Remote Identification of Impact Forces on Loosely Supported Tubes: Analysis of Multi-Supported Systems

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Impact forces are useful information in field monitoring of many industrial components, such as heat exchangers, condensers, etc. In two previous papers we presented techniques—based on vibratory measurements remote from the actual impact locations—for the experimental identification of isolated impacts (Araújo et al., 1996) and complex rattling forces (Antunes et al., 1997). In both papers a single gap support was assumed. Those results concern systems which are simpler than the actual multi-supported tube bundles found in heat exchangers. Impact force identification is a difficult problem for such systems, because 1) when sensed by the remote motion transducers, the traveling waves generated at several impact supports are mixed, and there is no obvious way to isolate the contribution of each support; 2) multi-supported tubes may be quite long, with significant dissipative effects (by interacting flows or by frictional phenomena at the clearance supports), leading to some loss of the information carried by the traveling waves; 3) in multi-supported systems, some of the supports are often in permanent contact, leading to nonimpulsive forces which are difficult to identify. In this paper, we move closer towards force identification under realistic conditions. Only the first problem of wave isolation is addressed, assuming that damping effects are small and also that all clearance supports are impacting. An iterative multiple-identification method is introduced, which operates in an alternate fashion between the time and frequency domains. This technique proved to be effective in isolating the impact forces generated at each gap support. Experiments were performed on a long beam with three clearance supports, excited by random forces. Beam motions were planar, with complex rattling at the supports. Experimental results are quite satisfactory, as the identified impact forces compare favorably with the direct measurements.

1 Introduction

Flow-induced vibration of heat-exchanger tube bundles is an important issue, concerning both component life and plant availability. Excitation by the flow turbulence and possible fluidelastic phenomena may lead to a premature failure of the component, due to material fatigue or to vibro-impact wear of the gap-supported tubes. Predictive methods have been developed to analyze the vibratory responses and wear, for realistic multi-supported configurations and flow excitations (Rogers and Pick, 1977; Frick et al., 1984; Rao et al., 1987; Axisa et al., 1988; Fisher et al., 1988; Mahutov et al., 1989; Antunes et al., 1990; Axisa et al., 1990; de Langre et al., 1991; Fricker, 1991; Antunes et al., 1992a; de Langre et al., 1992; Payen and de Langre, 1996; Sauvé, 1996; Tan and Rogers, 1996; Yetisir and Fisher, 1996; Zhou and Rogers, 1996). Experimental validation of these methods is currently pursued by several research groups, with considerable success (Axisa et al., 1984; Chen et al., 1984; Cha et al., 1986; Haslinger et al., 1987; Antunes et al., 1992b; Axisa and Izquierdo, 1992; Vento et al., 1992; Boucher and Taylor, 1996; de Langre and Lebreton, 1996; Fisher et al., 1996; Mureithi et al., 1996). However, experiments on vibro-impacting tubular bundles involves very carefully instrumented test tubes and tube supports, which is seldom possible in real field components, due to space limitations and severe environment conditions (temperature, radiation), which prevent an adequate instrumentation of the tube supports. Because the tube-support impact forces cannot be easily monitored under real operating conditions, there is a need for identification techniques that enable the diagnostic and field monitoring of tube-support interaction, using information from motion transducers located far from the impact locations. This problem is being addressed by the authors in a series of papers.

The main difficulty with inverse problems is ill conditioning—physical or numerical—of the propagation operators which describe the phenomena. This leads to inverse formulations which are very sensitive to noise contamination of the measured signals. Problems may be partially overcome by regularization of the transformation operators, using several methods; namely, singular value decomposition, incorporation of physical constraints and optimization techniques (see Jeffrey and Rosner, 1986; Dimri, 1992; Press et al., 1992; Groetch, 1993; Parker, 1994 and Hansen, 1994). In the context of vibro-impact system identification, ill-conditioning difficulties are enhanced due to the dispersive nature of flexural waves.

