Application of Nonlinear Receptance Coupling to Dynamic Stiffness Evaluation for Reconfigurable Machine Tools

Ahmet S. Yigit
Department of Mechanical and Industrial Engineering
Kuwait University
P.O. Box 5969 Safat 13060 Kuwait
and
A. Galip Ulsoy
Department of Mechanical Engineering
The University of Michigan
Ann Arbor, MI 48109-2125

Abstract

A systematic procedure is proposed to evaluate dynamic characteristics of design alternatives, for a Reconfigurable Machine Tool (RMT). The procedure is intended to be used in an automated design environment where various design alternatives are generated by kinematic synthesis based on a given task and specification. The evaluation procedure makes use of a substructuring method called nonlinear receptance coupling. The coupling method includes the effects of weakly nonlinear compliant joints through the use of describing functions for the nonlinearities involved. To demonstrate the utility of the proposed evaluation procedure design alternatives are generated based on a reduced order lumped parameter model of the RMT and examined with respect to the proposed criteria. It is shown that joint nonlinearities may affect dynamic stiffness considerably. It is also shown that the suggested criteria can distinguish various design alternatives with respect to their expected dynamic behavior.

1. Introduction

A new manufacturing paradigm called Reconfigurable Manufacturing Systems (RMS) is emerging to address the needs created by rapidly changing markets and rapid introduction of new products [1]. One of the primary goals in RMS is to reduce design lead-time, manufacturing set-up time and ramp-up time while providing a cost-effective solution. These new systems provide exactly the functionality that is needed exactly when it is needed [2].

Therefore, in RMS, reconfigurable machine tools (RMT) are designed to be easily reconfigured such that they process a family of parts and accommodate new and unanticipated changes in the product design and processing needs. In general, an RMT is made up of various replaceable modules. In the process of reconfiguration, new modules replace some old modules, and the degrees of freedom of the machine tool are also changed. Thus, RMTs undergo topological changes, i.e., size, type and number of modules and their interconnections [3].

A systematic design methodology which allows synthesis of the best possible RMT from a library of basic building blocks for processing a given family of parts is currently being developed [4]. In this methodology, various design alternatives (candidate designs) are generated based on kinematic considerations. To select the best possible design, the alternatives are evaluated with respect to several criteria such as the work envelope, degrees of freedom, the number of modules used, and the expected static and dynamic performance. The use of the first three criteria is quite straightforward and well-documented [5]. It is, however, necessary to develop practical criteria, which can be used to evaluate the dynamic behavior of the candidate machine tool designs. The accuracy of the work produced on a machine tool is determined by the deviations at the cutting point from the required relative motion between the tool and the workpiece. These deviations are caused – in addition to geometric and kinematic errors – by static and dynamic forces present in the machine tool [5]. Traditionally, the so-called dynamic stiffness (frequency dependent flexibility) is used to assess a machine's dynamic behavior. For an existing machine tool, experimental procedures called "acceptance tests" are developed to obtain the required metrics [6]. In the design stage, however, analytical or computational means have to be used. One obvious and accurate method would be to develop a detailed finite element model of the candidate design considered for the RMT and carry out a dynamic analysis. However, it is very cumbersome to develop a sufficiently accurate, and thus complicated, FEM model for each alternative design. Thus, to carry out a full dynamic analysis is not practical for evaluation of candidate designs, though it will eventually has to be done on the particular design chosen after the evaluation procedure. A viable alternative is to use the available module information to synthesize the dynamic stiffness of the whole RMT without requiring any new dynamic analysis. This can be achieved by employing one of the existing substructuring or modal synthesis method. However, most of the substructuring methods were developed by considering that the individual components are rigidly attached to each other to form the assembled system. In certain cases and especially in the case of RMT's the modules are connected by removable joints characterized by considerable amounts of compliance and damping. In this case, the coupling or assembly of the component dynamics should include the effects of these joints.

Identifying the stiffness and damping characteristics of various types of joints typically used in machine tools has been the subject of numerous investigations (see e.g., [7-9]). Based on experimental, analytical and empirical studies a number of simplified joint models have been proposed [8,9]. Yoshimura [9] used an experimental procedure to obtain equivalent linear stiffness and damping properties of bolted joints as a function of interface pressure and apparent contact area. Wang and Liou [10] developed an experimental synthesis procedure to calculate the dynamic stiffness of a structure including joint effects. Watanabe and Sato [11] proposed the so-called "Nonlinear Building Block (NLBBA)" approach to evaluate frequency response characteristics of a structural system including nonlinear springs. They used impedance coupling method by representing the joint nonlinearity with its describing function. Dowell [12] used component mode synthesis method with Lagrange multiplier to analyze a simply supported beam attached with a nonlinear spring-mass system. Inclusion of joint nonlinearities in the coupling procedure leads to nonlinear algebraic equations, which need to be solved iteratively [13]. Recently, Ferreria and Ewins [14] proposed a nonlinear receptance coupling approach which was originally developed to capture the first order nonlinear effects through the use of describing functions. The methodology has been extended to include multi harmonics [15].

The main objective of the current paper is to propose a general procedure to obtain and evaluate the dynamic stiffnesses of various alternative RMT designs from its modules including the effects of nonlinear joint characteristics. This procedure should help the (automated) design process in eliminating or ranking the alternatives in a cost efficient way. Due to its simplicity, an improved version of the impedance coupling that incorporates nonlinear joint dynamics is used. Section 2 includes a brief description of this method. A step by step evaluation criteria is proposed in Section 3, which includes the results and discussion for a prototype desktop RMT. Summary of the main results and anticipated future work are given in Section 4.

2. Receptance Coupling with Nonlinear Joints

It is assumed here that the machine tool is comprised of a number of modules connected together by means of joints at a finite number of known points. We also assume that the receptances (or any other related quantity such as mobilities or accelerances) between specified points of the individual modules along specified axes are available. This can be done experimentally, analytically with a lumped parameter model, or computationally by a separate finite element analysis of modules. For the sake of simplicity and without loss of generality it is assumed that in case of experimental data the masses of accelerometers are much smaller than that of modules so that no mass cancellation is necessary. Dynamic behavior of the individual modules is assumed to be linear. Therefore, nonlinearities are introduced in the system behavior through the joints only. The formulation used in this paper closely follows Reference [14]. For the sake of completeness a brief description will now be given.

