Timoshenko versus Euler–Bernoulli beam theories for high speed two-link manipulator

H. Zohoor a,b,∗, F. Kakavand c

a Center of Excellence in Design, Robotics and Automation, Sharif University of Technology, Tehran, P.O. Box 11155-9567, Iran
b The Academy of Sciences of IR Iran, Tehran, P.O. Box 19735-167, Iran
c School of Mechanical Engineering, Sharif University of Technology, Tehran, Iran

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Abstract In this paper, a two-link flexible manipulator is considered. For a prescribed motion, Timoshenko and Euler–Bernoulli beam models are considered. Using the Galerkin method, nonlinear equations of motion are solved. The Runge–Kutta method is employed for the time response integration method. A comparative study is made between the Euler–Bernoulli and Timoshenko beam models, with and without foreshortening effects. It is demonstrated that for two-link manipulators, both theories provide good models, and the results for both theories are very similar for all ranges of slenderness ratio. The findings suggest that for two-link manipulators with relatively high slenderness ratios, there is a remarkable difference between the models, considering the foreshortening effect and un-stiffened models. It is obvious that for high precision applications, the stiffened Timoshenko model is recommended. It is interesting to note that joint torques for the entire range of slenderness ratios are the same.

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1. Introduction

Flexible manipulators are found in robotic systems design, flexible gyroscopes and, in general, in flexible multibody systems. It is, therefore, necessary to have a simple and accurate dynamic model in order to estimate the dynamic behavior of such systems.

Dynamic analysis of flexible multibody systems has gained the attention of researchers over the past decades. Earlier models of flexible multibody systems by finite element or assumed mode methods were based on the assumption that small deformations of the flexible bodies do not affect the rigid body motion significantly [1]. There are many publications listed in [2] that offer solutions for the rotating beam problem by employing several methods, such as the finite element method and the assumed mode. The acceleration and reaction forces were obtained from rigid body motion analysis, and were introduced to the linear elasticity problem as external forces for computing deflections. The elastic deformation is then superimposed on the rigid body motion. These dynamic models, however, do not yield accurate results, since they do not provide for the coupling of the rigid and elastic motion.

A hybrid-coordinate formulation, based on identifying the configuration of each flexible body by means of two coordinate systems, is developed in [3]; a reference coordinate system and an elastic coordinate system. Reference coordinates define the location and orientation of a body reference, while elastic coordinates describe the body deformation with respect to the body reference. Then, the rigid body motion and elastic deformations are solved simultaneously. Kane et al. [4] and Yoo et al. [5] had shown that the conventional hybrid-coordinate formulation, in which the Cartesian deformation variables are employed with a linear Cauchy strain measure, fails to capture the motion-induced stiffness terms and provides erroneous dynamic results in cases of high rotating speed (large overall motion) [1]. Yoo et al. [5] had shown, in detail, that using conventional axial deformation causes a diverging solution at high speed, because of the fact that in linearization of the potential strain energy, some terms of retaining force are lost.
A further improvement in the formulation can be achieved by employing non-Cartesian deformation variables to derive equations of motion for a thin beam or a thin plate [3,5–7]. With the inclusion of the foreshortening deformation, the motion-induced stiffness term is derived, which is the lost term in previous modeling methods. Using a stretch variable provides a simple expression of strain energy. Thus, in linearization of strain energy there is no lost term and the required retaining force is available. Therefore, at high speed, the model gives accurate and converging solutions. It has been proved that this method is as efficient as the conventional linear modeling method and as accurate as the nonlinear modeling methods [5].

Bayo [8] used FEM to deal with multi-link flexible manipulators, considering the Timoshenko beam theory and including nonlinear Coriolis and centrifugal effects for the elastic behavior. An iterative solution scheme is proposed for finding the desired joint torques, where the solution of each linearization is carried out in the frequency domain.

