Use of Fan-governor for Safety and Efficiency in Cable Logging

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Contents

Introduction

Skyline cable logging have been quickly developed in Japan during past fifteen years. Recent statistics made by the Forestry Agency of the Ministry of Agriculture and Forestry shows that about twenty thousands cableways and ten thousands cable-yarders are employed for forest operations in the mountainous regions of this country. This fact, the tremendous number of the skyline cable installations, is one of the effective results of recent development of the road systems which considerably increased the possibilities of mechanized methods in forest exploitation.

Today, cableway and cable-crane transport of timbers from forests to truck-roads, mostly by downhill yarding, are very popular methods of hauling operation, especially in steep mountains[1]. These operations are performed by licensed engineers and well trained forest workers. By this general tendency, however, many loggers are exerting to try these operations under more and more difficult terrain conditions, with heavier loads and higher speed. But, usual machines and equipments of the conventional types are obviously insufficient to meet these new requirements. New problem has been arisen, how to maintain safety and efficiency of cable logging under such severe conditions.

As the results of observations and inspections of many existing installations and operations, as well as of analysis of accidents, it was found that the most important problem is lack of sufficient braking capacity of the machine. There observed severe overheat of brake-drum and rapid wear and slip of brake-lining in the case of ordinary brake system, such as band-brake or centrifugal brake. Damages ot other parts of machine and of wirerope are also caused by this overheat. Some improvements to avoid this defect of the traditional brake system were attempted in the past[2]. For instance, use of water-cooling system, forced air-cooling system, double brake system, etc. But, all these were practiced with little success.

The only way to solve the problem is to establish a new brake system, which should be of non-overheat, simple and light construction, easy operation and low cost. For this purpose, the author considered that application of *"fan-governor"* might be most reasonable, and his experimental studies started in 1955. Since then the research works were carried out by the staffs of the Institute of Forest Utilization, colaborated by the staffs of the Institute of Aerodynamics, Tokyo University. Also several machine-makers played their roles in the field of trial manufacture of the new machines. Some remarkable results were obtained by our experimental studies. Such results of various experiments were theoretically analysed. And finally we could develope a new brake system for cableways and cable-cranes. In this new system, the fan-governor ocupies the place of supplementary brake which is applicable for any kind of cable mashine. Today, some models of the new machines of this kind are available, and already successfully practised by progressive loggers. But, it is not so easy in each individual logging place to select a most adequate machine and to find out most suitable method and organization of operation. Therefore, sufficient knowledge about the new system is indispensable.

The author's aim of this article is to instruct forestengineers and logging operators by giving them fundamental knowledge about the fan-governor for the final purpose of maintaining perfect safety and highest efficiency in cable logging.

Chapter I. Merits of Fan-governor

A fan-governor is a rotary fan, which has more than two wings, and this fan is rotated by the pulling force of traction-cable of the cableway, or by that of operating cable (outhaul line or lifting line) of the cable-crane, providing that a transmission system is set between sheave-axle (or drum-axle) and fan-axle. When a loaded carriage is running downhill on the skyline by gravity, the traction-cable, or the operating cable, which is fastened to the carriage, is also running at the same speed. In the case of the lifting line in the Tyler system, rope speed corresponds to the falling down speed of the load hooked on the loading block. When the brake of the cable-sheave, or that of the cable-drum of the yarder, is released, the sheave, or the drum, must be rotated by the pulling force of the cable at a certain speed corresponding to the rope-speed. In order to control rope-speed, usually we have to operate the main brake attached to the sheave or the drum. If rotation of sheave- or drum-axle is transmited to fan-axle, and at the

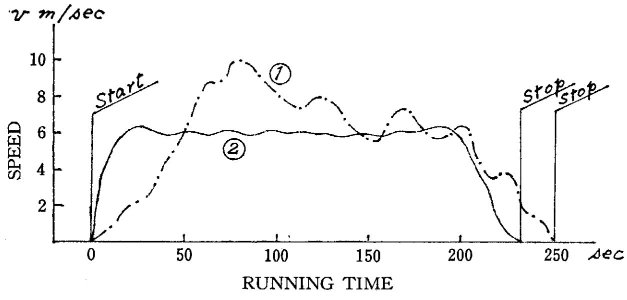
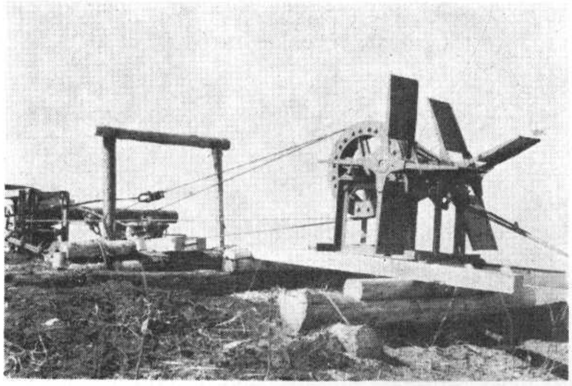


Fig. 1. Comparison of Running Speed of the Carriage by Downhill Gravity Yarding in the Tyler-system Skyline Logging.