The coupling process is related to modules and coordinates at two distinct stages: 1) pre-coupling, and 2) post-coupling. All variables in the pre-coupling stage are represented by lower case letters and by capital letters in post-coupling stage. The coordinates are divided into two groups: Those related to the connections are called “connection coordinates”, and those not related to connections are called “internal coordinates”. Each connection is represented by a pair of coordinates, (p_j,q_j), j=1,2,M each on a separate module as shown in Figure 1. Here the index j refer to the jth connection among a total of M. In case of a compliant joint, for each connection along or about an axis both of these coordinates represent a degree of freedom. In case of a rigid connection, however, after assembly, one of these coordinates will be redundant since both will have the same displacement. The coordinate sets c, and C include all non-redundant coordinates associated with the connections in the pre- and post-coupling stages. The response equation at the pre-coupling stage is given in matrix form as:

$$\begin{Bmatrix} \boldsymbol{x}_i \\ \boldsymbol{x}_p \\ \boldsymbol{x}_q \end{Bmatrix} = \begin{bmatrix} \mathbf{H}_{ii} & \mathbf{H}_{ip} & \mathbf{H}_{iq} \\ \mathbf{H}_{pi} & \mathbf{H}_{pp} & \mathbf{H}_{pq} \\ \mathbf{H}_{qi} & \mathbf{H}_{qp} & \mathbf{H}_{qq} \end{bmatrix} \begin{Bmatrix} \boldsymbol{f}_i \\ \boldsymbol{f}_p \\ \boldsymbol{f}_q \end{Bmatrix} \qquad (1)$$

where $\mathbf{H}_{mn}$ is the receptance matrix between the coordinate sets *m* and *n*. In the post-coupling stage, the response can be written as:

$$\begin{Bmatrix} \boldsymbol{X}_I \\ \boldsymbol{X}_P \\ \boldsymbol{X}_Q \end{Bmatrix} = \begin{bmatrix} \mathbf{H}_{II} & \mathbf{H}_{IP} & \mathbf{H}_{IQ} \\ \mathbf{H}_{PI} & \mathbf{H}_{PP} & \mathbf{H}_{PQ} \\ \mathbf{H}_{QI} & \mathbf{H}_{QP} & \mathbf{H}_{QQ} \end{bmatrix} \begin{Bmatrix} \boldsymbol{F}_I \\ \boldsymbol{F}_P \\ \boldsymbol{F}_Q \end{Bmatrix} \qquad (2)$$

The equilibrium conditions can be written as:

$$\begin{aligned} &\boldsymbol{F}_P = \boldsymbol{F}_Q = \boldsymbol{f}_p + \boldsymbol{f}_q \\ &\boldsymbol{f}_i = \boldsymbol{F}_I \end{aligned} \qquad (3)$$

The compatibility conditions can be written as:

$$\begin{aligned} &\boldsymbol{x}_i = \boldsymbol{X}_I \\ &\boldsymbol{x}_p = \boldsymbol{X}_P \\ &\boldsymbol{x}_q = \boldsymbol{X}_Q \end{aligned} \qquad (4)$$

and

$$\boldsymbol{x}_p - \boldsymbol{x}_q = -\boldsymbol{f}_p / G_{nj} \qquad (5)$$

$$\boldsymbol{x}_q - \boldsymbol{x}_p = -\boldsymbol{f}_q / G_{nj} \qquad (6)$$

where G_{nj} is the describing function for the joint impedance between the associated connection coordinate pair. The describing function is a quasi-linear representation for the nonlinear element subjected to a sinusoidal input. The use of describing functions in a modal analysis is analogous to using the harmonic balance method to solve nonlinear dynamic equations [16].

Substituting Equations (3)-(5) into (1) and (3), (4) and (6) into (1) and relating the forces of pre- and post-coupling stages and some elementary matrix manipulations together with comparisons with Equation (2) yield the following assembly equations

$$\begin{Bmatrix} \boldsymbol{X}_I \\ \boldsymbol{X}_P \\ \boldsymbol{X}_Q \end{Bmatrix} = \left\{ \begin{bmatrix} \mathbf{H}_{ii} & \mathbf{H}_{ip} & \mathbf{H}_{iq} \\ \mathbf{H}_{pi} & \mathbf{H}_{pp} & \mathbf{H}_{pq} \\ \mathbf{H}_{qi} & \mathbf{H}_{qp} & \mathbf{H}_{qq} \end{bmatrix} - \begin{Bmatrix} (\mathbf{H}_{ip} - \mathbf{H}_{iq}) \\ (\mathbf{H}_{pp} - \mathbf{H}_{pq}) \\ (\mathbf{H}_{qp} - \mathbf{H}_{qq}) \end{Bmatrix} [\mathbf{B}]^{-1} \begin{Bmatrix} (\mathbf{H}_{ip} - \mathbf{H}_{iq}) \\ (\mathbf{H}_{pp} - \mathbf{H}_{pq}) \\ (\mathbf{H}_{qp} - \mathbf{H}_{qq}) \end{Bmatrix}^T \right\} \begin{Bmatrix} \boldsymbol{F}_I \\ \boldsymbol{F}_P \\ \boldsymbol{F}_Q \end{Bmatrix} \qquad (7)$$

where

$$\mathbf{B} = \mathbf{H}_{pp} + \mathbf{H}_{qq} - \mathbf{H}_{pq} - \mathbf{H}_{qp} + \boldsymbol{\Delta} \qquad (8)$$

with

$$\boldsymbol{\Delta} = \mathrm{Diag}(1/G_{nj}) \qquad (9)$$

Equation (7) can be arranged in a more concise form, by using the non redundant coordinate sets as:

$$\begin{Bmatrix} \boldsymbol{X}_I \\ \boldsymbol{X}_C \end{Bmatrix} = \left\{ \begin{bmatrix} \mathbf{H}_{ii} & \mathbf{H}_{ic} \\ \mathbf{H}_{ci} & \mathbf{H}_{cc} \end{bmatrix} - \begin{Bmatrix} (\mathbf{H}_{ip} - \mathbf{H}_{iq}) \\ (\mathbf{H}_{cp} - \mathbf{H}_{cq}) \end{Bmatrix} [\mathbf{B}]^{-1} \begin{Bmatrix} (\mathbf{H}_{ip} - \mathbf{H}_{iq}) \\ (\mathbf{H}_{cp} - \mathbf{H}_{cq}) \end{Bmatrix}^T \right\} \begin{Bmatrix} \boldsymbol{F}_I \\ \boldsymbol{F}_C \end{Bmatrix} \qquad (10)$$

The submatrices in Equation (10) are generated by partitioning the coordinates involved in the modules. For example, for the system shown in Figure 1, the joints are represented by the connection pairs $(2,4)$, $(3,5)$. Points 1 and 6 are internal points. Note that in general every point has three degrees of freedom (u,v, and w along x,y and z directions). Therefore, the coordinate sets to be used in the synthesis procedure are identified to be:

$p = \{u_2, v_2, w_2, u_3, v_3, w_3\}$, and. $q = \{u_4, v_4, w_4, u_5, v_5, w_5\}$

The non-redundant connection coordinates are $c = p \cup q$. The internal coordinates, which are not related to any of the joints, are identified as $i = \{u_1, v_1, w_1, u_6, v_6, w_6\}$.

The describing function, G_{nj} is a function of relative displacement between the connection coordinates. therefore, Equation (7) is a system of nonlinear algebraic equations to be solved iteratively at each spectral line. In case of experimentally obtained receptances, the stiffness synthesis requires only one matrix inversion. The size of the matrix to be inverted is equal to the number of connection coordinates. These two features improve the computational efficiency of the synthesis procedure.

