Finite Element Modeling and Robust Control of Plant Protection Machine Boom

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***Abstract*—In order to reduce the horizontal deformation caused by the sprayer rigid frame of the plant protection machine, this paper establishes the structural dynamics model of the spray boom rigid frame based on the rigid frame structure commonly used in the plant protection machine spray boom, and transforms it into state space form. A robust controller based on state observer is designed. The boom frame with vibration deformation due to disturbance is controlled. Simulation results show that the method is effective in restraining the horizontal vibration deformation of the spray rod frame. The research results provide a method for restraining the vibration deformation of the spray rod frame structure of plant protection machine.**

***Index Terms*—spray boom, finite element, model space, vibration deformation, robust control**

I. Introduction

As we all know, Chinese population is huge, so Chinese agricultural development is particularly important.So the development of Chinese agriculture is particularly important currently.In the process of development, agricultural natural disasters plant diseases and insect pests have become one of the issues of importance. From a global perspective, chemical control and mechanical spraying are still the main means of plant protection operations. There are many methods of plant protection, mainly chemical control, physical control, biological control and comprehensive prevention and robust control [1]. The chemical control method has the characteristics of high efficiency, timely control, quick effect, good control effect and low cost [2], which has become the main method of plant protection. Plant protection machinery is used to apply chemicals to crops and is widely used in the plant protection process for agricultural production. These chemicals are usually distributed by plant protection sprayers [3].

The plant protection machine spray boom can be regarded as a rigid frame structure. This structure has large size, low frequency and low damping. When the plant protection machine is in operation, the sprayer spray boom will be elastically deformed and vibrated due to the shape of the soil, which will cause locality. Excessive and under-spraying affects the quality of the spray [4]-[6]. In terms of vibration reduction of the spray boom, even a small amplitude of vibration can cause excessive spray for a wide-angle sprayer [7]. Rong J.H (2000) proposed the theory of mitigating structural vibration and optimized structural parameters. This study provides a theoretical reference value for the lightweight design of sprayer booms compared to normal spray operation heights [8]. In 2003, Anthonis *et al.* designed an active control suspension to suppress the horizontal vibration of the sprayer and achieved certain effects [9]. Improving the boom suspension method is an effective method to reduce the vibration of the boom. However, this method cannot control the vibration deformation of the boom itself. More research es did not consider the vibration deformation of the boom and only considers it as a rigid body. Because the boom is essentially an elastomer, the vibration of the boom is still poor in this way.

In this paper, a finite element method is used to establish the dynamic model of the boom frame structure and convert it into a modal space equation. For the spatial rigid frame structure in this paper, considering the inevitability of parameter uncertainty in the modeling process, the robust deformation control of the boom rigid frame is carried out by robust control method, and it is simulated, tested the effectiveness of the method.

II. Geometric Description of Rigid Frame of Spray Rod of Plant Protection Machine

The model in this paper uses the rigid frame structure commonly used in large-sized plant protection machine boom of Fig. 1.

The entire structure is symmetrical with a total length L = 24m. The boom has has 27 nodes totally considering the displacement in terms of its level, each node has 6 degrees of freedom. Displacement and rotation angles in the x-axis, y-axis, and z-axis directions, respectively, and 7 and 8 nodes are fixed on the plant protection machine frame to limit all degrees of freedom. The total degree of freedom is 150, where 1, 2 and 13, 14 nodes are 1m apart, 2 to 13 nodes are 22m apart, 2, 27 and 13, 15 nodes are 0.5m apart, 7, 22 and 8, 22 nodes are 1.5m apart.

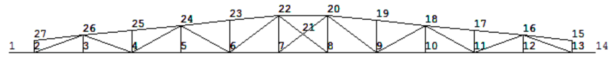

Figure 1. Spray rod rigid frame diagram and node distribution

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III. Finite Element Method Modeling of Spray Frame Frame Structure of Plant Protection Machine

When the beam unit is subjected to a load, a bending deformation can be generated, and the strain corresponds to a bending strain. In the case where the length of the section is much smaller than the length of the beam element, the shear strain has little effect on the deflection of the beam, so it can be ignored. This beam element is a typical Euler-Bernoulli beam element as shown in Fig. 2.

