

Comparing Three Methods for Evaluating Impact Wrench Vibration Emissions

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To provide a means for comparing impact wrenches and similar tools, the international standard ISO 8662-7 prescribes a method for measuring the vibrations at the handles of tools during their operations against a cotton-phenolic braking device. To improve the standard, alternative loading mechanisms have been proposed; one device comprises aluminum blocks with friction brake linings, while another features plate-mounted bolts to provide the tool load. The objective of this study was to evaluate these three loading methods so that tool evaluators can select appropriate loading devices in order to obtain results that can be applied to their specific workplace operations. Six experienced tool operators used five tool models to evaluate the loading mechanisms. The results of this study indicate that different loads can yield different tool comparison results. However, any of the three devices appears to be adequate for initial tool screenings. On the other hand, vibration emissions measured in the laboratory are unlikely to be fully representative of those in the workplace. Therefore, for final tool selections and for reliably assessing workplace vibration exposures, vibration measurements should be collected under actual working conditions. Evaluators need to use appropriate numbers of tools and tool operators in their assessments; recommendations are provided.

Keywords: hand-transmitted vibration; HAVS; impact wrench; nut runner; threaded fastener

INTRODUCTION

Threaded fastener tools such as impact wrenches are widely used in many occupational sectors. Workers using these tools are frequently exposed to hand-transmitted vibration, rotational forces, high hand-tool coupling forces, awkward postures, and repetitive motions that may lead to occupational injuries or disorders [National Institute for Occupational Safety and Health (NIOSH), 1997]. Threaded fastener tools with impact action have been specifically associated with vibration white finger (Aiba *et al.*, 1999); vibration white finger, along with other symptoms, is commonly associated with hand-arm vibration syndrome (HAVS) (Taylor and Brammer, 1982; Griffin, 1990; Bovenzi, 1998). In the interest of preventing HAVS and other occupational injuries and disorders, an effective approach is

to reduce the vibration exposure at its source by developing and selecting appropriate tools.

To assist in tool development and selection, the international standard ISO 8662-7 (1997) prescribes a tool screening testing method, which is based on the measurement of the vibration emission values at the handles of impact wrenches, nut runners, and screwdrivers with impact, impulse, or ratchet action. This vibration characterization technique involves the operations of sample tools against a special loading mechanism (braking device).

Unfortunately, the braking device prescribed in the standard may not provide accurate accounts of interactions between tools and actual work pieces (Marcotte *et al.*, 2008); the braking device may produce a poor simulation of workplace impact wrench operations. It is unknown if tool comparisons based on braking device tests would mirror comparisons based on vibration measurements collected during actual work tasks. Furthermore, the test data obtained from different laboratories may also vary greatly, as measured

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vibrations have been shown to vary with the subject's physical size and with the magnitude of applied hand forces (Shida *et al.*, 2001). Difficulties in fabricating a durable braking mechanism that provides consistent tool loading may also contribute to interlaboratory variances.

In view of these problems and other issues, the current standard is scheduled to be replaced. ISO Technical Committee 118/SC 3/WG 3—vibrations in hand-held tools—*ad hoc* group for wrenches is tasked with compiling and evaluating proposed changes to the current standard. As stated by the committee, the replacement standard, ISO 28927-2, is intended to

- (i) more closely simulate actual vibration in the workplace,
- (ii) assure that the end results would fall into the top quartile of what is actually experienced by a variety of workers in different work situations,
- (iii) produce a vibration result that could be easily translated into risk assessment analysis by the tool user, and
- (iv) be able to apply the revised procedure to a cross section of all threaded fastener tools from all power sources, including pneumatic, hydraulic, or electric (ISO, 2006).

As part of their evaluation process, the ISO technical committee has considered the retention of the existing cotton-phenolic braking device. They have also considered at least two new tool loading mechanisms to be used for tool vibration characterization. One of the proposed new loading devices comprises concrete-mounted steel plates that encompass 10 bolts and nuts. With this 10-bolt apparatus, tool vibration data are collected as a tool operator seats the 10 nuts in a prescribed timing sequence. This new tool loading method was evaluated in our recent study (McDowell *et al.*, 2008). The ISO Technical Committee is also considering a modified version of the existing braking device. The modified braking mechanism, detailed in the draft international standard ISO/DIS 28927-2 (2008), is identical to the braking device described in the existing standard (ISO 8662-7, 1997) except that it utilizes aluminum blocks with friction material bonded to the braking surface to provide the tool load instead of cotton-phenolic blocks. According to a recent report by a major tool manufacturer (Atlas Copco, 2008), the

aluminum blocks may have advantages over the phenolic blocks as (i) they appear to be less affected by heat, (ii) tool speed is more stable, and (iii) less clamping force is required.

The intention of the impact wrench vibration assessment standard is to provide a means to compare different power tools or different models of the same tool type (ISO, 1997, 2008). Theoretically, a tool evaluator could use any of the three tool loading mechanisms to rank order impact wrenches according to acceleration magnitude. However, little is known about how the results of tests using these three loading devices compare to one another. Therefore, to assist in the development of an improved impact wrench vibration testing standard, the objective of this study was to evaluate the three testing methods featuring these different loading devices using a common set of operators and tools.

