Mechatronic Optimization, Analysis and Simulation of Machines

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摘要

工具機或產業機械可以透過全部的次系統(subsystem)進行有限元素法最佳化分析，單獨利用控制相關的參數量測無法滿足最佳化分析的需求，因此我們談的先進技術是工具機械整機以建立FEM為途徑來分析模型，如此一來每個單獨機械元件亦可進行從動態的最佳化分析。本文引用兩個範例說明如何成功建立FEM分析模型的步驟與方法來進行最佳化分析。

In this article it is shown that a machine tool or production machine is a mechatronic system which can be optimized by involving all of the subsystems. Control-related measures alone are not sufficient, as is clearly shown in the second chapter. It then follows that a machine should be modeled so that the mechanical part of the machine can also be optimized. Two application examples indicate how the complete system can be successfully modeled. The limits of control-related measures are indicated and mechanical design steps are explained in order to address the causes of the disturbing machine properties.

INTRODUCTION

Generally, various companies are involved in the development of important parts of a machine: The machinery construction OEM – these are often medium-sized companies –have overall system responsibility for the mechanical sub- system and the automation components are provided by companies who offer motors, control systems and drives. The A&D MC business division of Siemens AG, with its

portfolio of state-of-the-art automation equipment, is not only in a position to offer intelligent control systems in conjunction with high dynamic performance motors and drives, but it can also support machinery construction OEMs when they are mechanically designing a machine. This is achieved by analyzing and optimizing the complete machine in a focused fashion.

This means that manufacturers of machine tools and production machines can use the mechatronic services in the form of machine optimization, analysis and simulation. This article describes the basics of machine simulation. The contents of the article focus on the area of suitable modeling of the complete mechatronic system and less on the detailed description of the individual subsystems.

Every machine tool and production machine is designed so that specific motion is impressed on a mechanical structure. This motion should be as fast as possible with the adequate precision. The second section will show that specific characteristics of the mechanical system to be moved limit the velocity which can be achieved with the specified accuracy.

The third section will show how a mechanical subsystem is modeled using discrete elements. And the fourth section will explain which systems require modeling with finite elements. The procedure which is applied to generate a model is then discussed using examples from industry.

MECHANICAL LIMITS - ACHIEVABLE MACHINE DYNAMIC RESPONSE

The ability to impress the re- quired movement on a mechanical structure is often restricted by the lowest natural frequencies of the mechanical structure. This can be clearly illustrated using a simple system: A load, with mass mL is shown in Fig.1 which is to be moved by a mo- tor with mass m_{Mot}. The coupling between the motor and load is established using a spring with stiffness c and a damping element with damping

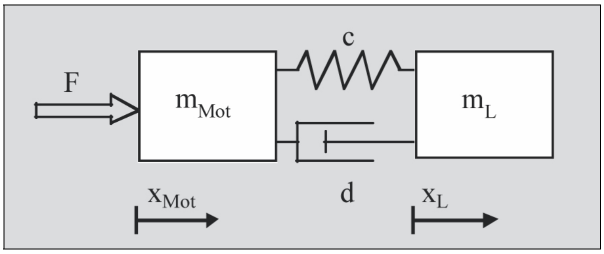

Fig. 1 Simple arrangement: A load (mass m)L moved by a motor (mass m_{Mot})

characteristic d. The goal of a closed-loop control with this configuration would be to define the force F as a function over time of the measured quantities, so that the position of the load follows, as precisely as possible, a specified reference characteristic.

The Laplace transformation of $G_{XLXM}(s)$ with the motor position, $X_{Mot}(s)$ as input quantity and the position of the load X_{Load} (s) as output quantity is given by:

$$G_{XLXM}(s) = \frac{X_L(s)}{X_{Mot}(s)} = \frac{2 \cdot D \cdot T \cdot s + 1}{T^2 \cdot s^2 + 2 \cdot D \cdot T \cdot s + 1} \; mit \quad T = \sqrt{m_L / c}, \quad D = \frac{d}{2\sqrt{c \cdot m_L}}, \quad (1)$$

The amplitude characteristic from the associated frequency response characteristic $G_{XLXM}(f) = |X(j2\pi f)/X_{Mot}(j2\pi f)|$ can be seen in Fig.2. In this case, the parameters are assumed as follows：

$m_L = m_{Mot} = 1$kg, c = 3948 N/m,d=77s/m.

In the vicinity of the resonant frequency$f_{res} = 1/(2\pi) \cdot \sqrt{(c/m_L)} = 10$ Hz, at which frequency the load oscillates with respect to the motor, a significant increase in the amplitude characteristic can be identified. When the motor moves at frequencies exceeding 10 Hz, then with increasing frequency these act less and less on the load. This can be clearly seen with the negative gradient of the amplitude characteristic with 40 dB/decade. At 33.5 Hz, a value of $G_{XLXM}(33.5\ Hz) = -20$dB can be read-off from the characteristic in Fig.2. When the motor moves with a frequency of 33.5 Hz, in a