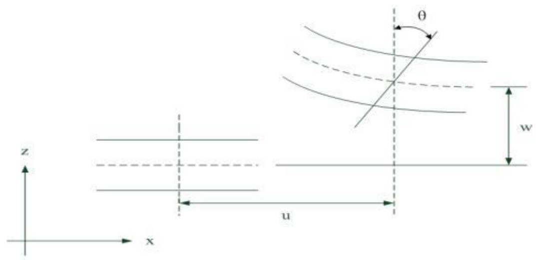

Figure 2. Euler-Bernoulli beam element

In this paper, the finite element method is used to parametrically model the boom frame. The unit type can be Euler-Bernoulli beam unit, which is the main basic structure of the rigid frame. For the beam element in space, each node has 6 degrees of freedom, namely tensile deformation, bending deformation and torsional deformation. For the spatial node unit, in the unit local coordinate system, the node variable can be expressed as

$$\delta^e = [x_i \ y_i \ z_i \ \theta_{x_i} \ \theta_{y_i} \ \theta_{z_i} \ x_j \ y_j \ z_j \ \theta_{x_j} \ \theta_{y_j} \ \theta_{z_j}]^T$$

where x_i, y_i and z_i are the displacements of the nodes along the local coordinate direction, respectively.

θ_{x_i}, θ_{y_i} and θ_{y_i} are the corners of the section at the node.

The strain and stress of the element can be expressed as

$$\varepsilon = B\delta^e(t) \tag{1}$$

$$\sigma = D\varepsilon = DB\delta^e(t) \tag{2}$$

where B is the element strain matrix and D is the elastic matrix of the elements, which are only related to the nature of the material. The mass matrix and stiffness matrix of the element can be derived from the kinetic energy and potential energy expressions of the element. The unit kinetic energy expression is

$$T_e = \frac{1}{2}\int_V \rho(\frac{\partial\delta^e(t)}{\partial t})^2 dV = \frac{1}{2}\dot{\delta^e}(t)\int_V \rho N^T N dV\dot{\delta^e}(t)$$

$$= \tfrac{1}{2}\dot{\delta^e}(t)^T M_e \dot{\delta^e}(t) \tag{3}$$

In the formula (3), N is a unit form function matrix, M_e is a unit mass matrix, and its calculation formula is

$$M_e = \int_V \rho N^T N dV \tag{4}$$

The expression of unit potential energy is

$$U_e = \frac{1}{2}\int_V \varepsilon^T\sigma dV = \frac{1}{2}\int_V \delta^e(t)^T B^T DBdV\delta^e(t)$$

$$= \tfrac{1}{2}\delta^e(t)^T K_e \delta^e(t) \tag{5}$$

In the formula (5), $\mathrm{K_e}$ is the stiffness matrix of the element, and its calculation formula is

$$K_e = \int_V B^T DBdV \tag{6}$$

Substituting the above kinetic energy into the Lagrangian equation, the finite element equation of the beam element is obtained

$$\overline{\mathrm{M}}\ddot{\overline{\delta}} + \overline{\mathrm{K}}\overline{\delta} = \mathrm{P}\overline{\mathrm{F}}(\mathrm{t}) \tag{7}$$

In the formula (7), P is a control force distribution matrix, and the number of rows is the number of degrees of freedom of the boom rigid frame, and the number of columns is the number of control forces. If there is a strong input in a certain degree of freedom, the corresponding degree of freedom and the column are 1 and the remaining positions are filled with 0. $\overline{\mathrm{F}}$ is the input force vector, which is a column vector whose number of lines is the number of input forces. $\overline{\mathrm{M}}$ and $\overline{\mathrm{K}}$ are the total mass matrix and the overall stiffness matrix of the boom frame, which are obtained from the unit mass matrix $\overline{M}_e$ and the unit stiffness $\overline{K}_e$ matrix, respectively. For complex structures, the direction of each beam element will be different. To do this, it is necessary to establish a coordinate transformation matrix to move the local coordinate system to the global coordinate system.

$$\overline{M}_e = \mathrm{T^T} M_e \mathrm{T} \quad \overline{K}_e = \mathrm{T^T} K_e \mathrm{T}$$

The lumping method is to expand the unit mass matrices $\overline{M}_e$ and $\overline{K}_e$ into a square matrix of 150×150 according to the total number of degrees of freedom. After expansion, the original unit mass and stiffness matrix are one-to-one corresponding according to their degrees of freedom, and the remaining positions are filled with 0. The expansion matrix of each element is summed to obtain an overall mass matrix $\overline{\mathrm{M}}$ and an overall stiffness matrix $\overline{\mathrm{K}}$.