Liu and Hong [1,9] have developed a matrix presentation of spatial and planar Euler–Bernoulli beams based on the assumed mode method. They employed a non-Cartesian deformation variable for taking into account the motion-induced stiffness. They used a forward recursive formulation for driving the dynamic equations of a flexible link system.

In a certain applications, where the rotary inertia and the shear deformation effects are not significant, an analysis based on the Euler–Bernoulli beam theory would be sufficient. However, the error of using this theory can prove to be significant where thicker beams are concerned. The error may also be significant in the calculation of natural frequencies of vibration at higher modes and time responses. Rao and Gupta [10] used the Timoshenko model for a rotating beam. They solved a twisted tapered Timoshenko beam. Kyung-Su Na, Ji-Hwan Kim [11] had considered joint torques, where the solution of each linearization is carried out in the frequency domain.

For the kinetic energy of link $i$, the position vector is as follows. The position vector of point $k$ on the central line of beam $i$ can be defined, with respect to the $O_i - X_i Y_i Z_i$, as:

$$V_{k} = V_{O_i} + V_r + \dot{\omega} \times \vec{r}_{O_i k},$$

where $V_{O_i}$ and $\dot{\omega}$ are the absolute velocity of base point of link $i$, and the absolute angular velocity of beam $i$, in terms of $O_i - X_i Y_i Z_i$ unit vectors, respectively, which can be written as:

$$V_{O_i} = V_{O_i x} \hat{i} + V_{O_i y} \hat{j},$$

$$\dot{\omega} = \omega_0 \hat{k},$$

Substituting Eqs. (1) and (3)–(5) into Eq. (2), the velocity of point $k$ leads to:

$$\vec{V}_k = (V_{O_i x} + \dot{u}_i - v_i \omega_0) \hat{i} + (V_{O_i y} + \dot{v}_i + (x + u_i) \omega_0) \hat{j}.$$  

Therefore, the kinetic energy of beam $i$ can be written as:

$$T_i = \frac{1}{2} \rho_i \int_V \vec{V}_k \cdot \vec{V}_k dV,$$

$$T_i = \frac{1}{2} \rho_i A_i \int_0^l [(V_{O_i x} + \dot{u}_i - v_i \omega_0)^2 + (V_{O_i y} + \dot{v}_i + (x + u_i) \omega_0)^2] dx.$$  

In which, $A_i$ and $\rho_i$ are the cross-section area and material density of the beam respectively. Since the study of the beam dynamic in large overall motion is desired, using the stretch variable in the driving of the Potential Energy is necessary. As Yoo et al. described in [5], any linearization of potential energy will make the dynamic of the system in divergence cases at high speeds. Using the Von-Karman relation [5]:

$$u_i = s_i - \dot{h}_i,$$
where \( s_i \) is stretch variable and:

\[
 h_i = \frac{1}{2} \int_0^x \left( \frac{\partial v_i}{\partial \eta} \right)^2 d\eta, \tag{9}
\]

in which \( \eta \) is a dummy variable. Similarly, the time derivative of \( u \) is given by:

\[
 \dot{u}_i = \dot{s}_i - \dot{h}_i, \tag{10}
\]

where the superposed dots indicate the derivative with respect to time and:

\[
 \dot{h}_i = \int_0^x \frac{\partial v_i}{\partial \eta} \frac{\partial \dot{v}_i}{\partial \eta} d\eta. \tag{11}
\]

Using Eqs. (8)–(11), the kinetic energy Eq. (7) leads to:

\[
 T_i = \frac{1}{2} \rho A_i \int_0^l \left[ (V_{Ox} + \dot{s}_i - \dot{h}_i - v_i \omega) \right]^2 dx + \left( (V_{Oy} + v_i + (x + s_i - h_i) \omega) \right)^2 dx. \tag{12}
\]

