Dynamic Finite-Element Method of Thin-Walled Beams

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This paper presents the dynamic finite-element method for thin-walled beams with constrained torsional vibration. Based on the differential equation for torsional vibration, which includes the effect of cross-sectional warping, the shape functions are determined and, in turn, the frequency-dependent mass and stiffness matrices are derived. As an application of the dynamic finite-element method, a numerical example is presented. The results show that this method gives more accurate results in the high-frequency range than those obtained by the static finite-element method.

Nomenclature

$\bar{a}$ = shape function
E = modulus of elasticity
e = indicating element
G = modulus of rigidity (or shear modulus)
I_p = polar moment of inertia of cross section
J_k = torsional constant
$J\omega$ = warping constant
K = stiffness matrix
m = mass density
M = mass matrix
$T(t)$ = kinetic energy
u = eigenvector
$\bar{U}$ = displacement vector
$\dot{\bar{U}}$ = velocity vector
$V(t)$ = potential energy
θ = angle of twist of beam at any cross section
λ = eigenvalue
σ = Poisson ratio
ω = natural frequency
$(')$ = d/dx
$('')$ = d^2/dx^2
(n) = d^n/dx^n
$(\ddot{o})$ = d^2/dt^2

Introduction

THIN-WALLED beams are widely used in a broad range of structural applications such as automobile and aircraft frames. Studies involving static or dynamic calculations of thin-walled beams have been completed by many authors, including Gere,[1] Bathe and Chaudhary,[2] and George.[3] The common features of these studies are that the static displacement interpolation functions are used for the dynamic analysis and that the eigenvalues obtained using the corresponding mass and stiffness matrices are unreliable in the high-frequency range. In order to improve accuracy, we have to abandon the traditional static displacement interpolation approach. Based on the motion equation of an element, we construct displacement interpolating functions that are dependent on the unknown frequency. This method, called the dynamic finite-element method, yields upon further calculation the dynamic mass and stiffness matrices. In order to avoid complicated calculations, we assume that the displacement can be expressed as an ascending power series of the unknown frequency.[4] This approach has been used in the derivation of dynamic mass and stiffness matrices for bars, beams, and plates in the past. However, the present work is the first to present dynamic mass and stiffness matrices for thin-walled beams with constrained torsional vibration. The validity of the present approach has been illustrated by way of an example problem. The results show that the dynamic finite-element method is highly accurate.

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Differential Equation of Thin-Walled Beams with Constrained Torsional Vibration

The differential equation for free torsional vibrations of thin-walled open or closed cross-section beams is

$$\theta^{(4)} - K^2\theta^{(2)} + P^2\ddot{\theta} = 0 \tag{1}$$

where

$$K^2 = \frac{GJ_k}{E_1 J\omega}, \qquad P^2 = \frac{\mu m I p}{E_1 J\omega}, \qquad E_1 = \frac{E}{1-\sigma^2}$$

$\mu = 1$ for open cross section

$\mu = 1 - J_k/I_p$ for closed cross section

Assume that

$$\theta = \bar{a}\bar{U} \tag{2}$$

where

$$\bar{a} = \bar{a}_0 + \bar{a}_1\omega + \bar{a}_2\omega^2 + \ . \ . \ . \tag{3}$$

$$\bar{U} = \begin{Bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \end{Bmatrix} = \begin{Bmatrix} q_1 \\ q_1 \\ q_3 \\ q_4 \end{Bmatrix} e^{i\omega t} \tag{4}$$

Also, u_i, $i = 1, 2, 3, 4$ are the end displacements of the beam shown in Fig. 1. Substituting Eqs. (3) and (4) into Eq. (2), one gets

$$\theta = \sum_{j=0}^{\infty} \bar{a}_j \omega^j \bar{q} e^{i\omega t} \tag{5}$$

Note that for simplicity we define $\bar{a} = [a_1, a_2, a_3, a_4]$. Substituting Eq. (5) into Eq. (1), we obtain

$$\sum_{j=0}^{\infty} \bar{a}_j{}^{(4)}\omega^j \bar{q} e^{i\omega t} - K^2 \sum_{j=0}^{\infty} \bar{a}^{(2)}\omega^j \bar{q} e^{i\omega t} - p^2 \sum_{j=0}^{\infty} \bar{a}_j \omega^{j+2} q e^{i\omega t} = 0 \tag{6}$$