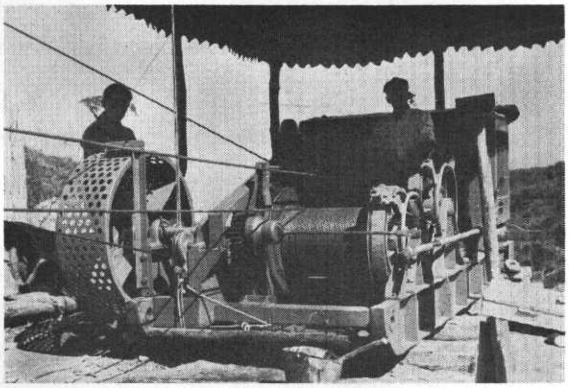
① Without fan-governor; Speed controlled by bandbrake only. (Carriage-load: 759 kg)

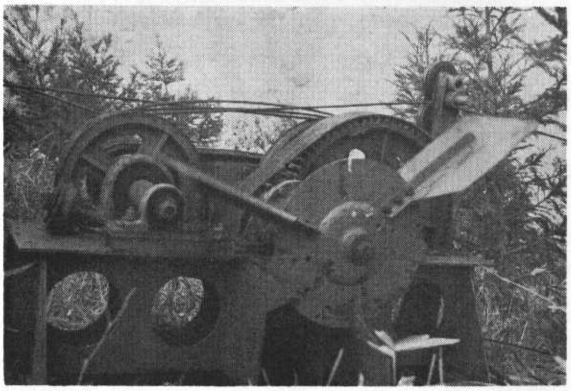
② With fan-governor; Speed controlled automatically by fan-governor. Band brake was operated only at the short time of starting and stopping. (Caıriage-load: 1076 kg)



① "Morito" fan-governor, installed for a logging cableway of medium size.

② "Nansei RK-62" fan-governor, attached to a "Nansei PK-62" cable yarder for heavier duty.





③ "Owase L-III" fan-governor, installed for a skyline crane.

same time, if any speed increasing mechanism is inserted between both axles, the fan may rotate at highspeed. In this case, wings of the fan are forced to face against a certain amount of air pressure. Actually, this resisting force of air is effectively utilized for braking force. Thus, rope-speed could be automatically controlled and braking capacity of machine could be considerably increased. When a fan-governor is most adequately designed, even it is possible to maintain an automationed operation except at the time of starting and stopping.

Fig. 1. shows a typical example of the result of comparative experiment with and without a fan-governor.

It is obvious in this figure that running speed is kept approximately constant in the case of ①, while it deviates irregularly in the case of ②, and time needed for one trip transportation, i.e. from start to complete stop, is much shorter in ① than in ②.

As a whole, we can state the merits of fan-governor:

a) Increasing safety.

b) Increasing timber hauling efficiency.

c) Increasing life of machine and wirerope.

d) Reducing hard labour and eliminating difficult technique of machine operators.

e) Decreasing hauling cost.

Some typical fan-governors in working conditions at logging sites are shown in the pictures.

Chapter II. Theoretical Principle of Fan-governor

As shown in Fig. 2., a cable sheave (a) is connected to a fan (b). Rotation of the sheave and the fan is caused by the pulling effort of the traction-cable. Assume that the sheave is rotating at a certain velocity corresponding to the rope-speed v (m/sec), and the driving force on the circumference of the sheave is F (kg), then the driving effort (kg. m/sec):

$$E=F\cdot v$$

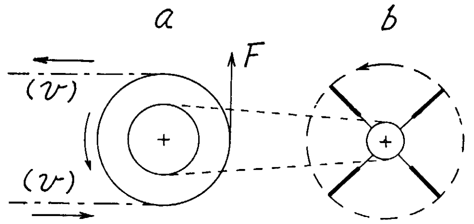


Fig. 2.

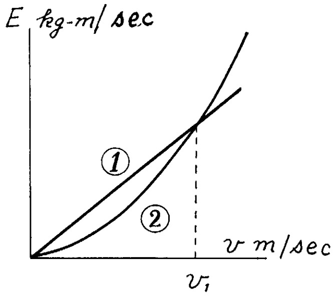


Fig. 3.

Therefore, E is proportional to v. This relation is shown by a straightline ① in Fig. 3.