MATERIALS AND METHODS

Six male impact wrench operators participated in the study. This was a different set of tool operators than that used in the earlier study (McDowell *et al.*, 2008). All study participants were experienced impact tool operators recruited from local bus garages and tire shops. Candidates were asked to have logged at least 100 h with threaded fastener tools to be eligible for the study. The tool operators participated in the experiment on a paid and informed consent basis. The tool operators followed a protocol that was reviewed and approved by the NIOSH Human Subjects Review Board.

The tool operators used five pneumatic impact wrench models during the experiment. Tool details are contained in Table 1. Each of these models features a 0.5-inch square drive. A 76-mm impact drive extension was used in all trials. All tools were brand new. Each tool was lubricated in accordance with its manufacturer's specifications and 'broken in' during trial runs in preparation for the study. The supplied air pressure was regulated to 620 kPa (90 psi). Each tool was adjusted to operate at its maximum torque setting.

Three tool loading devices were employed in simulated work tasks. Two of the loading devices used in the experiment were braking mechanisms specified in the draft international standard for evaluating

Table 1. Details of the five pneumatic threaded fastener tools used in the study

Tool	Manufacturer	Model	Maximum torque (Nm)	Tool weight (kg)
A	Chicago Pneumatic	CP749	612	2.4
B	Chicago Pneumatic	CP7733	746	2.5
C	Atlas Copco	LMS27-HR13	180	2.1
D	Atlas Copco	EP12PTS150-HR13-AT	150	2.5
E	Ingersoll-Rand	2135Ti	848	1.8

vibration emissions of impact wrenches (ISO/DIS 28927-2, 2008). The two braking devices are shown side by side in Fig. 1a. These braking devices comprise steel bases that are mounted on a solid reinforced concrete block. Each base holds a two-piece braking block; the braking block features a hole sized to fit a socket (50.8 mm OD) that is sandwiched between the two pieces of the braking block. Figure 1a shows the blocks with the sockets and impact drive extensions installed. The braking block provides a friction load to the outside surface of the socket as the socket rotates inside the braking block. The amount of friction applied to the socket is controlled by adjusting the clamping force via two sets of bolts, nuts, washers, and cup springs. Two types of braking blocks were used in this study. The two-piece block in the first device was fabricated from a cotton-phenolic laminate material. The block in the second version was constructed from solid aluminum with a bonded friction material serving as the brake lining. Both braking block versions are shown in Fig. 1b.

The typical subject posture for the braking device tests is shown in Fig. 2. At the beginning of the test,

the load-regulating bolts on the block were backed-off to allow the socket to spin freely between the two block pieces. The tool operator ran the tool at full speed by fully engaging the tool trigger. Four of the tools were operated in clockwise rotation. However, Tool D was equipped with an automatic shut-off mechanism; to prevent inadvertent tool shut-offs during the brake device tests, this tool was operated in reverse as is allowed in the draft standard (ISO/DIS 28927-2, 2008). The operator was instructed to apply the minimum amount of hand force required to ensure safe and stable operation of the tool. The speed of the rotating shaft was monitored via a hand-held laser tachometer; the tachometer readings were double checked with a hand-held stopwatch. The load-regulating bolts of the braking device were then adjusted to produce a rotational speed of the tool shaft/socket of 6 ± 1 r.p.m. Once the speed was within the prescribed window, tool vibration data were collected for 8 s. The operator then shut off the tool by releasing the trigger and rested for 45 s. At the end of the rest period, the operator repeated this process with the same tool for a total of five

