VIBRATION INDUCED FAILURE ANALYSIS OF A HIGH SPEED ROTOR SUPPORTED BY
ACTIVE MAGNETIC BEARINGS

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ABSTRACT
Active Magnetic Bearings (AMBs) are increasingly used in various industries and a quick re-levitation of AMBs supported high speed flexible rotor is necessary in case of vibration induced failure. A robust fault diagnosis algorithm is presented to detect suspected saturation type of nonlinearity associated with a power amplifier. A five degree-of-freedom AMB system consisting of four opposing pair of radial magnets and a pair of axial magnets is considered. In this paper failure of an industrial grade AMB system is investigated using Sinusoidal Input Describing Function (SIDF) method. SIDF predicts the gain and frequency at which failure occurs. It is demonstrated that the predicted frequency is in agreement with the frequency at which failure occurs.

Keywords: Active Magnetic Bearings (AMBs); vibration; describing function analysis; high speed rotating machines.

ANALYSE DE DÉFAILLANCE (À CAUSE DE VIBRATION INDUITE) D’UN ROTOR À HAUTE VITESSE SUPPORTÉ PAR DES PALIERS MAGNÉTIQUES ACTIFS

RÉSUMÉ
Les paliers magnétiques actifs sont de plus en plus utilisés dans diverses industries. Une nouvelle lévitation rapide du rotor flexible à haute vitesse est nécessaire en cas de défaillance à cause de vibration induite. Un algorithme robuste de diagnostic de panne présenté pour détecter une saturation soupçonnée de type non-linaire. Un système de paliers magnétiques actifs à 3-degrés de liberté consiste en quatre paires d’aimants radiaux et une paire d’aimants axiaux est étudié. Dans cet article une défaillance d’un palier magnétique actif de type industriel est investiguée en utilisant la méthode Sinusoidal Input Describing Function (SIDF). La méthode SIDF prévoit le gain et la fréquence à laquelle se produit la défaillance. Il est démontré que fréquence prédict correspond avec la fréquence à laquelle se produit la défaillance.

Mots-clés : palier magnétique actif; vibration; analyse de fonction descriptive; machines rotatives à grande vitesse.

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1. INTRODUCTION

Active Magnetic Bearings (AMBs) are frictionless bearings that do not use lubricating fluids and works on the principle of electromagnetic forces for supporting a high speed rotor. Due to absence of working fluid and its non-contact nature it has found applications in several industries. Nevertheless trip or failure of high speed rotating machines supported by AMBs is a common occurrence resulting in shut-down of process for several hours to several days. This aspect of AMB takes significance in particular when AMBs are deployed in areas such as natural gas compression, process industry and similar applications where even a short shut down time can result in huge financial loss. Successful redeployment of failed AMB supported rotating machinery depends crucially on accurate analysis of the trip data. However this can be very time consuming due to the large number of signals recorded for post-failure processing and analysis.

Lateral synchronous vibrations in high speed rotors are mainly due to imbalance force travelling at the same speed as that of the shaft. Synchronous vibrations are attenuated in AMB system with the implementation of rotor speed dependent notch filters. However, there are also non-synchronous shaft whirling and vibrations that may be caused due to closer seals clearances [1] friction, hysteresis, working fluid pressure or presence of other nonlinear forces [2, 3]. The power amplifier which drives the current to electromagnet windings also has nonlinear characteristics and constitutes an important element of feedback loop of AMB system. The amplifier saturation nonlinearity has a direct impact on achievable force slew rate and dynamic current. The large force demand at high rotational frequencies causes power amplifier saturation which can lead to deterioration of bearing closed loop characteristics and eventually to instability. Therefore, any of these forces or combination of these forces can lead to non-synchronous vibrations and may lead to failure. Non-synchronous vibrations are either sub-synchronous, i.e. below the rotational shaft frequency, or super-synchronous, i.e. above the rotational frequency of rotor.

