DYNAMIC ANALYSIS WITH GAPS

by

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ABSTRACT

During the 1988 MSC World User's Conference User Forum, a number of delegates addressed the issue of dynamic analysis of structures with variable connectivity or boundary conditions. In this presentation, a very effective and versatile alternative to using gap elements, where some difficulties have been reported, is illustrated using NOLIN "elements" in the dynamic analysis of industrial machinery.

A variety of nonlinear effects can be modeled by combining the NOLIN tables with linear springs and other elastic elements. The examples include results obtained using a modal model together with numerical integration.

1. INTRODUCTION

The design of a very large industrial machine was analyzed in MSC-NASTRAN to predict the potential for durability problems. At issue was the possibility that an interference could be introduced between two moving parts of the machine.

Due to confidentiality agreements, this paper does not include a detailed presentation and description of the machinery analyzed. For the same reason, the example results have been modified and serve for the purpose of illustration only. The paper thus merely concentrates on the approach and its advantages over "conventional" methods, rather than on an actual application.

There are many situations in Mechanical Engineering where interference or contact/no contact conditions may exist, such as in the case of cams in general, and the liftoff of parts held down by springs, or the introduction of foreign mechanical interferences into a moving mechanism, more specifically, where the techniques presented may be used.

One way to model problems of this nature in MSC-NASTRAN is to do a full non-linear transient analysis with gap elements in Solution 99. An alternate approach was used here, combining linear springs and NOLIN tables and performing the transient analysis on a modal model.

2. TYPES OF NONLINEARITIES

The simplest type of nonlinearity considered in the analysis is shown in Figure 1. Once part I (very stiff) gets into an interference position with part II (flexible) a force arises characterized by the spring constant K_S of the flexible part.

The corresponding curve (in terms as defined by NOLIN1 Tables) is depicted in Figure 2. Physically, this situation is equivalent to a gap element.

This type of condition may exist, e.g., in the case an interference of stiffness K_S is lodged between two rigid, moving parts of a machine (Figure 3).

A similar situation, e.g., that of a stiff part supported by a flexible bearing but that may lift off, can be idealized and modeled as a combination of a flexible spring and a NOLIN1 Table, as shown in Figure 4.

A more complicated situation is shown in Figure 5, where two stiffnesses are involved, those of a frame and of a spring, which is preloaded. This situation can be idealized and modeled as a combination of linear spring, NOLIN1 Table and load, as shown in Figure 6.

3. MODEL AND SOLUTION METHODS

The model consisted mainly of beams and rigid elements arranged in a semi-circle representing a stiff ring, which in nodal positions was "supported" by various elements and NOLIN Tables of the type shown. In one of the positions, an interference such as that of Figure 3 was included.

Principal natural modes of such a model were extracted and included for numerical integration to solve for the transients caused by the sudden introduction of an interference. In this way, very high-frequent modes of the flexible ring were eliminated, but the principal inertia characteristics of the same were duly represented.

Accordingly, Solution 72 (Modal Transient Response, Data Base) was used, instead of Solution 69 (Direct Transient Response).

4. TYPICAL RESULTS

Displacements and forces in selected parts, as caused by an interference to a rotating part, are shown in Figures 7 (a), (b) and (c). Note that actual results have been altered, both in magnitude and distribution.

5. CONCLUSIONS

It can be concluded that a modal model and NOLIN Tables can be combined very effectively for analyzing dynamic contact problems using Solution 72. Direct integration methods such as Sol. 69 give results that may include undesirable higher frequency mode effects. Similar situations may be expected from the more general Solution 99 (Transient Nonlinear Analysis) which in addition may be cumbersome, in particular when used in combination with gap elements. The stiffness of interferences modeled as NOLIN "elements" has an influence on the peakedness of the response, with higher stiffnesses giving sharper peaks in curves such as that shown in Figure 7(b). However, the area under these curves (impulse) is nearly the same for all cases. Care must be taken that relevant rigid body properties of the portions lifted by the interference are represented in the modal model, so that all inertia effects are duly included and reflected in the response forces obtained.

