Acta of Bioengineering and Biomechanics
Vol. 19, No. 3, 2017

Original paper
DOI: 10.5277//abb-00659-2016-03

Is bearing resistance negligible during wheelchair locomotion? Design and validation of a testing device

JOSEPH BASCOU[1, 2]*, CHRISTOPHE SAURET[2], FRANCOIS LAVASTE[2], HÉLÈNE PILLET[2]

[1] Institution Nationale des Invalides, Centre d'Etudes et de Recherche sur l'Appareillage des Handicapés, Woippy, France.
[2] Arts et Métiers, Institut de Biomécanique Humaine Georges Charpak, Paris, France.

Purpose: Among the different resistances occurring during wheelchair locomotion and that limit the user autonomy, bearing resistance is generally neglected, based on a few studies carried out in static conditions and by manufacturer's assertion. Therefore, no special attention is generally paid to the mounting and the maintenance of manual wheelchair bearings. However, the effect of inadequate mounting or maintenance on wheelchair bearing resistance has still to be clarified. This study aimed at filling this gap by developing and validating a specific device allowing the measurement of wheelchair bearing friction, characterized by low speed velocities, with an accuracy lower than 0.003 Nm.

Methods: The bearing resistance measured by the device was compared to free deceleration measurement, intra and inter operator reproducibility were assessed. A factorial experiment allowed the effects of various functioning parameters (axial and radial loads, velocity) to be classified.

Results: The device allowed significant differences in the bearing resistance of static and rotating conditions to be measured, even if a relatively high proportionality was found between both conditions. The factorial experiment allowed the expected impact of the radial load on bearing resistance as well as the predominant effect of the axial load to be demonstrated.

Conclusions: As a consequence, it appeared that the control of the axial load is compulsory for measurement purposes or during wheel mounting, to avoid significant increase of global resistance during wheelchair locomotion. The findings of this study could help enhancing the models which assess manual wheelchair mechanical power from its settings and use conditions.

Key words: *wheelchair, bearing resistance, radial load, setting influence, axial load, rotation velocity*

1. Introduction

The mechanical work necessary for the locomotion with a manual wheelchair (MWC) is produced by the user, mainly by the muscles of their upper limbs. This mechanical work allows the acceleration (or deceleration) of the MWC or the elevation of the center of mass when climbing a slope or a kerb. But a non-negligible part of the work produced is also dissipated by various resistances acting on the MWC, namely, rolling, bearing, turning and air resistance [2], [7]. These resistances result in unwanted energy losses in almost all daily and sport situations and potentially lead to upper limb joint overload. Musculoskeletal pain and long term injuries classically [16] related to joint overloading often result in the limitation of the user autonomy.

Among the above-mentioned resistances, the bearing resistance torque was quantified in static conditions to 0.1 Nm for a radial load of 100 N according to the results of Frank and Abel [5]. At the same time, bearing resistance has also been quantified by various authors [9], [10] for other applications than wheelchairs (aerospace, manufacturing machine, etc.), with rotational velocities up to 44 000 rotations per minute (rpm). Using online estimator programs and empirical formulas provided by manufacturers [12], [13], bearing friction torque could also be evaluated between 0.001 and 0.1 Nm for a radial load of 100 N. Using

* Corresponding author: Joseph Bascou, Institution Nationale des Invalides, Centre d'Etudes et de Recherche sur l'Appareillage des Handicapés, 47 rue de l'Echat, 94000 Créteil, France. Phone: 00 33 1 41 78 52 53, e-mail: joseph.bascou@invalides.fr

Received: May 27th, 2016
Accepted for publication: September 12th, 2016

these values, the power lost in bearings can be estimated between 0.017 and 1.7 W for a 6 cm radius wheel while the wheelchair is rolling at 1 m/s. As a comparison other resistances such as rolling resistance range from 2 to 15 W [11]. Finally, in a study focused on MWC propulsion modeling, (based on the work of Cooper [2]), Hofstad and Patterson [7] tested the influence of various parameters of the model on the tangential force that need to be applied on the wheelchair handrims for propulsion. This model included bearing resistance, but the values of bearing resistance were not reported. In the previous literature specifically focused on MWC, the bearing resistance has either been related to the angular velocity [2], [7] or to the radial load [5]. Concerning the axial load, its effect seems not to be reported in any paper, though it is mentioned by some manufacturers. Besides, if all these parameters are expected to increase the bearing resistance, their relative influences do not appear to have been investigated.

Based on these studies, bearing resistance is usually neglected when modeling MWC propulsion in daily life situations [2], [6], [11]. However, the testing conditions in the studies presented may not be exactly representative of MWC locomotion. Indeed, the experiments proposed by Frank and Abel [5] were limited to the assessment of the bearing friction torque under static conditions (which is easier to measure than rotating bearing friction torque) and without axial loading, whereas experiments conducted by [9], [10] were not representative of the MWC range of use, where rotational velocity does not exceed 2500 rpm, even for small front casters during sport activities. The results provided by manufacturers are also given for optimal functioning of the bearings (perfectly clean environment, perfect mounting, etc.), which can be slightly different from the actual functioning, with the bearings placed in the wheel hub and subject to daily life environment. Finally, neither Cooper [2] nor Hofstad and Patterson [7] reported the coefficients they used for calculating the bearing friction torques, which were also assessed along with other resistances (air, rolling resistance, etc.) through modelling only.

Therefore, adequate measures of MWC bearing resistance during motion in MWC daily life conditions are not currently available in the existing literature to the authors' knowledge. As a consequence, there is a lack of knowledge regarding the effect on bearing resistance of both abrasion and soiling of ball bearings, resulting from the use of an MWC. Lastly, the effect of the axial load, which could result from an inclination of the wheels such as for cambered wheels, needs to be clarified. This knowledge could lead to recommendations for both design and maintenance of MWCs.

In this framework, the primary purpose of this study was to propose and validate an adequate experimental device and protocol to quantify the frictional torque of MWC bearings in realistic conditions, that is to say, mounted on a wheel hub, during wheel rotation, and to define a range for bearing friction torques. The second goal was to validate a posteriori the underlying assumption made by Frank and Abel [5] that bearing friction torque measured under static conditions can be used to approximate the actual bearing friction torque, occurring during the daily use of an MWC. The third goal of this study was to evaluate how much radial and axial loads, related to the MWC settings and its maintenance, impact the ball bearing resistance. Overall, this study aims at providing quantitative data which will be potentially useful for the modeling of MWC propulsion, as well as objective information on the impact of the type of load on frictional torque that could have potential implications in ball bearing selection to facilitate MWC propulsion.