Previous work in this field include papers by Whiston (1984) and Jordan and Whiston (1984), who discussed theoretical and experimental aspects related to the remote identification of impact forces. These authors modeled the flexural propagation waves in the frequency domain, using a Timoshenko beam model without damping. In his book and in a series of related papers, Doyle (1989) followed a similar approach. These authors also presented satisfactory experimental results, provided

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by single impacts acting on long beams, in such way that wave reflections at the boundaries did not interfere seriously with the direct wave used for identification purposes. Lin and Bapat (1992, 1993a) presented methods for estimation of the impact forces and the support gap in a single degree of freedom system, respectively, for sinusoidal and random excitations. The extension of these methods to a beam with a single nonlinear gap-support was proposed using a modal approach in the frequency domain (Lin and Bapat, 1993b). Busby and Trujillo (1987) presented a similar approach, whereas the force identification was achieved in the time domain. The extension of these methods to multi-supported beams, which display an ill-defined (or even unknown) modal basis, is problematic. In a recent paper, Wu et al. (1994) discussed the problem of source separation, for several simultaneous impacts, using a time-domain approach. The so-called spectral methods of deconvolution, which may be quite useful when dealing with nondispersive phenomena, have been used very seldom for dispersive flexural waves (Kim and Lyon, 1992).

In a previous paper (Araújo et al., 1996), we presented techniques, based on response measurements at remote locations, for the experimental identification of the flexural wave-guide propagation parameters and for recovering an impact force and impact location. Experimental results showed that these inverse problems can be successfully attempted, and a good agreement was found between direct measurements and the remotely identified impact forces. However, as in most other work published, those numerical simulations and experiments were presented for simple *isolated impacts.* Such experiments were still remote from the operating conditions of real-life heat exchanger tubes, which display very *complex rattling* motions when subject to flow-induced vibrations. As a result of the very complex pattern of interacting traveling waves generated by rattling phenomena, the identification of impact forces becomes much more difficult. This problem was addressed recently by the authors (Antunes et al., 1997), using a signal processing method which enables the separation of the primary impact-generated flexural waves from the severe background contamination by secondary wave reflections. This technique uses the information provided by a limited number of vibratory transducers, enabling a straightforward identification of the rattling forces at a loose support. It can be applied to both nondispersive and dispersive waves, and is therefore useful for all kinds of beam motions.

The present paper addresses the identification of complex and simultaneous impact forces in *multi-supported systems.* If no restrictions are enforced, this is an extremely hard problem, due to the following difficulties:

1 When sensed by the remote motion transducers, the traveling waves generated at several impact supports are mixed, and there is no obvious way to isolate the contribution of each support.

2 Multi-supported tubes may be quite long, with significant dissipative effects (by interacting flows or by frictional phenomena at the clearance supports), leading to some loss of the information carried by the traveling waves.

3 In multi-supported systems, some of the supports are often in permanent contact, leading to nonimpulsive forces which are difficult to identify.

Only the first problem will be addressed here, as we assume that damping effects are small and also that all clearance supports are impacting. In the first part of this paper, for completeness, we briefly review the main results concerning the propagation and separation of flexural waves. Then, we move to the main contribution of the present work. An iterative multiple-identification method is introduced, which operates in an alternate fashion between the time and frequency domains. This technique proved to be effective in isolating the impact forces generated at each gap support. Experiments were performed using a long steel beam (approximate length 6 m) with nonanechoic boundaries. Excitation was provided by a pair of small inertial shakers driven by banded white noise, in order to simulate the flow turbulence. Impacts were generated at an instrumented support presenting a gap. Vibratory measurements and impact force identifications were based on the responses provided by accelerometers located far from the clearance supports. Results are quite satisfactory, as the identified impact forces compare favorably with the direct measurements.

