Retrofittable Machine Condition and Structural Excitation Monitoring From the Terminal Box

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***Abstract*— Retrofittable self-powered sensors for machine condition monitoring ease the burden of installation and decision-making for maintenance and acoustic performance assessment. Terminal box magnetic power harvesting sensors are nonintrusive. They require no special wiring and can simultaneously observe and correlate important variables for machine diagnostics, including vibration and speed. These correlated data can be used to detect and differentiate imbalances from failing structural mounts, among other possibilities. New hardware and algorithms are presented for enabling *in situ* vibration monitoring, with demonstrations on data sets from US Coast Guard vessels. A specific algorithmic focus of this paper is estimation of a machine's contribution to structure-borne noise and vibration, an important consideration for ship acoustic signature.**

***Index Terms*— Condition monitoring, capacitive sensors, energy harvesting, frequency response, vibration measurement.**

I. Introduction

VIBRATION monitoring of rotating machines and other electromechanical systems is widely used to make condition-based maintenance decisions. Many organizations, including the United States Navy (USN) and the United Stated Coast Guard (USCG), have adopted machine vibration standards [1]–[3]. The USN and USCG commonly contract specialized consultants to perform vibration inspections of critical shipboard equipment. Monitoring has diagnostic value [4], but intermittent monitoring of machinery does not catch problems as soon as they appear. In addition, isolated testing of individual systems does not accurately represent real (e.g., underway) operating conditions where coupling between machines may introduce significant mechanical interactions. Shipboard machine vibration also contributes to structural vibration and equipment operating "out of specification" can vitally alter the platform acoustic signature.

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Installation of vibration sensors for continuous monitoring may be cost prohibitive for all but the most critical machines. Permanent vibration monitoring can also generate an overwhelming amount of data. Consequently data analysis and interpretation is often reduced to examining averaged, RMS, or otherwise digested "figures of merit" that may fail to differentiate problems with structural coupling, e.g., a mounting resonance, versus a rotating imbalance.

Installed sensors, including vibration sensors, can be powered from the magnetic fields around power lines and electromagnetic equipment, permitting the creation of VAMPIRE, the Vibration Assessment Monitoring Point with Integrated Recovery of Energy [5]. VAMPIRE enables easy retrofit, requiring no special wiring for sensor power. VAMPIRE powered sensors enable the correlation of any sensed quantity, such as temperature, humidity, vibration, or other environmental information, with direct knowledge of the machine's operation based on the magnetic signals from the equipment's power cables.

Electrical measurements reveal machine power consumption, turn-on and turn-off times, steady-state speed and high fidelity spin-down speed after power-off. Taking advantage of electrical information to aid the interpretation of run-time vibration measurements offers two noteworthy advantages. First, intelligently targeting vibration analysis to probe diagnostically useful machine behaviors significantly reduces data redundancy. A second advantage is the use of electrical information to aid in discriminating interesting vibration signals from strong vibration disturbances present in noisy environments, such as the machine spaces of underway ships. For example, correlated electrical information permits a form of Frequency Response Function (FRF) estimation using system identification algorithms [6], [7] to disaggregate useful vibration signal from extraneous signals caused by nearby pumps or engines.

A. Structural Vibration

Existing methods for the analysis and interpretation of steady-state operating vibration signals are largely focused on the machine itself, and less able to distinguish related problems with structural mounts [8], [9]. Steady-state monitoring with a hand-held or temporary vibration meter offers little utility for distinguishing machine contribution to overall structural vibration. Our proposed electrically enhanced vibration monitoring can address both machine condition and structure borne vibration. The ability to give decision makers up-to-date knowledge of a machine's contribution to total structural vibration levels could be useful for condition-based maintenance and operating assessment, e.g., as a tactical decision aid on board a ship.

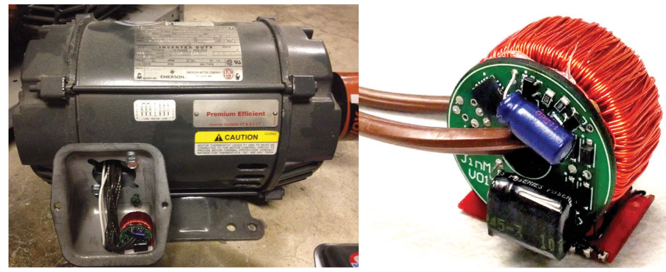
Machine vibrations are primarily transmitted to surrounding ship structure through their load-bearing connecting mounts (either resilient or rigid) and through other features such as pipes or ducts [10], [11]. Understanding the condition of the machine's mounts is essential to addressing the transmission of vibration forces to surrounding structure. A technique for distinguishing imbalance from structural or mounting changes can be developed by analyzing the turn-off spin down of the machine. We present techniques that passively track the shaft speed of a de-energized machine using electrical parasitics during spin down. Electrically determined shaft speed information can also serve directly as input to the family of order tracking methods [12].

