Designing Process and Theories of Rotary Blades for Better Rotary Tillage (Part 1)

By Jun SAKAI

Department of Agricultural Machinery, Faculty of Agirculture, Mie University

Traditional tillage works in paddy fields were performed mainly by animal plows and harrows. The rotary tillage with hand tractors, appeared in Japan in around 1950, showed that it achieves both plowing and harrowing at one pass of the machine on the field. The hand tractor coupled with the rotary tiller is called "koun-ki" in Japanese, power tiller in English, and rapidly spread in the whole farming area in Japan from about 1955 to 1965. Now, riding tractors with rotary tillers are becoming as popular as power tillers[1]. Thus plows and harrows so far used have been almost replaced by rotary tillers in Japan.

Rotary blades of rotary tiller are equivalent to the share and mouldboard of plow. It can be definitely said that to develop and use the rotary blade of rational shape is indispensable for effective tillage.

The author has studied to obtain the dimensions necessary for drawing rotary blade, as shown in Fig. 1, through theoretical equations composed of machine specifications and quantitative expressions of desirable farm works, but not through intuition.

The designing theories of rotary blade will be presented briefly following the actual process of designing.

Drawing of Japanese rotary blade

Plate 1 shows one of the recent blades made in Japan[2]. As a rule, the drawing expresses the industrial shape of the blade to be produced, although there are many ways of expressing dimensions according to designers. Fig. 1 shows one of them. Drawing forms of

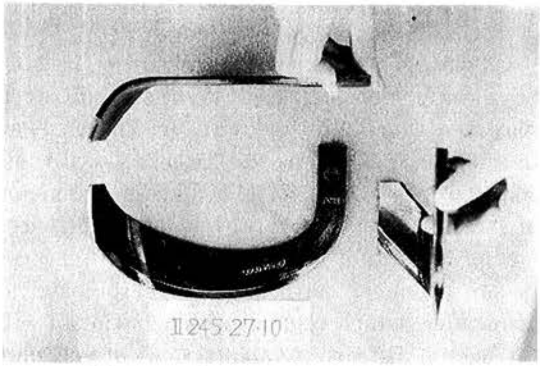

Plate 1. A rotary blade in Japan

blade dimensions in main Japanese manufacturers are becoming almost similar to that of Fig. 1.

The dimensions, which are closely interrelated each other, can be classified into four groups to express the following shapes:

1. Fixing (shank) portion
2. Edge curve and back curve of lengthwise portion
3. Scoop surface of sidelong portion (blade tip portion)
4. Section of blade including knife-edge

The drawing of rotary blade completes with these perfect dimensions and also some notice discriptions about its material, hardness distribution after heat treatment, surface finishing, marking of its brand name and part number, number of right- and left-facing blades necessary for each rotary tiller and so forth.

