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BE DE NL(71) Applicant: **mitsubishi JUKOGYO KABUSHIKI KAISHA**
5-1, Marunouchi 2-chome Chiyoda-ku
Tokyo 100(JP)(72) Inventor: **Sakamoto, Hidenori Nagasaki**
Shipyards & Engine Works
Mitsubishi Jukogyo K.K. 1-1, Akunoura-machi
Nagasaki-shi Nagasaki-ken(JP)(74) Representative: **Henkel, Feiler, Hänzeler & Partner**
Möhlstrasse 37
D-8000 München 80(DE)(54) **Rotary separator for use in pulverizer.**

(57) The known separator for use in pulverizer of the type that a plurality of blades are disposed about a substantially vertical center axis of rotation as separated by a predetermined radius therefrom and as directed in the vertical direction, is improved so that a cross-section area of passage of airflow between blades can be taken broad and yet it may not be varied even under influence of a centrifugal force. The improvements reside in that the blade is made of a flat plate rather than a shaped material, and that the separator is provided with an annular body for connecting the blades so as to surround the center axis of rotation. A tension to be generated in the annular body as a result of a centrifugal force and a reinforcement effect of the annular body are mathematically analyzed as a basis of design of the improved rotary separator.

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ROTARY SEPARATOR FOR USE IN PULVERIZER

BACKGROUND OF THE INVENTION:

Field of the Invention:

The present invention relates to a rotary separator in a very fine grain size coal pulverizer or other pulverizer.

Description of the Prior Art:

A known coal pulverizer provided with a rotary separator is shown in longitudinal cross-section in Fig. 4, and a rotary separator in the prior art taken out from the coal pulverizer in Fig. 4 is shown in Fig. 5. As shown in Fig. 5, in a rotary separator 01, a large number of blades are disposed so as to surround a center axis of rotation, and by making use of a centrifugal force generated by rotation of the separator and an accompanying airflow, the separator classifies the grain sizes of the powder pulverized by a bowl 03 and pulverizing rolls. For the purpose of increasing a rigidity, blades 02 are formed by employing a material having a flange as shown in Figs. 5 to 7. Or else an I-section material omitted from illustration could be employed.

The above-mentioned rotary separator in the prior art involved the following problems to be resolved:

(1) Due to the broad width of flanges of shaped materials of blades, a cross-section area of passage of airflow between blades is narrowed, and a flow of fine powder is hindered.

(2) As a result of collision of a fine powder flow against the flanges, abnormal abrasion would occur on the blades, and aging deterioration of a performance would arise.

(3) Because of the same reason as that described in paragraph (2) above, a mechanical strength of the blades would be lowered.

SUMMARY OF THE INVENTION:

It is therefore one object of the present invention to provide a rotary separator in which a powder flow is not hindered by blades and a classifying performance can be maintained over a long period without subjecting to aging deterioration.

According to one feature of the present invention, there is provided a novel rotary separator for use in pulverizer, in which a plurality of blades are

disposed about a substantially vertical center axis of rotation as separated by a predetermined radius therefrom and as directed in the vertical direction, and which comprises an annular body for connecting the blades so as to surround the center axis of rotation.

Since the rotary separator according to the present invention is constructed as featured above, it operates in the following manner. That is, in a rotary separator in the prior art, blades would deform outwards due to a centrifugal force accompanying rotation. The deformation presents an aspect just like the case where a beam supported at its opposite ends is subjected to a load distributed almost uniformly. In a normal beam whose span is not extremely short, flexure in the proximity of its center is generally large, hence a large skin stress caused by bending is generated in the cross-section, and the beam is liable to result in rupture. However, in the construction according to the present invention, since there is provided an annular body for connecting the respective blades, the flexure would cause a hoop tension in the annular body. As the hoop tension can be substituted by a simple tensile stress in the axial direction of the annular body, it can be made sufficiently large, thus since it can share a most part of the load caused by the centrifugal forces exerted upon the blades, the blades do not necessitate flanges or the like, and so, the gap clearance between the blades can be taken sufficiently large.

