AN ELIMINATION OF RESONANCE IN ELECTRIC DRIVES

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Abstract: *Flexible couplings together with resonance phenomenon are present mainly in the field of servodrives where high accuracy and dynamic requirements are crucial. When dynamics doesn't correlate with mechanical system design, unwanted frequencies in the system are exited. Sometimes we haven't conditions (whether material or space) to design mechanical system with resonant frequencies too high to be exited. In that case we must choose compensating methods which can eliminate these phenomenons. This paper is dedicated to them.*

Keywords

Servodrive, flexible coupling, filtration, observed, state control.

1. Introduction

A systematic production increase, frequent production changes and infiltration of machine production to new "territories" cause not only higher demand for controlled drives but also unceasing demand for increase of quality and production rate. The latest relate to improvement of mechanical construction quality and also entire electric drives by which machine tools are driven. In the field of electric drives it means increase of dynamic and accuracy but this increase can cause number of related problems. One of them is elimination of flexible coupling [1] between system components. Unfortunately, in many cases, there isn't only one problematic coupling but often a chain of flexible couplings which significantly influence the quality of product or production. The paper brings review of elimination procedures.

2. Elimination Possibilities

There are a lot of possibilities to cope with flexible couplings phenomenon, but only the most widely used methods are mentioned in this paper. As it was mentioned in [1], besides the change of mechanical

parameters the elimination can be provided also by proper control structure.

A bandwidth reduction (by means of gain reduction) can cause resonant frequencies will not be to exited, but on the other hand it cannot meet servodrive requirements, e.g. disturbance rejection capabilities. All depends on application and desired bandwidth. The welding robot with minimum mechanical damping has different behavior than machine tool- milling machine. In both cases there are different tuning rules. For example in the first case we have to choose lower gains because there aren't sources of mechanical damping to absorb oscillations.

The simplest way of suppressing of oscillations is to eliminate the cause of their generation, but now we are not talking about mechanical parameters but about excitation signal. This signal can cause "resonance excitation". In [1] resonance was caused by step change of control signal. This can be solved by means of utilization of signal with step change of acceleration or jerk and not position. The reference signal can be created with the help of limit values of jerk, acceleration, velocity and position. This type of reference is often used together with feedforward direct branches, where position and its derivations are references for all direct branches. But this is only the first condition. The most important parameter is jerk time (jolt time), defined by acceleration and jerk limit values. If the jerk time is too short, characteristic frequencies of mechanical system can be excited. As pointed out in [2], characteristic frequency of numerator of two mass system transfer function in [1], i.e. antiresonant frequency, is again critical. The lower the jerk time in comparison with inverted natural frequency, i.e. $t_{jerk} < \dfrac{2\pi}{\omega_{nzero}}$, the more significant are the oscillations. Above this value the overshoot is negligible. This is not exactly true in the systems where acceleration change quickly or in case of short distance positioning, when acceleration has not enough time to reach limit value and jerk time is shortened.

In general one can say that to excite resonant frequencies we need to generate reference excitation signal (or vector of kinematic variables), whose frequency spectrum doesn't contain natural frequencies

of mechanical system. When this condition is fulfilled the reference can be shaped arbitrarily.

Note: The feedforward based only on system equations without flexible couplings cannot deal with this problem. Gain design in the direct branches must respect flexible mechanical system.

A special treatment, not yet used in commercial drives, is based on 4th order feedforward using, where the additional direct path of jerk derivation is utilized [3]. When the system is identified well, there is a chance of elimination of those parts of characteristic equation which contains mechanical parameters and thus to avoid resonance. A drawback of this method, alongside the increase of complexity, is a necessity of accurate parameters identification.

3. Filtration

The most widely used way how to deal with drawbacks caused by flexible couplings is a filter utilization from the simplest form of low-pass filter through band stop to biquad filters.

The simplest is low-pass filter which attenuate frequencies above its characteristic frequency which is the only adjustable parameter. In our case, filter is used to attenuate amplitude around the resonance frequency as Fig. 1 shows. To augment final effect we can use a chain of these filters with the same parameters. One of the drawbacks of this approach is a phase lag injected by filter which results in phase margin reduction (see Fig. 1). To influence more resonance than system responsiveness (bandwidth), this filter should be used to damp higher resonant frequencies.

