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VIBRATION FATIGUE DESIGN METHODOLOGY OF A LARGE SCALE HEAVY DUTY ROBOT

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Abstract

The objective of the present study is to develop a design methodology for the large scale heavy duty robot to meet the design requirements of vibration and stress levels in structural components resulting from exposure of system modules to LCD (Liquid Crystal Display) processing environments. Vibrations of the component structures significantly influence on the motion accuracy and fatigue damage. To analyze and design a heavy duty robot for LCD transfer, FE and multi-body dynamic simulation techniques have been used. The links of a robot are modeled as flexible bodies using modal coordinates. Nonlinear mechanical properties such as friction, compliance of motors, gears and bearings were considered in the flexible multi-body dynamics model. Various design proposals are investigated to improve structural design performances using the dynamic simulation model. Design sensitivity analyses with respect to vibration and stresses are carried out to search an optimal design. An example of an 8th-Generation LTR (LCD Transfer Robot) is illustrated to demonstrate the proposed methodology. Finally, the results are verified by real experiments including strain measure test and vibration test.

1. INTRODUCTION

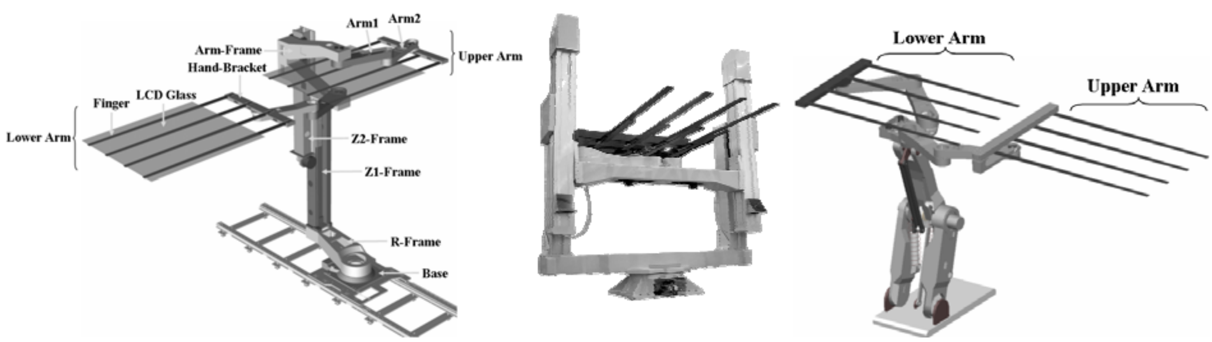
LCDs are widely used to TV's, computers, mobile phones, etc. because they offer some real advantages over other display technologies. They are thinner and lighter and draw much less power. Recently, the size of raw glass is greatly increased in a new generation LCD (Liquid Crystal Display) technology. In order to handle bigger and heavier glasses, it is necessary to develop a large scale LTR (LCD Transfer Robot) to support various complicated LCD fabrication processes. It will cause many difficult design problems such as vibration and high stresses due to heavier dynamic loads, resulting in inaccurate transfer motion and fatigue cracks. Therefore it is necessary to establish a methodology for predicting deflections, vibrations, and dynamic stress time histories using virtual computer simulation model. An integrated design simulation method would be useful to validate a baseline design and to propose new improved designs. In this paper an integrated computer simulation methodology is presented to predict deflections, dynamic stresses due to vibrations design, based on the existing FEM and flexible body dynamics technology.

The proposed methodology is applied to the LTR that handles 8th-Generation LCD glasses. Vibration analysis is performed and validated with the vibration modal test to identify and to recapture the inherent phenomenon in the system. Some flexible components in the LTR may experience severe vibration to cause fatigue damage due to large dynamic loads. Modal characteristics are used to consider structural flexibility in flexible multi-body dynamic simulations. Tip deflection of the end-effector can be calculated to see if design requirements are met. Dynamic loads and dynamic stress histories can be obtained from the dynamic simulation. Stress levels are investigated at the critical areas to predict if fatigue cracks might occur. If the stress level is not in a safe region, design change should be made based on the computer simulation results and design sensitivity study. Then a prototype LTR is built and tested for design validation. The present paper describes the CAE-based durability analysis that is being implemented and developed at SAMSUNG, to predict fatigue damages corresponding to durability tests. The proposed methodology can be used to develop a new large scale LTR in the early design stage.

