

Predicted Machining Dynamics for Powertrain Machining

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ABSTRACT

Owing to decreased development cycle timing, designing components for manufacturability has never been as important. Assessing manufacturing feasibility has therefore become an increasingly important part of new product engineering. This manufacturing feasibility is conventionally assessed based on static stiffness of components and fixture assemblies. However, in many operations, excess vibration represents the actual limitation on processing a workpiece. Limits on how far into components a tool can reach or the amount of processing time required to machine a feature is commonly decreased significantly due to vibration. Critical time is spent resolving these vibration problems during product launches. Depending on the machining configurations these vibrations can be due to the part & work support structure or due to the tooling & spindle assembly. This paper presents approaches for predicting the dynamic flexibility for either of these assemblies using purely analytical approaches. Results from experiential modal testing are subsequently used to validate the approaches shown. Also by predicting the frequency content of the cutting forces, the risk of resonance or harmonic resonance during machining can also be assessed. Combined, these methodologies represent a comprehensive approach to mitigating the risk of vibration issues during a machining launch using up front simulation during the part development and manufacturing process.

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1. INTRODUCTION

The periodic nature of cutting forces often results in severe vibration, which severely limit the metal removal rates. These vibrations can be categorized into three different types, free vibrations, forced vibrations and self-excited vibrations [1]. In powertrain applications free vibrations rarely cause a problem, however both forced vibrations and self-excited vibrations are common problems.

Typically a forced vibration problem occurs when the periodic forces from a discontinuous machining process coincide with a dynamically weak portion of the workpiece. The exciting force exceeds the dynamic stiffness of the workpiece and severe vibration occurs. On the other hand, self-excited vibrations occur most commonly when waviness from a previous pass interact to form a positive feedback loop with subsequent passes in a mechanism called regeneration of waviness or more commonly just regenerative chatter.

There is a substantial body of work that has been presented by researchers in quantifying and controlling machining vibrations, particularly regenerative chatter problems. These techniques predominantly focus on experimental modal analysis for system identification, and the development of stability lobes to map the stable regions based on the system identification. A review of the state of the art for detecting, identifying, avoiding or preventing chatter was published by Quintana and Ciurana [1]. Traditional approaches are described along with the recent trend in the online monitoring, measuring and control of the machining process.

Outside the academic setting, there are well established industrial solutions specifically aimed at system identification and control of vibration problems in cutting. MLI's TXF and MAL's CutPro are both fully mature systems that can be used in industrial settings to analyze and optimize vibration limited cutting [2, 3]. Other than the system identification approach, sound measurement can be used to identify forced vibration and chatter problems, as well as to guide the user to a stable cutting region based on the chatter frequencies experienced [4].

Thus, solutions for vibrations on the shop floor are common but this involves waiting until a vibration problem develops. Trouble shooting and optimizing during launch is not only expensive, but can lead to program delays. This reactive approach is also responsible for late breaking changes in cycle times as processes are tuned during launch. For any manufacturing organization it is therefore important to be involved in moving more analysis upfront, aimed at reducing the time spent solving vibration issues during launch as well as to optimize machining rates with an eye to increasing investment efficiency.

Unfortunately only heuristic guidelines have been used in the past to try avoiding vibration problems during the process planning phase, without significant engineering analysis. This paper presents the use of an up-front physics based approach to mitigate machining induced vibrations problems. The approach presented is separated into three sections, predicting the dynamic flexibility of the workpiece, predicting the tool point frequency response function and using time domain simulations to establish the excited frequencies for forced

vibration problems. Combining these three approaches enables the risk of severe vibration during machining to be assessed early in the product design cycle.

2. PREDICTING PART DYNAMIC FLEXIBILITY

When vibration problems occur during machining the phenomenon is driven either by the workpiece or the part. Commonly at low rotational speeds the problem is said to be part driven and at higher speeds the problem is due to the tooling. This is neither always true, nor helpful in early process design. Instead, a better approach is to investigate the dynamic flexibility of both the part and the tool. If fixturing and parts are already available, this is simply achieved by impact testing of both the tool and workpiece structure. The challenge arises when physical parts are not available yet. This section will focus on using finite element analysis to analytically find the frequency dependent flexibility (dynamic flexibility) of powertrain components before parts and fixtures are available. The subsequent section will focus on finding this same flexibility for the corresponding tooling.

2.1. Fundamental Assumptions

As is typical with most finite element modeling, the physical problem of predicting the dynamics of a workpiece is extremely complicated and fraught with interactions. In order to cast the problem to a manageable level of complexity a number of simplifying assumptions are made, which restrict the scope of the solutions obtained. With regards to vibrations during metal cutting this mostly revolves around deciding how much of the machine tool needs to be modeled.