B. In-Situ Monitoring

The power connections to an electromagnetic actuator are typically completed in a terminal box on the machine frame. This box is an opportunity. We have found it to be an excellent location for collecting vibration data with an accelerometer. It also provides a shielded and reasonably spacious location for installing an instrumented terminal box cover that includes sensors and a VAMPIRE inductively-coupled power tap. Section 2 describes sensors and a prototype power harvester. A custom built vibration data logger for monitoring motor terminal boxes in the field is used to evaluate and demonstrate the potential utility of a VAMPIRE monitor. Section 3 gives a brief summary of passive vibration isolation theory and describes some sources of vibration in rotating machines. Section 3 also describes the analysis of spin down vibration signals, FRF estimation, and simulated FRF examples demonstrate vibration response isolation for the monitored machine and the effects of rotating component and mounting condition. The final section presents estimated FRFs and other vibration data drawn from the USCG ships USCGC BERTHOLF (WMSL 750), USCGC STRATTON (WMSL 752), and USGCG SENECA (WMEC-906). The field visits are described in [13]. This section develops specific maintenance interpretations and observations for predicting structural vibration forces.

II. Hardware for Joint Vibration and Electric Sensing

Permanently installed vibration monitoring systems identify diagnostic conditions in mission-critical machinery like diesel engines and gas turbines [14]. These situations represent a minority of cases where vibration assessment could be useful. A challenge impeding continuous vibration monitoring for less critical machines is installation cost. Sensor support logistics like the routing and safety certification of power and data cables may cost more than the sensor hardware. Another challenge is secure sensor mounting in a retrofit application.



(a) (b)

Fig. 1. VAMPIRE hardware. (a) VAMPIRE power harvester, accelerometer and digital radio module located in motor terminal box with cover removed. (b) Close-up of power harvester module.

Within the volume of a motor terminal box, the power cable phase lines are separated from the main cable's overwrap insulation. This access to individual phases permits power harvesting using a magnetic flux from individual phase lines. The terminal boxes of most electric motors also contain two or more holes for threaded fasteners that are used to attach the terminal box cover. This permits secure attachment of the sensor package to the motor in the form of a replacement cover allowing retrofit installation. Repeatable mechanical registration also allows for comparison over time and across machines.

A. Power Harvester

The VAMPIRE power harvester is a current transformer that employs a high permeability magnetic core to couple to a phase line of the motor power cable. One phase of the machine's power cable serves as a primary winding, passing through an easily installed toroidal core. This core's many secondary turns power a circuit that harvests energy while current is flowing through the primary [15]. The secondary-side power electronic controls continuously monitor the secondary voltage and shape the secondary current waveform to maximize power harvesting efficiency. Power harvesting capability depends on the VAMPIRE core volume and the primary's current level. A prototype harvester, which achieves secondary powers of 5mW per cm^3 per RMS ampere in the primary, is shown installed in the terminal box of a two horsepower motor in Fig. 1. With typical motor currents ranging between one and several hundred amperes, secondary power can easily operate a suite of sensors, micro processors, and RF communications circuits with core volumes small enough to coexist with the cabling inside motor terminal boxes. The terminal box cover can also mechanically support the VAMPIRE device, and provide a non-metallic window or antenna for RF communication if desired. On-board energy storage can be included in the VAMPIRE device to power sensors and computation when the electromagnetic load is off; average available power would depend on the host machine duty cycle.

The power harvester also detects motor turn off. This permits the VAMPIRE microcontroller to effectively manage power, and to know when the machine is "coasting" or spinning down, a period that we have found useful for diagnostic assessment. Sudden decreases in motor current consumption,

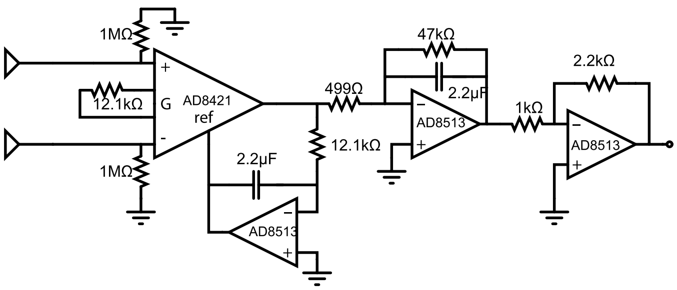


Fig. 2. Capacitive sensor schematic. The AD6421 differential amplifier rejects electromagnetic disturbances such as line frequency common to both electrodes. The remaining stages provide stabilization and amplification.

