Development of a 3 dof Parallel Kinematic Machine for an application in shoes manufacturing

R.Maj, M.Leonesio, F.Modica, G.Bianchi, F.Paolucci
ITIA-CNR Institute for Industrial Technologies and Automation
Viale Lombardia 20/a, 20131, Milano, Italy
r.maj@itia.mi.cnr.it

Abstract: ITIA-CNR has been investigating in the last years the peculiarities and potentialities of machines based on a Parallel Kinematic structure. Different physical prototypes have been built, but also design methodologies and software tools have been developed to support industrial companies willing to develop PK machines. This paper presents the development of an industrial prototype for light deburring in shoes manufacturing with three degrees of freedom based on a PK structure plus a wrist with two rotational dof. The analyses done on a Finite Element model of the machine are compared with that ones executed on kinematic and multi-body models.

1 Introduction

Parallel Kinematic Machines (herein "PKM") are claimed to offer interesting performances in respect to their serial counterparts, namely: higher structural stiffness; non cumulative joint error; modular structure; less moving mass, with the motors close to the fixed base; a simpler solution of the inverse kinematic problem. Conversely, they suffer from singular configurations, low workspace to footprint ratio, a more complicated direct kinematic solution, and more difficult control techniques. In order to fully evaluate their industrial potential, the time required to analyze a proposed design has to be shorten by the development of appropriate design methodologies and software tools.

This paper presents ITIA-CNR activity on these topics, describing the development of a PKM for multitasking operations in a flexible and highly automated production system for shoe manufacturing. The paper is organized as the following: sect. 2 presents the industrial application; sect. 3 the selected PK architecture and its kinematic optimization; sect. 4 summarizes the analyses done with a multi-body approach; sect. 5 to 8 present the analyses based on the Finite Element model; sect. 9 draws a quick comparison of the various approaches.

2 Industrial Applications & Process Needs

The Parallel Kinematic Machine has to execute 3 different tasks during manufacturing of a shoe with injected sole :

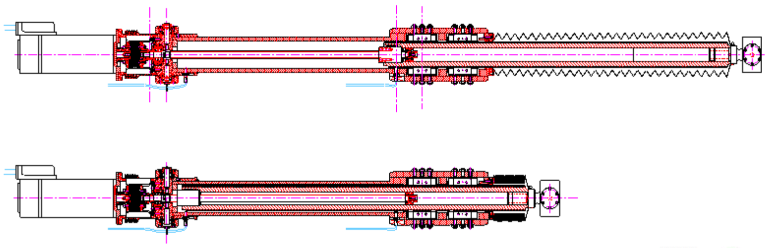
1. Deburring of the shoe profile: the tool, generally a mill for wood machining, removes a strip of leather 0.8 mm thick and 8 mm high;
2. Deposition, around the same profile, of a glue layer
3. Spraying a silicon mixture over the mould for PVA injection.

The strictest requirements are given by the first two operations, in terms of:

- working volume: 400 x 200 x 100 mm^3
- precision: +/- 0.1 mm
- max. speed: 30 m/min, acceleration: 5 m/s^2
- loads: 45 N deburring force

To execute the contour following in operation 2, three translational D.o.f. (degrees of freedom) for positioning and two rotations for tool orientation are required, with ±180° around the tool axis and at least ±90°around the other axis. These requirements are difficult to achieve using a fully parallel PKM. Furthermore, the issue of flexibility and future applications led to adopt an hybrid architecture (a 3 d.o.f. PKM plus a 2 d.o.f. serial wrist for orientation).

3 Parallel Kinematic Architecture and kinematic optimization



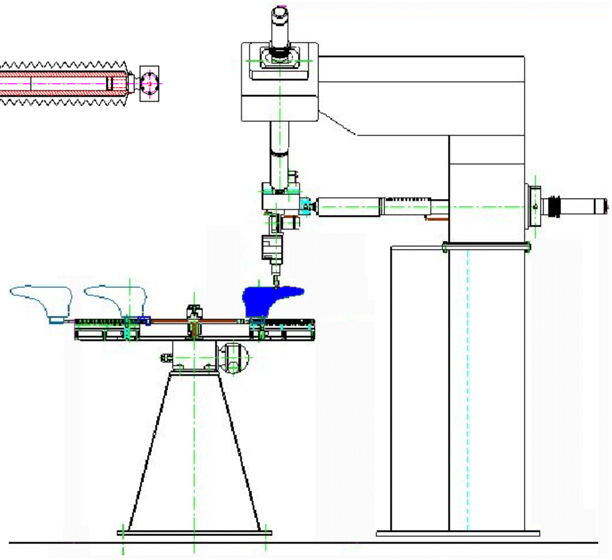


Figure 1– Section of one strut and the complete machining station for shoe deburring

The selected machine architecture is a variant of the PK manipulator originally proposed by Tsai ([1],[2]). The moving platform, that carries the End-Effector ("EE"), is connected to the fixed structure by three variable length struts (see). At both ends of each strut, a universal joint permits two relative rotations. A particular spatial disposition of the universal joints axes assures a pure translational motion of the headstock. The three struts differ only for the stroke, that is 350 mm for that ones in the vertical plane and 500 mm for the horizontal one. The prismatic coupling in the middle of each strut is realized by a set of four carriages with recirculating balls, that roll along two grooves on the rod. The same rod is also connected to the nut of the ball screw that realizes the telescopic motion of the strut.