The above-mentioned and other objects, features and advantages of the present invention will become more apparent by reference to the following description of one preferred embodiment of the invention taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS:

In the accompanying drawings:

Fig. 1 shows one preferred embodiment of the present invention, Fig. 1(a) being a plan view (partly omitted), and Fig. 1(b) being a side cross-section view;

Fig. 2 is an enlarged perspective view showing parts of blades 1, a reinforcement ring 2 and a blade bottom mounting plate 3 taken out from Fig. 1;

Fig. 3 is an illustration for explaining mechanical strengths of the blade 1 and the reinforcement ring 2 in the above-mentioned preferred embodiment, Fig. 3(a) being a perspective view, and

Fig. 3(b) being a diagrammatic view for calculating a bending radius from a bending curve (circular arc);

Fig. 4 is a schematic longitudinal cross-section view of a general coal pulverizer which also serves as illustration of the prior art;

Figs. 5(a) and 5(b) are a plan view and a side cross-section view showing a rotary separator in the prior art, which correspond to Figs. 1(a) and 1(b), respectively;

Fig. 6 is a partial enlarged view showing an encircled portion VI in Fig. 5(a); and

Fig. 7 is another partial enlarged view taken along line VII-VII in Fig. 5(b) as viewed in the direction of arrows.

DESCRIPTION OF THE PREFERRED EMBODIMENT:

Now one preferred embodiment of the present invention will be described with reference to Figs. 1 and 2. In these figures, blades 1 are mounted to circumferential edges of a lower cone 5 and an upper spoke 6 supported from a rotor 7 in a cantilever fashion via a lower blade mounting plate 3 and an upper blade mounting plate 4, respectively, so as to surround the rotor 7 as directed substantially in the vertical direction. The blade itself does not have a flange as is the case with the rotary separator in the prior art, but it is formed in a rectangular plate shape and takes an attitude with respect to the rotor 7 such that the plane of the plate shape may be directed in the radial direction of the rotor 7. An reinforcement ring 2 horizontally penetrates through the central portions of the respective blades 1 to connect them with one another. It is to be noted that the penetrating portions are appropriately welded and thus the blades 1 and the reinforcement ring 2 are integrated.

Since the above-described construction is employed, when the rotor 7 rotates, the blades 1 are subjected to an outward load due to a centrifugal force, and even though the belly, that is, the central portion of the blade 1 tends to flex, the reinforcement ring 2 restrains the flexure as if a barrel has been hooped, so that almost no flexure is produced in the blades 1. In other words, since a mechanical strength against a centrifugal force is largely shared by the reinforcement ring 2, the blade 1 does not necessitate a flange for the purpose of reinforcement and maintenance of rigidity, but it could be a simple plane plate, hence an aperture area between adjacent blades in the direction of passage of powder for classification of grain sizes can be made sufficiently large, and since obstacles against a flow are remarkably reduced, a classifica-

tion capability is greatly improved. In addition, the blades were provided with flanges in the prior art as described above, and since the flange is directed almost perpendicularly with respect to a flow of powder, the mass of the flange in the direction of the flow corresponds to the thickness of the flange, hence powder particles coming in flight would strike against the flange initially under the largest kinetic energy condition, thus as the rubbed location is borne by the short distance, that is, by the end surface in the direction of thickness, wear of the end surface is remarkable, and the flange is very quickly subjected to aging change as compared to the portion of web where the powder flow passes nearly in parallel to the surface. Whereas, according to the present invention, the blade 1 is not provided with a flange, and since it includes only a flat plate, aging change would scarcely occur. Here, the term "aging change" implies not only change of shape caused by wear. For instance, an I-section material has an extremely large section modulus as compared to the case where flanges are not provided, owing to the flanges provided at its upper and lower ends, and thus it can maintain a very large bending strength, but if the flanges upon which the strength depends should wear, the strength would be lowered greatly. As compared to wear in the proximity of the center of a cross-section of an I-section material, wear at the outermost end portions would remarkably affect the deterioration of a mechanical strength. This is also true with respect to an angle material, and loss of the flange from the web is very serious. Accordingly, in comparison to a blade which was designed so as not to have a flange initially, aging change of the flange of a blade which was designed expecting a flange effect has a very serious meaning in view of a mechanical strength. According to the present invention, not only such a flange is not provided, but also a mechanical strength relies upon the reinforcement ring 2, and therefore, aging change with respect to a mechanical strength is substantially zero. In this connection, the reinforcement ring 2 corresponds to a circular rod placed perpendicularly in a flow, and because of the facts that it does not have any protrusion which promotes wear and that its cross-section area can be made sufficiently large and the influence upon a mechanical strength is proportional to the cross-section area, wear itself would scarcely occur, and even if wear should arise, a fear of affecting an effective mechanical strength would be nearly zero as will be described later.