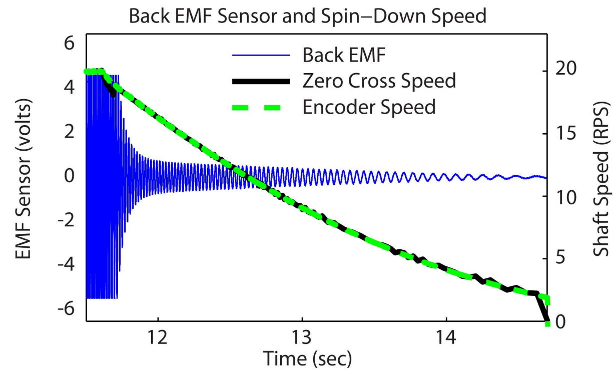


Fig. 3. Signal from noncontact back EMF sensor during spin-down, used for calculating shaft speed. Tachometer signal shown for comparison. The high gain sensor clips while the motor is powered, derived speed is accurate after sensor leaves saturation (at 11.6 sec in this case).

such as power off or abrupt load change, represent potentially valuable transient vibration monitoring opportunities. High data rate vibration signal collection, storage, and transmission can be selectively enabled at interesting times. Microelectromechanical Systems (MEMS) accelerometers are generally a low power [16] demand for the VAMPIRE harvester, enabling pre-trigger buffering to catch acceleration profiles before, after, and during interesting mechanical events like a spin down. Data processing and transmission may require more power, and the precise profile for VAMPIRE operation (e.g., what other sensors may be deployed, where data is stored, when and how much data is transmitted, and how the data is processed for diagnostic assessment) will depend on the duty cycle of the motor or actuator under observation and the current demand of the actuator. Depending on available power and user preference, any desired schedule of monitoring can be achieved in principle. In situations where duty cycle and current levels are both high (e.g., a pump on a ship), many options may be available for analyzing, compressing, and transmitting data to provide actionable diagnostic metrics, as will be shown in the following sections.

B. Back-EMF Sensing

A terminal box VAMPIRE installation permits non-contact voltage sensing. A capacitive sensor can determine the voltage on a wire with a plate or electrode outside of the wire insulation. A differential capacitive sensor can be employed to eliminate or greatly reduce interference from background electric fields and the fields due to neighboring wires. This non-contact voltage sensor requires relatively low power, well within the range of the energy harvester. Voltage sensing during motor operation enables detailed estimation of power and line synchronized harmonic content in the motor current. Even more exciting for vibration monitoring, the sensor can continue to detect the machine terminal voltage or "back electromotive force (back-EMF)" after the machine has been disconnected from the utility while it spins down. A design sensitive enough to detect back-EMF during spin-down has been developed. A differential measurement between two sensor electrodes secured to the power wires feeding the motor is used to reject such background interference as 60 Hz pickup. The circuit schematic for this electrostatic voltage sensor is shown in Fig. 2, with a detailed description in [17].

When a utility connected motor is switched off, the rotor will begin to slow its rotation. Residual magnetic flux in the rotor will continue to generate a speed voltage or back-EMF that can be used as an indicator of rotor speed during the spin-down of the machine. The open circuit stator voltage has an instantaneous frequency proportional to the shaft speed through the pole count of the motor as shown in Fig. 3. This signal can be processed to determine mechanical shaft speed. The spin-down rate is governed by residual load and friction forces, and may represent a valuable indicator of machine health. When combined with measured vibration during the spin-down, speed information can be used to generate diagnostic metrics for identifying imbalance or structural mounting pathologies, described in Section 3. Accurate shaft speed information also allows for high accuracy order tracking without the need to install a tachometer, a powerful feature for extracting rotation dependent vibration orders from noisy acceleration signals.

C. Accelerometer Data Logger

The terminal box is an atypical location for machine vibration measurement. However, the power harvesting and mounting advantages, access to internal machine variables like temperature and line voltage, and the ease of retrofit compel consideration of this location. We measured vibration at the terminal box with portable data logging vibration sensing units constructed for field tests. These units contained ADXL345 three axis MEMS accelerometers sampled at 3.2 kHz to a micro SD card. The ADXL345 was configured for $\pm$16g range on each axis. The loggers were attached to the external surface of motor terminal boxes. Test mass impulse response data indicate the vibration loggers have a flat frequency response of approximately $\pm$ 1 dB between DC and 200 Hz.

The three orthogonal axes of vibration are identified by their closest equivalent in the cylindrical coordinate frame of the motor (without trigonometric correction), illustrated in Fig. 4a. The logger's purpose was to accelerate algorithm and concept development, and explore the scope of vibration-related pathologies that could be diagnosed from the motor terminal box in conjunction with spin-down vibration measurements.

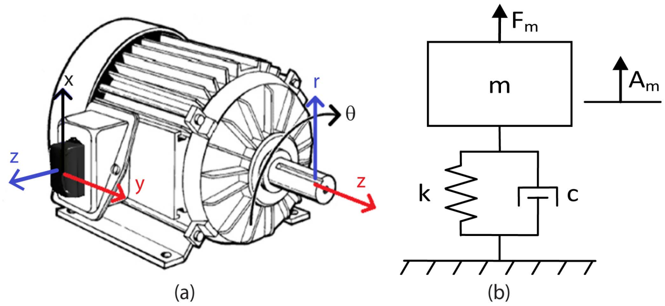


Fig. 4. Mounting and mount model. (a) Cartesian coordinate frame of terminal box mounted logger and cylindrical coordinate frame of the target motor. Like colors show equivalent measurement directions for reference. (b) Model of a single degree of freedom vibration isolation mount.

