

## Field of the Invention

10 Background of the Invention

With reference to FIGS. 1A and 1B, a prior art head suspension assembly 100 comprises a load beam 105, a gimbal 110 and a slider 115. A portion of the slider 115 provides the read/write head, and the slider 115 is secured to the gimbal 110. The gimbal 110, in turn, is secured to the distal end or free-end portion 117 of the load beam 105 by a number of welds 114 on the load beam 105. The proximal end or mounting end portion 118 of the load beam 105 has an opening 120 to which a base plate 125 is attached for securing the load beam 105 to the rotary arm (not shown). The load beam 105 comprises three

portions, a mounting portion 130, which encompasses the opening 120 at the proximal end 118, a support portion 135 that extends longitudinally to the distal end 117, and a relatively narrow resilient portion 140 between the mounting portion 130 and the support portion 135. At the resilient portion 140, an opening 142 improve the out-of-plane flexibility, or reduce the spring rate, of the load beam 105.

The head suspension assembly 100 provides a flexible means for positioning the magnetic read/write head, which allows the slider 115 to pitch or roll as it flies on a cushion of air over the undulating surface of a rotating disk (not shown).

When the disk is rotating, the slider 115 is supported by aerodynamic buoyancy produced by the rotating disk, and when the disk is not rotating the slider 115 is parked on a designated area on the surface of the disk, which is called the parking zone. At the parking zone, the slider 115 is pressed against the stationery disk surface by a spring force or preload force of 1 to 3 gram force (gf) exerted by the resilient portion 140 of the load beam 105. In addition, the load beam 105 has a spring rate of approximately 20 Newtons per meter (N/m). When the disk is rotating, the aerodynamic lift force on the slider 115 counters the preload force of the resilient portion 140, and causes the slider 115 to float or fly over the surface of the rotating disk.

In seek mode the head actuator assembly positions the read/write head over a desired data track on the disk. The time taken for the read/write head to get to the position over the desired data track, including time to settle over the track, is called the seek time. During seek mode, the read/write head is moved radially between selected data tracks to gain access to data stored in the tracks. With the advent of microprocessors that operate at higher speeds, and the ever growing amount of information that is stored in re-write-able storage media, such as magnetic disk drives, there is a need to reduce the time take to read/write information from/to the storage media. Consequently, there is a need to shorten the seek time to allow faster access to data.

During seek mode, the read/write head is initially accelerated and then coast for some period of time at maximum velocity before finally decelerating and settling down over the desired track. To shorten the seek time, the magnitude of acceleration, as well as that of deceleration, is inevitably raised. However, faster deceleration contributes towards higher transient inertial force, which may then adversely excite a range of resonant vibration modes of the head actuator assembly, particularly the resonant vibration modes of the load beam 105. In addition, at higher seek speeds, external wind disturbance induced by the rotating disk can also contribute to unwanted vibration of the load beam 105.

The resonant vibration can be divided into three primary modes according to their motion characteristic. Namely, they are lateral resonance, torsional resonance, transversely bending resonance of the load beam 105. Among the vibration modes, the lateral and twisting motion due to structural resonance is most undesirable since these can cause the read/write head to oscillate and be displaced away from the target track. Under moderate vibration, the read/write head will not settle quickly to its desired position. As a result, the access time will be extended to allow for time for another rotation of the disk before data can be transferred.

When the lateral or sway resonance is excited, the load beam 105 vibrates excessively in the same plane as the movement of the load beam 105 when moving to a position over a desired data track. Since the lateral resonance of the load beam 105 interferes directly with its controlled seeking motion, the accuracy with which the load beam 105 positions the read/write head over a desired track is adversely affected due to the excessive resultant lateral vibration.

When the torsional resonance is excited, the resultant torsional force causes the load beam 105 to twist about its longitudinal axis. Since the read/write head is mounted onto the slider 115 at a height below the load beam 105, the twisting motion of the load beam 105 can translate into lateral motion of the read/write head. Consequently, the

twisting of the load beam 105 displaces the read/write head away from the position over the desired data track on the rotating disk, and adversely affects the read/write operation.

As is known, the data track density of magnetic media has  
5 increased to 100 thousand tracks-per-inch (kilo TPI), the requirement for positioning the read/write head over a desired data track accurately has become more stringent. The conventional approach for suppressing unwanted vibration of the load beam 105 during lateral seeking motion of the read/write head is for the load beam 105 to be made with a very  
10 stiff structure to improve rigidity of the load beam 105 when it moves in the seeking or lateral direction.

Known methods of stiffening the structure of load beam are to employ a load beam, which has a thicker structure, and to add rails  
145 or other reinforcements to the support portion 134 of the load  
15 beam 105 structure. However these methods may increase the mass, and hence the inertia, of the load beam 105, and the increased inertia limits the seek speed for a given torque, which is applied by the rotary arm.

Typically, torsional resonance of the load beam 105 occurs at a  
20 lower frequency than lateral resonance of the load beam 105, however, torsional resonance of the load beam 105 has less gain than lateral resonance of the load beam.

In shortening the seek/settling time, it is desirable to raise the first torsional frequency of the load beam 105 to a greater value to prevent  
25 the first torsional resonance of the load beam 105 from adversely affecting the performance of the read/write head system 100 at speeds at, or below, the maximum desired seek speed.

As will be known to one skilled in the art, the mass and stiffness of the load beam 105 has a significant impact on the structural resonance  
30 frequencies and gain characteristics of the structural resonance frequencies of the load beam 105.

US Patent No. 4992898 by Wanlass, assigned to Applied Magnetic Corporation of the USA, teaches a load beam having a wide structure with a relatively high lateral stiffness, which provides a higher lateral resonance frequency. However, the torsional resonance frequency of Wanlass's load beam is relatively low. Conversely, a narrow load beam is taught in US Pat. No. 5027240 by Zarouri et al, assigned to Computer and Communications Technology Corporation of the USA, having a high torsional resonance frequency, but the lateral resonance frequency of Zarouri's load beam is low.

As disclosed in US Pat. No. 5850319, by Tangren, assigned to Hutchinson Technology Incorporated of the USA, a load beam is provided with an optimized upper profile, which distributes mass transversely and over the longitudinal length of the load beam based on empirical data. The resultant load beam has increased torsional resonance and lateral resonance frequencies. However, the improvement of the first torsional frequency of the Tangren's load beam is inadequate, and does not meet the seek time requirement of 0.2 millisecond (ms) or less, demanded in modern high-performance disk drives.

