Dynamic Emulation of Mechanical Loads Using a Vector-Controlled Induction Motor–Generator Set

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Abstract—This paper addresses the problem of electronic emulation of both linear and nonlinear mechanical loads using a vector-controlled induction machine dynamometer. It is shown that a basic emulation scheme where the dynamometer torque demand is derived from the inverse dynamics of the emulated load is not generally viable, especially if the emulated load is part of a closed-loop speed control system. A new feedforward speed-tracking control scheme for the dynamometer is presented, which preserves the load dynamics and allows for emulation of a large class of nonlinear loads. An experimental rig is described using vector-controlled induction machines as the drive motor and dynamometer, and experimental results showing excellent emulation of both linear and nonlinear load dynamics are presented.

Index Terms—Dynamic emulation, dynamometer, mechanical load emulation.

I. INTRODUCTION

The use of torque-controlled load dynamometers is common in engine test beds or in the testing of electrical machines [1]–[4]. In these applications, the engine or electrical machine is normally tested under steady-state or slowly changing conditions. Recent research, aimed at emulating loads having faster dynamics [5]–[7], has resulted in simulated load emulation under open-loop conditions, i.e., the emulated load is not part of a closed-loop speed or position control system. Dynamic load emulation under closed-loop conditions is desirable for evaluating motor drive controllers. Researchers reporting motor control methods generally validate results using either a load machine connected to a resistor bank (emulating viscous friction) or a torque-controlled load machine emulating gravitational loads or general torque disturbances. However, adaptive and robust control schemes are attracting considerable attention. To verify the effectiveness of these, it is desirable to provide a dynamometer load in which mechanical parameters (such as inertia and friction) can either be preprogrammed or else vary with speed or position (e.g., winding applications, robot arms). In such cases, it is very desirable that the emulation preserves the model mechanical dynamics. This paper addresses this problem. In fact, the work presented was motivated by the need to validate the authors’ own work into nonlinear and fuzzy control methods.

In addition to machine/engine testing, another application of mechanical load emulation (either in open or closed loop) is to provide off-site testing of converter drives driving real industrial applications. Many of these provide challenges for the application or commissioning engineer. Examples include high-stiction loads (e.g., reciprocating pumps, escalators), period impact loads (large washing machines, compressors), the catching of spinning loads (after power interrupt), and many underhauling/overhauling applications. If the parameters of such loads are even only approximately known, the ability to evaluate and test such applications off-site would be advantageous.

The arrangement of the experimental system used in this paper is shown in Fig. 1 and consists of two vector-controlled induction motors on a common shaft, which are controlled using a microprocessor system. A PC provides user interface and data capture facilities. Further details of the practical setup are provided in Section IV. The drive machine and its inverter provide the target system for research into motion control strategies. The load machine (dynamometer) is controlled so that the mechanical rig dynamics, defined as the speed response to a given drive torque, is equivalent to that of any desired linear or nonlinear mechanical load dynamics. In this way, the emulation preserves the physical causality of a mechanical system in which the motion variables are the output responses to a driving force or torque. In this paper, we concentrate on the control of the load machine to achieve this objective.
Previous dynamic emulation research [5]–[8] is based on the principle of inverse mechanical dynamics, in which the shaft speed is measured and used to derive the desired torque for the dynamometer. In Section II, we analyze and discuss this principle, showing that discretization effects can severely affect the emulation. In [6] and [7], a model-reference approach is presented, in which it is implied that the shaft speed or position could be used as a tracking variable and so avoid the inverse dynamics. However, this is not clear, since the authors present results in which the shaft torque is the tracked variable; neither is the preservation of the mechanical dynamics (pole–zero structure) addressed in [6] and [7]. In [9], an integrator backstepping design technique is presented, which claims to emulate a dynamic load under closed-loop conditions. However, the desired torque trajectory is still derived from an inverse mechanical model, and only simulation results are given. In fact, all the researchers in [5]–[9] present only simulation results. This is also discussed in Section II. In this paper, the results are experimental, simulation results being provided only for comparison.

