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Analytical modeling enables explanation of paradoxical behaviors of electronic and optical materials and assemblies

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Abstract. Merits, attributes and challenges associated with the application of analytical modeling in electronics and photonics materials science are addressed, based mostly on the author's research during his tenure with Bell Labs, University-of-California, Portland State University, and small business innovative research (SBIR) ERS Co., USA. The emphasis is on practically important, yet often paradoxical, i.e., intuitively non-obvious, material behaviors. It is concluded that when material reliability is crucial, ability to effectively quantify it is imperative, and that analytical modeling is the most suitable, although never straightforward, technique to understand, explain and quantify material behaviors, especially in extreme, extraordinary and paradoxical situations.

Keywords: analytical modeling; thermal stresses; assemblies comprised of dissimilar materials; electronics and photonics packaging

1. Introduction: analytical modeling, its role, significance, attributes, challenges

Modeling is the major approach of any science, weather pure or applied (see, e.g., Chen *et al*. 2016, Suhir *et al*. 2008, Suhir *et al*. 2011, Suhir 1997b, 1999b, 2000a, b, 2011a, b, c, d, 2012a). Modeling could be experimental, when the experimental setup is put together depending on what is anticipated to observe and measure (see, e.g., Bar-Cohen and Kraus 1988, Deletage *et al*. 2003, Zhang *et al*. 2006, Zhou *et al*. 2009, Fan and Lee 2010, Shi *et al*. 2010, Zhou *et al*. 2010, Fan 2014, Fan *et al*. 2015, Fan *et al*. 2015), or theoretical. Experimental models (specimens) are typically of the same physical nature and, in electronics and photonics, on the same scale, as the actual objects. Theoretical models use abstract notions. Their ultimate goal is to reveal non-obvious, latent, even paradoxical, relationships hidden in the input information. No theoretical model can provide results, which are not contained in the input data and in the taken assumptions and hypotheses. Experimental models, on the other hand, can occasionally lead to new results. A famous example is the surprise discovery of radiation by Antoine Henri Becquerel in 1896 that gave birth of nuclear chemistry.

Theoretical models can be either analytical or numerical (computational). Analytical models

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employ mathematical methods of analysis. The today's numerical models are, as a rule, computer-aided. The most widespread model in the stress-strain evaluations and physical design for reliability of electronic and photonic materials and systems is finite-element analysis (FEA) (see, e.g., Deshayes *et al*. 2003, Dandu *et al*. 2010, Fan and Suhir 2010, Fan and Ranouta 2012, Fan *et al*. 2014, Sun *et al*. 2016, Chen *et al*. 2017). This method strongly depends on the use of computers and went a long way from its initial application in the 1960s to avionic structures (by J.H. Argyris and his associates at the University of Stuttgart in Germany) to wide applications in material science and reliability physics in a wide variety of engineering disciplines and efforts (see, e.g., Sun *et al*. 2017, Sun *et al*. 2016, Shen *et al*. 2016).

Experimental and theoretical models should be viewed as equally important for the design of a viable, reliable, and cost-effective product. In aerospace, civil, ocean, and in many other areas of engineering, experimental and theoretical models are indeed considered as equal partners and complement each other in any significant effort. In electronics and photonics engineering the incentive for the development and use of theoretical models is not as critical, as it is in the macro-engineering world. The overwhelming majority of studies dealing with the physical design and performance of high-tech materials and products are experimental. There are several natural and good reasons for that:

1) Experiments could be often carried out with full autonomy, i.e. without necessarily requiring theoretical support;

2) Unlike theory, testing can be used as a final proof of the viability and reliability of a product and is therefore essential requirement, when it comes to making a viable electronic or a photonic device into a reliable product;

3) Experiments in the high-tech field, expensive as they might be, are considerably less costly than, e.g., those in naval architecture, or in aerospace engineering, where "specimens", ship or avionic structures, might cost millions of dollars;

4) High-tech experimentations are much easier to design, organize, and conduct than in the macro-engineering world;

5) Materials, whose properties are not completely known, are often and successfully employed in high-tech products; because every five years or so, new generations of electronic or photonic products are developed, there is often simply not enough time to establish all the properties and understand the behavior of materials in these fields of engineering; lack of information about the material properties is often viewed as an obstacle for carrying out theoretical modeling, including FEA, but not for implementing the materials themselves;

6) Many leading specialists in high-tech engineering (experimental physicists, materials scientists, chemists, chemical engineers) traditionally use experimental methods as their major research tool. It is not just a coincidence that eleven out of twelve Bell Labs Nobel laureates in physics were experimentalists.

Experimental investigations, unlike theoretical modeling, require, as a rule, considerable time and significant expense. What is even more important, is that experimental data inevitably reflect the effect of the combined action of a variety of factors affecting the material, the phenomenon or the product of interest, and, because of that, experimental evaluations, important as they are, are often insufficient to understand the underlying physics of the behavior and performance of a material or a device. Such a lack of insight inevitably leads to tedious, time-consuming and costly efforts. Experimental data cannot be simply extended to new situations or designs that are different from those already tested. It is always easy to recognize purely empirical relationships obtained by formal processing of experimental data: these relationships often contain fractional exponents and coefficients, odd units, etc. Although such relationships may have a certain practical value, the very fact of their existence should be attributed to the lack of knowledge in the given area of applied science. Typical examples are power law (e.g., the one used in proof-testing of optical fibers, when the time to delayed fracture, "static fatigue", is evaluated) and an inverse power law, such as, e.g., numerous relationships of Coffin-Manson type, when evaluating the lifetime of solder joint interconnections in electronic products.

A good theoretical and physically substantiated model could be as practical as the most thoroughly conducted experimentation. Here is what could be gained by using theoretical modeling in electronics and photonics materials science and engineering:

1) Predictive modeling, unlike experimentation, is able to shed light on the role of each particular parameter that affects the behavior and performance of the material or the product of interest;

2) Although testing can reveal insufficiently robust elements, it is usually incapable to detect superfluously reliable ones; over-engineered and superfluously robust products may have excessive weight and be more costly than necessary: in mass production of expensive high quality devices superfluous reliability may entail substantial and unnecessary additional costs;

3) Theoretical modeling can usually predict the result of an experiment in less time and at considerably lower expense than the actual experiment;

4) In many cases, theory serves to discourage wasting time on useless experiments; the classical example are numerous attempts to build impossible heat engines that have been prevented by a study of the theoretical laws of thermodynamics; while this is, of course, a classical and an outstanding example of the triumph of a theory, there are also numerous, though less famous, examples, when plenty of time and expense were saved because of prior theoretical modeling of a problem;

5) In the majority of research and engineering projects, a preliminary theoretical analysis enables one to obtain valuable information about the phenomenon or the object, and gives an experimentalist an opportunity to decide, what and how should be tested or measured, and in what direction success might be expected;

6) By shedding light on "what affects what", theoretical predictive modeling often serves to suggest new useful experiments. E.g., the theoretical analyses of stresses in bi-material assemblies and in semiconductor thin films (see, e.g., Suhir 1986a and Luryi and Suhir 1986) triggered numerous experimental investigations aimed at the rational physical design of semiconductor crystal grown systems;

7) Theory can be used to interpret empirical results, to bridge the gap between different experiments and to extend the existing experience on new materials, components and structures.

8) One cannot do without a good theory when developing rational (optimal) designs; the idea of optimization of structures, materials, and costs has penetrated many areas of modern engineering; no progress in this direction could be achieved, of course, without application of theoretical methods.

Analytical modeling (see, e.g., Huang *et al*. 2015, Luryi and Suhir 1986, Suhir 1989a, 1991a, 1997c, d, 2005a, 2009a, 2013a, 2014a, 2015a, b, Suhir *et al*. 2012, Ou *et al*. 2016, Ou *et al*. 2016) occupies a special place in the predictive modeling effort. Such modeling is able not only to come up with simple relationships that clearly indicate "what affects what", but, more importantly, can often explain the physics of phenomena and especially, as it is in this review, paradoxical situations much better than the FEA modeling, or even experimentation, can. It should be emphasized that since mid-1950s, FEA modeling has become the major simulation tool for theoretical stress-strain evaluations in materials, mechanical, structural, aerospace, maritime and other areas of engineering and applied science. FEA has become the major modeling tool in electronics and photonics materials science about twenty years later. This should be attributed, first of all, to the availability of powerful and flexible computer programs, which enable one to obtain, within a reasonable time, a solution to almost any stress-strain related problem, but partially also to the illusion that FEA is the ultimate and indispensable tool for solving any design or stress analysis problem. The truth of the matter is that FEA and broad application of computers has by no means made analytical solutions unnecessary or even less important, whether exact, approximate, or asymptotic. Simple and physically meaningful analytical relationships have invaluable advantages, because of the clarity and compactness of the information and clear indication on the role of various factors affecting the given phenomenon or the material's behavior or the device performance. These advantages are especially significant when the parameter under investigation depends on more than one variable. As to the asymptotic techniques, they can be successful in many cases, when there are difficulties in the application of computational methods, e.g., in problems containing singularities. Such problems are often encountered in high-tech materials engineering, because of the wide employment of assemblies comprised of dissimilar materials. But even when application of FEA encounters no difficulties, it is always advisable to investigate the problem analytically before carrying out FEA. Such a preliminary investigation helps to reduce computer time and expense, develop the most feasible and effective preprocessing model and, in many cases, avoid fundamental errors.

It is noteworthy that FEA has been originally developed for structures with complicated geometry and/or with complicated boundary conditions (such as, e.g., avionics or some civil engineering structures), when it might be difficult to apply analytical approaches. As a consequence, FEA has been especially widely used in those areas of engineering, in which structures of complex configuration are typical (aerospace, maritime and offshore structures, some civil engineering structures, etc.) In contrast, electronic and photonic structures are usually characterized by simple geometries and can be easily idealized as beams, flexible rods, rectangular or circular plates, composite structures of relatively simple geometry, etc. There is an obvious incentive therefore for a broad application of analytical modeling in electronics and photonics materials science and engineering. Additional incentive is due to the fact that adjacent structural elements in electronics materials engineering often have dimensions that differ by orders of magnitude. Examples are multilayer thin film structures fabricated on thick substrates or adhesively bonded assemblies, in which the bonding layer is typically significantly thinner than the bonded components of the assembly. Since the mesh elements in a FEA model must be compatible, FEA of such structures often becomes a problem of itself, especially in regions of high stress concentration. Such a problem does not occur, however, with an analytical approach.

Another consideration in favor of analytical modeling is associated, as has been mentioned above, with an illusion of simplicity in applying FEA procedures. Many users of FEA programs sincerely believe that the ”black box" they deal with will automatically provide the right answer, as long as they push the right key on the keyboard. At times, a hasty, thoughtless, and incompetent application of computers can result in more harm than good by creating an impression that a solution has been obtained, when, in effect, this “solution” might be simply wrong. It is well known that although it is usually easy to obtain ***a*** FEA solution, especially with the today's user friendly software, it might be quite difficult to obtain ***the*** right one. And how would one know that it is indeed the right solution, if there is nothing to compare it with? Clearly, if the FEA data are in good agreement with the results of an analytical modeling (which is typically based on different assumptions: FEA is a numerical continuum mechanics tool, while the available close-form analytical solutions use mostly approximate structural analysis and strength-of-materials methods), then there is a reason to believe that the obtained solution is accurate enough.

