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Dynamic behaviour and measurement accuracy of a bicycle brake hood force transducer

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Abstract

To measure road bike load inputs, it is customary to manufacture and instrument a bike component. The purpose of this paper is to investigate the influence of the dynamic behaviour of a brake hood force transducer on measurement accuracy. The majority of the studies reported in the literature on the development of transducers were designed with the objective of having high transducer stiffness in order to increase its first natural frequency to the highest possible degree. This paper shows that, in the case of force transmitted at cyclist's hands, this objective does not lead to the desired results. Rather, the best results are obtained when the natural frequency of the instrumented fabricated component matches that of its standard counterpart.

Keywords: Bike; vibration; brake hood force transducer; dynamic behaviour; accuracy

1. Introduction

Ride quality has become one of the most sought-after characteristics of a road bicycle by customers as well as by bicycle manufacturers. Vibration generated by road surface defects and transmitted by the bicycle to the hands and the buttocks is a significant source of discomfort, fatigue and disincentive to ride. In this context, in order to investigate the *in situ* vibrational transmissibility characteristics of a road bicycle, the dynamic loads and acceleration measurements at the cyclist's hands and buttocks are

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required. To develop an appropriate bicycle transducer for this application, the measurement issues surrounding the dynamic behaviour of the transducer itself must be fully understood.

Although several bicycle load transducers in the form of instrumented bicycle parts have already been described in the literature [1-6], an in-depth investigation of the influence of these transducers' dynamic behaviour on load measurement accuracy has yet to be presented. The purpose of this paper is to address some of the key whys and wherefores of the dynamic behaviour of a road bicycle load transducer. This will be accomplished through the description of an instrumented brake hood transducer that allows the measurement of vertical dynamic force at the cyclist's hands. In the case of a base excitation problem, we will show that the common practice of increasing the stiffness of an instrumented bicycle part in order to push its first natural frequency away from the upper limit of the frequency range of interest is in fact detrimental to measurement accuracy.

2. Methods

This section is divided into two parts: the first section is devoted to a general mechanical description of the brake hood force transducer (BHFT) including elements related to static calibration; and the second describes key steps to establishing the influence of the dynamic behaviour of the BHFT on measurement accuracy.

2.1. Mechanical description

The BHFT consists of two principal mechanical components: an instrumented body and a hand rest (Fig. 1 and 2). Both components are made of 6061-T6 aluminium. The instrumented body is a hollow cylinder with a handlebar mating shape at one end. The hand rest is attached to the free end of the instrumented body. The total mass of the BHFT is 312 g.

The BHFT can be fitted on a standard road bicycle handlebar like a standard brake hood. The vertical force F_z applied by the cyclist's hand to the hand rest is measured using four strain gauges arranged in a full Wheatstone bridge configuration. Theoretically, the position of the strain gauges and their interconnection give bridge signals that are independent of the location of the measured force as well as insensitive to other load components. Static calibration was performed by applying force and moment loads at the end the hub axles to measure the direct sensitivity and the non-calibratable cross-sensitivities (Rowe et al. [2]). The calibratable sensitivity (V/N, normalized for gain and input voltage) is $V_z = 0.165F_z$.

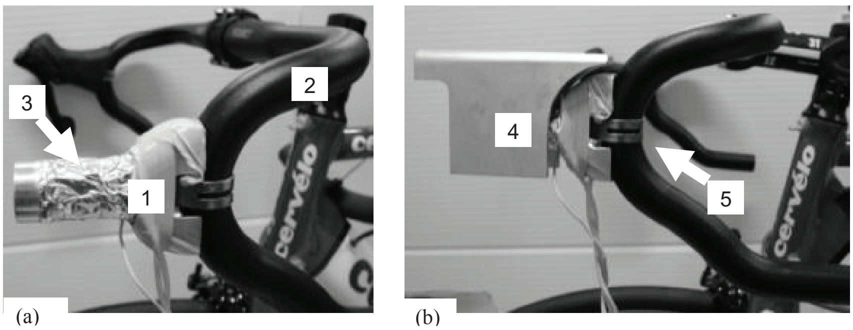

Fig. 1. Photographs of the brake hood force transducer. (a) Instrumented body mounted on handlebar and (b) hand rest covering the instrumented body (1- Transducer instrumented body, 2- Handlebar, 3- Aluminium foil covering strain gauges, 4- Hand rest, 5- Fixing clamp)