There are some reported research works on fault detection of AMB systems [4–6] employing linear system identification techniques to predict system faults, whereas industrial type AMBs as considered in this paper are far more complex with flexible rotors and severe nonlinearities. It is evident therefore from the above discussion that a set of robust algorithms are needed for post-processing of trip or failure data in order to detect the cause or type of fault of industrial type magnetic bearings. Development of such algorithms requires thorough understanding of magnetic bearings as well operational experience of AMB systems. The
paper therefore is concerned with the development of one such algorithm to detect and analyse a certain class of faults. In particular this work investigates and report failure of high speed rotor supported by AMB due to suspected saturation power amplifier nonlinearity. The sudden failure of AMB supported rotor running at several thousand rpm in industrial setting is taken as a case study. The sinusoidal input describing function (SIDF) approach is employed to conduct the nonlinear analysis of AMB system. In particular emphasis is placed on estimating the frequency at which limit cycle oscillation sets in leading to eventual instability. The predicted limit cycle frequency is compared with that of real world trip data.

The methodology presented will be of great help to practising engineers in the field of AMB systems to rapidly analyse and isolate one of the main causes of failure AMB systems. Timely analysis will be of huge significance to the end user (i.e. industry) as well as to the supplier of AMB systems. Despite the importance of quick failure analysis and re-levitation of AMB supported turbo machinery, the work presented here is the first of its kind to be reported in the literature to the best of the authors’ knowledge.

To this end, a brief account of AMB working principle and related nonlinearities are given in Section 2. Section 3 describes the SIDF technique for nonlinear system analysis and Section 4 presents nonlinear analysis of AMB-rotor system. In Section 5 conclusions are given.

2. AMB SYSTEM WITH NONLINEARITIES

Figure 1 shows a sketch of a typical AMB system and Fig. 2 provides a block diagram of different components within a magnetic bearing system. Basically, it consists of two radial magnetic bearings and a magnetic thrust bearing. Each radial bearing support the rotor weight as well as controls the two translational (horizontal and vertical) degrees of freedom of the rotor (Fig. 1, inset). While the thrust bearing controls the axial displacement of the shaft. Feedback signals from the rotor position sensors are utilised for stabilising and controlling an open loop unstable five degrees-of-freedom (DOF) rotor-magnetic bearing system. An entire active magnetic bearing system therefore comprises of magnetic bearings, current am-
plifier and compensator with position sensors. Further details about the AMB’s systems working principles and rotordynamics can be found in [7–9].

The AMB is highly complex mechatronic system with several nonlinearities, a detailed exposition on AMB nonlinearities is summarised in a survey paper by Ji et al. [10] and that of Skricka et al. [11]. A number of components presented in Fig. 2 are nonlinear and therefore the AMB system is inherently nonlinear. The following nonlinearities are inherent in any AMB system:

1. Dead band and saturation nonlinearity of power amplifier.
2. Saturation of magnetic core material.
4. Hysteresis of the core material.

These nonlinearities are more pronounced especially for large industrial type machines where large magnetic forces, large currents and small air gaps are needed. To compound the problem AMB system is open loop unstable, therefore some kind of feedback loop is necessary to stabilise the system as well as to meet the desired system performance requirements. Control techniques presently utilised in the industry for AMB systems ignores the nonlinear effects. The simplification is deemed necessary due to intractability of the actual complex nonlinear model as well as due to acceptance of AMBs in industries, necessitating International Standard Organisation (ISO) to formulate its own standards/guidelines on AMB systems. The ISO 14839-3 standard specific to AMBs clearly states the stability margins requirements for magnetic bearings based on linear control theory [12, 13]. AMB systems practitioner therefore inevitably relies on linearization of nonlinearities while designing the control system. However any deviation from the linear regime can lead the system to step into nonlinear operating regime where linear feedback control loop has little or no influence and inevitability loses stabilizing effect. Without going into the details of the linearization strategy adopted for various nonlinearities mentioned above, attention is instead focused on nonlinearities pertinent to the issue of failure of high speed rotor.