The coefficients of the same powers of ω in Eq. (6) must be zero; therefore, we obtain

$$\bar{a}_0^{(4)} - K^2\bar{a}_0^{(2)} = 0 \tag{7}$$

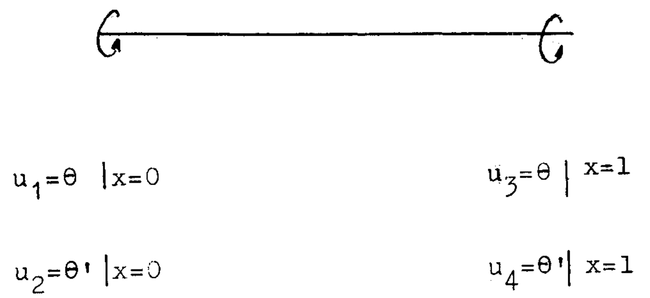

Fig. 1 The end displacements of a beam with constrained torsional vibrations.

$$\bar{a}_1^{(4)} - K^2\bar{a}_1^{(2)} = 0 \tag{8a}$$

$$\bar{a}_2^{(4)} - K^2\bar{a}_2^{(2)} = p^2\bar{a}_0 \tag{8b}$$

Equations (7) and (8) can be integrated directly. The first vector $\bar{a}_0$ is then used to satisfy the boundary conditions; i.e., $\theta = u_1$ and $\theta' = u_2$ at $x = 0$, $\theta = u_3$ and $\theta' = u_4$ at $x = 1$, while the remaining vectors $\bar{a}_1$, $\bar{a}_2$, $\bar{a}_3$, . . . must all vanish at $x = 0$ and $x = 1$. The general solution of Eq. (7) is as follows:

$$\bar{a}_0(x) = \bar{F}_1 + \bar{F}_2x + \bar{F}_3 Shkx + \bar{F}_4 Chkx \tag{9}$$

Requiring that Eq. (9) satisfy the boundary conditions yields

$$a_{01}(0) = 1,\ a_{01}(1) = 0,\ \frac{da_{01}}{dx}\Big|_{x=0} = 0,\ \frac{da_{01}}{dx}\Big|_{x=1} = 0 \tag{10a}$$

$$a_{02}(0) = 0,\ a_{02}(1) = 0,\ \frac{da_{02}}{dx}\Big|_{x=0} = 1,\ \frac{da_{02}}{dx}\Big|_{x=1} = 0 \tag{10b}$$

$$a_{03}(0) = 0,\ a_{03}(1) = 1,\ \frac{da_{03}}{dx}\Big|_{x=0} = 0,\ \frac{da_{03}}{dx}\Big|_{x=1} = 0 \tag{10c}$$

$$a_{04}(0) = 0,\ a_{04}(1) = 0,\ \frac{da_{04}}{dx}\Big|_{x=0} = 0,\ \frac{da_{04}}{dx}\Big|_{x=1} = 1 \tag{10d}$$

Substituting Eq. (9) into Eqs. (10), the coefficients in Eq. (9) can be found as follows:

$$F_1(1) = A/S,\ F_2(1) = -KShkl/s,\ F_3(1) = Shkl,$$

$$F_4(1) = B/S \tag{11a}$$

$$F_1(2) = D/K/S,\ F_2(2) = B/S,\ F_3(2) = A/K/S,$$

$$F_4(2) = -D/K/S \tag{11b}$$

$$F_1(3) = B/S,\ F_2(3) = KShkl/S,\ F_3(3) = -Shkl/S,$$

$$F_4(3) = -B/S \tag{11c}$$

$$F_1(4) = H/K/S,\ F_2(4) = B/S,\ F_3(4) = -B/K/S,$$

$$F_4(4) = -H/K/S \tag{11d}$$

where

$$S = KlShkl - 2Chkl + 2$$

$$A = KlShkl - Chkl + 1$$

$$B = 1 - Chkl$$

$$D = klChkl - Shkl$$

$$H = Shkl - kl$$

As $\bar{a}_1$ must vanish at $x = 0$ and $x = 1$, it follows from Eqs. (8) that

$$\bar{a}_1 \equiv 0 \tag{12}$$

In order to avoid tedious algebraic manipulation, the following procedure is adopted to solve Eq. (8b). From Eq. (9), the solution can be expressed as

