Size selection C

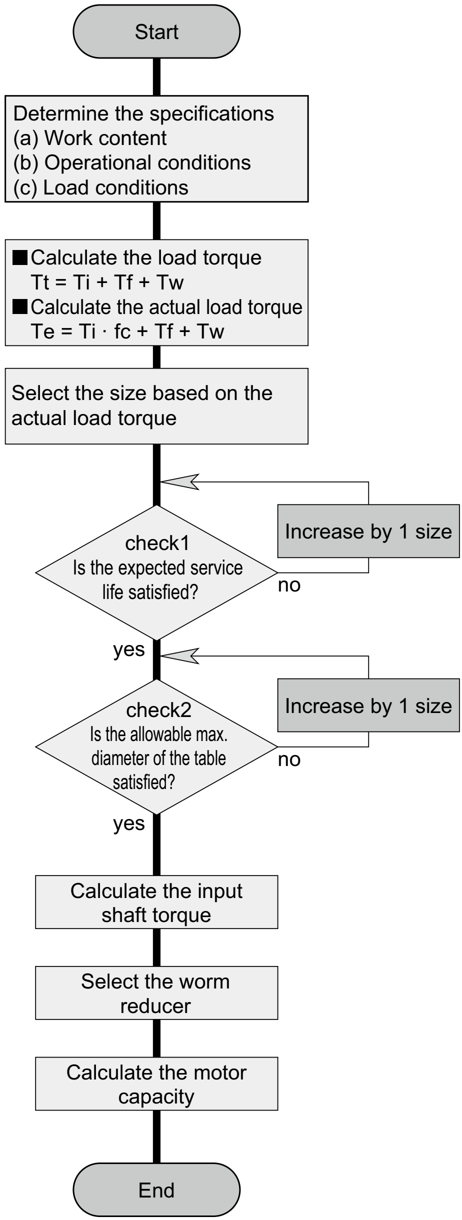
How to determine the specifications and select the size

When you select specifications and model of the index drive, determine the following primary specifications first.

Work content	Operational conditions	Load conditions
Table drive Conveyer drive Oscillatior drive	Index number Index angle Cam curve Rotational speed of input shaft	Load capacity Outside table diameter Table supporting method External load

Taking service life and allowable max. table diameter into consideration, the selection should be finalized comparing the actual load torque applied to the output shaft with the dynamic rated output torque of the index drives when the index drive is operated under these conditions.
When the index drive is selected, the design of the index angle should be as large as possible, the table diameter as small as possible and the weight as light as possible from the start. Taking all these factors into consideration in designing may help you reduce the size of the index drives, which can contribute to saving space and costs.

Size selection flow chart



1. Calculating the load torque

a. Load torque for the standard timing

When the table or conveyor is driven, 3 different types of torque is applied to the output shaft of index drives: inertia torque, frictional torque and work torque. The total of these 3 is called the load torque.

$$Tt = Ti + Tf + Tw \ (N \cdot m)$$

Here, Tt : Load torque (N·m)
Ti : Inertial torque (N·m)
Tf : Frictional torque (N·m)
Tw : Work torque (N·m)

(1) Inertial torque: Ti

The inertial torque is the torque required to accelerate/decelerate the table, the jigs and the workpieces mounted on the output shaft during indexing.
The inertia torque can be obtained by multiplying the moment of inertia with the output shaft max. angular acceleration.

$$Ti = I \cdot \alpha \ (N \cdot m)$$

Here, I : Total moment of inertia ($kg \cdot m^2$)
α : Max. angular acceleration of the output shaft (rad/s^2)

(1)-1 Total moment of inertia: I

This indicates the total moment of inertia for various objects with the identical center of rotation. (Refer to the moment of inertia formulas.)

$$I = I_1 + I_2 + ... + In \ (kg \cdot m^2)$$

(1)-2 Max. angular acceleration of the output shaft: α

The max. output shaft angular acceleration can be obtained with a formula which consists of the non-dimensional max. acceleration Am of the cam curve, Index number n, index angle θh and input shaft rotational speed N.

● Index

$$\alpha = Am \cdot \frac{2\pi}{n} \left(\frac{360}{\theta h} \cdot \frac{N}{60} \right)^2 \ (rad/s^2)$$

Here, Am : non-dimensional max. acceleration of cam curve

Type of curve	MS modified sign curve	MC modified constant velocity curve	MT modified trapezoidal curve	TR trapecloid
Am	±5.53	±8.01	±4.89	±6.17

n : Index number
θh: Index angle (degrees)

$$\theta h = \frac{\theta t}{z}$$

θt : Total index angle (degrees)
N : Input shaft rotational speed (rpm)
z : Dwell No.

● Oscillator

$$\alpha = Am \cdot \frac{\pi \cdot \psi}{180} \cdot \left(\frac{360}{\theta h} \cdot \frac{N}{60} \right)^2 (rad/s^2)$$

Here, ψ : Oscillating angle (°)

(2) Frictional torque: Tf

The friction torque is the torque applied to the output shaft due to friction of the bearing and the sliding surfaces. Friction torque can be obtained using the following formula.

$Tf = \mu \cdot Ff \cdot Rf$ (N·m)
$Ff = mg$ (N)

Here, μ : Coefficient of friction

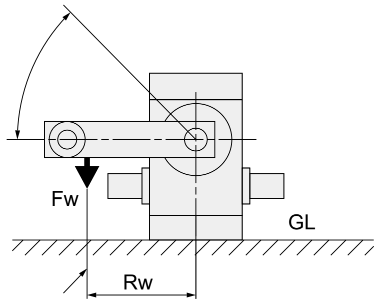
Rolling friction	Sliding friction
μ = 0.03 to 0.05	μ = 0.1 to 0.3

Ff : Force applied to a sliding surface and a bearing (N)
Rf : Average frictional radius (m)
m : Weight (kg)
g : Gravity acceleration (m/s^2)

How to determine the specifications and size selection

(3) Work torque: Tw

When the output shaft is in operation or when an external load acts on the output shaft as a load, the total of these is the work torque.
For example, when the output shaft is set up to be horizontal, the forces caused by the table (arm), the jigs and the workpieces as eccentric load are to be considered as the work torque.



Arms and workpieces
Position of center of gravity

Work torque can be obtained using the following formula.

$$Tw = Fw \cdot Rw \ (N \cdot m)$$

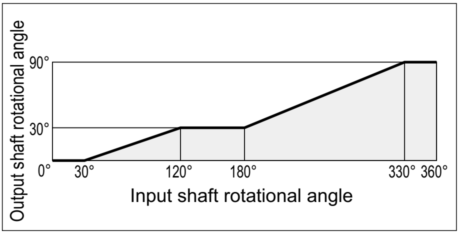
Here, Fw : Force required for work (N)
Rw : Radius for work (m)

CAUTION: Compare the torque which is applied when the input shaft stops to the rated static output torque.

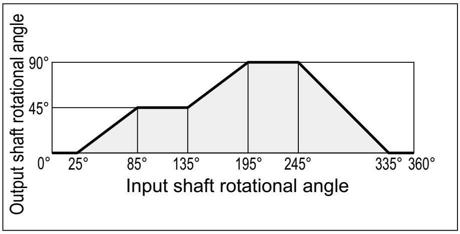
b. Load torque for the special timing

After obtaining load torque Tt for each index range, the max. value should be selected.

Example of special timing (index)



Example of special timing (oscillation)



c. Load torque for indirect drive

When you make index units indirectly drive the unit to its output shaft using gears and/or sprockets, calculate the equivalent load torque considering the reduction ratios (io) of various torques.

(1) Equivalent inertial torque for the output shaft: Tie

$$Ie = I \cdot io^2 \ (kg \cdot m^2)$$

Here, Ie : Equivalent moment of inertia for the output shaft

$$Tie = Ie \cdot \alpha \ (N \cdot m)$$

(2) Equivalent frictional torque for the output shaft: Tfe

$$Tfe = Tf \cdot io \ (N \cdot m)$$

(3) Equivalent work torque for the output shaft: Twe

$$Twe = Tw \cdot io \ (N \cdot m)$$

2. Calculating the actual load torque: Te

The load torque mentioned above is the theoretical torque. The actual load torque applying to the output shaft of the index drive is greater than this because of the rigidity of the driving system, presence of backlash and the coupling method. Furthermore, how the unit is used critically affects it.
Hence, you should calculate the actual load torque taking the usage factor for your operational conditions which you have obtained from experience into consideration.

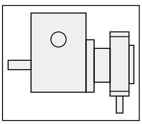
$$Te = Ti \cdot fc + Tf + Tw \ (N \cdot m)$$

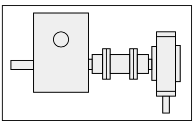
Here, Te: Actual load torque (N·m)
fc : Usage factor

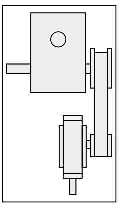
(Table C.1) Usage factors for various driving methods

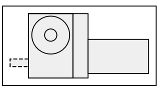
Driving method		Reduction ratio of the reducer	
Output system	Input system	≤ 1/20 [1/10, 1/20]	> 1/20 [1/30,1/40 1/50, 1/60]
Table arm directly connected	Direct worm (1)	1.6	1.5
	Direct worm (2)	1.7	1.6
	Indirect worm	2.1	1.9
	Geared motor	3.7	3.7
	Geared motor (hypoid gear)	2.0(2.5)	2.0(2.5)
	Geared motor (helical worm gear)	1.6	1.5
Table indirect	Direct worm (1)	2.0	1.8
	Direct worm (2)	2.2	2.0
	Indirect worm	2.7	2.5
	Geared motor	4.7	4.7
	Geared motor (hypoid gear)	2.5(3.1)	2.5(3.1)
	Geared motor (helical worm gear)	2.0	1.8
Conveyer drive	Direct worm (1)	1.9	1.7
	Direct worm (2)	2.0	1.9
	Indirect worm	2.5	2.3
	Geared motor	4.4	4.4
	Geared motor (hypoid gear)	2.4(3.0)	2.4(3.0)
	Geared motor (helical worm gear)	1.9	1.7

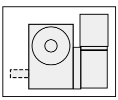
Note: Figures in ( ) are for RGIB series.


Direct worm (1)
Worm reducer is directly mounted on the housing.


Direct worm (2)
Connected with a coupling.


