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AN ELEVATOR HOISTING APPARATUS.

The present invention relates to an elevator hoisting apparatus, and more particularly to an elevator hoisting apparatus utilizing a parallel shaft type gear arrangement exhibiting superior low-noise and low-
5 vibration characteristics.

In the field of elevator hoisting apparatus for driving an elevator car or cars, it is known that a parallel shaft type gear arrangement is used as a reduction gear assembly, for example, as disclosed in Japanese
10 Patent Unexamined Publication No. 86586/1984.

Figs. 5 and 6 schematically show the structure of the prior-art elevator hoisting apparatus set forth in the aforesaid publication.

The illustrated elevator hoisting apparatus
15 includes a hoistway 1 for an elevator or elevators, a machine room 2 provided at the top of the hoistway 1, a floor 2A of the machine room 2, support frames 3A and 3B secured to the floor 2A, a machine platform 4 fixed to the respective of tops of the support frames 3A and
20 3B, a base 5 supported on the machine platform 4 with a rubber vibration insulator 6 interposed therebetween,

- 1 an electric motor 7 mounted on the base 5, a reduction
gear assembly 8 mounted on the same base 5, an input
shaft 8A extending from the shaft of the electric motor
7 and a first gear 8B constituted by a helical gear
5 fixed to the input shaft 8A, the first gear 8A being
engaged with a second gear 8D fixed to an output shaft 8F.

The above elevator hoisting apparatus further
includes a driving sheave 9 fixed to the output shaft
8F, a deflector-pulley support beam 10 fixed to the
10 machine platform 4, a shaft 11 supported at its opposite
ends by the deflector-pulley support beam 10, a deflector
pulley 12 rotatably fitted onto the shaft 11 and a main
rope 13 reeved over the sheave 9 and the deflector
pulley 12, one end of the main rope 13 being connected
15 to an elevator car or cars 17 with the other end being
connected to a counterweight 18. Although not particularly
shown, a brake system serving to bring the or each car
to a stop is provided at the input shaft 8A extending from
the electric motor 7 into the reduction gear 8.

20 In the above-described elevator hoisting
apparatus of the prior art, however, the brake device
for bringing the or each car to a stop is provided at
the shaft of the electric motor 7. Therefore, in the case
of a high-speed elevator system, impact produced upon
25 emergency stop greatly acts on the gear assembly together
with the inertial force of the sheave 9. In consequence,
an excessive load is applied to the respective gears and
this lowers the safety of the entire elevator system.

1 Also, the sheave 9 and the second gear 8D are
fixed to the output shaft 8F with the opposite ends
thereof being supported by bearings 8G. However, since
a long bearing interval exists between the opposite
5 bearings, the thus-produced deflection of the output
shaft 8F affects the state of engagement of the gear
assembly, and thus there is a tendency for the level of
noise to be increased. In order to prevent the increase
in the level of noise, it has heretofore been necessary
10 to increase the substantial diameters of the respective
gears to some extent.

In addition, in order to prevent transmission
of noise to the periphery of the machine chamber 2 and/or
the elevator car, the reduction gear assembly 8 is
15 required to, in particular, have low-noise character-
istics as compared with a reduction gear assembly of a
general type. For this reason, a worm gear type of
reduction gear assembly has heretofore been chiefly
employed. In addition, a helical gear type of reduction
20 gear assembly having a high transmission efficiency has
recently started to be used for the sake of saving of
energy. However, while such helical gear type reduction
gear assembly affords a high transmission efficiency,
the state of engagement of helical gears utterly differs
25 from that of the worm gear type. Therefore, as mentioned
in Japanese Utility Model Unexamined Publication No.
151371/1982, if the level of noise is to be reduced, it
is necessary to prepare a gear with a high precision

1 capable of satisfying the requirements of the first
or higher class specified under the Japanese Industrial
Standard and this leads to the problem that the production
cost is increased. To overcome the problem, the aforesaid
5 publication further states that the helix angle of
a helical gear is selected within the angular range of
20 to 30 degrees. This setting of helix angle acts to
improve a so-called overlapping contact ratio of gears,
thereby reducing the degree of fluctuations in the
10 respective spring constants of the mating gears. In
addition, such setting of helix angle acts to suitably
adjust the errors of the tooth profiles, pitches and
the like of individual teeth by engaging a plurality of
teeth.

