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Adaptive tuned mass damper with variable mass for chatter avoidance

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ABSTRACT

If the machine tool damping is not sufficient to reduce the vibration induced by machining, chatter can occur. This paper presents an innovative adaptive tuned mass damper (ATMD) with variable mass. The variation of the mass of the ATMD allows the adaption of its eigenfrequency to the dominant one of the machine tool, which is dependent on the axis position of the machine. The design of the ATMD and consequently its working range was optimized with a genetic algorithm. The functionality and performance of the ATMD were then validated under real cutting conditions on a machining centre.

1. Introduction

The interaction between the cutting forces and the resilience response of a machine tool is the limiting factor of its performance and its precision. If the machine damping is not sufficient, the machine starts vibrating and chatter can occur. Vibrations do not only limit the performance of the machine tool but are the cause of numerous problems such as tool breakage, increased material wear or poor surface quality [1–3].

In the last decades researchers and industry dedicated a lot of effort to improve the machining stability of machine tools. While some of them improved the machining process with spindle speed variation, new tool geometry, etc., others improved the dynamic behaviour of the machine tool with additional systems or by changing the structure of the machine tool.

The approach we present here is to improve the dynamic behaviour of the machine tool deals with an adaptive tuned mass damper (ATMD) with variable mass. Since the dominant eigenfrequency of the machine tool varies with the working position of the machine, the adaption of the mass allows the adjustment of the ATMD to this eigenfrequency in order to reduce the chances of chatter occurring.

2. State of the art

Given that the approach presented in this paper focuses on improving the dynamics of machine tools, so will the following account of the state of the art. The improvement of the machine tool dynamics with additional systems can be categorized in three classes:

- passive systems without any input of energy in the vibrating system,
- semi-active or semi-passive systems, with passive properties that can be changed by energy input,
- active systems with input of energy to generate forces [4].

Passive systems are dimensioned to improve the dynamic behaviour for a given eigenfrequency. The introduction of an additional passive system leads to the improvement of the frequency response function (FRF) of the machine tool (vibrating system) at the aimed eigenfrequency but also to a new eigenfrequency resulting in two new increases in amplitude of the FRF. Tuned mass dampers (TMDs) are the most commonly used passive systems. A combination of several TMDs can also be used. [5] presented an approach where the use of four TMDs fixed on the workpiece permits a reduction of vibration amplitude of 98% and [6] showed that the use of multiple identical TMDs gives better results that one larger TMD. [7] presented an approach for the dimensioning of multiple TMD (MTMD) and showed that the effectiveness of an optimally designed system is higher than the one of a single TMD having the same mass ratio.

Semi-active or semi-passive systems are passive systems whose properties can be changed with the input of energy. The changed properties are mainly the stiffness and the damping of the system. [8] presents an approach where the stiffness and therefore the eigenfrequency of a boring bar is changed by the use of a magnetorheological fluid. [9] presented an approach where an electrorheological fluid is used in a linear guide to increase the dynamic stiffness in the direction of the table feed, leading to the suppression of chatter vibrations. Furthermore [10] presented an approach to adapt the stiffness of an adaptive-passive absorber using shape-memory alloys. The adaption of the eigenfrequency is obtained by changing the temperature of the springs made of shape-memory alloys. [11] presented an approach to change the eigenfrequency of a machine tool carriage by adapting the mass of the carriage leading to the reduction of the vibrations by up to 45%.

Active systems are actuator-based systems with an integrated control loop. Usually piezo actuators are used to avoid chatter occurrence by compensating the vibrations with the actuator

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displacement, for instance in boring bars [12], spindles [13], workpiece clamping systems [14] or the strut of a parallel kinematics machine tool [15].

For chatter avoidance, a tuning objective is defined to calculate the mass, the stiffness and the damping of a TMD. The tuning objectives consist into minimizing the peak magnitude of the FRF or maximizing the most negative real part of the FRF [1,3].

3. Approach

The interaction between the tool and the workpiece limits the improvement of the performance and the precision of a machine tool. This interaction depends on the FRF of the machine tool and on the cutting process. The FRF of the machine tool changes with the axis-position of the machine tool [1,3].

In this way, the dynamic behaviour of a machine tool without any adaptive or active additional system cannot be optimal for all machining situations at all machining positions.

The approach presented here to avoid chatter occurrence is an ATMD with variable mass. The ATMD (semi-passive additional system) permits the compensation of the critical vibrations of the machine tool (vibrating system). When the working position of the machine tool changes, so does the value of the eigenfrequency we want to compensate. The ATMD is then adapted to compensate this eigenfrequency by changing its mass. In our approach, the ATMD is a hollow shape. The change of mass is therefore achieved through the filling of the inner space with fluid, for instance cooling lubricant. The slushing mechanism is for instance in buildings already known for suppressing wind-induced vibrations [16]. In our case it is not this mechanism that is used to avoid chatter but the mass of the fluid, with the tuning objective of minimizing the peak magnitude of the FRF having a negative real part.

