# Engineering

INDUSTRIAL MOTION CONTROL, LLC.

CAMCO FERGUSON

## Foreword

Cam-actuated motion control is a specialized business. In a 4 to 5-year university curriculum for mechanical engineering, most students spend only a few weeks studying cams and their related mechanisms. In addition to continuing academic research, many advances in cam technology have been made by companies and employees involved in the commercial application of these products. This engineering section provides the basic concepts necessary for machine designers to wisely choose the best cam solutions for their application.

There are some good publications available to the general public for those seeking a more in-depth understanding of the subject. Three that we can recommend are Clyde H. Moon's "Cam Design Manual for Engineers, Designers, and Draftsmen", published by Emerson Electric Co., Harold A. Rothbart's book, "Cam Design Handbook" published by McGraw-Hill and Robert L. Norton's book "Cam Design and Manufacturing Handbook" published by Industrial Press. Mr. Moon's book is available in Adobe Acrobat® PDF format on the IMC web site and can be easily downloaded at **www.camcoindex.com**.

We would like to thank all of the IMC employees and IMC manufacturer representatives that have contributed to our extensive cam knowledge base and helped collect the information presented in this catalog.

## Introduction

Industrial Motion Control, LLC is a joint-venture company formed in 2001 between Ferguson Machine Co. and Commercial Cam Co., also known as Camco.

Ferguson has been in continuous operation since 1930, with European operations established in 1961. Camco was established in 1939, first manufacturing the copper coils required for the then-emerging residential and commercial air-conditioning and refrigeration industries. Camco needed cam-actuated machinery to produce these products and eventually the business focused on the commercialization of cam-operated machinery, index drives and custom cams.

As divisions of larger, Fortune 500 companies, both Ferguson and Camco were able to invest in substantial amounts of equipment and facilities while developing a diverse line of products that include **index drives, custom cams, parts handlers, precision-link conveyors and servo-motor drive systems**.

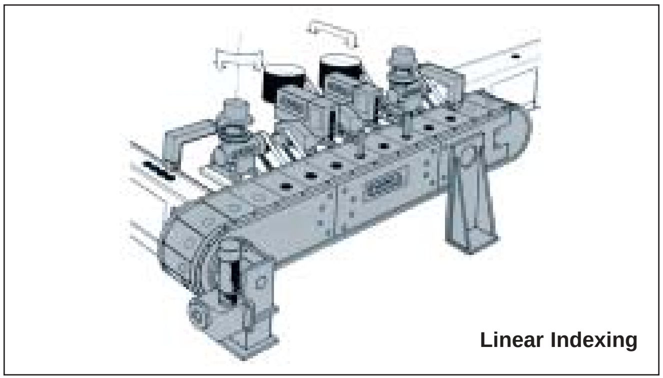
Today, as IMC, Ferguson and Camco are the world's largest producer of cam-actuated index drives, utilizing state-of-the-art production equipment to provide the highest quality cam-actuated and servo motor-actuated motion control products available.

## Solutions in Motion™

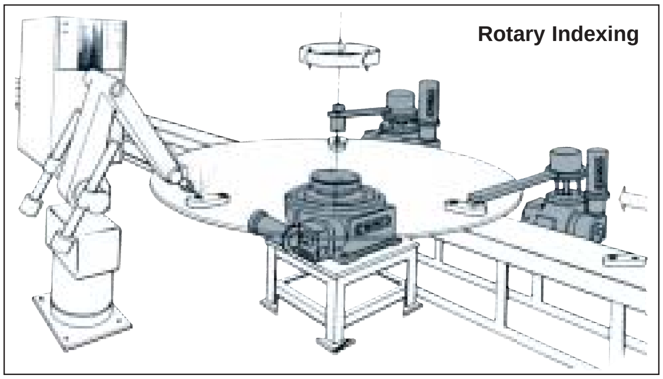
Choosing the proper type and size of index drive can be complicated. Over the years, IMC has developed a wealth of experience in selecting and applying the optimum product solutions to a wide variety of applications. IMC application engineers, sales engineers, catalog information and proprietary software all combine to make the task less daunting. This engineering section will help eliminate much of the mystery behind high-performance indexing and its application in specialty machinery design.

## What is Indexing?

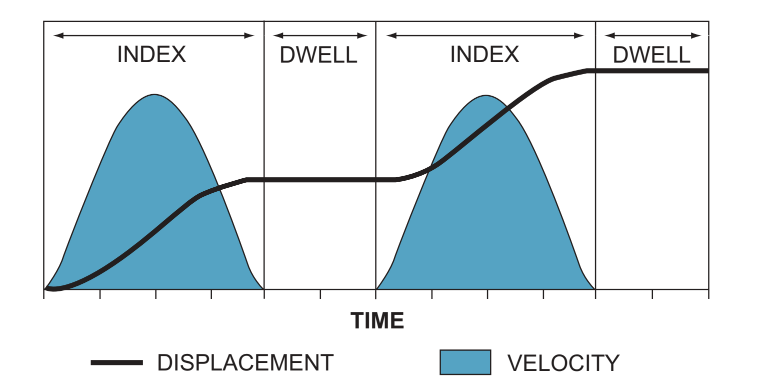
Indexing can be linear or rotary. As defined by IMC, indexing is the process of starting and stopping in precise intervals at precise locations.



