

# MAGNETIC FIELD AND FORCE IDENTIFICATION OF COMMERCIAL MAGNETS FOR BUILDING ELECTROMAGNETIC SHAKERS

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**Abstract.** *The identification process of dynamic systems requires that the system be excited, and the vibration response be measured. Regarding the excitation, it can be done by several means, being electrodynamic shakers and impact hammers the most common ones. Electromagnetic shakers, as excitation devices, present the advantages of not contacting the system (drawback of the electrodynamic shaker) and repeatability of force direction (drawback of the impact hammer). However, the design of electromagnetic shakers is more complex, and its operation requires a force control system to guarantee I/O fidelity (input signal/output force). In this work, one presents the experimental identification of magnetic field forces of an electromagnetic shaker composed by two commercially available electromagnets. The idea is to build a cost-effective electromagnetic shaker with commercial electromagnets, but for that, the relationship between force and magnetic field must be identified. The experimental identification is based on response surface models, and the force signal is correlated to the magnetic field signal. Based on this relationship, the force control system is designed, using the magnetic field measurements as the feedback signal (closed loop control). Results show the strong influence of input voltage and distance to object parameters on force/magnetic field signals.*

**Keywords:** *parameter identification, electromagnetism, electromagnetic shakers*

## 1. INTRODUCTION

Rotating systems such as pumps, compressors, turbines, machine spindles, are designed to present dynamic characteristics suitable for their respective application fields. These dynamic characteristics (stiffness, damping, frequency response function) are often defined early in design, based on correlated mathematical models or based on empirical experience. Hence, sometimes it is necessary to measure such characteristics, be it for model correlation and validation, be it for knowing a new system that has never been tested before.

The identification of dynamic characteristics of a structure requires the excitation of the structure and measurement of structure response. The excitation is done by applying known forces, whose frequency spectrum is also known. The measurement of the structure response is done by positioning sensors all over the structure, in points of interest, thus acquiring displacement, velocity or acceleration values. Hence, by knowing the force (input) and the response (output), one can determine the transfer function matrix of the structure that contains the dynamic characteristics between the input and output points (Ewins, 1984; Maia and Silva, 1997).

The excitation of the structure can be done in different ways, but the most common ones are by using impact hammers and electrodynamic shakers (Fig.1). In both cases, a piezoelectric load cell is fixed in the input point to measure the input excitation force. The use of an impact hammer has the advantages of easy handling, easiness of changing input points, lightness and compactness. However, only impulsive excitations can be performed by the impact hammer. Moreover, repeatability and force alignment (direction of impact) are jeopardized because they strongly depend on the precision of the hammer operator hand. Thus, the use of electrodynamic shakers is indicated when it is necessary to adopt non impulsive force excitation and when repeatability in a certain excitation direction is required. Electrodynamic shakers are composed of a solenoid whose magnetic field moves the central cursor. The magnetic field is proportional to the input signal, and generates the desired excitation force. The excitation of the structure is done by attaching the shaker to the structure. The work of Roger and Ewins (1989) presents an extended discussion on dynamic identification of rotating systems, focusing on advantages and disadvantages of impact hammers and electrodynamic shakers, and the necessary caution in the use of proximity probes in the measurement procedures.

The dynamic identification of rotating systems obey the same principles of the dynamic identification of static structures: system excitation and response measurement. However, the use of electrodynamic shakers in rotating systems results in an additional difficulty: how do you attach the shaker to something that is rotating? In most cases, the shaker is attached to a roller bearing which is mounted in the rotating shaft (Bucher et al., 1996; Bucher, 2001; Maia and Silva,



Figure 1. Most common exciters used in the dynamic identification of structures.