2. Material and methods

To achieve the objectives of the study, it was necessary to develop a custom device allowing the measurement of bearing friction torque while the wheel is turning around its rotation axle. This testing device was designed to be able to alter both radial and axial loads applied on the ball bearing as well as controlling the wheel rotation velocity.

2.1. Testing device and associated measurement protocol

The testing device (Fig. 1a) was a rigid frame allowing the rotation of a wheel around a fixed horizontal axis passing through the wheel center. To ensure the rotation of both wheel and bearings, the wheel was placed on a driving roller controlled by a 100 W DC motor that allowed maintenance of a constant wheel rotational velocity. The wheel was held in the device by two other rollers. The three rollers were built with concave shapes in order to increase the stability of the wheel in the testing device.

Radial loading of the bearings was performed using a specific axle (Fig. 1b, cut section Fig. 2) passing through the bearings, composed of two parts linked by

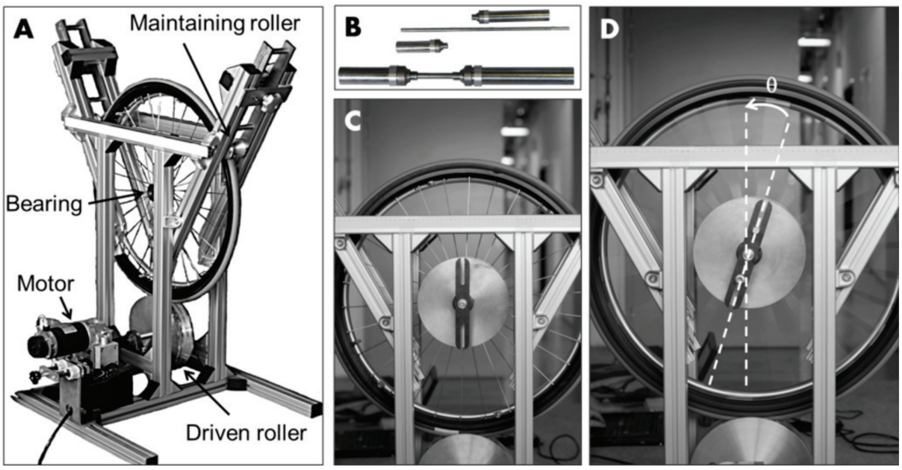

Fig. 1. (a) Global view of the testing device; (b) photograph of the specific axle; (c) photograph of the loaded bearings after placement and alignment of the slotted-plate; and (d) inclination (θ) of the slotted-plate at equilibrium during wheel rotation

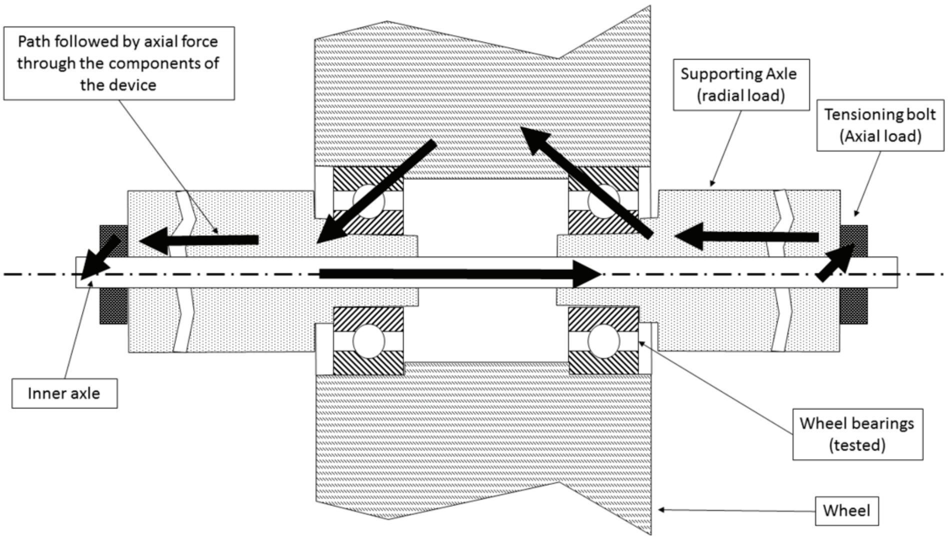

Fig. 2. Cut section of supporting axle and path followed by axial force

an inner axle and designed to support several discs of 2.45 kg each. In total, the maximal radial load on the bearings was created by adding an overall mass of 30.05 kg, including the mass of 12 discs and the mass of the specific supporting axle, and was distributed for half on each side of the wheel. The radial load applied on the bearings was easily determined from the total weight of the loading discs and the supporting axle inferred from their respective masses.

The design of the supporting axle also allowed the axial load to be modified by means of a tensioning bolt (Fig. 2) and controlled through a manual torque wrench (Rahsol Dremotec, Gedore, Germany, 1–12 Nm). The axial force is transmitted to the inner ring of one bearing through the first part of the supporting axle, then to the outer ring of this bearing, the wheel, and the outer ring of the second bearing; it is then transmitted to the inner ring of the second bearing, to the second part of supporting axle, to the inner axle of the supporting axle and then to the tensioning bolt (Fig. 2). The tightening torque was then used to determine the axial load from equation (1), described in the ISO 16047 standard (based on [4])

$$T = F_A\left(0.5\frac{P + 1.154\pi\mu_{th}d_2}{\pi - 1.154\mu_{th}\dfrac{P}{d_2}} + \mu_b\frac{D_0 + D_h}{4}\right), \quad (1)$$

where T is the tightening torque (in Nm); F_A is the axial load (in N); P is the thread pitch (in m); $(D_0 + D_h)/4$ is the effective radius of bolt head contact (in m); μ_b and μ_{th} are the bolt head and thread friction coefficients, respectively (without unit); and d_2 is the effective diameter of thread contact (in m), which is the half of the thread-pitch diameter.

