Blade Loss Transient Dynamic Analysis of Turbomachinery

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This paper reports on work completed to develop an analytical method for predicting the transient nonlinear response of a complete aircraft engine system due to the loss of a fan blade, and to validate the analysis by comparing the results against actual blade loss test data. The solution, which is based on the component element method, accounts for rotor-to-casing rubs, high damping, and rapid deceleration rates associated with the blade loss event. A comparison of test results and predicted response show good agreement except for an initial overshoot spike not observed in test. The method is effective for analysis of large systems.

Nomenclature

m_i = generalized mass
Q_i = generalized force
q_i = generalized coordinate or modal participation of the ith mode
Δt = time interval in the central difference solution

Superscripts

0 = current value
−1 = previous value
1 = next or future value

Introduction

THE purpose of this presentation is to report on work partially completed under NASA contract, to develop an analytical method to predict the transient structural response of an aircraft engine system when a blade is lost, particularly a fan blade, and to validate the analysis by comparing the results against actual blade loss test data. Because transient dynamics and modal synthesis are well documented in the technical literature,[1-5] the emphasis of this paper is on the correlation of calculated results with test data.

To analytically predict a complex event with confidence requires experimental validation of the method of calculation. The end product of this work is a computer program designed specifically to analyze aircraft gas turbine engines. The payoff is the avoidance of engine problems by permitting a more thorough design analysis, reduction of development costs by reducing required test/redesign iterations, and the optimization of structural design.

Modern commercial aircraft gas turbines are usually high bypass ratio turbofans. Typically, they are dual rotor machines, with the low-pressure (LP) rotor consisting of a fan, LP shaft, and LP turbine. The high-pressure (HP) rotor is coaxial with the LP rotor and consists of the HP compressor, shafting, and HP turbine.

For commercial aircraft gas turbines, Federal Air Regulations require the manufacturer to demonstrate that its engine can lose and contain a whole fan blade, without the engine becoming detached from the mounts or producing a hazard to the aircraft. The work reported here provides a tool for quantitatively predicting the dynamic response of the engine structure to a blade loss occurrence. The response involves vibration of the engine as a structural unit or system and excludes localized deformations or actual structural distress.

The blade loss event is rather complex. For loss of a complete blade, the unbalance is large and suddenly applied. The response produces large forces involving the total structural system, which must be represented fully in the analysis. Because of the large response and close running clearances, rotor-to-casing rubs may occur, which have a profound influence on the results. This is especially true in the fan rotor following a blade loss where an alternate nonlinear load path occurs at the rub. Damping is high, and a very rapid deceleration of the rotors usually occurs. It is important that most of these characteristics be incorporated in the analysis if accurate load prediction is to be obtained, and for structural integrity to be correctly assessed.

Analytical Approach

The analytical method used in this program to predict the transient nonlinear response of the complete engine system is the component element method[5] which is a form of the widely used modal synthesis approach.[2-4] The approach is relatively straightforward. The large structural system to be analyzed is broken into a number of components or substructures, called component elements. Each component element is defined in terms of its normal modes, either free-free or constrained. The complete system is reconstituted in terms of a finite number of modes and their generalized coordinates, thereby effecting a substantial savings in computer time in the dynamic analysis of large structural systems.

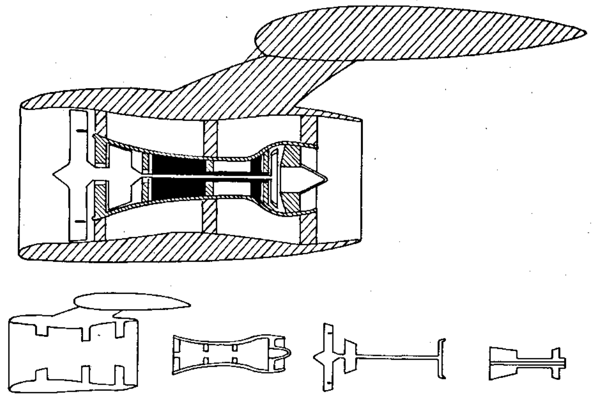

Fig. 1 Structural breakdown for TETRA analysis.