Linear Indexing



Rotary Indexing




## Why Cam-Actuated Index Drives?

The advantages of cam-controlled motion are obvious and effectively demonstrated in everyday life by the camshaft found in automobile engines. No other technology can provide comparable **speed, precision, repeatability, load capability and reliability**.

Cam-driven mechanisms require little or no maintenance and are capable of moving, with precision, a wide variety of products and components. For example – larger E-Series Index Drives rotate *several tons* of automotive body parts in seconds – and smaller P-Series and RG-Series index drives accurately index pharmaceutical components and electronic components in *milliseconds*. The mechanical technology typically requires no maintenance, other than routine checks for proper lubrication. Rolling pre-loaded contact between the cams and cam followers minimize wear and thermal inefficiencies. This preloading technique is also used on the input and output bearings of the index drive, achieving the most rigid, accurate and efficient mechanical actuator possible. With this inherent rigidity, settling time (the time to dampen any vibrations) in dwell is short or virtually non-existent — very important for many applications requiring a combination of speed and precise positioning.

Through careful design of the cam profile, velocity and acceleration are also controlled throughout the indexing cycle, minimizing vibration and providing a known, repeatable displacement-time relationship.

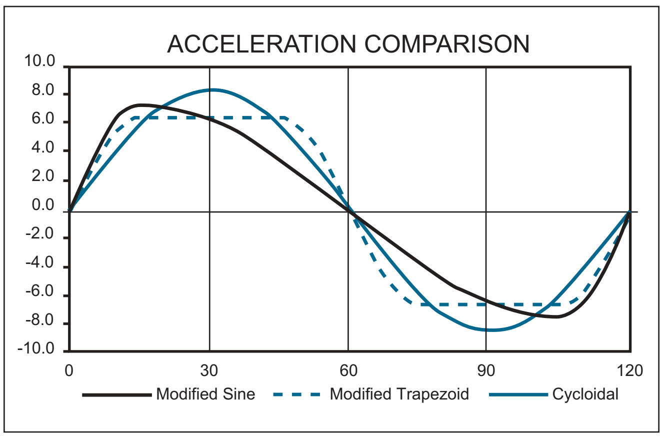
In summary, cam-operated indexing systems have the following features and benefits:

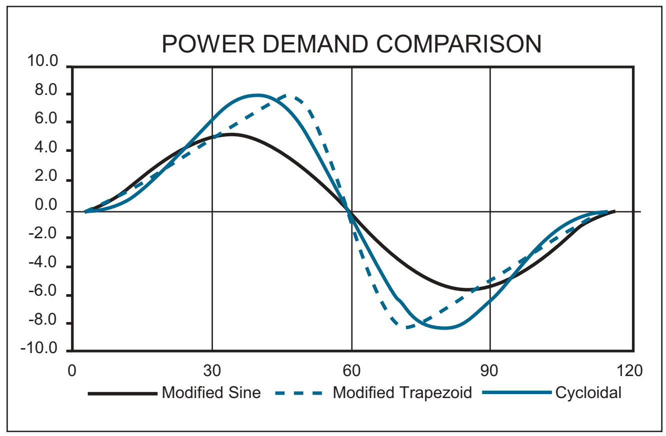
- Controlled Acceleration and Decelerations
- Repeatable, Accurate Positioning
- High Load Capacity
- High Speed Capability
- Smooth Motion
- Quick Settling Time in the Dwell Position
- Low Maintenance, Superior Life
- Known Displacement-Time Relationship
- Known Power Requirement

# Types of Motions

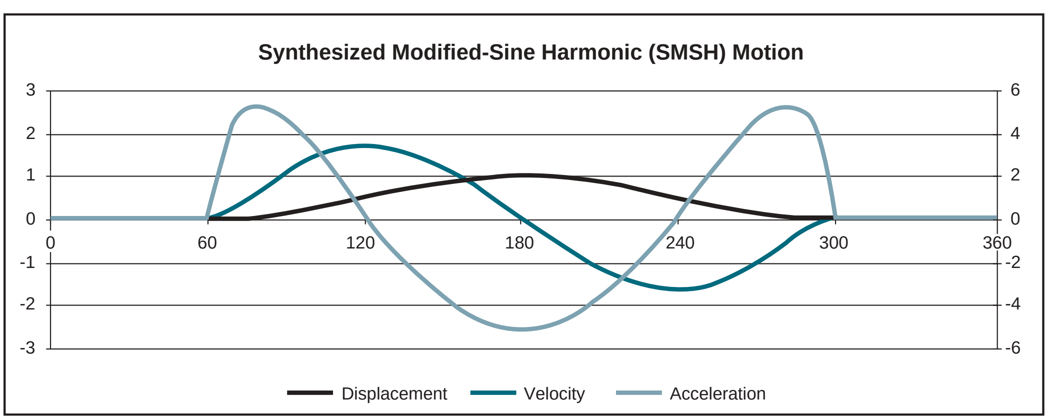
Controlled Indexing is comprised of three sections or phases: **acceleration, peak velocity** and **deceleration**. To optimize the transition from one phase to the next, several standard motion profiles have been developed. They include **Cycloidal, Modified Sine** and **Modified Trapezoidal**. In special circumstances, the motion required calls out for certain positions and/or velocities at certain times in the index cycle. Special **Polynomial** curves can be constructed for these applications. In other applications, the peak velocity needs to match the velocity of another component of the machine – and variations of Polynomial and Modified Sine curves can be customized to suit the requirements.