III. Underway Monitoring and Mount Transmissibility Identification

Steady-state shaft speed harmonics are useful indicators [8], [9] of rotating element condition, but require clean signals without coupling from other vibration sources. In an underway environment with many motors operating in a given space and similar speed ranges, overlap of local and extraneous shaft speed harmonics in the vibration spectrum lead to signal disambiguation and decision making challenges.

Instead, we examine non-stationary vibration during machine spin-down. Simultaneous planned power off events resulting in identical spin-down speed profiles are unlikely in noisy multiple machine environments. After power off, the rotation dependent vibration sources continue to excite the monitored machine until the shaft spins to a stop, while extraneous noise from neighboring machines will remain nominally steady. With proper processing this differentiating behavior allows one to distinguish signal content between the monitored machine and extraneous sources. Even for critical systems that must remain in continuous operation, individual machine power off events will occur with regularity due to load and operating wear sharing between the redundant machines nearly always included in the design of critical systems.

Machine spin-down vibration can also provide information on the condition of the machine's resilient mounts. Machine vibrations can be transmitted to the support structure and lead to noise or other disturbances to occupants of the space or adjacent machines. In hostile environments, ship acoustic signature is also a concern and is closely associated with structural vibration levels. The purpose of vibration isolation mounting is to reduce the transfer of vibration energy between machine and base structure [18]. Passive systems do this by lowering the natural frequency of the system below the main excitation frequency of the operating machine, usually the motor's rotation frequency.

Single degree of freedom analysis of vibration isolation can be applied to our low frequency single measurement point sensing architecture, where our sensor measures the quantity $\mathrm{A_m}$ shown in Fig. 4b. For low frequencies we assume that the machine exhibits rigid body behavior. This assumption is justified by machine design rules that require natural frequencies much higher than operating speeds to avoid excessive fatigue and radiated sound in motor housings. In addition, ship design guides advocate the mounting of vibration sources such as rotating machines on relatively stiff base structure such as above longitudinal deck plate stiffeners. This allows for neglecting base motion in the following development. The acceleration, A_m, of the rigid machine is then a result of internally generated forces, F_m, and is described by the Laplace domain transfer function (1), where m is machine mass, c is the mount damping parameter, k is the mount stiffness and s is the Laplace variable.

$$\frac{A_m(s)}{F_m(s)} = \frac{s^2}{ms^2 + cs + k} \tag{1}$$

At operating speed, the force transmitted from the machine to the base structure, F_T, passes primarily through the mount stiffness, equation (2), where f is frequency. The effect of damping may be neglected at operating speed for two reasons: material damping in vibration isolators is not velocity dependent as in the viscous model, but is displacement dependent and hysteretic in nature, leading to negligible effect where displacements are not large (i.e., away from resonance frequencies). In addition, equivalent viscous models of hysteretic damping in engineering systems rarely exceed critical damping rations higher than 0.1 [19].

$$\|F_T(f)\| = \frac{\|A(f)\|}{f^2}k \tag{2}$$

Estimation of transfer function (1) from data allows for the determination of the primary mounting resonance frequency ω_n, which is proportional to mount stiffness through (3).

$$k \propto \omega_n^2 \tag{3}$$

Change in condition or damage may affect the stiffness and damping properties of the resilient mount, resulting in a shift in the resonant frequency. Change in mount resonance frequency will affect both machine vibration and transmitted force. The ratio of force transmitted to the base structure before and after a change in mount resonant frequency is given in equation (4), with prime quantities representing post change values.

$$\frac{\|F'_T\|}{\|F_T\|} \propto \frac{\|A'_m\|\omega'^2_n}{\|A_m\|\omega_n^2} \tag{4}$$

Elastomeric vibration mounts were the most common mount type observed in field tests. The rubber components of vibration isolators can fail from creep or compression set, problems exacerbated by elevated temperatures, or through fatigue cracking and tearing. The effect of creep on mount stiffness is highly geometry dependent, but crack propagation in a rubber mount is generally a gradual process and will serve to reduce mount stiffness [20]. Chemical attack by hostile agents in the environment may serve to harden or soften rubber components in vibration mounts as well.