The machine layout has been initially optimized, as described in [3], by a kinematic analysis based on the jacobian matrix that relates actuators velocities to End-Effector velocities, in order to reach the best numerical condition. As it is well known (e.g. [4]), such analysis is of fundamental importance for PKMs: when the jacobian is almost singular, internal loads tend to infinity and the machine gains one or more degrees of freedom, becoming uncontrollable. Thanks to such analysis it was decided to have, in the center of the workspace, the three struts orthogonal to each other: this layout assures a perfect isotropic behavior from the kinematic point of view, with unitary Jacobian, as in a serial machine with orthogonal axes. The developed analytical formulation was also necessary to customize the machine Numerical Control.

4 Analysis with the Virtual Prototyping Environment for PKM

A second step of development has been done using an analysis package developed by ITIA-CNR: the so-called Virtual Prototyping Environment for PKM (described in [5]). The package is based on a customization of a commercial multi-body software (Adams^{TM}, MDInc) and on a numerical post-processor with highly efficient PKM-specific routines (implemented in

Matlab™, The MathWorks, Inc.). The tool evaluates, for any machine structure, the workspace but also the actuators' effort and the internal loads due to forces applied on the End Effector, to inertia, to friction and to machine weight. The effects of lumped structural compliances and the effects of manufacturing/assembly errors are also evaluated.

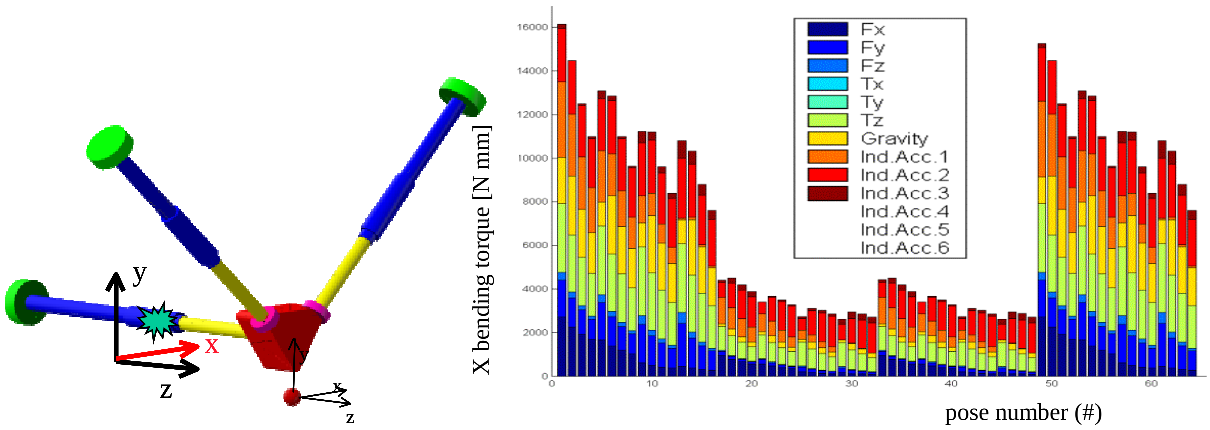


Figure 2 – Bending load at the prismatic joint of the horizontal strut

The analysis has shown the high level of torsional and bending loads in the struts (Figure 2), producing reference values for joint design and motor selection.

5 Finite Element Analysis

When the machine architecture was chosen, a first mechanical design was executed. A Finite Element model was then developed, to better analyze the static and dynamic behavior. A modeling difficulty specific to PKMs, due to the presence of closed chains in the structure, is that the relative positions of all machine parts (like joints rotations and strut elongation) are not independent, being coupled by the inverse kinematics of the machine: the usual modeling approach adopted for serial machines, were mesh points are positioned at the foreseen contact locations, is unfeasible.

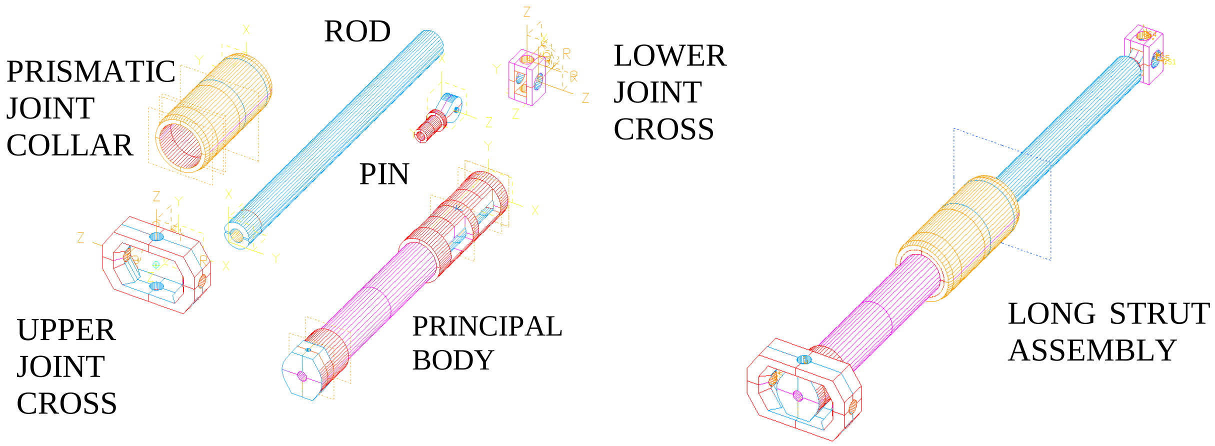


Figure 3 – The strut: main components and the assembly.