**IMC usually employs Modified Sine curves** due to their smooth transition from peak acceleration to deceleration and smooth power demand curves. Frequently, a period of peak, constant velocity is needed due to cam design or machine design requirements and a variation of this motion curve, **Modified Sine Constant Velocity** (abbreviated “msc”), is used.



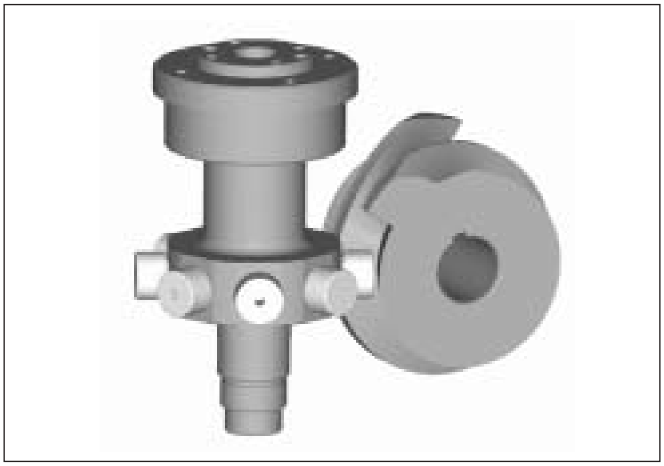

In addition to those motions already described, IMC also has several other special application motions. They include **Modified Sine Quick Return (MSQR)** and **Synthesized Modified Sine Harmonic (SMSH)**. **MSQR** is an oscillating motion with no dwells. It has a forward stroke with a matched peak velocity and a quick return stroke. It is used in applications where a constant speed conveyor or rotating dial is tracked (velocity is synchronized) in order to perform work during the synchronized movement. Examples are printing or moving a saw or cutting blade to cut parts to size. **SMSH** is a motion used in oscillating applications that require a dwell at one end of the stroke and no dwell at the other. This motion reduces the number of acceleration reversals. Please contact your local IMC sales representative or IMC application engineer for further details.



# Types of Index Drives

IMC manufactures all three types of index drive geometries: **Roller Gear, Right Angle, and Parallel.**

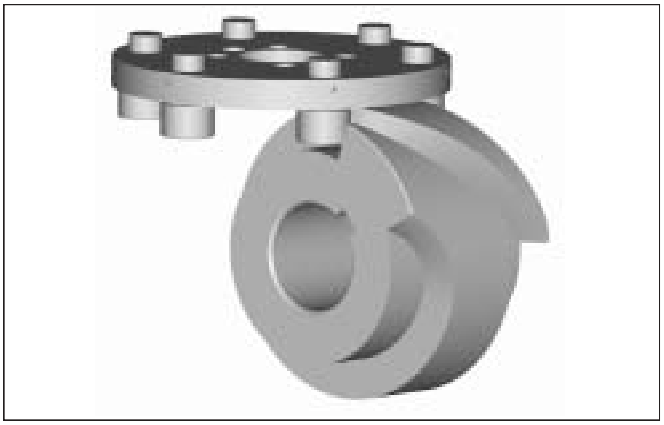
## Roller Gear



This family of indexers uses a globoidal cam in conjunction with followers mounted radially outward from the circumference of the follower wheel, much like the teeth of a gear. The input shaft is perpendicular to the output shaft. With this right angle configuration, it is possible to provide an optional large through-hole along the axis of the output shaft, or design a large output flange to accept dials (dial mounting). Large cam diameters relative to the output follower wheel allow for a wide rande of special motions, short motion periods and a large output displacement for relatively smaller input displacement. In summary, **Roller Gear** Indexers provide:

- Compact Low Profile Design
- Flanged Output Capability for Dial Mounting Applications
- Through-Hole Capability (for electric and pneumatic lines or stationary center post)
- Motion Flexibility (special and complex motions) due to relatively large cam
- 2 to 24 Stop Range