When designing a load beam, there is also a concern with shock resistance. Compared to desktop applications, mobile applications of magnetic disk drives, such as in laptop computers and portable audio-visual products, a higher non-operating shock characteristic for withstanding accidental drops and bangs is required. When the disk is stationary, in non-operating mode, as described earlier, the slider 115 is pressed against the parking zone by a preload force provided by the resilient portion 140 of the load beam 105. When a disk drive is subjected to a mechanical shock, the slider 115 may be momentarily separated from the surface of the disk. Due to the preload force of the resilient portion 14, the slider 115 returns towards, and collides with, the surface of the disk. Such collisions can damage the read/write head, rendering the disk drive unusable. To improve the shock

resistance, two known approaches are typically adopted. The first is to decrease the vertical stiffness of the load beam 105, and the second is to increase the bending resonance frequency of the load beam 105. A more rigid load beam provides a higher separation shock resistance, however, the flexibility and agility of the load beam are compromised. Conversely, a thin and lightweight load beam may also improve shock resistance, however, this tends to lower the torsional and lateral resonance frequencies.

Relatively simple design methods that modify the thickness, the span, and the width of load beam as taught by the prior art, do not address the conflicting requirements for a load beam with higher lateral and torsional resonance frequencies, as well as higher resistance to mechanical shock.

#### Brief Summary of the Invention

The present invention seeks to provide a load beam and method for designing same, which overcomes or at least reduces the abovementioned problems of the prior art.

Accordingly, in one aspect, the present invention provides a load beam comprising:

an elongate member being substantially tapered along its longitudinal axis between a mounting-end portion and a free-end portion, the elongate member having a lateral cross-section extending along at least a portion of the longitudinal axis, wherein the lateral cross-section has two substantially parallel sides comprising:

a shorter side; and

a longer side; and

wherein the lateral cross-section has two sides extending between the shorter side and the longer side.

In another aspect the present invention provides a method for designing a load beam comprising the steps of:

- a) defining a design domain beam to produce an image of a proposed design of the load beam;
- b) performing topology optimization on the image of the proposed design of the load beam to produce a topologically optimized image of the load beam; and
- 5 c) processing the optimized image of the load beam to produce a manufacturable design of the load beam.

### Brief Description of the Drawings

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An embodiment of the present invention will now be more fully described, by way of example, with reference to the drawings of which:

FIG. 1A shows a plan view of a prior art load beam;

FIG. 1B shows a side view of the prior art load beam in FIG. 1;

15 FIG. 2A shows a plan view of a load beam in accordance with a first embodiment of the present invention;

FIG. 2B shows a side view of the load beam in accordance with the first embodiment of the present invention in FIG. 2A;

20 FIG. 2C shows cross-sectional views of the load beam in accordance with the first embodiment of the present invention FIG. 2A;

FIG. 2D shows a chart comparing the performance of the first embodiment of the present invention in FIG. 2A with that of a prior art load beam;

25 FIG. 3A shows a plan view of a load beam in accordance with a second embodiment of the present invention;

FIG. 3B shows a side view of the load beam in accordance with the second embodiment of the present invention in FIG. 3A;

30 FIG. 3C shows cross-sectional views of the load beam in accordance with the second embodiment of the present invention FIG. 3A;

FIG. 3D shows a chart comparing the performance of the second embodiment of the present invention in FIG. 3A with that of a prior art load beam;

FIG. 4A shows a plan view of a load beam in accordance with a  
5 third embodiment of the present invention;

FIG. 4B shows a side view of the load beam in accordance with the third embodiment of the present invention in FIG. 4A;

FIG. 4C shows cross-sectional views of the load beam in accordance with the third embodiment of the present invention FIG. 4A;

10 FIG. 5A shows a plan view of a load beam in accordance with a fourth embodiment of the present invention;

FIG. 5B shows a side view of the load beam in accordance with the fourth embodiment of the present invention in FIG. 5A;

FIG. 5C shows cross-sectional views of the load beam in  
15 accordance with the fourth embodiment of the present invention FIG. 5A;

FIG. 6A shows a plan view of a load beam in accordance with a fifth embodiment of the present invention;

FIG. 6B shows a side view of the load beam in accordance with the  
20 fifth embodiment of the present invention in FIG. 6A; and

FIG. 7 shows a flowchart detailing a process for designing a load beam in accordance with the present invention.

#### Detail Description of the Drawings

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A load beam in accordance with the present invention has an elongate support portion extending between a proximal end and a distal end, where the width of the support portion tapers from the proximal end to the distal end, and where the thickness of the support portion  
30 tapers laterally from a thicker longitudinal central portion to thinner side portions. A lateral cross-section of the support portion extending along its longitudinal axis comprises two substantially parallel sides, a

shorter upper side, and a longer lower side; and two sides that extend between the shorter upper side and the longer lower side.

The load beam and the method for designing the load beam with such structure, in accordance with the present invention, distributes  
5 mass three dimensionally to form the thicker central portion and the thinner side portions. The thicker central portion advantageously contributes towards the load beam having a higher torsional resonance frequency, and the thinner side portions advantageously contribute towards the load beam having a higher lateral resonance frequency.

10 With reference to FIGS. 2A and 2B, a load beam 205 in accordance with a first embodiment of the present invention is an elongate member that has three portions. Namely, a mounting portion 230 with an opening 220 at the proximal end or mounting end portion 118; a support portion 235 extending to the distal end or free-end portion 117;  
15 and a resilient portion 240, which is located between the mounting portion 230 and the support portion 235. As is known in the art, the opening 220 is for attaching a base plate thereto (not shown) for securing the load beam 205 to the rotary arm (not shown). Typically, the load beam 205 is bent at the resilient portion 240 to generate a  
20 preload force, however, the load beam 205 is shown prior to bending.

The support portion 235 has a thicker central core 250 that extends along a longitudinal axis 251; and two outer rims 255 that have a reduced or lesser thickness relative to the central core 250. The width of the support portion 235 generally tapers from the proximal or  
25 mounting end 118 to the distal or free end 117, and more particularly tapers from the resilient portion 240 to the distal end 117. The support portion 235 has symmetrically shaped sides, each side comprising a pair of adjacent curved-in sections 260 and 262 that extend from a slightly tapered section 263, proximal the resilient portion 240, to an  
30 intermediate point 261, and then to the distal end 117.

The thicker central core 250 also has symmetrically shaped sides, each side comprising a curved-in section 264 that extends from the

resilient portion 240 to a first intermediate point 265, a first tapered section 266 that extends from the first intermediate point 265 to a second intermediate point 267, and a second tapered section 268 that extends from the second intermediate point 267 to the distal end 117.

- 5 Due to the increased width of the load beam 205 at the intermediate point 261, an enlarged portion 269 is formed for each of the outer rims 255. The enlarged portion 269 advantageously contributes towards the load beam 205 having a higher lateral resonance frequency.

FIG. 2C shows lateral cross-sectional views of the first embodiment  
10 of the load beam 205 along sectioning lines 271-274 in FIG. 2A. The thicker central core 250 is thicker about the longitudinal axis 251, and the two outer rims 255 extend away from the longitudinal axis 251. The upper side 253 of the central core 250 is shorter than the lower side 254 of the central core 250, and the upper and lower sides 253 and 254  
15 are substantially parallel to each other. Also, two sides 252 extend between the shorter upper side 253 and the longer lower side 254. In addition, the widths of both the central core 250 and the two outer rims 255 do not vary correspondingly along the longitudinal axis 251.

