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Numerical investigation of System Ringing and Stress Oscillations in High Speed Tensile Test on Polymers

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Abstract

The Finite Element (F.E.) analysis of a grip system of a servo hydraulic machine used for high speed tensile testing of polymers is here presented. State-of-the- art dynamometers use a slack adaptor to allow the machine to accelerate, in order to reach the nominal tensile speed before the load is applied, so that the entire test is conducted at constant speed. In this case the sudden application of the load causes the onset of stress waves that can excite the system, causing it to oscillate at its natural frequency. This phenomenon, called “system ringing”, is here examined through the analysis of the frequencies characterizing the oscillations in the initial transient of the tensile test. It will be shown that this analysis can be a suitable tool for assessing what is the maximum strain rate achievable with a given equipment. We also show that the stress oscillations contain information that can be related to material properties and, consequently, can be used for their measurement through simple spectrum analyses. Some examples are reported for an easy determination of the Young modulus of a Polypropylene (PP) based compounds from the Fast Fourier Transform (FFT) of the force signal.

Keywords: high speed tensile tests; system ringing; FFT; strain rate

1. Introduction

The experimental techniques for the characterization of polymers at high strain rates are becoming a topic of particular relevance due to the increasing request of strain rate dependent curves as input for dynamic finite element analysis, which has become the dominant method for design in several industrial environments, in particular in automotive. It is recognized [1,2,3,4,5] that one main issue in tensile testing is system ringing, which may pose limitations in the capability of generating acceptable data.

In the dynamic tensile test, when a slack system is used, the load is applied to the specimen only after the tensile machine has accelerated to the desired speed; in this case the sudden engagement of the mobile part of the machine with the load train (i.e. the load path from the test specimen to the load cell) can generate high amplitude stress waves which can excite the system, causing it to oscillate ai its natural frequency (system ringing).

Xiao et al [1] studied the response of a dynamic tensile test system subjected to the sudden application of the tensile load, modelling it as a simple mass-spring system excited by an impulsive force *f*. Under free vibration, the response of the system, assumed as initially at rest, is a sinusoidal oscillation at its natural circular frequency ω_n. Useful considerations highlighted in that study were that:

- the amplitude of the oscillation is linearly proportional to the loading rate (i.e. to the tensile speed).
- the higher the system natural frequency, the lower the magnitude of the oscillation.
- when damping is present, the natural frequency determines the decay of the ringing

These observations clarify why system ringing becomes critical at high speed and why higher system natural frequencies are sought to minimize ringing effects.

A more detailed analysis of the load train was given by Yang et al. [2]. Here, with the aid of a simple 1 DOF spring-mass-damper model to represent the load train, it was shown that, regardless of the damping ratio, the system overresponses are definitely quite limited when input signal frequencies are lower than about 0.25 times the system natural frequency. Otherwise, especially for relatively low damping ratios, ringing may become significant, causing the stress values returned by the load cell to be scarcely accurate. Based on these considerations, a critical frequency $_c$ was recognized as $\omega_c = \omega_n /4$, and the resulting criterion for not having ringing artifacts was then $\boldsymbol{\omega < \omega_c}$, or

$$\omega < \omega_n/4 \qquad (1)$$

In eqn. (1) the circular frequency ω is the frequency of the load signal, i.e. the frequency contained in the force signal acting on the load cell. Consequently, eqn. (1) poses limitations in the maximum strain rate which can be tested with a given apparatus without generating significant ringing. An approach for converting strain rate into frequency was also proposed in [2]. Here, the circular frequency of the testing signal was computed as:

$$\boldsymbol{\omega} = \frac{\boldsymbol{\pi}}{\boldsymbol{\Delta t}} \qquad (2)$$

where Δt represents the load rise time. The assessment of this value was there reconducted to a conventional quasi static test, based on the assumption that there is no change in the uniform elongation over the range of strain rate of interest, leading to the relationship:

$$\dot{\boldsymbol{\varepsilon}}_{\boldsymbol{d}} = \dot{\boldsymbol{\varepsilon}}_{\boldsymbol{q}} \frac{\boldsymbol{\omega_d}}{\boldsymbol{\omega_q}} \qquad (3)$$

where the suffix "*d*" refers to dynamic conditions (i.e. to the high strain rate) and the suffix "*q*" refers to the quasi static test.

