SIMULATION-DRIVEN DESIGN OF COMPLEX MECHANICAL AND MECHATRONIC SYSTEMS

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Karlskrona, December 2006,

Johan Wall

"*Pluralitas non est ponenda sine necessitate*" or "*Entities are not to be multiplied beyond necessity*" [1].

Ockham's (or Occam's) razor is a principle attributed to the 14th century Franciscan friar William of Ockham. When faced with multiple competing theories of equal predictive power, the principle recommends embracing the one introducing the fewest assumptions and hypothetical entities. It is in this sense that Ockham's razor is usually understood today.

Abstract

Effective and efficient product development is critical to business success on the increasingly competitive global market, and simulation has proven to support this in many sectors. The aim of this thesis is to study how properties of complex mechanical and mechatronic systems can be more efficiently and systematically predicted, described, assessed and improved in product development. The purpose is to elaborate an approach that can, rather than only verifying solutions that are already decided upon, support dialogues with customers, stimulate creation of new concepts and provide guidance towards more optimised designs, especially in early development stages. This is here termed simulation-driven design.

To be useful for this, product models and simulation and optimisation procedures must be efficient, that is, they must accurately answer posed questions and point towards better solutions while consuming an acceptable amount of time and other resources. In this thesis a coordinated approach to create such efficient decision support is elaborated. This is done by action research through two industrial case studies; an automobile exhaust system representing a complex mechanical system and a water jet cutting machine representing a mechatronic system.

The general knowledge gained from these case studies should be a good base for coming implementation of this approach as an inherent working routine in companies developing complex mechanical and mechatronic products.

A specific result is a validated virtual model of the exhaust system, which facilitates fast structural dynamics simulation of customer proposed design layouts. It is also shown that the non-linear flexible joint between the manifold and the rest of the exhaust system makes the system behaviour complex. This has resulted in an additional general research question, namely how systems that are linear, except for small but significant non-linear parts, can be simulated in an efficient way. Another specific result is a validated real-time virtual machine concept for simulation of the water jet cutting machine, which facilitates early-stage design optimisation. As the mechanics and the control system are considered simultaneously, interaction effects can be utilised. An introductory optimisation study shows a significant potential for improved manufacturing accuracy and a more light-weight design. This potential would not likely have been found through a conventional sequential design approach.

The results of this thesis indicate that there is a great potential for improved product development performance in small and medium-sized companies. By incorporating modern simulation support these companies can improve their competitiveness as well as contribute to improved resource efficiency of society at large. In doing so, it is important to find a good balance between model fidelity, validity and cost for achieving a relevant decision support. The coordinated approach to simulation-driven design elaborated in this thesis is a promising and systematic way of finding this balance.

Thesis

Disposition

This thesis comprises an introductory part and appended papers A-H. The papers have been reformatted from their original publication into the format of this thesis but the content is the same.

Paper A

Englund T., Wall J., Ahlin K. & Broman G., Significance of non-linearity and component-internal vibrations in an exhaust system, in: *Proceedings of the 2nd WSEAS International Conference on Simulation, Modelling and Optimization*, Skiathos Island, 25-28 September, 2002, pp. 1921-1926

Paper B

Englund T., Wall J., Ahlin K. & Broman G., Automated updating of simplified component models for exhaust system dynamics simulations, in: *Proceedings of the 2nd WSEAS International Conference on Simulation, Modelling and Optimization*, Skiathos Island, 25-28 September, 2002, pp. 1931-1936

Paper C

Wall J., Englund T., Ahlin K. & Broman G., Modelling of multi-ply bellows flexible joints of variable mean radius, in: *Proceedings of the NAFEMS World Congress*, Orlando, 27-31 May, 2003

Paper D

Englund T., Wall J., Ahlin K. & Broman G., Dynamic characteristics of a combined bellows and liner flexible joint, *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering* 218(5) (2004) pp. 485-493

Paper E

Wall J., Englund T., Ahlin K. & Broman G., Influence of a bellows-type flexible joint on exhaust system dynamics, *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering* 218(12) (2004) pp. 1473-1478

Paper F

Wall J., Englund T. & Berghuvud A., Identification and modelling of structural dynamics characteristics of a water jet cutting machine, in: *Proceedings of the International Modal Analysis Conference – IMAC*, Dearborn, 26-29 January, 2004, pp. 138-147

Paper G

Jönsson A., Wall J. & Broman G., A virtual machine concept for real-time simulation of machine tool dynamics, *International Journal of Machine Tools & Manufacture* 45(7-8) (2005) pp. 795-801

Paper H

Wall J., Fredin J., Jönsson A. & Broman G., Introductory design optimisation of a machine tool using a virtual machine concept. Submitted for publication.

The Author's Contribution to the Papers

The papers appended to this thesis are the result of joint efforts. The present author's contributions are as follows:

Paper A

Took part in the planning and writing of the paper. Carried out approximately half of the modelling, simulations and measurements.

Paper B

Responsible for planning and writing the paper. Carried out approximately half of the development of the updating routine and the simulations.

Paper C

Took part in the planning and writing of the paper. Carried out approximately half of the modelling, simulations and measurements.

Paper D

Took part in the planning and writing of the paper. Took part in the modelling and simulation. Responsible for the measurements.

Paper E

Responsible for planning and writing the paper. Carried out approximately half of the measurements.

Paper F

Responsible for planning and writing the paper. Responsible for the modelling and simulation. Carried out approximately half of the measurements.

Paper G

Took part in writing the paper. Responsible for developing FE-models, sub-structuring and reduction techniques.

Paper H

Responsible for planning and writing the paper. Responsible for modelling, simulation and optimisation.

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Appended Papers

1 Introduction

Many sectors of industry are facing increasing global competition. More effective and efficient product development is a way of addressing this challenge.

The time aspect is critical. On some markets, changes in consumer preferences are very rapid and consequently the window of market opportunity very narrow. Also, the average product lifetime has been compressed significantly. With the expanding world economy, prices for many product types have decreased at maintained or improved product quality in a classical sense. As a consequence of all this, there has over the last decades been, and still is, a strong focus in industry on cutting time-to-market, reducing development and product costs, and increasing product quality [2-4].

A way for Swedish industry to stay competitive is to focus more on the market of complex products (systems). High level knowledge and competence in using front-edge methods, tools and equipment is then a more important competitive advantage than, for example, low labour costs. Since most companies in Sweden are small and medium-sized (SMEs), it is particularly important that this sector strengthen its capacity for such advanced product development.

A market driver that is currently gaining significantly in importance is the increasing awareness among customers, authorities and governments about the sustainability problems of today's society and the role of products in this context. A product's socio-ecological impacts – positive and negative throughout its life-cycle – are largely determined during early product development phases. Integration of sustainability aspects in product development is therefore gaining more and more interest [5, 6]. This includes saving resources in the development process itself.

Occupational health is also an increasingly important driver for product development. A legislation example related to structural dynamics is Directive 2002/44/EC, known as the Physical Agents (Vibration) Directive [7]. This was adopted by the European Union in 2002, forcing companies to assess and control vibration levels in their products.

Swedish industry has a relatively good starting position and a good possibility to develop its competence further in both these fields into a strong competitive advantage.

All of the above speaks in favour of developing and using more of simulation support in product development to better predict, describe, assess and improve product properties, especially in early development phases and especially regarding complex products. Simulation support has already proven to have a positive effect on product development performance in many sectors; see, for example, [8]. The potential is probably especially prominent for SMEs.

Simulation-driven design has been defined by Sellgren [9] as: "*a design process where decisions related to the behaviour and the performance of the design in all major phases of the process are significantly supported by computer-based product modelling and simulation*". This definition comes close to the meaning of the term as of this thesis. However, it is here further emphasised that an approach that can, rather than only verifying solutions that are already decided upon, support dialogues with customers, stimulate creation of new concepts and provide guidance towards more optimised designs, especially in early development stages, should be strived for.

To be useful for this, product models and simulation and optimisation procedures must be efficient, that is, they must accurately answer posed questions and point towards better solutions while consuming an acceptable amount of time and other resources.

This thesis discusses the potential of simulation-driven design and elaborates on a coordinated approach to virtual and physical prototyping, experimentation (simulation), verification/validation, and optimisation/design, in response to the following general research question:

"*How can properties of complex mechanical and mechatronic systems be more efficiently and systematically predicted, described, assessed and improved in product development?*"

The research question is addressed by action research through two industrial case studies to assure relevance and co-learning; one in cooperation with Faurecia Exhaust Systems AB, Torsås, Sweden, studying an automobile exhaust system as an example of a complex mechanical system, and one in cooperation with Water Jet Sweden AB, Ronneby, Sweden and GE Fanuc Automation CNC Nordic AB, Sollentuna, Sweden, studying a water jet cutting machine as an example of a mechatronic system.

Specific aims of the first case study include developing a validated virtual model for fast structural dynamics simulation of the automobile exhaust system. The purpose is to support dialogues around customer-proposed design layouts and optimisation in early development phases.

Specific aims of the second case study include developing a validated virtual machine concept for real-time simulation of the water jet cutting machine, including validated models of the electrical and mechanical parts and their interaction with a real control system. The purpose is to support optimisation in early development phases, in particular regarding manufacturing accuracy, manufacturing speed and weight.

As a general background to the appended papers, an overview discussion of simulation-driven design in product development, and some of its related terminology, is given in chapter 2. The industrial case studies are described in more detail in chapter 3. A summary of the appended papers is provided in chapter 4, and conclusions and suggestions for future work follow in chapter 5.

2 Simulation-Driven Design

2.1 Simulation and Experimentation in Product Development

As a general remark it should be said that the terminology related to simulation-driven design and product development is not entirely distinct, neither in actual development work nor in the literature. Even for the concepts themselves there exist several alternative definitions. The meaning of simulation-driven design in this thesis is briefly discussed in chapter 1. It will be further motivated and elaborated in this chapter. As regards product development, Ulrich & Eppinger [10] define it as "*a set of activities beginning with the perception of a market opportunity and ending in the production, sale, and delivery of a product*". It might be viewed as an iterative decision making process. The ability to make well-informed decisions is, of course, therefore crucial. Well-informed decisions demand knowledge of the design problem at hand, which increases as the development project progresses. On the other hand, design changes generally become more difficult and costly (resource intensive) as the development project progresses. This paradox, often resulting in far from optimal designs, is schematically illustrated in figure 1.

Experimentation (simulation) offers a way of increasing the knowledge about the design problem in early development phases where the degree of design freedom is still high, and is therefore a key component to successful product development [8, 11, 12]. The term "experimentation" is defined here as the act of conducting a controlled investigation for the testing of an idea or hypothesis aiming at an increased knowledge of the studied system. This means running some kind of test procedure for various input data and to study how these data affect the output of the procedure. Some kind of visualisation of the output is often included, which could be in the span from simple hand-drawn diagrams to advanced virtual reality environments.

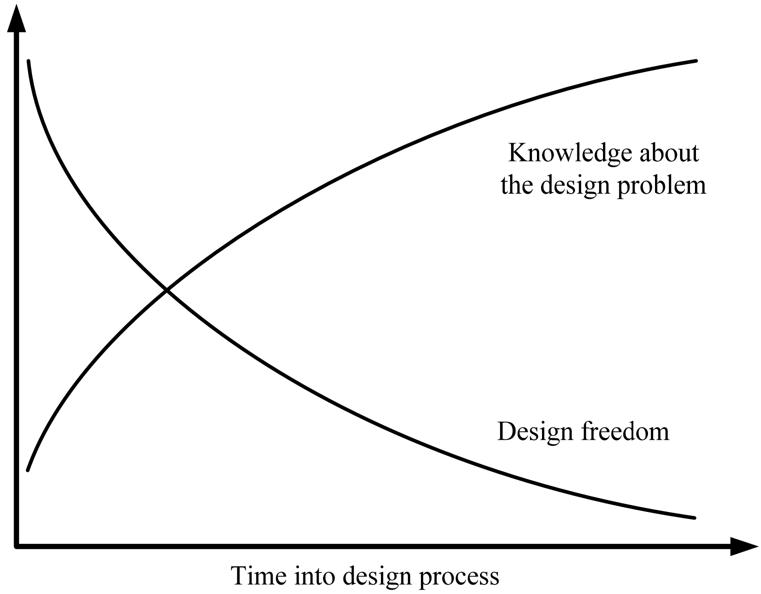

Figure 1. The product development paradox.

The terms "experimentation" and "simulation" are sometimes used in a synonymous way. It is, however, more common to use the term "simulation" when the test procedure is based on virtual models and to use the term "experimentation" when the test procedure is based on physical models. The term "virtual experimentation" is sometimes used in the first case and the term "physical experimentation" is sometimes used in the second case [13, 14], while some authors use the term "simulation" both for testing with virtual models and for testing with physical models [15]. According to, for example, Neelamkavil [16], "*simulation is the process of imitating important aspects of the behaviour of the system in real time, compressed time, or expanded time by constructing and experimenting with a model of the system*". Henceforth in this thesis, the term "experimentation" is used in the wider meaning, including tests with both physical and virtual models while "simulation" refers to tests based on virtual models.

Experimentation with the actual system is seldom a realistic alternative due to restricted resources or the fact that the system might not yet exist, might be deemed too complex to comprehend or might not be available. It might even be dangerous or destructive to use the real system in some cases. As indicated above, experimentation is therefore commonly done using models.

2.2 Modelling and Prototyping

A model might be defined in a number of ways. Neelamkavil [16] states that "*A model is a simplified representation of a system intended to enhance our ability to understand, predict and possibly control the behaviour of the system*". In engineering science two broad categories of models are identified: physical models, for example scale models, and virtual (symbolic) models, for example mathematical models. Depending on the type of mathematical model, analytical or numerical solution methods are used. Since it is often difficult to describe a product simply enough to find an analytical solution, numerical methods are most frequently used in product development.

Depending on the investigation objectives the same system may be represented with sufficient accuracy by models of different fidelity, giving different types and amounts of information; fidelity being the extent to which a given representation reproduces the studied system (real world). The investigation objectives should therefore always be unambiguously specified as they focus the modelling process on problem specific issues. Once these objectives are clearly stated, a suitable experimental frame can be developed. This is a specification of the conditions under which the system is studied [17]. The modelling process aims at developing a model that mimics the studied system with sufficient accuracy in the specified experimental frame.

In general, the modelling process consists of analysis, simplification, abstraction, and synthesis [18]. To analyse is to study the system to be modelled and to divide it into components that can be treated separately. Simplification is accomplished by stripping away unimportant details, preferably by starting with the simplest model possible and adding complexity until it represents the system characteristics of interest with sufficient accuracy for the intended application of the model, or/and through the assumption of simpler relationships than in the real world. The ability to build models by selecting the smallest subset of variables which adequately describes the studied system is, of course, highly desirable. Abstraction is the process of representing the qualities and behaviour of the studied system in a different form or manner. Synthesis is the process of assembling all component models into a complete model of the studied system. The quality of the assembled model is dependent on the quality of the component models and the validity of the assumed interactions between the components included. If the modelling process is carried out properly it will result in a model that mimics the behaviour of the studied system with sufficient accuracy when used for experimentation. Documentation should also be considered as an integral part of the modelling process [16].

The terms “model” and “prototype” are sometimes used in a synonymous way; see, for example, [19]. Ulrich & Eppinger [10] define a prototype as “*an approximation of the product along one or more dimensions of interest*”. This definition is broad, including all entities exhibiting some characteristic of the studied product of interest, and it overlaps greatly with the meaning of a model as discussed above. In this thesis, a prototype is a model of a product, or part of a product, that is used for the purpose of investigating certain characteristics of the product during product development. As all models in this thesis are used for this purpose, the terms “model” and “prototype” are used as synonyms. An overview of virtual prototyping is given in [20]. Virtual prototyping within mechanical engineering is discussed in [19].

Experimentation has traditionally been based on physical prototyping. This is often resource intensive, and as a consequence, the main objective has been verification in late development stages. Such a strategy may prohibit innovation and lead to a conservative design [14]. The increasing global competition has, however, changed product development practices drastically. An approach based only on physical prototyping is usually no longer a viable option due to time and cost constraints. Virtual prototyping is rapidly gaining attention. This “paradigm shift” is an important step towards more effective and efficient product development. A major advantage of virtual prototyping is that the number of design configurations (proposals) that could be tested within a limited resource frame can be significantly increased compared to physical prototyping. Once a virtual model has been developed, the additional cost for extensive experimentation is usually very low compared to building new physical models. This potentially widens the scope of experimentation from primary being a means of verification to also answer “what if” questions and to challenge presumed answers. This aids innovation and accelerates the learning process [8, 14, 18, 21, 22]. Furthermore, virtual prototypes can be utilised in early stages, in contrast to full physical prototyping that naturally need to be conducted at a late stage in the development process. Virtual prototyping can thus make the learning curve steeper in the beginning of a development project, when design changes are still economically viable, see figure 2.

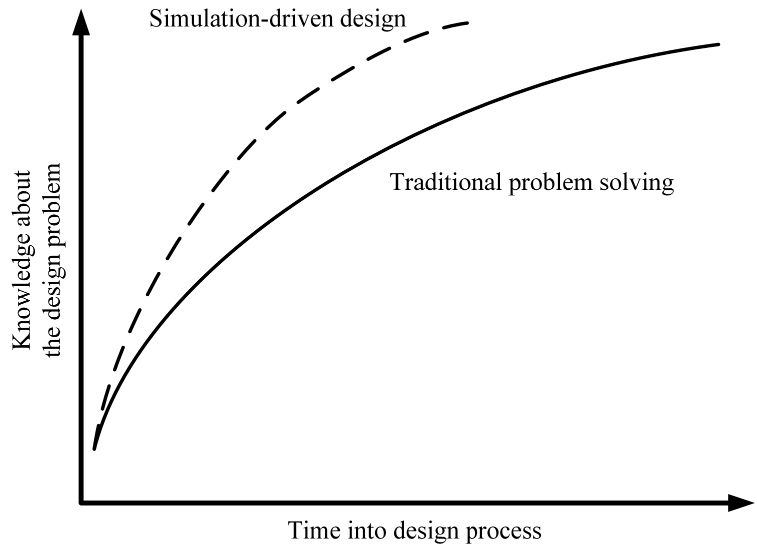

Figure 2. Benefit of simulation-driven design.

The radically changed economics of experimentation might also make a parallel strategy more attractive [8], conceptually shown in figure 3. In this way, larger investigations can be undertaken within a given time frame compared to a sequential strategy [8, 23]. This further increases the probability of discovering more radical improvements and innovations. Early parallel experimentation, delaying commitment to a single solution, also reduces the number of real world design iterations and the probability of major rework in later design phases [24]. Adopting a parallel strategy invites more failures. Failure should however not be confused with mistakes. Failures contribute just as much as success to the generation of useful information. It is just as important to understand what does not work as what does [8, 25]. Mistakes, on the other hand, produce very little new information.

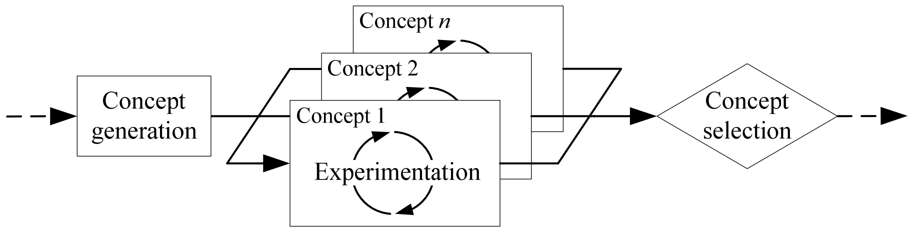

Figure 3. Parallel concept evaluation.

A drawback of virtual models may be a lower fidelity than that of full physical models. Lower fidelity implies that less knowledge might be gained from each iteration compared to experimentation with physical models (generally of higher fidelity). In spite of this, the overall learning speed is still often higher with virtual prototyping since each iteration is performed much faster, making many more iterations possible within a given time frame [26].

Simulation is increasing in almost all industrial sectors. This is particularly evident among automobile manufactures. Examples of case studies related to the automobile industry are those of [13] studying BMW, [26] studying Toyota, and [27] studying Audi. All these studies demonstrate the positive impact of simulation on product development performance. A shift from physical experimentation to simulation as such may no longer be a differentiator within the automobile industry, as it potentially still is in many other industrial sectors, especially for SMEs. An approach that can, rather than only verifying solutions that are already decided upon, support dialogues with customers, stimulate creation of new concepts and provide guidance towards more optimised designs, might however be an important differentiator.

It is important to note that what is said above does not in any way mean that physical experimentation is obsolete. It should not be excluded beforehand that a physical model may, in fact, still be the best option in a specific case, in spite of the currently available resources for virtual prototyping. Furthermore, when developing and validating the virtual models, limited physical models or the reuse of previous validation results are necessary. A balanced combination of virtual and limited physical models is usually the best approach, favouring virtual models of lower fidelity in the early development phases.

2.3 Verification and Validation

As mentioned above, models should be validated to be trusted as support for design decisions. In fact, a model has no value as decision support until it is judged valid. It might, however, still have other benefits, such as increasing the modeller's understanding of the studied system. Definitions of the terms "verification" and "validation" are also by no means consistent, neither in practice nor in the literature. Verification is often defined as the process of determining if the model works as intended, that is, that model coding and implementation is done correctly. Validation is often defined as a measure of usefulness in relation to the investigation objectives, and may be divided into model validation, data validation and operational validation [28]. Model validation is the process of determining if the model is reasonable for the intended usage, for example, checking the assumptions made. Data validation is the process of checking that the data necessary to build and run the model are adequate and correct. Operational validation is the process of determining if the implemented model is able to describe the physical phenomena of interest to the extent demanded by the investigation objectives.

When developing a model, generally only a subset of the components and interactions of the studied system is implemented. The rest is neglected to optimally use available limited resources. As already mentioned, simplification and abstraction is the very essence of modelling. The investigation objectives should drive the choice of which trade-offs to make. The relationship between fidelity and committed resources is schematically illustrated in figure 4.

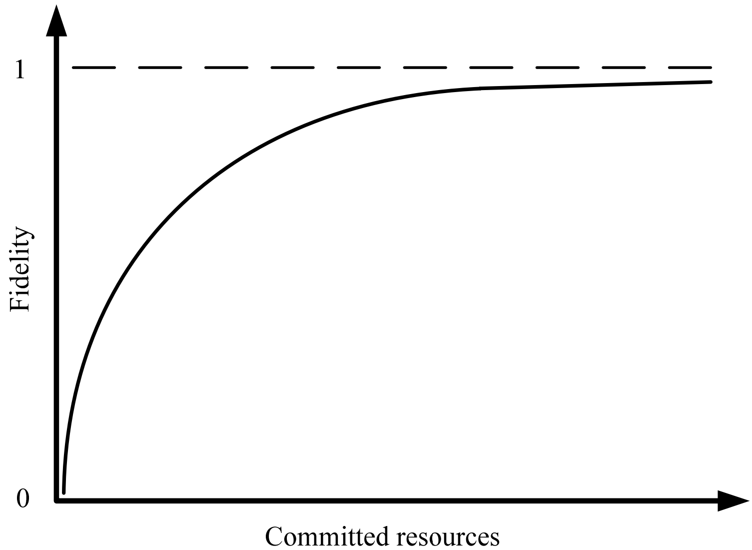

Figure 4. Fidelity versus committed resources, modified from [29].

There is a widespread acceptance for the importance of keeping models simple [16, 30-32], that is, the ability to build models by selecting the smallest subset of variables which adequately fulfils the investigation objectives. The availability and affordability of computer power combined with easy-to-use software, enabling building and solving complex models, do not change this. One reason is that the ever increasing computational power tempts analysts to redefine what is considered an “adequate” model. Examples of this can be the transitioning from linear to non-linear analysis [33] or to attack new types of problems such as multidisciplinary optimisation [34].

Fidelity, as used here, is divided into two basic components, dimensions (how much of the studied system is represented in the model) and attributes (what is the quality of these representations). This view of fidelity is closely related to the definition of complexity given in [17] where it is referred to as the size/resolution/interaction product, size being the number of components, resolution the number of variables and their precision and interaction describes the interaction between included components.

It is not unusual that models with higher fidelity than actually demanded are developed due to the assumption that they are more accurate and therefore better. A number of other reasons, technical as well as non-technical, to why models of unnecessary high fidelity are developed is given in [35]. Non-technical factors are: “show off” factor (impress manager), “include all” syndrome (usually insecure inexperienced modeller) and possibility factor (available computational resources). Technical factors are: lack of understanding of the real system, inability to model the problem correctly, inability to use software or write code, unclear investigation objectives, which may result in an unnecessarily wide experimental frame.

A major drawback of high fidelity is that it is generally harder to truly understand the relationships contained within the model. This makes verification and validation harder, resulting in a final model that is more likely to contain errors. Furthermore, it might also make interpretation of simulation results more difficult, thereby increasing the risk of the wrong conclusions being drawn [31]. The fidelity level does not only affect computational performance but also other aspects such as project management, communication time, resource constraints, et cetera. Based on this, a model of lower fidelity would, in the general case, be expected to be easier to understand, contain fewer errors, and be resource efficient considering development, implementation, simulation, et cetera. Following this thought, such a model is more likely to be adopted and used in practical applications [16]. Resource commitment must also be considered. A simple model makes it easier to admit failure and start over [30]. Reducing complexity is, however, a difficult problem as oversimplification renders the model useless while inclusion of trivial details makes the model unnecessarily large, complex and possibly intractable.

There are, of course, also negative aspects of highly simplified models. One problem is to obtain sufficient confidence. Furthermore, simplified models tend to be less portable than those of higher fidelity. A simple model may be hard to understand due to the level of abstraction. While being perfectly understandable to the modeller, an end user (person, or team of several persons, using the simulation results as decision support) might deem the model incomprehensible.

The aim of experimentation is to increase the knowledge of the studied system, attaining information useful to support decision making in the product development process. The information is useful if it decreases the risk that the product will be something else than what it is supposed to be. Risk is here quantified as the product of the probability of occurrence and its consequences [36]. A model’s performance might be defined as its ability to produce useful information. Such a performance measure is influenced by both fidelity and validity. A model’s value is proportional to its performance and inversely proportional to the cost of the resources spent achieving that level of performance.

Modelling is an iterative process. Models are ideally developed and refined gradually, starting with a model based on simple assumptions, learning from it, and adding complexity if necessary to successfully fulfil the investigation objectives [18]. The validity of the model generally increases through successive iterations. The input to the modelling process is experience from prior projects and resources, for example, in the form of personnel and equipment. The output is a model with a certain level of fidelity and validity. The more useful information that can be extracted from experimentation with the model, the lower the risk for the decision maker. However, the cost of resources spent accumulates through the modelling process. Hence, regarding the value of the model, there is an obvious trade-off between fidelity and validity (performance) and cost as shown in figure 5. The model should therefore be as good as necessary, not as good as possible.

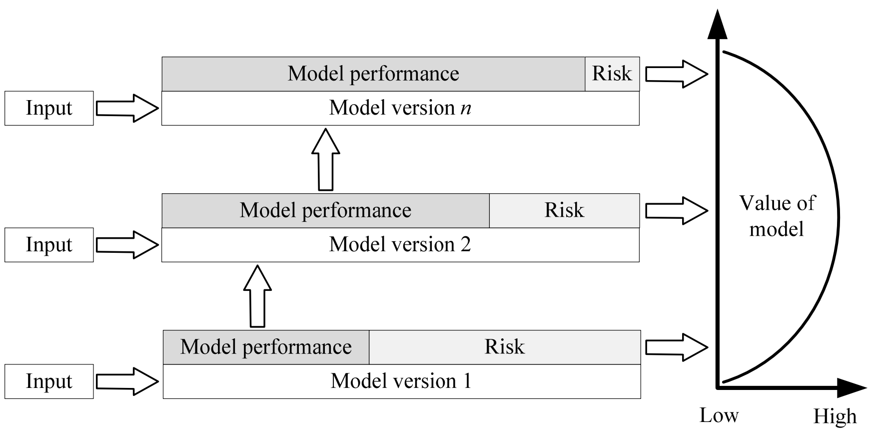

Figure 5. Value of model.

The obvious question of when to stop this iteration is far from trivial. In practice, modelling is often stopped first after the last step has shown to be a step taken too far. An iterative strategy with small development steps decreases the cost of the last, unnecessary, step.

2.4 Optimisation and Design

The whole purpose of developing and validating models and using them for experimentation is, as already stated, to support well-informed design decisions. The main parts of the basic design cycle are analysis (defining product criteria), synthesis (thinking up design proposals), experimentation (predicting product properties), and evaluation (comparing properties to criteria) [15]. Based on this evaluation it is decided whether the performance of the proposal is satisfactory. This basic design cycle is repeated in the different product development phases, usually with an increasing level of detail.

Obviously it is desirable to find the best design out of many options, or at least a better design than the already existing. Optimisation techniques can be of considerable help in dealing with this complex task. The design problem must, however, be unambiguously defined, including a clear definition of what is meant by "best design". A quantitative measure of performance, a so-called objective, has to be defined. The objective must be dependent on some design variables. By changing the design variables new designs are created. There are usually restrictions on how values for the design variables may be assigned, that is, they are constrained in some way. Furthermore, design problems are commonly characterised by a mix of continuous and discrete variables.

Once an optimisation problem is properly formulated an optimisation algorithm may be applied to solve it. The optimisation algorithm guides the search of the optimal solution through the design space by iteratively suggesting design variable values that improve the objective. There is, however, no universal optimisation algorithm. Numerous different algorithms exist, ranging from more general purpose algorithms to problem-specific ones. The strategy used to move from one iteration to the next distinguishes one algorithm from another. The algorithm should preferably be robust, efficient and accurate. These goals may, however, be conflicting and choosing an appropriate algorithm is therefore a difficult task, heavily influencing solution time, or in the worst case, preventing a solution to be found at all. Usually, the algorithms are complicated and need to be computerized. Since a large number of objective evaluations are usually necessary, it is important that product models and simulation procedures are efficient.

Real world engineering design problems are usually characterised by the presence of multiple objectives. These objectives may be competing, meaning that some trade-off is required. In automobiles for example, light-weight components are sought for better fuel economy. The structural integrity of these components must, however, also be considered for acceptable crashworthiness. How to handle this is a subject of current research; see, for example, [37, 38].

The need for effective and efficient product models and simulation procedures is even more pronounced for multidisciplinary products. Multidisciplinary design optimisation, incorporating all the relevant engineering disciplines simultaneously and thereby considering possible interaction effects is, of course, an attractive design approach. These are, however, very complex problems and the computational cost related to this type of problem is usually rather high. Strategies to handle this are discussed by, for example, [34].

Design optimisation in general is discussed by, for example, [39, 40]. Many books have been written on the more mathematical aspects of optimisation algorithms; see, for example, [41-43].

2.5 A Coordinated Approach

The core terms discussed above are gathered in a coordinated approach aiding simulation-driven design, see figure 6.

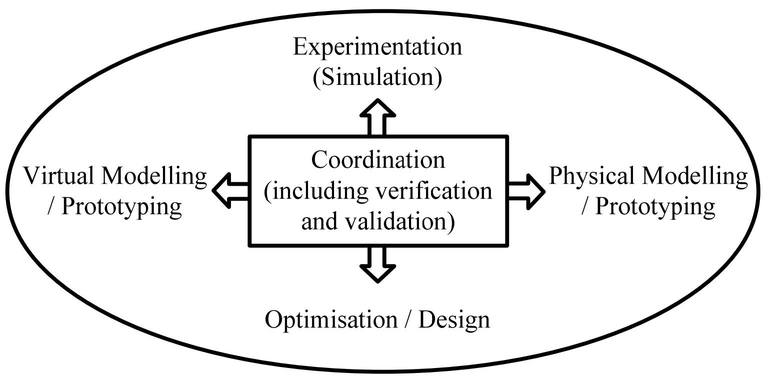

Figure 6. A coordinated approach.