FIG. 2D shows a chart 280 of a frequency response function (FRF)  
20 282 of a head suspension assembly (HSA) that employs the first embodiment of the load beam 205, and the FRF 284 of another HSA that employs a typical prior art load beam. Vibration is induced at the slider when a HSA, consisting of load beam, gimbal and slider, is accelerated in the seeking direction. When the driving frequency of the  
25 HSA coincides with its mechanical resonance, a sharp increase in the amplitude of the vibration is observed. For example, a spike 286 indicates the torsional resonance frequency of the HSA of the prior art load beam occurs at a lower frequency than a first instance of resonance 288 of the HSA with the load beam 205 of the first  
30 embodiment. In addition, the first instance of resonance 288 of the HSA with the load beam 205 of the first embodiment is the lateral resonance frequency of the load beam 205. Thus, the load beam 205 in accordance

with the present invention, advantageously improves the FRF of the HSA. This is accomplished with a load beam having a torsional resonance frequency that occurs at a higher frequency than that of the prior art.

5       A spike in FRF at the mechanical resonance frequency of a HSA, such as the spike 286, can adversely affect control of the positioning of the read/write head. Hence, a HSA with a relatively smoother FRF is preferred. From the FRF 280, the HSA with the load beam of the first embodiment 205, has a higher vibration amplitude than the prior art  
10       because of the reduction in mass and stiffness of the load beam 205. However, the HSA with the load beam 205 of the first embodiment advantageously provides a FRF that is smoother than that of the prior art. This enables better control of the positioning of the read/write head.

15       Referring now to FIGS. 3A and 3B, a load beam 305 in accordance with a second embodiment of the present invention also has three portions. A mounting portion 330 with an opening 320 at the proximal end 118; a support portion 335 extending to the distal end 117; and a resilient portion 340, which is located between the mounting  
20       portion 330 and the support portion 335. As before, the opening 320 is for attaching a base plate thereto (not shown) for securing the load beam 305 to the rotary arm (not shown). Typically, the load beam 305 is bent at the resilient portion 340 to generate a preload force, however, the load beam 305 is shown prior to bending.

25       The support portion 335 has a thicker central core 350 that extends along a longitudinal axis 351; and two outer rims 355 that have a reduced or lesser thickness relative to the central core 350. The width of the support portion 335 generally tapers from the resilient portion 340 to the distal end 117. The support portion 335 has symmetrically  
30       shaped sides, each side comprising a curved section 360 that extends from a slightly extended section 363, which is proximal the resilient

portion 340, to a tapered section 362, which extends to the distal end 117.

The thicker central core 350 also has symmetrically shaped sides, each side comprising a first tapered section 364 that extends from the resilient portion 340 to a first intermediate point 365, and a second tapered section 366 that extends from the first intermediate point 365 to the distal end 117.

FIG. 3C shows cross-sectional views of the second embodiment of the load beam 305 along sectioning lines 371-374 in FIG. 3A. The thicker central core 350 is thicker about the longitudinal axis 351, and the two outer rims 355 extend away from the longitudinal axis 351. The upper side 353 of the central core 350 is shorter than the lower side 354 of the central core 350, and the upper and lower sides 353 and 354 are substantially parallel to each other. Also, two sides 352 extend between the shorter upper side 353 and the longer lower side 354. In addition, the widths of both the central core 350 and the two outer rims 355 vary correspondingly along the longitudinal axis 351.

FIG. 3D shows a chart 380 of a frequency response function (FRF) 382 of a head suspension assembly (HSA) that employs the second embodiment of the load beam 305, and the FRF 384 of another HSA that employs a typical prior art load beam. As before, a spike 386 indicates the torsional resonance frequency of the HSA with the prior art load beam, which occurs at a lower frequency than a first instance of resonance 388 of the HSA with the load beam 305 of the second embodiment. In addition, the first instance of resonance 388 of the HSA with the load beam 305 of the second embodiment is the lateral resonance frequency of the load beam 305. Thus, as before, the load beam 305 in accordance with the present invention, advantageously improves the FRF of the HSA. This is accomplished with a load beam having a torsional resonance frequency that occurs at a higher frequency than the prior art.

A spike in FRF at the mechanical resonance frequency of a HSA, such as the spike 386, can adversely affect control of the positioning of the read/write head. Hence, a HSA with a relatively smoother FRF is preferred. From the FRF 380, the HSA with the load beam of the second  
5 embodiment 305, has a higher vibration amplitude than the prior art because of the reduction in mass and stiffness of the load beam 305. However, the HSA with the load beam 305 of the second embodiment advantageously provides a FRF that is smoother than that of the prior art. This enables better control of the positioning of the read/write  
10 head.

With reference to FIGS. 4A and 4B, a load beam 405 in accordance with a third embodiment of the present invention has two portions. Namely, a mounting portion 430 with an opening 420 at the proximal end 118 and a support portion 435 extending to the distal end 117. As  
15 is known in the art, the opening 420 is for attaching a base plate thereto (not shown) for securing the load beam 405 to the rotary arm (not shown).

The support portion 435 has a thicker central core 450 that extends along a longitudinal axis 451 with two outer rims 455 that have  
20 a reduced or lesser thickness relative to the central core 450. In addition, the support portion 435 has a thinner free-end portion 456 with a thickness that is less than that of the thicker central core 450. The width of the support portion 435 generally tapers from the proximal end 118 to the distal end 117. The support portion 435 has  
25 symmetrically shaped sides, each side comprising a pair of adjacent curved-in sections 460 and 462 that extend from a slightly tapered section 463, adjacent to the proximal end 118, to an intermediate point 461, and then to the distal end 117.

The thicker central core 450 also has symmetrically shaped sides,  
30 each side comprising a curved-in section 464 that extends from a point adjacent the proximal end 118 to an intermediate point 465, and a tapered section 466 that extends from the intermediate point 465 to a

core distal end 480, which stops short and does not extend to the distal end 117.

Typically, the thinner free-end portion 456 has substantially the same thickness as the two outer rims 455. This advantageously allows the third embodiment of the load beam 405 to have a reduced spring rate and reduced inertia relative to the prior art load beams, with minimal degradation of the torsional resonance frequency of the load beam 405. In addition, to generate a preload force, a resilient portion 482 of the free-end portion 456, proximal to the core distal end 480, can be bent. Alternatively, the load beam 405 can also be bent at a resilient portion 440 indicated by broken lines, and located between the mounting portion 430 and the support portion 435, to provide a preload force. Another alternative is to have bends at both the resilient portions 480 and 482. The load beam 405 is shown prior to bending.