A crucial requirement for an effective analytical model is its simplicity and clear physical meaning. A good analytical model should be based on physically meaningful considerations and produce simple relationships, clearly indicating the role of the major factors affecting the phenomenon or the object of interest. One authority in applied physics remarked, perhaps only partly in jest, that the degree of understanding of a phenomenon is inversely proportional to the number of variables used for its description. It takes a lot of imagination, intuition, appropriate assumptions and effort to come up with a meaningful analytical expression, while it is typically merely skill that is needed for the application of FEA simulation.

While an experimental approach, unsupported by theory, is blind, theory, not supported by an experiment, is dead. An experiment forms a basis and provides input data for a theoretical model, and determines its viability, accuracy, and limits of application. Limitations of a theoretical model are different in different problems and, in the majority of cases, are not known beforehand. It is the experimental modeling, which is often the "supreme and ultimate judge" of a theoretical model. The limitation of a particular theoretical model could be also assessed based on a more general model: limitations of a linear approach could be determined on the basis of a more general non-linear model. Experiment can often be rationally included into a theoretical solution to an applied problem. Even when some relationships and structural characteristics lend themselves, in principle, to theoretical evaluation, it is sometimes simpler to determine these relationships empirically. E.g., the spring constant of an elastic foundation provided by the primary coating of an optical fiber could be evaluated experimentally and then included into the analytical or a numerical predictive model.

In the review that follows some important, yet paradoxical, situations in the field of electronic and photonic materials science and engineering are addressed and explained using analytical modeling.

2. Review

2.1 Why do the interfacial thermal stresses concentrate at the assembly ends and, for a sufficiently large assembly, do not increase with the further increase in its size?

A microelectronic package (Fig. 1) is comprised of dissimilar materials with different coefficients of thermal expansion (CTEs). This results in elevated thermal stresses and, if special measures are not taken (such as, e.g., the use of metal frame in Fig. 1), also in elevated bow that can be an obstacle to the fabrication procedure and can affect the reliability of the second level of interconnections, the most vulnerable structural element in today's electronics technologies. The interfacial shearing and peelings stresses concentrate at the assembly ends (Figs. 2 and 3) and, for large enough assemblies, do not increase with the further increase in the assembly size (Fig. 4).

The engineering theory of bi-metal thermostats of infinite length (Timoshenko 1925) and of finite length (Suhir 1986a, 1989b) is widely used in electronics and photonics modeling effort. Timoshenko has indicated in his classical 1926 paper that while the stresses acing in the cross-sections of the thermostat strips can be predicted based on the strength-of-material (structural analysis) approach, the interfacial (shearing and peeling) stresses can be predicted only on the basis of the elasticity theory. Many attempts of mechanical engineers to do that led, however, to

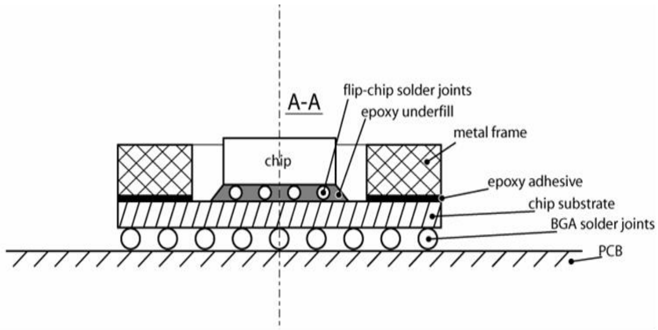

Fig. 1 Typical microelectronic package design. The surrogate metal frame is introduced to keep the package flat during processing and operation. The induced stresses in the package materials could be, however, quite high

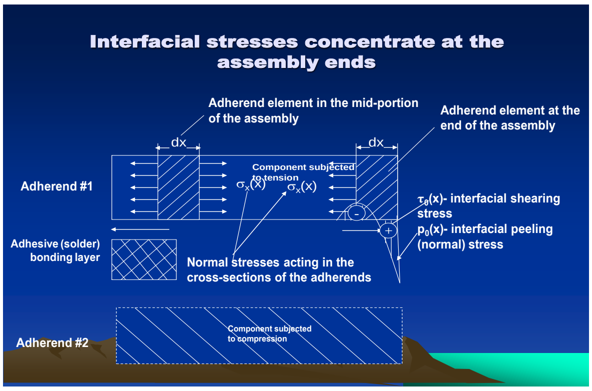

Fig. 2 The segment in the middle of the assembly is equilibrated by the normal stresses only, while the segment at the assembly ends needs interfacial shearing and peeling forces/stresses to be equilibrated

cumbersome analytical expressions. Suhir generalized Timoshenko theory for the case of finite size assemblies, which are typical in electronics and photonics. Suhir's theory (Suhir 1986a, 1989b) is based on the concept of interfacial compliance. Unlike Timoshenko's model, this theory enables to evaluate the interfacial shearing and peeling stresses, and to demonstrate that the maximum values of these stresses increase with an increase in the assembly size for short assemblies with compliant bonds, but remain unchanged for large assemblies with stiff interfaces. A simple, easy-to-use and physically meaningful formula $\tau(x) = kT\dfrac{\sinh kx}{\cosh kl}$ has been obtained for the interfacial shearing stress. Here $T = \dfrac{\Delta\alpha\Delta t}{\lambda}$ is the thermally induced force in the mid-portion of a long enough assembly, $\Delta\alpha$ is the CTE difference of the assembly components, Δt is the change in temperature, $\lambda = \lambda_1 + \lambda_2$ is the total axial compliance of the assembly, $\lambda_1 = \dfrac{1-\nu_1}{E_1 h_1}$ and $\lambda_2 = \dfrac{1-\nu_2}{E_2 h_2}$ are the axial compliances of its components, h_1 and h_2 are the component

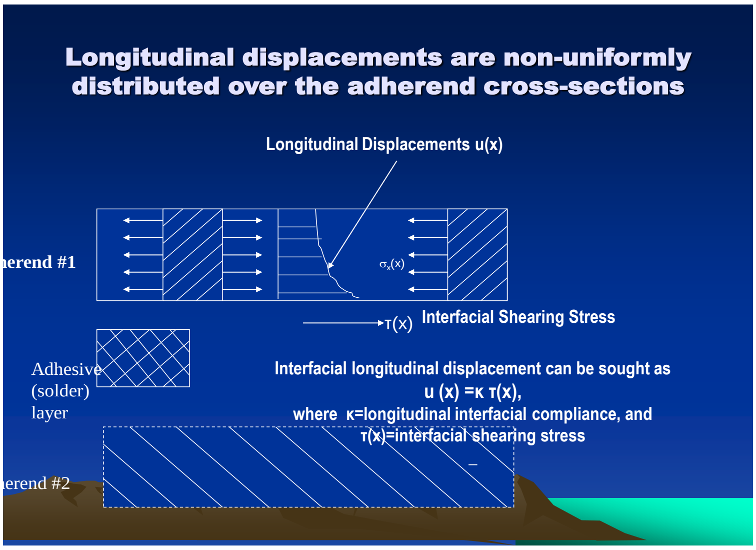

Fig. 3 Rationale behind the simplest analytical stress model: the longitudinal interfacial displacements are somewhat greater than the displacements of the inner points of the cross-section. The approximate stress model assumes that these displacements can be evaluated as a product of the interfacial position-independent compliance of the assembly and the interfacial shearing stress in the given cross-section. This stress changes from zero at the mid-cross-section of the assembly to its maximum value at the assembly end

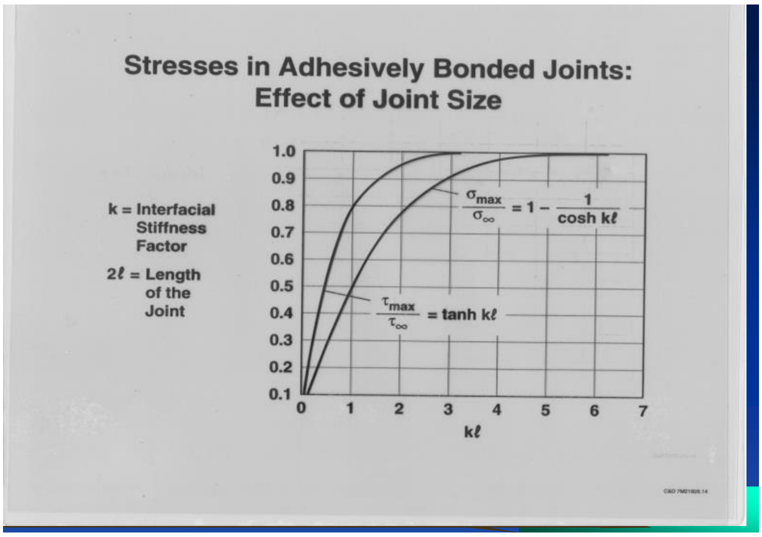

Fig. 4 If the induced stresses are already high, they do not increase with the further increase in the assembly size, if the product *kl* of the parameter *k* of the interfacial shearing stress and half the assembly length *l* exceeds the level $kl \approx 2.5$.

thicknesses, E_1 and E_2 are Young's moduli of the component materials, ν_1 and ν_2 are their Poisson's ratios, $k = \sqrt{\frac{\lambda}{\kappa}}$ is the parameter of the interfacial shearing stress, $\kappa = \kappa_0 + \kappa_1 + \kappa_3$ is the total interfacial compliance of the assembly, $\kappa_0 = \frac{h_0}{G_0}$ is the interfacial compliance of the bonding layer, $\kappa_1 = \frac{h_1}{3G_1}$ and $\kappa_2 = \frac{h_2}{3G_2}$ are the interfacial compliances of the assembly components, $G_0 = \frac{E_0}{2(1+\nu_0)}$, $G_1 = \frac{E_1}{2(1+\nu_1)}$ and $G_2 = \frac{E_2}{2(1+\nu_2)}$ are shear moduli of the materials, and l is half the assembly length. The origin of the longitudinal coordinate x is at the mid-cross-section of the assembly. The maximum shearing stress in the approximate and simplified analysis takes place at the end of the bonded assembly and is expressed as $\tau(l) = \tau_{\max} = kT\tanh(kl)$. For assemblies characterized by the kl values exceeding, say, 2.5, this formula yields: $\tau(l) = \tau_{\max} = kT$, so that the maximum shearing stress becomes assembly size independent.

2.2 Is there an incentive for using low modulus bonding materials and materials with low yield stress in electronic assemblies?

Since the interfacial stresses concentrate at the assembly ends and because compliant bonds act as effective strain buffers between dissimilar materials in assemblies subjected to the change in temperature, there is an obvious incentive for using low modulus bonding materials and thick bonding layers, especially at the assembly ends, where the interfacial stresses concentrate. But how low is "low", how thick is "thick", and how large is the expected size of the inelastic deformations, if any, at the peripheral portions of the assembly? It has been shown (Suhir 2003a) that this size, which leads to low cycle fatigue condition during temperature cycling tests, can be established from the condition that the forces at the ends of the assembly's elastic mid-portion are equal to the forces applied from the inelastic peripheral portions and that the latter forces can be evaluated as products of the yield stress in shear of the bonding material and the length of the peripheral inelastic region. The calculated distribution of the interfacial shearing and peeling stresses, when a low-yield-stress material is employed as a suitable bond to attach a highly vulnerable low-expansion photonic chip to a high-expansion copper sub-mount (heat sink), is shown in Fig. 5. The situation is similar, if it is not a bond, but a thin coating layer (Suhir 2001d) is considered, or when a low-yield-stress solder is used as the bonding material (Suhir 2006). In the latter case the bonding material relieves the stresses at the assembly ends to the level of the yield stress. This might not be favorable for the bonding material, but might be nevertheless advisable for a vulnerable chip.

