

## Analysis of Vibration Characteristics and Dynamic Load Identification in Continuous Miner Cutting Arm

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**Abstract** — Sensor disposition optimization based on modal analysis is made to analyze the vibration characteristics and to identify dynamic loads of a continuous miner cutting arm in the working condition, and results obtained of the modal movement energy as the main reference and results of modal strain energy as the auxiliary reference. The statistical parameters of dynamic vibration characteristics were obtained and vibration behavior was described according to vibration signals in the field test of continuous miner cutting arm. Then dynamic loads of cutting arm on key measurement points on different conditions were identified using integration superposition method in time-domain. The results showed that vibration amplitude in articulated position of motor and cutting drum and in the articulated position of motor were more than the others. Moreover, response of cutting conditions in different points were also verified. Vibration characteristics and dynamic load identification of continuous miner cutting arm provide a theoretical basis for further development of load spectrum and optimization in structural design.

**Keywords** - Continuous Miner, Vibration Characteristics, Load Identification, Sensors Disposition Optimization

### I. INTRODUCTION

Mechanical vibration is a process in which displacement amount of a certain observation point in the machine changes over time around the mean or relative reference when mechanical equipment is in operating state. Zhongbin Wang[1] proposed a novel approach through integration of improved particle swarm optimization and wavelet neural network in order to accurately identify the change of shearer cutting load. E. Mustafa Eyyuboglu[2] also conducted load tests on roadheader to determine the optimal structure form. Vibrations on the continuous miner could be caused by random working load, assembly error, and non-uniform distribution of picks on the roller on the circumference, etc. And they had a serious impact on the performance of the machine's life and working performance. Because cutting arm is the main working load-bearing parts of continuous miner, vibration testing and analysis for the cutting arm would lay the foundation for improving the working performance of the machine as well as dynamic design.

Dynamic load identification technology in terms of heavy-duty mining equipment has been currently in the initial research stage[3-6]. Traditionally, dynamic load identification was divided into two categories: frequency domain method and time domain method. The basic idea of time domain method is to establish vibration differential equation of the system in the time domain and identify the system load by calculating the natural frequency, damping ratio and signal superimposing of displacement, velocity, acceleration. With the continuous development of engineering signal processing and pattern recognition methods, many methods, such as wavelet analysis, fractals, neural networks, have become the dynamic load identification methods. In the specific study on load identification methods, Kaminski M. [7] analyzed the

frequency domain method and time domain method to identify the load by response data, who pointed out that instability of the inverse matrix caused by rank deficiency of the frequency response function at individual frequency and response measurement noise were the main problems on load identification. Watanabe H [8] pointed out that the main reason for slow development of load identification technology was measurement noise, sick near natural frequency and measurement error of the frequency response function. For these reasons, the authors used integration superposition method in time domain to identify vibration loads of system, which based on inherency frequency of vibration and estimated damping ratio of continuous miner.

### II. SENSORS DISPOSITION OF CONTINUOUS MINER OPTIMIZATION

In order to obtain a full range information of equipment operating status, sensors were usually arranged in the key point of core components and points which were prone to deterioration phenomena in the device, which could ensure the effectiveness of the vibration signal measurements. Disposition optimization guidelines currently have developed many kinds, such as error minimizing recognition criteria and interpolation fitting criteria, etc. Methods for sensor disposition optimization include genetic algorithm and simulated annealing algorithm, etc.[9-10]. The authors used the ADAMS software to optimize sensor placement for continuous miner cutting arm. The simulation modal parameters and modes for the part could be got through software emulation before testing, which covered the deficiency of previously conventional method of determined empirically measuring points.

The physical map of EML340 continuous miner was as shown in Figure 1. Because the frame, track frame and rear tailstock was a rigid bolt connection, it was treated as a

whole rigid body and fixed with the base. Racks, tanks and cutting arm, cutting arm and cutting drum were flexible couplings. So measuring points arrangement of sensor on flexible cutting arm was the main consideration. Simultaneously, sweep excitation was simulated in virtual prototype. Therefore, settings for each virtual measurement points of sensors were as shown in Figure 2. Then vibration characteristics and vibration response of each measuring point of the model were calculated.

