A DESIGN METHODOLOGY FOR HAPTIC DEVICES

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ABSTRACT
This paper presents a design methodology for optimal design of haptic devices, considering aspects from all involved engineering domains. The design methodology is based on parametric modeling with an iterative and integrated design approach that leads to easier design space exploration for global optimal design and initial verification in the conceptual design phase. For design optimization, performance indices such as; workspace volume, isotropy, stiffness, inertia and control of the device, from all involved engineering domains were considered. To handle this complex and non-linear optimization problem, a multi-objective algorithm together with a new proposed optimization function was used, to obtain an optimum solution. A case study, where the methodology has been applied to develop a parallel haptic device is presented in detail in this paper. The simulation and experimental results obtained from this test case show significant improvements in the performances of the device.

Keywords: Design methodology, haptic devices, parallel mechanism, optimization and performances.

1 INTRODUCTION
A haptic device is a robot-like mechanism that provides an extra sense of touch; force/torque feedback capability to an operator based on what he/she discovers and interacts within a virtual world or remote environment. Application of these devices is emerging in various fields such as medicine, telerobotics, engineering design, and entertainment [1, 2]. The work presented in this paper is related to the design methodology for design and development of these devices. Basically, haptic devices present a difficult mechatronic design problem, as they are required to be backdrivable and light (low inertia and friction), as well as being able to provide enough stiffness, feedback forces and torques when reflecting forces from stiff contacts. It is also desired that motion, forces and stiffness provided by the device are isotropic (same in all direction). Furthermore, structural transparency and stability is required so that the operator feels free space motion as free, while during interaction with virtual objects feels the dynamics of the manipulated objects, not of the structure of the haptic device.
The design of the haptic devices is an iterative process, and an efficient design requires a lot of computational efforts and capabilities for mapping design parameters into design criteria, hence turning out to be a multi-objective design optimization problem. Thus it presents a high level of computational complexity for finding an optimal design solution. The main focus of this research is to *develop a methodology for design and optimization of haptic devices*. The methodology will be based on parametric, iterative and integrated modeling design approach that leads to easier design space exploration and early verification during product development.
In traditional mechatronic design methodologies, the mechanical system is developed independently of the electronic and control system, and at a later stage they are integrated with each other [3]. For example the sequential design approach as shown in Figure 1a [4] has the advantage of dividing a large and complex design problem into several smaller design problems. Here the mechanism, actuation and control design are designed independently, which reduce the computational complexity of the problem. However, neglecting to include aspects from dynamics and control point of view into the design of mechanical system, may result in a system with non-optimal dynamic performance. This may, in the worst case, require major redesigns of the electromechanical system late in the design process, e.g. as reported in [3, 4, 5, 6, 7].
Fathy et al [6] identify four different design approaches for integrated optimization of mechanical and control system design: sequential, iterative nested and simultaneous (Figure 1b). The first two approaches have the potential of finding designs that are optimal within each domain, but sub-optimal on the system level. The forth one “simultaneous” consider the whole system at a time for optimization, it can provide the global optimal solution, but at a high computational cost for complex systems.

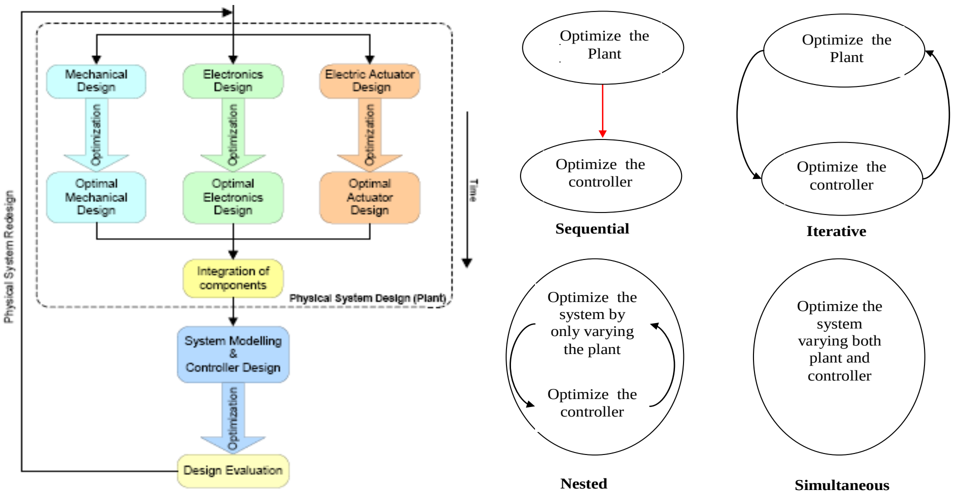


Figure 1. a) Traditional mechatronic design approach [2] and b) Different design methodologies [4].