It has been shown also (Suhir 2001e) that one effective way to bring down the thermally induced interfacial shearing stress at the ends of an adhesively bonded or soldered assembly is to employ a design, in which a bonding material with a high modulus is used in the mid-portion of the assembly, while a low modulus bond is used at its peripheral portions. No significant stress relief can be expected, if Young's modulus of the peripheral low modulus bond is too low (in the extreme case of a zero modulus, this material will not play any role at all) or too high (say, equal to the Young's modulus of the high modulus adhesive material in the mid-portion). The most effective compromise, as far as the maximum stress is concerned, should consider the lengths and the moduli of the bonding materials in the mid-portion and at the peripheral portions of the assembly. What is even less obvious, is that the CTE of the bonding material plays no role, as long as the bonding layer is significantly thinner than the bonded components and/or has a considerably

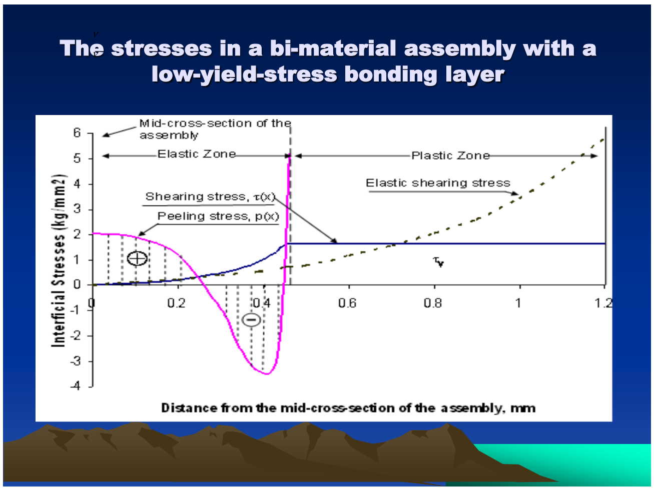

Fig. 5 The interfacial shearing stress increases at the assembly mid-portion from zero at the assembly mid-cross-section to its yield value, and then remains equal to the yield stress in shear

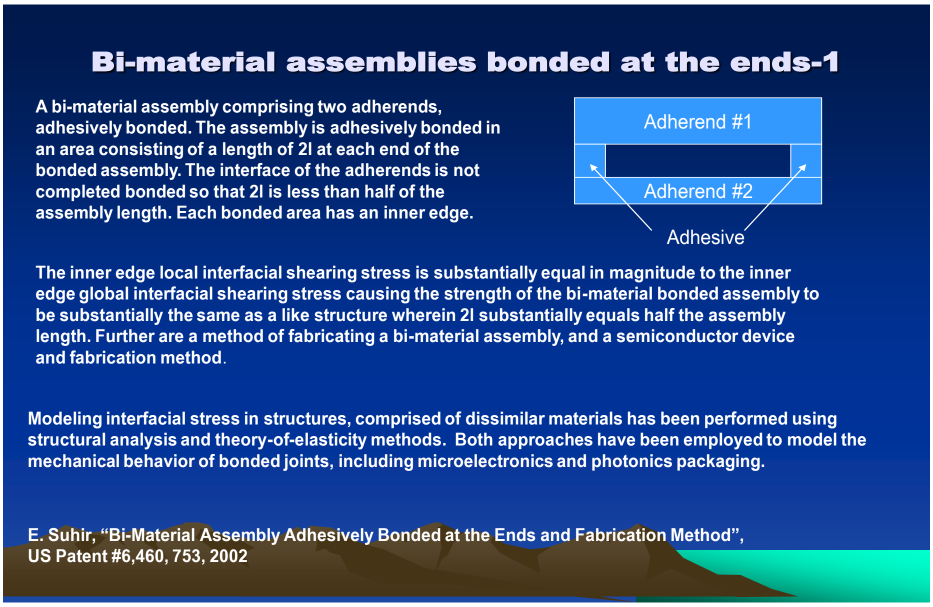

Fig. 6 Bi-material assembly bonded at the ends: predictive stress modeling

lower modulus. This is indeed the case in a typical adhesively bonded or soldered assembly in electronics and optics. This is not true in the case of a polymer coated fiber with a low modulus coating at the ends, when both the fiber and its coating are equal “players”.

The case of an assembly bonded at its ends (Fig. 6) could be regarded in a way as a sort of an opposite situation in comparison with an assembly with a low modulus bond at its ends. In an assembly bonded at the ends the Young’s modulus is finite, of course, at the peripheral regions, and is zero in the mid-portion of the assembly. It has been shown (Suhir 2002a) that if the peripheral bonding layers are long enough and comprised of high modulus materials, the induced stresses will not be different of those in assemblies with continuous bonding layers. This conclusion is particularly important when manufacturers try to necessarily bring by all means an encapsulation (underfill) material for the entire under-chip space. Such an effort makes sense, if good thermal management is important, but is not necessary from the standpoint of the level of the induced stresses.

2.3 Could transverse grooves in bonded components relieve interfacial stresses?

Jorma Kivilahti and Tomi Reinikainen (Espoo, Finland) have observed, using FEA modeling, significant decrease in the magnitude and the non-uniformity of the interfacial shearing stresses in test samples with transverse grooves in the bonded components ("pins"). This paradoxical phenomenon has been explained using analytical stress modeling (Reinikainen and Suhir 2009, Suhir and Reinikainen 2008, 2009, 2010): the detected stress relief is due to the increase in the interfacial compliance owing to the "conversion" of pin inner portions into additional layers of the bond. If the grooves are deep enough, the additional interfacial compliance of the bonding structure has a favorable effect on the magnitude and the distribution of the interfacial stresses. On the other hand, the grooves isolate the inner portions of the pins from the direct action of the external tensile forces, thereby increasing the axial compliance of the joint (see section 2.1). While the increase in the axial compliance has a negative effect on the interfacial stresses, the carried out numerical example has indicated that the favorable effect of the increased interfacial compliance suppresses the adverse effect of the increase in the axial compliance of the pins, so that the cumulative effect of the grooves is highly favorable. This effect, as determined by the analytical model, is, however, not as strong as the FEA predicts, but is still appreciable. This is because the FEA overestimates the stresses at the edge regions: at the very ends of the assembly even singularities (infinitely high stresses) might occur.

The observed and explained effect of the reduction in the level and uniformity of the interfacial stresses in grooved joints can be effectively used in the next-generation shear-off test methodologies for solder joint interconnections: such methodologies would enable obtaining more consistent and more stable test data.

2.4 Could thermostatic compensation in temperature-sensitive devices be achieved by employing regular materials?

A novel method for the compensation of temperature induced changes has been suggested in Suhir (2002a). In a preferred embodiment, the method and apparatus are directed toward compensation of fiber optic cable to remove the effects of temperature-induced changes, on system performance. The method can be applied to temperature-sensitive devices other than refractive index (Bragg) gratings in optical fibers: applications to a musical instrument string or a pressure relief valve are just a few examples. In general, any temperature-sensitive device, the performance of which may be adjusted by tension or compression, may be mounted on a thermostatic compensation device made according to the method of the invention to compensate for the change of performance with temperature. Materials suitable for use in fabricating the invention include ceramics, glass, Kovar, and Invar. To maximize the curvature of the thermostatic elements, bi-material thermostats which include a ceramic with a negative CTE may also be employed. The use of a ceramic material increases the difference in CTE of the two materials constituting the thermostatic elements and thus provides for greater curvature in response to temperature changes.

There are many applications where this method could be applied. E.g., the pitch of a string on a musical instrument can be controlled by maintaining string tension despite temperature-dependent lengthening or shortening of the string. The maximum pressure allowed by a release valve, such as those that may be found on common water heaters, can be controlled by mounting one side of the valve in juxtaposition with the thermostatic structure disclosed herein. In a preferred embodiment, a thermostatic structure, according to the invention, can provide a compensating adjustment in a length of optical fiber substantially equal, and opposite, of the temperature induced change in the length of the fiber. A particularly beneficial use of such an application is found in the stabilization of optical fiber refractive index gratings. One common use of such gratings is to reflect a specific narrow-band wavelength for separation of wavelengths in WDM fiber optic communication systems. Because the period of an optical fiber refractive index grating is subject to change with changes in length of the optical fiber, undesired temperature induced changes in the optical fiber, and the grating, adversely influence the separation of wavelengths and the subsequent performance of WDM systems. Application of the thermostatic structure of the invention to compensate for temperature induced changes in the optical fiber operates to stabilize the grating period and the subsequent performance of the grating.

2.5 Could compliant external leads reduce the strength of a surface mounted device?

Compliant external leads were widely used in mid-1980's to act as strain buffers between a rigid IC package and a flexible PCB subjected to bending. A paradoxical situation was observed, however, during testing of compliant leaded hybrid integrated circuits (HIC): in some tests the bending moment, applied to the printed circuit board (PCB) and causing HIC fracture, turned out smaller (not greater!), when leads of greater compliance were installed. It was shown (Suhir 1988c,d) that such a paradoxical situation was due to the redistribution of the lead reactions at certain combinations of HIC length, HIC and PCB flexural rigidity, and spring constant of the elastic attachment provided by the compliant leads. The analysis (confirmed by FEA) has indicated that only highly compliant leads can essentially reduce the stresses, while leads of moderate compliance can result in even greater stresses in the HIC than ideally stiff leads.

2.6 Could the curvature of a coated fiber be different from the curvature of its coating?

When proof-testing of dual-coated optical fibers one can easily measure the curvature of the secondary coating. But is the curvature of the glass fiber always and everywhere the same? Analytical modeling indicates that the curvature at the end portions of a bent coated fiber could be considerably different of the observed curvature of its coating. The analytical model based on the theory of beams of finite length lying on a continuous elastic foundation has shown that fiber curvature at the fiber specimen ends can be considerably larger than the observed curvature of its secondary coating (Suhir *et al*., Suhir 1993). The spring constant of the elastic foundation was evaluated based on an appropriate analytical model (Suhir 1988).The situation is, in a way, similar to the situation addressed in the section 2.5.

2.7 Is it possible to design, fabricate and operate a bow-free assembly?

It is certainly important to be able to predict and, if necessary, to minimize the warpage of an electronic or a photonic assembly (Suhir and Weld 1997, Suhir 1992c, 1995a, 2000b). It turns out that it is indeed possible to design and operate a bow free assembly (Suhir 1998a, 2001a, 2003b).

The bow-free condition for a tri-material assembly is

$$E_0^* E_1^* h_0 h_1 (h_0 + h_1)(\alpha_1 - \alpha_0) - E_0^* E_2^* h_0 h_2 (h_0 + h_2)(\alpha_2 - \alpha_0) + E_1^* E_2^* h_1 h_2 (h_1 + 2h_0 + h_2)(\alpha_1 - \alpha_2) = 0.$$

Here the zero component is the inner one, and the components #1 and #2 are the outer ones. The following notation is used here: $h_i, i = 0,1,2,$ are the components' thicknesses, $\alpha_i, i = 0,1,2,$ are the materials CTEs, $E_i^* = \frac{E_i}{1-\nu_i}, i = 0,1,2,$ are their effective Young's moduli, $E_i, i = 0,1,2,$ are the actual Young's moduli, and $\nu_i, i = 0,1,2,$ are their Poisson's ratios. A bi-material assembly with a thin and low modulus bonding layer is statically determinate and cannot be made bow-free. bending moment. A multi-material body should contain at least three components and should be statically indeterminate to be made bow-free. Then the assembly could be designed in such a way that the resultant moment created by the three thermally induced forces is zero, and no elastic moment is needed to equilibrate this "external" moment. It is imperative that the component materials are elastic and remain elastic during the assembly testing and operation, otherwise the bow-free condition will be compromised. When the two outer components are identical (Suhir 1999a, 2000c, 2009c, Suhir *et al*. 2015), the bow-free condition is fulfilled automatically, of course, for any inner component. Such structures were addressed, with an emphasis on the behavior of the bonding material, in connection with the design and use of holographic memory devices. The adhesive in this case was an optically sensitive holographic material (Suhir *et al*. 2011).

