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The Effect of Linear Guide Representation for Topology Optimization of a Five-axis Milling Machine

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Abstract

Topology optimization is a countermeasure to obtain lightweight and stiff structures for machine tools. Topology optimizations are applied at component level due to computational limitations, therefore linear guides' rolling elements are underestimated in most of the cases. Stiffness of the entire assembly depends on the least stiff components which are identified as linear guides in the current literature. In this study, effects of linear guide's representation in virtual environment are investigated at assembly level by focusing on topology optimization. Two different contact models are employed for rolling elements in the linear guides. Reliability of the contact models are verified with experiments. After the verification, heavy duty cutting conditions are considered for the system and topology optimization is performed for two different contact models to reduce the mass of the structure. The difference caused by the representation of rolling elements is demonstrated for the same topology algorithm and the optimization results are compared for the models. Lastly, the effect of using more stiff linear guides in the five-axis milling machine is investigated by increasing the stiffness of the contact elements.

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1. Introduction

Competition in the market is steadily forcing machine tool developers to increase productivity while reducing machine costs by creating a descending trend in the machine tools energy consumption. Today, lightweight design of machine tool structures is mainstreamed for energy efficiency, but it is also important to note that the ability to reach the upper limits of servo drivers is another major contributor while developing efficient machine tools. However, to be able to design such a machine tool is not an easy task. Lightweight machine tool structures provide extended working bandwidths for servo drivers compared to the massive ones due to mass reduction. Also, these lightweight structures push the low modes to higher frequencies allowing higher gains to be used in the control loops. The first natural frequencies of lightweight machine tool structures and the drivers are in a similar bandwidth. Therefore, a greater risk may occur during design stage for overlapped modes at low frequencies [1]. In order to overcome the mentioned drawbacks, the everlasting objective should be increasing stiffness globally while reducing or keeping the same component weights [2]. However, entire machine structure stiffness depends on the weakest components of assembly which are usually linear guides and bearings [3, 4].

Topology optimization is one of the most powerful tools for designing lightweight and stiff structures at the early design stage; however, it has its own drawbacks. A typical topology optimization application is carried out in virtual environment by employing FE models of the machine. These models have proved their suitability and significance for subsystem level design analyses such as modeling of ball-screw feed-drive systems [5], spindles [6] and full machine assembly design analyses. However, FE analyses of full machine models are computationally costly. For instance, an FE model of typical machine assembly has one million

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degrees of freedom (DOF) or more [7]. In order to reduce DOF and model complexity, most of the FE models ignore contact elements and connection parameters. In reducing computational cost, two approaches are common. The first one is to define critical structural components and optimize topology for these components separately. The second is to use the full assembly model for topology optimization with co-FEM or Model Order Reduction techniques [5, 7].

The first approach -defining critical parts and optimizing them- has generally been applied when different considerations are taken into account for topology optimization. In a machine tool structural optimization problem, the objective might not only be the static stiffness; the end user may also care about chatter and surface quality of the workpiece. Hence, the problem statement must also include dynamic rigidity concerns, and therefore employing a soft-kill BESO method [8] proposed for the component or sub-assembly level. For most practical design problems, 'self –weight' and 'design depended loading' issues drive the objective as minimizing mass while satisfying stress constraints. Due to stress singularity in the computational process reaching a global optimum for a stress-based topology optimization is not guaranteed, therefore it is applied locally [9]. Additionally, it is well known that continuous topology optimization problem forms like SIMP and RAMP methods tend to offer composite material structure in terms of element density [2, 10]. At this point manufacturability is the greatest obstacle for the stiffness objective, although most dominant topology optimization software has casting, drawing and extrusion constraints with the help of MMA methods [11]. Manufacturing constraints pose innumerable computational effort therefore, these constraints strictly limits the assembly optimization initiatives [12].

The second approach- entire assembly optimization - gives superior results while simulating real behavior of the machine tool structure, by representing the contact interfaces. However, simulation of full FE model, is a really time consuming process and is inefficient for a FE solver [7]. Therefore, CMS and Model Order Reduction techniques are applied together [13]. Also co-FEM methods like Multi Body Simulation techniques are coupled with topology optimization to decrease the computational cost [3, 14].