IV. Free Vibration Modal Analysis of Spraying Frame of Plant Protection Machine

Based on ANSYS software and Block Lanczos method, the low-order natural vibration modes of the model are solved. The first four natural vibration frequencies are shown in Table I.

TABLE I. Natural Vibration Frequency

Serial number	1	2	3	4
Frequency (Hz)	0.449	0.451	2.857	2.865

It can be seen from Table I that the first 4 natural frequencies of the structure are all lower than 3 Hz, showing the characteristics of low frequency. Since the plant protection machinery is driven in the field to generate vibrations of less than 5 Hz, the controller designed for the vibration control of the rigid frame structure should have a control effect on the entire frequency band.

Through modal analysis, it can be seen that in the first 4th mode, when the boom is free to vibrate, the boom produces horizontal vibration deformation. The amplitude of the distal end of the boom frame can be seen from the first 4th mode diagram. Maximum, so the amplitude of the boom nodes 1 and 14 can be selected to describe the vibration of the entire boom and the suppression of the vibration of the entire boom when the control is applied [10].

V. Robust Controller Design Based on State Observer

A. The Equation of Motion is Transformed into a State Space

For a vibrating system with n degrees of freedom, the physical parameter model is described by n independent physical coordinates. In the linear range, the free vibration response in the physical coordinate system is a linear superposition of n main vibrations. Each main vibration is a specific form of free vibration. The vibration frequency is the main frequency (natural frequency) of the system~~,~~ and the vibration form. That is, the main mode of the system (modal or natural mode). Firstly, the kinetic equations established by the finite element method are subjected to modal coordinate transformation.

$$x = \theta n \tag{8}$$

where θ is the undamped modal matrix of the system obtained from the eigenvectors by mass normalization, and n is the generalized modal coordinate vector of the rigid frame. According to the modal coordinate transformation [11], the equation (7) can be changed to the following dynamic equations in generalized modal coordinates.

$$\bar{M}^{*}\ddot{n} + \bar{K}^{*}n = U^{*}(t) \tag{9}$$

in

$$\bar{M}^{*} = \theta^{T}\bar{M}\theta,\ \bar{K}^{*} = \theta^{T}\bar{K}\theta,\ U^{*}(t) = \theta^{T}P\bar{F}(t)$$

If the first m-order mode is taken to approximate the vibration of the original system, then m<n, the dynamic model under the generalized mode coordinates can be described as

$$\bar{M}_{m}{}^{*}\ddot{n} + \bar{K}_{m}{}^{*}n = U_{m}{}^{*}(t) \tag{10}$$

The spatial rigid frame has a complicated structure and a high dimensionality. If the dynamic equation is used, the design of controller becomes difficult, the calculation amount is too large, and the time is too long, which are unfavorable for vibration control. Therefore, in this paper do modal truncation of the mathematical model in the modal space, that is, retain the low-frequency mode, that is, the first four-order mode, ignoring the high-frequency mode. The modal displacement and the modal velocity are combined into a state variable, and the model is transformed into a state space representation, and a generalized state space model of the spatial rigid frame structure is established [12].

Since the stiffness matrix and mass matrix of the rigid frame structure are time-invariant, the structural vibration control system belongs to the linear steady system, and the equation (7) is transformed into the state space.

$$\dot{x} = Ax + Bu \tag{11}$$

where $x = [x_1 \quad x_2]$, in $x_1 = \bar{\delta}$, $x_2 = \dot{x}_1$

$$A = \begin{bmatrix} 0 & I \\ -M^{-1}K & 0 \end{bmatrix} \tag{12}$$

$$B = \begin{bmatrix} 0 \\ M^{-1}P \end{bmatrix} \tag{13}$$

$$C = I_{8\times 8} \text{ (Unit quality)} \tag{14}$$

It is known from MATLAB that (A, B) is controllable and (A, C) is considerable.