Indirect worm

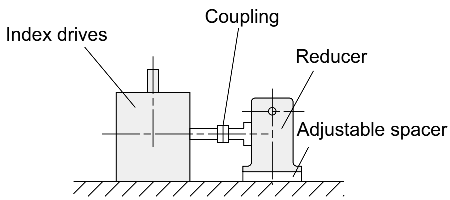

Geared motor


Geared motor
(Hypoid gear)/(Helical worm gear)

CAUTION: When a system is designed, make sure to give sufficient rigidity for the drive system from the motor to the input shaft of index drives minimizing backlash. This allows you to reduce the usage factors and helps you select the suitable index size.
Backlash in the system may cause vibration while the table is rotating, shortening the service life of the unit, and damaging the components.

(1) Avoid inserting a camshaft in series between the motor and the input shaft of the index drives. If it cannot be avoided, design the system taking the following factors into consideration.
- -1 Increase the rigidity of the cam shaft.
- -2 Select the connection method of min. backlash.
- -3 Select the motor capacity with sufficient margin.
- -4 Use a timing belt of high rigidity.

(2) When the input shaft of the index drives are connected directly to the output shaft of a reducer with a coupling, be sure to use a coupling of high rigidity which does not have backlash. Consideration should be given to making the center height adjustable.



Example of a drive

How to determine the specifications and size selection

3. Size selection

Provisionally select the index drives based on the actual load torque Te and verify that this satisfies the expected service life and the allowable max. diameter of the outside table.

Check. 1 Service life: Lh

We design our products to have a service life of 10,000 hours based on the service life for rolling fatigue of the race surface of the cam follower at the input shaft rotational speed at the rated torque.
When you want service life longer than 10,000 hours, determine the required torque using the following formula taking the service life coefficient into consideration.

$$fh = \frac{Tr}{Te}$$

$$Lh = 10000 \cdot fh^{(10/3)} (h)$$

Here, Tr : Rated dynamic output torque (N·m)
fh : Service life coefficient
Lh: Service life hours (h)

Verify the service life is above the expected service life. If not, the size of the index drives should be increased.
Service life of repair parts such as oil seals depends on the Operational conditions, and it may not be satisfied in some cases.

(Table C.2) Service life and the coefficient of service life

Lh(h)	fh	Lh(h)	fh	Lh(h)	fh
10,000	1.000	25,000	1.316	60,000	1.712
12,000	1.056	30,000	1.390	70,000	1.793
14,000	1.106	35,000	1.456	80,000	1.866
16,000	1.151	40,000	1.516	90,000	1.933
18,000	1.193	45,000	1.570	100,000	1.995
20,000	1.231	50,000	1.621		

If the expected service life Lh is less than 10,000 hours, contact us.

If you choose the service life longer than necessary when you select a model, the size of the index drives will become large, which is uneconomical. Considering the actual time when the input shaft of the index drive is in operation, choose the most economical service life when you design the system.

Check. 2 Allowable max. diameter of the table: Dm

For the table drive, if the outside diameter of the table is large compared to the difference between the shafts of the index drives, a small impact may generate a force greater than the static torque of the index drives, leading to damage of the cam follower and the cam. This may also result in the increased residual vibration due to the outside diameter of the table being too large, and positioning will take too much time.
Based on our experience, the allowable diameter of the table where the index is mounted is calculated using the following formula as a guide. Note that this formula only applies to the standard unit. You should be careful when there is a factor which negatively affects the rigidity of the system such as an extended output shaft.

$$Dm \geq De$$

Here, Dm : Allowable max. diameter of the table (mm)

$$Dm = \frac{C \cdot ft}{io}$$

De: Max. diameter of the table (mm)
ft : Table coefficient (refer to Table C.3.)
C : Index drive size (Shaft interval) (mm)
io : Reduction ratio of the output shaft

(Table C.3) Formula for calculating the coefficient of table

Type	Formula	ft max. value	
Compact	ft=1.5·fh+2.5	ft=7	
Standard	ft=2.5·fh+2.5	ft=12	*1
Wide angle	ft=1.5·fh+1.5	ft=6	
Table	ft=2·fh+4	ft=12	*1
Compact multi-index			
Flat			
Basic			

*1
- We recommend this to be used with De ≤ $\frac{10/C}{io}$.

For the service life coefficient fh, refer to Table C.2.

[For using a parallel cam drive]

The parallel cam is easy to make because of its simple construction, but the rigidity will be low as the pressure angle at the dwell interval is high.
If the parallel cam is used to drive a table whose inertia load is large, residual vibration may be generated because of its low rigidity at the dwell interval. Residual vibration has a negative effect on the index accuracy, as well as on the service life of the index drives. In order to drive the table, we recommend the roller gear which has high rigidity at the dwell interval.
The driving method suitable for the parallel cam is the conveyor drive. The conveyor drive has the friction load in addition to the inertia load. Friction load suppresses residual vibration. Parallel cams with small index number can be manufactured, and a single indexing is offered as standard. Using the index drives of a small index number for conveyor drive allows large feed pitch motion. Using a gear reducer also reduces the load torque applied to the output shaft, which makes it possible to reduce the size of index drives. In addition, since the input and output shafts are arranged in parallel, this mechanism is suitable for conveyors.

4. Calculating the input shaft torque: Tc

The input shaft torque is the sum of the 1 required to drive the load which applies to index drives output shaft and the internal frictional torque. The former can be further divided into the inertia load torque and the friction/work load torque.

$$Tc = Tci + Tcw$$

Here, Tci : Input shaft torque (N·m) due to inertia load
Tcw : Input shaft torque due to frictional load (N·m)

$$Tci = \frac{\psi}{\theta h} \cdot Qm \cdot Ti$$

Here, ψ : Oscillating angle $\psi = \frac{360}{n}$(°)

θh : Index angle (degrees)
Qm: Torque coefficient

$$Tcw = \frac{\psi}{\theta h} \cdot Vm \cdot (Tf + Tw) + Tin$$

Here, Vm: Non-dimensional max. speed of cam curve
Tin : Internal frictional torque (N·m)

(Table C.4) Qm and Vm list

	MS modified sign curve	MC modified constant velocity curve	MT modified trapezoidal curve	TR trapecloid
Qm	±0.99	±0.72	±1.65	±1.76
Vm	1.76	1.28	2.00	2.18

CAUTION: The input shaft torque obtained here is the one required to drive the index drives itself. The torque must be calculated separately due to the external load.

How to determine the specifications and size selection

5. Selecting the worm reducer

Calculate the torque Ter of the output shaft of the reducer using the following formula. (when it is directly connected to the input shaft of the index drives)

$$Ter = Tc \times fr \ (N\cdot m)$$

Here, Ter : Load torque (N·m) of the reducer
Tc : Index input shaft torque
fr : Reducer usage factor
(Refer to Table C.5.)

Compare Ter obtained here in the worm reducer rated output torque table to verify that the reducer can be used combined with the index drives. If Ter is greater than the worm reducer rated output torque in the standard combination, the size of the reducer should be increased. For details (dimensions for mounting a reducer), contact us.

(Table C.5) Reducer usage factors: fr

	Operational hours per day		
	2 hours	10 hours	24 hours
Continuous operation	0.90	1.25	1.50
Intermittent operation	1.25	1.50	1.75

6. Calculating the motor capacity

(1) When the index drive is driven with a worm reducer

●Calculating the motor capacity: P
You can obtain the motor capacity for the index drives itself from the input shaft torque and input shaft rotational speed of the index drives.

$$P=\frac{Tc\cdot N}{9550\cdot\eta}(kW)$$

Here, P : Motor capacity (kw)
Tc: Input shaft torque (N·m)
N : Input shaft rotational speed (rpm)
η : Efficiency of the reducer (η < 1)
(Refer to "Section D: Reducer Specifications.")

●Calculating the actual motor capacity: Pe
The capacity calculated above is the peak value for index driving. The actual capacity is roughly half of the above generally required for the inertia torque. In addition, considering the motor capacity of the reducer unit, a value is added to the motor capacity of the index drive.

$$Pe = \frac{1}{2}\times \text{Motor capacity due to inert load}$$
$$+ \text{Motor capacity for frictional load}$$
$$+ \text{Motor capacity for reducer unit}$$

$$Pe = \frac{1}{2}\cdot\frac{Tci\cdot N}{9550\cdot\eta}+\frac{Tcw\cdot N}{9550\cdot\eta}+Pr$$
$$=\frac{N}{9550\cdot\eta}\left(\frac{1}{2}\cdot Tci+Tcw\right)+Pr(kW)$$

Here, Pe : Actual motor capacity (kW)
Pr : Motor capacity for the worm reducer unit itself (kW)

$$Pr=\frac{Tinr\cdot Nr}{9550}\ (kW)$$

Tinr: Internal frictional torque of the reducer (N·m)
(Refer to Table C.6.)
Nr : Worm shaft rotational speed (rpm)

Table C.6 indicates the relationship between the oil temperature of the reducer and its internal friction torque for each size. When we make calculations, we assume the oil temperature to be 10 degrees C unless otherwise requested.

(Table C.6) Internal frictional torque of the reducer: Tinr (N·m)

Reducer size		Oil temperature			
		5°C	10°C	15°C	20°C
HO	32	0.30	0.24	0.19	0.16
	40	0.53	0.42	0.34	0.29
	50	0.92	0.72	0.59	0.50
	60	1.5	1.1	0.93	0.79
	80	2.9	2.2	1.8	1.4
	100	4.0	3.1	2.5	2.0
	135	5.7	4.5	3.6	2.9
TE	35	0.38	0.33	0.29	0.26
	42	0.61	0.52	0.45	0.40
	51	1.25	1.00	0.85	0.72
	63	2.03	1.63	1.34	1.14
	80	3.56	2.72	2.19	1.83
	100	6.11	4.56	3.55	2.90
	150	10.6	7.96	6.15	4.95
CRG	25	1.96×10^{-2}			
	32	1.96×10^{-2}			
CAUTION: The following standard lubricating oil should be used. HO reducer Bonnoc TS320 TE reducer Daphne Alpha Oil TE260 CRG reducer grease No. 2 (sealed)					

CAUTION:

When you use a worm reducer other than the above, add to Pr the value obtained by converting the internal frictional torque given in the technical data into motor capacity.

In addition, since the viscosity of lubricating oil in the reducer becomes high in the cold climate region or during cold winter morning, higher motor capacity will become necessary. As a result, the motor capacity may become insufficient failing to obtain the expected speed. In the worst case, the motor may get stuck.

As you can see, select a motor with sufficient margin for the calculation.

