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Procedia Structural Integrity 5 (2017) 1326–1333

Structural Integrity Procedia

www.elsevier.com/locate/procedia

2nd International Conference on Structural Integrity, ICSI 2017, 4-7 September 2017, Funchal, Madeira, Portugal

Fatigue damage in spline couplings: numerical simulations and experimental validation

Francesca Curà[a]*, Andrea Mura[a], Federica Adamo[a]

[a]Politecnico di Torino, Department of Mechanical and Aerospace Engineering, C.so Duca degli Abruzzi 24, Torino 10129, Italy

Abstract

Spline couplings are often over dimensioned concerning fatigue life, but they are subjected to wear phenomena. For as concerns fatigue life, standard design methods consider only a part of the spline teeth to be in contact and this brings to underestimate the components life, so a better understanding about component fatigue behavior may allow to a weight reduction and a consequent increasing of machine efficiency. On the other hand, wear damage may cause spline coupling run outs; this phenomenon is generally caused by the relative sliding between engaging teeth; the sliding may be due to kinematic conditions (angular misalignment between shafts) of teeth deflection. In order to obtain component optimization, both fatigue and wear behavior have to be taken into account. Standard spline coupling design methods do not properly consider wear damage and they evaluate fatigue life with big approximations. In this work fatigue damage are experimentally and numerically investigated while wear damage has been experimentally evaluated. Experimental results have been obtained by a dedicated test rig. Fatigue tests have been performed by means of a special device connected to a standard fatigue machine. Tests have been done by varying the most important working parameters (torque and misalignment angle). Experimental results have been compared with standard design methods to evaluate if and how they may over dimension the components. Results show that concerning the fatigue life, the actual component life is higher respect to that calculated by standard methods. Regarding wear behavior, results shows that whenever a relative motion between engaging teeth is present, wear damage appears.

Peer-review under responsibility of the Scientific Committee of ICSI 2017

Keywords: Spline coupling; fretting; wear; fatigue; test rig

* Corresponding author. Tel.: +39 011-0906930; fax: +39 011-0906999.
E-mail address: andrea.mura@polito.it

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Peer-review under responsibility of the Scientific Committee of ICSI 2017
10.1016/j.prostr.2017.07.141

1. Introduction

One of the most common way to connect two rotating shafts, where high power has to be transmitted, is to use spline couplings. These components transmit motion by means of a certain number of engaging teeth, that are subjected to both fatigue and wear caused by variable amplitude loadings and relative sliding.

Generally, in industrial transmissions, splines transmit more torque for their size than any other type of coupler or joint and, if they operate with a relative small shaft diameter, a common kind of failure in this case is due to the shaft shear stresses.

In case of high power transmissions, as requested in recent decades due to the increase in machines performance, shaft and hub are designed from both static and fatigue point of view, by means of classical formula, and the problem of sudden failures due to shear stresses is practically non-existent.

For as concerns fatigue life, standard design methods consider only a part of the spline teeth to be in contact and this brings to underestimate the components life, as an example in Dudley (1957) approach. Dudley approach involves the calculation of shear stresses in spline shaft and in spline teeth, compressive stresses and bursting stresses in internal spline parts.

More recently DIN Standards (2000-2006), based on Niemann et al. (2006) studies, provide design methods based on the calculation of the contact pressure on spline teeth in case of maximum, nominal and equivalent torque respectively, compared to the corresponding admissible pressure expressed in terms of limit values for the material, corrected by influence parameters and safety factors.

Referring to high power transmissions, generally spline couplings are not critical in terms of fatigue behavior, because they are carefully designed due to the necessity to a weight reduction and a consequent increasing of machine efficiency. So, about this topic, the more interesting question is: how do splines fail?

The answer is very simple, wear damage may cause spline coupling run outs; this phenomenon is generally caused by the relative sliding between engaging teeth, due to kinematic conditions (angular misalignment between shafts) or teeth deflection caused by variable amplitude loadings.

The causes of spline wear, discussed in detail by Ratsimba et al. (2004), can be summarized as an inability of the coupling to adequately accommodate misalignment, a difficulty in maintaining sufficient lubrication, and a basic susceptibility to the process of fretting. Then, there is some motion between teeth which makes them vulnerable to wear.

