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CONTINENTAL O-1430-1 ENGINE

MINIMUM WEIGHT VALVE SPRINGS

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MINIMUM WEIGHT VALVE SPRINGS

Object

The purpose of this analysis was to determine the minimum weight of springs that would satisfy the requirements on the CONTINENTAL O-1430-1 Engine valve Gear

Summary

By selecting a material with a high elastic limit and utilizing its maximum capability of work, a minimum weight of material is required. In the following pages the necessary mathematical equations are derived and the subject springs are calculated.

Conclusions

The springs as used on the single cylinder test engine, and now specified for the 12 cylinders under construction, weigh .756 pounds for each valve, toteling 18.25 pounds for 12 cylinders (24 valves). The springs as called for in the following design will weigh only .206 pounds for each valve or a total of 4.95 pounds for the 12 cylinders, showing a weight saving of 13.3 pounds.

Endurance runs on the single cylinder are to be made to test the connecting rods. At the same time, these springs will be tested and if proven satisfactory may be then used on the multi-cylinder engine.

Procedure

Figure 1 (taken from Reference 1) shows the fatigue limit diagram for the Swedish Chrome Vanadium Steel selected. The X axis represents the initial stress to which the test springs were subjected and the Y axis the final stress. The scale of the graph was laid out so that the initial stresses would all fall on a straight line at 45 degrees to the axis. In this method of plotting, the initial stress may be read on either the X or Y axis and the intercept on the Y axis shown by the vertical solid line indicates a safe endurance limit of the spring in question.

Springs of this material will operate satisfactorily if the stress variation is below the line denoting the elastic limit and between the two angular lines on the chart. Springs which go above the upper slanting line will fail in less than ten million reversals.