Undamped inherency frequency and damping ratio of system were shown at Table 1. Cutting drum was subject to the low-frequency excitation in the cutting process, considering the first six order modal.

$$[Q] = [q_1(n), \dots, q_i(n), \dots, q_n(n)]^T \quad (1)$$

As shown in equation 1, modal coordinates could be represented as weighting factor of each order modal shape in the displacement response vector.



Figure 1. EML340 continuous miner

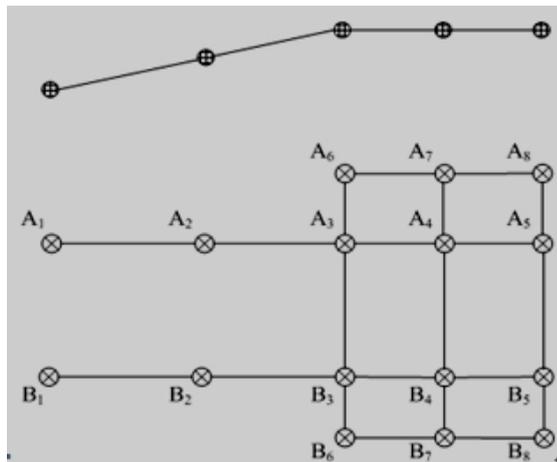


Figure 2. Initial Sensor Disposition on Cutting Arm.

TABLE 1 Inherency frequency and damping ratio of machine.

Modal order	Undamped natural frequency(Hz)	Damping ratio
1	4.777255E+000	1.965258E-001
2	8.399263E+000	2.333704E-001
3	2.226598E+001	3.690606E-001
4	2.834237E+001	4.355037E-001

5	4.948740E+001	3.009011E-001
6	8.255728E+001	3.404916E-001

representing the contribution of the *i*-th mode shape made. The first six modal coordinates were obtained that the second-order modal shared largest proportion. And the second-order modal shape showed cutting arm and cutting drum bobbing and was consistent with the load condition in actual working process.

Figures 3 and 4 were modal kinetic energy diagram and modal strain energy diagram for each measuring point of the second-order modal. We could clearly see the distribution of kinetic energy and strain energy from them.

Energy proportion table of each placement measuring point for 1-6th order modals was as shown as Table 2. Taking symmetry of the model into account, here listed only value of  $A_i (i = 1 \sim 8)$ .

Percentage superposition of energy of each measuring point at each modal at Table 2 was cumulated. It could be seen that the measuring points

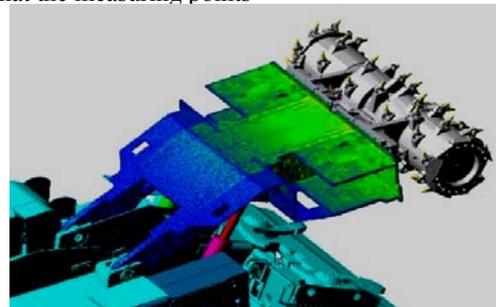


Figure 3. Kinetic energy diagram of second-order modal of machine

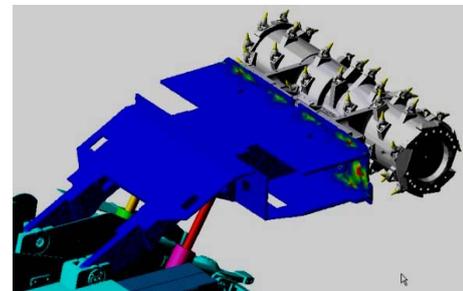


Figure 4. Kinetic energy diagram of second-order modal of machine.

whose proportion was from large to small were A1, A8, A5, A6, A4, A7, A3, A2 for modal kinetic energy and measuring points A1, A5 and A8 were the most important for modal strain energy. Then disposition for measuring points was optimized with results of the modal movement energy as the main reference and results of modal strain energy as the auxiliary reference. We selected A1, A6, A8 and symmetrical measuring points B1, B6, B8 as the experimental arrangement position of the sensors. Optimized measuring points disposition was as shown in Figure 5. Vibration characteristics could be reflected completely and vibration behavior could be described by the optimized measuring points in the frequency domain. Finally, direction and the channel list of the sensors were as shown at table 3. Then we arranged with the X and Y directions of A1, the Y

and Z directions of A8, and the Y direction of A6, for the structure was symmetrical.