2. INTRODUCTION OF LCD-TRANSFER ROBOT

Figure 1 shows various types of LTRs. Telescope type LTR consists of a base frame, a R-frame, two Z-frames, two articulated arms with slender hands as shown in Fig.1(a). The frame structures are fabricated with cast iron and aluminium. Hands with slender fingers are made of lightweight composite materials. It also has two arms (upper and lower arms) to handle 2 glasses simultaneously. The LTR has a cylindrical workspace to transfer glasses for various processes. For precision control of handling the glasses, static deformation at the tip of the finger must be less than 10mm. Since the joints which connect the arms and links include bearings and gears, joint compliance must be considered to predict the static deformation at the tip. Flexibilities of the arm itself are also important to both static and dynamic deformation, because the arm is a kind of cantilever type structure with a large lumped mass at the tip. Figure 1 shows various types of LTRs.

LTR is supposed to repeat millions of cycles to perform LCD fabrication processes in life. Therefore, it has to pass the physical tests to ensure the survivability of the robot system when subjected to static and cyclic loadings. The durability test is a cyclic loading apparatus that evaluates the durability characteristics of the component structure. Among the many different tests, one of the most critical is the hand motion of stretching out and pulling in with z-frame's vertical motion. The critical motion simulates the jerking and twisting impact that an arm support bracket might experience when running with large glasses loaded. The arms and hands are synchronized and moved at a speed of about 4m/s.



(a) Telescopic type (b) Gate type (c) Link type

Figure 1. LCD Transfer robots (LTR)

Since LTR repeats millions of cycles of particular loading and unloading with various configurations, it may result in fatigue failures at the critical stress area. In this paper, to predict static and dynamic deformation at the tip of the finger and critical stress levels including vibration of the LTR, flexible multi-body dynamic simulations are presented. Frames, links, arms are modelled as flexible bodies. Static and dynamic deformation is assumed to be very small, therefore, within the elastic range. To represent the flexibility, vibration normal modes and static correction modes are obtained from the finite elements vibration and static analysis for each flexible component. To represent the joint compliance, spring and damper force elements are used instead of kinematic joint elements.

3. FLEXIBLE MULTI-BODY DYNAMICS

The main advantage of using modal coordinates in flexible multi-body dynamics is the reduction in the number of generalized coordinates that must be included in the analysis. Two types of modes are used in component mode synthesis for flexible multi-body dynamics [1-2]. One is a normal mode. The other is a static mode. All used normal modes and static modes must be normalized to have the same magnitude and be orthogonalized to be independent to each other.

3.1 Kinematics of Flexible Components

A typical flexible component is shown in Fig.2.

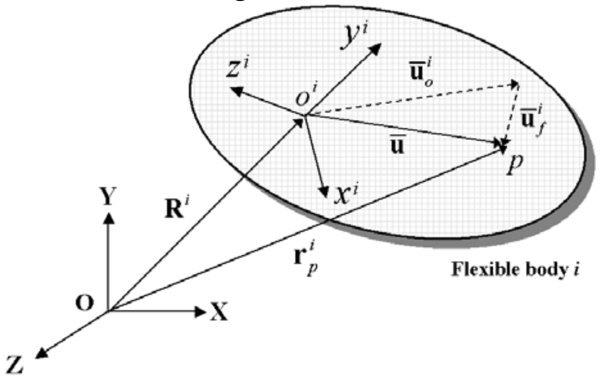


Figure 2. Global displacement of a point *p* in a flexible component *i*

The flexible component *i* is discretized into a large number of finite elements. The global position of a point *p* in a flexible part *i* can be represented as

$$\mathbf{r}_p^i = \mathbf{R}^i + \mathbf{A}^i \bar{\mathbf{u}}^i = \mathbf{R}^i + \mathbf{A}^i(\bar{\mathbf{u}}_o^i + \bar{\mathbf{u}}_f^i) \tag{1}$$