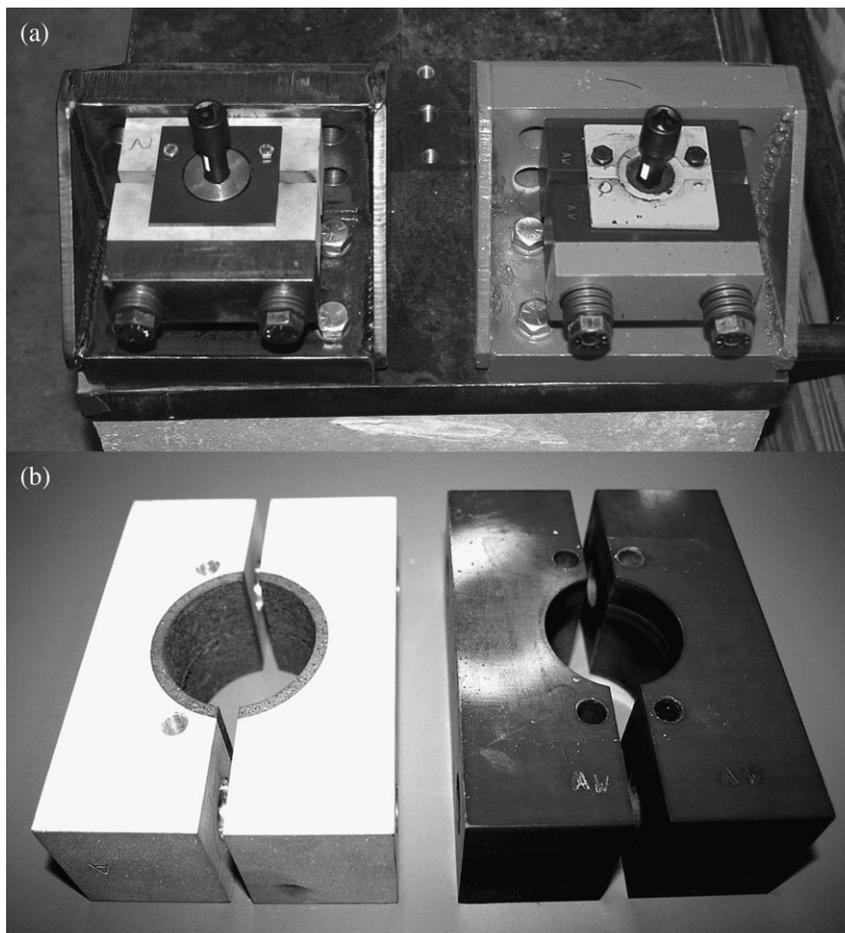


Fig. 1. (a) The two braking devices mounted side by side on a steel mounting plate and concrete pedestal. (b) The two types of braking blocks used in the study: the aluminum block with a brake lining is on the left and the cotton-phenolic block is on the right.

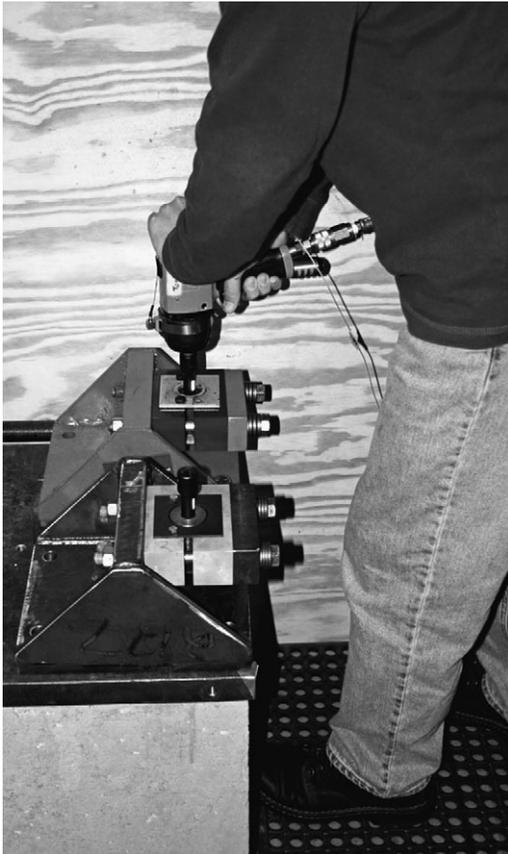


Fig. 2. Tool operator using a tool with the cotton-phenolic braking device.

trials. Once the five trials were completed, the operator rested for at least 2 min while the next tool and load combination was prepared for the evaluation.

The third loading device was constructed based on the requirements of the test setup and procedures provided by ISO Technical Committee 118/SC 3/WG 3 (ISO, 2006). This test apparatus is depicted in Fig. 3. This test is described in detail in the previous report (McDowell *et al.*, 2008). Briefly, the test setup consists of two hardened steel plates vertically mounted on a concrete block. Each plate supports 10 steel bolts. The 10 bolts on each plate are arranged in two evenly spaced rows of five bolts each. Each bolt is fitted with a nut, two Belleville washers, and a matching flat washer. Vibration measurements were made over the course of a series of 30-s trials that involve the seating of 10 nuts onto the plate-mounted bolts.

In order to control the work pace with the 10-bolt apparatus, a custom program using National Instruments LabVIEW™ software was developed (McDowell *et al.*, 2008). Basically, the program's display consisted of 10 dial gauges that represented the 10 nuts to be tightened during a trial. The dial needle swept around a gauge in 2 s, then the subject was prompted to move to the next nut; 1 s later, the needle



Fig. 3. Tool operator using an impact wrench with the 10-bolt apparatus. The task-pacing program is being displayed on the computer at the operator's eye level.

swept around the next gauge and so on. The pacing program ran on a notebook computer that was placed in front of the test subject at eye level. The computer setup can be seen in Fig. 3. This pacing mechanism proved to be successful as trials during the tests were consistently within 30 ± 1.5 s for all study participants. After a 10-bolt trial, the subject rested while a test engineer backed-off the nuts to their starting positions. The subject was then prompted to complete the next trial on the alternate plate. In this fashion, about half of the test session's trials were completed on one plate and half on the other.