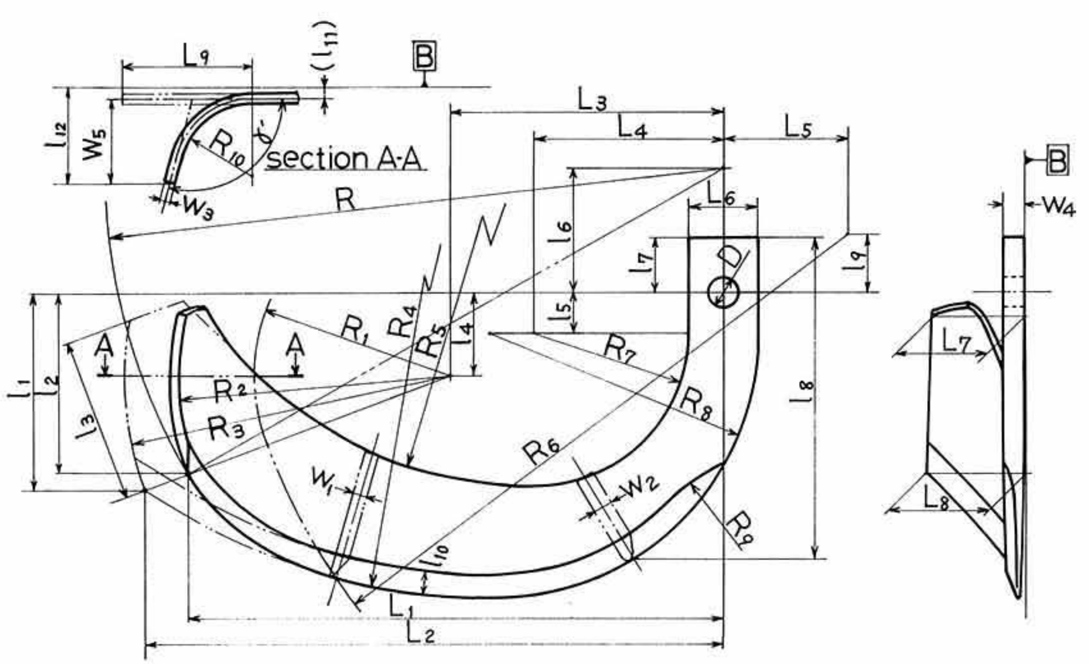

Fig. 1. An example of Japanese rotary blade drawing

Process of planning design for drawing

The drawing of a blade as shown in Fig. 1 cannot be drafted at one time, but in general the designing proceeds through the following stages:

1. Planning design
2. Part drawing

In other words, such a drawing as shown in Fig. 1 is drafted after the planning design is finished and reasonable readjustments of the original dimensions of blade shape drafted on planning design sheet are made to facilitate a smooth manufacturing process. There are also several methods of planning design. Fig. 2 shows one of the flow charts expressing the development of a new rotary blade adapted to a given tractor specifications. The author's research has been done with the idea of selecting important design factors from the flow chart of an actual blade designing.

Decision of tillage performance and specification

The travelling speed v cm/sec of the rotary tiller is controlled by walking ability of drivers in case of hand tractor. As for the riding tractor, the speed as fast as possible within the range of its engine horsepower and the stability of the machine will be required. However in general, a travelling speed higher than the hand tractor is not so practical, since tillage width of the rotary tiller is required to be at least equal or wider than the outside width of both drive wheels.

General values of v is in the range of 25 to 70 cm/sec. The speed range of 25 to 40 cm/sec is available for hard or heavy soil condition, 40 to 50 cm/sec for normal soil condition, and 50 to 70 cm/sec for such light tillage as inter-row cultivation or puddling etc.[3,4].

Rotation speed n rpm of a rotary axle of the tiller is usually in the range of 130 to 400 rpm. The tangential force T kg at the tip of the rotary blade, which is an imaginary

