A study on the slip of the centrifugal clutch of the fan-governor applied for the automatic braking devise of the logging skyline-crane

Yasushi MINAMIKATA*

I. Introduction

In Japan there are many dowhill cableways and skyline cranes of gravity system, for which the fan-governors could be employed for the supplementary automatic brakes.

Some types of the fan-govermors for the gravity cableways have been already proved to be very effective in many cases of the practical logging operations[1]. But, untill recently, the fan-governors are very seldom utilized for the skyline yarding cable-cranes, because there are some difficulties in design and structural setting of the fan-governor which always satisfies the various requirements caused by the various conditions of the yarding operations.

In 1962, a unique type of the improved fan-governor, to which the centrifugal clutch was attatched, had been made by OWASE-KOSAKUSHO factory under the guidance of Prof. Dr. S. Kato, and his staffs of the Institute of Forest Utilization of Tokyo University. Some machines of this type were used in the logging sites of the Tsuruga and Owase National Forests for the purpose of experimental logging operations. In this chance, the auther could have the opportunity to carry out some field experiments about the efficiency of the fan-governor. In this paper, some important problems on the centrifugal clutch are discussed.

The author acknowledges with many thanks the guidance given by Prof. Dr. S. Kato and Asst. Prof. M. Maruyama for this contribution; also he is very much obliged to the kind helps by Mr. H. Hukusada for the field experiments.

II. The mechanism of the fangovernor and the result of the field observation

The some fundamental theory of the fangovernor[2] could be applied for the braking devise, whethere it is installed for a cableway or a yarding skyline-crane. But the mechanical structures in detail should differ from one another, because the operating system of the cable way and that of the skyline-crane are quite different.

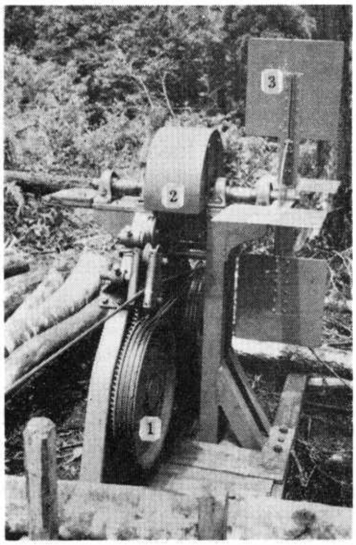
The most important behaviors which characterise the operation of the downhill yarding skyline-crane, could be noted as follows:

1) Only one skyline is used for the carrier cable, which shoud be montaged and demontaged within a short period. This means that the length and gradient of the skyline vary considerably, and also even the amount of the unit load changes in wide

* Inst. of forest Utilization faculty of Agriculture University of Tokyo

range. So that the varying requirements for the braking cappacity should be always taken into account. For this reason, it is necessary for the fangovernor to have not only the changable fan-plates but also a powerful clutch, and some transmission geare too.

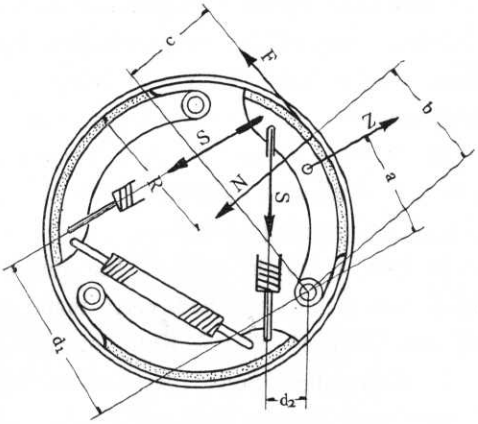
2) Only one carriage runs on the skyline in the to and fro system, i. e. the carriage brings down the logs and the logs are unloaded at the lower end, then it is hauled back on the same cable by the motor power upto the loading point. Therefore, fan-governor should resist against the movement of the downhill loaded carriage, while