The dynamometer torque response imposes a limit on the mechanical dynamics that can be emulated. For the rig described in the paper, the torque bandwidth is fairly modest at about 200–250 Hz (limiting the emulated mechanical dynamics to frequencies up to 50–100 Hz). While this can be improved with direct torque control methods [10], high-frequency vibrational modes (including backlash at modest to high speeds) will always remain beyond dynamometer emulation. A lesser restriction is that an electrical torque signal of the drive motor should be available. This is the case for nearly all sensoed drives (dc, vector-induction, permanent magnet (PM), and switched reluctance (SR) drives) and is consistent with the main aim of this paper in providing a test bed for evaluating motor drive control strategies. A shaft torque transducer is, thus, not required. Those cases in which a good drive torque signal is not available (i.e., constant volts-per-hertz (V-f) drives or present-generation sensorless vector drives operating at very low speed) are discussed in Section VII.

II. EMULATION USING ONLY INVERSE LOAD DYNAMICS

We analyze here the principle of using the inverse load dynamics and consider the basic problem of emulating a linear load with inertia $J_{em}$ and friction $B_{em}$. The principle is shown in Fig. 2(a). $T_e$ and $T_{\ell}$ are the electrical torques of the drive and load machine, respectively. $G(s) = 1/(Js+B)$ is the total dynamics of the drive and load machine and connecting shaft. The speed is measured and the inverse dynamics $G^{-1}_{em}(s) = J_{em}s + B_{em}$ yields $T_{\ell}$ after compensation for the drive and load machine dynamics. Then, it is easily shown that

$$\frac{\omega(s)}{T_{\ell}(s)} = G_{em}(s) = \frac{1}{J_{em}s + B_{em}}$$

as required. However, in practice, the inverse dynamics will be implemented on a $\mu$P and sampling effects need to be
\[ \omega(k) = \frac{\omega(k) - \omega(k-1)}{T_s} \] and \[ \omega(z) = \frac{(z-1)}{T_s z} \omega(z) \] (2)
where \( T_s \) is the sampling time, yields the sampled-data system of Fig. 2(b). \( G_{th}(s) \) is the zeroth-order hold (ZOH). \( \omega(z) \) can be written as
\[ \omega(z) = \frac{T_s G(z)}{1 + G_{th}(s) G_{tl}(z)}. \] (3)
It should be noted that the numerator is \( T_s G(z) \), not \( T_s(z)G(z) \), so that the zeros of (3) depend on \( T_s \). Defining \( \Delta J = J_{em} - J \), \( \Delta B = B_{em} - B \), and \( A = \exp(-BT_s/J) \), the characteristic equation \( 1 + G_{th}(s)G_{tl}(z) = 0 \) of (3) can be derived as
\[ z^2 + \beta_1 z + \beta_2 = 0 \] (4)
where
\[ \beta_1 = \frac{(\Delta J + T_s \Delta B)(1 - A)}{BT_s} - A \] and \[ \beta_2 = -\frac{\Delta J (1 - A)}{BT_s}. \]
In general, \( B \) will be small, so that \( A \approx 1 - (BT_s/J) \) and \( \beta_1, \beta_2 \) become
\[ \beta_1 = \frac{J_{em} - 2J + B_{em} T_s}{J} \] and \[ \beta_2 = -\frac{\Delta J}{J}. \]
If \( B_{em} \) is zero or small (corresponding to the emulation of an inertial load), then the roots are \( z_1 = 1 \) and \( z_2 = -\Delta J/J \), which means the system is unstable if \( \Delta J/J > 1 \) or \( J_{em} > 2s \). Alternatively, if \( \Delta J = 0 \) and \( B_{em} \) is not zero or small, then the roots can be shown to be \( z_1 = 0 \) and \( z_2 = 1 - (B_{em} T_s/J) \). This means the system is unstable when \( (B_{em} T_s/J) > 2 \).
If the term \( \Delta J \) is included, the required relation \( \omega(s)/T_e(s) = G_{comp}(s) G_{em}(s) G_{comp}^{-1}(s) \) is obtained, and the load’s pole–zero structure is retained. Unfortunately, \( G_{comp}(s) \) contains the emulated load dynamics. This is undesirable, especially for nonlinear emulated loads. A solution is shown in Fig. 3(b), in which the inverse dynamics are fed forward from the ideal speed \( \omega_{em} \). It can now be shown that
\[ \frac{\omega(s)}{T_e(s)} = \frac{G_{em}(s) G_{comp}(s)}{1 + G(s) G_{comp}(s)} \] (7)
where
\[ G_{comp}(s) = \frac{1 + G(s) G_{comp}(s)}{G(s) G_{comp}(s)}. \] (8)
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\[ \frac{\omega(s)}{T_e(s)} = \frac{G_{em}(s) G_{comp}(s)}{1 + G(s) G_{comp}(s)} \] (7)
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\[ G_{comp}(s) = \frac{1 + G(s) G_{comp}(s)}{G(s) G_{comp}(s)}. \] (8)