Characteristics of our tensioning loading mechanism (P = 0.001 m; D_h = 0.01 m; D_0 = 0.0065 m; d_2 = 0.00535 m and $\mu_b = \mu_{th} = 0.2$), allow equation (1) to be written as:

$$F_A \approx 620\,T\,. \tag{2}$$

In order to balance the misalignment between the loading disc center of mass and rotation axis, a static balance procedure of the device was defined. To this aim, a lightweight rectangular slotted-plate was attached symmetrically in a plane normal to the rotation axis (Fig. 1c). When the system was static (i.e., the disc center of mass was under the rotation axis), the slotted-plate was manually aligned with the vertical axis using a laser vertical alignment device (Mark 2 LC, David White, USA) and a counterweight was placed in its upper slot. The position and weight of the counterweight were chosen in order to equilibrate the system: whatever the rotation angle of the discs, no movement should be observed in the equilibrated system. In our experiments, the balance was obtained by placing the counterweight, which had a mass in the range from 7 to 20 grams, at a distance ranging from 4 to 8 cm from the rotation axis.

At this step, the external ring of the ball bearing was linked to the wheel, whereas the internal ring was fixed to the axle supporting the radial load. Under static conditions, both the wheel and the support axle were stationary. In order to measure the static friction torque, an opposite torque was created by placing a weight pawn at a distance l from the axis of rotation in the lower slot of the slotted-plate. Then the axle was gradually turned until the static equilibrium was maintained. The bearing friction torque (T_b) was then calculated from the following expression

$$T_b = mgl\sin\theta\,, \tag{3}$$

where m is the weight pawn mass (in kg); g is the gravitational acceleration (in m.s^{-2}); l is the distance from the axis of rotation to the weight pawn (in m) and θ is the inclination of the slotted-plate with respect to the vertical (in radians).

The mass (m) and the distance (l) were chosen in order to obtain equilibrium with an inclination ranging from 20 to 70° (Fig. 1d**)**. The distance l was measured with a tape measure and the slotted-plate inclination (θ) using a digital inclinometer (SPI Pro 3600 31-040-9, Swiss Precision Instrument, Switzerland).

During wheel rotation, the dynamic bearing friction torque tended to rotate the axle/discs system. The dynamic bearing friction was measured using the same procedure as for static friction torque: the weight pawn position was changed until the axle/discs no longer rotated, meaning that the dynamic bearing friction torque was compensated. The dynamic bearing friction torque was also calculated using equation (3).

2.2. Accuracy and reproducibility

To validate the measurement protocol, the results were tested against another methodology for the determination of the bearing friction torque. Bearing friction torque could be inferred from the measurement of the angular deceleration of the loaded axle with respect to the wheel, knowing the moment of inertia of the loaded axle. For that purpose, a marker-based optoelectronic system (Vicon V8i, Oxford Metrics, UK), composed of 8 cameras, was used at a working rate of 100 Hz. Four reflective markers were glued on the loading discs and six on the wheel (Fig. 3), allowing angular parameters (positions, velocities and accelerations) to be calculated for both the discs and the wheel. Although only 3 markers would have been sufficient to calculate these quantities, the additional markers assisted in preventing marker occlusion (concealed by the testing device) and in improving data accuracy. For these experiments, the wheel was rotated at a constant velocity imposed by the motor. The loaded axle was left free to accelerate thanks to the bearing friction torque until reaching, after approximately half a minute, an angular velocity close to the wheel angular velocity. Then, the wheel rotation was stopped by cutting off the motor and the loaded axle was left free to decelerate, solely under the action of the bearing friction torque. In this case, the bearing resistance torque was calculated using the following expression

$$T_b = I\ddot{\theta}\,, \tag{4}$$

where I is the mass moment of inertia of the loaded axle (in kg·m^2) and $\ddot{\theta}$ is the angular deceleration of the discs with respect to the wheel (in rad·s^{-2}).

The bearing friction torque was determined using both methodologies (deceleration test and testing device) using 6 repeated series of measurements for each test type. Both static and "dynamic" measurements were performed with the testing device. For every series, the wheel velocity and the radial load were identical (4.4 rad.s^{-1} and 154 N, respectively) whereas the axial load was increased via the tensioning bolt (preload torques of 0, 1, 3, then 5 Nm, resulting in axial loads of approximately 0, 620, 1860 and 3100 N). Due to the manual adjustment of the torque wrench, the precision of the axial preload torque was about 0.3 Nm (about 180 N of axial load).

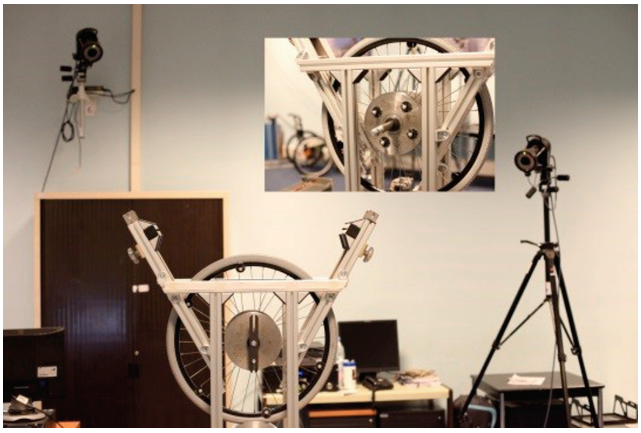

Fig. 3. Photographs of both the wheel and the discs equipped with reflective markers

Statistically, the accuracy of the testing device with respect to the deceleration methodology was evaluated by means of (1) the coefficient of determination (r^2), (2) a Wilcoxon signed-rank test ($\alpha = 0.05$) to test for statistical difference between both measurement series and (3) the standard error of the estimate (SEE) to quantify the gap between both measures. The same statistical procedure was performed between static and "dynamic" measurements with the testing device.

After evaluating the accuracy of the current testing device and the associated protocol, the reproducibility of the method was evaluated. First, an experimenter performed a series of 10 trials preserving both axial and radial loads, and the balance. During these 10 trials, only the weight pawn and its position from the axis of rotation were changed, inducing changes of the slotted-plate inclination to reach equilibrium. Second, the same experimenter performed the whole procedure, including balance and loading, 10 times. Finally, a second experimenter also performed the whole procedure 10 times. For all these measurements, the velocity, radial load and the axial load remained unchanged: 4.4 rad·s^{-1} for velocity (42 rpm), 154 N for the radial load and an axial preload torque of 1 Nm. These data were used to assess the intra-operator repeatability of the measurement method and both intra and inter-operator reproducibility of the complete procedure.

