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INVESTIGATION INTO SURFACE TREATMENT ON FATIGUE LIFE FOR CYLINDER BLOCK OF LINEAR ENGINE USING FREQUENCY RESPONSE APPROACH

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Abstract

Aluminum alloys are one of the most promising materials selections for automobiles parts and electrical components to reduce their weight and to increase their specific strength. This paper presents the effect of surface treatment on the fatigue life of the vibrating cylinder block for a new two-stroke free piston engine using random loading conditions. The finite element modeling and analysis were performed using a computer aided design and finite element analysis codes respectively. In addition, the fatigue life prediction was carried out using finite element based fatigue analysis code. The material aluminum alloys are considered in this study. The frequency response approach was applied to predict the fatigue life of cylinder block using different load histories. Based on the finite element results, it is observed that there is a significant variation between the nitriding treatment and untreated cylinder block of free piston engine. The obtained results indicate that the nitrided treatment produces longest life for all loading conditions. Therefore, the nitriding process is one of the promising surface treatments for aluminum alloy parts to increase the fatigue life of the linear engine cylinder block.

1. INTRODUCTION

Due to market pressures for improvements in productivity, reliability, ductility, wear resistance as well as the profitability of mechanical systems, manufacturers are placing increasing demands on available materials. In order to enhance the surface properties of today's materials, producers of components are turning to different surface treatments [1]. There are several techniques available for mechanically improving the surface properties of the components, such as polished, ground, machined, hot rolled, forged, cast, etc. Some of these techniques produce an improve surface by plastic deformation of surface irregularities [2]. Various methods have so far been employed in order to improve fatigue strength, including optimization of geometric design, stronger, materials and surface processing such as nitriding, cold rolled, shot peening etc. Among them shot peening has long been widely used as a low cost and simple method for increasing the fatigue strength of the component [3].

Light metals have been utilized for automotive parts to reduce the weight of automobiles, aiming at the significant reduction of CO_2 emission and environmental burdens [4]. The use of aluminum (Al) instead of steel for lightening of vehicle components or machine parts has recently increased. Al and its alloys have advantages over non metallic materials: aluminum alloys have a high melting point, a good corrosion resistant, a good workability and have a good thermal conductivity. However, the hardness and wear resistance of Al alloys are respectively lower and inferior to those of steel; therefore, there is a limit in their application to moving parts. Hence, research has been carried out in surface modification technology to increase the applicability of Al alloys as moving parts. Surface modification technologies for Al alloys can be classified into three main groups. Alloying is the first method and this forms a hard film on the Al surface [5]. The second group is coating method, which covers the Al surface with hard materials [6]. The third is a heat treating process, such as nitriding [7].The objective of the current study is specifically to investigate into the effect of surface treatments on the fatigue life improvement of the cylinder block of linear engine. Numerical investigates are performed to characterize completely the different induced effects before and after surface treatments. The numerical results were discussed and analysed. Their influences on HCF behaviour of the vibrating component made from AA 6061-T6.

2. FREQUENCY RESPONSE METHODS

This section describes a variety of approaches for computing fatigue life or damage directly from the PSD of stress as opposed to a time history. The stress power spectra density represents the frequency domain approach input into the fatigue [8]. This is a scalar function that describes how the power of the time signal is distributed among frequencies [9]. Mathematically this function can be obtained by using a Fourier Transform of the stress time history's auto-correlation function, and its area represents the signal's standard deviation. It is clear that PSD is the most complete and concise representation of a random process.

Bendat [9] presented the theoretical basis for the first of these of these frequency domain fatigue models, so called Narrow band solution. This expression was defined solely in terms of the spectral moments up to m_4. However, the fact that this solution was suitable only for a specific class of response conditions was an unhelpful limitation for the practical engineer. Bendat showed that the probability density function of peaks for a narrow band signal tended towards Rayleigh distributions as the bandwidth reduced. To complete his solution method, Bendat used a series of equations derived by Rice [10] to estimate the expected number of peaks using moments of area beneath the PSD.

