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(54) FASTENER ELEMENT

(71) We, YAMAMOTO BYORA COMPANY LIMITED, a Japanese Corporation of 2-2, 4 chome, Arakawa, Arakawa-Ku, Tokyo, Japan, do hereby declare the invention for which we pray that a Patent may be granted to us and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to a fastener element in particular, though not exclusively, such as bolt, screw or nut.

Conventionally, a screw is known which is formed with a slotted head or cross recessed head. This type of screw has the drawbacks that the screw is limited in transmission efficiency of torque and that the inner wall of the slot or recess is likely to be worn or damaged.

In the customary bolt or nut including a hexagonal head or body, a so-called "driving angle", which is defined between a radial line passing through a force-applied point and a tangential line to the driver contour at the force-applied point, is large, for example 60°. For this reason, a driver for the fastener is apt to slip on the outer wall of the bolt head when driving the bolt, so that torque loss is increased causing the angular portions to be worn. As the result, said angular portions are likely to be rendered circular. In addition, the bolt head is made from raw material having a circular cross section and has to be subjected to trimming into a hexagon cross section when worked, which unavoidably results in a complicated manufacturing process and increased manufacturing cost.

For avoiding the above-mentioned drawbacks, there has been proposed fastener element of the type with a head having at its periphery a plurality of grooves and lobes, each lobe being defined between respective adjacent grooves. In this type of head, however, the "driving angle" is still relatively large and therefore the aforesaid drawbacks are not fully eliminated. Further, no consideration is given to the dimensions of the groove and lobe with respect to the mechanical strengths of head and fastener driver. There-

fore, the head lobe is very likely to be damaged or worn.

Accordingly, an object of the invention is to provide a fastener element of the aforesaid type, having a plurality of lobes and grooves at the periphery of its head, which is capable of attaining an extremely high torque transmission efficiency and so designed as to prevent the occurrence of such wear or damage.

Another object of the invention is to provide a fastener element designed so as to prevent the occurrence of damage even when full torque is transmitted to it.

A still further object of the invention is to provide a fastener element capable of attaining a ready attachability to or detachability from a fastener driver and a good engagement therewith.

According to the invention there is provided a fastener element for engagement with a fastener driver for transmission of torque, the fastener element having a portion of its external surface provided with a plurality of grooves extending substantially parallel to a central axis of the portion and a plurality of lobes defined between the respective adjacent grooves, wherein said portion forms a notional first circle at its cross section taken transverse to said axis, each of said grooves forms at its cross section taken transverse to said axis a partial circle inside said first circle, and the center of the circle of which the partial circle is part is situated on a radial line extending from the center of the first circle and being located within or at the limits of a range of said radial line defined between said latter center and a first intersection where two tangential lines intersect, said tangential lines being tangents to the first circle at second intersections respectively where the first circle intersects the circle of which the partial circle is part.

In this fastener element, a component of force is not produced which causes the fastener driver to slip on the element. Therefore, driving torque is wholly transmitted to

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the element from the fastener driver and in addition the element is little worn or damaged.

Where the center of the partial circle is positioned radially outside of the first intersection, as in the case of the prior art, the driving angle has a value greater than 0° with respect to the driving direction of the fastener driver, resulting in the drawbacks as described above.

The center of the partial circle of the groove is preferably situated between the first intersection and a chord connecting the second intersections. Where the center of the partial circle is positioned inside of the chord with respect to the center of the cross sectional circle of the fastener element, the side corners of the lobe become acute and fragile. A further consequence is that the positioning of the fastener driver relative to the fastener element becomes troublesome.

The number of said grooves is preferably in the range from four to eight, and most preferably is six. Where the groove number decreases to below four, it is troublesome to position the driver relative to the fastener element. Where the groove number rises to above eight, the lobe defined between the adjacent grooves becomes small to render the lobe easily wearable and in addition to cause errors to be easily produced in the measurement and positions of the grooves during manufacture.