Note that eqn. (2) expresses the concept, as remarked in standard [8], that the load signal up to yield is approximated as the first quarter of a sine wave. Based on these considerations, the standard [8] reported, for several specimen geometries, a value of a maximum suggested nominal strain rate achievable with a system having a given natural frequency. Accordingly, for an ISO 8256 Type3 specimen, in order to achieve a nominal strain rate higher than 100 s^{-1}, a system natural frequency higher than 4000 Hz was recommended.

Many studies about the development of dynamic tensile test methodologies have been focusing on overcoming the limitations and issues described above. They have mainly been focusing on the design of specimens, measurement devices, testing procedures and equipment which is suitable for very high-speed tensile tests. Basic studies proposed methods for the identification of the system natural frequency, mainly based on two different experimental techniques. In one case [3] a modal analysis was proposed, by which the dynamometer was excited at some specific locations with an instrumented hammer, and then the machine vibrations were detected by means of accelerometers attached to the structure. The frequency analysis of the response, obtained through the FFT (Fast Fourier Transform) of the measured signal, allowed the identification of the system frequencies.

An alternative approach for determining the natural frequency of the system was used by Cheresh et al [10], based on the analysis of the fluctuations in the force signal recorded after the fracture of a strong but brittle specimen. After specimen break, the system would continue to oscillate at its natural frequency. The analysis of the recorded data allowed determining the system natural frequencies.

The analysis of tensile machines was in some cases oriented to improving their design, as by Fitoussi et al. [6], where a damping system based on a nitrile damping joint was used to limit the measurements perturbation, or by Quin et al. [11], who proposed a load sensor based on a Digital Image Correlation method applied to a gage obtained on a portion of the specimen.

The investigation in this latter paper made also use of F.E. (Finite Elements) analysis. Finite Elements simulation have also been used as investigation tool in other papers. Grams [12] modelled the entire train load with Finite Elements, and the transient subsequent to the load application was studied in the time domain, resulting in an optimal design of the specimen for minimizing the oscillations. A similar purpose was pursued in [6], where the dumbbell optimal shape obtained from the optimization is verified with two composite materials. Here, the propagation of the stress waves within the specimen during the tensile test was also studied numerically. Finally, in [11], a numerical simulation of a simplified test system was used to verify system ringing in impact tests triggered by the yield point of a material, which in that case was a steel. Interesting is that this is one of the few papers where it is recognized that system ringing may be affected by the properties of the tested material, which could emphasize the oscillations making them difficult to overcome.

The problem is definitely complex and is related to the propagation of stress waves in the specimen, for which we report the basic formulas that will be discussed in the following.

The time of travel for one reflected stress wave in the specimen can computed as [1,8]:

$$t_{wave} = \frac{2L_g}{c} \qquad (4)$$

where L_g denotes the distance between the grips, c is the speed of the sound in the specimen material, which corresponds to the speed of the elastic stress wave and can be accordingly computed as $c=(E/\rho)^{1/2}$ for an infinite medium, being E the Young modulus and the density of the tested material. More precisely taking into account geometric effect, the speed of the longitudinal wave c_L is [15]:

$$c_L = \sqrt{\frac{E_L(1-\upsilon)}{\rho(1+\upsilon)(1-2\upsilon)}} \qquad (5)$$

where E_L is the elastic modulus in the longitudinal direction and νthe transverse to longitudinal contraction ratio (Poisson coefficient). Similarly, the same formula with adjusted parameters can be used to determine the speed of the transversal wave c_T. Alternatively, the transverse wave speed can be computed from [15]:

$$\nu = \frac{1-2(c_T/c_L)^2}{2-2(c_T/c_L)^2} \qquad (6)$$

Note that eqn. (5) and (6) are only valid within the elastic domain. In plasticity wave propagation needs to be taken into account differently [7].

In this paper we present the F.E. analysis of the initial transient of a high speed tensile test carried out with a servo hydraulic dynamometer, where the system under study comprises the grip system. The equipment here modelled is the one currently used in our testing activity on thermoplastic materials, as polypropylene compounds, polyamide compounds or others. The main tool for this investigation is the analysis of the spectrum of the stress fluctuations recorded in the transient.

One purpose of this study is to verify whether the same system can be operated at higher speeds, from the viewpoint of the issues presented in the previous chapter. One other purpose of the study is verifying if the numerical tool can be suitable for the design of a grip system and which information can be obtained therefrom.