2 Theoretical Formulation

As shown by Araújo et al. (1996) and Antunes et al. (1997), simple Bernoulli-Euler theory for flexural vibrations proved to be adequate for identification of the tube-support impacts. Assuming a viscous damping model, the small-amplitude flexural response of a beam (with constant cross section) is described by Morse and Ingard (1968) and Graff (1975)

$$EI\frac{\partial^4 y}{\partial x^4} + \rho A\frac{\partial^2 y}{\partial t^2} - N\frac{\partial^2 y}{\partial x^2} + \eta\frac{\partial y}{\partial t} = F(t) \qquad (1)$$

where $F(t)$ is the external force and $y(t)$ is the dynamic vibratory response, E is the Young modulus and ρ is the mass density of the beam, A is the area and I is the moment of inertia of the cross section, N is the axial tension on the beam and η is a viscosity coefficient. Here, parameters E, ρ, A, I, N, and η are assumed constant along the beam. Neglecting the axial tension and damping effects, a solution for Eq. (1) may be obtained in the form

$$y(x,t) = \sum_n \begin{pmatrix} C_{1n}e^{-ik_n x} + C_{2n}e^{ik_n x} + \\ + C_{3n}e^{-k_n x} + C_{4n}e^{k_n x} \end{pmatrix} e^{i\omega_n t} \qquad (2)$$

where, for each circular frequency ω_n, parameter k_n is given by the so-called dispersion relation

Nomenclature

A = area of beam cross section
C_n = wave amplitude parameters
$\mathscr{C}$ = propagation constant
E = Young modulus of beam
$F_y(t)$ = external force
F_n = spectral coefficients of external force
$\mathscr{F}$, $\mathscr{F}^{-1}$ = direct and inverse Fourier transforms
$G(x, \omega_n)$ = wave-to-wave transfer function
$G(x, \omega_n)$ = force-to-wave transfer function
I = moment of inertia of cross section
$k_n = \mathsf{F}(\omega_n)$ = dispersion relation of waves
m = iteration no.
N = axial tension on beam
s = support no.
S = no. of supports
t = time
x = location along beam
$y(x, t)$ = flexural beam response
y_b = backward-going wave
y_f = forward-going wave
Y_n = spectral coefficients of wave
ϵ_m = convergence error
ρ = mass density of beam
ω_n = circular frequency

$$k_n = \left[\frac{\rho A}{EI}\right]^{1/4} \sqrt{\omega_n} \equiv \mathscr{C}\sqrt{\omega_n} \quad (3)$$

and parameters C_{1n} to C_{4n} are frequency dependent. The first and second terms of solution (2) are propagating waves, while the third and fourth terms are nonpropagating (evanescent). As far as the remote identification problem is concerned, the nonpropagating terms can be disregarded, provided all motion transducers are located far from singularities such as the tube boundaries and excitation locations.

Assuming that the beam response $y_0(t) \equiv y(0, t)$ is measured at location $x = 0$ during time T, the spectral coefficients Y_n of $y_0(t)$ may be computed from Fourier analysis. Then the propagated forward and backward traveling waves can be predicted at any other location x using

$$y_f(x, t) \cong \sum_n Y_n e^{-ik_n x + i\omega_n t}$$

$$y_b(x, t) \cong \sum_n Y_n e^{ik_n x + i\omega_n t} \quad (4)$$

On the other hand, assuming that a force $F(t)$ is applied at location $x = 0$ during a time T, the beam response at location x is given by Junger and Feit (1986) and Doyle (1989)

$$y_f(x, t) \cong \frac{1}{4EI} \sum_n \frac{F_n}{k_n^3} i e^{-ik_n x + i\omega_n t}$$

$$y_b(x, t) \cong \frac{1}{4EI} \sum_n \frac{F_n}{k_n^3} i e^{ik_n x + i\omega_n t} \quad (5)$$

where F_n are the spectral coefficients of $F(t)$. These equations will be used to convert from the impact forces to response measurements and also for force estimation.

In practice, manipulation of the preceding formulations can be conveniently achieved by fast Fourier transforming all the time-domain signals. Then propagation phenomena in the frequency domain are computed by simple products of functions. Finally, the time-domain estimated results are obtained by inverse Fourier transforms. The same routine applies when computing the inverse problems. Alternative equations to (4) and (5) may be obtained when dealing with velocity, acceleration, and strain signals. If the axial tension N and damping effects are included, the dispersion relation is more complex than (3) and k_n will display both real and imaginary parts. Damping effects are often quite small—such is the case of the experiments presented here—and can be safely neglected. However, when damping effects are significant, use of the simple dispersion relation (3) will lead to a systematic underestimation of the impact forces. As discussed by Araújo et al. (1996), significant damping terms may cause unstable identification results. Indeed, in the inverse problem, damping terms will unduly amplify any noise contamination of the measurements.