3. Dynamic Stiffness Evaluation Criteria

Design alternatives generated by the kinematic synthesis procedure should be evaluated to choose the best possible design. It is assumed here that there exist evaluation criteria for geometric and kinematic errors [4] and they are already applied to eliminate the unacceptable alternatives. The focus of the current work is on the dynamic effects. The objective is a smooth stable operation with minimum relative displacement between the tool and the workpiece. We also would like to have a chatter-free cutting condition. With respect to the first objective, a machine with a lower compliance (high dynamic stiffness) at a particular operating condition is to be preferred. Since the machine may be subject to a broad band excitation within the frequency range of interest (FROI), a measure of dynamic stiffness should be maximized in addition to a high static stiffness, which in general result in a higher fundamental frequency. Therefore, with respect to the stable operation, the evaluation criteria should distinguish the alternatives for their static stiffness, the fundamental frequency, and a measure of dynamic stiffness within the FROI.

The basic cause of chatter is the dynamic interaction of the cutting process and the machine tool/workpiece structure. It is a self-excited vibration caused by an instability mechanism due to the nonlinearities and delays present in the system. The chatter vibrations occur at a particular frequency (generally close to one of the natural frequencies) [17]. The chatter performance of a machine tool is measured with respect to the maximum dept of cut possible without leading to chatter at a particular cutting speed. The usual practice is to examine the frequency response since it is closely related to the stability of the system. Since, for most cutting operations, the cutting force acts in a direction different from the normal to the cut surface; the relevant frequency response is the cross response between these two directions and between the tool and the workpiece. This receptance is called the operative receptance [17]. The onset of chatter can be determined by examining the stability charts obtained from the resulting analysis. Determination of the stability charts for an existing machine tool for a particular process has been the subject of numerous studies (see e.g., [18]). For the purpose of comparing two alternative designs with respect to their chatter performance a simpler procedure can be employed based on an approximate, but conservative value of the horizontal asymptote for which the process is predicted to be stable at all spindle speeds. It has been shown that this threshold is related to the maximum negative in-phase displacement in the polar operative receptance plot [6]. A coefficient, called the coefficient of merit (COM) is defined which can be used to compare the chatter performance of similar machines. Let the cross-receptance value X between the cutting force and the relative displacement in the direction normal to cut surface is given as

$$X(\omega) = [\alpha(\omega) + j\beta(\omega)]F \qquad (11)$$

where $j = \sqrt{-1}$, and F is the magnitude of the harmonic force applied at the cutting point. Then, the coefficient of merit is defined as

$$COM = \frac{1}{2\alpha_{\min}} \qquad (12)$$

Clearly a machine with a higher value of COM is to be preferred with respect to chatter performance.

In light of the above discussion the following evaluation criteria can be used to distinguish and enumerate alternative designs. A higher value of the following criteria indicates a more favorable design:

Criterion 1. Static stiffness.

Criterion 2. Fundamental frequency (lowest natural frequency of the machine tool).

Criterion 3. Minimum dynamic stiffness within FROI obtained from the operative receptance curve.

Criterion 4. Mean value of the dynamic stiffness within FROI.

Criterion 5. Coefficient of Merit.

3. 1 A Procedure for Dynamic Stiffness Evaluation

Based on the discussion above the following sequential evaluation procedure is proposed:

1. For a given task, identify candidate designs through kinematic synthesis (e.g., see [4]).
2. Identify process parameters, especially cutting forces in terms of the directions, magnitudes (both static and dynamic parts) and the frequency content. Determine the frequency range of interest.
3. Identify the important receptance directions (e.g., the direction of cutting force and the normal to cutting surface)
4. From the given configuration, and the cutting force data, determine joint pressure distributions, and consequently joint parameters.
5. Synthesize required receptances by using FRF coupling with joints.
6. Apply evaluation criteria on three levels.
   a) Level 1: Compare the lowest natural frequencies and the static stiffness, and eliminate unacceptable designs, use Criteria 1 and 2.

b) Level 2: Apply Criteria 3, and 4, and eliminate unacceptable designs.

c) Level 3: Obtain refined polar plots for the operative receptances around resonance frequencies and compare the chatter performance of remaining designs, use Criterion 5.

3.2 Illustrative Example

In order to demonstrate how the procedure is employed, the proposed method is applied to the prototype RMT built by ERC for Reconfigurable Manufacturing Systems. Figure 2 shows the essential features of this machine that is built to drill cam tower holes on different cylinder heads. The configuration shown on Figure 2 is for V8 cylinder heads where a two-axis spindle assembly is sufficient to locate and drill the holes. To process V6 cylinder heads, however, a three-axis reconfiguration (not shown) is necessary, since the holes do not lie in a single horizontal line.

Since the process considered is drilling, the cutting direction is the same as the normal to the cut surface. Therefore, the operative receptance required to analyze the dynamic stiffness of this machine is obtained as the complex relative displacement between the tool and the workpiece along the cutting direction for a harmonic force applied between the same two positions.

In order to carry out the modal synthesis, the required receptance matrices for each component, or module should be available. As mentioned earlier, experimental, analytical or computational methods can be utilized to obtain these receptances. If the purpose of the evaluation is to obtain the values of natural frequencies and the overall behavior (e.g., Criterion 2), experimental data can be used without significant errors. However, the resulting noisy behavior makes it difficult to apply some of the evaluation criteria proposed; especially Criterion 5, since the phase information is corrupted by the artificial resonances created by the noisy data [19]. More importantly, it is quite difficult to require extensive experiments to pre-compile FRF matrices for all possible modules prior to design stage. For complicated assemblies it is necessary to include the rotational degrees of freedoms associated with the connection coordinates, which are very difficult to measure experimentally. Clearly, once the design is completed, and an RMT is built, well known acceptance and performance tests will be used to evaluate the final design [5]. However, there is little merit to require this extensive experimental work for all possible modules for all possible sizes. Typically, however, a lumped parameter model, or an FEM model (or at least a solid model) is provided in the module library. Therefore, during the enumeration of the alternative designs in the automated synthesis, it is preferred to use module data from an analytical model, or from an FEM analysis, which can be provided with the module information in the module library, or can be carried out prior to synthesis stage In this paper, to illustrate the effectiveness of the proposed procedure, the FRF matrices are obtained from a simple lumped parameter model of the system components which represent their dynamic behavior along the cutting direction (see Figure 3).

For the purpose of simple modeling, it is assumed that the spindle unit is represented by a two degree of freedom system that is to be attached to the rigid machine bed by a nonlinear joint. The structure, which is used to carry the workpiece, is assumed to be represented by a single degree of freedom system (one-mode approximation) and it is attached rigidly to the machine bed. Finally, the workpiece is assumed to be rigid and attached to the structure by a nonlinear joint. While there are many possible joint models, for the purpose of illustration, a hardening spring with a linear viscous damper in parallel is assumed to represent the joint characteristics. This type of nonlinearity has been well studied in the literature [20] and the nature of response is well understood. This is important to assess the effectiveness of nonlinear receptance coupling. It should be noted that it is straightforward to include a more realistic joint model once its describing function is determined either analytically, or experimentally for a particular joint.

