

THE UNIVERSITY OF MICHIGAN
INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

ESTABLISHMENT OF RELIABILITY STANDARDS
THROUGH TESTING

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The performance of an automotive part in service is judged by the number of failures or time to failure of a small number of components rather than by the mean life. This, in turn depends on the degree of scatter. Thus in a meaningful design it is essential to incorporate a reliability factor which would provide for this scatter.

Reliability can be defined as the probability of a product performing a specified function, under given conditions, for a specified period of time, without failure. Thus, the function of the part, the operating conditions, and the time of operation are all important aspects of reliability.

In statistical terms reliability is the converse of the probability of failure. If the probability of failure is one the part will fail and, therefore, the reliability is zero. Similarly, if the probability of failure is zero the part will not fail and reliability is one or 100, if expressed in percent. Thus, 80 percent reliability means 20 percent failure, 95 percent reliability means 5 percent failure, and so forth.

Thus, reliability implies an avoidance of failure. In the case of automotive components this means principally fatigue failures as most automotive parts are subjected in service to repetitive loading.

The Problem of Scatter

It is the fundamental characteristic of manufactured parts that they exhibit variation in life when subjected to identical loading conditions. Aside from the variations resulting from the human error in testing, or from the limitations of the test equipment, the principal variation lies in the parts themselves. No two pieces produced are exactly the same, no matter how refined is the process. Although the differences may be small, they nevertheless exist.

Faced with this problem of scatter it has been intuitively recognized that it will not suffice to evaluate an automotive part from a single sample and a recourse must be taken to an average (mean) of a number of samples. This has been a widely accepted method of evaluating data.

The use of the average alone has a serious drawback. Half of the specimens have a life lower than the average and, therefore, a design based on the mean value implies a 50 percent reliability, that is, half of the parts will fail. This would be an intolerable condition and the reason that 50 percent failures do not occur in actual practice is because in design calculations based on mean values generous factors of safety are provided.

Meaningful analysis of scatter data necessitates the use of statistical methods. The dispersion, or scatter, is described by a numerical factor. The usual measure of standard deviation can be used in connection with a normal distribution of occurrences, or the log-standard deviation can be used with the log-normal distribution. More refined methods, such as the extreme value distribution, the probit method, or the staircase method of fitting a regression line to the distribution did not offer any higher accuracy in the present case.

For this study the normal distribution was used. Figure 1 shows a Normal Distribution Curve. If one standard deviation each side of the average life is measured off, then 68.27 percent of the population will lie within $\pm 1\sigma$. This means that 31.73 percent will lie outside these limits, half to the left and half to the right. Now, in reliability we set a certain design goal, say 5000 hours for an engine or at least 100,000 miles for a truck axle. This means that we are interested principally in