3 Identification of Impact Forces

Equations (5) relate excitation forces to the *primary* generated waves, and are only strictly applicable to infinite beams. For finite-length systems, the straight inversion (5) of the wave response would lead to completely erroneous impact force identifications. To overcome this problem, it is essential to isolate the primary waves from the background secondary reflections. Antunes et al. (1997) showed that, even for complex rattling motions, such wave separation can be effectively achieved using two sets of motion transducers. As shown in Fig. 1(*a*), these encompass a section of the beam which contains the gap support (located at x_0). The right-traveling and left-traveling waves sensed by a pair of transducers—see Fig. 1(*b*)—can then be separated using a simple frequency-domain formulation, based on Eqs. (4).

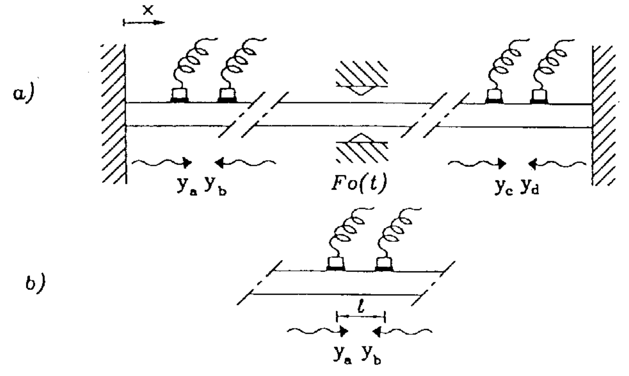

Fig. 1 (*a*) Typical setup for the wave separation and impact-force identification; (*b*) detail of a pair of close transducers

Each transducer j senses a signal $Z_j(\omega)$, which is the sum of the right-traveling wave $Y_{aj}(\omega)$ plus the left-traveling wave $Y_{bj}(\omega)$. These measurements can be written in terms of the waves sensed by the first transducer

$$\begin{Bmatrix} Z_1(\omega) \\ Z_2(\omega) \end{Bmatrix} = \begin{bmatrix} 1 & 1 \\ e^{-ik(\omega)(x_2-x_1)} & e^{ik(\omega)(x_2-x_1)} \end{bmatrix} \begin{Bmatrix} Y_{a1}(\omega) \\ Y_{b1}(\omega) \end{Bmatrix} \quad (6)$$

and the separated waves are obtained by inversion of (6)

$$\begin{Bmatrix} Y_{a1}(\omega) \\ Y_{b1}(\omega) \end{Bmatrix} = [\mathscr{M}_{12}(x_1, x_2, \omega)]^{-1} \begin{Bmatrix} Z_1(\omega) \\ Z_2(\omega) \end{Bmatrix} \quad (7)$$

In a similar way, for the pair of transducers located in the right side of the beam

$$\begin{Bmatrix} Y_{c4}(\omega) \\ Y_{d4}(\omega) \end{Bmatrix} = [\mathscr{M}_{34}(x_1, x_2, \omega)]^{-1} \begin{Bmatrix} Z_3(\omega) \\ Z_4(\omega) \end{Bmatrix} \quad (8)$$

