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Identification of optimum *Calophyllum inophyllum* bio-fuel blend in diesel engine using advanced vibration analysis technique

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ABSTRACT

Bio-fuel causes less pollution to the environment as compared with diesel fuel. However, sound and vibration resulting from the combustion in bio-fuel-powered engines adversely affect users. In this study, non-edible bio-fuel, i.e., *Calophyllum inophyllum*, and its blends were investigated for their performance as alternative fuel and were compared by using vibration as the engine parameter. Previous studies used overall engine vibration to determine the optimum bio-fuel blends with minimal vibration. Overall vibration could consist of vibration components due to combustion as well as other external vibration sources. This condition limits its effectiveness in identifying an optimum fuel blend mainly due to vibration caused by combustion effect. Advanced vibration analysis technique, which consists of time-domain, frequency-domain, and motion visualization analyses (i.e., operating deflection shape analysis), is first performed to isolate the effect of external vibration source due to torque from the overall vibration. Bio-fuel-powered engines at full speed and with the lowest engine torque are more suitable for the identification of an optimum fuel blend with minimal engine vibration. Bio-fuel blend B20 has the least vibration in overall root mean square acceleration at full speed compared with pure diesel and other bio-fuel blends.

1. Introduction

Bio-fuel is an alternative fuel extracted from fat tissues and vegetable oils. It can be used in combustion engines by using a suitable portion of bio-fuel to be mixed with diesel fuel. The advantages of bio-fuel have been proven by previous studies [1,2]. Exhaust from bio-fuel combustion produces less pollution as compared with that from diesel fuel.

A critical issue was raised regarding the disagreement on food versus fuel for edible oil sources. Hence, researchers have paid attention into finding the most efficient bio-fuel from non-edible feed-stocks [3]. Various non-edible plants have been tested and considered suitable for producing bio-fuel. Various feed-stocks for bio-fuel are being studied to avoid global dependence on certain sources, which might eventually cause high demand low supply. Generally, most of non-edible feed-stocks are high in acid content [4]. Methods for reducing acid content need to be identified and tested before any non-edible feed-stocks can be proposed. Furthermore, the properties of non-edible feed-stocks should be within acceptable ranges that comply with ASTM D6751 and EN 12412 standards [5]. Some of the known advantages of non-edible bio-fuel are its portability, availability, renewability, high heat content, low sulfur content, low aromatic content, and biodegradability. A previous study [6] found that non-edible bio-fuel can be an alternative to existing fossil fuels.

An example of a non-edible plant that can be produced as bio-fuel is *Calophyllum inophyllum* (CI). A previous study [7] investigated the origin of CI and its application in medicine and consumers products. The production of CI bio-fuel, its advantages and disadvantages, and the properties and performances was previously studied [8]. Reportedly, CI bio-fuels can improve engine performance and fuel uptake, as well as reduce carbon monoxide emission and smoke opacity. Aside from all of the tests and characteristics mentioned, a vibration test to evaluate the engine vibration using CI bio-fuel has not been conducted.

Sound and vibration resulting from engine combustion directly affect users. One of the drawbacks of diesel fuels is loud noise and strong vibration, which can lead to harmful physical effects. Noise is

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most observable among engines with high compression ratios and fast-rising combustion pressure [9]. An engine consists of both rotating and reciprocating motion, which produce vibration. The implementation of condition-based monitoring (CBM) on machine vibration to monitor and diagnose rotating machinery problems are considered successful [10]. Several studies and industrial applications have shown the effectiveness of CBM in maintaining the lifetime of rotating machinery. However, the implementation of CBM to the internal combustion (IC) engine is not as effective as those for other rotating machineries. The difficulties in implementing the CBM to monitor IC engine is due to the complexity of the vibration signals involved. Unlike other rotating machineries, the vibration of IC engines may come from complex problems, such as injection pump malfunctions, injector clogging, excessive clearances of cylinders, and burn out of valves, which do not have any specific vibration characteristic corresponding to each problem. Therefore, the vibration of the combustion engine must be evaluated. Previous studies have shown alternatives for evaluating the vibration of IC engines [11]. However, skilled and experienced personnel in vibration are required to analyze and understand vibration data (e.g., Wigner-Ville Distribution). Meanwhile, other studies had considered other parameters, such as cylinder pressure, acoustic signal, injection parameter, pistons slap, and knock detection, to be compared with IC engine vibration for fault diagnosis [12–16].