In short, virtual models for description of interesting product characteristics are developed, verified and used for initial simulation. These results are then compared to results from experimentation with physical models, developed in parallel, or are assessed by using an analogy with previous products (experience). As modelling is an iterative process, the model is revised and refined until it is sufficiently valid. The coordination also means that simulation is used to design good physical models and measurement strategies, increasing the likelihood of obtaining useful information (virtual testing [44]). This iteration is an important part of modelling [14]. The validated virtual model can then be used together with an optimisation algorithm for finding an improved design. Should optimisation imply design changes that significantly change the relevance of the assumptions of the virtual model, the whole procedure is repeated. When a completely new product is developed, many complete iterations are usually needed. When a new variant of a product is developed, much knowledge inherent in existing virtual models can usually be re-used.

Apart from the case studies presented in this thesis, the coordinated approach has evolved through several other studies [45-47].

3 Industrial Case Studies

3.1 Automobile Exhaust System

An automobile exhaust system has several functions. Originally, it was used for silencing the noise caused by high pressure exhaust gases leaving the engine and for transporting these hot and toxic gases away from the driver's compartment. Nowadays, it is also an important and integral part of combustion and emission control. For this to work properly, there must be no leakage upstream of the catalytic converter. The durability of that part of the system is therefore crucial. Customer demands for comfort and a long product life guarantee, also for the exhaust system as a whole, are additional reasons for the increasing importance for design engineers to be able to predict, describe and assess the dynamics of various system design proposals during product development [48]. The above considerations converge into the critical objective of obtaining low vibration levels in the exhaust system.

A modern exhaust system generally consists of a manifold, a flexible joint, a catalytic converter, mufflers, and pipes. A typical system is shown in figure 7 (manifold not included).

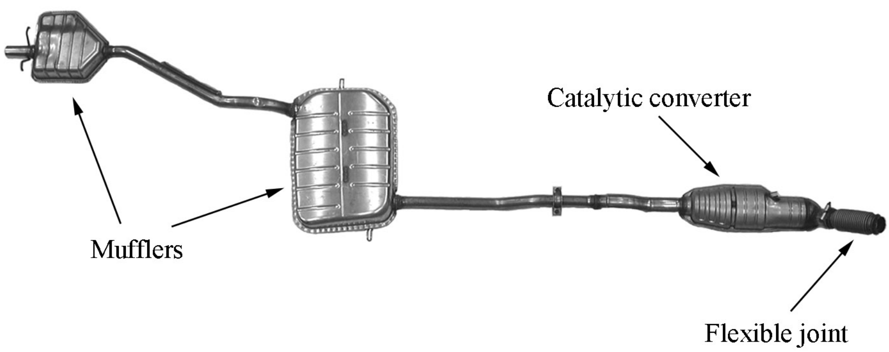

Figure 7. Typical exhaust system design (modified Volvo S/V 70 system).

The catalytic converter is included to convert toxic and environmentally harmful gases into less harmful gases. The mufflers are included to reduce the noise level. The flexible joint is included primarily to reduce transmission of engine movements to the exhaust system. As it is usually located between the manifold and the catalytic converter it needs to be gas-tight. It must also withstand high temperatures, and should combine high flexibility with high strength and durability. A steel bellows-type joint is therefore commonly used. It generally consists of a multi-ply bellows combined with an inside liner and an outside braid. The liner is included to reduce the temperature of the bellows and improve flow conditions. The braid is included for mechanical protection and to limit the possible extension of the joint. The parts are connected with end caps. A schematic of the joint is shown in figure 8.

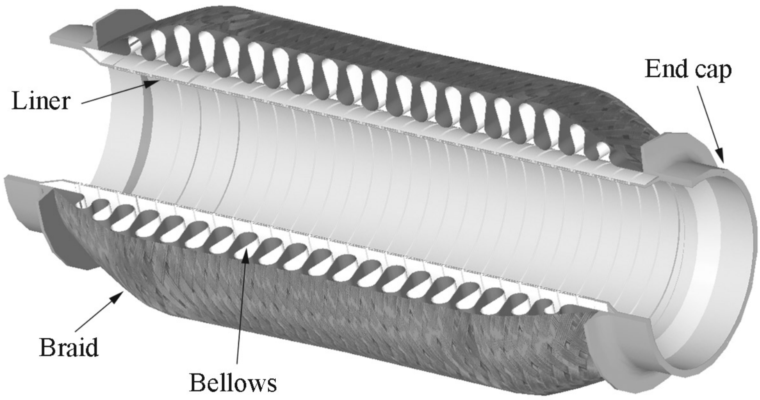

Figure 8. Basic design of the flexible joint.

Experience shows that this joint sometimes causes complex dynamic behaviour of the exhaust system and this has caused car and component manufacturers severe problems [49].

A few publications concerning dynamics analysis of exhaust systems have been found in the literature. Verboven et al. [50] discuss measurements of exhaust system dynamics in general while focusing on a comparison of an ordinary experimental modal analysis with a running mode analysis. Deweer et al. [51] perform an experimental modal analysis and emphasise the usefulness of simulation support during the pre-test phase to make better measurements. Belingardi and Leonti [52] deal with problems related to building finite element models with which to study the dynamic behaviour of exhaust systems. Piombo et al. [53] point out the direct coupling between the dynamic behaviour of the exhaust system and its fatigue resistance. Ling et al. [54] perform finite element simulations of exhaust system dynamics to study vibration transmission to the car body and show the potential benefits of incorporating simulation into the exhaust system design process.

The above references do not include studies of the influence of the bellows-type flexible joint on the dynamics of exhaust systems; neither do they include discussions of non-linear dynamics analyses.

The flexible joint has been studied by, for example, Broman et al. [55] and Cunningham et al. [56] in relation to the automobile exhaust system application. In addition to these references, there are numerous publications regarding bellows flexible joints in general. For a review, see [55].

3.2 Water Jet Cutting Machine

A water jet cutting machine is a type of machine tool. Since the industrial revolution machine tools have had a huge impact on the modern society. Mass production turned the attention to manufacturing accuracy and speed, resulting in a shift from manually to automatically controlled machines. The introduction of computer numerically controlled (CNC) machines improved the flexibility and accuracy. Advanced controllers do not change the fact that each machine design has absolute mechanical limits. Its dynamic behaviour has a direct effect on manufacturing accuracy as machine tool vibration results in a degraded quality on the machined parts. Other consequences are for example shorter tool life and unpleasant noise. These unwanted effects are often more pronounced as manufacturing speeds are increasing.

CNC machine tools are typical mechatronic systems. A generally accepted definition of a mechatronic system has not yet been adopted [57]. At the centre of the modern understanding of mechatronics is the synergetic effect of integrating different technologies such as mechanical engineering with electronics and intelligent computer control. The focus is on the combination of the individual technologies, making it possible for a system to perform its task optimally or for new functionalities to be realized. A parallel multidisciplinary design approach, simultaneously analysing the control, structural and electronic components and thereby utilising interaction effects, is believed to be superior to the traditional sequential design approach [58].

Water jet cutting is a manufacturing technique that uses the erosion power of water to shape the work piece. The basic principle is to channel highly pressurised water (400 MPa or more) through a narrow nozzle in the cutting head, concentrating a high amount of energy in a small area and thereby creating massive cutting power. Cutting with a jet of only water works for soft materials. Abrasive particles are added to the water jet for the cutting of hard materials, such as metals. In an abrasive water jet the abrasive particles, not the water, erode the material. More information about water jet cutting in general can be found in [59]. An example of a water jet cutting machine is shown in figure 9.

Figure 9. Water jet cutting machine.

Very little research has been published on the machine itself and its dynamic behaviour. The major part of the published research related to water jet cutting machines is focused on the cutting process; see, for example, [60-62]. Machine tool dynamics in general is, however, receiving interest within the research community; see, for example, [63-67]. Machine tool dynamics is becoming even more important as the popularity of high speed machining is increasing.

A comprehensive review of virtual prototyping within machine tool technology is given in [68]. The necessity of considering the machine tool as a complete system, that is, simultaneously considering design of structural components and control, is discussed. This is confirmed in, for example, [66, 67, 69]. The coupling of physical control systems to simulation of machine tool models is not discussed in [68]. This has been done in, for example, [70, 71]. However, none of these incorporates detailed time-varying structural dynamics simulation capabilities.

4 Summary of Papers

The first case study (the automobile exhaust system) is described in papers A, B, C, D and E. In paper A the dynamic characteristics of a typical exhaust system are investigated. A simplified virtual model is suggested and validated. In paper B an automated model updating procedure is developed. The exhaust system model from paper A is used as a test case showing that the procedure works well. In paper C a simplified virtual model of a multi-ply bellows is developed and validated. The flexible joint model of paper C is improved in paper D by also including the liner. The combined bellows and liner joint model shows a significantly non-linear behaviour. The influence of the combined bellows and liner joint model on the complete exhaust system dynamics is investigated in paper E, which shows the importance of including a model of the liner in the complete system model when present in the real system.

The second case study (the water jet cutting machine) is described in papers F, G and H. In paper F a virtual model describing the dynamic behaviour of the water jet cutting machine is developed. The model is developed and validated using the same coordinated approach as in the first case study. In paper G a virtual machine concept is presented. The concept consists of a real control system, a machine simulation (including the model developed in paper F), and a 3D machine visualisation. Initial simulation is performed to show the relevance and potential of the virtual machine concept. The virtual machine concept developed in paper G is used in paper H to perform an introductory design optimisation study. It is shown that interaction effects exist between mechanics and control, which emphasises the usefulness of the virtual machine as a design tool when developing mechatronic systems.

4.1 Paper A

In this paper the dynamic characteristics of a typical exhaust system are investigated through both virtual and physical models. The flexible joint is not included in the study. It is shown that shell vibrations of the catalytic converter and the mufflers as well as ovalling of the pipes are negligible in the frequency interval of interest. This implies that the pipes can be modelled by using beam elements, and that the catalytic converter and the mufflers can be modelled by using lumped mass and mass moment of inertia elements. Additional short beam elements are used with these models to account for the flexibility at the connections. It is also shown that non-linearity in the part of the system downstream of the flexible joint is negligible. This implies that this part can be considered as a linear subsystem in dynamics studies of the complete exhaust system.

4.2 Paper B

In this paper an automated model updating procedure is developed. A model for simulation of the part of the exhaust system that is downstream of the flexible joint is built in the commercial finite element software ABAQUS using the simplified component models suggested in paper A. The sum of the differences between simulated and measured natural frequencies is chosen as the objective function to be minimised. Constraints are used on the correlation between simulated and measured mode shapes, considering the modal assurance criterion matrix, to ensure that correlated mode pairs are compared. The properties of the short beam elements used to model flexibility at the connections between the mufflers/catalytic converter and the pipes are used as the variables to be adjusted during updating. Updating is performed by using the sequential quadratic programming algorithm in the Optimization Toolbox in MATLAB. To obtain an automated procedure, in-house developed tools for data exchange between ABAQUS and MATLAB are used. The correlation between simulation results using the updated model and results from measurements is very good, indicating the usability of these component models and also that the updating procedure works well.

4.3 Paper C

In this paper a highly simplified virtual model of the bellows flexible joint is presented. A straightforward way of modelling the bellows is to use shell finite elements. Due to the convoluted geometry of the bellows this would, however, require a high number of elements. The bellows model would thus constitute a large part of the model of the complete exhaust system. For more efficient dynamics simulations, a beam finite element representation of the bellows has been presented in prior work. This paper suggests adjustments by which this procedure can be extended to model also multi-ply bellows of a variable mean radius. Physical models are used for validation of the virtual model. The correlation between simulation and measurement results is very good. The measurements reveal, however, that the bellows is slightly non-linear, but this non-linearity is weak and may be neglected in the present application. Nonetheless, a hypothetical qualitative explanation for the non-linearity is provided.

4.4 Paper D

In this paper the bellows of paper C is combined with an inside liner. The braid is not included. An approach for modelling the combined bellows and liner joint is developed and validated for the axial and bending load cases. The correlation between simulation and measurement results is good. It is shown that the dynamic characteristics of the joint are strongly dependent on the relation between the excitation force level and the friction limit of the liner. Peak responses are, for example, significantly reduced when the excitation level approximately corresponds to the friction limit. This is due to friction-based damping. The liner thus makes the dynamics of the joint significantly non-linear and complex, and it is therefore important to consider these effects in joint design.

4.5 Paper E

In this paper the influence of the flexible joint on the dynamics of the exhaust system is investigated. The ability of different joint configurations to reduce vibration transmission from the engine to the exhaust system is studied. Measurements show the great reduction of vibration transmission to the exhaust system that the bellows-type joint, with and without an inside liner, gives in comparison to a stiff joint. However, for the combined bellows and liner joint, vibration transmission is higher than for the bellows alone. This makes the choice of including a liner in the real application a complex issue. It is also shown how the combined bellows and liner joint makes the exhaust system dynamics significantly non-linear, leading to the conclusion that a model of the liner needs to be included in the model of the system when the liner is present in the real system. Furthermore, the non-linearity of the double-plied bellows reported in paper C is seen to have a minor influence. This confirms the validity of using the linear beam model of the bellows itself (without the liner) when studying exhaust system dynamics.

4.6 Paper F

In this paper a virtual model of the dynamic behaviour of a water jet cutting machine is developed. The studied machine shows a complex dynamic behaviour. Hence, developing valid virtual models is a difficult task. The model is developed using an iterative coordinated approach, aiming at a validated, computationally efficient, virtual model of the studied machine. A high correlation between results obtained from theoretical and experimental modal analysis implies that the developed model can be used with confidence in future studies of the machine's total system behaviour.

4.7 Paper G

In this paper a fully automated virtual machine concept for simulation of the dynamic behaviour of a machine tool is presented. The concept consists of a real control system, a machine simulation, and a 3D machine visualisation. The specific application is a water jet cutting machine, as an example of a modern CNC machine tool. The structure of the concept allows for easy adjustment or replacement of component models, which facilitates transfer and reuse of knowledge between development projects. Virtual models are developed, primarily of the mechanical structure, and simultaneously physical models are developed for the purpose of validation. The validation process indicates good agreement between simulation and measurement results, but suggests further studies on servo motor, connection and flexibility modelling. However, already from the initial simulation it can be concluded that the influence of structural flexibility on manufacturing accuracy is of importance at desired feeding rates and accelerations, and that this fully automated real-time virtual machine concept is a promising base for dealing with this trade-off between productivity and accuracy of the manufacturing process through multidisciplinary design optimisation.

4.8 Paper H

In this paper the usefulness of the virtual machine concept developed in paper G as a design tool is shown. An introductory optimisation study is conducted aiming at improving the existing design of a water jet cutting machine regarding weight and manufacturing accuracy at maintained manufacturing speed. The design problem can be categorised as constrained multidisciplinary multi-objective multivariable optimisation. An optimisation approach using a genetic algorithm is therefore deployed. The outcome of the study, a significantly improved machine tool design, is presented and compared to the original design. It is also shown that interaction effects exist between structural components and control. Hence, this design improvement would most likely not have been possible with a conventional sequential design approach within the same time, cost and general resource frame. This indicates the potential of the virtual machine concept for contributing to improved efficiency of both complex products and the development process for such products.

5 Conclusions

The potential of simulation-driven design for supporting effective and efficient product development is shown through two industrial case studies. In the first case study an automobile exhaust system is studied as an example of a complex mechanical system. In the second case study a water jet cutting machine, a type of machine tool, is studied as an example of a mechatronic system.

Simulation-driven design is here described as a design process where simulation, rather than only verifying solutions that are already decided upon, support dialogues with customers, stimulate creation of new concepts and provide guidance towards more optimised designs. A full embracing of simulation-driven design potentially provides the leverage companies need on today's increasingly competitive market.

To be useful for simulation-driven design, product models and simulation and optimisation procedures must be efficient, that is, they must accurately answer posed questions and point towards better solutions while consuming an acceptable amount of time and other resources. A coordinated approach to virtual and physical prototyping, experimentation (simulation), verification/validation, and optimisation/design is elaborated to support such efficiency.

There are many drivers for more effective and efficient product development. Not the least the increasing awareness among customers, authorities and governments about the sustainability problems of today's society and the role of products in this context. A product's socio-ecological impacts – positive and negative throughout its life-cycle – are largely determined during early product development phases. This favours the adoption of simulation-driven design.

The complex dynamic behaviour of modern exhaust systems, partly due to the non-linear flexible bellows-type joint between the manifold and the catalytic converter, emphasises the need for a coordinated approach to simulation-driven design of these systems. The presented simplified, validated, finite element model of the automobile exhaust system, including a combined bellows and liner joint, enables fast structural dynamics simulation that may support dialogues around customer proposed design layouts and optimisation.

The complexity of the water jet cutting machine (mechatronic system) makes design complicated and difficult as trade-offs between design objectives are far from intuitive. This emphasises the need for a coordinated approach to simulation-driven design of these systems. The presented real-time virtual machine concept for simulation of the water jet cutting machine, including detailed mechanical component models, is unique. It shows that the influence of structural flexibility on manufacturing accuracy is of importance at desired feeding rates and accelerations. Furthermore, the existence of interaction effects between the control system and the structural components is shown, clearly indicating the relevance of the virtual machine concept for design optimisation.

The main contribution of this thesis to science and technology is summarised as follows.

On a general level this thesis:

- Elaborates a coordinated approach to simulation-driven design, including virtual and physical prototyping, experimentation (simulation), verification/validation, and optimisation/design.
- Confirms the conclusion made by others regarding the benefits of simulation support in product development, while emphasising the importance of finding a good balance between model fidelity, validity and cost for achieving a relevant decision support.
- Points to the potential of simulation-driven design to improve product development performance, in particular regarding complex products and in particular in small and medium-sized companies.
- Has resulted in an additional general research question, namely how systems that are linear, except for small but significant non-linear parts, can be simulated in an efficient way. (This research question is pursued in a new separate project.)

On a more specific level this thesis:

- Presents a simplified, validated, virtual model of an automobile exhaust system, including a combined bellows and liner flexible joint, which could support dialogues with customers, stimulate creation of new concepts and aid the search for more optimised designs.
- Demonstrates the highly non-linear behaviour of the combined bellows and liner joint, and the importance of including a model of the liner in the model of the complete system when the liner is present in the real system.

- Presents a validated real-time virtual machine concept for simulation of a water jet cutting machine with the ability to handle interaction effects, which could support dialogues with customers, stimulate creation of new concepts and aid the search for more optimised designs.
- Shows, through an introductory design optimisation study, the value of this kind of design tool for mechatronic products. Significant improvements are obtained already in this introductory study as regards manufacturing accuracy and weight of the water jet cutting machine.

Interesting for future work is the implementation of the coordinated approach to simulation-driven design as an inherent working routine in companies developing complex mechanical and mechatronic products. The general knowledge gained from the presented case studies should be a good base for this.

Regarding the exhaust system, interesting questions for future research may, besides the one mentioned above, include how more realistic excitations and boundary conditions, as well as the high temperatures and the flow of the exhaust gases, and the braid of the flexible joint, affect the dynamics of the exhaust system.

Regarding the water jet cutting machine, a complete validation against a real water jet cutting machine should be carried out. The servo motor behaviour should be further investigated, friction at connections should be considered and the influence of flexibility of the parts now considered rigid should be investigated. Furthermore, the influence of the forces due to the water jet should be investigated.

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Paper A

Significance of Non-linearity and Component-internal Vibrations in an Exhaust System

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Significance of Non-linearity and Component-internal Vibrations in an Exhaust System

Thomas L Englund, Johan E Wall, Kjell A Ahlin, Göran I Broman

Abstract

To facilitate overall lay-out optimisation inexpensive dynamics simulation of automobile exhaust systems is desired. Identification of possible non-linearity as well as finding simplified component models is then important. A flexible joint is used between the manifold and the catalyst to allow for the motion of the engine and to reduce the transmission of vibrations to the rest of the exhaust system. This joint is significantly non-linear due to internal friction, which makes some kind of non-linear analysis necessary for the complete exhaust system. To investigate the significance of non-linearity and internal vibrations of other components a theoretical and experimental modal analysis of the part of a typical exhaust system that is downstream the flexible joint is performed. It is shown that non-linearity in this part is negligible. It is also shown that shell vibrations of the catalyst and mufflers as well as ovalling of the pipes are negligible in the frequency interval of interest. The results implies, for further dynamics studies, that the complete system could be idealised into a linear sub-system that is excited via the non-linear flexible joint, that the pipes could be modelled with beam elements and that the other components within the linear sub-system could also be modelled in a simplified way. Such simplified component models are suggested. The agreement between theoretical and experimental results is very good, which indicates the validity of the simplified modelling.

Keywords: *Correlation, Dynamics, Exhaust system, Linear sub-system, Modal analysis, Non-linear joint.*

1 Introduction

There is a trend to use more computer simulations in the design of products. This is mainly due to demands on shortened time to market, higher product performance and greater product complexity. To be useful in the design process it is important that the simulation models are kept as simple as possible while still being accurate enough for the characteristics they are supposed to describe. To reveal weaknesses in the simulation models experimental investigation is often necessary. The simulation models can then be updated to better correlate with experimental results.

This study is a part of a co-operation project between the Department of Mechanical Engineering at the Blekinge Institute of Technology, Karlskrona, Sweden and Faurecia Exhaust Systems AB, Torsås, Sweden. The overall aim of the project is to find a procedure for effectively modelling and simulating the dynamics of customer-proposed exhaust system lay-outs at an early stage in the product development process, to support the dialogue with the costumer and for overall lay-out optimisation. Demands on, for example, higher combustion temperatures, reduced emissions, reduced weight, increased riding comfort and improved structural durability have made the design of exhaust systems more delicate over the years and more systematic methods have become necessary.

Examples of linear studies of exhaust systems are the works by Belingardi and Leonti [1] and Ling et al. [2], who focus on simulation models, and the work by Verboven et al. [3] who focus on experimental analysis. An introductory study of the present exhaust system is that of Myrén and Olsson [4].

Most modern cars have the engine mounted in the transverse direction. A flexible joint between the manifold and the rest of the exhaust system is therefore included to allow for the motion of the engine and to reduce the transmission of vibrations to the rest of the exhaust system. Recent suggestions of a stiffer attachment of the exhaust system to the chassis, as discussed by for example DeGaspari [5], with the purpose of reducing weight, makes this joint even more important. The commonly used type of joint is significantly non-linear due to internal friction, which makes some kind of non-linear dynamics analysis necessary for the complete system.

Thus a more comprehensive approach seems necessary. This paper represents an early step and focuses on the part of the exhaust system that is downstream the flexible joint. The purpose is to verify the assumption that this part is essentially linear so that, in the further studies, the complete system could be idealised into a linear sub-system that is excited via the non-linear flexible joint. The purpose is also to find a computationally effective and experimentally verified finite element (FE) model of this linear sub-system. This includes simplified modelling of the components.

2 Exhaust System Design and Excitation

The studied automobile exhaust system is shown in figure 1. The mass of the system is about 22 kg and it has a length of approximately 3.3 m.

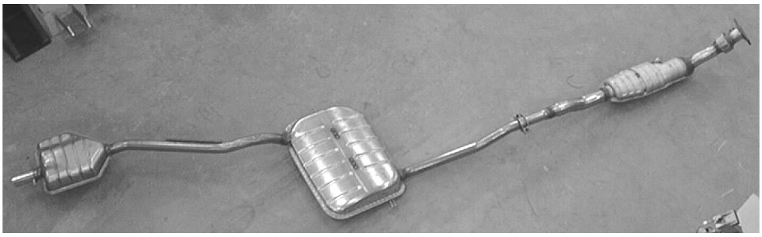

Figure 1. The studied exhaust system.

The system consists of a front assembly and a rear assembly connected with a sleeve joint. Both are welded structures of stainless steel. The front part is attached to the manifold by a connection flange. The engine and manifold are not included in the study.

Between the manifold and the catalyst there is a flexible joint, consisting of a bellows expansion joint combined with an inside liner and an outside braid. This joint is significantly non-linear due to internal friction. More information on this type of joint is given by, for example, Cunningham et al. [6] and Broman et al. [7].

The front assembly, see figure 2, consists of this joint, the catalyst and pipes.

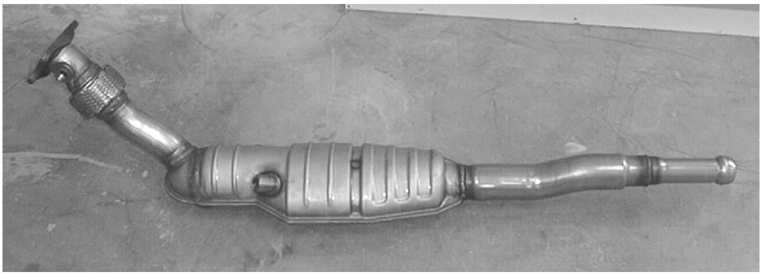

Figure 2. Front assembly.

The catalyst includes a honeycomb ceramic and the outside shell structure is rather complicated. Thus, detailed modelling would be computationally expensive.

The rear assembly, see figure 3, consists of pipes, an intermediate muffler and a rear muffler.

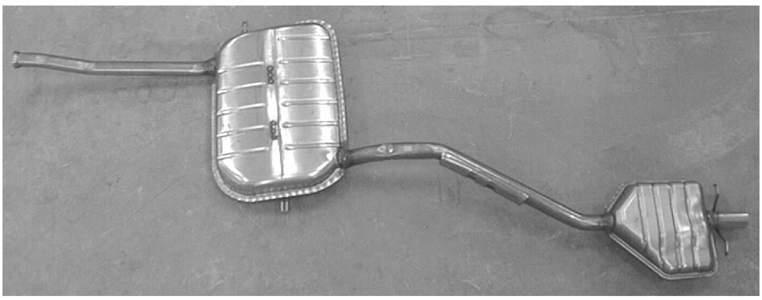

Figure 3. Rear assembly.

Perforated pipes pass through the mufflers. The mufflers are filled with sound silencing material. Their outside shell structure is also rather complicated.

Besides the connection to the manifold the exhaust system is attached to the chassis of the car by rubber hangers. Two hanger attachments are placed at the intermediate muffler and a third is placed just downstream the rear muffler.

The frequency interval of interest for the modal analysis is obtained by considering that a four-stroke engine with four cylinders gives its main excitation at a frequency of twice the rotational frequency. Usually the rotational speed is below 6000 rpm. Excitation at low frequencies may arise due to road irregularities, as discussed by, for example, Belangardi and Leonti [1] and Verboven et al. [3]. Thus, the interval is set to 0-200 Hz.

Free-free boundary conditions are generally desired to facilitate a comparison between the FE-results and the experimental results. This also makes it possible to easily exclude the influence of the non-linear joint in the present analysis. It is assured that the flexible joint does not have any internal deformations. Thus it will move as a rigid body in the present analysis.

3 Initial Finite Element Model

An initial FE-model of the exhaust system is built in I-DEAS [8]. The outside shell structure of the mufflers and the catalyst are modelled with linear quadrilateral shell elements using the CAD-geometry. The mass of the internal material is distributed evenly to the shell elements. The pipes are modelled using parabolic beam elements. The flexible joint is modelled by stiff beam elements with a fictive density to reflect its mass and mass moment of inertia. Lumped mass elements are used to model the connection flange, attachments for the hangers, nipples and the heat shield. Connection between the beam elements representing the pipes and the shell elements representing the mufflers/catalyst is obtained by rigid elements.

By comparing different mesh densities it is found that approximately 140 beam elements and 1900 shell elements are sufficient. The total number of nodes are approximately 2200. This initial model is used as a basis for determining suitable transducer locations for the experimental modal analysis of the exhaust system.

The natural frequencies are solved for by the Lanczos method with free-free boundary conditions.

4 Experimental Modal Analysis

To sufficiently realise the free-free boundary conditions in the experimental modal analysis (EMA) the exhaust system is suspended, at the hanger attachments and at the connection flange, using soft adjustable rubber bands as shown in figure 4.

Figure 4. The measurement set-up.

From the initial FE-analysis it is known that the motion is mainly in the plane (y-z) perpendicular to the length-direction (x) of the system. To be able to excite the system in both the y- and z- directions in one set-up the shaker is inclined. After consulting the FE-model several possible excitation points are tested. The final excitation point is taken just upstream the intermediate muffler, as seen in figure 4. The shaker is connected to the exhaust system via a stinger and a force transducer. A burst random signal is used to excite the exhaust system to avoid possible leakage problems. An HP VXI measuring system with 16 available channels is used. Five triaxial accelerometers could therefore be used in each measuring round. The accelerometers are attached on top of the exhaust system. To minimise the influence of the extra mass loading the accelerometers are evenly spread over the exhaust system in each measuring round.

Again considering the results from the initial FE-model it is concluded that 25 evenly distributed measuring points should be sufficient to represent the mode shapes in the frequency interval of interest. Using the AutoMAC, see figure 5, the chosen measurement points are checked to avoid spatial aliasing. The small off-diagonal terms in the AutoMAC indicate that the chosen measurement points sufficiently well describe the modes in the frequency interval of interest.

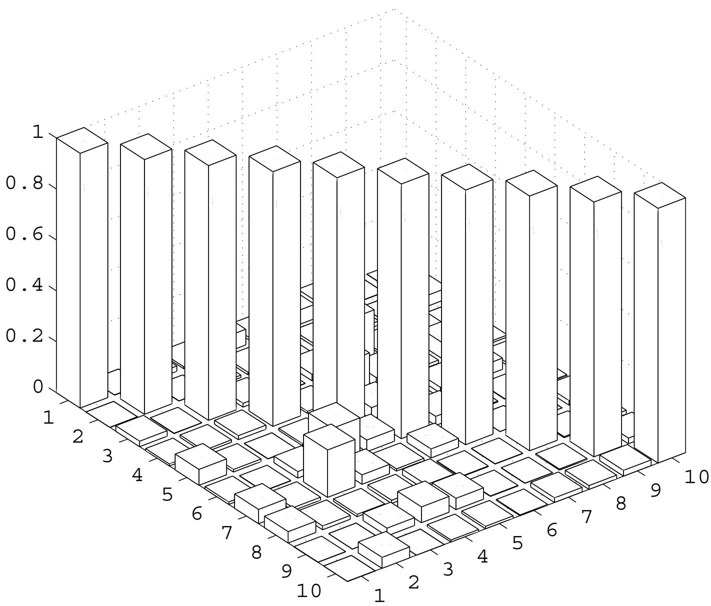

Figure 5. The AutoMAC-matrix.

The quality of the experimental set-up is further assured by a linearity check, a reciprocity check and by investigating the driving point frequency response function (FRF). Also the coherence of some arbitrary FRFs is investigated. All the quality checks show satisfactory results.

Due to the long and slender geometry of the exhaust system concerns may arise that the static preload could have an undesired influence when the system hang horizontally. To ensure that this is not the case the exhaust system is also hanged vertically and some arbitrary FRFs are measured. The difference in natural frequencies is negligible between the two set-ups and it is therefore concluded that the initial set-up is satisfactory.

I-DEAS Test [9] is used to acquire the FRFs. The FRFs are exported to MATLAB [10] where they are analysed using the experimental structural dynamics toolbox developed by Saven Edutech AB [11]. The poles are extracted in the time domain using a global least square complex exponential method. The residues are found using a least squares frequency domain method. To improve the quality of the extracted modal parameters only data in the *y*- and *z*-directions are used. To get as good a fit as possible the curve fitting procedure is conducted in two steps; first in the interval 5-90 Hz and then in the interval 90-150 Hz. Above 150 Hz no significant modes are found, as seen in a typical FRF shown in figure 6.

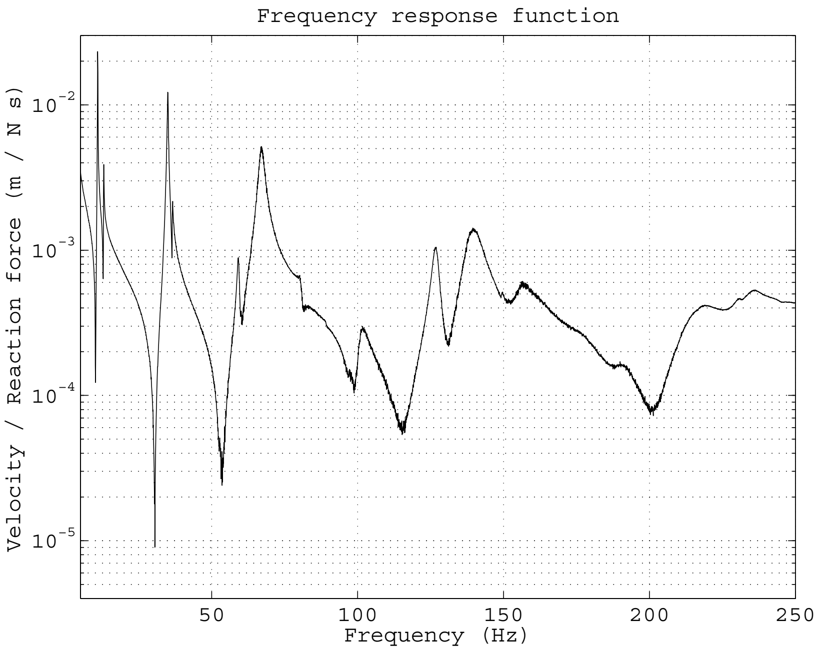

Figure 6. Typical FRF.