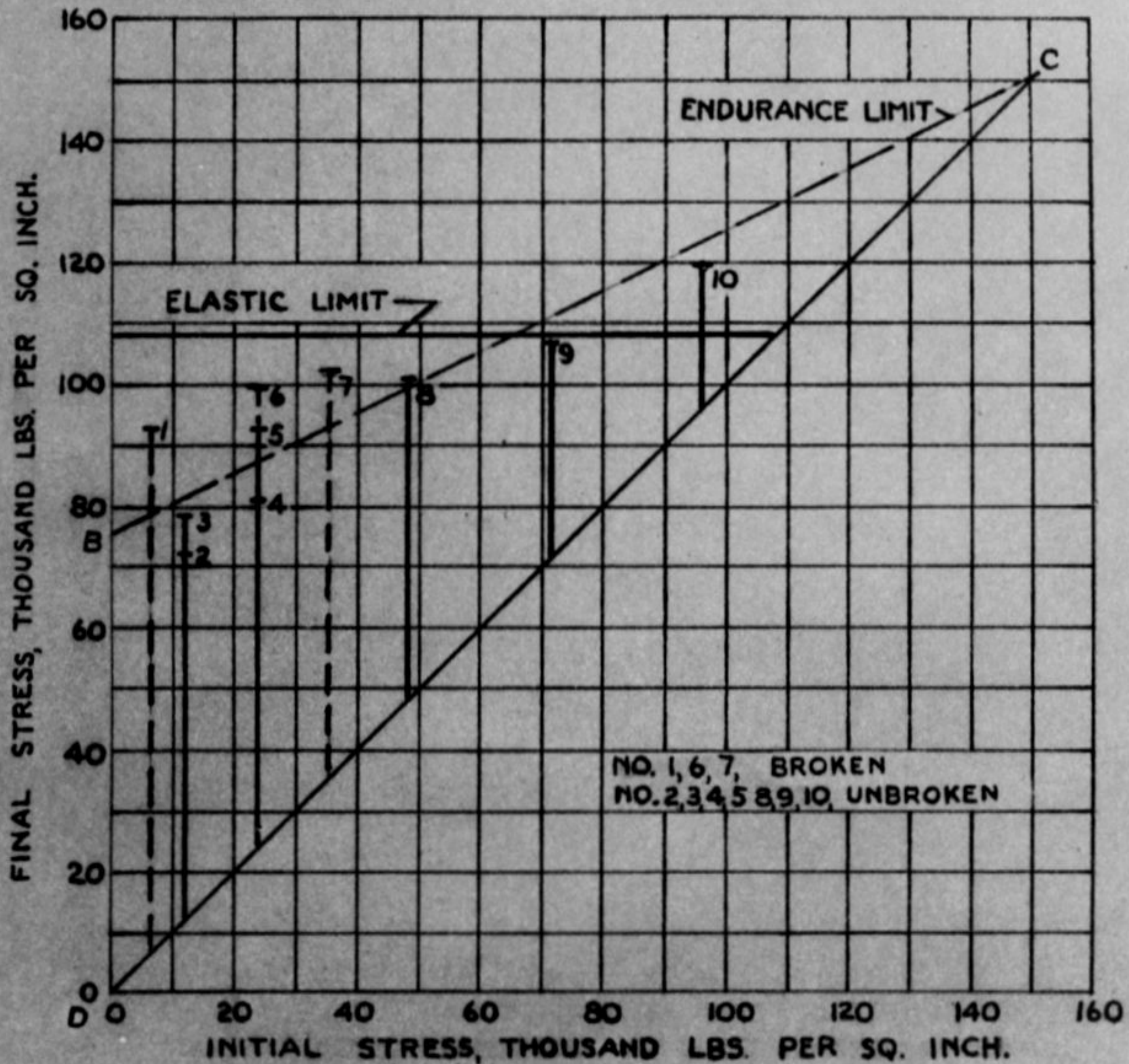

FIG.1. - FATIGUE LIMIT DIAGRAM FOR SWEDISH CHRO. VAN. STEEL.

A = .490 **B = 76,000**

All Stresses Calculated By Wahl Formula

Note: The above chart taken from U. of M. Engineering Research Bulletin No. 26 - July 1934.

Let A = Slope of the dotted line BC in Figure 1 = .490

Let B = Maximum stress range or from 0 to 76,000

Then B = 76,000 pounds per square inch

Let S_1 = Initial stress

Let S_2 = Final stress

Then $S_2 = AS_1 + B$ (1)

All of the actual stress calculations are based on the formulae by Mr. A. M. Wahl (See Reference 2). This gives a summation of torsional and shearing stresses.

$$S_2 = \text{Maximum stress} = \frac{8\,PD}{\pi d^3}\left[\frac{(4C-1)}{(4C-4)} + \frac{0.615}{C}\right]$$

Where P = Axial load in pounds

D = Mean diameter of the spring

d = Wire diameter

$C = \frac{D}{d}$ i.e. Ratio of mean diameter to wire diameter

However as the Wahl correction will be taken into consideration in the actual design, it may be omitted from the general case listed below.

We then have for any spring:

$$S_2 = \frac{8PD}{\pi d^3} \qquad (2)$$

$$f_2 = \frac{\pi D^2 S_2}{Gd} \qquad (3)$$

$$H = nf_2 \qquad (4)$$

Where f_2 = Deflection per coil

G = Torsional modulus of elasticity = 11,500,000 #/□"

n = Number of active coils

H = Total deflection

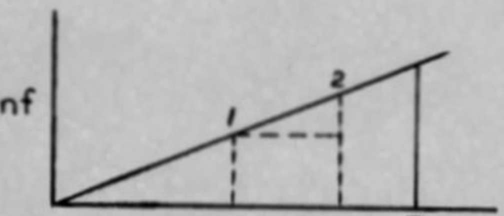

FIG.2

Plotting the deflection against the load as shown in Figure 2, the work done is the area under the curve or

$$W = 1/2\ Pnf \qquad (5)$$

The work done between any two points as 1 and 2 on the curve is

$$W_1^2 = 1/2\ (P_2nf_2 - P_1nf_1)$$

From equation (2) $P = \dfrac{S_2 \pi d^3}{8D}$

Substituting in equation (5) for P_2 and f_2

$$W_2 = 1/2\ \frac{\pi d^3 S_2}{8D} \cdot \frac{\pi D^2 S_2}{Gd}\ n = 1/2 \left[\frac{\pi d^2}{4}\ n\ \pi D\right] \frac{S_2^2}{2G}$$