The expected total fatigue damage for narrow band Gaussian process,

$$E[D] = \sum_i \frac{n_i}{N(S_i)} = \frac{S_t}{K} \int S^b \; p(S)\, dS$$
$$= \frac{E[P]T}{K} \int S^b \left[\frac{S}{4\, m_0} \; e^{\frac{-S^2}{8m_0}} \right] dS \tag{4}$$

where $N(S_i)$ is the number of cycles of stress range S occurring in T seconds, n_i is the actual counted number of cycle, S_t is the total number of cycles equals to $E[P]T$. In order to compute fatigue damage over the lifetime of the structure in seconds the form of materials S-N data must be defined using the parameters k and b.

Many expressions have been proposed to correct the conservatism associated with this solution. The solutions of Wirsching and Light [11], Tunna [12], Chaudhury and Dover [13], Steinberg [14], and Hancock [15] were all derived using this approach. They are all expressed in terms of the spectral moments up to m_4 .

Tunna Solution: $$p(S)_T = \left[\frac{S}{4\gamma\, m_0} e^{-\frac{-S^2}{8\gamma\, m_0}} \right] \tag{6}$$

Wirsching Solution : $$E[D]_{Wirsching} = E[D]_{NB} \left[a + [1-a](1-\varepsilon)^c \right] \tag{7}$$

where $\varepsilon = \sqrt{1-\gamma^2}$ = a spectral bandwidth parameter which is an alternative version of the irregularity factor, and a and c are best fitting parameters expressed as:

$$a = 0.926\text{-}0.033b;\ c = 1.587b - 2.323 \tag{8}$$

where b is the slope of the S-N curves which is defined in Eq. 5.

This solution is given in the form of an equivalent stress range parameter S_{eq},

where $$S_{eq} = \left[\int_0^\infty S^b\ p(S)\, dS \right]^{\frac{1}{b}} \tag{9}$$

Hancock 's equivalent stress : $$(S_{eq})_{Hancock} = \left(2\sqrt{2\, m_0}\right)\left[\gamma\, \Gamma\left(\frac{b}{2}+1\right)\right]^{\frac{1}{b}} \tag{10}$$

Chaudhury & Dover equivalent stress:

$$(S_{eq})_{C\&D} = \left(2\sqrt{2\, m_0}\right)\left[\frac{\varepsilon^{\,b+2}}{2\sqrt{\pi}}\Gamma\left(\frac{b+1}{2}\right) + \frac{\gamma}{2}\Gamma\left(\frac{b+2}{2}\right) + erf^*(\gamma)\,\frac{\gamma}{2}\,\Gamma\left(\frac{b+2}{2}\right)\right]^{\frac{1}{b}} \tag{11}$$

where,

$$erf^*(\gamma) = 0.3012\gamma + 0.4916\gamma^2 + 0.9181\gamma^3 - 2.354\gamma^4 - 3.3307\gamma^5 + 15.6524\gamma^6 - 10.7846\gamma^7$$

Steinberg Solution:

$$(S_{eq})_{Steinberg} = \left[0.683\left(2\sqrt{m_0}\right)^b + 0.271\left(4\sqrt{m_0}\right)^b - 0.043\left(6\sqrt{m_0}\right)^b \right]^{\frac{1}{b}} \tag{11}$$

The fatigue damage can then easily be obtained by substituting this into the general damage equation used when deriving the narrow band solution

$$E[D] = \frac{E[P]\, T}{k} S_{eq} \tag{12}$$

Probably the most famous empirical formula for approximating the rainflow amplitude distribution is that proposed by Dirlik [16], which uses a combination of an exponential and two Rayleigh densities. In the Dirlik model, approximate closed-form expression for the probability density function of rainflow ranges, which was obtained using Monte Carlo technique. The Dirlik solution is expressed by the Eq. (13) and details the specific literature reported in the refs. [16].

$$N(S) = E[P]T\ p(S) \tag{13}$$

where $N(S)$ is the number of stress cycles of range S (N/mm^2) expected in T seconds, $E[P]$ is the expected number of peaks and $p(S)$ is the probability density function.