The time-domain separated waves are easily obtained by inverse Fourier-transforming the arrays $Y_{a1}(\omega)$, $Y_{b1}(\omega)$, $Y_{c4}(\omega)$, and $Y_{d4}(\omega)$. From Eqs. (5), (7), and (8), two estimates of the impact force can be obtained (Antunes et al., 1997)

$$F_0^{(1)}(t) = \mathscr{F}^{-1}\left[\frac{Y_{b1}(\omega) - Y_{d4}(\omega)G_{41}(\omega)}{\tilde{G}_{01}(\omega)}\right] \quad (9)$$

$$F_0^{(2)}(t) = \mathscr{F}^{-1}\left[\frac{Y_{c4}(\omega) - Y_{a1}(\omega)G_{14}(\omega)}{\tilde{G}_{04}(\omega)}\right] \quad (10)$$

where the propagation functions $G_{ij}(\omega)$ and $\tilde{G}_{0j}(\omega)$ are given as

$$G_{ij}(\omega) = e^{-ik(\omega)(x_i - x_j)} \quad (11)$$

$$\tilde{G}_{0j}(\omega) = \frac{i}{4EIk(\omega)^3} e^{-ik(\omega)(x_0 - x_j)} \quad (12)$$

This formulation was established assuming that the waves are sensed by displacement transducers. For other types of transducers, simple modifications apply; for example, a factor $-\omega^2$ is introduced in formulation $\tilde{G}_{0j}(\omega)$ when accelerometers are used. At least two pairs of transducers are required to apply the aforementioned technique. However, as shown by Antunes et al. (1997), better numerical conditioning of the wave separation is achieved if three well-located transducers are used for each set. Then, matrixes $\mathscr{M}_{12}$ and $\mathscr{M}_{34}$ become rectangular and should be pseudo-inverted in the Moore-Penrose sense (Groetch, 1993).

The preceding formulation will be used as a departure point, when discussing the identification of multi-supported systems in Section 4. The previous remarks are illustrated in Fig. 2, for a vibro-impacting beam. The sample acceleration signal (a) was sensed by one of the remote transducers. The direct measurement of the impact force (b) compares very well with the estimates of the impact force (c–d) obtained from Eqs. (9) and (10). However, a straight identification using Eq. (5) will lead to the totally erroneous result (e). Details of the experimental setup are provided by Antunes et al. (1997).

4 Extension to Multi-Supported Systems

In the case of multi-supported beams or tubes, the waves sensed by the remote transducers are generated at several impact locations. This introduces a further difficulty in the identification problem. An obvious solution is to isolate each clearance support with two nearby sets of motion transducers. However, this is unsatisfactory, because it prevents true remote identifications and also leads to time-consuming experiments. In this section, we introduce a constrained-inversion approach to deal with the simultaneous multiple-identification problem. This iterative algorithm operates in an alternate fashion between the time and