Roos [4] proposed a new integrated design methodology for design of electro mechanical servo systems. This approach is based on two types of models; static and dynamic models. Static model include parameters related to the physical model, and dynamic model include the dynamic parameters (required for control design) in the design process. Roos applies a simultaneous design approach to the whole system to find an optimal solution. This design methodology works efficiently for simple design problems, but its performance becomes worse for complex design problems due to the increased level of computational complexity. A similar approach, based on design decision variables from all involved disciplines for optimal design of product has been proposed by Bart at al [7].
The approach taken in this paper for a methodology can be categorized to the "Nested approach" by Fathy et al. [6] and [7]. The motivation for developing a methodology for development of haptic devices, specifically using parallel kinematic structures, is that this type of device has complex structures which give many structural advantages like high stiffness and low inertia, but also give complex optimization problem and a complicated control system. The remaining part of the paper is organized in sections. Section 2 explains the design methodology, Section 3 presents the case study, and section 4 presents results and discussion respectively.

2 DESIGN METHODLOGY

In this work, a methodology has been developed for design and optimization of haptic devices. This methodology provides a model based parametric, iterative design approach that leads to an easier design space exploration and initial verification during process development as shown in Figure 2.

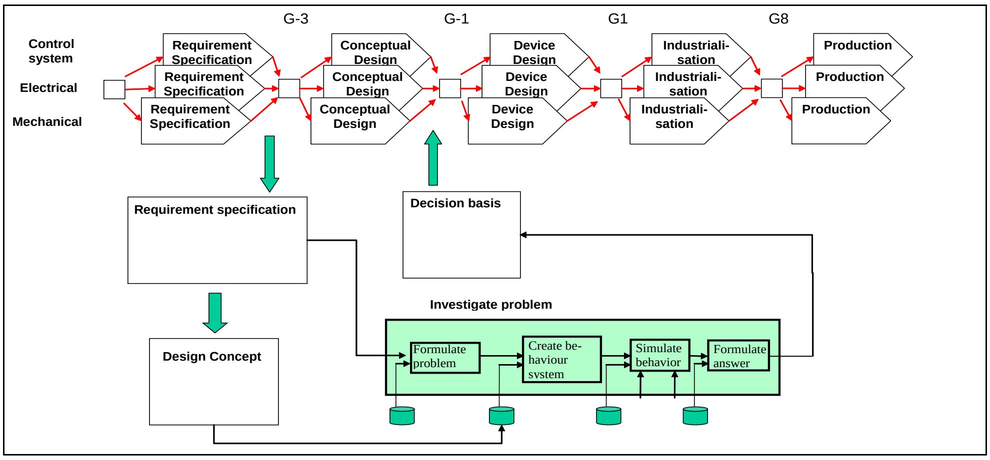


Figure 2. A general design process model for haptic device development, after Andersson [4].

The first stage of the methodology is to define the more direct device requirements and market. These requirements include on an abstract level: Degree Of Freedom (DOF), workspace, force/torque capability, stiffness and control dynamics. The second stage of the methodology is conceptual design; here the methodology should include preliminary analysis of the number of DOF, workspace, actuator requirements and singularity points (which shouldn't exist within the workspace). In parallel, a rough layout of the mechanical structure with preliminary material properties should be made as well as an investigation of possible control strategies and components to use. Next is device design which includes design of the mechanical structure, actuation, transmission, and also analysis of workspace, stiffness, inertia, force/torque capabilities and backdrivability. In parallel with designing the mechanics and actuation, the models necessary for control design are derived. For the control design, sensors and control strategies are selected and designed. Before the device is finally built and the control implemented, thorough work should be made for optimal design using simulation and rapid prototyping to verify performance and if necessary iterate within the design process.

Apparently there is a large number of design parameters that needs to be fixed before a final design is achieved. In addition to the direct specifications it is important to consider other design criteria towards an overall optimal design. Such criteria can include: (1) minimum footprint/size to workspace ratio; (2) uniform motions, forces and stiffness capabilities over the workspace (kinematic isotropy); and (3) minimum inertia of structure, transmission and actuation capabilities (dynamic and control characteristics). All these design parameters are almost mutually dependent, thus leads to a large complex design problem with high computational complexity. To cope with this problem, a global optimal solution is determined using a multi-objective optimization criteria based on efficient computational tool such as multi-objective genetic algorithm. The different phases of the design methodology are discussed in detail in following sections.

2.1 Requirement Specification

Design starts with a need, when satisfied, results in a product that fits into existing market or creates market for its own [7]. In the first step, a literature review and market analysis should be performed in order to identify the potential users and their requirements. From the statements of needs a requirement specification is formulated.

As a minimum for haptic devices, this should include requirements for size (footprint), workspace, Degrees Of Freedom (DOF), force/torque capability, and stiffness.

2.2 Conceptual design

In the conceptual design phase the development of structure, mechanical device and control system should be performed in parallel since the performance of a haptic device is highly dependent on the interaction between all of these systems.