## Right Angle



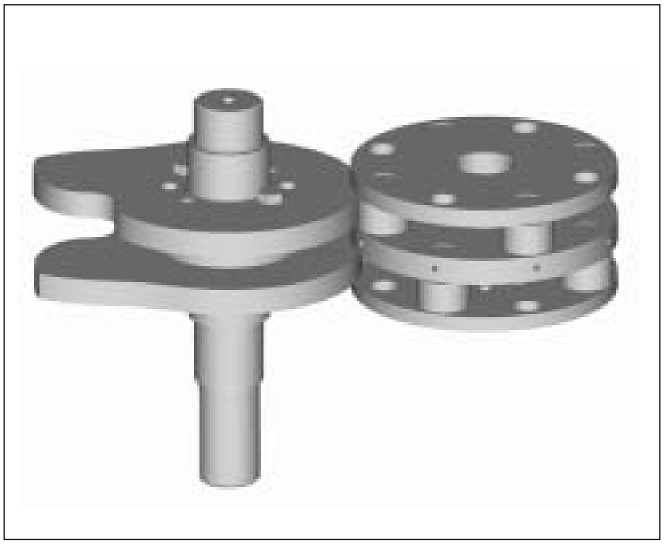
This family of indexers uses a cylindrical or barrel cam in conjunction with followers mounted parallel to the axis of the output. Similar to the Roller Gear, the input shaft is perpendicular to the output shaft. The cam is tucked partially underneath the output wheel, offering a more compact arrangement. For a given torque requirement, Right Angle indexers usually occupy the least amount of floor space and volume. IMC production equipment allows us to produce very large index drives in this geometry. Control of the cam rib thickness allows for preloading. Center distances between input shaft and output shaft can be fixed accurately. The minimum cam rib requirements limit the range of motions (output motions as a function of input motion) when compared to Roller Gear indexers. In summary, **Right Angle** Indexers provide:

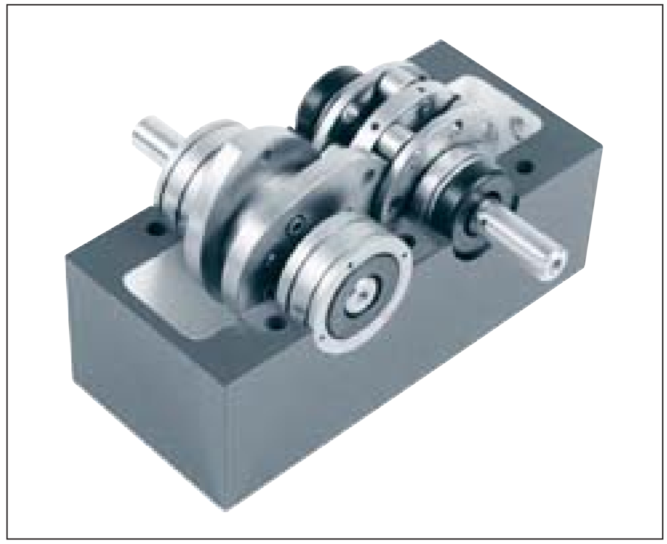
- Most Compact Design for Given Output Capacity
- Fixed Center Distance Between Output and Input Shafts (tighter tolerance on the distance between input and output shafts)
- Flanged Output Capability for Dial Mounting Applications (E-Series & RAD Series)
- Through-Hole Capability (E-Series & RAD series)
- 3 to 24 Stop Range
- Very Large Index Drives for Automotive Assembly and Large (up to 40 feet) Dial Diameters

## Types of Index Drives

(continued)

### Parallel





This family of indexers use a pair of conjugate plate cams with yoke-mounted followers mounted parallel to the axis of the output. The input shaft is parallel to the output shaft. With this parallel configuration, there are no ribs on the cam as found on Roller Gear and Right Angle indexers. Also unique to the Parallel family is no reversal of the cam followers. Since they rotate in the same direction throughout the index cycle, index rates of over 1000 indexes per minute are possible. Without minimum rib requirements (no rib), larger followers can be used, providing high torque capability. Parallel indexers produce high output displacements for relatively smaller input displacements. The yoke-mounted geometry also makes the Parallel family more resistant to shock loading (more robust). Double output shafts are also available. In summary, **Parallel** Indexers provide:

- High Speed Capability (with Non-Reversing Followers)
- High Load Capability (with Oversized Followers)
- Shock Resistance (More Robust)
- Motion Flexibility (special and complex motions) due to conjugate cam geometry
- 1 to 8 Stop Range

## Model Code Designation

**Camco Model Code Designation**

Number of Stops

**663RAD 6 H36-180**

Model

Cam Follower diameter in 32nds of an inch (H36 = 1.125")

Index Period

**For Indexers Only**

**Ferguson Model Code Designation**

Model

Index Period

**FD162F 8/2 F100-90**

Number of Stops (8/2 = 4 stops)

Cam Follower diameter in 100ths of an inch (F100 = 1")

**For Indexers Only**

# Approach to Sizing Index Drives

Within each family type (**Roller Gear, Right Angle, Parallel**), IMC offers more than a dozen different sizes of index drives. The first consideration when choosing an **index drive type** is mounting requirements and the geometry of the driven member. The mounting requirements usually determine the type of indexer and then size is selected. Often the geometry (size of dial, for example) helps determine the initial choice. The **index drive size** is verified through data sheet calculations.

All IMC indexers are designed and rated to have a $\mathbf{B_{10}}$ life of 8,000 hours on the followers and over 100,000 hours on the other major components. The $\mathbf{B_{10}}$ life is an estimate of time between cam follower replacement. For example, a $\mathbf{B_{10}}$ life estimate of 15,000 hours means that we can expect 10% of the followers to begin to show wear after 15,000 hours of operation. For this case, IMC would recommend replacing all of the followers after 15,000 hours of continuous operation.