Where $\mathbf{R}^i$ is the global position vector of the X'-Y'-Z' body reference frame, $\mathbf{A}^i$ is the coordinate transformation matrix from the body reference frame to the global inertial frame, $\bar{\mathbf{u}}_o^i$ is the initial position vector of the point *p* from the body reference frame, and $\bar{\mathbf{u}}_f^i$ is the displacement vector due to deformation. The displacement vector $\bar{\mathbf{u}}_f^i$ can be approximated by a linear combination of deformation modes like Equation (2).

$$\bar{\mathbf{u}}_f^i = \mathbf{\Psi}^i \boldsymbol{\eta}^i = \sum_{j=1}^{M} \psi_j \eta_j \tag{2}$$

Where $\mathbf{\Psi}^i = \mathbf{\Psi}^i(x^i, y^i, z^i) = [\mathbf{\Psi}_t^i, \mathbf{\Psi}_r^i]$ is a modal matrix and ψ_j is the corresponding deformation mode of a flexible part i. $\boldsymbol{\eta}^i = \boldsymbol{\eta}^i(t)$ is a 6N×1 modal vector and η_j is modal coordinates, M is the number of modal coordinates. The deformation modes can be normal modes, static modes, or combination of normal and static modes. Used M modes should be linearly independent to each other.

3.2 Flexible Multi-body Dynamic Equations

As shown in Fig. 2, the nodal position vector of a typical point *p* in the global reference frame can thus be written as Equation (3) using Equation (2)

$$\begin{aligned} \mathbf{r}_p^i &= \mathbf{R}^i + \mathbf{A}^i \bar{\mathbf{u}}^i = \mathbf{R}^i + \mathbf{A}^i(\bar{\mathbf{u}}_o^i + \mathbf{\Psi}_t^i \boldsymbol{\eta}^i) \\ \boldsymbol{\pi}_p^i &= \boldsymbol{\pi}^i + \mathbf{\Psi}_r^i \boldsymbol{\eta}^i \end{aligned} \tag{3}$$

Where $\bar{\mathbf{u}}^i = \bar{\mathbf{u}}_o^i + \mathbf{\Psi}_t^i \boldsymbol{\eta}^i$ and the rotational displacement $\boldsymbol{\pi}_p^i$ of nodal point *p* is defined by $\mathbf{\Psi}_r^i \boldsymbol{\eta}^i$. The combined set of kinematic and driving constraints of the multi-body dynamic system may be written in the form [1-2]

$$\mathbf{\Phi}\,(\mathbf{q}, \mathrm{t}) = \mathbf{0} \tag{4}$$

Where the generalized coordinates $\mathbf{q} = [\,\mathbf{q}_r^T,\ \mathbf{q}_f^T\,]^T = [\,\mathbf{r}^T,\ \boldsymbol{\pi}^T,\ \boldsymbol{\eta}^T\,]^T$, $\mathbf{t}$ is the time, $\mathbf{\Phi}$ is the constraint equation. Using the Lagrange Multiplier Theorem [1], variational equations of motion of the multi-body system may be obtained by summing all bodies and constraints in the system as in the matrix form of Equation (5).

$$\begin{bmatrix} \mathbf{M}^* & \mathbf{\Phi}_{\mathbf{q}}^T \\ \mathbf{\Phi}_{\mathbf{q}} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}} \\ \boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{Q}^* \\ \boldsymbol{\gamma} \end{bmatrix} \tag{5}$$

This is a mixed system of differential-algebraic equations of motion for considering the elastic effect of the mechanical system. To solve mixed differential algebraic equations, many numerical algorithms have been developed [1]. Using the Equation (5), dynamic stress history of a flexible component can be calculated [3].