(2) When the index drive is driven with a geared motor

● Calculating the actual motor capacity: Pe

When a motor drives a mechanism whose torque fluctuates significantly such as index drives, the motor may be driven reversely in the deceleration range.

Furthermore, compared to the worm reducer, the geared motor has larger backlash, creating a situation where it will be directly affected by this. Hence, the peak value for index drive is taken as the actual motor capacity Pe.

Pe = Motor capacity due to inert load
+ Motor capacity for frictional load

$$Pe=\frac{Tci\cdot N}{9550\cdot\eta}+\frac{Tcw\cdot N}{9550\cdot\eta}$$

$$=\frac{N}{9550\cdot\eta}(Tci+Tcw)\ (kW)$$

Here, Pe : Actual motor capacity (kW)

CAUTION:

Some manufacturers of geared motors indicate the rated torque of the gear head. You should verify that the selected motor frame number satisfies the following conditions.

Index drives Input shaft torque	<	Gear head Rated torque

How to determine the specifications and size selection

7. Selecting the hollow shaft geared motor

(1) Verify the motor power (Pm) in the characteristics table for the selected index drives to confirm that it exceeds the motor capacity (Pe) obtained in 6.

$Pm \geq Pe$

(2) Select the input shaft rotational speed (N) from the output shaft rotational speed (Nrs) in the characteristics table to determine the reduction ratio (ir).
When you control the unit with an inverter, select the reduction ratio (ir) so that the specified rotational speed is obtained with the setup frequency of the inverter to be 40-60 Hz for continuous operation.

(3) Verify that the allowable output torque (Trr) of the geared motor exceeds the geared motor load torque (Ter).

$Trr \geq Ter$

$Ter = Tc \times fr$

fr: Factor when geared motor is used

(Table C.7) Usage factor for geared motor: fr

Motor type	Operation	Operational hours per day		
		2 hours	10 hours	24 hours
Hypoid gear	Continuous	1.30		1.75
	Intermittent			
Helical worm gear	Continuous	0.90	1.25	1.50
	Intermittent	1.25	1.50	1.75

8. Selecting the inverter

When the inverter controls the starting/stopping of the index drive, a suitable inverter should be chosen for the task.
Referring to the catalog of the chosen inverter, verify its capacity.
If the capacity is satisfied, any inverter can be used irrespective of manufacturer and model.

(1) Calculating the required capacity of the inverter

Capacity required for starting/stopping the motor: Pinv

-1 Motor braking time: td

It is required that the total of the braking time required for starting and that required for stopping should be within the dwell time (t_2).

$$t_2 = \frac{60}{N} \times \frac{360-\theta h}{360} \text{ (s)}$$

t_2: Index drives dwell time (s)

$td \leqq \frac{1}{2} t_2$ td : Motor braking time (s) required for starting/stopping

Or when sensor signals are taken out using a detection cam on the input shaft of index drives

Within the braking angle determined by the braking time, the adjusting angle required for the detection cam should be 50% of it.

$$td \leq \frac{1}{4} t_2$$

Note: The braking time (td) is assumed to be the min. of 0.1 sec.

(2) Converted moment of inertia for the motor shaft: JMc

$$JMc=\frac{Ic}{ir^2}+JM\ (kg\cdot m^2)$$

JMc : Converted moment of inertia for the motor shaft ($kg\cdot m^2$)
JM : Moment of inertia of the motor ($kg\cdot m^2$)
ir : Reduction ratio
Ic : Moment of inertia of the input shaft ($kg\cdot m^2$)
Note: For Ic, refer to the Characteristics table of individual models.

(3) Required capacity of the inverter: Pinv

$$Pinv=\frac{JMc\cdot Nms^2}{91190\cdot td}\ (kW)$$

Pinv: Required capacity of the inverter (kW)
Nms: Motor speed (rpm)

(2) Selection of the inverter

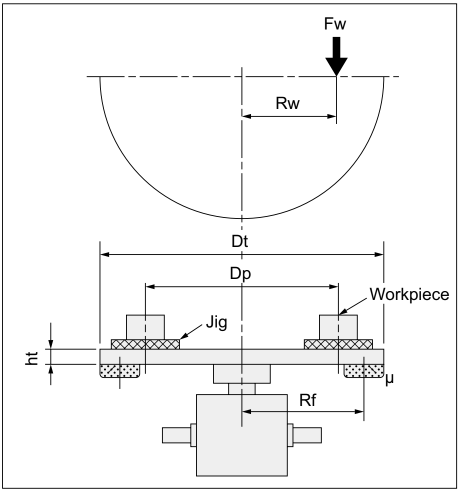
-1 Select the inverter whose capacity is greater than the required capacity of inverter (Pinv) obtained with the motor output (Pm) in the Characteristics table and calculation.

Inverter capacity ≥ Pm or Pinv

Note1: About braking time (td)
When you set up braking time (td) to be 0.1 sec. or less, the capacity of the geared motor may not be sufficient.
If you want the specification of 0.1 sec. or less, you should re-select the geared motor. Contact CKD if necessary.

Note2: About starting and stopping
Be sure to make the Index drives start and stop within the dwell range. If it is unavoidable to make it stop/start while rotating, allow enough braking time.

Example of calculating the table drive (direct drive)



Selection conditions

(1) Model: RGIS
(2) Shape of the output shaft: flange
(3) Index number: n = 6
(4) Input shaft rotates intermittently
(5) Machine cycle time: tm = 5 s
(6) Movement time (index time): t_1 = 0.6 s
(7) Index angle: θh = 270 degrees
(8) Unit cycle time: t_0

$$t_0=0.6\times\frac{360}{270}=0.8\ \text{s}$$

(9) Input shaft rotational speed: N

$$N=\frac{60}{0.8}=75\ \text{rpm}$$

(10) Cam curve: MS modified sign curve
(11) Table diameter: Dt = 300 mm
(12) Table thickness: ht = 20 mm
(13) Material of table:
Steel (Material density ρ = 7.86 g/cm^3)
(14) Number of workpieces: nw = 2
(15) Total weight of workpieces: m_2 = 1.5 kg × 2 = 3 kg
(16) Number of jigs: np = 6
(17) Total weight of jigs: m_3 = 6 kg × 6 = 36 kg
(18) Diameter of jigs and workpieces mounting center: Dp = 250 mm
(19) With a support underneath the table and roller bearing
μ=0.03
(20) Average frictional radius: Rf = 125 mm
(21) Force required for work (external load 1 kgf)
Fw=1×9.81=9.81N
(22) Radius for work: Rw = 125 mm
(23) When index drive is driven with a worm reducer
(24) Index drives should be installed in Position #1
(25) Material of housing: Fc
(26) The cam of index drive is spiraled in the left direction (standard)
(27) Expected service life: 12000 hours
Select the size for the above conditions.

CAUTION:
You should distinguish the cycle time of the index drives and that of the machine.

● Unit cycle time: t_0 (s)

$$t_0=t_1+t_2=\frac{60}{N\times z}$$

Here, t_1 : Index time (s)
t_2 : Dwell time (s)
N: Input shaft rotational speed (rpm)
z : Dwell No.

● Machine cycle time: tm (s)

$$tm=t_0+ts$$

Here, t_0 : Unit cycle time (s)
ts: Stop time (s) (input shaft stop time)

In addition, when the input shaft operates intermittently, a large index angle is generally chosen.

Size selection

1. Calculating the load torque: Tt

● Calculating the weight which is a load

Weight of the table

$$m_1=\frac{\pi}{4}\cdot Dt^2\cdot ht\cdot\rho\cdot 10^{-6}$$

$$=\frac{\pi}{4}\times 300^2\times 20\times 7.86\times 10^{-6}$$

$$=11.1\ \text{kg}$$

(1) Inertial torque: Ti

(1)-1 Total moment of inertia

Moment of inertia of the table

$$I_1=\frac{m_1\cdot R^2}{2}=\frac{11.1\times 0.15^2}{2}=0.125\ \text{kg}\cdot\text{m}^2$$

Moment of inertia of the workpieces

$$I_2=m_2\cdot Re^2=3\times 0.125^2=0.0469\ \text{kg}\cdot\text{m}^2$$

Moment of inertia of the jigs

$$I_3=m_3\cdot Re^2=36\times 0.125^2=0.563\ \text{kg}\cdot\text{m}^2$$

● Total moment of inertia

$$I=I_1+I_2+I_3=0.125+0.0469+0.563$$

$$=0.735\ \text{kg}\cdot\text{m}^2$$

(1)-2 Max. angular acceleration of the output shaft

$$\alpha=Am\cdot\frac{2\pi}{n}\cdot\left(\frac{360}{\theta h}\cdot\frac{N}{60}\right)^2$$

$$=5.53\times\frac{2\pi}{6}\times\left(\frac{360}{270}\times\frac{75}{60}\right)^2$$

$$=16.1\ \text{rad/s}^2$$

● Thus, the inertial torque is;

$$Ti=I\cdot\alpha$$

$$=0.735\times 16.1$$

$$=11.8\ \text{N}\cdot\text{m}$$

(2) Frictional torque: Tf

● Calculating the force applied to a sliding surface and a bearing

$$Ff=m\cdot g$$

$$=(11.1+3+36)\times 9.81$$

$$=491\ \text{N}$$

● Calculating the frictional torque

$$Tf=\mu\cdot Ff\cdot Rf$$

$$=0.03\times 491\times 0.125$$

$$=1.84\ \text{N}\cdot\text{m}$$

(3) Work torque: Tw

$$Tw=Fw\cdot Rw$$

$$=9.81\times 0.125$$

$$=1.23\ \text{N}\cdot\text{m}$$

(4) Load torque: Tt

$$Tt=Ti+Tf+Tw$$

$$=11.8+1.84+1.23$$

$$=14.9\ \text{N}\cdot\text{m}$$

2. Calculating the actual load torque: Te

● Determine the usage factors

fc=1.6

(Refer to Table C.1. Usage factors for various driving methods.)

● Actual load torque

$$Te=Ti\cdot fc+Tf+Tw$$

$$=11.8\times 1.6+1.84+1.23$$

$$=22.0\ \text{N}\cdot\text{m}$$

Example of calculating the table drive (direct drive)

3. Index drive size selection

You can find the index drives which rated output torque satisfies the actual load torque of Te = 22.0 N·m to be RGIS040 or greater in the rated output torque table. Provisionally select RGIS040 (Tr = 26.1 N·m, C = 40) and verify that this satisfies all the conditions.