Some activities (and suggested tools to use) that should be performed during this phase are listed below;

- Selection of alternative structures to examine for further development. This information is given by the literature review and market analysis that have been performed earlier, when stating the requirement specification. This should be complemented with a more detailed study of possible structures for the intended application for the device.
- Modeling and analysis to determine numbers of DOF, preliminary actuator requirements and preliminary dimensions for the wanted workspace for selected structures. These are some of the basic requirements for a haptic device to achieve capabilities for feedback in the required DOFs and workspace. For these types of analysis MBS modeling and analysis software, e.g. Adams View® [8], is recommended.
- Investigation and preliminary selection of motors based on the calculated actuator requirements. In addition evaluation and preliminary selection of encoder and transmission should be made.
- Inverse and direct kinematic modeling of the selected structures. Development of inverse and direct kinematic models is a pre-condition for performing kinematic optimization and is also needed for development of the control system. For this type of modeling and analysis, Matlab [9] is recommended.
- Optimization of the kinematic structure. This is a crucial task for haptic devices that are based on parallel kinematic structures. The optimization turns out to be a multi-criteria optimization problem. For these types of problem the use of a genetic algorithm has been proved to be

successful in finding a global optimum solution. The goal function should include indices for workspace, isotropy, torque/force and stiffness requirement and inertia of the device. Suggested software to use here is Matlab [9] and MOGA (Multi Objective Genetic Algorithm) toolbox [10].

- Rough layout of the mechanical design based on the MBS analysis and optimization results. This is a traditional engineering design task to make a preliminary assembly layout of the device based on optimization results and MBS analysis. Tasks to consider in this phase are selection of motor, transmission and search for standard components to use for e.g. joints, as well as basic design and preliminary material of support structure. For these tasks any CAD 3D modeling tool is feasible.
- Alternative control strategies for the haptic device. The requirement on the device is to get a frictionless feeling when moving the device in free space and to achieve force/torque feedback when entering contact with an object. This means that the control system have to compensate for the inertia and friction that always occur in real systems. The task here is to investigate optimal control strategies and different approaches to compensate for these effects.

After selecting the candidate structures to consider for the device in hand, above steps can be done in parallel assuming that a parametric modeling approach is used for all these activities.

2.3 Device design

The outlined activities during the conceptual design phase all follow the verification process described in Figure 2 which has the purpose to produce a decision basis to decide how to proceed to the next design phase. This results in selection of one (or maybe two) candidate structures for further development and final design. The following design phase is the device design phase. Some activities (and suggested tools to use) that should be performed during the device design phase are listed below.

- Mechanical design to make the detail design of the device based on the optimization results. This includes careful selection of standard components, if possible (e.g. joints, electric motors), detail design, material selection and manufacturing documents of components to be manufactured. For these tasks any CAD 3D modeling tool is feasible.
- Prototype creation. Once the mechanical design is determined a physical prototype should be built. This includes the manufacturing of some components and ordering of standard components.
- Control design. As soon as we have a physical prototype we can start testing different control strategies being investigated during the previous design phase. For the initial tasks dSpace [11] can be used but for the final implementation a suitable micro controller should be selected as well as a software development tool for implementing the control system in the micro controller.
- Testing of the prototype. After the prototype being built we should start with the testing of the device. Initially mechanical stiffness and clearance can be tested using a CMM (Coordinate Measuring Machine). After that, testing of the complete device should be made in a controlled and restricted environment. First, simple tests of contact conditions and free space motion should be made and thereafter more complicated contact conditions, requiring many DOF's feedback as a result of a contact, should be investigated.

3 APPLICATION EXAMPLE: DESIGN OF A 6 DOF HAPTIC DEVICE

The proposed design methodology in section 2 has been applied to the development of a parallel 6-DOF haptic device. The intended application of the device is a milling simulator that will be used in curriculum for surgical training of vertebral operations [1]. In this scenario a haptic device is used to achieve manipulation capabilities and force/torque feedback in 6-DOF during simulation of vertebral operations to achieve a user interaction that gives a realistic impression due to the milling process of a virtual modeled bone tissue. Such procedures involve removing bone by drilling or milling, including processing of channels and cavities, hence requiring 5-6 degrees of freedom and stiff contact feedback to the user.

3.1 Requirement specification

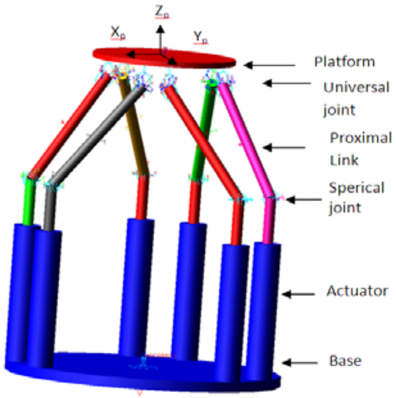
In this first step, a literature review and market analysis has been performed in order to identify the potential users and their requirements. From the statements of needs a requirement specification is formulated. The preliminary specifications given here have been obtained in dialogue with a tentative user, in this case a surgeon. The application domain is completely new and unique, thus it is difficult to obtain specific requirements. The initial requirements for the haptic device are as follows [12].

- The device should have 6 actuated degrees of freedom.
- The whole device should fit within the space of 250x250x300 [mm].
- The translational workspace should be a minimum of 50x50x50 [mm].
- The stiffness of the device including actuation and control should be a minimum of 50 [N/mm].
- The TCP peak force/torque performance should be at least 50 [N] and 1 [Nm] in all directions.
- It should be possible to place it on a table in front of the operator, easy to access for the user.

The outcome from this stage is a requirement specification on an abstract level, based on identified users of the device.