1. Driving pulley
2. Centrifugal clutch
3. Fan-plate

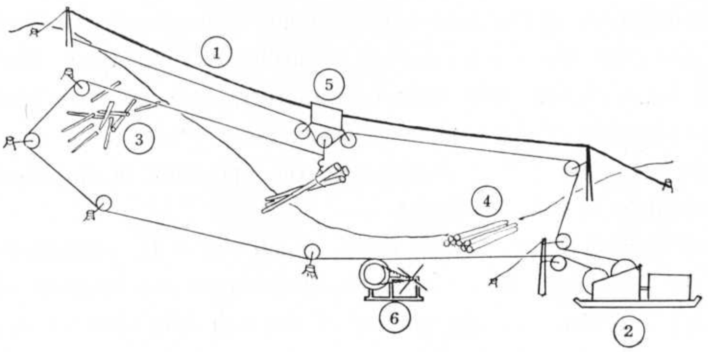
Diameter of fan (D): 1.0 m
Width of fan plate (B): 0.3 m
High of fan-plate (H): 0.3 m
Number of fan-plate: 4
Torque coefficient (C_Q): 1.05
Air density: 0.125
Increasing gear ratio: 4

Fig. 1



G: Weight of a shoe (=1.3 kg)
R: Radius of clutch drum (=0.14 m)
μ: Coefficient of friction (=0.3)
a: 0.087 m b: 0.087 m
c: 0.09 m
d_1: 0.14 m d_2: 0.031 m

Fig. 3



① Track cable
② Yarder
③ Logs to be hauled
④ Logs unloaded from carriage (to be loaded on trucks)
⑤ Carriage
⑥ Fan-governor

Fig. 2

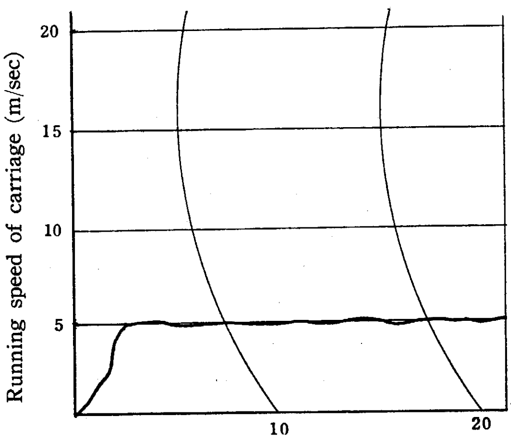
it should be kept free when the empty carriage is going uphill. For this reason, application of an adequate clutch is inevitable.

Considering the characteristic requirements for the skyline-crane, a unique mechanism, shown in Fig. 1. was given to the new machine. This governor is not an ordinary fan attatched to the yarder, but an independent governor-set, which could be placed at any where, and through which the haul back line (braking line) runs. The whole yarding system of the skyline-cane is chown in Fig. 2.

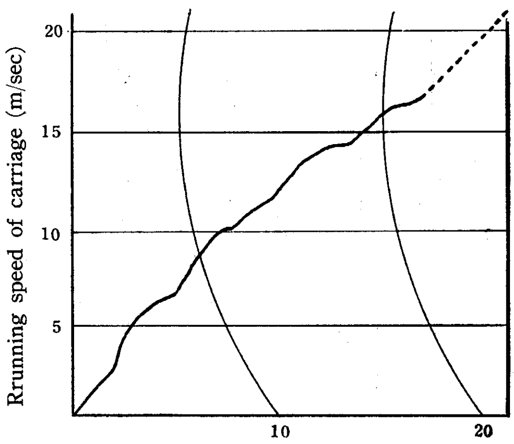
For the clutch of this machine, an automatic centrifugal clutch, shown in Fig. 3, of the IWATE FUJI Y-25 yarder was employed, because it is usually available in the market and seems to be far more convenient than any other hand-operating cone clutch or disk clutch.

The field experiments were carried out for the purpose of confirming effect of the fan-governor mentioned above.

In these experiments, the downhill running speed of the loaded carriage was recorded on the ossirograph-charts for the following both cases, (1) when the traction-cable was braked by fan-governor and (2) when the traction cable was made free from the governor, so that the carriage would run freely by its own gravity. Fig. 4–a, b are the representative examples of the records.



Lapse time after the start of carriage (sec)
Load: 380 kg, braked by the fan with four wings
Fig. 4, (a)



Lapse time after the start of carriage (sec)
Load: 100 kg, free from the fan
Fig. 4, (b)

It is obvious from the comparison of these two charts that the fairly good braking effect of the fan-governor could be recognized. But, during the time of the controlled speed experiment, undesirable phenormena of overheating of the centrifugal clutch was always observed.