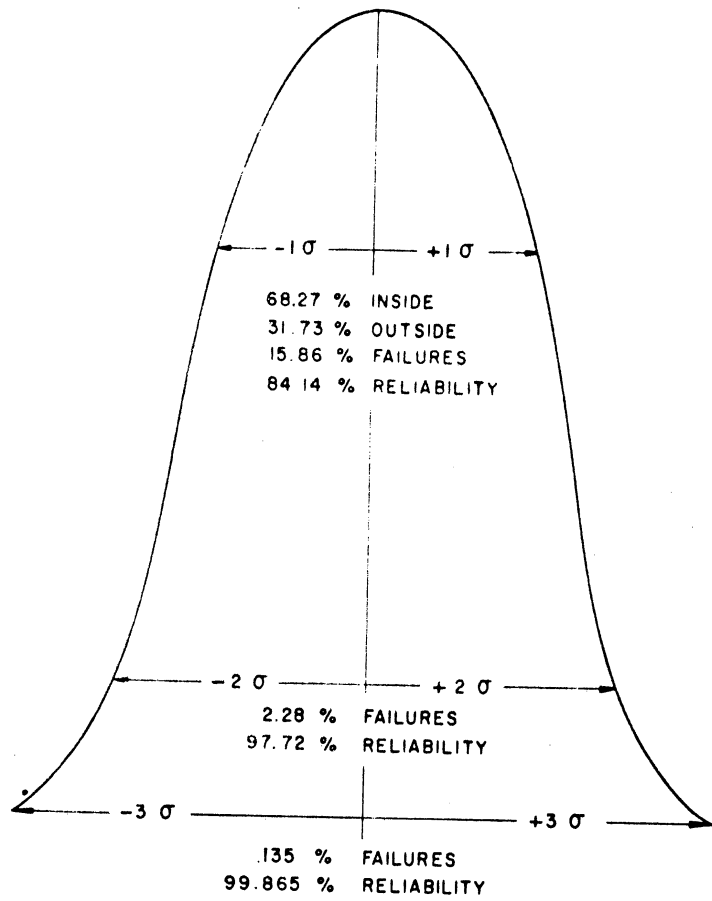


Figure 1. Normal Distribution Curve.

those parts which may not meet this limit, to the left of the design value, because these are the ones that are likely to fail.

Thus, we take one-half of 31.73 percent, that is 15.86 percent, and this then is the percentage of probable failures. The converse of failure is reliability. Thus, by subtracting 15.86 percent from 100 we obtain 84.14 percent as Percent Reliability.

To summarize, if we design our part to a value of $\pm 1\sigma$ off the average, we can predict a 84.14 percent reliability, with 2σ we obtain 97.72 percent reliability, and with 3σ we have 99.865 percent reliability.

Before we apply these values to the prediction of reliability let us consider a fatigue curve, such as in Figure 2. Suppose we focus our attention on one stress level and consider the scatter of the test points. We can then measure off the average life \bar{x} one standard deviation (standard deviation for a sample is denoted by s_1 ; for a population by σ), two standard deviations and three standard deviations, as shown. However, one standard deviation represents 84.14 percent reliability and we can obtain similar data for two standard deviations, three standard deviations, or any number or fraction of standard deviations. We can thus relate the design life in cycles, hours, and so forth, with percent reliability and percent failures. However, instead of expressing the design life as an absolute quantity, such as cycles which then will apply only to a specific problem, we prefer to express it in terms of a dimensionless quantity by dividing each life by the average life.

Average, by its definition, corresponds to 50 percent reliability, and each of the design life points on the abscissa corresponds to some reliability, as we have just determined. Thus we obtain a fraction: life

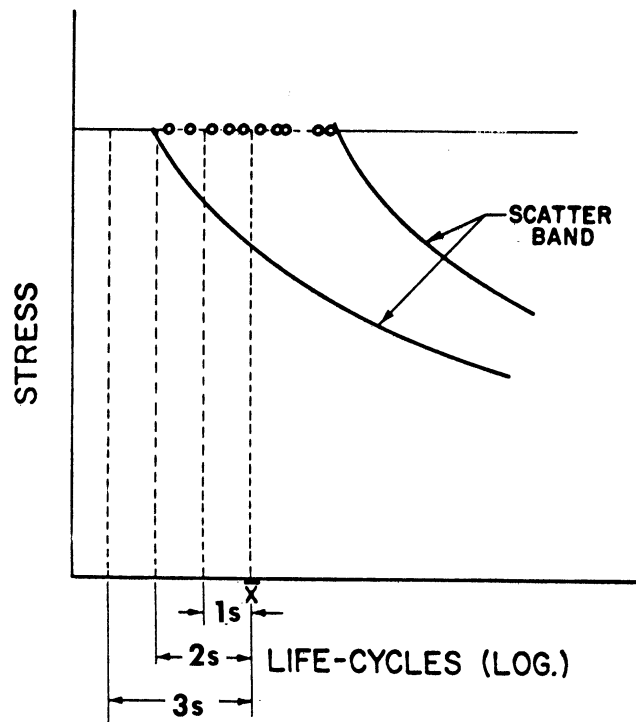


Figure 2. Fatigue Data.

at a given reliability divided by a life at 50 percent reliability, that is, life as a fraction of the average life. This we call the Design Life Factor.