$$a_{0i} = F_1(i) + F_2(i)x + F_3(i)Shkx + F_4(i)Chkx \tag{13}$$

where $i = 1,2,3,4$. Substituting

$$Shkx = (e^{kx} - e^{-kx})/2,\ Chkx = (e^{kx} + e^{-kx})/2$$

into Eq. (13) gives

$$a_{0i} = F_1(i) + F_2(i)x + [F_3(i) + F_4(i)]e^{kx}/2$$

$$+ [F_4(i) - F_3(i)]e^{-kx}/2 \tag{14}$$

Finally, substituting Eq. (14) into Eq. (8b) and integrating Eq. (8b) leads to

$$a_{2i} = F_5(i) + F_6(i)x + F_7(i)x^2 + F_8(i)x^3 + F_9(i)e^{kx}$$

$$+ F_{10}(i)e^{-kx} + F_{11}(i)xe^{kx} + F_{12}(i)xe^{-kx} \tag{15}$$

where

$$F_5(i) = p^2F_4(i)/K^4 + C_1(i) \tag{16a}$$

$$F_6(i) = p^2F_3(i)/K^3/2 + C_2(i) \tag{16b}$$

$$F_7(i) = -p^2F_1(i)/K^2/2 \tag{16c}$$

$$F_8(i) = -p^2F_2(i)/K^2/6 \tag{16d}$$

$$F_9(i) = -p^2[F_3(i) + F_4(i)]/K^4/2 + C_3(i) \tag{16e}$$

$$F_{10}(i) = -p^2[F_4(i) - F_3(i)]/K^4/2 + C_4(i) \tag{16f}$$

$$F_{11}(i) = p^2[F_4(i) - F_3(i)]/K^3/2 \tag{16g}$$

$$F_{12}(i) = -p^2[F_4(i) - F_3(i)]/K^3/4 \tag{16h}$$

and where

$$C_3(i) = p^2(2 - kl)[F_3(i) + F_4(i)]/K^4/4 \tag{17a}$$

$$C_4(i) = p^2(2 + kl)[F_4(i) - F_3(i)]/K^4/4 \tag{17b}$$

$$C_1(i) = -C_3(i) - C_4(i) \tag{17c}$$

$$C_2(i) = KC_4(i) - kC_3(i) \tag{17d}$$

Using Eqs. (11–17), $\bar{a}_0$ and $\bar{a}_2$ can be easily calculated on a digital computer.

Dynamic Mass and Stiffness Matrices

This section presents the calculation of the element mass and stiffness matrices. From the theory of constrained torsional vibration, the element kinetic energy is

$$T(t)^e = ½\int_0^1 mI_p\dot{\theta}^2 dx \tag{18}$$

Substituting Eq. (2) into Eq. (18), we obtain

$$T(t)^e = ½\int_0^1 mI_p\dot{U}^T\bar{a}^T\bar{a}^T\bar{a}\dot{U}\,dx \tag{19}$$

which in turn can be written as

$$T(t)^e = \tfrac{1}{2}\ \dot{U}^T M^e \dot{U} \tag{20}$$

Comparing Eq. (19) with Eq. (20), we note that the element mass matrix is

$$M^e = \int_0^1 m I_p \bar{a}^T \bar{a}\ \mathrm{d}x \tag{21}$$

The calculation proceeds by substituting Eq. (3) into Eq. (21) to yield

$$M^e = \int_0^1 m I_p (\bar{a}_0 + \bar{a}_2\omega^2 + \ . \ . \ .)^T (\bar{a}_0 + \bar{a}_2\omega^2 + \ . \ . \ .)\ \mathrm{d}x$$

$$= M_0^e + M_2^e\omega^2 + \ . \ . \ . \tag{22}$$

where

$$M_0^e = \int_0^1 m I_p \bar{a}_0^T \bar{a}_0\ \mathrm{d}x \tag{23}$$

$$M_2^e = \int_0^1 m I_p (\bar{a}_0^T \bar{a}_2 + \bar{a}_2^T \bar{a}_0)\ \mathrm{d}x \tag{24}$$

From the theory of constrained torsional vibration, the element potential energy is

$$V(t)^e = E_1 J\omega \int_0^1 (\theta'')^2\ \mathrm{d}x/2 + GJ_k \int_0^1 (\theta')^2\ \mathrm{d}x/2 \tag{25}$$