$$p(S) = \frac{\frac{D_1}{Q} e^{\frac{-Z}{Q}} + \frac{D_2 Z}{R^2} e^{\frac{-Z^2}{2R^2}} + D_3 Z\, e^{\frac{-Z^2}{2}}}{2\sqrt{m_0}} \tag{14}$$

where, $p(S)$ is the probability density function, S is the stress amplitude, m_0 is the zeroth order spectral moment and

$$x_m = \frac{m_1}{m_0}\sqrt{\frac{m_2}{m_4}}; \qquad \gamma = \frac{m_2}{\sqrt{m_0\, m_4}}; \qquad D_1 = \frac{2(x_m - \gamma^2)}{1+\gamma^2}$$

$$R = \frac{\gamma - x_m - D_1^2}{1 - \gamma - D_1 + D_1^2}; \qquad Z = \frac{S}{2\sqrt{m_0}}; \qquad D_2 = \frac{1-\gamma - D_1 + D_1^2}{1-R}$$

$$D_3 = 1 - D_1 - D_2; \qquad Q = \frac{1.25(\gamma - D_3 - D_2\ R)}{D_1}$$

where x_m, D_1, D_2, D_3, Q, Z and R parameters depending on the m_0, m_1, m_2 and m_4; Z is a normalized variable. m_0, m_1, m_2 and m_4 are the zeroth, 1st, 2nd and 4th order spectral moments, respectively. It should be noted that, if compared to the Wirsching-Light model, this approach gives an amplitude distribution (and thus a rainflow damage) depending on just four spectral moments (i.e. m_0, m_1, m_2 and $m_{4)}$, including in particular a dependence on m_1 moment. All of above discussed methods are determining fatigue life from PSDs stress.

3. RESULTS AND DISCUSSION

A geometric model of the cylinder block of the free piston engine is considered in this study. Three-dimensional model geometry was developed in CATIA® software. Since the tetrahedral is found to be the best meshing technique, the 4 nodes tetrahedral (TET4) element version of the cylinder block was used for the initial analysis. In addition, the TET4 compared to the 10 nodes tetrahedral (TET10) element mesh using the same global mesh length for the highest loading conditions (7.0 MPa) in the combustion chamber. The investigating the results, it can be found that the TET10 mesh predicted higher von Mises stresses than that the TET4 mesh. The TET10 mesh is presumed to represent a more accurate solution since TET4 meshes are known to be dreadfully stiff [17]. TET4 employed a linear order interpolation function while TET10 used quadratic order interpolation function. For the same element size, the TET10 is expected to be able to capture the high stress concentration associated with the bolt holes. A TET10 was then finally used for the solid mesh. Mesh study is performed on the FE model to ensure sufficiently fine sizes are employed for accuracy of calculated results but at competitive cost (CPU time). In the process, specific field variable is selected and its convergence is monitored and evaluated. Sensitivity analysis was performed to obtain the optimum element size. The analysis was performed iteratively at different element lengths until the solution obtained appropriate accuracy. Convergence of the stresses was observed, as the mesh size was successively refined. The element size of 0.20 mm was finally considered. A total of 35415 elements and 66209 nodes were generated with 0.20 mm element length. Compressive loads were applied as pressure (7.0 MPa) acting on the surface of the combustion chamber and preloads were applied as pressure (0.3 MPa) acting on the bolt-hole surfaces. In addition, preload was also applied on the gasket surface generating pressure of 0.3 MPa. The loading and constraints on the cylinder block are shown in Fig. 1. The constraints were applied on the bolt-hole for all six degree of freedoms.

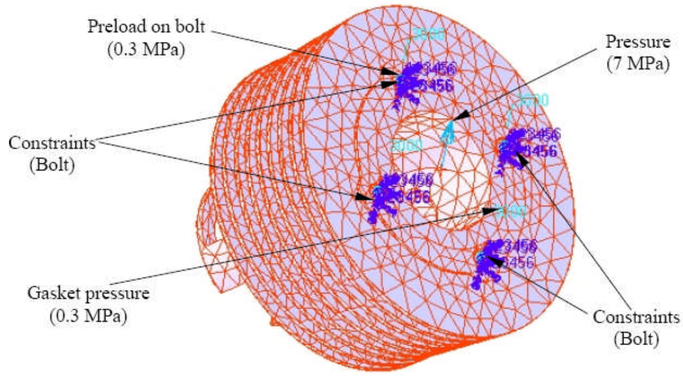

Figure 1. Loading and constraints on the cylinder block.