As discussed before, traditional spline couplings design methods allow to perform static and fatigue dimensioning, but do not take properly into account the effect of wear.

A very interesting study related to the application of spline couplings used in modern wind turbine gearboxes to connect planetary and helical gear stages has been carried on by Guo and al. (2013); through the developed model, a greater understanding of the behavior of spline connections has been achieved and recommendations to improve design standards have been provided.

The primary necessity to obtain the optimization of these components is that both fatigue and wear behaviors have to be taken into account.

Standard spline coupling design methods don't properly consider wear damage and they evaluate the fatigue life with strong approximations. For this reason, a better understanding of both fatigue and wear phenomena on spline couplings has to be pursued to develop better design practices.

The heart of this research is to analyze how fretting wear damage may arise from two different causes related to two different working conditions. More in detail, fretting wear may come from misaligned working conditions, when shaft and hub run with an angle between the corresponding rotation axis. But they may be subjected to wear damage even if they work in perfectly aligned conditions, due to the flexibility of teeth, in case of strongly variable torque and then in case of classical fatigue damaging.

So, the first aim of this work is to compare fretting wear phenomena coming from two completely different behaviors.

In particular, in this work the fatigue damage has been numerically and experimentally investigated, while wear damage has been experimentally evaluated.

Experimental results have been obtained by a dedicated test rig that allows to perform wear tests on component angularly misaligned. Fatigue tests have been carried on by means of a special device connected to a standard fatigue machine. Tests have been done by varying the most important working parameters (torque and misalignment angle). It is important to highlight that also after fatigue tests wear damage appear, this is due to the relative sliding caused by tooth deflection.

Experimental results have been compared with standard design methods to evaluate if and how they may over dimension the components.

2. Numerical models

FEM simulations have been performed by means of Abaqus software, to calculate contact pressure, teeth sliding and stress state of an involute crowned teeth spline coupling. The spline coupling considered in this work has the following geometrical parameters: number of teeth z=26, modulus m=1.27mm, pressure angle α=30°, mean radius of the shaft R_m=16.51mm, length width L=12.5mm, tooth contact height h_w=1.63 mm, ρ_{max1}=0.122mm and ρ_{max2}=0 are the maximum curvatures for respectively shaft and hub, ρ_{min1}=0.0014mm and ρ_{min2}=-0.120 are the minimum curvatures for respectively shaft and hub.

The component is made of 42CrMo4 steel (tensile stress R_m = 1000MPa, yield stress R_{P02} = 700 MPa, fatigue limit σ_{D-1} = 420MPa, Young modulus 210 GPa, 0.3 Poisson coefficient).

The component has been modelled with second order tetrahedral solid elements obtaining a total of 480772 elements (Fig. 1). The element size varying from 2mm to 0.15 mm on the contact zones. Elements have been defined creating contact surfaces characterized by a friction coefficient of 0.11.

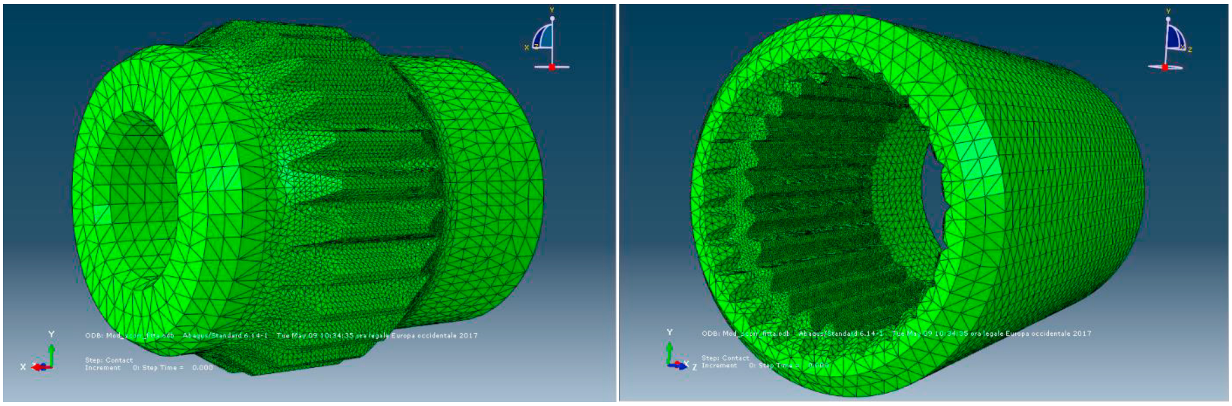

Fig. 1. Numerical model of the spline coupling: shaft (left) and hub (right).