TABLE 2 Proportion of energy placement measuring points for the second-order modal.

Measuring point		A1	A2	A3	A4	A5	A6	A7	A8
1	KE	1.9	1.2	3	8.3	12	3.8	7.4	12.5
	MSE	42.5	0	0	0	0	0	0	7.5
2	KE	2.4	0.6	4.1	9.7	12.8	3.4	6.3	10.6
	MSE	16.1	0	0	0	7.6	0	0	26.3
3	KE	13.5	5.2	4.5	3.7	2.5	10.9	4.9	4.8
	MSE	9.1	0	0	3.1	2.9	0	0	34.9
4	KE	15.5	3.9	3.5	6.3	8	2.5	3.7	6.5
	MSE	0	0	0	0	30.5	0	0	19.5
5	KE	29.8	3.8	1.6	2	2.3	1.5	3.2	5.7
	MSE	0	0	0	0	0	0	0	50
6	KE	37.4	1.6	0.8	2.5	3.4	1	0.7	2.6
	MSE	0	0	0	0	50	0	0	0

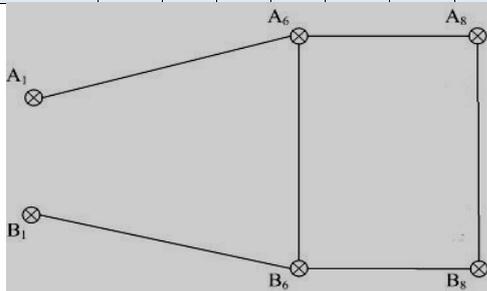


Figure 5. Optimized sensor disposition of cutting arm

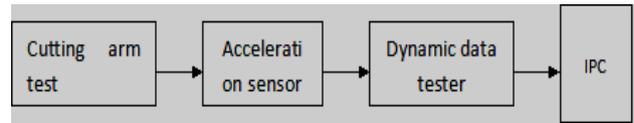


Figure 6. Test system figure

TABLE 3 Disposition direction and channel number of sensors.

Measuring point	Sensor arrangement direction	Acquisition channel number
A1	-X	1
A1	Y	2
A6	Y	3
A8	Y	4
A8	Z	5

### III. VIBRATION CHARACTERISTICS ANALYSIS OF CUTTING ARM

*A. Test and Data Extraction.* Because accelerometer could be made very small, it could be

attached to the measured surface taking advantage of fixed ways such as magnetic, adhesive or screws, etc.. Therefore, it was easy to use and the disturbance caused to the measured system was small. So the acceleration sensor was chosen as the vibration test sensor for cutting arm. Besides, a MDR dynamic data tester was also needed. The test system was as shown in Figure 6.

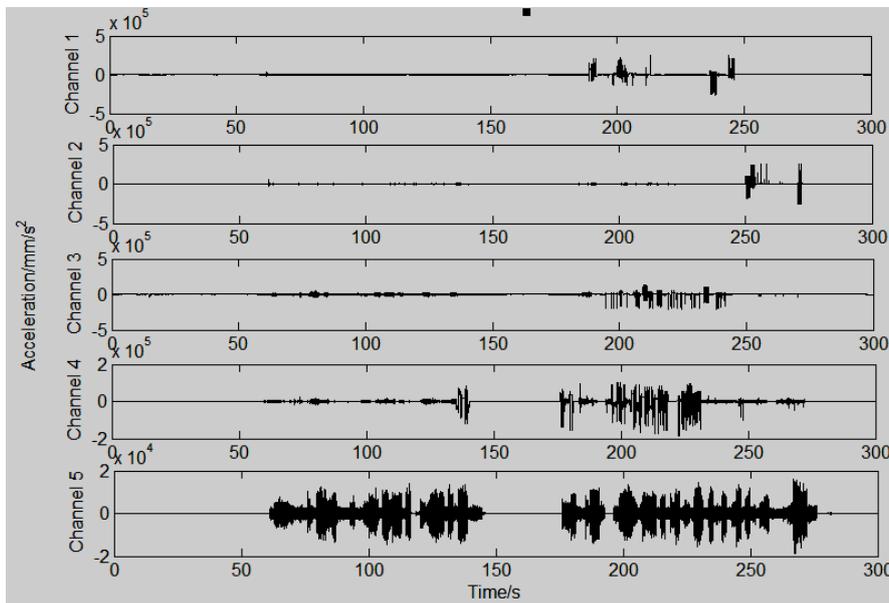


Figure 7. Acceleration time-domain waveform.