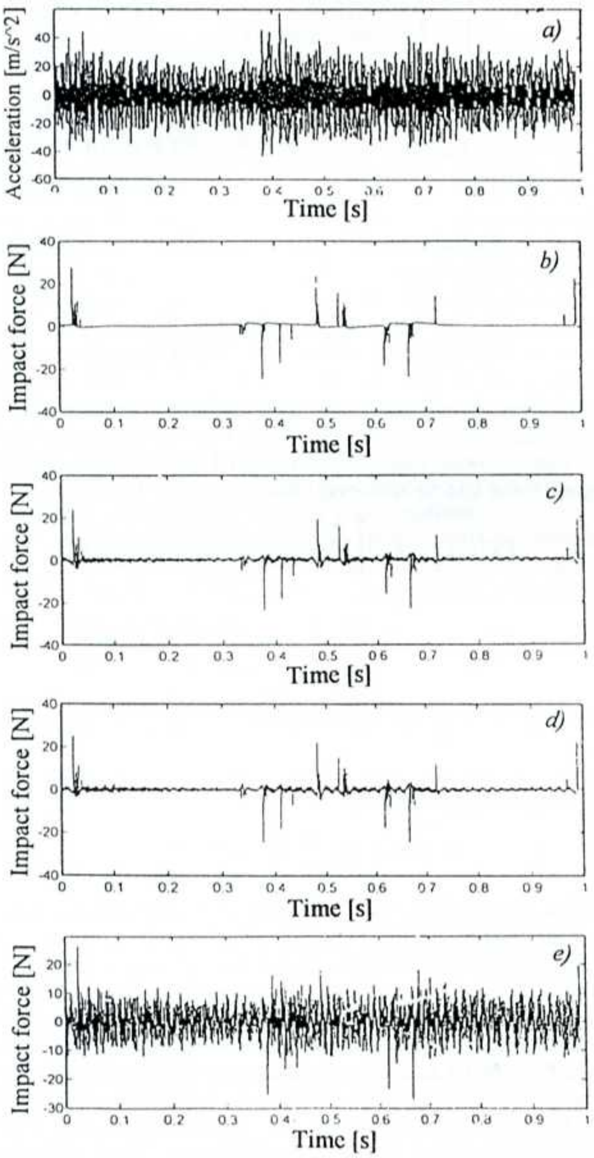

Fig. 2 (a) Sample of a response time-history; (b) measured impact force time-history; (c–d) two identifications of the impact force based on the wave separation technique; (e) direct naïve force identification

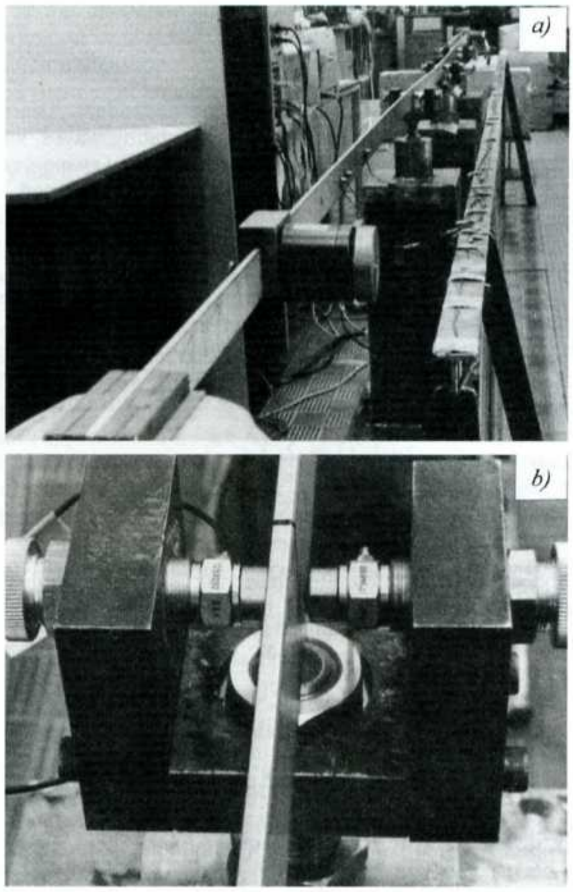

Fig. 3 (a) General view of the experimental setup; (b) detail of one instrumented support

frequency domains; to be specific, wave propagation is performed in the frequency domain and solution constraints are imposed in the time domain, at each iteration.

We showed in Section 3 that *two* estimates of the impact force $F_s(t)$ at a given location s can be computed. These estimates may obviously be used to isolate the impact forces in systems with only two gap supports. This approach will not obviously work whenever we have three or more clearance locations, which is the general case in heat exchangers. However, two estimates $F_s^{(1)}(t)$ and $F_s^{(2)}(t)$ of the impact force can be most useful to check the *consistency* of the results. The basic idea is that, at a given clearance support s, the estimates should be well correlated *only* when impacts arise at that particular location. Bad correlation between the force estimates means that (i) the system is not impacting, or that (ii) impacts were generated at a different location.

Let us define the "moving cross-correlation" function

$$\mathcal{R}_s(t) = \frac{\int_{t-\Delta T}^{t+\Delta T} (F_s^{(1)}(\tau) - \bar{F}_s^{(1)})(F_s^{(2)}(\tau) - \bar{F}_s^{(2)})d\tau}{\left[\int_{t-\Delta T}^{t+\Delta T} (F_s^{(1)}(\tau) - \bar{F}_s^{(1)})^2 d\tau \int_{t-\Delta T}^{t+\Delta T} (F_s^{(2)}(\tau) - \bar{F}_s^{(2)})^2 d\tau\right]^{1/2}} \quad (13)$$