The joint force developed due to a deflection x is given by

$$f = kx + k' x^3 + b\dot{x} \tag{13}$$

where k, k', are the stiffness coefficients and b is the damping coefficient for the joint considered. The describing function is defined as the complex ratio of the fundamental component amplitude of the displacement to the amplitude of the sinusoidal input force and given as [16]

$$G_n(\omega, A) = k + \frac{3}{4} k' A^2 + ib\omega \tag{14}$$

Where A is the amplitude of the relative displacement at the joint, and ω is the frequency.

The values of the model parameters are given in Table 1. These parameters are selected in an *ad hoc* manner to have a good qualitative match with the experimentally obtained frequency responses of the actual system within the FROI [19]. The model appears to capture the essential dynamic behavior within the FROI. Note that the purpose here is not to obtain an accurate model for the RMT considered. Using a set of realistic parameters, however, makes the conclusions drawn from the model, especially those related to the effect of nonlinearities, relevant to the typical applications.

From Figures 2 and 3, it is possible to identify two compliant joints: one between the spindle unit and the machine bed, and the other one between the structure and the workpiece with the following coordinate pairs: $(0,1), \quad (4,3)$. Therefore, the coordinate sets used in the synthesis procedure are identified to be $p = \{0,4\}, \quad q = \{1,3\}$. The non-redundant connection coordinates are $c = \{1,3,4\}$. Note that the coordinate 0 is on the machine bed which is assumed to be rigid (i.e., zero displacement). Therefore, the receptance associated with this coordinate is assumed to be zero. The internal coordinate, which is not related to any of the joints, is identified as $i = \{2\}$.

The operative receptance is obtained by applying equal and opposite forces at the coordinates 2 and 3, and evaluating the relative displacement $X_{rel} = X_2 - X_3$.

Figure 4 shows the resulting relative compliance for various force magnitudes. Joint nonlinearities do not significantly affect the natural frequencies. However, the effect on the amplitudes at certain frequencies may be significant. Amplitude dependent nature of the nonlinear behavior can be observed in the figure. Also, the well-known jump phenomenon, commonly observed with cubic nonlinearities, can also be predicted at certain frequencies. As expected, for low values of the input force, the behavior approaches to linear case. In fact, the case of 10 N is almost identical with the linear case. Thus, the receptance coupling procedure has been shown to capture the essential characteristics of nonlinear response. It should be noted, however, that the modal analysis for the nonlinear system is effective as long as the deviation from linear behavior is small. In other words, the use of the dynamic stiffness concept is limited to RMTs with weakly nonlinear joints.

In order to demonstrate how the proposed evaluation criteria are used three other alternative designs are generated with the same model (see Table 1). Since the application of the criteria related to the static stiffness and the fundamental frequency is straightforward, it is more informative to focus on the last three criteria. Therefore, the alternatives are selected in such a way that it is not possible to distinguish them by using the first two criteria. The results of application of the criteria are summarized in Table 2. Figure 5 compares the dynamic compliances of Design No. 1 with Design No. 2. These two designs have the same static stiffness and the fundamental frequency. Therefore, with respect to Criteria 1 and 2 both designs are equivalent. However, when Criteria 3 and 4 are applied, Design 1 is superior to Design No. 2 since both the value of the minimum dynamic stiffness and the value of the mean dynamic stiffness within the FROI are larger. This is to be expected since Design 2 has more flexibility in the spindle unit (lower k_2 value). Though this stiffness loss is compensated by a high stiffness of the joint, it resulted in a lower dynamic stiffness around the second resonance. Another set of design alternatives is generated which involves both stiffness and mass alterations. Design 4 has a stiffer spindle unit with a higher mass around the tool. Design 3 is essentially similar to Design 1 with a lower force amplitude. Figure 6 compares these designs. Both alternatives have almost the same static stiffness and the fundamental frequency. The minimum dynamic stiffness value is higher in Design 3 due to its lower mass at the tool. However, according to Criterion 4, Design 4 is superior to Design 3 since it has a higher mean dynamic stiffness. Since there is a conflict between the two criteria, it may be worthwhile to compare the chatter performance of both designs using Criterion 5. Figure 7 shows the polar operative receptance plots for both design. It is seen that Design 4 is superior to design 3 since it has a higher COM. It should be noted that, the alternative designs considered here are merely selected to demonstrate the effectiveness of the evaluation procedure in distinguishing similar designs with respect to their dynamic characteristics. In a real implementation, these design alternatives are generated by the automated synthesis procedure [4], and evaluated within the same environment. Clearly, the final design selected can further be improved by applying well-known design principles [6,9]. It should be re-iterated here that the focus of this work is not on improving a single design. It is rather on selecting the best alternative configuration in an automated modular design environment.

4. Summary and Conclusions

A systematic procedure was proposed to evaluate design alternatives for an RMT with respect to dynamic stiffness. The procedure is intended to be used in an automated design environment where various design alternatives are generated by kinematic synthesis based on a given task and specification. The procedure makes use of a substructuring method called nonlinear receptance coupling, which incorporates the effects of weakly nonlinear compliant joints. To demonstrate the utility of the proposed evaluation procedure design alternatives are generated based on a lumped parameter model of the RMT and examined with respect to the proposed criteria. It was shown that joint nonlinearities may affect dynamic stiffness considerably if the harmonic component of the excitation is large. It was also shown that by sequentially applying the suggested criteria, various design alternatives could be distinguished with respect to their dynamic characteristics. Though the proposed procedure is straightforward, its success depends on 1) the accuracy and completeness of the receptance matrices of each module; 2) the accuracy of the dynamic characterization of the joints. The accuracy and completeness of the receptance matrices can be achieved by requiring a reasonably detailed FEM model of each module. For an accurate joint characterization, however, a simple empirical or experimental procedure should be available to determine the model and its parameters for a given joint and a set of operating conditions. Future work will focus on incorporation of the proposed procedure into the automated design environment.

Acknowledgements

This work has been carried out during the first author's sabbatical leave at The University of Michigan. The authors are pleased to acknowledge financial support from Kuwait University, and the National Science Foundation Engineering Research Centers program under grant EEC9529125.

References

1. Koren, Y. Heisel, U., Jovane, F. Moriwaki, Pritchow, G. Van Brussel, H., and Ulsoy, A.G. Reconfigurable Manufacturing Systems. *CIRP Annals,* 1999, Vol. 48(2), 527-540.

2. Mehrabi, M.G., Ulsoy, A.G., and Koren Y. Reconfigurable Machining Systems: Key to Future Manufacturing. *Proc. Of the Japan-USA Symposium on Flexible Automation,* Otsu, Japan, July 1998, pp. 677-682.

