Pescara benchmarks: nonlinear identification

E Gandino¹, L Garibaldi and S Marchesiello
Dynamics & Identification Research Group,
Department of Mechanics, Politecnico di Torino,
C.so Duca degli Abruzzi 24, Torino, I-10129, Italy

E-mail: edoardo.gandino@polito.it

Abstract. Recent nonlinear methods are suitable for identifying large systems with lumped nonlinearities, but in practice most structural nonlinearities are distributed and an ideal nonlinear identification method should cater for them as well. In order to extend the current NSI method to be applied also on realistic large engineering structures, a modal counterpart of the method is proposed in this paper. The modal NSI technique is applied on one of the reinforced concrete beams that have been tested in Pescara, under the project titled ”Monitoring and diagnostics of railway bridges by means of the analysis of the dynamic response due to train crossing”, financed by Italian Ministry of Research. The beam showed a softening nonlinear behaviour, so that the nonlinearity concerning the first mode is characterized and its force contribution is quantified. Moreover, estimates for the modal parameters are obtained and the model is validated by comparing the measured and the reconstructed output. The identified estimates are also used to accurately predict the behaviour of the same beam, when subject to different initial conditions.

1. Introduction
In many applications nonlinear effects may affect significantly the dynamics, even when the amplitude of the motion is sufficiently small. These dynamical effects must be accounted for in order to accurately understand and robustly model the dynamics.

Nonlinear system identification has been thoroughly investigated in recent years and many efforts have been spent leading to a large number of methods. A comprehensive list describing the past and recent developments is given in [1]. Among them, the Conditioned Reverse Path method [2, 3] is based on the construction of a hierarchy of uncorrelated response components in the frequency domain. The Nonlinear Identification through Feedback of the Outputs (NIFO) [4] exploits the spatial information and interprets nonlinear forces as unmeasured internal feedback forces. Starting from the basic idea of NIFO, the Nonlinear Subspace Identification (NSI) method has been developed [5], showing a higher level of accuracy with respect to NIFO. NSI is a time domain method which exploits the robustness and the high numerical performances of the subspace algorithms.

This method has been improved for identifying large systems with lumped nonlinearities [6], but in practice most structural nonlinearities are distributed and an ideal nonlinear identification method should cater for them as well [7]. Moreover, any nonlinear method is requested to correctly identify the types of nonlinearity present and possibly to quantify the extent of their force contributions. Therefore, the identification of a whole parametric nonlinear model is an important instrument for

¹ To whom any correspondence should be addressed.
many purposes. For example, it would allow for treating nonlinearities in possibly damaged structures [8], or attaining improved predictions of vibration response amplitude, which is an issue for accurate long term fatigue estimates.

In order to extend the current method to be applied also on realistic large engineering structures, a model reduction is needed and can be performed by selecting a set of modes that span the dominant dynamics. To this purpose, a modal counterpart of the NSI method has been developed, together with ideas for handling a large complex nonlinear system as simple modal single degree of freedom systems.

In this paper, the modal NSI technique is described and demonstrated through an experimental application set up in Pescara, Italy, where some tests on reinforced concrete beams have been performed. In a companion paper [9] the set up is fully described and a softening nonlinear behaviour is shown, thus justifying the application of the proposed method. Here, for one of those “Pescara beams”, a nonlinearity concerning the first mode is characterized and its force contribution is quantified. Moreover, estimates for the modal parameters are obtained and the model is validated by comparing the measured and the reconstructed output. The identified estimates are also used to accurately predict the behaviour of the same beam, when subject to different initial conditions.

2. Nonlinear Subspace Identification in modal space

2.1. Nonlinear modal model
A dynamical system with \( N \) degrees of freedom and with lumped nonlinear springs and dampers can be described by the following equation of motion [5, 7]:

\[
\ddot{\mathbf{z}}(t) + \mathbf{C}_v \dot{\mathbf{z}}(t) + \mathbf{K} \mathbf{z}(t) = \ddot{\mathbf{f}}(t) - \sum_{j=1}^{R} \mathbf{p}_j \mathbf{E}_{\mathbf{g}} \mathbf{g}_j(z, \dot{z}) = \ddot{\mathbf{f}} + \ddot{\mathbf{f}}_n(t),
\]