In the last part of the paper, a quite different viewpoint will be adopted. In fact, machine designers always consider the oscillations in the test as noise, and as such the main purpose of any design improvement is reducing them or, at least, getting rid of them. Vice versa, as previously highlighted, stress oscillations may be affected by test parameters and material properties. Hence, what on one side is seen as noise may be seen as information on the other side. A method will be shown by which information on the material elastic modulus can be extracted from the study of the oscillations in the test response.

2. Materials and Methods

The approach of this study is numerical, as based on F.E. modeling of the tensile specimen mounted on grips. The system considered in the simulation is depicted in fig.1; this is mounted on a MTS 810 frame, equipped with control electronics and hydraulics suitable for a maximum speed of 200 mm/s, which is not modelled but considered as perfectly rigid.

The embodiment in fig. 1 is represented along the horizontal direction only for sake of simplicity, but it will be obviously mounted on the tensile dynamometer with his axis along the vertical direction. A specimen **6** is firmly held by an upper grip **3** and a lower grip **2**, through the screws **8**. A pin **7** is used to connect the upper grip **3** to a rod **4**, which on its opposite side will be firmly connected to the load cell **5**. The length of the rod **4** will be sufficient to allow the load cell **5** to be mounted on the outside of a climatic chamber (not shown in the drawing). A connector **11** is used to couple the lower grip **2** to a load transferring device **1**. Said device **1** is connected on the opposite extremity to the mobile part of the tensile machine, and hence is moved by the hydraulic actuator. The device **1** is designed such to allow the actuator to move for a certain time before engaging the connector **11** and hence the specimen; by so doing the dynamometer has time enough to accelerate to reach the desired nominal speed before the load is applied to the specimen.

The system in fig. 1 was accurately measured with a caliper, and a CAD geometry was then created therefrom and meshed. The mesh size was about 1.5 mm to 2 mm in the metallic part, about 0.5 mm in the specimen, whose geometry corresponds to an ISO 8256 Type 3 with thickness of 3.15 mm. Tetrahedral solid elements were used in the metal, while mostly brick elements were used for the specimen when allowed by the local geometry. An example of the mesh in the area of the grip and of the load cell is in fig. 2. The effect of the mesh size and element formulation was studied as well. Parabolic elements were chosen when more accuracy in the results was needed.

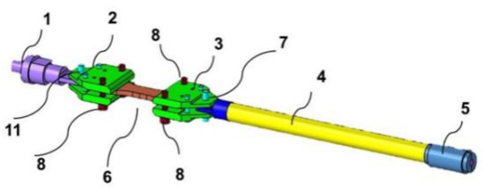

Fig. 1. Grip system

The engagement of the grip system with the load transferring device through the connector **11** were modelled in the F.E. simulation through contacts; the details of the coupling are not disclosed in this paper. However, we limit ourselves in mentioning that this is modelled through a contact that suddenly onsets after a certain free movement of the machine, occurring only after the load transferring device has reached the test speed; as such, the contact provides an impulsive loading on the specimen grips which models what effectively occurs in the reality. The shoulders of the specimen were coupled with the internal surfaces of the grips through rigid links or tyings [16], which were also used to model the fixing of screws on the grips. A boundary condition of imposed velocity was set at the load transferring device **1**.

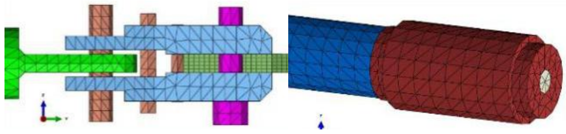

Fig. 2. Detail of the mesh in the area of the grips (left) and of the load cell (right)

The output which was considered was the reaction force at the nodes in correspondence with the load cell extremity and at the nodes in correspondence with the load application as well. The output curves were sampled every 1.E-06 s.; when FFT (Fast Fourier Transform) was applied thereon, the software OriginPro 8.6 by OriginLab was used. Before the application of the FFT, a Hanning window [13] for reducing frequency leakage was manually applied on the sampled data as:

$$w(i) = 0.5 - 0.5 \cos\left(\frac{2\pi i}{M}\right) \qquad (7)$$

where *w(i)* is a multiplicative term, *i* is an index which progressively numbers the data and *M* the total number of data (*i*=1..*M*).