3. Moon Y.M., and Kota, S. Generalized Kinematic Modeling Method for Reconfigurable Machine Tools. *Proc. 25th Biannual Mechanism Design Conference*, Atlanta, GA, 1998.

4. Moon,Y.-M., and Kota, S., A Methodology for Automated Design of Reconfigurable Machines. *Proceedings of the 32nd CIRP International Seminar on Manufacturing Systems*, Leuven, Belgium, 1999, pp. 297-303.

5. Weck, M. *Handbook of Machine Tools*. Vols. 1 and 4. Wiley-Heyden Ltd., Chichester, 1984.

6. Sadek, M.M., and Tobias, S.A. Comparative dynamic acceptance tests for machine tools applied to horizontal milling machines. *Proc. Instn. Mech. Engrs*, 1970-71, Vol. 185, pp. 319-337.

7. Gaul, L. Wave transmission and energy dissipation at structural and machine joints. *Journal of Vibration, Acoustics, Stress, and Reliability in Design*, 1983, Vol. 105, pp. 489-496.

8. Ito, Y., and Tsutsumi, M. Determination of mathematical models in structural analysis of machine tools –2nd Report; determination of mathematical models for normal static stiffness of joints. *Bulletin of the JSME*, 1981, Vol. 24, pp. 2234-2239.

9. Yoshimura, M. Computer-aided design improvement of machine tool structure incorporating joint dynamics data. *Annals of CIRP*, 1979, Vol. 28(1), pp. 241-246.

10. Wang, J.H., and Liou, C.M. Experimental substructure synthesis with special consideration of joints effects. *The international Journal of Analytical and Experimental Modal Analysis*, 1989, Vol. 5(1), pp.13-24.

11. Watanabe, K., and Sato, H. Development of nonlinear building block approach. *Journal of Vibration, Acoustics, Stress, and Reliability in Design*, 1988, Vol. 110, pp. 36-41.

12. Dowell, E.H. Component mode analysis of nonlinear and nonconservative systems. *Journal of Applied Mechanics*, 1980, Vol. 47, pp. 172-176.

13. Budak, E., and Ozguven, H.N. Iterative receptance method for determining harmonic response of structures with symmetrical non-linearities. *Mechanical Systems and Signal Processing*, 1993, Vol. 7, pp. 75-87.

14. Ferreira, J.V., and Ewins, D.J. Nonlinear receptance coupling approach based on describing functions. *Proc. 14th International Modal Analysis Conference*, Feb. 12-15, 1996, Dearborn, MI, pp.1034-1040.

15. Ferreira, J.V., and Ewins, D.J. Multi-harmonic nonlinear receptance coupling approach based on describing functions. *Proc.15th International Modal Analysis Conference*, Feb 3-6, 1997, Orlando, FL, pp. 27-33.

16. Atherton, D.P. *Nonlinear Control Engineering.* Van Nostrand Reinhold Company, London, 1975.

17. Boothroyd, G., and Knight, W.A. *Fundamentals of Machining and Machine Tools*. 2nd Ed., Marcel Dekker, Inc., New York, 1989.

18. Chen, S.G,, Ulsoy, A.G., and Koren, Y. Computational stability analysis of chatter in turning. *Journal of Manufacturing Science and Engineering*, 1997, Vol. 119, pp. 457-460.

19. Yigit, A.S., and Ulsoy, A.G., Dynamic Stiffness Evaluation for Reconfigurable Machine Tools Including Weakly Nonlinear Joint Characteristics. Submitted for publicaion in *Journal of Engineering Manufacture*, 2000.

20. Nayfeh, A.H., and Mook, D.T. *Nonlinear Ocsillations.* John Wiley&Sons, New York, 1979.

Table 1: Parameters of alternative designs

	Design 1	Design 2	Design 3	Design 4
m_1	30 kg	30 kg	30 kg	30 kg
m_2	30 kg	30 kg	30 kg	60 kg
m_3	80 kg	80 kg	80 kg	80 kg
m_4	50 kg	50 kg	50 kg	50 kg
k_1	5000 kN/m	25000 kN/m	5000 kN/m	5000 kN/m
k_1'	5×10^{12} N/m^3	5×10^{12} N/m^3	5×10^{12} N/m^3	5×10^{12} N/m^3
k_2	3000 kN/m	2026 kN/m	3000 kN/m	6000 kN/m
k_3	5000 kN/m	5000 kN/m	5000 kN/m	5000 kN/m
k_3'	5×10^{12} N/m^3	5×10^{12} N/m^3	5×10^{12} N/m^3	5×10^{12} N/m^3
k_4	7000 kN/m	7000 kN/m	7000 kN/m	7000 kN/m
b_1	300 Ns/m	300 Ns/m	400 Ns/m	4000 Ns/m
b_2	30 Ns/m	30 Ns/m	400 Ns/m	400 Ns/m
b_3	300 Ns/m	300 Ns/m	300 Ns/m	300 Ns/m
b_4	30 Ns/m	30 Ns/m	300 Ns/m	300 Ns/m
F	100 N	100 N	10 N	10 N

Table 2: Results of dynamic stiffness evaluation procedure

	Design Alternative 1	Design Alternative 2
Minimum Dynamic Stiffness	$1.82 \times 10^4 N/m$	$6.85 \times 10^3 N/m$
Mean Dynamic Stiffness	$6.14 \times 10^5 N/m$	$4.24 \times 10^5 N/m$

	Design Alternative 3	Design Alternative 4
Minimum Dynamic Stiffness	$3.95 \times 10^4 N/m$	$3.77 \times 10^4 N/m$
Mean Dynamic Stiffness	$6.69 \times 10^5 N/m$	$9.86 \times 10^5 N/m$
Coefficient of Merit	$3.96 \times 10^4 N/m$	$4.06 \times 10^4 N/m$