Sample normality was tested through Shapiro–Wilk tests. Both procedure- and operator-dependences were evaluated using the Mann Whitney U-tests ($\alpha = 0.05$) due to the small samples sizes ($n = 10$). For all tests, the significant level was set to $\alpha = 0.05$.

2.3. Loads and velocity effects on bearing resistance

In an attempt to assess the relative influence of various parameters such as axial load, radial load and velocity, a full factorial experiment [1], [15] was performed on the same pair of ball bearings (608-2RS, LFD Wälzlager GmbH, Dortmund, Germany) mounted in a wheel hub without spacer. Every factor (radial load, axial load and angular velocity) was evaluated using two levels, assuming a linear relationship with the bearing friction torque. The levels were chosen to be as close as possible to usual manual wheelchair conditions. The usual manual wheelchair conditions were considered to be: 100 kg for the {User+MWC} total mass; 50 to 80 % of the total weight on drive wheels (resulting in a radial load ranging from 250 to 400 N on every drive wheel, and ranging from 100 to 250 N on every front wheel); velocity ranging from 0.5 to 2 m/s, resulting in angular velocities ranging from 1.6 to 6.7 rad/s for a 0.6 m diameter drive wheel, and from 8.3 to 33.3 rad/s for 0.12 m diameter caster. The radial load could be imposed by the drive wheel camber (max = 170 N, for 22° camber angle with 450 N applied vertically on the axle) or a screw torque. The screw torque range was estimated between 0 and 3 Nm: the "zero" value corresponded to the use of a quick release system (which imposes a space between the released ball and the bearing inner ring) and the 3 Nm value was tested by the authors and considered plausible. However, all conditions could not be fulfilled due to limitations in the testing device possibilities the safety aspects. Five measurements for every conditions (total: $5 \times 2^3 = 40$ measurements) were performed, including the balancing procedure. The respective lower and higher levels were chosen as: 106 N and 297 N for the radial load; 1 Nm and 3 Nm of axial preload torque for the axial load (resulting in axial force of about 620 and 3100 N; a minimum torque of 1 Nm was chosen because a 0 N axial load was difficult to obtain considering the residual frictions between the parts of the supporting axle); and 2.6 rad·s^{-1} (24 rpm) and 4.4 rad.s^{-1} (42 rpm) for the angular velocity. These angular velocities applied to the rear wheels corresponded to MWC linear velocities of 0.85 and 1.45 m·s^{-1}, respectively. Unfortunately, the higher velocity was imposed by the maximal power of the motor, which was severely limited by the high rolling resistance that occurred between the wheel and the three concave rollers. In addition, for safety reasons, bearings were not tested on caster wheels, but the authors tried to reproduce their functioning conditions (higher velocity, lighter radial load and higher axial load than drive wheels). Furthermore, to avoid the uncertainties due to the manual torque wrench setting, all the conditions requiring the same axial load were performed successively.

To map the results of the factorial experiment, the values −1 or +1 were attributed for the lower and higher levels of each factor defined above. A level was also attributed for factor combinations by multiplying the value attributed to each combined factor. Thus, the resultant bearing friction torque (T_b) can be expressed by the following multilinear equation

$$T_b = a + bA + cR + dV + eAR + fAV + gRV + hARV\,, \quad (5)$$

where A, R and V are the levels (–1 or +1) of the axial load, radial load and velocity, respectively; and a, b, c d, e, f, g and h are the coefficients weighting the level of the factors and combinations of factors.

From the 8 experimental conditions which were performed, a system comprising Eq. (5) written 8 times was then available and included 8 unknown variables, which are the weighting coefficients. This system can then be written in a matrix form as follows:

$$\underbrace{[M_B]}_{8\times1} = \underbrace{[M_E]}_{8\times8}\underbrace{[M_C]}_{8\times1}, \quad (6)$$

where $[M_B]$ is the matrix containing the bearing friction torque measured for the 8 experimental conditions; $[M_C] = [a, b, c, d, e, f, g, h]^T$ is the unknown matrix containing the coefficients weighting the individual and combined effects of the different factors; and $[M_E]$ is the matrix that describes the conditions of each experiment

$$[M_E] = \begin{bmatrix} +1 & -1 & -1 & -1 & +1 & +1 & 1 & -1 \\ +1 & -1 & -1 & -1 & 1 & -1 & -1 & +1 \\ +1 & -1 & +1 & +1 & -1 & +1 & -1 & +1 \\ +1 & -1 & +1 & -1 & -1 & -1 & +1 & -1 \\ +1 & +1 & +1 & +1 & +1 & -1 & -1 & -1 \\ +1 & +1 & +1 & -1 & +1 & +1 & +1 & +1 \\ +1 & +1 & -1 & +1 & -1 & -1 & +1 & +1 \\ +1 & +1 & -1 & -1 & -1 & +1 & -1 & -1 \end{bmatrix}, \quad (7)$$

where each row corresponds to one combination of experimental conditions (here, 8 combinations) and where the columns correspond to the level (–1 or +1) of: (1) the mean overall torque value from all experiments (always +1), (2) the axial load (A), (3) the radial load (R), (4) the angular velocity (V), and the interaction terms: (5) axial and radial loads (A R), (6) axial load and velocity (A V), (7) radial load and velocity (R V) and (8) axial load, radial load and velocity (A R V).

Finally, the weighting coefficients included in $[M_C]$ can be computed from the following expression

$$[M_C] = [M_E]^{-1}[M_B]\,. \quad (8)$$

3. Results

3.1. Accuracy

Results obtained from the marker-based optoelectronic system showed a constant rotational deceleration of the loading disks for all trials ($r^2 > 0.98$) (Fig. 4), which allowed rotational decelerations to be computed from a linear regression. Within all trials (6 series of 4 different axial load conditions), decelerations ranged from 0.06 to 2.30 rad·s^{-2}. Considering the mass moment of inertia obtained from geometrical modeling and homogenous density (7.42×10^{-2} kg·m^2), bearing friction torques ranged from 4 to 170 Nmm. In comparison, results provided by the testing device exhibited differences with the validation tests, which used deceleration measurement, that reached 6.5 Nmm (mean difference: 2.3 Nmm) for the "dynamic" condition. It can also be noted that the arithmetic average of the differences was not null and equal to 1.1 Nmm. Statistical analysis showed a coefficient of determination (r^2) of 0.997 between results from the deceleration methodology and those from the testing device in "dynamic" condition (Fig. 5a). The Wilcoxon signed rank test did not show significant differences between results provided by both methods ($p = 0.130$). Finally, the SEE was 3.0 Nmm.