Prior to testing, there was a short practice session (~15 min) to allow the subjects to become familiar with the three loading devices and to get them accustomed to the sequences and pacing of the simulated work tasks. Following practice, each subject completed five trials with each tool on each loading device for a total of 75 trials in a test session that usually lasted ~2 h. The testing order of the 15 tool-load combinations was randomized for each subject.

Vibration was measured at the front portion of the tool housing. While the draft standard calls for acceleration measurements at two tool locations (ISO/DIS 28927-2, 2008), the results of the previous study showed that conclusions based on measurements at the tool handle were identical to those based on measurements at the front of the tool housing (McDowell *et al.*, 2008). Therefore, we concluded that one accelerometer is sufficient for tool screening.

Thus, the least invasive of the two accelerometer mounting positions was chosen for this study. All vibration measurements were collected via calibrated PCB Model 356B11 triaxial accelerometers. Each accelerometer was mounted on an aluminum mounting block. Hose clamps were used to secure the accelerometer assemblies to the tools; the accelerometer mounting method can be seen in Fig. 4. The measurement of vibration of percussive tools often yields significant drift (i.e. DC shifts) in the piezoelectric accelerometer output (Kitchener, 1977). While preparing for the previous study with the 10-bolt apparatus (McDowell *et al.*, 2008), evidence of such DC shifts in some of the measurements was observed. In order to alleviate this DC shifting problem, thin layers of rubber were placed underneath the accelerometer mounting blocks and hose

clamps to provide mechanical filtering in a similar fashion to that detailed in another study (Dong *et al.*, 2004). This filtering method, shown in Fig. 4b, has proven to be effective at eliminating the DC shifts.

As is specified in ISO 5349-1 (2001), vibrations were measured in three axes. Unweighted vibration data were collected for each one-third octave band with center frequencies from 6.3 to 1250 Hz. The triaxial accelerometer signals were conditioned via PCB 480E09 ICP sensor signal conditioners with gains set to '1' (no amplification). The vibration signals were then fed into a portable six-channel B&K PULSE system (Model 3032A) where a separate text file was generated for each test trial.

A Microsoft Excel spreadsheet was used to process the ISO-weighted root-mean-square (r.m.s.) accelerations for each trial. Immediately following each five-trial tool run, the coefficient of variation (CV) of the root-sum-of-squares value (total value) for the five consecutive trials was calculated. Consistent with the draft ISO standard (ISO/DIS 28927-2, 2008), the five-trial data were checked for errors if the CV was ≥ 0.15 . Measurements found to be obvious outliers were excluded from the dataset, and the tool operators performed replacement trials to complete the prescribed matrix. Processed vibration values proved to be reasonably consistent with $<5\%$ of all test trials requiring replication.

Unweighted vibration magnitude values were collected for each one-third octave band from 6.3 to 1250 Hz for each axis. The ISO-weighted root-sum-of-squares vibration magnitude total values were calculated after applying the one-third octave band weighting coefficients presented in ISO 5349-1 (2001). The 'total' ISO-weighted r.m.s. value was then calculated using the following formula:

$$a_{\text{total}} = \sqrt{a_x^2 + a_y^2 + a_z^2}, \quad (1)$$

where a_{total} is the ISO-weighted root-sum-of-squares total value, and a_x , a_y , and a_z are the ISO-weighted r.m.s. values for the x -, y -, and z -axis, respectively. All vibration magnitude values are expressed in meters per second square.

Univariate mixed-model analysis of variance (ANOVA) tests for ISO-weighted total acceleration were performed to identify significant study factors. A two-factor within-subject model (tool + loading device) was used to examine the entire acceleration dataset, while a single-factor model (tool) was applied to each of the three loading device subsets. In the analyses, tool operator was included in the model and treated as a random factor while tool and load were treated as fixed factors. Tukey's honestly significant difference (HSD) *post hoc* pairwise comparisons were also performed. All ANOVAs and Tukey's HSD tests were performed using SPSS statistical software (SPSS Inc., version 14.0). Analysis