III. PROPOSED EMULATION CONTROL SCHEME
Consider Fig. 3(a) with \( G_{comp}(s) = 1 \). The motor drive torque \( T_e \) is used to drive the load model \( G_{em}(s) \) to yield a desired shaft speed \( \omega_{em} \). The desired speed is used as a demand for a speed-tracking loop with a controller \( G_T(s) \) outputting the dynamometer torque \( T_f \). From Fig. 3(a), we have
\[ \frac{\omega(s)}{T_e(s)} = \frac{G_{em}(s) G_{comp}(s)}{(1/G(s)) + G_{T}(s)} \] (5)

where
\[ G_{comp}(s) = \frac{1}{(1/G(s)) + G_{T}(s)} \] (6)

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where
\[ G_{comp}(s) = \frac{1 + G(s) G_{comp}(s)}{G(s) G_{comp}(s)}. \] (8)
are not explicitly required. In fact, Fig. 3(b), with \( G_{\text{comp}}(s) \), is equivalent to Fig. 3(c). The net torque of both machines reduces to \( T_t \), which is that required to make the shaft speed travel at \( \omega_{\text{em}} \). The demand speed for the speed-tracking loop is modified by \( G_{\text{comp}}(s) \), which is the inverse of the closed-loop speed-tracking control. The response of the speed-tracking loop is, thus, not of primary concern. \( G_{\text{comp}}(s) \) is, however, not proper. This can be solved by considering the sampled-data representation for a \( \mu P \) implementation. This is shown in Fig. 4. The compensation term is nominally

\[
G_{\text{comp}}(z) = \frac{1 + G_t(z)G(z)}{G_t(z)G(z)} \tag{9}
\]

where \( G(z) = Z\{G_h(s)G(s)\} \) (the ZOH equivalent or the step invariance discretization of \( G(s) \)) and \( G_t(z) \) is a discrete controller. If \( G(s) \) is first order and \( G_t(z) \) a discrete proportional integral (PI) controller, then \( G_{\text{comp}}(z) \) of (9) can be made proper by introducing a single delay

\[
G_{\text{comp}}(z) = \frac{1 + G_t(z)G(z)}{G_t(z)G(z)} \frac{1}{z}. \tag{10}
\]

This delay can itself be compensated by setting \( G_{\text{em}}(z) \) as

\[
G_{\text{em}}(z) = z \ast Z\{G_h(s)G_{\text{em}}(s)\}. \tag{11}
\]

For a linear \( G_{\text{em}}(s) \), (11) corresponds to discretization by pole–zero matching [12]. This is quite elegant, since the \( z \)-transfer function \( \omega(z)/T_e(z) \) reduces to

\[
\frac{\omega(z)}{T_e(z)} = G_{\text{em}}(z)G_{\text{comp}}(z) \frac{G_t(z)G(z)}{1 + G_t(z)G(z)} = Z\{G_h(s)G_{\text{em}}(s)\} \tag{12}
\]

so that the emulation is equivalent to the standard \( z \)-transfer function (step invariance) discretization of \( G_{\text{em}}(s) \). For a nonlinear load, the advance operator \( z \) in (11) corresponds
to using the latest value of $T_e$ in the nonlinear difference equations (see Section VI).

Current loop delays can be included in the speed-tracking loop in series with $G_{	ext{el}}(z)$. They will form part of the closed-loop speed tracking loop and, thus, will appear in $G_{\text{em,p}}(z)$. They are difficult to model accurately, due to converter delays (although a model fit can be obtained from frequency response tests). However, if the frequencies of the emulated mechanical dynamics are kept within reasonable limits (see Section I), their neglect has little effect on the quality of emulation.