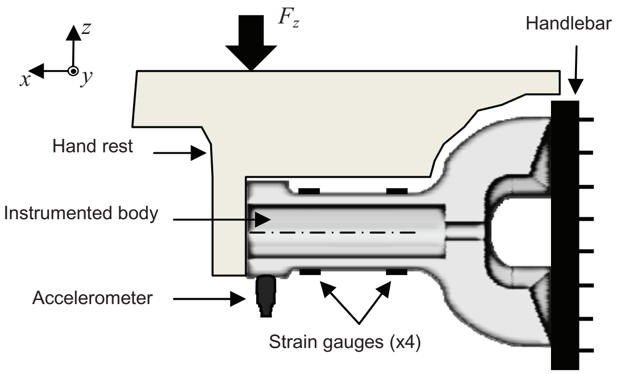

Fig. 2. Sectional view of brake hood force transducer. Strain gauges locations are indicated

2.2. Dynamic behaviour

In order to establish the influence of the dynamic behaviour of the BHFT on measurement accuracy, a three-step approach was used. First, an analytical model of a brake hood was devised. Next, experimental measurements of brake hoods were performed to determine the analytical model parameters. Two brake hoods were experimentally tested: the BHFT and a standard brake hood (Shimano model 105). Finally, the analytical model was used to analyse and compare the dynamic behaviour of both brake hoods.

Analytical model: Road induced vibrations transmitted to the cyclist's hands at brake hoods is a base excitation (or displacement transmissibility) problem. This is illustrated in Fig. 3 where $X(\omega)$ and $Y(\omega)$ represent the vertical displacement of the hand side and the handlebar side of the brake hood, respectively. For analytical analysis purposes and for sake of simplicity, it is proposed in this paper to model the brake hoods as a one degree of freedom (DOF) cantilever beam model. The beam has a stiffness k and a damping coefficient c. A mass m corresponding to that of a hand is suspended at its free end. It is assumed (1) that the transducer strain is proportional to the relative displacement Frequency Response Function (FRF) $H_r(\omega) = X(\omega)/Y(\omega)$ and (2) that the force measurement provided by the BHFT corresponds to the transmitted force $F_s(\omega)$ applied by the spring to the mass (Fig. 3c). By measuring the accelerance FRF $H_a(\omega)$ using the test setup described in the next section, it will be possible to estimate the model parameters m, k and c, and to calculate H_r and F_s normalized by $Y(\omega)$ as follows:

$$H_r(\omega) = X(\omega)/Y(\omega) = \left[(k^2 + (c\omega)^2)/((k - m\omega^2)^2 + (c\omega)^2)\right]^{1/2} \quad (1)$$

$$F_s(\omega)/Y(\omega) = k\left[1 - H_r(\omega)\right] \quad (2)$$

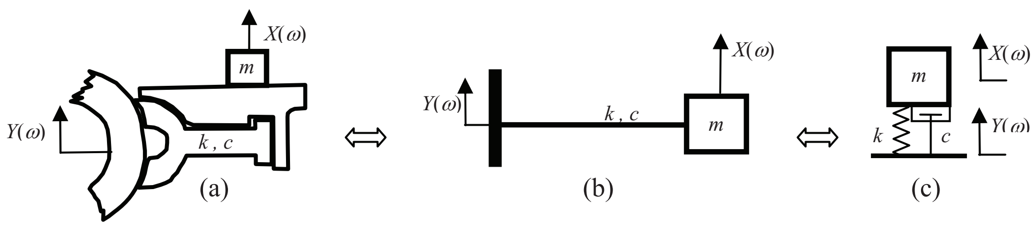

Fig. 3. Model of brake hood base excitation problem ((a) brake hood fitted to handlebar; (b) one DOF cantilever beam model; (c) spring-mass-damper system equivalent to (b))