The basic material properties considered in the first part of the work are extracted from stress-strain curves of some PP-based compounds in the portfolio of LyondellBasell [17], mainly consisting in the time for reaching the stress peak under static conditions, which are referred to tests at low strain rate as 1e-3 (1/s). Relevant properties and materials considered for this study are summarized in table 1.

Table 1. Virtual Materials and Related Properties Used in the Study of System Ringing

Material typolog	Orientation	T (°C)	Δt static (s)
GF30-PP	L	23	9
GF10-COPO	T	23	11
Talc filled, Impact modified COPO	T	23	12
GF30-PP	T	85	13
GF40-PP	L	23	9,25

40% talc filled PP	T	23	4,8

For the analysis of the influence of material properties on stress oscillations, specimens of several polymeric compounds were considered, including talc-reinforced, impact modified Polypropylene copolymer ("TPO"), a 50% glass reinforced Polypropylene ("50%GF-PP") and a 65% Glass reinforced humid Polyamide PA66 ("65%GF-PA66"). For the analysis of the frequency spectrum, linear elastic models have been used for these materials, with parameters mentioned in table 1, which were identified from F.E. simulation of the tensile test. All the data were derived from specimens cut from an injection molded plaque along the direction of the flow. A linear elastic model was used for the load train metallic parts, with a modulus of 210 GPa and a density of 7800 Kg/m^3. For all the simulations, explicit calculations were conducted using the solver Abaqus Explicit, release 2018 by Dassault Systemes.

Table 2. Virtual materials and related properties used in the study of stress oscillations

Material	E (Mpa)	Density (Kg/m^3)	Poisson ratio
TPO	1252	1040	0.34
50% GF PP,	9284	1340	0.3
65% GF PA66, moist	12921	1720	0.3

3. Results and Comments

3.1. Assessment of the maximum strain rate achievable

One goal of this study is to assess the suitability of the current grip system to execute tensile tests at high speed. In light of the limitations expressed by eqn. (5), the system natural frequency had to be determined. This has been done by introducing a failure criterion for the specimen, and then recording the force computed on the nodes of the load cell, hence simulating the praxis used in [10]. The resulting oscillating signal has been then submitted to FFT. The spectrum obtained can be seen in fig. 3. There, some peaks are clearly identifiable, at the frequencies indicated in the picture. It can be easily observed that those values are unusual. multiple of a first peak observed at a frequency of 4181 Hz within an acceptable degree of approximation. These peaks are then recognized as the odd harmonics related to a fundamental natural frequency of the system, namely f_n=4181 Hz. Even harmonics are not expected to develop due to physical boundaries.

According to eqn. (1), the critical circular frequency for this system is:

$$\omega_c = \frac{\omega_n}{4} = \frac{2\pi f_n}{4} = 6564 \text{ rad/s} \quad (8)$$

This value can be used to verify whether the system, which is considered here, is suitable – from the point of view of system ringing -for running tests at higher speeds. The type of material should also be considered.

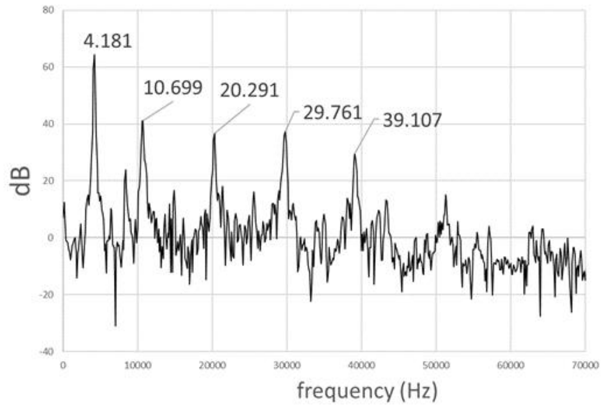

Fig.3. FFT Spectral analysis of the force signal at the load cell after specimen break. Results from FE analysis

The fundamental data have been collected in table 3, computed from the properties listed on table 1. The key values are the circular frequencies ω_d, which depend on the material, to be compared with the value of the system critical frequency

Table 3. Example of critical circular frequencies computed as in [2] from static data for the materials of Table 1. Cells with red background are those with circular frequencies exceeding the critical circular frequency