4. DYNAMIC MODELING OF AN LCD-TRANSFER ROBOT

4.1 Modeling of a Flexible Parts

The LTR system shown in Fig. 1(a) is modeled with 86 rigid bodies, 30 flexible bodies, kinematic joints, and force elements [1]. For parallel rectilinear motion of the finger and hand bracket, a timing belt at each joint is modeled to drive at constant speed ratio. As shown in the Fig.3, to represent the elasticity and damping of the belt, spring and damping forces are approximated to be proportional to displacement and velocity of the belt length change. Even the joint compliance is modeled in a similar way with spring and damper elements. Major components such as arms and links are made of cast iron or cast aluminium. Those structural components can be assumed to be linear elastic during normal operation. However, such a small elastic deformation may cause vibration and repeated dynamic stresses resulting in inaccurate transfer motion and fatigue cracks. Therefore it is necessary to establish a methodology for predicting the deformation, vibration, and dynamic stress time histories using virtual computer simulation model.

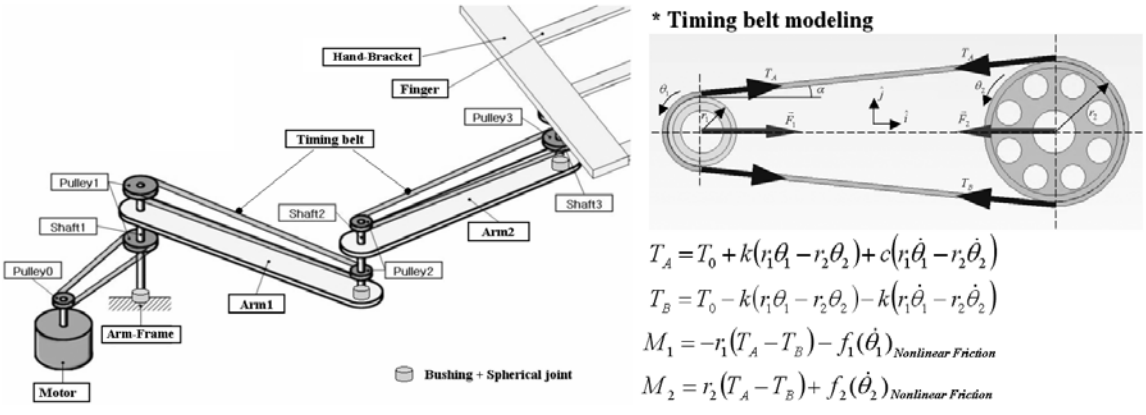
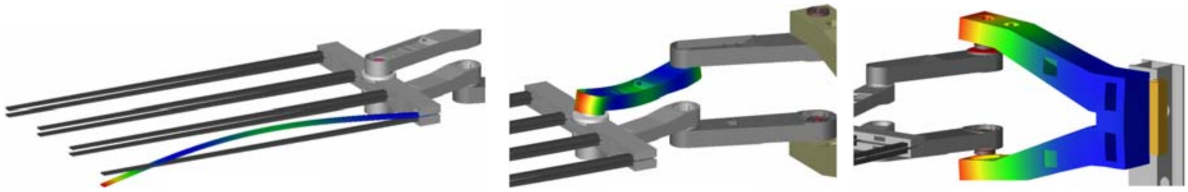