IV. EXPERIMENTAL MOTOR–DYNAMOMETER DRIVE RIG

The drive rig consists of two 0.55-kW vector-controlled induction motor drives, as shown in Fig. 5. Both 6-kHz power
MOSFET inverters are connected to the same dc bus, allowing
circulation of the generated power such that the mains supplies
only the drive losses. The kinetic energy of the rig at rated
speed can be transferred to the dc-link capacitors with only a
few volts increase in the dc-link voltage. The vector control
is based on indirect rotor flux orientation (IRFO) [13], with
the current loops operating at 2 kHz. A 10,000 line encoder
is mounted at the drive motor end. The speed loops and
load emulation operate at 400 Hz. The PI speed control-2
corresponds to \( G_e(z) \) of Fig. 4 and is designed to give a
closed-loop natural frequency for the speed tracking of 20
rad/s. The modest response of this loop is of no concern, since
the closed-loop dynamics are compensated for by \( G_{\text{comp}}(z) \),
as explained. However, it is vital that PI speed control-2 does
not saturate, otherwise, the compensation will not be valid. PI
speed control-1 is a speed controller for the whole rig (i.e., for
speed control of the emulated load). For research into control
methods, this controller is replaced by the controller under
evaluation. The output of speed control-1 is also fed as a (-ve)
torque demand to the dynamometer. The drive parameters are
given in the Appendix. It is important that the rig inertia
be known accurately. The value of \( J \) is a nominal average
over the speed range; it has been found that errors in this
parameter have only a small effect on the quality of load
emulation.

V. RESULTS FOR EMULATION OF A LINEAR LOAD

In this section, experimental results will be shown for the emulation of the linear load \( G_{\text{em}}(s) = 1/(J_{\text{em}}s + B_{\text{em}}) \). The
derived load is included in a speed control loop with PI speed control-1 operational. The results will be compared with the simulation (using the Simulink/MatLab package) of the closed-loop system, as shown in Fig. 6(a). The block \( G_e(z) \) corresponds to PI speed control-1 and is designed to give a closed-loop natural frequency of 70 rad/s for \( J_{\text{em}} = J, B_{\text{em}} = B \). Antiwindup is included in both experimental and simulated controllers.

The experimental closed-loop speed responses for three different loads of \( J_{\text{em}} = J, J_{\text{em}} = 4 \times J \), and \( J_{\text{em}} = 10 \times J \)
\((B_{\text{em}} \text{ is kept constant and equal to } B)\) and the corresponding simulation responses are shown in Fig. 6(b). \( G_e(z) \) is the same in all three cases. Also shown is the experimental torque measure \( K_T \cdot i_{d1} \cdot i_{q1} \) (where \( K_T \) is the torque constant and \( i_{d1}, i_{q1} \) are the measured stator currents of the driving machine) in comparison with the simulated \( T_e \) of Fig. 6(a). The comparison is felt to be excellent and shows that the load dynamics are preserved in the emulation. The small differences between the real and simulated responses are thought to be due to the current control loops that are ignored in the simulation model.

Fig. 7 shows the simulated and experimental speed and electrical torque responses to a 50% emulated load disturbance \( T_{\text{em}}(z) \) added before \( G_{\text{em}}(z) \) in Fig. 5 \((J_{\text{em}} = 6 \times J)\). Emulation of external torque disturbances presents no problem. Since \( G_e(z) \) is designed for \( J_{\text{em}} = J \), the increased inertia is seen to degrade the closed-loop response, as expected.

VI. NONLINEAR LOAD EMULATION

In this section, we describe the emulation of two nonlinear loads: a load with coulomb friction (stiction) and a load having an inertia as a function of speed. In both cases, \( G_{\text{em}}(z) \) becomes a set of nonlinear difference equations with \( T_e \) as the input.
A. An Inertial Load With Stiction

Stiction is a common problem in the position control of mechanical systems. Mathematically, it is difficult to model because the stiction $T_{s} = \omega$ characteristic tends toward a delta function at zero speed. However, an approximate model can be implemented. Fig. 8(a) shows a possible model function for combined viscous friction and stiction. The function is repeated in the $(-T_{s}, -\omega)$ quadrant and is only shown for $+\omega$ speed. If the stiction torque is modeled by a sine function, then with $J_{\text{em}}$ included we have

$$T_{e} = J_{\text{em}} \frac{\omega}{dt} + T_{s}(\omega)$$

$$T_{s}(\omega) = \begin{cases} B_{s} \omega + T_{s} \sin \left( \frac{\pi \omega_{s}}{\omega_{s}} \omega \right), & |\omega| \leq \omega_{s} \\ B_{s} \omega, & \text{otherwise.} \end{cases} \quad (13)$$

The value of $\omega_{s}$ should be small to obtain realistic emulation. In the rig, the minimum value of $\omega_{s}$ is determined by the resolution of the shaft encoder, which is 2.4 r/min (0.25 rad/s). Equation (13) can be discretized using backward differences to the difference equation (14), shown at the bottom of the page. The digital implementation will impose further restrictions on $T_{s}$ and $\omega_{s}$ of Fig. 8(a). Numerical instability can occur for large slopes of the $T_{s} = \omega$ characteristic. The maximum slope occurs when $\omega = 0$, and it can be shown that the system is locally unstable in the stiction region when

$$\frac{T_{s}}{J_{\text{em}}} \left( B_{s} + \frac{\pi T_{s}}{\omega_{s}} \right) > 2$$

(15)

The ratio of $T_{s}/\omega_{s}$ is, thus, limited; $\omega_{s}$ cannot be very low if $T_{s}$ is chosen high.