Experimental estimation of brake hood dynamic characteristics: This section explains the experimental setup and the procedure used to obtain the BHFT parameters m, k and c. All the moving parts of the Shimano brake hood were removed and only the main body attached to the handle bar was kept for testing. The dynamic response of both brake hoods was measured using the setup shown in Fig. 4. A 0.5 kg mass was attached to the free end of both brake hoods to mimic mass loading of the hand. To enforce fixed-free boundary conditions for the brake hoods, they were bolted to a 50 kg steel block secured to a heavy steel bench. A shaker (Brüel and Kjær model 4809) suspended by bungees was used for brake hood free end vertical excitation. The input excitation force F(ω) was measured with a uniaxial piezoelectric force transducer (PCB model 208C04). A uniaxial accelerometer (PCB model 353B18) was fixed near the free end of the brake hoods to provide their dynamic acceleration output response A(ω). A data acquisition system consisting of LMS SCADAS recorder (model SCR01-08B) and LMS Test.Lab software was used for recording and analysing the signals.

A 10-500 Hz burst random signal generated by the LMS system was used to drive the shaker. A sampling frequency of 4096 Hz was set with a frequency resolution of 0.5 Hz. Proper excitation signal duration adjustment allowed the use of uniform time windows while avoiding frequency leakage. Power and cross-spectrum of both signals were measured and the accelerance Ha(ω) = A(ω)/F(ω) was calculated. Fig. 5 shows the accelerance measured on both brake hoods. The first peak of each curve corresponds to their respective first mode with a natural frequency fn at 144 Hz for the standard hood and at 202 Hz for the BHFT. This shows that the BHFT stiffness is higher than the stiffness of the standard brake hood. Using Polymax LMS modal parameter estimator, the model characteristics for both brake hoods were obtained and the results are presented in Table 1. The model predictions are also shown in Fig. 5.

Table 1. Model parameters for standard brake hood and brake hood force transducer

Brake hood type	*m* (kg)	*k* (N/mm)	*c* (Ns/m)	f_n (Hz)
Standard	4.0	3270	137	144
BHFT	3.8	6140	134	202

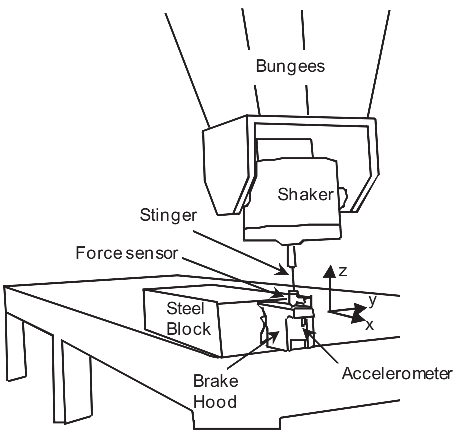

Fig. 4. Experimental setup for measurement of brake hood dynamic response

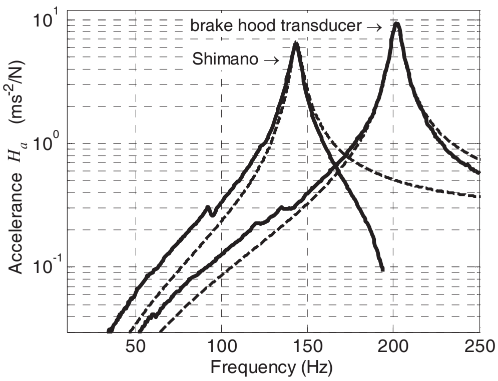

Fig. 5. Accelerance of Shimano brake hood and brake hood force transducer (- - - - - - model; ▬▬▬ experimental measurements)

3. Results

Using previously estimated model parameters of both brake hoods (Table 1), their respective transmitted force F_s can be calculated using Eq. 2. Note that F_s for the BHFT corresponds to the measured force. The force ratio R_f between the BHFT and the standard brake hood can also be calculated as well and the results are shown in Fig. 6 (solid line). The ideal situation would be to obtain a value of 1 at all frequencies. This would be obtained if both brake hoods had the exact same dynamic behaviour. As shown by the solid line curve in Fig. 6, the measured force systematically underestimates the actual transmitted force. The discrepancy augments with frequency and is equal to 0.62 at 100 Hz. At low frequency where the dynamic behaviour is not dominant, the ratio is lower than 1. This is related to the mass difference between both hoods.