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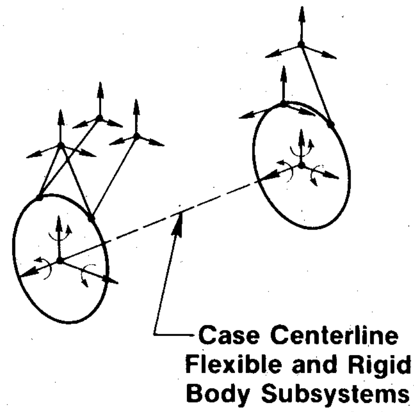

Fig. 2 Engine support-link element multidirection, multipoint connection.

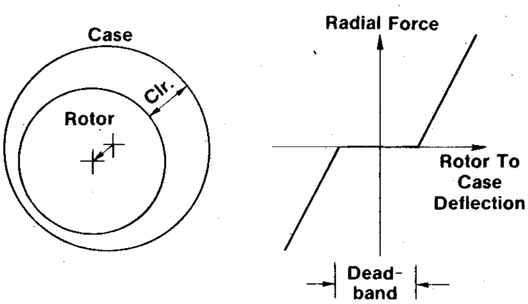

Fig. 3 Deadband representation of rotor-to-casing rub.

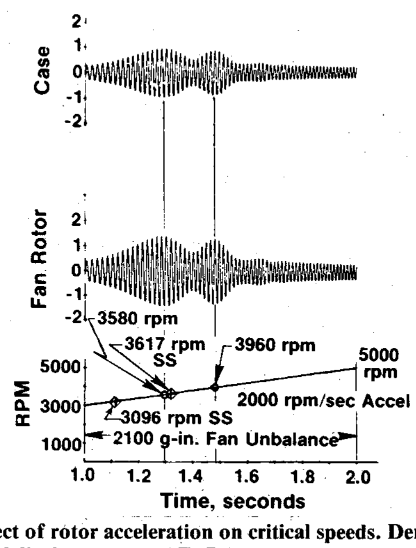

Fig. 4 Effect of rotor acceleration on critical speeds. Demonstration model: radial displacements—TETRA.

In the analysis and program[8] developed here, the number of subsystems or component elements utilized to represent the complete system were limited for convenience to four, but more could easily have been used without adding complexity. The four elements are sufficient to represent the two rotors, the engine static structure, and the aircraft pylon. These components are those normally associated with most turbofan engines (Fig. 1). Having selected the number of component elements, the analyst must then decide on the number of modes required for each element to adequately define the system dynamically. In addition, the physical connecting elements which tie the components together must also be defined. These include connectors such as links, springs and trusses, etc.

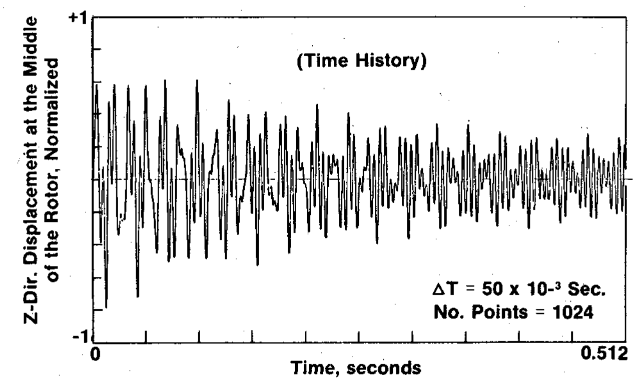

Fig. 5 Demonstration model: transient response at 12,000 rpm and sudden unbalance.