On the other hand, the amount of air resistance against the wings of the fan is approximately proportional to the square of revolution of the fan-axle, and a certain braking effort E (kg. m/sec) is caused by this resistance. Since there is a fixed relation between rope-speed and revolution per unit time of the fan-axle according to the given transmission mechanism, E is approximately proportional to v^3. This relation is shown by a curve ② in Fig. 3. There is a breakeven point of ① and ② corresponding to a certain rope-speed v_1. If it is assumed that the tractive force of the traction-cable is constant, and the movement of the cable begins from $v=0$, the value of the driving effort ① is higher than the braking effort ② until the rope-speed reaches to $v=v_1$. The rope-speed can be no more accelerated, but kept constant at v_1, because there is no surplus driving power when the rope-speed reaches to v_1. In other words, the rope-speed has its upper limit when the sheave is restrained by a fan-governor. This means that the running speed of the carriage has also its upper limit, because the rope is fixed to the carriage. Actually, as soon as the carriage leaves the loading point, it accelerates very quickly and then it keeps its limited maximum speed. Thus the carriage can travel along the skyline with approximately constant speed until it is braked by the traction-cable at the unloading point.

Chapter III. Different Types of Fan-governors

There are several types of fan-governors which are available and already practically used for cable logging. From the view point of mechanism of the machine, we can classify them in following four types (Fig 4. A, B, C, D):

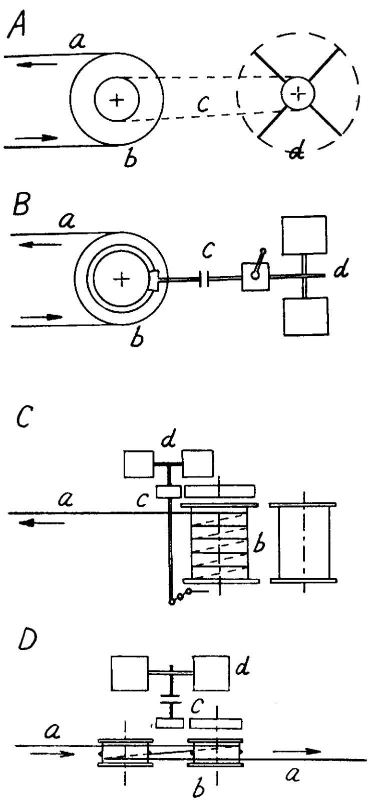


Fig. 4.

A, B for cableway.
- a : Traction-cable
- b : Cable sheave
- c : Transmission mechanism
- d : Fan

C for crane. Fan is attached to the cable yarder.
D for crane. Independent fan-governor unit.
- a : Haulout cable or lifting cable.
- b : Winchdrum or independent sheave(s)
- c : Transmission mechanism
- d : Fan

Chapter IV. Method of Calculation

1. General Procedure.

Principal function of fan-governor is to keep automatically the most adequate speed of running carriage(s) when the main brake of the machine is released. Therefore, to find out this probable running speed of carriage(s), which could be assumed to be equal to rope-speed, is the aim of calculation. For this purpose, the general procedure of calculation is to solve the following equation, which represents the general condition for existence of balanced relation between driving power and braking power. If we denote,

Q_1: Driving torque (kg.m) of the sheave-axle.

Q: Resisting torque (kg.m) of the fan-axle.

a: Speed ratio of rotation (fan-axle: sheave-axle), a certain definite value for the given machine.

η: Mechanical efficiency of transmission system, which may be estimated at the value of 0.75~0.90 according to the applied mechanism of transmission system.

$$\eta\cdot Q_1=\cdot aQ \quad \cdots\cdots (1)$$

where $\eta=0.75\sim0.90$

To solve this equation, at first Q_1 and Q should be investigated independently.

2. Calculation of driving torque Q_1[1,2,4]

(1) Cableway with endless traction-cable—Driving Torque of Sheave-axle.

Fig. 5. shows the main sheave of a cableway. The sheave is driven by the traction-cable and rotating counter-clockwise when the main brake of the sheave is released.

In this case,

T_1: Tractive force of traction-cable, which is pulled by loaded carriage(s) (kg).

T_2: Tractive force of traction-cable, which is pulled by the empty carriage(s) (kg).

G: Tensioning force applied at sheave-axle (kg).

R: Friction of sheave axle (kg).

μ: Frictional coefficient of axle bearing.

r_1: Radius of sheave (m).

r_2: Radius of sheave-axle (m).

Then, the driving force F (kg) is given by

$$F=(T_1-T_2)-\frac{r_2}{r_1}R \quad \cdots\cdots (2)$$

where, $R=\mu G$

The second term in eq. (2) is small enough and practically negligible when it is compared with the first term. Therefore, approximately,

$$F=T_1-T_2 \quad \cdots\cdots (3)$$

Hence, the driving torque:

$$Q_1=r_1\cdot F=r_2(T_1-T_2) \quad \cdots\cdots (4)$$

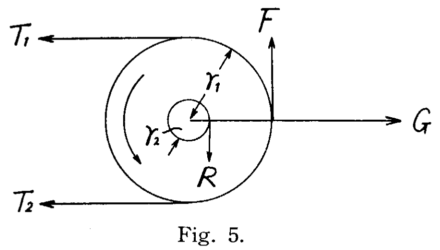


Fig. 5.