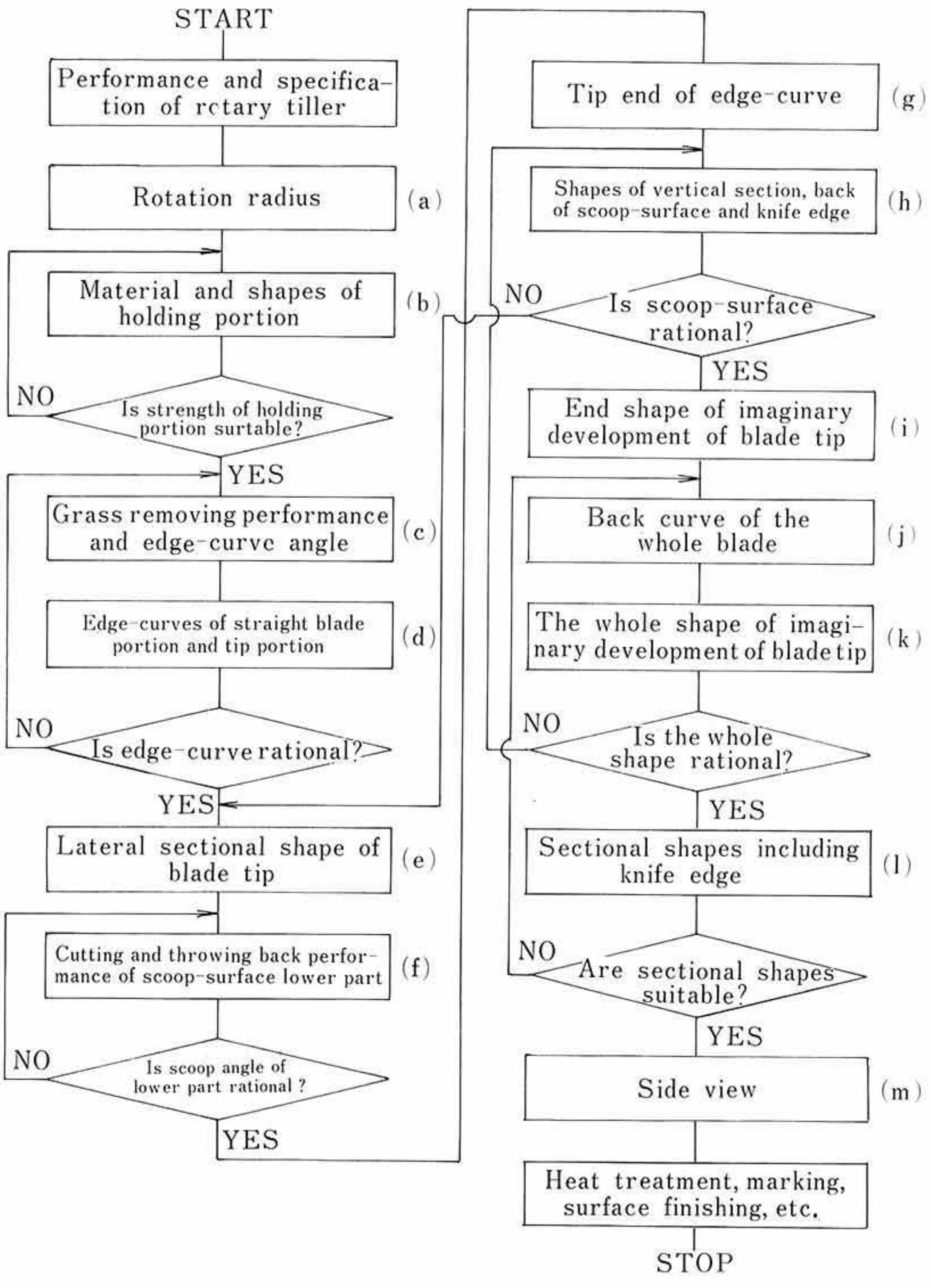

Fig. 2. A flow chart of planning design
(The marks from (a) to (m) mean those of Figs. 3 & 7)

force and named tillage force[3], is calculated by:

$$T = \frac{71620\ Ne\ Pe}{nr} \quad \cdots\cdots\cdots\cdots\cdots (1)$$

Where

- Ne: engine output, ps
- Pe: power transmission efficiency from the tractor engine to the rotary axle (in case of the closed-loop power transmission system)[5].
- r: turning radius of the blade, cm

Smaller values of n rpm result in rough pulverizing of soil and stronger tillage force, causing bigger driving-forward force P_1 and lifting-up force R_1 of the soil to the machine[3].

These two forces are the external forces acting on the machine and are contributing to the motion of the machine as a free body. This means that the lowest rotational speed of the rotary axle must be selected so as to be in the range of keeping the machine stable for the certain soil condition and machine specifications. These two external forces P_1 and R_1 are presumed to be affected by the radial-suction force τ_{12} of the blade which is caused by the shape of the scoop-surface of it, and by the knife factor τ_{13} which depends on the shape of knife edge[6].