This function was used by Antunes et al. (1997) as a final "cleaning" procedure in their identifications. Here, Eq. (13) will be used to help separate the impact forces generated at each gap support. This is simply a scaled correlation coefficient ($0 < \mathcal{R}_s(t) < 1$) between $F_s^{(1)}(t)$ and $F_s^{(2)}(t)$, which is computed within a moving window of size $2\Delta t$. The window size is based on the time scale of individual force spikes, in order to get an adequate time resolution. Estimates of Δt can be easily

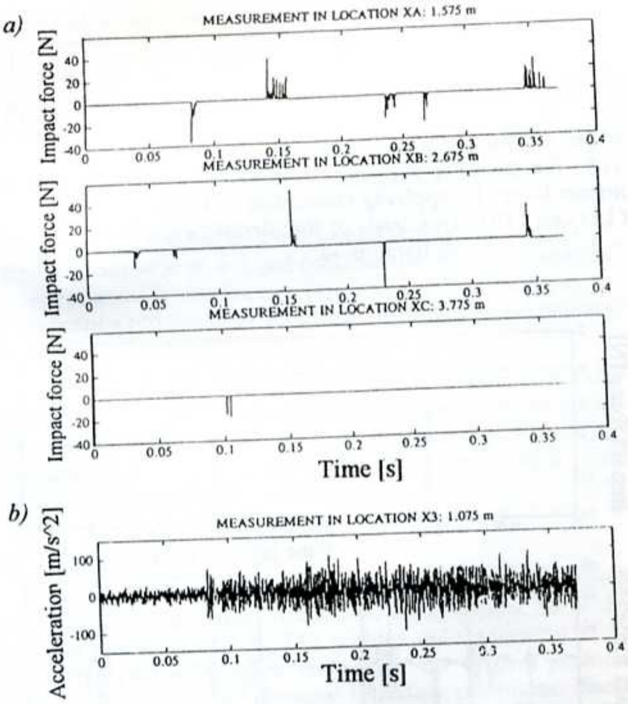

Fig. 4 (*a*) Measured impact forces at the three clearance-supports; (*b*) motion response from a remote accelerometer

computed from the tube/support contact stiffness and the tube mass (Axisa et al., 1988), or by simple experiments. Then, the following physically plausible *constraints* can be imposed to the identified impact forces:

1 If $\mathscr{R}_s(t)$ is higher than a suitable value $\mathscr{R}_{\text{Lim}}$, impacts were generated at support s, and a better estimate of the impact force is given as

$$F_s(t) = \frac{F_s^{(1)}(t) + F_s^{(2)}(t)}{2} \quad (\text{if } \mathscr{R}_s(t) > \mathscr{R}_{\text{Lim}}) \tag{14}$$

2 If $\mathscr{R}_s(t)$ is lower than $\mathscr{R}_{\text{Lim}}$, there are no impacts at support s and the impact force is null

$$F_s(t) = 0 \quad (\text{if } \mathscr{R}_s(t) < \mathscr{R}_{\text{Lim}}) \tag{15}$$

As pointed by Antunes et al. (1997), a good value for $\mathscr{R}_{\text{Lim}}$ is given by the level corresponding to a minimum in the histogram of the "moving-correlation" function. This will be illustrated in Section 6.

Let us now consider a beam with S clearance supports. Based on the preceding arguments, the following iterative identification method can be proposed.

(A) Initial Identification (Iteration $m = 0$)

1 For each gap support s, compute starting estimates of the impact forces from the separate left and right traveling waves

$$F_{s0}^{(1)}(t) = \mathscr{F}^{-1}\left[\frac{Y_{b1}(\omega) - Y_{d4}(\omega)G_{41}(\omega)}{\tilde{G}_{s1}(\omega)}\right]; \quad s = 1, 2, \ldots, S \tag{16}$$

$$F_{s0}^{(2)}(t) = \mathscr{F}^{-1}\left[\frac{Y_{c4}(\omega) - Y_{a1}(\omega)G_{14}(\omega)}{\tilde{G}_{s4}(\omega)}\right]; \quad s = 1, 2, \ldots, S \tag{17}$$

2 For each gap support s, improve the initial estimates of the impact forces by applying constraints (14) and (15) to results (16) and (17). This leads to the identification $\hat{F}_{s0}(t)$ (with $s = 1, 2, \ldots, S$), at iteration $m = 0$.