3.2 Conceptual design

From the literature review in the first stage of methodology, haptic devices that are currently available in the market or at a prototype stage, both serial and parallel structures are being used [2, 13-23]. However, since parallel structures have some significant advantages as compared to serial ones, e.g. high stiffness, high accuracy and low inertia, we have chosen two concepts based on parallel kinematic structures. In the next step, these concepts were investigated for structural analysis such as numbers of DOF's, workspace and force/torque requirements. For structural analysis, these concepts were modeled using Adams View® MBS software [8] as a main tool.
The first concept is a modified Stewart Gough mechanism [21, 23, 24], which consists of a fixed base, a moving platform, and six identical legs connecting the platform to the base shown in Figure 3a. Each leg consists of an active linear actuator fixed to the base, a spherical joint, a constant length proximal link, and a universal joint. This 6-PSU (active Prismatic, Spherical and Universal) joint configuration was used to get 6 DOF. For parametric design of this structure, six design parameters were considered: range of actuators motion (L_{min}, L_{max}), length of proximal link c_i, radius of base r_b, radius of platform r_p, angle between the base pair of joints 2α and angle between the platform pair of joints 2β , see Figure 3b. The attachment point pairs are symmetrically separated 120^o and lie on a circle, both on the base and the platform. The platform attachment points are rotated 60^o clockwise from the base attachment points.



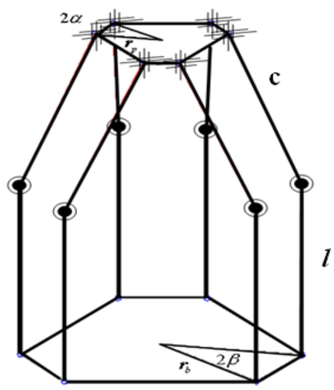


Figure 3. a) Conceptual model of concept 1 in Adams View b) Kinematic structure of the concept 1.

The second mechanism is based on a hybrid parallel kinematic structure called TAU, shown in Figure 4a. This concept consists of fixed I-column, a moving platform and three parallel chains (1, 2 and 3) which connect the base frame to the moving platform. In this structure chain 1 and chain 2 are symmetrical while chain 3 is unsymmetrical as shown in Figure 4. Each symmetrical chain has two active rotational actuators, one attached to the I-column while another one is mounted on the upper link U_1, U_2. Furthermore chain 1 and 2 have extra two proximal links connecting the platform to upper links U_1 and U_2 to increase the structural stiffness. The third chain, chain 3, has also two active rotational actuators, one attached to the I-column and the other mounted at the top of the device.
For parametric design of this structure, five design parameters were considered: position of each parallel chain with respect to the base coordinate system {N} is at 1.5d, 3d and 4.5d, which is function of parameter *d*, length L_1 of the upper arm, length L_2 of proximal links in each chain, radius of platform R_p, elevation angle θ_{32nom} (nominal angle for θ_{32}) of the upper arm U3 of chain 3 with orientation of the base frame as given in Figure 4b.

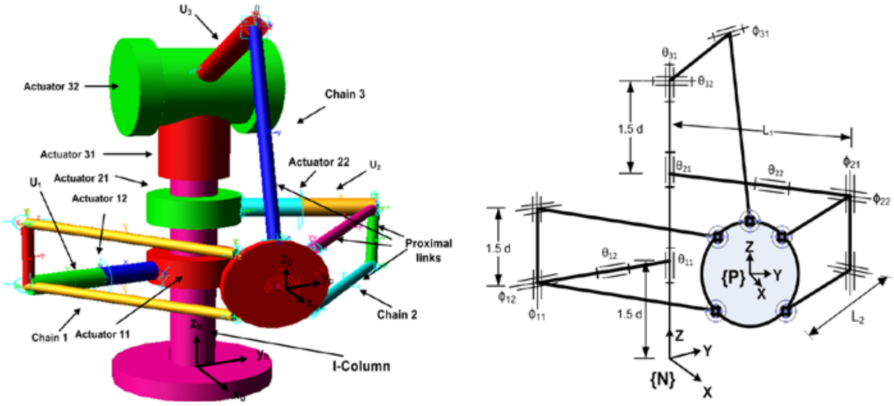


Figure 4. a) Conceptual model of concept 2 in Adams View b) Kinematic structure of the concept 2.

In the next step of conceptual design phase, we investigated the basic performance of these concepts, utilizing the verification process proposed by Andersson et al [12]. First we assign initial dimensions to the device that fulfills device size requirements. Thereafter, we focus on investigating three main properties; No's of DOF, device workspace, and actuator performance giving wanted force/torque performance around TCP.

The first concept (1) provides 6-DOF motion at TCP. The translation workspace provided by the concept is $\pm$ [50, 50, 50] mm in X, Y and Z direction as shown in Figure 5a. The maximum range of rotation measured at the center and at each corner of the selected cube within translational workspace was $\pm$ 40^o around X, Y, Z direction, while in combination it ranges from $\pm$35^o around all directions.