Preferably, said grooves are equidistantly spaced circumferentially of the male portion, and the diameter of the partial circle of the groove is defined by the equation:

$$\frac{1}{5} \cdot \frac{\pi}{n} \cdot D_0 \leq d_0 \leq \frac{1}{2} \cdot \frac{\pi}{n} \cdot D_0 \quad (1)$$

where n represents the number of grooves, d_0 the diameter of the groove partial circle and D_0 the diameter of the notional first circle.

When the diameter of the groove partial circle is so determined, even full torque transmission causes no occurrence of damage either in the fastener element or in the driver. The reasons are as follows.

Usually, material constituting a bolt, screw or nut is so standardized as to have a tensile strength σ_B of 45—180 kg/mm² while material constituting a fastener driver is so standardized as to have a tensile strength σ_B of 120—180 kg/mm². Since the shearing strength of this type of material, particularly steel, can be regarded as proportional to the tensile strength thereof, the ratio of the shearing strength of the fastener element to that of the fastener driver ranges from 1.5:1 to 1:4. And, since the shearing strength of the fastener element does not usually exceed that of the fastener driver, said ratio comes to range from 1:1 to 1:4. It is understood from the above that if the width ratio of the force-transmitted portion (i.e., the element lobe

defined between two adjacent grooves of the fastener element) to the force-transmitting portion (i.e., the driver protrusion to be engaged with the element groove) ranges from 1:1 to 4:1, the shearing resistances of both the element lobe and the driver protrusion will be equalized. Since the width of the driver protrusion is substantially equal to the width of the element groove and this groove width approximates to the diameter d_0 of the groove partial circle, the diameter d_0 is desirably defined by the equation:

$$\frac{1}{4} \cdot l \leq d_0 \leq l \quad (2)$$

where l represents the lobe width of the fastener element.

On the other hand, the lobe width is substantially equal to the arc length of the outer peripheral edge of said lobe, and can therefore be expressed as follows.

$$l \doteq \frac{\pi}{n} \cdot D_0 - d_0 \quad (3)$$

Accordingly, the equation (1) is derived from the equations (2) and (3). Where the groove number n is set to, for example, six, the relationship between the diameter of the groove partial circle and the diameter of the cross sectional circle of the male portion will be expressed approximately by the equation:

$$0.1D_0 \leq d_0 \leq 0.25D_0$$

Actually, the fastener element whose tensile strength bears the ratio of 1:2 to 1:3 to that of the fastener driver is most widely in use. When, therefore, consideration is given to practicability and economy, the diameter of the groove partial circle preferably is defined as follows.

$$\frac{1}{4} \cdot \frac{\pi}{n} \cdot D_0 \leq d_0 \leq \frac{1}{3} \cdot \frac{\pi}{n} \cdot D_0$$

Preferably the grooves are tapered at an angle of 1° to 10° with respect to the central axis of the fastener element and are more inclined than the corresponding engaging socket of the fastener driver. And the cross section of the lower end of the driver socket corresponds to a cross section of the male portion slightly above the lower end thereof. This type of element configuration enables an extremely easy release of the fastener element from the mold at the manufacturing time and an extremely easy engagement or disengagement of the element with or from the fastener driver. When engaged with the element, the driver is always tightly fitted to said element

at a point slightly above the element lower end. For this reason, the fastener element is prevented from falling off the driver by its own weight, which offers convenience particularly where the element should be brought to a hand-unreachable narrow fastening place.

This invention will now be described by way of example with reference to the accompanying drawings in which:

Fig. 1 is a plan view of a bolt according to an embodiment of the invention;

Fig. 2 is a side view of Fig. 1;

Fig. 3 is a longitudinal sectional view illustrating the condition in which the bolt of Fig. 1 is fitted to a corresponding fastener driver;

Fig. 4 is a cross sectional view on line 4—4 of Fig. 3;

Figs. 5A and 5B respectively illustrate the relationship of the "driving angle" with the groove position of the bolt of Fig. 1;

Fig. 6 is a perspective view of a nut according to another embodiment of the invention;

Fig. 7 is a longitudinal sectional view of the nut of Fig. 6; and

Fig. 8 is a perspective view of a nut according to another embodiment of the invention.