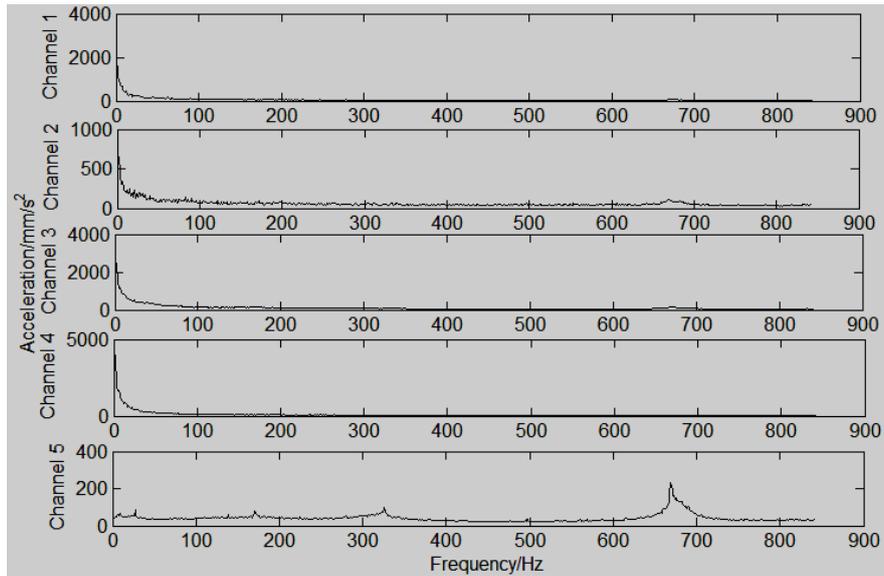


Figure 8. Acceleration frequency domain waveform.

The test condition according to the actual work of continuous miner was determined in chronological order: Cutting start → Shipment start → Body position adjust → Cutting down → Cutting up → Cutting down → Cutting up → Body position adjust → Shipment stop.

Signal pretreatment could make the averaging process for the original signals. It could improve the signal-to-noise ratio, effectively reduce noise interference on the signal and facilitate to reach the right results using limited data by electing the signals in appropriate frequency range, or windowing to prepare for subsequent signal analysis. Specially, the ways of eliminating abnormal points and Butterworth filter were used in the continuous miner testing.

Data processing of random vibration waveform and deterministic waveform were different. The characteristics of the former could not be described by single peak and signal amplitude at a certain time, and the probabilistic method should be used for analysis and processing. To facilitate the analysis, it was often assumed that these random data were ergodic. Finally, acceleration time-domain waveform and frequency-domain waveform of each channel were as shown in Figures 7 and 8.

*B. Test Analysis.*

(1) From the statistical characteristics of the load in Figure 8, we could find that the vibration of A1, A6 and A8 in the Y direction was intense, while vibration of A8 in Z direction was less intense. This was because A1 and A6 were in a closer distance to cutting drum, the force of coal seam on the cutting arm was relatively larger than measuring point A8.

(2) From time course of each measuring point of load, we could find this: when the shipping motor started, vibration loads were small; when the cutting motor started, vibration loads increased rapidly; besides, shipping motor had a large impact on A8 in the Z direction, while it had little effect on other measurement points. This showed that

the vibration of A8 was mainly due to combination of shipping motors and cutting motor, while A1 and A6 were more affected by the cutting motor.

(3) From the frequency domain of each measuring point, we could find this: operating frequencies of A1 and A6 were small, which was consistent with the foregoing analysis on inherency frequency; while A8 in direction Z had a remarkable resonance frequency at 669.1Hz. After a preliminary analysis, the frequency was close to the fifth order frequency of the inherency frequency of shipping motor. In order to verify this analysis, it was needed to make coherent analysis for these two frequencies. We did not repeat them here.

IV. DYNAMIC LOAD IDENTIFICATION OF CUTTING ARM

In this part, collected data of measurement points on the continuous miner cutting arm in the whole no-load testing process was processed. Then integration superposition method in time-domain was used to identify dynamic load of measurement points in key components.

