Optimization of interconnections between packaging levels

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In large-scale integrated circuits, the interface between ceramic modules and the next level—epoxy-glass circuit boards or cards—contains a large number of pin arrays. Because the modules and the card are usually quite rigid and mechanically strong, the interface between the module and the card is commonly the weakest region in the assembly system. This interface is where the differential deformations between the two levels of packaging are accommodated. This paper describes a theoretical and experimental program to understand the loadings and stresses present, and to optimize the design of the connecting pin in order to distribute the stresses more evenly across the surfaces of the braze joint that connects the pin to the ceramic module. This work was done jointly by members of the East Fishkill and Endicott laboratories.

Introduction

With the accelerated pace of large-scale integration, the interface between ceramic modules and the next packaging level—epoxy-glass circuit boards or cards—contains increasing numbers of pin arrays. These pins are used for communication between the ceramic modules and the next level. Usually one end of each pin would be brazed to the module; the other end of the pin would be inserted into a plated through-hole in the card and then wave-soldered in place, as indicated in **Figure 1.**

Since the modules and the card are usually quite rigid and structurally strong, the interface between the module and the card is commonly the weakest region in the module–card assembly system, since it serves not only the electrical function of communication but also the mechanical function of accommodating the differential deformations between the two levels of packaging. This accommodation becomes more critical with greater areas of pin arrays, because the stresses become correspondingly larger. It is clear that making the strongest, and hence the stiffest, pin-joint design would not be the optimum design direction. This design would transfer high stresses onto the ceramic modules or the cards themselves and thus increase the risk to the soldered or brazed connections. For optimum mechanical design, the entire system of pin, brazed joint, and soldered joint must be sufficiently strong to sustain all the processes, tests, and thermal stresses arising during machine operation, while still being sufficiently flexible so as not to induce high stresses from differential deformations.

Design optimization requires a knowledge of the strengths of the materials and the joints employed, the complete set of loadings to which the assembly will be subjected, and the stresses arising from these loadings. This paper describes an analytical and experimental program set up to understand

1. The mechanical deformation of the pin and the strength of the brazed joint.
2. The deformation in the card–pin–module system, and the stress distribution in the pins and brazed joints, due to various mechanical and thermal loadings on the assembly.

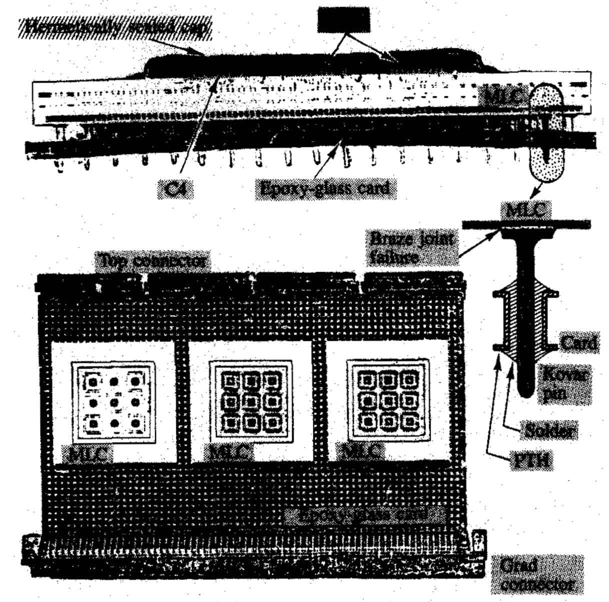


Figure 1

50-mm MLCs on a GRAD card. There are nine chips and 19 × 19 brazed pins per MLC.

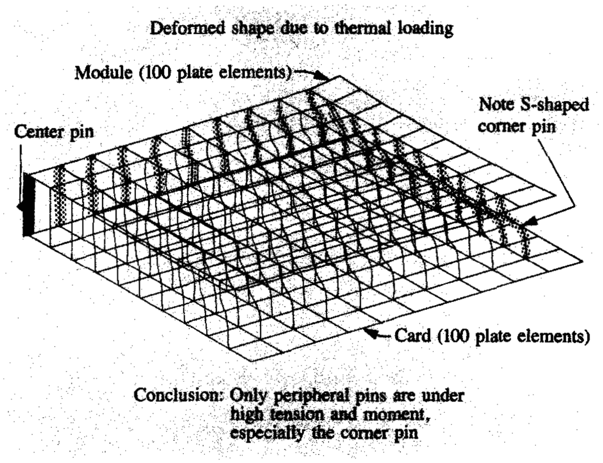


Figure 2

Macro Model I for the wave solder process.