A. Virtual Inputs and Mount FRF Identification

The frequency sweeping excitation of machine spin-down permits estimation of FRFs that can serve to track overall mount condition and locate the primary mounting resonance. Motor spin up vibration signals are often significantly shorter in duration than spin-down signals, and are affected by strong electromagnetic vibration disturbances from large current transients, making them less desirable for FRF estimation. We estimate the excitation signal from electric signals provided in the proposed terminal box architecture. However, the phase of the virtual input signal generally will not match that of the real spin-down excitation source. Therefore, all FRF in this paper are presented as magnitude only.

$$I(t) = \frac{f_s(\mathrm{t})^2}{f_s(t=0)^2}\cos\left(\int_0^t 2\pi f_s(\tau)d\tau\right) \quad (5)$$

First order virtual inputs, $I(t)$, are generated from the spin-down speed curve given in Hz as $f_s(t)$, via a simple model of rotating imbalance, as shown in equation (5). The input signal is normalized to magnitude one at the machine turn off instant because the amount of imbalance will also not generally be known. Higher order virtual inputs linked to integer multiples of the shaft speed frequency can be generated for other vibration source phenomena, like coupling misalignment, but selection of suitable speed dependent amplitude functions is a challenge due to complicated manifestations of higher order excitation sources. For any given machine the particular type(s) of excitation sources present may not be well known. However, if only resonance locations are of interest, we have found a simple model with amplitude linearly dependent on speed suffices, and is even supported in some cases by other researchers modeling coupling misalignment [21]. The FRF function is calculated using the H_1 estimator, selected for its strong rejection of uncorrelated content in the output signal. See [6] and [7] for a description of the estimator.

The H_1 FRF estimator excels at removing output content uncorrelated with input from the results, but the presence of strong narrowband disturbances that overpower the measured vibration response at a given frequency will cause some extraneous correlation with the virtual input signal and possibly corrupt the FRF at the affected frequency. This problem is a significant challenge in all single vibration sensor algorithms that lack outside information on the properties of the disturbance. The most common source of narrowband disturbances in ship machinery spaces is operation of neighboring machines, leading to these disturbances having a steady or slowly varying character. Field trials in noisy spaces indicate that it is rare for disturbances of this type to overpower the local machine vibration response causing the effect on the estimated FRF to be negligible. Despite this, the user may detect a significant narrowband disturbance affecting an estimated FRF by estimating a verification FRF from the same virtual input and a start time delayed 10-20% of the average spin-down duration after t = 0. All legitimate machine vibration response information should be absent from the verification FRF and any strong peaks present in both the verification FRFs and the primary FRF would likely be the result of persistent narrow band disturbances from neighboring machines.

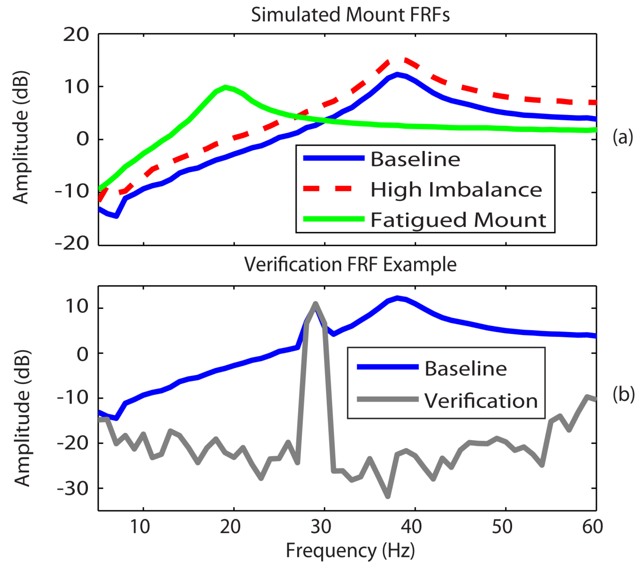


Fig. 5. Simulated FRFs. (a) FRF showing effect of mount condition change vs. increased imbalance relative to baseline. (b) Time delayed verification FRF indicates the presence of a persistent narrow band disturbance.

B. Simulation

This section provides FRF examples generated from simulated machines. The simulation model assumes rigid machine and base structure dynamics and includes a low level of imbalance (even in the baseline case). Simulated machine acceleration during a spin-down was contaminated with additive white Gaussian noise and used to calculate mount FRFs. This model simulates a baseline scenario, and compares results to two fault scenarios: increased imbalance and decreased mounting stiffness. While the absolute magnitude of the estimated FRFs have no meaning, the mean relative magnitude shifts in estimated FRF are easy to distinguish from changes in FRF shape indicating different fault types as shown in Fig. 5a. A separate estimated FRF of the baseline scenario in the presence of strong narrow band noise at 29 Hz is shown in Fig. 5b, along with a verification FRF to indicate the noise affected frequency.