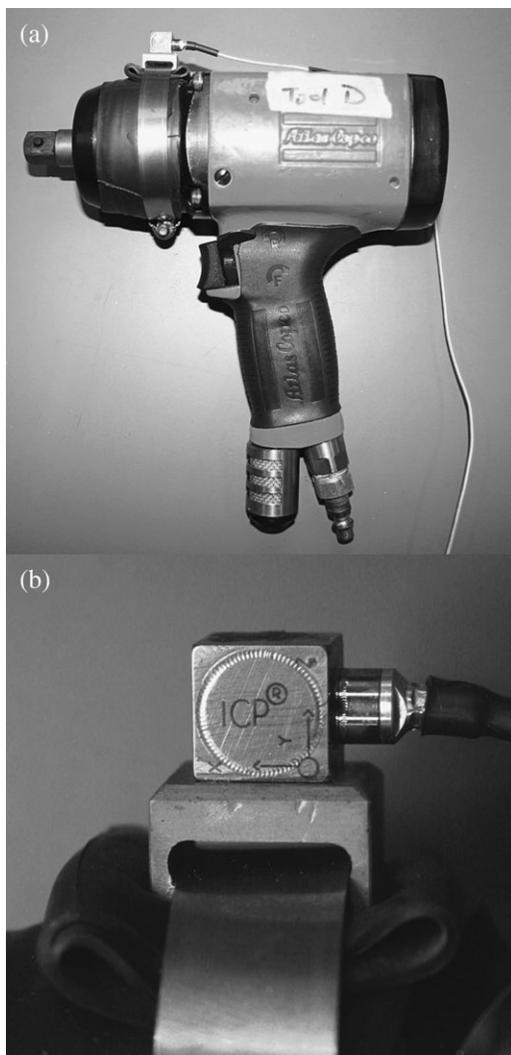


Fig. 4. (a) Typical arrangement of the accelerometer, mounting block, and hose clamp used for mounting the accelerometer on the tool near the front of the tool housing. (b) To prevent DC shifting of the acceleration measurements, layers of rubber were placed under each hose clamp and accelerometer mount.

results were considered significant at the $P < 0.05$ level.

RESULTS

Representative unweighted vibration spectra measured on the five tools under each loading condition are presented in Fig. 5. The basic trends of the spectra measured with the two braking devices are similar. Probably because the socket is tightly constrained in the blocks of the braking device, the low-frequency vibration components (<16 Hz) of the tools are generally lower than those measured with the 10-bolt apparatus. However, the basic trends are similar for all spectra at frequencies >31.5 Hz.

Distributions of measured ISO-weighted acceleration for each tool and load combination are depicted in Fig. 6. Figure 7 shows the distributions of measured acceleration for each loading condition. As can be seen in these figures, the two braking devices pro-

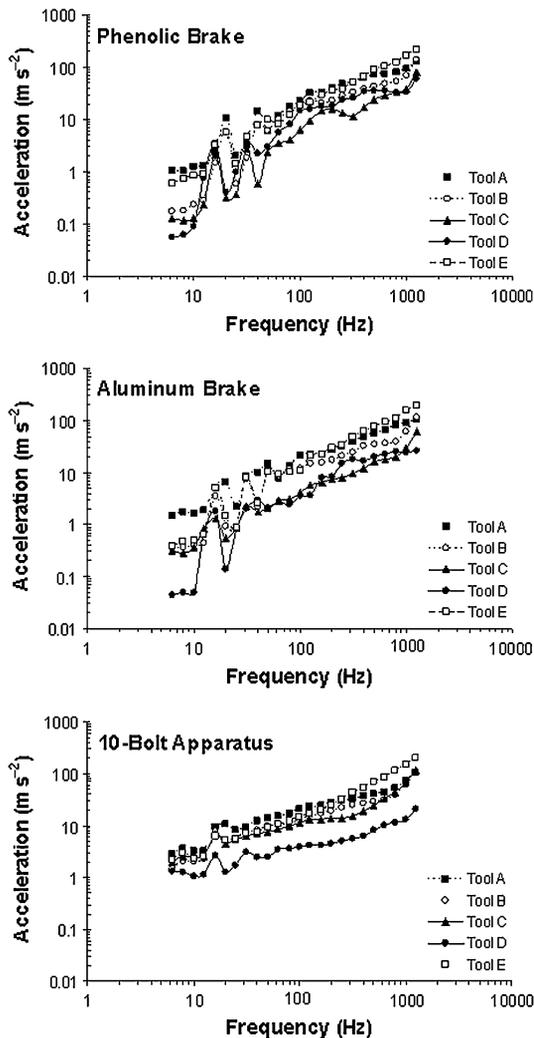


Fig. 5. Typical unweighted vibration spectra measured on each of the five tools under each of the three loading devices.

duced similar vibration patterns, while the 10-bolt apparatus produced somewhat unique results in terms of both the magnitude and vibration emission variations.

ISO-weighted acceleration means and standard deviations (SDs) for each tool and load combination as well as overall means and SDs are presented in Table 2a; the tools are rank ordered from low to high vibration for each loading condition. The tool vibration rank orders for the two braking devices are identical. While Tool D ranked the lowest on all three loading devices, the braking devices and the 10-bolt apparatus produced different rank orders of the other four tools.

Table 2b presents the unweighted acceleration means and SDs in the same fashion as Table 2a. The rank orderings for unweighted acceleration are consistent with those for ISO-weighted vibration; the only differences are that Tools A and E swap positions for the 10-bolt apparatus and for the overall averages.

Table 3 contains the results of the two-factor mixed-model ANOVA for ISO-weighted acceleration for all loading conditions. The results reveal that both tool and loading condition have significant influences on measured tool acceleration. The tool-load interaction is also significant. Pairwise comparisons show that the mean acceleration for the 10-bolt apparatus (11.05 m s^{-2}) is significantly higher than the mean for the phenolic brake (8.93 m s^{-2}), which is significantly higher than the mean for the aluminum brake (8.14 m s^{-2}). As indicated in Table 2a, the overall means for Tools A and E are not significantly different, but all other tool mean pairs are significantly different.