3. The residual loads on each of the pins in one pin array after a module has been soldered to a card.
4. The effects on the above deformations and loads that result from changes in design parameters.

The results of the study were an optimized pin design and a quantitative understanding of the stresses induced by manufacturing process, test, and handling loadings which can be safely sustained by the assembly.

Finite-element modeling was employed to simulate mathematically the structure of the module–card assembly and to calculate the stress distributions present at the critical regions. An experimental program was undertaken in concert with the finite-element analyses to verify the model and to refine the numerical predictions. Observation and measurement of the parts in the actual processing and testing steps further verified the real-life loading conditions induced in the manufacturing line.

Strategy

The overall approach taken combined various analytical and experimental techniques. Two finite-element models were used: a macro model simulating the module–pin–card assembly and a micro model representing the pin–braze joint. The former modeling was done in Endicott, where the module-to-card assembly took place, and the latter modeling was done in East Fishkill, where the pin-to-module braze operation was carried out. Both analyses used the NASTRAN program [1] to allow for a common basis in the finite-element representations, for the ability to run one another's idealizations, and for possible integration into a single model if necessary.

An important aspect of the strategy was the series of experiments and laboratory measurements to verify and refine the models and to obtain some of the basic material constants needed for the NASTRAN idealizations. First, a holographic experiment was performed to measure the displacements of the card–module assembly during a temperature drop. The displacement of 0.01 micrometer per degree Celsius obtained from the pattern of interference fringes was compared with the results from the NASTRAN calculations for the same change in temperature. This was a key experiment to verify and refine the models, since the comparison of results could be made not only at one point, but also over total surfaces. Second, a card–module assembly was instrumented by mounting strain gauges on a few selected pins. This allowed for verification of the calculated loadings on individual pins from mechanical loadings applied to the assembly. Strain gauge measurements were in excellent agreement with the calculated values.

In the brazed joint, the ability to predict how the braze would fracture was critical. It was found that the shank of the pin would break, rather than the braze, under an applied axial tensile load. A 20-degree pull test was instituted to provide comprehensive statistical data for the breaking strength of the brazed joints. The idealization used in the NASTRAN modeling of the brazed joint could analyze the state of stress in the braze both under the conditions of the module–card assembly and of the 20-degree pull test. This

Table 1 Material constants used in idealizations.

	Module	*Braze*	*Pin*	*Card*
Young's modulus	340 GPa (50×10^6 psi)	75 MPa (11×10^6 psi)	136 GPa (20×10^6 psi)	9.5 GPa (1.4×10^6 psi)
Poisson's ratio	0.28	0.3	0.3	0.3
Thermal expansivity	6.4×10^{-6}/K	—	6.2×10^{-6}/K	9×10^{-6}/K below 120°C 6.5×10^{-6}/K above 120°C

was used to correlate the minimum breaking strength to forces and moments.

Finite-element models

• *Macro models*

Two macro models were used. The first idealized one-fourth of a single module-and-card assembly (**Figure 2**), and the second represented the full assembly of three modules and the printed-circuit card. The one-fourth model allowed the use of more elements to represent each pin. The alumina substrate of the module and the printed-circuit card were each idealized as assemblages of plate elements, each with four nodes. The elements were not isoparametric, but did combine membrane and bending modes. The pins were idealized as beams of circular section.

In NASTRAN terminology, the card and module were both represented by CQUAD2 elements, and the pins by the CBAR element with OFFSET, to accommodate the fact that the nodes defining the ends of the pins were not at the same physical locations as the nodes in the midplanes of the plate elements to which they were connected. Also, the shear parameter K was set to unity to activate the shear behavior in CBAR.

The pins were idealized as consisting of four elements each, not to improve the formulation, but to allow the display of the actual flexural response of each pin: a beam built-in at both ends with a zero moment at its center. The material constants used in the idealizations are given in **Table 1**.