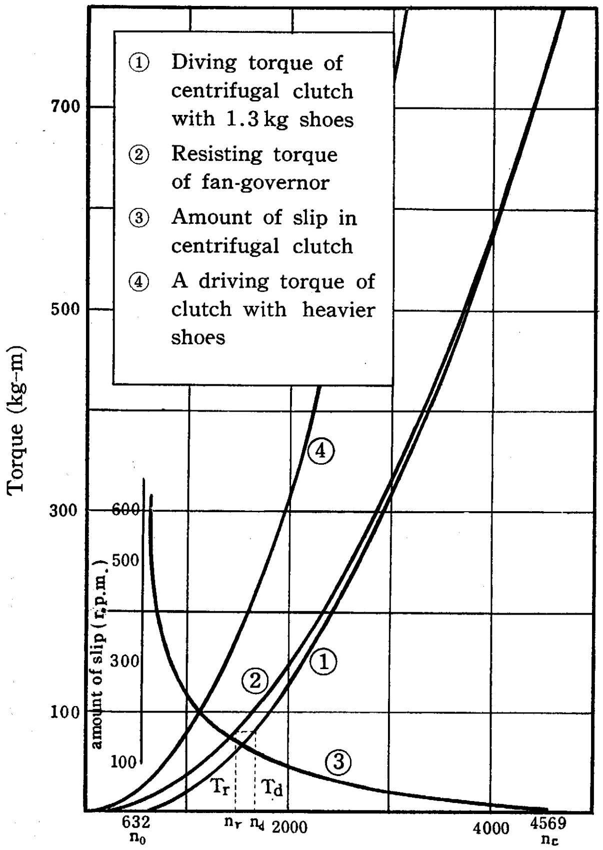
The sourse of overheating is, no doubt, due to the slipping friction between the outer lace of the clutch and the clutch-shoes. Therfore, the auther tried to make some theoretical studies on this problem.

III. Theoretical analysis of the centrifugal clutch

1) Driving torque of the centrifugal clutch

For the limb type centrifugal clutch, shown in Fig. 3, the driving torque of the clutch with 3 shoes is theoretically given in the following formula.

$$T_d=3\cdot R\cdot\frac{a\cdot z-s(d_1+d_2)}{b/\mu\pm c} \tag{1}$$



(r. p. m.)
Fig. 5

Where

T_d: driving torque of the clutch (kg–m)

z: centrifugal force of a shoe (kg)

$$=G/g\cdot R\cdot\frac{n^2\cdot\pi^2}{900}$$

R: radius of the clutch drum (m)

G: weight of a shoe (kg)

S: tensile force of a spring (kg)

μ: coefficient of friction

n: r.p.m. of the shoe axle

a, b, c: distance from each forces to the shoe pin, needable for the calculation of the moment of forces about shoe pin (m)

$d_1 d_2$: distance between shoe pin and spring (m)

So that the values of the driving torque corresponding to r.p.m. of the clutch shoe axle can be calculated by this formula. The numerical result of calculation for the clutch used for the experiment is shown by the curve ① in Fig. 5. We can recognize that the driving torque is shown in a parabolic curve with regard to (r.p.m.) of shoes and is nearly proportional to the weight of a shoe.

2) Resisting torque of the fan-governor.

The resisting torque caused by the rotation of the fan has been proved to be proportional to the (r.p.m./60)^2 and D^5, and given by the formula:

$$T_r=C_Q\cdot\rho\cdot(n_r/60)^2\cdot D^5 \tag{2}$$

where

T_r: resisting torque of fan-governor (kg–m)

n_r: r.p.m. of fan exle

D: diameter of fan (m)
C_Q: coefficient of torque
ρ: air density

The resisting torque corresponding to any given value of r.p.m. of the fan can be calulated by the formula (2). The mumerical result of the calculation for the fan used for the experiment is shown by the curve ② in Fig. 5. It is also evident that resistig torque is a parabolic curve with regard to n_r (r.p.m.) of the fan exle.

3) Slip of the centrifugal clutch.

When the resisting torque caused by the fan is larger than the driving torque of the clutch, there occurs slip in the centrifugal clutch. The amount of the slip could be derived as follows.