Displacements on all directions have been blocked on one of the two extremities of the hub. On the opposite extremity on the component, a torque has been applied about the shaft axis (Fig. 2). Numerical simulations have been run with different torque values: 200 Nm and 700 Nm. These values correspond to the minimum and maximum load applied on the real component during experimental tests described in the next section.

3. Theoretical models

Theoretical models taken into account in this work are the well known Hertzian theory (Giovannozzi (1965)) and the standard design formula related to the experience of Niemann et al. (2006) and to the DIN 5466-1 (2000).

Pressure values coming from the Hertzian theory (and related to both materials and geometrical characteristics already reported in Section 2., Numerical models) consist in maximum and mean pressure values and the corresponding contact area entity. These value substantially correspond to those calculated by FEM simulations from the physical point of view.

For as concerns standard design formula, the equation for calculating the mean pressure between teeth in an involute spline coupling is the following:

$$p = \frac{T}{zLh_w R_m} k_{\varphi\beta} k_l \quad (1)$$

where p is the contact pressure (no crowning radius is taken into account in this approach), T the applied torque, $k_{\beta\varphi}$ and k_l respectively the load sharing and longitudinal factors.
This pressure value p (that may be calculated from a maximum or nominal or equivalent torque) has to be compared to the admissible pressure related to the material characteristics.
In general, the admissible pressure is obtained by static characterization of the material, corrected by parameters or safety factors related to the required number of cycles.

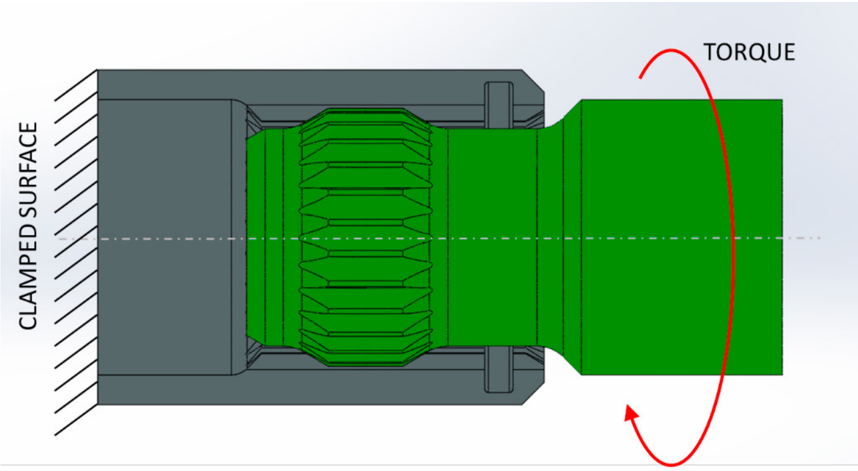

Fig. 2. Model boundary conditions and loading.

4. Experimental set up

Experimental tests have been performed in order to reproduce real working condition of the components. In particular, a dedicated device (Fig. 3a) has been designed in order to allow testing the component with variable amplitude torque with a standard fatigue machine (Figure 3b).