15 According to the above-described arrangement,
the selection of a helix angle within the range of 20
to 30 degrees serves to reduce the acceleration of the
circumferential vibration of the driving sheave 9. How-
ever, this circumferential acceleration results chiefly
20 from vibrations as in the direction of rotation of a
rotary shaft line connecting the electric motor 7 and
the sheave 9. Accordingly, in order to further reduce
the level of vibration noise, a high-precision helical
gear must finally be used.

25 It is therefore an object of the present invention

to provide an elevator hoisting apparatus which is reduced in both size and weight with the safety and reliability of elevator systems being ensured and in addition which enables a highly efficient operation with a low level of noise.

5 According to a first aspect of the present invention there is provided an elevator hoisting apparatus having a reduction gear assembly connected to a driving electric motor and a sheave, a main rope being reeled around said sheave for supporting a car in a suspended manner so
10 that said car may be moved upwardly and downwardly, said reduction gear assembly including therein a plurality of helical gears whose diameters are of different sizes from one another and which are mounted on respective parallel shafts, at least small and large gears of said helical gears
15 being connected respectively to said input shaft and said output shaft in said reduction gear assembly, said driving electric motor being located on the side of said input shaft while said sheave and a brake system are located on the side of said output shaft, one bearing of said sheave being
20 integral with a gearbox of said reduction gear assembly, and said helical gears having a pressure angle of 15 to 18 degrees.

 According to a second aspect of the present invention there is provided an elevator hoisting apparatus
25 having a reduction gear assembly connected to a driving electric motor and a sheave, a main rope being reeled around said sheave for supporting a car in a suspended manner so that said car may be moved upwardly and downwardly, said

reduction gear assembly including therein a plurality of helical gears whose diameters are of different sizes from one other and which are mounted on respective parallel shafts, at least small and large gears of said helical gears being connected respectively to said input shaft and said output shaft in said reduction gear assembly, said driving electric motor being located on the side of said input shaft while said sheave and a brake system being located on the side of said output shaft, one bearing of said sheave being integral with a gearbox of said reduction gear assembly, and said helical gears having a dedendum of 1.1 to 1.3 module which is the value obtained by subtracting an addendum and a root from the depth of the teeth.

These arrangements enable close layout of the electric motor, the reduction gear assembly and the brake system, thereby achieving a decrease in the size of elevator hoisting systems. Moreover, the brake system is provided at the output shaft of the reduction gear assembly. Therefore, the impact generated by the emergency stop of, for example, a high-speed elevator never acts directly on the respective gears, so that it is possible to improve the safety and reliability of the overall elevator system. Furthermore, the fluctuating load provided by the errors of individual mating gears can be suppressed by reducing the spring constant of their teeth. Also, since the height (or depth) of each tooth is increased, even when there are variations in the number of tooth pairs which are engaged at the same time, abrupt variations in the spring constant of the teeth

can be substantially prevented.

The present invention will now be described in greater detail by way of example with reference to the accompanying drawings, wherein:-

5 Fig. 1 is a front elevational view of the preferred embodiment of the present invention;

 Fig. 2 is a top plan view, partially in cross-section, of the embodiment shown in Fig. 1;

 Fig. 3 is a top plan view of the state of

1 installation of the embodiment shown in Fig. 1;

Fig. 4 is a front elevational view of the embodiment shown in Fig. 3;

Figs. 5 and 6 schematically illustrate the construction of an example of an elevator hoisting apparatus of the prior art, in which;

Fig. 5 is a longitudinal section of the conventional example including a machine chamber;

Fig. 6 is a top plan view, partially in cross-section, of the conventional example shown in Fig. 5;

Fig. 7 is a schematic, cross-sectional view, taken in the direction perpendicular to the rotating axis of a pair of mating gears constituting the preferred embodiment of the present invention;

Fig. 8 is an enlarged view of a portion of the mating gears shown in Fig. 7 but showing a portion of teeth in engagement;

Fig. 9 is a chart showing the relationship between pressure angles and the spring constant of the teeth;

Fig. 10 is a chart showing variations in the spring constant of the teeth in contact as in a case where the engagement of the gears proceeds;

Fig. 11 is a chart showing variations in the spring constant of the teeth and the tooth-profile constant of the same in a case where the pressure angle varies;

Fig. 12 is a chart showing the relationship

between the noise and the load factor of the mating gears.