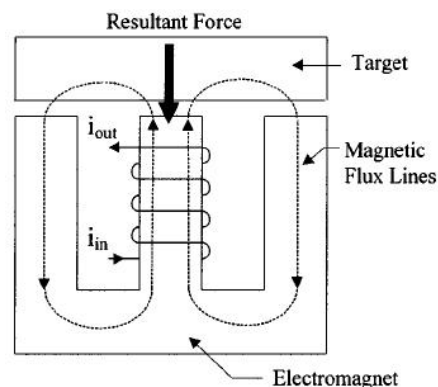


Figure 2. Schematic view of an electromagnetic shaker.

1997; Nicoletti and Santos, 2005). The adoption of a roller bearing to couple the electrodynamic shaker to the rotating shaft is a viable solution which gives good results in the dynamic identification of the system. The disadvantage lies on the fact that the roller bearing can be a limiting factor in the operational range of the shaker. This limitation can be related to the load (amplitude of excitation force), noise generated by the rollers, and shaft rotating velocity. The higher the shaft rotating velocity is, the smaller the roller bearing must be for a safe operation within the bearing manufacturer specifications (the inertia of the rollers becomes an important factor). Hence, this solution is not feasible for the identification of rotating systems under high rotating velocities (over 10000 *rpm*). Under these conditions, the roller bearing recommended by the manufacturer is often smaller than the shaft radius of the machine. Alternative ways of attachment or excitation must be adopted.

An alternative way of exciting high velocity rotating systems could be by changing the roller bearing to a hydrodynamic bearing. This change would need a continuous supply of oil to the hydrodynamic bearing which would result in an excessive increase of system complexity. Moreover, the oil film interface in the bearing has its own dynamic characteristics (equivalent stiffness and damping) which will interfere in the identification procedure. Hence, other alternatives must be chosen, being the use of electromagnetic shakers one of them.

Electromagnetic shakers are solenoids (magnets) whose electromagnetic forces are used to excite the structure (Fig.2). These exciters have the advantage of applying excitation forces without direct contact to the structure or the rotating shaft. Thus, they can be used in the dynamic identification of rotating systems regardless the rotating velocity of the shaft. One can find in literature many examples of the use of electromagnetic shakers (Joh and Lee, 1993; Arumugam et al., 1995; Jiang et al., 1997; Jing et al., 1998), where active magnetic bearings are also used for shaft supporting and excitation (Kasarda, 2000; Lindlau and Knospe, 2002).

The great challenge in developing electromagnetic shakers lies on the fact that the excitation force cannot be directly measured during the identification procedure, but only be estimated. Contrary to electrodynamic shakers where a load cell can be attached between the shaker and the structure, no force transducer can be mounted in the gap between the electromagnetic shaker and the structure/shaft, otherwise one would miss the non contact advantage of such devices. The solution is measuring the magnetic field in the gap, and using this information to calculate the magnetic excitation forces.

The correlation between the magnetic field and the magnetic force can be done either analytically or experimentally. If the correlation is done analytically, one may obtain poor results between real and estimated magnetic forces due to the successive simplifying hypotheses that are usually adopted to model the magnet. Better results can be obtained experimentally, by adopting response surface models, where the magnetic field and the magnetic force are simultaneously measured in a test rig and correlated (Dyck and Murray, 2001).

In this work, one presents the magnetic field and force identification of commercial electromagnets aiming at building an electromagnetic shaker with commercially available electromagnets. The response surfaces are obtained experimentally and correlated, in order to be used as a force estimator in the force control strategy of the shaker. The force control strategy to be used in the shaker is also presented and discussed.

### 1.1 The Electromagnetic Shaker with Commercial Electromagnets

The electromagnetic shaker in study is composed of a pair of electromagnets located in both sides of the shaft, in the point of excitation (Fig.3). The desired excitation force (reference) is obtained by controlling the input signals to the electromagnets. The magnetic field in the gap between the electromagnets and the shaft is measured by Hall sensors, whose signals are used as a feedback in the force control system.