where \( \mathbf{M} \), \( \mathbf{C}_v \) and \( \mathbf{K} \) are the mass, viscous damping and stiffness matrices respectively, \( \mathbf{z}(t) \) is the generalised displacement vector and \( \ddot{\mathbf{f}}(t) \) the generalised force vector, both of dimension \( N \), at time \( t \). Each of the \( R \) nonlinear components depends on the scalar nonlinear function \( \mathbf{g}_j(z, \dot{z}) \), which specifies the class of the nonlinearity (e.g., Coulomb friction, clearance, quadratic damping, etc.). The vector \( \mathbf{E}_{\mathbf{g}} \), whose entries may assume the values 1, -1 or 0, is related to the location of the nonlinear element.

The transformation between physical and modal space, after selecting \( N_R \) retained modes, is defined by

\[
\mathbf{z}(t) = \mathbf{\Phi} \mathbf{p}(t),
\]

where \( \mathbf{p}(t) \) is the generalised modal displacement vector and \( \mathbf{\Phi} \) is the \( N \times N_R \) modal matrix of the underlying linear system. The modal approach is a form of model order reduction, since the number of retained modes, in general, will be much smaller than the number of physical degrees of freedom, so that \( N_R << N \).

Substituting equation (2) into the equation of motion (1) and pre-multiplying by \( \mathbf{\Phi}^T \) yields:

\[
\mathbf{\Phi}^T \ddot{\mathbf{M}} \ddot{\mathbf{\Phi}} \mathbf{p}(t) + \mathbf{\Phi}^T \mathbf{C}_v \dot{\mathbf{\Phi}} \mathbf{p}(t) + \mathbf{\Phi}^T \mathbf{K} \mathbf{\Phi} \mathbf{p}(t) = \mathbf{\Phi}^T \ddot{\mathbf{f}}(t) - \mathbf{\Phi}^T \sum_{j=1}^{R} \mathbf{p}_j \mathbf{E}_{\mathbf{g}} \mathbf{g}_j(z, \dot{z}) = \mathbf{\Phi}^T \ddot{\mathbf{f}} + \mathbf{\Phi}^T \ddot{\mathbf{f}}_n(t).
\]

By using the orthogonality of the modes, equation (3) becomes:

\[
\mathbf{M} \ddot{\mathbf{p}}(t) + \mathbf{C}_v \dot{\mathbf{p}}(t) + \mathbf{K} \mathbf{p}(t) = \mathbf{f}(t) - \sum_{j=1}^{R} \mathbf{\mu}_j \mathbf{g}_j(p, \dot{p}) = \mathbf{f}(t) + \mathbf{f}_n(t),
\]

where the matrices \( \mathbf{M} \), \( \mathbf{C}_v \) and \( \mathbf{K} \) are \( N_R \times N_R \). The modal mass matrix \( \mathbf{M} \) and the linear modal stiffness matrix \( \mathbf{K} \) are diagonal, while the modal damping matrix \( \mathbf{C}_v \) is diagonal for proportionally
damped systems only. \( f(t) \) is the \( N_r \times 1 \) applied modal force vector and \( f_{nl}(t) \) is the \( N_r \times 1 \) vector of internal feedback modal forces due to nonlinearities.

From the equation of motion (4), the following continuous modal model can be derived:

\[
\dot{x} = A_r x + B_r u \\
y = C x + D u
\]  
(5)

The continuous model of (5) may be converted into a discrete modal model and then processed by means of the subspace methods. The nonlinear identification procedure is based on the computation of system parameters, once the state space matrices \( A_r, B_r, C \) and \( D \) have been estimated by a subspace method in the time domain [10]. In fact, system parameters (included in \( M, C_r, K \) and \( \mu_j \)) are contained in the matrix

\[
H_N(\omega) = D + C(\omega \mathbf{i} - A_r)^{-1}B_r ,
\]  
(6)

which is invariant under the similarity transformation corresponding to the application of a subspace method [5].