Check. 1 Service life: Lh

Service life coefficient: fh

$$fh = \frac{Tr}{Te} = \frac{26.1}{22.0} = 1.19$$

$$Lh = 10000 \cdot fh^{(10/3)}$$
$$= 10000 \times 1.19^{(10/3)}$$
$$= 17900 \text{ h}$$

Hence, this satisfies the expected service life (12000 hours).

Check. 2 Allowable max. diameter of the table: Dm

Table coefficient: ft

$$fh < 3$$
$$ft = 2.5 \cdot fh + 2.5$$
$$= 2.5 \times 1.19 + 2.5$$
$$= 5.48$$
$$Dm = C \cdot ft$$
$$= 40 \times 5.48$$
$$= 219 \text{ mm}$$

Hence, this does not satisfy Dm ≥ Dt = 300 mm.
Here, you should reselect the drive to a 1 step larger in size, RGIS050 (Tr = 43.1 N·m, C = 50).

Check. 1 Service life: Lh

Service life coefficient: fh

$$fh = \frac{Tr}{Te} = \frac{43.1}{22.0} = 1.96$$

$$Lh = 10000 \cdot fh^{(10/3)}$$
$$= 94200 \text{ h}$$

Hence, this satisfies the expected service life (12000 hours).

Check. 2 Allowable max. diameter of the table: Dm

Table coefficient: ft

$$ft = 2.5 \cdot fh + 2.5$$
$$= 7.4$$
$$Dm = C \cdot ft$$
$$= 50 \times 7.4$$
$$= 370 \text{ mm}$$

Hence, this satisfies Dm ≥ Dt = 300 mm.
Based on the above, RGIS050-006270S1F1 is chosen.

4. Calculating the input shaft torque: Tc

$$Tc = Tci + Tcw$$

$$Tci = \frac{\Psi}{\theta h} \cdot Qm \cdot Ti$$

$$Tcw = \frac{\Psi}{\theta h} \cdot Vm \cdot (Tf + Tw) + Tin$$

For the internal frictional torque Tin, refer to the characteristics table in the catalog.

$$Tci = \frac{60}{270} \times 0.99 \times 11.8$$
$$= 2.6 \text{ N·m}$$

$$Tcw = \frac{60}{270} \times 1.76 \times (1.84 + 1.23) + 3.0$$
$$= 4.2 \text{ N·m}$$

$$Tc = 2.6 + 4.2$$
$$= 6.8 \text{ N·m}$$

5. Selecting the worm reducer (HO series)

● Determine the reducer usage factors. (Table C.5)

fr=1.5

(Continuous operation: Operation exceeding 10 hours per day)

● Calculate a torque Ter of the output shaft of HO reducer.

$$
\begin{aligned}
Ter &= Tc \cdot fr \\
&= 6.8 \times 1.5 \\
&= 10.2 \text{ N·m}
\end{aligned}
$$

Verify Ter obtained here to be within the dynamic rated output torque Trr of the standard combination reducer.
The standard HO reducer for RGIS050 is HO32.
The reduction ratio required for 75 rpm of the input shaft rotational speed is 1/20 for 60 Hz.
Worm shaft rotational speed: Nr = 75 × 20 = 1500 rpm. Thus, calculate the dynamic rated output torque Trr when the worm shaft rotational speed of HO32-1/20 is 1500 rpm.

Trr=16 N·m
This is acceptable because Trr ≥ Ter.

Since the operation is intermittent for this specification, the clutch and brake are required. Hence, the reducer HO32-1/20 with C/B is chosen.

6. Calculating the motor capacity: P

When index drive is driven with a worm reducer (HO series), calculate the motor capacity.

● Calculating the actual motor capacity: Pe
To calculate the internal frictional torque of the worm reducer, suppose the oil temperature to be 10°C.

Tinr=0.24 N·m

(Refer to Table C.6. Internal frictional torque of the HO reducer.)

$$
Pr = \frac{Tinr \cdot Nr}{9550} = \frac{0.24 \times 1500}{9550} = 0.038 \text{ kW}
$$

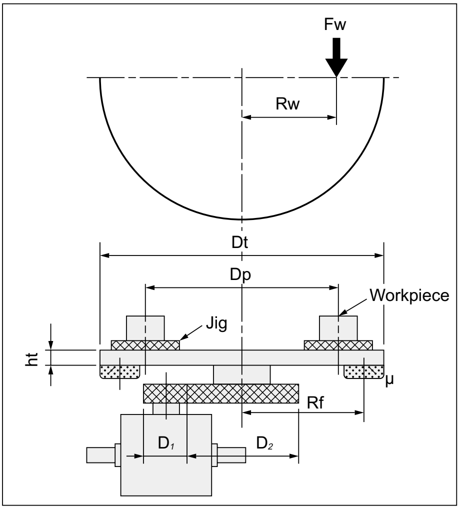
$$
Pe = \frac{N}{9550 \times \eta}\left(\frac{1}{2} \times Tci + Tcw\right) + Pr
$$

$$
= \frac{75}{9550 \times 0.68}\left(\frac{1}{2} \times 2.6 + 4.2\right) + 0.038
$$

$$
= 0.102 \text{ kW}
$$

Hence, the motor whose capacity is 0.102 kW or greater should be chosen.

Example of calculating the table drive (indirect drive)



Selection conditions

(1) Model: RGIS
(2) Shape of the output shaft: straight
(3) Number of stations: ns = 6
(4) Input shaft rotates intermittently
(5) Machine cycle time: tm = 6 s
(6) Movement time (indexing time): t_1 = 1.5 s
(7) Index angle: θh = 300 degrees
(8) Unit cycle time: t_0

$$t_0=1.5\times\frac{360}{300}=1.8\ \text{s}$$

(9) Input shaft rotational speed: N

$$N=\frac{60}{1.8}=33.3\ \text{rpm}$$

(10) Cam curve: MS modified sign curve
(11) Table diameter: Dt = 1200 mm
(12) Table thickness: ht = 25 mm
(13) Material of table: aluminum
(Material density ρ = 2.7 g/cm^3)
(14) Number of workpieces: nw = 6
(15) Total weight of workpieces: m_2 = 10 kg × 6 = 60 kg
(16) Number of jigs: np = 6
(17) Total weight of jigs: m_3 = 30 kg × 6 = 180 kg
(18) Diameter of jigs and workpieces mounting center: Dp = 1000 mm
(19) Gear on the table side:
Weight … m_4 = 45 kg
Number of teeth … Z_2 = 180
Pitch circle diameter … D_2 = 540 mm
(20) Index drives output shaft gear:
Weight … m_5 = 5 kg
Number of teeth … Z_1 = 60
Pitch circle diameter … D_1 = 180 mm
(21) Reduction ratio of the output shaft:

$$io=\frac{Z_1}{Z_2}=\frac{60}{180}=\frac{1}{3}$$

(22) Index number:

$$n=ns\cdot io=6\times\frac{1}{3}=2$$

(23) With a support underneath the table.
Roller bearing μ = 0.03
(24) Average frictional radius: Rf = 500 mm
(25) Suppose there is no force required for operation.
(26) When index drive is driven with a worm reducer.
(27) Index drives should be installed in Position #1.
(28) Material of housing: FC
(29) The cam of the index drive is spiraled in the left direction (standard).
(30) Expected service life: 10000 hours
Select the size for the above conditions.

Size selection

1. Calculating the equivalent load torque: Tte

(1) Equivalent inertial torque for the output shaft: Tie

● Calculating the weight which is a load

Weight of the table

$$m_1=\frac{\pi}{4}\cdot Dt^2\cdot ht\cdot \rho\cdot 10^{-6}$$

$$=\frac{\pi}{4}\times 1200^2\times 25\times 2.7\times 10^{-6}$$

$$=76.3\ \text{kg}$$

(1)-1 Calculating moment of inertia

Moment of inertia of the table

$$I_1=\frac{m_1\cdot R^2}{2}=\frac{76.3\times 0.6^2}{2}=13.7\ \text{kg}\cdot\text{m}^2$$

Moment of inertia of the jigs

$$I_2=m_3\cdot Re^2\cdot 180\times 0.5^2\cdot 45.0\ \text{kg}\cdot\text{m}^2$$

Moment of inertia of the workpieces

$$I_3=m_2\cdot Re^2=60\times 0.5^2=15.0\ \text{kg}\cdot\text{m}^2$$

Moment of inertia of the gear on the table side

$$I_4=\frac{m_4\cdot R^2}{2}=\frac{45.0\times 0.27^2}{2}=1.64\ \text{kg}\cdot\text{m}^2$$

Moment of inertia of the index drives output shaft gear:

$$I_5=\frac{m_5\cdot R^2}{2}=\frac{5.0\times 0.09^2}{2}=0.0203\ \text{kg}\cdot\text{m}^2$$

● Equivalent inertial torque for the output shaft

$$Ie=(I_1+I_2+I_3+I_4)\cdot io^2+I_5$$

$$=(13.7+45.0+15.0+1.64)\times\left(\frac{1}{3}\right)^2$$

$$+0.0203$$

$$=8.39$$

(1)-2 Calculating the max. angular acceleration of the output shaft

$$\alpha=Am\cdot\frac{2\pi}{n}\cdot\left(\frac{360}{\theta h}\cdot\frac{N}{60}\right)^2$$

$$=5.53\times\frac{2\pi}{2}\times\left(\frac{360}{300}\times\frac{33.3}{60}\right)^2$$

$$=7.71\ \text{rad/s}^2$$

● Calculating the equivalent inertial torque for the output shaft

$$Tie=Ie\cdot\alpha=8.39\times 7.71=64.7\ \text{N}\cdot\text{m}$$

(2) Equivalent frictional torque for the output shaft: Tfe

● Calculating the force applied to a sliding surface and bearing

$$Ff=m\cdot g$$

$$=(76.3+180+60+45)\times 9.81$$

$$=3540\ \text{N}$$

● Calculating the frictional torque

$$Tf=\mu\cdot Ff\cdot Rf$$

$$=0.03\times 3540\times 0.5$$

$$=53.1\ \text{N}\cdot\text{m}$$

● Calculating the equivalent frictional torque for the output shaft

$$Tfe=Tf\cdot io$$

$$=53.1\times\frac{1}{3}$$

$$=17.7\ \text{N}\cdot\text{m}$$

(3) Equivalent work torque for the output shaft: Twe

Since no force is required for work, the equivalent work torque Twe is zero.

(4) Equivalent load torque: Tte

$$Tte=Tie+Tfe+Twe$$

$$=64.7+17.7+0$$

$$=82.4\ \text{N}\cdot\text{m}$$

2. Actual load torque: Te

● Determine the usage factors

$$fc=1.8$$

(Refer to Table C.1. Usage factors for various driving methods.)