List of Figure Captions

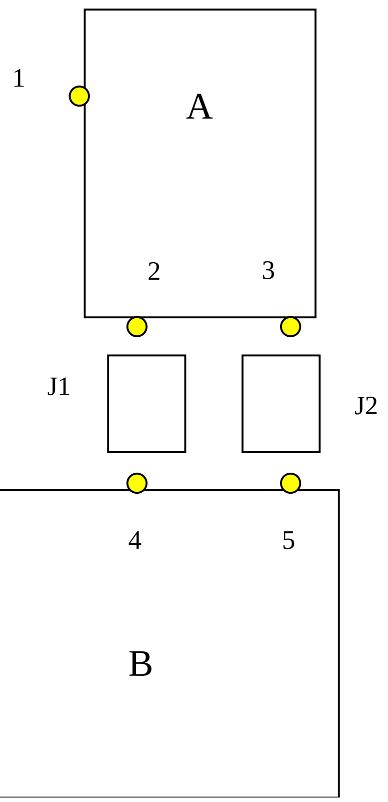

Figure 1: A sketch showing two modules to be connected by two joints

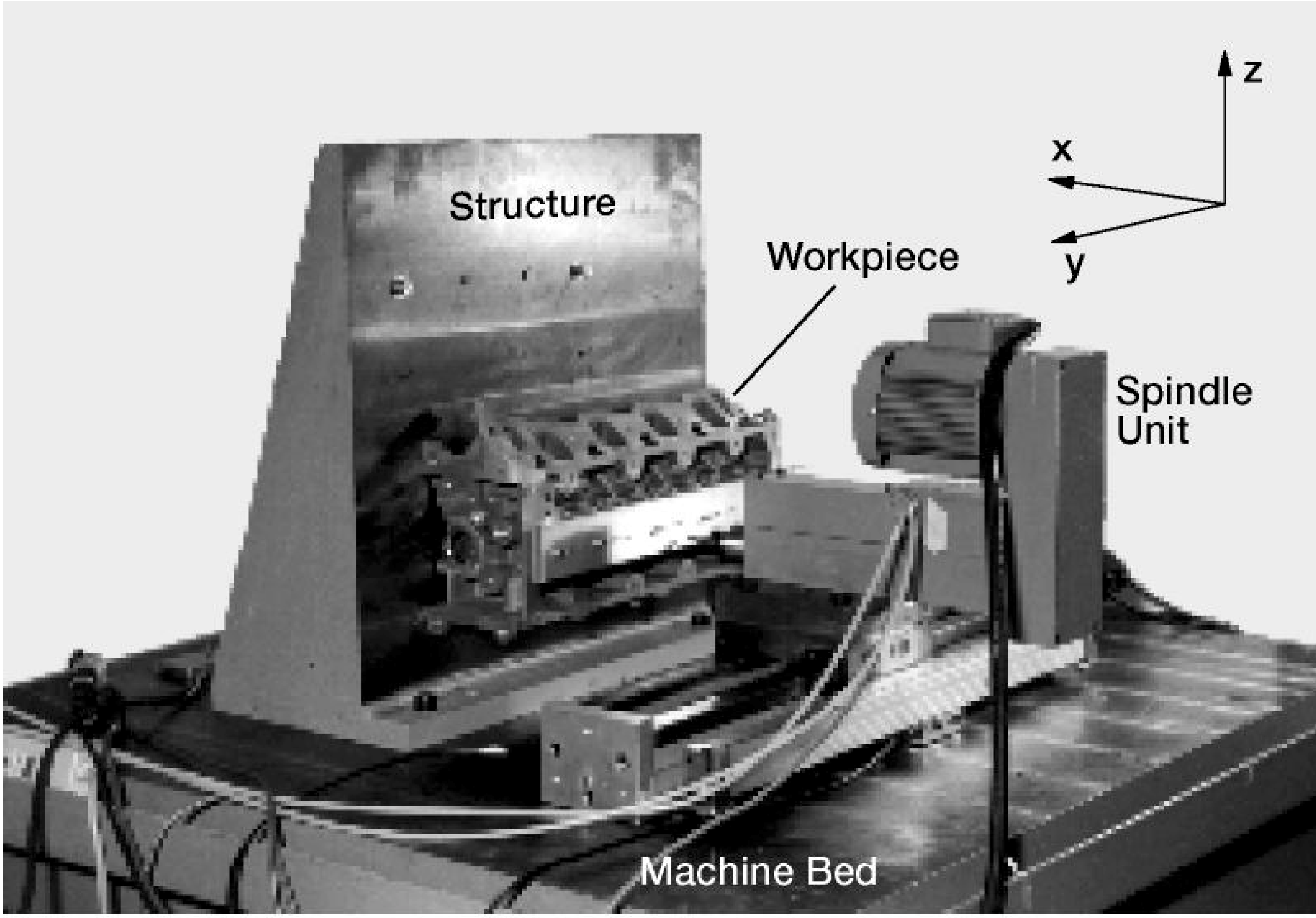

Figure 2: Prototype RMT

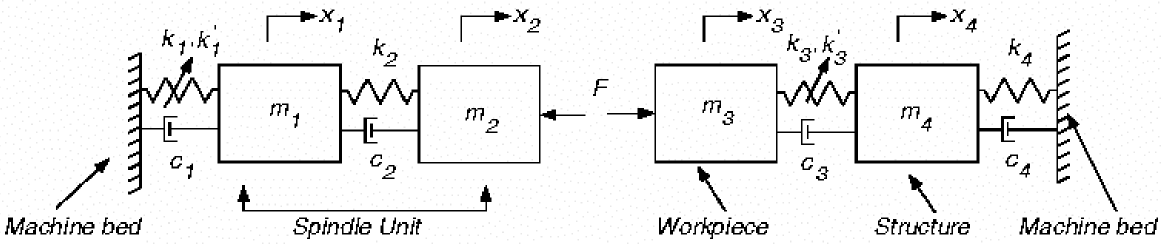

Figure 3: Sketch of the lumped parameter model

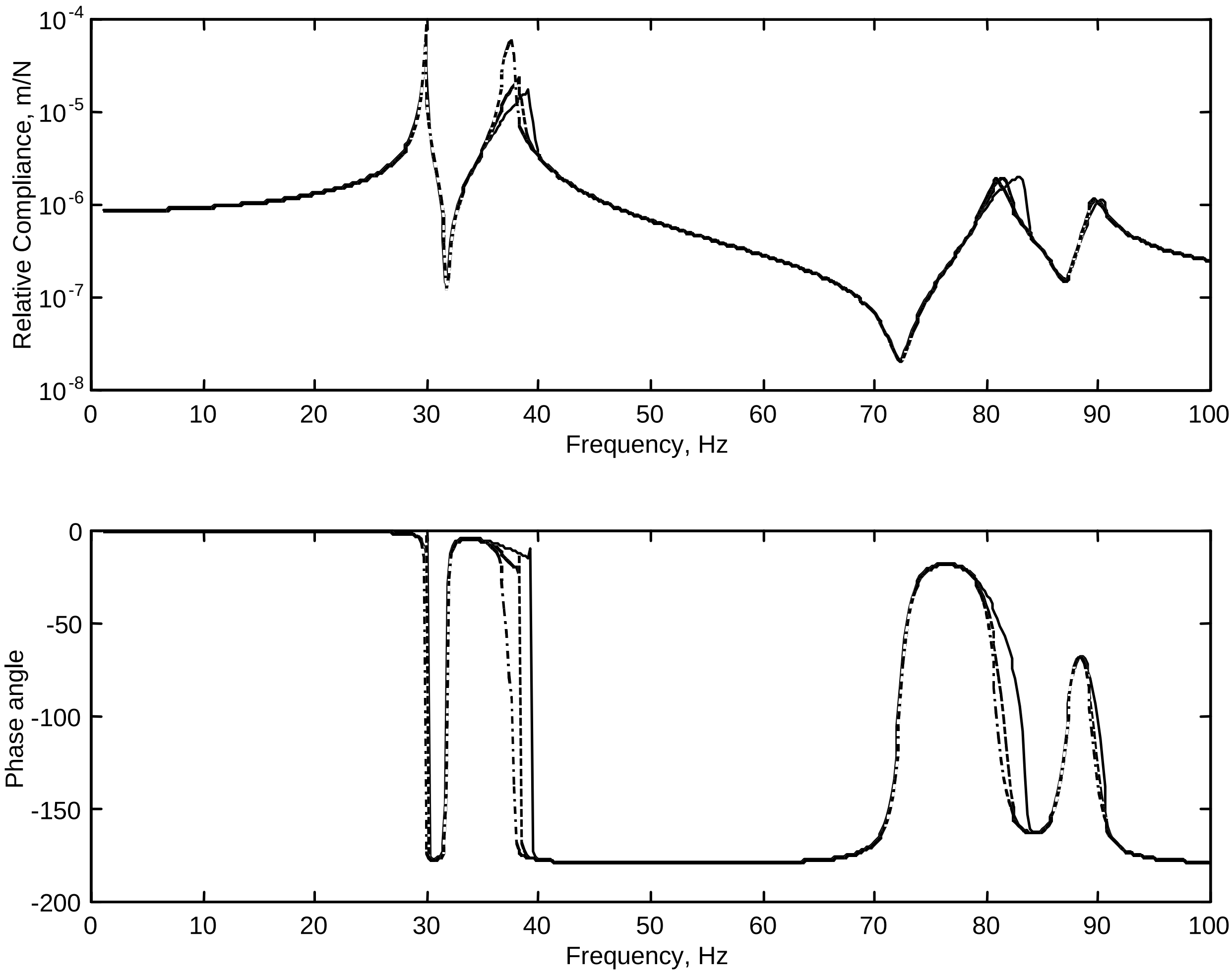