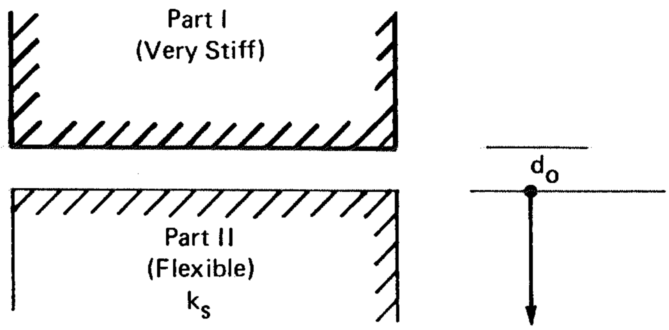

Figure 1
Simple Contact Condition

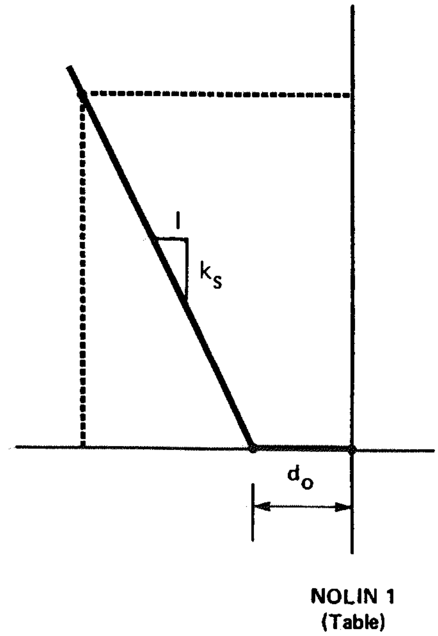

Figure 2
Contact Condition of Figure 1 in Terms of NOLIN1 Table

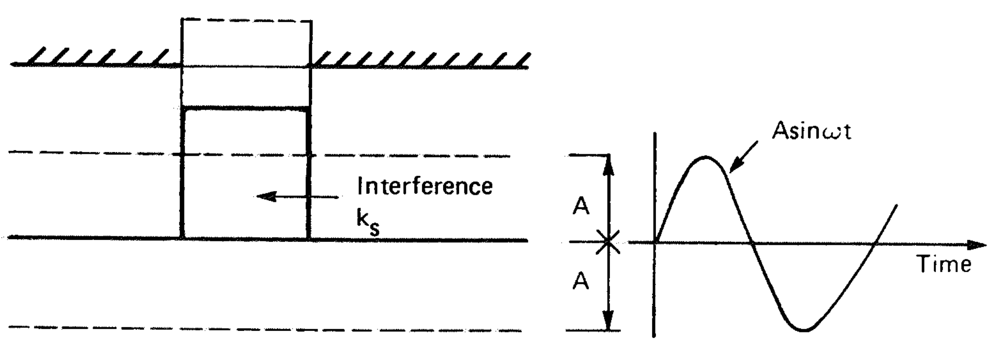

Figure 3
Contact Due to Interference Between Two Moving Parts

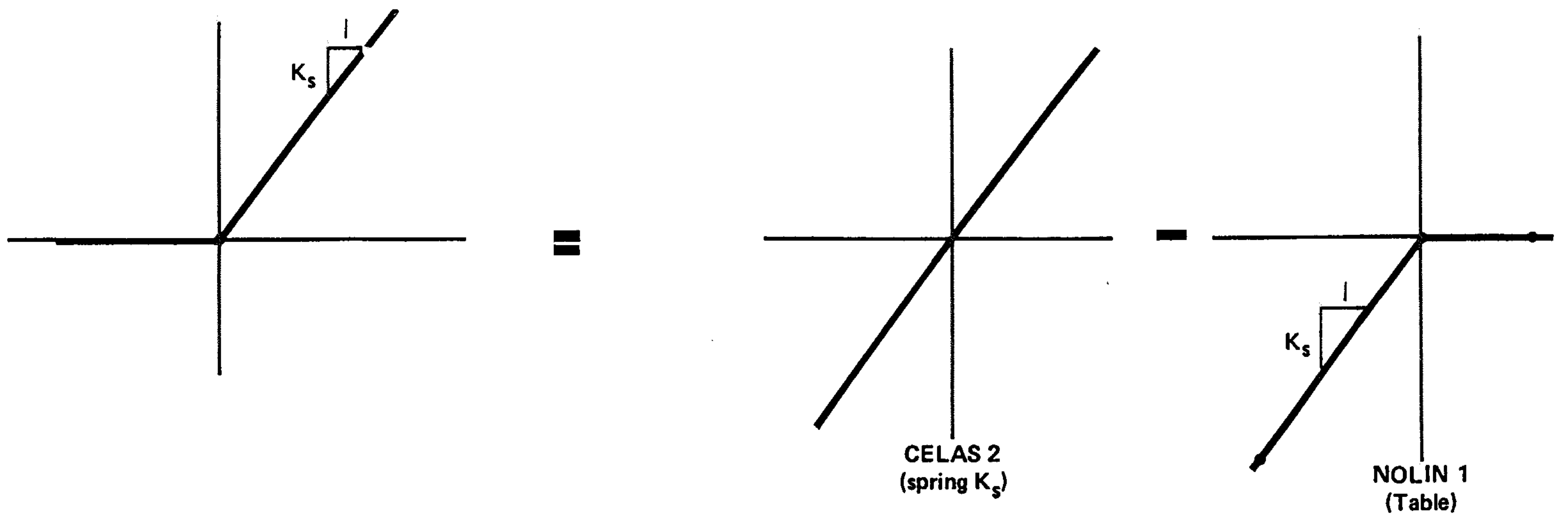

Figure 4
Modeling a Rigid Part Resting Loosely on a
Flexible Support (i.e. Compression Only)