The rolling elements of linear guides have rarely been simulated in a FE model of milling machine assembly until now, due to the computational limitations. Besides, the design tendency for stiff structures have directed designers to create massive structures without considering the least stiff components of the machine tool assembly. This study is aimed to reveal significance of the contact elements. Especially, the linear guides are considered in the given entire machine tool assembly. The optimized topologies are compared with respect to their static and dynamic behaviors. For this purpose, two different linear guide representations of a five-axis milling machine are plugged in the entire assembly of an FE model, and then the reliability of these sub FE models are verified with experiments. Rolling elements of the linear guides are represented as surface contacts in the first equivalent contact model, while the same components are represented as springs in the second equivalent contact model. Furthermore, linear guide's stiffness is increased and its effects on mass reduction are demonstrated within this study.

The paper is organized as follows; two different representation of linear guides based on FE models of the entire assembly are presented, the reliability of these FE models are verified with static experiments and then, the loading conditions are explained for topology optimization in Section 2. In Section 3, topology objectives and constraints are stated then, the results of topology optimization are compared for two different linear guide's representation. Furthermore, linear guides' stiffness is increased and resultant topologies are demonstrated in Section 3. Conclusions are shared in Section 4.

2. FE simulation of machine tool structures

A competitive five –axis machine in the market must have superior design features. To design a lightweight, fast and precise five-axis machine tool, FE simulations and topology optimizations are vital. These methods provide predictions about precision and accuracy limits of the machine tool at early design stages. In order to obtain the best reliable results from topology optimization, the FE models of machine tools and the simulation conditions should be close to real ones. However, computational limitations drive machine tool designers to made simplifications on the machine tools and analyze them in component level. Therefore, all contact surfaces are neglected or underestimated. In this part, different representation of linear guides based on FE models of the entire assembly are presented. Two approaches are employed for the rolling contact elements at the assembly level. Reliability of these FE models are verified with static tests. For topology optimization, loading conditions are explained. The results of the loading conditions are used as constraints in Section 3.

2.1. FE models

Five-axis milling machine FE models are generated by using its respective CAD models. Each structural component of the model is meshed with tetra elements, with total of ~ $4x10^6$ elements and ~$1x10^6$ nodes, after a convergence test. Three material properties, for steel and cast iron assigned to different components of the model are given in Table 1.

Table 1. Material properties assigned to components

Material	Elasticity Modulus	Density	Poisson Ratio
Steel	210 GPa	7850 kg/m^3	0.3
Cast Iron	140 GPa	7200 kg/ m^3	0.3

The crucial part of the modeling is the representation of the rolling elements within ball grooves of the linear guide components, which is significant to obtain a realistic machine tool structure.

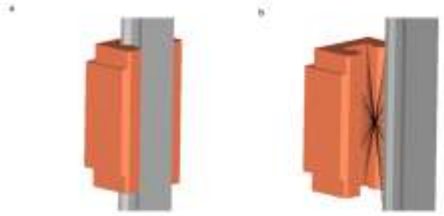

Fig. 1. (a) Model 1 surface contacts for rollers elements, (b) Model 2 springs for roller elements

These roller elements are modelled as surface contact in Model 1. For Model 2 non-linear spring elements are employed. The stiffness value of bearing element is taken as 82 N/µm from the manufacturer catalogues.

2.2. Reliability of FE models

In order to understand directional stiffness behavior of the full assembly, the spindle tip is loaded in various directions in real and virtual environment. Verification experiments are conducted [16] to measure the static deformation of the machine tool spindle in five different positions.

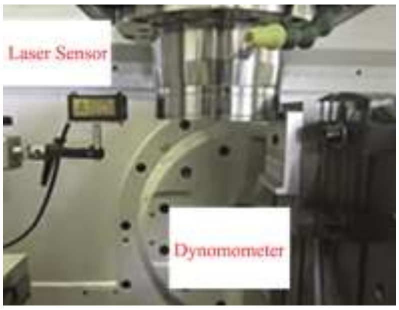

Fig. 2. Experimental Set-up for Static Experiments

The machine tool spindle tip is loaded in machine's X direction during the first three experiments while it is loaded in the machine's Z direction during the fourth and the fifth experiments. Both experimental and FE results are tabulated in Table 2 and Table 3 for equivalent contact model 1 and 2, respectively.