4. Design of the ATMD

For a given machine and cutting operation, one can calculate the machining stability; the result of the calculation is a stability lobe diagram, made of multiple lobes. The dominant eigenfrequency defines in the diagram the lobes and the critical depth of cut [3,17]. The aim of the ATMD is to compensate this dominant eigenfrequency in order to increase the critical depth of cut. Nevertheless, the dominant eigenfrequency will vary with the working position of the machine. Thus, the possible mass variation of the ATMD (defined by its design) has to take into consideration the variation range of the dominant eigenfrequency.

One can design a traditional TMD based on the mass and the frequency of the vibrating system. [18] describes the design procedure. In our case, the vibrating system has an eigenfrequency that varies; therefore, the ATMD has to adapt to compensate the vibrations of the system. Our approach is the adaption of the ATMD through the variation of its mass. The ATMD is consequently set up to compensate one eigenfrequency that can vary in a variation range. The lower mass of the ATMD (when the ATMD is empty) has to permit the compensation of the highest frequency (for instance at working position 2 in Fig. 1) whereas the higher mass compensates the lowest frequency (at working position 1 in Fig. 1). Thus, we defined a design procedure for our ATMD, see Fig. 1.

The first step of the design procedure permits the determination of empty mass and the full mass of the ATMD. We first define an approximate empty mass of the ATMD to calculate the approximate stiffness of the ATMD. We then define the stiffness of the ATMD by combining die springs available on the market in parallel, in order to approach the approximated stiffness. Finally, we calculate the empty mass and the full mass of the ATMD.

The second step of the procedure allows to determine the dimensions of the ATMD in order to obtain the aimed empty and full mass. The full mass of the ATMD is equal to the sum of the empty mass and the mass of the filled inner space of the ATMD. The shape of the ATMD defines the empty mass of the ATMD and the

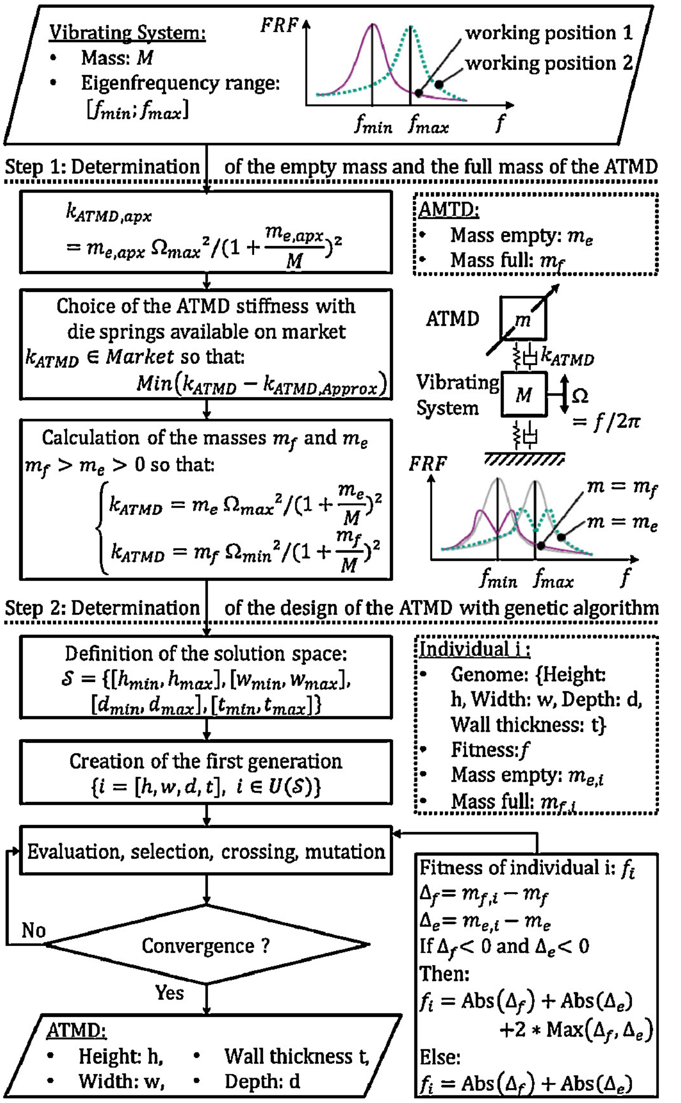

Fig. 1. Design procedure of the ATMD.

volume – and therefore the mass – of the inner space. Consequently, the design optimization of the ATMD consists in finding the optimal shape with the desired empty mass and the desired inner volume. We used a genetic algorithm in the second step of this procedure to solve this optimization problem. The height, width, depth and the wall thickness of the ATMD define the genome of an individual. The allowed variation of these parameters defines the solution space. The algorithm first generates a first population generation of individuals in the solution space. The algorithm then evaluates the population: the sum of the differences between the individual's empty and full mass compared to the aimed empty and full mass defines the fitness of an individual. If the empty and the full mass of the solution are equally smaller than the aimed empty and full masses, the solution is also suitable by adding the missing mass to the external space of the ATMD. This case is also considered in the calculation of the fitness value. After selection, crossing and mutation, a new generation of individual arises. This process loops until the algorithm converges. Finally, the algorithm returns the fittest individual, corresponding to the most suitable solution.

5. Results

5.1. Presentation of the machine tool considered

The machine tool we used for investigating our approach of ATMD with variable mass is a LiFLEX II 766 i B2 5 axes machining