Comparison of the testing device in static condition with deceleration measurements showed differences that reached 44.6 Nmm (mean difference: 10.7 Nmm). Coefficient of determination (r^2) was 0.980 but the Wilcoxon signed-rank test showed significant differences between both methods ($p = 0.047$). SEE was 16.0 Nmm with respect to the deceleration methodology. Similar results were obtained between the static and the "dynamic" conditions using the testing device. Indeed, r^2 was 0.980 (Fig. 5b); the Wilcoxon signed-rank test also showed significant difference between both methods ($p < 0.001$); and SEE was 17.0 Nmm. It can also be noted that static measurement resulted in lower bearing friction torque than the one obtained in "dynamic" condition (22 times out of 24 measurements). On average, the bearing friction torque obtained during "dynamic" measurement was about 20% higher than the one under static condition. Assessing "dynamic" measurements from static data multiplied by 120% allowed the SEE to decrease until 8.7 Nmm. Finally, considering only trials with low axial preload torques (0 and 1 Nm), r^2 remained high (r^2 = 0.974), the Wilcoxon signed-rank test showed significant differences, but SEE dropped to 2.2 Nmm and to 2.0 Nmm when multiplying static data by 120%.

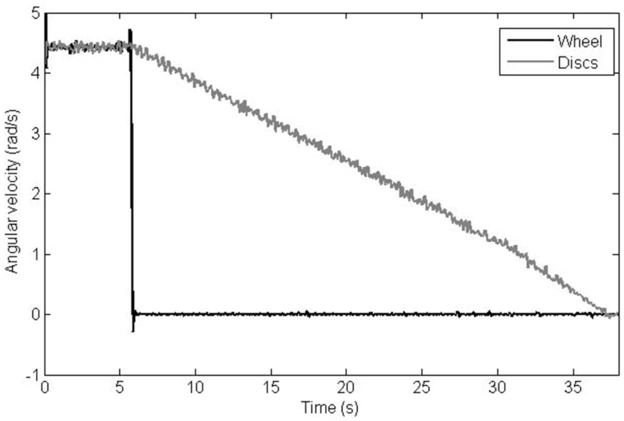

Fig. 4. Time course of both wheel and discs angular velocities computed using data from the marker-based optoelectronic system

6.6 Nmm (mean: 5.1 ± 0.9 Nmm). Hence, the two series did not exhibit high difference on average (0.1 Nmm) but the standard deviation of the second series was twice that of the first series. For each series, a normal distribution of the data was observed from the Shapiro–Wilk test: W = 0.94 and p = 0.58 in both cases. The Mann–Whitney U-test did not allowed the two samples to be differentiated (p = 0.82) even if their standard deviations differed.

Besides, even when decreasing the sample sizes until 5 trials, the Shapiro–Wilk test made it possible to conclude to normal distributions (first series: W = 0.84–0.95 and p = 0.10–0.71; second series: W = 0.94–0.91 and p = 0.69–0.40) and the Mann –Whitney U-test did not reveal significant differences between both series until reaching sample sizes of n = 5 (p = 0.20–0.82). In the same way, the Mann–Whitney U-test did not reveal any significant differences for every series between the complete sample and the reduced sample, until n = 5 (first series: p = 0.39–0.99; second series: p = 0.56–0.77) where the bearing friction torque was 5.0±0.4 Nmm for the first series and 4.9 ± 0.9 Nmm for the second one.

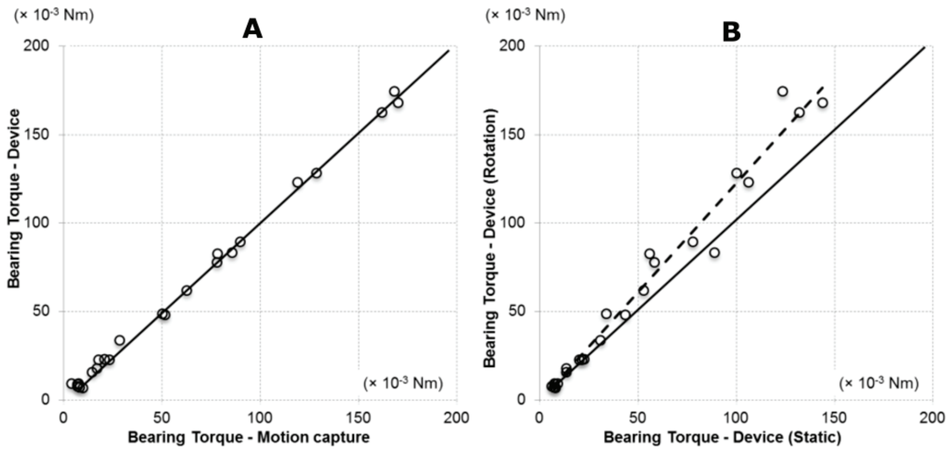

Fig. 5. Bearing friction torques (A) measured using the dedicated device in "dynamic" condition vs. calculated from the marker-based optoelectronic system; and (B) measured in static vs. "dynamic" condition using the dedicated device in both cases

3.2. Reproducibility

Within the first series of 10 repeated measures performed by the first experimenter without altering the balance and loading procedures, the bearing friction torque ranged from 4.3 to 5.9 Nmm (mean: 5.2 ± 0.4 Nmm). The second series performed by the same experimenter and including the whole procedure provided bearing friction torque ranging from 4.0 to 6.6 Nmm (mean: 5.1 ± 0.9 Nmm).

In the third series, performed by the second experimenter, bearing friction torques ranged from 3.0 to 6.5 Nmm (mean: 4.8 ± 1.0 Nmm). The Shapiro–Wilk test made it possible to conclude to a normal distribution of the data (W = 0.96 and p = 0.77). This was also the case for every sample larger than n = 5 (W = 0.86–0.97 and p = 0.23–0.86). The Mann –Whitney U-test did not allow samples from the two experimenters to be distinguished, for every sample size ranging from 5 to 10 (p = 0.47–0.93).

3.3. Effects of the type of load and rotational velocity

The 40 measures of the factorial experiment showed bearing friction torques ranging from 12.6 to 92.3 Nmm. Considering the mean value for every condition (Table 1), the bearing friction torques ranged from 15.9 to 82.5 Nmm for an overall average of 47.4 Nmm.