2.8 What are the levels of thermal and lattice mismatch stresses in semiconductor crystal grown assemblies?

Using the analytical modeling approach addressed in Suhir (1986a), Luryi and Suhir (1986) suggested a novel approach to high-quality dislocation free epitaxial growth of lattice mismatch materials (see Fig. 7 and Suhir, 2011e, 2013c, 2015c, d). The pioneering publications Suhir (1986a) and Luryi and Suhir (1986) have triggered hundreds of experimental and theoretical (mostly FEA based) investigations. The following major conclusions can be drawn from the existing practice:

1) Whenever possible, one should try to prevent the formation of firm grain boundaries that occur when crystals with appreciable lattice misfit grow together: although some insignificant density of dislocations might be acceptable, extensive development of dislocations has a detrimental effect on the device's functional performance;

2) Thermal stresses should be addressed only if the adverse effect of lattice misfit is minimized; this does not seem to be the case, however, with the currently used technologies, when the lattice misfit strains and stresses overwhelmingly prevail;

3) Thermal stresses in the today's SCG assemblies are the highest: high fabrication temperature is needed for crystal growth, and the thermal stresses are proportional to the change in temperature from the manufacturing temperature of about 1000°C or so, to the low (operation or testing) temperature. Calculations and measurements indicate that even such high thermal stresses are considerably lower than lattice-misfit stresses.

Important information has been obtained, using analytical modeling and theory-of-elasticity based approach, for circular (wafer-type) assemblies (Suhir 2013c, 2016a, Suhir *et al*. 2015). The

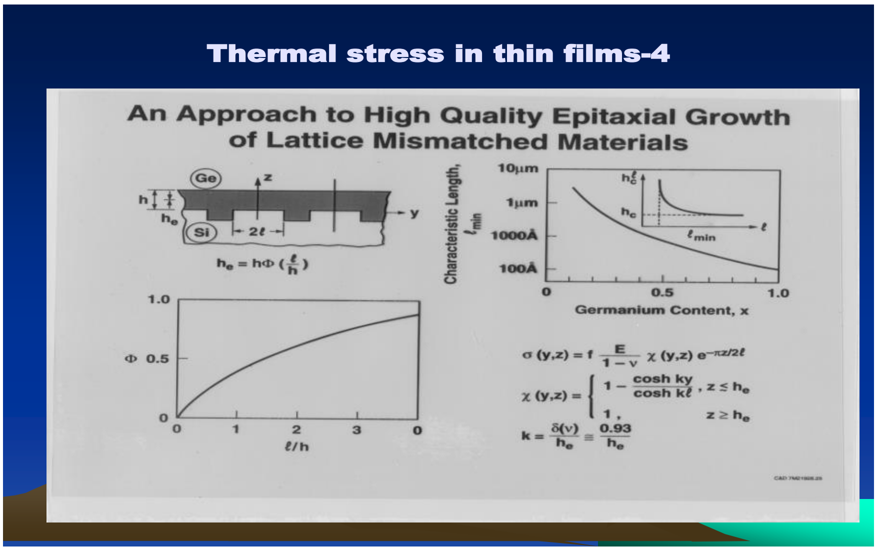

Fig. 7 Novel approach to the high-quality epitaxial growth of lattice-mismatched materials. It was suggested that a semiconductor film was grown on a specially engineered substrate with little “towers” on its surface

curvilinear configuration of the wafer edge plays an important role, as far as the level of the interfacial shearing and peeling stresses is concerned. The tangential (circumferential) normal stresses at the wafer edge could be much higher than the normal stresses in the semiconductor film at the assembly mid-portion.

2.9 Could shock tests adequately mimic drop test conditions?

Shock testers are less expensive than drop testers and shock test response data is much easier to organize and to interpret. Shock and drop tests are close relatives. But how close are they? And how could the shock tests be designed and how should the shock tester be tuned to adequately interpret drop test data? Analytical stress models for the dynamic response of electronic products subjected to drop and shock tests were developed (Suhir 2002c, Zhou *et al*. 2009) and the obtained results, in terms of the maximum accelerations, were compared and discussed. It was concluded that drop tests could be indeed adequately mimicked using a shock tester, when the tester is tuned in such a way that the maximum induced acceleration and the loading time are adequately reproduced.

2.10 Is the maximum acceleration an adequate criterion of the dynamic strength of a product?

The quality of an electronic or a photonic product is determined by its functional (electrical or optical) performance, its physical (mechanical, structural) reliability and its environmental durability, i.e., its ability to withstand harsh environmental conditions, such as high humidity, high or low temperatures, vibration environment, etc. It is shown (Suhir 1985, 1997a, Zhou *et al*. 2009, Suhir *et al*. 2011) that while the functional performance of a product can be usually satisfactorily characterized by the level of the induced accelerations, its physical (structural) reliability is due to

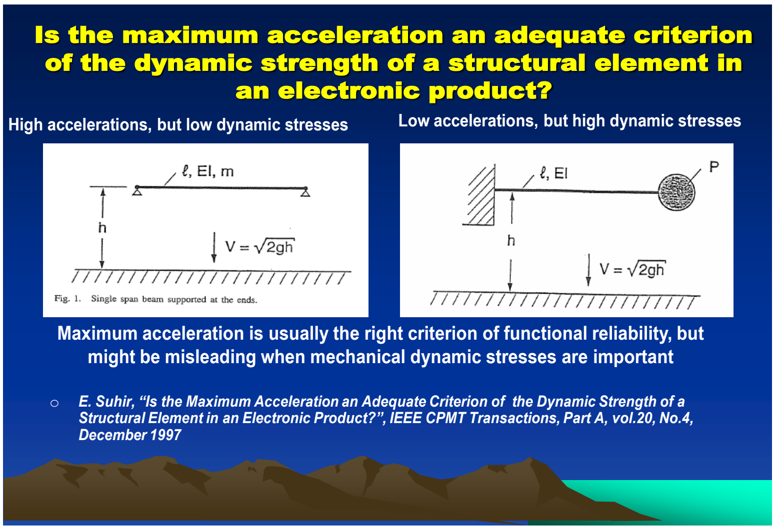

Fig. 8 Structural element idealized as a simply supported beam (on the left) experiences, as a result of drop tests, high accelerations. These are proportional to the frequency of vibrations squared, and the frequency of the drop induced vibrations is high. The induced stresses are, however, low. On the other hand, the cantilever beam with a lump mass at the end experiences high stresses at its clamped end, while the induced frequencies and, hence, the accelerations might be low

the dynamic mechanical stresses, which are not always proportional to the high accelerations (Fig. 8).

2.11 Could a static load be more damaging than a dynamic one of the same maximum value?

If the duration of the loading is much shorter than the quarter of the period of free vibrations, the system reacts to the impulse of the applied force, and not to its maximum value (Suhir 1996a, b). The static factor can be defined, in the linear approximation, as the ratio of the maximum static displacement or stress caused by the maximum value of the applied force to the maximum dynamic (actual) displacement and stress caused, in effect, by the impulse of this force and not by the force itself. The dynamic factor is defined as the ratio of the displacement or stress caused by the dynamically applied force to the displacement or stress caused by the same force, if this force is applied statically, i.e., when the time of the action of this force exceeds considerably the quarter period of the system's free (natural) vibrations.

If, for instance, a suddenly applied lateral loading stays on the beam that vibrates under the action of this force, then the maximum displacements and, hence, the induced stresses turn out to be twice as high as if the same force is applied statically. The "static loading", when the system is supposed to be able to withstand a high static force, can be indeed more damaging than "dynamic loading", when the system is expected to react to the impulse of this force, i.e., to the integral of the time-dependent force taking for the time duration between zero and the short duration of the loading. The system needs time to react to the applied loading and the maximum displacements and curvatures of a flexible structural element take place after a quarter period of vibrations, when the maximum deflections and the maximum accelerations (decelerations) are the largest. Because of that the application of the maximum force, instead of its impulse, to the structural element of interest can lead to a misleading conclusion that the element is not able to withstand such a force, while in effect, by the time when the element, subjected to an impact load, reaches its maximum deflection and maximum acceleration, the applied force will practically disappear. This is the situation, when the static load can be more detrimental (damaging) than the actual short-term dynamic loading. For a constant suddenly applied and suddenly removed load of the duration t_c, the dynamic factor for the time $t \le t_c$ is the largest at the moment $t = t_c$ of time, takes place, when the duration $t = t_c$ of loading is equal to the half the vibration period T_i of the system and is equal to $K_d = 2$. For a sine-shaped impact load, the dynamic factor for the time $t \le t_c$ is the largest at the last moment $t = t_c$ of time and takes place when the duration t_c of loading is equal to half the period T_i of vibrations and is equal to $K_d = \frac{\pi}{2}$. As to the "static factor", it can be defined as the ratio $K_s = \frac{f_{is}}{f_{iS}}$ of the static response f_{is} to the dynamic response f_{iS} caused by an instantaneous impulse. This factor is equal to $K_s = \frac{\pi}{\omega_i t_0} = \frac{T_i}{2t_0}$ in the case of a sine-shaped load and to $K_s = \frac{1}{\omega_i t_0} = \frac{T_i}{2\pi t_0}$ in the case of a suddenly applied and suddenly removed load of the duration t_0. When the duration of the impulse is next-to-zero, the "static factor" can be significant. For a load of the duration $t_0 = \frac{T_i}{8}$ that can be, according to our analysis, "harmlessly" substituted with an instantaneous impulse, the "static factor" is as high as $K_s = 4$, in the case of a sine-shaped force, and is $K_s = \frac{4}{\pi} = 1.2732$, in the case of a suddenly applied and suddenly removed force. Thus, the dynamic factor for the same magnitude of the applied stress can be much less detrimental to the structural strength of an electronic product, if the duration of the dynamic loading is short, lower than a quarter of the period of natural/free vibrations.

