Mechanism and Machine Theory 50 (2012) 150–158

Contents lists available at SciVerse ScienceDirect

Mechanism and Machine Theory

journal homepage: www.elsevier.com/locate/mechmt

Determination of optimal position for both support bearing and unbalance mass of balance shaft

Chan-Jung Kim [a,b], Yeon June Kang [a], Bong-Hyun Lee [b], Hyeong-Joon Ahn [c,*]

[a] Institute of Advanced Machinery and Design, School of Mechanical and Aerospace Engineering, Seoul National University, Gwanangno 599, Sillim-9Dong, Gwanak-gu, Seoul 151-744, Republic of Korea
[b] ICE/EV Convergence Technology Research Center, Korea Automotive Technology Institute, 74 Yongjung-Ri, Pungse-Myun, Chonan, Chungnam 330-912, Republic of Korea
[c] Department of Mechanical Engineering, Soongsil University, 511 Sangdo-Dong, Dongjak-Gu, Seoul 156-743, Republic of Korea

ARTICLE INFO

Article history:
Received 2 September 2010
received in revised form 7 September 2011
accepted 8 November 2011
Available online 10 December 2011

Keywords:
Conceptual design of balance shaft
Global optimal location
Elastic strain energy
Kinematic energy
Unbalance mass
Supporting bearing

ABSTRACT

A balance shaft module is a smart device that removes harmonic vibrations of rotating machinery directly by generating an excitation with the same magnitude and an opposite phase of the vibration. However, the unbalance of a balance shaft causes a considerable bending deformation of the balance shaft as well as measurable power consumption. This paper presents an optimal conceptual design of a balance shaft by determining locations of both unbalance and supporting bearing. The optimal strategy is to minimize a normalized energy sum of both the elastic strain energy and the kinematic energy of a balance shaft. Then, an optimal design of the balance shaft is derived by the explicit formulation of the global optimum location of both the supporting bearing and the unbalance mass. The optimal design is verified with a simulation of a conceptual design of a balance shaft for a specific target engine.

1. Introduction

Rotating machines are very common in industry and carry out effective actions or useful work by converting energy into rotation using rotating elements such as gears or chains. However, harmonic excitations of rotating elements are hard to avoid due to the nature of the operating mechanism [1–4]. The harmonic vibrations caused by a small unbalance or eccentricity of rotating elements may cause significant problems under persistent and high-speed rotation.

A balance shaft module generates a direct mechanical counterforce for a harmonic vibration, which is one of the most efficient solutions to remove such problems. The balance shaft module is an intuitive device that quenches harmonic vibrations directly by generating an excitation with the same magnitude as the vibration but with an opposite phase from unbalances on a rotor [5–16]. This approach is superior to other indirect methodologies such as an isolation of the vibration path using mounting systems [5–7,17–19] because the induced excitation will no longer be transmitted as the original vibration source is directly removed using the balance shaft module.

A creative mechanical rotating balancer, or balancing shaft, was first proposed by Lanchester [20], which consists of two oppositely rotating unbalanced rotors to reduce the cyclic engine vibration efficiently. In addition, Mewes [21] suggested the position of unbalances in the offset slider-crank mechanisms. Two classic papers contributed to develop the novel mechanism of a balance shaft in practice; however, there were few technical contents to extend for the optimal design issues of a balance shaft.

One of the well-known target systems is a vehicle engine that inherently induces the secondary harmonic excitation from a reciprocating movement of pistons [5–7]. More recently, the Lanchester-type balance shaft module has been widely used for the inline 4-cylinder vehicle engine, as shown in Fig. 1. If the equivalent reciprocal mass in Fig. 1 is given as m_R and the

* Corresponding author. Tel.: +82 2 820 0654; fax: +82 2 880 0654.
E-mail address: ahj123@ssu.ac.kr (H.-J. Ahn).

0094-114X/$ – see front matter © 2011 Elsevier Ltd. All rights reserved.
doi:10.1016/j.mechmachtheory.2011.11.006

Nomenclature

m_R, m_B	Equivalent mass of a reciprocal part and a single balance shaft
m_{SYM}, m_{ASYM}	Mass of a symmetric part and an asymmetric part
I_{SYM}	Mass moment of inertia of a symmetric part
I_Z	Mass moment of inertia with respect to the Z-axis
I_X	Second moment of inertia with respect to the X-axis
f_R, f_B	Force from a secondary reciprocal part and a single balance shaft
f_{b1}, f_{b2}	Reaction force from a front supporting bearing and a rear supporting one
r_B, r_C	Equivalent radius of a single balance shaft and a crankshaft
l_P	Equivalent length of a connecting rod
r_{SYM}, r_{ASYM}	Effective radius of a symmetric mass and an asymmetric mass
l_B	Length of a balance shaft
x_m, x_b	Position of an unbalance mass and a supporting bearing
k	Radius of gyration
k_0, k_1	Linear coefficient of first term and second term of a radius of gyration
M	Bending moment
E	Young's modulus
θ_C	Rotating angle of a crankshaft
ω_C, ω_B	Angular velocity of a crankshaft and a balance shaft
J	Objective function of
V	Elastic strain energy
T	Kinematic energy
γ_m	Mass ratio between a symmetric part and an asymmetric part
γ_f	Load capacity of a supporting bearing
γ_w	Linear weighting factor
x_m^*, x_m^{**}	Global optimum of unbalance mass for unlimited supporting bearing and for limited supporting bearing
x_{b_i}	Position of a supporting bearing at # i
x_b^*, x_b^{**}	Global optimum of supporting bearing for unlimited supporting bearing and for limited supporting bearing
F_{err}	Error function about a approximated radius of gyration
⊕	Geometric center of a balance shaft

corresponding unbalances of a balance shaft is designed as $m_B r_B$ to avoid the secondary vibration from the engine, the relation between m_R and $m_B r_B$ can be formulated over the angular velocity of crankshaft, ω_C ($d\theta_c/dt$) as shown in Eq. (1) [5–8].

$$\begin{aligned} &4m_R\left(\frac{{r_C}^2}{l_P}\right){\omega_C}^2 = 2\left(m_B r_B (2\omega_C)^2\right) \\ &\Rightarrow m_B r_B = \frac{m_R {r_C}^2}{2 l_P} \end{aligned} \tag{1}$$

The structural fragility of a balance shaft module stems from the unbalance on the shaft that is an indispensible structural part for counter harmonic excitations. Such the unbalance causes not only the considerable bending deformation of the shaft but also the measurable driving torque consumption during operation. The former may cause a rub or interference between the rotor and the support bearing, and the latter decreases the fuel efficiency of the vehicle [5–7].

This paper presents an optimal conceptual design of a balance shaft by determining locations of both unbalance and supporting bearing. It begins with the formulation of an objective function that consists of the elastic strain energy and the kinematic energy of the balance shaft. Then, an optimal designed is derived with a function of both the mass ratio between symmetric and asymmetric part of the shaft and with load capacity of the support bearing. This study excludes any external force induced or transmitted by a balance shaft, so the analytical models are supposed to express a free-free condition [22–26]. In addition, the two support bearings are assumed to symmetrically be located with respect to the geometric center of a balance shaft. Finally, we perform a simulation of a balance shaft for a specific target-vehicle engine and demonstrate that the derived optimal design agree well with the simulation results.

2. Optimal design formulation of balance shaft

2.1. Formulation of objective function

An optimum balance shaft design starts from the definition of objective function that should seamlessly address the two main problems such as the bending deformation of the rotating shaft and the driving torque consumption. The objective function is derived with kinetic and strain energy formula, which is explained using a simple rotor model shown in Fig. 2.