If the driving torque of the inner race at n_d (r.p.m.) of shoe exle is smaller than the resisting torque of outer race at the same n_d (r.p.m.), the drived axle (fan axle) keeps its rotation at the value of n_r (r.p.m.) corresponding to the resisting torque (T_r) which is equal to driving torque (T_d). Assuming that the difference between n_d and n_r is due to the slip of the clutch, then the amount of slip per minute (α) is given by

$$\alpha=n_d-n_r \tag{3}$$

if we put $T_d=T_r$ in equations (1) and (2),

$$3\left\{\frac{aGR^2n_d^2-900gRS(d_1+d_2)}{900g\,(b/\mu\pm c)}\right\}=C_Q\rho(n_r/60)^2D^5 \tag{4}$$

and

$$n_r=\left\{12\frac{aGR^2\pi^2n^2-900gRS(d_1+d_2)}{g(b/\mu\pm c)\,C_Q\rho D^5}\right\}^{1/2} \tag{5}$$

From (3) and (5), we can obtain

$$\alpha=n_d-\{An_d^2-B\}^{1/2} \tag{6}$$

$$A=\frac{12aG\pi^2R^2}{g(b/\mu\pm c)\,C_Q\rho D^5}$$

$$B=\frac{10800gSR(d_1+d_2)}{g(b/\mu\pm c)\,C_Q\rho D^5}$$

Equation (6) is the general formula for finding the amount of the slip of clutch in the case of $T_r>T_d$.

IV. Discussion on the slip of the centrifugal clutch

Fig. 5 showes the theoretical relations among T_r, T_d, and α. Observing the curves ① and ④, it is obvious that the driving capacity is insuficient because the both curves do not cross each other.

If the curves of T_d and T_r cross like the curve ① and ② at the point $n=n_c$, we can recognize that when the fan rotates at n (r.p.m.) larger than n_c (i.e. $n\geqq n_c$), the clutch acts perfectly without slip. When $n_c>n\geqq n_0$, it rotates with slip. When $n<n_0$, it does not rotate at all. The amout of the slip per minute is shown by the curve ③.

When the drivnig shoe axle rotate at n_d r.p.m. ($n_0 < n_d < n_c$), the resisting torque T_r can not be larger than the driving torque T_d. Therefore the drived axle (fan-axle) does not revolve at n_d but n_r.

Using the curve ③, it is able to know the amount of slip of clutch at every r.p.m. of driving shaft. In designing the fan-governor with a centrifugal clutch, we must consider to allow the slip area of clutch as small as possible.

V. Practical discussion for the improvement of the mechanism

1) Use of the heavier shoes in the clutch.

For the purpose of gaining the larger driving torque, it may be an advantage to increase the weight of shoes, because the driving torque is nearly proportional to the shoe weight. Using the 1.5 kg shoes instead of 1.3 kg shoes, the values of the driving torque in the inner race change from curve ① to curve ⑤, and becomes about 1.15 times larger than that of the clutch with 1.3 kg shoes. Therefore, the boundary of slip as shown in Fig. 6 is decreased from $632 \leqq n \leqq 4569$ to $590 \leqq n \leqq 1520$. Yet, the new boundary of slip is still too fast. Heavier shoes are required for the practical opereation. But too heavy shoes are also unpractical.

2) Adjustment of the rate of rotation between the fan and the clutch.

The centrifugal clutch is to be used under the condition of high rotation. It is not adequate to use it at low rotation less than 1,000 r.p.m., so that the fan-governor