● Actual load torque

$$Te=Tie\times 1.8+Tfe+Twe$$

$$=134\ \text{N}\cdot\text{m}$$

Example of calculating the table drive (indirect drive)

3. Index drive size selection

You can find the index drives which rated output torque satisfies the actual load torque of Te = 134 N·m to be RGIS110 or greater in the rated output torque table. Provisionally select RGIS110 (Tr = 265 N·m, C = 110) and verify that this satisfies all the conditions.

Check. 1 Service life: Lh

Service life coefficient: fh

$$fh = \frac{Tr}{Te} = \frac{265}{134} = 1.98$$

$$\begin{aligned} Lh &= 10000 \cdot fh^{(10/3)} \\ &= 10000 \times 1.98^{(10/3)} \\ &= 97500 \text{ h} \end{aligned}$$

Hence, this satisfies the expected service life (10000 hours).

Check. 2 Allowable max. diameter of the table: Dm

Table coefficient: ft

$$\begin{aligned} ft &= 2.5 \cdot fh + 2.5 \\ &= 2.5 \times 1.98 + 2.5 \\ &= 7.45 \end{aligned}$$

$$\begin{aligned} Dm &= \frac{C \cdot ft}{io} \\ &= \frac{110 \times 7.45}{\frac{1}{3}} \\ &= 2460 \text{ mm} \end{aligned}$$

Hence, this satisfies Dm ≥ Dt = 1200 mm.

Based on the above, RGIS110-002300S1S1 is chosen.

4. Calculating the input shaft torque: Tc

$$Tc = Tci + Tcw$$

$$Tci = \frac{\Psi}{\theta h} \cdot Qm \cdot Tie$$

$$Tcw = \frac{\Psi}{\theta h} \cdot Vm \cdot (Tfe + Twe) + Tin$$

For the internal frictional torque Tin, refer to the characteristics table in the catalog.

$$\begin{aligned} Tci &= \frac{180}{300} \times 0.99 \times 64.7 \\ &= 38.4 \text{ N·m} \end{aligned}$$

$$\begin{aligned} Tcw &= \frac{180}{300} \times 1.76 \times (17.7 + 0) + 15 \\ &= 33.7 \text{ N·m} \end{aligned}$$

$$\begin{aligned} Tc &= 38.4 + 33.7 \\ &= 72.1 \text{ N·m} \end{aligned}$$

5. Selecting the worm reducer (HO series)

● Determine the reducer usage factors. (Table C.5)

$fr=1.5$

(Continuous operation: Operation exceeding 10 hours per day)

● Calculate a torque Ter of the output shaft of HO reducer.

$$Ter=Tc\cdot fr$$
$$=72.1\times1.5$$
$$=108\ \mathrm{N\cdot m}$$

Verify Ter obtained here to be within the rated torque Trr of the standard combination reducer. The standard HO reducer for RGIS110 is HO60. The reduction ratio required for 33.3 rpm of the input shaft rotational speed is 1/40 for 60 Hz. Worm shaft rotational speed: Nr = 33.3 × 40 = 1332 rpm. Thus, calculate the dynamic rated output torque Trr when the worm shaft rotational speed of HO60 is 1332 rpm.

$Trr=108\ \mathrm{N\cdot m}$

This is acceptable because $Trr \geq Ter$.

Since the operation is intermittent for this specification, the clutch and brake are required. Hence, the reducer HO60-1/40 with C/B is chosen.

6. Calculating the motor capacity: P

When index drive is driven with a worm reducer (HO series), calculate the motor capacity.

● Calculating the actual motor capacity: Pe

To calculate the internal frictional torque of the worm reducer, suppose the oil temperature to be 10°C.

$Tinr=1.1\ \mathrm{N\cdot m}$

(Refer to Table C.6. Internal frictional torque of the HO reducer.)

$$Pr=\frac{Tinr\cdot Nr}{9550}=\frac{1.1\times1332}{9550}=0.15\ \mathrm{kW}$$

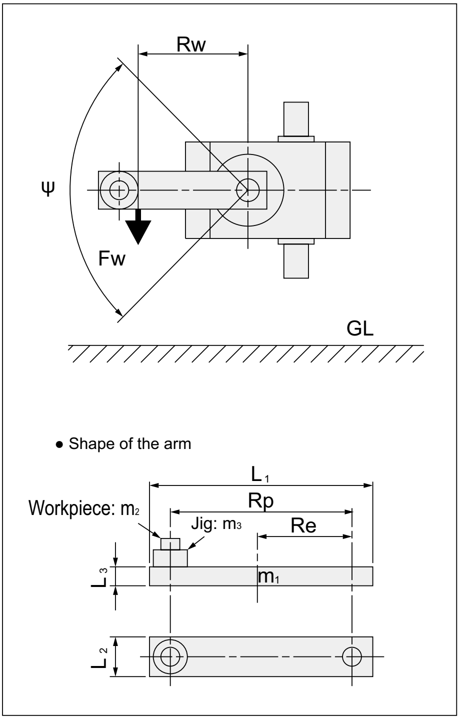
$$Pe=\frac{N}{9550\times\eta}\left(\frac{1}{2}\times Tci+Tcw\right)+Pr$$

$$=\frac{33.3}{9550\times0.6}\left(\frac{1}{2}\times38.4+33.7\right)+0.15$$

$$=0.457\ \mathrm{kW}$$

Hence, the motor whose capacity is 0.457 kW or greater should be chosen.

Example of calculating the oscillator drive



Selection conditions

(1) Model: RGOL

(2) Shape of the output shaft: straight

(3) Oscillating angle: ψ = 90 degrees

(4) Input shaft rotates continuously

(5) Unit cycle time: t_0

$$t_0=1 \text{ s}$$

(6) Total index angle: θt = 120 degrees

(7) Index angle: θh

$$\theta h=\frac{\theta t}{z}=\frac{120}{2}=60°$$

Note: For the oscillation drive of the standard specification, the number of dwell points is 2.

(8) Movement time (indexing time): t_1

$$t_1=1\times\frac{60}{360}=0.167 \text{ s}$$

Note: Time required to oscillate the output shaft for 90 degrees

(9) Input shaft rotational speed

$$N=\frac{60}{1}=60 \text{ rpm}$$

(10) Cam curve: MS modified sign curve

(11) Shape of the arm

Total length of the arm: L_1 = 200 mm

Arm width: L_2 = 40 mm

Arm thickness: L_3 = 20 mm

Weigh of the arm: m_1 = 1.3 kg

Amount of eccentricity of the arm: Re = 70 mm

(12) Weight of workpiece: m_2 = 0.5 kg

(13) Weight of jig: m_3 = 1.5 kg

(14) Workpiece and jig mounting radius: Rp = 150 mm

(15) Suppose there is no load due to friction.

(16) Installation of index drives should be Position #3.

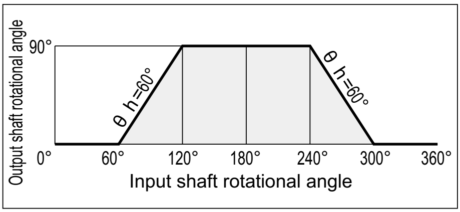
(17) Since the installation is of Position #3, the weights of the arm, the jig and the workpiece all add as eccentric load to the force required for operation. Suppose there is no other force required for operation.

(18) When index drive is driven with a worm reducer.

(19) Orbit pattern should be T.

(20) Expected service life: 10000 hours

(21) Timing chart (standard)



Select the model suitable for the above conditions.

Size selection

1. Calculating the load torque: Tt

(1) Inertial torque: Ti

(1)-1 Calculating moment of inertia

Moment of inertia of the arm

$$I_1=m_1\cdot\left(\frac{L_1^2+L_2^2}{12}+Re^2\right)$$

$$= 1.3 \times \left(\frac{0.2^2+0.04^2}{12}+0.07^2\right)$$

$$= 0.0109\ \mathrm{kg\cdot m^2}$$

Moment of inertia of the workpieces

$I_2=m_2\cdot Rp^2=0.5\times0.15^2=0.0113\ \mathrm{kg\cdot m^2}$

Moment of inertia of the jigs

$I_3=m_3\cdot Rp^2=1.5\times0.15^2=0.0338\ \mathrm{kg\cdot m^2}$

● Total moment of inertia

$I=I_1+I_2+I_3$

$= 0.0109 + 0.0113 + 0.0338$

$= 0.056\ \mathrm{kg\cdot m^2}$

(1)-2 Calculating the max. angular acceleration of the output shaft

$$\alpha=Am\cdot\frac{\pi\cdot\psi}{180}\cdot\left(\frac{360}{\theta h}\cdot\frac{N}{60}\right)^2$$

$$=5.53\times\frac{\pi\times90}{180}\times\left(\frac{360}{60}\times\frac{60}{60}\right)^2$$

$$=313\ \mathrm{rad/s^2}$$

● Calculating the inertial torque

$Ti=I\cdot\alpha$

$=0.056\times313$

$=17.5\ \mathrm{N\cdot m}$

(2) Frictional torque: Tf

Since no frictional load exists, the frictional torque Tf is zero.

(3) Work torque: Tw

As described before, when the orientation is #3, the arm and the jigs become the eccentric load acting as work torque.

● Work torque due to eccentric load on the arm: Tw_1

Force required for work

$Fw_1=1.3\times9.81=12.8\mathrm{N}$

Radius for work

$Re=0.07$

$\therefore Tw_1 = Fw_1\cdot Re = 12.8 \times 0.07$

$= 0.896\ \mathrm{N\cdot m}$

● Work torque due to eccentric load of the jigs and workpieces: Tw_2

Force required for work

$Fw_2=(1.5+0.5)\times9.81$

$=19.6\ \mathrm{N}$

Radius for work

$Rp=0.15\ \mathrm{m}$

$\therefore Tw_2 = Fw_2\cdot Rp = 19.6 \times 0.15$

$= 2.94\ \mathrm{N\cdot m}$

Hence, the total work torque

$Tw=Tw_1+Tw_2=0.896\times2.94$

$=3.84\ \mathrm{N\cdot m}$

(4) Load torque: Tt

$Tt=Ti+Tf+Tw=17.5+0+3.84$

$=21.3\ \mathrm{N\cdot m}$

2. Calculating the actual load torque: Te

● Determine the usage factors

$fc=1.6$

(Refer to Table C.1. Usage factors for various driving methods.)