In order to be able to quickly analyze the machine anywhere in the workspace, a specific approach was developed, exploiting the capabilities of the selected solid modeler and FE pre-processor (I-Deas® Master Series, SDRC): components meshes are defined only once and associated to each body in the kinematic model of the machine. The model is moved in the desired position and then all the meshes are connected with set of springs representing the joints, using non-localized multi-point couplings for the linear guiding elements that realize the struts prismatic joints. A complete model of the kinematic chain, composed by nut, screw, thrust bearings, coupling and motor, has been defined based on specifications given by the suppliers, in order to correctly reproduce not only its static stiffness, but also its dynamic behavior, as required for the subsequent modal analysis.

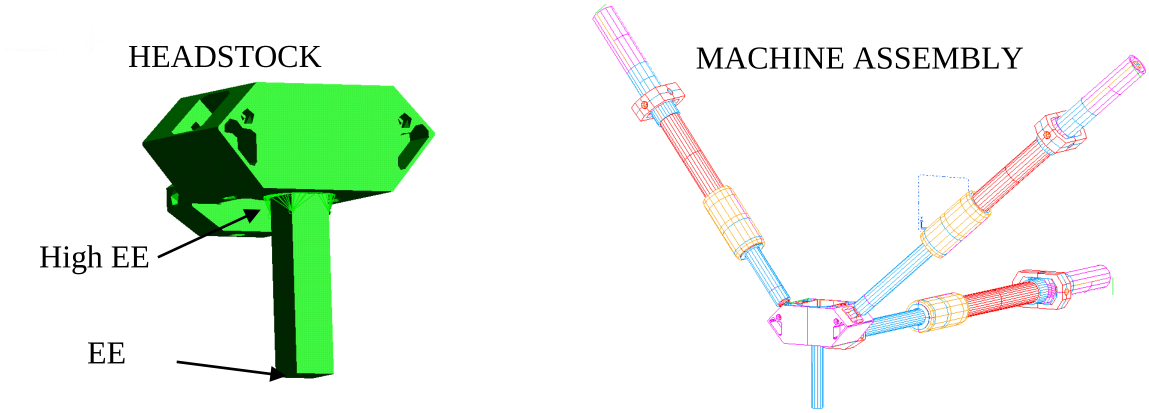


Figure 4 – Head with schematic tool and Parallel Kinematic structure.

The resulting FE model has approx. 42.300 elements, 34.200 nodes, for a total of 103.000 dof. The FE model of the fixed structure has also been developed, but this paper refers only to the PK part of the machine: the external bearings of the universal joints near the motors in this case are directly connected to ground. The herein presented analyses consider the machine with the End-Effector moved by ΔX=-100, ΔY=-100, ΔZ=70 mm in the respect of the workspace center (where the three struts are mutually orthogonal): this position was used for some experimental tests. The "EE" point represents the location of the cutting tool with the bi-rotative wrist used for shoe deburring, while the "High EE" point in Figure 4 has been considered in section 5 to evaluate heavier machining operations.

table 1 Force direct.	compliance at the End Effector [μm/N]		
	X	**Y**	**Z**
X	2,621	0,026	0,079
Y	0,026	2,021	-0,151
Z	0,079	-0,151	0,074

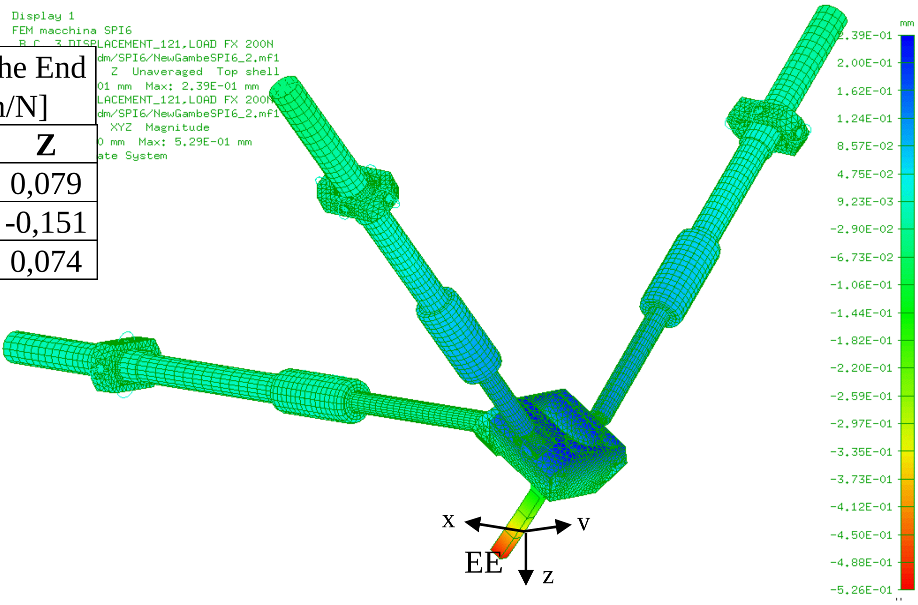


Figure 5 – Machine deformation due to a static load along X: F_X = 200N on the End Effector.