In Figure 3 we plot Percent Reliability on the upper abscissa. The lower abscissa gives the corresponding Percent Failures. On the ordinate are plotted Design Life Factors, just defined.

If test data are plotted on this chart a straight line results, as shown for a particular part (ball bearings tested at 475 ksi) in Figure 3. The chart indicates that if you desire a 90 percent reliability, that is a maximum of 10 percent failures, you should design these bearings to 0.4 of the average life. If you want fewer failures, say one percent failures, you must design more precisely, to .18 of the average life.

Let us illustrate the use of this chart by the following example. Suppose you are a manufacturer of chassis springs and you set a goal of 100,000 miles with not more than 10 percent failures. This particular chart, if it applied to springs, would state that for 90 percent reliability the design life must be 0.4 of the average life. Solving the equation in Figure 4 we conclude that the average life must be 250,000 miles.

That is, if you want 100,000 miles of failure-free operation, with not more than 10 percent failures, your springs must have an average life of 250,000 miles; because if you design to an average life of 100,000 miles and your goal is still 100,000 miles, you will record 50 percent failures.

In this manner, a great deal of test data was accumulated and plotted in the manner of Figure 3. From this, Design Life Factors at four different percent reliabilities were tabulated as shown in Table 1.

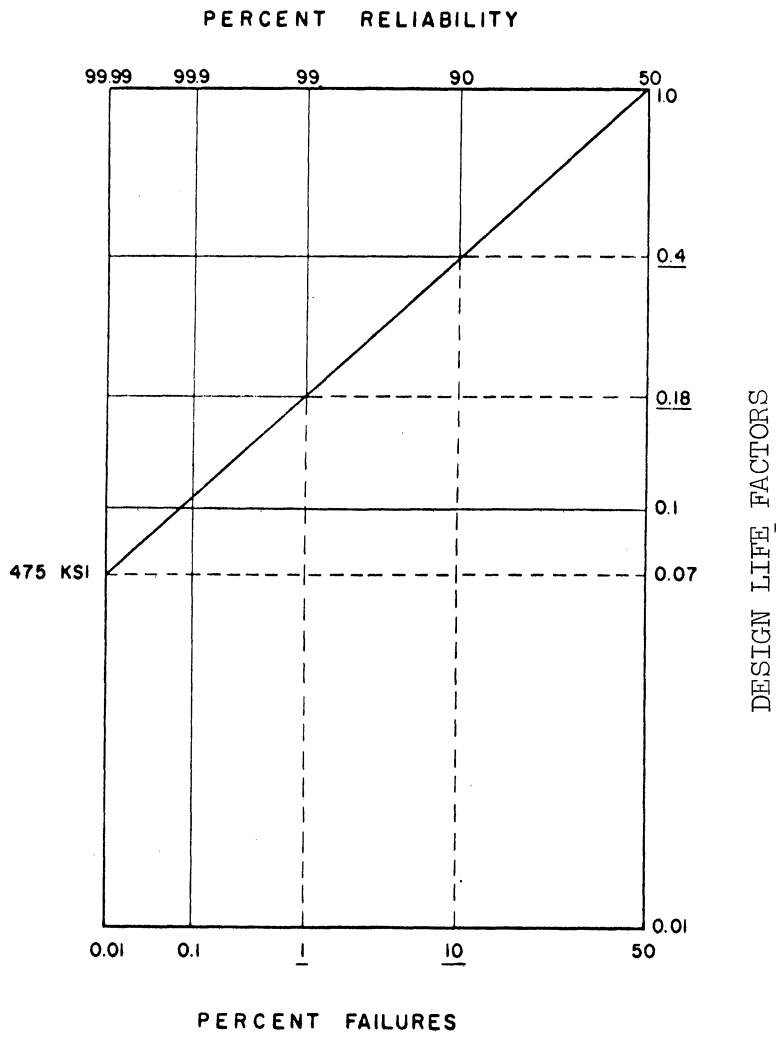


Figure 3. Design Life Factors for a Ball Bearing.

$$\frac{\text{LIFE AT GIVEN RELIABILITY}}{\text{LIFE AT 50\% RELIABILITY}} = .4$$

$$\frac{\text{LIFE AT 90\% RELIABILITY}}{\text{AVERAGE LIFE}} = 4 = \frac{100,000 \text{ MILES}}{\text{AVERAGE LIFE}}$$

$$\text{AVERAGE LIFE} = \frac{100,000}{4} = \underline{25,000 \text{ MILES}}$$

Figure 4. Average Life vs. Design Life.