Figs. 1 and 2 are respectively one preferred embodiment of an elevator hoisting apparatus in accordance with the present invention, Fig. 1 being a front elevational view of the embodiment with Fig. 2 being a top plan view, partially in cross-section of the same. In Figs. 1 and 2, like reference characters are used for the sake of simplicity to denote like or corresponding elements which constitute each of the components shown in the above-described example of the prior art.

The illustrated embodiment includes the electric motor 7 and the reduction gear assembly 8. The input shaft 8A of the reduction gear assembly 8 is connected to the shaft of the electric motor 7 by couplings 7A and 7B, and the first gear (or small gear) 8B constituted by a helical gear secured to or integral with the input shaft 8A, the input shaft 8A being supported at opposite end portions thereof by the bearings provided at the stationary portion of the reduction gear assembly 8. The second gear (large gear) 8D, the driving sheave 9 and a brake drum 9A are fixed to the output shaft 8F, the second gear 8D being engaged with the first gear 8B and the brake drum 9A being integral with the driving sheave 9. The sheave 9 and the electric motor 7 are disposed on opposite sides of

1 the central reduction gear assembly 8, one end of the
output shaft 8F being supported by a bearing 15 with
the other end thereof being rotatably supported by the
sheave 9 positioned in the vicinity of the reduction
5 gear assembly 8. The second gear 8D is supported in an
overhanging manner by the portion of the output shaft
8F which is near the reduction gear assembly 8, the
second gear 8D being engaged with the first gear 8B.
The electric motor 7, the reduction gear assembly 8 and
10 the bearing 15 are fixed to the common base 5, and also
the common base 5 supports a brake element 14 attached
to the brake drum 9A by means of the bearing 15.

Referring illustratively to a high-speed
elevator in the class of a 240 m/min elevator speed, if
15 a 2:1 roping system in which a pulley is disposed on a
car is applied to such an elevator, it is possible to
increase the rotational speed of its sheave. If the
diameter of the sheave is set to forty times 12 mm
which is a rope diameter, that is, 500 mm and also the
20 rotational speed is set to 1500 rpm for the sake of
reducing the size of an electric motor, the resultant
reduction ratio is 4.9, and this means that, in the case
of helical gears, a one-step speed reduction is enabled.
Also, in the case of elevator having a speed of less
25 than 240 m/min, if the rotational speed of the electric
motor is controlled by inverter control or the like
within a lower range in which its economic efficiency
is not lowered, a one-step speed reduction is enabled.

- 1 This is more advantageous than a two-step speed reduction from the viewpoints of reduction in size and cost.

By so doing, since the shaft of the electric motor 7 is connected to the input shaft 8A by the
5 couplings 7A and 7B, if the couplings 7A and 7B are, for example, of a flexible type made of rubber, it is possible to absorb the vibration transmitted from the electric motor 7, thereby facilitating an alignment operation.

- 10 Also, since the input shaft 8A provided with the first gear 8B is rotatably supported at its opposite ends, the first gear 8B is not affected by the vibration transmitted from the electric motor 7, so that the first gear 8B and the second gear 8D can be engaged with
15 each other in a stable manner.

The output shaft 8F provided with the second gear 8D has a strength (or large diameter) sufficient to bear the loads such as the sheave 9 and the brake drum 9A, the output shaft 8F being secured in the
20 vicinity of the bearing 15 in an overhanging manner. Therefore, the level of fluctuations in engagement is small with respect to variations in the level of load and thus the noise can be maintained at a low level.