The finite element results of time domain (Pseudo-static method) i.e. the maximum principal stresses distribution is presented in Fig. 2. The fatigue life of time domain histories are performed using the stress-life method employed rainflow cycle counting technique [18]. Time domain fatigue approach consists of a number of steps. The first is to count the number of stress cycles in the response time history. This is performed through a process of rain flow cycle counting. Damage from each cycle is determined, typically from an *S-N* curve. The damage is then summed over all cycles using linear damage summation techniques to determine the total life.

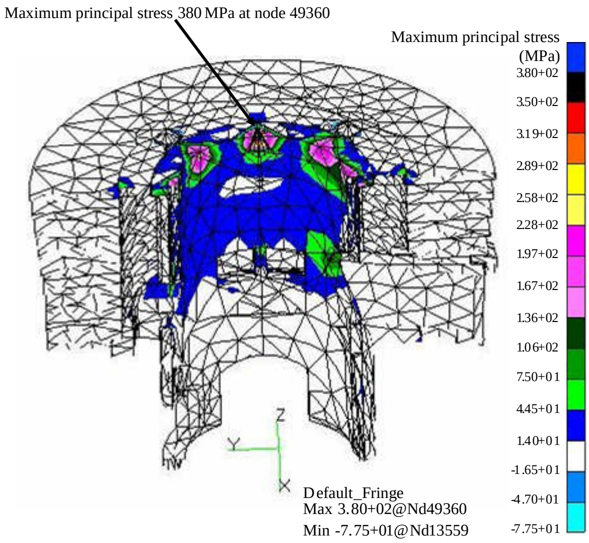

Figure 2. Maximum principal stresses distribution for Pseudo-static linear analysis.

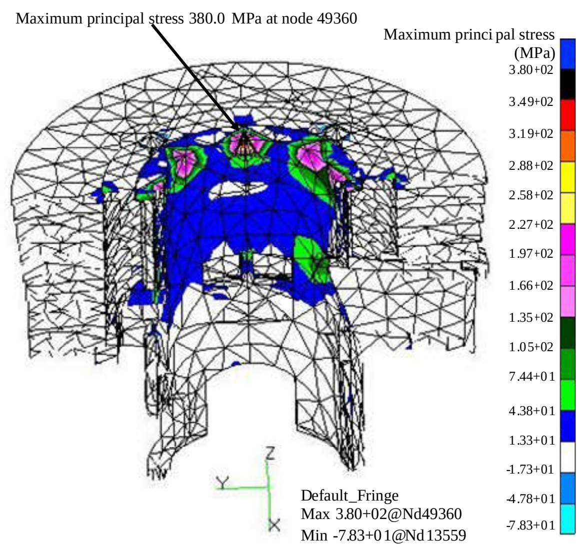

Figure 3. Maximum principal stresses distribution for frequency response analysis with zero Hz.

The frequency response analyses were performed using the finite element analysis code. The frequency response analysis used the damping ratio of 5% of critical. The damping ratio is the ratio of the actual damping in the system to the critical damping. Most of the experimental modal reported that the modal damping in terms of non-dimensional critical damping ratio expressed as a percentage [19]. In fact, most structures have critical damping values in the range of 0 to 10%, with values of 1 to 5% as the typical range [20]. Zero damping ratio indicates that the mode is undamped. Damping ratio of one represents the critically damped mode. The result of the frequency response finite element analysis with zero Hz i.e. the maximum principal stresses distribution of the cylinder block is presented in Fig. 3 respectively. From the results, the maximum and minimum principal stresses of 380.0 MPa and –77.5 MPa for the Pseudo static analysis, and 380.0 MPa and -78.3 MPa for the frequency response analysis for zero Hz were obtained respectively. These two maximum principal stresses contour plots are almost identical.

The variation of the maximum principal stresses with the frequency is shown in Fig. 4. It can be seen that the maximum principal stress varies with the higher frequencies. This variation is due to the dynamic influences of the first mode shape. It is also observed that the maximum principal stress occurs at a frequency of 32 Hz. The maximum principal stresses of the cylinder block for 32 Hz is presented in Fig. 5. From the results, the maximum and minimum principal stresses of 561.0 MPa and -207.0 MPa were obtained at node 49360 and 47782 respectively.