● Actual load torque

$Te=Ti\cdot fc+Tf+Tw$

$=17.5\times1.6+0+3.84$

$=31.8\ \mathrm{N\cdot m}$

Example of calculating the oscillator drive

3. Index drives size selection

You can find the index drives which rated output torque satisfies the actual load torque of Te = 31.8 N·m to be RGOL140 or greater in the rated output torque table. Provisionally select RGOL140 (Tr = 45.9 N·m, C = 140) and verify that this satisfies all the conditions.

Check. 1 Service life: Lh

Service life coefficient: fh

$$fh = \frac{Tr}{Te} = \frac{45.9}{31.8} = 1.44$$

$$\begin{aligned} Lh &= 10000 \cdot fh^{(10/3)} \\ &= 10000 \times 1.44^{(10/3)} \\ &= 33700 \text{ h} \end{aligned}$$

Hence, this satisfies the expected service life (10000 hours).

Check. 2 Allowable max. diameter of the table: Dm

Table coefficient: ft

$$\begin{aligned} ft &= 1.5 \cdot fh + 1.5 \\ &= 1.5 \times 1.44 + 1.5 \\ &= 3.66 \end{aligned}$$

Max. diameter of the table (assumed)

$$De = L_1 + Re \times 2 = 200 + 70 \times 2 = 340 \text{ mm}$$

$$\begin{aligned} Dm &= C \cdot ft \\ &= 140 \times 3.66 \\ &= 512 \text{ mm} \end{aligned}$$

Hence, this satisfies Dm ≥ De = 340 mm.

Based on the above, RGOL140-090120STS3 is chosen.

4. Calculating the input shaft torque: Tc

$$Tc = Tci + Tcw$$

$$Tci = \frac{\Psi}{\theta h} \cdot Qm \cdot Ti$$

$$Tcw = \frac{\Psi}{\theta h} \cdot Vm \cdot (Tf + Tw) + Tin$$

For the internal frictional torque Tin, refer to the characteristics table in the catalog.

$$\begin{aligned} Tci &= \frac{90}{60} \times 0.99 \times 17.5 \\ &= 26.0 \text{ N·m} \end{aligned}$$

$$\begin{aligned} Tcw &= \frac{90}{60} \times 1.76 \times (0 + 3.84) + 29 \\ &= 39.1 \text{ N·m} \end{aligned}$$

$$\begin{aligned} Tc &= 26.0 + 39.1 \\ &= 65.1 \text{ N·m} \end{aligned}$$

5. Selecting the worm reducer (HO series)

● Determine the reducer usage factors. (Table C.5)

fr=1.5

(Continuous operation: Operation exceeding 10 hours per day)

● Calculate a torque Ter of the output shaft of HO reducer.

$$Ter = Tc \cdot fr = 65.1 \times 1.5 = 97.7\ \text{N·m}$$

Verify Ter obtained here to be within the rated torque Trr of the standard combination reducer. The standard HO reducer for RGOL140 is HO60. The reduction ratio required for 60 rpm of the input shaft rotational speed is 1/20 for 60 Hz. Worm shaft rotational speed: Nr = 60 × 20 = 1200 rpm. Thus, calculate the dynamic rated output torque Trr when the worm shaft rotational speed of HO60 is 1200 rpm.

Trr=105 N·m

This is acceptable because Trr ≥ Ter.

Since the operation is intermittent for this specification, the clutch and brake are required. Hence, the reducer HO60-1/20 with C/B is chosen.

6. Calculating the motor capacity: P

When index drive is driven with a worm reducer (HO series), calculate the motor capacity.

● Calculating the actual motor capacity: Pe

To calculate the internal frictional torque of the worm reducer, suppose the oil temperature to be 10°C.

Tinr=1.1 N·m

(Refer to Table C.6. Internal frictional torque of the HO reducer.)

$$Pr = \frac{Tinr \cdot Nr}{9550} = \frac{1.1 \times 1200}{9550} = 0.14\ \text{kW}$$

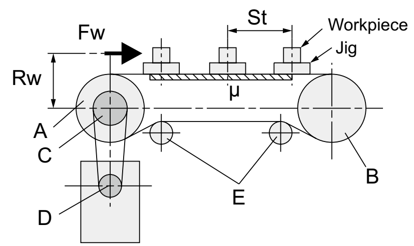
$$Pe = \frac{N}{9550 \times \eta}\left(\frac{1}{2} \times Tci + Tcw\right) + Pr$$

$$Pe = \frac{60}{9550 \times 0.71}\left(\frac{1}{2} \times 26.0 + 39.1\right) + 0.14$$

$$= 0.601\ \text{kW}$$

Hence, the motor which capacity is 0.601 kW or more should be chosen.

Example of calculating the conveyor drive



Selection conditions

(1) Model: PCIS
(2) Input shaft rotates continuously
(3) Input shaft rotational speed: N = 80 rpm
(4) Index angle: θh = 300 degrees
(5) Unit cycle time: t_0

$$t_0=\frac{60}{80}=0.75\text{ s}$$

(6) Movement time (indexing time): t_1

$$t_1=0.75\times\frac{300}{360}=0.625\text{ s}$$

(7) Cam curve: MS modified sign curve
(8) Approximate conveyor feed pitch:
St'=190 mm
(9) Chain size: #40
Chain pitch: Sch = 12.7 mm
Weight per unit: mch = 0.64 kg/m
(10) Number of chain threads: single
(11) Number of conveyor feed teeth: nc

$$\frac{St'}{Sch}=14.96$$

Determine nc = 15
(12) Fixed conveyor feed pitch:
St=Sch·nc=12.7×15
=190.5 mm
(13) Sprocket A (driving), B:
Weight … m_4 = 0.7 × 2 = 1.4 kg
Number of teeth … Za = 30
Pitch circle diameter…Da = 121.50 mm
(14) Sprocket C:
Weight … m_5 = 0.3 kg
Number of teeth … Zc = 20
Pitch circle diameter…Dc = 81.18 mm
(15) Sprocket D:
Weight … m_6 = 0.08 kg
Number of teeth … Zd = 10
Pitch circle diameter…Dd = 41.10 mm
(16) Reduction ratio of the output shaft:

$$io=\frac{Zd}{Zc}=\frac{1}{2}$$

(17) Index number:

$$n=\frac{Za}{nc}\cdot io=\frac{30}{15}\times\frac{1}{2}=1$$

(18) Number of jigs (pallets): np = 20
(19) Number of jigs (pallets):
m_3=0.8kg×20=16 kg
(20) Number of workpieces: nw = 5
(21) Total weight of workpieces:
m_2=0.4kg×5=2 kg
(22) Coefficient of friction: μ = 0.3
(23) Sprocket E (idler):
Weight … m_7 = 0.075 kg
Number of teeth … Ze = 10
Pitch circle diameter…De = 41.10 mm
Quantity … 2
(24) Suppose there is no force required for operation.
(25) When index drive is driven with a worm reducer.
(26) Index drives should be installed in Position #5.
(27) The shaft layout of index drive is S1.
(28) Expected service life: 20000 hours
Select the model suitable for the above conditions.

Size selection

1. Calculating the equivalent load torque: Tte

(1) Equivalent inertial torque for the output shaft: Tie

● Calculating the weight which is a load

Chain weight

$$m_1=St\cdot np\cdot mch\cdot\frac{1}{1000}$$

$$=190.5\times20\times0.64\times\frac{1}{1000}=2.44\ kg$$

(1)-1 Calculating moment of inertia

Moment of inertia of the chain

$$I_1=2.44\times0.061^2$$
$$=0.0091\ kg\cdot m^2$$

Moment of inertia of the sprockets A and B

$$I_4=\frac{1.4\times0.061^2}{2}=0.0026\ kg\cdot m^2$$

Moment of inertia of the sprocket C

$$I_5=\frac{0.3\times0.041^2}{2}=0.0003\ kg\cdot m^2$$

Moment of inertia of the 2 idlers

$$I_7=\frac{0.075\times0.021^2}{2}\times\left(\frac{Za}{Ze}\right)^2\times2$$

$$=\frac{0.075\times0.021^2}{2}\times3^2\times2=0.0003\ kg\cdot m^2$$

Moment of inertia of the jigs (pallets)

$$I_3=16\times0.061^2$$
$$=0.0595\ kg\cdot m^2$$

Moment of inertia of the workpieces

$$I_2=2\times0.061^2=0.0074\ kg\cdot m^2$$

Moment of inertia of the sprocket D

$$I_6=\frac{0.08\times0.021^2}{2}=0.00002\ kg\cdot m^2$$

● Equivalent inertial torque for the output shaft

$$Ie=(I_1+I_2+I_3+I_4+I_5+I_7)\cdot io^2+I_6$$
$$=0.0792\times(1/2)^2+0.00002$$
$$=0.0198\ kg\cdot m^2$$

● Calculating the max. angular acceleration of the output shaft

$$\alpha=Am\cdot\frac{2\pi}{n}\cdot\left(\frac{360}{\theta h}\cdot\frac{N}{60}\right)^2$$

$$=5.53\times\frac{2\pi}{1}\times\left(\frac{360}{300}\times\frac{80}{60}\right)^2$$

$$=88.9\ rad/s^2$$

(1)-2 Calculating the equivalent inertial torque for the output shaft

$$Tie=Ie\cdot\alpha=0.0198\times88.9$$
$$=1.76\ N\cdot m$$

(2) Equivalent frictional torque for the output shaft: Tf

● Calculating the force applied to a sliding surface and bearing

$$Ff=m\cdot g=\left(\frac{m_1+m_3}{2}+m_2\right)\times9.81$$

$$=110\ N$$

● Calculating the frictional torque

$$Tf=\mu\cdot Ff\cdot Rf=0.3\times110\times0.061$$
$$=2.01\ N\cdot m$$

● Calculating the equivalent frictional torque for the output shaft

$$Tfe=Tf\cdot io$$

$$=2.01\times\frac{1}{2}$$

$$=1.01\ N\cdot m$$

(3) Equivalent work torque for the output shaft

Since no force is required for work, the equivalent work torque

Twe will be zero.

(4) Equivalent load torque: Tte

$$Tte=Tie+Tfe+Twe$$
$$=1.76+1.01+0$$
$$=2.77\ N\cdot m$$

2. Actual load torque: Te

● Determine the usage factors

fc=1.9

(Refer to Table C.1. Usage factors for various driving methods.)