Table 2. Equivalent contact model 1 surface contact for rolling elements

Experiment Number	Applied Force	Experiment Result	Analysis Result	Difference
1	520N	20 µm	8.4 µm	58%
2	455N	20 µm	8.3 µm	59%
3	375N	24 µm	10 µm	58%
4	485N	13 µm	3.7 µm	71%
5	265N	6.5 µm	1.5 µm	77%

Table 3. Equivalent contact model 2 spring for rolling elements

Experiment Number	Applied Force	Experiment Result	Analysis Result	Difference
1	520N	20 µm	21.2 µm	6%
2	455N	20 µm	19.6 µm	2%
3	375N	24 µm	22.9 µm	4%
4	485N	13 µm	11.8 µm	9%
5	265N	6.5 µm	5.9 µm	9%

The comparison of the experimental and FE results for both models indicate significant discrepancy which is caused by representation of the rolling elements. Although the first equivalent model has its own stiffness value, the rolling elements are underestimated. Instead of these underestimated rolling elements, built-in contact elements are employed which is computationally less expensive. However, these substitutions are not sufficient. The performance evaluation can be observed in Table 2. On the contrary, the second equivalent model is employed springs to directly represent rolling elements even though this method is computationally costly. However, the return is significant as indicated in Table 3.

2.3. Loading Conditions for Topology Optimization

For topology optimization, heavy cutting conditions are applied for a tapered helical ball end mill cutter [15], which are commonly used in machining of complex surfaces such as airfoils, in the FE simulations. Titanium Ti6Al4V alloy is chosen as the workpiece material. Axial depth of the cut is 20mm and feed rate is 0.050mm/tooth. The cutting forces are obtained via CutPro software for the conditions indicated in [15]. Static and modal analysis are performed by using the resultant cutting loads. FE simulations are repeated for two contact models. Based on the static analyses, total spindle deflection at the spindle tip is determined 22 µm for the first contact model while it is 55 µm for the second model. Additionally, based on modal analysis, the natural frequencies of both models obtained by the finite element solution are illustrated in Fig 3.

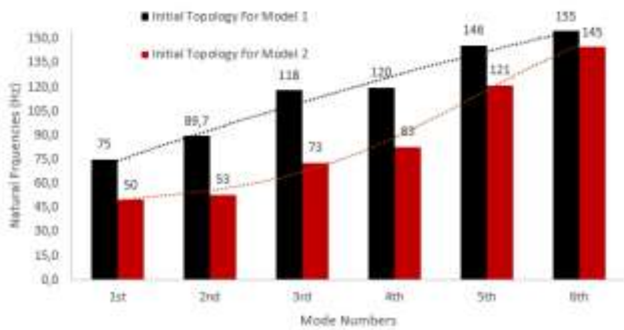

Fig. 3. Natural Frequencies of Given Initial Design for the First Six Modes

As can be seen from Fig.3, the differences are considerable for the first four modes for Model 1 and Model 2. The gap closes dramatically when the fifth and sixth modes are considered. The gap reduces nearly zero for the higher modes, but as mentioned before [1] at low frequencies servo drive and machine structure modes may overlap and cause instability The usual way to overcome this is to reduce gains for the servo drivers which limits the running range of the servos reducing acceleration/deceleration rate. Therefore, in order to reach upper limits of servo drivers and increase speed performance of a machine tool, simulation of the structural models with realistic predictions especially for the low modes, are vital during design stage. Furthermore, pushing the low modes to higher frequencies as much as possible through mass reduction would not only increase the servo performance and acceleration and jerk limits of the machine axes but also reduce energy consumption.