Figure 4: Relative compliance at the tool/workpiece interface for various force levels; (solid: 100 N, dashed: 50 N, dashed-dot:10 N)

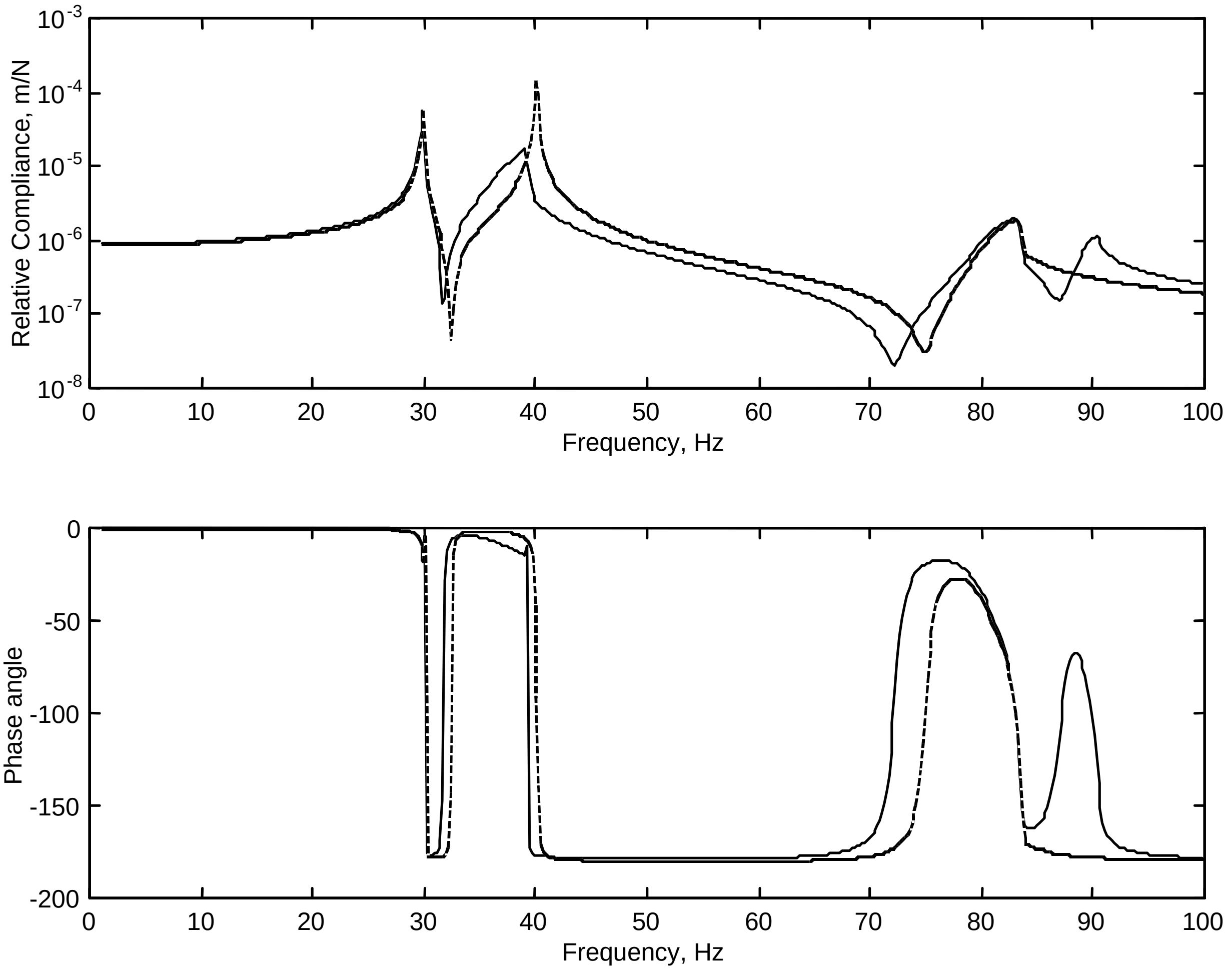

Figure 5: Comparison of two alternative designs with respect to the relative compliance; (solid:Design 1, dashed:Design 2)