On the other hand, the higher rotational speed n rpm of the rotary axle produces the smaller tillage force T kg. (refer to Equation 1). However, the increasing work of high pulverizing effect to the soil slices is apt to consume bigger horsepower. There is a possibility of causing the machine vibration because of unbalanced mass and/or force of rotating parts. The highest rotational speed of n rpm has to be decided so as to meet farmers' demand for the pulverizing effect to soils.

The maximum depth of tillage is also decided by farmers' practices (more than 20 cm in upland farming in Europe and America whereas less than 18 cm in Asian rice cultivation and usually 13 to 15 cm in Japan)[1].

Turning radius of blade[3]

The turning radius R of the blade consists of the tillage depth and an allowable dimension "a" that cannot sink in hard soil mechanically, such as the bottom radius of a transmission case to the rotary axle. The equation is:

$$R = H + a \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots (2)$$

(a) in Fig. 3 can be drafted after this calculation. Actual values of R is in the range of 200 to 255 $^{+0}_{-4}$ mm.

Material and mechanism of holding and fixing

There are two types of blade holding mechanism: the flange-disk type for European blades and the holder type for Japanese ones. The results of some studies on the statical strength of Japanese-typed holding were as follows[4].

According to the simplest equation of the cantilever of rectangular equal section, its safety factor should be 5.8 or more practically.

One of more accurate methods is to obtain the maximum stress σ_{max} of the curved beam of equal rectangular section with bending and twisting stress. The safety factor in this case is 4 or more. These calculations and dimensions have to be obtained on the basis of the tension strength of the blade material. Spring steel, generally SUP-6 under JIS, is recommended for the material, which has 125 kg/mm^2 of tension strength after heat treatment.

The sectional dimensions of "width × thickness" are:

$$(20 \text{ to } 35^{+0}_{-0.7}) \times (8 \text{ to } 12^{\pm 0.5}).$$

A bolt of 10 mm dia. is recommended to the blades for power tillers and 12 mm dia. for riding tractors. Drafting (b) of Fig. 3 will be done after the decision of the relative length of the holding portion and location of a hole for the bolt, as shown in Fig. 4.