### 2.2. Nonlinearities in modal coordinates

Let’s focus on how \( f_{nl}(t) \) in equation (4) can be obtained from the nonlinear term of equation (3). A vector \( L_{nl} = \Phi^T \bar{L}_{nl} \), whose entries depend on the matrix \( \Phi \), can be defined. Moreover, it can be assumed that

\[
\sum_{j=1}^{h} \bar{g}_j(\Phi p, \Phi \dot{p}) = \sum_{j=1}^{h} \varphi_j g_j(p, \dot{p}),
\]  
(7)

where the coefficients \( \varphi_j \) and the number of nonlinear modal components \( h \) depend on \( \Phi, \bar{h}, N_r \) and on the nonlinear functions \( g_j(z, \dot{z}) \). In equation (7), \( h \) new nonlinear modal functions \( g_j(p, \dot{p}) \) are derived. Then,

\[
\sum_{j=1}^{h} \bar{f}_j(\Phi p, \Phi \dot{p}) = \sum_{j=1}^{h} \bar{L}_{nl} \varphi_j g_j(p, \dot{p}) = \sum_{j=1}^{h} \mu_{nj} g_j(p, \dot{p}).
\]  
(8)

A simple example is given, in case of a 3DOFs system with a single nonlinear function \( \bar{g}_j(z) = (z_2 - z_1)^3 \), a cubic stiffness between DOFs 1 and 2. In this case, \( N = 3 \) and \( \bar{h} = 1 \). From equation (2), \( \bar{g}_1 \) can be written as:

\[
\bar{g}_1 = \left( \sum_{r=1}^{N} \Phi_{2,r} p_r - \sum_{r=1}^{N} \Phi_{1,r} p_r \right)^3 = \left( \sum_{r=1}^{N} (\Phi_{2,r} - \Phi_{1,r}) p_r \right)^3 = \left( \sum_{r=1}^{N} \alpha_r p_r \right)^3.
\]  
(9)

By considering just the first mode, \( N_r = 1 \), equation (9) becomes

\[
\bar{g}_1 = \alpha_1^3 p_1^3 = \varphi_1 g_1(p),
\]  
(10)

so \( h = 1 \).

By taking into account \( N_r = 2 \), equation (9) turns into a more complicate expression:

\[
\bar{g}_1 = (\varphi_1 p_1 + \varphi_2 p_2)^3 = \alpha_1^3 p_1^3 + 3\alpha_1^2 \varphi_2 p_1^2 p_2 + 3\alpha_1 \varphi_2^2 p_1 p_2^2 + \varphi_2^3 p_2^3 = \varphi_1^3 p_1^3 + \varphi_2^2 p_2^2 + \varphi_1 p_1^2 p_2 + \varphi_2 p_2^2 + \varphi_1 p_1 p_2^2 + \varphi_2 p_1^2 p_2 + \varphi_1^2 p_1 p_2 + \varphi_2^2 p_1 p_2
\]  
(11)

and in this case \( h = 4 \).

Written as in equation (11), the transformation from physical to modal coordinates can be seen as an unattractive step, since it can introduce more nonlinear terms and then increasing difficulties and computational efforts. This is true for numerical applications in which small systems with known lumped parameters are analyzed: the class and characterisation of nonlinear terms are known and their
expression in modal coordinates is in general disadvantageous. For these systems, other methods [4, 5] in physical coordinates are most suited, although the modal space representation behaves well anyway. The present modal approach can be much effective when analyzing real, continuous (not lumped parameter) and complex structures, for which the characterisation of distributed nonlinearities is not known but can be easily approximated with some nonlinear modal functions \( g_j(p, \dot{p}) \).

2.3. Estimating the modal model
The modal coordinates \( p(t) \) of equation (2) will in practice be obtained from the measurements of \( N_m \) measured physical coordinates \( z_m \), where usually \( N_R < N_m << N \). Then [7]

\[ p = (\Phi_m^T \Phi_m)^{-1} \Phi_m^T z_m, \]  \hspace{1cm} (12)

where \( \Phi_m \) is the \( N_m \times N_R \) modal matrix corresponding to the measured set of responses.

For systems with a significant class of stiffness nonlinearities (such as polynomials, sine and piecewise linear functions) it is possible to obtain nearly linear responses as long as the excitation applied to the system has a sufficiently low amplitude, so the system behaves in an essentially linear manner [7]. This allows for the identification of the modal matrix \( \Phi_m \) and consequently of the modal model of the underlying linear system.