5 Simplification and Correlation

Determining the natural frequencies of the mufflers and the catalyst experimentally it is seen that no significant local modes are present in the frequency interval of interest. This was also found by Verboven et al. [3]. Therefore the modelling of the mufflers and the catalyst, which are responsible for most of the model size in the initial FE-model, can be significantly simplified. The mufflers and the catalyst are modelled by lumped mass and mass moment of inertia elements. The properties of these elements are obtained from the original FE-model. If more suitable in a general case these properties can also be obtained directly from the CAD-model or experimentally. The lumped mass and mass moment of inertia elements are connected to the beam elements representing the pipes by rigid elements.

The natural frequencies of the pipes are also investigated experimentally. No significant ovalling modes are found in the frequency interval of interest, which confirms the validity of modelling the pipes by beam elements.

To simulate the flexibility of the connections between the pipes and mufflers/catalyst, short beam elements with individual properties are used. These elements are located at the true connection locations, that is, with reference to the real system. Thus, they are placed between the rigid elements that are connected to the lumped mass and mass moment of inertia elements and the beam elements representing the pipes.

These individual beam properties are updated so that the difference between theoretical and experimental results is minimised. The updating procedure uses MATLAB [10] and ABAQUS [12] and is described in an accompanying paper (Englund et al. [13]).

The updated FE-model has approximately 200 nodes. Thus a reduction of over 90 % compared to the initial FE-model is obtained. Simplifications of this type are important if direct time integration becomes necessary for the non-linear dynamics analysis of the complete system. It is also important when a large number of simulations are necessary for overall exhaust system lay-out optimisation.

The FE modes are calculated without consideration of damping and are therefore real-valued. To be able to compare these modes with the modes obtained experimentally, which are complex due to damping, the experimental modes are converted into real-valued modes.

6 Results and Correlation

To correlate the mode shapes from the updated FE-model and the experimental mode shapes a MAC-matrix is calculated, see figure 7.

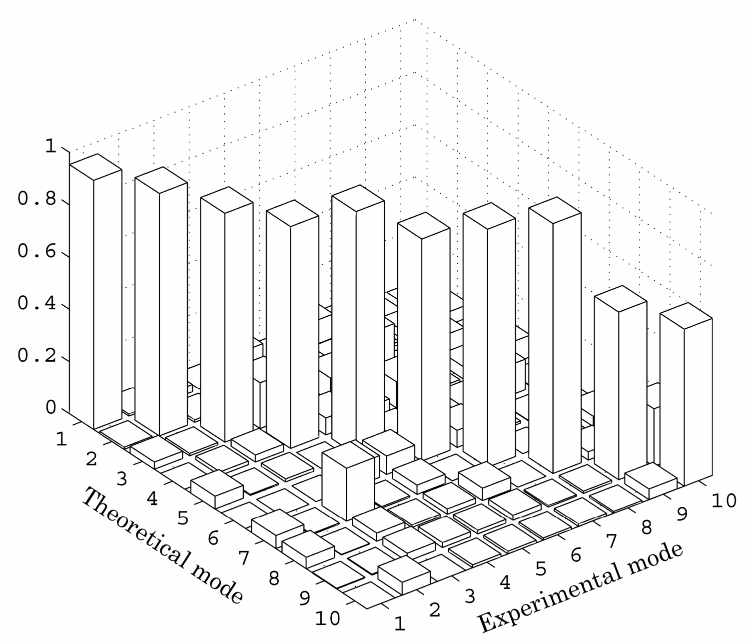

Figure 7. The MAC-matrix.

Except for mode nine and ten the diagonal MAC-values are above 0.85, which indicates good correlation. All the off-diagonal values in the MAC-matrix are below 0.2. This indicates that the different mode shapes are non-correlated.

A comparison between theoretical and experimental natural frequencies is shown in figure 8. The 45-degree line represents perfect matching. The crosses indicate the frequency match for each correlated mode pair.

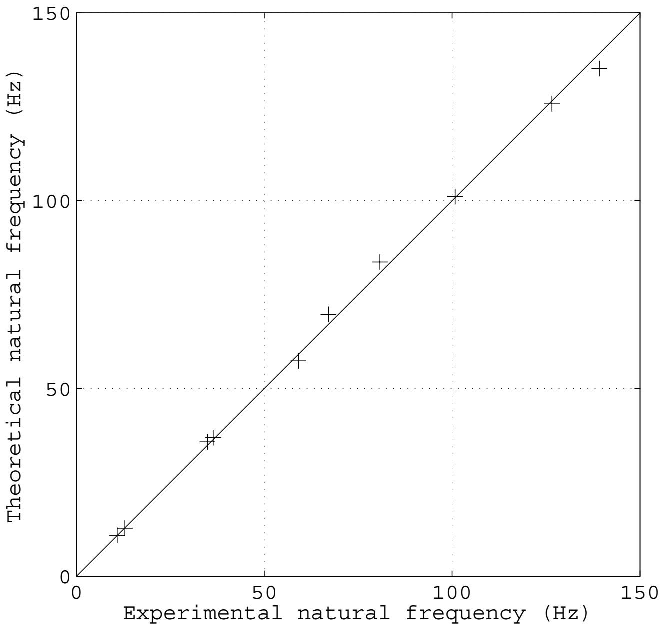

Figure 8. Theoretical and experimental natural frequencies.

The maximum difference in corresponding natural frequencies is below four per cent. The small and randomly distributed scatter of the plotted points is normal for this type of modelling and measurement process [14].

The results are summarised in table 1.

Table 1. Results.

Mode	Experimental Frequency (Hz)	Experimental Damping (%)	Theoretical Frequency (Hz)	Correlation[a] (%)	MAC
1	10.9	0.32	10.9	0.24	0.95
2	12.9	0.52	12.8	-1.0	0.93
3	34.9	0.49	35.8	2.6	0.88
4	36.4	0.30	36.9	1.3	0.85
5	59.1	0.69	57.3	-3.0	0.93
6	67.1	1.5	69.7	3.9	0.85
7	80.8	0.79	83.7	3.6	0.91
8	101	1.6	101	0.30	0.96
9	127	0.91	126	-0.60	0.64
10	139	2.3	135	-2.9	0.60

[a] The correlations are calculated before rounding off.

The damping values are given as the fraction of critical damping and the correlation value is the relative difference between experimental and theoretical natural frequencies. Above 150 Hz no significant modes are found.

7 Conclusions

A dynamics study of an exhaust system that consists of a non-linear flexible joint and a main part including pipes, mufflers and a catalyst is presented. The good agreement between the theoretical and experimental modal analysis, as well as the satisfactory results of the linearity check, implies, for further dynamics studies, that the complete system could be idealised into a linear sub-system that is excited via the non-linear flexible joint.

It is also shown that shell vibrations of the catalyst and mufflers as well as ovalling of the pipes are negligible in the frequency interval of interest. This implies that the pipes could be modelled by beam elements and that the other components within the linear sub-system could be modelled by lumped mass and mass moment of inertia elements. The mass and inertia properties can be obtained either from a CAD-model or experimentally. Short beam elements with individual properties can be used successfully to model the flexibility of the connections between the mufflers/catalyst and the pipes. Automated updating of these individual properties is recommended since doing it manually is time consuming and difficult.

The agreement between results from the updated FE-model and the experimental investigations is very good. This implies that such simplified modelling is a valid approach and it may turn out important in coming non-linear analyses, since such analyses are often computationally expensive.

Acknowledgements

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Paper B

Automated Updating of Simplified Component Models for Exhaust System Dynamics Simulations

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Automated Updating of Simplified Component Models for Exhaust System Dynamics Simulations

Thomas L Englund, Johan E Wall, Kjell A Ahlin, Göran I Broman

Abstract

To facilitate overall lay-out optimisation simplified component models for dynamics simulations of automobile exhaust systems are desired. Such optimisation could otherwise be computationally expensive, especially when non-linear analyses are necessary. Suggestions of simplified models of the mufflers and the catalyst are given. To account for the flexibility at the connections between those components and the pipes short beam elements with individual properties are introduced at these locations. An automated updating procedure is developed to determine the properties of these beam elements. Results from an experimental modal analysis are used as the reference. The theoretical model of the exhaust system is built in the finite element software ABAQUS. The updating procedure uses the sequential quadratic programming algorithm included in the Optimization Toolbox of the software MATLAB to minimise the sum of the differences between experimentally and theoretically obtained natural frequencies. Constraints are used on the correlation between the experimentally and theoretically obtained mode shapes by considering the MAC-matrix. Communication between the two software packages is established by an in-house MATLAB script. The correlation between results from the updated theoretical model and the experimental results is very good, which indicates that the updating procedure works well.

Keywords: *Correlation, Dynamic, Exhaust system, Modal analysis, Optimisation, Updating.*

1 Introduction

Demands on shortened time to market, higher product performance and greater product complexity in combination with the fast development of computers have resulted in more simulations for prediction and evaluation of product performance. Simplified modelling and inexpensive simulation procedures are often desired early in the product development process to study certain product characteristics and for overall introductory systems optimisation. The models and simulations should reflect the interesting characteristics of the real system accurately enough to support relevant design decisions. To gain confidence of this some kind of experimental verification is often necessary. If the correlation is not good enough the models need to be updated. Doing this manually is usually a time consuming and difficult task, especially if there are many parameters to be updated in the theoretical models.

Procedures for more automated updating have therefore attained interest within the analysis community. See for example the works by Van Langenhove et al. [1] and Deweer et al. [2] regarding updating of dynamic systems. Avitabile [3] discusses different updating criteria and points out the importance of the choice of parameters in the updating procedure. Chen and Ewins [4] describe the effect of discretisation errors when updating finite element models.

This study is a part of a co-operation project between the Department of Mechanical Engineering at the Blekinge Institute of Technology, Karlskrona, Sweden and Faurecia Exhaust Systems AB, Torsås, Sweden. The overall aim of the project is to find a procedure for effectively modelling and simulating the dynamics of customer-proposed exhaust system lay-outs at an early stage in the product development process, to support the dialogue with the costumer and for overall lay-out optimisation. An accompanying paper is that of Englund et al. [5], which focuses on simplified and experimentally verified modelling of a typical automobile exhaust system. The updating of the simplified models of the components within that system is performed according to the procedure described in the present paper. The MATLAB Optimization Toolbox [6] is used for the updating procedure and ABAQUS [7] is used to solve for the natural frequencies and mode shapes. Communication between the two different software packages is established by an in-house MATLAB script to obtain automated updating.

2 Exhaust System Design

The studied automobile exhaust system is shown in figure 1. The mass of the system is about 22 kg and it has a length of approximately 3.3 m.

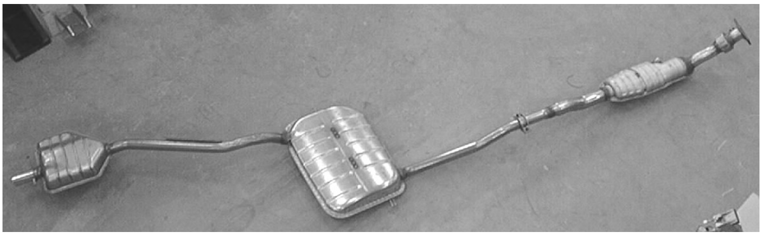

Figure 1. The studied exhaust system.

The system consists of a front assembly and a rear assembly connected with a sleeve joint. Both are welded structures of stainless steel. The front part is attached to the manifold by a connection flange. The engine and manifold are not included in the study.

Between the manifold and the catalyst there is a flexible joint. This joint is significantly non-linear due to internal friction. More information on this type of joint is given by, for example, Broman et al. [8] and Cunningham et al. [9].

The front assembly consists of this joint, the catalyst and pipes. The rear assembly consists of pipes, an intermediate muffler and a rear muffler. Perforated pipes pass through the mufflers. The mufflers are filled with sound silencing material.

Besides the connection to the manifold the exhaust system is attached to the chassis of the car by rubber hangers. Two hanger attachments are placed at the intermediate muffler and a third is placed just downstream the rear muffler, see figure 1.

3 Theoretical and Experimental Analysis

A theoretical model of the exhaust system is built in ABAQUS [7]. The pipes are modelled using quadratic beam elements and the mufflers and the catalyst are modelled using lumped mass and mass moment of inertia elements. Such simplified modelling is valid in the frequency interval of interest [5]. These elements are connected to the beam elements representing the pipes by rigid elements. The properties of the lumped mass and mass moment of inertia elements are obtained from a finite element (FE) model where these parts are modelled with shell finite elements [5]. If more suitable in a general case these properties can also be obtained directly from the CAD-model or experimentally.

To simulate the flexibility of the connections between the pipes and mufflers/catalyst, short beam elements with individual properties are used. These elements are located at the true connection locations, that is, with reference to the real system. Thus, they are placed between the rigid elements that are connected to the lumped mass and mass moment of inertia elements and the beam elements representing the pipes.

Lumped mass elements are used to model the connection flange attached to the flexible joint, attachments for the hangers and the heat shield. Free-free boundary conditions are used and the natural frequencies and mode shapes are solved for by the Lanczos method. More information about the theoretical model can be found in [5].

The results from the theoretical model are compared with natural frequencies and mode shapes obtained experimentally. The theoretical mode shapes are calculated without consideration of damping and are therefore real-valued. To be able to compare these modes with the modes obtained experimentally, which are complex due to damping, the experimental modes are converted into real-valued modes.

The experimental modal analysis (EMA) is performed using free-free boundary conditions. To exclude the influence of the non-linearity of the flexible joint it is assured that it does not have any internal deformations. Thus it will move as a rigid body in the present analysis. More about the EMA can be found in [5].

The frequency interval of interest is 0-200 Hz but actually no significant modes occur above 150 Hz for this particular exhaust system [5].

4 Updating

The experimentally obtained natural frequencies and mode shapes are used to update the theoretical model. If, in a general case, a full physical prototype does not exist results from a detailed finite element model can be used as the reference.

The selection of parameters to be included in the updating procedure is important. This is true whether the updating is based on frequency differences, mode shape differences or frequency responses [3]. Except for the connections between the mufflers/catalyst and the pipes the theoretical model of the exhaust system is straightforward. Properties influencing the flexibility (stiffness) of these connections are used when updating the theoretical model. There are six connections marked in figure 2 and 3. Each of them includes the following three stiffness related properties; the two area moments of inertias and the polar area moment of inertia of the short beam elements representing the connections. All connections have individual properties. Altogether this gives 18 independent parameters to consider when updating the theoretical model.

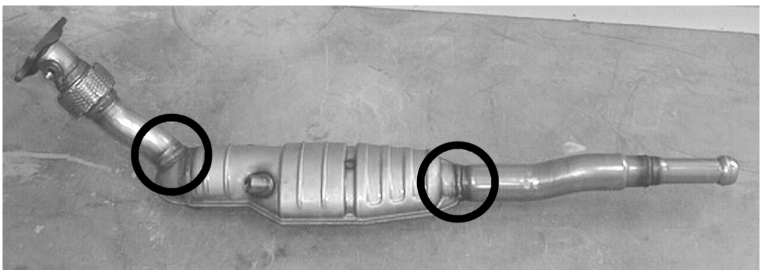

Figure 2. Front assembly.

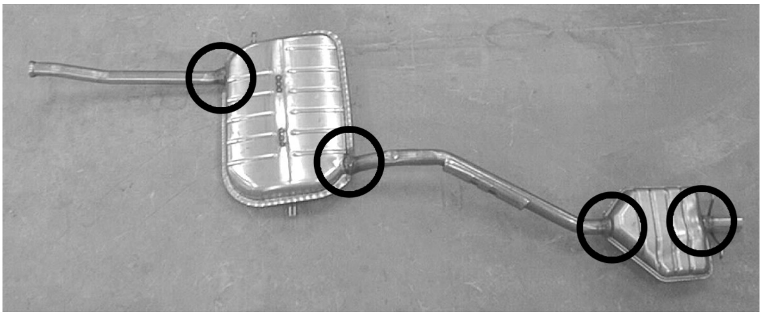

Figure 3. Rear assembly.

To sort out the important ones, a simple parameter study is performed. The parameters are modified by a factor ten, one at a time, and the natural frequencies are calculated. It can then roughly be concluded which parameters that are important to consider when updating the theoretical model of the exhaust system. Ten parameters are found to be significantly more important than the others. Using this approach the possibility to detect interaction between parameters is lost. Considering also these effects can be very time consuming. The procedure used in this work is a compromise between accuracy and time consumption. The aim is not necessarily to find the global optimum, but rather a solution that is good enough. Since many parameters are still involved an automated updating procedure using the Optimization Toolbox in MATLAB is developed. A constrained optimisation is performed using a sequential quadratic programming (SQP) algorithm [6]. The optimisation algorithm is supplied with start-values, bounds, constraints and optimisation criterion. The optimisation criterion chosen, which is to be minimised, is the sum of the differences in natural frequency within each correlated mode pair. Constraints are used on the correlation between theoretical and experimental mode shapes using the diagonal values of the MAC-matrix. The modal assurance criterion (MAC) is a technique to quantify the correlation between two sets of mode shapes. This constraint is important since it forces the algorithm to use correlated mode pairs when calculating the optimisation criterion. Good agreement is sought for both natural frequencies and mode shapes. Using constraints and bounds limits the search space, which usually reduces the number of function evaluations, that is, the problem converges faster [6].

Since natural frequencies and mode shapes must be solved for many times during the updating procedure ABAQUS and MATLAB interact with each other. An in-house MATLAB script, taking advantage of MATLAB's ability of reading and writing ASCII-files, is used to transfer data between the two different software packages. The optimisation procedure is schematically shown in figure 4.

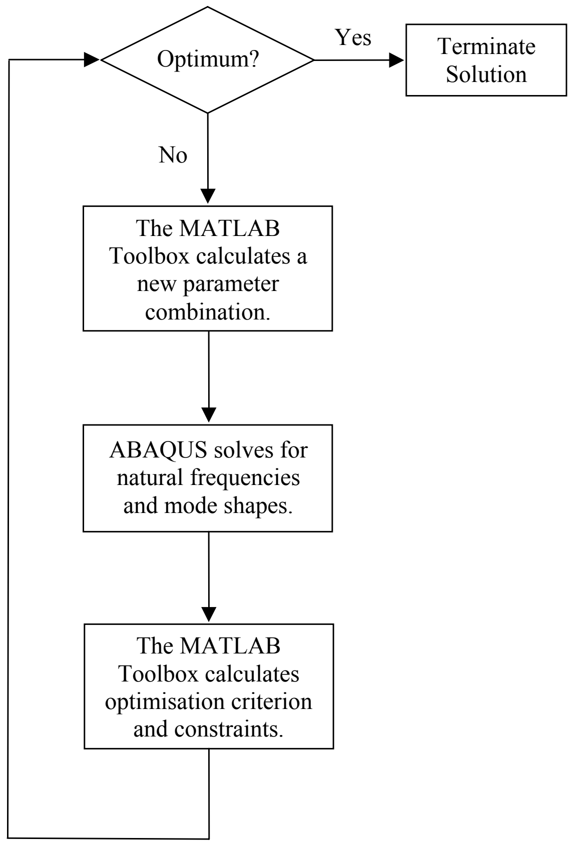

Figure 4. Automated updating procedure.

Setting appropriate tolerances for the search algorithm in the Optimization Toolbox is not a trivial task. It usually has to be tuned for specific problems. Setting the tolerances to tight forces the algorithm to make a large number of function evaluations without finding a much better solution. On the other hand setting them to loosely the search algorithm might not find the correct optimum. To be able to set the tolerances for the optimisation algorithm in a straightforward way all the ten parameters are scaled to be between zero and unity.

An important aspect to consider is that SQP is a gradient-based optimisation routine. This means that it only finds local optima, that is, different optima can be found depending on the start-values. Some kind of multi-start procedure can be used to reduce this problem. Another way is to use some kind of derivate-free optimisation method. In this work the start-values for the short beam element properties are taken from the beam elements representing the pipes at the connections in the theoretical model. If the start-values are good, that is, are near an optimum, the search algorithm finds this optimum faster.

5 Results and Discussion

A comparison between the results from an initial theoretical model, that is, a model without the short beam elements accounting for the flexibility at the connections between the mufflers/catalyst and the pipes, and the experimental results shows that this model is far too stiff. Some of the theoretical natural frequencies are more than fifty per cent higher than the corresponding natural frequencies obtained experimentally.

In a first step to achieve a theoretical model that correlates better with the experimental results Young's modulus, of the fictive material of the short beam elements representing the connections, is updated. The same value of this parameter is used for all connections. The comparison between results from this roughly updated model, and the experimental results are summarised in table 1. The correlation is still not considered good enough.

As seen in table 1 mode six and seven is not correlating. This is due to a mode switch between these modes, see figure 5. Furthermore, it can be seen in the figure that some of the off-diagonal values are high. This also indicates bad correlation.

Table 1. Results after the first update.

Mode	Experimental Frequency (Hz)	Theoretical Frequency (Hz)	Correlation[a] (%)	MAC
1	10.9	10.6	-2.4	0.95
2	12.9	13.1	1.2	0.93
3	34.9	35.9	2.8	0.58
4	36.4	42.1	16	0.67
5	59.1	50.0	-15	0.84
6	67.1	74.6	11	
7	80.8	82.9	2.6	
8	101	86.2	-14	0.91
9	127	116.5	-7.9	0.72
10	139	141.2	1.5	0.70

[a] The correlations are calculated before rounding off.

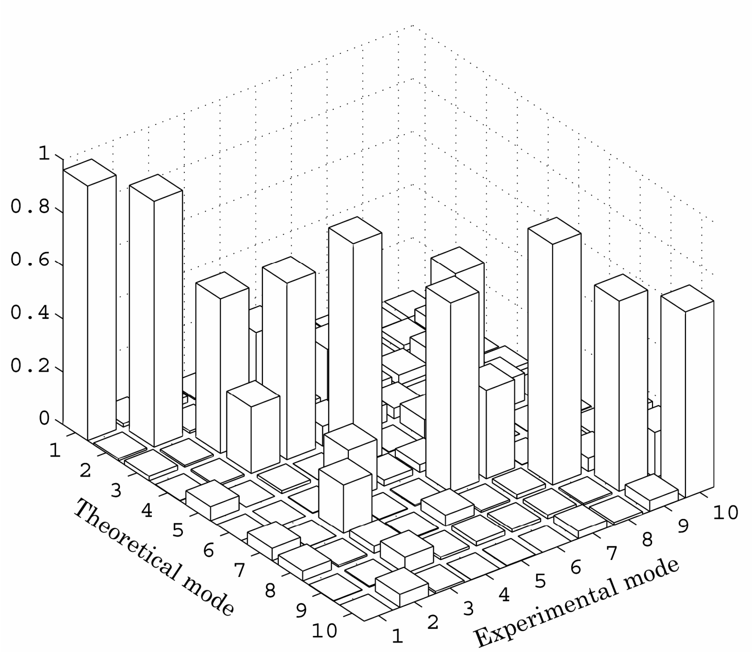

Figure 5. The MAC-matrix after the first update.

In a final step the ten independent parameters are included in the automated updating procedure. The correlation between modes of this theoretical model and the experimental modes are calculated using the MAC-matrix, see table 2 and figure 6.

The correlation is very good. All diagonal MAC values are above 0.85 except for mode nine and ten. Furthermore all the off-diagonal terms in the MAC-matrix are below 0.2. As also seen all differences in natural frequencies are below four per cent.

Table 2. Results after the final update.

Mode	Experimental Frequency (Hz)	Theoretical Frequency (Hz)	Correlation[a] (%)	MAC
1	10.9	10.9	0.24	0.95
2	12.9	12.8	-1.0	0.93
3	34.9	35.8	2.6	0.88
4	36.4	36.9	1.3	0.85
5	59.1	57.3	-3.0	0.93
6	67.1	69.7	3.9	0.85
7	80.8	83.7	3.6	0.91
8	101	101	0.30	0.96
9	127	126	-0.60	0.64
10	139	135	-2.9	0.60

[a] The correlations are calculated before rounding off.

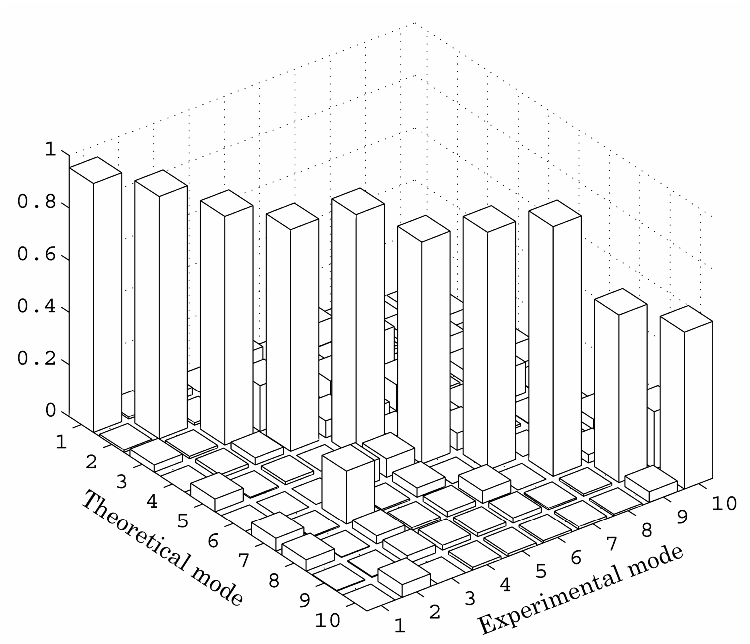

Figure 6. The MAC-matrix after the final update.

6 Conclusions

Updating of simplified component models for simulation of the dynamic behaviour of an automobile exhaust system is the subject of this paper. Results obtained from an experimental modal analysis are used as the reference. If, in a general case, a full physical prototype does not exist results from a detailed finite element model can be used as the reference.

The simplified component models can be used for, otherwise computationally expensive, overall lay-out optimisation and they can also be re-used when the same or similar components are to be included in other exhaust system assemblies.

An automated updating procedure is developed. The sequential quadratic programming algorithm in MATLAB's Optimization Toolbox is used to minimise the difference between theoretical and experimental natural frequencies. Constraints are used on the correlation between the theoretical and experimental mode shapes using the MAC-matrix. The natural frequencies and mode shapes are solved for by ABAQUS. Communication between the two software packages is established by an in-house MATLAB script.

The very good correlation between the updated theoretical model and the experimental results shows that the updating procedure works well.

Acknowledgements

The support from Faurecia Exhaust Systems AB is gratefully acknowledged, especially from Håkan Svensson. The authors also gratefully acknowledge the financial support from the Swedish Foundation for Knowledge and Competence Development.

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Paper C

Modelling of Multi-ply Bellows Flexible Joints of Variable Mean Radius

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Modelling of Multi-ply Bellows Flexible Joints of Variable Mean Radius

Johan E Wall, Thomas L Englund, Kjell A Ahlin, Göran I Broman

Abstract

Bellows flexible joints are included in automobile exhaust systems to allow for engine movements and thermal expansion and to reduce vibration transmission. Generally the joint consists of a flexible bellows, an inside liner and an outside braid. In this work the bellows is considered. A straightforward way to model the bellows is to use shell finite elements. Due to the convoluted geometry of the bellows that procedure requires however a high number of elements, meaning that the bellows model would constitute a large part of the model of the exhaust system. For more effective dynamics simulations a beam finite element representation of the bellows has been presented in a prior work. This modelling procedure was implemented in the commercial software I-DEAS and was verified against experimental results available in the literature for single-ply bellows of constant mean radius. This paper suggests adjustments by which this procedure can be extended to model also multi-ply bellows of variable mean radius. Experimental investigations of a double-ply bellows having decreasing mean radius towards its ends are included for verification. The agreement between theoretical and experimental results is very good, implying that the suggested extension of the modelling procedure is valid. It is also shown that the procedure can easily be implemented into other commercial software (in this case ABAQUS). The experimental investigation reveals an intriguing resonance frequency shift at small excitation force levels. Although considered to be of minor significance for the present application of the bellows, a hypothetic qualitative explanation to the observed phenomenon is given.

Keywords: *Beam model, Bellows, Dynamic, Experimental investigation, Flexible joint, Frequency shift, Multi-ply, Variable mean radius.*

Notation

A	Area [m^2]
E	Modulus of elasticity [Pa]
G	Shear modulus [Pa]
h	Height [m]
I	Area moment of inertia [m^4]
K	Polar area moment of inertia [m^4]
L	Length [m]
R	Radius [m]
r	Radius [m]
t	Thickness [m]
ν	Poisson's ratio
ρ	Density [Kg/m^3]

Indices

conv	Convolution
m	Middle
p	Pipe

1 Introduction

Bellows flexible joints are important components in automobile exhaust systems. A flexible connection between the manifold and the rest of the exhaust system is necessary to allow for deflections induced by engine movements and due to thermal expansion and to reduce vibration transmission. Recent suggestions of a stiffer attachment of the exhaust system to the chassis, as discussed by for example DeGaspari [1], with the purpose of reducing weight, makes this component even more important.

Proper dimensioning of the flexible joint requires understanding of its dynamic characteristics and interaction with the rest of the exhaust system. This is studied in a co-operation project between the Department of Mechanical Engineering at Blekinge Institute of Technology, Karlskrona, Sweden and Faurecia Exhaust Systems AB, Torsås, Sweden. The overall aim of the project is to find a procedure for effectively modelling and simulating the dynamics of customer-proposed exhaust system lay-outs at an early stage in the product development process, to support the dialogue with the costumers and for overall lay-out optimisation. To be suited for that the simulation procedure cannot be too computationally expensive. This is especially important when the dynamics is non-linear, which will be considered in later studies. The models of the components of the exhaust system must therefore be as simple as possible while still giving a proper description of the dynamics of the system. The bellows flexible joint is the component within the exhaust system that is most difficult to describe inexpensively.

Broman et al. [2] presented a method for determining the dynamic characteristics of single-ply bellows of constant mean radius by manipulated beam finite elements of commercial software based on the assumption that the bellows is linear. Compared to a shell elements model, which would be the most straightforward way of modelling the bellows, the model size is reduced considerably by using this beam element model. Axial, bending and torsion degrees of freedom can be studied simultaneously and the modelling technique facilitates the interaction between the bellows and the rest of the exhaust system, usually also modelled by finite elements. A short historical background and further references on bellows studies can be found in [2].

In this paper it is investigated if the beam element procedure can be extended to model also a multi-ply bellows of variable mean radius. Experimental investigations are performed for verification. It is also tested if the procedure can easily be implemented in other commercial software (in this case ABAQUS [3]) than the one used in [2] (I-DEAS [4]).

2 Basic Design of Flexible Joint and Excitation

The basic design of the flexible joint is shown in figure 1. It consists of a gas-tight bellows combined with an inside liner and an outside braid. The liner was originally introduced for reduction of bellows temperature and for improved flow conditions. It also further reduces vibrations. The braid is used for mechanical protection and to limit the extension of the joint. The parts are connected with end-caps. The complete joint is significantly non-linear. In this paper the bellows is considered. More information on this type of joint is given by, for example, Cunningham et al. [5].