Now considering about a classification performance, aging change of the blade having a flange in the prior art, that is, wear of the flange edge portion would broaden a passage cross-section area for a powder flow between the adjacent

blades, and so, the initially set classification performance would be varied. Accordingly, a best classification performance cannot be maintained over a long period, and it is compelled to change rotation and other specifications very frequently. However, according to the present invention, flanges are not present and an interval between the adjacent blades 1 is always substantially constant, and a little wear in the radial direction would scarcely affect the classification performance. Accordingly, not only with respect to a mechanical strength, but also in view of the aspect of a classification performance, a highly reliable and excellent rotary separator can be provided. By the way, commenting on the mechanical strength, the following mathematical analysis is offered. In Fig. 1, if the blades 1 rotate jointly with the rotor 7, since the blades 1 have their lower and upper ends supported by the lower blade mounting plate 3 and the upper blade mounting plate 4, respectively, the portions in the proximities of their centers would expand outwards due to centrifugal forces. With reference to Fig. 1, the radius from the center axis of the rotor 7 to the blade 1 is large at the upper end and small at the lower end, hence if the blade 1 has a uniform cross-section area, that is, a uniform mass along the vertical direction, the centrifugal force would become large as the position shifts upwards, and so, the load does not become a vertically uniformly distributed load. However, if such gradient of a centrifugal force is neglected for the purpose of simplicity, then the blade 1 can be deemed as a beam supported at its both ends and subjected to a uniformly distributed load. When the rotor 7 rotates, since the blades 1 rotate equally on a fixed radius, the centrifugal forces generated in the respective blades 1 have all the same value, and moreover, since the distance between the adjacent ones is relatively short and constant, one can deem that a centrifugal force is distributed uniformly on a radius. Now representing the radius from the center axis of the rotor 7 to the reinforcement ring 2 by r , and representing a centrifugal force (per unit length) distributed on a minute portion of the ring 2 having a minute arc angle $d\theta$ with respect to the center of rotation as viewed in a plan by p , then a composite component force T of a centrifugal force acting upon two cross-sections which cut the reinforcement ring 2 passing through the center axis of the rotor 7 is:

$$T = 2 \int_0^{\frac{\pi}{2}} p \cdot r \cdot \cos \theta \cdot d\theta$$

$$= 2 pr$$

(where θ represents the angle formed by the radius of the ring portion having the minute angle $d\theta$ with respect to a line passing through the center of the rotor that is perpendicular to the cut plane of the above-mentioned ring passing through the center of the rotor). Accordingly a force $T/2$ acting upon one cross-section is:

$$\frac{T}{2} = pr$$

Since this is valid for every cross-section which cuts the reinforcement ring 2 passing through the center of the rotor 7, the reinforcement ring 2 is subjected to a tension of pr in its tangential direction at its every cross-section, and thus forms a hoop tension. As every material would always extend if it is pulled, the reinforcement ring 2 also extends, and its radius also extends a little as compared to that upon stoppage of rotation. Since this extension is naturally equal to a flexure of the blade 1 at that location, schematic illustration of the relation between the blade 1 and the reinforcement ring 2 representing that amount as a flexure δ is given in Fig. 3(a).