*A. Meaning of measuring point load.*

Loads of continuous miner discussed in this paper were not the internal forces of measuring points, but dynamic loads causing the vibration of measuring points in some coordinate direction. It was the resultant of internal forces  $F_1(t)$ ,  $F_2(t)$  in this direction. Take for instance the measuring point was in tension. As shown in Figure 9:

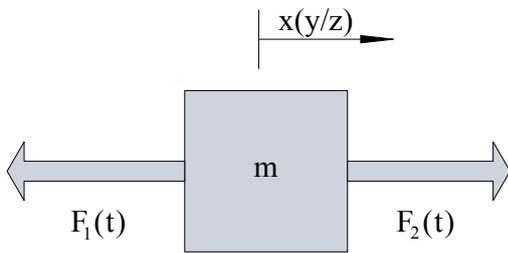


Figure 9. Stress analysis chart of measuring point in electric control boxes, travel unit.

Symbol meanings:

m: Measuring point (It was deemed as a particle);

$F_1(t), F_2(t)$ : Internal forces of measuring point in the measuring direction ;

$x(y/z)$ : Measuring direction, x-axis direction, or y-axis or z-axis direction.

*B. Integral Superposition in Time Domain.*

Each measuring point of continuous miner was regarded as a particle and was isolated from the continuous miner by isolation method. Because what each acceleration sensor measured was vibration of measuring point in a certain direction. The formula obtained by the basic theory for forced vibration of freedom system of a single degree could be written as:

$$m \times \ddot{x} + c \times \dot{x} + k \times x = F(t) \quad (2)$$

Symbol meanings:

m: Quality;

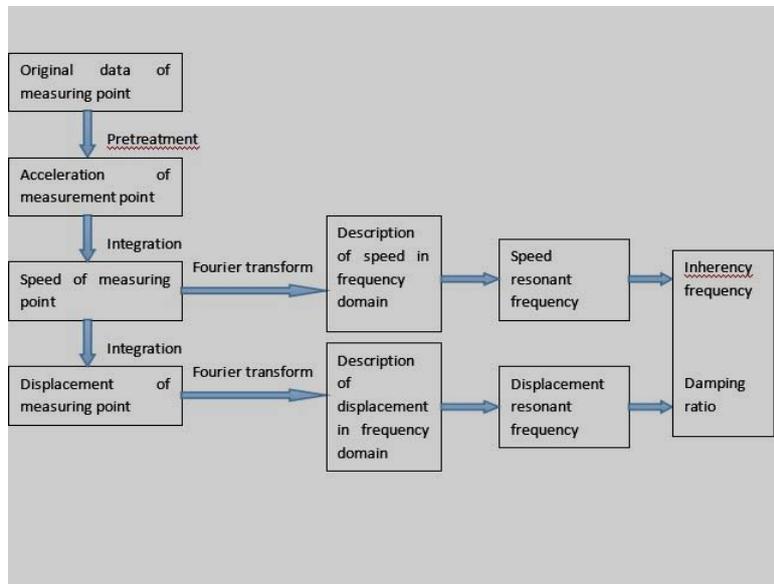


Figure 10. Calculation process of inherency frequency, damping ratio.

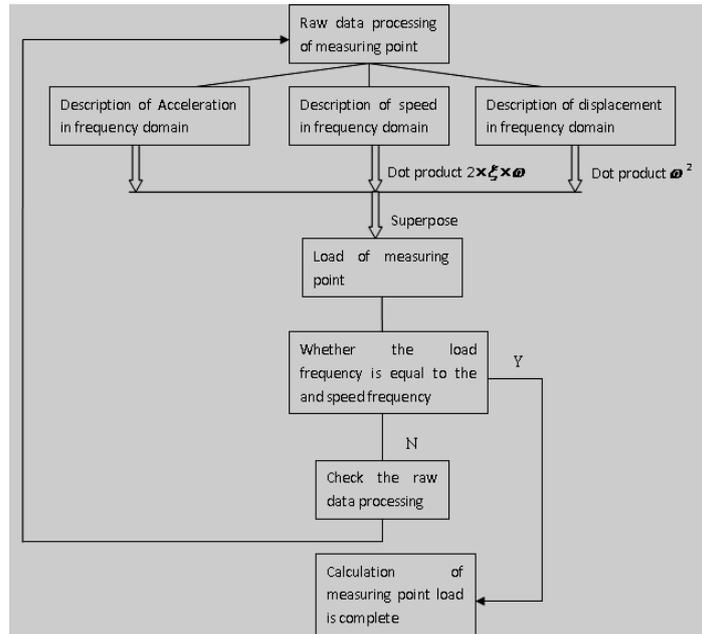


Figure 11. Calculation process of measuring point loads of electronic control box,travel unit.