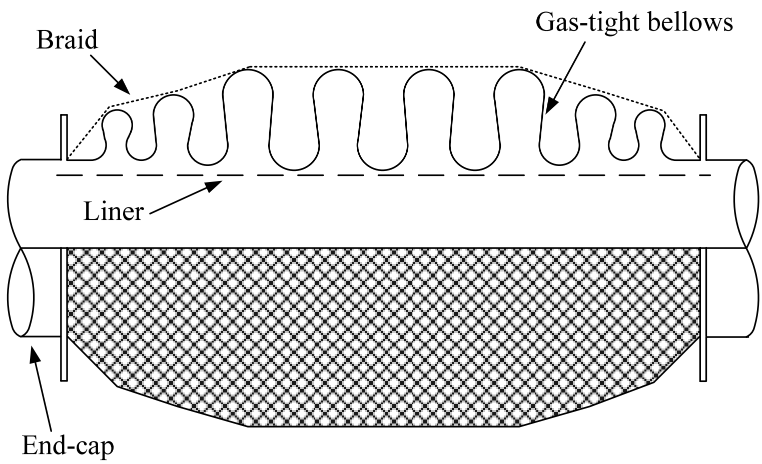

Figure 1. Basic flexible joint design.

The bellows of this paper is double-plied and it has smaller mean radius closer to the ends. For a given strength a multi-ply bellows has lower stiffness than a single-ply bellows. Low stiffness is desired to decouple the engine from the rest of the exhaust system.

The frequency interval of interest for the analysis is obtained by considering that a four-stroke engine with four cylinders gives its main excitation at a frequency of twice the rotational frequency. Usually the rotational speed is below 6000 rpm. Excitation at low frequencies may arise due to road irregularities, as discussed by, for example, Belangardi and Leonti [6] and Verboven et al. [7]. Thus, the interval is set to 0-200 Hz.

3 Modelling of Bellows

Broman et al. [2] described how to model the dynamic characteristics of a bellows using a pipe analogy and by manipulating certain parameters of the beam finite element formulation in the software I-DEAS. This procedure is adopted and extended in this paper.

While the current bellows has a variable mean radius, with smaller radii closer to the ends, different equivalent pipes are used for different parts of the bellows. These equivalent pipes have different equivalent density, ρ_p, shear modulus, G_p, modulus of elasticity, E_p, area, A_p, area moment of inertia, I_p, and polar area moment of inertia, K_p. Three different equivalent pipes, with assumed constant mean radii, are used, see figure 2.

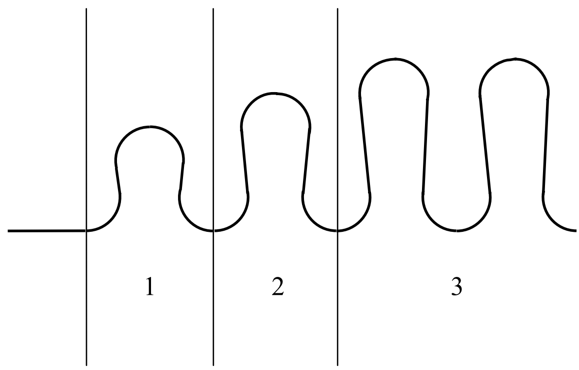

Figure 2. Three sections to be represented by different equivalent pipe models.

Other differences are that the end caps of the bellows are included in the analysis and that the bellows has two plies instead of one ply. Furthermore the convolution profile is slightly different from the U-shaped profile considered in [2].

The bellows is made of stainless steel. The material properties are E = 193 GPa, $\rho = 8000$ kg/m^3, and $\nu = 0.29$.

The characteristic dimensions of the convolutions can be seen in figure 3. The convolution dimensions of the three different pipe sections are presented in table 1.

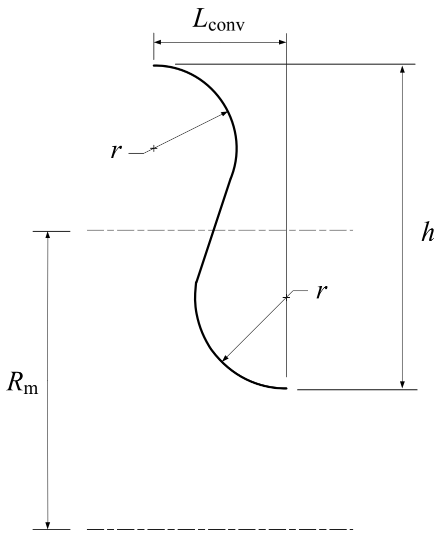

Figure 3. Convolution dimensions.

Table 1. Convolution dimensions (mm).

Dimension	Section		
	1	2	3
R_m	61.2	64.9	67.9
L_{conv}	7.60	7.60	7.60
r	2.20	2.40	2.65
h	6.70	10.4	13.4
t	0.193	0.193	0.193

The thickness of each ply in the bellows is reduced during the forming process. The standard of the Expansion Joint Manufacturing Association, EJMA [8], suggests how to account for this for U-shaped bellows. For the present convolution profile this correction is however insufficient, resulting in a too heavy and stiff bellows. A different approach is therefore used. The mass of the bellows is measured. As the density and the remaining dimensions are known, the material thickness, t, can be calculated. It is assumed that the thickness is constant all over the bellows and the same for each ply.

The equivalent parameters, E_p, G_p, ρ_p, A_p, I_p, and K_p, are calculated in the way described in [2] for the three different equivalent pipes, see table 2. The axial and torsion stiffness are calculated using the linear static solver in I-DEAS considering one ply. These stiffness values are then multiplied by a factor two, to account for the two plies. This means that the plies are assumed to work independently of each other, in agreement with the EJMA-standard [8].

Table 2. Properties of the equivalent pipes.

Property	Section		
	1	2	3
E_p [MPa]	35.6	9.64	7.03
G_p [GPa]	12.4	8.21	6.28
ρ_p [kg/m^3]	7120	$1.02 \cdot 10^4$	$1.29 \cdot 10^4$
A_p [m^2]	$1.92 \cdot 10^{-4}$	$2.04 \cdot 10^{-4}$	$2.13 \cdot 10^{-4}$
I_p [m^4]	$9.00 \cdot 10^{-8}$	$1.07 \cdot 10^{-7}$	$1.23 \cdot 10^{-7}$
K_p [m^4]	$1.80 \cdot 10^{-7}$	$2.15 \cdot 10^{-7}$	$2.46 \cdot 10^{-7}$

As this is an early step in the analysis of the complete flexible joint, in which also the non-linear characteristics of the joint will be considered, an analysis procedure that with small adjustments can handle also this is sought. The analysis is therefore performed in ABAQUS instead of I-DEAS, because direct time integration might be needed when considering also the non-linearity of the flexible joint.

Both I-DEAS and ABAQUS have finite element formulations based on Timoshenko beam theory, which includes the influence of shear deformation and in dynamics the influence of rotary inertia. The latter must be considered for the pipe equivalents of the bellows but the influence of shear deformation on the bellows in bending is very small according to, for example, Morishita et al. [9] and Jakubauskas and Weaver [10], and should therefore be suppressed in the beam formulation.

The model is solved in ABAQUS using the mode-based steady-state dynamics solver. This solver calculates the steady-state displacement amplitude as a function of frequency based on modal superposition. The first step is to extract sufficiently many eigenmodes so that the dynamic response of the system is adequately modelled. To get a realistic result the modal damping of the system is specified. These damping ratios are obtained from experimental results. The damping ratios used in this work are given in table 3.

The model is clamped at one end and is free at the other end. Two different load cases are considered. In the first case the bellows is excited with an axial harmonic force at the free end. In the second case a transverse harmonic force is used. The amplitude of the harmonic force is 0.1 N in both cases. The model is solved with a frequency step of 0.1 Hz. By comparing different mesh densities it is found that 20 linear beam elements are sufficient.

The end-caps are modelled by lumped mass and mass moment of inertia elements. These elements are connected to the beam elements representing the bellows by rigid elements. During the experimental investigation a steel plate is welded on to the free end of the bellows. This plate is modelled with a lumped mass and mass moment of inertia element. The transducers are accounted for by lumped mass elements.

4 Experimental Investigations

A measurement set-up that is useful also for the complete flexible joint in coming studies is desired. A sinusoidal excitation is used because it is well suited for non-linear analysis. The input signal level can be accurately controlled and the signal to noise ratio is good because all the input energy is concentrated at one frequency at the time. The main drawback is that it is time consuming because the measurements are performed frequency by frequency and that time is needed for the test specimen to reach steady state at each frequency.

The signal analyser I-DEAS Test is used [11], which has a function called step sine closed loop control in the sine measurements module. Using this function the amplitude and/or the phase of the excitation force can be controlled.

In the first case the bellows is excited in the axial direction and in the second case it is excited in the transverse direction. An HP VXI measuring system is used to acquire the experimental data. The experimental set-ups for the two different cases are described bellow. Due to the low stiffness of the bellows in the transverse direction it is mounted so that its axial direction coincides with the gravitational force, to avoid undesired influence from gravitation on the dynamic behaviour.

To secure that the experimental set-ups are satisfactory the influence of vibration feedback through the rigid table is investigated. No significant feedback is found.

4.1 Axial Measurements

The bellows is rigidly mounted at one end in a chuck from a lathe, and is excited in the other end. The lathe chuck is mounted in a frame made of steel beams, which is assumed to be rigid. This frame is attached to a rigid table of considerable mass. The shaker is also mounted on this rigid table. The free end is connected to the shaker through a stinger and a force transducer. The measurement set-up can be seen in figure 4.

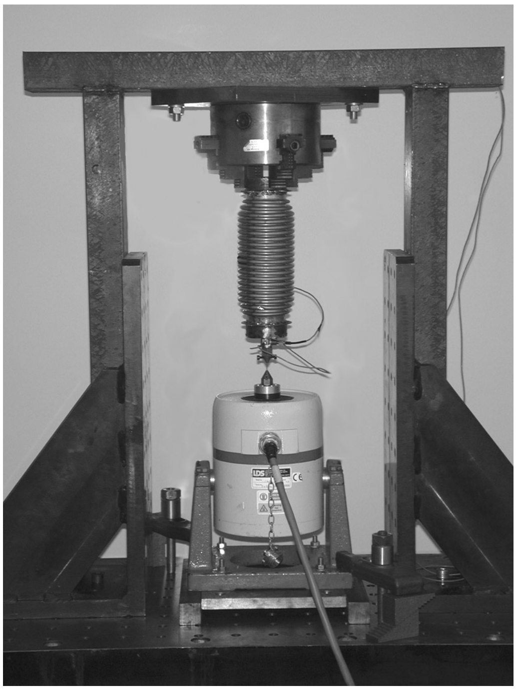

Figure 4. Experimental set-up for the axial measurements.

To be able to excite the bellows in this set-up it has to be slightly modified. A metal plate is welded on to the free end. On this plate an accelerometer and a force transducer are mounted, see figure 5.

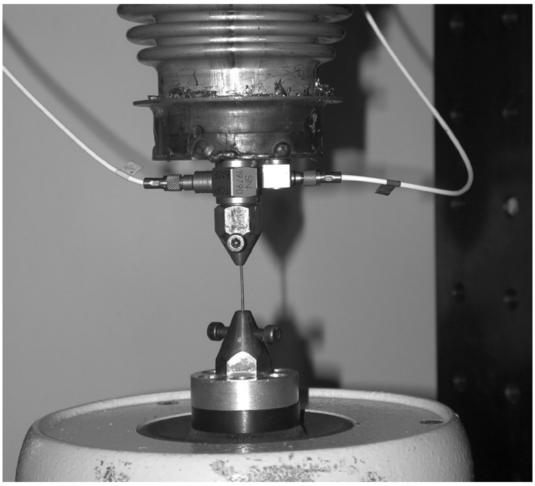

Figure 5. Transducer placement in the axial set-up.

The plate is rigid in the frequency interval of interest. The bellows is excited with a sinusoidal signal with an amplitude of 0.1 N. The frequency step is adapted so that a smaller increment is used near resonance frequencies, where the response changes rapidly.

4.2 Transverse Measurements

The bellows is rigidly mounted at one end in a chuck from a lathe, and is excited in the other end. The lathe chuck is mounted on a rigid table of considerable mass. The shaker is also mounted on this table via a rigid angle bracket made of thick reinforced steel plates. The free end is connected to the shaker through a stinger and a force transducer. An accelerometer is placed on the opposite side of the bellows, see figure 6. The measurement set-up can be seen in figure 7.

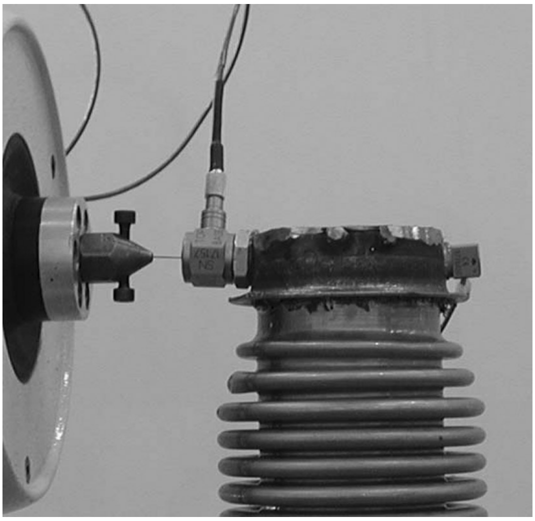

Figure 6. Transducer placement in the transverse set-up.

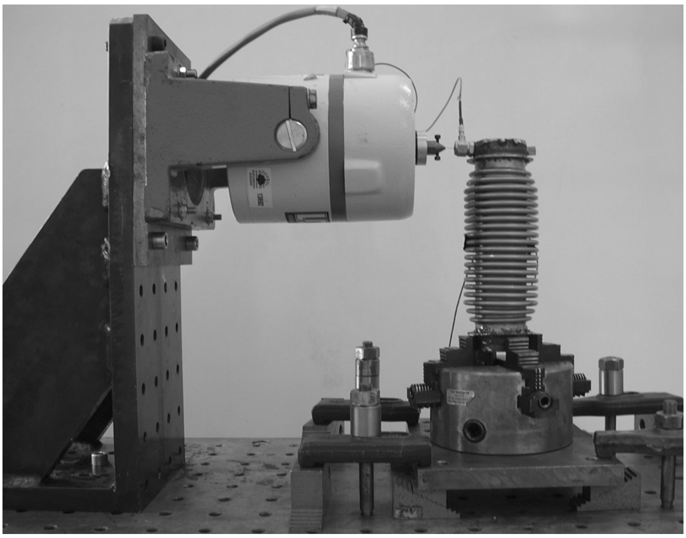

Figure 7. Experimental set-up for the transverse measurements.

An excitation amplitude of 0.1 N is used and the frequency increment is adapted in the same way as in the axial case.

4.3 Linearity Check

The validity of the linearity assumption is checked. The bellows is excited at different force levels, the response is measured and the frequency response function (FRF) is calculated. For a linear structure these FRFs are independent of excitation level. If non-linearity is present this is not true. Therefore the bellows is excited at different levels, in both the axial and transverse case.

An example of such a test can be seen in figure 8, which shows FRFs corresponding to different excitation levels in the axial case. Depending on the excitation level a frequency shift of roughly 8 Hz at the third axial resonance frequency occurs. A discussion of this phenomenon is included in the next section.

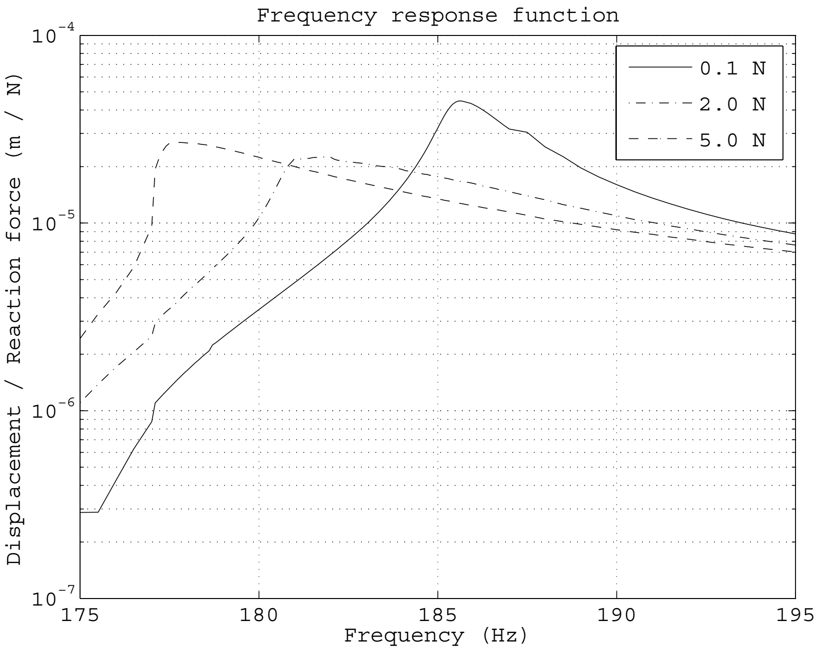

Figure 8. FRFs for different excitation force levels.

5 Results and Discussion

The resonance frequency shift observed in the linearity check is very interesting and somewhat intriguing. It appears at such small excitation levels and deflections that, for example, plastic material behaviour or geometric non-linearity should not be the reason. A completely verified explanation cannot be given at the moment, but a hypothetic qualitative explanation is presented in the following.

A similar experiment performed with a single-ply bellows does not show such shift. This indicates that the non-linearity could be related to the multiple plies. Although it is common to consider the plies to work independent of each other [8], the authors suspect that there are spots of contact between them. As the convolutions are exposed to bending, both in the axial and the transverse load case, relative (micro-)motion between the inner surface of the outer ply and the other surface of the inner ply, is induced, and thus there will be friction dissipation when the bellows vibrate. Friction does however usually not give a frequency shift - only a reduction of the vibration amplitude - but, if the friction limit, below which the two plies stick together, is slightly different between different locations along the bellows, the shift could be understood. Viewing the bellows as a chain of many "links", among which most of them have a friction limit between the plies close to zero, but some of them have a higher friction limit, the chain will have a different equivalent stiffness for different excitation force levels. At low levels some of the "links" will have a much higher stiffness than the others, because the two plies are then stuck together and they act at these locations as one ply of double thickness (the bending stiffness is proportional to the thickness in cube). At increasing excitation force level, fever and fever "links" will have this higher stiffness, since the friction limit is exceeded in more and more "links", and the equivalent stiffness of the chain then decreases. This goes on until the plies do not stick in any "links". For further increasing excitation force levels, there will then be no further shift due to this friction-related non-linearity, only a reduction of vibration amplitude.

For the present application of the bellows (exhaust system component) this non-linear phenomenon is probably of minor significance. The excitation levels should in practice mostly exceed the ones including this phenomenon. In any case, the frequency shift only seems to amount to a few percent, close to the theoretical linear resonance frequency. Furthermore, when the bellows is combined with the liner, this introduces a more significant non-linearity.

The issue is therefore not further scrutinized in this paper, and the measured resonance frequencies are accepted as "linear" in the comparison bellow.

The FRF for the axial load case is shown in figure 9 for an excitation force level of 0.1 N. The FRF for the transverse load case is shown in figure 10 for an excitation force level of 0.1 N. The solid lines show the theoretical results and the dashed lines shows the experimental results.

The results are summarised in table 3. The damping value is given as the fraction of critical damping and the correlation value is the relative difference between theoretical and experimental resonance frequencies.

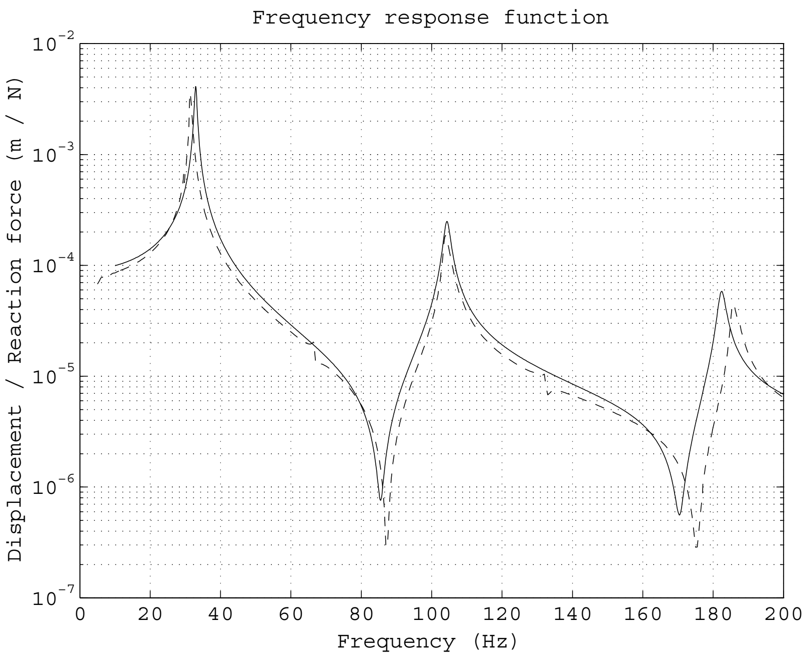

Figure 9. Theoretical and experimental FRFs in axial direction.

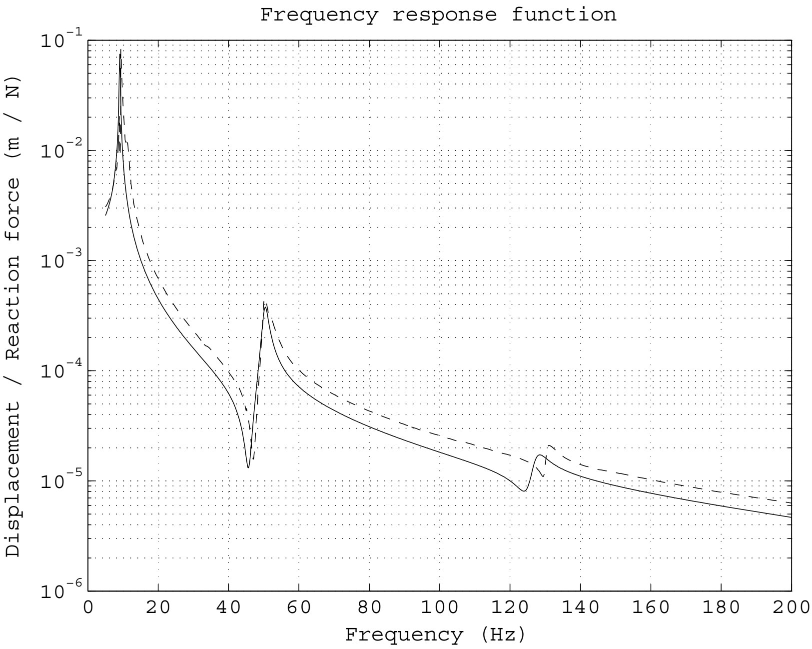

Figure 10. Theoretical and experimental FRFs in transverse direction.

Table 3. Results.

Mode	Experimental Frequency (Hz)	Experimental Damping (%)	Theoretical Frequency (Hz)	Correlation[a] (%)	Direction[b]
1	9.40	1.2	9.10	-3.2	T
2	31.4	1.0	32.9	4.8	A
3	50.2	1.7	50.3	0.2	T
4	104	0.9	104	0.4	A
5	131	2.0	127	-2.6	T
6	186	0.6	182	-1.8	A

[a] The correlations are calculated before rounding off.
[b] T=Transverse, A=Axial.

The damping ratios are amplitude dependent due to the non-linear behaviour of the bellows. The damping ratios increase with increasing excitation force.

It should be mentioned that for mode five the damping ratio cannot be calculated by established methods from experimental data because this mode is not significant, see figure 10. This damping ratio is instead estimated on a trial and error basis so that the theoretical FRF resembles the experimental FRF near this mode.

6 Conclusions

Simplified modelling of bellows flexible joints, used in for example automobile exhaust systems, is the subject of this study.

Broman et al. [2] presented a method for determining the dynamic characteristics of single-ply bellows of constant mean radius by manipulated beam finite elements of a commercial software. This paper suggests how that procedure can be extended to model also a multi-ply bellows of variable mean radius. Experimental investigations are performed for verification.

At small excitation force levels the response of the bellows is observed to be excitation dependent, that is, it is non-linear. The authors find this observation interesting and important to report, and a hypothetic qualitative explanation is discussed. For the present application of the bellows (exhaust system component) this non-linear phenomenon is however considered to be of minor significance, and its influence on the measured results seems to be small and is therefore neglected.

This said, the correlation between theoretical and experimental results is very good, which verifies the extended modelling procedure.

Furthermore, it is shown that the procedure can easily be implemented in another commercial software (in this case ABAQUS) than the one used in [2] (I-DEAS).

Acknowledgements

The support from Faurecia Exhaust Systems AB is gratefully acknowledged, especially from Håkan Svensson. The authors also gratefully acknowledge the financial support from the Swedish Foundation for Knowledge and Competence Development.

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Paper D

Dynamic Characteristics of a Combined Bellows and Liner Flexible Joint

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Englund T., Wall J., Ahlin K. & Broman G., Dynamic characteristics of a combined bellows and liner flexible joint, *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering* 218(5) (2004) pp. 485-493

Dynamic Characteristics of a Combined Bellows and Liner Flexible Joint

Thomas L Englund, Johan E Wall, Kjell A Ahlin and Göran I Broman

Abstract

A bellows combined with an inside liner and an outside braid is commonly used as a flexible joint in automobile exhaust systems to reduce transmission of engine movements to the exhaust system. It greatly influences the dynamics of the complete system. Understanding of its dynamic characteristics and a modelling method that facilitates systems simulation are therefore desired. This has been obtained in earlier works for the bellows itself. In this work an approach to the modelling of the combined bellows and liner joint is suggested and experimentally verified. Simulations and measurements show that the liner adds significant non-linearity and makes the characteristics of the joint complex. Results are presented for the axial and the bending load cases. In torsion, influence of the liner is negligible. Peak responses are significantly reduced when the excitation level approximately corresponds to the friction limit of the liner. The complexity of the combined bellows and liner joint is important to know of and consider in exhaust system design and proves the necessity of including a model of the liner in the theoretical joint model when this type of liner is present in the real joint to be simulated.

Keywords: *Beam model, Dynamic, Experimental investigation, Flexible joint, Friction, Liner, Non-linear.*

Notation

A	Area [m^2]
[**C**]	Damping-matrix
E	Young's modulus [Pa]
F	Force [N]
[**K**]	Stiffness-matrix
k	Stiffness [N/m]
[**M**]	Mass-matrix
M	Moment [Nm]
L	Length [m]
R	Radius [m]
α	Damping parameter
β	Damping parameter
ε	Strain
ξ	Damping ratio
σ	Stress [Pa]
ω	Angular frequency [rad/s]

Indices

f	Friction
i	Number
l	Liner
y	Yield

1 Introduction

With the introduction of transverse engines and catalytic converters a highly flexible gas-tight joint became necessary in automobile exhaust systems to reduce transmission of engine movements to the exhaust system. A bellows combined with an inside liner and an outside braid is commonly used for this purpose between the manifold and the rest of the exhaust system. As shown by Wall et al. [1], this joint greatly influences the dynamics of the complete system. An understanding of its dynamic characteristics and a modelling method that facilitates simulations of the exhaust system at an early stage of the product development process are therefore desired. Therefore, the simulation procedure must be computationally inexpensive. The component models should therefore be as simple as possible while yet giving a proper description of their dynamic characteristics and mutual influence within the exhaust system. Such models have been suggested and experimentally verified by Englund et al. [2, 3] for the pipes, the mufflers and the catalytic converter.

A method of modelling the bellows by manipulated beam finite elements, which considerably reduces the model size compared to a straightforward shell elements model, was presented by Broman et al. [4]. In this reference also a short historical background and further references on bellows joints can be found. This method was extended to apply also for multi-ply bellows of variable mean radius by Wall et al. [5].

This paper presents an approach to the modelling of the combined bellows and liner joint. Simulations are performed to gain understanding of its dynamic characteristics and experimental investigations are included for verification.

2 Basic Design

The design of a typical flexible joint is shown in figure 1. It consists of a gas-tight bellows combined with an inside liner and an outside braid. The gas-tightness is crucial for emission control since the joint is often upstream of the catalytic converter. The ends of the bellows, the liner and the braid are rigidly connected with the end-caps. General information on the design of this type of joint is also given by, for example, Broman et al. [4] and Cunningham et al. [6].

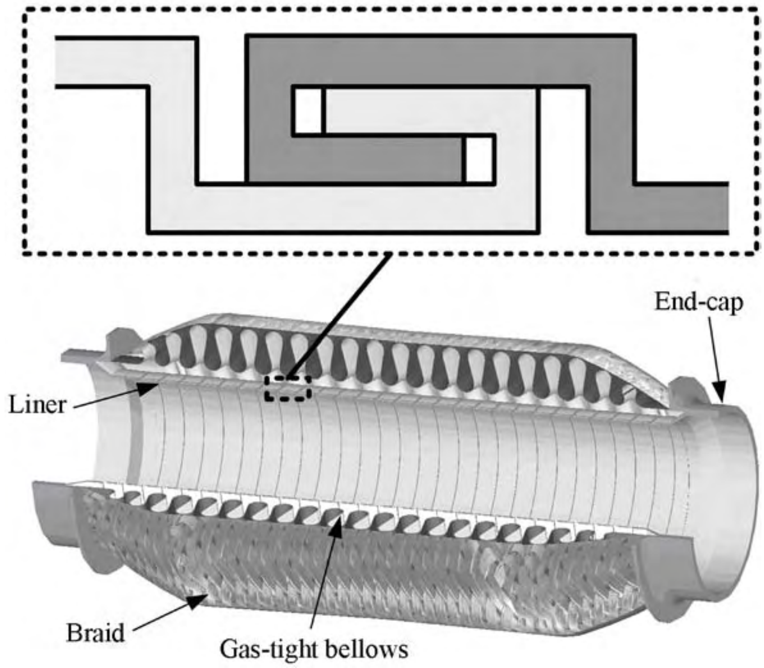

Figure 1. Basic design of the flexible joint.

The bellows studied in this paper is double-plied and has a smaller mean radius closer to the ends. Low stiffness is beneficial to decouple the engine from the exhaust system and a double-plied bellows has a lower stiffness than a single-ply bellows for a given strength. The liner is used for reduction of bellows temperature and for improved flow conditions. The liner consists of a strip of sheet metal that is winded cylindrically and folded. The cross-section of the winding is schematically shown in figure 1. When the coils move relative to each other there is resistance due to friction, which depends on how hard the folding is and the coefficient of friction.

The braid is used for mechanical protection and to limit the extension of the joint. It is not included in this study.

3 Modelling

The bellows is modelled in ABAQUS [7] using a pipe analogy and a beam finite element formulation with certain parameters manipulated [4, 5]. Considering that the studied bellows has different mean radius three different equivalent pipes are used. By comparing different mesh densities it is found that 20 beam elements are sufficient.

The liner is also modelled by beam elements with a pipe section. The wall thickness and the density of the material are set to give the correct mass of the liner. The liner and the bellows are only connected at the outer nodes of the elements at the ends. It is assumed that there is no contact elsewhere. The friction in the liner is modelled as a Coulomb-type dry friction. When the force is below the friction limit, the liner behaves elastically (sticking). Above the friction limit the liner is slipping. The axial stiffness, k, and the axial friction limit, F_f, are determined experimentally (see section 5). The friction force is assumed to be symmetric and independent of frequency. To model the above in ABAQUS an ideal-plastic isotropic material is defined. The principal behaviour of this material is shown in figure 2, where E is Young's modulus, σ_y is the yield stress and ε is the strain. The shear stress that may arise in the beam elements is not taken into account in the plasticity calculations.

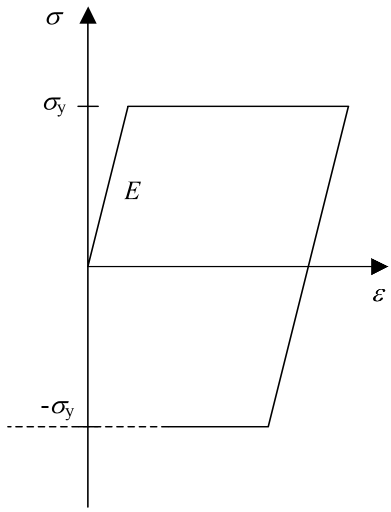

Figure 2. Behaviour of the ideal-plastic material.

The fictive Young's modulus and the fictive yield stress are

$$E = \frac{kL}{A_1} \tag{1}$$

$$\sigma_y = \frac{F_f}{A_1} \tag{2}$$

where A_1 is the area of the pipe cross section representing the liner and L is its length.