Let us introduce the following relations as the velocity components of an arbitrary point of the beam:

\[
 V_u = V_{Ox} + \dot{s}_i - \dot{h}_i - v_i \omega, \tag{13}
\]

\[
 V_v = V_{Oy} + v_i + (x + s_i - h_i) \omega. \tag{14}
\]

For the Timoshenko model, the kinetic energy of a link is as follows [15]:

\[
 T_i = T_{ke} + \frac{1}{2} \int_0^l p_i \dot{\psi}_i^2 dx, \tag{15}
\]

where \( T_{ke} \) is the kinetic energy of the Euler–Bernoulli model (Eq. (12)). \( \psi \) is the rotation angle of each cross section and also \( l \) is the moment of inertia of the cross section.

In the case of a two-link manipulator, it is obvious that the base velocity of the outer link is the velocity of the end point of the inner link. Therefore, the base velocity components of the outer link in its body-fixed coordinate system are as follows:

\[
 V_{Ox} = V_{11x_{k-1}} \cos \lambda_2 + V_{11x_{k-1}} \sin \lambda_2, \tag{16a}
\]

\[
 V_{Oy} = -V_{11x_{k-1}} \sin \lambda_2 + V_{11x_{k-1}} \cos \lambda_2, \tag{16b}
\]

where \( \lambda_2 \) is angle between links. The \( \lambda_2 \) is as follows:

\[
 \lambda_2 = \theta_2 + \zeta, \tag{17a}
\]

where \( \zeta \) is the rotation angle of the cross section on the tip of link 1, and \( \theta_2 \) is the relative angle of link 2, with respect to link 1, in a rigid case. The \( \zeta \) for the Euler–Bernoulli beam model is as follows:

\[
 \zeta = \left. \frac{\partial v_i}{\partial x} \right|_{x_{k-1}}. \tag{17b}
\]

and for the Timoshenko model is:

\[
 \zeta = \left. \psi \right|_{x_{k-1}}. \tag{17c}
\]

The absolute angular velocity of the outer link can be written as:

\[
 \omega_2 = \omega_1 + \dot{\lambda}_2. \tag{17d}
\]

Strain energy for the Euler–Bernoulli model is as follows:

\[
 U_i = \frac{1}{2} E_i \int_0^l \left[ A_i \left( \frac{\partial s_i}{\partial x} \right)^2 + l_i \left( \frac{\partial^2 v_i}{\partial x^2} \right)^2 \right] dx, \tag{17e}
\]

and for the Timoshenko model is:

\[
 U_i = \frac{1}{2} E_i \int_0^l \left[ A_i \left( \frac{\partial s_i}{\partial x} \right)^2 + l_i \left( \frac{\partial \psi_i}{\partial x} \right)^2 \right] dx + \frac{1}{2} \mu_i A_i \int_0^l \left( \frac{\partial \psi_i}{\partial x} \right)^2 dx. \tag{17f}
\]

In Eq. (17), \( \mu_i \) and \( G_i \) are the shear factor and shear modulus of elasticity, respectively.

Using the Lagrange method and considering the sum of kinetic energies and strain energies for the manipulator, the equation of motion and the boundary value problem of the system can be obtained.

3. Discretization of equations of motion

The solution of the boundary value problem can be approximated by a finite set of ordinary differential equations by means of the Galerkin method. To do that, we introduce the Galerkin expansion:

\[
 s_i(x, t) = \sum_{k=1}^{n} q_i^k(t) \phi_i^k(x), \tag{18a}
\]

\[
 v_i(x, t) = \sum_{k=1}^{n} q_i^k(t) \phi_i^k(x), \tag{18b}
\]

\[
 \psi_i(x, t) = \sum_{k=1}^{n} q_i^k(t) \phi_i^k(x), \tag{18c}
\]

with the eigenfunctions of a linear stationary cantilever as comparison functions, where \( n \) is the number of modes that are chosen, and \( q_i^k \)'s, \( \phi_i^k \)'s and \( \psi_i^k \)'s are the \( k \)th modal coordinates of stretch, lateral deflection and shear of the \( i \)th link, respectively.