Many helpful software programs have been developed by IMC to assist with the selection process. The following examples will show both a manual method of calculating and a faster method using special software.

All sizing for rotating equipment (motors, gear reducers and indexers) rely on the basic Newtonian Mechanics equation:

$$\text{Torque} = T_i = I\ \alpha$$

Where **I** is the Rotational Mass Moment of Inertia and $\alpha$ is the peak angular acceleration (radians/sec$^2$).

Additional work or friction torque is also added, giving the full equation:

$$T_{Total} = I\ \alpha + T_w$$

Where $T_w$ = Work Torque = $\mu$ x R x F

$\mu$ = coefficient of friction, **R** = radius to Work Force and **F** = Force

For smaller diameter dial applications, Work Torque is negligible. For larger diameter dial applications, Work Torque can be significant. The inefficiencies of speed reducers also add to the total Work Torque.

After Torque is calculated we then determine the power requirements through:

$$\text{Power} = T \times \omega = I \times \alpha \times \omega$$

Where $\boldsymbol{\omega}$ is the rotational velocity (radians/sec). Note that with an indexing application, $\boldsymbol{\alpha}$ and $\boldsymbol{\omega}$ are a function of time or $\boldsymbol{\alpha}$ = f(t) and $\boldsymbol{\omega}$ = f(t).

Since **I** is usually constant, power peaks when the product of $\boldsymbol{\alpha}$ and $\boldsymbol{\omega}$ peak. Software automatically chooses this peak product, and the manual data sheet methods rely on $\mathbf{K_i}$ and $\mathbf{K_f}$ factors to determine peak power. $\mathbf{K_i}$ and $\mathbf{K_f}$ are explained later in this Engineering catalog section.

Input (camshaft) torque requirements are calculated through the conservation of energy equation, Power in = Power out, or:

$$T_{in} \times \omega_{in} = T_{out} \times \omega_{out}$$

$$\text{Restated, } T_{in} = T_{camshaft} = T_c = T_{out} \times \omega_{out} / \omega_{in}$$

Note that:

$K_i \equiv \omega_{out} / \omega_{in}$ at peak value of the product of $\alpha_{out} \times \omega_{out}$ so we have:

$$T_{c(inertia)} = T_{inertia\ out} \times K_i \text{ (for inertia)}$$

Similarly,

$$T_{c\ (work)} = (T_{work\ out} + T_{friction\ out}) \times K_f$$

(for friction and work torque)

Where $K_f \equiv \omega_{out} / \omega_{in}$ at $\omega_{out}$ (maximum).

Total Camshaft Torque

$$T_c = T_{c(inertia)} + T_{c(work)}$$

Horsepower is calculated based on Camshaft Torque and Speed

$$\text{Power} = \text{HP} = \frac{T_c \times N}{63025 \times E} \text{ (Horsepower)}$$

Where **N** = Camshaft speed in RPM

**E** = Efficiency of the gear reducer

$\mathbf{T_c}$ is in units of in-lbs.

# Derivation of Torque Demand Equation for Indexing Dials

Inertia Torque, $T_i$, is defined by:

$$T_i = I \alpha$$

Where **I** = Rotational Mass Moment of Inertia (in-lb-sec$^2$)

**α** = Peak angular acceleration (radians/sec$^2$)

From the "Cam Design" manual by Mr. Clyde H. Moon:

$$\alpha = C_a \frac{\theta_O}{t_2^2}$$

Where $C_a$ = Acceleration Coefficient (5.528 for modified sine motion)

$\theta_O$ = Output Angle or Angle of Index (radians)

$t_2$ = Index time (seconds)

The Output Angle, θ, is calculated based on the number of stops

$$\theta_O = \frac{2\pi}{S}$$

Where **S** = Number of stops

If the modified sine motion has constant velocity the acceleration factor, $C_a$, must be modified by a ratio of the $C_a$ for the constant velocity relative to the $C_a$ for a modified sine motion without constant velocity.

$$C = \frac{C_{a(cv)}}{C_a}$$

A service factor, SF, of 1.3 is added into the equation

Substituting, acceleration becomes

$$\alpha = \frac{C_a \text{ x } C \text{ x } SF \text{ x } 2\pi}{S \text{ x } t_2^2}$$

IMC calculates weight moment of inertia and then converts to mass moment of inertia:

$$I = \frac{Wk^2}{g}$$

Where $Wk^2$ = Weight Moment of Inertia (lb-in$^2$)

**g** = Acceleration due to gravity (386.4 in./sec$^2$)

The final torque equation is then

$$T_i = \frac{Wk^2}{g} \quad \frac{C_a \text{ x } C \text{ x } SF \text{ x } 2\pi}{S \text{ x } t_2^2}$$

Substituting the constants

$$T_i = \frac{Wk^2 \text{ x } 5.528 \text{ x } C \text{ x } SF \text{ x } 2\pi}{386.4 \text{ x } S \text{ x } t_2^2}$$

$$T_i = \frac{.09 \text{ x } SF \text{ x } Wk^2 \text{ x } C}{S \text{ x } t_2^2}$$