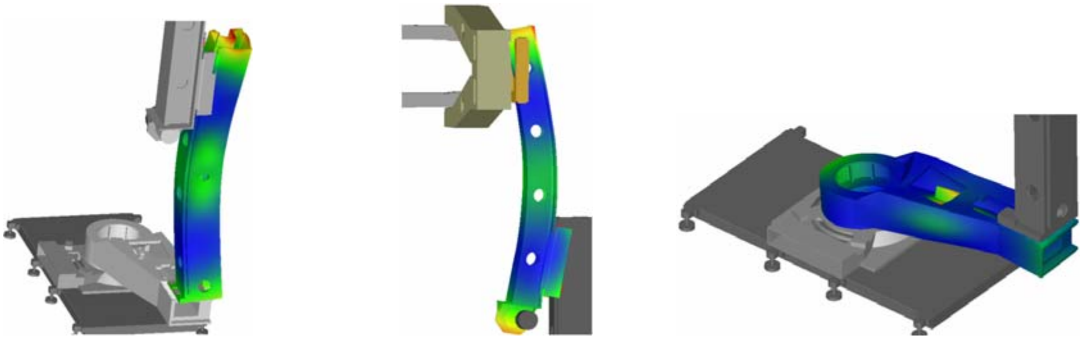
Figure 3. Dynamic modeling of LTR arm system

Component mode synthesis technique [1-3] explained in the previous section can be used for efficient computer simulation in large rigid body gross motion with small elastic deformation. Figure 4 shows the 1st vibration mode shapes of flexible components in the telescopic LTR in Fig .1(a). Also

Fig.4 shows the component mode numbers used in the flexible multi-body dynamic analysis, as explained in previous section.



(a) Finger (36) (b) Arm1 (18) (c) Arm-Frame (42)



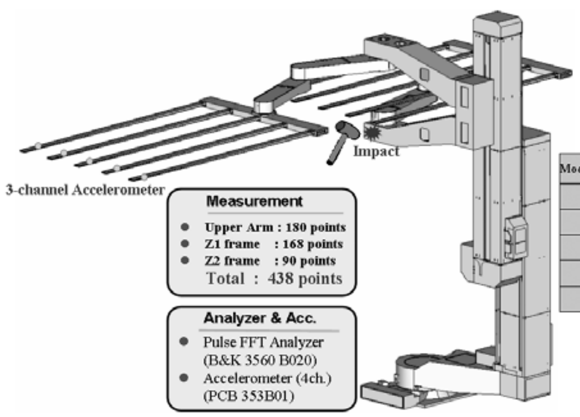
(d) Z1-Frame (24) (e) Z2-Frame (24) (f) R-Frame (36)

Figure 4. 1st Vibration modes of flexible components (Component modes used for dynamic simulation)

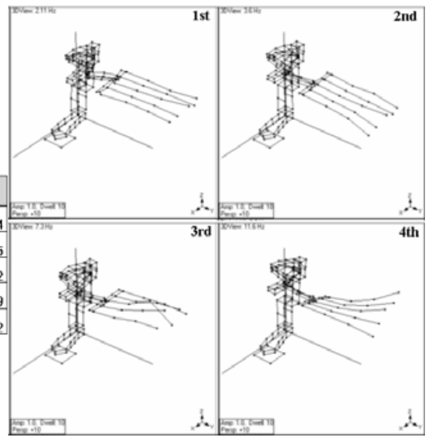
5. ANALYSIS AND EXPERIMENT FOR 8G-LTR DESIGN

5.1 System Vibration Modes

Since major structural components such as arms and links are modelled as flexible bodies with a few vibration normal modes with proper kinematic joints and force elements, fundamental vibration modes of the total LTR system can be investigated.



Mode No.	Experiment	Simulation
1	2.11	2.4
2	3.6	3.5
3	7.3	7.2
4	11.6	10.9
5	14.5	14.2



(a) Experiment set-up for modal analysis (b) Vibration mode shape from test results

Figure 5. Modal deformation of the telescopic LTR by modal test

Analytical vibration modes are compared with the experimental test results for validation. Figure 5 shows the test set up for modal test of the LTR system. Comparison with the modal test results showed that simulation results correlate well with the test results.