In this figure, reference characters A and B designate support points at the upper and lower ends of a blade 1, reference character l designates a length of the blade 1, and character b designates its width. In addition to the support points A and B, the reinforcement ring 2 also serves as a support point, and so, the beam becomes unstationary. In order to calculate the flexure δ caused by rotation, a centrifugal force of the reinforcement ring 2 must be also taken into consideration, and as this is troublesome. Accordingly, in the unstationary beam, still by giving a small flexure δ of such extent that a flexure curve can be deemed as a pseudo-circle, the respective strains are calculated from the flexure curves of the blade 1 and the reinforcement ring 2, and thereby an initial stress ratio is derived. In Fig. 3(a), if a bending radius of the flexure curve (pseudo-circle) of the blade 1 when a flexure δ is produced is represented by R and the radius of the reinforcement ring 2 is represented by r , from Fig. 3(b) is derived the following equation:

$$R^2 = (R - \delta)^2 + \frac{l^2}{2}$$

$$\therefore R = \frac{\delta}{2} + \frac{l^2}{8\delta}$$

Now choosing a middle size class of rotary separators as a model, then as numerical values relatively close to practical ones, l is 800 mm and b is about 70 mm, and so, the bending radius R in the case where a flexure δ of 2 mm, for example, is

generated in this model would become as follows:

$$R = \frac{2}{2} + \frac{800}{8 \times 2}$$

$$\doteq 40000 \text{ (mm)}$$

On the other hand, for instance, in the case where steel is employed as a material of the blades 1, within an elastic limit of steel since a compression stress and a tensile stress caused by the same strain are nearly identical, an neutral axis in the direction of width b of the blade 1 caused by bending would appear at the position $b/2$ apart from the inner end or the outer end. Accordingly, for instance, a strain ratio ϵ_t at its outer end is represented by the following equation:

$$\epsilon_t \doteq \frac{b}{2R} = \frac{70}{2 \times 40000}$$

$$\doteq 8.8 \times 10^{-4}$$

Representing the corresponding skin stress (tensile stress) by σ_t and a Young's modulus of steel by E ($= 21000 \text{ Kg/mm}^2$), then the following value is derived:

$$\sigma_t = E \cdot \epsilon_t$$

$$= 21000 \text{ Kg/mm}^2 \times 8.8 \times 10^{-4}$$

$$= 18 \text{ Kg/mm}^2$$

Whereas, a tensile stress σ_r generated in the reinforcement ring 2 is calculated, assuming the radius r of the reinforcement ring 2 to be 900 mm, as follows:

$$\sigma_r = E \cdot \epsilon_r$$

$$= E \cdot \frac{\delta}{r}$$

$$= 21000 \text{ Kg/mm}^2 \times \frac{2}{900}$$

$$= 47 \text{ Kg/mm}^2$$

Then a stress proportion m of the reinforcement ring 2 is calculated as follows:

$$m = \frac{\sigma_r}{\sigma_t + \sigma_r}$$

$$= \frac{47}{65}$$

$$= 0.72 \text{ (= 72\%)}$$

Thus, this value is remarkably large as compared to the blade 1. Accordingly, with respect to the reinforcement ring 2, the blade 1 always stands on the safety side either in view of a mechanical

strength or in view of a deformation (strain). In other words, it is seen that the reinforcement ring 2 largely contributes to the mechanical strength of the blade 1. In this connection, since a strain multiplied by an area is a force, if a hoop tension generated in the reinforcement ring 2 (that is, the previously calculated $T/2 = pr$) is divided by σ_r , then a cross-section area of the reinforcement ring 2 can be obtained. The diameter of the reinforcement ring 2 can be determined within the scope allowed by the width b of the blade 1, and within what numerical value range the flexure δ should be contained, can be also arbitrarily chosen. At that time, so long as attention is paid to the mechanical strength of the reinforcement ring 2, there is no need to be anxious about rupture of the blades 1 at all.

While the reinforcement ring 2 was employed one at the middle position of the blades 1 in the above-described embodiment, if necessary, a plurality of reinforcement rings could be employed. In addition, the blades 1 are not limited to the flat plate shape. Though the reinforcement ring 2 and the blades were welded with each other in the above-described embodiment, unless the problems of noises and abrasion are present, the reinforcement ring 2 could be simply penetrated through the blades 1. The cross-section of the reinforcement ring 2 is also not limited to a circular shape.