With a 1.3 service factor, the Inertia Torque Demand Equation is:

$$T_i = \frac{.117 \text{ x } Wk^2 \text{ x } C}{S \text{ x } t_2^2} \text{ (in.-lbs.)}$$

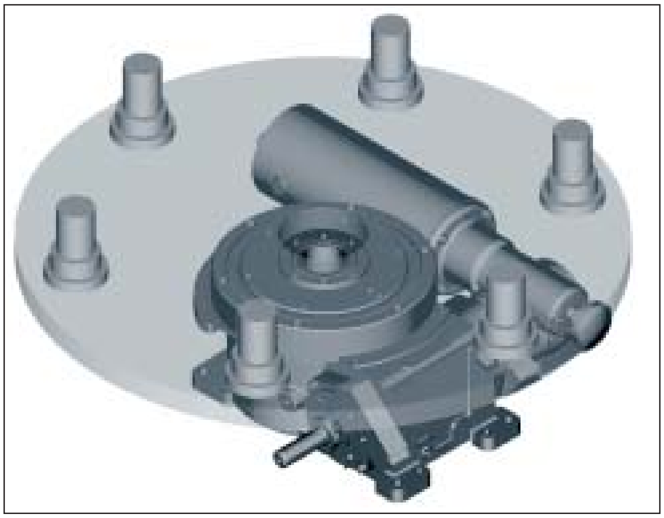
We will use a dial and conveyor application to illustrate.

# Type I and Type II Indexers Explained

Some index drives produce two indexes for every one rotation of input shaft. This is due to the geometric constraints of certain motion period and output displacement combinations. If there is a double index, we call this a **Type II indexer** (and M = 2). If there is only one index per revolution of input camshaft, we call this a **Type I indexer** (and M = 1). **Type I indexers are more common**. If IMC has a Type II indexer with a "270 degree motion period", the first index is achieved with 135 degrees of real input camshaft rotation (270 / 2 = 135°).

## Dial Application Example

| | | |
|---|---|---|
| Number of Stations | (S) | 6 |
| Weight of Single Station | ($W_s$) | 5 lbs. |
| Radius to Station Center | ($R_s$) | 10 in. |
| Dial Plate Diameter | ($D_d$) | 24 in. |
| Dial Plate Weight | ($W_d$) | 33.6 lbs. |
| Dwell Time | ($t_1$) | 2 sec. |
| Index Time | ($t_2$) | 0.5 sec. |



Since dwell time is more then 3 times greater then the index time, the application will be cycle-on-demand.

## Dial Example

**Index Period**

$\beta$ = 270° Since this is cycle-on-demand, choose a long, standard motion period.

**Index Rate**

$$N = \frac{\beta}{6 \times t_2} = \frac{270}{6 \times 0.5} = 90 \text{ index/min.}$$

**Inertia Loading**

Dial Plate Inertia

$$W_d \times \frac{D_d^2}{8} = 33.6 \times \frac{24^2}{8} = 2419 \text{ lb.-in.}^2$$

Station Inertia

$$W_s \times S \times R_s^2 = 5 \times 6 \times 10^2 = 3000 \text{ lb.-in.}^2$$

Total $Wk^2$ External

$$(Wk^2_{(ext.)}) = 2419 + 3000 = 5419 \text{ lb.-in.}^2$$

**Preliminary Output Torque**

$$T_i = \frac{.09 \times SF \times Wk^2_{(ext.)}}{S \times t_2^2} = \frac{.09 \times 1.3 \times 5304}{6 \times .5^2}$$

$$= 423 \text{ in.-lbs.}$$

The model 601RDM6H24-270 with a 4.0D overload clutch is the preliminary selection. $B_{10}$ capacity is 5625 in-lbs. @ 50 index/min. Indexer internal inertia is 110 lbs.-in.$^2$ and the overload clutch inertia is 69 lbs.-in.$^2$.

Correcting $B_{10}$ @ 50 to $B_{10}$ @ 90 to obtain the capacity at the required operating speed of 90 index/minute.

$$B_{10} @ 60 = B_{10} @ 50 \times \left(\frac{50}{90}\right)^{0.3}$$

$$= 5625 \times \left(\frac{50}{90}\right)^{0.3}$$

$$= 5326 \text{ in.-lbs.}$$

**Inertia Torque Calculation**

The following formula includes a safety factor of 1.3.

$$T_i = \frac{.09 \times SF \times (Wk^2_{(ext.)} + Wk^2_{(int.)})}{S \times t_2^2}$$

$$= \frac{.117 \times (5419 + 110)}{6 \times .5^2}$$

$$= 431 \text{ in.-lbs.}$$

**Camshaft Torque**

$$K_f = \frac{C_v \times 360 \times M}{\beta \times S} = \frac{1.7596 \times 360 \times 1}{270 \times 6} = 0.391$$

$$K_i = 0.56 \times K_f = 0.56 \times 0.391 = .219$$

$$T_c = T_i \times K_i = 431 \times .219 = 94 \text{ in.-lb.}$$

Note: $C_v \equiv$ Velocity coefficient for modified sine motion.