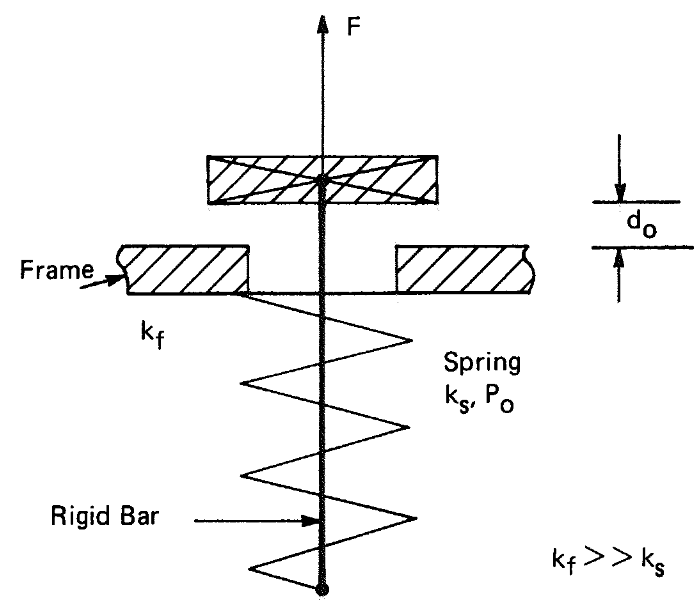

Figure 5
Contact Condition With Two Stiffnesses and Preload

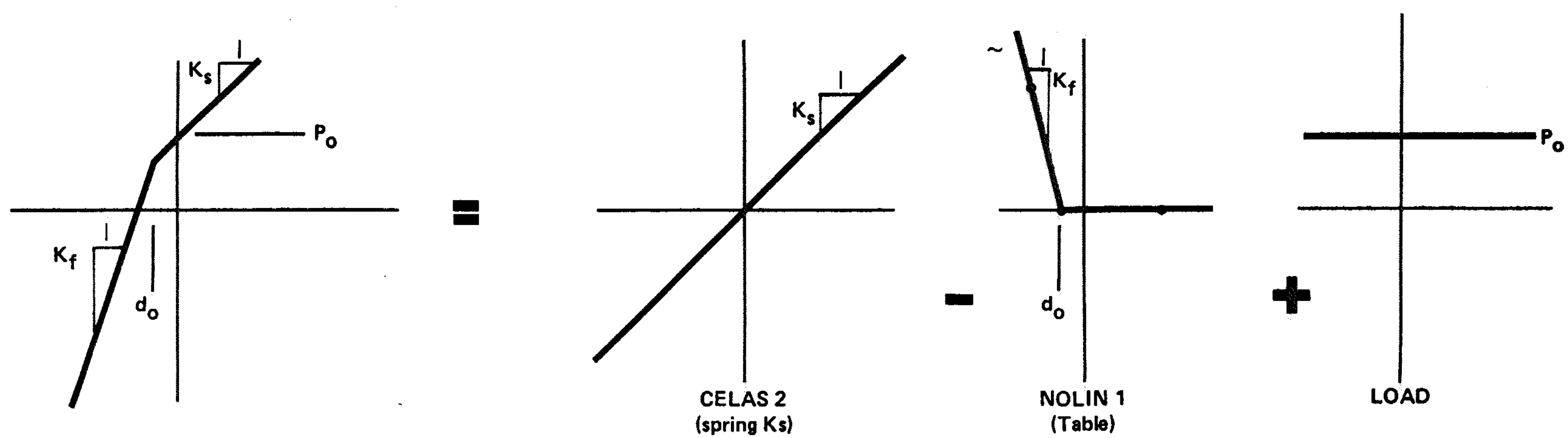

Figure 6
Modeling Contact Conditions of Figure 5

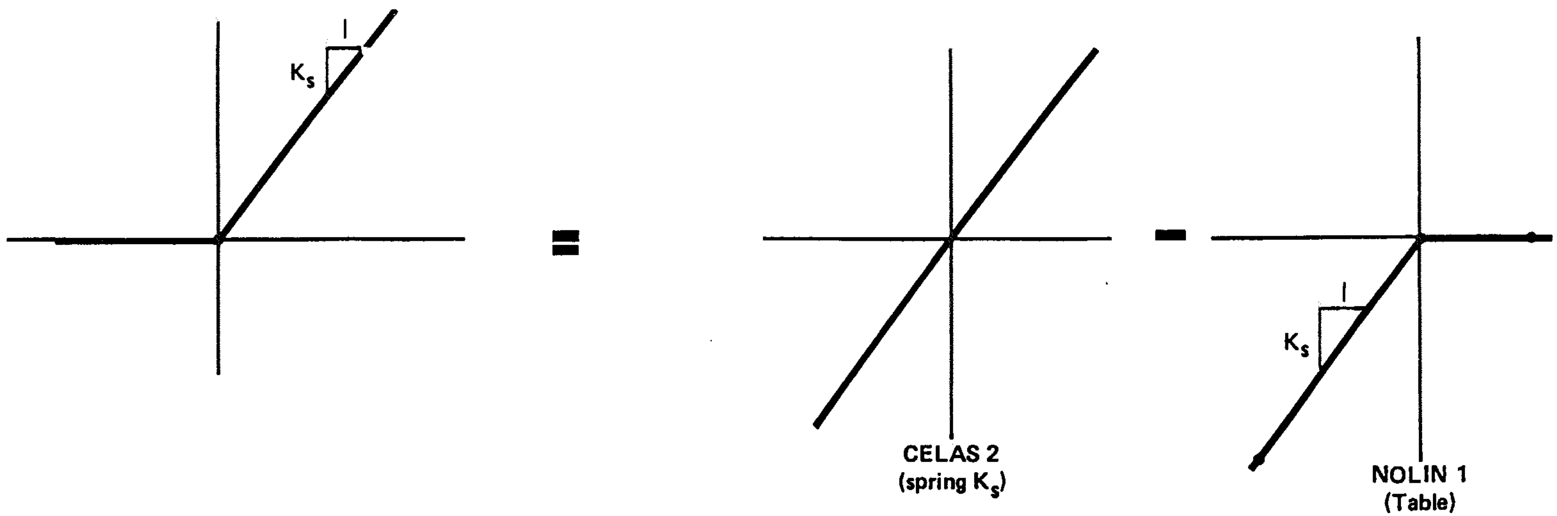

Figure 4
Modeling a Rigid Part Resting Loosely on a
Flexible Support (i.e. Compression Only)