In the above-described embodiment, since a flat plate is used as the blade 1, and the reinforcement ring 2 penetrates through the middle portions of these blades and connects them together, powder can easily pass through the gaps between the adjacent blades 1, and since any protrusion such as a flange traversing a flow of powder is not present, wear would hardly occur. Accordingly, aging deterioration of a classification performance would not occur, and aging deterioration of a mechanical strength also would hardly arise. In addition, as the reinforcement ring 2 largely contributes to the mechanical strength of the blades 1 and suppresses deformation of the blades upon rotation, there is also provided an advantage that the scope of selection for the thickness and shape of the blades 1 can be remarkably broadened.

Since the rotary separator according to the present invention is constructed as described above, the following effects and advantages are provided:

(1) Under an action of a centrifugal force generated by rotation, since most of the mechanical strength of the blades is shared by the annular body, the anxiety about rupture of the blade is eliminated, and so, a freedom for the shape, length and the like of the blades is greatly enhanced.

(2) Since it becomes unnecessary to provide a flange on the blade, the gap distance between the blades is broadened, hence a flow of powder becomes not to be hindered, and a classification performance is improved.

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(3) As the blade does not necessitate a flange which is liable to wear, aging change of the gap distance between adjacent blades is not present, and so, a classification performance can be maintained high over a long period.

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(4) Since the blade does not necessitate a flange which is liable to wear, and since the mechanical strength depends mostly upon the annular body which is hardly subjected to aging change, deterioration of a mechanical strength caused by aging would not occur.

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While a principle of the present invention has been described above in connection to one preferred embodiment of the invention, it is a matter of course that many apparently widely different embodiments of the present invention could be made without departing from the spirit of the invention.

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Claims

1. A rotary separator for use in pulverizer in which a plurality of blades are disposed about a substantially vertical center axis of rotation as separated by a predetermined radius therefrom and as directed in the vertical direction, characterized in that said separator is provided with an annular body for connecting said blades so as to surround the center axis of rotation.

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2. A rotary separator for use in grinders as claimed in Claim 2, characterized in that said flat blade is disposed along the radial direction of the center axis of rotation.

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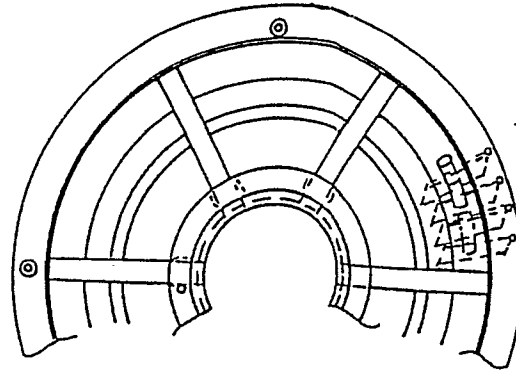
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Fig. 1

(a)



(b)