# Dial Example

*(continued)*

### Camshaft RPM

$$N_c = \frac{\beta}{6 \times t_2 \times M} = \frac{270}{6 \times .5 \times 1} = 90 \text{ RPM}$$

Where M=1 for Type 1 indexers (see p. A-14)

### Reducer Selection

Assuming an 1800 RPM motor speed, the model R180 reducer with a 20:1 reduction ratio is selected.

### Horsepower

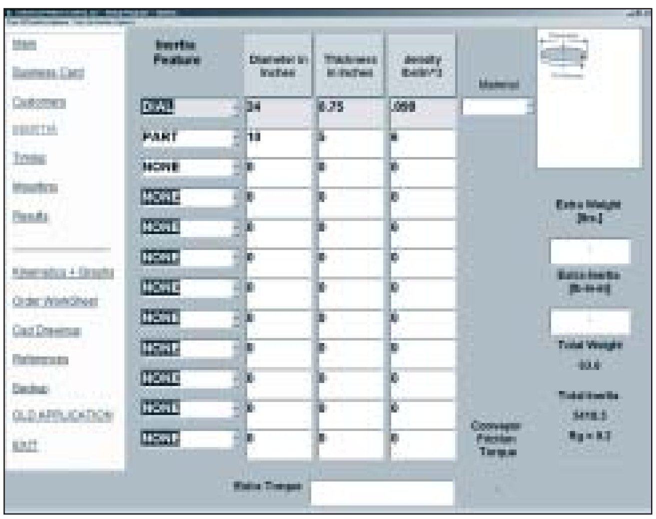
$$Hp = \frac{T_c \times N_c}{63{,}025 \times E} = \frac{94 \times 90}{63{,}025 \times .85} = 0.16 \text{ Hp}$$

Due to component compatibility and horsepower requirements, a $^1/_3$ horsepower motor is selected for this application.

# Dial Example Using IMC Software

With the advent of user-friendly, Windows-based software, we can input the data and quickly get the results. The program takes into account additional factors such as internal friction and cam stresses for more precise calculations. Shown below are the input screens and output screens for the same dial application:

### Dial Application Input Screen



### Dial Application Results Screen

# Conveyor Application Example

## Over/Under Precision Link Converter

| | | |
|---|---|---|
| Index Distance | ($S_x$) | 3.00 in. |
| Index Time | ($t_2$) | 0.375 sec. |
| Dwell Time | ($t_1$) | 3.0 sec. |
| Sprocket Pitch Dia. | ($D_s$) | 7.8394 in. |
| Sprocket Weight | ($W_{ds}$) | 18.0 lbs. |
| Number of Teeth on Sprocket | (n) | 8 |
| Indexed Parts Weight | ($W_p$) | 64 lbs. |
| Chain & Fixture Weight | ($W_c$) | 128 lbs. |
| Coefficient of Friction | ($\mu$) | 0.3 |
| Chain Pitch | (p) | 3.0 in. |

# Conveyor Example

### Index Period

For this cycle-on-demand application, the index period should be 270°, or larger.

Calculating the number of stops.

$$S = \frac{n \times p}{S_x} = \frac{8 \times 3}{3} = 8$$

For cycle-on-demand applications, the index rate for a continuous run should be used for indexer selection.

$$N = \frac{\beta}{6 \times t_2} = \frac{270}{6 \times .375} = 120 \text{ index/min.}$$

### Inertia Calculations

Drive Sprocket

$$W_{ds} \times \frac{D_s^2}{8} = 18 \times \frac{7.8394^2}{8} = 138 \text{ lb.-in.}^2$$

Note: Most IMC Precision Link Conveyors use a chordal compensation cam at the take-up end. No take-up sprocket is necessary.

Chain and Fixtures

$$W_c \times \frac{D_s^2}{4} = 128 \times \frac{7.8394^2}{4} = 1967 \text{ lb.-in.}^2$$

Parts

$$W_p \times \frac{D_s^2}{4} = 64 \times \frac{7.8394^2}{4} = 983 \text{ lb.-in.}^2$$

### External Inertia

$$Wk^2_{(ext.)} = 138 + 1967 + 983 = 3088 \text{ lb.-in.}^2$$

### Preliminary Inertia Torque

$$T_i = \frac{.09 \times SF \times Wk^2_{(ext.)}}{S \times t_2^2} = \frac{.09 \times 1.3 \times 3088}{8 \times .375^2}$$

$$= 321 \text{ in.-lbs.}$$

The equation used to calculate $T_i$ includes a service factor of 1.3

### Friction Torque

$$T_f = (W_c + W_p) \times \frac{D_s}{2} \times \mu = (128 + 64) \times \frac{7.8394}{2} \times 0.3 = 226 \text{ in.-lbs.}$$

### Work Torque

The parts are being translated horizontally, therefore there is no work torque.