(B) Identification Loop (Iterations $m = 1, 2, \ldots$)

1 For each gap-support s, compute new estimates of the impact forces by correcting Eqs. (16) and (17) with the identi-

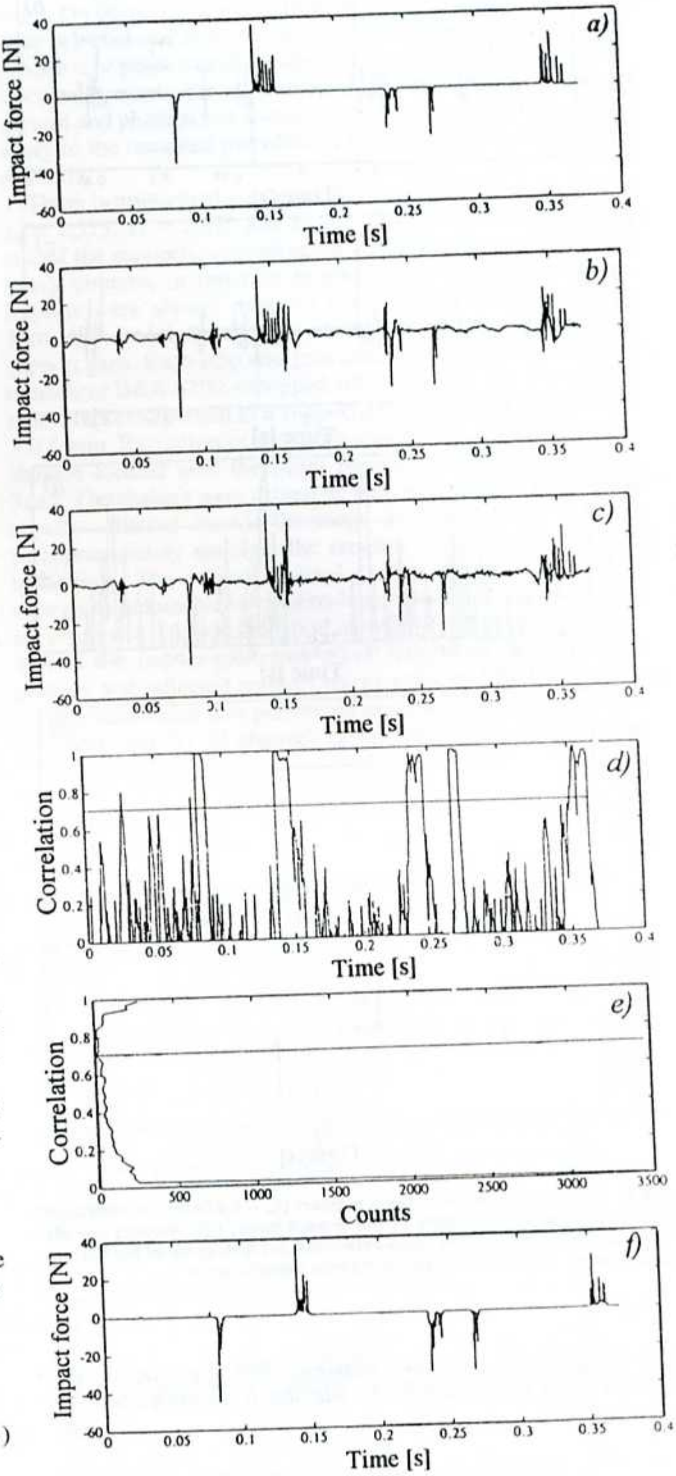

Fig. 5 Impact force at the first support (x_a = 1.757 m): (*a*) measured force; (*b–c*) initial estimates of the impact force; (*d*) "moving correlation" function between the force estimates; (*e*) histogram of the correlation function; (*f*) initial constrained force identification