6 Static compliance with locked motors

The first analysis executed was a set of static solutions to evaluate the effect of forces applied on the End-Effector. Figure 5 shows the result of a 200 N force along X: a strong local deformation of the universal joints connected to the head is evident. The static compliance analysis is used also to perform a sensitivity analysis, in order to evaluate the influence of material and coupling stiffness on machine behavior, but final results were not available for this publication.

7 Dynamic analysis

A first analysis of machine dynamic behavior evaluates the capability of the structure of "keeping in place the tool", against the reaction forces generated by the machining process. A simplified analysis can be based on the stability criterion proposed by Tlusty and Koenigsberger ([4]): applying the Nyquist stability criterion for a closed loop system, it is possible to relate, for a given material, the maximum chip depth reachable in stable cutting conditions, as a function of the dynamic compliance at the tool tip.

In order to execute such calculation, the corresponding Frequency Response Functions have been computed, using a modal model, obtained computing 25 dynamic modes, with locked motors. The modal model is accurate only up to around 250 Hz, which is sufficient for the analyses presented. A generic (and quite conservative) modal damping of 2% was considered.

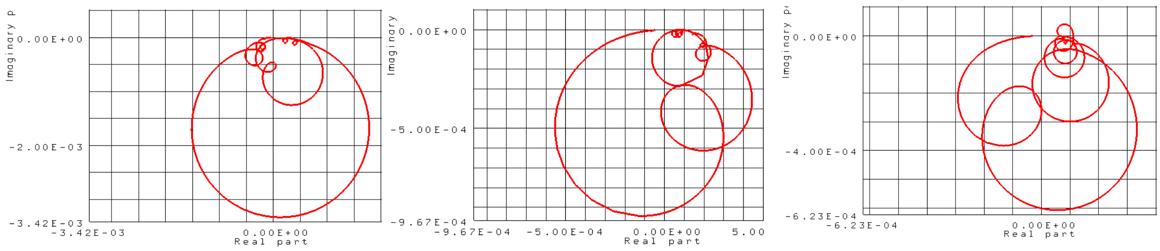


Figure 6 - Freq.Resp.Function (Nyquist plot) displacement/force at the EndEffector

This analysis is not done to evaluate if the required deburring operation will be stable,

$$b_{\lim} = \frac{1}{2 \cdot \mathrm{Min}_{\omega}\left(\mathrm{RealPart}(\omega)\right) \cdot k_s}$$

because it will performed on leather, using a tool with pneumatically controlled compliance: the goal is to obtain a generic quality indicator for the machine structure. For this reason, machining of a generic finishing operation on medium hardness steel was considered (specific pressure k_s= 2500 Mpa, at a chip thickness of 0.2 mm).

These values are only indicative, because an accurate tool description would be required to evaluate the stability of a specific machining operation, nevertheless, they provide a useful evaluation criteria to examine the structural vibration modes. We can see in Figure 7 which resonances contribute to the largest circles in Figure 6: these modes mostly limit the machining capability. It is therefore interesting to examine, in Figure 8, the difference between mode shapes at various frequencies.