3. Single degree of freedom approach
Starting from the equation of motion (4), the equation for a particular mode for a proportionally damped nonlinear system is in the form of a single degree of freedom (SOF) system:

\[ m_r \ddot{p}_r + c_r \dot{p}_r + k_r p_r = f_r(t) + \sum_{j=1}^{k} \mu_{r,j} g_{r,j}(p, \dot{p}) = f_r(t) + f_{r, nl}(t), \]  \hspace{1cm} (13)

where \( p_r \) is the \( r \)-th modal displacement, \( m_r \), \( c_r \) and \( k_r \) are the \( r \)-th mode modal mass, damping and stiffness and \( f_r \) is the applied modal force [7]. The term \( f_{r, nl} \) refers to the \( r \)-th mode internal feedback modal force and in general is a function of several modal coordinates to allow for nonlinear cross-coupling. Note that the subscript \( r \) has been introduced here also to denote \( h_r \) and \( g_{r,j}(p, \dot{p}) \), in order to underline the possibility to choose a suitable number and type for each of the \( r \)-th modal nonlinear terms, since the equations are separately dealt with. This can be useful in particular for real continuous structures, for which the main nonlinear modes can be identified in more detail.

Nonproportional damping would lead to the presence of modal damping coupling terms, so equation (4) should be identified by subspace methods as a whole \( N_R \) degrees of freedom system instead of \( N_R \) SDOF systems.

3.1. Free response
In absence of an applied force, the analysis is performed by considering the system as subject to initial conditions or an impulsive excitation. A free response analysis can be useful in order to perform a characterisation of modal nonlinearities, in particular for large structures, when forced tests are often uneasy.

Consider the equation (13) for the \( r \)-th mode; assuming that the measurements concern displacements only, the modal formulation corresponds to a state vector chosen as \( x = [p_r, \dot{p}_r]^T \) and to an input vector \( u = [-g_{r,1}(t) \ldots -g_{r,k}(t)]^T \), as the following:
\[
\begin{bmatrix}
\dot{p}_r \\
\ddot{p}_r
\end{bmatrix} =
\begin{bmatrix}
0 & 1 \\
-k_r/m_r & -c_r/m_r
\end{bmatrix}
\begin{bmatrix}
p_r \\
\dot{p}_r
\end{bmatrix} +
\begin{bmatrix}
0 & \cdots & 0
\end{bmatrix}
\begin{bmatrix}
\mu_{r,sl}/m_r \\
\mu_{r,sh}/m_r
\end{bmatrix}
\begin{bmatrix}
g_{r,i}(t) \\
g_{r,h}(t)
\end{bmatrix} 
\] (14)

\[
y = [1 \ 0]
\begin{bmatrix}
p_r \\
\dot{p}_r
\end{bmatrix} + [0 \ \cdots \ 0]
\begin{bmatrix}
-g_{r,i}(t) \\
-g_{r,h}(t)
\end{bmatrix} 
\] (15)

and matrices \( A_r, B_r, C \) and \( D \) of equation (5) are consequently defined.

From the eigenvalues of the system matrix \( A_c \) it is possible to obtain estimates for the eigenfrequencies \( f_{n,r} \) of the undamped system and for the damping ratios \( \zeta_r \). Then, by estimating the modal mass or assuming a unitary modal mass, estimates for the remaining modal parameters \( c_r \) and \( k_r \) can be computed.

Consider now the identification of the \( r \)-th mode nonlinear coefficients \( \mu_{r,ij} \). The computation of matrix \( H_{r,E}(\omega) \) can be performed as in equation (6), and the particular case \( \omega = 0 \) can be easily computed from equations (14) and (15):

\[
H_{r,E}(\omega = 0) = [0 \ \cdots \ 0][-1 \ 0]
\begin{bmatrix}
-c_r/k_r & -m_r/k_r & 0 & \cdots & 0 \\
1 & 0 & \mu_{r,sl}/k_r & \cdots & \mu_{r,sh}/k_r
\end{bmatrix}
\] (16)

In case of monotonic nonlinear functions, such as polynomials, these estimated values can be used to evaluate the ratio between the \( j \)-th nonlinear modal feedback and the linear stiffness force contribution:

\[
\lambda_j = \frac{\mu_{r,sl}}{k_r} \frac{\max(g_{r,i}(p, \dot{p}))}{\max(p)} , \quad \text{for} \quad j = 1, 2, \ldots, h_r 
\] (17)

Then, the absolute contribution of each nonlinear term with respect to the whole nonlinear force can be evaluated in percentage as follows:

\[
\delta_j = 100 \frac{|\lambda_j|}{\sum_{i=1}^{k_r} |\lambda_i|} , \quad \text{for} \quad j = 1, 2, \ldots, h_r 
\] (18)

The investigation of \( \delta_j \) leads to the elimination of the minor or negligible nonlinear terms and then to a suitable choice of the \( h_r \) nonlinear modal functions for further detailed identification procedures.