The idea of using commercial electromagnets is motivated by the simplicity of assemblage, mounting, availability

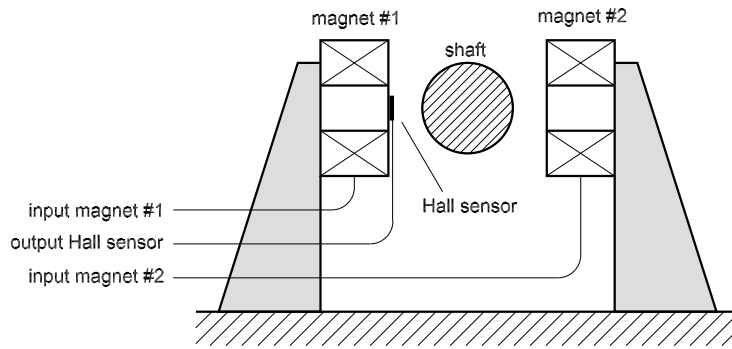


Figure 3. Schematic view of the electromagnetic shaker with commercial magnets.

of models, and cost. One can build a drive to operate and control the electromagnets, and have shakers with different characteristics by simply changing the electromagnets, which are commercially available in a variety of models. In this work, the electromagnets are provided by Metalmag Produtos Magnéticos Ltda. A picture of the electromagnet is shown in Fig.4, and the characteristics for the different electromagnet models are presented in Tab.1.

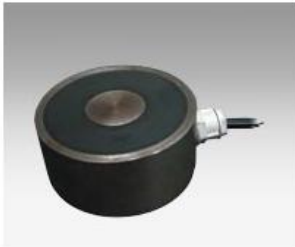


Figure 4. Metalmag commercial magnet.

Table 1. Characteristics of Metalmag electromagnets.

| model  | diameter (mm) | height (mm) | maximum force <sup>1</sup> (N) |
|--------|---------------|-------------|--------------------------------|
| 30/40  | 30            | 40          | 100                            |
| 40/40  | 40            | 40          | 210                            |
| 50/45  | 50            | 45          | 380                            |
| 75/50  | 75            | 50          | 930                            |
| 100/60 | 100           | 60          | 1680                           |

<sup>1</sup> force to detach a carbon steel piece from the electromagnet.

## 2. MAGNETIC FIELD FORCE IDENTIFICATION

As mentioned before, the magnet force imposed by the shaker to the shaft must be estimated because it is not possible to measure the force directly. This is done by correlating the force with the magnetic field developed by the electromagnet. The magnetic field depends on two parameters: the gap between the electromagnet and the object, and the input voltage of the electromagnet. Hence, in order to measure the magnetic field and resultant magnetic force for different gaps and input voltages, one has built the test rig shown in Fig.5.