Direction	Max. cutting depth $b_{\lim}$ [mm]
X	0.14
Y	0.34
Z	0.52

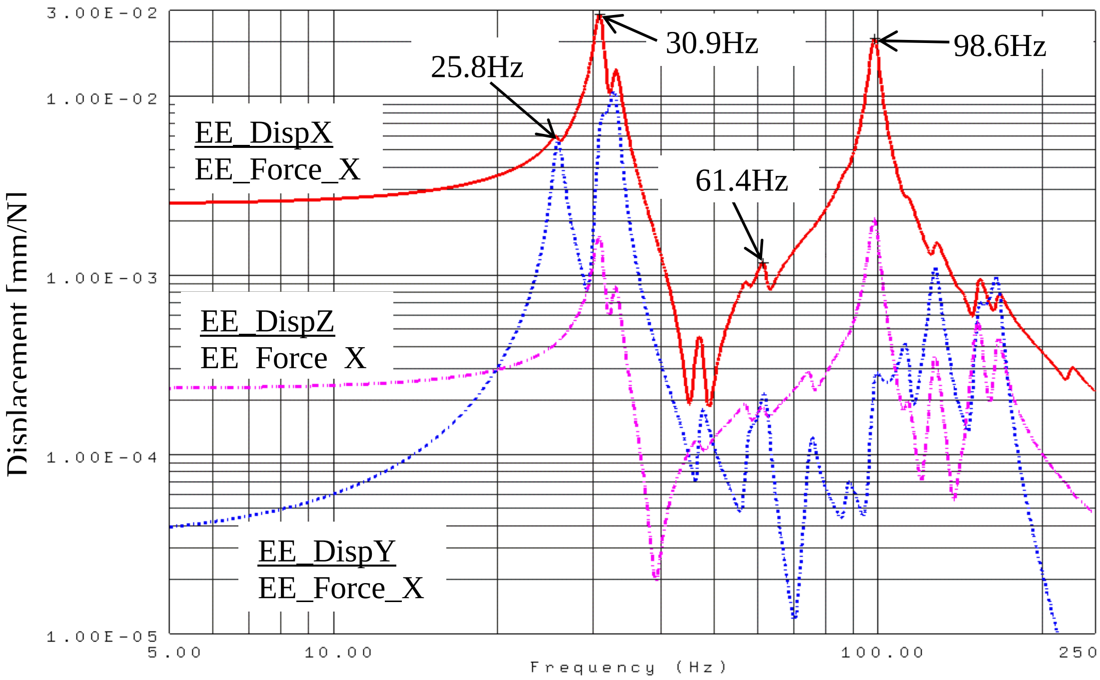


Figure 7 – FRF: (EE Displacements X,Y,Z)/ (EE Force in X direction)

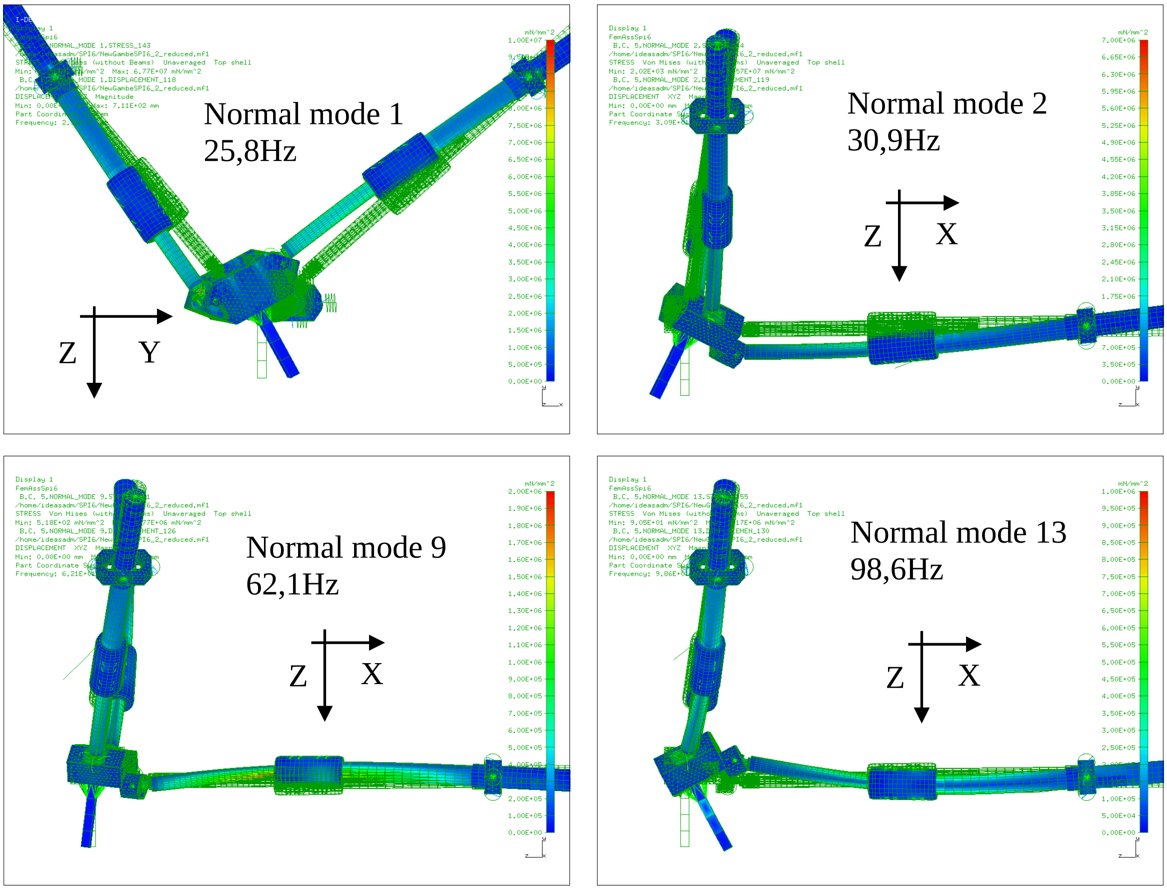


Figure 8 - Samples of different mode shapes typologies

vibration modes with locked motors														
Mode	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Frequency	25,8	30,9	32,9	46,6	46,7	47,2	47,3	57,7	62,1	75,3	87,7	88,7	98,6	113

In the first vibration modes the elastic energy is related to strut torsion and local deformation in the joints, always around the strut axes (see modes 1 and 2 in Figure 8): as a consequence large head rotations are involved, provoking large displacements in X and Y directions at the End-Effector, located far away from the strut neutral axes. The resulting compliance in X and Y direction is quite large. A second set of modes, starting from 57 Hz (see mode 9 in Figure 8), is characterized by local bending deformation in the struts: it is interesting to note, in the FRF of Figure 7, that these modes generate a moderate contribution to the EE dynamic compliance. This behavior is typical of a reticular structure, because a bending mode has a very limited (second order) effect on strut length, which is the only parameter influencing the global deformation of the reticular structure. For PKMs, this is the case of an hexapod architecture. A third set of higher frequency modes (see mode 13 in Figure 8) is a combination of both previous cases and contributes significantly to EE compliance at high frequency (in Figure 7).