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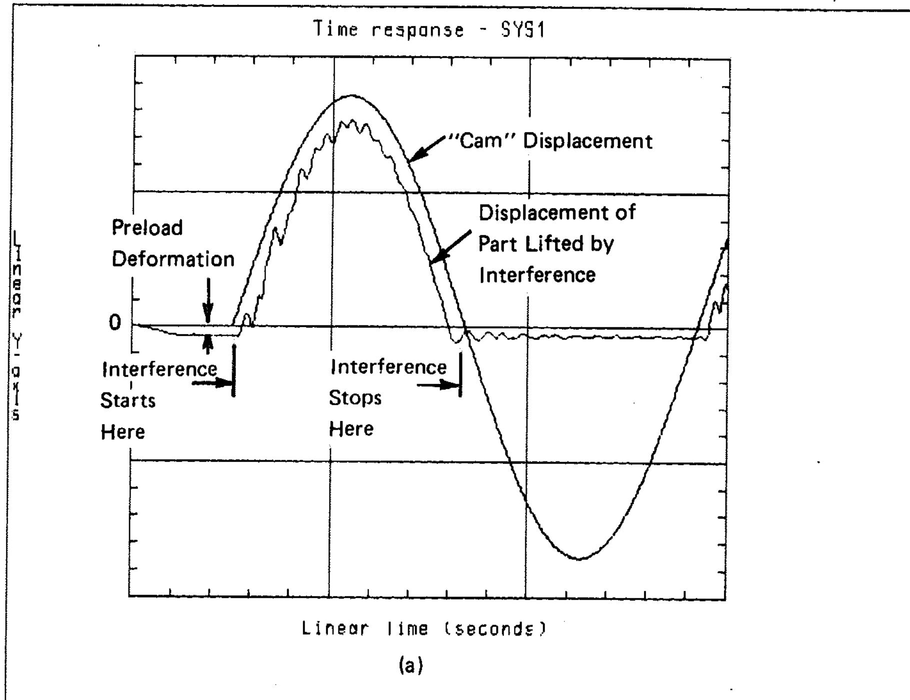
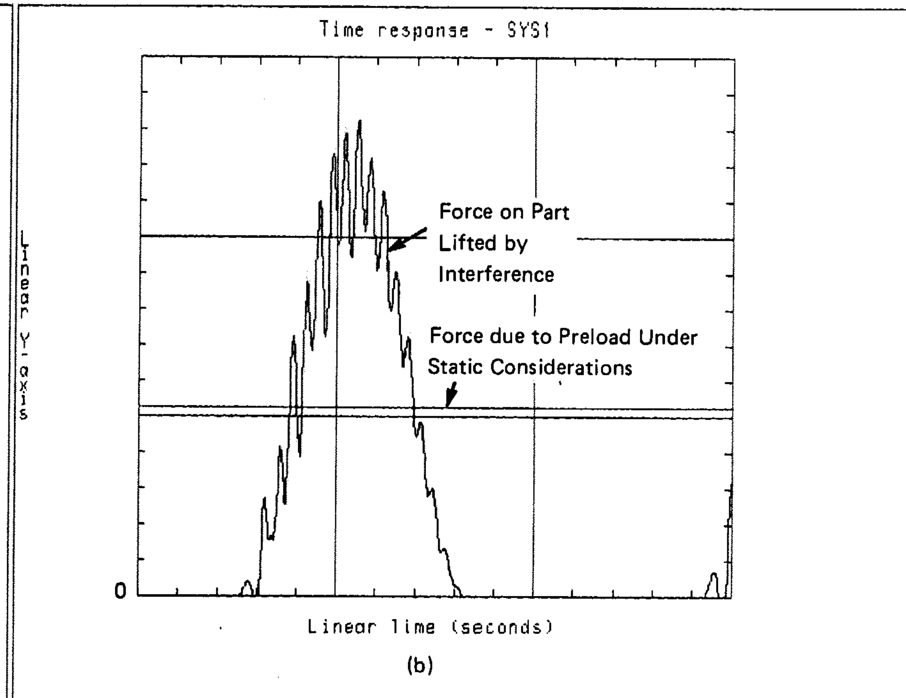
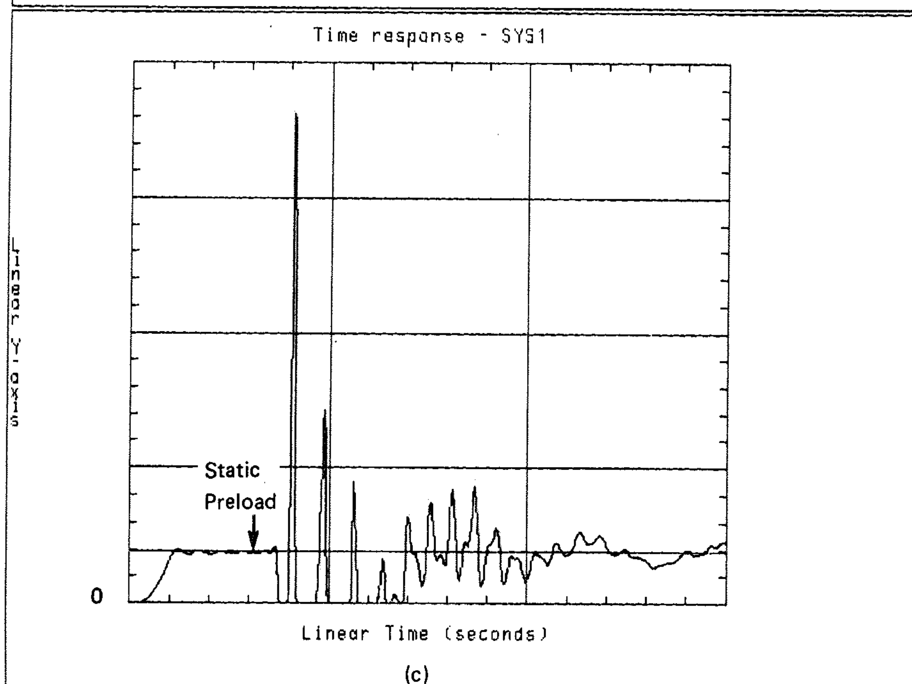


(a)



(b)



(c)

Figure 7
Response of Selected Items Due to
Interference in a Rotating Mechanism

(a) Displacement of part lifted by interference.

(b) Force between lifted part and interference.

(c) Force on free support bearing, including impacts due to contact/no contact situations.

Notes:

1. Actual results have been modified in both distribution and magnitude.

2. All time scales are the same.