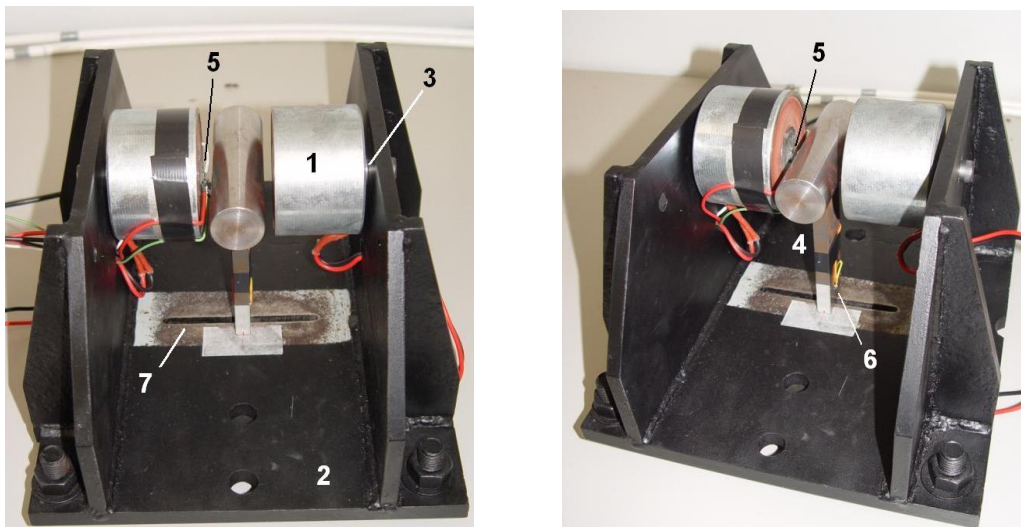


Figure 5. Test rig for measuring the magnetic field and resultant magnetic force of the shaker for different gaps and input voltages: (1) electromagnet, (2) base, (3) spacer, (4) beam load cell, (5) Hall sensor, (6) strain gauge, (7) groove.

The test rig is composed of a base (Fig.5 (2)) where the electromagnets (Fig.5 (1)) are fixed with spacers (Fig.5 (3)). By changing the spacers, it is possible to change the distance between the electromagnets. A Hall sensor is fixed in the surface of the electromagnet (Fig.5 (5)) and it is used to measure the magnetic field in the gap between the electromagnet and the shaft. The shaft is attached to a steel beam load cell (Fig.5 (4)) of cross section  $10 \times 30 \text{ mm}$ , and two  $120 \Omega$  strain gauges (Fig.5 (6)) fixed in each side of the beam. The beam load cell is bolted to the base, and its position can be altered by using the machined groove of the base (Fig.5 (7)). By using the groove, one can change the gap between the shaft and the electromagnets (off-center position). The calibration curve of the load cell is shown in Fig.6, from which the sensitivity of  $17.85 \text{ N/V}$  is obtained.

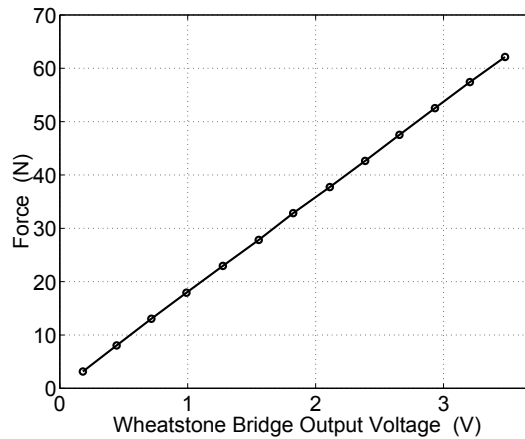


Figure 6. Calibration curve of the steel beam load cell.

The electromagnets of the test rig are the Metalmag model 75/50 (Tab.1), whose input signal for maximum force is  $24 \text{ V}$ . The input signal is controlled by a drive which generates pulse width modulated (PWM) signals to the electromagnets (Fig.7). In Fig.7, the control voltage ( $V_C$ ) can vary from  $-10$  to  $+10 \text{ V}$ . If  $V_C = -10 \text{ V}$ , then the drive generates a PWM signal to magnet #1 ( $V_1$ ) with 0% pulse width and a PWM signal to magnet #2 ( $V_2$ ) with 100% pulse width (amplitude of  $24 \text{ V}$ ). Likewise, if  $V_C = +10 \text{ V}$ , the drive generates a PWM signal to magnet #1 ( $V_1$ ) with 100% pulse width and amplitude of  $24 \text{ V}$ , and a PWM signal to magnet #2 ( $V_2$ ) with 0% pulse width. If  $V_C = 0 \text{ V}$ , then the drive generates PWM signals with 50% pulse width ( $24 \text{ V}$  amplitude) for both electromagnets. The Hall sensor generates the output signal  $V_B$ , proportional to the magnetic field in the gap between magnet #1 and the shaft. The resultant force is measured by the beam load cell, whose Wheatstone bridge gives the output signal  $V_S$ .