Due to the increased width of the load beam 405 at the intermediate point 461, an enlarged portion 469 is formed for each of the outer rims 455. The enlarged portion 469 advantageously contributes towards the load beam 205 having a higher lateral resonance frequency. However, this is achieved at the cost of reducing a torsional resonance frequency. Hence, an appropriate size for the enlarged portion 469 is to be determined to balance the requirements of both higher torsional and lateral resonance frequencies.

FIG. 4C shows cross-sectional views of the third embodiment of the load beam 405 along sectioning lines 471-474 in FIG. 4A. The thicker central core 450 is thicker about the longitudinal axis 451, and the two outer rims 455 extend away from the longitudinal axis 451. The upper side 453 of the central core 450 is shorter than the lower side 454 of the central core 450, and the upper and lower sides 453 and 454 are substantially parallel to each other. Also, two sides 452 extend between the shorter upper side 453 and the longer lower side 454. In addition, the widths of both the central core 450 and the two outer rims 455 do

not vary correspondingly along the longitudinal axis 451 up to the free-end portion 456.

Referring now to FIGS. 5A and 5B, a load beam 505 in accordance with a fourth embodiment of the present invention has two portions. A  
5 mounting portion 530 with an opening 520 at the proximal end 118 and a support portion 535 extending to the distal end 117. As before, the opening 520 is for attaching a base plate thereto (not shown) for securing the load beam 505 to the rotary arm (not shown).

The support portion 535 has a thicker central core 550 that  
10 extends along a longitudinal axis 551 with two outer rims 555 that have a reduced or lesser thickness relative to the central core 550. In addition, the support portion 535 has a thinner free-end portion 556 with a thickness that is less than that of the thicker central core 550. The width of the support portion 535 generally tapers from the proximal  
15 end 118 to the distal end 117. The support portion 535 has symmetrically shaped sides, each side comprising a curved section 560 that extends from a slightly extended section 563, adjacent to the proximal end 118, to a tapered section 562, which extends to the distal end 117.

20 The thicker central core 550 also has symmetrically shaped sides, each side comprising a first tapered section 564 that extends from a point adjacent the proximal end 118 to an intermediate point 565, and a second tapered section 566 that extends from the intermediate point 565 to a core distal end 580, which stops short and does not extend to  
25 the distal end 117.

Typically, the thinner free-end portion 556 has substantially the same thickness as the two outer rims 555. This advantageously allows the fourth embodiment of the load beam 505 to have a reduced spring rate and reduced inertia relative to the prior art load beams, with  
30 minimal degradation of the torsional resonance frequency of the load beam 505. In addition, to generate a preload force, a resilient portion 582 of the free-end portion 556, proximal to the core distal end 580,

can be bent. Alternatively, the load beam 505 can also be bent at a resilient portion 540 indicated by broken lines, and located between the mounting portion 530 and the support portion 535, to provide the preload force. Another alternative is to have bends at both the resilient portions 580 and 582. The load beam 505 is shown prior to bending.

FIG. 5C shows cross-sectional views of the fourth embodiment of the load beam 505 along sectioning lines 571-574 in FIG. 5A. The thicker central core 550 is thicker about the longitudinal axis 551, and the two outer rims 555 extend away from the longitudinal axis 551. The upper side 553 of the central core 550 is shorter than the lower side 554 of the central core 550, and the upper and lower sides 553 and 554 are substantially parallel to each other. Also, two sides 552 extend between the shorter upper side 553 and the longer lower side 554. In addition, the widths of both the central core 550 and the two outer rims 555 vary correspondingly along the longitudinal axis 551 up to the free-end portion 556.

Referring now to FIGS. 6A and 6B, a load beam 605 in accordance with a fifth embodiment of the present invention is substantially similar to the load beam 505 of the fourth embodiment. The load beam 605 of the fifth embodiment includes an elongate opening 690 that extends from the resilient portion 682 towards the distal end 117. The opening 690 reduces the spring rate of the fifth embodiment of the load beam 605 relative to the spring rate of the fourth embodiment of the load beam 505. A similar opening can also be incorporated in the third embodiment of the load beam 405 with the opening extending from the resilient portion 482 towards the distal end 117.

With reference to TABLE 1 below, a comparison of the performance specifications of HSAs with the load beams, 205 of the first embodiment, 305 of the second embodiment, 405 of the third embodiment, 505 of the fourth embodiment, and the prior art load beam, are provided. These performance specifications are obtained from finite element simulation. Supporting portions of all embodiments are

within a design domain of 15 mm in length, 5 mm wide and 0.076mm thick.

| Parameters                           | Prior Art | 1 <sup>st</sup> (205) | 2 <sup>nd</sup> (305) | 3 <sup>rd</sup> (405) | 4 <sup>th</sup> (505) |
|--------------------------------------|-----------|-----------------------|-----------------------|-----------------------|-----------------------|
| 1 <sup>st</sup> bending (kHz)        | 1.662     | 1.326                 | 1.317                 | 0.968                 | 0.978                 |
| 1 <sup>st</sup> torsion (kHz)        | 3.695     | 5.904                 | 5.234                 | 5.921                 | 5.269                 |
| 1 <sup>st</sup> lateral/sway (kHz)   | 18.365    | 11.959                | 12.675                | 12.695                | 13.861                |
| 1 <sup>st</sup> cantilever (kHz)     | 0.359     | 0.341                 | 0.312                 | 0.342                 | 0.326                 |
| spring rate (N/m)                    | 45.012    | 20.531                | 18.981                | 15.442                | 14.979                |
| lift-off (or shock resistance) (G/g) | 104.759   | 223.593               | 202.465               | 299.026               | 80.099                |

TABLE 1

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A comparison of the performance of the first, second, third, and fourth embodiments are provided in TABLE 1 above, and details of the parameters now follow.

10 The first bending resonance frequency is due to out-of-plane resonant vibration. When determining the first bending resonance frequency, the mounting portion of the load beam is clamped while the slider at the tip of load beam is restrained from moving out-of-plane or in z-direction.

15 The first torsional resonance frequency is due to undue twisting of the load beam about its longitudinal axis. When determining the first torsional resonance frequency, the boundary conditions used are the same as that used when determining the first bending resonance frequency. Specifically, the mounting portion of the load beam is clamped while the slider's surface in contact with a disk is restrained  
20 from moving out of plane.

The first lateral/sway resonance frequency is due to the combined motion of twisting and in-plane swinging. Such mode of vibration can

be catastrophic to positioning of the read/write head, however, the first lateral/sway resonance frequency occurs at a relatively higher frequency than the first torsional resonance frequency, which predominantly limits the bandwidth of position control. When  
 5 determining the first lateral/sway resonance frequency the same boundary conditions as that used when determining the first bending resonance and first torsional resonance frequencies are used.

The first cantilever resonance frequency is due to out-of-plane vibration of the load beam. When determining the first cantilever  
 10 resonance frequency, the mounting portion of the load beam is clamped while its tip is free to move out of plane.