The second concept (2) also provides 6-DOF motion at the TCP. The translation workspace provided by this concept is $\pm$ [85, 85, 100] mm in X, Y and Z direction, shown in Figure 5b (right). The results from the rotation analysis show that the rotation angles for X and Y axis are $\pm$52^o in all eight corners, when rotating one axis at a time. While in combination the range of rotation is decreased to $\pm$ 30^o in all the corners. Around the Z-axis the structure can provide rotation up to $\pm$ 40^o.

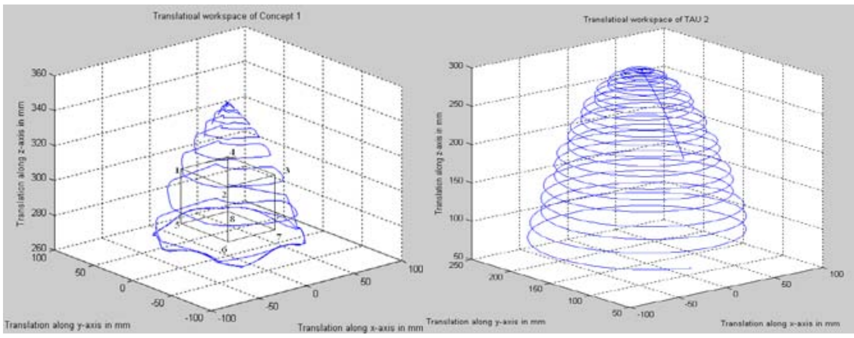


Figure 5. a) Workspace for concept 1(left) and b) for concept 2 (right) in 3D space.

To measure the force and torque capability, a constant force of 50 N was applied on TCP, then TCP was moved on a specified circular path within workspace and reaction forces was measured on each actuator. The force/torque analysis of concept 1 shows that the measured reaction forces at active linear joints increased as the TCP moves along the specified path to the outer circle see Figure 6a. The torque analysis of the second concept shows that higher torque is required on actuator 32, see Figure 6b with a few high peaks. These peaks occur as a result of an incorrect modeling of the load when moving in all directions (xyz) at the same time and should be disregarded.

The outcome of these preliminary analyses in the conceptual design phase is used as a decision basis to select the mechanism that we will consider further for design optimization. Based on the torque requirements and low inertia due to the fixed motors, concept 1 was selected. Next, it is important to consider other design criteria towards device design optimization. Such criteria can include: (1) minimum footprint/size to workspace ratio (workspace); (2) uniform motions, forces and stiffness capabilities over the workspace (kinematic optimization); and (3) minimum inertia of structure, transmission and actuation capabilities (control design). The kinematic and control optimization were performed in parallel based on the defined performance indices in the following section.

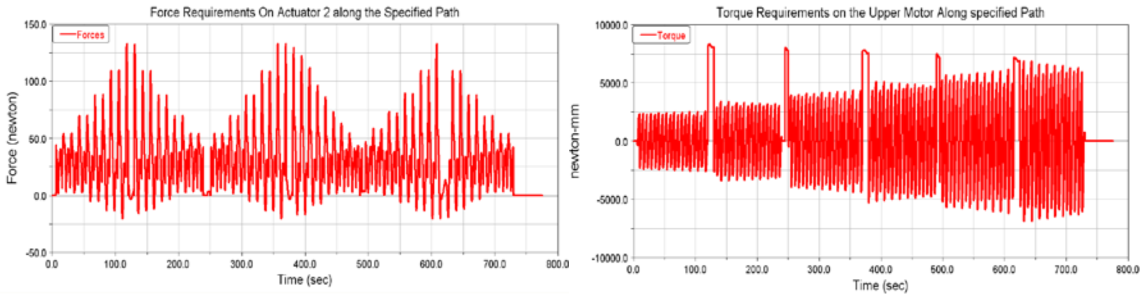


Figure 6. Forces requirements (left) concept 1 and (right) concept 2 for 50[N] applied force on TCP within workspace.

1. *Workspace Index*

Workspace is the working space that the haptic device can operate within. It is defined as a three dimensional space that can be reached by TCP. The boundaries of this space were determined using inverse kinematics. A Cartesian workspace within a range of ±75 mm along all three axes was scanned using an evenly spaced grid. Finally, the volume of the workspace can be calculated as $v = \int_v dv$. Where *dv* is the volume of a grid element. The optimization criterion is to maximize the workspace volume while keeping the footprint (size) of the device as a constraint.

2. *Isotropy Index*

The kinematic isotropy index (II) indicates how evenly the device produces motions (velocities) in all directions in the workspace. A haptic device is called "isotropic" if at least in one point of the workspace some of its kinematic properties are homogenous with respect to all directions. The isotropy index is defined as the ratio of minimum singular ($\sigma_{\min}$) to maximum singular ($\sigma_{\max}$) values of the Jacobian matrix (J) [24], according to

$$II = \frac{\sigma_{\min}(J,w)}{\sigma_{\max}(J,w)}, 0 \le II \le 1, \tag{1}$$

where w is the pose of TCP in workspace. If, at a certain point, the isotropy index approaches unity, the haptic device can produce a more uniform motion in all directions. While on the other hand if the isotropy index approaches zero, it indicates operation close to singular points in the workspace, which needed to be excluded from workspace($II \ge 0.005$). To represent the average of the device isotropy index over the whole workspace, a global isotropy index is defined as

$$GII = \frac{\int_v II \cdot dv}{v}. \tag{2}$$

A higher value of GII represents a mechanism with a better isotropy characteristic within its workspace, and thus the criterion is to maximize this index.