Experimental results will be shown for various values of $T_{s}$ for the driving torque profile shown in Fig. 8(b), where $T_{e} = 0$ for $t > 2T_{p}$. No outer speed control is placed around the load for these initial investigations. The simulation model is done in Simulink using (13).

Fig. 9 (a)–(c) shows results for $T_{s}$ = 0.7, 0.94, and 0.95, respectively, with other parameters set constant at $J_{\text{em}} = 2 \times 10^{-3}$, $B_{s} = 0.01$ N⋅m/s, $T_{p} = 0.5$ s, $\omega_{s} = 1$ rad/s, $T_{e,\text{max}} = 1$ N⋅m, and $T_{e} = 2.5$ ms. The figures also compare the experimental torque measure $K_{T} \cdot \dot{\omega}_{\text{eq}}$ with the simulated electrical torque (not shown in Fig. 9(c) for clarity).

Note that $\omega_{s}$ is set at four times the resolution of the speed encoder; if less than twice, the response is found to be dominated by resolution effects. The results of Fig. 9 are good, especially since the value of stiction is between 70%–95% of the input torque. In Fig. 9(c), the input torque only just overcomes stiction, and the system speed does not even reach $\omega_{s}$; however, even though resolution effects dominate, the shaft speed is still seen to follow the desired emulation speed. Finally, Fig. 10 shows the effect of local instability when the stiction parameters are set to $\omega_{s} = 0.5$ rad/s, $T_{s} = 2$ N⋅m, and $T_{e,\text{max}} = 1.3$ N⋅m, so violating the stability condition of (15). The local instability is clearly seen as the speed attempts to settle about zero.

B. Emulation of a Watt Governor

The Watt governor is a good example of a nonlinear device having an effective inertia (and friction) variation during motion. The physical structure is shown in Fig. 11, and consists of two pendulums fixed at $O$; when the shaft rotates, the balls fly outwards due to the centrifugal force. The original governor contains further mechanics for the regulation of the shaft speed (e.g., the balls are connected by a link to a sleeve sliding parallel on the shaft). The extra linkages are of little or no interest if the aim is only to emulate a realistic variable inertial load.

The motor torque sees an effective inertia and friction

$$T_{e} = J_{\text{eq}} \dot{\omega} + B_{\text{eq}} \omega$$

(16)

$$\omega(k) = \begin{cases} \left( 1 - \frac{B_{s} T_{s}}{J_{\text{em}}} \right) \omega(k - 1) + \frac{T_{s}}{J_{\text{em}}} T_{e}(k) - \frac{T_{s}}{J_{\text{em}}} \sin \left( \frac{\pi}{\omega_{s}} \omega(k - 1) \right), & |\omega| \leq \omega_{s} \\ \left( 1 - \frac{B_{s} T_{s}}{J_{\text{em}}} \right) \omega(k - 1) + \frac{T_{s}}{J_{\text{em}}} T_{e}(k), & |\omega| > \omega_{s} \end{cases} \quad (14)$$
Fig. 9. Experimental and simulated speed and torque responses for (a) $T_{ad} = 0.7$ N·m, (b) $T_{ad} = 0.94$ N·m, and (c) $T_{ad} = 0.95$ N·m.

Fig. 10. Experimental and simulated speed responses for a local instability case.

where $J_{ef} = J_{em} + 2mL^2 \sin^2 \theta$ and $B_{ef} = B_{em} + 2mL^2 \dot{\theta} \sin(2\theta)$ and $J_{em}$ and $B_{em}$ are the inertia and friction assigned to the shaft itself. Defining the states as $x = [\omega, \dot{\theta}, \theta]$ and $B_o$ as viscous friction at $O$ (desirable if a damped system is required), the state equations can be derived as

$$
\begin{align*}
\dot{x}_1 &= -\frac{B_{em} + 2mL^2 \omega \sin(2\theta)}{J_{em} + 2mL^2 \sin^2(2\theta)} - x_1 \\
&\quad + \frac{1}{J_{em} + 2mL^2 \sin^2(2\theta)} T_e \\
\dot{x}_2 &= -\frac{B_o}{mL^2} x_2 + \frac{1}{2} \omega^2 \sin(2\theta) - \frac{g}{L} \sin(\theta) \\
\dot{x}_3 &= x_2
\end{align*}
$$

(17)

Fig. 11. The physical structure of a Watt governor.
which can be discretized in the normal manner. Note that, due to nonzero radius of the balls, the equations must be supplemented with the condition \( |x_3(k)| < \alpha \text{abs}(\theta_{\text{emin}}) \) then \( x_3(k) = \alpha \text{abs}(\theta_{\text{emin}}) \).