When subjected to excessive shock, the magnetic read/write head, which is originally pressed against the disk, will be lifted up and then bounce back colliding into the disk. This is often referred to as head  
 15 slap. The risk of head slap can be reduced if the head suspension assembly has a low equivalent mass, as this reduces the inertia force of the head.

Lift-off acceleration is defined as the acceleration of shock at which the slider loses contact with the disk surface i.e. the acceleration which  
 20 lifts the slider away from the disk surface, such that the reaction force between the slider and the disk surface is approximately zero. In this context, the lift off is calculated based on the following equation as suggested by Jen et al , "On Suspension Dynamics for Pico-sliders," IEEE Transactions on Magnetics, Vol. 33, No. 5, September 1997.

25

Equivalent mass,  $m_{eq}$

$$= \text{spring rate, } k_s / (2\pi \times \text{the 1}^{st} \text{cantilever frequency, } f_c)^2$$

$$= \text{spring rate, } k_s / (1^{st} \text{ cantilever eigenvalue, } h_c)$$

30  $m_{eq} = k_s / h_c$

$$\begin{aligned} \text{Lift-off acceleration, } a_s &= 1g / m_{eq} \\ &= 10^{-3} \times G / m_{eq} \\ &= (10^{-3} / m_{eq}) \times G \end{aligned}$$

Spring rate is the out-of-plane stiffness measured at the dimple point. Out of plane deflection occurs when a unit force is applied at the dimple in the z-direction.

5

|                                    | Embodiments    |                                     |                                               |                                   |
|------------------------------------|----------------|-------------------------------------|-----------------------------------------------|-----------------------------------|
| Parameters                         | (Prior Art)    | 1 <sup>st</sup> (205)               | 3 <sup>rd</sup> (405)                         | (Prior Art)                       |
| Thickness of rims                  | Full thickness | Thinned rims, 1/3 of full thickness | Thinned rims and front, 1/3 of full thickness | Zero thickness, or no rims at all |
| 1 <sup>st</sup> bending (kHz)      | 1.380          | 1.326                               | 0.968                                         | 1.405                             |
| 1 <sup>st</sup> torsion (kHz)      | 5.143          | 5.904                               | 5.921                                         | 6.747                             |
| 1 <sup>st</sup> lateral/sway (kHz) | 13.821         | 11.959                              | 12.695                                        | 9.847                             |
| 1 <sup>st</sup> cantilever (kHz)   | 0.345          | 0.341                               | 0.342                                         | 0.355                             |
| Spring rate (N/m)                  | 25.952         | 20.531                              | 15.442                                        | 20.289                            |
| Lift-off (G/g)                     | 181.062        | 223.593                             | 299.026                                       | 245.22                            |

TABLE 2

|                                    | Embodiments    |                                     |                                               |                                   |
|------------------------------------|----------------|-------------------------------------|-----------------------------------------------|-----------------------------------|
| Parameters                         | (Prior Art)    | 2 <sup>nd</sup> (305)               | 4 <sup>th</sup> (505)                         | (Prior Art)                       |
| Thickness of rims                  | Full thickness | Thinned rims, 1/3 of full thickness | Thinned rims and front, 1/3 of full thickness | Zero thickness, or no rims at all |
| 1 <sup>st</sup> bending (kHz)      | 1.332          | 1.317                               | 0.978                                         | 1.451                             |
| 1 <sup>st</sup> torsion (kHz)      | 4.457          | 5.234                               | 5.269                                         | 6.751                             |
| 1 <sup>st</sup> lateral/sway (kHz) | 16.273         | 12.675                              | 13.861                                        | 7.990                             |
| 1 <sup>st</sup> cantilever (kHz)   | 0.343          | 0.312                               | 0.326                                         | 0.324                             |
| Spring rate (N/m)                  | 29.022         | 18.981                              | 14.979                                        | 18.489                            |
| Lift-off (G/g)                     | 160.037        | 202.465                             | 80.099                                        | 224.149                           |

TABLE 3

With reference to TABLE 1 above, the first torsional resonance frequency of a HSA with the load beam 205 of the first embodiment, as well as that of the HSA with the load beam 305 of the second embodiment, is better than that of the HSA with the prior art load beam. Also, the third and fourth embodiments of the load beam 405 and 505 have lower spring rates. In addition, the third and fourth embodiments of the load beam 405 and 505 can achieve higher torsional and lateral resonance frequencies and also maintain sufficient out-of-plane flexibility, which is the inverse of spring rate. The performance provided in TABLE 1 is accomplished with a stainless steel substrate having a thickness of 0.076mm. If a thicker substrate with a shorter length is adopted, embodiments in accordance with the present invention can achieve even higher torsional and lateral resonance frequencies, and shock resistance.

With reference to TABLES 2 and 3 above, the effect of reduction in thickness of the rims and frontal edge of a load beam, in accordance

with the present invention, is illustrated. In general, the thinner the rim the higher the torsional resonance frequency of the load beam.

The present invention, as described, provides a load beam, which is suitable for use in very-high density magnetic storage drives. The load beam has a reduced height, is lighter, and provides improved dynamic performance relative to prior art load beams, particularly in relation to first torsional frequency, and shock resistance. Unlike conventional load beams, the load beam of the present invention, as described, does not adopt side rails, ribs or other stiffening reinforcements to improve dynamic performance. Instead, the load beams of present invention utilizes optimal top profile and regions of reduced thickness to enhance performance. Such designs are based on the principle that reducing the mass at the region contributing to a high level of kinetic energy can preferably increase the resonant frequencies. This shift in design paradigm is also supported by the fact that stiffness reinforcement has had a limited impact in enhancing resonant frequencies of higher order in load beams of the prior art.

As described, the load beam of the present invention comprises two portions, namely, a thicker portion, and a thinner portion. The thicker portion has a profile that is defined by its side when viewed from the top, and the thinner portion substantially surrounds the thicker portion. Viewed from the bottom, the outer edges of the load beam, which forms the sides of the thinner portion, define a wider profile than the sides of the thicker portion. As the thicker portion is located centrally on the load beam, the thicker portion is referred to as the thicker central core or central core. Similarly, the thinner portion is referred to as the outer rims of the load beam.

The thicker central core of the load beam in accordance with the present invention, as described, improves the first torsional resonance frequency, however the thicker central core is not sufficiently stiff when moving laterally. The outer rims of the load beam in accordance with the present invention, as described, advantageously improves the

lateral stiffness of load beam. As will be appreciated by one skilled in the art, that the outer rims can adversely affect the first torsional resonance frequency of the load beam. Hence, a design objective is a compromise between the conflicting requirements for high torsional  
5 resonance frequency and high lateral resonance frequency. The outer rims with reduced thickness relative to the thicker central core extend from the thicker central core to advantageously provide a load beam with a higher torsional resonance frequency and at least an adequate level of lateral resonance frequency.