2.12 Could a closed-form solution be obtained for highly nonlinear vibrations?

The PCB's contour is considered non-deformable, which is indeed the case in many practical situations. This circumstance, if the drop height and/or the induced inertia forces are significant, leads to elevated in-plane (membrane) stresses in the PCB and, as a result of that, -to a non-linear response of the board to the impact load: the relationship between the magnitude of the load (determined by the initial impact velocity) and the induced PCB deflections becomes geometrically non-linear, with a rigid cubic characteristic of the restoring force. The developed model is based on the exact solution to the nonlinear equation of motion, no matter how highly nonlinear the dynamic response might be. The carried out numerical examples, although reflect the characteristics of the PCB and loading conditions in an actual experimental setup, are merely an illustration to the general concept and are intended to demonstrate for the abilities of the suggested method and model. Predictions based on this method agree well with the FEA data and with experimental evaluations. The developed models can be and, in effect, have been, helpful in understanding the physics of the addressed problem. The obtained results can be easily generalized, if necessary, for PCB's of different aspect ratios and with other boundary conditions, for different distributions of the added masses, etc. The results can be applied, with adequate modifications, to PCB's in actual use conditions as well. The developed model can be used, along with FEA simulations, in the analysis, structural ("physical") design and meaningful accelerated

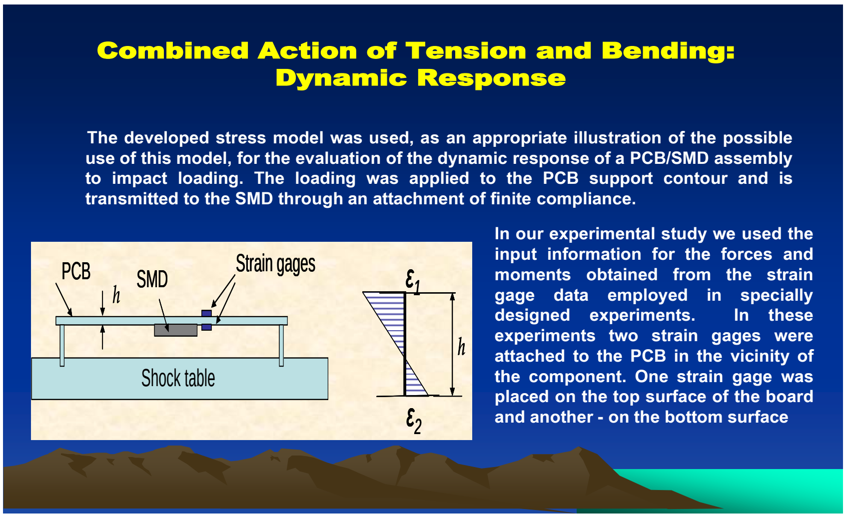

Fig. 9 Experimental setup for drop tests

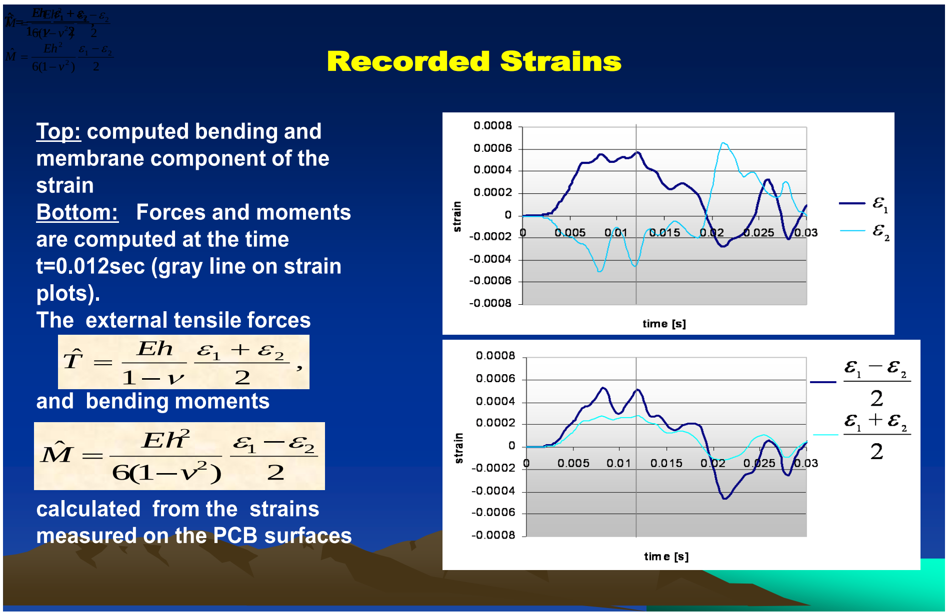

Fig. 10 Combined action of tensile and bending deformations (left sketch) and the recorded normal strains on both sides of the PCB (right sketch)

testing of electronic systems of the type in question, and particularly of "flexible-and-heavy" PCBs, both in accelerated tests and in actual operation conditions.

The dynamic response of a "heavy-and-flexible" PCB to the combined action of tensile and bending deformations and the measured strains recorded on both sides of the PCB are illustrated by Fig. 9. The recorded data indicate that significant in-plane tensile forces act, in addition to bending moments, in the PCB, thereby making the PCB dynamic response highly nonlinear. These

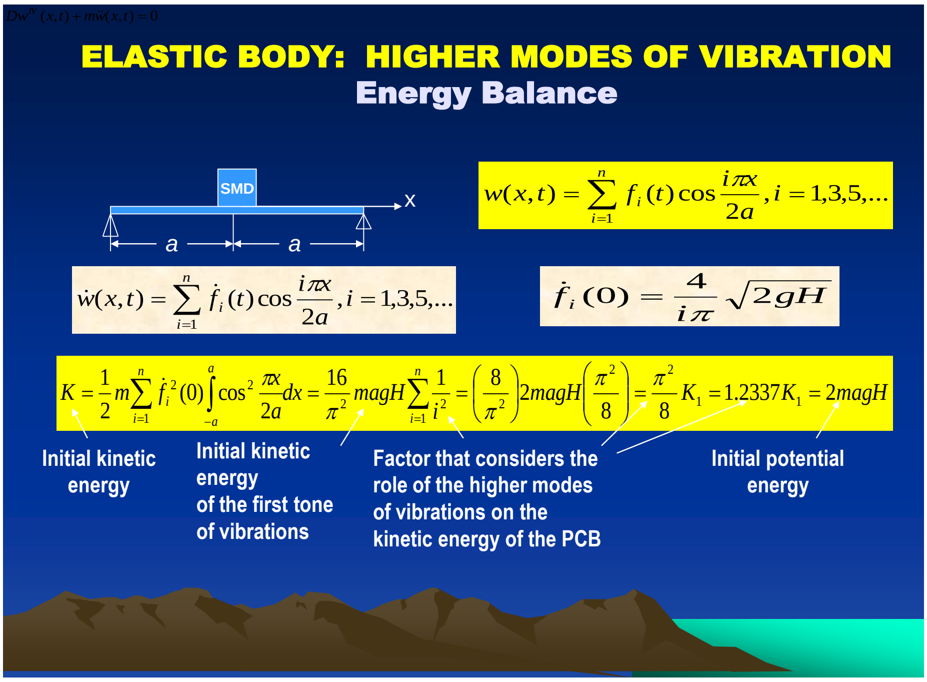

Fig. 11 How significant is the role of higher modes of the induced vibrations?

forces and moments can be either predicted theoretically or calculated from the measured tensile strains, as shown in the formulas on the sketch in Fig. 10.

The derivation in Fig. 11 is based on the linear model of the dynamic response of a PCB to an impact shock load applied to the PCB's support contour. An elongated PCB is considered. Its long edges are unsupported, and its short edges are freely supported. The dynamic deflection curve is sought, in accordance with the method of principal coordinates, as a sum of the time-dependent principal coordinates, $f_i(t)$, thus far unknown, and the coordinate functions $\cos\frac{i\pi x}{2a}$ that correspond to the free supports of the short edges of the board. Here a is half the board length, and the origin of the coordinate x is at the mid-cross-section of the board. The induced velocities of the board points can be found by differentiation. While the initial velocities of all the points of the board are the same and equal to $\sqrt{2gH}$, where H is the drop height, the initial velocities that correspond to different modes of vibrations are inversely proportional to the number $i = 1,3,5,...$ of the vibration mode. The last formula in Fig. 13 shows the evolution of the total initial kinetic energy (that should be equal to the initial potential energy $2mgH$) and, as the derivation indicates, is equal to $\frac{\pi^2}{8}K_1 = 1.2337K_1$, where K_1, as one could see from the equation in question, is the kinetic energy of the fundamental mode of vibrations. Thus, the higher modes of vibrations are responsible for only about 23.4% of the total vibration energy. In reality this percentage will be even lower, because higher modes dissipate much stronger and much sooner than the principal mode. For nonlinear vibrations this percentage will be even smaller, since the fundamental mode is

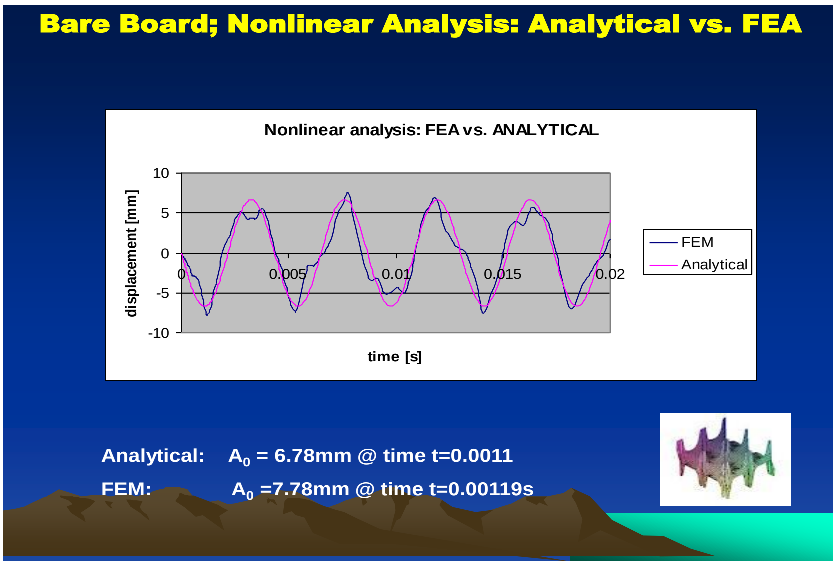

Fig. 12 Analytical vs. FEA data for the nonlinear dynamic response of a PCB to a drop impact

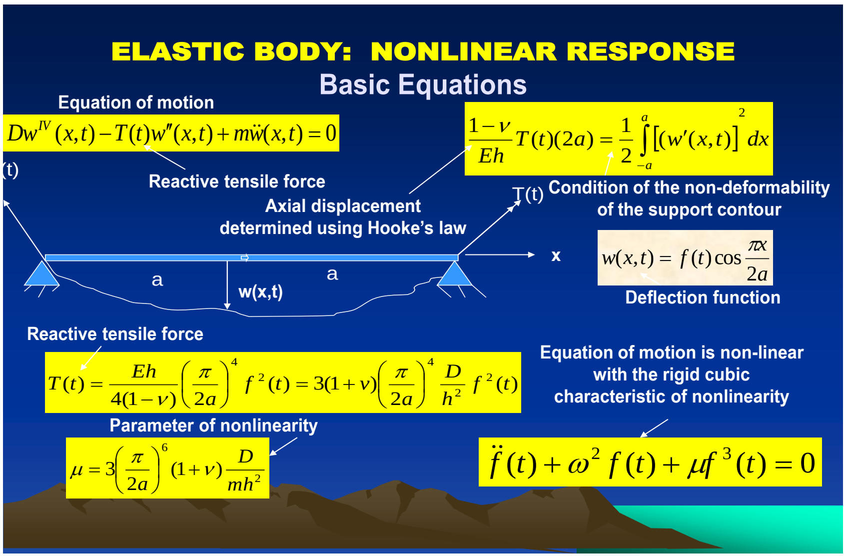

Fig. 13 Nonlinear dynamic response of a PCB to a shock load

a combination of the energies due to both bending and tensile in-plane deformations. This reasoning justifies that in practical applications the analysis of the nonlinear dynamic response of the PCB could be limited to the fundamental mode of vibrations only. This conclusion is confirmed by the comparison of the results of the analytical model data and FEA data shown in Fig. 12. Although FEA results consider and show the input of the higher modes, it is clear that in practical applications one could get away with taking into account the fundamental mode only and use the suggested analytical model as an adequate evaluation tool.