For the Euler–Bernoulli model, the following mode shapes for the cantilever beam can be considered:

\[
 \phi_i^k(x) = \sin \left( \frac{2k-1}{2} \frac{\pi x}{L_i} \right), \tag{19a}
\]

\[
 \psi_i^k(x) = \sin \beta_k x \sin \gamma_k x, \tag{19b}
\]

\[
 \phi_i^k(x) = \left[ \cos \beta_k x - \cosh \beta_k x \right] \sin \gamma_k x + \left[ \sin \beta_k x - \sinh \beta_k x \right] \cos \gamma_k x, \tag{19c}
\]

where \( L_i \) is the beam length and \( \beta_k \)'s are the roots of the Euler–Bernoulli frequency equation [16]:

\[
 1 + \cos \beta_k \cosh \beta_k = 0. \tag{19d}
\]

For the Timoshenko model, the following mode shape for the cantilever beam can be considered [16]:

\[
 \phi_i^k = L_i \left[ C_1 \sin \frac{ax}{L_i} + C_2 \cos \frac{ax}{L_i} \right. \\
 \left. + C_3 \sin \frac{bx}{L_i} + C_4 \cos \frac{bx}{L_i} \right], \tag{20a}
\]

\[
 \phi_i^k = D_1 \sin \frac{ax}{L_i} + D_2 \cos \frac{ax}{L_i} + D_3 \sin \frac{bx}{L_i} + D_4 \cos \frac{bx}{L_i}. \tag{20b}
\]

The wave numbers, \( a, b, \) and the constants, \( C_i, D_i, \) are dependent on each link property, therefore, they differ for each link (for details, see Appendix).
For the Euler–Bernoulli model and the Timoshenko model, the modal vector of a two-link manipulator can be defined as follows:

\[ Q_e = [q_1^1, q_1^2, q_1^3, q_1^4, q_2^1, q_2^2, q_2^3, q_2^4, q_3^1, q_3^2, q_3^3, q_3^4, q_4^1, q_4^2, q_4^3, q_4^4]. \]

\[ Q_T = [q_1^1, q_1^2, q_1^3, q_1^4, q_2^1, q_2^2, q_2^3, q_2^4, q_3^1, q_3^2, q_3^3, q_3^4, q_4^1, q_4^2, q_4^3, q_4^4]. \]

The Lagrangian of the system can be written as:

\[ \mathcal{L} = \sum_{i=1}^{2} T_i - \sum_{i=1}^{2} U_i. \quad (23) \]

Applying Relations (18)–(22) into (23) and using the Lagrange method as follows, the equations of motion will derive:

\[ \frac{d}{dt} \left( \frac{\partial \mathcal{L}}{\partial \dot{q}_i} \right) - \frac{\partial \mathcal{L}}{\partial q_i} = 0, \]

\[ \frac{d}{dt} \left( \frac{\partial \mathcal{L}}{\partial \dot{\theta}_i} \right) - \frac{\partial \mathcal{L}}{\partial \theta_i} = 0. \quad (24) \]

The equations of motion which are derived from Eqs. (24) are a set of ordinary differential equations. The Runge–Kutta method is employed for integration.

4. Numerical results

Now, consider a two-link manipulator which is shown in Figure 2.