The plasticity calculations are performed numerically in so-called section points (see figure 3). When studying axial vibrations one section point is sufficient, but to get a good approximation of the bending resistance, M_f, for any axis of rotation more section points are necessary.

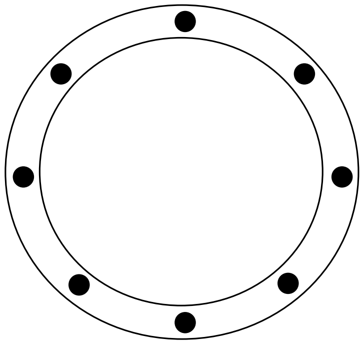

Figure 3. Section points.

If the liner is exposed to pure bending and uniform friction resistance is assumed along the circumference the exact bending resistance for any axis of rotation is

$$M_{\mathrm{f}} = \frac{2F_{\mathrm{f}}R_{\mathrm{l}}}{\pi} \tag{3}$$

where R_{l} is the radius of the liner. Using a finite number of section points an approximate bending resistance is obtained. The maximum error in bending resistance is a function of the number of section points. Twenty points are used in this work.

Axial and bending vibrations are studied. In torsion, influence of the liner is negligible compared to the bellows and thus the prior model is sufficient [5]. In axial loading, the liner slips at the coil that happens to have the weakest folding. This location is probably rather randomly distributed between different liners due to variations in fabrication and is thus hard to predict. In bending loading, the liner slips at the coil that has the highest ratio of bending moment over friction bending resistance. When the joint is a part of the whole exhaust system in general motion, this location is hard to predict. On the other hand, the exact slip location is then not so critical for the system and joint behaviour. Furthermore, numerical problems arise when connecting ideal-plastic elements in series. Therefore it is assumed that in the general case the liner slips at the mid-coil. This can be modelled by using a short ideal-plastic element in this mid-location, connected on each side to an elastic element to represent the rest of the liner.

For the specific boundary conditions and bending loading used in the experimental set-up it is, however, most likely that the liner bends close to its clamped end. Therefore the ideal-plastic element is located at this end in the simulations of this paper.

The end-caps are considered as rigid pipes and are modelled with lumped mass and rotary inertia elements. Transducers and connecting devices used in the experiments are modelled with lumped mass elements.

4 Simulation

The frequency interval of interest for the analysis is obtained by considering that a four-stroke engine with four cylinders gives its main excitation at a frequency of twice the rotational frequency. Usually the rotational speed is below 6000 r/min. Excitation at low frequencies may arise due to road irregularities, as discussed by, for example, Belangardi and Leonti [8] and Verboven et al. [9]. Thus, the interval is set to 0-200 Hz.

Since the joint is strongly non-linear, direct time integration, using the explicit solver of ABAQUS, is performed. In both the axial and bending cases the joint is clamped at one end and excited with a sinusoidal force at the other end. The response is taken in the excitation point. A frequency step of 2 Hz is used except at the first axial and bending peak responses, where a smaller step is used since the response here changes more rapidly. The model is simulated until steady state is reached for each excitation frequency.

When the excitation force level is approximately the same as the friction limit, damping associated with the material is negligible compared to the friction-based damping, but otherwise it affects the results. In ABAQUS it is possible to define so-called Rayleigh damping, according to

$$[\mathbf{C}] = \alpha[\mathbf{M}] + \beta[\mathbf{K}] \tag{4}$$

where [**C**] is the damping matrix, [**M**] is the mass matrix, [**K**] is the stiffness matrix and α and β are damping parameters. For a linear system these parameters relates to the modal damping ratios, ξ_i, as (see, for example, Bathe [10])

$$\varsigma_i = \frac{\alpha}{2\omega_i} + \frac{\beta\omega_i}{2} \tag{5}$$

where ω_i is a certain natural frequency. In this work the values of α and β are calculated so that $\xi_1 = \xi_3 = 2$ per cent (ω_1 equals the excitation frequency and $\omega_3 = 3\omega_1$ is the most dominant higher harmonic). This gives a system of two equations that is solved for each excitation frequency to obtain α and β. A damping ratio of 2 per cent gives a good overall agreement with experimental results when the system is excited with a force well below the friction limit.

Since the calculation of the damping parameters, the increment in excitation frequency and storing of the simulated time response must be performed for each excitation frequency it would be very time-consuming to perform these tasks manually. Instead an automated procedure is used where MATLAB [11] and ABAQUS interact with each other. The damping parameters and the excitation frequency are determined in MATLAB and this information is used as input to ABAQUS, where the calculations are performed. The calculated time response is then transferred back to MATLAB, where it is stored. This procedure is automatically repeated for the excitation frequencies of interest. The interaction between the two software packages is performed by taking advantage of MATLAB's ability to read and write ASCII files.

Since the system is non-linear the response generally includes higher harmonics. This is especially clear in the axial case when the excitation force level is approximately the same as the friction limit. To be able to analyse and compare simulated and experimental results in a straightforward way only the first harmonic is considered. The time responses are therefore transformed to the frequency domain, using the fast Fourier transform (FFT) algorithm in MATLAB, where the amplitude of the first harmonic is obtained. A flat-top window is used to avoid possible leakage problems. The normalized response, defined as the amplitude of the first harmonic over the excitation force amplitude, is then calculated.

Since the problem is non-linear the magnitude of the excitation force is significant. The behaviour of the joint is dramatically different depending on whether the excitation force is above or below the friction limit, so it is important that the force interval used in the simulation covers this shift.

5 Experimental Investigation

A sinusoidal excitation is used so that the input signal level can be accurately controlled. The signal-to-noise ratio is good because all the input energy is concentrated at one frequency at the time. Another advantage is that an adaptive frequency increment can be used so that rapid changes in the response can be accurately captured. The main drawback is that it is time consuming because the measurements are performed frequency by frequency and that time is needed for the test specimen to reach steady state at each frequency.

A Hewlett Packard VXI measuring system is used to acquire the experimental data and I-DEAS Test [12] is used as signal analyser. The I-DEAS Test has a function called step sine closed loop control in the sine measurements module. The force level is iteratively adjusted until the target amplitude and/or phase is reached within a specified tolerance. The steady state acceleration amplitude of the fundamental frequency in the response is then saved and the procedure is repeated for the next excitation frequency. Retaining only the fundamental frequency of the response gives so-called first-order frequency response functions (FRFs). For a further discussion see, for example, reference [13]. The measurement set-up for axial vibrations is shown in figure 4.

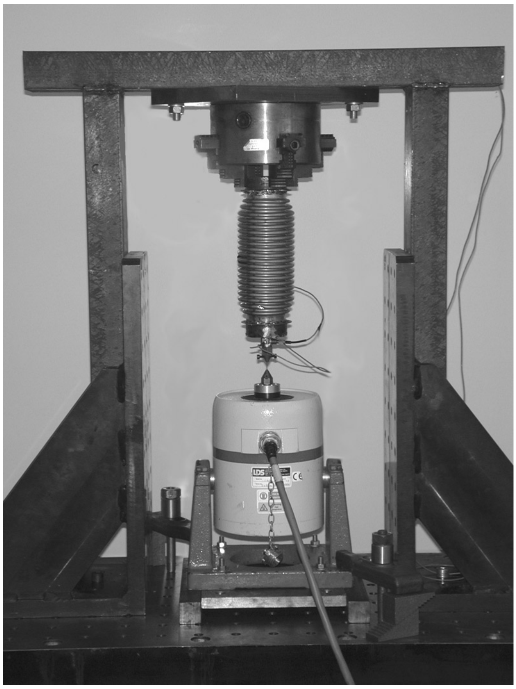

Figure 4. Experimental set-up for the axial measurements.

The joint is rigidly mounted at one end in a chuck from a lathe and is excited at the other end. The lathe chuck is mounted in a frame made of steel beams, which is assumed to be rigid. This frame is attached to a rigid table of considerable mass. The shaker is also mounted on this table. A metal plate is welded onto the free end of the joint and on this plate an accelerometer and a force transducer are mounted (see figure 5). The plate is rigid in the frequency interval of interest. The force transducer is connected to the shaker through a stinger. The measurement set-up for bending vibrations is shown in figure 6.

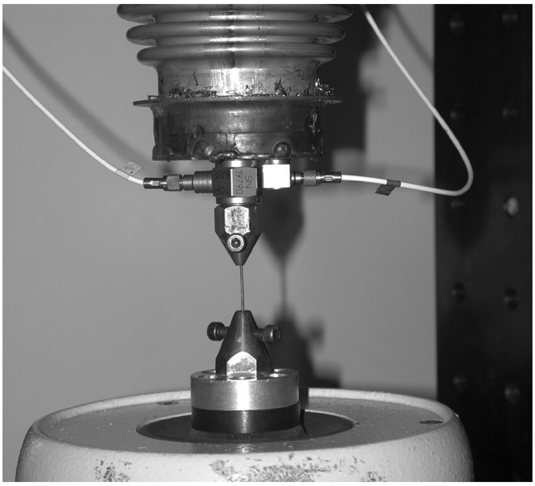

Figure 5. Transducer placements in the axial set-up.

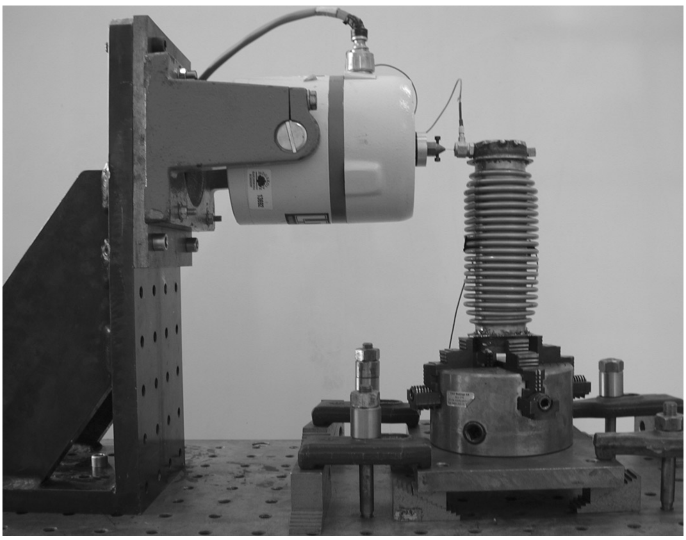

Figure 6. Experimental set-up for the bending measurements.

Due to the low stiffness of the joint in the transverse direction it is mounted so that its axial direction coincides with the gravitational force, to avoid undesired influence from gravitation on the dynamic behaviour. The joint is rigidly mounted at one end in a chuck from a lathe and is excited at the other end. The lathe chuck is mounted on a rigid table of considerable mass. The shaker is also mounted on this table via a rigid angle bracket made of thick reinforced steel plates. The free end of the joint is connected to the shaker through a force transducer and a stinger and an accelerometer is placed on the opposite side (see figure 7).

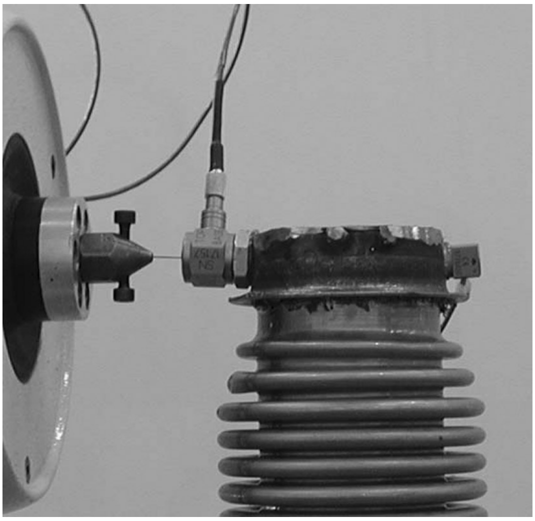

Figure 7. Transducer placements in the bending set-up.

In both set-ups it is assured that vibration feedback through the rigid table is insignificant.

The stiffness and the friction limit of the liner are determined experimentally by using principally the above described axial set-up. The axial stiffness, k, is obtained by exciting the liner with a force well below the friction limit. Plotting the FRF in a log-log Bode diagram format the stiffness is obtained from the low frequency asymptote (Ewins [14]). To determine the axial friction limit, F_f, the liner is excited with a sinusoidal force of low frequency and the force and displacements are registered. The amplitude of the excitation force is increased step by step until the friction limit is clearly reached. A plot is shown in figure 8.

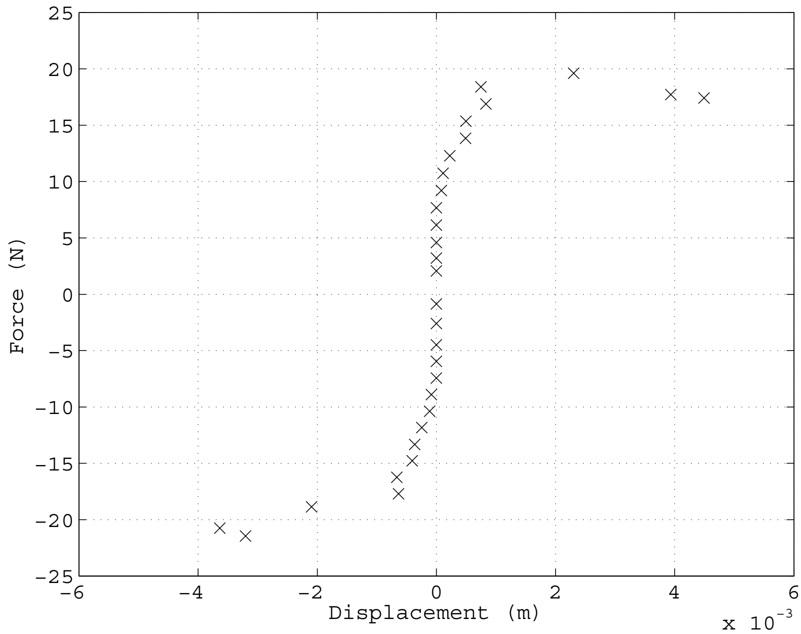

Figure 8. The axial friction limit of the liner.

6 Results and Discussion

Theoretical and experimental results are compared in figure 9 for two different excitation force levels in the axial case, one below (10 N) and one above (25 N) the friction limit of the liner (17 N). Theoretical and experimental results are compared in figure 10 for two different excitation force levels in the bending case, one below (0.25 N) and one above (2.25 N) the friction bending resistance of the liner (corresponding to a transverse force of 1.0 N). Normalized responses (magnitude of first order FRFs) are plotted against excitation frequency for different excitation force levels.

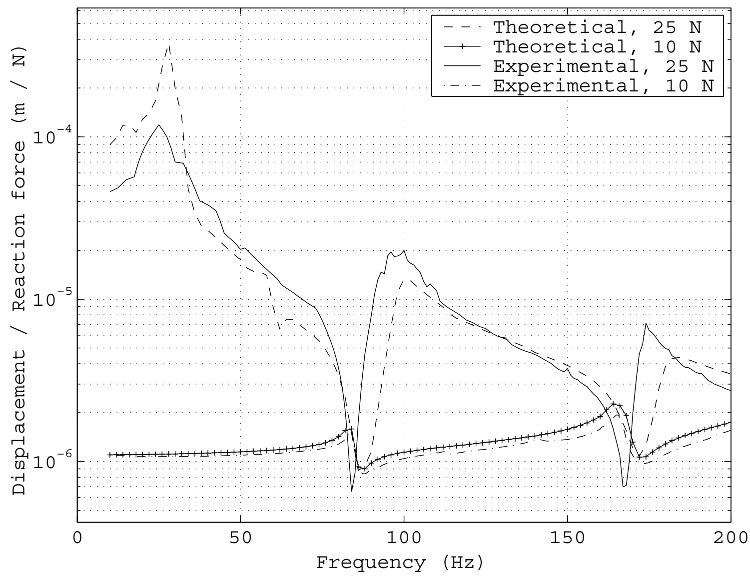

Figure 9. Theoretical and experimental axial results.

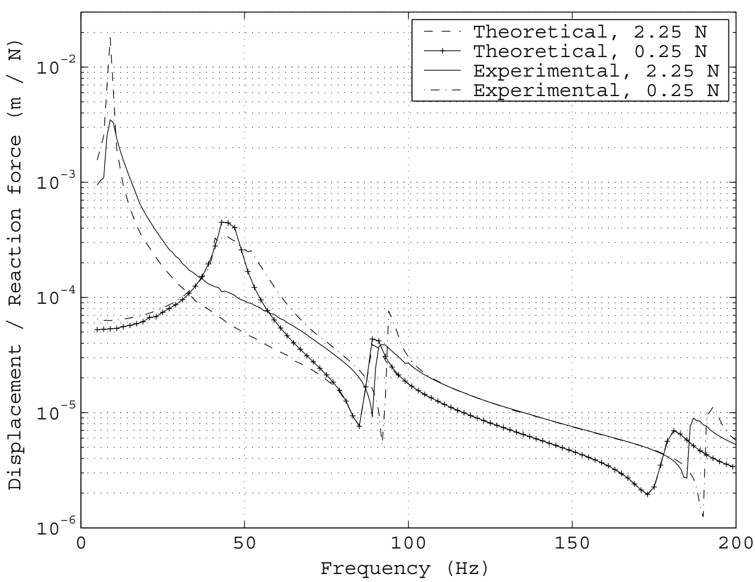

Figure 10. Theoretical and experimental bending results.

In the experimental investigation force drop-out occurs at frequencies with large responses due to the weakness of the joint above the friction limit, making it impossible to excite the joint with a specified constant force amplitude. The excitation force here also includes higher harmonics. The magnitude of the experimental normalized response is therefore uncertain at these peaks. Otherwise the agreement between theoretical and experimental results is good, indicating that the simulation model can be used for further studies.

Results for the simulation model excited with 10, 15, 20, 25 and 30 N in the axial direction, covering the friction limit, are shown in figure 11.

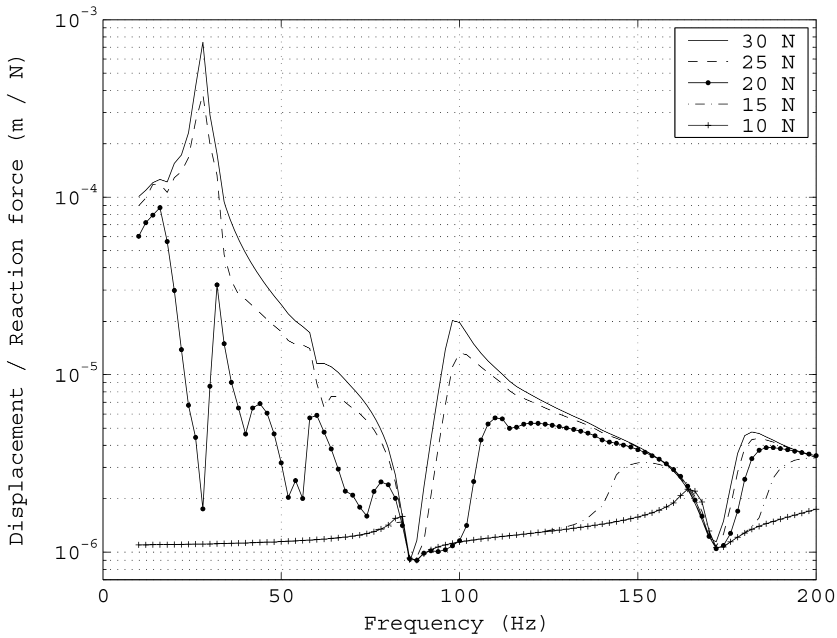

Figure 11. Simulated axial results for different force levels.

The normalized response is small and the peak response frequencies are high at the lowest force level (10 N). This is because the liner is sticking, i.e. no slipping occurs, and the stiffness of the liner is high compared to the bellows. At the intermediate force levels (15 and 20 N) both sticking and slipping take place during different parts of the motion and large friction-based damping results. The normalized response is higher and the peak response frequencies are shifted downwards. Higher harmonics are significant in the response (see figure 12).

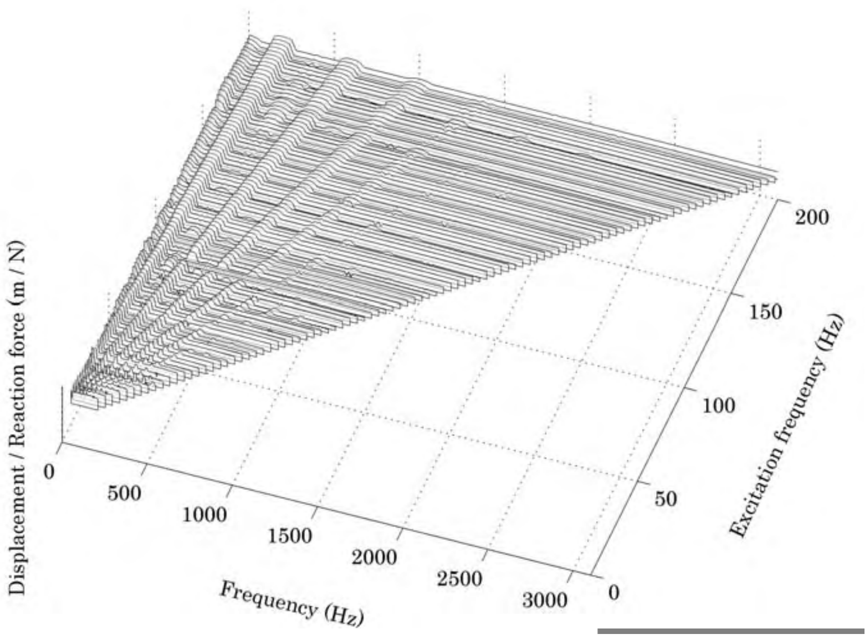

Figure 12. Waterfall diagram when exciting the joint with 20 N in the axial case.

At the higher force levels (25 and 30 N) the liner is mostly slipping and the influence of friction on the peak response frequencies becomes more and more negligible, so they approach those of the bellows itself [5] (with a small difference due to the mass of the liner). The normalized response is higher than for the low and intermediate excitation force levels but is lower than the response of the bellows itself [5].

Results for the simulation model excited with 0.25, 0.75, 1.25, 1.75 and 2.25 N in the transverse direction, covering the friction bending resistance, are shown in figure 13.

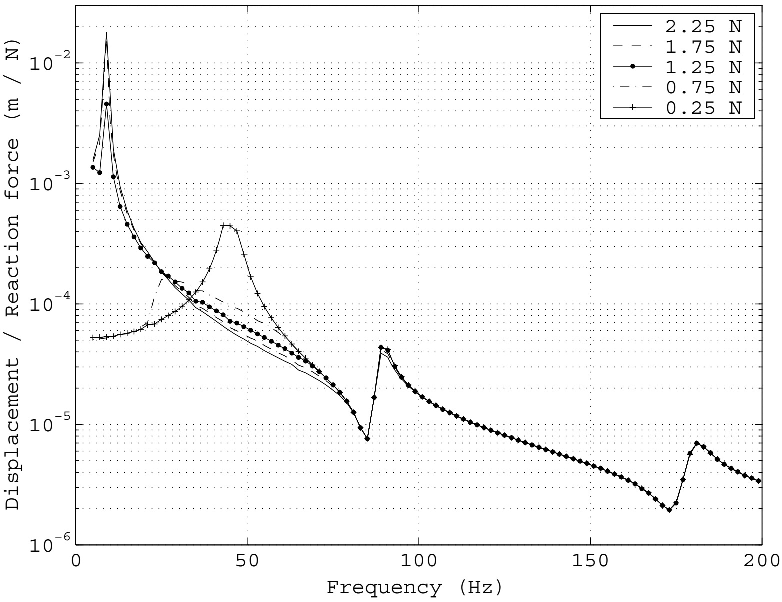

Figure 13. Simulated bending results for different force levels.

Comments regarding the normalized response are principally the same as for the axial case. The presence of higher harmonics at intermediate excitation levels is, however, less and at the higher excitation levels slipping occurs only for the lowest peak response.

7 Conclusions

Dynamic characteristics of a combined bellows and liner flexible joint, commonly used in automobile exhaust systems, is the subject of this paper. An approach to the modelling of this joint is suggested and implemented in the commercial software ABAQUS. The simulation model is experimentally verified. Axial and bending vibrations are studied and it is shown that the liner adds significant non-linearity due to friction, which makes the characteristics of the joint complex.

At excitation levels well below the friction limit of the liner the joint is very stiff (and essentially linear). This is not desired in applications since the purpose of the joint is to add flexibility. At intermediate excitation levels, close to the friction limit of the liner, stick-slip motion occurs, resulting in high-energy dissipation through friction-based damping. At excitation levels well above the friction limit of the liner the behaviour of the joint approaches that of the bellows itself (and it becomes again more and more linear). Additional complications are that differences in the friction limit have been observed between joint specimens and that it also seems that the friction limit decreases after some time of use.

The complexity of the combined bellows and liner joint is important to know of and consider in exhaust system design and proves the necessity of including a model of the liner in the theoretical joint model when this type of liner is present in the real joint to be simulated.

Acknowledgements

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Paper E

Influence of a Bellows-type Flexible Joint on Exhaust System Dynamics

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Influence of a Bellows-type Flexible Joint on Exhaust System Dynamics

Johan E Wall, Thomas L Englund, Kjell A Ahlin, Göran I Broman

Abstract

Most modern cars have a bellows-type flexible joint between the manifold and the catalytic converter to allow for thermal expansion and to decouple large engine movements and vibrations from the rest of the exhaust system. To obtain better understanding of the influence of this joint, the dynamic response of a typical exhaust system is studied when excited via different joint configurations. Measurements show the great order of reduction in vibration transmission to the exhaust system that a bellows joint, with and without an inside liner, gives in comparison with a stiff joint. For the combined bellows and liner joint, vibration transmission is, however, higher than for the bellows alone. Together with some other aspects this makes the choice of including a liner in the exhaust system application complex. For a system in general the possibility of tuning the friction limit of the liner, to minimize overall vibrations through friction-based damping, depends on how close to ideal the excitation source is and its location. Anyhow, the combined bellows and liner joint makes the exhaust system behaviour significantly non-linear, whereas the system behaviour proves to be essentially linear when the bellows has no liner, which imply that the liner needs to be included in theoretical models when present in the real system.

Keywords: *Bellows, Dynamics, Exhaust system, Experimental investigation, Flexible joint, Vibration transmission.*

1 Introduction

Increased customer awareness, demands on shortened time to market and the greater complexity of today's cars has made the traditional "cut-and-try" product design strategy obsolete. Together with the strong development of computer capacity this has made dynamics analyses of car components considerably more comprehensive over the years. The exhaust system is of particular interest since it is one of the main sources of structure-borne noise and needs to fulfil high demands on safety and durability.

Front-wheel-driven cars have become common on the market. These cars have the engine mounted in the transverse direction and have a flexible joint included between the manifold and the rest of the exhaust system to allow for thermal expansion and to decouple large engine movements and vibrations from the rest of the exhaust system. The commonly used bellows-type joint has, however, caused car and component manufacturers severe problems in some cases and it is generally recognised that there is a need for better understanding of the influence of this type of joint on the system dynamics.

This study is a part of a co-operation project between the Department of Mechanical Engineering at the Blekinge Institute of Technology, Karlskrona, Sweden and Faurecia Exhaust Systems, Inc., Torsås, Sweden. The overall aim of the project is to find a procedure for effectively modelling and simulating the dynamics of customer-proposed exhaust system lay-outs at an early stage in the product development process, to support the dialogue with the costumer and for overall lay-out optimisation. To be suitable for that it is important that the theoretical model is as computationally inexpensive as possible while still being accurate enough for the characteristics that it is supposed to describe.

Wall et al. [1] showed that the multi-plied bellows of this work can be modelled by a computationally effective beam elements model suggested by Broman et al. [2]. Englund et al. [3] investigated how the inside liner affects the dynamic behaviour of the flexible joint. They showed that it becomes significantly non-linear due to friction. Further information about bellows flexible joints can be found in Cunningham et al. [4].

The part of the exhaust system that is downstream of the flexible joint has been studied by Englund et al. [5]. They showed that this part could be modelled as a linear subsystem that is excited via the flexible joint.

This paper aims at an initial understanding of the influence of a bellows joint and a combined bellows and liner joint respectively on the dynamics of a typical exhaust system when excited via this joint, as a basis for design improvements and the development of theoretical models in line with the overall project aim. A rough idea of the reduction of vibration transmission through a bellows joint and a combined bellows and liner joint, compared to a stiff joint, is desired.

2 Basic Design of the Exhaust System

The exhaust system studied in this work is shown in figure 1. The weight of the system is approximately 20 kg and the length is about 3 m. It consists of a front and a rear assembly, which are connected with a sleeve joint. Both assemblies are welded structures made of stainless steel. The front assembly consists of pipes, a catalytic converter and a flexible joint. The rear assembly consists of pipes, a larger intermediate muffler and a smaller rear muffler. The manifold and the engine are not included in this study.

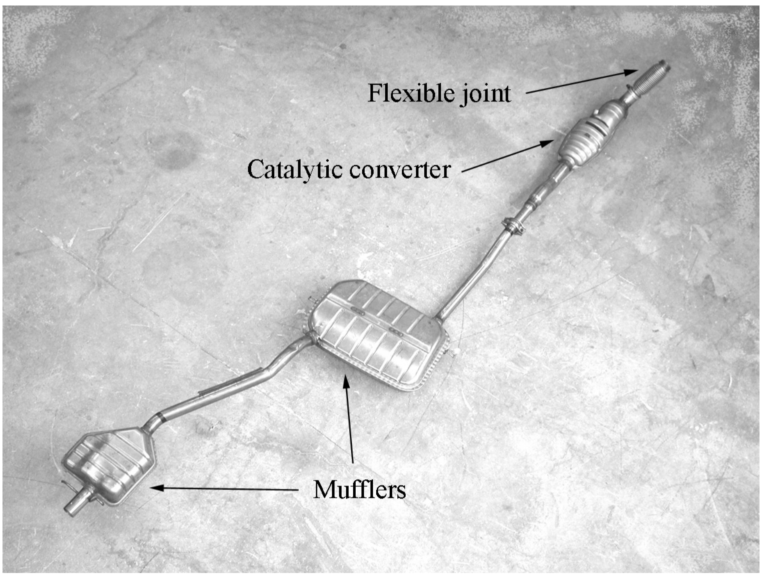

Figure 1. The exhaust system design.

In general the flexible joint consists of a gas-tight multi-ply flexible bellows, an inside liner and an outside braid. The liner was originally introduced for reduction of bellows temperature and improved flow conditions. The braid is used for mechanical protection and to limit the possible extension of the joint. The parts are connected with end caps. A model of the flexible joint is shown in figure 2. The braid is not included in this study as the focus of this work is on the influence of the liner.

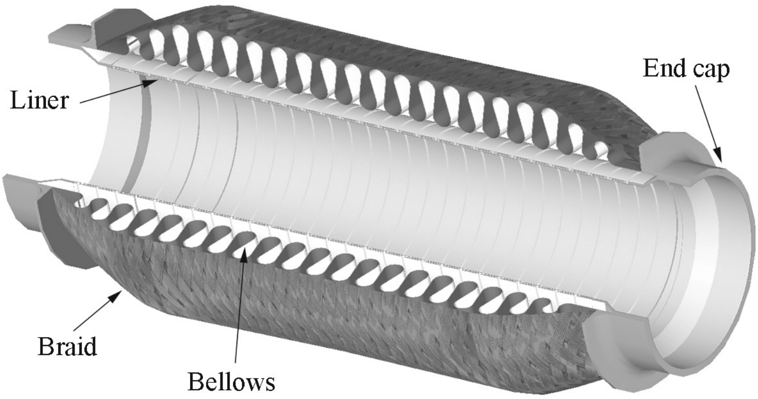

Figure 2. Basic design of the flexible joint.

The exhaust system used in this work has been modified to facilitate change between different joints. The free end of the pipe just upstream of the catalytic converter has been customized so that the end of the joint can be fitted into the pipe. The pipe and the joint are fixed to each other using a tube clamp. In the original design the joint is welded to the pipe.

Three different joint configurations are considered. In the first case the "joint" is simply a pipe with approximately the same dimensions as the pipe just upstream of the catalytic converter, see figure 3, to study the dynamics of the system when no flexibility is added at the joint location.