An important consideration was that during a large temperature cycle such as the one seen by the assembly during wave soldering, the thermal loadings would take the Kovar pins into the plastic range. An analysis was done to calculate the moment-flexure behavior of the Kovar pin in the inelastic range. It was found that for the range of deflection considered, the differences in the moment and in the axial loads from the purely elastic case were not large. For this reason, the behavior of all the materials contained in the idealizations was taken to be linearly elastic.

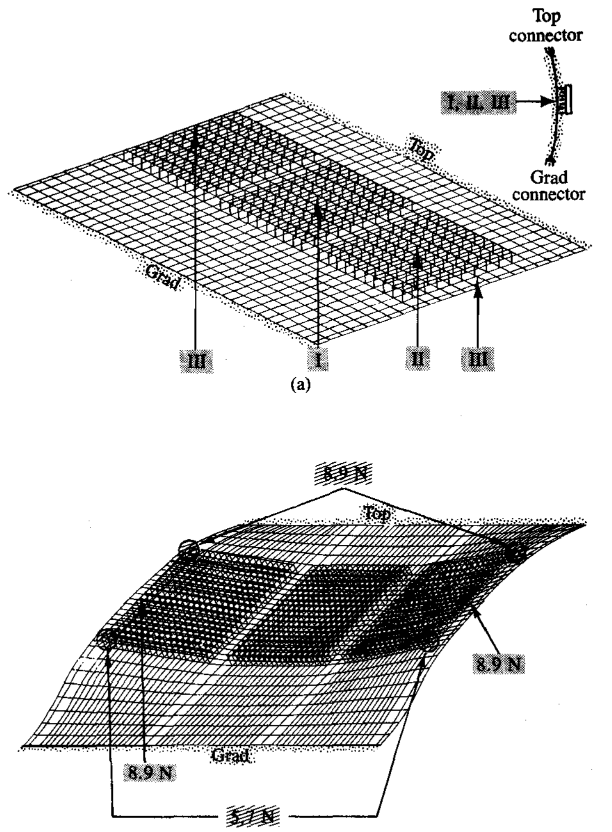
• *Micro model*

In order to characterize the state of stress in the braze attaching the Kovar pin to the MLC module, several idealizations were constructed. They consisted of assemblages of linear elastic elements (NASTRAN CTRIA2 and CQUAD2 elements were used). The structure analyzed was a two-dimensional slice through the head of the pin and the braze, since the finite-element program being used could not consider an idealization consisting of an axisymmetric structure to which nonaxisymmetric loads were applied. It was felt that this was a reasonable assumption because the strength of the braze would be calculated using this model under those pin loadings known to cause failure. These strengths could then be compared with stresses calculated under the load conditions obtained from the macro model. The thickness used for the slice was the value which provided a moment of inertia for the shank equal to that of the actual pin. The interest was in the stresses set up in the braze; therefore, only a small portion of the shank of the pin was included in the idealization. The braze was assumed to have a thickness range of 0.04 to 0.08 mm (0.0015 to 0.003 in.) and was attached to a rigid base. Idealizing the ceramic module as a rigid structure permits a conservative estimate of the stresses in the braze to be made.

Summary of results

• *Module-and-card assembly: The macro model*

The first result obtained was for the idealization of one-fourth the model-and-card assembly shown in Fig. 2, assuming a uniform drop in temperature after the wave-soldering operation. The key question was "Is there something very different about the loading on the corner pin?" It was found that in the thermal drop after wave soldering, all pins were loaded both with bending moments and axial forces. Although the bending moment decreased slowly in going from the outermost row toward the center of the model, the axial loading was tensile in the outermost row only. All pins other than those in the outermost row were in axial compression. The largest tensile force was present at the corner pin. However, under the thermal loading this axial force was less than 8.9 N (two lb). The braze joint is easily capable of sustaining this applied load. From the sharp reversal in sign of the axial force at the outermost row of pins, it was evident that the total axial load applied to the pins was important, and the analysis had to consider how additional axial loads could be generated in subsequent processes. This required consideration of the more complex, full assembly of three modules mounted to a card.



Identified concern and need to limit pin load to 22.3 N
(b)

Figure 3

Macro Model II for the MTTS (functionality) tester: (a) simulated handling load of the MTTS operator. (b) Loading case III.