In errors occur during the procession of mathematical modeling, and external factors can cause large parameter uncertainties. In the physical space model, the parameter uncertainties are mainly reflected in the error of the mass matrix, the stiffness matrix and the damping matrix, so they can be considered as uncertain. In the state space model, it mainly reflects the uncertainty of modal frequency, mode shape and modal damping ratio. These uncertainties have an impact on the stability and control performance of the control system.

For linear time-invariant systems, when considering structural uncertainty, the equation will be transformed into the following form:

$$\dot{x} = (A + \Delta A)x + (B + \Delta B)u$$
$$y = (C + \Delta C)x$$

Among them, ΔA, ΔB, ΔC are the uncertainties of the structural parameters. For the space rigid frame structure in this paper, only the uncertainty caused by the modal frequency is considered. Thus, the uncertainty exists only in ΔA, so it is simplified. State space model:

$$\dot{x} = (A + \Delta A)x + Bu$$
$$y = Cx$$

For the structural model change caused by the uncertainty of structural parameters, verify the designed controller, that is, controller is designed by applying to the nominal model, and then consider the influence of modal frequency uncertainty to draw the closed-loop singular value of the uncertainty system. Whether the curve satisfies $\|T_{ZW} < \gamma\|$ to judge the robustness of the controller. The modal frequency uncertainty can be additive uncertainty or multiplicative uncertainty [13].

B. Design of a State Observer Based Robust Controller

Consider the stability of the system for the uncertainties that exist in the system: for linear uncertain systems:

$$\begin{cases} \dot{x}(t) = [A + \Delta A]x(t) + Bu(t) \\ \qquad y = Cx(t) \end{cases} \tag{15}$$

where $x(t) \in R^n$ is the state matrix of the system, $u(t) \in R^m$ is the control input, and A and B and C are the real constant matrices describing the system model, assuming that ΔA=EΣF.

Define the following state observers:

$$\begin{cases} \dot{\hat{x}}(t) = A\hat{x}(t) + Bu(t) + L[y(t) - C\hat{x}(t)] \\ u(t) = -K\hat{x}(t) \end{cases} \tag{16}$$

And can be gained, obtained

$$e(t) = x(t) - \hat{x}(t) \tag{17}$$

Obtained by the above formula:

$$\dot{x}(t) = [A + \Delta A - BK]x(t) + BKe(t) \tag{18}$$

$$\dot{e}(t) = \dot{x}(t) - \dot{z}(t) = (A - LC)e(t) + \Delta Ax(t) \tag{19}$$

From the above formula can be obtained:

$$\begin{cases} \dot{x}(t) = [A + \Delta A - BK]x(t) + BKe(t) \\ \dot{e}(t) = (A - LC)e(t) + \Delta Ax(t) \end{cases} \tag{20}$$

Lemma 1: L, E, F are suitable dimension matrices, and $F^TF \le I$, I is a suitable dimension unit matrix, then there are $\alpha > 0$, $\alpha \in R$, so that the following inequality holds.

$$LFE + (LFE)^T \le \alpha^{-1}LL^T + \alpha E^TE$$

Theorem 1: For linear uncertain systems (15), there is linear feedback based on state observers, so that the sufficient condition for robust stability of closed-loop systems is that there are positive definite symmetric matrices $X > 0$, $P_2 > 0$ and matrices W, Z, K And the constant ε, which makes the following linear matrix inequality.

$$\begin{bmatrix} XA^T + AX - BW - W^TB^T & BK & E & XF^T & 0 \\ K^TB^T & A^TP_2 - C^TZ^T + P_2A - ZC & 0 & 0 & P_2E \\ E^T & 0 & -\varepsilon I & 0 & 0 \\ FX & 0 & 0 & -\frac{1}{2}\varepsilon^{-1}I & 0 \\ 0 & E^TP_2 & 0 & 0 & -\varepsilon I \end{bmatrix} < 0$$

Proof: The following Lyapunov function is constructed for the closed-loop augmentation system (20).