2.1. Electromagnetic Forces Nonlinearity

It is a common practice to support the rotor by a pair of opposing magnets. AMBs are said to be operating in differential mode when opposite pair of magnets generates opposing forces in radial directions. The net force generated by a pair of opposite magnets can be represented as

\[
F_{\text{net}} = K_b \left[ \left( \frac{i_1}{s_0 - \Delta x(t)} \right)^2 - \left( \frac{i_2}{s_0 - \Delta x(t)} \right)^2 \right],
\]

(1)
are fairly small compared to nominal air gap change in shaft position from equilibrium, \( i_1 \) and \( i_2 \) are the currents to opposite pair of magnets respectively, refer [8, 14] for further exposition and significance of terms in Eq. (1). From Eq. (1), the net force is a quadratic function of the currents \( i_1 \) and \( i_2 \). This nonlinear relationship is not amenable for linear control design. However, if each of these two currents are decomposed as a fixed bias \( I_b \), variable \( \Delta i_s(t) \) dynamic and static compensation \( i'_0 \) currents respectively, then linearization of Eq. (1) is permissible. The symbol \( \Delta \) indicates deviation from the equilibrium condition

\[
I_1 = I_b + i'_0 + \Delta i_s(t),
\]

\[
i_2 = I_b - i'_0 - \Delta i_s(t).
\]

Equation (1) then becomes

\[
F_{\text{net}} = k_b \left[ \left( \frac{I_i + i'_0 + \Delta i_s(t)}{s_0 - \Delta x(t)} \right)^2 - \left( \frac{I_i - i'_0 - \Delta i_s(t)}{s_0 - \Delta x(t)} \right)^2 \right].
\]

(3)

If the rotor displacement \( \Delta x(t) \) and dynamic currents \( \Delta i_s(t) \) are fairly small compared to nominal air gap \( s_0 \) of the rotor and the bias current \( I_b \) respectively, then the net magnetic force equation (3) is approximately linear. That is, after linearization via the Taylor series expansion of Eq. (3) the actuator force is given by [9]

\[
F_{\text{net}} \approx 4K_b \frac{I_b}{s_0} \Delta i_s(t) + 4K_b \frac{I_b^2 + i'_0^2}{s_0^2} \Delta x(t),
\]

(4)

\[
F_{\text{net}} \approx k_i \Delta i_s(t) + k_s \Delta x(t),
\]

(5)

where

\[
k_i = \left( \frac{\partial F_{\text{net}}}{\partial \Delta i_s} \right)_{\Delta x = 0} = 2K_b \left[ \frac{(I_b + i'_0)}{(s_0)^2} + \frac{(I_b - i'_0)}{(s_0)^2} \right],
\]

(6)

\[
k_s = \left( \frac{\partial F_{\text{net}}}{\partial \Delta x} \right)_{\Delta i_s = 0} = 2K_b \left[ \frac{(I_b + i'_0)^2}{(s_0)^3} + \frac{(I_b - i'_0)^2}{(s_0)^3} \right].
\]

(7)

Note that \( I_b \) is a positive number and the dynamic current \( \Delta i_s(t) \ll I_b \), where \( k_i \) is electromagnetic constant of bearing, \( k_i \) is the force-current factor and \( k_s \) is the force-displacement factor or the (negative) bearing stiffness constant. Net force equation (5) is linear with respect to the control current and the rotor displacement. In this format it is amenable to be used in mathematical model formulation and control synthesis of AMB system. Whereas the force equation (4) is exactly linear provided that the rotor displacement \( \Delta x(t) = 0 \) that is when rotor is suspended perfectly at the centre of bearing, then

\[
F_{\text{net}} = 4K_b \left( \frac{I_b}{s_0} \right) \Delta i_s(t).
\]

(8)

The linearized system is also open loop unstable nonetheless Eqs. (5) and (8) are utilised for synthesizing the controller gains \( K_p, K_d \) of the closed loop system.

### 2.2. Power Amplifier Nonlinearity

The function of the power amplifier is to induce current in the coils of electromagnets. Power amplifier together with electromagnets constitutes a magnetic bearing (see Fig. 1). In AMB system power electronic devices apply voltage to the windings to generate current in the windings. This current is proportional to force required to keep the object suspended as given by Eq. (4). It is assumed that the rate of change of
(slew rate) current should be fast enough to follow the current demand from the controller. In case of zero displacement $\Delta x(t) = 0$, the maximum rate of change of force is bounded by [15]

$$\left| \frac{dF_{\text{net}}}{dt} \right| \leq k_i \left| \frac{di_x}{dt} \right|_{\text{max}},$$

whereas a high frequency control current is required to dampen high frequency flexible rotor bending modes. This requirement in turn places a bound on the rate of change of dynamic current commensurate with the available maximum supply voltage $(V_{\text{max}})$ to power amplifier and winding inductance. The rate of change of dynamic current is thus given by the following relationship:

$$\left| \frac{di_x}{dt} \right| \leq \frac{V_{\text{max}}}{L_w},$$

where $L_w$ is the bearing winding inductance. Thus, the maximum allowable current at any frequency is function of $V_{\text{max}}$ and inductance $L_w$. The maximum available dynamic current in conjunction with the bias current therefore represents the saturation level or maximum current capacity of the amplifier.