IV. Examples From Fleet Survey and Interpretations

The custom vibration data logger was used in a series of field tests to acquire vibration data mounted to the terminal box of a range of pumps and fans present on various vessels of the USCG. The data were collected for concept development and validation, and did not include shaft speed sensing abilities. All speed signals used to generate virtual inputs for FRF estimation from the fleet survey were estimated by human annotation of the time frequency plot of vibration after shut off. Clearly discernible vibration linked to the shaft speed during spin-down was used to select a list of time speed points on the spin-down trajectory. These points guided a smoothing spline fit designed to remain in the center line of the time frequency spin-down trace. This procedure was verified against an encoder measurement in Fig. 6, which contains vibration

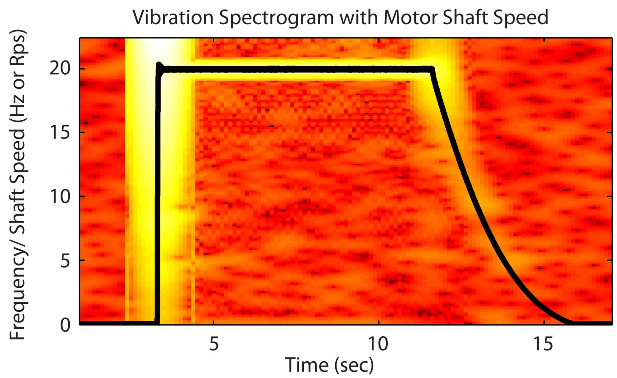


Fig. 6. Time frequency plot of lightly imbalanced motor vibration and motor shaft speed measured via tachometer (black line).

data from the same spin-down measured via back emf sensing in Fig. 3.

Turn off shaft speed was found by locating the strongest vibration spectrum peak in the feasible area defined by the motor name plate. The greatest source of error in human annotation is selection of the turn off instant. The highly selective correlation discriminator in the H1 estimator makes the result strongly dependent on having accurate relative timing between virtual input and output signals. However, the results are accurate enough for demonstration purposes, and would be much improved in full implementation involving internal terminal box access and direct spin-down speed sensing.

A. Ventilation Fans

Ships often use axial flow ducted fans for ventilation. These machines are commonly direct drive induction motor designs with fan heads with high rotational inertia. The fan heads are prone to imbalance issues. USCG documentation [3] contains procedures for field balancing of axial flow fan heads with guidelines on acceptable imbalance thresholds. Even machines that conform to stated guidelines possess enough imbalance to probe the mount condition during spin-down. A collection of ducted axial fans was monitored on three Coast Guard ships. The USCGC SENECA gave permission for temporary modifications of mount condition and imbalance magnitude to demonstrate the practical use of estimated mount FRFs for informing maintenance decisions and addressing structural vibration concerns.

1) SENECA: The USCGC SENECA (WMEC-906) contains ventilation fans in a space below the bridge deck. The fan pictured in Fig. 7a was monitored for steady state and spin-down vibration under a range of different imbalance and vibration mount condition experimental trials. During these trials, structural vibration was also recorded on the deck below the ventilation fan as a proxy for transmitted force. Despite the non-linear hysteretic nature of structural damping, ship structural steel has low damping characteristics and this allows for the assumption deck acceleration will vary linearly with applied the force. The results of four tests are presented here; a baseline test, partial mount stiffening, full mount stiffening, and increased imbalance with normal mount condition.

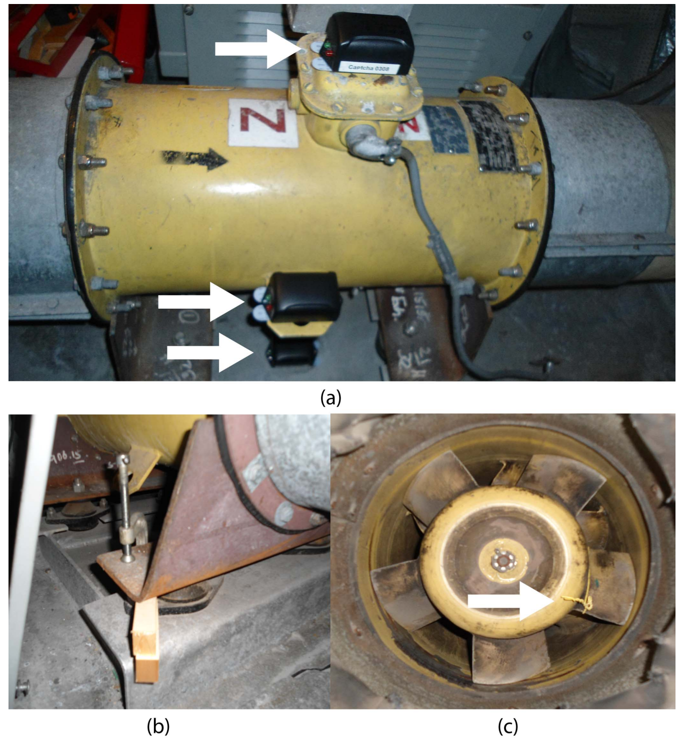


Fig. 7. SENECA axial fan. (a) Data loggers (at arrows) on fan and deck. (b) Mounts stiffened by clamping. (c) Fan and hookup wire imbalance (at arrow).