5.2 Design Analysis and Improvement for Vibration

Design problems such as tip deflection and fatigue crack can be investigated with the valid simulation model. Among the various process events for LCD glass transfer motion, stretching out and pulling in motions of the hands with glass loaded are the most critical motion to cause severe vibration and high stresses at the supporting bracket structure. Using the proposed flexible multi-body simulation technology, the critical motion is regenerated to investigate how large deflection and stresses occur during the operation. Since we have a valid simulation model, we can investigate various design proposals. Through design sensitivity studies the R-frame at the base of the LTR is known to be a critical component. To increase bending and twisting stiffness, height and width of the beam cross section is enlarged, and ribs are added as explained in Fig.6. Even aluminium material is replaced with high strength steel to increase the Young's modulus.

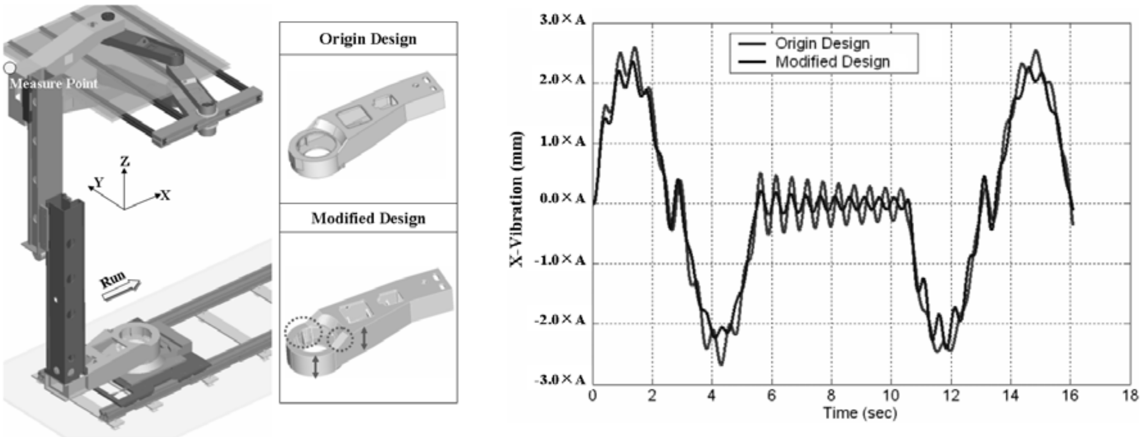


Figure 6. Design study to reduce the vibration by dynamics simulation

Figure 6 shows the comparison of vibration displacements during the simulated motions between the original baseline design and the new improved design. 50% reduction of the vibration level is observed even at the prototype test.

5.3 Stress Analysis and Design Improvement by Dynamic Simulation

As the size of raw glass tends to become larger for productivity and manufacturing cost competitiveness, LTRs need to be faster and bigger to handle the larger and heavier glasses with higher speed. It may result in increased dynamic loads causing fatigue cracks due to dynamic stresses. Figure 7 shows an example of the fatigue cracks due to dynamic loads at the supporting frame structure in the LTR fatigue test. Using the flexible multi-body dynamic simulation, cause and effect for the fatigue crack can be analyzed prior to adopt in an actual spot. To reduce the level of dynamic stress at the critical area, the shape and thickness of the structure must be redesigned based on the validated simulation model. Experimental tests are executed to validate the accuracy of dynamic stresses predicted in virtual computer simulations, as shown in Fig.8