Equations (17) are implemented in the \( G_{\text{em}}(z) \) block (the emulated load block) of Fig. 5. Fig. 12(a) and (b) shows the experimental speed and electrical torque responses in comparison with Simulink simulation when the system is driven by a 1 N\( \cdot \)m step torque input (the parameters are \( m = 0.1 \text{ kg}, \ell = 0.15 \text{ m}, B_o = 0.1 \text{ N} \cdot \text{m} \cdot \text{s}, J_{\text{em}} = 0.007 \text{ kg} \cdot \text{m}^2 \), and \( B_{\text{em}} = 0.01 \text{ N} \cdot \text{m} \cdot \text{s} \)). The simulation model is done in Simulink using (17). A glitch occurs at about 0.5 s; this is due to the sudden increase in \( \theta \) caused by the centrifugal force lifting the balls. As the balls fly outwards, the effective friction \( B_{\text{ef}} \) rapidly increases and slows down the increase in shaft speed. Again, excellent agreement between the simulated response and the experimental response is achieved. Fig. 12(c) shows the variations of the angle \( \theta \), \( J_{\text{ef}} \), and \( B_{\text{ef}} \) during the shaft acceleration. These values are, of course, taken from the governor model implementation of the emulated load block \( G_{\text{em}}(z) \) of Fig. 5, since they do not exist as real mechanical parameters/variables in the experimental rig.

VII. CONCLUSION

This paper has developed a dynamometer control strategy for the dynamic emulation of both linear and nonlinear mechanical load dynamics, such that these dynamics (or pole–zero structure for a linear load) are preserved during the emulation. The emulation can, thus, be used for the testing of motor drive control strategies. The emulation strategy is based on a speed-tracking control with implicit feedforward of the inverse dynamics and compensation for closed-loop tracking control dynamics. The inverse dynamics do not need to be implemented in practice; no derivative computations are necessary, and the scheme does not suffer from noise.

Experimental validation is based on the principle of output speed equivalence to a given input drive torque. Experimental results on a vector-controlled induction motor–dynamometer rig have shown excellent equivalence with simulation. The equivalence achieved when the emulated dynamics are placed in a speed control loop are also impressive and indicate that the mechanical dynamics are preserved for frequencies within the control loop bandwidth. Experimental emulations of stiction and a load having centrifugal components have also yielded good equivalence. For these nonlinear loads, it would be very
difficult to obtain such equivalence using the inverse dynamic approach.

The validation has not been done on a real mechanical rig. However, it is noted that any electronic emulation can only be an emulation of a load model. If the emulation is validated against the model (as this paper has shown), then it follows that using a real load would serve only to validate the model and not the emulation.

The emulation requires the drive motor torque reference signal. This is not a restriction given the aim of this work to provide a test bed for motor drive control. If a torque reference signal is not available, the authors would recommend, where possible, that the motor drive voltages and currents be measured and fed to an electrical torque observer based on the model equations of the drive machine. It is felt that errors in the estimated torque would be less problematic than the discretization and noise effects arising from the inverse dynamics.

The emulation of further mechanical load dynamics is beyond the scope of this paper. It is noted, however, that if a compliant load is being emulated, then shaft position can be used as the tracking variable instead of speed. Position-tracking controllers will be implemented on the rig; emulation of vibrational effects, as well as adaptive and robust fuzzy controller studies using the emulation system, will be the subject of future publications.

APPENDIX

A. Inverters

Input supply single phase, 240 V, 50 Hz; output supply three phase, 240 V, 50 Hz; output rated power 1.1 kW.

B. Induction Machines

Output rated power 0.55 kW; rated speed 1400 r/min; total shaft inertia (including both machines and coupling) \( J = 0.0035 \text{ kg-m}^2 \); viscous friction (total) \( B = 0.0007 \text{ N-m-s} \).

REFERENCES


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