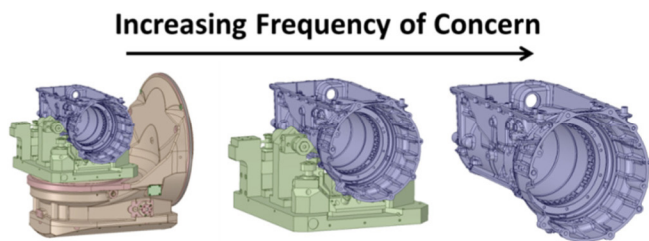


Figure 1. Various levels of M/C are modeled based on minimum frequency of concern.

Low frequency modes, that involve the entire machine tool, are typically stiffer than the tooling. Also for most powertrain machining cutting speeds are quite high. This all means that problems with low frequency vibration dominated by the machined part are quite rare. When the part dominates the machining dynamics it is usually a local mode that exhibits at relatively high frequency. This means that it is often only the part, or the part and fixture, which need to be modeled. The trunion can mostly be treated as rigid. Three different such configurations are shown in Figure 1. The decision whether the fixture needs to be considered is part dependent, and has to be evaluated for each component. For example, when considering the engine joint face for a transmission case, the fixture compliance is insignificant. However if considering the machining of spool bores on a valve body, the mass and stiffness of the fixture plays a significant role in the frequency ranges of interest.

A further complication in machining analysis can be that the structure being simulated changes as it is being machined. This requires a model that adjust to the evolving geometry [5], however since most powertrain components are cast to near net shape there is very little change in part flexibility in a single machining pass. This means that the part geometry can be treated as time invariant for most powertrain operations, which greatly reduces model complexity.

2.2. FEA Model

Many cutting induced vibration problems occur at frequencies greater than 1000Hz. This means that it is important to capture not just the first few structural modes, but the high frequency modes driven by local structure as well. This means finer meshes are required in the vicinity of points of interest. For thin walled components second order elements further improve the accuracy of locally dominated modes. As with all finite element, the mesh density and quality has a significant effect on results. This means, the higher the frequency of interest the better the local mesh needs to be.

To find the dynamic flexibility over a wide frequency range, computational expense can become a problem. Using a modal frequency solution has some numerical advantage in this instance. In addition, using modal damping allows for easier tuning of damping values based on experimental validation.

Load and output points are best chosen to match possible experimental test locations. This allows results to be easily compared with experimental values obtained once physical components are available. A transmission case is used as an example to demonstrate this process. The specific goal of this example is to evaluate the engine joint face of a rear wheel drive (RWD) transmission case for risk of vibration at a number of locations, whereby only the most flexible location is shown. The excitation location and response measurement location are indicated in Figure 2. The excitation was in the radial direction compared to the main barrel, a 1N amplitude was used and swept from 1Hz to 8000Hz. Modal damping was used at 1% over the entire frequency range, and all modes up to 18000Hz were considered.

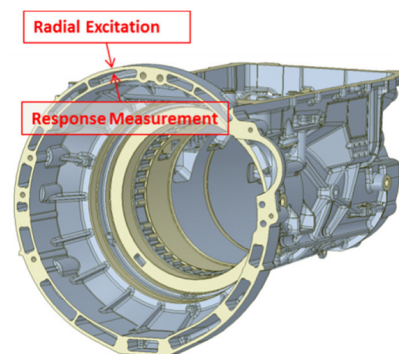


Figure 2. Excitation Location & Response Location for Transmitted FRF

The frequency step was 10Hz in an initial sweep and the step size was decreased in the vicinity of narrow modes in order to ensure a minimum of 5 values in any half amplitude bandwidth. The frequency response function (FRF) for the location indicated in Figure 2 is shown below in Figure 3. The cutting frequencies, based

on the cutting parameters are indicated in green. This is discussed in greater detail in section 4; the focus of the present section is simply predicting the dynamic flexibility.

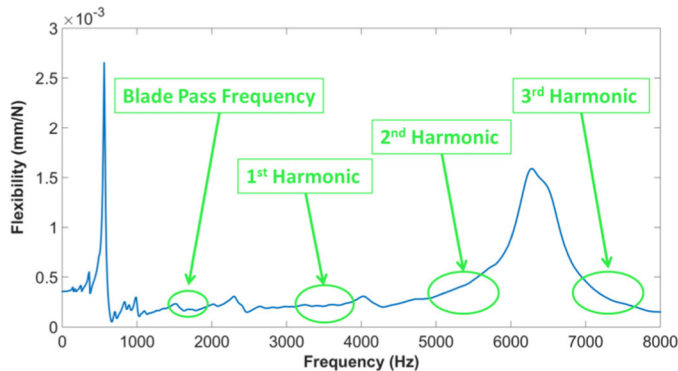


Figure 3. FRF for Transmission Case Front Face two clamping configurations.