Material typology	ωq (rad/s)	ωd (rad/s)		
		100 mm/s	1000 mm/s	5000 mm/s
GF30-PP	0,35	349	3490	17444
GF10-COPO	0,29	285	2855	14273
Talc filled, Impact modif. COPO	0,26	262	2617	13083
GF30-PP	0,24	242	2415	12077
GF40-PP	0,34	339	3395	16973
40% talc filled	0,65	654	6542	32708

From the data in table 3 it can be observed that, based on the criteria therein considered, the execution of tests at 1000 mm/s would not suffer from significant system ringing for all the materials listed, while the same grip are not assessed as suitable for tests at 5000 mm/s. As easily seen from the values in the table, a limit value for these grips is around 2000 mm/s. We note that the outcome of the procedure, which is used to build up table 3, is definitely affected by the definition of the yield strain or stress. In table 3, according to [2], yield is associated to the end of uniform elongation on the specimen under static loading, which in turn is recognized as the point in the stress-strain curve where said curve starts decreasing. Different definitions of yield conditions, e.g. as the point on the stress-strain curve beyond which permanent deformations in the dumbbell are observed, would lead to different assessments. Therefore, the results reported on table 2 could be questionable, since they are definitely affected by the criterion for the assessment of the signal frequency which is used in the literature and standards and which appears to be vague and does not take into account the possibility of an earlier fracture. This causes some "dark" areas and inconsistencies in the response, as in the evaluation of the feasibility of the test on the talc filled material which appears being more critical than the glass-fiber reinforced grades, due to the occurrence of fracture. However, this approach can be used for comparing different solutions that could be considered in the machine design.

The system natural frequency, here found rigorously from the load signal spectrum after specimen break, is expected to characterize the system oscillations during the tensile test starting from the load application [18].

Considering the signal from the nodes of the load cell, we limited the attention to the transient for times lower than 0.005 s, where no plasticity is believed to occur and hence the approximation of linear elastic material laws is sensible. The force signal was interpolated through a fourth order polynomial; thereafter, from the signal the interpolant was subtracted and the remaining oscillatory signal was analyzed via FFT. The spectrum obtained with the materials of table 2 is overlapped in fig. 4 with the spectrum of the force signal after break, showing the details for a range of frequencies from 0 to 20000 Hz. As visible, the main peak previously identified at about 4181 Hz is easily recognizable also in the initial transient of the tensile tests of all the materials examined, as those peaks related to its harmonics. We observe therefore that the analysis of the very initial transient can be useful to provide information on the system natural frequency, and said frequency appears clearly as the first peak in the spectrum.

3.2. Assessing if the proposed methodology can be used as design tool

A further goal of this study is to assess whether the chosen methodology, based on the F.E. analysis of the grip system in the initial transient of the tensile test, can be used as a support in the design of the testing machine. It can be easily shown how the numerical tool can be used for exploring the design space in the effort of improving the tensile machine performance, through modifications in the embodiment of the grip system, by changing the material or the geometry. For a demonstrative example, assigning to the grips, originally made of steel, the properties of carbonium or – in an even confrontational study – of lead caused the system to change its natural frequency to 5200 Hz or 3000 Hz respectively. Similarly, a modification in the geometry halving the length of the rod connecting the upper grip to the load cell lead to an increase of the system natural frequency to 5230 Hz. Although these can be considered as mere exercises, they can show the potentiality of the method for estimating if a prescribed strain rate can be achieved with a given equipment, or which modifications in the testing apparatus are needed to reach the target.

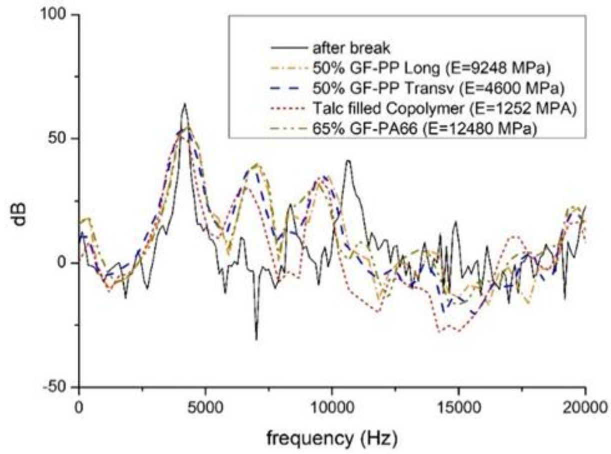

Fig. 4 Spectrum from load signal after specimen break compared to the spectrum from initial transient on 50% GF-PP material (E=9284 MPa).