Table 1. Bearing friction torques obtained during the 8 series performed during the factorial experiment

Series No.	Level (A = Axial load, R = Radial load, V = Velocity)			Mean ±SD ($\times 10^{-3}$ Nm)
	A	R	V	
1	–1	–1	–1	15.9 (±2.4)
2	–1	–1	1	16.1 (±2.5)
3	–1	1	–1	26.7 (±9.4)
4	–1	1	1	27.4 (±8.9)
5	1	1	–1	81.6 (±9.9)
6	1	1	1	82.5 (±10.6)
7	1	–1	–1	63.5 (±7.1)
8	1	–1	1	65.5 (±8.0)

These data allowed determining the respective influences of the radial load, axial load, velocity and every possible interaction (Fig. 6). The main impact estimated to 25.9 Nmm was attributed to axial load, followed by the radial load even if 3 times lower (7.2 Nmm). The interaction of axial and radial loads was found at the third rank (1.6 Nmm) but its effect remained limited (16 and 4 times lower than the impact of axial and radial loads, respectively). The influence of rotational velocity was found at the fourth rank with a weighting value of 0.4 Nmm. Finally, the impacts of other interactions were lower than 0.3 Nmm.

The bearing resistance could then be computed for a set of velocity, axial and radial loads as follows:

$$M_{bearing} = 47.4 + 25.9 * [Axial\ load\ level]$$
$$+ 7.2 * [Radial\ load\ level]$$
$$+ 1.6 * [Axial\ load\ level] * [Radial\ load\ level], \quad (9)$$

where $M_{bearing}$ is the bearing resistance expressed in Nmm, [*Axial load level*] is the value representing the axial load (being –1 for 1 Nm screw torque, +1 for 3 Nm screw torque, 0 for mean axial load), [*Radial load level*] is the value representing the radial load (being –1 for 106 N radial load, +1 for 297 radial load, 0 for mean radial load of (106 + 297)/2 = 201.5 N). The levels for intermediate values of radial and axial load can be computed using proportionality. Ex: radial load level for 150 N radial load is –1 + (150 – 106)*2/(297 – 106) = –0.54.

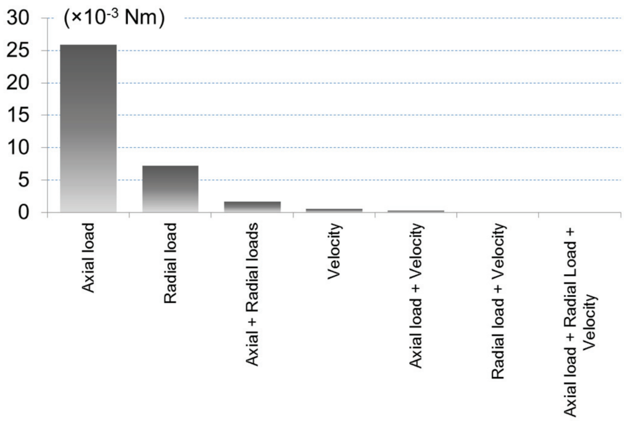

Fig. 6. Respective influences of the radial load, axial load, velocity and every possible interaction sorted out by importance

4. Discussion

Concerning the first goal of this study, which was to validate a device able to measure wheelchair bearing resistances, the results provided by our testing device in "dynamic" condition did not show significant differences from the ones from the reference methodology (deceleration methodology). This allowed confidence in the use of this device – with its associated protocol – to assess the friction torque of ball bearings mounted in a wheel hub. The accuracy of the measurement, estimated from the SEE, was 3.0 Nmm. However, a limited but non-null arithmetic difference was found (1.1 Nmm) between both mean values, revealing the presence of a systematic error. It is important to remind that the reference methodology (deceleration methodology) required the assessment of the mass moment of inertia of the loading discs and supporting axle. Thus, the observed systematic difference could result from an underestimation of 0.004 kg·m² of the moment of inertia, representing about 6% of the estimated mass moment of inertia.

As regards concerns reproducibility, the data from the two series of 10 repeated measures performed by the same experimenter exhibited normal distributions allowing a random noise on the data to be assumed. Besides, the fact that the standard deviation of the series concerning the whole procedure (balance + measure) was twice the standard deviation of the series without balance procedure allows the half of the noise to be attributed to the balance procedure and the other half to the measure itself. Thus, making several measurements without re-balancing would result in a systematic error. As a consequence of this analysis, data accuracy should be improved by carrying out several measurements (by re-balancing every time) for the same condition and considering the average value as the result of the measurement. The results obtained by decreasing the sample size indicated that 5 repeated measurements for each condition represent a good trade-off between time consumption (about 30 minutes) and accuracy improvement for future studies. Finally, as no significant difference was observed between data from the two experimenters, the possible operator-dependence of the measurement can be neglected, for every sample size from 5 to 10. The device then showed good intra and inter operator reproducibilities, considering at least 5 trials were made for each condition. It is important to note that the bearing resistances, obtained for each series of 5 measurements at 0, 1, 3 and 5 Nm of axial preload torque were scattered. These discrepancies can be attributed to the uncertainty in selecting the axial preload with the manual torque wrench used. It was then mandatory for the experimenters to limit the number of times this torque was changed during the factorial experiments to obtain reproducible results.

Concerning the second goal of the study which was to ascertain if static bearing resistance allows determining the bearing resistance in dynamic condition – our results demonstrated that this was not the case. Indeed, whereas static and dynamic measurements of bearing resistance showed good linear relationship, multiplying static resistance by 120% would be a better estimator of dynamic resistance, with a mean error of 2.2×10^{-3} Nm. However, this proportional coefficient could be specific to the bearings tested and ought to vary using other bearings. Finally, static bearing friction torque can only be considered as an approximation of the dynamic friction torque, but dynamic measurements are preferable for better accuracy.