Substituting Eq. (2) into Eq. (25), we obtain

$$V(t)^e = E_1 J\omega \int_0^1 \bar{U}^T (\bar{a}''^T \bar{a}'') \bar{U}\ \mathrm{d}x/2$$

$$+ GJ_k \int_0^1 \bar{U}^T (\bar{a}'^T \bar{a}') \bar{U}\ \mathrm{d}x/2 \tag{26}$$

which may be written as

$$V(t)^e = \tfrac{1}{2} \bar{U}^T K^e \bar{U} \tag{27}$$

Again, comparing Eq. (26) with Eq. (27), we obtain the element stiffness matrix as

$$K^e = \int_0^1 E_1 J_\omega \bar{a}''^T \bar{a}''\ \mathrm{d}x + \int_0^1 GJ_K \bar{a}'^T \bar{a}'\ \mathrm{d}x \tag{28}$$

Furthermore, substituting Eq. (3) into Eq. (28), we get

$$K^e = \int_0^1 E_1 J_\omega (\bar{a}_0'' + \omega^2 \bar{a}_0'' + \ . \ . \ .)^T (\bar{a}_0'' + \omega^2 \bar{a}_2'' + \ . \ . \ .)\mathrm{d}x$$

$$+ \int_0^1 GJ_k (\bar{a}_0' + \omega^2 \bar{a}_2' + . \ . \ .)^T (\bar{a}_0' + \omega^2 \bar{a}_2' + . \ . \ .)\mathrm{d}x$$

$$= K_0^e + \omega^4 K_4^e + . \ . \ . \tag{29}$$

where

$$K_0^e = \int_0^1 E_1 J\omega \bar{a}_0''^T \bar{a}_0''\ \mathrm{d}x + \int_0^1 GJ_k \bar{a}_0'^T \bar{a}_0'\ \mathrm{d}x \tag{30}$$

$$K_4^e = \int_0^1 E_1 J\omega \bar{a}_2''^T \bar{a}_2''\ \mathrm{d}x + \int_0^1 GJ_k \bar{a}_2'^T \bar{a}_2'\ \mathrm{d}x \tag{31}$$

The following formulas are used to calculate M_0, M_2, K_0, and K_4:

$$M_0^e(i,j) = \int_0^1 m I_p a_{0i} a_{0j}\ \mathrm{d}x \tag{32}$$

$$M_2^e(i,j) = \int_0^1 m I_p (a_{0i} a_{2j} + a_{2i} a_{0j})\ \mathrm{d}x \tag{33}$$

$$K_0^e(i,j) = \int_0^1 E_1 J_\omega a_{0i}'' a_{0j}''\ \mathrm{d}x + \int_0^1 GJ_k a_{0i}' a_{0j}'\ \mathrm{d}x \tag{34}$$

$$K_4^e(i,j) = \int_0^1 E_1 J_\omega a_{2i}'' a_{2j}''\ \mathrm{d}x + \int_0^1 GJ_k a_{2i}' a_{2j}'\ \mathrm{d}x \tag{35}$$

where $i, j = 1, 2, 3, 4$. Matrices M_0^e and K_0^e represent the static inertia and stiffness of the thin-walled beam element with contrained torsional vibration, while M_2^e and K_4^e and higher-order terms represent dynamic corrections.

Iterative Perturbation for Solving the Nonlinear Eigenproblem

The eigenproblem for thin-walled beams with constrained torsional vibration is as follows:

$$(K_0 + \lambda^2 K_4 + . \ . \ .)\ u = \lambda(M_0 + \lambda M_2 + . \ . \ .)\ u \tag{36}$$

where $\lambda = \omega^2$. Neglecting the higher-order terms, we arrive at

$$(K_0 + \lambda^2 K_4)\ u = \lambda(M_0 + \lambda M_2)\ u \tag{37}$$

Equation (37) can be rewritten as

$$K_0 u = \lambda[M_0 + \lambda(M_2 - K_4)]\ u \tag{38a}$$

or

$$K_0 u = \lambda(M_0 + \Delta M)u, \qquad \Delta M = \lambda(M_2 - K_4) \tag{38b}$$

Equations (38) is a nonlinear eigenproblem. The method for solving Eqs. (38) has been presented by Wittrick and Williams,[6] based on the Strum sequence search in every neighborhood of interested frequencies. However, it is difficult to search a cluster of natural frequencies. In order to avoid the difficulties of clustered frequencies, an iterative perturbation

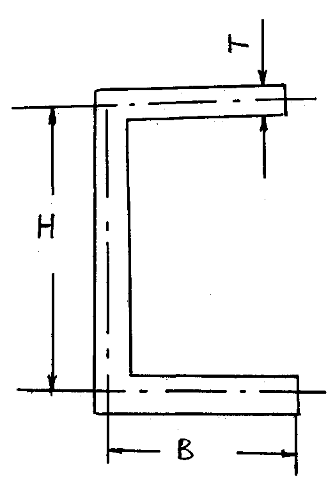

Fig. 2 The cross section of a channel beam.