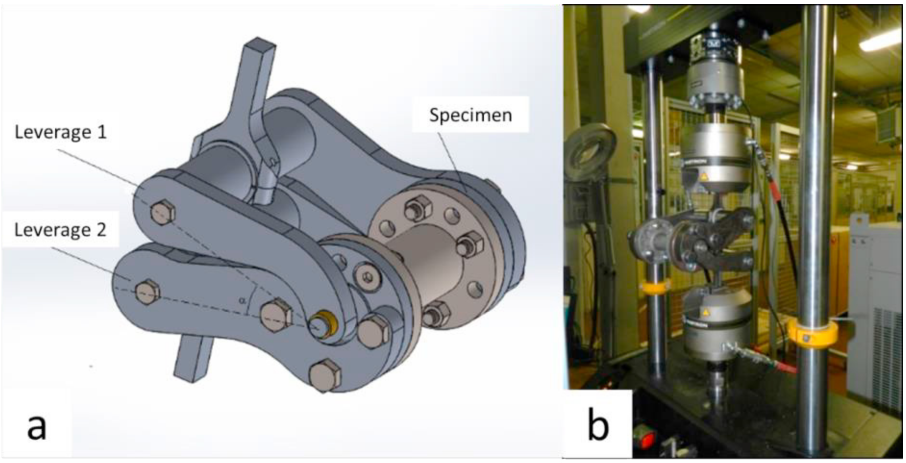

Fig. 3. Testing device to apply torque to the spline coupling (a) and fatigue testing machine (b).

The testing device (Fig. 3a) consists in two couples of leverages connected to the component to be tested. The leverages are then fixed on the fatigue machine clamps and allow to transform the axial force generated by the fatigue machine to a torque. The geometries of the spline coupling and the loading conditions are the same as described in the previous section and related to numerical simulations.

The test has been run with a loading frequency of 10 Hz and during 9x10^6 cycles.

5. Results

Fig. 4 shows an example of numerical results in terms of distribution of contact pressure along the teeth.

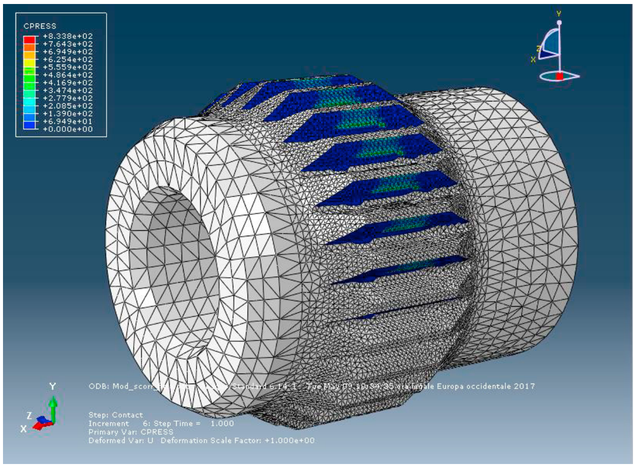

Fig. 4. Example of simulation results in terms of contact pressure.

The contact pressure distribution on the contact surface is best displayed in Fig. 5. On the left it can be seen the distribution in case of 200 Nm of applied torque, while on the right that in case of 700 Nm.

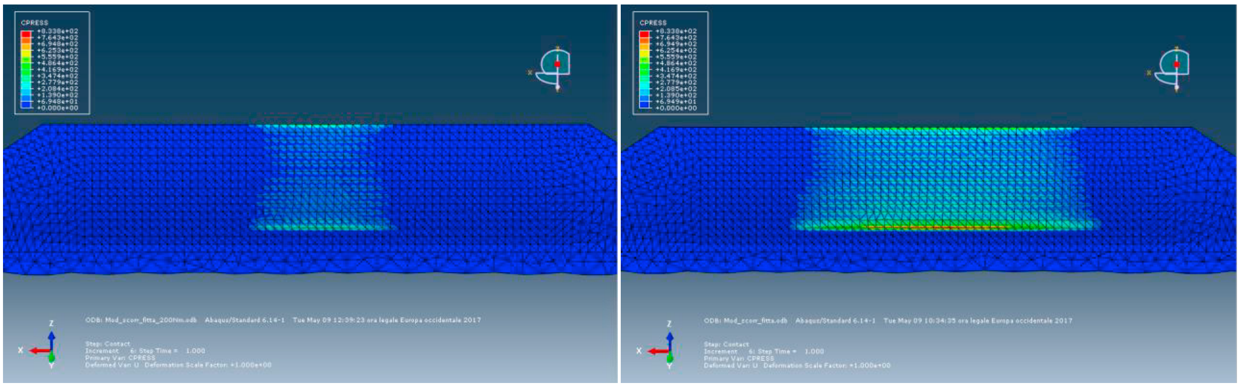

Fig. 5. Contact pressure distribution on contact surface for different values of torque: 200 Nm (left) and 700 Nm (right).

Fig. 6 shows the distribution of the relative tangential motions (slidings) of the surfaces during contact, respectively along x local tangent direction (up) and y local tangent direction (down) for both models.