To avoid phase lag injection lead-lag filter can be utilized which can bring a similar result regarding resonant peak damping but on the other side the change of phase is less aggressive (the measure of phase change depends on ratio of characteristic frequencies of transfer functions numerator and denominator (1)). As it was said, two adjusting parameters (characteristic frequencies $\omega_{0,zero}$, $\omega_{0,pole}$), defining pole and zero placement, brings higher flexibility of adjusting.

$$G(s)_{lead=lag} = \frac{s + \omega_{0,zeroa}}{s + \omega_{0,pole}} . \qquad (1)$$

In this case they adjust a band, in which amplitude decrease (filter has integrating nature). All can be seen on Fig. 1, where amplitude falls as low as in case of low-pass filter utilization, but system is characterized by higher phase margin.

$$G(s)_{bandstop} = \frac{s^2 + 2\xi_{zero}\omega_{0,zero} + \omega_{0,zero}^2}{s^2 + 2\xi_{pole}\omega_{0,pole} + \omega_{0,pole}^2} . \qquad (2)$$

Characteristic frequency defines filters activity frequency range and damping ratios defines sharpness

and depth of amplitude fall. The damping ratio of zeros defines depth (in some cases it can reach zero-the maximum depth) and damping ratio of poles defines sharpness of fall. The goal of filter tuning is primarily elimination of poles, which are located near imaginary axis (with low damping ratio resulting in resonance occurrence) and thus to increase system stability. Filter poles location depends on specific application and response demand.
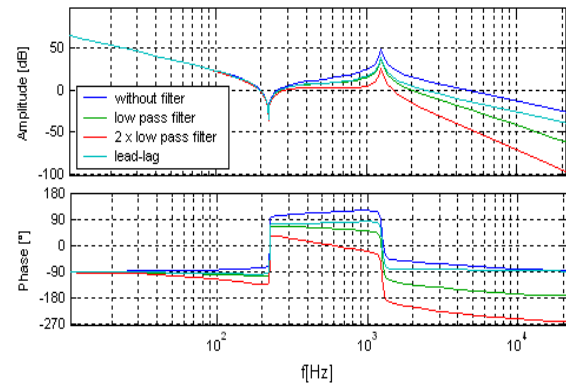

Fig. 1: Open loop bode plots of velocity controlled servodrive equipped by different types of filters.

An example of negation of mechanical system poles or more precisely negation of resonant peak caused by resonance frequency by means of bandstop filter utilization shows Fig. 2. We can see, that well tuned filter can fully remove the resonant peak. The filter, evidently, influence amplitude only in the defined band and at the same time phase margin fall is not significant. Effect of filter impact can be seen also in time domain on Fig. 3, which shows effort of elimination of oscillations caused by antiresonant frequency.

All aforementioned filtration types can be realize by means of biquadratic filter using. One of the possible expressions of this filter is similar to bandstop filter (2), but characteristic frequencies of numerator and denominator can be adjust separately (ω_0, zerpa≠ω_0, pole). As it has a similar properties to mentioned filters, the reader will be referred to aforementioned information. In addition, this filter can be adjusted as inverse transfer function of mechanical system ((3) in [1]) and thereby to filter the both antiresonant and resonant frequencies.

The filters bring also few drawbacks. One of them is related to current limiter in cascade control structure. If the current reference is higher than motor or inverter limit, the reference is set to this value and has clear DC character. Any type of filter is in this case ineffective and resonance occurrence is more probable. This situation can be avoid by filtration of feedback variable instead of direct path signal. The result is smooth current reference and lower chance of current limit reaching.