Figure 3. Modified exhaust system with the pipe.

In the second case a bellows flexible joint (without liner and braid) is attached to the system. In the third case a combined bellows and liner joint (without braid) is attached. The second and third cases are shown in figure 4.

Figure 4. Modified exhaust system with the flexible joint.

This work focuses on vibrations originating from the engine so the excitation frequency interval is obtained by considering that a four-stroke engine with four cylinders gives its main excitation at a frequency of twice its rotational frequency. Usually the rotational speed is between 900 and 6000 r/min. Thus the frequency interval of interest is set to 30 – 200 Hz. In general, also excitation at lower frequencies (and higher amplitude) may arise owing to road irregularities, acceleration, breaking, and gear shifting of the car, but this is not included in the present study.

3 Experimental Investigations

The overall experimental set-up is shown in figure 5. To facilitate the comparison with simulation results in a later stage of the project free-free boundary conditions are used during all the experimental investigations. This condition is sufficiently realised by hanging the exhaust system in soft rubber bands. Five suspension points are used to distribute the mass of the system in an appropriate way.

Figure 5. The experimental set-up.

A shaker is used to excite the system at the free end of the joint to simulate engine excitation roughly, as shown in figure 6. The shaker is attached to the system via a force transducer and a stinger.

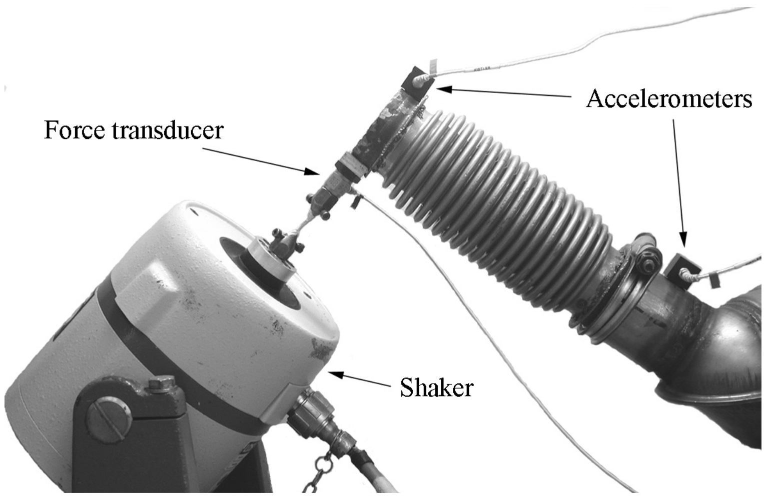

Figure 6. Excitation upstream of the flexible joint.

The response is measured using piezoelectric accelerometers in the excitation point and at a point just downstream of the joint. In this way the transmissibility of the joint can be calculated. The transducer locations are shown in figure 6. The response is also measured in a point in the middle of the exhaust system.

The signal analyser I-DEAS Test [6] and a Hewlett-Packard VXI measuring system are used to acquire the experimental data. The acquired frequency response functions (FRFs) are exported to MATLAB [7] where they are analysed.

In the first two cases a burst random excitation force is used to excite the system, since in these cases the system is considered as linear (see section 5). As the combined bellows and liner joint (third case) is significantly non-linear due to friction in the liner, a sinusoidal excitation force is used. Sinusoidal excitation is well suited for analysis of non-linear structures as the input signal can be accurately controlled. Furthermore the signal-to-noise ratio is good since all energy is concentrated at one frequency at a time. The main drawback of the method is that it is very time consuming. A function called step sine closed-loop control in I-DEAS Test is used. This function makes it possible to control the force amplitude so that the system can be excited with a constant force over the whole frequency interval of interest. This is desired to expose the non-linear behaviour in a satisfactory way. The fundamental frequency component of the excitation force and the response amplitudes is retained, giving so-called first-order FRFs (see, for example, Maia et al. [8]). Since no significant higher harmonics are present during transverse excitation of the combined bellows and liner joint [3], first-order FRFs are considered to give an accurate picture of the dynamic behaviour of the system. The frequency step is adapted so that a smaller frequency step is used when rapid changes in the responses occur and a larger step is used when the response varies more slowly.

4 Results

To verify the linearity assumption for the first and second case the system is exited with different excitation force levels (the force level is doubled in each measurement round) and FRFs between the excitation point and a point in the middle of the system are calculated and overlaid. For a linear structure the FRFs are independent of excitation level. This is clear for the first case. For the second case the results are shown in figure 7.

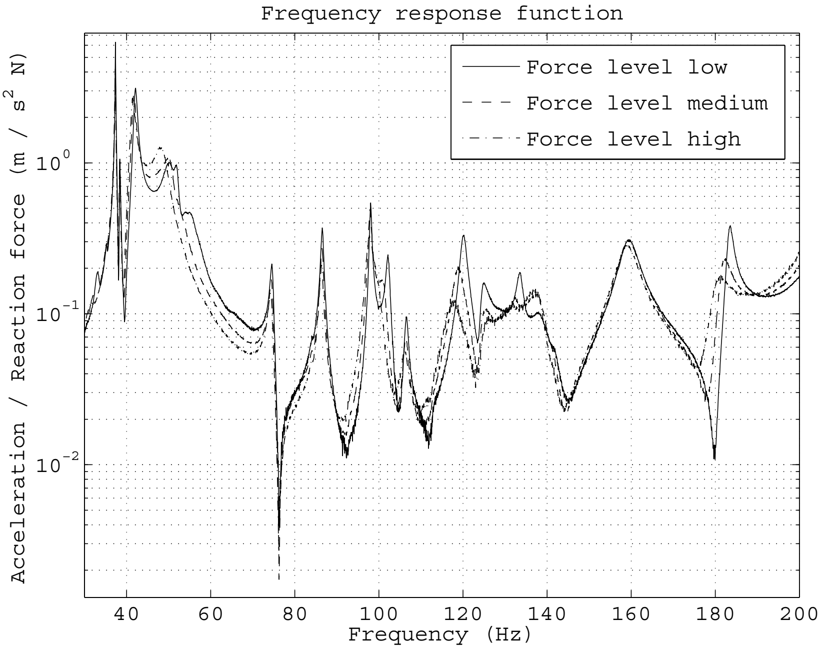

Figure 7. Linearity check for the exhaust system including the bellows joint.

As seen the damping increases slightly and the natural frequencies decrease slightly with increased excitation level. This is in agreement with the results found by Wall et al. [1] for the bellows itself. The non-linearity is however weak so the system is considered linear also for the second case of this work.

The transmissibility between the excitation point and a point just downstream of the joint is shown in figure 8 for the first and second case. The solid curve represents the transmissibility when the pipe "joint" is attached (first case) and the dashed curve represents the transmissibility when the bellows joint is attached (second case).

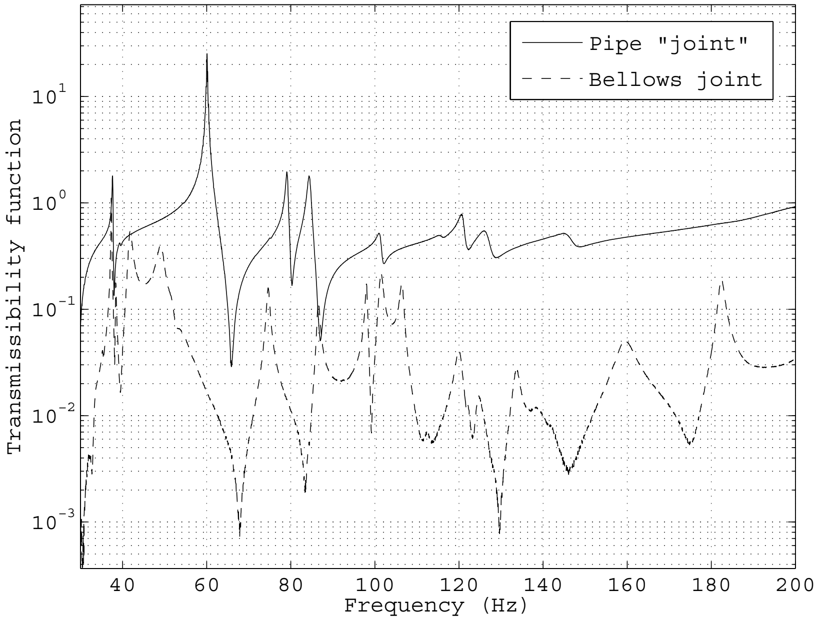

Figure 8. Transmissibility for the bellows joint and the pipe, respectively.

As seen, the bellows joint significantly reduces the vibration transmission compared to the case with the pipe "joint".

The transmissibilities between the same points for the second and third case is shown in figure 9. The solid curve represents the transmissibility when the bellows joint is attached (second case) and the dashed and dot-dashed curves represent the transmissibilities when the combined bellows and liner joint is attached (third case). To reveal the non-linear behaviour of the combined bellows and liner joint different excitation force levels are used. In figure 9, curves for excitation force levels of 0.1 N and 4 N are included.

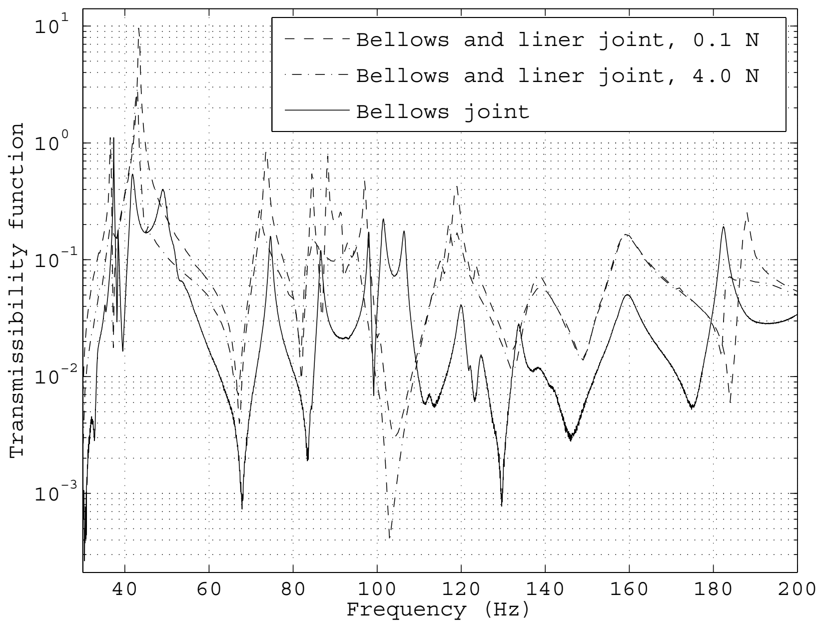

Figure 9. Transmissibility for the combined bellows and liner joint and the pipe, respectively.

As seen the combined bellows and liner joint generally has a higher vibration transmission than the bellows joint. The transmissibility in this third case is, however, still significantly lower than for the first case (pipe "joint").

Figure 10 shows typical FRFs between the excitation point and a point in the middle of the exhaust system when exited with different excitation force levels between 0.1 N and 4 N. This further reveals the non-linear behaviour for the third case (compare figure 7 and 10). As seen, the exhaust system dynamics depends heavily on excitation level when the combined bellows and liner joint is attached. When the friction limit is exceeded, friction based damping arise and the resonance peaks are reduced significantly. The response is, however, generally higher than for the second case (bellows without a liner).

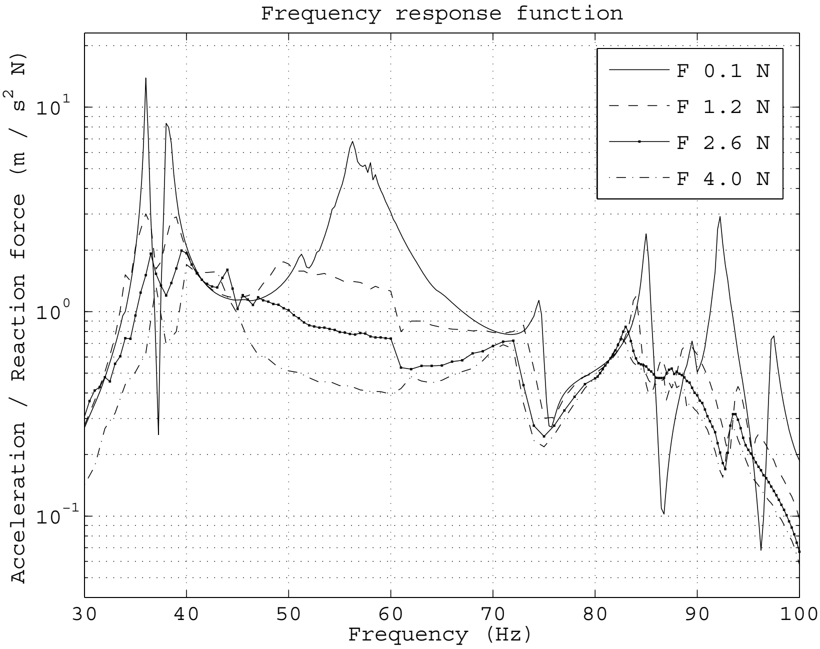

Figure 10. Typical FRFs at the middle of the system including the combined bellows and liner joint.

5 Discussion and Conclusions

The influence of a bellows joint and a combined bellows and liner joint, respectively, on the dynamics of a typical exhaust system is the subject of this work.

It is seen that the non-linearity due to the double plies of the bellows, reported by Wall et al. [1], does not significantly affect the dynamic behaviour of the exhaust system. This non-linearity can therefore be neglected in theoretical models of the assembled exhaust system.

The great order of reduction of vibration transmission through a bellows joint, both with and without an inside liner, in comparison with a stiff joint, is demonstrated. The transmissibility for the combined bellows and liner joint is, however, generally higher than for the bellows without the liner. This should be even more pronounced when the excitation source is closer to ideal (not affected by the excited system), since the energy dissipation due to friction in the liner is then of less importance. The engine of a car is probably close to an ideal excitation source of its exhaust system. Thus, from the viewpoint of reduction of transmission of engine vibrations through the joint a bellows without a liner, or with a liner with as low a friction limit as possible, should be preferred. Regarding other sources of excitation of the exhaust system, or regarding a system in general (non-ideal excitation and excitation at other locations), the liner may, however, be advantageous by reducing the response through friction-based damping. The friction limit could then theoretically be tuned to maximise the damping over some interesting frequency interval and excitation force level interval.

In addition to the above it should be considered that the liner reduces the bellows temperature and improves flow conditions. Friction in the liner should, however, not be necessary to fulfil these objectives, but a liner with no (or very low) friction may cause additional undesired noise.

The choice of including a liner in the exhaust system application is thus complex. Anyhow, it is shown that the system is significantly non-linear due to friction when the liner is included and so it is important to include the liner in theoretical models if it is present in the real system.

Acknowledgements

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Paper F

Identification and Modelling of Structural Dynamics Characteristics of a Water Jet Cutting Machine

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Identification and Modelling of Structural Dynamics Characteristics of a Water Jet Cutting Machine

Johan E Wall, Thomas L Englund, Ansel J Berghuvud

Abstract

Dynamic characteristics of a water jet cutting machine, to be used in a virtual machine implemented in an analysis tool for engineering design, are derived. Machine users need for more cost effective production put demands on faster cutting. Faster cutting results in higher dynamic loads. As a consequence, problems with unwanted vibrations that decrease cutting precision may occur. Prediction of such potential problems is facilitated by an analysis tool for evaluation of suggested design solutions early in the product development process. The present work contributes to ongoing development of such an analysis tool for design engineers. An iterative approach including both theoretical and experimental analysis is applied in order to derive a structural dynamics model of the studied machine. A complex dynamic behaviour of the machine is found. High correlation between results obtained from theoretical and experimental modal analysis implies that the developed model can be used with confidence in future studies of the machine's total system behaviour.

Keywords: *Experimental investigation, Modal analysis, Modelling, Structural dynamics, Vibrations.*

1 Introduction

Water jet cutting machine users desire higher productivity for better competitiveness. A way to achieve this is faster cutting and better cutting precision. However, increased cutting speed gives higher dynamic loads on the machine. Problems with unwanted vibrations that decrease cutting precision may follow. Machine developers strive to both fulfil the increased customer demands and to decrease total development costs. This, and a shortened time-to-market, is believed to be achievable by prediction of the machine behaviour earlier in the product development process through theoretical modelling and simulation. The use of theoretical modelling and simulation facilitates design optimisation and minimises the number of needed physical prototypes. In general, enabling an early prediction of the system behaviour also facilitates integration of specialised disciplines in a concurrent engineering process, as for example described by Andreasen [1] and Olsson [2]. An analysis tool for appraisal of suggested design solutions is therefore desirable. This is of particular interest in the current case since the produced machines often are unique as they are modified to suite particular customer needs.

Water jet cutting is an erosion process. A high velocity water jet is created by letting out water through a small orifice from a pressurised vessel. Cutting with a jet of water purely is appropriate for soft materials. Abrasives are added to the water jet for cutting of hard materials. More information about this technology is given by for example Draughon [3], Öjmertz [4] and Water Jet Sweden AB [5].

The studied type of product is an example of a mechatronic system. Analysis considering the characteristics of both the control system and the mechanical parts are therefore needed to enable optimisation towards the desired behaviour of the machine.

The work presented here is a part of a co-operation project between the Department of Mechanical Engineering at Blekinge Institute of Technology, Karlskrona, Sweden and Water Jet Sweden Inc., Ronneby, Sweden. The overall long-term goal is to derive a virtual machine that can be used by design engineers to optimise accuracy and cutting speed, predict durability, evaluate safety, etc., during the development of new machine designs. The selected approach for studies of the mechatronic system behaviour includes a mixture of hardware and software in the loop. This puts demands on simulation and interaction with hardware in real-time. Simulation models should therefore be as computationally efficient as possible while still being accurate enough for the characteristics they are supposed to describe. The development of the virtual machine is further described by Bathelt and Jönsson [6].

The present work focuses on the mechanical parts of a typical water jet cutting machine from Water Jet Sweden Inc. An iterative approach including theoretical modelling, simulation, experimental investigation and model updating is applied in order to gain understanding of the dynamic behaviour of the system and to develop a structural dynamics model of the studied machine, see figure 1.

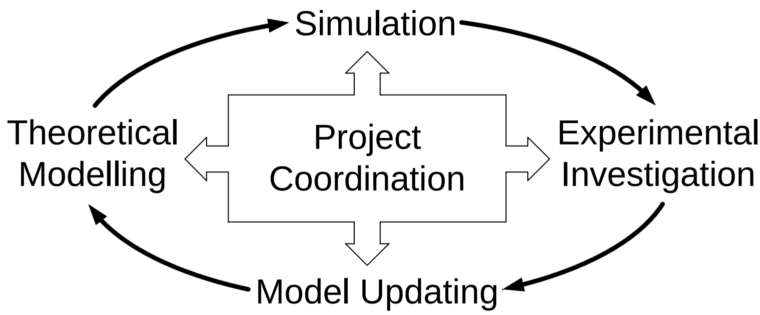
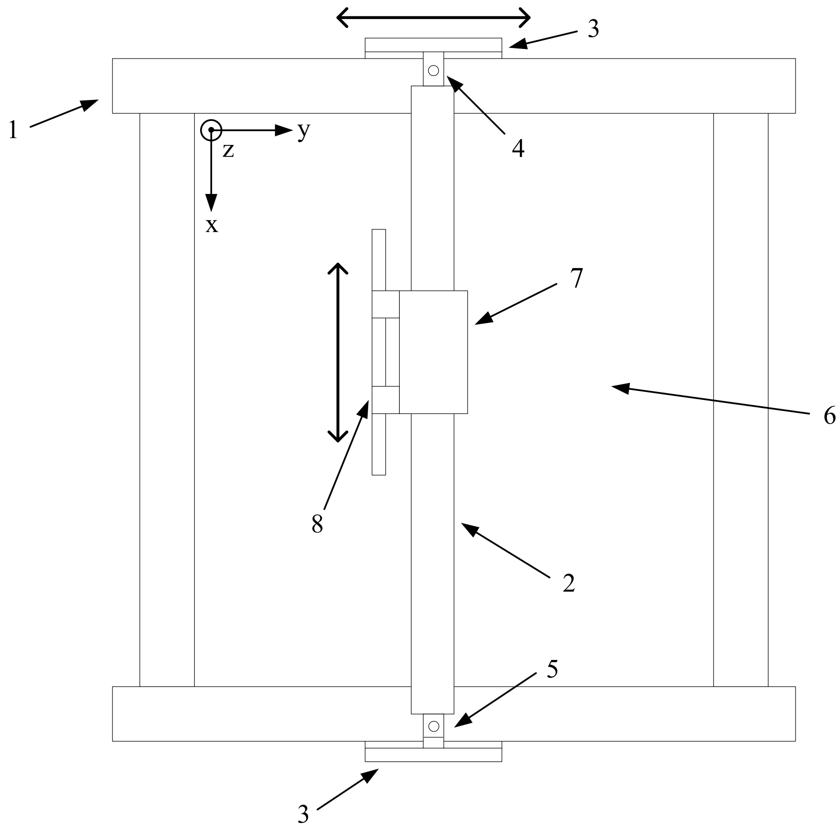


Figure 1. An overview of the iterative approach applied.

2 Design of the Machine

Knowledge and understanding about the general design of the studied machine is needed for both the theoretical modelling and experimental investigation. The focus is put on the mechanical parts that are considered to influence cutting precision.

A schematic picture of the machine can be seen in figure 2. The main parts are indicated in the figure. The machine has two axles of motion in the horizontal plane and a working area of about three by three metres.



1. Stand.
2. Boom.
3. Carriage.
4. Rotational joint.
5. Combined rotational and translational joint.
6. Water container.
7. X-unit.
8. Z-unit.

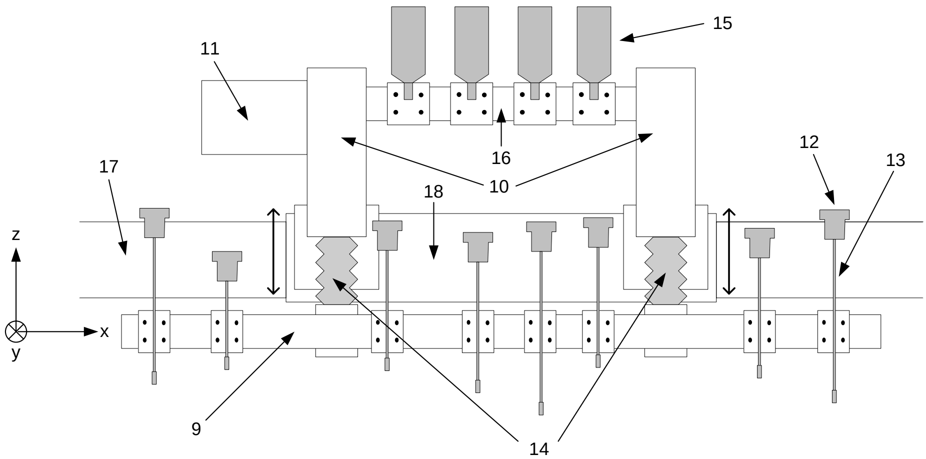
Figure 2. Top view of the studied machine.

The foundation of the machine is the stand (1), which consists of beams that are welded together. The boom (2) is coupled to the stand via carriages (3). The two carriages are attached to the stand using runner blocks and y-guides. They are individually controlled via ball screws driven by electric motors. The boom is connected to the carriages with joints, (4) and (5). The first end has a rotational joint around the z-direction. The second end has a rotational joint around the z-direction combined with a translational joint along the x-direction. This design facilitates the positioning precision of the machine.

The water in the container (6) absorbs the kinetic energy remaining in the water jet after cutting through the work piece. The work piece is placed on a grid located on the top of the water container. The water container is not mechanically connected to the rest of the machine.

The main part of the x-unit (7) is the x-unit casing, which consists of plates that are assembled with screws. The x-unit also includes other parts such as an electronics box and pressurised water distributors. Runner blocks and an x-guide allows for translation along the boom. This motion is driven by an electric motor via a ball screw. The x-guide and the boom are covered by flexible bellows during normal operation.

The z-unit (8) is attached to the x-unit with screws. The main part of z-unit is the z-unit casing, which consists of plates that are screwed together. A schematic picture of the z-unit can be seen in figure 3.



9. Cutting head carrier beam.
10. Z-unit casings, containing racks, z-guides and runner blocks.
11. Electric motor, z-direction.
12. Cutting heads.
13. Acceleration pipes.
14. Bellows.
15. Abrasive medium dispensing apparatuses.
16. Dispensing apparatus carrier beam.
17. Boom.
18. X-unit.

Figure 3. Front view of the studied machine.

The z-guides, attached to the cutting head carrier beam (9), allow for motion in the z-direction via runner blocks, located inside the z-unit casings (10). This makes it possible to adjust the distance between the cutting heads and the work piece. The motion of the z-guides is driven by an electric motor (11) via racks and pinions.

The cutting heads (12), attached to the cutting head carrier beam, direct the water jets against the work piece. The height of each cutting head can be individually adjusted by changing the mounting position on its acceleration pipe (13).

The parts of the z-guides that are located between the z-unit casings and the cutting head carrier beam are covered by flexible bellows (14). Abrasive medium dispensing apparatuses (15), from here on called dispensing apparatuses, are placed at the top of the z-unit. These are attached to the z-unit via a carrier beam (16).

3 Modelling

A theoretical model of the studied system is developed using the commercial finite element software I-deas (EDS PLM Solutions [7]). Two models are developed, an initial model with high abstraction level is built before the experimental investigations are carried out to give a rough understanding of the machine dynamics and to be used for pre-test decisions (see chapter 4). The knowledge gained during the experimental investigation is used to build a more realistic model of the machine, incorporating more parts and using a more detailed description of the physical relationships between the included parts. This model is updated to correlate better with experimental results. The final model is described below.

A straightforward modelling procedure to be suited for analysis early in the product development process is strived for. The aim is to find a procedure that is suited for computationally efficient simulations in future studies of this type of machine. Further, the focus is on vibration characteristics and not on detailed stress analysis. Beam elements, lumped mass and mass moment of inertia elements, springs and rigid elements are therefore used. A graphical representation of the developed model can be seen in figure 4.

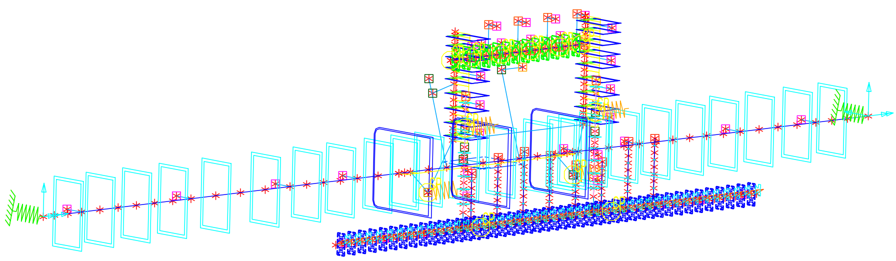


Figure 4. Graphical representation of the finite element model.

The boom, the z-guides, the cutting head carrier beam, the dispensing apparatus carrier beam and the z-unit casings are all long and slender structures and are therefore modelled using beam elements.

All internal parts in the z-unit casings, such as for example racks and runner blocks, are modelled using lumped mass elements as they are assumed to be rigid. The dispensing apparatuses are modelled using lumped mass and mass moment of inertia elements. These elements are coupled to the dispensing apparatus carrier beam using rigid elements and rotational springs. The spring elements allow rotation around the x-direction and accounts for the flexibility of the brackets holding the dispensing apparatuses. The spring coefficient is determined experimentally. The cutting heads, except for the acceleration pipes, and their mountings on the cutting head carrier beam are modelled using lumped mass and mass moment of inertia elements. The acceleration pipes are modelled using beam elements. The studied system includes four dispensing apparatuses and eight cutting heads.

The x-unit casing is modelled using beam elements although its flexibility is not considered to influence the results. Beam elements are used because it is an easy way to get a correct mass and mass moment of inertia distribution. Parts attached to the x-unit, such as the electronics box and the pressurised water distributors, are accounted for by lumped mass elements. The stiffness of the connections between the x-unit and the boom (bolted joints and couplings between runner blocks and x-guide) and the coupling between the runner blocks and the z-guides are both modelled using rotational springs.

The stiffnesses of these springs are adjusted when updating the model since these are considered to be the most uncertain parameters in the model. The springs are only allowed to rotate around the x-direction. Transducers used during the experimental investigation are accounted for by lumped mass elements. In total, the finite element model consists of 399 elements and 386 nodes.

The boundary conditions for the boom are applied to comply with its joints described in chapter 2, with the exception of the y-direction where springs are inserted to account for the flexibility in the remaining parts not included in the model (for example bending stiffness of carriages and axial stiffness of ball screws). The Lanczos solver is used to solve for the natural frequencies and corresponding mode shapes of the derived model.

4 Experimental Investigations

An experimental modal analysis is performed in order to investigate the dynamic characteristics of the studied machine. The objectives are increased knowledge of the dynamic behaviour of the system and to obtain reference data for updating of the theoretical model.

The experimental set-up is shown in figure 5. The actual boundary conditions present during normal operation of the machine are used during all measurements.

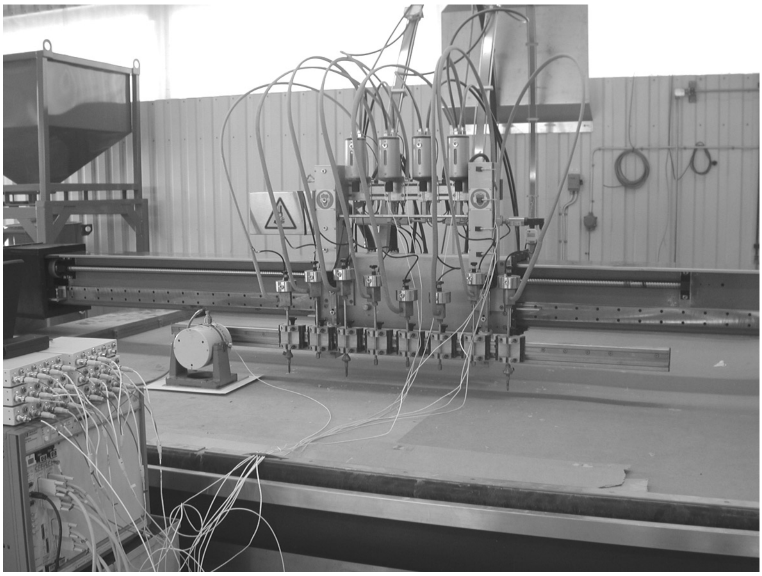


Figure 5. The experimental set-up.

The machine is excited with a shaker via a force transducer and a stinger. To avoid possible leakage problems a burst random force signal is used. The excitation point is chosen consulting the initial finite element model of the machine. The strategy, described by Ahlin and Brandt [8], is to select the excitation point that best excites the least significant mode. The excitation point is chosen on the cutting head carrier beam, see figure 5.

The responses are measured using piezoelectric accelerometers. The number and locations of measurement points are chosen on basis of an AutoMAC calculated using the initial finite element model. The Modal Assurance Criterion (MAC) is a tool to numerically quantify the degree of conformance between two sets of mode shapes. A value of one indicates perfect correlation while a value of zero indicates no correlation. Using the AutoMAC the mode shapes are correlated against themselves. The criterion used when deciding on suitable measurement points is that the off-diagonal terms of the AutoMAC should be as low as possible to get a good separation of the modes. In total 48 evenly distributed measurement points are chosen.

A Hewlett Packard VXI measuring system is used to acquire the experimental data and I-DEAS Test [9] is used as a signal analyser to obtain Frequency Response Functions (FRFs). Due to hardware limitations only 7 out of the 48 measurement points is covered in one measurement round while triaxial accelerometers are used. Since different mass loading of the structure may cause serious problems when extracting the modal parameters (Maia et al. [10], [8]), dummy masses are used.

No significant modes are present above 100 Hz, see figure 6. The frequency range of interest is therefore set to be between 0 and 100 Hz. In this frequency interval the coherence is generally good except at some anti-resonances.