3. *Force requirement Index*

The force requirement index (FI) is defined as the maximum magnitude of an actuator force required for a unit applied load on the tool center point (TCP). As the applied load on the TCP is related by the Jacobian matrix to the forces required on the actuators, the force requirements index is defined as the maximum singular value of the Jacobian matrix as $FI = \sigma_{\max}(J,w)$. A global force requirement index which represents the average of the device force/torque performance over the selected workspace is defined as

$$GFI = \frac{\int_v (FI) dv}{v}. \tag{3}$$

A smaller value of the force requirement index implies that less capacity of the actuators is required i.e. this index should be minimized.

4. *Stiffness Index*

From mechanics point of view, stiffness is the measured ability of a body or structure to resist deformation due to the external forces. For the selected mechanism, the stiffness at a given point in the workspace can be characterized by its stiffness matrix [25]. This matrix relates the forces and torques applied at TCP to the corresponding linear and angular Cartesian displacement. If F represents the external applied forces on TCP, then the corresponding linear and angular Cartesian displacement can be determined from ellipsoid sphere with the lengths of horizontal axis and vertical axis being the maximum value and minimum value of the deflection, respectively. The direction with largest deflection of the moving platform has the lowest stiffness. Thus, the maximum value of deflection of the moving platform can be regarded as the evaluating index of stiffness when a unit force F acts on the moving platform. The maximum and minimum deformations can be obtained from the eigenvalues of the stiffness matrix $(K^{-1})^T$ K^{-1}as $\|p_{\max}\| = \sqrt{\max(\lambda_p)}$ and $\|p_{\min}\| = \sqrt{\min(\lambda_p)}$. The global stiffness index representing the average stiffness within the workspace is defined as

$$GSI = \frac{\int_V \| p_{\max} \| dv}{\int_V dv} \tag{4}$$

Here the criterion is to minimize the global stiffness index and so maximize the stiffness of the structure.

5. *Inertial Index*

An inertial index is based on the mass matrix of the device that represents the dynamic characteristic of the device. The mass matrix is obtained by computing the masses and inertia of all the moving components (platform, actuators including motor inertia and proximal link) in the task space [26]. In the case of a haptic device it is needed to minimize inertial effects (minimizing the maximum singular value of the mass matrix). Thus the inertial mass index can be defined, using the maximum singular value of the mass matrix (M) as

$$IMI = \frac{1}{1 + \sigma_{\max}(M, w)}. \tag{5}$$

The criterion here is to maximize the inertial index (minimize max. singular value), to obtain lower dynamic effects in the workspace.

6. *Multi-objective optimization*

As in our case all the actuators are identical to each other, thus they have the same stiffness and thus the stiffness matrix K reduce to a diagonal matrix, which simplify the criteria as $K=kJ^TJ$ in task space. Thus the condition number or singular value of the matrix J^TJ *need to be optimize instead of* kJ^TJ [27]. Also in the case of isotropy index we minimize the maximum singular value of the Jacobian matrix, the same criteria as for force and stiffness indices (dependent on Jacobian matrix), thus we effectively reduce this MOO problem to three main indices see equation (6). Furthermore, the selected indices are normalized such that all indices contribute equally in the optimization process. In this normalization each index is divided by a numerical value, calculated from the mid values of the given design parameters space according to equation (8) and their design parameters input space. Finally, a multi-criteria design objective function is defined based on these indices as

$$GDI = \min\left[\frac{VI}{VI_m}\frac{GII}{GII_m}, \frac{IMI}{IMI_m}\right], \tag{6}$$

where subscript m indicates mid values of the parameter space. The main advantage of this new approach as compared to the traditional objective function presented in [28-30], is to assure that all design indices are equally active in the optimization process. For optimization we also need to define the constraints and allowed range for the design parameters (DP) as per the specification of the device for all sub levels.

Finally the optimization problem can formulated as

$$\begin{aligned} &\text{maximize} \quad GDI \\ &\text{subject to} \quad J(X), M(X) \succ 0 \quad \mathrm{X} \in \mathrm{v} \\ &\qquad\qquad L_{\mathrm{i}}\_\mathrm{min} \le L_{\mathrm{i}} \le L_{\mathrm{i}}\_\mathrm{max} \\ &\qquad\qquad \mathrm{Dp\_min} \le \mathrm{Dp} \le \mathrm{Dp\_max}, \end{aligned} \tag{7}$$

where (L_i) represents the stroke of actuator. To solve the above described nonlinear and non convex MOO problem, we applied three different approaches/algorithms; Weighted sum, MOGA-II [31] and NSGA-II [32] to find the Pareto optimal solution [33]. These approaches were implemented in Matlab and run with 100 as initial population size and maximum number of generations as 100.

7. *Result from Optimization and pareto fronts*

The Pareto front resulted from the above described optimization approaches is shown in Figure 7a, and b. The pareto optimal solution obtain from MOGA-II is shown as dense points in Figure 7b, where the performance of all the indices are best and can't be improved more, unless it deteriorate the other one. The solution obtained from these three approaches is approximately the same.