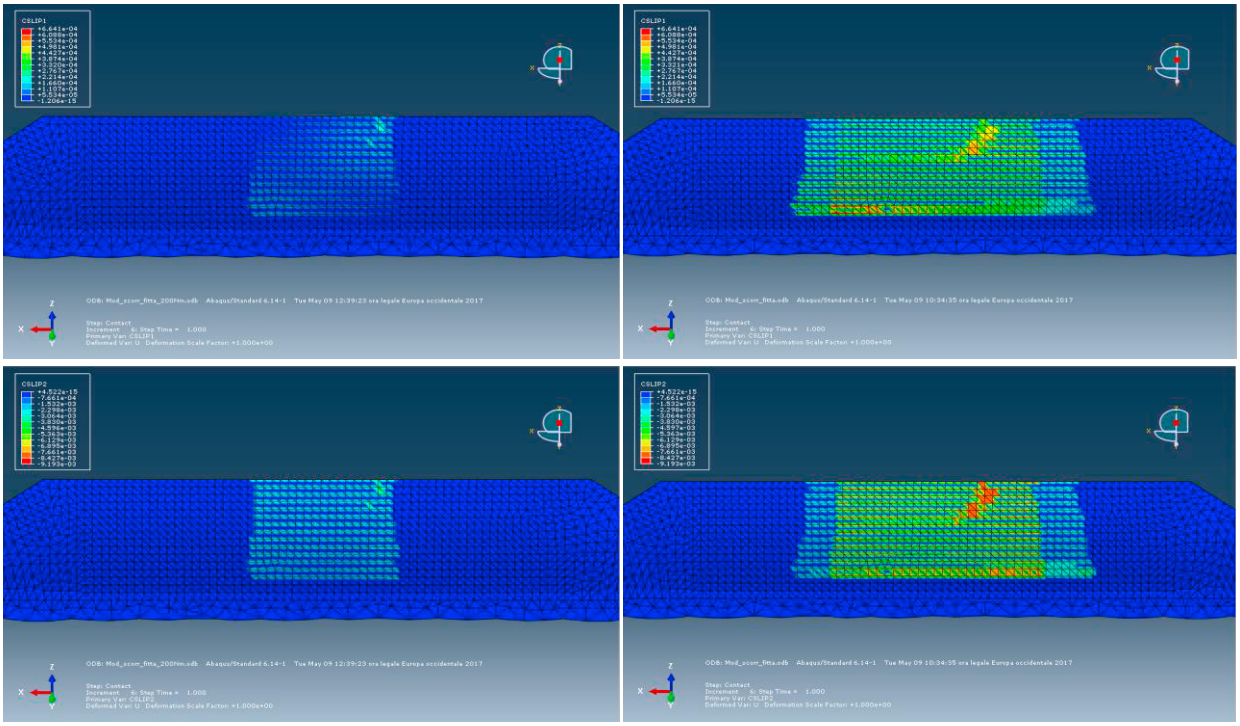

Fig. 6. Slidings on contact surface for different values of torque: 200 Nm (left) and 700 Nm (right).
Relative tangential motion in x-direction (up) and y-direction (down).

Table 1 resumes the parameters values calculated from numerical simulations, Hertzian theory and Niemann approach.

In particular, for minimum and maximum applied torque values (200 and 700 Nm, first column), columns 2, 3, 4 report the results related to FEM analysis, columns 5, 6, 7 show the results obtained by Hertzian calculations and column 8 refers to Niemann formula (Equation (1)).

Table 1. Results of FEM simulations and Hertzian calculations.

	FEM			HERTZ			NIEMANN
Torque [Nm]	Contact area [mm^2]	p,mean [MPa]	p, max [MPa]	Contact area [mm^2]	p, mean [MPa]	p, max [MPa]	p [MPa]
200	2.69	179.8	370.8	5.66	95.0	142.5	30.3
700	6.05	275.0	833.8	13.05	144.3	216.4	106.1

Referring to Equation (1), values for involved parameters are: $k_{\beta\varphi}$=1.02 (due to geometrical dimensions of shaft and hub), k_l=1.3 for H8/IT8 tolerances (see DIN 5480-1 (2006)).