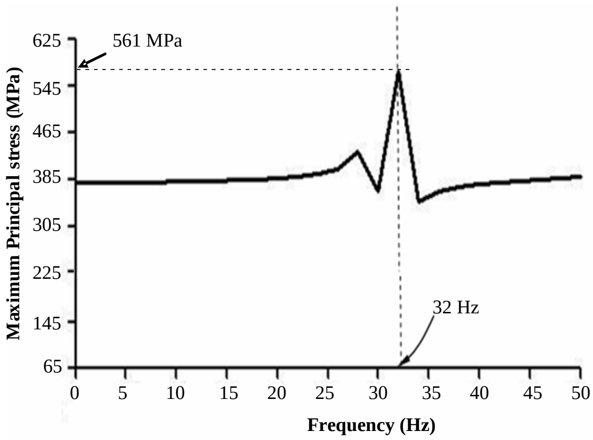

Figure 4. Maximum principal stresses plotted against frequency.

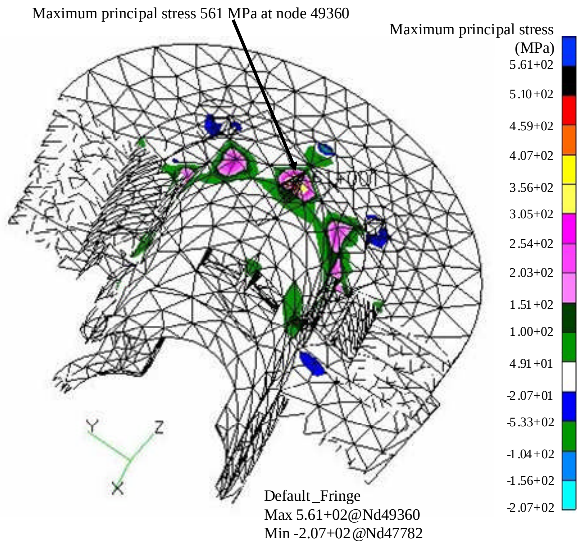

Figure 5. Maximum principal stresses contour for the frequency response analysis with 32 Hz.

Fig. 6 shows the time–load histories signals of the narrow band (1 peak), and two wide band of bimodal (2 peaks) and trimodal (3 peaks) loading histories [21]. Fig. 7 shows the power spectral density function function of those signals.

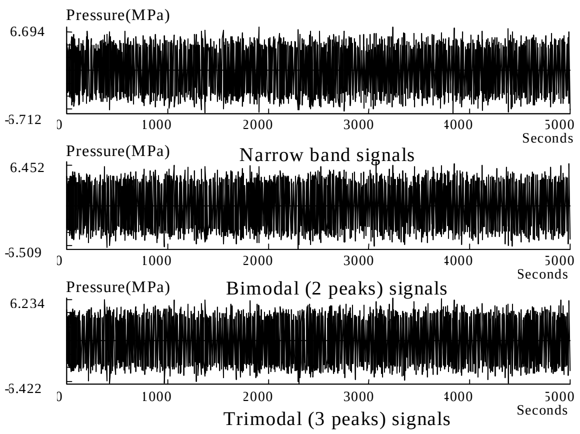

Figure 6. Time-loading histories of narrow band, bimodal and trimodal signals.

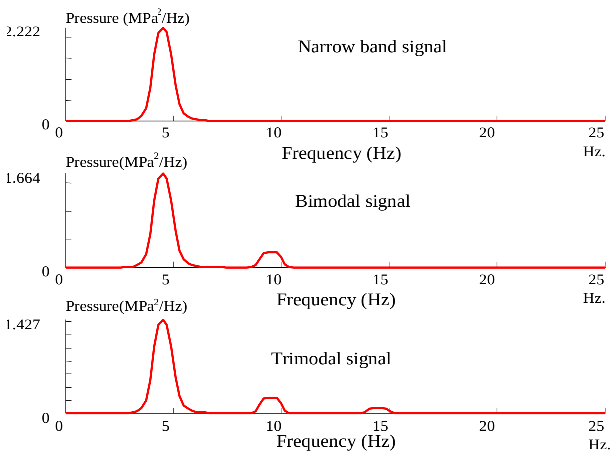

Figure 7. Power spectral density functions of narrow band, bimodal and trimodal signals.