In order to validate this model, experimental modal analysis was performed on this case on the plant floor. An impact hammer was used to excite the case at the excitation location and an accelerometer was mounted in the response location. The frequency response function for the deflection (dynamic flexibility) was then calculated from these impact tests and is compared to simulation in Figure 4.

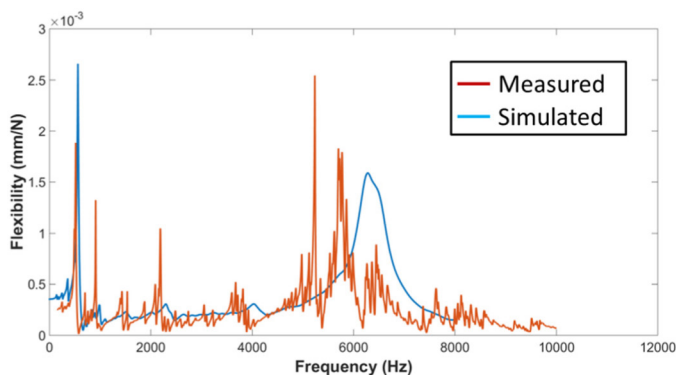


Figure 4. Transmission Case Roughing Front Face FRF Simulated and Measured.

The comparison between the experimentally measured response and those simulated is not perfect, but there is a strong correlation, particularly for frequencies greater than 500 Hz. Given the frequency range of interest, the result is well suitable. The low frequency modes, which are not captured well, are modes associated with the machine tool base and slides. Since these are not modelled at all, they are not expected to be captured. If frequencies below 500 Hz were of concern, then more of the machine tool would need to be simulated. This is however rarely necessary. Starting at 500 Hz the modes are predominantly formed by the part itself, and thus more accurately captured.

Given the uncertainties of damping coefficients for these models it is best to validate the model setup on an existing product first, and only then model the new product for comparison. This ensures sufficient confidence in the model to draw usable conclusions from the analysis.

The frequency response of the workpiece is however only part of the equation, the other half is the frequency response of the tooling, which is covered in the next section.

3. TOOL DYNAMIC FLEXIBILITY PREDICTION

It is more challenging to predict the frequency response of tooling than that of the workpiece. This is because spindle modes are much more likely to be in the frequency range of interest, which means that the machine tool spindle cannot be ignored. Building an accurate model of the spindle is time intensive, fraught with inaccuracy and requires information about the spindle that is only available to the machine tool supplier.

Principally, spindle dynamic prediction is performed using 2D Timoshenko or Euler-Bernoulli beam analysis [6]. However full 3D models for the spindle and slides can also be assembled to capture more complex modes [7]. But this requires in-depth knowledge about the design of the spindle, information that is not typically available.

Substructure analysis presents an alternative which allows for different portions of the tool and spindle assembly to be studied separately and combined into a total response [6, 8, 9, 10]. Receptance coupling substructure analysis (RCSA) is the analytical combination of the receptances of each sub-structure of a whole system. This allows the separate treatment of the machine tool spindle and tool assembly. More precisely it allows the spindle behavior to be measured and the tool assembly to be simulated, and the combined response simulated. This is shown schematically in Figure 5 [8, 9, 10].

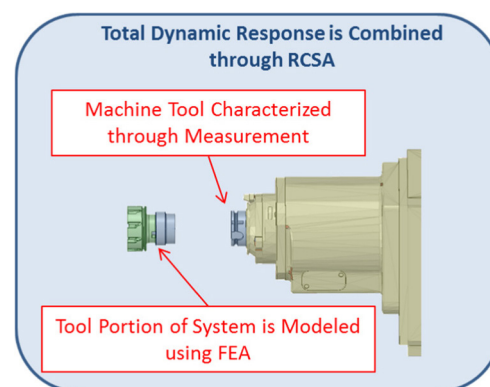


Figure 5. Receptance Coupling Schematic

In an automotive company, where a limited variety of machine types are purchased, it is straight forward to develop a catalogue of standard spindle behaviors. This can then be combined with simulated cutting tools to model the tool-spindle assemblies early in the product design cycle. This approach is relatively mature and supported in the commercial MLI's TXF [2] commercial package, which is being used at Ford Motor for these purposes.