10 As is known, a load beam can be produced by etching a stainless steel foil. A load beam of the present invention can also be produced from stainless steel foil, where the outer rims can advantageously be realized by partially etching the stainless steel foil in conjunction with a photochemical machining process. For example, predetermined  
15 patterns of photoresist are formed on both sides of the foil after which a photochemical etching process is applied. The portions of the foil that are covered by the patterned photoresist are masked or protected from the etching process, and the masked portion corresponds to the inner core of a load beam in accordance with the present invention. Thus,  
20 after the etching process, the masked portion of the foil remains intact, and has the original full thickness of the foil. On the other hand, the portions of the foil that are not masked by the patterned photoresist, will are etched. A through hole is formed in a portion of the foil that is etched from both sides, while a partially etched portion is one that is  
25 etched from one side. A partially etched portion, produced in this way, has a substantially uniform reduced thickness, which may be selectively produced to be approximately 50%, 30% or 25% of the full  
thickness of the foil, based on the degree of etching that is allowed. Aside from uniform reduced thickness, it is anticipated that a load  
30 beam in accordance with the present invention can be fabricated to have a tapering thickness, exponentially decreasing thickness, and a variety of thickness distributed along its length.

Alternatively, adhering two or more pieces of patterned stainless steel foil together can produce a load beam in accordance with the present invention. For example, a first piece of foil having a larger thickness is photo-chemically etched or cut into a profile defined for the thicker central core of the load beam. Then, a second piece of foil having a relatively smaller thickness is photo-chemically etched or cut into the profile defined for the outer rim of the load beam. After separately fabricating the thicker central core and the outer rim, the two pieces of stainless steel foil are aligned and lapped together. Subsequently, the two pieces are adhered to each other by means of laser welding or by means of adhesive bonding. If laser welding is adopted, the two pieces are welded together at several critical locations. On the other hand, if adhesive bonding is adopted, adhesive is dispensed on the abutting areas on the surfaces of the two pieces, which are then held together until the adhesive between them is cured.

According to the first and the second embodiments of the load beam 205 and 305, the thickness of the load beam 205 and 305 ranges between 60  $\mu\text{m}$  to 102  $\mu\text{m}$ . The length of the load beam 205 and 305, from the center of the opening 220 and 230, respectively, to a dimple (not shown) at the distal end 117, is about 18 mm. Since the load beam has a small ratio of thickness to span and does not possess side rails, as the load beams of the prior art, the load beam in accordance with the present invention, as described, is relatively flexible in the transverse direction. The flexibility is distributed throughout its length. In addition, the risk of over stress near the spring region (or resilient portion) owing to physical bending of load beam during assembly process is reduced.

As compared to a prior art load beam with identical thickness, width, and length, the load beam of the present invention in accordance with the first and the second embodiments 205 and 305, respectively, achieve more than 20 % enhancement for the 1st torsional resonance frequency, and more than 100% enhancement for the non-operating separation acceleration.

In addition to improving the first torsional resonance frequency, the load beam in accordance with the present invention, also improves the gain characteristics of a HSA in which the load beam is utilized. As illustrated in FIG. 2D, the HSA with the prior art load beam has significant gain, or abrupt peak, at its torsional resonance when subjected to in-plane acceleration. However, the HSA with the load beam of the present invention shows little gain at its torsional resonance frequency, when subjected to the same acceleration. This is due, at least in part, to the absence of side rails on the load beam of the present invention.

The height of side rails on a prior art load beam is relatively large as compared to its thickness, this tends to cause the side rails to become excited into transverse bending when the load beam of the prior art is at its torsional resonance frequency. Located at outer edges of prior art load beam, side rails contribute to higher torsional kinetic energy, vibrate excessively and cause lateral movement of the slider.

In contrast, the load beam of the present invention has outer rim portions with reduced thickness relative to the thicker central core, which tends to reduce the resonant energy stored over the portion of the load beam close to the outer rims. Hence, the resulting twisting motion of the load beam of the present invention is not of a significant magnitudes to be translated into significant lateral motion of the slider.

Given identical profiles and equal spans, a load beam as thick as 102  $\mu\text{m}$  can achieve higher torsional frequency than a thinner load beam. However, the increment in thickness may adversely render higher inertia, higher spring rate, and lower shock resistance to the load beam. To overcome the disadvantages of the thicker load beam, the profile of the load beam may be defined in a variety of ways that distribute the mass of the load beam.

To achieve higher flexibility of the load beam, in accordance with the present invention, without necessarily sacrificing a higher torsional resonant frequency, the thickness of the free-end portion 456 and 556,

of the third and fourth embodiments, is made as thin as that of the outer rim 455 and 555, respectively. Also, to enhance flexibility of the load beam, in accordance with the present invention, the resilient portion 482 and 582, of the third and fourth embodiments, is employed. As, the  
5 resilient portion 482 and 582 is located close to the distal end 118, the thinner free-end portion 456 and 556 can bend more easily and thus allow improved flexibility.

With a span of 14.5 mm and a thickness of 102  $\mu\text{m}$ , the first torsional resonance frequency of the third embodiment is 10.7 kHz. Due  
10 to the thinner free-end portion 456, the first lateral resonance frequency is improved as well. In addition, the shock resistance of the load beam in accordance with third embodiment improves three times higher than that of the prior art, while the spring rate and mass remain relatively low.

15 To further enhance the flexibility of the load beam, in accordance with the present invention, the length of thinner free-end portions 456 and 556 with respect to the thicker central core 450 and 550, of the third and fourth embodiments are extended. In addition, an opening 690 can be made extending from the resilient portion 582 towards the  
20 distal end 117, to enhance the flexibility of the load beam of the fifth embodiment. Yet another method of enhancing flexibility of the load beam, in accordance with the present invention, is to manipulate the profile of the thicker central core and outer rim accordingly, to reduce vertical stiffness.

25 With reference now to FIG. 7 a process 700 for designing a load beam in accordance with the present invention, starts 705, with defining 710 a design domain for the load beam. This step is akin to defining a design problem and providing initial information pertaining to the design problem. Design objectives and constraint requirements  
30 are formulated and provided as response parameters. A design domain, which encompasses the spatial distribution of all possible designs, is defined with boundary condition parameters and external loading

parameters that will be encountered by the load beam during operation. In addition, material for the load beam is selected as a material parameter. Spatial thickness of the design domain is taken as design parameters or design variables. Initially, the spatial thickness may  
5 start with uniform or random thickness. The spatial thickness is allowed to vary over iterations of optimization.

For example, when designing a load beam for a hard disk drive having disks with a diameter of 2.5 inches, a design domain in a rectangular shape is defined with dimensions of 12mm\_4mm\_0.06mm  
10 for span, width and thickness, respectively. The design domain is modeled as a cantilever with one end, the distal end, free to move while the opposite end, the proximal end is clamped. The design domain is also loaded with a lateral load at its distal end to simulate inertia force produced by the combination of a gimbal and slider, as provided by  
15 loading parameters, and stainless steel is selected as the material from which the load beam is to be constructed by material parameters.