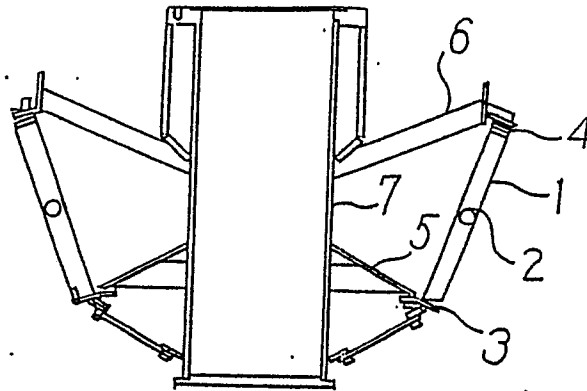


Fig. 2

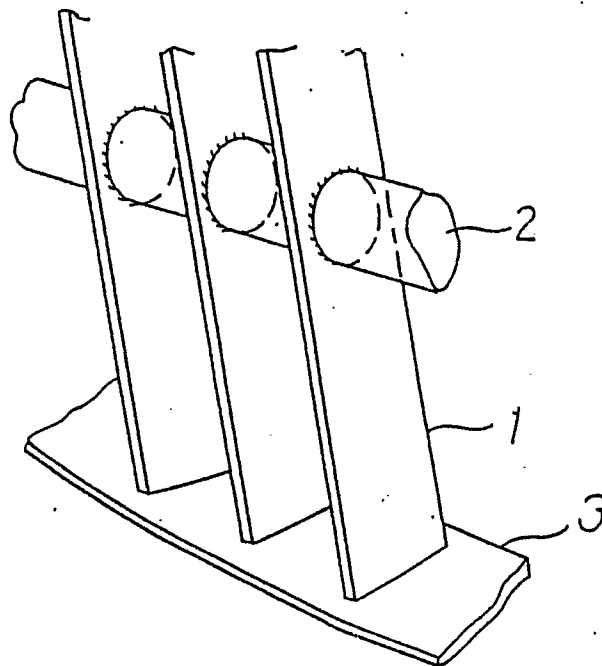
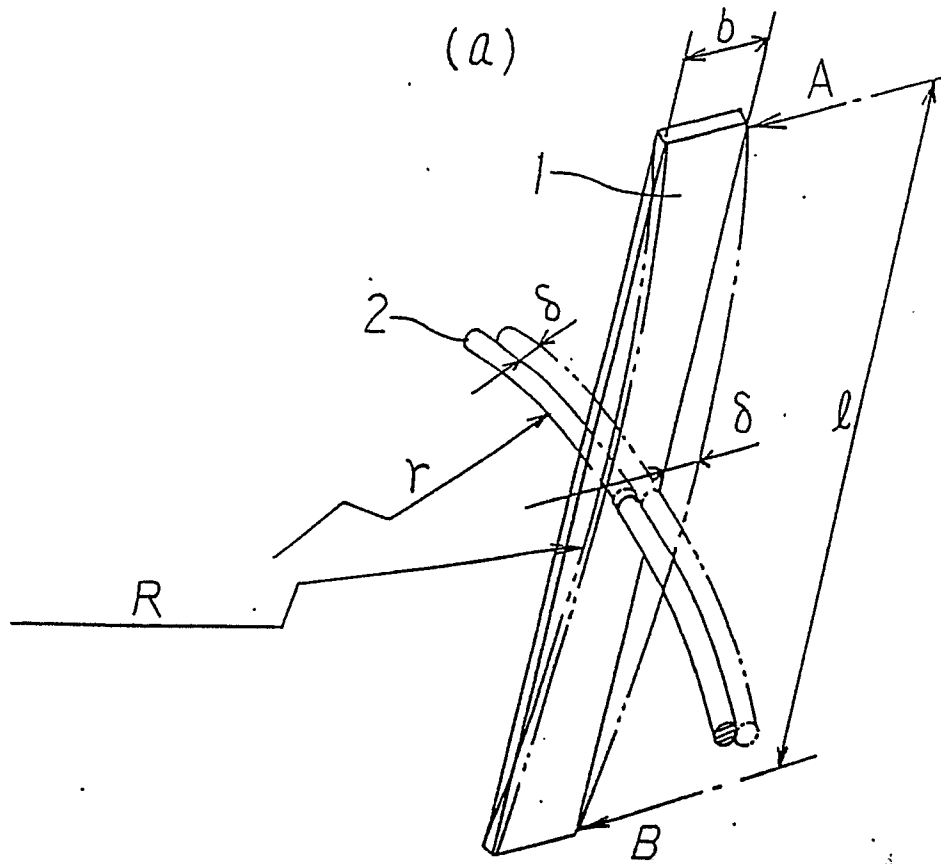


Fig. 3



(b)