Also, since the first gear 8B and the second
25 gear 8D are disposed in horizontal relationship and the casing of the reduction gear assembly 8 is constituted by a pair of upper and lower members, the center distance between the mating gears can be minimized in accordance

1 with the sizes of their bearings, and yet it is easy to
adjust tooth contact.

The brake drum 9A is not disposed on the shaft
of the electric motor 7, but is integral with the sheave
5 9 about the output shaft 8F of the reduction gear
assembly 8. This arrangement serves to prevent the
impact generated upon emergency stop from acting directly
on the respective gears. Should any of the gears be
broken, the brake system is actuated to bring the sheave
10 to a stop, thereby preventing drop of the car or a
counterweight. In addition, the brake system is disposed
on the side of the sheave 9 and the dimension between
the braking system and the shaft end of the electric
motor 7 are substantially equal to that between the same
15 and the end portion of the output shaft 8F (or shaft
of the sheave 9). In consequence, as shown in Fig. 3,
the elevator hoisting apparatus can be disposed in the
substantial center of the hoistway 1 in top plan.

In addition, as shown in Fig. 4, the hoisting
20 machine is fixed to the machine-room floor 2a by support
frames 3a and 3b. If one of the support frames 3a and
3b is spaced apart from the center of the sheave 9 by
a distance equivalent to the diameters of pulleys 16a
and 16b respectively supporting the car and the counter-
25 weight, the support frame 3a functions as a support
member for supporting one end of the main rope 13 in
the 2:1 roping method.

In addition, before explaining how to improve

1 the state of engagement of the gears, the generation
and transmission of the noise of the helical gears will
first be described below. In general, gears are arranged
in such a manner that several pairs of teeth are brought
5 into engagement one after another in order to transmit
rotation. During this time, in the case of a typical
helical gear, the number of simultaneously mating gears
varies as the rotation proceeds. Since these teeth are
resilient members, each pair of engaged teeth becomes
10 a vibrating unit whose spring constant is periodically
varied. Also, since the teeth cannot be produced so as
to have ideal tooth profiles, a driven gear undergoes a
forced displacement which may be caused mainly by the
tooth profile error. Furthermore, since the point of
15 engagement of typical helical gears shifts in the axial
direction as well, the shafts of the gears serve as a
beam subjected to a traveling load. In this fashion,
in cases where power is transmitted through the helical
gears, the shaft line of the helical gears is vibrated
20 by a combination of three forced terms such as a
coefficient excitation, a forced displacement and travel
of a loading point. In response to this vibration, for
example, the circumferential vibration of the sheave
occurs or the sheave produces noise.

25 The noise will be described below in detail.
In the case of a gear assembly, in particular of the
type in which gears are enclosed in a gearbox, a noise
emanates into the air from the engaged portion of the

1 mating gears, passing through the gearbox and entering
human ears. However, only 10% of the total noise merely
reaches them, and the remaining 90% emanates outwardly
when a fluctuating torque caused by the vibration of
5 the shaft line of the aforesaid gears in the direction
of its rotation is converted into a radially and axially
fluctuating force. Therefore, in order to reduce the
level of vibration noise of the helical gear assembly,
the values of the aforementioned three terms must be
10 reduced. In this case, the term of forced displacement
cannot be reduced unless the accuracy of production of
gears is enhanced. As described above, what directly
affects the level of noise is the fluctuating load
on a tooth surface which is obtained by multiplying the
15 forced displacement and the spring constant of teeth
together. It will be found, therefore, that, if the
spring constant of teeth is reduced, the accuracy of
production of gears need not necessarily be enhanced.
On the other hand, in order to reduce the term of
20 coefficient vibration, the contact ratio of the mating
gears may be increased so as to reduce the range of
fluctuation of the spring constant of teeth. In this
case, it is preferred that the transverse contact ratio
of various contact ratios is enhanced in terms of
25 excitation effects based on the travel of a loading
point. The spring constant of teeth can be calculated
using a formula for obtaining the level of deflection
of teeth which is described in "a collection of papers

1 issued by Japan Mechanics Association, Vol. 15, No. 59".

The pressure angle and the height (or depth) of teeth were variously changed. The results will be described below with reference to Figs. 7 to 10.