The maximum operating current needed for a particular AMB system is decided by the maximum load or unbalanced forces and its frequencies, i.e. Eq. (10). Whereas maximum current slew rate needed is dictated by damping requirement of high frequencies rotor bending modes, i.e. Eq. (9). Breach of either of these two conditions can lead to saturation of power amplifiers rendering opening of the stabilising closed-loop control. As discussed earlier AMB system is an open loop unstable system, opening of the feedback path therefore will lead the system to trip. As a consequence the saturation nonlinearity is critical to developing an understanding of its implication on the overall systems performance. This is considered next.

3. THE DESCRIBING FUNCTION ANALYSIS

While many dynamical systems or plants are linear or can be linearized near the operating region, actuators and sensors associated with the plant are not. Magnetic actuators in particular exhibit nonlinear behaviour as discussed earlier. If nonlinear behaviour is not accounted for, it can degrade the closed loop system performance. Sinusoidal Input Describing Function (SIDF) also referred as Describing Function (DF) is employed to characterise the nonlinear behaviour of power amplifier.

If the input to a nonlinear element is sinusoidal $e(T) = e_m \sin(\omega T)$ then the resultant output $y(t)$ of the nonlinear element is periodic and has the same frequency as that of input sinusoid. The output in addition to fundamental component has higher harmonics. However in DF analysis it is assumed that only the fundamental harmonic component of the output is significant. This assumption is often valid since higher harmonic component in the output of a nonlinear element are generally of smaller amplitude than the fundamental component. As a consequence replacement of the nonlinear element in the control loop by its describing function is justified.

Therefore DF is defined as

$$N(E_m, \omega) = \frac{Y_1}{E_m} \sin(\omega t + \phi_1),$$

where $N(E_m, \omega)$ represents the DF and $E_m$ represents the amplitude of the input sinusoid signal. The amplitude and phase shifts of the fundamental harmonic component of the nonlinear element output signal are $Y_1$ and $\phi_1$ respectively. In general $N$ depends on the amplitude and the frequency of the input signal to the nonlinear element. The amplifier nonlinearity does not involve any energy storage and therefore is merely amplitude dependent, that is $N = N(E_m)$. 

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Analytical equations of standard nonlinear elements can be found in many texts such as that of Dutton et al. [16]. The saturation nonlinearity under consideration is represented by the following DF:

\[ N = k \quad \text{for} \quad E_m \leq E_s \]
\[ N = \frac{2k}{\pi} \left( \sin^{-1} \frac{E_s}{E_m} + \frac{E_s}{E_m} \sqrt{1 - \left( \frac{E_s}{E_m} \right)^2} \right) \quad \text{for} \quad E_m > E_s, \]  

(12)

where \( k \) is the slope of linear part of nonlinear element, \( E_m \) is the amplitude of the input signal, \( E_s \) is the value of saturation and \( N \) the DF of the saturation.

The DF is treated as an element with amplitude and frequency dependent gain and phase changing with input signal level and frequency. Therefore DF is essentially a frequency response function and not a transfer function. If all linear elements of Fig. 2 are collectively represented by \( G(j\omega) = G_c G_p \), then the closed loop frequency response becomes

\[ \frac{X(j\omega)}{R(j\omega)} = \frac{N(E_m)G(j\omega)}{1 + N(E_m)G(j\omega)}, \]  

(13)

\[ 1 + N(E_m)G(j\omega) = 0. \]  

(14)

If the nonlinearity can be sufficiently described by the function \( N \), then the loop gain is simply \( NG(j\omega) \). Thus the study of stability (i.e. prediction of limit cycle) involves estimation of the value of amplitude of the input signal which satisfies

\[ G(j\omega) = -\frac{1}{N(E_m)}. \]  

(15)

4. ANALYSIS OF AN AMB SYSTEM WITH SATURATION NONLINEARITY

In this section, the DF method is utilised to analyse an AMB system with saturation nonlinearity. A first-order AMB rotor equation-of-motion (e.o.m) is presented followed by utilisation of nonlinear theory for analysis of real world trip data. The example presented here is an industrial case study hence only publishable data available is in the form of open loop gain \( L \) obtained via experimentation. Nevertheless for completeness underlying equations are presented albeit in a simplified form.