Figs. 1 and 2 respectively show a bolt according to the invention. The bolt comprises a substantially cylindrical male portion, i.e., head 12 forming a circle 18 (Fig. 4) at its cross section, a flange 13 radially extending from the bottom portion of said head, and a threaded shank 14 axially protruding from said flange opposite to the head. Said head 12 is formed in its outer peripheral wall with six substantially semicylindrical grooves 15 disposed equidistantly from each other. Each groove forms, in transverse cross section, a partial circle inside the circle 18. Between the respective adjacent grooves are formed lobes 16 whose outer walls 17 present arcuate cross sections respectively. Said respective grooves each have a constant width throughout their length and extend from the upper end of the head to the lower end thereof substantially parallel to the central axis of the head 12. The cross section of each groove presents a semicircle 19 concaved inwardly of the circle 18. The center of the circle 18 is denoted by O.

Referring to Figs. 3 to 5B, the bolt head 12 is fitted, in use, into an engaging recess 22 formed in a socket portion 21 of a driver 20. Such a driver is described in the specification of our co-pending British Patent Application No. 32928/74 (Serial No. 1,458,089). The recess 22 presents a cross sectional configuration complementary to that of the head 12. Said socket portion 21 is formed at its upper part with a hole 23 having a rectangular cross section. A handle grip, not shown, is adapted to be inserted into the hole 23 so as to rotate the driver.

Engagement protrusions 24 formed in the inner wall of said recess respectively present a substantially semicircular cross section complementary to that of the grooves 15 and are engaged with the respective grooves. The effective transmission of driving torque from the driver engagement protrusion 24 to the bolt head lobe 16 requires for a "driving angle" to be taken into account. This "driving angle" is the one defined between a radial line passing through a point on the partial circle of a groove (said point being a force-applied point at which the driver is allowed to press the bolt head when fastening or releasing the bolt) and a tangential line to the partial circle of the groove at said force-applied point. Where the "driving angle" is greater than 0° with respect to the direction of drive, a component of force is produced which interrupts torque from being sufficiently transmitted to the bolt head and further causes the driver to slip on the outer wall of the bolt head. Due to this slippage the possibility of the side corners of the bolt head lobe being worn or damaged is increased. On the other hand, where the "driving angle" is 0° with respect to the driver-driven direction, the above-mentioned slippage does not occur, thereby effecting an extremely efficient torque transmission and attaining a good engagement of the driver with the bolt head. In view of this, the shape of the bolt head groove 15 in the fastener of the invention has been designed so that the "driving angle" is 0° with respect to the driver direction.

Figs. 5A and 5B respectively schematically show the relationship between various centers of the partial circle of the groove 15 and the driving angle. Since the cross sectional configuration of the bolt head can be regarded as substantially the same as that of the driver recess 22, the outer wall of the head and the inner wall of the recess are drawn by the same curved lines in Figs. 5A and 5B.

Referring to Fig. 5A, the center point O_1 of the partial circle 19 of the engagement protrusion 24, accordingly of the groove 15, is situated on the intersection of the lines 25 and 25a which are tangential to the circle 18 of the head 12 at the intersections P and P₁, where the circle of which the partial circle 19 is part intersects the larger circle 18. In this case, the intersection P is regarded as the force-applied point F when the head is driven in the direction indicated by arrow A, and the tangential line to the drive protrusion 24 (and thus also to the partial circle of the groove 15) at the point P, coincides with the radial

line $\overline{OP}$. Accordingly, the "driving angle", that is the angle between said tangential line

and the radial line $\overline{OP}$ is 0° . This enables full transmission of torque from the protrusion 24 to the lobe 16 to occur.