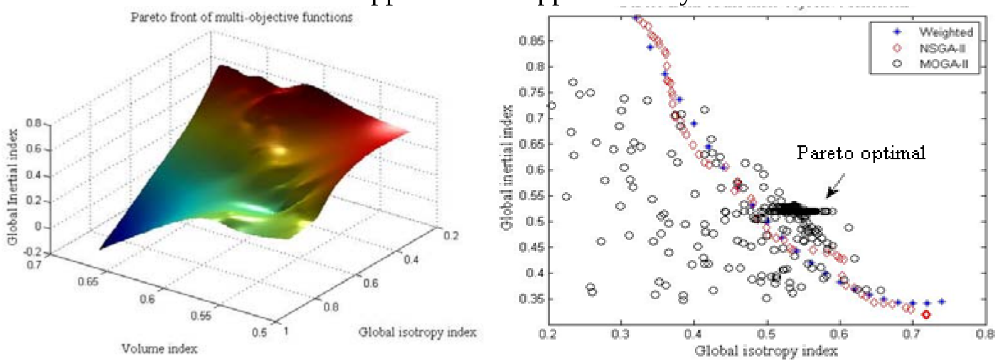


Figure 7. a) Pareto front of the volume, global isotropy and inertial indices (left) and b) pareto optimal solution obtained from the applied approaches (right).

The results from the design optimization process using MOGA-II, with design parameters are presented in Table 1.

Table 1. Design parameters bounds and optimal values

Parameters	Min	Max	Optimal
l *[mm]*	120	150	129.4159
c *[mm]*	120	150	125.4555
R_b *[mm]*	100	125	118.1799
R_p *[mm]*	40	60	54.9920
β *[deg]*	10	30	18.1519
α *[deg]*	10	30	10.5485
Volume index, VI	-	-	0.9790
Global isotropy index, GII	-	-	0.255
Inertial index IMI	-	-	0.8522

Furthermore the set of optimal design parameter values, obtained from genetic algorithm was used to evaluate the performance of the device. In order to visualize the variation of isotropy and force requirements indices in the optimized workspace, the TCP is moved in a circular path in the x-y plane with small incremental changes in radius. When the radius reaches the maximum, the TCP is shifted to the next x-y plane with a small increment in the z-direction. At each small grid isotropy and force requirements indices are measured. Figure 8a shows that the device has good "isotropic" behavior around the central position of the workspace. The force requirements is small for unit applied force around the center of workspace while it increases as the TCP moves away from central point (see Figure 8b). This characteristic is also quite obvious from the isotropy definition of the device. From the index values corresponding to the optimal parameter set and by analysis made in Adams View®, it is concluded that workspace and isotropy requirements as represented in section 3.1 are fulfilled. The variation of stiffness K within the workspace (see Figure 8c), which shows the structure is stiffer when the actuators are at lower limits, and less stiff when the actuator reaches its maximum position.

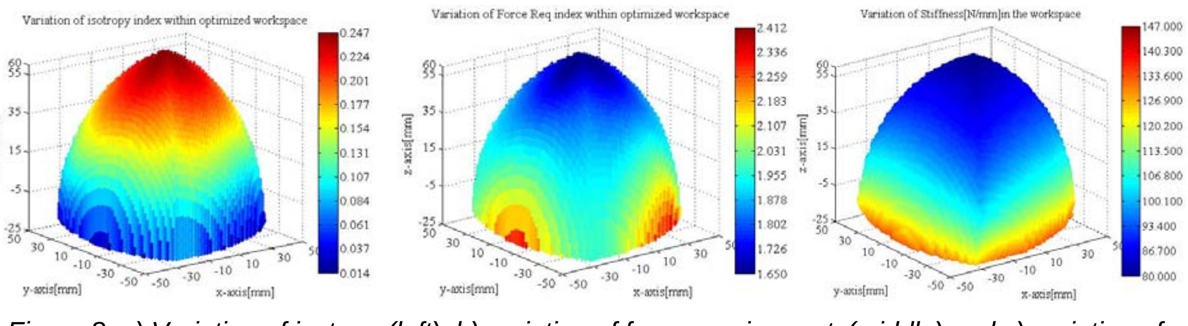


Figure 8. a) Variation of isotropy(left), b) variation of force requirements(middle) and c) variation of stiffness(right) within in the workspace.

In conceptual design phase, we also perform in parallel control design optimization, to obtain a structure optimized both from kinematic, dynamic and actuation point of view. Here the main performances that needed to be considered are transparency and stability of the device. The requirement on transparency means that motion in free space should feel free while motion in contact with a virtual or remote object should result in feedback forces and torques as close as possible to those appearing in the remote or virtual world.
In free space motion, transparency is affected by the dynamics (moving inertia, friction) of the device and dynamics of the operator. Keeping the device inertia as low as possible as well as compensating for it in control design will increase the transparency of device. The task here is to investigate optimal control strategies and different approaches to compensate for these effects. The modeled optimal design control strategy is shown in Figure 9. The control design is based on optimal load from the optimal kinematic structure of the device (complete integrated system).