4.1. AMB Dynamic Model and Suspension Control

The bearing force given by Eq. (5) and assuming linear amplifier model the magnetic bearing can be modelled as a linear device. The symbol \( \Delta \) is used to indicate deviation from the equilibrium condition. The e.o.m in deviation variable form of a single lumped mass \( m \) supported by magnetic bearing using Newton’s law can be written as

\[ m\Delta \ddot{x} = k_i \Delta i_x(t) + k_s \Delta x(t), \]  

(16)

Ignoring the delta term and converting it into a Laplace domain, Eq. (16) becomes

\[ \frac{x(s)}{i_x(s)} = \frac{k_i}{m s^2 + k_s}. \]  

(17)

The position stiffness or the spring constant \( k_s \) is negative hence the pole is on the positive real axis, the AMB system is unstable and therefore a stabilizing feedback loop is required. Additionally closed-loop feedback control is needed to attenuate imbalance induced vibrations as well as to reject disturbance arising from sensor run-out. The proportional-integral-derivative type controllers and its variants have found applications in many systems supported by magnetic bearings [4, 17, 18]. The controller used for magnetic bearing
supported rotor in the current study is the classical proportional-derivative (PD) type controller. This control scheme ensures that the dynamic current in the magnetic bearing coils is adjusted in real-time based on the rotor sensor position.

Assuming $K_p, K_d$ as the proportional and derivative gains of PD controller used to stabilise this system, the control current therefore is

$$i_x(t) = -K_a(K_p x + K_d \dot{x})$$

(18)

where $K_a$ is the power amplifier gain.

Substituting (18) in (16) and also considering the unbalance force term yields

$$m\ddot{x} + (k_i K_a K_d) \dot{x} + (-k_s + k_i K_a K_p) x = f_{un},$$

(19)

where $F_{un}$ is the rotating unbalance disturbance force. Rendering above equations in compact form

$$m\ddot{x} + c\dot{x} + kx = f_{un} \quad \text{where} \quad c = k_i K_a K_d \quad \text{and} \quad k = (-k_s + k_i K_a K_p).$$

(20)

Similarly the e.o.m for a multi degree of freedom elastic rotor-bearing system can be expressed as

$$M\ddot{q} + C\dot{q} + Kq = F_{un},$$

(21)

where $M$, $C$ and $K$ are the mass, external viscous damping and stiffness matrices respectively. $F_{un}$ represents the unbalanced force vector, while vector $q$ contains lateral and angular position displacements of the rotor elements also referred as the generalised coordinates of rotor elements.

### 4.2. System Loop Gain

For a single degree of freedom AMB system described above, open loop gain $L$ in a Laplace domain is given by

$$L = (K_a K_p + K_a K_d s) \times \left( \frac{k_i}{ms^2 - k_s} \right).$$

(22)

Equation (22) represents a simple forward path transfer function comprising of compensator and the plant. However in an industrial grade AMB system, the control system contains PID controller, series of second order filters, amplifier’s transfer function and in some cases notch filters. Since it is difficult to model all the elements of the control system, system loop gain is obtained via experimentation. Analogous to single degree-of-freedom loop gain, the loop gain for multi degree-of-freedom flexible-rotor AMB system can be derived or as often the case in the industry is obtained via experiments as presented in this work. While conducting the experiments, the rotor is statically suspended in magnetic bearings and a swept sine wave with a frequency range of 5 Hz to 10 kHz is injected in one of the axis of magnetic bearing to excite the rotor’s eigen-frequencies. A dynamic spectrum analyser is utilised to inject the excitation signal and receive the corresponding system response. The measured closed-loop frequency response function between the system response or shaft position and the input current measured in terms of voltage. Subsequently, the open-loop gain $L(j\omega) = G_c G_p$, is extracted by transformation of the closed-loop frequency response data for linear and nonlinear stability analysis. This is discussed next.