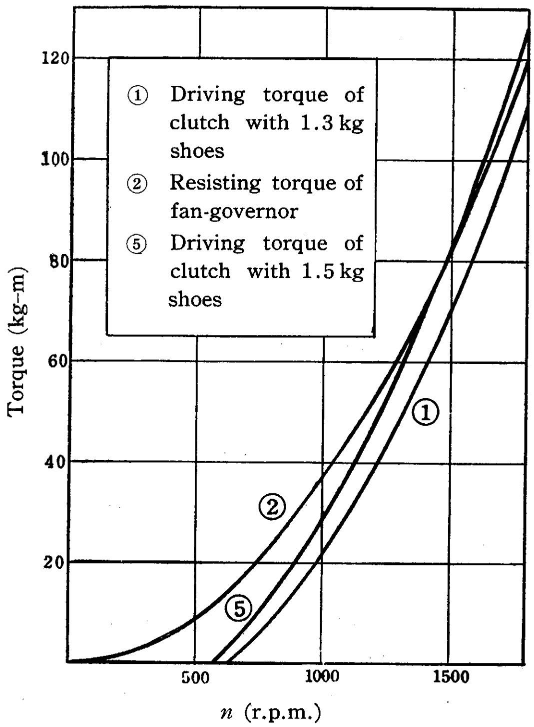


Fig. 6

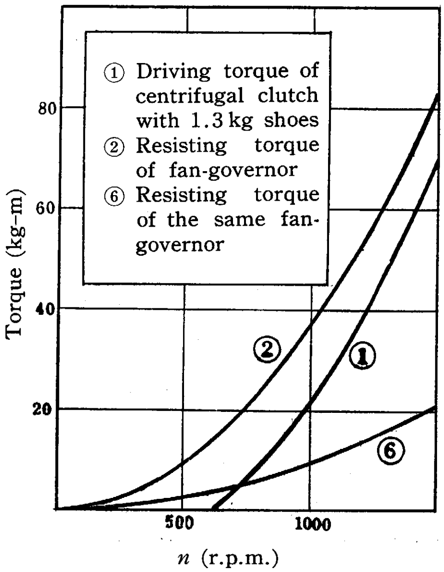


Fig. 7

should have the mechanism which makes the inner race of clutch rotate as fast as possible, and at the same time the rotation of the fan-axle should be kept at less than 1,000 r.p.m. By using such a mechanism, the centrifugal clutch and the fan could be used at the best condition, and the slip area of the clutch could be extremely decreased as shown in Fig. 7.

In order to obtain such a mechanism, following improvements are recomended

a) In the case of using the centrifugal clutch witch is available in the market, its inner race should be rotated as fast as possible by increasing the gear ratio between pulley II and III shown in Fig. 8, and the rotating speed should be kept at not less than 1,500 r.p.m. at least.

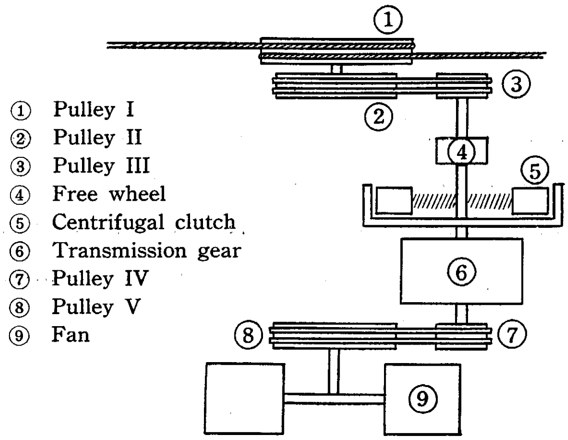


Fig. 8

b) The rotation of fan axle should be kept within 1,000 r.p.m. by decreasing its speed by means of pulley IV and V, because this value of rotation is advisable from the view point of operational safty.

3) Other improvements recomended.

a) It is recomended to set a free wheel between pulley III and clutch in order to keep the fan-governor free when the unloaded carriage goes back uphill.

b) It might be more effective to set a transmission gears between the pulley IV and clutch if it is wanted to enlarge the applicability of the fan-governor to sutisfy the various conditions of the logging operations.

Literature

1) Kato, S. and others: A practical experiment on governing effect of fan-governor set applied to the logging cableway. Transactions of the 63rd meeting of the Japanese Forestry Society, (1961)

2) Kato, S. and others: An experimental study on the fan-governor applied to a supplementary braking equipment of the logging cableway (I). Journ. Jap. For. Soc. 43, (1961)

3) Czitary, E.: Seilschwebebahnen, (1951)

集材機用風圧ガバナーに用いられた遠心クラッチのスリップに関する研究

南　方　　康

要　　旨

架空線集材用に製作された遠心クラッチ付風圧ガバナーの新機種について現地実験を行なつた結果その制御力はクラッチのトルク伝達能力に左右され，この伝達トルクとガバナーの抵抗トルクとの間の調整を計らなければ遠心クラッチはスリップを生じ，制御力の減退，クラッチの過熱等，種々問題を生ずることが判明した。本稿はこの種の遠心クラッチのスリップ領域並びに遠心クラッチを使用する風圧ガバナーの基本的な機構に関し理論的な考察を行なつたものである。