Finally, we show that this methodology can be used to make sure that the test embodiment here considered is suitable for achieving a condition of dynamic equilibrium, which is critical for ensuring the validity of test results[1,5,7]. In fact, if the loading time can be comparable to the time needed for stress waves to propagate through the specimen, the stress no longer has time to evenly distribute itself over the specimen, thus determining non homogeneous stress distribution along it. This typically occurs for very high testing speeds. The numerical simulation shows that effectively this situation is achieved for the class of materials here examined, as visible from the stress distribution plotted along the specimen axis in fig. 5. The values are here normalized with respect to the peak stress, and taken at t=8.0E-05 s after the onset of the load application in a case at 1000 mm/s, which corresponds to an engineering strain of about 2E-03, i.e. within material elastic behavior.

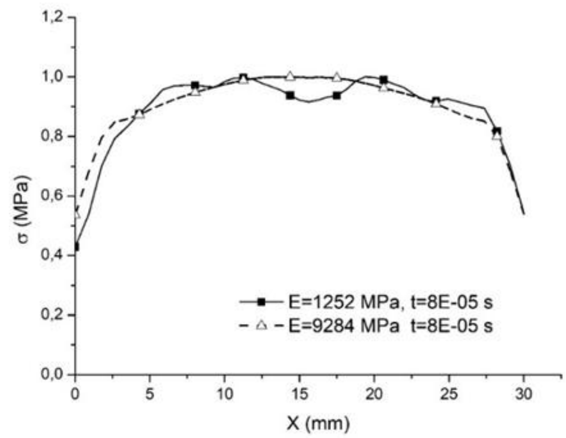

Fig. 5. Stress distribution in the specimen for two grades after 8E-05 s in the test at 1000 mm/s

3.3 Identification of material properties through the analysis of stress fluctuations

In this section we will examine in detail the response of the system subjected to the impulsive loading which is generated when the load is suddenly applied after machine acceleration. Reference is made to fig. 6-a, which shows the frequency spectrum observed at the load cell in the simulation of the tensile test of the GF50% grade where linear properties are assumed. The data refer to the first 5 ms of the transient.

The system's natural frequency is now assessed from FEM analysis as f_1=3846 H. This value is slightly different from the one previously reported due to the use of parabolic rather than linear finite elements. This was mandatory for this specific study, despite the larger computational times requested, in order to have more accurate results, which is requested for the identification of the peaks in the spectra.