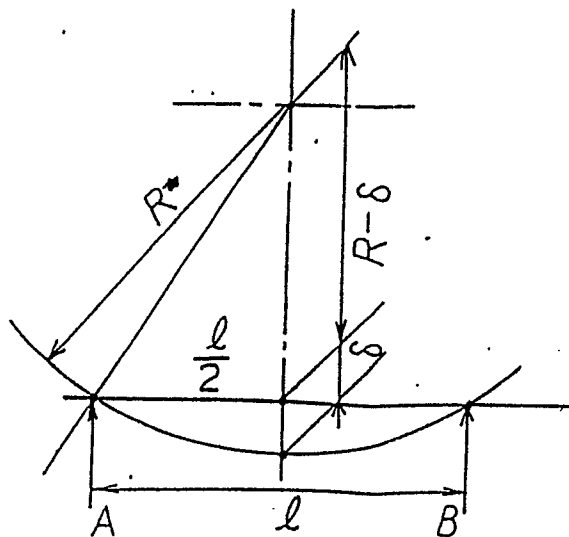


Fig. 4 (Prior Art)

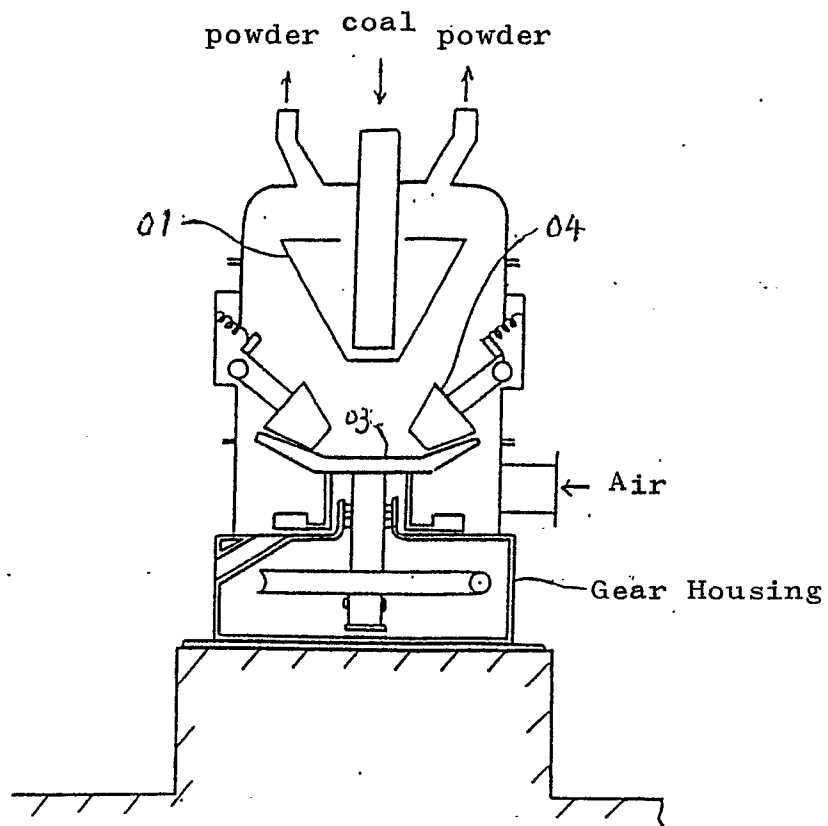


Fig. 5 (Prior Art)