It will be noted that those parts which are subjected in service to contact loading (ball bearings, hypoid and bevel gears, engine valves) were found to have lower Design Life Factors than parts under flexural (bending, torsion, axial) loading (crankshafts, springs, fans, and so forth). This is shown in Figure 5.

TABLE 1

DESIGN LIFE FACTORS

Component	Reliability - Percent			
	.90	.95	.99	99.9
Wheel	.72	.66	.55	.45
Leaf spring	.68	.60	.49	.38
Fan	.65	.57	.45	.35
Transmission gear	.57	.49	.35	.26
Crankshaft - design A	.52	.46	.31	.22
Crankshaft - design B	.49	.40	.27	.18
Front wheel spindle	.41	.33	.21	.13
Engine exhaust valve	.30	.20	.11	.05
Hypoid gear, 93 ksi	.42	.33	.20	.13
Hypoid gear, 105 ksi	.47	.37	.25	.16
Bevel gear, 75 ksi	.40	.32	.19	.11
Bevel gear, 93 ksi	.36	.27	.16	.084
Ball bearing, 388 ksi	.41	.34	.21	.13
Ball bearing, 430 ksi	.35	.26	.15	.08
Ball bearing, 475 ksi	.39	.30	.19	.11
Ball bearing, 510 ksi	.41	.34	.21	.14
Ball bearing, 540 ksi	.32	.24	.13	.07
Axle shaft, automobile, 27 ksi	.42	.35	.22	.14
Axle shaft, automobile, 38 ksi	.56	.46	.34	.24
Axle shaft, automobile (shot peened)	.44	.37	.24	.16
Axle shaft, farm tractor, 30 ksi	.65	.57	.45	.34
Axle shaft, farm tractor, 54 ksi	.60	.52	.40	.29
Roller chain, 650 lb.	.45	.35	.23	.14
Roller chain, 800 lb.	.41	.32	.20	.12