TABLE I
SENECA FANS TRANSMITTED FORCE RESULTS

Test	RMS Acceleration Fan		RMS Acceleration Deck		Mount FRF Resonance	ΔF_T ratio (equation 4)	
	g	dB	g	dB		ratio	dB
Baseline	2.01e-2	0 dB	7.01e-4	0 dB	21 Hz	1	0 dB
2 Clamp	2.85e-2	3.04 dB	3.22e-3	13.2 dB	37 Hz	4.41	12.9 dB
4 Clamp	5.84e-2	9.26 dB	1.05e-2	23.5 dB	48 Hz	15.2	23.6 dB

The fan was supported by four vibration isolation mounts, which were stiffened by C-clamping wood blocks to "short" the mount (Fig. 7b). Imbalance was increased in one test by twisting 30cm (12in) of 22 gauge solid core hookup wire around one fan blade as shown in Fig. 7c.

Fan and deck RMS acceleration at the operating speed of 58.9 Hz was determined via flat-top windowed power spectral density estimates in Figs. 8a, with results listed in table 1. Spin-down FRFs were estimated for each test condition and shown in Fig. 8b. From the FRF, the primary mounting resonance is quickly determined and equation (4) is employed to predict the change in deck acceleration with comparison to measured results in table 1. It is valuable to note that using only steady RMS vibration levels to gauge force transmission to ship structure would lead to inaccurate results.

2) NSC Cutters: We compare nominally identical ventilation fans from a pair of sister ships, the USCGC BERTHOLF and USCGC STRATTON. The fan design closely resembles the SENECA's ducted fan. A "high vibes" condition was reported by the BERTHOLF crew for their lower forward

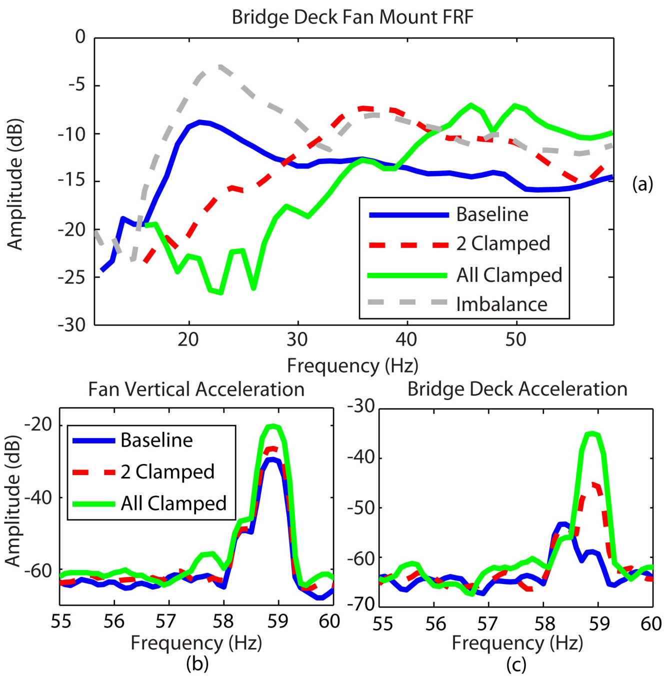


Fig. 8. (a) FRF in vertical direction for SENECA ventilation fan mount stiffening and imbalance experiments. (b) Steady vertical vibration power spectral density of bridge fan (top). (c) Deck plate below fan (bottom).

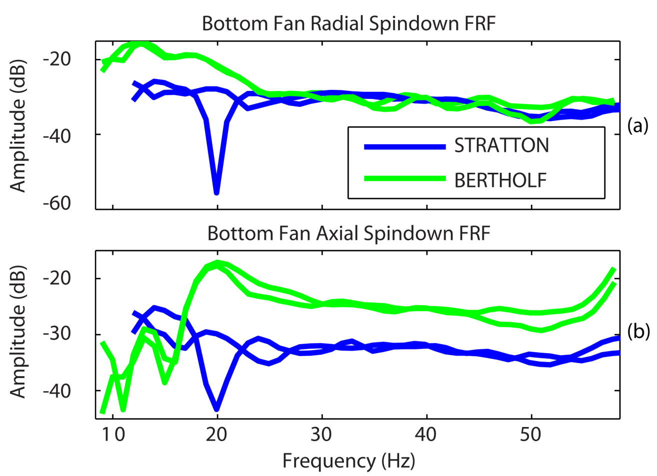


Fig. 9. Mount FRF estimates from sister ship's lower forward fans. (a) Radial direction. (b) Axial direction. There is indication of resonance near operating speed in the axial FRF from BERTHOLF.

ventilation fan. Measured RMS acceleration at the "high vibes" fan ranged from a factor of 4.8 to a factor of 18 higher than acceleration of the same fan on the STRATTON, depending on axis. The crew hypothesis for the cause was high imbalance. Superimposed spin-down FRFs from two consecutive spin-downs of each fan are shown in Fig. 9a. In the motor's cylindrical coordinates, the radial direction is the most sensitive to imbalance vibration, yet the radial direction FRF from both fans show nearly identical behavior. Instead, significant difference between the STRATTON and BERTHOLF fan is observed in the axial vibration direction in Fig 9b. The increase in axial vibration response of the