Concerning the third goal of this study, which was to assess the influence of various parameters (axial load, radial load and velocity), the results demonstrated the critical influence of the axial load on bearing resistance. The radial load, which ranked at the second place, exhibited a noticeable impact on bearing resistance but its effect did not exceed the third of the axial load effects. Velocity did not exhibit a convincing effect on bearing resistance in this experiment, when the wheel was in motion. However, it is important to keep in mind that the calculated effects are directly dependent on the range of variation chosen for the different factors. Axial load conditions imposed were chosen to be 1 and 3 Nm: the “zero axial load” condition could be difficult to obtain due to residual axial forces often existed between the various parts of the supporting axle (Fig. 2), thus a minimum compression of the bearings was considered preferable for our experiments. Thus, the range of variation of bearing resistance would probably be greater than the one observed if the range of axial loads had been extended to 0–3 Nm instead of 1–3 Nm. The results could also be altered by the precision of the torque wrench that imposed axial load, though not sufficiently to alter the outcomes of this study. Maximum radial load imposed in the experiment (297 N) did not reach the highest value of the range of radial load experienced by MWC driving wheel (250 to 400 N for 100 kg distributed 60 to 80% on the rear wheels). So a higher effect could be expected if the range was extended to 100–400 N instead of 100–300 N. However, considering the respective effects of axial and radial loads, the change of the ranges of variation of these parameters would not challenge the conclusion of this study. For the velocity, it is likely that the range of variation of the experiment was too small to cause significant impact on bearing resistance. The maximal velocity used in this study (4.4 rad·s^{-1} or 42 rpm) was imposed by the maximal power delivered by the motor but remained close to the usual conditions of rear wheels. However, front casters would turn at higher velocities. Hence, motor with higher power-delivering capability and/or a decrease of rolling resistance by re-designing both driving and maintaining rollers (with a lower concave shape, for instance) would be necessary for further experiments including a greater range of rotation velocities. However, as the velocity effect was about 60 times lower than the axial load effect, its influence on the bearing resistance in the wheelchair velocity usual range of use may still be low compared to the axial and radial loads. Finally, all other interactions between factors exhibited negligible impacts on bearing friction torque.

Knowing the major effect of axial loading on bearing resistance, its control is compulsory for both experimental measurements and wheelchair setting. Concerning the measurement device, the axial load control was performed by means of a tensioning bolt using a manual torque wrench, making the precise control of the axial load difficult. In wheelchair practice, the control of the axial load of a wheel using a screw system could also lead to higher bearing resistance values, if not conducted with special care. The use of a brace – or spacer – between two bearings of a wheel to prevent excessive axial load could then be recommended for manufacturer, if the wheel has to be mounted rapidly or repetitively with no torque control. This solution is often used in practice for the mounting of front casters of sport or high-end wheelchairs. Spacers between inner rings of bearings avoid axial load only if the spacer is wider than the wheel (imposing a distance between outer rings of the bearings), which leads to an axial movement of the bearings. If the spacer is narrower than the wheel, axial load appears with tensioning bolt. So, the spacer needs to accurately fit the wheel hub width to limit axial load and axial movement, which means an additional cost for manufacturing explaining why no spacer, are provided with the wheels in many standard wheelchairs. Therefore, without a proper spacer, a trade-off between bearing resistance and caster fixation in the fork must be sought. On the contrary, this manual adjustment usually cannot be done for rear wheels, which often require to be easily put and removed. In many cases, rear wheels are mounted without spacer using a quick release system. This solution limits the axial loading but allows a little translation along the rotation axle. In few cases, a pre-set of the axial loading is done, but it increases greatly the cost of the wheel and the complexity of its manufacturing.

The results of the factorial experiment presented showed that bearing friction torque could be considered only as a function of both axial and radial loads, neglecting the impacts of velocity and every interaction factor. However, the experiments were conducted on a single wheel, mounted with a couple of standard bearings. Further experiments, involving more bearings with various properties, are still required in order to provide a simple tool similar to the one presented by Sauret et al. [11] for rolling resistance.

Besides, the bearing friction torques obtained within these experiments were consistent with data obtained from online simulator [12], [13]. Results obtained for axial preload of 1800 N on static conditions (preload torque: 3 Nm) were also consistent with data related by Frank and Abel [5], that could be explained by the use of front caster wheel in their experiment, requiring a manual screwing that potentially had led to uncontrolled axial loading.

Looking at the modelling of wheelchair propulsion, bearing resistance is often neglected, based on the results of Frank and Abel [5]. Using the results of our study and computing the global wheelchair power lost in bearing resistance for extreme conditions, it could range between 0.45 W (conditions: 1 m/s, wheel radii: 0.3 m and 0.05 m, bearing torque of each wheelchair wheel: 15.9 Nmm) and 5.4 W (conditions: 1 m/s, wheel radii: 0.3m and 0.05m, 50% load on front wheels, axial load imposed by 3 Nm torque on tensioning bolt, resulting bearing torque of each wheelchair wheel: 73.3 Nmm). This maximum power could also rise for higher velocities, as it is related to the wheel angular velocity. As a comparison, the power loss due to rolling resistance ranges between 4 W (conditions: high performance wheelchair rolling on the favorable condition of hard smooth surface, MWC velocity: 1 m/s) and 60 W (conditions: low performance wheelchair on carpet surface, 1m/s) [11]. Then, the hypothesis of a negligible bearing resistance could be considered valid in many cases, in comparison to rolling resistance. However, in the case of a combination of unfavorable factors in the choice of wheelchair settings (high axial load, small wheel radius, high load on front wheels), the bearing resistance could become non negligible compared to rolling resistance.

Considering the axial load induced by a high wheel camber, inclined at 20°, and a normal ground reaction force of about 300 N, the load axially applied to the bearings would not exceed 100 N, resulting in a bearing friction torque of 30 Nmm (corresponding to a power loss of 1.4 W by bearing friction for 0.3 m and 0.05 m wheel radii, MWC velocity: 1 m/s). As a consequence, the additional resistance due to the axial loading of the bearing could not explain the higher movement resistance observed for cambered wheels, which includes rolling and bearing resistances [3], [11].

The major limit of this study is to not have been able to test the higher values of radial load and velocity observed in usual manual wheelchair use, especially the velocity values for small front casters. These issues should be addressed by improving the testing device. The torque wrench used to impose axial load should also be more precise in future experiments. However, taking into account the high discrepancies between factor effects, the rank of their influence may not be altered, even considering a wider range of variation for the factors.

Better knowledge of the bearing energetic behavior – especially in the particular conditions of manual wheelchair use – linking it to the manual wheelchair settings and use conditions, will help complete the existing manual wheelchair mechanical models. These models allow assessing the external efforts applied by the ground on the wheelchair and its internal power only by knowing its movements and allow assessing user efforts with limited instrumentation. Thanks to these models, kinetic outputs of the wheelchair/user can be computed more precisely [2] and potentially linked to the user effort/energy consumption [8], [14].