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Referring to Fig. 5B, the central point O_1 of the partial circle 19 is situated on the chord 26 connecting together the intersections P and P_1 of the circle 18 and the circle of which the partial circle 19 is part. In this case, the point F at which force is applied is situated slightly inside of the circle 18 and the tangential line to the partial circle 19 at

the point F coincides with the radial line $\overline{OF}$. Therefore, the driving angle is 0° as in the above case.

When the center point O_1 of the partial circle 19 is situated as described above, the "driving angle" is 0° with respect to the direction in which the driver is driven. For this reason, the torque transmission efficiency is increased to an extreme extent and the occurrence of the driver slippage is also prevented.

Where the center O_1 of the partial circle 19 is situated inside of the chord 26 with respect to the center O of the circle 18, the driving angle will also be 0° . However, the side corners of the lobe 16 are rendered acute in the proximity of the force-applied point, so that the occurrence of lobe damage is liable to be caused.

In this invention, consideration is given to the material quality of bolt and driver in addition to properly positioning the center of said partial circle, so as to prevent the occurrence of driver or bolt damage even when torque is fully transmitted from the driver to the bolt.

As previously mentioned, where the shearing strengths of materials used to make bolt or driver are taken into account, it is understood that the ratio of the width of the driver protrusion 24 to that of the bolt lobe 16 has to be made to range from 1:1 to 1:4, and preferably from 1:2 to 1:3 in order to equalize the shearing resistances of the bolt and the driver in the vicinity of the force-applied and force-applying points. Again in Fig. 4, the width L of the engagement protrusion 24 is substantially equal to the width L_1 of the groove 15 and the groove width L_1 approximates to the diameter d_0 of the partial circle 19. Therefore, the relationship between the width l of the lobe 16 and the diameter d_0 of the partial circle is defined as follows.

$$\frac{1}{4} \cdot l \leq d_0 \leq l$$

and preferably

$$\frac{1}{3} \cdot l \leq d_0 \leq \frac{1}{2} \cdot l$$

On the other hand, the width l is substantially equal to the arc length of the circle

18 defined between two adjacent grooves 15. Therefore,

$$l \approx \frac{1}{n} \cdot \pi D_0 \approx d_0$$

(where n is the number of protrusions 24 or grooves 15)

When the latter equation is substituted in the former equation,

$$\frac{1}{5} \cdot \frac{\pi}{n} \cdot D_0 \leq d_0 \leq \frac{1}{2} \cdot \frac{\pi}{n} \cdot D_0$$

and preferably

$$\frac{1}{4} \cdot \frac{\pi}{n} \cdot D_0 \leq d_0 \leq \frac{1}{3} \cdot \frac{\pi}{n} \cdot D_0$$

Since, in this embodiment, the number n of grooves 15 is six, the diameter d_0 of the partial circle 19 has to bear the following relationship with the diameter D_0 of the circle 18.

$$0.1D_0 \leq d_0 \leq 0.25D_0$$

and preferably

$$0.12D_0 \leq d_0 \leq 0.17D_0$$

When the diameters of the partial circle 19 and the circle 18 of diameter D_0 are set as above, the respective shearing resistances of the driver engagement protrusion 24 and the bolt lobe 16 are mutually equalized. Even if, therefore, torque transmission is fully effected from the driver to the bolt, neither will be damaged.

in contrast, where

$$d_0 > \frac{1}{2} \cdot \frac{\pi}{n} \cdot D_0$$

the bolt becomes liable to be damaged. Conversely, where

$$d_0 < \frac{1}{5} \cdot \frac{\pi}{n} \cdot D_0$$

the driver becomes liable to be damaged.

In Fig. 3, the groove 15 and lobe 16 of the bolt head are respectively inclined at angles of α and β with respect to the central axis $O-O$ of the head. The angles α and β are preferably equal and defined within the range of $1^\circ > \alpha = \beta \leq 10^\circ$.