While not presented here, a two-factor mixed-model ANOVA was also performed for unweighted acceleration. The list of significant factors and interactions was identical to that for ISO-weighted acceleration (see above). As indicated in Table 2b, the results of *post hoc* pairwise comparisons for unweighted tool acceleration means were slightly different than the ISO-weighted comparison results; for unweighted acceleration, all tool means were significantly different from each other for each tool loading condition.

Given the significant tool-load interaction, follow-up single-factor mixed-model ANOVAs for ISO-weighted acceleration were performed for each load to further evaluate tool influences on tool vibration. Those results are listed in Table 4. For all loading devices, tool was found to be a significant factor; the pairwise comparison results are indicated in Table 2a. As noted, the tool acceleration means for the phenolic brake settled into three subsets; with the aluminum brake, each of the five tool acceleration means was significantly different from the others. Like the aluminum brake, all five tool means for the 10-bolt apparatus were significantly different from each other, albeit in a different rank order.

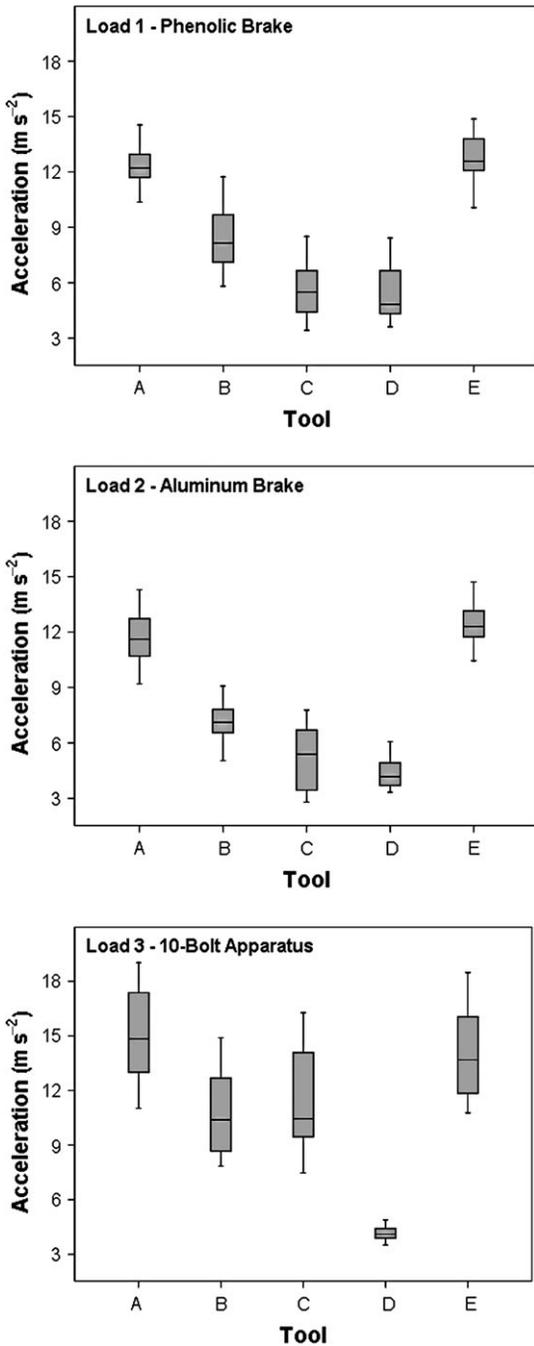


Fig. 6. Distributions of ISO-weighted total r.m.s. vibration values ($n = 30$) measured during all trials with each tool by all six operators for each loading device. The boxes represent the interquartile range for each distribution; the line within each box is the median value. The whiskers extend to the 5th and 95th percentiles.

DISCUSSION AND CONCLUSIONS

Tool vibration magnitudes vary greatly in real working conditions. It is impossible to design a standardized laboratory test method that can represent all workplace environments. Operating tools against

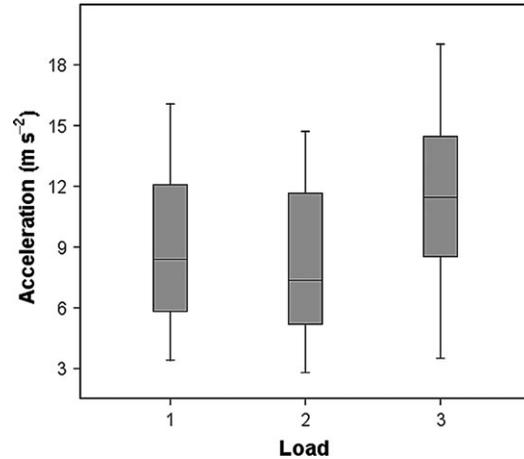


Fig. 7. Distributions of ISO-weighted total r.m.s. vibration values ($n = 150$) measured during all trials, tools, and operators for each of the loading devices (Load 1 = phenolic brake; Load 2 = aluminum brake; and Load 3 = 10-bolt apparatus). The boxes represent the interquartile range for each distribution; the line within each box is the median value. The whiskers extend to the 5th and 95th percentiles.

a stable load seems to be a reasonable approach for standardizing tool tests for the purpose of comparing and screening tools. From this point of view, all three loading methods examined in this study are acceptable. However, as demonstrated in this study, tool vibration emissions can differ by load. Each loading method comprises some unique features. The findings of this study can be used to (i) identify major differences among loading techniques, (ii) help tool evaluators select appropriate methods, and (iii) assist in the development of standardized tool testing methods.