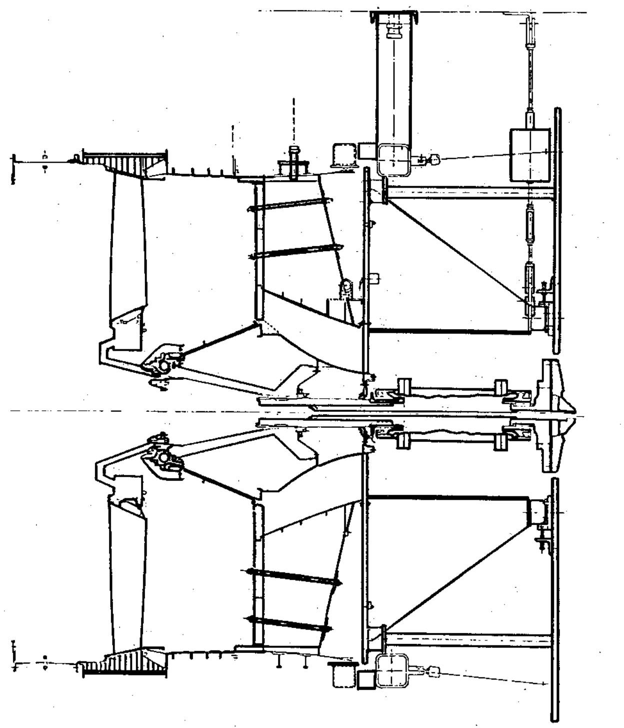

Fig. 6 Cross section of blade loss test vehicle.

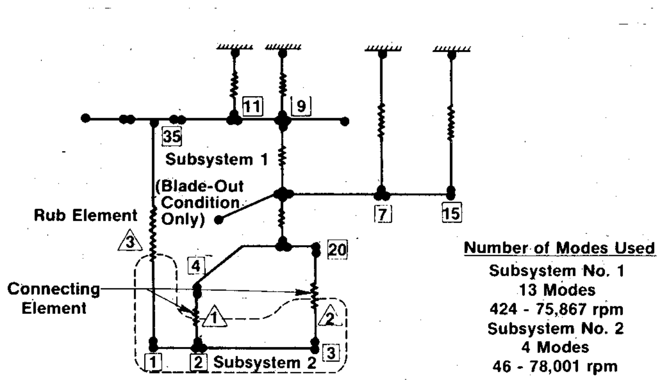

Fig. 7 Blade loss test vehicle TETRA schematic.

The time history solution, which uses a central-difference integration scheme to compute the transient response, was incorporated in a computer program called the Turbine Engine Transient Response Analysis (TETRA). Several unique features were incorporated in this numerical solution method. The computer program is modular in construction, and uses a building block system of modeling. This enables additional capabilities to be added easily as modules without

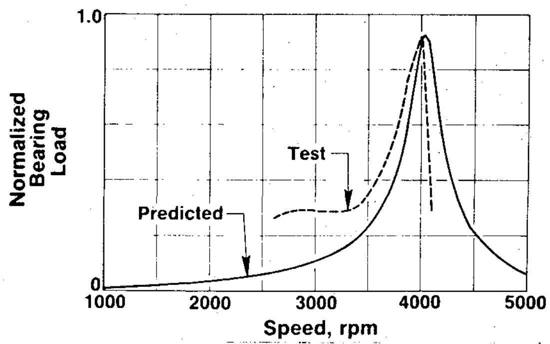

Fig. 8 Test vehicle: comparison of direct solution with test data (steady-state/nominal unbalance).

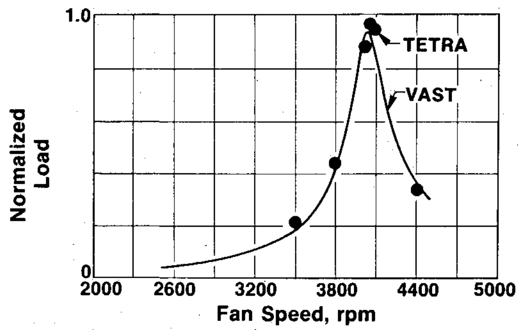

Fig. 9 Test vehicle: comparison of TETRA solution with direct solution (steady-state/nominal unbalance).

the necessity for rewriting the program. Gyroscopic effects, which are very important for large turbofan engines, were included. Time-dependent connecting force elements, such as the nonlinear blade tip rub or "dead-band" element, are permitted between rotor-to-rotor or rotor-to-stator locations. Discrete and distributed damping are permitted and time-dependent rotor speeds and unbalance forces can also be specified. A key feature of the component-element method and the free-free mode representation is that no global matrix inversion is required, and is the basis for the economy of this approach.

For user convenience, a number of physical connecting elements were developed, which are used in the analysis by simply specifying the required key identifying elements. These include the spring-damping element with which the physical properties of the elements connecting the components are specified, the space link element for representing the three-dimensional structure usually used to join the engine to the pylon (Fig. 2), the rub or deadband element defining the nonlinear load path which occurs between components when relative deflections are large enough to induce a rub (Fig. 3), and the gyroscopic element which introduces the cross-coupling load terms at selected rotor stations associated with angular velocity at the station. Each of these elements, defined as a stiffness and damping matrix, establishes the forces between the component elements.