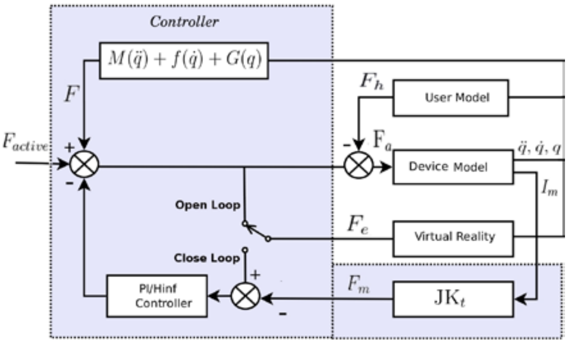


Figure 9. Optimal control structure of the 6-DOF haptic device.

As shown in the Figure 9, the control design is based on computed torques of the device dynamics and current feedback. We measure the current I_m in each motor and thus indirectly torque and forces produced by the haptic device (using motor torque constant K_t and Jacobian matrix J). A force/torque error feedback control is obtained using a PI controller with low pass filter. Input to the PI control is the error between reference force from virtual world F_e and filtered measured force F_m. Then a compensation for the dynamic influence F of the device is added to the control signal as a feed-forward term. The aim of this feed-forward term is to increase the transparency of the device, i.e. the user should not feel the inertia and friction of the device itself, only of the tool.
Outcome from the conceptual design phase is the complete optimal design of the 6-DOF haptic device.

3.3 Device Design

The CAD model for the prototype was developed based on the final set of design parameters from GA, pareto diagram, control strategy model and sensitivity plots [24, 26]. The developed model is shown in Figure 10 below. The size of the model is 250x250x300 mm. Six Dc motors model GR 53x58, 60W were fixed at base and a cable transmission mechanism with pulley was used to convert the angular motion to the linear actuator motion. The cable transmission makes the system backdrivable. The developed 6-DOF haptic device is connected to a personal computer using a dSpace 1103 board as shown in Figure 10. The proposed control structure is implemented in Simulink on that PC and the target controller code is executed on the dSpace board with 1kHz sampling rate. The haptic collision detection and force torque feedback program is implemented on the same computer. The position measurement resolution in each actuator leg is 0.01mm and the update rate of the controller is 1kHz.

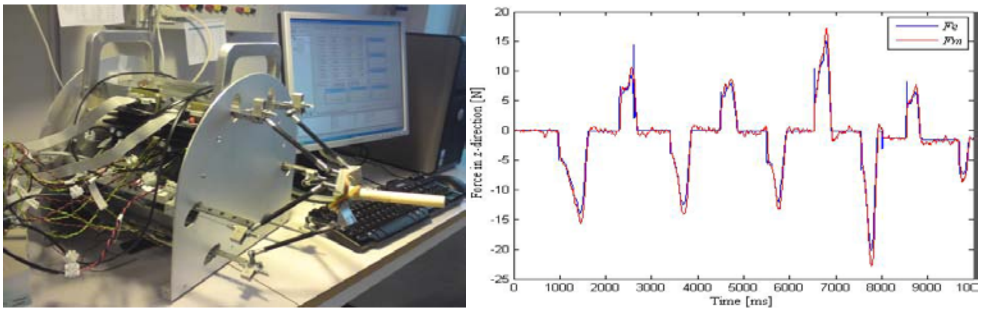


Figure 10. a) Prototype b) comparison of reference force and measured response from the device.

Figure 10b presents the response of the system (measured forces) and the reference forces from the virtual environment both in free space and while interaction with virtual objects. It has been observed that the optimal controller and optimal structure improves the performance of the 6-DOF device, as desired and thus its transparency, as shown in Figure 10b and Table 2.

3.4 Design validation

In the final stage, experiments for workspace, forces and torque capabilities, stiffness capability of the prototype were performed. The experimental results are shown in Table 2.

Table 2.

Characteristics	Values
No. of DOF	6
Dimension	250x250x300 mm
Workspace	Translation:75x75x100 mm Rotation: Pitch=Yaw=± 45° Roll=±40°
Maximum and continuous forces	52N and 20 N
Maximum and continuous torques	1.2Nm and 0.85 Nm
Stiffness	54 N/mm
Resolutions	Linear 0.01mm and Angular 0.01deg
Time step	1 ms

4 CONCLUDING REMARKS

The design process of the haptic devices particularly based on parallel mechanism, presents a complex design, due to multi-disciplinary mechatronic product design. It was concluded from this research work that following a systematic design methodology, one can develop an optimal haptic device, from the prospect of all involved engineering aspects. The proposed design methodology is based on parametric modeling, iterative and integrated design approach that leads to simple design space exploration of a pareto optimal design solution and initial verification in the conceptual phase of the product development. The methodology has been applied on a test case, i.e. the design of a parallel 6-DOF haptic device for a milling simulator for surgical training of vertebral operations. It has been concluded both from simulation and experimental results that the performance of the optimally designed device has been improved and satisfies the user requirements. This indicates that the methodology can support development of an optimal haptic device. However, more test cases are needed to verify this methodology.

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