### 4.3. Analysis of Trip Data

Industrial grade AMB systems are equipped with trip data logging facility to record various system signals in the event of a trip in order to investigate the cause(s) of failure. Typical data that is recorded are positions, currents, angular displacement and speed of rotor. A re-circulating buffer continuously logs various signals. The trip data for dynamic current is shown in Fig. 3 and illustrates the dynamic current signal in one of the axis of the 5-axis active magnetic bearing system. This signal is responsible for compensating imbalance,
Fig. 3. Time domain trip data.

Fig. 4. Variation of the amplitude spectrum of the shaft displacement with rotor spin speed.
disturbances and other forces acting on the high speed rotor. It is clear that at ‘0’ second current has reached its maximum saturation limit of –0.9 Amps and therefore unable to provide any additional level of control signal resulting in opening of the stabilising feedback loop. The system eventually becomes unstable and fails. An estimate of frequency is calculated from a short time window of Fig. 3 just before the failure occurs which revealed an oscillation frequency of 64.5 Hz.

Additional information can be gleaned by converting the trip data in frequency domain. The signal’s harmonic spectrum can be investigated using waterfall plots. A waterfall is a harmonic spectrum in three dimensions with rotor spin rate as the additional dimension. The waterfall plot plays a vital role in rotor dynamics which shows the changes in frequency spectrum with rotation speed. Figure 4 is the waterfall plot of the closed-loop system. It is clear from this plot that the main contribution to the control coil voltage and hence maximum rotor displacement comes from 64 Hz spin rate. Further analysis (Fig. 4) of the same signals reveals a peak around 64 Hz confirming that this indeed the frequency at which the AMB system tripped. Another prominent frequency of 450 Hz in waterfall plot is the one due to machine operating speed of 27000 rpm or 450 Hz.

4.4. Stability Analysis in Gain-Phase Plane

In order to ascertain the cause of failure, theory delineated above is employed. Figure 5 shows the plot of $N$ as a function of input amplitude. Describing function $N$ reflects variation of gain with the amplitude of the input signal to the nonlinear device, this aspect is evident in Fig. 5. Next Nichols plot of AMB system given by Eq. (22) is plotted in Fig. 6 and superimposed on this diagram is the inverse of variable gain ($-N$) in decibels. Note the saturation nonlinearity does not introduce phase shift hence it is amplitude dependent alone and frequency does not play any part. Hence, the $-N$ locus lie entirely on the vertical 180 degree line of gain-phase plane. The Nichols plot is the frequency locus while the inverse DF is amplitude locus. Whether the closed-loop system is stable or unstable depends on the relative position of the two loci. The system descends into limit cycle oscillations when two loci intersects, satisfying Eq. (15).

The two loci of Fig. 6, depict the characteristics of an unstable system as the two loci intersect. The point of intersection predicts at what frequency and amplitude limit cycle will set in. A slight increase in
the amplitude of the input signal to the nonlinear element will result in a loop gain greater than unity and as consequence results in an increase in oscillation amplitude. This will eventually lead to an unstable limit cycle. The predicted frequency from the intersection of the two loci in Fig. 6 is 64 Hz. However, the analyses of the time and frequency domain trip data (Figs. 3 and 4) show an oscillation frequency of 64.5 Hz at the time of failure. This frequency is very close to the one predicted by the DF method.

5. CONCLUSIONS

Several types of algorithms as well as practical insight are required to decipher the cause of failure from the trip data. In this paper an SIDF based approach is presented as a tool for the analysis of AMB trip data when the cause of failure is due to saturation nonlinearity of the power amplifier. AMB failures can be expensive particularly for process industry where downtime of critical equipment such as rotating machine supported by AMBs can lead to huge financial losses. The presented approach not only points to the cause of the failure but also provides a valuable insight to control engineers to re-shape the closed loop system’s loop gain at the predicted limit cycle frequency, thereby improving AMB system’s robustness to failure. Thus the DF analysis can be utilised not only for post-processing of failure data but also for the control design stage. Utilisation of presented approach will yield a robust control design and is likely to eliminate failures due to amplifier saturation or other nonlinearities if accounted for in DF analysis.

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