3. Topology Optimization of Machine Tool Structures

3.1 Optimization Problem Statement

The most common topology optimization formulation is developed to obtain stiffer structure by minimizing the compliance subject to a given amount of material, [2]. Basically, minimizing compliance equals to minimizing the energy of deformation at the equilibrium state of the structure. This problem in a continuous form can be stated as the following;

$$
\begin{aligned}
min_{\rho} &: \ \varphi(\rho) = \boldsymbol{F^T U} \\
s.t. &: \ \sum_{e=1}^{N} v_e \rho_e = \boldsymbol{v}^T \boldsymbol{\rho} \ \leq \ V^*, e = 1, \dots, N \\
&: \ g_i(\rho) \leq \ g_i^*, \ \ i = 1, \dots, M \\
&: \ 0 \leq \ \rho_{min} \leq \rho \leq 1 \\
&: \ \boldsymbol{K(\rho)U = F}
\end{aligned}
$$

within a given domain (Ω) by discretizing N finite elements. Here, the density depends on compliance as $\varphi(\rho)$ objective function with a volume constraint V^*, where, $\boldsymbol{F}, \boldsymbol{K}$ and $\boldsymbol{U}$ stand for force vector, global stiffness matrix, and nodal displacement vector, respectively. The displacements of the components are limited with a displacement constraint, which is represented by g_i^* in the problem statement. The displacements of the components are limited; (1) on the spindle tip, (2) on the maximum deflected areas of the moving components, which are spindle head, ram and sliding carriage. The optimization results show that the plotted topologies were exactly same for the (1) and the (2) displacement limitations. Therefore, the displacements are limited on the spindle tip during the whole optimization.

For the volume constraint, an iterative volume fraction process is applied to explore mass reduction capacities of the given five-axis milling machine. In the optimization, volume fraction rate is set to 20%, 25% and 30%, respectively. It was seen that higher than 30% volume fraction rate caused violation of displacement constraints.

3.2 Topology Optimization Results

The Optimal Topology for Model 1

The moving components of the initial design are optimized to obtain minimum compliance for the given constraints in the problem statement. The re-designable components are chosen as spindle head, ram and sliding carriage which are the most active parts in the given assembly. As a result of the optimizations, element densities are shown in Fig.4.

Fig. 4. Topology Optimization Results for the Model 1 with volume fraction constraint (a) 20%, (b) 25% and (c) 30%.

Blue regions indicate optimum mass reduction areas while red regions illustrate compulsory areas for the stiffened structure. The elements with low density are removed and the resulting structure with 30% volume fraction is shown in Fig 5.

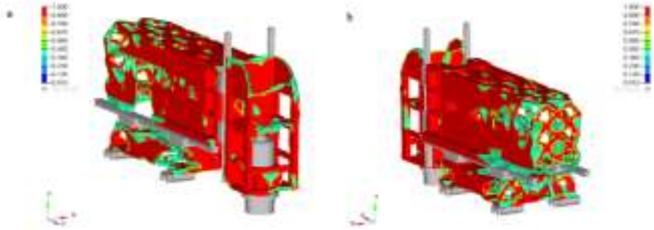

Fig. 5. (a) Front View of Top. Opt. for Model 1, (b) Back View of Top. Opt. for Model 1.

The Optimal Topology for Model 2

Same as Model 1, optimization results for Model 2 are given in Fig.6. Even though, there are ~ 60% difference in static response behavior and around ~40% difference in dynamic response behavior between Model 1 and 2, their optimal topologies are similar. Nevertheless, the optimized topologies of connection areas with linear guides are very different. The reason for that is the higher rigidity of Model 1 due to neglected contact stiffness. The resulting structure with 30% volume fraction is shown in Fig 7.

Fig. 6. Topology Optimization Results for the Model 2 with volume fraction constraint (a) 20%, (b) 25% and (c) 30%.

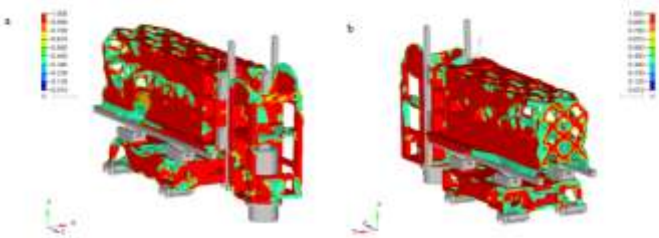

Fig. 7. (a) Front View of Top. Opt. for Model 2, (b) Back View of Top. Opt. for Model 2.