8 Structural loads and deformations due to acceleration

The structural deformations provoked by acceleration are somehow underestimated in classical PKM development that usually is focused on static performance indices, like those ones related to the machine Jacobian matrix. When a high dynamics machine is under development, it is instead very important to consider inertial loads, that provoke important strut bending even when, like in an hexapod, the adopted joints can only transmit axial loads to the struts.

The analysis goals were: to evaluate the loads on the recirculating elements of the prismatic joints and to estimate the tool displacement due to machine deformation during acceleration. The first objective, for our iso-static machine, was reachable using the VPE-PKM tool and a rigid structural model. The second analysis requires a flexible model and it is usually executed, e.g. for machine tool design, computing static solutions under force fields representing the inertial loads acting on the moving parts. In our case, such methodology was applicable only on single struts, because when the complete machine moves, each component undergoes a complex acceleration field, as described by the inverse kinematics, and it was impossible to describe such a complex force field (e.g. compared to the uniform acceleration due to a linear axis of a machine tool) in the selected FE package.

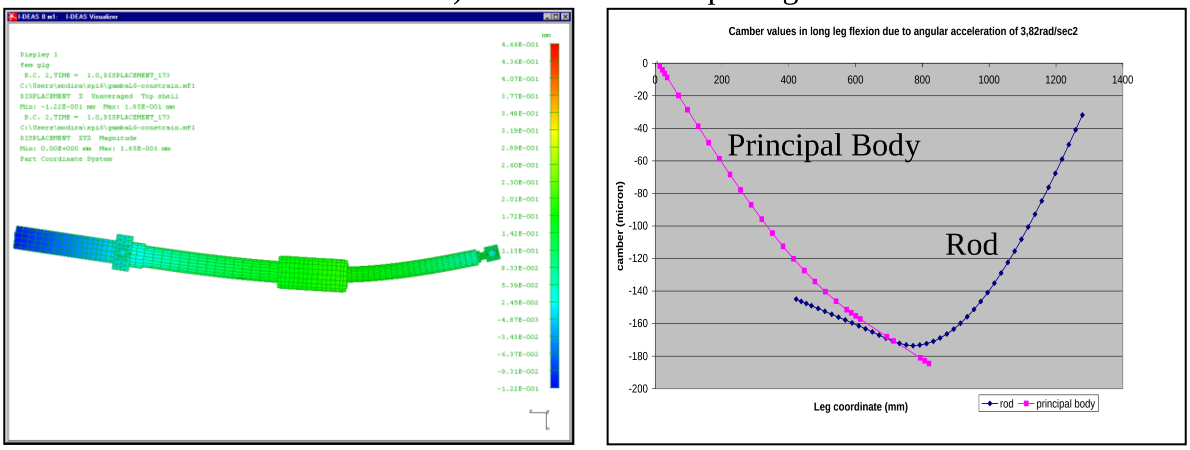


Figure 9 - Long strut deflection due to angular acceleration of 3,82rad/sec2.

Figure 9 shows the strut deformation under a constant angular acceleration of 3,82 rad/sec^2, which is the maximum value for our machine. The joint-to-joint distance variation is a second order effect, that can be computed by a non-linear analysis, but it is usually negligible.

To evaluate deformation of the complete machine, a specific post-processor has been developed (in Matlab®, by The Mathworsk), to handle the complex acceleration field. The FE package computes a reduced order model of the machine with free motors and transmit it to the post-processor using the Modal Neutral File (MNF, by MDInc.) format. The reduced order model, is obtained computing six static modes (corresponding to actuators and End-Effector motion) and 19 constraint dynamic modes, with a Craig Bampton approach [7]. The obtained model, because of the chosen level of reduction, is accurate only up to around 250 Hz, which is sufficient for the analyses presented.

The routine identifies, in the reduced model, the rigid modes, which correspond to null eigen-frequencies, that describe the ideal machine motion, given by the kinematic equations. The user specifies the maximum allowed acceleration at the end-effector and the routine computes, knowing the mode shapes and the inertia matrix, the corresponding inertial forces on the whole machine. Then the corresponding machine deformation is computed, through a static solution with blocked actuators. The resulting displacement of all FE nodes is shown, with element colors related to the entity and direction of nodes displacement. Figure 10 shows the machine deformation due to a 5 m/s^2 acceleration along the Y axis:

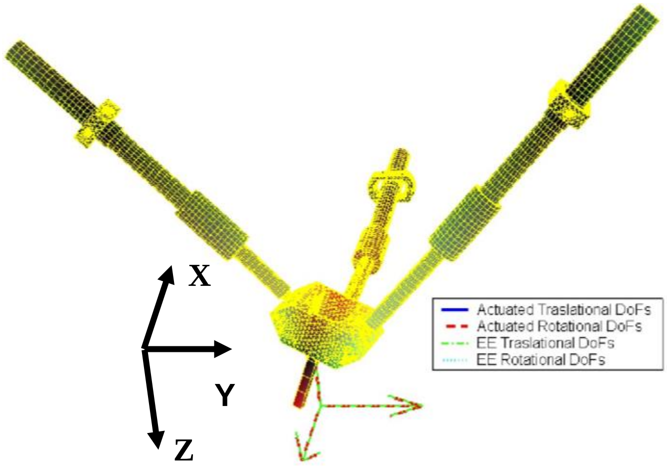


Figure 10 - Machine deformation under acceleration

Acceleration 0,5g along Y axis	
X displacement	-0.0106mm
Y displacement	-0.1260mm
Z displacement	0.0104mm

Knowing the linear relationship between end-effector acceleration and nodes displacement ($\{\delta_{nodes}\} = \mathbf{W}\{a_{ee}\}$), a worst case analysis, indicating the acceleration direction provoking the maximum end-effector motion, can be done by Singular Value Decomposition (SVD) of the **W** matrix.