In this equation, the tractive force in kg is given by

$$T_1-T_2=(T-T_R)-(T'+T_R') \quad \cdots\cdots (5)$$

where,

T: Rope-pull due to loaded carriage(s) (kg).

T': Rope-pull due to empty carriage(s) (kg).

T_R: Running resistance of loaded carriage(s) (kg), including that of running rope on the coresponding side.

T_R': Running resistance of empty carriage(s) (kg), including that of running rope on the coresponding side.

It is evident that the predominant factor in eq. (5) is T. Approximate average values for T and T' could be calculated by the following formulae.

(Fig 6).

$$\left.\begin{aligned} T&=P(i_1\cdot\sin\alpha+i_2\cdot\sin\alpha_2+\cdots)=P\sum i\cdot\sin\alpha \\ T'&=P'(i_1\cdot\sin\alpha_1+i_2\cdot\sin\alpha_2+\cdots)=P'\sum i\cdot\sin\alpha \end{aligned}\right\} \quad\cdots\cdots\cdots\cdots(6)$$

in which,

P: Weight of each loaded carriage (kg).

P': Weight of each empty carriage (kg).

α_1, $\alpha_2\cdots$: Inclination in degree of the span, on which carriage(s) is (are) situated.

i_1, $i_2\cdots$: Number of carriage(s) on the corresponding span.

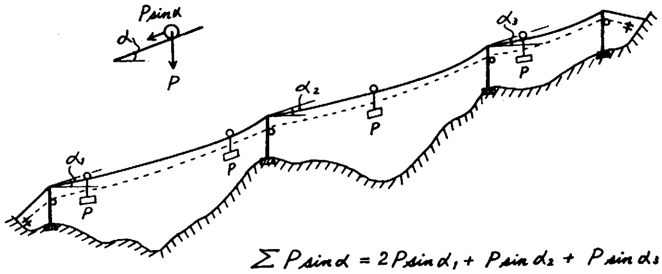


Fig. 6.

For the calculation of more accurate values of T and T_1, the author recomends the use of the "*Method of Influenceline*"[2,4].

T_R and T_R' are the resistances caused by all moving parts of the cable system, which could be calculated by the formulae,

$$\left.\begin{aligned} T_R&=f(\sum P+W) \\ T_R'&=f(\sum P'+W) \end{aligned}\right\} \quad\cdots\cdots\cdots\cdots(7)$$

where,

f: Coefficent of running resistance.

$\sum P$: Total weight of the loaded carriage(s) (kg).

$\sum P'$: Total weight of the empty carriage(s) (kg).

W: Weight of the traction-cable on each side (kg).

The value of the coefficient f in the above formulae is estimated at 0.04~0.05, in extreme cases $f=0.02$ to $f=0.08$ or more.

(2) Cable-crane—Driving Torque of the Drum-axle of Cableyarder.

As shown in Fig. 7., consider that the main brake of the drum is released, and if we denote

T_1: Pull of the operating line (kg).

G: Weight of the drum (kg), including weight of the wirerope reeled in.

R: Friction of the drum axle (kg).

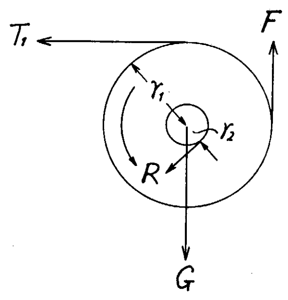


Fig. 7.

μ: Frictional coefficient of the axle bearing.

r_1: Radius of drum, which is variable according to the amount of reeled—in wire-rope (m).

r_2: Radius of drum axle (m).

F: Driving force at the external surface of the drum (kg).

then,

$$F=T_1-\frac{r_2}{r_1}R \quad \cdots\cdots (8)$$

where $R=\mu\cdot\sqrt{T_1^2+G^2}$

$\frac{r_2}{r_1}R$ in eq. (8) is practically negligible when compared with T_1.

Hence,

$$F=T_1 \quad \cdots\cdots (9)$$

Therefore, the driving torque:

$$Q=r_1T_1 \quad \cdots\cdots (10)$$

And,

$$T_1=T-(f\cdot P+f'W') \quad \cdots\cdots (11)$$

in which

T: Pulling force of the operating line due to the gravity component of the loaded carriage (kg).

P: Weight of the loaded carriage (kg).

f: Coefficient for the running resistance of the carriage. ($f=0.03\sim0.05$)

W': Weight of the operating cable between the end block and the drum (kg).

f': Coefficient for the moving resistance of the cable ($=0.05\sim0.1$)

The value of T in eq. (11) is variable according to the situation of the carriage. When the carriage, running on the skyline, is at the middle of the span, T takes its average value, while value of T is maximum or minimum when the carriage is at the upper or lower end of the span respectively. These, different values of T are calculated by the following formulae (Fig. 8).