4. The Other Possibilities of Resonance Curing

Another way of curing of resonance is utilization of observer. Observer, besides phase margin improving, increase also robustness against resonance and in addition provides signal of (practically) immeasurable variables such as load torque. The major advantage is that observer can substitute dual system of position measuring, this means that we cannot use load position sensor.
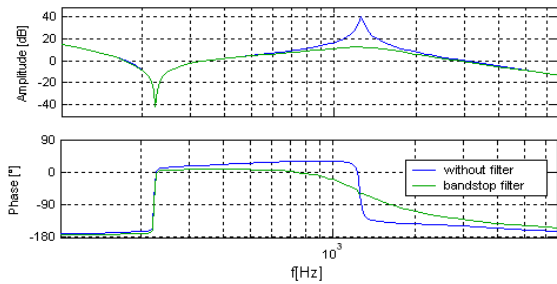

Fig. 2: Open loop bode plots of velocity controlled servodrive with and without band stop filter.
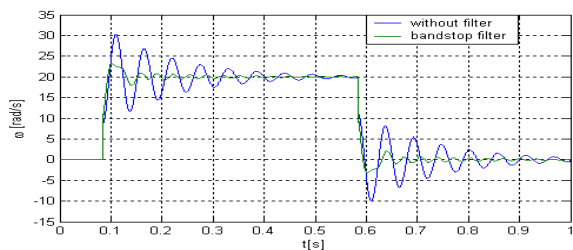

Fig. 3: Open loop bode plots of velocity controlled servodrive with and without band stop filter.

The observer can be realize as a classical Luenberger observer, observer based on artificial intelligence, sliding mode observer or so called ESO observer [4]. The most widely used is Luenberger observer. In a system without flexible couplings has observer same structure as observed system with inputs of current and velocity or position [5]. This type of observer can be utilized (under certain condition) also in systems with flexible couplings. It is clear, that observer has a filtering nature, which can be used in velocity signal detection (which is why it is sometimes called filtration observer). If gain crossover frequency corresponds to resonance frequency, effect of observer is minimal. In the situation, when bandwidth is out of resonant frequencies, observer filtrate system oscillation which aren't caused by flexible couplings. If resonance is located in the bandwidth, observer brings, (through observed load torque signal [6]), such amount of damping (this benefit is also in rigid systems). When the mechanical system parameters are known, there is a possibility of observer extension with mechanical system (with flexible couplings) model to prevent resonance to occur. One of the possible observer structures is shown on Fig. 4.

Despite of complexity of this scheme, it provides alternative to dual sensor scheme of servodrive with additional advantages in significant increase of system damping. This scheme can be simplified by neglecting of damping coefficient, which usually reaches only a small values and moreover its identification is complicate.

The last approach of resonance elimination is based on state control which (in many cases) utilize mentioned observer. One the most typical state control schemes shows Fig. 5. The observer (not shown) provides state variables for feedback vectors. One of the well known drawbacks of state control is necessity of system parameters adjusting in one step which can cause, in case of incorrect setting, damages. The main advantage of state control related to almost unlimited freedom in pole placement which eliminates resonance frequencies. So, even lightly damped system can be controlled without compromises, according to application needs with limits in system components delay (inverter, sampling, filters, etc.) and motor and inverter capabilities.

Fig. 4: Servodrive with Luenberger observer based on system with flexible coupling element.

Fig. 5: Block diagram of state control of position controlled drive with flexible coupling where state variable are provided by state observer (Fig. 4).

It is evident that success in filter as well as observer tuning depends on mechanical system parameters knowledge. Incorrect adjusting makes situation even worse. But in numerous applications parameter identification isn't easy and designer has no choice only to tune it with trial and error method. Another problem relates to variation parameters during operation when system should to react to these changes. These changes are moment of inertia (due to the technological procedure or production changes) or damping coefficient variation (due to the temperature fluctuation or aging). In both cases, for tuning and on-line retuning needs adaptive structure utilization (such as in [7]) is required. This approach is based on comparison of energies of signals which pass through two parallel band pass filters tuned to frequencies below and above expected resonance frequencies. The result tells us whether there is a need of filter readjusting or no. There are also methods based on frequency analysis of error signal by means of fast Fourier transformation [8].

5. Conclusion

There are a lot of problems to solve when designing a control of servodrive with flexible couplings and range of this paper can't cover all aspects of this phenomenon. The aim was to describe basic fundaments and to indicate approaches the most widely used. There are a lot of unmentioned approaches but these are used mainly in

laboratories and not in real applications. Among them are methods based on mathematical optimization, such as H_∞ or H_2 control (LQG). Application of these techniques requires good knowledge of mathematic, which can cause problems in implementation in real applications. This assumption is confirm by the fact that these techniques are known five decades.

Acknowledgments

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