3.3 Comparison of the Results for Model 1 and Model 2

Although, there is a remarkable difference between the static and dynamic response behaviors of the models, the resultant optimized topologies are similar. However, difference occurs in the neighborhood of the linear guides. According to topology optimization results, the volume fraction intensity in the neighborhood of linear guides is noticeable for Model 2, while it is the reverse for Model 1. The differences are illustrated clearly in Fig.8;

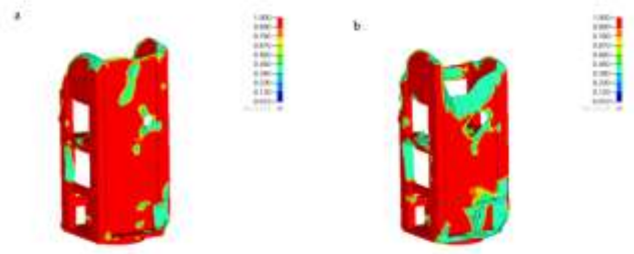

Fig. 8. (a) Spindle Head Top. Opt. for Model 1, (b) Spindle Head Top. Opt. for Model 2.

On the contrary, local displacements are transmitted with two linear guides at sliding carriage and ram. Hence, there is considerably lesser volume fraction intensity at these local displacement areas in Model 2. The differences between the models can be seen more clearly in Fig. 9 and 10.

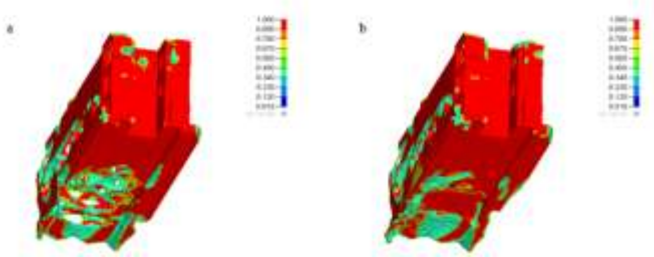

Fig. 9. (a) Ram Top. Opt. Model 1, (b) Ram Top. Opt. for Model 2

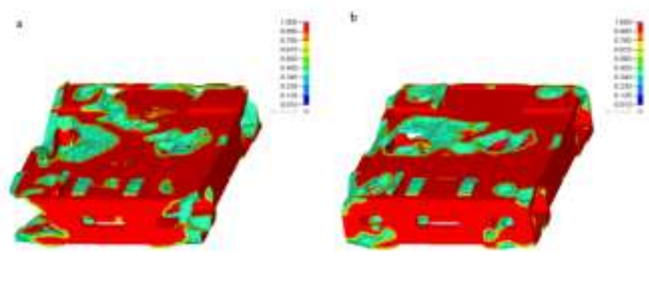

Fig. 10. (a) Sliding Carriage Top. Opt. for Model 1, (b) Sliding Carriage Top. Opt. for Model 2

The difference in topology causes different modal behaviors in Model 1 and 2. The change of modal behavior of the models is shown in Fig. 11 for 30% volume fraction.

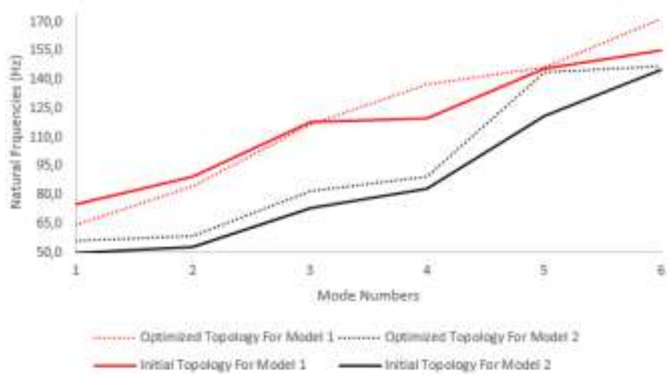

Fig. 11. Comparison of Natural Frequencies of Given Initial Design and Optimized Design

Fig.11 reveals that all the modes are shifted 10% for Model 2, but this trend is fluctuated for Model 1, and it is hard to predict modes behavior previously before the mass reduction. The difference in the predicted modes is around 40% between these two models. It is noteworthy that for an ordinary servo driver the first mode is around 45-60 Hz. For instance, Kroll and et al. [1], showed that the first mode of a Siemens drive (1FT6086-8AF7x model) is 44.8Hz. After 30% mass reduction, and by increasing the gains, the natural frequency shifted to 58.5 Hz. Thus, bandwidth of the dynamic control was extended for the related axis. Therefore, they operate the drivers at higher angular frequencies easily. The importance of this example is, although different configurations of the assemblies, the first natural frequencies of the machine tool structure and the drivers are in a similar bandwidth. Hence, unrealistic representation of the linear guides, possibly will lead the overlapped modes at low frequencies and then, the gains must be limited at feed drives to defeat this situation. In other words, the highest angular frequencies will be limited and reaching upper limits for the drivers will not be possible.