Next, topology optimization 715 is applied to the defined design domain along with the design parameters. Topology optimization is typically performed by a computer program, which essentially solves the  
20 structural design of a load beam as a material distribution problem in the defined design domain. The objective of topology optimization is to satisfy the design requirements by varying the design parameters as they apply to various locations within the design domain. Conventionally, a physical prototype design of a load beam is obtained  
25 through material addition to, or material removal from, a building block of raw material. Similar to the physical process for producing the prototype, simulated material distribution is used to virtually manipulate material by varying the thickness of the structure of a load beam at various locations within the design domain. After an optimum  
30 thickness for the design of the load beam is determined, an optimum design with complicated shapes and holes is automatically generated as a contour image of thickness distribution. This is accomplished by

computer procedures or software, which couples finite element analysis, sensitivity analysis and numerical optimization.

In one embodiment, the design domain is numerically discretized into numerous smaller elements, whose thickness is taken as a design variable, and finite element analysis is used to determine the performance and sensitivity of a design. The response and sensitivity information is then numerically obtained and subsequently used to produce an improved design with a corresponding thickness distribution, which is then used for subsequent analysis and optimization.

After multiple iterations of topology optimization, as described above, an optimized material distribution for a design of a load beam is obtained, in which the thickness of the load beam in the design domain is spatially plotted in grayscale contours. This can be provided either as a display on a user terminal or as a print out. The varying contours indicate portions of the design of the load beam having full thickness, portions having intermediate thickness, and portions having zero thickness. Physically, the portions of full thickness represents solid portions, while portions of zero thickness indicate material removal or holes within the design domain. Portions of intermediate thickness indicate relatively thinned portions that are thinner than the full thickness portions.

#### MODE TRACKING

As is known, vibration of certain resonant modes, for example, torsional and lateral resonance frequency modes adversely affect positioning accuracy of the read/write head, while other resonance frequency modes, such as transverse bending, does not. Consequently, an objective of the topology optimization step is to increase torsional and lateral resonance frequencies of a designed load beam, in order to produce a designed load beam that allows a read/write head to be

positioned more accurately over a wider frequency band, thus supporting faster seek time.

However, transversely bending frequencies need to be maintained at low out-of-plane flexibility for consistent flying height of the read/write head over a disk. Mode tracking is employed in topology optimization to distinguish the resonance frequency modes that are more relevant for improving the performance of a load beam in the design domain from the resonance frequency modes that are less relevant. With mode tracking, the topology optimization step advantageously provides a topology for the load beam that raises the frequencies of the more relevant resonant modes and reduces the less relevant resonant modes. Consequently, the topology optimization software program that is employed incorporates mode tracking that is based on modal assurance criteria (MAC), which was originally proposed by D. J. Ewins, *Modal Testing: Theory and Practice*, Research Studies Press, 1984, for experimental modal testing. MAC is defined as follows:

$$\text{MAC} = |\Phi_{\text{ref}} \bullet \Phi_m|^2 / (|\Phi_{\text{ref}} \bullet \Phi_{\text{ref}}| \bullet |\Phi_m \bullet \Phi_m|)$$

where  $\Phi_{\text{ref}}$  is the reference mode shape; and

$\Phi_m$  is the m-th mode shape to be monitored for a modified structure.

The value of MAC varies between 0 and 1. When the MAC value is equal to 1, the vector  $\Phi_m$  is identical to the mode shape  $\Phi_{\text{ref}}$ , and when the two vectors are orthogonal to each other the MAC value is equal to 0. In mode tracking, the mode with the highest value of MAC is identified as the targeted mode.

After the topology optimization step 715, a topological image in grayscale is produced. This image shows spatial distribution of material, in term of thickness, within a design domain. However, the design of a load beam with continuously varying thickness and complex geometry is very difficult to be realized using standard fabrication

technique of photochemical machining or lapping of multiple layers. Hence, the step of post-processing 720 of the topological contoured image, which is now described.

## 5 CONVERSION OF CONTINUOUS THICKNESS INTO DISCRETE THICKNESS

In order to realize a useful and practical design, the topological image is first manually interpreted. Interpretation converts the image into multiple layers of distinct profiles that are tractable by standard  
10 fabrication techniques. The number of layers that are allowable depends on the availability of particular fabrication techniques. For example, if photochemical machining without partial etching is used, a single layer is possible. When partial etching is adopted, two layers can be realized. On the other hand, a load beam that is made up of more layers with  
15 distinct profiles is possible when of process of joining the layers is also employed. The joining process can include welding, adhesive or three-dimensional printing, as in rapid prototyping processing. However, the final choice of fabrication technique(s) is determined by the cost afforded.

20 For example, the conversion process for a topological image into a design that utilizes a single layer of material now follows. Two design states are allowed within a single layer, namely a solid and a void. To convert a grayscale image into a monochrome image with two states, image filtering is adopted. First, a threshold that distinguishes a void  
25 from a solid portion is set, say at 1% of the substrate thickness. This setting implies that, any location with the material thickness that is less than 1% is deemed as a void. Otherwise, it is deemed as a solid. After an initial image filtering process, adjacent solid portions and voids may alternate in repeating patterns. In this case, the thickness of the  
30 repeating patterns is averaged before proceeding with a subsequent image filtering process. After image processing of the designed load beam in accordance with the post-processing step 720, an image with

the two states is obtained. In other words, a design with primarily solid portions and voids is obtained. To further simplify the designed load beam, small gap(s) or scattered voids in solid portions of the designed load beam are consolidated. Finally, a post-processed image of the designed load beam, that is more suited for manufacturing, is obtained.

When partial etching is employed, a topological image of the designed load beam can be realized having two layers. Here, there are three design states, namely solid, thinned solid, and void. Image filtering is performed using two distinct thresholds; one for defining voids and another for defining thinned solids. For example, a location of the designed load beam having a thickness below 1% is deemed as a void; a location with a thickness that is less than 60% but more than 1% is deemed as a thinned solid portion, and a location having a thickness which is greater than 60 % is deemed a solid portion. Next, the portions of the designed load beam having alternating solid portions and voids are averaged and subjected to another image filtering process. Finally, the solid portions, thinned solid portions, and voids are consolidated separately and result in a load beam design having two layers, where the two-layered design is suitable for manufacturing.

The distinct profiles of the two layers can be realized, respectively, by etching from each of the upper and lower surfaces of a substrate, such as one of stainless steel foil. Where a joining process is employed, the two layers may be fabricated separately, and subsequently aligned and joined together to form a load beam. Following the same approach, a topological image can be converted into a design of multiple layers using multiple thresholds for image filtering. After image filtering is completed, the profiles of the converted image(s) are reconstructed using simple lines and curves. Simple geometric construction of the load beam design is necessary for successful fabrication.