The device presented in this study may proved its interest on assessing bearing resistance in the manual wheelchair range of use, it will allow testing various bearing types on driving wheels and hopefully front casters with future versions of the device. The effect of clogging on the bearings or the effect of them growing older could be investigated, for example, with this device: this would help define at what time changing the bearings would provide a noticeable improvement energy losses. This knowledge is also useful to help optimizing the settings, the maintenance of manual wheelchair and to limit unnecessary energy consumption. This study also presented various values of bearing resistances, linking them to the wheelchair use conditions, which, to the author's knowledge, was not done before.

5. Conclusions

This paper aimed at validating a specific device dedicated for the measurement of the bearing friction torque of bearings mounted in a wheel hub, in order to explore the effect of MWC settings, such as velocity, axial and radial load, on bearing resistance. Specifically, this testing device should allow measurements while the bearings are rotating. This study allowed validating the use of the testing device with an uncertainty of 3 Nmm. This accuracy could also be improved by carrying out several measurements for each testing condition and considering the average value. The accuracy, however, could not be sufficient to distinguish different bearings with close resistance properties, but should allow quantifying the expected impacts of both abrasion and soiling, for instance. A better control of the axial loading of the bearings should also improve the accuracy determined in this study. This device allows further studies to be conducted, in order to give recommendations for wheelchair bearing maintenance or to give simple predictive methods for assessing bearing friction torques.

Furthermore, this study had tempered the underlying assumption made by Frank and Abel [5] that bearing friction torques of functioning bearings could be assessed from measurements under static condition. Nevertheless, static measurements, which are easier to achieve, can be used to obtain approximations of the bearing friction torques, which could be sufficient in many cases.

This study also highlighted the impact of the axial preload, which appeared as the main factor affecting bearing resistance. The precise control of the axial preload during measurements is thus essential, but can be technically difficult to achieve. In practice for the use of an MWC, the quick release axles allow this axial preload to be cancelled but, at the same time, let an additional degree of freedom (translation along the wheel rotation axis) that might favor vibrations and reduce wheelchair maneuverability. Radial load has also shown a significant impact on bearing resistance, confirming the necessity of taking this parameter into account when modeling precisely MWC bearing resistance, like it has been done by Cooper et al. or Hofstadt et al. [7]. However, in comparison to rolling resistance, the results of this study show that the bearing resistance can be neglected when modeling MWC in many cases [2], [6], [11], considering that the bearings support a low axial load. However, for high performance wheelchairs and low front wheel radii, the bearing resistance can become non negligible and should then be taken into account. This study did not find a significant impact of the angular velocity on bearing friction torque when the wheel is in motion, although a study on higher ranges of velocities could temper this assumption. Future measurements on a wide range of bearings and clogging conditions will also complete this dataset.

Financial disclosures

The authors have declared that no competing interests exist.

Acknowledgement

This material was based on work supported by the SACR-FRM project, French National Research Agency (ANR-06-TecSan-020) and the Centre d'Etudes et de Recherche sur l'Appareillage des Handicapés, from Institution Nationale des Invalides (especially the design office for both its help in the computer-aided design and the production of the testing device required to fulfil this work).

References

[1] Burton M., Subic A., Mazur M., Leary M., *Systematic design customization of sport wheelchairs using the Taguchi method*, Procedia Eng., 2010, 2, 2659–2665.

[2] Cooper R.A., *A systems approach to the modeling of racing wheelchair propulsion*, J. Rehabil. Res. Dev., 1990, 27, 151–162.

[3] Faupin A., Campillo P., Weissland T., Gorce P., Thevenon A., *The effects of rear-wheel camber on the mechanical parameters produced during the wheelchair sprinting of handibasketball athletes*, J. Rehabil. Res. Dev., 2004, 41, 421–428.

[4] Kellerman R., Klein, H.-C., *Untersuchungen über den Einfluß der Reibung auf Vorspannung und Anzugsmoment von*

Schraubenverbindungen, in Konstruction, Vol. 2, Springer Verlag, 1955.

[5] FRANK T.G., ABEL E.W., *Measurement of the turning, rolling and obstacle resistance of wheelchair castor wheels*, J. Biomed. Eng., 1989, 11, 462–466.

[6] HOFFMAN M.D., MILLET G.Y., HOCH A.Z., CANDAU R.B., *Assessment of wheelchair drag resistance using a coasting deceleration technique*, Wheelchairs, 2003.

[7] HOFSTAD M., PATTERSON P.E., *Modelling the propulsion characteristics of a standard wheelchair*, J. Rehabil. Res. Dev., 1994, 31, 129–137.

[8] MEDOLA F.O., DAO P.V, CASPALL J.J., SPRIGLE S., *Partitioning Kinetic Energy During Free wheeling Wheelchair Maneuvers*, IEEE Trans. Neural Syst. Rehabil. Eng., 2014, 22, 326–333.

[9] PALEU V., CREȚU S., DRĂGAN B., BĂLAN R., *Test Rig for Friction Torque Measurement in Rolling Bearings*, Mach. Des., 2004, 85–91.

[10] POULY F., *Modélisation thermomécanique d'un roulement à billes grande vitesse*, Recherche, 2010.

[11] SAURET C., BASCOU J., RMY N. DE SAINT, PILLET H.H., VASLIN P., LAVASTE F.F. et al., *Assessment of field rolling resistance of manual wheelchairs*, J. Rehabil. Res. Dev., 2012, 49, 63.

[12] Schaeffler Group Industrial, Media tool, (n.d.).

[13] SKF, SKF web tools, (n.d.).

[14] SPAEPEN A.J., VANLANDEWIJCK Y.C., LYSENS R.J., *Relationship between energy expenditure and muscular activity patterns in handrim wheelchair propulsion*, Int. J. Ind. Ergon., 1996, 17, 163–173.

[15] TAGUCHI G., *The System of Experimental Design: Engineering Methods to Optimize Quality and Minimize Costs*, 1987.

[16] VAN DER WOUDE L.H.V., DE GROOT S., JANSSEN T.W.J., *Manual wheelchairs: research and innovation in sports and daily life*, Sci. Sports, 2006, 21, 226–235.