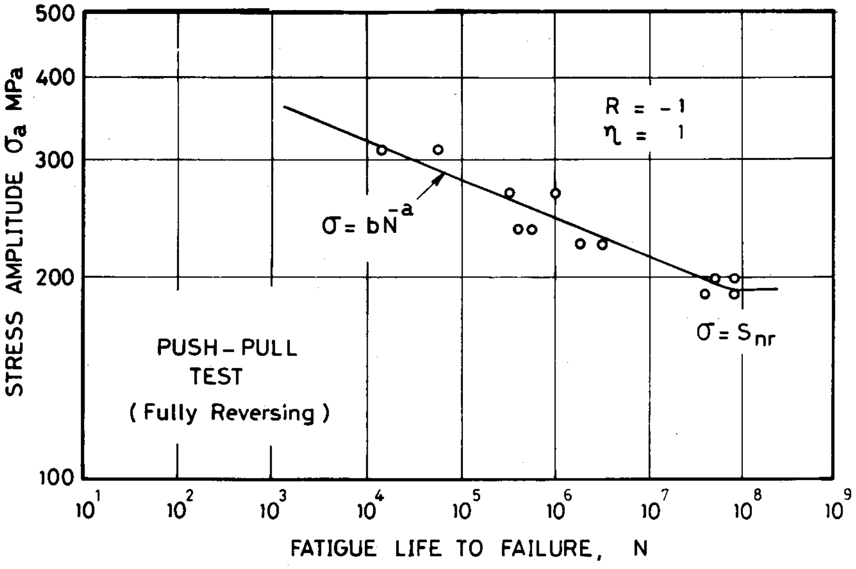


**Figure 10: Fatigue test Data for Brass Ms 58 (DIN 1709) under Stress-Controlled Conditions.**

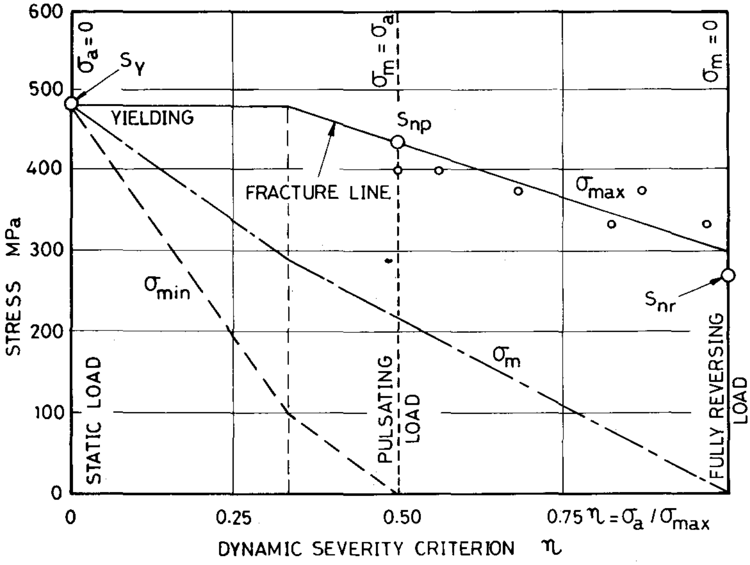


**Figure 11: Experimental Results Interpreted in Terms of the Dynamic Severity Criterion $\eta$ (Material : Steel C 60).**

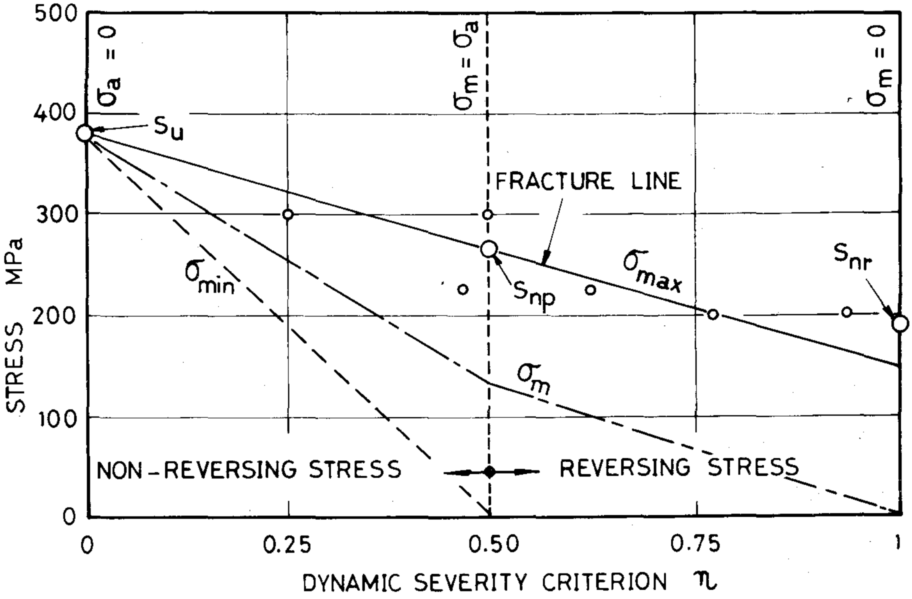


**Figure 12: Experimental Results Interpreted in Terms of the Dynamic Severity Criterion $\eta$ (Material : Brass Ms 58).**

Should test results obtained under conditions of sinusoidally fluctuating loads be represented in terms of the "Dynamic Severity Criterion" $\eta$ as put forward by the author [25-28], Figures (11) and (12) would evolve. It is worth mentioning that the $\sigma$ - $\eta$ diagram gives a clear picture of how the strength of the material varies from $S_u$ at $\eta = \sigma_a / \sigma_{max} = 0$, to $S_{np}$ at $\eta = 0.5$ ($\sigma_{max}=2\sigma_a$), and finally to $S_{nr}$ at $\eta = 1$ ($\sigma_{max}=\sigma_a$, $\sigma_m = 0$) as the alternating load component (and stress) become more and more dominant.

It is quite evident that the $\sigma$ - $\eta$ plot presents quite an effective tool for the designer. The design stress boundary can well be determined once the factor of safety and the influences of surface finish, stress concentration and similar factors be taken into account.

The consistency of test results and their general agreement with published data inspire confidence in the testing machine and measuring equipment. Further experiments may well be conducted to study various facets of fatigue behaviour such as the effect of wave form and load frequency, cumulative damage, surface roughness, size, geometrical imperfections and stress concentrations.

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