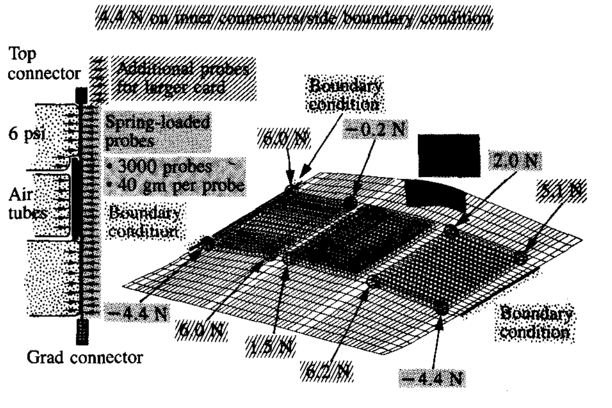


Figure 4

Macro Model II for the NMTE (continuity) tester.

Once it was confirmed that the one-fourth model and the full three-module model gave similar loads on the pins for a uniform change in temperature, efforts were concentrated on understanding various handling conditions wherein the loadings on the pins might be excessive.

Intuitively, high axial loadings on the peripheral pins, but not on the inner pins, of a module would be possible under some conditions of bending. But which process or handling conditions were the most likely culprits? An investigation for possible clues started in the test areas.

Each module-and-card assembly is tested for continuity and for functionality prior to installation in a computer. In the MTTS (functionality) tester, the assembly is plugged into edge connectors at both the top and bottom of the card. It was observed that if the card failed to make good electrical contact to the connectors, the operator would try to shake the card back and forth. Further investigation showed that this shaking force sometimes exceeded 53.5 N (12 lb). This effect was modeled as a module-and-card assembly, with the card clamped at top and bottom, and with forces applied perpendicular to the card as indicated in **Figure 3.** After the NASTRAN model was exercised, it was observed that the corner pins at the outside corners are subjected to an axial load of 44.5 N (10 lb) each when a 44.5-N load is applied to the card. Since this force is applied instantaneously, the solder joint connecting the pin to the card does not have time to relax.

The module-and-card idealization was next used to understand the loadings on the pins generated by the NMTE (continuity) tester shown in **Figure 4.** In this case, the module side of the card is pushed by airbags toward a bank of spring-loaded pin probes. It was thought at first that the applied loadings on the module pins must be compressive. However, upon further study, it was found that initially the connector at the top edge of the card is loaded by several rows of pin probes. At this time, the card is supported at some discrete lengths at the two short edges. When this loading condition was entered into the NASTRAN model, it became clear that the two top outside corner pins on the outside modules were in axial tension and the lower ones in compression, as suggested in Fig. 4. This correlated very well with the failure data, a histogram of which is shown in **Figure 5.** It was estimated that the additional tensile loads, which were to be superposed on the residual loads from wave soldering, were about 44.5 N. As a check to the model, a fully strain-gauged card was prepared. Since measured loads on the corner pins of a module varied from 35 to 45 N (8 to 10 lb), it was found that the loads calculated from the model and the experimental measurements correlated very well. In fact, an earlier experimental study in which two diagonally opposite corner pins were instrumented had concluded that the loading on the pins was indeed compressive. Since this observation agreed with the intuitive feeling of how this tester operated, the corner pins on the other diagonal were not instrumented. In hindsight, we found that, without benefit of a model, the strain gauges had been placed only on those pins which experienced a compressive axial force.

With the results from this investigation of the two testers, immediate actions were taken to establish a design that would minimize the tensile forces which could be applied to the pins. Conveniently, the NASTRAN models were able to quantify the benefit of the corrective actions to improve handling.

- *Brazed joint stress analysis: The micro model*

With a finite-element model constructed for the pin head and brazed joint, the question was "How shall the loading be applied?"

Loadings applied to the model illustrated in **Figure 6** were those forces and moments calculated by the macro model for the most highly stressed pins. The values used are given in **Table 2.** Moments were applied as the statically equivalent vertical (axial) forces distributed across the nodes of the shank of the pin. To facilitate the analysis, the thickness of the idealization was taken to be that value which gave the same cross-sectional area in the shank region as the actual shank. The forces from wave soldering, from handling and tester conditions, and from pull tests were also distributed across the line of nodes at the top of the shank portion of the model.