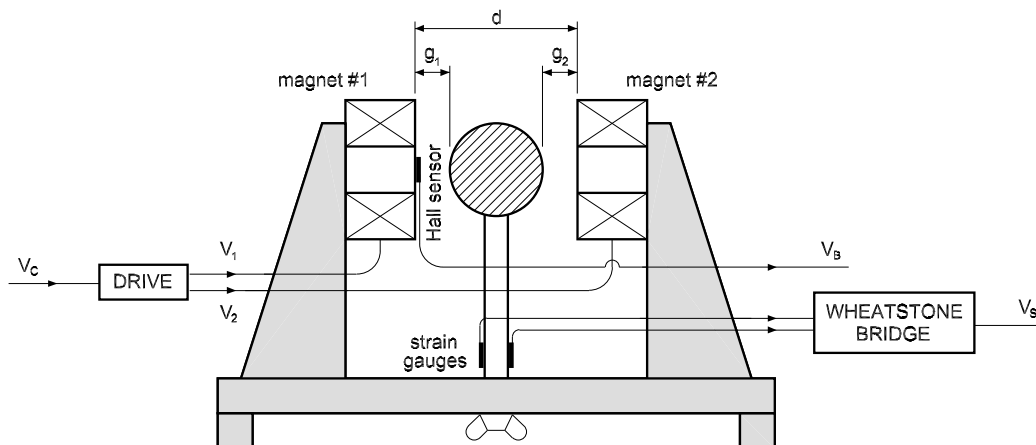
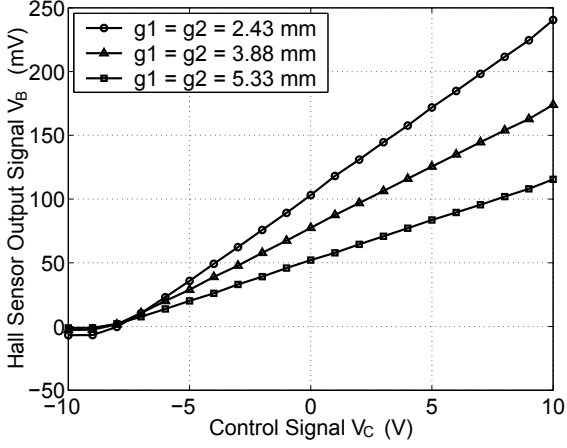


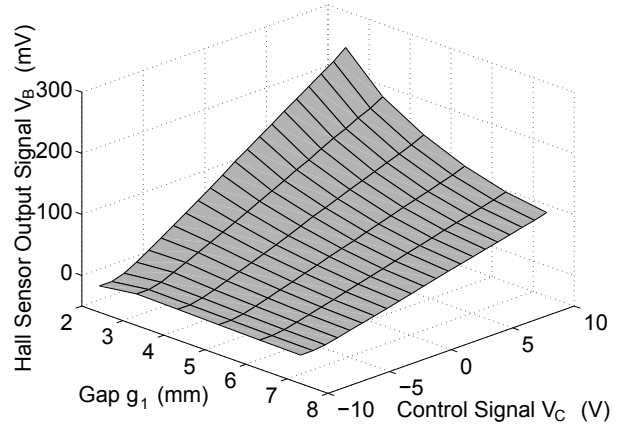
Figure 7. Schematic view of the test rig (input and output signals).

Figure 8 presents the experimental results of the Hall sensor output signal, which is proportional to the magnetic field, for different gap conditions. In Fig.8(a), the shaft remained centered in the test rig and the electromagnets were mounted with different spacers, thus equally changing the gaps  $g_1$  and  $g_2$ . In Fig.8(b), one varies the position of the beam load cell using the groove in the base, thus changing  $g_1$  and  $g_2$  ( $g_1 \neq g_2$ ). As one can see in Fig.8(a), the Hall sensor signal  $V_B$  linearly varies by increasing the control voltage  $V_C$ . This is consistent because the Hall sensor is located on magnet #1,

and this electromagnet receives more power as  $V_C$  tends to +10 V (the Hall sensor is not affected by the magnetic field generated by magnet #2). In the range between -10 and -8 V, the electromagnet receives no power, showing that the drive is not using the whole range of -10 to +10 V to control the magnets (circuit non linearity). By decreasing the gap  $g_1$ , one achieves a nearly quadratic increase of output signal, which reflects the increase of magnetic field in the gap (Fig.8(b)).