The inner link (link 1) is deployed from 90 to 0° in \( T_i \) seconds. The angular velocity history of the inner link is given by:

\[ \dot{\theta}_1(t) = \begin{cases} -\omega_0 \frac{\theta_0 - \theta_i}{T_i} \left( 1 - \cos \frac{2\pi t}{T_i} \right) & t < T_i \\ \frac{\theta_0 - \omega_0}{T_i} \left( 1 - \cos \frac{2\pi t}{T_i} \right) & t \geq T_i \end{cases}, \]

where \( \theta_0 = \pi/2 \), and the spin-up motion of the outer link is given by:

\[ \dot{\theta}_2(t) = \begin{cases} \omega_0 \frac{T_i}{2} \left( 1 - \cos \frac{2\pi t}{T_i} \right) & t < T_i \\ \omega_0 \left( T - \frac{T_i}{2} \right) & t \geq T_i \end{cases}. \]

For a thin beam case, say \( \omega_0 = 1 \) rad/s, and the geometric property and material data of the inner link are: mass density \( \rho_1 = 2766.7 \) kg/m³, the modulus of elasticity \( E_2 = 68.952 \) Gpa, the area moment of inertia \( I_2 = 1.5 \times 10^{-3} \) m⁴, cross-section area \( A_2 = 7.3 \times 10^{-3} \) m², length \( L_2 = 8 \) m.

Figrure 3 and 4 show the tip deflection of the inner and outer, respectively, for a thin beam. It is obvious that the results of the Euler–Bernoulli beam are completely coincident to the results of [1].

The torques at the joints can be gained as follows:

\[ \frac{d}{dt} \left( \frac{\partial \mathcal{L}}{\partial \dot{\theta}_1} \right) - \frac{\partial \mathcal{L}}{\partial \theta_1} = T_1, \quad (25) \]

\[ \frac{d}{dt} \left( \frac{\partial \mathcal{L}}{\partial \dot{\theta}_2} \right) - \frac{\partial \mathcal{L}}{\partial \theta_2} = T_2, \quad (26) \]

where \( T_1 \) and \( T_2 \) are the applied joint torques at the inner and outer joints, respectively. Figures 5 and 6 show the applied torque at the joints. It is interesting that the results for the Timoshenko model and the Euler–Bernoulli model are completely coincident, but the results are different in cases of models in which the foreshortening effect is included, and models without a stiffening effect.

For a thick beam case, say \( \omega_0 = 100 \) rad/s, and the geometric property and material data of the inner link are: mass density \( \rho_1 = 2766.7 \) kg/m³, the modulus of elasticity \( E_2 = 68.952 \) Gpa, the area moment of inertia \( I_2 = 1.5 \times 10^{-2} \) m⁴, cross-section area \( A_2 = 3.84 \times 10^{-4} \) m², length \( L_2 = 0.25 \) m, whereas for the outer link, they are: mass density \( \rho_2 = 2766.7 \) kg/m³, the modulus of elasticity \( E_1 = 68.952 \) Gpa, area moment of inertia \( I_1 = 1.5 \times 10^{-2} \) m⁴, cross-section area \( A_1 = 3.84 \times 10^{-4} \) m², length \( L_1 = 8 \) m. For a thick beam case, they are: mass density \( \rho_2 = 2766.7 \) kg/m³, the modulus of elasticity \( E_1 = 68.952 \) Gpa, area moment of inertia \( I_1 = 1.5 \times 10^{-2} \) m⁴, cross-section area \( A_1 = 3.84 \times 10^{-4} \) m², length \( L_1 = 0.25 \) m, whereas for the outer link, they are: mass density \( \rho_2 = 2766.7 \) kg/m³, the modulus of elasticity \( E_2 = 68.952 \) Gpa, area moment of inertia \( I_2 = 1.5 \times 10^{-3} \) m⁴, cross-section area \( A_2 = 7.3 \times 10^{-3} \) m², length \( L_2 = 8 \) m.

Figures 5 and 6 show the torque at the inner and junction joints, respectively, for a manipulator with high slenderness ratios. It is evident that the results of the Euler–Bernoulli and Timoshenko models are completely coincident, except for the junction joint.

Figures 7 and 8 illustrate the tip deflection of the inner and outer links, respectively, for a thick beam. The results of the Euler–Bernoulli and Timoshenko beams are completely coincident, except for the deflection of the inner link.