As shown in Fig. 5, the unweighted vibration spectra of the five impact wrenches generated with the phenolic and aluminum brakes are similar. The rank orders, in terms of the frequency-weighted acceleration, produced by the two braking devices are also the same, as listed in Table 2a. These observations suggest that tool vibration emissions generated with these two braking devices are likely to be similar. However, Tukey's pairwise comparisons for the phenolic braking tests yielded only three subsets for the ISO-weighted tool vibration means, whereas all five tool vibration means were independent for the aluminum brake. Therefore, the aluminum brake appears to be more sensitive for tool screening. This, along with the other advantages mentioned above, might make the aluminum brake the better choice between the two braking blocks.

Physically, the 10-bolt apparatus provides a reasonable simulation of the working conditions and procedures for impact wrench operations at many workplaces. Despite the large variation in vibration emissions generated with the 10-bolt apparatus, the ISO-weighted vibration means of the five tools were

Table 2. Means and standard deviations (SDs) expressed in $m\ s^{-2}$ for each tool operated for five trials by each of the six operators under each load ($n = 30$) and under all loads ($n = 90$) for a) ISO-weighted acceleration and b) unweighted acceleration

Phenolic brake				Aluminum brake			10-Bolt apparatus			All loads		
Rank	Tool	Mean	SD	Tool	Mean	SD	Tool	Mean	SD	Tool	Mean	SD
a) ISO-weighted acceleration												
1	D	5.42 ^a	1.41	D	4.37	0.83	D	4.15	0.36	D	4.65	1.11
2	C	5.57 ^a	1.51	C	5.21	1.62	B	10.75	2.18	C	7.37	3.41
3	B	8.34	1.54	B	7.14	0.99	C	11.34	2.48	B	8.74	2.22
4	A	12.56 ^b	1.50	A	11.60	1.31	E	13.99	2.30	E	13.03 ^c	1.72
5	E	12.73 ^b	1.18	E	12.37	0.94	A	15.04	2.35	A	13.07 ^c	2.28
b) Unweighted acceleration												
1	D	90.5	19.7	D	79.8	17.0	D	32.0	6.1	D	67.4	29.8
2	C	155.0	41.4	C	147.9	50.0	B	141.1	18.8	C	155.2	39.2
3	B	183.6	14.1	B	164.1	17.7	C	162.7	20.0	B	162.9	24.2
4	A	221.1	31.0	A	202.6	33.9	A	171.3	16.2	A	198.4	34.7
5	E	372.7	36.0	E	348.0	41.5	E	299.8	27.8	E	340.2	46.5

Tools are ranked by vibration from lowest to highest for each loading condition. Means with a common superscript are not statistically different.

Table 3. Two-factor mixed-model ANOVA for ISO-weighted total vibration measured on each tool under each load by the six tool operators

Source	Error term	SS	DF	MS	F	P
Tool (T)	a	4841.2	4	1210.3	104.3	0.001 ^a
Load (L)	b	682.8	2	341.4	13.7	0.001 ^a
T × L	c	491.0	8	61.4	6.5	0.001 ^a
Operator (O)		94.4	5	18.9		
T × O		232.1	20	11.6	(a)	
L × O		249.6	10	25.0	(b)	
T × L × O		377.0	40	9.4	(c)	

^aSignificant factors ($P < 0.05$).

Table 4. Single-factor mixed-model ANOVAs for ISO-weighted total vibration measured on all tools under each loading condition by the six tool operators

Load	Source	Error term	SS	DF	MS	F	P
Phenolic brake	Tool (T)	a	1549.9	4	387.5	47.5	0.001 ^a
	Operator (O)		78.2	5	15.6		
	T × O		163.1	20	8.2	(a)	
Aluminum brake	Tool (T)	b	1609.2	4	402.3	63.1	0.001 ^a
	Operator (O)		17.5	5	3.5		
	T × O		127.5	20	6.4	(b)	
10-Bolt apparatus	Tool (T)	c	2173.2	4	543.3	34.1	0.001 ^a
	Operator (O)		248.2	5	49.6		
	T × O		318.6	20	15.9	(c)	

^aSignificant factors ($P < 0.05$).

reliably differentiated, which suggests that the 10-bolt test is also a good choice in terms of its sensitivity. However, this method could be more costly in terms

of both the apparatus itself and in the experienced manpower required for tool testing. Another major concern is that operators may have difficulties in operating heavy tools during the 10-bolt tests. Therefore, if the tool rank order obtained via the braking device is similar to that of the 10-bolt apparatus, the braking device approach may be a better choice for a standardized tool-screening test.