● Actual load torque

$$Te=Tie\cdot fc+Tfe+Twe$$
$$=1.76\times1.9+1.01+0$$
$$=4.35\ N\cdot m$$

Example of calculating the conveyor drive

3. Index drive size selection

You can find the index drives which rated output torque satisfies the actual load torque of Te = 4.35 N·m to be PCIS050 or greater in the rated output torque table. Provisionally select PCIS050 (Tr = 7.26 N·m, C = 50) and verify that this satisfies all the conditions.

Check. 1 Service life: Lh

Service life coefficient: fh

$$fh = \frac{Tr}{Te} = \frac{7.26}{4.35} = 1.67$$

$$Lh = 10000 \cdot fh^{(10/3)}$$
$$= 10000 \times 1.67^{(10/3)}$$
$$= 55300 \text{ h}$$

Hence, this satisfies the expected service life (20000 hours).

Based on the above, PCIS050-001300S115 is chosen.

4. Calculating the input shaft torque: Tc

$$Tc = Tci + Tcw$$

$$Tci = \frac{\Psi}{\theta h} \cdot Qm \cdot Tie$$

$$Tcw = \frac{\Psi}{\theta h} \cdot Vm \cdot (Tfe + Twe) + Tin$$

For the internal frictional torque Tin, refer to the characteristics table in the catalog.

$$Tci = \frac{360}{300} \times 0.99 \times 1.76$$
$$= 2.09 \text{ N·m}$$

$$Tcw = \frac{360}{300} \times 1.76 \times (1.01 + 0) + 1.7$$
$$= 3.83 \text{ N·m}$$

$$Tc = 2.09 + 3.83$$
$$= 5.92 \text{ N·m}$$

5. Selecting the worm reducer (HO series)

● Determine the reducer usage factors. (Table C.5)

$fr=1.5$

(Continuous operation: Operation exceeding 10 hours per day)

● Calculate a torque Ter of the output shaft of HO reducer.

$$Ter=Tc \cdot fr$$
$$=5.92\times1.5$$
$$=8.88\ N\cdot m$$

Verify Ter obtained here to be within the rated torque Trr of the standard combination reducer. The standard HO reducer for PCIS050 is HO32. The reduction ratio required for 80 rpm of the input shaft rotational speed is 1/20 for 60 Hz. Worm shaft rotational speed: Nr = 80 × 20 = 1600 rpm. Thus, calculate the dynamic rated output torque Trr when the worm shaft rotational speed of HO32 is 1600 rpm.

$Trr=21\ N\cdot m$

This is acceptable because Trr ≥ Ter.

Since the operation is continuous for this specification, the clutch and brake are not required.

Hence, the reducer HO32-1/20 without C/B is chosen.

6. Calculating the motor capacity

When index drive is driven with a worm reducer (HO series), calculate the motor capacity.

● Calculating the actual motor capacity: Pe

To calculate the internal frictional torque of the worm reducer, suppose the oil temperature to be 10°C.

$Tinr=0.24\ N\cdot m$

(Refer to Table C.6. Internal frictional torque of the HO reducer.)

$$Pr=\frac{Tinr\cdot Nr}{9550}=\frac{0.24\times1600}{9550}$$
$$=0.0402\ kW$$

$$Pe=\frac{N}{9550\times\eta}\left(\frac{1}{2}\times Tci+Tcw\right)+Pr$$

$$=\frac{80}{9550\times0.69}\left(\frac{1}{2}\times2.09+3.83\right)+0.0402$$

$$=0.099\ kW$$

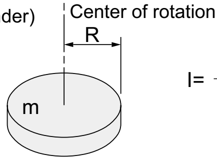
Hence, a motor whose capacity is 0.099 kW or greater should be chosen.

Moment of inertia formulas

[m: Weight of body (kg)]

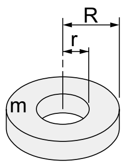
● A: when the rotational center is its shaft

1. Disc (Cylinder)



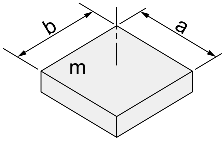
$$I=\frac{mR^2}{2}$$

2. Hollow disc (Hollow cylinder)



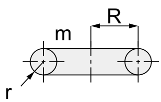
$$I=\frac{m\,(R^2+r^2)}{2}$$

3. Direct hexagonal side finish body



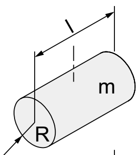
$$I=\frac{m\,(a^2+b^2)}{12}$$

4. Ring



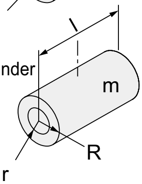
$$I=\frac{m\,(4R^2+3r^2)}{4}$$

5. Cylinder



$$I=\frac{m\,(3R^2+l^2)}{12}$$

6. Hollow cylinder

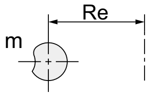


$$I=\frac{m\,(R^2+r^2+l^2/3)}{4}$$

● B: when the rotational center is not its shaft

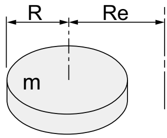
1. Any shape (as long as sufficiently compact)

Center of rotation



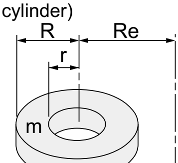
$$I=mRe^2$$

2. Disc (Cylinder)



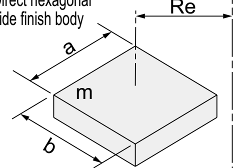
$$I=m\left(\frac{R^2}{2}+Re^2\right)$$

3. Hollow disc
(Hollow cylinder)



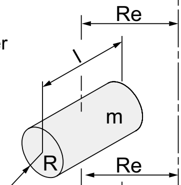
$$I=m\left(\frac{R^2+r^2}{2}+Re^2\right)$$

4. Direct hexagonal side finish body



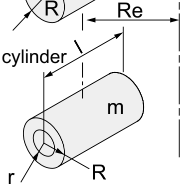
$$I=m\left(\frac{a^2+b^2}{12}+Re^2\right)$$

5. Cylinder



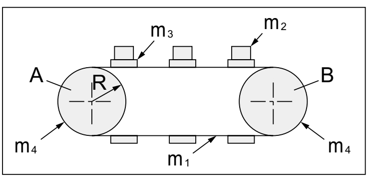
$$I=m\left(\frac{3R^2+l^2}{12}+Re^2\right)$$

6. Hollow cylinder



$$I=m\left(\frac{R^2+r^2+l^2/3}{4}+Re^2\right)$$

● Conveyor



m_1: Weight of chain
m_2: Total weight of workpieces
m_3: Total weight of jigs (pallets)
m_4: Total weight of Sprocket A (driving) + B
R: Driving sprocket radius

$$I=\left(m_1+m_2+m_3+\frac{m_4}{2}\right)\cdot R^2$$

List of symbols and formulas

Symbol	Name	Unit	Formula
A	Non-dimensional acceleration		
Am	Non-dimensional max. acceleration		
C	Shaft interval of the index drives	mm	
De	Max. diameter of the table	mm	
Dm	Allowable max. diameter of the table	mm	
Dt	Outside diameter of the table	mm	
Ff	Force applied to a sliding surface and bearing	N	Ff=m·g
Fw	Force required for work	N	
fa	Acceleration coefficient		
fc	Usage factor of the index drives		Refer to Table C.1.
fh	Service life coefficient		Refer to Table C.2.
ft	Table coefficient		Refer to Table C.3.
fr	Reducer usage factor Usage factor of the geared motor		Refer to Table C.5. Refer to Table C.7.
g	Gravitational acceleration	m/s^2	$1g=9.81m/s^2$
id	Reduction ratio (Reducer series)		
io	Reduction ratio of the output shaft		
ir	Reduction ratio		
I	Moment of inertia	$kg \cdot m^2$	Refer to the moment of inertia formulas
Ic	Moment of inertia of the input shaft	$kg \cdot m^2$	
Ie	Equivalent moment of inertia for the output shaft [For indirect drive]	$kg \cdot m^2$	$Ie=I \cdot io^2$
J	Non-dimensional jerk		
JMc	Converted moment of inertia for the motor shaft	$kg \cdot m^2$	
JM	Moment of inertia of the motor	$kg \cdot m^2$	
Jm	Non-dimensional max. jerk		
Lh	Service life	h	$Lh=10000 \cdot fh^{(10/3)}$
m	Weight	kg	
mch	Weight of chain per unit length	kg/m	Refer to the catalogs of chain manufacturers.
N	Rotational speed of input shaft	rpm	● Index: $N=\frac{60}{t_0 \cdot z}$ ● Oscillator: $N=\frac{60}{t_0}$
Nms	Motor speed	rpm	
Nr	Rotational speed of worm shaft	rpm	
Nrs	Rotational speed of output shaft	rpm	
n	index number		n=ns·io [For indirect drive]
nc	Number of conveyor feed teeth		
np	Number of jigs (pallets)		
ns	Number of stations		

List of symbols and formulas

Symbol	Name	Unit	Formula
n_w	Number of workpieces		
P	Motor capacity	kW	$P=\frac{Tc\cdot N}{9550\cdot\eta}$
Pe	Actual motor capacity	kW	●Worm reducer drive $Pe=\frac{N}{9550\cdot\eta}(\frac{1}{2}\cdot Tci+Tcw)+Pr$ ●Geared motor drive $Pe=\frac{N}{9550\cdot\eta}(Tci+Tcw)$
Pinv	Required capacity of the inverter	kW	
Pm	Motor output	kW	
Pr	Worm reducer unit itself Motor capacity		$Pr=\frac{Tinr\cdot Nr}{9550}$
Qm	Torque coefficient		Refer to Table C.4.
Rf	Average frictional radius	m	
Rw	Radius for work	m	
S	Non-dimensional displacement		
Sch	Chain pitch	mm	Refer to the catalogs of chain manufacturers.
St	Conveyor feed pitch	mm	
T	Non-dimensional time		
Taj	Release (Trip) torque	N·m	Taj=Te×1.3
Tc	Input shaft torque	N·m	Tc=Tci+Tcw
Tci	Input shaft torque due to inertia load	N·m	$Tci=\frac{\psi}{\theta h}\cdot Qm\cdot Ti$
Tcw	Input shaft torque due to frictional load	N·m	$Tcw=\frac{\psi}{\theta h}\cdot Vm\cdot(Tf+Tw)+Tin$
td	Motor control time	s	
Te	Actual load torque	N·m	Te=Ti·fc+Tf+Tw
Ter	Load torque of the reducer Geared motor load torque	N·m	Ter=Tc·fr
Tf	Frictional torque	N·m	Tf=μ·Ff·Rf
Tfe	Equivalent frictional torque for the output shaft (for indirect drive)	N·m	Tfe=Tf·io
Ti	Inertial torque	N·m	Ti=I·α
Tie	Equivalent inertial torque for the output shaft (for indirect drive)	N·m	Tie=Ie·α
Tin	Internal frictional torque of the index drives	N·m	Refer to the characteristics table of each model.
Tinr	Internal frictional torque of the reducer	N·m	Refer to Table C.6.
Tr	Dynamic rated output torque of the index drives	N·m	
Trr	Rated output torque of the reducer Allowable torque of the geared motor output shaft	N·m	
Tt	Load torque	N·m	Tt=Ti+Tf+Tw
Tte	Equivalent load torque for the output shaft	N·m	Tte=Tie+Tf+Tw
Tw	Work torque	N·m	Tw=Fw·Rw
Twe	Equivalent work torque for the output shaft (for indirect drive)	N·m	Twe=Tw·io