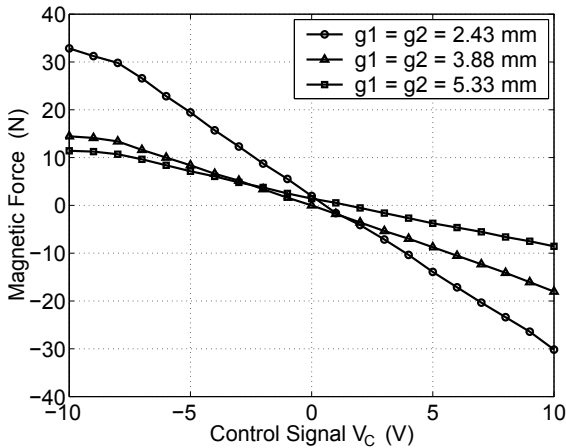
(a) centered shaft



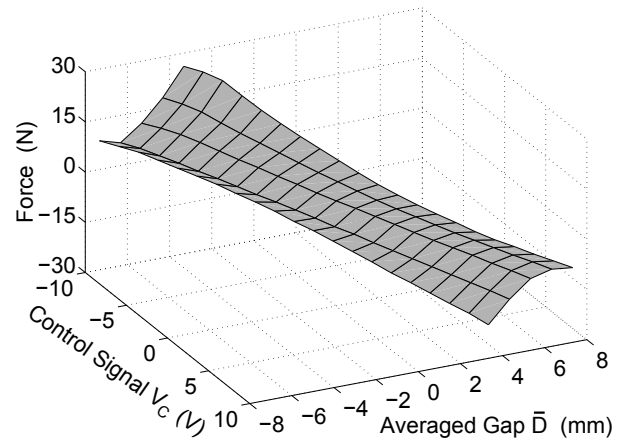
(b) off-centered shaft

Figure 8. Experimental results of the Hall sensor output signal for different gap conditions.

Figure 9 presents the experimental results of the magnetic force for the same gap conditions of Fig.8(a). In Fig.9(a), the shaft remained centered in the test rig and the electromagnets were mounted with different spacers, thus equally changing the gaps  $g_1$  and  $g_2$ . In this case, the magnetic force is positive when  $V_C < 0$  V, which means that the net force acting on the shaft is towards magnet #2. This is consistent to the fact that, for  $V_C < 0$  V, more power is supplied to magnet #2, i.e. the force generated by magnet #2 is greater than the force generated by magnet #1. Similarly, as  $V_C$  becomes greater than zero, the net magnetic force is negative, which means that the applied force is towards magnet #1. Hence, the magnetic force is linearly proportional to the control voltage  $V_C$ , with a non linear region between -10 and -8 V caused by the drive circuit non linearity.



(a) centered shaft



(b) off-centered shaft

Figure 9. Experimental results of the magnetic force for different gap conditions.

In Fig.9(b), one varies the position of the beam load cell using the groove in the base, thus changing  $g_1$  and  $g_2$  ( $g_1 \neq g_2$ ). In this case, the averaged gap parameter  $\bar{D}$  is introduced to help defining the surface response of the system. The averaged gap parameter is defined by:

$$\bar{D} = P_1 g_1 - P_2 g_2 \quad (1)$$

where  $P_1$  and  $P_2$  are the percentage of power supplied to magnets #1 and #2, respectively, for a given value of control voltage  $V_C$ . This parameterization is necessary because the measured force depends on the force generated by both

electromagnets. Considering that, the force generated by the electromagnet depends not only on the percentage of power supplied, but also on the respective gap to the shaft, this parameterization helps defining a unique surface response for the system that takes into account all variables. The surface response of the force, shown in Fig.9(b), is valid for any  $\bar{D}$ , i.e. for any combination of gaps and percentage of power supplied to the electromagnets.