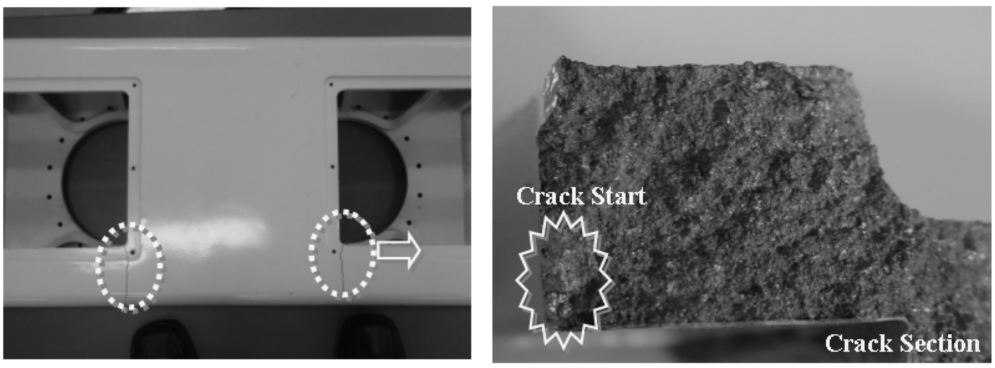


Figure 7. An example of the crack fatigue

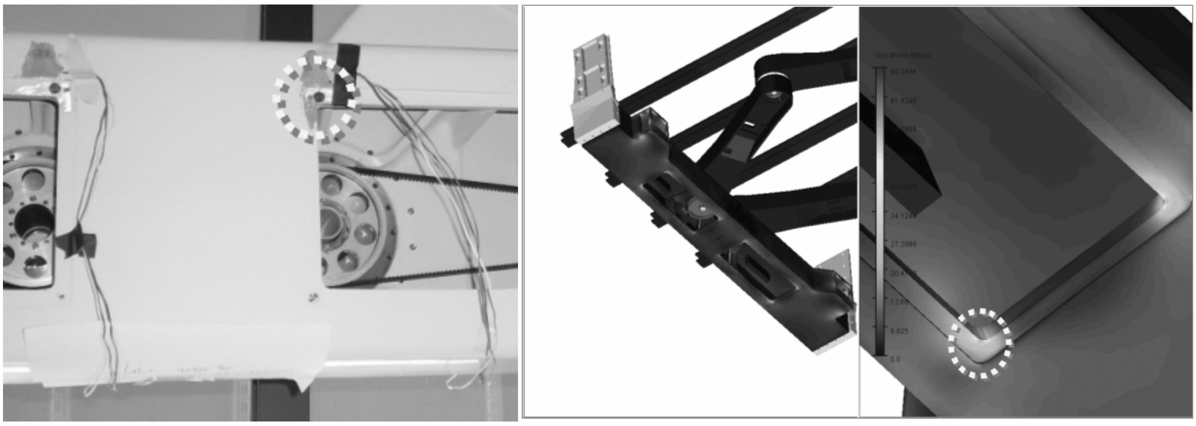


Figure 8. Stress experiment and dynamic analysis for fatigue design

Figure 9 shows the comparison of dynamic stresses between the original design and various design proposals with different design shape and metal thickness. The stress measure point of the part is dot circle area in the Fig.8.