Symbol	Name	Unit	Formula
t_0	Unit cycle time	S	● Index: $t_0=t_1+t_2=\frac{60}{N \cdot z}$ ● Oscillator: $t_0=2\cdot(t_1+t_2)=\frac{60}{N}$
t_1	Indexing time	S	$t_1=\frac{\theta h}{360}\cdot\frac{60}{N}$
t_2	Dwell time	S	
ts	Stop time	S	
tm	Machine cycle time	S	$tm=t_0+ts$
V	Non-dimensional speed		
Vm	Non-dimensional max. speed		
Zp	Number of conveyor feed teeth		$Zp=\frac{St}{Sch}$
z	Dwell No.		
α	Max. angular acceleration	rad/s^2	● Index $\alpha=Am\cdot\frac{2\pi}{n}\cdot\left(\frac{360}{\theta h}\cdot\frac{N}{60}\right)^2$ ● Oscillator $\alpha=Am\cdot\frac{\pi\cdot\psi}{180}\cdot\left(\frac{360}{\theta h}\cdot\frac{N}{60}\right)^2$
η	Efficiency of the reducer		Refer to "Section D: Reducer Specifications."
θh	Index angle	°(degrees)	$\theta h=\frac{\theta t}{z}$ (Applicable only to the standard specification)
θt	Total index angle	°(degrees)	
μ	Coefficient of friction		
ψ	Oscillating angle	°(degrees)	$\psi=\frac{360}{n}$

● How to read Greek letters

Upper case	Lower case	How to read	Normal applications	Upper case	Lower case	How to read	Normal applications
Α	α	alpha	Angle, coefficient	Ο	ο	omicron	
Β	β	beta	Angle, coefficient	Π	π	pi	Pi (3.14159...) Angle, (upper case) symbol for product
Γ	γ	gamma	Angle, weight per unit area (upper case) relation				
Δ	δ	delta	Compact changes, density, displacement	Ρ	ρ	rho	Radius, density
Ε	ε	epsilon	Compact amount, strain	Σ	σ	sigma	Stress, standard deviation, (upper case) sum
Ζ	ζ	zeta	Variable	Τ	τ	tau	Time constant, time, torque
Η	η	eta	Variable				
Θ	θ	theta	Angle, temperature, time	Υ	υ	epsilon	
Ι	ι	iota		Φ	φ	phi	Angle, function, diameter
Κ	κ	kappa	Radius of gyration	Χ	χ	khi	
Λ	λ	lambda	Wave length, eigenvalue	Ψ	ψ	psi	Angle, relation
Μ	μ	mu	Coefficient of friction 10^{-6} (micro)	Ω	ω	omega	Angular speed = 2πf (Upper case) Ω = Electrical resistance unit
Ν	ν	nu	Frequency				
Ξ	ξ	xi	Variable				

List of symbols and formulas of the index and table (direct/indirect) drives

Selected model: ☐RGIS ☐RGIT ☐RGIL ☐RGIB ☐Other [ ]
Shape of the output shaft: ☐S (straight shaft) ☐F (flange shaft)
Material of housing: ☐ Fc ☐ Aℓ
Installation position: ☐1 ☐2 ☐3 ☐4 ☐5 ☐6

■ Operational conditions

1. Index number: n = [ ]

2. For the indirect drive of the table

• Output shaft reduction ratio: $io = \frac{D_1}{D_2} =$ [ ] • Number of stations: $ns = \frac{n}{io} =$ [ ]

3. Total index angle (θt) = index angle (θh) × dwell No. (z)

[ ] °= [ ] °× [ ]

4. Cycle time

☐For continuous drive	Unit Cycle time (t_0) = indexing time (t_1) + dwell time (t_2) [ ] seconds = [ ] seconds + [ ] seconds
☐For intermittent drive	Machine Unit Cycle time (tm) = cycle time (t_0) + stopping time (ts) [ ] seconds = [ ] seconds + [ ] seconds *ts: Time when the input shaft is stopped because of C/B

5. Input shaft rotational speed: N

$$N = \frac{60}{\text{Unit cycle time } (t_0) \times \text{Dwell No. } (z)} = \boxed{\quad} \text{ rpm}$$

6. Cam curve: ☐ MS (standard) ☐ MC ☐ MT ☐ Other ( )

7. Input shaft driving method
☐ Direct worm (1) [HO, TE, etc. installed]
☐ Direct worm (2) [connected via coupling]
☐ Indirect worm [Chain & belt drive]
☐ Gear motor

8. Expected service life
[ ] h
Unless otherwise specified, this will be calculated as 10000 hours.

CKD
TO: ( )

Company name		Name	
Department/ Section			
TEL		FAX	

[Options]

☐ Worm reducer: HO, CRG, TE series

• C/B ☐ Yes ☐ No • Reduction ratio 1/ • Mounting direction

☐ Overload protection unit: TSF, TST, TGX —

■ Load conditions

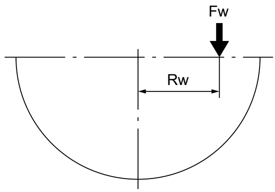
1. Table

• Diameter: Dt = ø mm

• Thickness: ht = mm

• Material: ☐ Steel ☐ Aluminum

☐ Other (Material: Material density:)



2. Workpieces and jigs

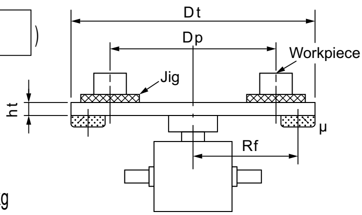
• Number of workpieces: nw =

• Total weight of workpieces: m_2 = × nw = kg

• Number of jigs: np =

• Total weight of jigs: m_3 = × np = kg

• Mounting center diameter of workpieces and jigs: Dp = P.C.D mm

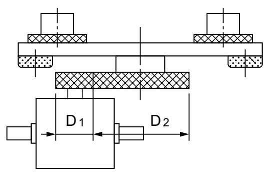


■ Direct drive

3. Specification of the lower surface of the table (Yes / No)

• Frictional radius: Rf = mm

• Coefficient of friction: μ =



■ Indirect drive

	Index gear	Gear on the table side
P.C.D	D_1=	D_2=
Tooth width		

4. External load during indexing (when the output shaft is operating)

• External load: Fw = N or kgf

• Action radius Rw = mm

■ For a case other than the above conditions, contact our sales representative.

List of symbols and formulas of the index and conveyor drives

Selected model: ☐PCIS ☐RGIS ☐RGIL ☐Other [ ]

Shape of the output shaft: ☐S (straight shaft) ☐F (flange shaft)

Material of housing: ☐ Fc ☐ Aℓ

Installation position: ☐1 ☐2 ☐3 ☐4 ☐5 ☐6

■ Operational conditions

1. Index number: n = [ ]

2. Reduction ratio of the output shaft: $io = \frac{Zd}{Zc} =$ [ ]

3. Total index angle (θt) = index angle (θh) × dwell No. (z)

[ ] ° = [ ] ° × [ ] *Set Z=2 for 6,8 index of PCIS

4. Cycle time

☐ For continuous drive	Unit Cycle time (t_0) = indexing time (t_1) + dwell time (t_2) [ ] seconds = [ ] seconds + [ ] seconds
☐ For intermittent drive	Machine Unit Cycle time (tm) = cycle time (t_0) + stopping time (t_s) [ ] seconds = [ ] seconds + [ ] seconds *ts: Time when the input shaft is stopped because of C/B

5. Input shaft rotational speed: N

$$N = \frac{60}{\text{Unit cycle time } (t_0) \times \text{Dwell No. } (z)} =$$ [ ] rpm

6. Cam curve: ☐ MS (standard) ☐ MC ☐ MT ☐ Other ( )

7. Input shaft driving method
- ☐ Direct worm (1) [HO, TE, etc. installed]
- ☐ Direct worm (2) [connected via coupling]
- ☐ Indirect worm [Chain & belt drive]
- ☐ Gear motor

8. Expected service life

[ ] h

Unless otherwise specified, this will be calculated as 10000 hours.

C K D TO: ( )	Company name		Name	
	Department/ Section			
	T E L		F A X	

[Options]

☐ Worm reducer: HO, CRG, TE series

• C/B ☐ Yes ☐ No • Reduction ratio 1/ • Mounting direction

☐ Overload protection unit: TSF, TGX —

■ Load conditions

1. Conveyor feed pitch: St = mm

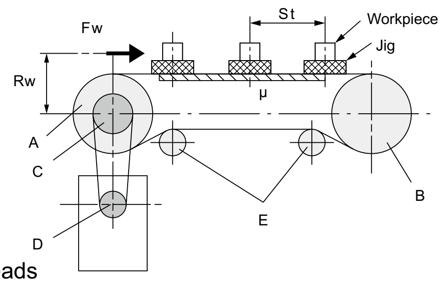
2. Chain size

• Chain model number

• Chain pitch: Sch = mm

• Weight per unit: mch =

3. Number of chain threads: NB = threads



4. Sprocket

	A (drive)	B	C	D	E
Number of teeth (Z)					
Pitch circle diameter (D)					
Tooth width (h)					
Number per thread	1	1			

5. Workpieces and jigs

• Number of workpieces: nw = • Total weight of workpieces: m_2 = × nw = kg

• Number of jigs: np = • Total weight of jigs: m_3 = × np = kg

• Coefficient of friction: μ =

6. External load during indexing (when the output shaft is operating)

• External load: Fw = N or kgf • Action radius: Rw = mm

■ For cases other than the above conditions, contact our sales representative.