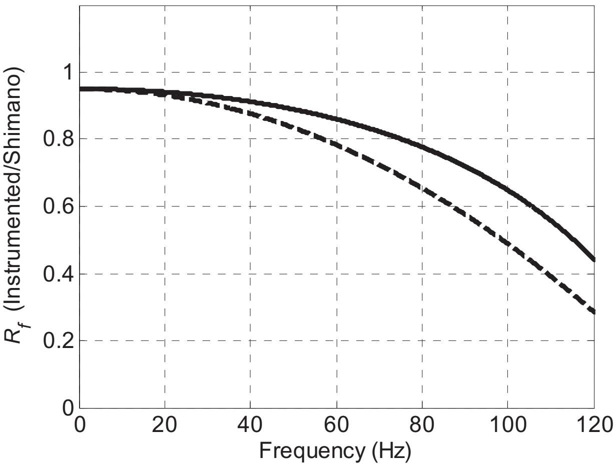

Fig. 6. Force ratio R_f between measured force by the BHFT and the standard brake hood (——— actual BHFT; - - - - - - stiffer BHFT)

In most published papers on bicycle component transducer development, the design goal is to get the first natural frequency of the transducer as high as possible, which is as far away as possible from the upper limit of the frequency range of interest. This is obtained by designing an instrumented body as stiff as possible. In order to test the validity of this approach, the model stiffness of the instrumented brake hood was increased by a factor of 100. A new force Ratio R_f was calculated and is shown in Fig 6 (dashed line). The discrepancy is augmented, indicating that this design strategy is not appropriate.

4. Discussion

In this paper, a one degree of freedom model was used in order to develop a comparison basis between the BHFT and its standard counterpart. This model is obviously a simplification of the real behaviour of a brake hood. For the real component, the mass is distributed over all the length of the brake hood and is not localised at the brake hood's free end as assumed in the model. The testing procedure to evaluate the model parameters also raises some issues. The hoods are both clamped to a steel block using their anchor bolt. Getting perfect clamping conditions is very difficult and one can expect that there might be some local effect at the contact matting surface between the brake hoods and the steel block. The results were however very consistent when controlling the anchor bolt torque. The real mass of both hoods is less than

1 kg. The modal mass is approximately 4 kg, which shows that a one DOF model is an approximation of the real behaviour of the tested parts. Nevertheless, the Frequency Response Functions showed a well-defined first natural frequency. Animation of this first mode also shows a cantilever type mode shape. These results support the idea that the one DOF assumption allows us to adequately investigate the dynamic behaviour of the brake hoods.

Using a strain gauge instrumented bicycle component with the same dynamic behaviour as that of its standard counterpart appears to be the best way to obtain the same system behaviour. At first glance, the fact that an instrumented bicycle component experiences dynamic amplification is likely to be surprising and probably counterintuitive. Fabricating a stiff instrumented bicycle component to push away the first natural frequency from the upper limit of the frequency range of interest is a design misconception for a vibration base excitation problem. Matching the dynamic behaviours of instrumented bicycle component and of its standard counterpart is therefore a design objective that will allow accurate dynamic load measurements.

5. Conclusion

In this paper, we investigated the influence of the dynamic behaviour of a brake hood force transducer on measurement accuracy. We emphasized that the vibration loads transmitted to the hands constitutes a base excitation problem. It was shown that in this case and to obtain accurate dynamic load measurements, the dynamic behaviour of a bicycle load transducer (in the form of instrumented bicycle part) must match that of its standard counterpart. This result is a contradiction of the traditional design objective in the literature for instrumented bike components. Finally, this recommendation can be extended to other fields of application with a similar instrumentation strategy.

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