c: Damping;

k: Stiffness coefficient;

F (t): Excitation force of the rest of continuous miner on the measuring point

t: Time;

$\ddot{x}$  ,  $\dot{x}$  , x: Acceleration, velocity, displacement of measuring point.

The formula obtained by the relationship of natural frequency, damping ratio with mass, stiffness, damping could be written as :

$$\xi = \frac{c}{2 \times \sqrt{m \times k}} \quad (3)$$

$$\omega_n = \sqrt{\frac{k}{m}} \quad (4)$$

The formula obtained by the formulas 2-4 could be written as :

$$\ddot{x} + 2 \times \xi \times \omega_n \times \dot{x} + \omega_n^2 \times x = \frac{F(t)}{m} \quad (5)$$

Make  $m = 1$ , the above formula could be written as:

$$\ddot{x} + 2 \times \xi \times \omega_n \times \dot{x} + \omega_n^2 \times x = F(t) \quad (6)$$

Calculation of the natural frequency and damping ratio: the inherency frequency was decided by the mass  $m$  and stiffness  $k$  of the measured object. However, it had a certain relationship with the resonance frequency of displacement signal, velocity signal, acceleration signal of the measured object, that was the frequency where maximum amplitude of the signal was. The relationship of displacement circular frequency, speed circular frequency and the inherency frequency were as following:

$$\omega_v = \omega_n \quad (7)$$

$$\omega_r = \omega_n \sqrt{1 - 2\xi^2} \quad (8)$$

The relationship of circular frequency and frequency could be written as:

$$\omega = 2 * \pi * f \quad (9)$$

Symbol meanings :  $\omega$  :circular frequency;

f: frequency.

Firstly, speed signal was obtained by integral of acceleration signal for each measuring point in time domain, and the displacement signal was obtained by integral of the velocity signal for each measuring point in time domain. Secondly, description of the speed, displacement was converted from time domain to frequency domain. Speed resonant frequency  $f_v$  and displacement resonant frequency  $f_r$  were read out from spectrum of velocity and displacement of each measuring point. Thirdly, inherency frequency and damping ratio of each measuring point were calculated by formulas 7 to 9. Then, the velocity signal and the displacement signal of each measuring point were magnified respectively by  $2\xi\omega$  and  $\omega^2$  times. Next, load of each measuring point was got by superimposition of the acceleration signal, the speed signal, the displacement signal. Finally, the frequency of load of measuring points was compared with speed frequency. If the two were basically the same, load calculation was complete. If not, raw data of measuring point was reprocessed.

The process of load calculation were as shown in Figures 10 and 11.

C. Load Identification of Cutting Arm.

Load characteristics of each measuring point calculated by the data processing were as shown at table 4. We could find this: the maximum load occurred in the X direction of A1 , and the minimum load occurred in the Z direction of A8; maximum standard deviation occurred in X direction of A8, which indicated that it suffered a large dynamic component loads; inherency frequency and results of simulation analysis were approximate.

V. CONCLUSION

Structure optimization of continuous miner was based on vibration tests of cutting arm, and vibration of system could be described quantitatively by statistical analysis for test data. Load distribution of continuous miner cutting arm in different locations and associated characteristic parameters were obtained by identification of natural frequency and damping ratio of vibration, which laid the foundation for the further establishment of the load spectrum.

TABLE 4 Load tables of measuring point.

Measuring point	Direction	Maximum load (N)	Standard deviation(N)	Inherency frequency (Hz)	Damping ratio
A1	-X	252.6	10.9	2.7	0.67
	Y	100.8	6.4	3.3	0.67
A2	Y	125.4	16.7	1.9	0.57
A3	Y	102.4	17.3	5.7	0.68
	Z	16.2	1.7	669.1	0.68

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