The percentage of power supplied to the electromagnets is the ratio between the instantaneous power, which depends on the control signal  $V_C$ , and the maximum power of the electromagnet. Considering the linear region of the system (Figs.8(a) and 9(a)), one can represent the percentage of power supplied to the electromagnets by the expressions:

$$P_1 = \frac{V_C + 8}{18}, \quad \text{valid for } -8 < V_C < +10 \text{ V} \quad (2)$$

$$P_2 = \frac{10 - V_C}{18}, \quad \text{valid for } -8 < V_C < +10 \text{ V} \quad (3)$$

From Fig.8(a), when the control signal  $V_C$  is equal to  $-8 \text{ V}$ , electromagnet #1 is at 0% power and no magnetic field is detected by the Hall sensor ( $P_1 = 0$ , Eq.(2)). When the control signal  $V_C$  is equal to  $+10 \text{ V}$ , electromagnet #1 is at 100% power and maximum magnetic field is detected by the Hall sensor ( $P_1 = 1$ , Eq.(2)). A similar, but inverse, relationship is presented for electromagnet #2 (Eq.(3)).

Hence, the identification of the field force of an electromagnetic shaker composed of commercial electromagnets resulted in the surface responses shown in Figs.8(b) and 9(b). These surfaces were obtained simultaneously for the same operational conditions. By using these surfaces, one can correlate the resultant magnetic force acting on the shaft with the Hall sensor output signal. This will be useful in the estimation of the excitation force of the shaker, necessary for controlling.

### 3. FORCE CONTROL STRATEGY

As mentioned before, in the dynamic identification of structures/rotating systems, the shaker must excite the system with a desired force, whose frequency spectrum is known. This means that, the shaker must apply a time varying force with precise amplitude. In the case of electromagnetic shakers, the force that is applied to the structure depends not only on the control signal, but also on the distance to the structure. If the structure moves due to the action of the exciting force, the force itself will change due to the variation of the gap between the magnets and the structure. Hence, in order to guarantee that the applied force is the desired time varying force with correct amplitude, a control system feedback must be employed.

Ideally, by measuring the force that is applied to the structure/rotating system, one can build a closed loop system. The control system takes into account the error between the desired force (reference) and the measured force (real), and changes the control signal to the magnets by adopting a control law. However, in a electromagnetic shaker, it is not possible to measure the applied force without changing significantly the structure. Hence, the force that is being applied to the structure must be estimated.

A block diagram of the electromagnetic shaker with commercial electromagnets is presented in Fig.10. The desired force ( $F_{ref}$ ) is compared to the estimated force, giving the error  $e$ . By adopting a control law (e.g. PID), one calculates the necessary control signal  $V_C$ . By knowing  $V_C$ , the input signals to the electromagnets ( $V_1$  and  $V_2$ ) are generated by the drive circuit. The resultant magnetic fields, created by the electromagnets, force the structure (plant), whose output response is  $x$ . The magnetic field ( $B$ ) is measured by the Hall sensor located on the surface of magnet #1, whose output signal is  $V_B$ . By taking the signals  $V_C$  and  $V_B$ , one can estimate the magnetic force ( $F$ ) acting on the structure and close the control loop.