9 Analyses comparison

Having used different methodologies to analyze the designed PKM, it is interesting to compare the results obtained using a kinematic, multi-body or Finite Element model of the structure.

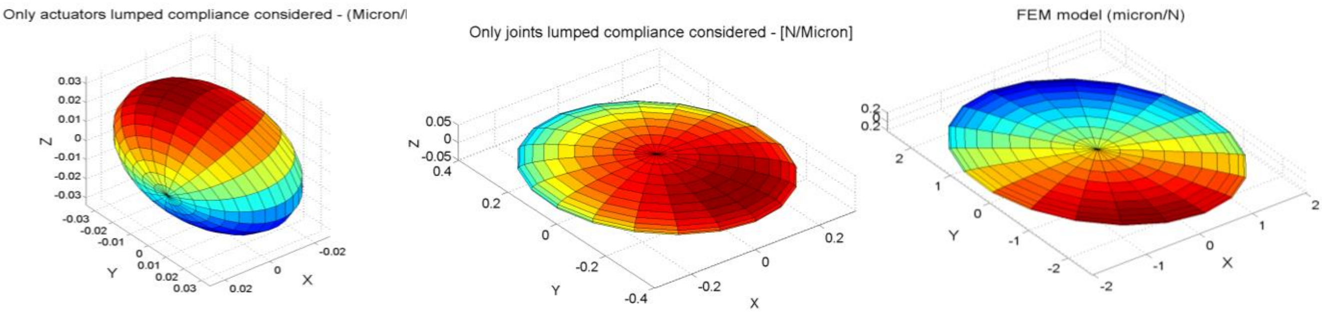


Figure 11 - Static compliance at the EE by kinematic. multi-body and F.E. analyses

Figure 11 shows, by means of ellipsoids with the axes corresponding to the principal directions of the compliance matrix, a comparison of the static compliance at the End-Effector computed with the three approaches: for this machine, the kinematic approach, that doesn't consider any rotation of the headstock, produces an extremely large error, underestimating by a factor of 100 the compliance in the XY plane. The same compliance is underestimated also by the multi-body approach, but "only" by a factor of 3. We can conclude that for an accurate estimation of structural properties, a FE model is required, but the VPE-PKM can produce reasonable predictions during the first design phases, in a much short time and with an automatic exploration of the machine workspace.

Considering, as in the following table, many different aspects covered during machine design, it is clear how the best choice is to use an intelligent combination of the three approaches, in different design phases.

Analysis Methodology:	**analytic**	**multi-body**	**Finite Element**
development time for new architectures	high	low	high
computation time on parametrised architectures	low	medium	not easy
useful for control implementation	required	not useful	not useful
deep comprehension of the kinematic behavior	yes	medium	low
inertial and gravity effects	complex	yes	very well
friction effects	complex	yes	difficult
internal structural loads	complex	yes	very well
effect of manufact. errors, lumped compliances	complex	yes	difficult
dynamic analysis with flexibility	very difficult	difficult	very well
sensitivity for structure optimization	no	poor	yes

10 A first comparison with experimental data

A first experimental campaign on the machine prototype has been recently executed, measuring geometrical accuracy on linear and circular paths, static stiffness at the tool and few Frequency Response Functions exciting the structure with an instrumented hammer and measuring the generated accelerations. A quantitative comparison is today impossible because a machine model complete with the fixed structure and active control would be necessary. At the moment a preliminary evaluation can be done (Figure 12), comparing the experimental FRFs (acceleration/force on the headstock) with that ones computed on the FE model with both free and locked motors: we can already appreciate a good agreement between measured and computed data.

11 Conclusions and future activity

The presented design study underlines the advantages deriving from the usage of a mix of different analysis methodologies, respectively based on kinematic ([3]), multi-body ([5]) and F.E. machine models. Tools like that ones developed by ITIA CNR permit to considerably shorten the development of a PKM for industrial applications. The importance of a correct structural evaluation is also highlighted, especially for machines characterized by passively constrained degrees of freedom and/or high acceleration.

The already mentioned reduced order model, completed with the fixed structure, will be used to model the machine with active control. A deeper experimental test is also planned and quantitative comparisons with numerical results will be done.