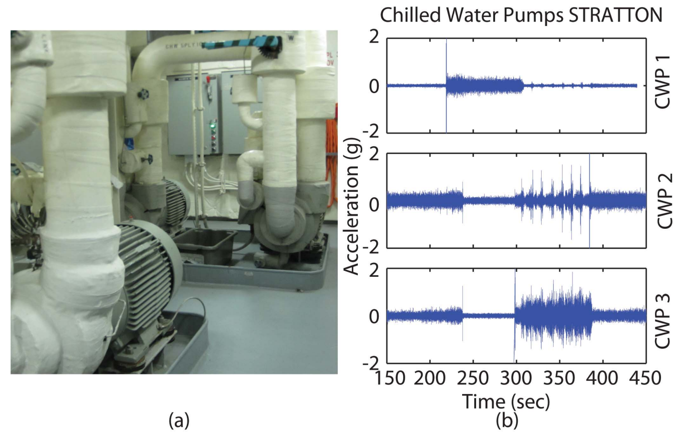


Fig. 10. (a) Chilled water pumps in close proximity on USCGC STRATTON. (b) Terminal box acceleration measurement for time referenced CWP tests.

BERTHOLF fan as operating frequency is approached at the right of Fig 9b suggests the "high vibration" condition is a result of operation near a structural resonance in the axial direction. A similar behavior is not observed for the STRATTON fan axial FRF.

B. Pump Motors

Ships contain many large induction motor driven centrifugal pumps for coolant, fire suppression, or other services. While underway, some of these services are kept under constant pressure, requiring a minimum of one operational pump at all times, and overlap during pump hand over. Their rotating components are generally higher precision and are not as susceptible to imbalance as ventilation fans, and therefore represent a more challenging case for spin-down FRF identification. Significant fluid induced vibration from the always flowing main service line can also lead to high levels of broadband vibration transmitted to the motor through the connecting pipes, even when the check valve is closed. In some cases the vibration signals during spin-down of large fluid pumps may contain significant disturbances or high noise content that obscure the generally weaker imbalance generated vibration signal critical for estimating mount FRFs The exact spin-down speed from the non-contact electric sensors should alleviate the problem of high noise and weak imbalance excitation in full implementations of the hardware.

The USCGC BERTHOLF and STRATTON each contain three chilled water pump (CWP) motors for providing continuous flow of coolant water to various thermal loads on the ships in a constantly pressurized service, shown in Fig. 10a. An acceleration logger was affixed to the terminal box of each CWP and data was recorded simultaneously. During the trial each pump was operated in series except for dual pump operation at hand over. CWP2 and 3 are located in proximity to each other and are connected via a short length of pipe leading to strong coupling of operating vibration between the terminal boxes, as observed in Fig. 10b. Also seen in Fig. 10b is significant periodic impulse vibration measured at CWP2 during operation of CWP3 (time 300 sec to 390 sec).

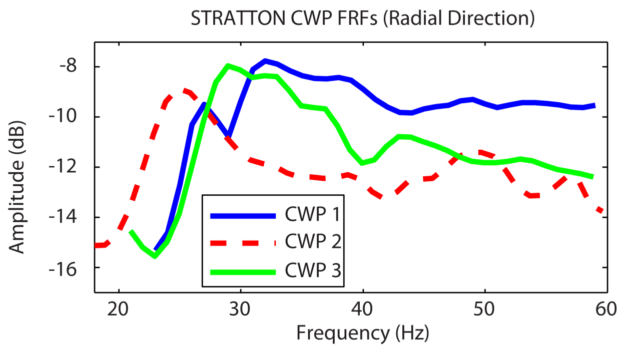


Fig. 11. STRATTON CWP mount FRF clearly showing the fundamental mounting resonance.

This machine interaction would be easily missed in conventional one at a time dock side vibration testing. FRFs estimated from the spin-down of the STRATTON CWPs are shown in Fig. 11. The primary mounting resonance is clearly visible, despite flow-induced noise coupled to the pump motors and check valve operation. This example shows the robustness of the proposed methods and hardware in noisy multi machine environments.

V. Conclusion

We present new sensor technology and algorithms to bring the benefits of highly capable permanent vibration based condition monitoring to work-horse electric pumps and ventilation fans with drastically lower installation cost per unit. Elements of the self-powering wireless design have been tested in the lab and on the deck plates of active duty vessels. Underway monitoring in noisy environments was addressed by exploiting motor spin-down shaft speed, an easily sensed signal given the close proximity to the motor's power cables afforded by the magnetic flux power harvesting design. Algorithm development focused on combining vibration and electric signals to characterize machine mount condition and provide estimates of individual machine contribution to overall structural vibration, an important factor in ship acoustic signature.

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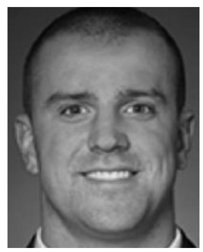

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