Calculated pressure p has to be compared with the admissible pressure coming from the material characteristics. Admissible pressure may be obtained from the yield stress, corrected by the coefficient f_s (see DIN 6892 (2012). In this case f_s=1.2; so, the permissible pressure value is 840 MPa. This value may be then corrected by a durability factor, f_L, taking into account the expected number of cycles of the component.

In this case of study, the loading conditions produce so low pressure values that the component may be considered as in infinite life condition.

If FEM results are analyzed, it may be observed that maximum contact pressure values are substantially higher than those calculated by theoretical models. In particular, maximum pressure conditions in both models (200 and 700 Nm) is localized in zones not close to the mean radius, but at the extremities of the contact area. This stress condition is due to the fact that no micro geometries have been considered in designing the specimen, allowing to translate the contact zone close to the mean radius.

Theoretical values of the contact pressure are more close each other. It can be noted that pressure p is related to straight teeth, while in the present case a crowning radius is present. Then, Hertzian pressure values refer to a very small contact area; on the contrary, Niemann formula considers the pressure quite constant along the whole tooth surface.

Table 2 resumes the values of stress at the bottom of the tooth and the slidings on x ($Sliding_1$) and y ($Slidings_2$) local tangent directions.

Table 2. Values of stresses and slidings on tooth surface.

T [Nm]	Stress [MPa]	$Sliding_1$ [mm]	$Sliding_2$ [mm]
500	183.8	0.0001911	0.003496
700	468.4	0.0006641	0.009193

(1)

Fig. 7 shows the spline coupling after $9x10^6$ fatigue cycles, emphasizing an evident zone of wear on the teeth surface. This damage is substantially due to the variation of the applied torque and to the corresponding slindings.

Slindings calculated by FEM simulations confirm this phenomenon caused by the deflection of teeth during the variable amplitude fatigue test.

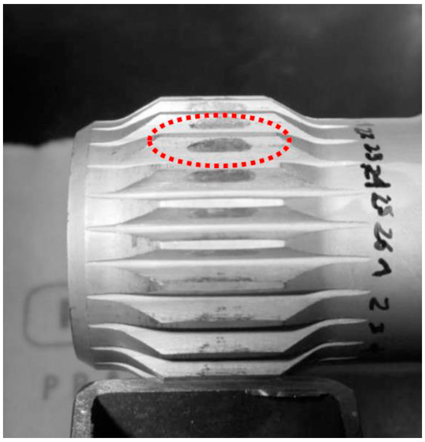

Fig. 7. Damaged zones due to wear.

Finally, in order to provide a global estimation of the wear damage, the variation of the angular rotation before and after the test has been measured (Cuffaro et al. (2014)). In this case, the percent variation is about 200%, while in case of angular misalignment of 10' between shaft and hub axis, with the same torque value (700 Nm, constant), the variation of the angular rotation is about 14% (Curà et al. (2017)).

6. Conclusions

In this paper the wear damage in spline couplings has been investigated, focusing on the problem that this kind of damage is substantially neglected in traditional standard approaches.

This research activity was born from the experience of the research group about fretting damage in these components, due to misaligned working conditions between shaft and hub.

The basic idea was to analyze how fretting wear damage may arise from different causes, related not only to misalignments, but also due to the flexibility of teeth, in case of strongly variable torque, even if perfectly aligned conditions are present.

This study has been firstly carried on from both numerical and theoretical point of view in order to verify standard formula. Then, an experimental activity has been performed in order to achieve a better understanding about component fatigue behavior.

From the analysis of the obtained results, some conclusions can be drawn as follows.

Firstly, FEM analysis showed the necessity to realize micro geometries in splined couplings, in order to localize the damaged area in the middle zone of the teeth and to avoid pressure peaks at the extremities that may increase the surface damage.

Secondly, FEM analysis provided the entity of slidings that are not negligible and that are responsible of the wear damage due to flexibility of teeth, as experimentally verified.

The experimental activity has emphasized that this damage may cause spline couplings run out even if no fracture is present. As indicated by the variation of the angular rotation, a lot of material has been removed during the fatigue test performed in aligned conditions.

Finally, it may be concluded that standard formula provide stress states very approximated, leading to an expected fatigue life very overestimated.

The future goal will be to define a method not only for the analysis of fatigue parameters, but also based on wear damage conditions.

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