On the other hand, the engagement protrusion 24 of the driver and the engagement groove 27 formed between adjacent two of

said engagement protrusions are respectively tapered at angles of γ and θ with respect to the central axis O—O and bear the following relationship with the angles α and β .

$$\alpha = \beta > \gamma = \theta$$

Further, a maximum distance C_0 between the bottom of the groove 15 and the central axis O—O of the bolt head bears the following relationship with a maximum distance C_1 between the apex of the engagement protrusion 24 and the central axis O—O of the driver.

$$C_0 \leq C_1$$

The maximum outer diameter D_0 of the bolt head bears the following relationship with the maximum inner diameter D_1 of the recess 22 of the driver.

$$D_0 \leq D_1$$

Further, the flange 13 is tapered at an angle of S . When, under this condition, the height of the bolt head, namely, the distance between the upper end of the flange 13 and the upper end of the head, is designated by H , and the distance between the head upper end and the engagement point P_0 of the head with the driver is represented by Q , the following relationship is always established.

$$Q \leq H$$

For this reason, when the driver is fitted to the head, unless $Q=H$, the driver does not abut against the upper surface of the flange but reliably fitted to the bolt head at a point slightly above the flange upper end, thereby to be prevented from being readily disengaged from the bolt head. A screw of this type is molded by pressing or cold forging. At this time, the high mold-releasability of bolt from the metal mold is attained on account of the tapered configuration of the molded article, i.e., the bolt head.

Figs. 6 and 7 respectively show a nut according to the invention. This nut comprises a body 28 formed at its central part with a screw hole 29. A plurality of grooves 30, lobes 31 and flange 32 formed in the outer periphery of the body have the same configurations as those shown in Figs. 1 to 5C.

Fig. 8 shows another nut. This nut is the same as the nut illustrated in Figs. 6 and 7 excepting that it has no flange.

WHAT WE CLAIM IS:—

1. A fastener element for engagement with a fastener driver for transmission of torque, the fastener element having a portion of its external surface provided with a plurality of

grooves extending substantially parallel to a central axis of the portion and a plurality of lobes defined between the respective adjacent grooves, wherein

said portion forms a notional first circle at its cross section taken transverse to said axis;

each of said grooves forms at its cross section taken transverse to said axis a partial circle inside of the first circle, and the center of the circle of which the partial circle is part is situated on a radial line extending from the center of the first circle and being located within or at the limits of a range of said radial line defined between said latter center and a first intersection where two tangential lines intersect, said tangential lines being tangents to the first circle at second intersections respectively where the first circle intersects the circle of which the partial circle is part.

2. A fastener element as claimed in Claim 1 wherein said center of the circle of which the partial circle is part is situated between the first intersection and a chord connecting the second intersections.

3. A fastener element as claimed in claim 2 wherein the grooves of said portion are equidistantly spaced circumferentially of said portion, and the diameter of the groove partial circle is defined by the equation:

$$\frac{1}{5} \cdot \frac{\pi}{n} \cdot D_0 \leq d_0 \leq \frac{1}{2} \cdot \frac{\pi}{n} \cdot D_0$$

where D_0 represents the diameter of the first circle, d_0 the diameter of the partial circle, and n the number of the grooves of said portion.

4. A fastener element as claimed in claim 3 wherein the number of said grooves is in the range from four to eight.

5. A fastener element as claimed in claim 3 wherein said portion constitutes a bolt head and has a coaxial threaded shank.

6. A fastener element as claimed in claim 5 wherein a flange is formed between said head and shank.

7. A fastener element as claimed in claim 3 wherein said portion constitutes a nut body and is formed with an internally threaded central screw hole.