5 Fig. 9 shows variations in the spring constant of a pair of engaged teeth at a pitch (t) between normals when a pressure angle (α) is changed in a case where an addendum (h) and a dedendum (h) except a root portion (or clearance) (c) are regarded as one
10 module, that is, full-depth teeth.

Fig. 10 shows variations in the spring constant (k) of a pair of mating teeth (curve (a)) and the variations of the pair of mating gears (curve (b)) obtained from the former variations while the engagement
15 between the gears is proceeding, that is, the contact point is changing. In this case, the teeth has a JIS standard tooth profile in which the pressure angle (α) is 20 degrees, the height (or depth) (h) of teeth being full-depth teeth. In Fig. 10, the length indicated by
20 (l) represents the distance of travel of the contact point measured along the line of action of the teeth, (t) representing the pitch between adjacent normals. (l/t) is represents the transverse contact ratio, in this example approximately 1.68. While the engagement
25 is advancing from point (A) to point (B), the number of a pair of simultaneously engaged teeth varies in the manner of two pairs, one pair and two pairs. This causes abrupt variations in the spring constant (k) as shown.

1 The broken line shown in Fig. 10 represents the result
obtained from the calculation of the height (or depth)
(h) of teeth when the addendum and dedendum are
regarded as a 1.1 module. Since the height (h) is
5 increased, the distance of travel of the contact point
(between (A) and (b')) is lengthened, i.e. (l').
Accordingly, variations in the spring constant (k) of
teeth as between a two-pair engagement (curve (b')) and
a one-pair engagement (curve (a)) are not so abrupt as
10 that of an arrangement in which the height (h) is
equivalent to one module.

The gears incorporated in the reduction gear
assembly 8, as shown in Fig. 7, each has a pressure
angle (α) of 17 degrees and a depth (h) equivalent to a
15 1.25 module.

As is evident from Figs. 8 to 10, if the value
of the spring constant (k) of teeth and the degree of
smoothness of its variations are merely considered, the
smaller the pressure angle (α), the better while the
20 higher the height (h), the better. However, there the
pressure angle (α) is limited in terms of the strength
of teeth and the height (h) of teeth is limited in terms
of interference during the cutting of teeth and/or the
operation of the elevator system.

25 Fig. 11 is a graph in which the pressure angle
(α) is plotted along the horizontal axis, the spring
constant (k) of teeth being plotted along one vertical
axis with reference to a pressure angle (α) of 20 degrees

1 and the tooth profile constant (Y) as the ratio of a
addendum stress and a tangential load per unit module
and per unit face width being plotted along the other
vertical axis. In Fig. 11, the spring constant (k) of
5 teeth is shown in solid line while the tooth profile
constant (Y) is shown in broken line. It is to be
noted that the tooth profile constant (Y) cited in this
specification is a so-called stress-tooth-profile
constant, and thus as the value of this constant is
10 increased, an increased level of stress is produced
with respect to the same level of load. As can be seen
from Fig. 11, when the pressure angle (α) is in the
range of 15 to 18 degrees, the level of stress does not
excessively increase (for example, equal to or less
15 than 1.7 times that generated at a pressure angle of
20 degrees) and yet the value of the spring constant (k)
can be relatively reduced.

The interference between engaged teeth depends
upon the selection of the number of teeth of gears, a
20 addendum modification coefficient, and in addition the
relation with the pressure angle (α). Although it is
therefore difficult to illustrate and depict such
interference, calculation was made within the aforesaid
range of pressure angles from 15 to 18 degrees under the
25 condition that the number of teeth of the small gear
is limited to a maximum of 35. It is preferred that
this number is reduced since it determines a contact
frequency constituting the basic frequency of the vibration

1 noise of the teeth. It has been found from the result
that a 1.35 module is the upper limit. In addition
to the gear according to this embodiment having a
pressure angle of 17 degrees and a height of 1.25 module,
5 were prepared a conventional gear having the number of
teeth, face width, helix angle and accuracy that are
equal to those of the aforesaid embodiment, the conven-
tional gear having been so formed as to have a pressure
angle of 20 degrees and a full-depth teeth type (that
10 is, the height (or hight) is 1 module.) The present
embodiment and the conventional gear were incorporated
in the same gearbox for the purposes of measuring the
level of noise produced by the respective gears. In
consequence, the result plotted in Fig. 12 was obtained.