Figures 9 and 10 display the applied torque at the joints. It is interesting that the results of the Timoshenko and Euler–Bernoulli models are completely coincident, and there is no difference in results in the stiffened model and the model without stiffening.

5. Conclusion

A two-link flexible manipulator is studied. For a prescribed motion, Timoshenko and Euler–Bernoulli models are considered. Using the Galerkin method, the nonlinear equations of motion are solved using three modes expansion. The Runge–Kutta method is employed as the time response integration method.
technique. A two-link flexible manipulator, which was studied by Liu and Hong [1], is considered, and the Timoshenko and Euler–Bernoulli beam models have been numerically examined in two cases, with and without foreshortening effects, and the results are compared with Ref. [1]. To capture the shear effect, the problem has been solved for various ranges of slenderness ratio, and the results for a thick beam are presented. In the above numerical studies, the time histories and joint torques are compared. It is demonstrated that for two-link manipulators, both theories provide good models and the results for both theories are very similar for the entire range of slenderness ratios. It is known that for the high slenderness ratios, both theories act similarly. Links with small slenderness ratios are necessary when high speeds and high precision are required. In these cases, the rigid body mode dominates the total response and the difference between the theories is still negligible. It is found that for two-link planar manipulators with relatively high slenderness ratios there is a remarkable difference in models, considering the foreshortening effect and un-stiffened models. It is obvious for high precision applications that the stiffened Timoshenko model is recommended, and for low precision applications in low and medium ranges of speed, the simpler Euler–Bernoulli model is suitable for control of elastic deformations. It is interesting that the joint torques in the entire range of slenderness ratios are the same.

Appendix

Han et al. [16] have derived, in detail, the mode shapes of a Timoshenko beam for various types of boundary condition. For a Timoshenko beam, the mode shapes are as follows:

\[ \psi^v = L \left( C_1 \sin \frac{ax}{L} + C_2 \cos \frac{ax}{L} + \frac{C_3}{L} \psi a x + C_4 \cosh \frac{bx}{L} \right), \quad (A.1) \]

\[ \psi^\alpha = D_1 \sin \frac{ax}{L} + D_2 \cos \frac{ax}{L} + D_3 \sin \frac{bx}{L} + D_4 \cosh \frac{bx}{L}, \quad (A.2) \]

where:

\[ a = \left[ \left( 1 + \frac{1}{\mu G} \right) \frac{\rho^* \omega^2}{2} + \left[ \left( 1 - \frac{1}{\mu G} \right) \frac{\rho^* \omega^4}{4} + \rho^* A \omega^2 \right] \right]^{\frac{1}{2}}, \]

\[ b = \left[ \left( 1 + \frac{1}{\mu G} \right) \frac{\rho^* \omega^2}{2} + \left[ \left( 1 - \frac{1}{\mu G} \right) \frac{\rho^* \omega^4}{4} + \rho^* A \omega^2 \right] \right]^{\frac{1}{2}}, \]

\[ D_1 = \alpha C_2, \]

\[ D_2 = -\alpha C_1, \]

\[ D_3 = \beta C_4, \]

\[ D_4 = \beta C_3. \]

\[ a = \frac{a^2 + \gamma^2 b^2}{a (1 + \gamma^2)}, \]

\[ \beta = \frac{b^2 + \gamma^2 a^2}{b (1 + \gamma^2)}, \]

\[ \gamma = \frac{2 (1 + \gamma)}{\mu}. \]

The variables with an asterisk are the non-dimensional variables. The non-dimensional cross-section area, shear modulus, density and moment of inertia are introduced below, respectively:

\[ A^* = \frac{A}{L^2}, \]

\[ G^* = \frac{GL^4}{EI}, \]

\[ \rho^* = \frac{\rho L^5}{EI}, \]

\[ I^* = \frac{I}{L^4}. \]

where \( A \) is the cross-sectional area, \( a \) and \( b \) are wave numbers, \( E \) is Young’s modulus, \( G \) is shear modulus, \( L \) is length of beam, \( \nu \) is Poisson ratio, \( I \) is moment of inertia and \( \mu \) is shear factor.