Table 1 Comparison of eigenvalues of a simply supported beam with constrained torisional vibration

Modal number	Exact solution	10 elements SFEM error, %	10 elements DFEM error, %	6 elements SFEM error, %	6 elements DFEM error, %
1	0.137116E+6	0.137270E+6 (1.1169E−1)	0.137116E+6 (0)	0.138005E+6 (0.648356)	0.137118E+6 (0.14586E−2)
2	0.564519E+6	0.567104E+6 (4.5786E−1)	0.564524E+6 (9.317E−4)	0.579400E+6 (2.636)	0.564607E+6 (0.01588)
3	0.133037E+7	0.134453E+7 (1.0678)	0.133044E+7 (4.735E−3)	0.141089E+7 (6.0592)	0.1331728E+7 (0.1020766)
4	0.251495E+7	0.256495E+7 (1.988)	0.251537E+7 (1.66126E−2)	0.278887E+7 (10.8917)	0.252837E+7 (0.5336)
5	0.423064E+7	0.43693E+7 (3.2774)	0.423267E+7 (4.779E−2)	0.490321E+7 (15.8976)	0.433979E+7 (2.57998)

scheme[5] for solving the nonlinear eigenproblems, Eqs. (38), can be used. If ΔM is dropped, Eqs. (38) become

$$K_0 u = \lambda M_0 \tag{39}$$

This is the same eigenproblem obtained in the static finite-element method. Equations (38) can be referred to as the perturbation equation of Eq. (39). Then the perturbed eigensolutions are

$$\lambda_i' = \lambda_i + \Delta\lambda_i \tag{40}$$

$$u_i' = u_i + \Delta u_i \tag{41}$$

where λ_i and u_i are the eigensolutions of Eq. (39). The quantities $\Delta\lambda_i$ and Δu_i can be easily computed by the matrix perturbation,

$$\Delta\lambda_i = -\lambda_i u_i^T \Delta M u_i \tag{42}$$

$$\Delta u_i = \sum_{s=1}^{n} C_{is} u_s \tag{43}$$

In practical calculations, the iterative perturbation starts with Eq. (39) in conjunction with the assumption that

$$\omega_{1i}^{(1)} = 0.0,\ \omega_{2i}^{(1)} = \lambda_i^{(1)} = \lambda_i,\ u_i^{(1)} = u_i \tag{44}$$

For $p = 1, 2, \ldots$, it follows that

$$\Delta_M^{(p)} = \omega_{2i}^{(p)}(M_2 - K_4) - \omega_{1i}^{(p)}(M_2 - K_4) \tag{45}$$

$$\Delta\lambda_i^{(p)} = -\lambda_i^{(p)}(u_i^{(p)})^T \Delta M^{(p)} u_i^{(p)} \tag{46}$$

$$C_i^{(p)} = -\frac{\lambda_i^{(p)}}{\lambda_i^{(p)} - \lambda_s^{(p)}}(u_s^{(p)})^T \Delta M^{(p)} u_i^{(p)} \tag{47a}$$

where $i \neq s$, $i, s = 1,2\ldots$, or

$$C_{is}^{(p)} = \tfrac{1}{2}(u_i^{(p)})^T \Delta M^{(p)} u_i^{(p)} \tag{47b}$$

where $i = s$, and

$$\lambda_i^{(p+1)} = \lambda_i^{(p)} + \Delta\lambda_i^{(p)} \tag{48}$$

$$\Delta u_i^{(p)} = \sum_{s=1}^{n} C_{is}^{(p)} u_i^{(p)} \tag{49}$$

$$u_i^{(p+1)} = u_i^{(p)} + \Delta u_i^{(p)} \tag{50}$$

$$\omega_{1i}^{(p+1)} = \omega_{2i}^{(p)} \tag{51}$$

$$\omega_{2i}^{(p+1)} = \lambda_i^{(p+1)} \tag{52}$$

$$\frac{|\Delta\lambda_i^{(p)}|}{\lambda_i^{(p+1)}} \le \epsilon \tag{53}$$

where ϵ is the convergence tolerance desired. Usually the convergence can be achieved after 4–5 iterations.