It has been found (Suhir 1991b, 1992a, b, 1995b, Suhir and Reinikainen 2009b) that the nonlinear dynamic response of a PCB to a drop impact applied to its support contour can be

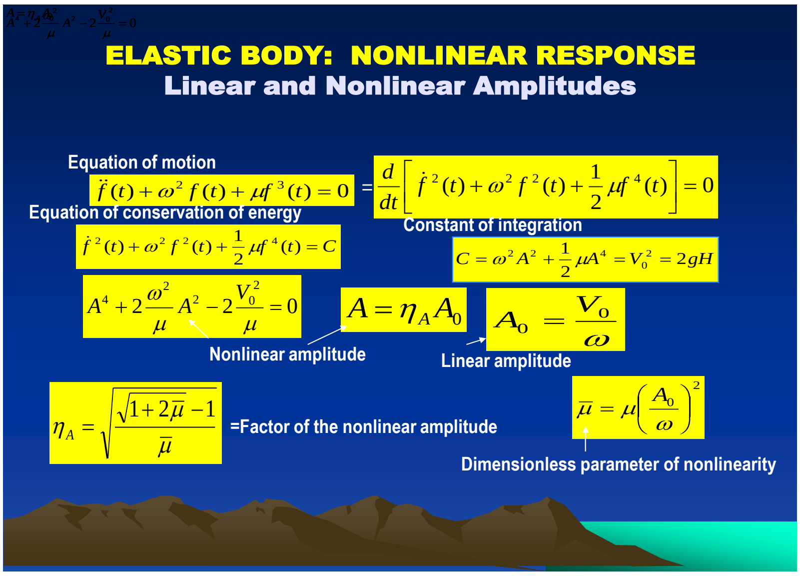

Fig. 14 Nonlinear dynamic response: equation of motion, linear and nonlinear amplitudes

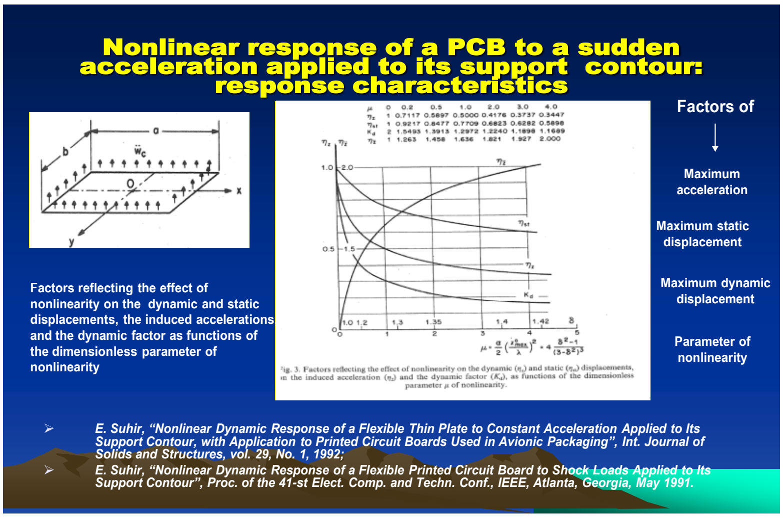

Fig. 15 Nonlinear response of a PCB to a sudden constant acceleration applied to its support contour

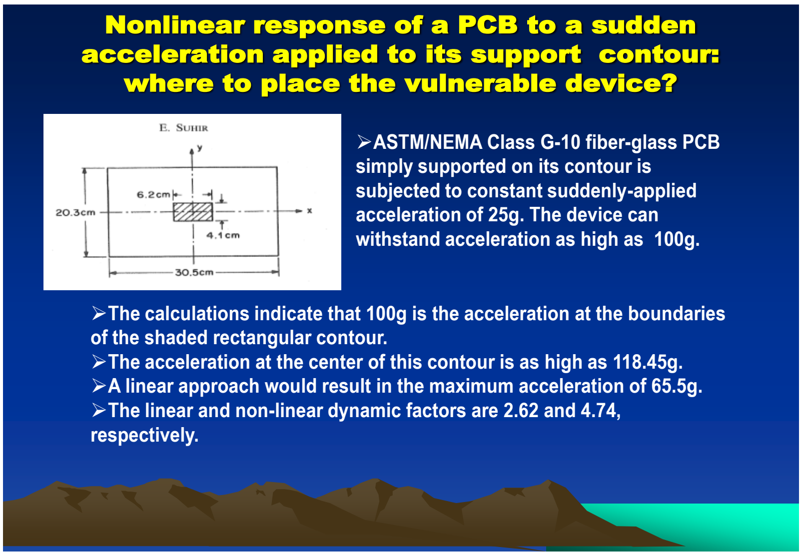

Fig. 16 The vulnerable electronic device should be placed outside the shaded area

described by the Duffing type of an equation (vibrations with the nonlinear cubic characteristic of the restoring force) for the principal coordinate (last formula in Fig. 13). Although the complete and exact solution to the Duffing equation exists and is expressed in elliptic functions, the nonlinear amplitudes can be found based on rather elementary considerations shown in Fig. 14.

Nonlinear response of a PCB to an impact load that is suddenly applied to and stays on its support contour is shown in Fig. 15. The situation reflects a PCB installed on a missile with a vertical take-off when experiencing constant acceleration. The board experiences highly nonlinear vibrations, whose accelerations (affecting the functional performance of devices mounted on the board) could be significantly higher than the accelerations of the board’s support contour. Based on the carried out analysis, the recommendation shown in Fig. 16 has been provided. Analytical modeling was applied also to determine if nonlinearity had an effect also on the coordinate function (Suhir and Arruda 2009). It has been found that this effect does not exist for simply supported PCBs, but does exist for clamped or otherwise supported boards.

2.13 Could a short impact load be substituted with an instantaneous impulse?

It is much simpler to evaluate the dynamic response of a system to an instantaneous impulse or to a static load, i.e., to two extreme cases of the dynamic response, than to do that for an arbitrary time-dependent dynamic loading. There is a reason to believe that a short enough loading will be adequately mimiced by an instantaneous impulse. In the case of a linear system, the error from substituting a short-term impact load of finite duration with an instantaneous impulse is shown in Fig. 17.

The computed data indicate that such a substitution could be done, with a rather satisfactory accuracy, if the duration of the actual loading does not exceed a quarter of the period of free vibrations, especially in the case of a sine-shaped impact. This is not surprising though (see section 2.11): it the time close to the quarter of the period of shock-induced (i.e., in effect, free) vibrations that is needed for the system to reach its amplitude and, in the case of a linear system, also its

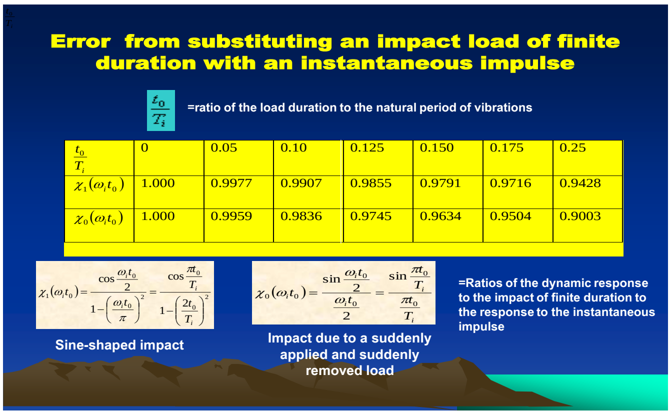

Fig. 17 Factor considering the error from substituting an impact of finite duration with an instantaneous impulse

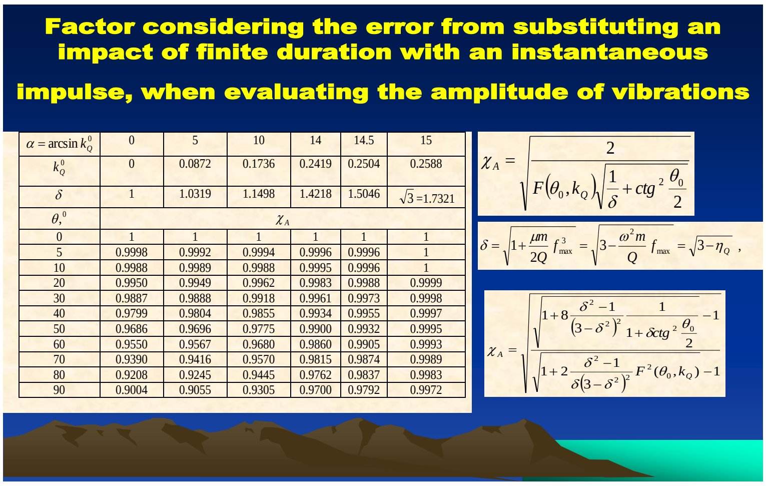

Fig. 18 Error from substituting an impact load of finite duration with an instantaneous impulse

maximum acceleration. If the duration of loading does not exceed this time, then the actual loading could be replaced by an instantaneous impulse. As to highly nonlinear systems with the rigid cubic characteristic of the restoring force (which is the case of a PCB whose ends cannot move closer during the induced vibrations) these systems are characterized by high inplane stresses, and, as the consequence of that, -by very high induced frequencies. Because of that the action of short-term loads is perceived as static loads. This suprising and favorable situation could be seen from Fig. 18, where the case of the vibration amplitudes is considered (Suhir and Arruda 2010). It is important, however, to have in mind that, unlike in a linear system, the nonlinear response is different for the amplitudes, velocities and accelerations, and therefore the possibility of the substitution of the type in question should be addressed and considered separately.

2.15 Could larger stand-off heights relieve stress in solder joint interconnections?

Simple, easy-to-use and physically meaningful analytical stress models are developed for the assessment of thermally induced stresses in ball-grid-array (BGA) and column-grid-array (CGA) interconnections currently employed in IC device packaging. It is suggested (Suhir 1992c, 2015g, h, 2016b, c, Suhir *et al*. 2013, Suhir *et al*. 2013, Suhir *et al*. 2015, 2016a, b, Suhir *et al*. 2016) that such an assessment is carried out in three steps. At the first step the effective interfacial shearing and peeling stresses, as well as the corresponding displacements, in BGA/CGA interconnections (second level of interconnections) are determined for the package-PCB system, assuming that no inelastic stresses and strains take place. At the second step the stresses and strains for particular (peripheral) joints are evaluated from the in-plane (shearing) and through-thickness displacements computed at the first step. These displacements consider the role of the compliance of the assembly components and the joints themselves at the joints' ends, thereby taking into account that the boundary conditions at the joints' ends could be somewhat different of simple clamped-clamped conditions and that the offset of the joint ends might be different than the one computed as the product of the thermal expansion mismatch strain $\Delta\alpha\Delta t$ and the distance of the given joint from the mid-cross-section of the assembly. If the calculations performed at the first two steps and based on the assumption of the elastic behavior of the materials suggest that such a behavior is not compromised, then no third step is needed. If, however, the computed stresses exceed the yield stress of the solder material, then the inelastic strain zone should be evaluated at the third step. This could be done assuming that the solder material is linearly elastic below the yield strain and ideally plastic above this strain. Clearly, if the actual solder material exhibit elasto-plastic behavior above the yield strain, the corresponding states of stress and strain are between the extreme cases of the linearly-elastic and ideally plastic solutions. The effective stresses and displacements are assessed at the first step using an assumption that the actual (inhomogeneous) BGA/CGA system can be substituted, in an approximate analysis, by a homogeneous (continuous) solder layer (attachment) of the same stand-off (thickness) as the actual BGA/CGA system. It is shown that such a substitution is acceptable, if the gaps between the supports (BGA balls or CGA columns) are small, and the product kl of the parameter k of the interfacial shearing stress and half the assembly length l is significant. Such a simplification is acceptable, if the ratio $\frac{p}{2l}$ of the pitch p (distance between the joint centers) to the joint widths $2l$ is below 5, and the computed product kl is above 2.5, which is indeed the case in actual BGA and CGA systems. This conclusion is made using the general relationships obtained for an arbitrary more or less complex inhomogeneosly bonded assembly (IBA). It is shown also that consideration of the finite rigidity (bow) of the assembly results in significant relief in the predicted interfacial shearing stress. The numerical example indicates that the maximum peeling stress is about half of the maximum shearing stress. All the analyses have been carried out assuming that the longitudinal compliance of the solder system is considerably larger than the compliance of the bonded components-the package and the PCB. The developed models include also the estimate of the length of the inelastic zone, if any. If inelastic strains cannot be avoided, then one of the numerous modifications of the Coffin-Manson relationships can be used to assess the low-cycle fatigue life of the peripheral joints subjected to such strains. It is clear that both the length of the inelastic zone and the level of the inelastic strains are important from the standpoint of the expected lifetime of the solder material. The main effort is to determine if one could possibly avoid inelastic strains in the solder material. If this is possible, then the low cycle fatigue conditions would be replaced with the elastic fatigue situation, thereby leading to considerable longer fatigue life of the material. The well-known Palmgren-Miner rule of the linear accumulation of damages, instead of Coffin-

Manson models, could be employed in such a case. The application of this rule is even more justified, if the loading is random (Suhir 2016g). In such a case the elastic and inelastic strains occur interchangeably and so does the weakening and strengthening of the material, so that the "history" of the loading is not pronounced as strongly as in the case of a deterministic (nonrandom) loading.