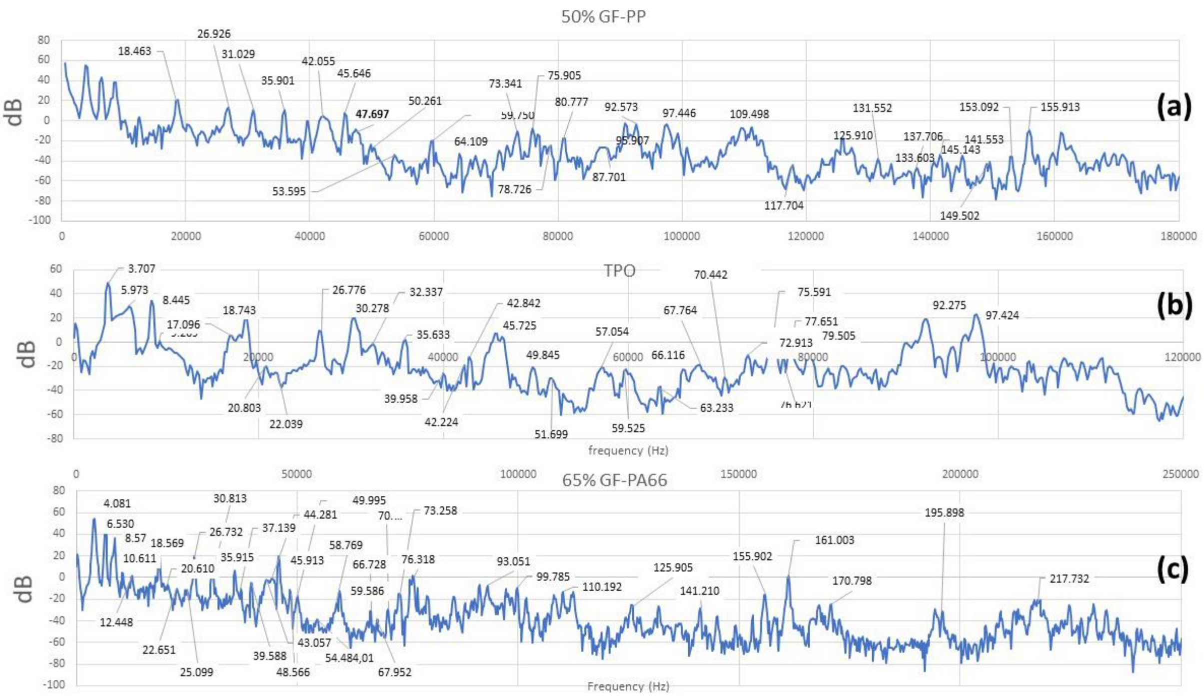

Fig. 6. Identification of the peaks in the FFT of the force for the test on (a) 50%GF PP- (b) TPO; (c) 65% GF-PA66

We use the value of the natural frequency to compute the related odd harmonics listed in Table 3. The resulting frequencies have been identified on the signal spectrum, with an error in general less than 5%, as visible from the related column in said table. In this list, frequencies with discrepancies higher than 5% were not associated to any harmonics.

At the end of this procedure, the examination of the frequency spectra in fig. 6- allows associating most of the peaks with odd harmonics of the three lowest frequencies. It may happen that some harmonics cannot be identified; these are only a few, and they may be hidden by harmonics of another of the frequencies considered. Vice versa, some peaks cannot be reconducted to harmonics of the three first frequencies. A detailed discussion is reported from the spectrum related to 50% GF_PP, in fig. 6(a), but same conclusions, omitted for brevity, can be drawn from the analysis of the spectra of the other two materials in fig. 6 (b) and 6 (c). For the spectrum in fig. 6(a) clear undefined peaks are at 12309 Hz, 47607 Hz, 75905 Hz, 145143 Hz and 153092 Hz, while a “twin peaks” unresolved region is at 90778 Hz and 92573 Hz.

Interesting is that these peaks can be related to the properties of the tested material assuming a maximum deviation from theoretical values of 5%. Given a theoretical specimen stiffness as $k= A_0 E/L_0$, with the data of this material a characteristic angular frequency for the specimen as $w=(\mathrm{k}/m)^{1/2}$ is computed as 85644 rad/s, resulting in a specimen natural frequency of 13630 Hz, which could be associated to the peak observed from FEM analysis at 12309 Hz. For this calculation the mass *m* of the specimen is considered as the mass of the specimen not including the specimen shoulders. Harking back to the considerations related to the stress waves propagation in the sample, it can be observed that some peaks in the FFT spectrum can be related to the wave speed travelling back and forth between the grips. From eqn. (5) the longitudinal wave speed can be computed, obtaining c_L=3048 m/s. It was also verified that this value is effectively occurring in the F.E. simulation, considering the stress distribution with respect to a coordinate X disposed along the specimen axis at different time instants during the initial transient; the wave propagation speed was obtained as the ratio between the difference in the X coordinate of two stress peaks taken at two consecutive instants. By so doing we obtained a value of about 3100 mm/s.

Table 4. Odd Harmonics of the system natural frequency and related peaks identified in FFT spectrum within a 5% deviation from test with GF-50% PP Longitudinal

Harmonic Frequency (Hz)	*Harmonic index*	Observed frequency (Hz)	difference /%)
3846	1		
11538	3	*12309*	*7*
19230	5	18463	4
26922	7	26926	0
34614	9	35901	4
42306	11	42055	1
49998	13	50261	1
57690	15	59750	4
65382	17	64109	2
73074	19	73341	0
80766	21	80777	0
88458	23	87701	1
96150	25	97446	1
103842	27	99241	4
111534	29	109498	2
119226	31	117704	1
126918	33	125910	1
134610	35	133603	1
142302	37	141553	1
149994	39	149502	0
157686	41	155913	1

Table 5. Odd Harmonics of the system second frequency and related peaks identified in FFT spectrum within a 5% deviation from test with GF-50% PP Longitudinal

Harmonic Frequency (Hz)	*Harmonic index*	Observed frequency (Hz)	*difference (%)*
6411	1		
19233	3	20258	5
32055	5	31029	3
44877	7	45646	2
57699	9		
70521	11	70520	
83343	13		
96165	15		

From eqn.(4), the frequency associated to the longitudinal stress wave propagation can be computed as the reciprocal of the period, giving a frequency f_L=50900 Hz. This value can explaine the unresolved peak at 47697 Hz; twice this value, corresponding to the first even harmonic, namely about 95394 Hz, can explain the second unresolved peak region found at about 92573 Hz. Its third harmonic (FEM value 1343091 Hz) could be related to the peak at 145000 Hz about. Note that for this physical phenomenon the symmetry of the problem is believed to allow the onset of even harmonics.