Numerical Example

Using the given formulas and the iterative perturbation, numerical computations have been completed for the simply supported channel beam shown in Fig. 2. The physical parameters of this beam are: length $L = 300$ cm, $H = 4.5$ cm, $B = 2.75$ cm, and $T = 0.5$ cm. The eigenvalues were calculated and are listed in Table 1, in which the solutions of the static finite-element method were found from Eq. (39), the solutions of the dynamic finite-element method were obtained from Eqs. (38), and the exact solutions are from Ref. 1. From Table 1, it can be seen that the dynamic finite-element method (DFEM) gives excellent results and that the accuracy of the static finite-element method (SFEM) is limited in the high-frequency range. For example, after four iterations, the frequency error is reduced to 0.53% from the original 10.891% in the fourth mode of the six-element model. For the DFEM, the error in the fourth mode frequency is 0.5336% for the six-element model, which is much smaller than the 1.988% error of the SFEM for the 10-element model. This indicates that the DFEM has greater accuracy. For this specific example, the accuracy of solutions obtained by the DFEM for the six-element model is equivalent to that obtained by the SFEM for the 10-element model. For this specific example, the CPU time required to compute the first five frequencies and mode shapes of the six-element model was 2.4 s using the SFEM. Taking the solution of SFEM as the starting value, it took another 2.9 s of iterative perturbation to arrive at the solution for the DFEM. On the other hand, the CPU time is 8.5 s to calculate the first five frequencies and mode shapes for the 10-element model using the SFEM. Therefore, similar accuracy is obtained using the DFEM as compared to the SFEM with a concurrent savings of about 35% in CPU time. Experience shows that the more degrees of freedom in a structure, the greater will be the saving in CPU time.

Conclusions

The dynamic finite-element method for a thin-walled beam with constrained vibration has been presented and illustrated by a numerical example of a simply supported channel beam. Based on the numerical results, it is possible to draw the following conclusions. First, because of the omission of the inertial effects, the eigenvalues obtained by the static finite-element method become highly unreliable in the high-frequency range. Second, in the high-frequency range, the dynamic finite-element method gives more accurate results. Finally, the iterative perturbation scheme proves to be effective and economical.

References

[1]Gere, J. M., "Torsional Vibration of Beams of Thin-Walled Open Section," *Journal of Applied Mechanics*, Dec. 1954, pp. 334-338.

[2]Bathe, K. J., and Chaudhary, A., "On the Displacement Formulation of Torsion of Shafts with Rectangular Cross-sections," *International Journal of Numerical Methods,* Vol. 18, No. 10, 1982, pp. 1565-1573.

[3]George, R. J., "Determination of Natural Frequencies and Mode Shapes of Chasis Frames," SAE Paper 730504.

[4]Przemieniecki, J. S., *Theory of Matrix Structural Analysis,* McGraw-Hill, New York, 1968.

[5]Zhang, Z., and Chen, S. "Iterative Perturbation for Solving Nonlinear Eigenvalue Problem of Dynamic Reduction," *Proceedings of the International Conference on Mechanical Dynamics*, 1987, pp. 682-687.

[6]Wittrick, W. H., and Williams, F. W., "A General Algorithm for Computing Natural Frequencies of Elastic Structures," *Quarterly Journal of Mechanical Applications in Mathematics,* Vol. 24, 1971, pp. 263-284.

Candidates Solicited for *JA* Editor-in-Chief Post

On January 1, 1991, AIAA will appoint an Editor-in-Chief of its *Journal of Aircraft (JA)* for a three-year term and solicits candidates for this prestigious editorial post.

Several goals have been established for *JA* beginning in FY'91:

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- To increase by 50% the number of papers published in *JA* by the end of FY'91 (with no reduction in quality).
- To make *JA* the first choice of submitting authors.
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To apply for the editorship, submit four copies of an application citing qualifications, your objectives for the journal, and how you will meet the above goals. Send them to

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B.M.M.