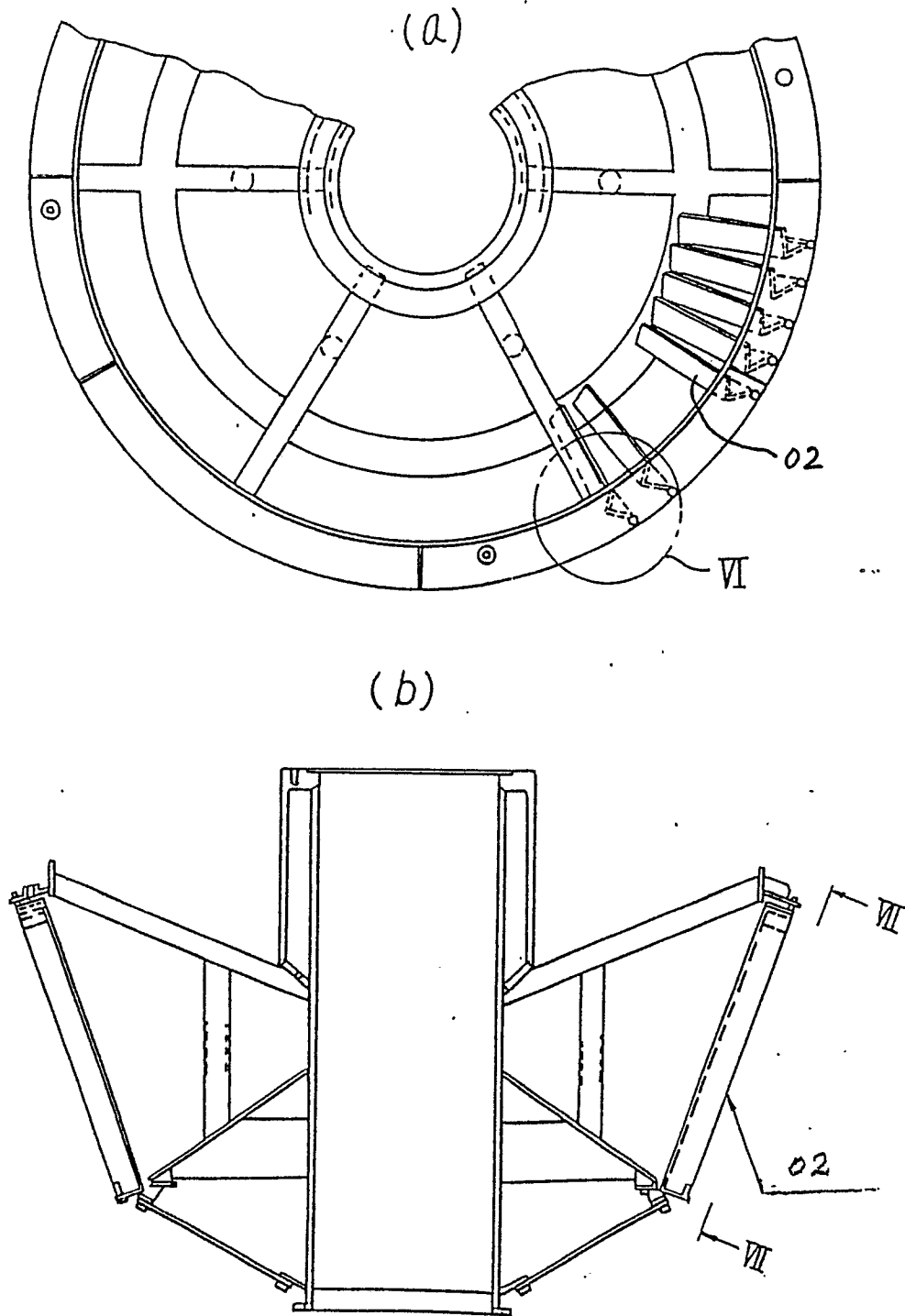


Fig. 6 (Prior Art)

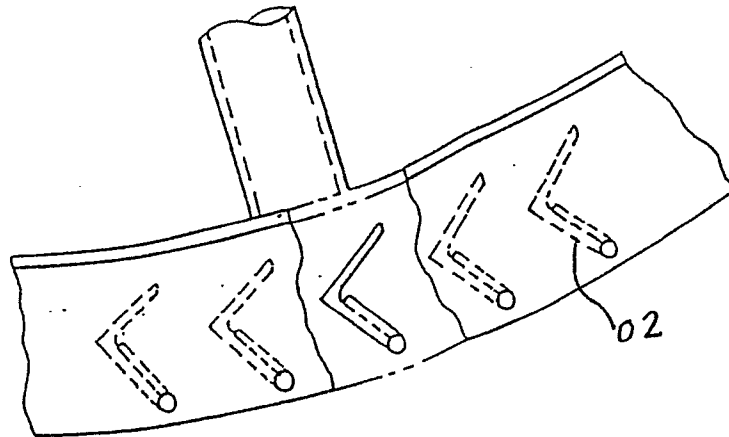


Fig. 7 (Prior Art)