Table 6. Odd Harmonics of the system second frequency and related peaks identified in FFT spectrum within a 5% deviation from test with GF-50% PP Longitudinal

harmonic frequency (Hz)	harmonic index	*observed from FEM frequency (Hz)*	*difference (%)*
8719	1		
26157	3	24874	5
43595	5		
61033	7	60262	1
78471	9		
95909	11	90776	5

In the range of frequencies considered, the peak at 75905 Hz could be also explained through stress wave propagation. In fact, the velocity of transverse waves c_T can be obtained from eqn.(6), obtaining c_T = 1407 m/s, which leads to a frequency f_{T1}= 70347 Hz, which explains the unresolved peak. The other unresolved peak at about 153 kHz can be recognized as the second harmonic of this one. If this could be experimentally verified, the frequency related to the transverse wave propagation is expected to be definitely lower due to material anisotropy.

At the end of this procedure, a set of frequencies related to the tested material properties have been identified. The analysis of the spectrum related to the TPO material leads to similar conclusions. The frequencies associated to the stress wave propagation have been identified with analogous considerations as f_L=22039 Hz (longitudinal wave, theoretical value 22687 Hz) and f_{T1}=67764 Hz (transversal wave along specimen width, theoretical value 68085 Hz). Additionally, the relatively low modulus allows showing, within the sampled frequencies range, a peak at about 90 kHz, which can be reconducted to the propagation of the stress wave along the thickness of the specimen, theoretically predicting a value of f_{T2}=85800 Hz

The third test on 65%GF-PA6, to which the spectrum in fig. 6 (c) refers, allowed to identify frequencies associated to the stress wave propagation as f_L=49995 Hz (longitudinal wave, theoretical value 52697 Hz), f_{T1}=66728 Hz (transversal wave along specimen width, theoretical value 72832 Hz)), f_{T2}=217 kHz (transversal wave along specimen thickness, theoretical value 231214 Hz).

The examples here reported show that F.E. analysis allows highlighting the information related to the material properties which is inherent in the force oscillations observed after the sudden application of the load. This analysis is certainly simplified, since a very rough linear material model is used. However, this allows identifying each single frequency and defining some possible strategies for deriving material properties therefrom. Peak frequencies sensitive to the material modulus could only be used to derive its value; other peaks could be used to investigate the Poisson ratio. When the material will enter into plastic behavior, the analysis will become more complex; the speed of the waves changes [7], and the peaks corresponding to wave propagation are expected to be found at different frequencies; additionally, second order harmonics from the natural system frequency could be found due to viscoelasticity. Plasticity, viscoelasticity and anisotropic material behavior will make the experimental spectra definitely more complex. However, the present analysis can provide some guidance in the application of this methodology to real testing. The identification of the peaks in the FFT spectrum carried out here could help identifying regions in the spectrum where the desired information is to be sought.

4. Conclusions

We showed that the Finite Element simulation of the tensile test on a polymeric sample, including in the analysis the grip system, can be a suitable tool for providing information about the strain rate achievable in a high-speed tensile test without the occurrence of significant system ringing. The methodology here presented is based on the analysis of the frequency spectrum of the force signal from the load cell and the identification of the system natural frequency therefrom. Said frequency is then compared with the load frequency, following criteria available in the technical literature and standards.

Some basic examples were proposed to show that this methodology can also be a useful tool for the design of specimens, measurement devices, testing procedures and equipment suitable for very high speed tensile tests.

Finally, we showed how the information related to material properties as modulus and Poisson ratio can be identified from the spectrum of the force signal and accordingly be used for determining said properties. Although this investigation was limited to a simple linear elastic material model, it is believed that this could be provide guidance and be exploited in an experimental measurement.

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