Four of the five tool models exhibited higher weighted vibrations with the 10-bolt apparatus than with either of the braking devices. Tool D exhibited lower vibration and much less variability with the 10-bolt load. Tool D was the only model equipped with an automatic shut-off mechanism. To avoid inadvertent shut-offs during the braking device tests, this tool was operated in reverse as is specified in the ISO standard (ISO 8662-7, 1997). Thus, its automatic shut-off mechanism was disabled during the braking device tests, but not during the 10-bolt tests. This likely explains why this tool exhibited lower vibrations and less variation during the 10-bolt apparatus tests.

With the exception of Tool D, ISO-weighted vibration magnitudes were significantly higher with the 10-bolt apparatus than with either of the braking devices. There may be several factors contributing to these differences. First, the torque resistance, stiffness, and damping properties of the 10-bolt apparatus are likely different than those of the braking devices. Second, as the operator's posture changed from task to task, changes in the impedance of hand and arm system could affect tool vibration emissions (Aldien *et al.*, 2005; Adewusi *et al.*, 2008). Likewise, changes in the gripping and pushing forces applied to the tool handle are known to influence the vibration signature (Shida *et al.*, 2001; Dong *et al.*, 2004); while not evaluated in this study, it is likely that the hand forces

required for the braking device tests are different from those for the 10-bolt apparatus. Further, tool speed was different for the 10-bolt apparatus tests than the braking device tests. Moreover, as shown in Fig. 5, some substantial differences between the vibration spectra measured on the 10-bolt apparatus and the braking devices are evident at frequencies <20 Hz. This is likely because low-frequency vibrations in the 10-bolt tests are generated as the operator moves the tool from bolt to bolt and positions the socket onto each nut. These vibrations, which are not observed in the braking device tests, are heavily weighted by the ISO standard (ISO 5349-1, 2001). Also, vibration frequency spectrum components that may be unique to the run-down portions of the 10-bolt tests would be excluded from the braking device tests (M. Persson, personal communication).

However, these obvious differences do not dramatically change the rank order of the five tools. Tool D is consistently ranked as the tool with the lowest vibration. The difference in ISO-weighted vibration between Tools A and E is <7%, regardless of test method. From a practical standpoint, these two tools can be considered similar for most applications; their observed rank order differences may not be that important. Even though the rank order of Tools C and B was shown to depend on the loading method, these two tools are consistently ranked in the middle positions above Tool D and below Tools A and E. Therefore, any of the three loads seems acceptable for an initial tool screening or ranking.

On the other hand, the results of this study also clearly indicate that vibration emissions measured with either of the two braking devices are unlikely to be fully representative of those in the workplace, as evidenced by their differences from those measured with the 10-bolt apparatus (see Fig. 5 and Table 2). Therefore, tool rank orders based on actual workplace loads may also be different than those based on laboratory test results. Thus, it may be a good practice to select a small group of candidate tool models through laboratory tool screenings. Then, final tool selections can be made via tool tests during actual work tasks.

It is also emphasized that evaluators need to use appropriate numbers of tools and tool operators in their assessments. The current standard (ISO 8662-7, 1997) and draft standard (ISO/DIS 28927-2, 2008) call for tool testing with three tool operators. For initial tool screenings, this recommendation is probably adequate. However, the results of the previous study indicate that a sample of only three tool operators may be insufficient for achieving a tool screening sensitivity of 0.5 m s^{-2} ISO-weighted acceleration in impact wrench vibrations while limiting Type I and II error risk probabilities to $\alpha = 0.05$ and $\beta = 0.20$, respectively (see McDowell *et al.*, 2008 for a more detailed discussion). The results of the

present study reinforce the previous recommendations; this study employed a different set of tool operators, yet the observed variance and ANOVA results were similar to those of the previous study. Therefore, for final tool selections and for assessing actual work task vibrations, the results of these two studies indicate that five tool operators using three samples of each tool model would be the recommended minimum sample size.

Tool acceleration magnitudes observed with the 10-bolt apparatus in this study were slightly higher than those of our previous study that utilized the same tool loading apparatus (McDowell *et al.*, 2008). This increase is likely due to the addition of the impact extension inserted between the tool and the socket. Misalignment between the tool and fastener is known to be a potential source for increased vibration magnitudes (ISO/DIS 28927-2, 2008). The use of the extension increases the chance for such misalignment. Furthermore, the use of an extension may decrease the rigidity and/or alter the damping properties of the system resulting in higher tool vibration levels. Therefore, the use of an extension would need to be standardized in order to increase the validity of laboratory-to-laboratory comparisons. It should be noted, however, that if the standardized procedure is being used by a single laboratory simply to compare or rank order tools according to vibration (which is the stated purpose of the ISO standard), the use of the extension should not affect the outcome.

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