(a) Carriage is at the upper end.

$$T_{max}=P\sin\varphi_{max}=P\sin\{\tan^{-1}[\tan\alpha+4s(1+2n)]\} \quad \cdots\cdots (12a)$$

(b) Carriage is at the middle point of the span.

$$T_{av.}=P\sin\varphi=P\sin\alpha \quad \cdots\cdots (12b)$$

(c) Carriage is at the lower end.

$$T_{min}=P\sin_{max}=P\sin\{\tan^{-1}[\tan\alpha-4s(1+2n)]\} \quad \cdots\cdots (12c)$$

(d) Carriage is at any point.

$$T=P\sin\varphi=P\sin\left\{\tan^{-1}\left[\tan\alpha-4s\,(1+2n)\,(1-2k)\,\frac{1+6\,(n+n^2)\,(k-k^2)}{[1+12(n+n^2)\,(k-k^2)]^{2/3}}\right]\right\}\quad\cdots\cdots(12d)$$

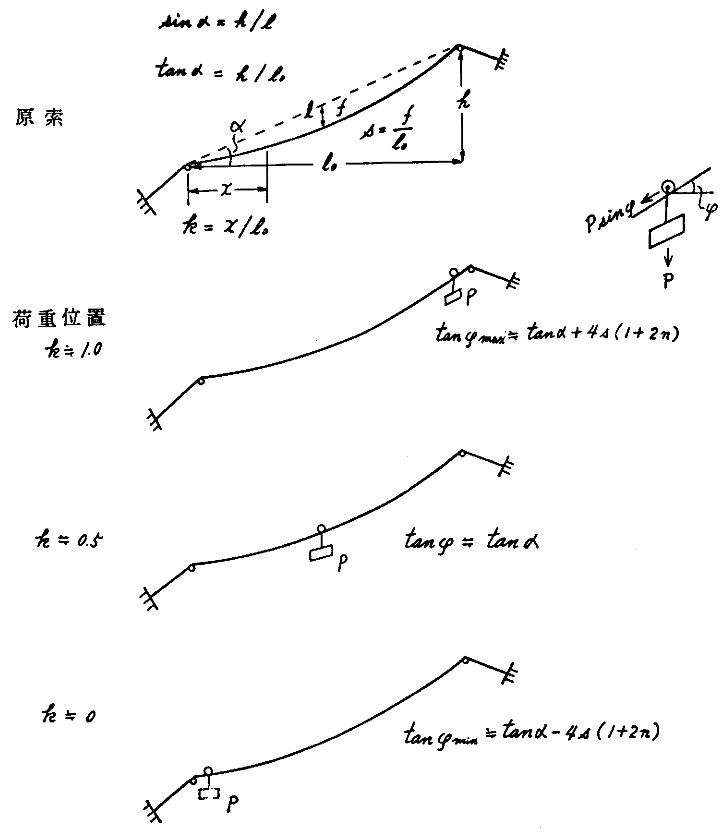


Fig. 8.

In these formulae (See Fig. 8),

φ: Inclination in degree of the locus curve of the running carriage (°).

α: Inclination in degree of the span (°).

s: Central sag-span ratio of the skyline cable.

n: Load ratio $P/W=\dfrac{\text{Weight of carriage load}}{\text{Weight of skyline cable}}$

k: Distance indicator for the situation of the carriage (0~1.0).

(3) Cable-crane—Driving Torque of the Sheave-axle of the Independent Governor unit. As shown in Fig. 9.,

T_1: Pull of the operating line (reel-out side) (kg).

T_2: Pull of the operating line (reel-in side) (kg).

r_1: Radius of the sheave (m).

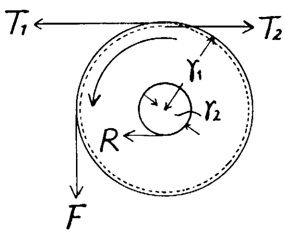


Fig. 9.

then,

$$F = T_1 - T_2 \qquad \cdots\cdots (13)$$

Hence, the driving torque

$$Q_1 = r_1 \cdot F = r_2 (T_1 - T_2) \qquad \cdots\cdots (13)$$

In this formula

$$T_1 = T - (f \cdot P + f' \cdot W') \qquad \cdots\cdots (15)$$

$$T_2 = f' \cdot W'' + f'' \cdot G \qquad \cdots\cdots (16)$$

where,

T: Pulling force of the operating line due to the gravity component of the loaded carriage (kg).

P: Weight of the loaded carriage (kg).

f: Coefficient of running resistance of the carriage ($f = 0.03 \sim 0.05$).