3.4 Increased Stiffness Results for Model 2

Verified by the experiments, Model 2 provides precise results as the entire model behavior depends on the least stiff component. Based on this, the effect of linear guide stiffness is demonstrated within this study. The rolling elements' stiffness is increased 20%, under the same loading conditions. The iterative volume fraction process is repeated to explore mass reduction capacities of the given five-axis milling machine. The optimizations are performed with increasing volume fraction rate from 20% to 40%. For the greater volume fraction rate than 40%, the allowed displacement constraints are violated. This means that, nearly 20% stiffness increase in rolling element makes additional 10% volume reduction possible. The obtained structure for increased stiffness with a 40% volume fraction is plotted in Fig 12.

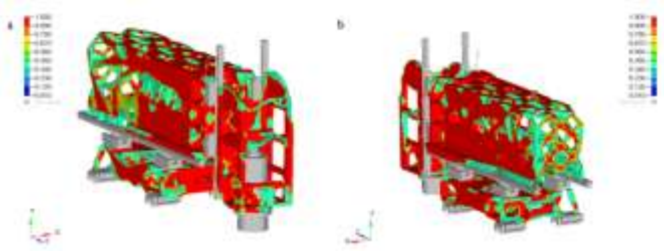

Fig. 12. Top. Opt. with %40 volume fraction for increased stiffness of the bearings; (a) Front View (b) Back View

Furthermore, the first six mode shapes are slightly changed for the optimized topologies. In Fig.13, the change of the mode shapes are displayed for the optimized topologies. The gap is nearly diminished between original stiffness and increased stiffness model results while the 10% additional volume fraction is posed to the model. This result is important, because it is possible to preserve the modal and static responses of the entire model while reducing mass by increasing stiffness of the linear guides.

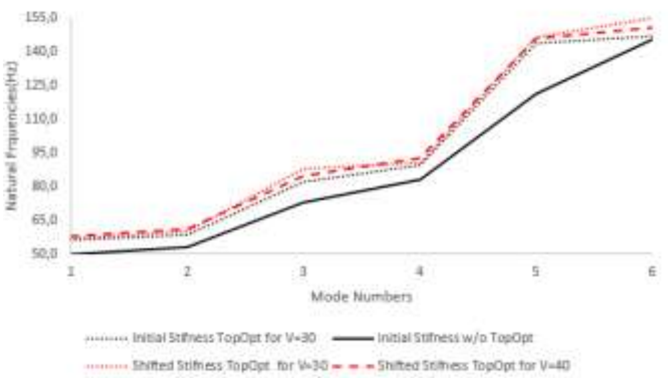

Fig. 13. Comparison of Natural Frequencies of After Top. Opt Original Stiffness and Increased Stiffness of Linear Guides

4. Conclusions

In this study, an extensive optimized topology comparison is presented for modeling the entire assembly for machining centers in order to obtain lightweight structures. Effect of bearing and interface parameters on the modes and on the displacements are analyzed and vital conclusions are derived.

- The rolling elements in the linear guides are significant during the process of FE modeling in virtual environment. Representing them directly by employing Model 2 in the virtual environment gives realistic predictions. Moreover, realistic prediction of structural modes prevents feed drives running bandwidth limitations at early design stage. In this way, reaching upper limits for the drivers will be possible for lightweight machining centers.
- Restricting maximum deflection as a topology optimization constraint gives the same result for spindle tip and for the other moving components.
- Choosing stiffer linear guides is a much more effective way than creating massive structures for increasing global stiffness of the model. By employing this approach, it is possible to preserve the modal and static response of the entire structural model while reducing mass and by increasing the stiffness of the linear guides.

This conclusions may not be generalized for the all machining center configurations, but they can give an insight into FE model creation and topology optimization process and the importance of linear guide's representation.

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