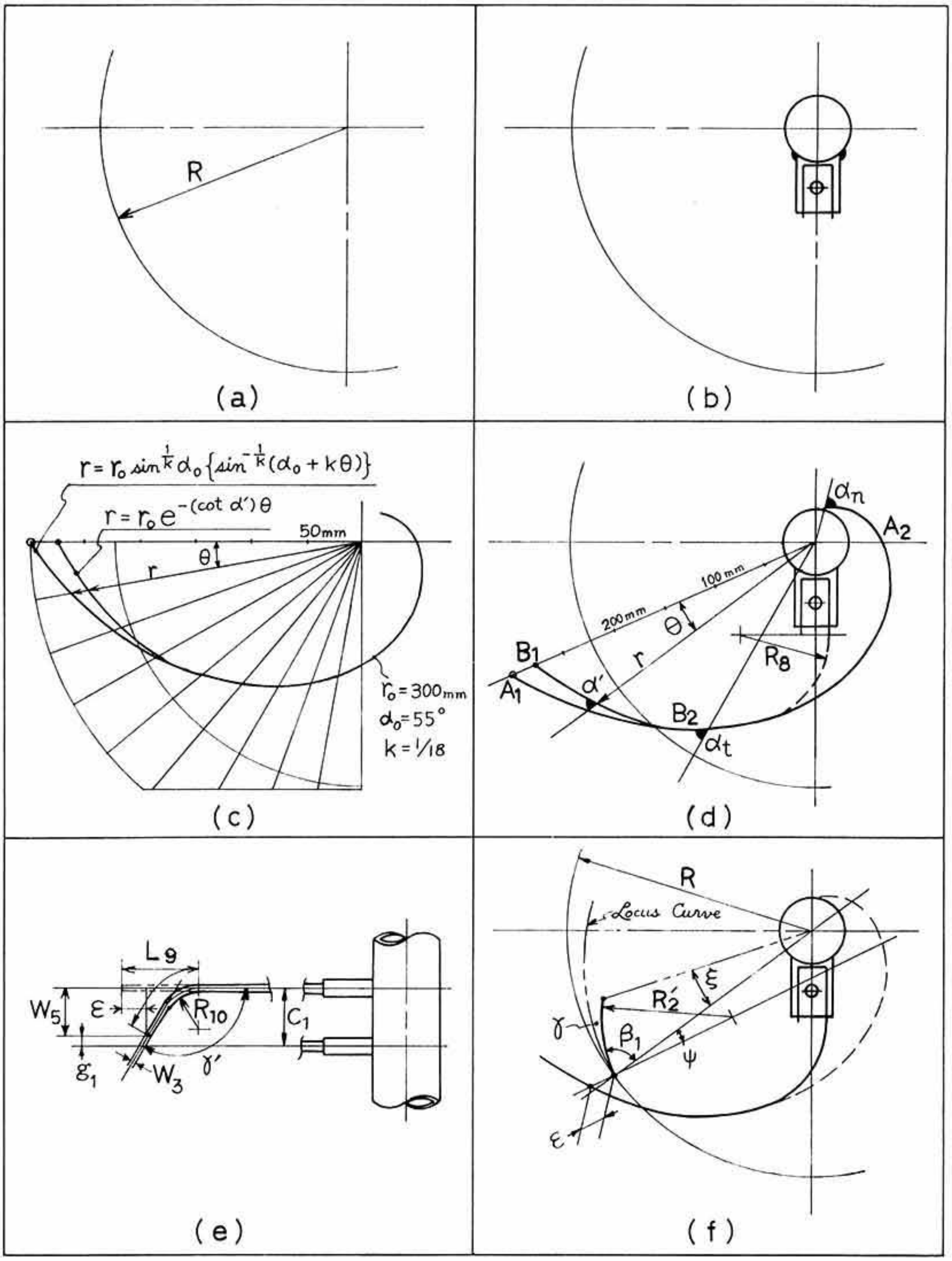

Fig. 3. Drawing process of planning design

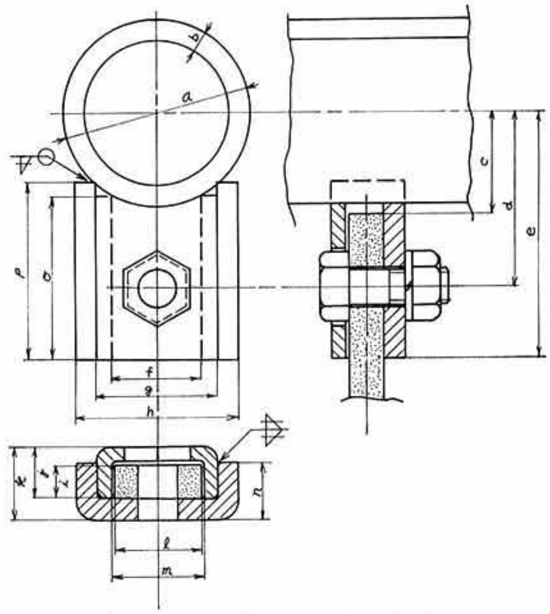

Fig. 4. Planning of a blade holder

Edge-curve and grass removing performance of blade

The coiling trouble of grass and straw to the rotary tiller occurs more easily on the soft soil of paddy fields than the hard soil of upland fields. The grass removing ability of blade will be better with the bigger edge-curve angle, α, of Fig. 5. According to the experiments done in Japan and the Philippines, the edge-curve angle α_t at the tip portion of such

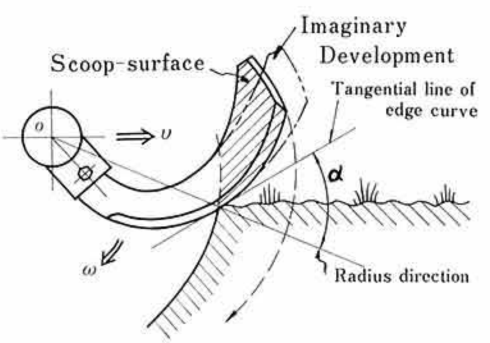

Fig. 5. Edge-curve angle α

a straight blade as shown in (d) of Fig. 3 is recommended to be 57.5°, and α_n at the rotary axle is 67.5°[7,8].