2.16 Stretchable electronics: Does one really need good thermal expansion (contraction) match between the die and the carrier?

Large area electronics is viewed as a promising technology for many attractive applications. An assembly comprised of thin silicon or organic die embedded into, or surface-mounted onto, a flexible (stretchable, large area) film-like plastic sheet (carrier) is a typical structural element in these technologies. The mechanical integrity of such an element could be compromised, when the carrier experiences elevated tensile stresses (deformations) in actual use or during qualification testing. Typical observed failures are die fracture and interfacial delaminations. Simple and physically meaningful analytical models are developed (Suhir 2009d, e, 2010a, 2011f) to evaluate the stresses in a die/carrier assembly employed in stretchable electronics. These stresses include normal stresses acting in the cross-sections of the die (and responsible for its short and long term strength), and the interfacial shearing and peeling stresses (responsible for the adhesive and the cohesive strength of the bonding material). A situation when the die/carrier assembly is manufactured at an elevated temperature, then cooled down to a low (say, room) temperature and then subjected to an external tensile load applied to the carrier is addressed and analyzed. The induced stresses are due therefore to the combined action of the thermal and the mechanical loading. The developed models can be used in the stress/strain analyses and physical design of the assemblies of the type in question in large area ('stretchable") electronics. The models indicate particularly that, contrary to the recommendations contained in (Christiaens *et al*. 2006), no low expansion carriers are needed. Just the opposite: appreciable stress relief could be expected if regular (high expansion) carrier materials are employed. In such a situation the compressive thermal stresses in the die will be compensated, to a greater or lesser extent, by the tensile external loading. The developed model can be used also in the design of the experimental setups (e.g., when deciding on the appropriate location of the strain gages), and in the analysis (interpretation) of the test data.

2.17 Could the threshold of the added transmission losses be predicted based on mechanical considerations?

It has been observed (Suhir 1990a) that the threshold of elevated transmission losses in jacketed optical fibers coincides with the temperature at which the hoop stresses applied from the high expansion (contraction) jacket to the low expansion silica fiber start to dramatically increase (Fig. 19). This phenomenon can be effectively used to predict the threshold of added transmission losses in jacketed optical fibers from the mechanical calculations and measurements.

2.18 Is there an incentive for pre-stressing accelerated test specimens?

In some today's and future electronic and optoelectronic packaging systems (assemblies) the package (system's component containing active and/or passive devices and interconnects) is

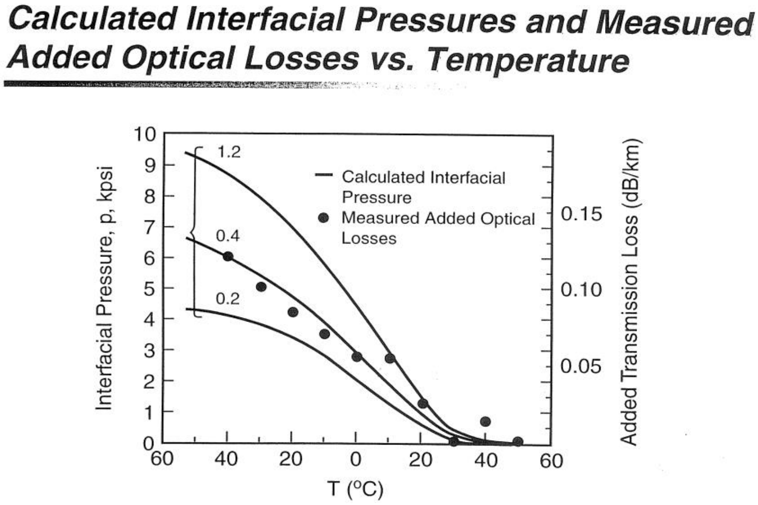

Fig. 19 Calculated pressures and measured added optical losses vs. temperature in jacketed optical fibers

placed (sandwiched) between two substrates, which, in an approximate stress analysis, could be considered, from the mechanical (physical) standpoint, identical. Such systems (assemblies) are certainly bow-free, provided that all the stresses are within the elastic range and remain elastic during testing and operation. The highest thermal stresses in such a tri-component (one inner and two outer components) bi-material (the composite material of the inner component-package and of the material of the outer components-substrates) assemblies are caused by the thermal contraction mismatch of the dissimilar materials of the assembly components. These stresses occur at low temperature conditions, and include normal stresses acting in the cross-sections of the components and interfacial shearing and peeling stresses. The normal stresses in the component cross-sections determine the reliability of the component materials and of the devices embedded into the inner component (package). The interfacial stresses affect the adhesive and cohesive strength of the assembly, i.e., its integrity.

It should be pointed out that although the assembly as a whole is bow-free, the peeling stresses in it, whether thermal or mechanical, are not necessarily low: the two outer components (substrates) might exhibit appreciable bow with respect to the bow-free inner component (package). While there is an incentive for using bow-free assemblies, there is also an incentive for narrowing the temperature range of the accelerated reliability testing: elevated temperature excursions during accelerated testing might produce an undesirable shift in the modes and mechanisms of failure, i.e., lead to failures that might hardly occur in actual operation conditions.

Failure oriented accelerated test (FOAT) specimens are particularly vulnerable, since the temperature range in these tests should be broad enough to eventually lead to failures, and if a shift in the modes and mechanisms of failures takes place during significant temperature excursions, the physics of such failures might be quite different than in actual operation conditions. An appropriate mechanical pre-stressing can be an effective means for narrowing the range of temperature excursions during accelerated testing and, owing to that, -for obtaining consistent and trustworthy test data (Suhir 2011g, Suhir and Nicolics 2014, Suhir *et al*. 2015). If such a pre-stressing is considered and implemented, the ability to predict the thermo-mechanical stresses in the test specimen is a must. Accordingly, simple, easy-to-use and physically meaningful and practically useful closed form solutions were obtained for the evaluation of stresses in a bow-free test specimen comprised of an electronic or optoelectronic package sandwiched between two identical substrates. The role of compliant attachments, if any, between the inner and the two outer components was considered. The numerical data indicated that in the absence of compliant attachments the magnitudes of all the stress categories (normal stresses in the cross-sections of the assembly components and the two types of the interfacial stresses) are comparable. The predicted maximum interfacial stresses turned out even comparable with the maximum normal stresses acting in the component cross-sections. The developed analytical model can be used at the design and accelerated test stages of the development of bow-free electronic and optoelectronic products. These compliant attachments could be particularly comprised of beam-like solder joint interconnections that, if properly designed, have a potential to relieve the thermal stresses to an extent that the low-cycle-fatigue state-of-stress in the solder material is avoided.

2.19 Could thinner and longer legs relieve stress in a thermoelectric module design?

A physically meaningful analytical model has been developed for the prediction of the interfacial shearing thermal stress in an assembly comprised of two identical components subjected to different temperatures (Suhir and Shakouri 2012, 2013, Suhir 2013d). The bonding system is comprised of a plurality of identical column-like supports located at equal distances (spaces) from each other. The model is developed in application to a thermoelectric module (TEM) design where bonding is provided by multiple thermoelectric material supports (legs). It has been shown that thinner (dimension in the horizontal direction) and longer (dimension in the vertical direction) TEM legs could result in a significant stress relief, and that such a relief could be achieved even if shorter legs are employed, as long as they are thin and the spacing between them is significant. It is imperative, of course, that if thin legs are employed for lower stresses, there is still enough interfacial “real estate,” so that the adhesive strength of the assembly is not compromised. On the other hand, owing to a lower stress level in an assembly with thin legs and large spacing, assurance of its interfacial strength is less of a challenge than for a conventional assembly with stiff, thick, and closely positioned legs.

It is shown also that the thermal stresses not only in conventional TEM designs (using Be2Te3 as the thermoelectric material, and Sn-Sb solder), but also in the future high-power (and high operating temperatures) TEM design (using Si or SiGe as the thermoelectric material and Gold100 as the appropriate solder), might be low enough, so that the long-term reliability of the TEM structure could still be assured. It has been found, however, that thin-and-long legs should be considered for lower stresses, but not to an extent that appreciable bending deformations of the legs become possible. Future work will include, but might not be limited to, the finite-element computations and to experimental evaluations (e.g., shear-off testing) of the stress-at-failure for the TEMs of interest.

2.20 What could possibly be done to reduce the bending stress in an optical fiber?

Numerous analytical stress models have been suggested to design, test and operate optical fibers and particularly optical fiber interconnects (Suhir 2014c, 1988e, f, 1998b, c, d, e, 2000d, 2003c). Here are some challenging problems addressed and solved by using analytical modeling.

Dual-coated optical fibers are fabricated at elevated temperatures and operated at low temperatures, such as, e.g., undersea conditions. It is imperative that the fibers remain stable at low temperatures, i.e., do not buckle (“micro-bend”) within the primary coating as a result of the thermal contraction mismatch of the high expansion (contraction) polymeric secondary coating and the low expansion (contraction) silica fiber.