The solution is obtained by calculating the current generalized acceleration of each mode in terms of the current generalized forces at any mass point.

$$\ddot{q}_i = Q_i / m_i \text{ current generalized acceleration}$$

The current accelerations are used to calculate future generalized displacements and forces. The future displacement is

$$q_i = 2q_i^0 - q_i^{-1} + \ddot{q}_i^0 (\Delta t)^2$$

A Runge-Kutta integration scheme could have just as easily been used.

The analysis and computer program were checked out using simple degenerate test cases, for which closed-form analytical solutions were developed. Demonstration cases were also run using a simple single rotor-single casing engine model. Five rotor and three casing modes were used in the analysis which encompass all modes up to two times maximum rotational speed. Several loading conditions were run for correlation with results from a standard Proh-Mykelstad[6] steady-state solution. Figure 4 shows the normalized predicted response for a rapid acceleration with unbalance in the rotor. The expected physical shift of the critical speeds due to acceleration rate is noted. Figure 5 shows the transient displacement response due to sudden unbalance, while a spectral analysis of the time history record was made to obtain the frequency content. The frequencies agreed with the critical frequencies obtained from the direct solution within 0.2%.

TETRA Validation with Test Vehicle Analysis

The validation of the TETRA computer program depended ultimately on the favorable comparison between its calculated results and the test data from a blade loss test vehicle. To verify its capability to predict real events involving realistic structures, the method was applied to a test vehicle which was constructed with actual engine hardware (Fig. 6).

Blade Loss Test Vehicle Modeling

Figure 7 shows a schematic of the test vehicle showing the idealized interconnections between the static structural elements and the rotor. To establish a valid analytical model, the vehicle was first analyzed by the direct method[7] to obtain its linear steady-state response for small unbalance. Agreement of these predictions with test data (as shown in Fig. 8) confirmed the vehicle idealization to be a good analytical model.

Subsequently, the TETRA model was derived from the direct solution model. The vehicle was subdivided into two subsystems: the static structure is referred to as subsystem 1 and the rotor as subsystem 2, as shown in Fig. 7. The normal modes of the ground-fixed subsystem 1 were obtained in the usual manner[7] for 13 modes, the maximum being 75,867 cycles/min. The fan rotor and driveshaft subsystem 1 was modeled as a free-free beam with four modes with the maximum modal frequency of 78,000 cycles/min. Subsequently, TETRA was run to reconstruct the system response of the entire vehicle in terms of the two subsystems' normal modes.

To verify the TETRA model, its steady-state response at several speeds was calculated at small unbalance and compared with results from the direct solution. As shown in Fig. 9, the agreement is excellent. Having established the validity of the test vehicles' TETRA model, using a linear response correlation, the actual blade loss test simulation could then be made.

TETRA Analysis—Test Correlation

To obtain data for correlation, the blade loss test vehicle was fully instrumented with accelerometers and dynamic strain gages. The transient responses measured by these sensors were recorded on magnetic tape for post-test data processing.

An explosive charge was electronically detonated to release one fan blade from its root, at a prescribed time and at a rotor speed of 4200 rpm. The dynamic bearing loads as well as the strains were recorded as functions of time and the rotor speed. From these, the blade loss load-time-speed history was reconstructed which ultimately was used to correlate with the results obtained from the TETRA analysis of the blade loss test vehicle.

A deadband or clearance between the rotor blade tips and the containment casing of 0.89 cm was assumed. This represents approximately the radial thickness of the abradable

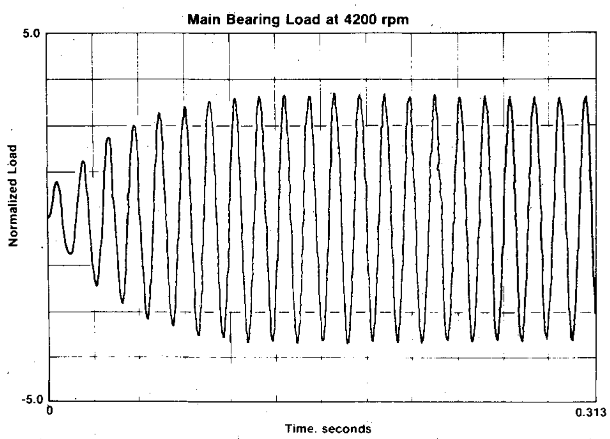

Fig. 10 Blade loss test vehicle, TETRA analysis without rub element at constant speed.