4. Experimental application

The proposed modal NSI method can be effectively applied to real, continuous structures, as demonstrated by the following experimental application.

In the framework of the project titled "Monitoring and diagnostics of railway bridges by means of the analysis of the dynamic response due to train crossing", financed by Italian Ministry of Research, some experimental tests on reinforced concrete beams have been performed in Pescara, Italy. These beams have shown a softening nonlinear behaviour, so the application of the proposed method is justified and demonstrated in the following as well. A detailed description of the tests is given in [9].
Table 1. Characteristic of the beam.

| Characteristic               | Value            |
|------------------------------|------------------|
| Total length                 | $l = 3.2$ m      |
| Length between the supports  | $L = 3$ m        |
| Area                         | $A = 0.0072$ m$^2$ |
| Moment of inertia            | $I = 4.32 \cdot 10^{-6}$ m$^4$ |
| Young modulus                | $E = 50.481$ GPa |
| Density                      | $\rho = 2597.2$ kg/m$^3$ |

In figure 1, a scheme of the simply supported beam analyzed (called T4-1, series 4, number 1) is shown, and the characteristics of the beam are presented in table 1 as a general reference. It is assumed that the “underlying linear” beam can be approximated by a simply supported beam, so the mode shapes of the undamped system can be obtained from the theoretical relationship

$$\phi_r = \sin \left( \frac{\pi x}{L} \right), \quad \text{for } r = 1, \ldots, N_R$$

where $L$ denotes the length of the beam and $x$ is the position over the beam. Since the contribution of other modes is negligible with respect to the first one [9], $N_R = 1$ is selected and the normalized modal matrix is:

$$\Phi = \begin{bmatrix} 0.5000 & 0.7071 & 0.5000 \end{bmatrix}^T.$$  

The beam has been excited by an impulsive force given by lifting it a few centimetres by one end and then by releasing it, such that it bumped against the support; accelerations in points 1, 2 and 3 (placed respectively at $\frac{L}{4}$, $\frac{L}{2}$ and $\frac{3L}{4}$) have been measured. For the present analysis, displacements are obtained through a numerical integration. Then, equation (12) is applied in order to obtain the modal coordinate $p_1(t)$.

A SDOF equation for the first mode is obtained as in equation (13); in order to perform a free response analysis as described previously, a ninth-order odd polynomial stiffness has been attempted, with the following four modal nonlinear terms (the subscript $r = 1$ is omitted from now on):

$$g_1(t) = p_1(t)^3, \ g_2(t) = p_1(t)^5, \ g_3(t) = p_1(t)^7, \ g_4(t) = p_1(t)^9$$

Table 2. Nonlinear terms contribution to the nonlinear force, in percentage.

| $\delta_1$ | $\delta_2$ | $\delta_3$ | $\delta_4$ |
|------------|------------|------------|------------|
| 1.74       | 10.45      | 45.88      | 41.93      |
Table 3. Estimates for the natural frequency, damping ratio and normalized modal parameters, in case of 4 and 3 modal nonlinear terms.

| Number of nonlinear terms | $f_1$ (Hz) | $\zeta_1$ | $c_1$ (Ns/m) | $k_1$ (N/m) |
|---------------------------|------------|-----------|-------------|-------------|
| 4                         | 18.9465    | 0.0134    | 89.4591     | $3.9751 \times 10^5$ |
| 3                         | 18.9218    | 0.0143    | 97.7688     | $3.9543 \times 10^5$ |

A subspace identification is carried out by considering $s = 6 \times 10^3$ samples and $i = 150$ block rows [5]. The matrix $H_e(\omega = 0)$ is obtained as in equation (16); the absolute contribution of each nonlinear term with respect to the whole nonlinear force is evaluated as in equation (18) and shown in table 2.