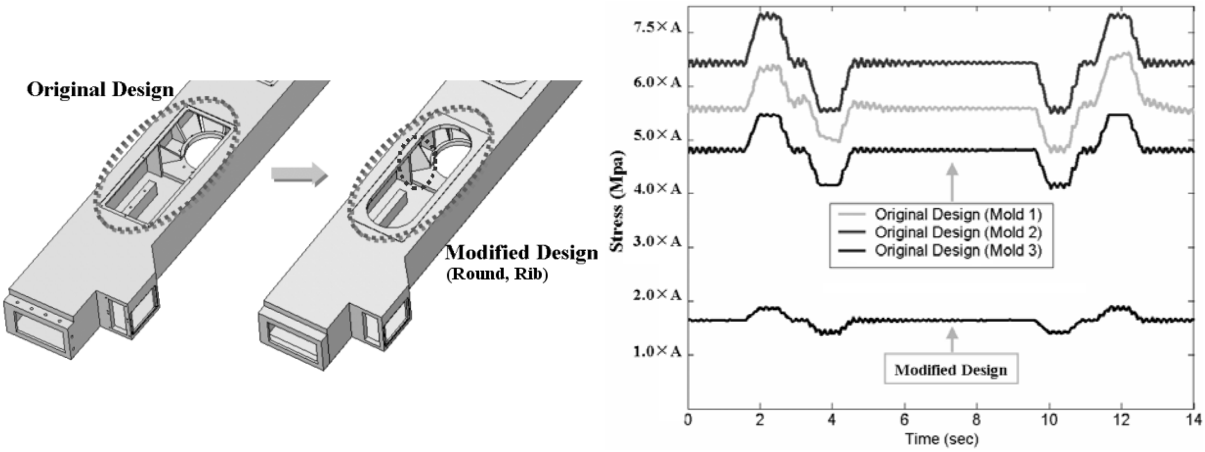


Figure 9. Stress analysis and design improvement for fatigue

5.4 Handling Accuracy and Design Optimization

If dynamic loads are increased, it might deteriorate the accuracy of the precision transfer motion due to deflection and deformation of major structural components [4]. The proposed simulation methodology can be used to evaluate the deflection at the tip of the finger. By investigating the simulation results for the baseline design, significant design parameters are identified and an optimal design can be found to improve the accuracy of the precision transfer motion. Figure 10 shows the vertical deflections at the tip of the hand finger for various designs during the specified LTR motion. Tip deflection for the baseline design of the LTR is 40 mm that exceed the design specification requirement 10 mm. Experimental tests using the laser tracker are carried out to validate the simulations. The dynamic simulation and DOE (Design of experiments) method were used to find an optimal design satisfying the design target of maximum deflection within 10 mm.

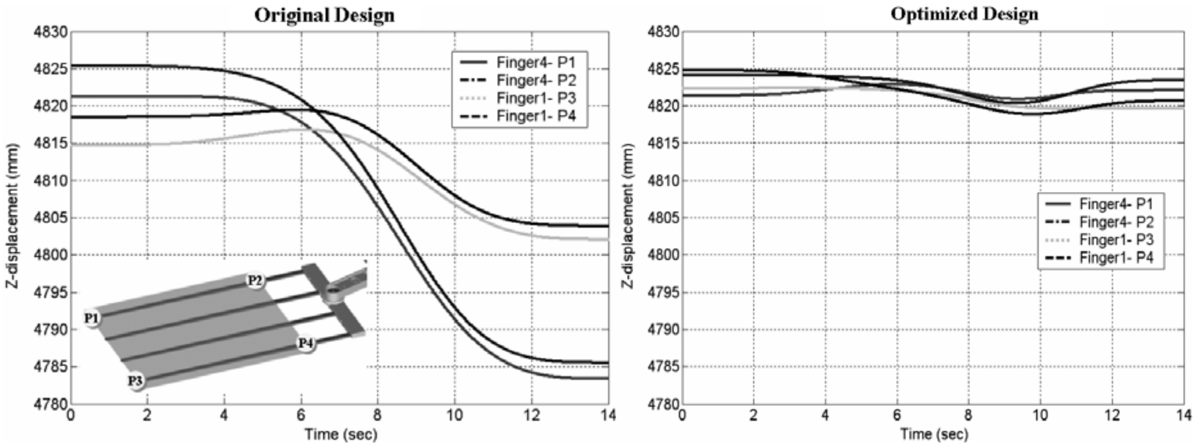


Figure 10. Optimized design for handling accuracy by simulation

6. CONCLUSIONS

A computer simulation methodology was presented for vibration and fatigue analysis of the LTR system. Variable amplitude multi-axial loading conditions can be generated to investigate any structural deflection, vibration, and fatigue. Flexible components are modelled using component mode synthesis technique. To represent the joint compliance and belt flexibility, spring and damper force elements are introduced with proper approximation. To have a valid simulation model, vibration modes of the total LTR system is compared with the modal test results. Comparison of the analysis and test results shows that they correlate well with each other. Deflection of the tip of the end-effector is investigated with the proposed methodology. To reduce the tip defection, a better design can be developed with the simulation model. Fatigue crack failure can be predicted with the baseline design. To prevent the fatigue failure at the critical area, stresses are reduced by changing the structural design. The results of the virtual durability assessment are quite good, and show good correlation with the areas of failure on the test. The value of being able to predict service lives based on results obtained exclusively in the virtual domain is obvious. The proposed methodology can be used to develop another type of LTR system.

ACKNOWLEDGMENT

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