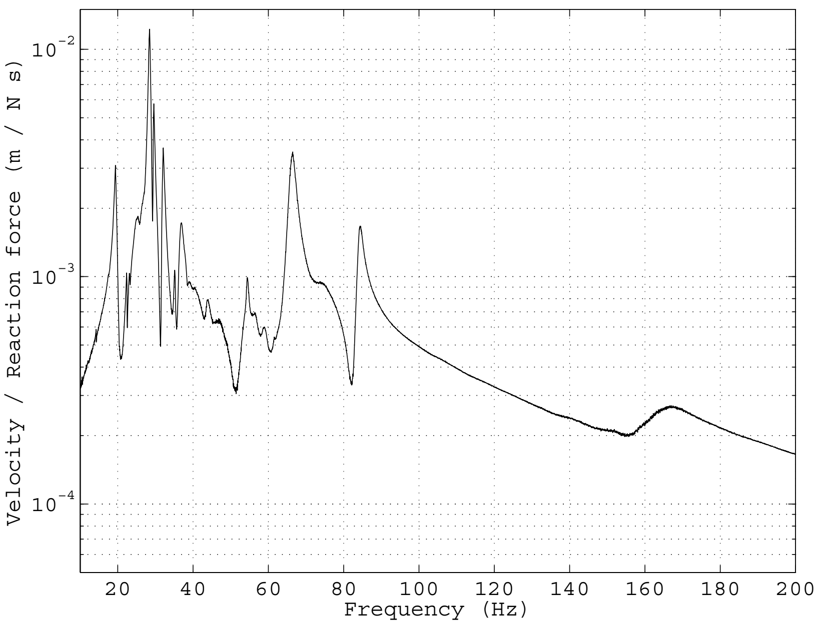


Figure 6. A typical FRF for the studied machine.

The quality of the experimental set-up is further investigated by checking the driving point FRF for consistency and by performing a linearity check. FRFs are independent of the excitation level for linear structures. However, this is not the case for non-linear structures. Three measurements with successively doubled excitation force are performed and the measured FRFs are overlaid to investigate the linearity assumption. The difference in the responses shown in figure 7 indicates that the dynamic behaviour of the studied machine is slightly non-linear. However a linear approximation is considered appropriate for the present study.

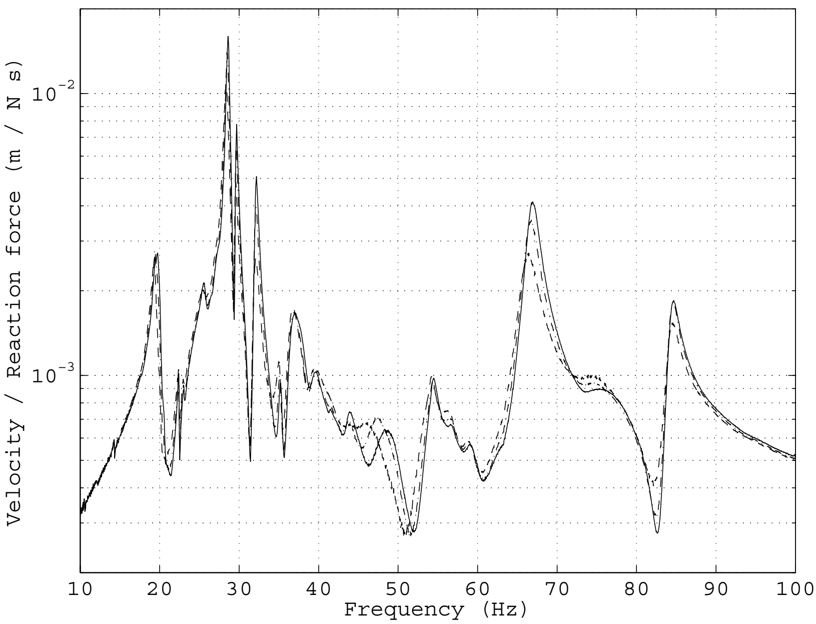


Figure 7. FRFs overlaid for low (solid line), medium (dash-dot line) and high (dashed) excitation force.

The measured FRFs are exported to MATLAB [11] where the modal parameter extraction is performed using the ModalTools Toolbox [12] developed by Saven EduTech AB. To obtain parameters of good quality only a part of the complete frequency range of interest is analysed at a time. Measurement data of low signal-to-noise ratio are not included in the modal parameter extraction to further improve the quality of the parameters.

5 Results and Discussion

MAC values, natural frequencies and visual examination of mode shapes are used for comparison of the theoretical and experimental results.

A MAC-matrix is calculated for appraisal of the correlation between identified experimental modes and their theoretical counterparts, see figure 8.

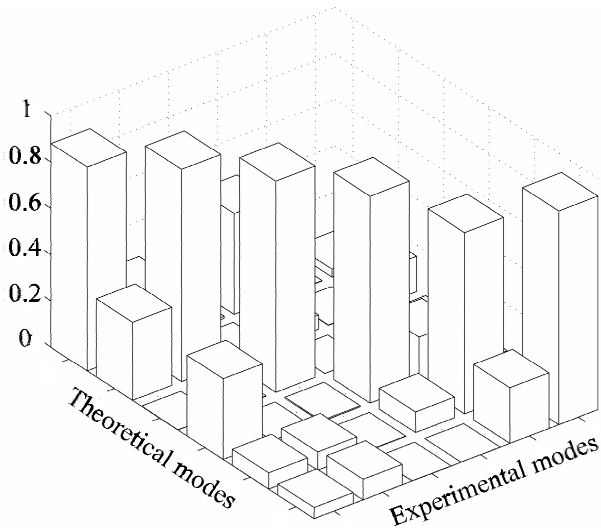


Figure 8. The MAC-matrix showing correlation between theoretical and experimental mode shapes.

It is found very difficult to obtain good correlation between experimental and theoretical results considering the cutting heads and dispensing apparatuses. These local responses are also found large in comparison with the rest of the machine. The MAC-matrix is therefore very dependent on these responses if they are included. Due to the above reasons they are not taken into account when calculating the MAC-matrix for appraisal of the global response correlation of the machine.

As a consequence, some of the off-diagonal terms become rather large, which can be seen in figure 8. The reason for this is that responses of the cutting heads and the dispensing apparatuses are important to separate the mode shapes from each other. The diagonal MAC-values are however between 0.78 and 0.92 for the selected modes, which indicates good correlation considering global responses.

A comparison between theoretical and experimental natural frequencies is shown in figure 9. The diagonal line represents perfect matching. The crosses indicate the frequency match for each correlated mode pair.

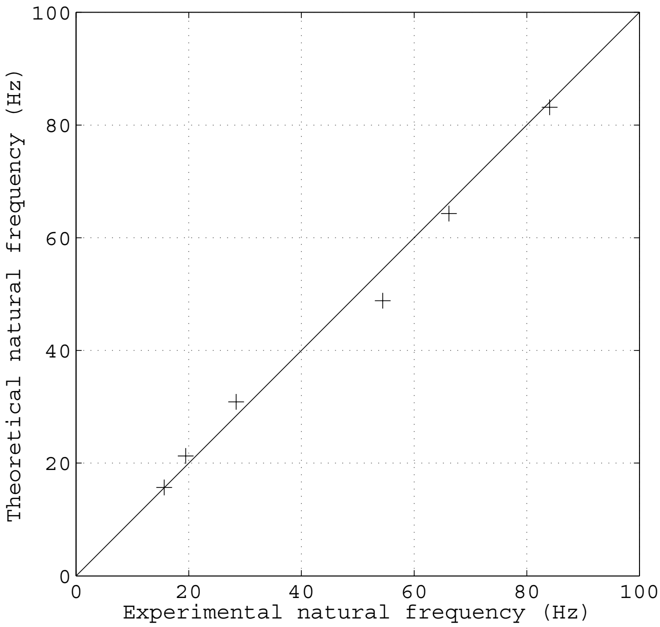


Figure 9. Comparison of theoretical and experimental natural frequencies.

The maximum difference in corresponding natural frequencies is below 10 per cent except for one mode. The small and randomly distributed scatter is normal for this type of modelling and measurement process (Ewins [13]). The results are summarised in table 1.

Table 1. Results.

Mode	Experimental		Theoretical	Correlation[a] (%)	MAC
	Frequency (Hz)	Damping (%)	Frequency (Hz)		
1	15.6	2.85	15.7	0.36	0.88
2	19.5	1.14	21.3	9.42	0.92
3	28.4	0.86	30.9	8.68	0.91
4	-	-	39.8	-	-
5	54.4	0.63	48.8	-10.3	0.90
6	66.2	1.53	64.3	-2.87	0.78
7	-	-	72.2	-	-
8	84.1	1.00	83.2	-1.05	0.92

[a] The correlations are calculated before rounding off.

The damping values are given as the fraction of critical damping and the correlation values are the relative differences between experimental and theoretical natural frequencies.

Mode four and seven predicted by the theoretical model are not found experimentally. The reason for this is believed to be that the main movement of these modes are in the x- and/or z-direction(s) while the machine is only excited in y-direction during the experimental investigations.

Except for the modes presented above several modes associated with large local responses of the cutting heads and dispensing apparatuses are seen both during the experimental and theoretical modal analysis in the frequency interval between 20 and 50 Hz.

6 Conclusions

The dynamic characteristics of a water jet cutting machine are investigated in this work. An iterative approach combining theoretical and experimental analysis is used in order to develop a structural dynamics model of the machine. Due to the complex dynamic characteristics of the machine it is very difficult to create a model that reflects reality without performing experimental investigations, which are often a complex task in its own. It is therefore believed that the iterative approach used in this work is efficient when analysing this kind of systems.

The good correlation between experimental and theoretical results implies that the developed model can be used with confidence as a basis for further work on creating a reliable model to be used in an analysis tool for predicting the dynamic characteristics of new designs.

The final aimed for application of the theoretical model built in this work is to be used in real-time simulations of the virtual machine. To be suited for this the model must be computationally inexpensive. The presented model therefore consists of computationally effective finite elements, such as beam elements and lumped mass elements. The good correlation implies that the level of detail in the model is sufficient to describe the complex dynamic characteristics of the machine. However, further simplifications should be investigated and strived for in the future.

To enable experimental identification of modes also in the x- and z-directions a different excitation direction and possibly also multiple shaker excitation should be considered.

Actual mechanical properties of couplings between machine parts are identified as both difficult and necessary to consider in the modelling. On basis of the performed linearity check the studied machine is considered as linear in the present study. The validity of this assumption should however be more thoroughly investigated in future studies.

Acknowledgements

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Paper G

A Virtual Machine Concept for Real-Time Simulation of Machine Tool Dynamics

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A Virtual Machine Concept for Real-Time Simulation of Machine Tool Dynamics

Anders Jönsson, Johan Wall, Göran Broman

Abstract

When designing CNC machine tools it is important to consider the dynamics of the control, the electrical components and the mechanical structure of the machine simultaneously. This paper describes the structure and implementation of a concept for real-time simulation of such machine tools using a water jet cutting machine as an application. The concept includes a real control system, simulation models of the dynamics of the machine and a virtual reality model for visualisation. The real-time capability of the concept, including the simulation of electrical and rather detailed mechanical component models is proofed. The validation process indicates good agreement between simulation and measurement, but suggests further studies on servo motor, connection and flexibility modelling. However, already from the initial simulation results presented in this paper it can be concluded that the influence of structural flexibility on manufacturing accuracy is of importance at desired feeding rates and accelerations. The fully automated implementation developed in this work is a promising base for dealing with this trade-off between productivity and accuracy of the manufacturing process through multidisciplinary optimisation.

Keywords: *Dynamics, Machine tools, Mechatronics, Real-Time simulation, Virtual machine.*

1 Introduction

In general, early prediction of product characteristics facilitates integration of specialised disciplines in a concurrent engineering process and promotes better design decisions; see, for example, [1]. Efficient tools for assessment of suggested design solutions are therefore desired for various product categories. For machine tool systems, a virtual machine concept holds the potential of greatly facilitating testing of a large number of design suggestions at an early stage of the development process. It could also facilitate, for example, safety evaluation, early education of operators and marketing. Building, testing and adjusting full physical prototypes for the above purposes is very costly and impractical.

More specifically, increasing market demands on productivity and accuracy of the manufacturing process makes it necessary to use new methods and tools when developing CNC machine tools [2-8]. Increased feeding rates and accelerations of the machine parts imply stronger excitation of oscillation. This makes it more difficult to follow a desired path, which is, of course, inherently negative for the manufacturing accuracy. The virtual machine concept presented in this paper aims at dealing with this trade-off during product development. This includes the design of the electrical and mechanical parts as well as the control system, and, of course, whole systems optimisation.

In [2], the benefits of using a virtual machine concept in the later stages of the product development process are shown. In that work, a real-time simulation is implemented together with a detailed virtual reality (VR) environment. The relatively simple simulation models are mostly event-driven and the motion of the mechanical parts is determined only from kinematics. Examples of virtual machines built using that concept are Programmable Logic Controller controlled textile machines, packaging machines, CNC laser cutting machines and grinding machines. In [3], benefits of a concept for control system development are focused.

In [2, 3], the need to improve the structural dynamics simulation capabilities is pointed out as crucial for the success of a virtual machine concept. This can also be concluded from, for example, [4-8]. In this work, the virtual machine concept will therefore be enhanced by more detailed structural models, including finite element (FE) models. The challenge is to combine this higher level of detail with short simulation cycle times. To achieve this in a meaningful way, modelling and simulation procedures have to be as computationally inexpensive as possible while still being accurate enough for the characteristics they are supposed to describe, which will also facilitate optimisation and real-time virtual reality visualisation. To be useful for multidisciplinary design optimisation (MDO), the aim is a fully automated implementation.

2 Concept Description

Figure 1 shows the principal parts of the virtual machine concept. Since a real control system is used in this work, the machine simulation must run in real-time. This implies that the simulation cycle time needs to be equal to or less than the control system cycle time, which puts special requirement on the simulation efficiency.

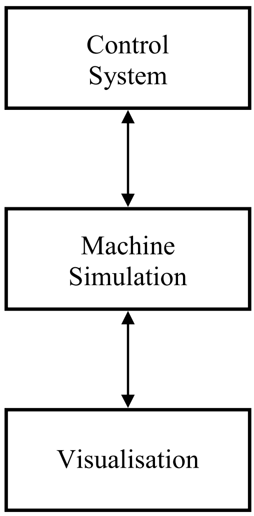

Figure 1. The virtual machine concept.

The specific application studied in this work is a water jet cutting machine, including a control system, amplifiers, motors, mechanical parts and sensors. In the virtual machine, these parts are replaced by virtual parts represented by models, see figure 2. Simulink is used for designing the simulation models. The models have defined inputs and outputs, which makes it possible to replace a model with a new improved model without changing the machine simulation structure. This enables an easy reuse of the simulation models.

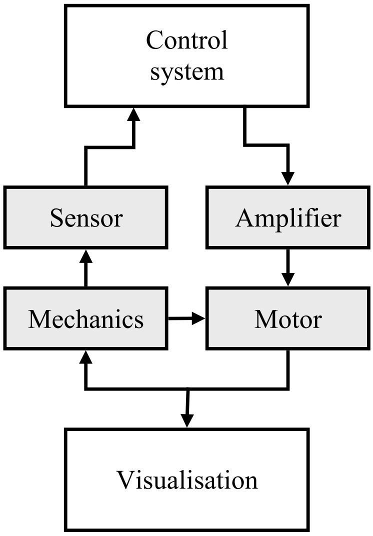

Figure 2. Simulation models used in the concept, marked as shaded.

The control signal from the control system is the input signal for the real-time machine simulation. The amplifier is idealised as a perfect amplifier. The motor is modelled by using data from the manufacturer. The motor torque is dependent on the rotational velocity of the motor, which implies a direct mutual coupling between this torque and the mechanics of the machine that this torque is driving.

The rotation of the motor axes transforms through belt drives and ball screws into linear motion of the carriages, which slides along guide ways that are considered rigid, see figure 3. The carriages, the boom and the cutting head assembly are treated as flexible and are described by FE models. In this work, the x- and y-directions are studied.

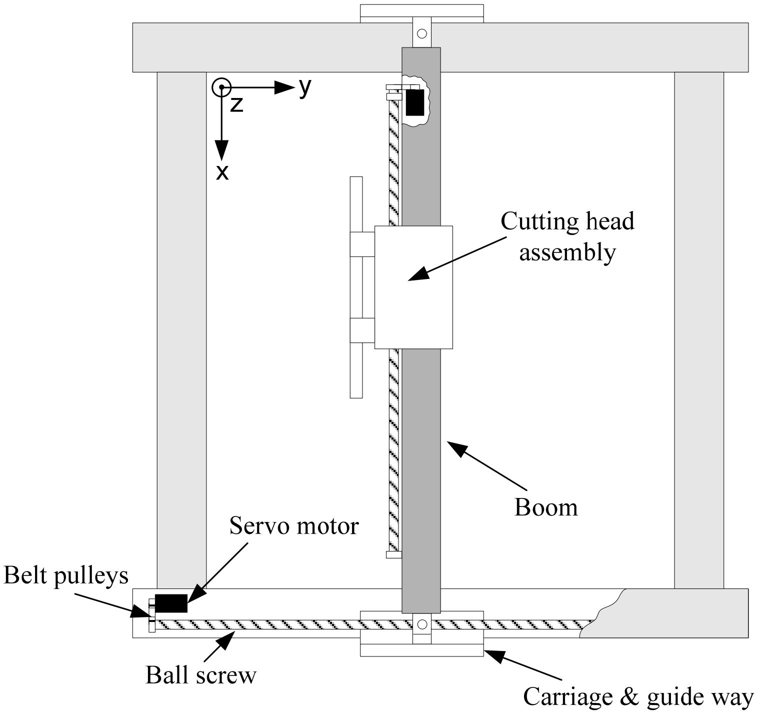

Figure 3. Schematic top-view of the water jet cutting machine.

The rigid parts of the structural model include the rotary moment of inertia of the servomotor, J_m, belt pulleys, J_1 and J_2, and the ball screws, J_b. The belt between the pulleys is considered mass less and inelastic. Friction in the ball screws and all bearings is neglected.

With the assumptions above, and considering the belt pulley radius ratio, R_2 / R_1, and the pitch of the ball screws, p, the rigid body parts can be described as an equivalent mass, m_{eq}, added to the model of the flexible structure and acted upon by an equivalent force, F_{eq}, see figure 4.

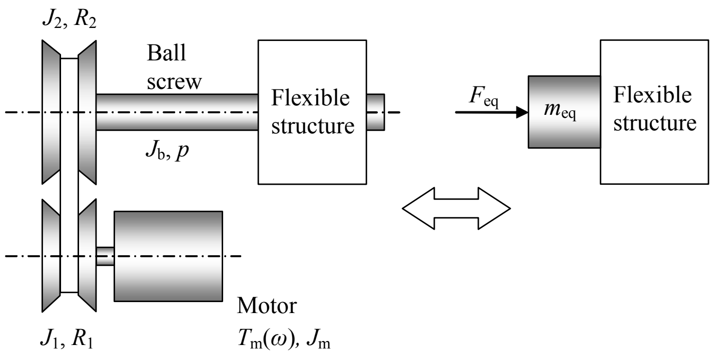

Figure 4. Equivalent mass and force model.

The latter quantities are then

$$m_{eq} = J_{tot}\left(\frac{2\pi}{p}\right)^2 = \left(J_b + J_2 + (J_1 + J_m)\left(\frac{R_2}{R_1}\right)^2\right)\left(\frac{2\pi}{p}\right)^2 \tag{1}$$

$$F_{eq} = T_m(\omega)\left(\frac{R_2}{R_1}\right)\left(\frac{2\pi}{p}\right) \tag{2}$$

where $T_m(\omega)$ is the torque of the motor and ω is the angular velocity of the motor. The resulting angular velocity from the simulation is output as a standard incremental encoder sensor signal to the analogue interface.

During machine operation, the cutting head assembly move relative to the boom, implying that the dynamic properties of the machine vary. To deal with this problem a sub-structuring approach [9] is applied where the cutting head assembly and the boom are described as subsystems, schematically shown in figure 5.

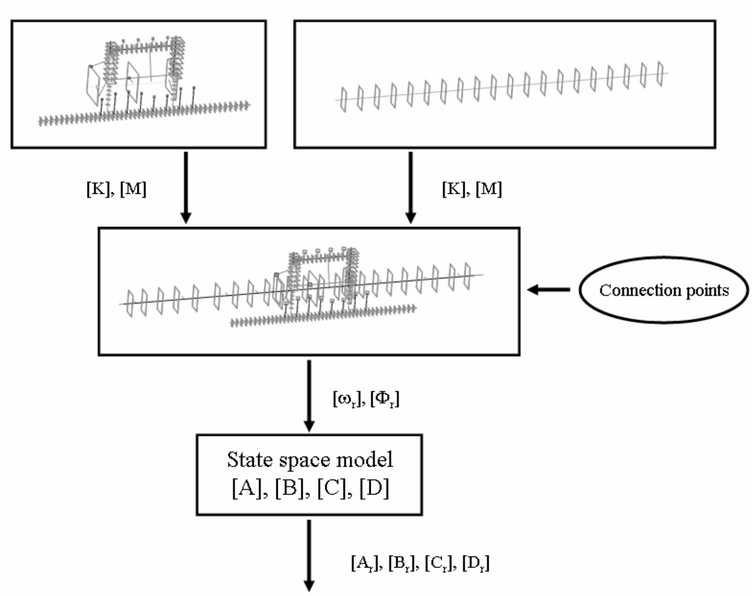

Figure 5. Sub-structuring and reduction process.

The contact points between the cutting head assembly and the boom is assumed to depend only on the motor rotation driving that motion. The complete model in modal form is assembled from the subsystems at the current points of contact. To achieve computational efficiency, the complete model is reduced by only retaining those modes that have a major influence on the dynamic response in the frequency range of interest. The modal model is converted into a state space model [10] to facilitate the implementation in Simulink. The state space model is reduced further by only retaining the excitation and response DOFs of interest. These two reduction steps decrease the number of DOFs in the final model significantly.

The reduced model is verified by comparison to the full model for different positions of the cutting head assembly. An example of this verification is shown in figure 6, where the cutting head assembly is at the middle of the boom. Modal damping from experimental investigations is added to the model.

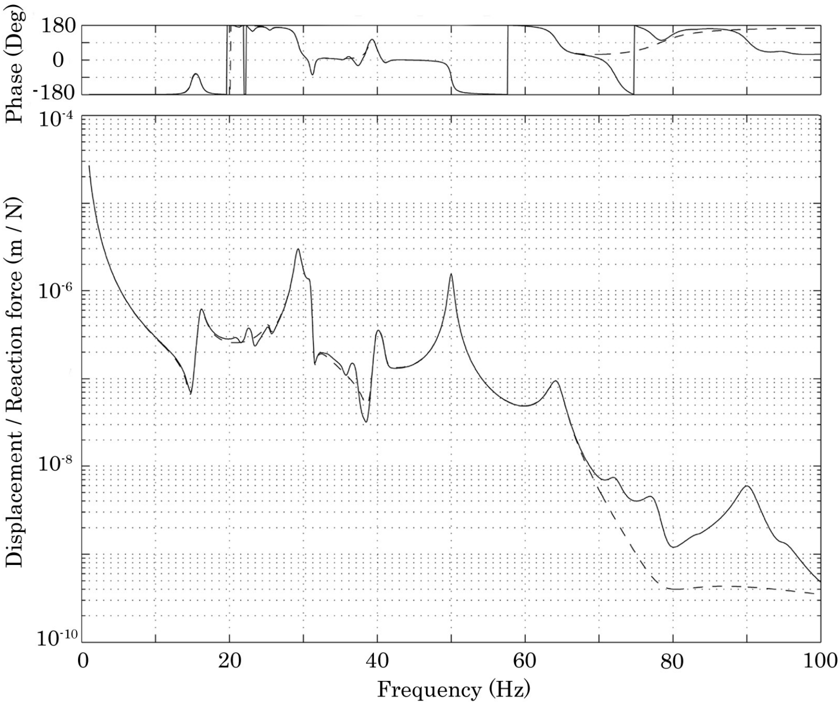

Figure 6. FRF for reduced model (dashed) and full model (solid).

The above FE-modelling and reduction procedure is performed beforehand for several positions of the cutting head assembly. At every instant of time during the simulation, the matrices of the state space model are selected among the matrices obtained beforehand. Doing these more computationally expensive calculations beforehand, helps significantly in keeping the cycle time down during the actual simulation.

A usual cycle time for the control used in this work is 250 μs. Therefore, the simulation is run on a real-time operating system, xPC Target. Furthermore, the Simulink model is converted into C-code using Real Time Workshop, and compiled into a real-time executable. As the simulation runs with a fixed cycle time, only fixed time-step solvers are feasible. The classical fourth-order Runge-Kutta integration scheme is used in this work.

3 Concept Implementation

The concept implementation is fully automated; featuring: building of simulation models, setting of simulation and control parameters, start and stop of simulation and NC-program, and post processing of simulation results. This implies that the parameters in the models are changeable and that the simulations run without human interaction. This includes parameters for the CNC, rigid body model, servomotor and sensor model and the FE-model. Figure 7 shows an overview of the hardware structure of the implementation.

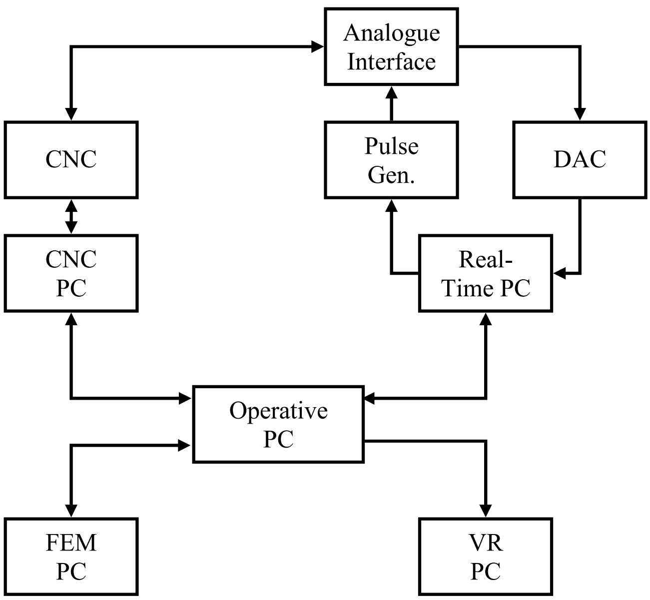

Figure 7. Concept implementation overview.

A PC is used for managing the total virtual machine concept. On that operative PC, a program runs that determines which parameters that is to be set, if a new FE-model has to be calculated, when to run the NC-program, when to start and stop the real-time simulation, etc. This PC also stores and post processes and links simulated motion data to the VR software.

A common control system for water jet cutting machines, GE Fanuc 160i-mb, is used in the set-up. The communication with the control system is implemented in the concept using an in-house developed application that runs on the PC connected to the CNC. This application starts the test cycle, samples test data from the CNC, and writes data to a file for further processing. This software communicates with the CNC using the Fanuc Open CNC Drivers and Libraries (FOCAS2) on a High Speed Serial Bus (HSSB). The servo control information is output as digital signals on the Fiber-optic Serial Servo Bus (FSSB). The FSSB is also used to input the encoder velocity information from the analogue interface.

A standard GE Fanuc analogue interface is used between the CNC and the simulation model. The analogue interface outputs a voltage velocity control signal. The analogue velocity control signal from the analogue interface is digitalised using a National Instruments PCI6052-E data acquisition card (DAC). This digitalised signal is the input to the real-time simulation of the machine dynamics.

The inputs to the analogue interface are the pulse signals from the pulse generator. Since a real encoder signal contains parallel signal pulses up to several MHz, a special hardware pulse generator produces the output information from the simulation [11]. In this work, a DEVA011 quadrature signal generator card accomplishes this. This card can produce standardised encoder pulse signals for three axes up to 10 MHz.

To be able to use this card on an xPC Target PC, a device driver is developed for this hardware [11]. In the mask of the S-Function, all constant parameters for the card – such as the pulses per revolution and the width of the reference marker of the simulated encoder – are defined.

The resulting motion is visualised using a VR-model of the machine, see figure 8. The VR-model gives an instant feedback of the motion of the machine, making it easy to detect if something is obviously incorrect. The VR-model is based on CAD data from the machine manufacturer. The data is converted into VRML format and imported into the VR-software Mediator. During simulation, a separate Simulink model runs that gathers motion data from the real-time simulation and transmits it to the VR-software.

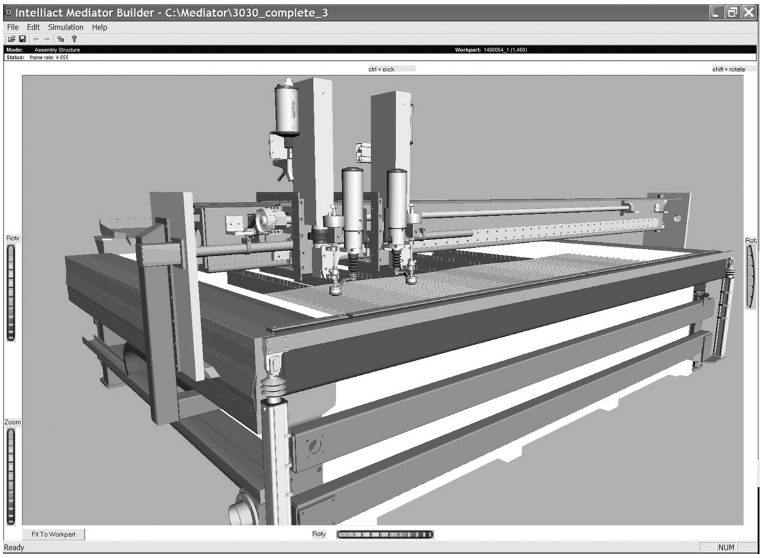

Figure 8. VR-model for visualisation of simulated motion of the machine.

The level of detail of the VR-model is quite high in this work, but, of course, an even more refined model, including, for example, textures and interaction can be used if desired. This can be the case if the virtual machine is to be used for marketing purposes or operator training.

4 Validation and Simulation

For the real-time simulation to be useful, it has to be validated. Validation of multidisciplinary systems as the water jet cutting machine is a delicate task. In this work a modular validation approach is used. Instead of trying to validate all models at the same time, the validation procedures of the servo motors, the mechanical structure and the sensors are separated. The validated models are then combined and used in the virtual machine simulation.

The sensor model and its driver is validated by running the virtual servo motor with a certain speed and observing the pulse frequency output from the pulse generator card using a digital oscilloscope. The validation shows very good agreement between the pulse frequency from the sensor model and the demanded pulse frequency.

To validate the servo motor model, a virtual servo motor axis is run and compared to results from a real servo motor axis. The servo parameters in the control system are identical for the virtual and the real servo motor axis. A virtual and a real disc are attached to the respective axes, corresponding to an equivalent medium rotary moment of inertia that needs to be driven in a typical water jet cutting machine.

The virtual servo motor axis is simulated within the same concept as the complete virtual machine. The real servo motor axis set-up consists of a digital amplifier, a servo motor with an integrated encoder, and a metal disc. In the real set-up, a digital interface from the control to the amplifier is used. Comparison between the real and virtual axes motion shows good agreement. The noticed difference is assumed to originate mainly from the usage of an analogue interface for the virtual axis and a digital interface for the real axis.

To validate the modelling and simulation of the flexible parts, measurements on a real water jet cutting machine is performed [12]. To prepare for the first measurement set-up, an initial model with high abstraction level is built and simulated. The knowledge gained from this first measurement, is used to develop a more detailed model, including more parts and the connections between the parts. An improved second measurement set-up is designed with the knowledge from earlier measurement and the new model, see figure 9. The second measurement is then used to build a further improved model. The correlation between this latter model and the real machine is good. For future work, the investigation so far indicates that further model improvement is primarily related to the servo motors, the connections between the machine parts and the flexibility of the parts now considered rigid.