W': Weight of the operating cable between the endblock and the sheave of the governor unit (kg).

W'': Weight of the operating cable between the sheave of the governor and the drum of the yarder (kg).

f': Coefficient for the moving resistance of the operating cable ($f' = 0.05 \sim 0.1$).

G: Weight of the drum of the yarder (kg), including the weight of the cable reeled in the drum.

f'': Coefficiant for the frictional resistance of the drum-axle ($f'' = 0.01 \sim 0.02$).

3. Calculation of Resisting Torque (Q)[5~7]

(1) Speed Ratio of Rotation (Sheave-axle: Fan-axle) (a)

The value of speed ratio (a) depends upon the mechanism of the transmission system. Revolution of the fan-axle is limited to 500~1500 r.p.m. from the view points of mechanical corstruction and safety. Therefore, the mechanism is usually designed as to give the value of "a" at 5~10. This means the final speed ratio, and this value is attributed to the individual model of the governor.

(2) Resisting Torque of Fan (Q).

The fan is rotating against strong resistance of air. And resisting torque of the fan-axle should be calculated by the empirical formula:

$$Q = C \cdot \rho \cdot n^2 \cdot D^5 \qquad \cdots\cdots (17)$$

where, Q: Resisting torque (kg.m).

C: Torque-coefficient, determined by the experiments.

ρ: Density of air.

n: Fan speed (r.p.s. = 1/60 r.p.m.).

D: Diameter of fan (m).

(3) Density of air (ρ)

Although the value of air density varies according to the atmospheric pressure or air temperature, cable logging is actually operated under moderate conditions, not under extreme conditions such as more than 2000 m above sea level. Therefore, it is sufficient for our purpose to use the approximate mean value in m.kg.sec. unit;

$$\rho=0.12 \qquad \cdots\cdots (18)$$

(4) Relation between Linear Ropespeed and Revolution per Second of Fan-axle.

If it is assumed that there occurs no slip of operating rope at any place of the cable system[8~10], the rotation of the fan is completely depends upon the movement of the operating rope.

Then,

$$\left.\begin{aligned} n&=\frac{a}{2\pi r_1}\cdot v \\ \text{or}\quad v&=\frac{2\pi r_1}{a}\cdot n \end{aligned}\right\} \qquad \cdots\cdots (19)$$

where,

a: Speed ratio.
n: Revolution of fan (r.p.s.).
v: Linear rope speed (m/sec).
r_1: Radius of sheave or drum (m).
π: 3.14

(5) Diameter (D) and Structure of the Fan

It is evident from the formula (17) that Q is proportional to D^5. Very high value of the resisting torque can be obtained when relatively big fan is applied. But use of too big fan may cause the disadvantage of undesirable vibration and fast wearing of the machine parts. Therefore, usually, diameter of the fan is selected at $D=0.5\sim1.5$ m.

Structure of the fan must be strong enough against air pressure and centrifugal force. For this purpose, disc-type or arm-type fan is suitable. Fig. 10 (a) shows the disc-type, by which fan wings are fixed on a circular disc made of rather thin steel or juralmin plate, while Fig. 10 (b) shows the arm-type, by which the fan-wings are fixed to the steel arms. Also a combined disc- and arm-type fan is available.

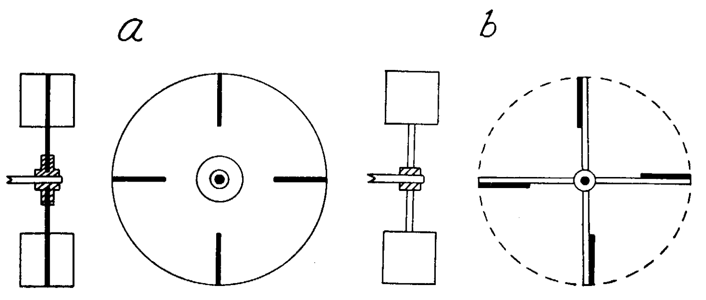


Fig. 10.

Wing plate should be very light and resistible, and usually it is made of thin steel or juralmin plate. And also it is convenient to let the wing plate convertible. Fig. 11. shows some examples of the convertible wing.

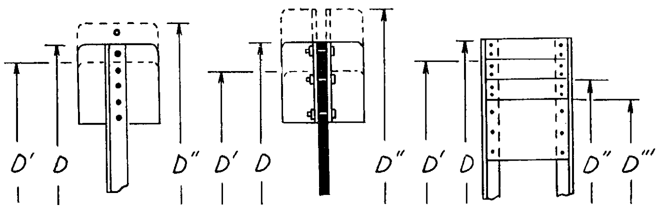


Fig. 11.

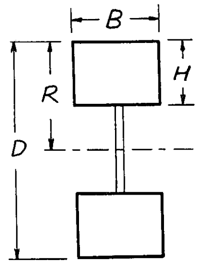


Fig. 12.