Table 1 shows comparison between the narrow band (1 peak), and wide band of bimodal (2 peaks) and trimodal (3 peaks) loading histories with the time domain from all the frequency domain approach methods. From the results, it can be seen that the Dirlik method gives the best results for all three loading conditions (narrow band and wide band signals). The narrow band gives good results from the narrow band signal (1 peak), but becomes too conservative when the signal is bimodal and trimodal signals (2 peaks and 3 peaks). Tunna breaks down completely for a wide band signal. Wirsching is nonconservative and then too conservative as is Steinberg. Hancock, and Chaudhury & Dover do reasonably well but not as well as Dirlik model. When the signal is wide band (bimodal and trimodal), the narrow band tends to turn any signal into a narrow band signal making the resulting prediction fatigue life extremely and sometimes overly conservative. The Dirlik method found to gives the best results when compared with the corresponding time domain result and others. From the above discussions, it is concluded that the Dirlik method is recommended for general usage.

Table 1. Predicted fatigue life at critical location (Node 49360) for various frequency response approaches using the narrow and wide band signal.

Load- time histories signals	Predicted Fatigue life at critical location in seconds (10^6)							
	Time Domain	Narrow Band	Dirlik	Tunna	Wirsching	Hancock	Chaudhury & Dover	Steinberg
Narrow band	2.66	2.61	2.65	5.83	3.83	2.47	2.46	3.18
Bimodal	4.48	2.49	4.30	1.11	4.23	3.81	4.53	3.51
Trimodal	6.16	2.55	6.38	3.79	4.32	5.19	7.46	3.67

Power spectral density with FFT buffer size 16384:0.03052 Hz width is consider in this analysis. The material used in this study is AA6061-T6. A high proportion of all fatigue failures nucleate at the surface of components and thus surface conditions become an extremely important factor influencing fatigue strength. Surface effects are caused by differences in surface roughness, microstructure, chemical composition, and residual stress [22]. The surface factors as a function of ultimate tensile strength involving different surface finish conditions such as grinding, machining, hot rolling, and as-forged [23]. The correction factors for surface treatment are obtained Juvinall and Marshek [28] empirical data and are related to the ultimate strength of the material. The effect of surface treatment at different loading conditions for the various surface finishes of the vibrating cylinder block using the 3 peaks wide band signal time-loading histories is summarized in Table 2.

Table 2. The effect of the surface finish and treatments on the fatigue life at critical location (node 49360) for 3 peaks loading conditions.

Surface Finish Processes	Predicted vibration fatigue life in years			
	Untreated	Shot peened	Cold Rolled	Nitrided
Polished	0.202	0.644	10.430	808.600
Ground	0.104	0.425	0.104	170.600
Good Machined	0.063	0.450	6.532	54.540
Poor Machined	0.026	0.149	1.475	8.689
Forged	0.010	1.265	0.010	1.265
Cast	0.009	0.064	0.009	1.091

It can be seen that the fatigue life for nitriding surface treatments is surprisingly increases than those other surface treatment processes due to the development of compressive residual stresses and the increase of hardness near the surface. It can be seen that there are no effect on ground, forged and cast surface finish condition with the cold rolled surface treatments. Forging can cause surface decarburization and the loss of carbon atoms from the surface material causes it to have a lower strength and may also produce residual tensile stresses. Both of these factors are especially detrimental to fatigue strength. Thus the nitriding could be applied to the fatigue critical area of the subject to improve the fatigue life and also improve the fretting resistance, since the cylinder block is in contact with other components.

4. CONCLUSIONS

This paper conveys important findings on the influence of the surface treatment process parameter on the fatigue lives. Frequency domain fatigue analysis has been applied to a typical cylinder block of two-stroke free piston engine. The results show that all surface treatment processes can be applied to increase the fatigue life of the aluminium alloys component. The surface compressive residual stress has the greatest effect on the fatigue life. According to the results, it can be concluded that the polished and nitriding combinations found the great influences on the fatigue life improvement. Nitriding treatment is to produce compressive forces in the outer layers of the component. In addition, the vibration fatigue analysis can improve understanding of the system behaviors in terms of frequency characteristics of both structures and loads and their couplings.

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