Thus, in an actual design, these angles, α_t, α_n, α', are rationally decided in accordance with grass and straw condition on the given field, and the edge-curve will be obtained by the spiral equation of polar coordinate.

The edge-curve of the straight blade portion is[7]:

$$r = r_0 \sin^{\frac{1}{k}} \alpha_0 \{\sin^{-\frac{1}{k}}(\alpha_0 + k\theta)\} \qquad ......(3)$$

where

- r_0: maximum radius at the tip point of edge curve
- α_0: edge-curve angle at r_0
- k: a constant to give changing angles from α_t to α_n If an increase of 10 degrees is expected from α_t to α_n within the θ range of 180°, k is 10/180, namely, 1/18.
- r: calculated radius of the spiral, that depends on the variable θ in the function, as shown in (c) of Fig. 3.

The imaginarily developed edge-curve angle, α', shown in Fig. 5 and (d) of Fig. 3 is approximately calculated from the following equation[8].

$$\alpha' = \alpha_t + \left[\left\{\tan^{-1}\frac{n\pi\sqrt{H(2R-H)}}{30v - n\pi(R-H)}\right\} + 90°\right] ...(4)$$

where

- n: rpm of the rotary axle
- H: maximum depth of cut, cm
- v: machine travelling speed, cm/sec
- R: rotation radius of the blade, cm
- α': this will be 4° or more bigger value than α_t.

The edge-curve of the sidelong portion is obtained by substituting the angle α' into the following equation[7,8]:

$$r = r_0 e^{-(\cot\alpha')\theta} \ ....(5)$$

The connected edge-curve $B_1 B_2 A_2$ shown in (d) of Fig. 3 is the basic one with a reasonable grass removing performance along the whole edge. The curve of A_1 portion may be erased. The shape of the shank portion expressed with dotted lines in (d) of Fig. 3 has to be drafted in consideration of the production system and strength of its material.

Sectional shape of blade tip portion

Such a sectional shape of the tip scoop surface as shown at the left top of Fig. 1 has to be decided. The necessary dimensions for it are the radius R_{10} of the lateral curvature of the scoop-surface, angle γ' and cutting width W_5 (see (e) of Fig. 3).

R_{10} of smaller radius and the angle γ' which approaches 90° will have possibilities of increasing tillage resistance and of causing trouble in production process, though the surface of hardpan will become smoother and flatter owing to them. R_{10} and γ' of bigger values will cause smaller tillage resistance and easier production process, but uneven hardpan surface, and soil slices will be thrown sideways. These dimensions are selected in accordance with the purpose of the given tillage work. These values in general are in the range of 25 to 50 mm for R_{10} and 100° to 130° for γ'.

Cutting width W_5 is determined by the clearance c_1 of the blade holders on the rotary axle[9]. A proper dimension is obtained with the idea of gap g_1 between the tip end of the sidelong blade and the center line of its neighboring blade (see (e) of Fig. 3). The holder clearance c_1 of Japanese rotary tiller is usually 40 to 60 mm.

The bigger gap means shorter cutting length of the blade in the soil and produces smaller tillage resistance. This means a bigger gap is effective to have smaller cutting resistance in case of hard soil. The usual gap is 5 to 10 mm in Japan.

When the section shape like (e) of Fig. 3 is drafted, the length L_9 of the center line of the scoop-surface has to be calculated. This length is useful to determine the necessary length L_9 of Fig. 1 and the imaginarily developed shape of blade in the process (i) of Fig. 2.