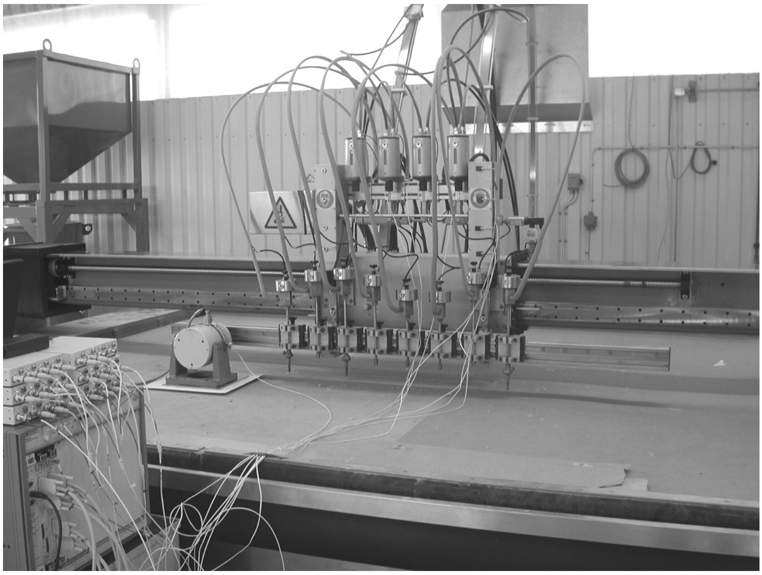

Figure 9. Measurement set-up for validation of flexible structure model.

The validated models of the sensors, the motors and the flexible structure are used in the concept and enable a full machine simulation. The concept is thus well suited to determine the accuracy of the cutting head motion, since simulation results can be extracted from any position on the machine. In figure 10, the position of a point on the cutting head holder beam (the cutting head position) as experienced from the motor encoder is compared to the actual (simulated) position of this point for a simple forward-rest-backward motion scheme. It is clear that in this case the flexibility is significant for the accuracy.

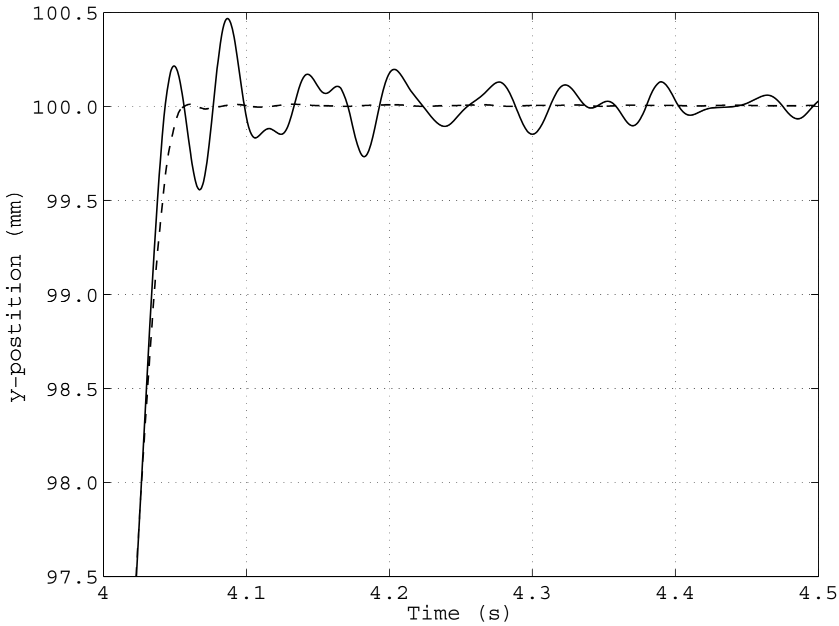

Figure 10. Experienced (dashed) and actual (solid) position of the cutting head.

5 Discussion and Conclusion

This work brings the virtual machine concept significantly towards becoming a useful tool for supporting machine tool systems development.

The concept facilitates total systems simulation, including the dynamics of the control and the electrical and mechanical parts of the machine. This makes it possible to assess systems design suggestions at an early stage of development and to perform systems optimisation. The structure of the concept allows for easy adjustment or replacement of component models, which facilitates transfer and reuse of knowledge between development projects. More refined virtual reality models can also easily be included to facilitate, for example, safety evaluation, early education of operators and marketing.

Through the specific combination of in-house developed and commercial software and hardware, and through the employed modelling and simulation techniques, it has been possible to achieve the short cycle time necessary for real-time simulation even with the significantly more detailed models of the mechanical parts used in this work, compared to the ones used in earlier works. This proves the potential of using the virtual machine concept for studying the influence of the flexibility of the mechanical structure on the manufacturing accuracy for various manufacturing conditions.

The validation process indicates good agreement between simulation and measurement, but suggests further studies on servo motor, connection and flexibility modelling. However, already from the initial simulation results presented in this paper it can be concluded that this influence is of importance at desired feeding rates and accelerations, which are directly related to the demands on productivity.

With expected increasing demands on both productivity and accuracy, it becomes even more important for design engineers to be able to study those aspects at an early stage of machine tool development. The fully automated implementation of the virtual machine concept developed in this work is a promising base for dealing with this trade-off between productivity and accuracy through MDO. An unexploited potential for improving both productivity and accuracy becomes clear immediately from the possibility to now let the control system work with respect to the actual (simulated) motion of the cutting head instead of, as presently, with respect to the motion of the cutting head as experienced through the motor encoder signal (and thus not accounting for the mechanical flexibility).

To improve the concept further, a complete validation against a real water jet cutting machine should be carried out. The servo motor behaviour should be further investigated, friction at connections should be considered and the influence of flexibility of the parts now considered rigid should be investigated. Furthermore, the influence of the forces due to the water jet should be investigated.

Acknowledgements

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Paper H

Introductory Design Optimisation of a Machine Tool Using a Virtual Machine Concept

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Introductory Design Optimisation of a Machine Tool Using a Virtual Machine Concept

Johan Wall, Johan Fredin, Anders Jönsson, Göran Broman

Abstract

Designing modern machine tools is a complex task. A simulation tool to aid this, a virtual machine, has therefore been developed in earlier work. The virtual machine considers the interaction between the mechanics of the machine (including structural flexibility) and the control system. This paper exemplifies the usefulness of the virtual machine as a tool for product development. An optimisation study is conducted aiming at improving the existing design of a machine tool regarding weight and manufacturing accuracy at maintained manufacturing speed. The problem can be categorised as constrained multidisciplinary multi-objective multivariable optimisation. Parameters of the control and geometric quantities of the machine are used as design variables. This results in a mix of continuous and discrete variables and an optimisation approach using a genetic algorithm is therefore deployed. The accuracy objective is evaluated according to international standards. The complete systems model shows non-deterministic behaviour. A strategy to handle this based on statistical analysis is suggested. The weight of the main moving parts is reduced by more than 30 per cent and the manufacturing accuracy is improvement by more than 60 per cent compared to the original design, with no reduction in manufacturing speed. It is also shown that interaction effects exist between the mechanics and the control, i.e. this improvement would most likely not been possible with a conventional sequential design approach within the same time, cost and general resource frame. This indicates the potential of the virtual machine concept for contributing to improved efficiency of both complex products and the development process for such products. Companies incorporating such advanced simulation tools in their product development could thus improve its own competitiveness as well as contribute to improved resource efficiency of society at large.

Keywords: *Machine tools, Mechatronics, Non-deterministic, Optimisation, Product development, Virtual machine.*

1 Introduction

On the increasingly competitive global market, users of machine tools demand increased accuracy and efficiency. This forces machine tool developers to incorporate new methods and tools in their development processes. Virtual experimentation (advanced simulation tools) seems promising for addressing these new demands while at the same time attaining other benefits, such as shortened time-to-market. This has been shown in other areas of engineering; see, for example, [1].

A virtual machine concept to support simulation-driven mechatronic design of CNC machine tools has therefore been developed in earlier work [2]. The virtual machine includes a real control system and simulation models of the machine having real-time capabilities. This parallel multidisciplinary design approach, simultaneously analysing the mechanics and the control, and thereby utilising interaction effects, is believed to be superior to the traditional sequential design approach [3]. Other works related to the idea of a virtual machine are, for example, [4, 5]. However, none of these incorporates detailed time-varying structural dynamics simulation capabilities.

The aim of this paper is to show the usefulness of the virtual machine concept for machine tool design optimisation. Designing a machine tool includes a wide variety of tasks ranging from selecting off-the-shelf products to designing unique parts from scratch. The design problem therefore usually consists of a mixture of continuous and discrete variables. Hence, a non-gradient based optimisation algorithm is well suited for the problem. Furthermore, in this type of problem many, often conflicting, objectives are usually present, i.e. it is a multi-objective problem. Methods able to handle this type of optimisation problem are discussed in, for example, [6, 7].

The complete multidisciplinary model of the studied mechatronic system shows a non-deterministic behaviour. Methods and strategies for non-deterministic simulations have in recent years received increased attention within the research community; see, for example, [8, 9]. The focus has been on systems with uncertain or variable model properties. However, in the virtual machine simulation, the source of non-determinism is inherent to the set-up. Statistical methods that consider this are suggested.

2 Virtual Machine Overview

The virtual machine includes a real control system, a hardware-in-the-loop (HIL) simulation of the machine and a virtual reality model for visualisation, see figure 1.

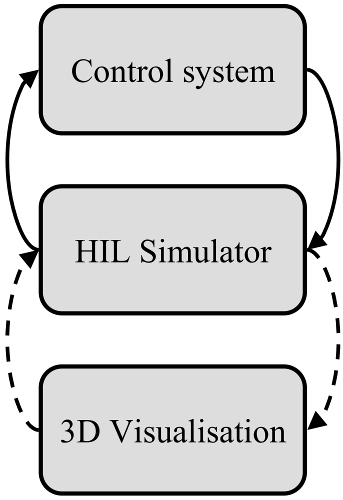

Figure 1. Overview of the virtual machine.

The HIL simulator contains a machine simulation model, I/O hardware for reading actuator control signals as well as hardware for emulation of sensors, see figure 2.

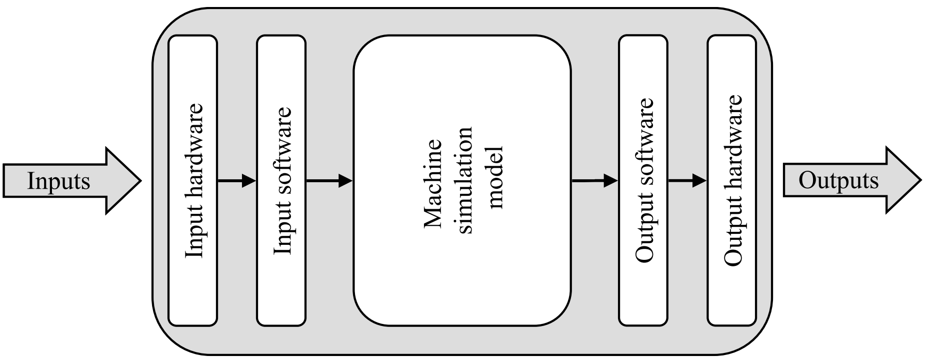

Figure 2. HIL simulator.

The machine simulation model is capable of describing the time-varying structural dynamic response of the studied machine in real-time. The need for real-time performance is because the inputs and outputs to and from the simulation have to be synchronised with a real control system. Therefore the cycle time of the simulation has to be the same as, or lower than, the cycle time of the control, in this work 250 μs. For a detailed description of the virtual machine see [2].

3 Case Study; Design Optimisation of a Water Jet Cutting Machine

This chapter describes the case study specifically while also giving general information about the chosen optimisation approach.

3.1 Water Jet Cutting Machine

Water jet cutting is a manufacturing technique that uses the erosion power of water to shape the work piece. The basic principle is to channel highly pressurised water (400 MPa or more) through a narrow nozzle in the cutting head, concentrating a high amount of energy in a small area and thereby creating massive cutting power. More information about water jet cutting can be found in [10].

A schematic of the studied water jet cutting machine can be seen in figure 3. The machine has two axes of motion in the horizontal xy-plane.

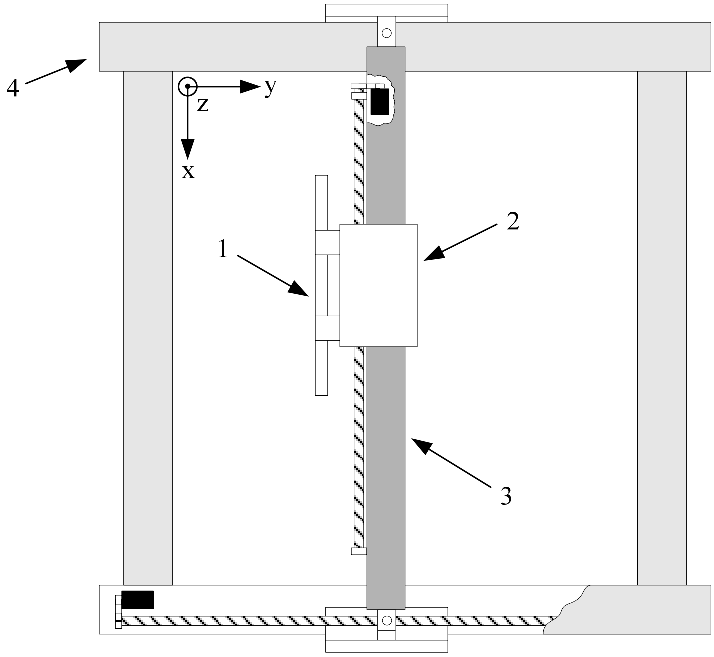

Figure 3. Top view of the studied machine.

A typical machine contains several cutting heads. In the studied machine design, the cutting heads are attached to the cutting head holder beam (1) which is mounted on the X-unit (2). The X-unit is able to move along the boom (3) enabling motion in the x-direction. The boom is able to move along the stand (4) enabling motion in the y-direction. Both axes are driven by electric motors via ball screws. A more thorough description of the machine design is given in [11].

3.2 Simulation Model

The machine simulation model contains several sub-models; an ABAQUS finite element (FE) model simulating the flexibility of the moving mechanical parts, a Simulink motor model and a Simulink multi-body model of the transmission. The complete model is built in and controlled from MATLAB.

For the simulation model to be functional in an optimisation study it has to be parameterised and automated, i.e. the optimisation algorithm must be able to influence the model by varying certain aspects of it. While this is straight forward for the Simulink (MATLAB) sub-models, tools enabling data exchange with ABAQUS is needed. This is realized through the software packages' ability to read and write ASCII-files.

The parameterisation of the FE model is based on a substructuring approach. The unique parts of the machine are isolated as subsystems. Models for these subsystems are developed and validated. Some subsystem models are dynamic in the sense that they for arbitrary model parameters, for example, geometric quantities or material properties, may be changed and re-built. Which subsystem models that are allowed to be dynamically changed and which are kept unchanged (static) depend on the choice of variables in the optimisation study. The subsystem models are then assembled into the complete FE model of the machine in MATLAB. The model is sent to ABAQUS and solved. The results are imported back into MATLAB and used as a part of the machine simulation model.

The described simulation environment, combining ABAQUS with MATLAB, is very flexible, allowing automatic simulation and assessment of different machine configurations.

3.2.1 Simulation Model Behaviour

Simulation results show a non-deterministic model behaviour. Typical results of manufacturing accuracy from a typical machine configuration can be seen in figure 4 for 1000 simulation runs.

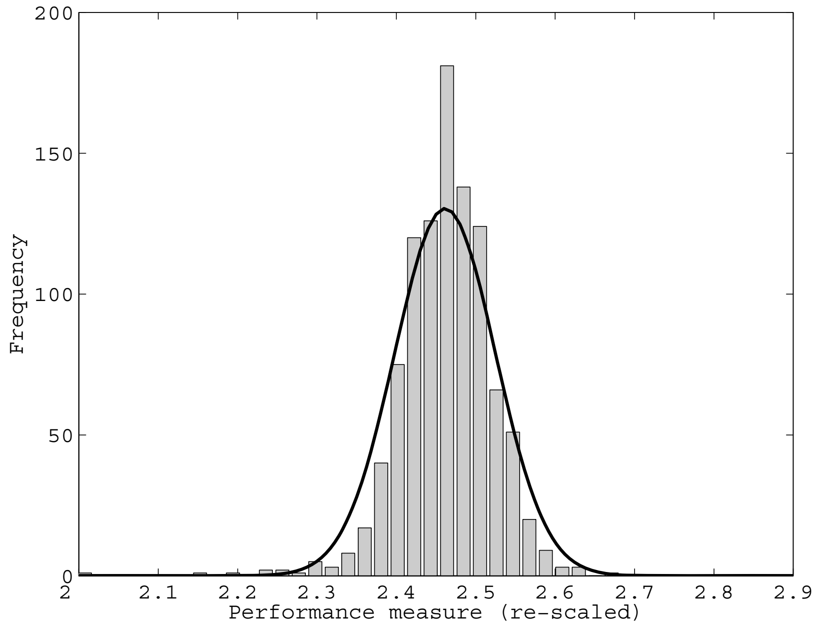

Figure 4. Histogram showing variance of typical simulation results.

The solid line is an estimated probability density function, assuming a Gaussian distribution, with the sample mean and sample standard deviation. Based on this, Gaussian distributed simulation results are assumed. The performance measure presented in figure 4 is re-scaled on request of the industrial partner. It is unit-less and does not explicitly represent the performance of the actual machine.

The variation must, of course, be considered when assessing machine tool performance. To get stable results, sufficiently many simulation runs with a given machine configuration must be carried out. To ensure this, the confidence interval of the predicted mean value is calculated. If the calculated interval is larger than a given threshold level (related to the expected magnitude of studied manufacturing accuracy) for a certain confidence level (99%), additional simulations are performed until the mean value is predicted with acceptable certainty. The confidence interval is calculated according to equation 1 [12]:

$$\overline{X} \pm t_{n-1}\left(\frac{\alpha}{2}\right)\frac{s}{\sqrt{n}} \tag{1}$$

where $\overline{X}$ is the sample mean, t_{n-1} is Student's t distribution with n-1 degrees of freedom, 1-α is the probability that the true mean value, μ, is contained within the calculated interval, s is the sample standard deviation, and n is the sample size.

From a robust design point of view, the variance of the performance measure used must also be considered in the optimisation study.

3.3 Optimisation Problem

The problem can be categorised as constrained multidisciplinary multi-objective multivariable optimisation with a mix of discrete and continuous variables. The problem is multidisciplinary since the simulation model is connected to a real control system.

3.3.1 Objectives

Obvious performance related objectives are accuracy, manufacturing speed and repeatability. Also of interest is the stroke of the *x*-axis, implying a trade-off between how large work pieces that can be machined in one set-up and the ability to cut several work pieces at the same time. A light weight design is also desirable, not the least from a general societal resource efficiency point of view. Three objectives are pursued in this introductory study: the weight of the main moving parts (which should also benefit energy and cost efficiency), the manufacturing accuracy and the manufacturing speed (i.e. the time it takes to cut the work piece). Since the feed rate is not a design variable in this study, the goal as regards the manufacturing speed is only to not have it significantly reduced.

The weight of the system is calculated by the finite element software. The manufacturing speed is easily obtained since the simulations are performed in real-time. The manufacturing accuracy is assessed according to the International Standard 230-4 [13]. A circular test is performed and the radial deviation is calculated. A fictitious test case is shown in figure 5.

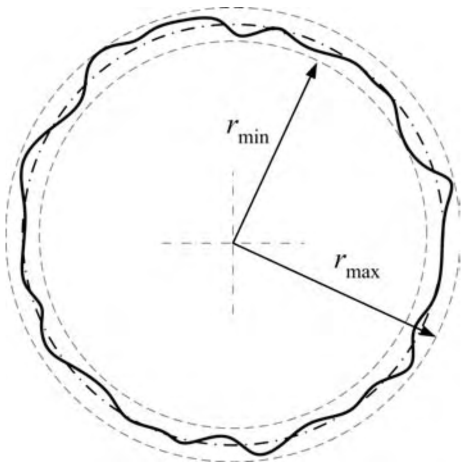

Figure 5. Test case.

The dash-dot line is the nominal path of radius r_{nom} that the machine tool is programmed to follow. The solid line represents the actual path produced by the machine tool (simulated motion of the tool centre point) when trying to follow the nominal path. The dashed lines are minimum and maximum concentric circles of radius r_{min} and r_{max}, respectively, enveloping the actual path. r_{max} and r_{min} are compared to r_{nom} and these deviations are used as a measure of the manufacturing accuracy (F_{tot}) according to equation 2.

$$F_{tot} = \left| r_{\max} - r_{nom} \right| + \left| r_{nom} - r_{\min} \right| \qquad (2)$$

In this study the radius of the nominal path is 5 mm and the feed rate is 5000 mm per minute. This feed rate is higher than feed rates normally used in the industry today. The purpose of this is to provoke larger differences between machine configurations.

3.3.2 Variables

The virtual machine, including the machine model as well as the control, contains thousands of parameters. It is of course not practically possible to vary all of them in an optimisation study. Some key parameters in the machine and the control are chosen as variables. This selection was guided by experience among the industrial partners as well as from previous simulations.

The following parameters in the machine model are considered as variables in the current study: the cross section of the cutting head holder beam, the cross section (box type) and length of the boom and finally the width of the X-unit. These are parameters that are easily changed in practice. Some of them vary within the industrial partners' current product range.

The controller used is a closed-loop servo system. It includes a position loop as well as a velocity loop. The position loop gain is chosen as a variable in the study. The loop gain determines how hard the servo tries to reduce possible errors. As the loop gain increases, the response is improved. A too large loop gain, however, might make the servo system unstable. The time constant used for acceleration/deceleration of the cutting feed is also used as a variable. A low time constant gives a higher manufacturing speed while also inducing more vibrations in the system.

The chosen variables with allowed values for the discrete ones and bounds for the continuous ones are given in table 1. Variable values for the original design is given in bold.

Table 1. Design variables.

Variable description	Type	Values (original values in bold)
Cross section of cutting head holder beam	Discrete	40x80 light version, **40x80**, 80x80 (mm)
Width boom cross section	Discrete	0.125, **0.150**, 0.175 (m)
Height boom cross section	Discrete	0.225, **0.250**, 0.275 (m)
Thickness of material in boom cross section	Discrete	0.005, **0.010**, 0.015 (m)
X-unit width	Continuous	Bounds: 0.55 - 1.0 (m) (**0.8**)
Length of boom	Continuous	Bounds: 3.875 - 4.325 (m) (**4.125**)
Time constant used for acceleration/deceleration of cutting feed (#1622)	Discrete	50, 75, 100, 125, **150**, 175, 200, 225, 250 (ms)
Loop gain for position control (#1825)	Discrete	1000, 1750, 2500, 3250, 4000, 4750, 5500, **6250**, 7000 (0.01 s^{-1})

The number of possible combinations of the variables presented in table 1 depends on how the continuous variables are encoded (see chapter 3.4). With the "resolution" used, over 6 700 000 combinations are possible. Thus, the problem is well suited for numerical optimisation.

3.3.3 Constraints

There are restrictions on how values for the design variables may be assigned. In the current study the following constraints are applied: A minimum axis stroke is given as well as domain constraints (lower and upper bounds for chosen design variables). The calculation of the axis stroke is approximated as boom length minus X-unit length. A minimum value of 3.325 meters is given.

The domain constraints are enforced automatically by the optimisation algorithm.

3.4 Optimisation Algorithm

A genetic algorithm (GA) is chosen since such have the ability to solve problems including both discrete and continuous variables. An in-house developed GA code is implemented in MATLAB. Real coded chromosomes are used for the discrete variables and binary coding is used for the continuous ones. Reproductive operators are single-point crossover, mutation and elitism. Duplicate chromosomes are not allowed in the population. Parents are chosen by proportionate selection, i.e. based on their fitness relative to all other individuals in the population. GA's in general are described in detail in for example [14].

While the purpose of the current work is to show the potential of the virtual machine concept and not necessarily to develop a perfect machine tool, a simple strategy to handle the multiobjective aspect of the problem is adopted. The different objectives are aggregated to one single figure of merit by a weighted sum approach. Weights are assigned to each objective by the decision maker. The sum of all objectives adjusted by their respective weight factor is used as the figure of merit according to equation 3 [6].

$$f_{\mathrm{w}}(\mathbf{x}) = \sum_{i=1}^{m} \left(\frac{f_{\mathrm{i}}(\mathbf{x})}{f_{\mathrm{i}0}} \right)^{\gamma_{\mathrm{i}}} \tag{3}$$

where f_{w} is the aggregated figure of merit, m is the number of objectives, γ_{i} is the weight factor, f_{i} the i:th objective function, $f_{\mathrm{i}0}$ the i:th objective function value for the best known solution so far and $\mathbf{x}$ the variable set.

The constrained problem is converted into an unconstrained problem through penalization of infeasible solutions. If a constraint is violated, a penalization term is added to the objective function. Penalizing a solution, still keeping it in the population, ads diversity compared to just removing the chromosome in question. This helps the GA avoid premature termination. A through discussion about constraint handling in GA can be found in [15].

3.5 Simulation Scheme

A worst case function call may take up to seven minutes to complete. This includes that a new FE model needs to be built and solved, variable values changed that forces a re-start of the control system and that many samples are needed to get a stable mean. An efficient simulation scheme is therefore necessary.

This is achieved by carefully planning the order in which the individuals in each generation are simulated. This might be seen as an optimisation problem in it self. Here, however, a simple rescheduling is applied where the individuals are sorted in groups related to the variable that is most time consuming to change. Within these groups the individuals are sorted once again in respect to the variable that is the next most time consuming to change. This procedure is continued until the generation is sorted for all variables.

When a variable combination is simulated the results are saved in a data base. If this variable combination appears again in a subsequent generation the results are loaded from the data base avoiding time consuming simulation of known data. The same is true for the FE model, once a model is built it is saved in a data base and re-used if needed.

3.6 Results

The optimisation algorithm converged to a design containing the following variable setting. Cross section 3 is selected for the cutting head holder beam. This cross section is stiffer than the original one. The X-unit becomes 0.61 meters long. Hence it is close to its lower bound (0.55 meters). A cross section of 175x250x5 (mm) is selected for the boom and it is given a length of 3.99 meters. This significantly lighter boom combined with the chosen X-unit results in an axis stroke of more than 3.325 meters, i.e. satisfying the minimum stroke constraint. The time constant (control variable # 1622) is set to 100 (reduced by 33 %) and the loop gain (control variable # 1825) is unchanged and remains at 6250.

The normalised aggregated objective function shows a decrease from 1 to 0.60 which is a considerable improvement. The improvements of the individual objective functions are shown in table 2.

Table 2. Optimisation results.

Objective	Relative improvement compared to original design [%]
Weight of the system	31
Manufacturing accuracy	64
Manufacturing speed	2.0

A typical simulation of the test case used comparing the improved machine design to the original one can be seen in figure 6. The dash-dot line is the nominal path (normalised) that the machine tool is programmed to follow. The solid and dashed lines represents the actual path produced using the improved and the original design, respectively.

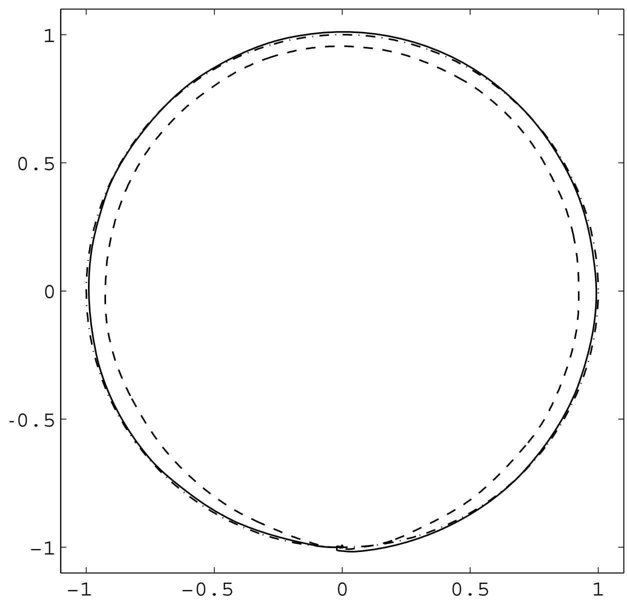

Figure 6. Typical simulation results.

One could argue that the large difference may to some extent be explained by a misfit of the original control parameters and the chosen test case. However, a design combining the original mechanics with the optimised control parameters is 52 per cent more accurate than the original design, i.e. it is still less accurate than optimised design (see table 2, 64 per cent improvement). It is also interesting to note that the opposite combination, i.e. a design combining the optimised mechanics with the original control parameters, is 4 per cent less accurate than the original design. These comparisons indicate that interaction effects between the mechanics and the control exist. This is also illustrated by carrying out a sequential optimisation. That is, first optimising the mechanics using the original control (not varying any control parameters) and then optimising the control using optimal mechanics obtained from the first step. This sequentially optimised design is 44 per cent more accurate than the original design, i.e. far less accurate than the design obtained from the simultaneous optimisation.

4 Discussion and Conclusion

A virtual machine is used in an introductory design optimisation study to improve an existing water jet cutting machine design. The weight of the main moving parts of the machine, the manufacturing accuracy and the manufacturing speed at a specified feed rate are used as objective functions. A genetic algorithm is used because of the discrete nature of some of the chosen design variables, and this method performs well in the presented test case.

In-house developed tools for data exchange between ABAQUS and MATLAB enable parameterisation of the simulation model, which yields a flexible simulation environment that works very well in the presented test case. Furthermore a strategy to handle non-deterministic simulation results based on statistical methods for Gaussian distributed data shows good performance in the presented test case.

Already in this limited introductory study a significant potential for design improvements is revealed. The weight of the main moving parts is reduced by more than 30 per cent, the manufacturing accuracy is improved by more than 60 per cent and the manufacturing speed is increased by 2 per cent (i.e. at least maintained as desired).

It is also shown that interaction effects exist between the mechanics and the control, i.e. this improvement would most likely not been possible with a conventional sequential design approach within the same time, cost and general resource frame. This indicates the potential of the virtual machine concept for contributing to improved efficiency of both complex products and the development process for such products. Companies incorporating such advanced simulation tools in their product development could thus improve its own competitiveness as well as contribute to improved resource efficiency of society at large.

The positive results already from this introductory study encourage further work with the virtual machine concept. The HIL simulator as well as the optimisation algorithm will be refined in preparation of more comprehensive optimisation studies, in parallel with physical testing and redesign of real machine tools.

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ABSTRACT

Effective and efficient product development is critical to business success on the increasingly competitive global market, and simulation has proven to support this in many sectors. The aim of this thesis is to study how properties of complex mechanical and mechatronic systems can be more efficiently and systematically predicted, described, assessed and improved in product development. The purpose is to elaborate an approach that can, rather than only verifying solutions that are already decided upon, support dialogues with customers, stimulate creation of new concepts and provide guidance towards more optimised designs, especially in early development stages. This is here termed simulation-driven design.

To be useful for this, product models and simulation and optimisation procedures must be efficient, that is, they must accurately answer posed questions and point towards better solutions while consuming an acceptable amount of time and other resources. In this thesis a coordinated approach to create such efficient decision support is elaborated. This is done by action research through two industrial case studies; an automobile exhaust system representing a complex mechanical system and a water jet cutting machine representing a mechatronic system.

The general knowledge gained from these case studies should be a good base for coming implementation of this approach as an inherent working routine in companies developing complex mechanical and mechatronic products.

A specific result is a validated virtual model of the exhaust system, which facilitates fast structural dynamics simulation of customer proposed design layouts. It is also shown that the non-linear flexible joint between the manifold and the rest of the exhaust system makes the system behaviour complex. This has resulted in an additional general research question, namely how systems that are linear, except for small but significant non-linear parts, can be simulated in an efficient way. Another specific result is a validated real-time virtual machine concept for simulation of the water jet cutting machine, which facilitates early-stage design optimisation. As the mechanics and the control system are considered simultaneously, interaction effects can be utilised. An introductory optimisation study shows a significant potential for improved manufacturing accuracy and a more light-weight design. This potential would not likely have been found through a conventional sequential design approach.

The results of this thesis indicate that there is a great potential for improved product development performance in small and medium-sized companies. By incorporating modern simulation support these companies can improve their competitiveness as well as contribute to improved resource efficiency of society at large. In doing so, it is important to find a good balance between model fidelity, validity and cost for achieving a relevant decision support. The coordinated approach to simulation-driven design elaborated in this thesis is a promising and systematic way of finding this balance.

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