(6) Torque-coefficient (C)

The value of the torque-coefficient C depends upon size and form of the fan-plate as well as upon type of the fan structure. From the results of our accurate laboratory experiments and field measurement, we could find out the facts;

a) The value of C is higher in the case of plane wings than in the case of curved wings, concave or convex.

b) The value of C is approximately 10% higher in the case of disc-type than in the case of arm-type.

c) The value of C varies considerably according to the form of the wing plates (rectangular plates). As to the form of the wing,

$R=D/2$: Radius of the fan.
B: Width of a wing plate.
H: Height of the wing plate.

are shown in Fig. 12. High values of C were obtained when $H/R=0.6\sim0.7$. The relation between C and B/R were not so evident. But from the view point of practical application, desirable proportion could be represented by

$$H/R=0.5\sim0.8 \text{ and } B/R=0.4\sim0.8$$

From this relation, form of the wing plate is reasonable when

$$(H:B)=(1:0.5)\sim(1:2)$$

d) Number of wings

The value of C varies according to the number of wing-plates of the fan, and it is not always proportional to the number of wing-plates. If 4-wing fan is considered to be the standard (C=100%), the percentage of the values of C in 2-, 3-, 6- and 8-wings fans can be given by the figures shown in the Table 1.

Table 1. Relation between Torque-coefficient and Number of Wing-plates

Number of wing-plates	Value of C (%)
2	50
3	75
4	100
6	120
8	150
10	170

e) Effects of side wall and safety cover

There observed almost no effect of any side wall, which may shut out the sidewise flow of air, even when the wall was set adjacent to the fan. Therefore,

Table 2. Values of Torque-coefficient for Standard 4-wing Arm-type Fan

Relative dimension of fan			Torque-coefficient C_4
H/R	B/R	B/H	
0.5	0.4	0.80	0.68
	0.5	1.00	0.84
	0.6	1.20	0.99
	0.7	1.40	1.15
	0.8	1.60	1.30
0.6	0.4	0.67	0.72
	0.5	0.83	0.86
	0.6	1.00	1.05
	0.7	1.17	1.24
	0.8	1.33	1.38
0.7	0.4	0.57	0.72
	0.5	0.71	0.88
	0.6	0.86	1.03
	0.7	1.00	1.20
	0.8	1.14	1.35
0.8	0.4	0.50	0.67
	0.5	0.85	0.85
	0.6	0.94	0.94
	0.7	1.05	1.05
	0.8	1.00	1.10

Supplementary table for calculation of C for other types of fans

Types of fan	Arm-type	Disc-type
For 2-wing	$C_2=C_4\times0.5$	$C_2'=C_2\times1.1$
〃 3-wing	$C_3=C_4\times0.75$	$C_3'=C_3\times1.1$
〃 6-wing	$C_6=C_4\times1.2$	$C_6'=C_6\times1.1$
〃 8-wing	$C_8=C_4\times1.5$	$C_8'=C_8\times1.1$
〃 16-wing	$C_{16}=C_4\times1.75$	$C_{16}'=C_{16}\times1.1$

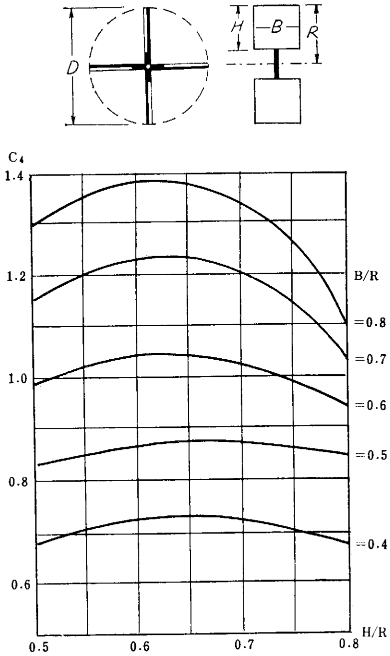


Fig. 13. Torque-coefficient C_4 for standard 4-wing arm-type fan.

no correction is required even if the fan is set at the side of a yarder or drum.

When the fan was covered by a plate, which was set just adjacent to the fan wings, we could find the tendency of increasing the value of C, in extreme case about 50%. But, the safety cover is usually set some reasonable distance apart from the wings. So that, the effects of the safety cover can be also neglected from the view point of practical calculation.

From these results of our experiments, the values of C are compiled on the Table 2. and shown in Fig. 13, in which C_4 means the values of C for standard 4-wing arm-type fan. The supplementary table could be utilized for calculation of the values of torque-coefficient corresponding to the other types of fans.