8. A fastener element as claimed in claim 7 wherein said nut body has a flange at one end.

9. A fastener element as claimed in any one of the preceding claims, wherein the grooves and lobes of the element are inclined to the central axis of said portion at angles defined by the equation:

$$1^\circ < \alpha = \beta \leq 10^\circ,$$

where α represents the inclined angle of the groove of said portion of the fastener element



said engagement protrusions are respectively tapered at angles of γ and θ with respect to the central axis O—O and bear the following relationship with the angles α and β .

$$\alpha = \beta > \gamma = \theta$$

Further, a maximum distance C_0 between the bottom of the groove 15 and the central axis O—O of the bolt head bears the following relationship with a maximum distance C_1 between the apex of the engagement protrusion 24 and the central axis O—O of the driver.

$$C_0 \geq C_1$$

The maximum outer diameter D_0 of the bolt head bears the following relationship with the maximum inner diameter D_1 of the recess 22 of the driver.

$$D_0 \leq D_1$$

Further, the flange 13 is tapered at an angle of S . When, under this condition, the height of the bolt head, namely, the distance between the upper end of the flange 13 and the upper end of the head, is designated by H , and the distance between the head upper end and the engagement point P_0 of the head with the driver is represented by Q , the following relationship is always established.

$$Q \leq H$$

For this reason, when the driver is fitted to the head, unless $Q=H$, the driver does not abut against the upper surface of the flange but reliably fitted to the bolt head at a point slightly above the flange upper end, thereby to be prevented from being readily disengaged from the bolt head. A screw of this type is molded by pressing or cold forging. At this time, the high mold-releasability of bolt from the metal mold is attained on account of the tapered configuration of the molded article, i.e., the bolt head.

Figs. 6 and 7 respectively show a nut according to the invention. This nut comprises a body 28 formed at its central part with a screw hole 29. A plurality of grooves 30, lobes 31 and flange 32 formed in the outer periphery of the body have the same configurations as those shown in Figs. 1 to 5C.

Fig. 8 shows another nut. This nut is the same as the nut illustrated in Figs. 6 and 7 excepting that it has no flange.

WHAT WE CLAIM IS:—

1. A fastener element for engagement with a fastener driver for transmission of torque, the fastener element having a portion of its external surface provided with a plurality of

grooves extending substantially parallel to a central axis of the portion and a plurality of lobes defined between the respective adjacent grooves, wherein

said portion forms a notional first circle at its cross section taken transverse to said axis,

each of said grooves forms at its cross section taken transverse to said axis a partial circle inside of the first circle, and the center of the circle of which the partial circle is part is situated on a radial line extending from the center of the first circle and being located within or at the limits of a range of said radial line defined between said latter center and a first intersection where two tangential lines intersect, said tangential lines being tangents to the first circle at second intersections respectively where the first circle intersects the circle of which the partial circle is part.

2. A fastener element as claimed in Claim 1 wherein said center of the circle of which the partial circle is part is situated between the first intersection and a chord connecting the second intersections.

3. A fastener element as claimed in claim 2 wherein the grooves of said portion are equidistantly spaced circumferentially of said portion, and the diameter of the groove partial circle is defined by the equation:

$$\frac{1}{5} \cdot \pi \cdot D_0 \leq d_0 \leq \frac{1}{2} \cdot \pi \cdot D_0$$

where D_0 represents the diameter of the first circle, d_0 the diameter of the partial circle, and n the number of the grooves of said portion.

4. A fastener element as claimed in claim 3 wherein the number of said grooves is in the range from four to eight.

5. A fastener element as claimed in claim 3 wherein said portion constitutes a bolt head and has a coaxial threaded shank.

6. A fastener element as claimed in claim 5 wherein a flange is formed between said head and shank.

7. A fastener element as claimed in claim 3 wherein said portion constitutes a nut body and is formed with an internally threaded central screw hole.

8. A fastener element as claimed in claim 7 wherein said nut body has a flange at one end.

9. A fastener element as claimed in any one of the preceding claims, wherein the grooves and lobes of the element are inclined to the central axis of said portion at angles defined by the equation:

$$1^\circ < \alpha = \beta \leq 10^\circ,$$

where α represents the inclined angle of the groove of said portion of the fastener element

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and β represents the inclined angle of the lobe of said portion.

5 10. A fastener element, substantially as hereinbefore described with reference to and as shown in the accompanying drawings.

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