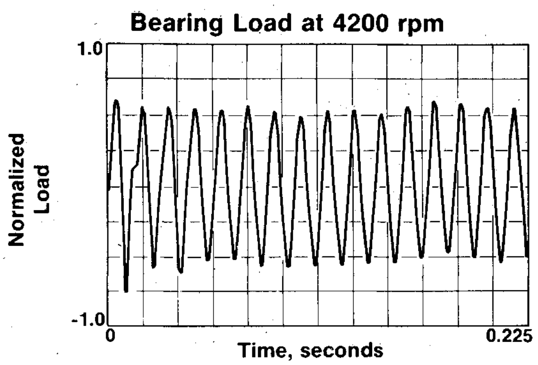

Fig. 11 Blade loss test vehicle, TETRA analysis with rub element at constant speed.

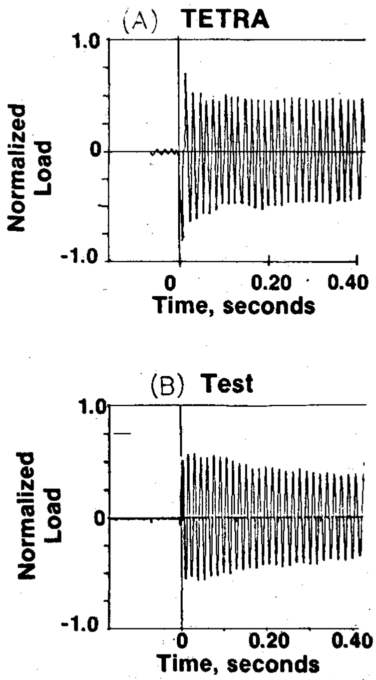

Fig. 12 Blade loss vehicle transient response for fan blade release: a) prediction from TETRA, and b) test data.

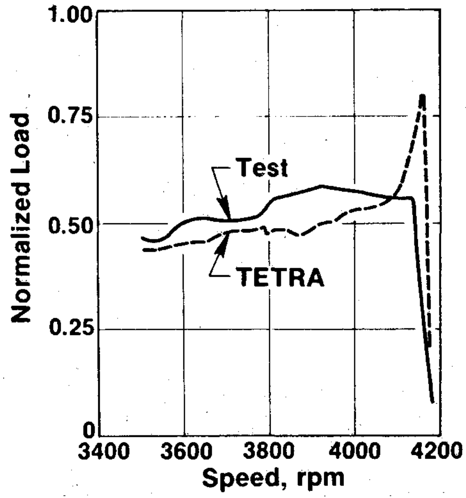

Fig. 13 Blade loss test vehicle—main bearing load transient response envelope: TETRA vs test.

shroud material, which is assumed to be rubbed out with negligible force. The casing local stiffness was predicted to be 1.75×10^8 N/m and was used to establish the load deflection characteristic in the deadband calculations. Based on prior experience, a modal damping Q of 7.5 was used. The rotor deceleration history measured in the vehicle test was used in the analysis, although an exact history is not critical to the results. With these physical conditions, the vehicle time history was predicted using the TETRA program.

To demonstrate the important influence on response of rotor-to-casing rubs, a solution was run with and without the tip rub restraint option for a full blade loss condition. For this comparison, the blade release was assumed to occur at a speed of 4200 rpm. The rotor speed was taken to remain constant. Figure 10 gives the time history of the main bearing load in the vertical direction for the nonrubbing condition. Note that the load builds up rapidly to a steady maximum normalized value of about 6.8 units peak-to-peak.

Figure 11 shows the results for the same conditions, but now with the tip-rub deadband characteristics described previously. In this case, the main bearing load builds up to a steady value of about 1 unit peak-to-peak with some initial overshoot. The effect of rotor blade-to-casing rub in this case essentially reduced the peak bearing load by approximately a factor of 6.8. In addition, the total system response is altered significantly. A key factor in the prediction of response for the blade loss event is proper accounting for the deadband associated with rubs.