Since $\delta_1$ is negligible with respect to the others, the nonlinear term $g_1(t)$ is discarded and a new identification procedure is performed with the remaining three modal nonlinear terms $g_2(t)$, $g_3(t)$ and $g_4(t)$. Estimates for the natural frequency, damping ratio and modal parameters are given in table 3 for both the identification procedures carried out (in case of 4 and 3 nonlinear terms): the modal parameters are computed, by assuming that the modal mass is theoretically equal to $m = \frac{\rho AL}{2}$.

Moreover, both the estimated nonlinear modal stiffness characteristics are shown in figure 2, against the linear one: they are almost overlaid, this furtherly justifying neglecting the cubic term. The estimated modal nonlinear internal force, evaluated at the maximum measured modal displacement, is equal to 67% of the corresponding linear internal stiffness force.

In order to verify the accuracy and validity of the identified model, it is possible to perform a numerical simulation starting from the estimated parameters, by considering the system as subject to known initial conditions. This way, a reconstruction of the output is obtained and it can be compared with the measured modal displacement. In figure 3 this comparison is shown, together with the output reconstructed by carrying out a classical linear identification by SSI [10]. An excellent level of accuracy is observed for the nonlinear reconstruction, while the linear one is inadequate in estimating the amplitudes and the frequency of the system, especially towards the end of the decay as shown in detail in figure 3.

Figure 2. Estimates of the nonlinear contribution to the modal stiffness curve. The two nonlinear functions are almost overlaid. The linear one is also shown.
In the end, it is possible to notice that the identified model can then be used to predict the behavior of the system starting from different initial conditions. For example, a second “lift and release” excitation has been produced for the same beam, with a similar level of response. In figure 4 the measured modal displacement is compared against the one predicted by considering the previously estimated parameters. A good agreement can be observed.

5. Conclusions
In practice most structural nonlinearities are distributed and an ideal nonlinear identification method should cater for them as well as for lumped nonlinearities. In order to extend the current method to be applied also on realistic large engineering structures, a modal counterpart of the NSI method is proposed in this paper, together with ideas for handling a large complex nonlinear system as simple modal single degree of freedom systems.

The modal NSI technique is applied on a reinforced concrete beam, showing a softening nonlinear behaviour, set up in Pescara, Italy. The identification procedure has been performed at first to investigate about the characterization of the modal nonlinearity and its force contribution extent. Then, estimates for the modal parameters are obtained and used to compare the measured and the reconstructed output. A good level of accuracy is observed, while a linear identification is inadequate in estimating the amplitudes and the frequency of the system. In the end, the identified estimates are also used to successfully predict the behaviour of the same beam, when subject to different initial conditions.

References
[1] Kerschen G, Worden K, Vakakis A F and Golinval J C 2006 Past, present and future of nonlinear system identification in structural dynamics Mechanical Systems and Signal Processing 20 505-92
[2] Richards C M and Singh R 1998 Identification of multi-degree-of-freedom non-linear systems under random excitations by the reverse-path spectral method Journal of Sound and Vibration 213 673-708
[3] Garibaldi L 2003 Application of conditioned reverse path method Mechanical Systems and Signal Processing 17 227-235
[4] Adams D E and Allemang R J 2000 A frequency domain method for estimating the parameters of a non-linear structural dynamic model through feedback Mechanical Systems and Signal Processing 14 637-656
[5] Marchesiello S and Garibaldi L 2008 A time domain approach for identifying nonlinear vibrating structures by subspace methods Mechanical Systems and Signal Processing 22 81-101

[6] Gandino E and Marchesiello S 2010 Identification of a Duffing oscillator under different types of excitation Mathematical Problems in Engineering Volume 2010 Article ID 695025, 15 pages, doi:10.1155/2010/695025

[7] Platten M F, Wright J R, Dimitriadis G and Cooper J E 2009 Identification of multi-degree of freedom non-linear systems using an extended modal space model Mechanical Systems and Signal Processing 23 8-29

[8] Bornn L, Farrar C R and Park G 2010 Damage detection in initially nonlinear systems International journal of Engineering Science 48 909-20

[9] Bellino A, Garibaldi L and Marchesiello S 2011 Determination of moving load characteristics by output-only identification over the Pescara beams Proc. of Damas 2011 submitted

[10] Van Overschee P and De Moor B 1996 Subspace Identification for Linear Systems: Theory, Implementation, Applications (Boston - London - Dordecht: Kluwer Academic Publishers)