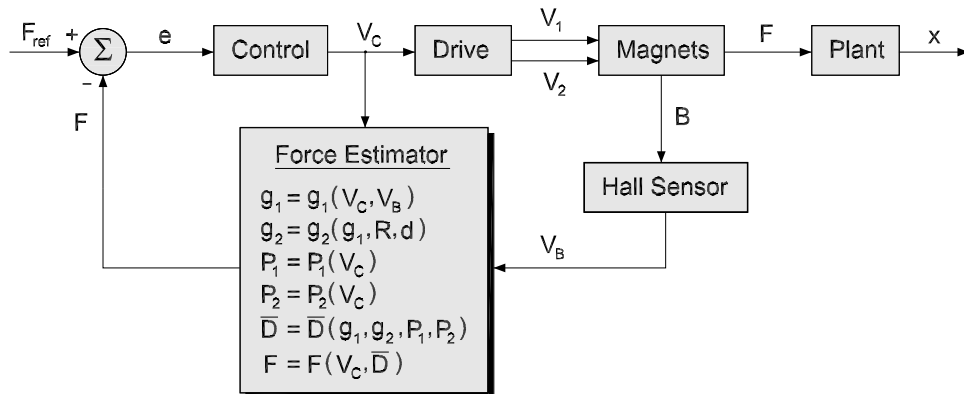


Figure 10. Block diagram of the electromagnetic shaker (control system strategy).

The estimation of the force begins with the determination of the gap  $g_1$  between magnet #1 and the structure. This is done with help of the response surface shown in Fig.8(b), where one can easily see that  $g_1$  is a unique function of the control voltage  $V_C$  and the Hall sensor output signal  $V_B$  ( $g_1 = g_1(V_C, V_B)$ ). By knowing  $g_1$ , one can determine the gap  $g_2$  between magnet #2 and the structure by the following relationship:

$$g_2 = g_2(g_1, R, d) = d - g_1 - 2R \quad (4)$$

where  $R$  is the shaft radius (or half the thickness of the structure), and  $d$  is the distance between the electromagnets.

The next step is finding the percentage of power supplied to the electromagnets, given by Eqs.(2) and (3). By knowing  $g_1$ ,  $g_2$ ,  $P_1$ , and  $P_2$ , it is possible to calculate the averaged gap  $\bar{D}$  with Eq.(1). The estimated force is finally determined with help of the surface response shown in Fig.9(b), where the force is a unique function of the control voltage  $V_C$  and the averaged gap  $\bar{D}$  ( $F = F(V_C, \bar{D})$ ). Considering that functions  $g_1$ ,  $P_1$ ,  $P_2$  and  $F$  are empirical (obtained experimentally in test rig), and functions  $g_2$  and  $\bar{D}$  are analytical, the error in the estimation of the excitation force is minimized, and the force control system of the electromagnetic shaker can work more efficiently.

#### 4. CONCLUSIONS

An electromagnetic shaker can be a suitable solution when a non contacting excitation device is needed in dynamic identification of structures or rotating systems. The idea of designing an electromagnetic shaker with commercial electromagnets is motivated by the potential simplicity of assemblage and mounting, and low cost. Due to the gamut of electromagnet models commercially available, it is possible to have shakers with different load capacities, using the same drive circuit, by simply changing the electromagnets.

Whatever the electromagnet model is, the magnetic force of the shaker must be measured (or estimated) during the identification procedure, otherwise it will not be possible to guarantee the desired frequency spectrum of the excitation force. The force control system of the shaker needs a feedback of the exerted force. In the electromagnetic shaker with commercial electromagnets, one proposes the use of surface response functions, obtained experimentally in test rig. By measuring the magnetic field and magnetic force under certain controlled conditions, it is possible to build a force estimator based on the correlation of the experimental data.

The obtained results show that both magnetic field and force are linear functions of the control voltage, and nearly quadratic functions of the gap between the electromagnets and the structure. Moreover, the maximum force obtained with the adopted magnets (Metalmag model 75/50 - Tab.1) was 30 N for a gap of about 2 mm. If a bigger excitation force is needed, one can mount the electromagnets closer to the structure (gap smaller than 2 mm), or change the electromagnets to more powerful ones (e.g. Metalmag model 100/60 - Tab.1).

#### 5. ACKNOWLEDGEMENT

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