The growing interest in the production of a suitable alternative bio-fuel blend necessitates the evaluation of IC engine vibration. A previous study [17] attempted to obtain the optimum bio-fuel blend from canola and soybean by evaluating engine vibration. The optimum fuel blend was determined from the overall root mean square (RMS) vibration value. Another study [18] used palm oil for engine vibration testing. The testing was done on one engine speed, 2000 rpm, i.e., maximum torque with different injection pressure. The optimum bio-fuel blend was determined through the overall RMS vibration value. The technique was simple but no in-depth studies were conducted on the effect of vibration caused by combustion of bio-fuel blend itself. Overall vibration in RMS could consist of vibration components due to combustion as well as other external vibration sources. The overall vibration may possibly be contaminated by these external vibration sources, which could limit the effectiveness of identifying the optimum fuel blend in engine vibration. Torque is one of the external vibration sources that could contribute to the overall vibration of an engine. According to Fig. 6 in Ref. [17], maximum torque must be observed at low engine speed, whereas minimal torque is required at full speed. Therefore, these two speeds are used to study the effect of torque as the external factor in overall engine vibration. The cause of external vibration source, such as torque should be eliminated before determining the optimum bio-fuel blend in engine vibration. In the current study, an advanced vibration analysis technique, which consists of time-domain, frequency-domain, and motion visualization analyses, i.e., operating deflection shape (ODS) analysis, is introduced. Vibration analysis through the time domain can determine the effect of bio-fuel combustion. Meanwhile, other causes of vibration, such as torque, can be separated and differentiated by performing frequency-domain analysis. Aside from these methods, visualizing the engine motion through ODS analysis is simpler and easier. ODS analysis requires both vibration and phase measurement. ODS analysis is a technique that can be used to determine the actual movement of a machine during operation [19]. The ODS technique can show the full motion (developed using DASYLab® software) of the system under testing. In the current study, the ODS technique as well as advanced vibration analysis are implemented to monitor the IC engine vibration fueled by the blend of CI and pure diesel. Each bio-fuel blend was tested and the engine vibration was compared through ODS visualization to determine the optimum bio-fuel blend.

2. Background theory

An ODS analysis relies on applied forces or loads. Therefore, the forced vibration response of a linear structure for a general system [20] can be expressed as follows:

$$[\mathrm{M}]\left\{\ddot{X}(t)\right\}+[\mathrm{C}]\left\{\dot{X}(t)\right\}+[\mathrm{K}]\{x(t)\}=[\mathrm{F}](t) \quad (1)$$

or the equivalent

$$[\mathrm{M}]\left\{\phi\ddot{U}(t)\right\}+[\mathrm{C}]\left\{\phi\dot{U}(t)\right\}+[\mathrm{K}]\{\phi U(t)\}=[\mathrm{F}](t) \quad (2)$$

where $x = \phi U(t)$ is the forced response, [M] is the mass matrix, $[C] = \alpha[M]+ ß[K]$ is the damping matrix, and [F](t) is the excitation force vector and the modal matrix representing the mode shape vectors. Indicating Equation (2) as a decoupled system, it can then be written as follows:

$$[\bar{I}]\left\{\ddot{U}(t)\right\}+[2\sigma_r]\left\{\dot{U}(t)\right\}+\left[\omega_r^2\right]\{U(t)\}=[\phi]^{\mathrm{T}}[\mathrm{F}](t) \quad (3)$$

where $[\bar{I}]$, $[2\sigma_r]$, and $[\omega_r^2]$ represent the diagonal matrices of elements, $\bar{I}$ is the identity matrix, ω_r is the eigenvalue of mode r, σ_r is the decay rate, and ϕ_r is the normalized mode shape or the eigenvector of the mode r. For a system subjected to harmonic excitation with harmonic force defined by $\{F\} = \{f\}e^{i\omega t}$, a solution form can be derived as $\{x\} = \{X(\omega)\}$, where ω is the excitation frequency, t is time, and $X(\omega)$ is the location amplitude vector defined as:

$$X(\omega)=\sum_{r=1}^{n}\frac{\{\phi\}_r^T\{F\}\{\phi\}_r}{\omega_r^2-\omega^2+2\sigma_r\omega i} \quad (4)$$

which represents the ODS of the system subject to a harmonic excitation.

In this research, the operating condition of the steady-state machinery was not changed during the measurement. Therefore, the cross-power spectrum between each roving point DOF and the reference point DOF was adequate. The cross-power spectrum can be written as follows [21]:

$$G_{ij}(\omega)=X_i(\omega)X_j^*(\omega) \quad (5)$$

where X^* is the complex conjugate, i is the measured response, and j is the referent response. Only the phase difference between the reference point and the roving point is important. Therefore, the Auto-Power Spectrum can be written as follows [22]:

$$G_{jj}(\omega)=X_j(\omega)X_j^*(\omega) \quad (6)$$

However, the variation in the amplitude of the auto-power spectrum output is used to measure the absolute amplitude of all points. Combining both the phase differences and amplitudes for all points when linked to the geometry will provide the operating deflection.

3. Methodology

3.1. Yanmar diesel engine

In this research, a single-cylinder diesel engine manufactured by