For a cantilever beam, boundary conditions are as follows:

\[ \nu(0) = 0, \]

\[ \psi(0) = 0, \]

\[ \frac{\partial \psi}{\partial x} (L) = 0, \]

\[ \frac{\partial \psi}{\partial x} (L) - \psi (L) = 0. \]

Applying boundary conditions on Eqs. (A.1) and (A.2) for a cantilever beam, the frequency equation can be obtained as [16]:

\[ (a^2 - b^2) \sin a b \cos a b \cos b \]

\[ + a^2 + a^2 \gamma^2 + 4 a^2 b^2 \gamma^2 + b^2 + b^2 \gamma^2 - 2 a b = 0. \]

For mode shapes taking \( C_1 = 1 \), therefore:

\[ C_3 = \frac{a}{\beta}, \]

\[ C_4 = \frac{2 a \sin a b + b b^2 - b}{2 a a \cos a b - \beta b b^2 - b b^2}, \]

\[ C_2 = -C_4, \]

\[ D_1 = \alpha C_2, \]

\[ D_2 = -\alpha, \]

\[ D_3 = \beta C_4, \]

\[ D_4 = \beta C_3. \]

Applying Eq. (A.4) into Eqs. (A.1) and (A.2) the mode shapes can be defined.

References


Hassan Zohoor was born in Esfahan, Iran, in 1945. He obtained his Ph.D. degree from Purdue University, USA, and is currently Professor of Mechanical Engineering at Sharif University of Technology, Tehran. He is also Fellow (Academician) and Secretary of the Academy of Sciences of IR Iran (IAS). He is author or co-author of over 350 scientific papers, two chapters of two books published by UNESCO, two chapters of two other books, and author of four technical pamphlets. He was also coordinator for compiling one e-book and four CDs for four courses. He has also supervised over 150 graduate theses. He has conducted more than twenty research funded projects, including an Iranian project in the area of energy, and holds one patent approved by the Office of Patent Management, Purdue Research Foundation, USA. He was Founder, President and Developer of the principal codes and regulations of the Payame Noor University in Iran. He was also Head of the Department of Engineering Sciences at IAS; Deputy for Infrastructure Affairs at the Budget and Planning Organization, Iran; Head of the Institute of Research and Planning in Higher Education, Iran; Academic Vice-Minister at the Ministry of Science and Higher Education, Iran; Acting President of Azhara University, Iran, and President of Shiraz University, Iran. He has received several honor plaques and awards, including the Top Student Award from Shiraz University; two Ross Ade Awards from Purdue University; an Award for Distinguished Professorship, Iran; the Lasting Personalities Award, Iran; an Honor Plaque for the most competent fellow from the IAS; Finalist in the Best Paper Award Competition from the American Society of Mechanical Engineers (ASME), USA; Honor Award from the International Council for Open and Distance Education (ICDE) Conference, for sustained contributions to Distance Education and Open Learning, India; A Golden Plaque from Payame Noor University for the best contribution to Open and Distance Education, Iran; an Honor Plaque for Distinguished Professorship from Sharif University of Technology (on the occasion of its 40th Anniversary); and an Honor Plaque for Distinguished Professorship in Mechanical Engineering from the Iranian Society of Mechanical Engineers (ISME).

Farshad Kakavand received a B.S. degree in Mechanical Engineering from Guilan University, Iran, in 1998, and an M.S. degree in Mechanical Engineering from Sharif University of Technology, Tehran, Iran, in 2000, where he is currently a Ph.D. degree student under the supervision of Dr. Zohoor. Since 2005, he has been Lecturer at the Islamic Azad University, Takestan Branch, Iran.