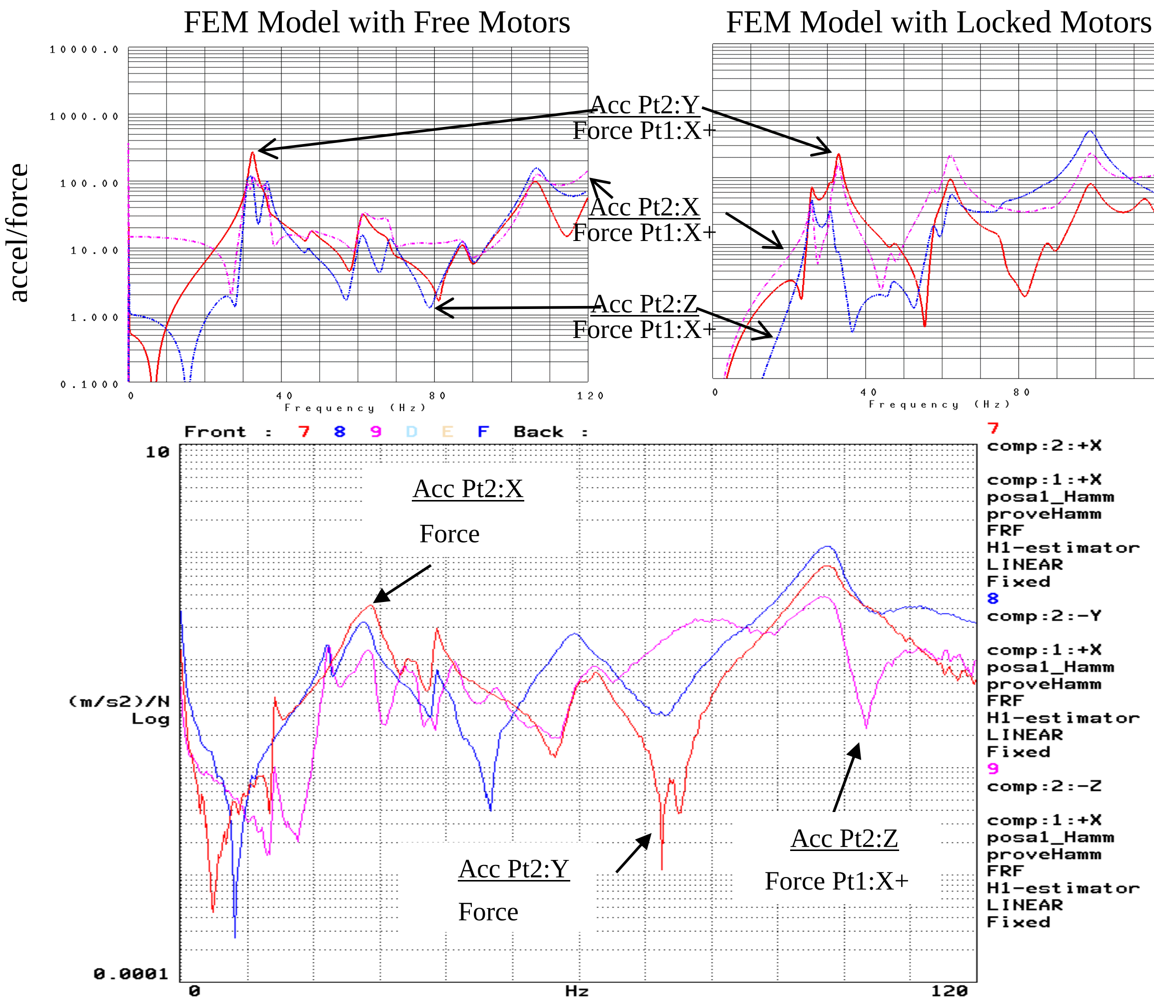


Figure 12 - FEM FRF's and Experimental FRF with controlled axes

12 Acknowledgments

The financial support of the Italian Ministry of University and Scientific and Technologic Research (MURST) is gratefully acknowledged. The presented PKM has been developed within the National Research Project on Innovative Production Systems SPI6: “Fully Automatic and Integrated Shoe Production Systems”, under the responsibility of the ITIA-PKM team, composed by L.Molinari Tosatti, I.Fassi, S. Negri.

13 References

[1] L. W. Tsai, R. Stamper, 1996, A parallel manipulator with only translational degrees of freedom, ASME 96 DETC-MECH-1152, Irvine (CA), USA

[2] R. Di Gregorio, V. Parenti-Castelli, 1998, A Translational 3-DOF Parallel Manipulator, Lenarčič and Husty (eds.), Advances in robot Kinematics: Analysis and Control, pp. 49-58

[3] S. Negri, G. Bianchi, L. Molinari-Tosatti, I. Fassi, G. Di Bernardo, “Analysis and Design of a 3 dof Parallel Kinematic Machine for Shoes Manufacturing”, 9th International Workshop on Robotics in Alpe-Adria Region (RAAD), June 1-3, 2000, Maribor, Slovenia, pp.313-318

[4] C. Gosselin, 1990, Stiffness Mapping for Parallel Manipulators, IEEE Trans. On Robotics and Automation, vol.6, n. 3, pp.377-382

[5] L. Molinari-Tosatti, G. Bianchi, I. Fassi, R. Maj; “Analysis and Design of a Translational 3 dof PKM for Light Deburring Operations”; Matador Conference 2000; Manchester UK

[6] Tlusty, Koenigsberger; “Machine Tool Structures”, Pergamon Press

[7] Craig R.R., Bampton M.C.C, Coupling of Substructures for Dynamic Analysis, AIAA Journal, Vol 6, No 7, July 1968, pp. 1313-1319