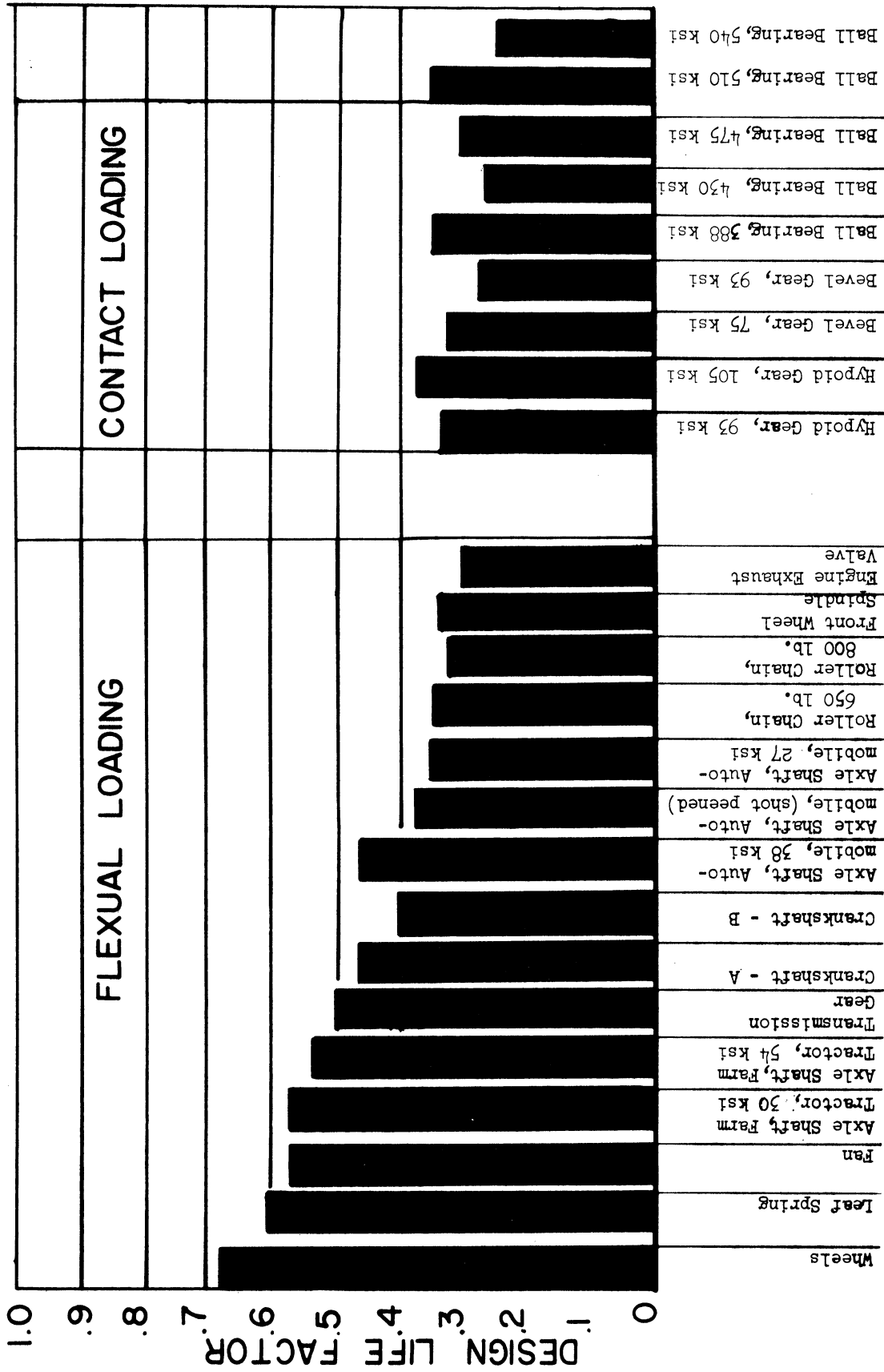


Figure 5. Design Life Factors at 95 Percent Reliability.

APPENDIX

Some additional information on the characteristics of the automotive components studied:

Ball bearings: 1/2 in. radial, SAE 52100.

Bevel and hypoid gears:

Wheels: truck, malleable iron.

Leaf springs: truck, SAE 4068, 444-477 BHN.

Gears: truck, transmission helical, SAE 8620, carburized, 58-63R_C.

Crankshaft: automobile, SAE 1046, 228-269 BHN.

Front wheel spindles: automobile, SAE 1046, 248-293 BHN.

Engine valves: truck, exhaust valves.

Axle shafts: farm tractor, SAE 8635, 269-321 BHN.

Axle shafts: automobile, SAE 8650, 388-444 BHN.

Roller chains: American standard No. 40.

Fan: automobile, 6-blade spider type, SAE 1020.

BIOGRAPHY

Charles Lipson

Professor Lipson graduated from Muhlenberg College with a B.S. degree in 1930 and from New York University with a Ph.D. in physics in 1935. From 1930 to 1946 he was employed by Chrysler Corporation on problems of design, development, testing, weight reduction, prevention of failure, and vibration. In 1946 he entered a consulting practice. In 1955 he was appointed Professor of Mechanical Engineering at The University of Michigan.

He is an author of numerous papers, books and pamphlets in the field of stress analysis, reliability, design of experiments and prevention of failures. He is co-founder, past president and editor of the Society for Experimental Stress Analysis; member of the New York Academy of Science; member of the Panel No. 3 of National Research Council of the National Academy of Sciences; member of the U.S. Army Mobility Equipment Command Scientific Advisory Group; editor of the Proceedings of the Society of Industrial Mathematics, 1956; national lecturer of the American Society of Mechanical Engineers, 1951-1953; and Woodside lecturer of the American Society for Metals, 1966.