After post-processing and reconstruction, the performance of the load beam design may not be as good as that suggested by the optimum design in variable thickness. Hence, there is a need to refine the load

beam design by tweaking its length, width and other dimensions. In other words, sizing optimization is used to improve the performance of the interpreted design. This is accomplished by a refinement step 725 of the load beam design.

5        Subsequently the performance of the refined load beam design is determined 730 using numerical simulation and experimental testing, and the performance is compared 735 to a desired performance. When the performance of the refined load beam design satisfies the requirements defined in step 710, then the refined design is adopted,  
10        and the design process 700 ends 745. However, when the performance of the refined load beam design does not satisfy the requirements defined in step 710, then the requirements as defined in step 710, are varied 740, and the process 700 returns to step 715 of topology optimization based on the varied requirements, and proceeds as  
15        described. Step 740 of varying the requirements comprises changing the initial design domain and design parameters.

      The load beam of the present invention, as described, advantageously has improved torsional and lateral resonance frequency performance relative to prior art load beams.

20        This is accomplished by a load beam having an elongate support portion extending between a proximal end and a distal end, where the width of the support portion tapers from the proximal end to the distal end, and where the thickness of the support portion tapers laterally from a thicker longitudinal central portion to thinner side portions. A  
25        lateral cross-section of the support portion extending along its longitudinal axis comprises two substantially parallel sides, a shorter upper side, and a longer lower side; and two sides that extend between the shorter upper side and the longer lower side. This load beam structure distributes mass three dimensionally to form the thicker  
30        central portion and the thinner side portions. The thicker central portion advantageously contributes towards the load beam having a higher torsional resonance frequency, and the thinner side portions

advantageously contribute towards the load beam having a higher lateral resonance frequency.

Thus, the present invention, as described provides a load beam and method for designing same, which overcomes or at least reduces  
5 the abovementioned problems of the prior art.

It will be appreciated that although only particular embodiments of the invention have been described in detail, various modifications and improvements can be made by a person skilled in the art without departing from the scope of the present invention.

## Claims

1. A load beam for a head suspension assembly of a magnetic disk storage drive, comprising:
  - 5 an elongate member being substantially tapered along its longitudinal axis between a mounting-end portion and a free-end portion, the elongate member having a lateral cross-section extending along at least a portion of the longitudinal axis, wherein the lateral cross-section has two substantially parallel sides comprising:
    - 10 a shorter side; and
    - a longer side; andwherein the lateral cross-section has two sides extending between the shorter side and the longer side;  
wherein the shorter side and the two sides define a thicker central core portion,  
15 and wherein the longer side and the two sides define a thinner outer rim portion.
2. A load beam in accordance with claim 1 wherein at least one of the two sides tapers relative to the longer side as the at least one of the two sides extends from the shorter side to the longer side.
- 20 3. A load beam in accordance with claim 1 wherein at least one of the two sides is substantially concave between the shorter side and the longer side.
4. A load beam in accordance with claim 1 wherein at least one of the two  
25 sides comprises at least one step between the shorter side and the longer side.
5. A load beam in accordance with claim 1 wherein at least one of the two sides comprises a contoured surface between the shorter side and the longer side.

6. A load beam in accordance with claim 1 wherein the two sides extending between the shorter upper side and the longer lower side are symmetrical.
7. A load beam in accordance with claim 1 wherein the thicker central core  
5 comprises at least one concave portion that extends at least partially between the mounting end and the free end.
8. A load beam in accordance with claim 1 wherein the thinner outer rim portion comprises at least one concave portion that extends at least partially  
10 between the mounting end and the free end.
9. A load beam in accordance with claim 1 wherein the thinner outer rim includes an enlarged portion that extends away from the longitudinal axis at a distance that is greater than that between the longitudinal axis and any point on  
15 the outer rim portion that is adjacent to the enlarged portion.
10. A load beam in accordance with claim 1, the elongate member has at least one opening therethrough.
- 20 11. A load beam in accordance with claim 10 wherein the at least one opening is proximal the free-end for increasing the flexibility of at least a portion of the load beam.
12. A load beam in accordance with claim 11 wherein the at least one opening  
25 extends from the shorter side to the longer side.
13. A load beam in accordance with claim 10 wherein the at least one opening is proximal the mounting end for mounting the load beam.
- 30 14. A method for designing a load beam of claim 1, comprising the steps of:

- a) defining a design domain beam to produce an image of a proposed design of the load beam;
- b) performing topology optimization on the image of the proposed design of the load beam to produce a topologically optimized image of the load beam; and
- 5 c) processing the optimized image of the load beam to produce a manufacturable design of the load beam.

15. A method for designing a load beam in accordance with claim 14 further comprising the step of:

- 10 d) refining the manufacturable design of the load beam to produce a refined design of the load beam.

16. A method for designing a load beam in accordance with claim 15 further comprising the step of:

- 15 e) determining the performance of the refined design of the load beam.

17. A method for designing a load beam in accordance with claim 16 further comprising the step of:

- f) comparing the performance of the refined design of the load beam with  
20 desired performance;
- g) when the performance of the refined design of the load beam substantially meets the desired performance, the refined design of the load beam is adopted as the design for manufacturing the load beam; and
- h) when the performance of the refined design of the load beam does not  
25 substantially meet the desired performance, the refined design of the load beam is changed and steps (b) – (h) are repeated until a subsequent refined design of the load beam substantially meets the desired performance.

18. A method for designing a load beam in accordance with claim 14 wherein step (a) comprises the step of providing at least boundary condition parameters, external loading parameters, and material parameters.
- 5 19. A method for designing a load beam in accordance with claim 14 wherein step (b) comprises the step of modeling a structure of the load beam as material distribution in the defined design domain.
- 10 20. A method for designing a load beam in accordance with claim 19 wherein step (b) further comprises the step of varying the boundary condition parameters, external loading parameters, and the material parameters at various locations within the design domain.
- 15 21. A method for designing a load beam in accordance with claim 14 wherein step (b) comprises the steps of determining an optimum thickness of the proposed design of the load beam, and subsequently generating an optimum design of the load beam.
- 20 22. A method for designing a load beam in accordance with claim 14 wherein step (b) comprises the step of topology optimization that employs mode tracking.
- 25 23. A method for designing a load beam in accordance with claim 14 wherein step (c) comprises the step of manually interpreting a topological image of the load beam to convert the topological image into multiple layers, wherein each of the multiple layers having distinct profiles that are tractable by standard fabrication techniques.
24. A method for designing a load beam in accordance with claim 15 wherein step (d) comprises the step of optimizing the size of the load beam.

25. A method for designing a load beam in accordance with claim 24 wherein step (d) further comprises the step of varying the length and width of the load beam.