Figure 12a shows the predicted vehicle blade loss event, including the effect of rotor deceleration rate. Note that an initial overshoot is present, and that the bearing load amplitude remains relatively constant over many revolutions despite the significant speed change.

In the actual blade loss test, the load time history at the main rotor bearing of the test vehicle was measured using an instrumented and calibrated bearing housing. The normalized load component in the vertical direction is given in Fig. 12b (Fig. 12a shows the predicted TETRA results). Note that the test data show no initial overshoot spike as predicted theoretically. This lack of initial overshoot is not unusual, and has been observed in other tests. Aside from the discrepant overshoot, the analytical and experimental results agree quite well. This is illustrated in Fig. 13 which shows a plot of the envelope of the predicted and measured responses.

The reason for the predicted overshoot initially was thought perhaps to be a result of truncation error. This was not believed likely, since a larger number of modes were used in the solution than were thought necessary. The solution included all component element modes up to 18 times the maximum rotor speed. Nonetheless, since no guidelines were available to the analyst on the number of modes to be included in this type solution, an investigation was conducted to evaluate this effect. Rather than include a greater number of modes (since a larger number was already being used), the influence of a reduced number was studied instead. Table 1 summarizes the results of this study. Tabulated are the

Table 1 Effect of number of modes on transient response

Max casing modal freq., cycles/min	Max rotor modal freq., cycles/min	Max case freq. / Max rotor speed[a]	Max rotor modal freq. / Max rotor speed[a]	Max modal freq. / Max rotor speed[a]	Load variation from test data, %	
					Initial overshoot	After overshoot
75,861 (13)[b]	78,000 (4)	18.1	18.6	18.6	30	2
39,995 (9)	17,626 (3)	9.6	4.2	9.6	30	2
18,878 (7)	17,626 (3)	4.5	4.2	4.2	30	3
6,535 (4)	146 (2)	1.6	0.04	1.6	45	7

[a] Maximum rotor speed = 4200 rpm. [b] Number in parentheses denotes number of modes.

maximum casing and rotor modal frequencies used in the solution (in cycles/min), together with the number of modes used, indicated in brackets. Also tabulated is the ratio of the maximum casing and rotor frequencies used in the solution to the maximum rotor rotational frequency of 4200 rpm.

The effect on the results of the number of modes used in the solution were expressed in two ways; the influence 1) on the initial overshoot, and 2) on the subsequent response at least three revolutions after the blade release. These are expressed in terms of percent variation of the calculated overshoot from the measured data, and percent variation of the calculated from the measured data just after the overshoot.

The ratios of maximum modal frequency to maximum rotational frequency that were evaluated were 18.6, 9.6, 4.2, and 1.6. For the first three ratios, essentially no change in overshoot or following transient is noted. For the ratio 1.6, substantial changes in both occur. These results lead to several conclusions: 1) truncation error was not the cause for the discrepant initial overshoot observed, 2) a ratio of maximum modal frequency to maximum rotational frequency of about four will give reasonably good results, and 3) the predicted overshoot is probably due to a mechanism not included in the analysis.

To try further to sort out the overshoot problem, the effect of damping was evaluated. A case was run in which the modal damping was doubled. This had no significant effect on the initial overshoot spike.

Since the deadband effect is the single most important factor influencing the response, it is felt that the discrepancy is related to the idealization used. The use of a more sophisticated load-deflection curve may be required to represent the rotor blade-to-casing rub phenomenon. Nonetheless, the results with the current model are quite good, and agree well with measured results except for the overshoot.

A comparable study also was conducted for a much more complex complete engine system, but with less favorable correlation between test and prediction. However, available resources did not permit as detailed an investigation as was reported here.

Conclusions

In conclusion, the method developed is both fast and economical and the accuracy is user dependent. The truncation error can be minimized by keeping the ratio of the maximum modal frequency to the maximum rotational frequency greater than four. Except for the initial overshoot, good agreement is obtained when compared with test. The overshoot is probably due to an imperfect rub model used in the analysis. The approach described here is a practical and effective method for predicting the three-dimensional nonlinear transient response of large systems.

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