$$V(x(t), t) = x^T(t)P_1x(t) + e^T(t)P_2e(t)$$

The Lyapunov function defined above is derived for t

$$\dot{V} = \dot{x}^T(t)P_1x(t) + x^T(t)P_1\dot{x}(t) + \dot{e}^T(t)P_2e(t)e^T(t)P_2\dot{e}(t)$$

$$= [(A + \Delta A - BK)x(t) + BKe(t)]^TP_1x(t) + x^T(t)P_1[(A + \Delta A - BK)x(t) + BKe(t)] + [(A - LC)e(t) + \Delta Ax(t)]^TP_2e(t) + e^T(t)P_2[(A - LC)e(t) + \Delta Ax(t)]$$

$$= [Ax(t) + \Delta Ax(t) - BKx(t) + BKe(t)]^TP_1x(t) + x^T(t)P_1[Ax(t) + \Delta Ax(t) - BKx(t) + BKe(t)] + [(A - LC)e(t) + \Delta Ax(t)]^TP_2e(t) + e^T(t)P_2[(A - LC)e(t) + \Delta Ax(t)]$$

$$= x^T(t)(A - BK)^TP_1x(t) + x^T(t)P_1(A - BK)x(t) + x^T(t)P_1BKe(t) + e^T(t)(BK)^TP_1x(t) + x^T(t)\Delta A^TP_1x(t) + x^T(t)P_1\Delta Ax(t) + e^T(t)(A - LC)^TP_2e(t) + x^T(t)\Delta A^TP_2e(t) + e^T(t)P_2(A - LC)^Te(t) + e^T(t)P_2\Delta Ax(t)$$

By Lemma 1

$$x^T(t)(\Delta A^TP_1 + P_1\Delta A)x(t)$$
$$\le x^T(t)(\varepsilon^{-1}P_1EE^TP_1 + 2\varepsilon F^TF)x(t) + x^T(t)P_1BKe(t) + e^T(t)(BK)^TP_1x(t) + e(t)[(A - LC)^TP_2 + P_2(A - LC) + \varepsilon^{-1}P_2EE^TP_2]e(t)$$
$$= \begin{bmatrix} x(t) \\ e(t) \end{bmatrix}^T \Omega \begin{bmatrix} x(t) \\ e(t) \end{bmatrix}$$

In Ω=

$$\begin{bmatrix} (A - BK)^TP_1 + P_1(A - BK) + \varepsilon^{-1}P_1EE^TP_1 + 2\varepsilon F^TF & P_1BK \\ (BK)^TP_1 & (A - LC)^TP_2 + P_2(A - LC) + \varepsilon^{-1}P_2EE^TP_2 \end{bmatrix}$$

Since the sufficient condition for the asymptotic stability of the closed-loop system is $\dot{V} < 0$, it is only necessary to prove that $\Omega < 0$. On the both sides of Ω, respectively, multiply and multiply $\mathrm{diag}(X, I)$, let $X = P_1^{-1}$, $W = KX$, $Z = P_2L$ can be obtained:

$$\begin{bmatrix} X(A - BK)^T + (A - BK)X + \varepsilon^{-1}EE^T + 2\varepsilon XF^TF & BK \\ K^TB^T & A^TP_2 - C^TZ^T + P_2A - ZC + \varepsilon^{-1}P_2EE^TP_2 \end{bmatrix}$$

Applying schur to the above matrix:

$$\begin{bmatrix} XA^T + AX - BW - W^TB^T & BK & E & XF^T & 0 \\ K^TB^T & A^TP_2 - C^TZ^T + P_2A - ZC & 0 & 0 & P_2E \\ E^T & 0 & -\varepsilon I & 0 & 0 \\ FX & 0 & 0 & -\frac{1}{2}\varepsilon^{-1}I & 0 \\ 0 & E^TP_2 & 0 & 0 & -\varepsilon I \end{bmatrix}$$

So prove the theorem 1.

C. *Horizontal Vibration Control Simulation of Boom Frame*

The boom test is performed on the controller obtained above. When the system is simulated, $\Delta A = E\Sigma F$ is taken.

$$E = 0.01 * I_{8\times8}\ ,\ \Sigma = I_{8\times8}\ ,\ F = 10 * I_{8\times8}$$

Limit all the degrees of freedom of the 7th and 8th nodes of the boom frame in the Fig. 1, and select the first 4th mode to control. Using LMITOOL solution of MATLAB: Since t=-9.3618e-10, $t < 0$, LMI is feasible. So get from the above formula.