4. Calculation of Rope-speed (v)

The rope-speed, which is equal to the speed of the running carriages, is calculated by following procedures. The basic condition is already given by the formula (1);

$$\eta \cdot Q_1 = a \cdot Q \qquad \cdots\cdots (20)$$

Substituting formula (17) in this, we have,

$$\eta \cdot Q_1 = a \cdot C \cdot \rho \cdot n^2 \cdot D^5 \qquad \cdots\cdots (21)$$

Hence,

$$n = \sqrt{\frac{\eta \cdot Q_1}{a \cdot C \cdot \rho \cdot D^5}} \qquad \cdots\cdots (22)$$

Substituting (19) for n of eq. (22), we obtain the rope speed controlled by the fan-governor,

$$v = \frac{2\pi r_1}{a} \sqrt{\frac{\eta \cdot Q_1}{a \cdot C \cdot \rho \cdot D^5}} \qquad \cdots\cdots (23)$$

in which,

π: 3.14.

r_1: Radius of the sheave (drum) (m).

a: Speed ratio (Fan-axle: Sheave-axle).

η: Mechanical efficiency of the transmission system$=0.75\sim0.90$.

Q_1: Driving torque of the sheave- (drum-) axle (kg.m).

C: Torque-coefficient.

ρ: Density of air$=0.12$.

D: Diameter of the fan (m).

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Appendix: Specifications of Available Fan-governors

Maker (Sales agent)	Specification			
Nansei Kosakusho Co. (Nansei Kikai Hanbai Co.)	Type	RK-62	FG-75	BV-305
	Application	Attached to PK-62 cable yarder	Independent unit for skyline cranes	Independent unit for cableways
	Weight	3,000 kg	250 kg	500 kg
	Diameter of main sheave (drum)	300 mm	500 mm	700 mm
	Final ratio of speed increase	5.2 : 1	6.6 : 1	7.3 : 1
	Transmission mechanism	Drum—Gear—Sliding gear clutch—Fan	Sheave—Gear—Cone clutch—Fan	Sheave—Gear—Cone clutch—Fan
	Diameter of fan	800 mm	750 mm	880 mm
	Number of fan-plates	8	8	8
	Size of fan-plate	250×290×3 mm	250×250×3 mm	230×240×3 mm
Owase Kosakusho Co. (Hokutan Norin Co.)	Type	L-I	L-III	
	Application	Independent unit for skyline cranes		
	Weight	450 kg	470 kg	
	Diameter of main sheave	500 mm	500 mm	
	Final ratio of speed increase	5 : 1	5 : 1	
	Transmission mechanism	Sheave—Gear—One-way Clutch—Fan		
	Number of fan-plates	2, 3, 4, 6	2, 3, 4, 6	
	Size of fan-plate	300×300 mm	300×300(400) mm	
Morito Kikai Seisakusho Co.	Type	Medium	Large	
	Application	Independent unit for cableways		
	Weight	160 kg	200 kg	
	Diameter of main sheave	800 mm	1,000 mm	
	Final ratio of speed increase	6.7 : 1	8.5 : 1	
	Transmission mechanism	Sheave—Chain—Fan		
	Diameter of fan	600 mm (min.) 1,000 mm (max.)	700 mm (min.) 1,400 mm (max.)	
	Number of fan-plates	4×2	4×2	
	Size of fan-plate	250×150 mm 250×350 mm	300×150 mm 300×500 mm	

Maker (Sales agent)	Specification			
Maruyama Tekkosho Co.	Type	M L-15		
	Application	Attached to brake devises of cable-ways		
	Weight	1,500 kg		
	Diameter of main sheave	1,000 mm		
	Final ratio of speed increase	6 : 1 12 : 1		
	Transmission mechanism	Sheave—Bevel gear—Shaft—Gear box—Clutch—Fan		
	Diameter of fan	1,000 mm		
	Number of fan-plates	4		
	Size of fanplate	300×350 mm		

架線集運材の安全と能率向上に役立つ風圧ガバナーの使用

加　藤　誠　平

概　要

架線集運材作業は本邦山岳林における機械化作業体系の中で重要な地位を占めているが，その技術の普及発達に伴い，索道や集材機が急勾配，重荷重，高速運転という苛酷な条件下において使用されることが多くなり，そのために機械の制動容量が不足して各種の障害が発生するという新しい問題が提起された。これを克服するために著者は風圧ガバナーの利用に着想し．1955年に基礎的研究に着手，その後多くの共同研究者の協力と機械製作者及び使用者側の助力を得て，実用化試験を続行した結果，作業の安全と能率向上に十分役立つような風圧ガバナーが開発され，市販されるに至り，漸く普及の段階に入った。本文はこの種の機械の特徴と理論的な原理を明らかにすると共に，実用上当面する機種の選定に際し，あるいは与えられた機械を操作するときに，予め的確に知っておかなければならない諸量，すなわち制動力，制御速度などの算定法を解説したものである。