Now the volume of metal in the spring $V = \dfrac{\pi d^2}{4}\ n\ \pi D$

Therefore $W_2 = \dfrac{V}{4G} S_2^2$

Similarly $W_1 = \dfrac{V}{4G} S_1^2$

Therefore work $W_1^2 = \dfrac{V}{4G}\left(S_2^2 - S_1^2\right) \qquad (6)$

We now wish to introduce into the above equation, the factors for the stress range of the material

$$S_2 = AS_1 + B \qquad \therefore S_1 = \frac{S_2 - B}{A}$$

Substituting this value in equation (6)

$$W = \frac{V}{4G}\left(S_2^2 - \frac{S_2^2 - 2\,BS_2 + B^2}{A^2}\right)$$

Now we wish this spring to do the maximum amount of work it is capable of doing within the known values, A and B taken from Figure 1

This condition is satisfied when $\dfrac{dW}{dS_2} = 0$

$$\therefore \frac{dW}{dS_2} = \frac{V}{4G}\left(2S_2 - \frac{2S_2}{A^2} + \frac{2B}{A^2}\right) = 0$$

$\dfrac{V}{4G}$ is a constant quantity therefore

$$S_2 - \frac{S_2}{A^2} + \frac{B}{A^2} = 0$$

$$A^2 S_2 - S_2 + B = 0$$

$$S_2 = \frac{B}{1 - A^2}$$

Where ℓ = Valve lift

$$\frac{S_2}{S_1} = \frac{H}{H - \ell}$$

For the Chrome Vanadium wire selected, A = .49 B = 76,000. However, this 76,000 #/□" represents the highest permissible stress range and a spring designed using this value for the constant B in the above equations would lie exactly on the dotted line in Figure 1. The probable error in the fatigue tests from which these values were determined, has been established as plus or minus 3000 #/□" and it is well to allow a slight margin to cover any inaccuracies in the wire, etc. Therefore, we will use a value of 70,000 #/□" for B leaving the "A" value .49. This will give us a spring lying slightly below the dotted line in Figure 1.

ℓ = Valve lift = .5625

$$H = \frac{\ell}{1 - A} = \frac{.5625}{1 - .49} = 1.102$$

$$S_2 = \frac{B}{1 - A^2} = \frac{70,000}{1 - .49^2} = 92,200 \ \#/\square''$$

The following springs satisfy the design requirements:

<u>Inner Spring</u>

D = 1.237 d = .135 d^3 = .00246

$$C = \frac{1.237}{.135} = 9.16$$

$$\frac{4C - 1}{4C - 4} + \frac{.615}{C} = \frac{35.64}{32.64} + \frac{.615}{9.16} = 1.157$$

S_2 corrected for Wahl equation = $S_{2c} = \frac{92,200}{1.157} = 79,500 \ \#/\square''$

$$P = \frac{S_{2c}\pi d^3}{8D} = \frac{79{,}500 \times 3.1416 \times .00246}{8 \times 1.237} = \frac{614}{9.9} = 62.1 \text{ pounds}$$

$$f = \frac{\pi D^2 S}{Gd} = \frac{3.1416 \times 1.530 \times 79{,}500}{.135 \times 11{,}500{,}000} = \frac{3820}{15510} = .246 \text{ inch}$$

H = Total Deflection = 1.102

$$n = \text{Number of coils} = \frac{1.102}{.246} = 4.48 \text{ active coils}$$

<u>Outer Spring</u>

$D = 1.574$ $\quad$ $d = .162$ $\quad$ $d^3 = .00425$

$$C = \frac{1.574}{.162} = 9.72$$

$$\frac{4C-1}{4C-4} + \frac{.615}{C} = \frac{37.88}{34.88} + \frac{.615}{9.72} = 1.149$$

$$S_{2c} = \frac{92{,}200}{1.149} = 80{,}200 \text{ \#/□"}$$

$$P = \frac{80{,}200 \times 3.1416 \times .00425}{8 \times 1.574} = \frac{1071}{12.592} = 85.1 \text{ pounds}$$

$$f = \frac{3.1416 \times 2.477 \times 80,200}{11,500,000 \times .162} = \frac{62.3}{186.1} = .335 \text{ inch}$$

$$n = \frac{1.102}{.335} = 3.3 \text{ Active coils}$$

The above springs as designed may have excessive surge which if present, is apt to over stress the material causing failure. It is proposed to make up two sets of springs one as described above, the other essentially the same containing a dampening coil. These springs may be tested simultaneously and should give satisfactory results

REFERENCES

"Permissible Stress Range for Small Helical Springs"

By: F. P. Zimmerli
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"Stresses in Heavy, Closely - Coiled Helical Springs"

By: A. M. Wahl
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