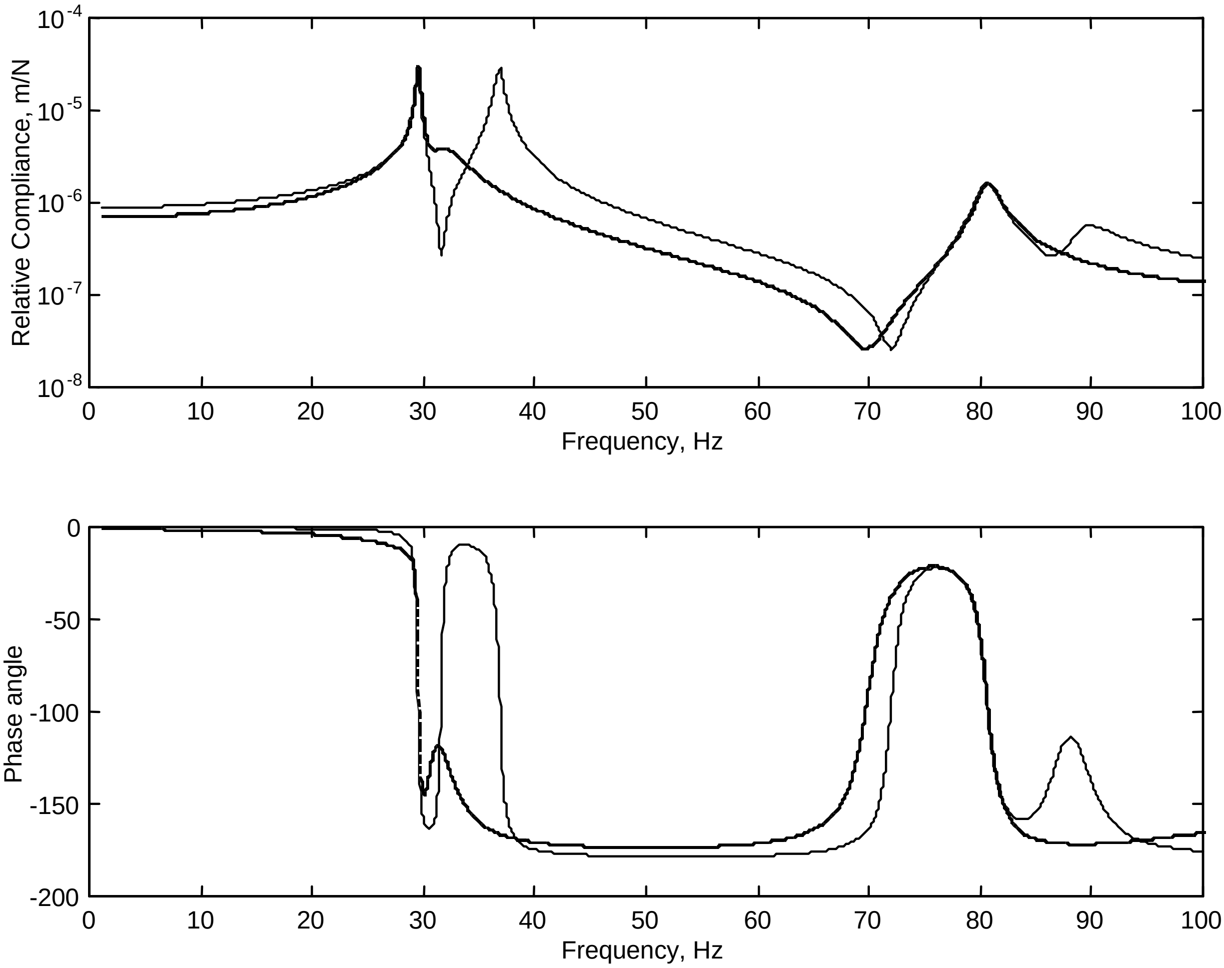

Figure 6: Comparison of two alternative designs with respect to the relative compliance; (solid:Design 3, dashed:Design 4)

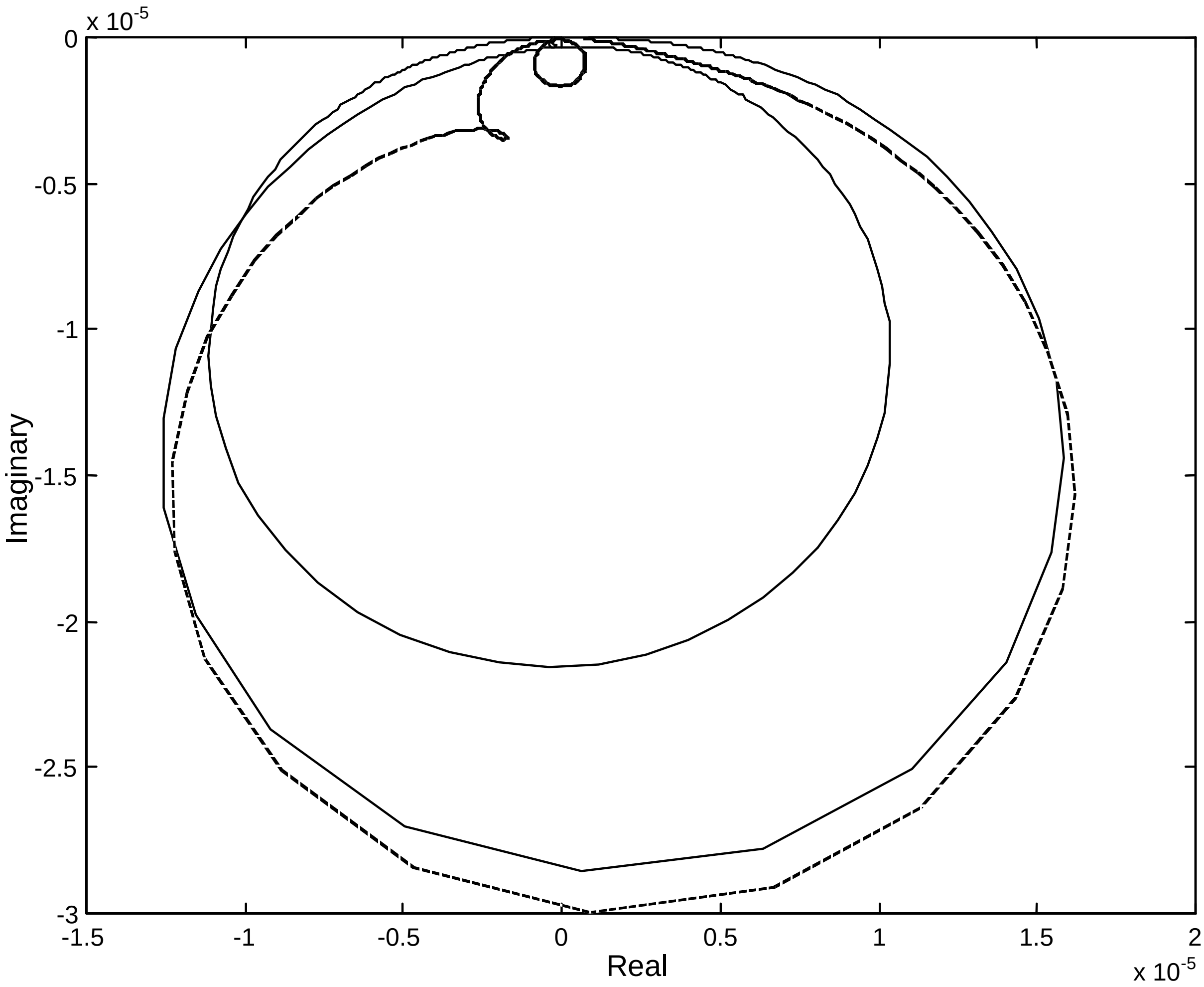

Figure 7: Comparison of two alternative designs with respect to the coefficient of merit; (solid:Design 3, dashed:Design 4).