Low-temperature micro-bending (Fig. 20), while harmless from the standpoint of the level of bending stresses, can result in substantial added transmission losses. This phenomenon is a good illustration of a situation, when it is the need for a failure-free functional (optical), not physical (structural) performance that determines the requirements for the adequate structural (physical) design. Simple analytical models suggest that the fiber prone to low temperature micro-bending is

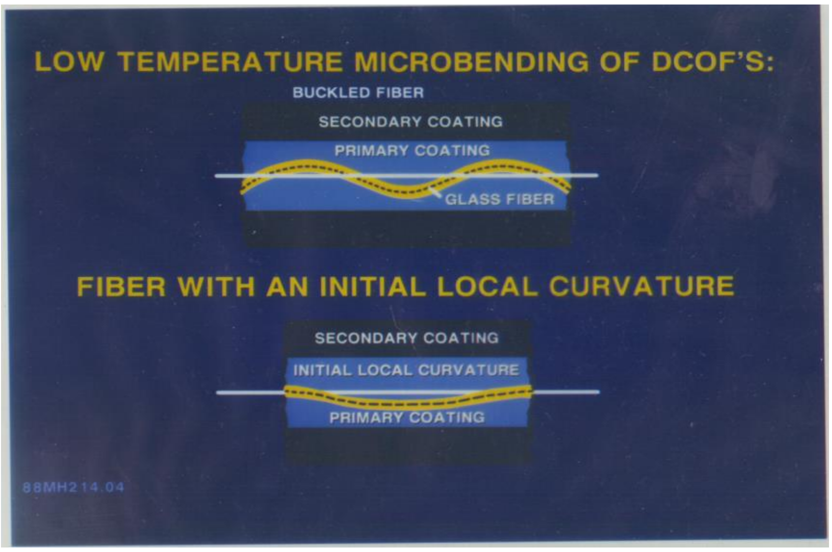

Fig. 20 Low temperature micro-bending of dual-coated optical fibers (Suhir 2014c)

treated as an infinitely long beam lying on a continuous elastic foundation provided by the coating system. As long as such a model is considered, particular attention should be paid to how the spring constant of the elastic foundation is determined (Suhir 1988e).

Another significant finding, as far as the low-temperature micro-bending phenomenon is concerned, has to do with the role of the initial local curvatures (Suhir 1988e, f). While the initial curvatures do not change the magnitude of the critical (Euler) force, they affect the pre-buckling behavior of the compressed fiber. When the compressive thermally-induced force increases, an initially straight fiber remains straight up to the very moment of buckling, while the localized curvatures in a fiber gradually increase with an increase in the compressive force. This situation could cause appreciable additional deflections of the glass fiber and, as the consequence of that, -added transmission losses even at moderately low temperatures, well below the buckling temperatures. It has been shown particularly that, from the standpoint of the pre-buckling behavior and propensity of a fiber to buckling, certain curvature lengths are less favorable than the others: the fiber behaves with respect to the distributed localized initial curvatures like a narrowband filter that enhances the curvatures, which are close to the post-buckling configuration of the fiber, and suppresses all the other, ‘non-resonant”, curvatures.

The developed analytical models are simple, easy-to-use, and clearly indicate the role of various factors affecting the pre-buckling behavior of the fiber. The obtained solutions indicate what could possibly be done to bring down, if necessary, the low temperature induced curvatures and the resulting added transmission losses, and to design the coating system so that the elastic stability of the fiber is not compromised. It has been observed (Suhir *et al*. 1995) that external periodic loading with the period of about 100 nm can cause appreciable micro-bending losses in dual-coated fibers, and therefore should be avoided in actual designs. It is noteworthy that this period is rather close to the predicted "critical" periods in low-temperature micro-bending.

Pigtails in laser package designs provide particular challenge, as far as their bending is concerned. Various situations encountered when a pigtail is employed to connect a laser package to the "outside world" have been addressed and analyzed using analytical modeling (Suhir 2014c). It

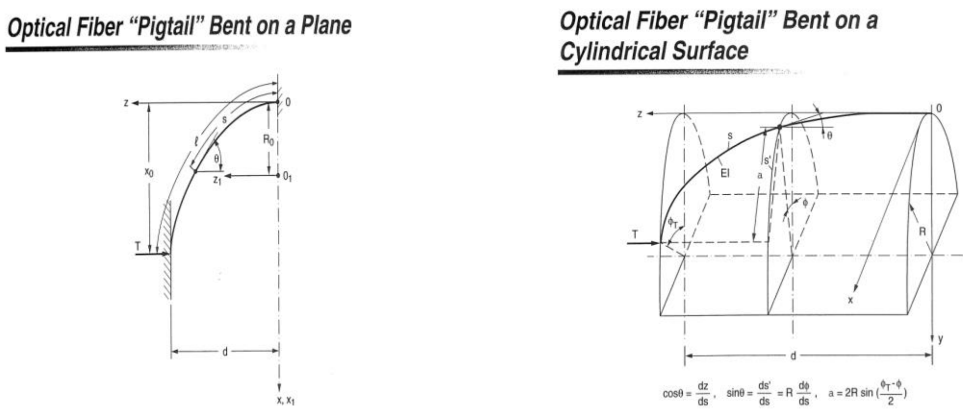

Fig. 21 Optical fiber pigtail configured as a quarter of a circumference (a) and optimized configurations of an optical fiber pigtail for undersea applications (b)

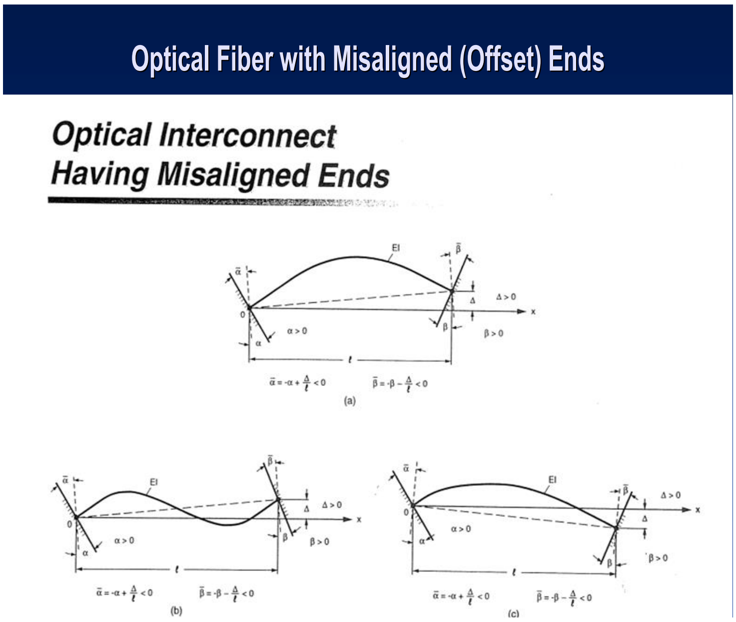

Fig. 22 Optical fiber interconnect subjected to the ends offset

has been shown particularly that by rotating the package inside its enclosure, one could reduce dramatically the induced curvatures. This should be done with caution, however. Ideally straight pigtails cannot be typically recommended: while the initial bending stress in them is indeed zero, the situation could be worsened dramatically if the structure with a high expansion enclosure is heated up, thereby resulting in undesirable and significant tensile stresses in the pigtail. It is usually preferred that the pigtail in such a design is "loose". Such a pigtail is able to accommodate appreciable axial deformations in tension or compression without being stressed.

If a pigtail experiences two-dimensional bending on a plane (Fig. 21(a)), significant bending stress relief can be obtained by forcing the pigtail to be configured as a quarter of a circumference, so that all its points have the same curvature and, hence, experience the same bending stress. This stress can be considerably lower than the maximum bending stress at the clamped end of a clamped-free pigtail. More practical and more complicated situations take place when a pigtail is bent on a cylindrical surface (Fig. 21(b)). Such a design was considered for lasers intended for AT&T undersea long haul communication technologies. Achieving an optimized geometry of such a pigtail is definitely a challenge.

The thermal stresses in an optical fiber interconnect can be brought down dramatically, if its ends are properly rotated (Fig. 22) or subjected to an appropriate compression by employing an enclosure with an adequate CTE.

2.21 Analytical modeling could be of help when designing, conducting and analyzing optical fiber proof-tests

The stress-strain related problems that arise during proof-testing of coated optical fibers and that were addressed, based on the analytical predictive modeling, include the role of the length of a test specimen in pull-testing, the buffering effect of the coating in pull (proof) and bending tests, the magnitude and the distribution of the interfacial stresses during pull-out testing, as well as stresses in coated fibers stretched on a capstan during the manufacturing process (Suhir 2014c, 1994a, b, 1990b, 1992d, e, 1998f, Suhir and Bechou 2013a). Some more or less paradoxical physics of the associated phenomena are the following:

1) It is well-known in materials science that if one intends to determine the Young's modulus of a material and/or its flexural strength through three- or four-point bending, the specimen should be long enough (say, its length should be at least 12-15 times larger than it height), so that lateral shearing deformations do not occur in the specimen. An appropriate analytical model enables one to design the right test specimens and to establish the right testing conditions (Suhir 2014c);

2) A problem encountered during pull testing of a glass fiber whose end is soldered or adhesively bonded, and another end is subjected to a pulling (tensile) force is somewhat different: considering that the pulling force will always form a certain angle with the fiber axis, what could be done to minimize the effect of the associated bending stresses? Accordingly, an analytical model has been developed (Suhir 1994a) for the evaluation of the bending stress caused by the misalignment of the ends of a glass fiber specimen soldered into a ferrule and subjected to tension during pull testing. It is shown that this stress can be reduced considerably by using sufficiently long specimens. It is also shown how the uncertainty in the prediction of the misalignment of the fiber ends can be considered when establishing the appropriate specimen length. The stresses in the bonded joint can be determined and, if necessary, minimized using the model (Suhir 1994b).

3) The tensile force experienced by a dual-coated optical fiber specimen during its reliability (proof) testing is applied to the fiber's secondary coating and is transmitted to the glass fiber at a

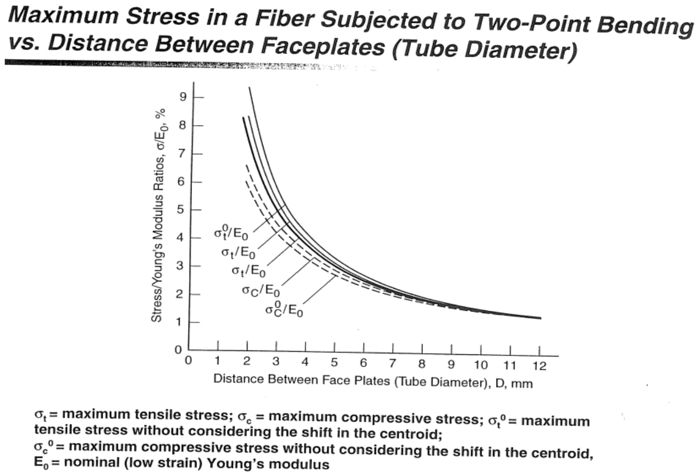

Fig. 23 Two-point bending of optical fibers

certain distance from the specimen's ends. Although it is true that, in accordance with the Saint-Venant's principle, the glass fiber will be subjected, at a certain distance from the specimen ends, to the same stress that it would experience if the external force were applied to both the fiber and the coating, it is also true that, because of the buffering effect of the coating, the effective length of the fiber under testing, when the testing force is applied to the coating only, might be reduced appreciably in comparison with the fiber actual length. A simple analytical stress model for the evaluation of this effect has been developed (Suhir 1990b, Suhir and Bechou 2013a) and used to establish the appropriate minimum length of the test specimen, so that the experimental data are consistent and physically meaningful.

4) The maximum stress in an optical fiber interconnect subjected to two-point bending (Fig. 23) should be determined before the testing is conducted in order to attribute the measured time of the delayed fracture to the corresponding induced stress, and this should be dine with consideration of the nonlinear stress-strain relationship of the silica material (Suhir 1992d, e).

3. Conclusions

All the three basic approaches in microelectronics and photonics materials science and engineering-analytical (mathematical) modeling, numerical modeling (simulation) and experimental investigations-are equally important in understanding the physics of the materials behavior and in designing, on this basis, viable and reliable electronic devices and products. Analytical modeling is a powerful tool that enables one to explain paradoxical situations in the behavior and performance of electronic materials and products, and make a viable device into a reliable product.

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