Idealizations were generated for two different pin geometries. The first was the pin then in use, and a new design with increased head thickness to distribute the loadings across a greater area of the braze because of the stiffer pin head. The second featured an increased head diameter to move the stress concentration effect of the braze fillet away from the point of highest stress. The actual dimensions used are given in **Table 3.**

The effect of increased head thickness was suggested by an analysis using a beam-on-elastic foundation model by L. S. Goldmann [2]. Load sets were applied to both idealizations for the following cases:

1. Card join loads.
2. Card join loads plus 22.3-N (five-lb) tensile pull from handling the card.
3. Straight pull test.
4. Inclined pull test at an angle of 20 degrees.

During the debugging of the idealization, load sets were considered which consisted of only horizontal forces, of only vertical forces, and of only the statically equivalent forces for moments. The value of the moment used in the case of interest was that for the fully plastic response of the pin (0.0069 N-m or 0.06 in.-lb). The analysis of the straight pull test showed that the maximum stress in the braze was located beneath the center of the head of the pin. For all

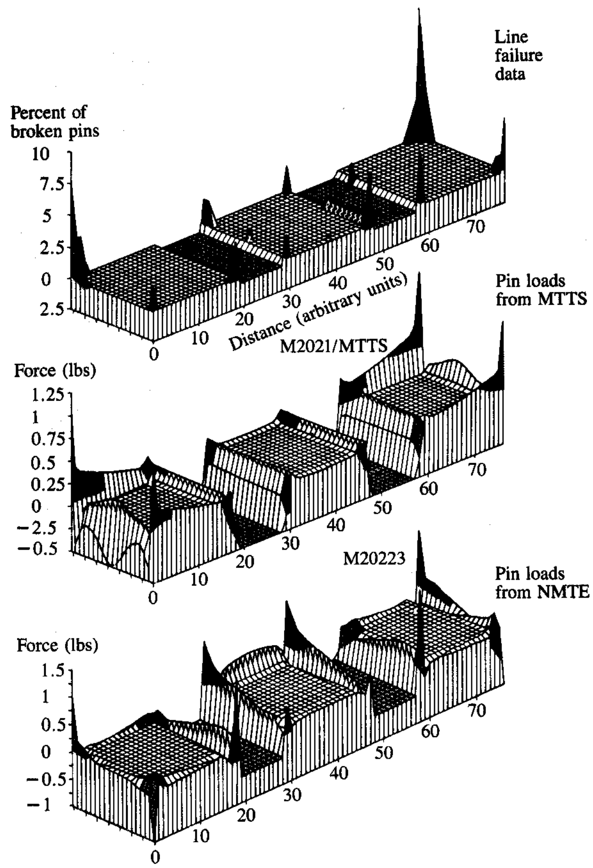


Figure 5

Correlation of broken pins vs axial force.

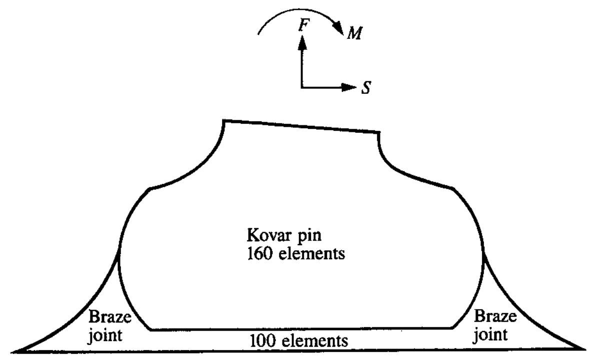


Figure 6

Loadings applied to the micro model, showing the deformed state.

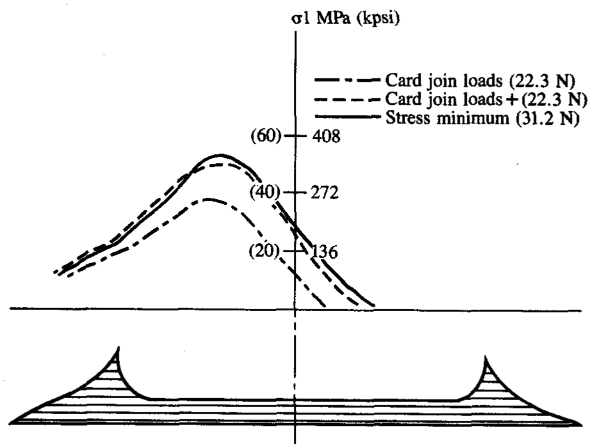


Figure 7

Braze stress in the original design.