$$K = WX^{-1}, L = ZP_2^{-1}$$

MATLAB solution

K =

$$\begin{bmatrix} 493937.3 & -191063.4 & -1893346.5 & -2096220.3 & 163020.6 & -233587.7 & 63319.9 & -40453.0 \\ -1062099.2 & -1021938.2 & 5596066.1 & 6499159.2 & 226411.2 & 256503.3 & -255752.5 & 179451.2 \end{bmatrix}$$

$$L = \begin{bmatrix} R_1 & R_2 \\ R_3 & R_4 \end{bmatrix} \quad R_1 = \begin{bmatrix} 0.32 & 0 & 0 & 0 \\ & 0.32 & 0 & 0 \\ & & 0.18 & 0 \\ & & & 0.18 \end{bmatrix}$$

$$R_2 = \begin{bmatrix} -1.41 & 0 & 0 & 0 \\ & -1.42 & 0 & 0 \\ & & -1.98 & 0 \\ & & & -1.98 \end{bmatrix}$$

$$R_3 = \begin{bmatrix} -2.89 & 0 & 0 & 0 \\ & -2.89 & 0 & 0 \\ & & -126.01 & 0 \\ & & & -126.76 \end{bmatrix}$$

$$R_4 = \begin{bmatrix} 1.13 & 0 & 0 & 0 \\ & 1.14 & 0 & 0 \\ & & 29.54 & 0 \\ & & & 29.71 \end{bmatrix}$$

Select the 6th and 9th nodes on the boom frame, and the corresponding nodes apply to the horizontal direction control force. Apply a momentary disturbance of 0.25m in the positive direction of the z-axis to the 1st and 14th nodes of the boom frame. Observe 1 Deformation in the horizontal direction of the 14-node to understand the horizontal deformation vibration of the entire boom frame.

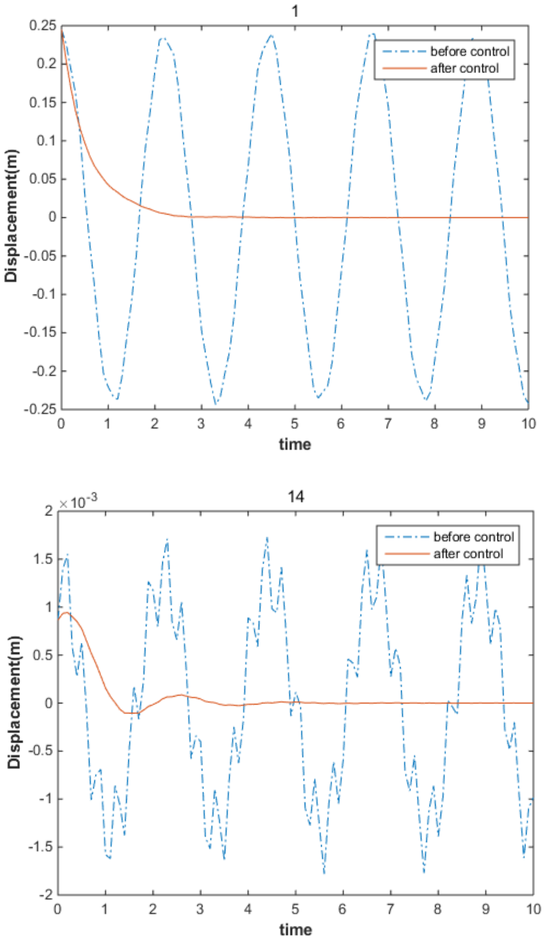

Figure 3. Displacement of Nodes 1 and 14 in the horizontal direction

As shown in Fig. 3, the curve fluctuates greatly at the beginning. When it is acted by the controller, the curve tends to be stable gradually. The simulation results show that the designed state observer can stabilize the spray rod.

VI. Result

In this paper, the finite element modeling of the common boom frame structure of the plant protection machine is carried out, and the problem of controlling the vibration deformation on the rigid frame structure is studied. The boom frame is regarded as the combination of beam elements. The finite element method is used to model the dynamics, and the obtained model is transformed into the modal space. The first 4 modes are used to suppress the vibration deformation of the spray booms in the horizontal direction.The robust stability problem is analyzed. Based on the linear matrix inequality method and the Lyapunov stability principle, the relevant criteria of the system stability are given and simulated. The effectiveness of the control method is demonstrated.

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