Clods throwing capacity of scoop-surface

When the scoop-surface of the blade cuts into the soil, it should create an action to throw back soil clods against soil slices. (The back-surface of the scoop-surface must never push untilled soil)[3]. The strength of the throwing effect depends on the scoop-angle β_1[3,4,10], as expressed in (f) of Fig. 3.

The smallest throwing effect is expected from the scoop-surface that moves along the direction angle β of locus curve. The scoop-angle β_1 smaller than β produces stronger throwing effect. However, too small scoop-angle will cause big tillage resistance[3,10].

The angle β between the radius direction and tangent of locus curve is calculated by the following equation[3,4].

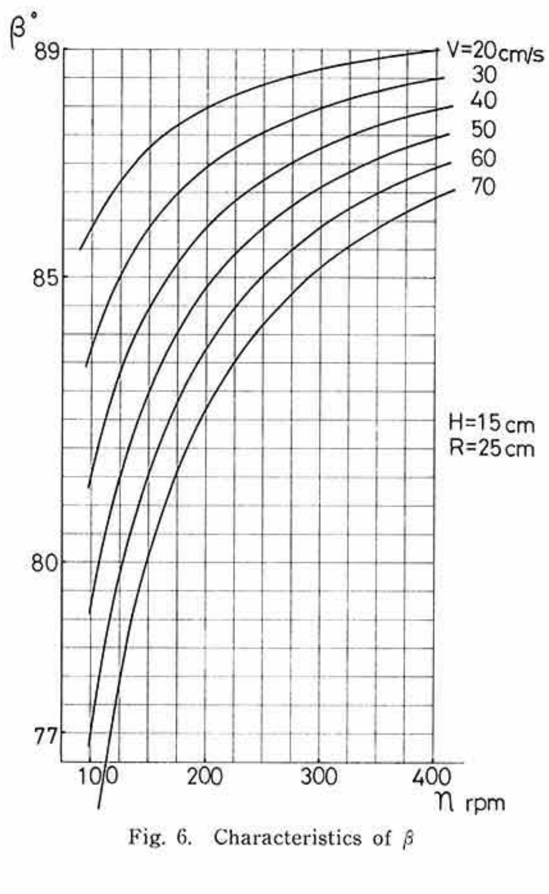

Fig. 6. Characteristics of β

$$\beta=\cos^{-1}\left\{\frac{30v}{R}\sqrt{\frac{H(2R-H)}{(30v)^2-60n\pi v(R-H)+(Rn\pi)^2}}\right\} \qquad \cdots\cdots(6)$$

Fig. 6[4] shows characteristics of β. The $(\beta-\beta_1)$ is relief angle γ of the back of the scoop-surface, as shown in (f) of Fig. 3. Namely:

$$\beta_1=\beta-\gamma \qquad \cdots\cdots(7)$$

When smaller cutting resistance of the scoop-surface edge is expected, for example in case of hard and heavy soil, γ will be 3° to 4° with the idea of angular allowance in production process. The γ of 20° to 30° is useful to give strong throwing-back and turning-over actions to the soft soil. Values of 10 to 20 degrees are common for γ.[3]

Thus, angle ψ shown in (f) of Fig. 3 is obtained from the following equation, in order to find the location of the radius center of the scoop surface curvature:

$$\psi=90°-(\beta-\gamma) \qquad \cdots\cdots(8)$$

Summary

The designing process together with summarized design theories on individual parts of the blade is presented on the basis of the author's research related to design theories of rotary blades done in last several years.

An example of the flow chart for planning design which is necessary to complete the drawing of the blade is proposed in Fig. 2. However, only an initial half of the whole series of drafting steps, i.e., from the start to (f) of the flow chart shown in Fig. 2 is given in (a) through (f) of Fig. 3, because this paper constitutes part 1 of the whole paper.

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