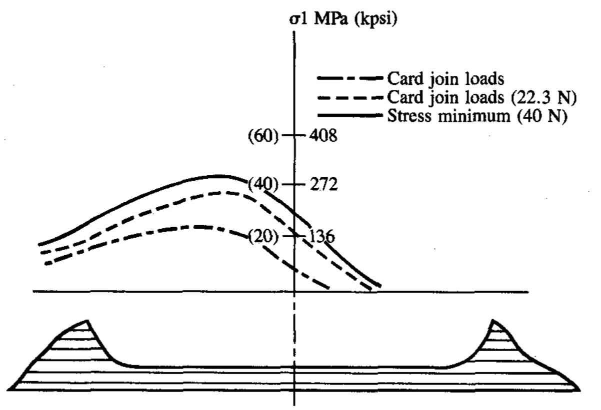


Figure 8

Braze stress in the new design.

other load cases, where horizontal forces and moments are present, the maximum principal stress in the braze was located at a point beneath the edge of the shank of the pin (see **Figure 7**, for example). For this reason an angled pull test was recommended to monitor the integrity of the brazed pin structure: The location of the stress maxima occurred at the same location in the braze on this test as it did in the brazed joints of card-mounted modules.

Because of the low ductility of the intermetallic phases in the pin brace, it was felt that a linear elastic analysis was reasonable to represent this material. For information concerning crack propagation in the braze, attention was focused on the most tensile principal stress, since failures were cohesive failures in the braze. Plots of the magnitude of this principal stress are shown in **Figures** 7 and **8.** They show a dramatic reduction of stress in the braze for the larger-headed pin.

Statistical analysis of pin pull data taken on the original pin design showed that the 3-sigma lower bound of the pin pull strength was 32 N (7.2 lb) for the 20-degree pull test. As can be seen from Fig. 7, this provided no safety factor for a handling load of 22.3 N (five lb) superposed on the card join loads from the wave-soldering operation. With the larger-headed design (Fig. 8 and Table 3) a minimum pull strength of 40 N (nine lb) could reasonably be set as a specification. In point of fact, pull tests on pins with the large-headed design almost invariably have failures occurring in the shank of the pin at a tensile load of 71.3 to 80.2 N (16 to 18 lb) when the pins are pulled at an angle of 20 degrees.

The foregoing analysis and experiments concern the stress state in the joint connecting the pin to the module. A stress analysis of the joint connecting the pin to the card is given in Ref. [3].

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Table 2 Loadings on pins from wave-soldering operation.

Location	*Axial force*		*Horizontal force*		*Bending moment*	
	(N)	(lb)	(N)	(lb)	(N-m)	(in.-lb)
Corner pin (calculated)	9.8	2.2	8.5	1.9	0.0069	0.061
Corner pin (measured)	8.9	2.0			0.0063	0.056
Adjacent pin in outside row (calculated)	3.1	0.7	8.0	1.8	0.0065	0.057
Adjacent pin along diagonal (calculated)	−3.1	−0.7	7.6	1.7	0.0061	0.054

Table 3 Pin dimensions.

	Original design		New design	
	(mm)	(in.)	(mm)	(in.)
Length (between module and card)	1.57	0.062	1.57	0.062
Shank diameter	0.42	0.016	0.41	0.016
Head diameter	0.81	0.032	0.97	0.038
Head thickness	0.20	0.008	0.36	0.014

sites contributed significantly towards the total program. J. Macek did the early analysis and many of the strain gauge measurements. Doug Strope, Tom Wray, Doug Thorne, and Macy Potter did the major share of experimental laboratory measurements for verification of the models. H. Nobel analyzed and provided the failure data for model correlation. To all of them and many other colleagues in development, manufacturing, test quality, and product assurance we express our thanks for their cooperation and contributions. Lastly, we would like to thank L. K. Schultz and R. H. Massey for their support all along the way.

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