15 Fig. 12 shows variations in the level of sound
pressure relating to a contact frequency component which
is the fundamental frequency associated with a load
factor. A line including symbols (⊙) corresponds to
the presently preferred embodiment while another line
20 including symbols (X) corresponds to the conventional
gear. As is evident from Fig. 12, when the load factor
is 30% or more, there is a difference of 10 to 14 dB
between the two.

As will be readily understood from the foregoing
25 description, in the elevator hoisting apparatus in
accordance with the present invention, the reduction gear
assembly including a pair of gears whose axes are parallel
to each other is located in the substantial center, the

1 electric motor and a pair of the sheave and the brake
system being respectively disposed on the input shaft
and output shaft of the reduction gear assembly. This
arrangement enables a reduction in the size and weight
5 of the apparatus and the compact layout of each components.
Also, since the brake system is integral with the sheave,
a burden applied to the gears is smaller than that
applied to prior-art gears. Therefore, the present
invention is superior in safety and reliability.

10 In addition, it is possible to reduce the
levels of noise and vibration attributed to engagement
of the gears of the elevator hoisting machine having a
helical gear type reduction gear assembly without
lowering the strength of the gears and providing dis-
15 advantages when the gears are produced. It is therefore
possible to reduce the level of noise and vibration
transmitted to the elevator cars and to a residential
room in the vicinity of the uppermost floor of a buiding
structure. In the consequence, the quantity of various
20 measures for sound insulation can be reduced as compared
with the prior art, and furthermore, if the conventional
sound insulation measures are not modified, the level of
noise and vibration can be further lowered, whereby it
is possible to provide low-cost elevator hoisting
25 apparatus featuring a silent operation.

**CLAIMS:-**

1. An elevator hoisting apparatus having a reduction gear assembly connected to a driving electric motor and a sheave, a main rope being reeled around said sheave for supporting a car in a suspended manner so that said car may be moved upwardly and downwardly, said reduction gear assembly including therein a plurality of helical gears whose diameters are of different sizes from one another and which are mounted on respective parallel shafts, at least small and large gears of said helical gears being connected respectively to said input shaft and said output shaft in said reduction gear assembly, said driving electric motor being located on the side of said input shaft while said sheave and a brake system are located on the side of said output shaft, one bearing of said sheave being integral with a gearbox of said reduction gear assembly, and said helical gears having a pressure angle of 15 to 18 degrees.
2. An elevator hoisting apparatus according to claim 1, wherein said large gear is supported within said reduction gear assembly in a cantilever manner by said input shaft of said reduction gear assembly.
3. An elevator hoisting apparatus having a reduction gear assembly connected to a driving electric motor and a sheave, a main rope being reeled around said sheave for supporting a car in a suspended manner so that said car may



be moved upwardly and downwardly, said reduction gear assembly including therein a plurality of helical gears whose diameters are of different sizes from one other and which are mounted on respective parallel shafts, at least
5 small and large gears of said helical gears being connected respectively to said input shaft and said output shaft in said reduction gear assembly, said driving electric motor being located on the side of said input shaft while said sheave and a brake system being located on the side of said
10 output shaft, one bearing of said sheave being integral with a gearbox of said reduction gear assembly, and said helical gears having a dedendum of 1.1 to 1.3 module which is the value obtained by subtracting an addendum and a root from the depth of the teeth.

15 4. An elevator hoisting apparatus according to claim 3, wherein said large gear is supported within said reduction gear assembly in a cantilever manner by said input shaft of said reduction gear assembly.

20 5. An elevator hoisting apparatus according to claim 3 or 4, wherein said helical gears have a pressure angle of 15 to 18 degrees.

6. An elevator hoisting apparatus constructed substantially as herein described with reference to and as illustrated in Figs. 1 to 4 and Figs. 7 to 12 of the
25 accompanying drawings.