### Preliminary Output Torque

$$T_o = T_i + T_f + T_w = 321 + 226 + 0 = 547 \text{ in.-lbs.}$$

Using the appropriate catalog section, select an index drive corresponding to the preliminary torque requirements.

$$B_{10} = \frac{T_o}{\left(\frac{50}{N}\right)^{0.3}} = \frac{547}{\left(\frac{50}{120}\right)^{0.3}} = 711 \text{ in.-lbs.}$$

Select 401RA8H24-270, modified sine motion (ms), $Wk^2_{(int.)}$ = 15 lbs.-in.$^2$, $B_{10}$ capacity = 1463 in.-lbs.

## Conveyor Example

*(continued)*

### Overload Protection

Output overload protection should be used with this application. A large instantaneous gear ratio at the start of index makes output overload protection the preferred method for protecting the index drive. With an output overload clutch, jams or overloads at the start of index can easily be detected prior to damaging the indexer.

From your IMC indexer catalog, select the appropriate clutch model for the index drive being used. Clutch model 2.3FC-SD with $Wk^2_{(cl.)}$ = 31 lbs.-in.$^2$ is chosen.

### Inertia Torque

The actual inertia torque including indexer internal inertia and clutch inertia can now be calculated.

$$T_i = \frac{.09 \times SF \times (Wk^2_{(ext.)} + Wk^2_{(int.)} + Wk^2_{(cl.)} \times C}{S \times t_2^2}$$

$$= \frac{.09 \times 1.3 \times (3088 + 15 + 31) \times 1}{8 \times .375^2}$$

$$= 326 \text{ in.-lbs.}$$

### Output Torque

$$T_o = T_i + T_f + T_w = 326 + 226 + 0 = 552 \text{ in.-lbs.}$$

### C, $K_i$ and $K_f$

Values for C, $K_i$ and $K_f$ can be calculated or found in the table on page A-19.

$C = 1.0,\ K_i = 0.16,\ K_f = 0.29$

### Camshaft Torque

$$Tc = (T_i \times K_i) + (T_f \times K_f) + (T_w \times K_f)$$

$$= (326 \times 0.16) + (226 \times 0.29) + 0$$

$$= 118 \text{ in.-lbs.}$$

### Camshaft RPM

$$N_c = \frac{\beta}{6 \times t_2 \times M} = \frac{270}{6 \times .375 \times 1} = 120 \text{ RPM}$$

This is a type 1 unit, therefore M = 1.
For type 2 or 3, M = 2, M = 3.

From the Right Angle Series indexer catalog section, an R225 reducer with a 15:1 reduction ratio is chosen.

### Horsepower

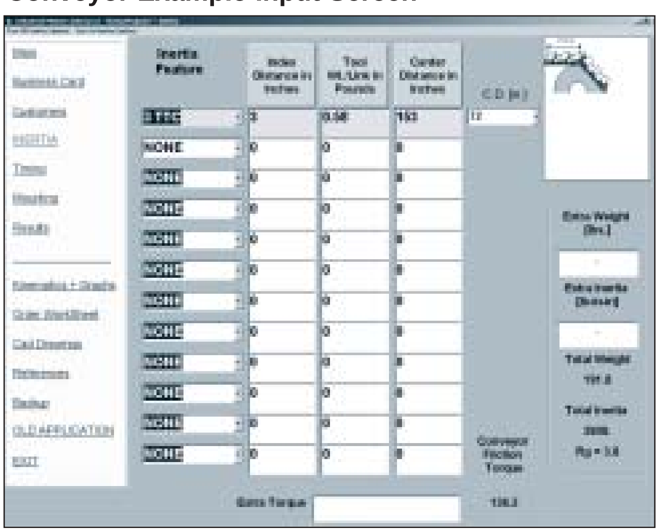
$$Hp = \frac{T_c \times N_c}{63{,}025 \times E} = \frac{118 \times 120}{63{,}025 \times .75} = 0.30 \text{ Hp}$$

Due to component compatibility and horsepower requirements, a one horsepower motor is chosen for this application.

## Conveyor Example Using IMC Software

As demonstrated for the dial application, we can input the conveyor data into the software program and quickly get results. Shown below are the input screens and output screens for the conveyor application:

### Conveyor Example Input Screen

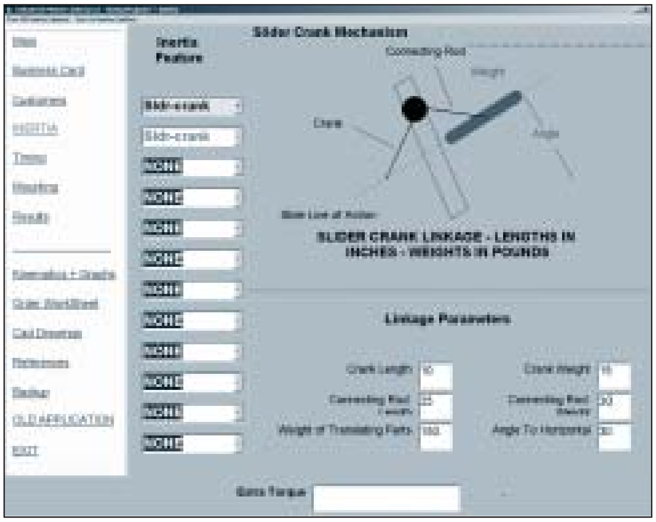


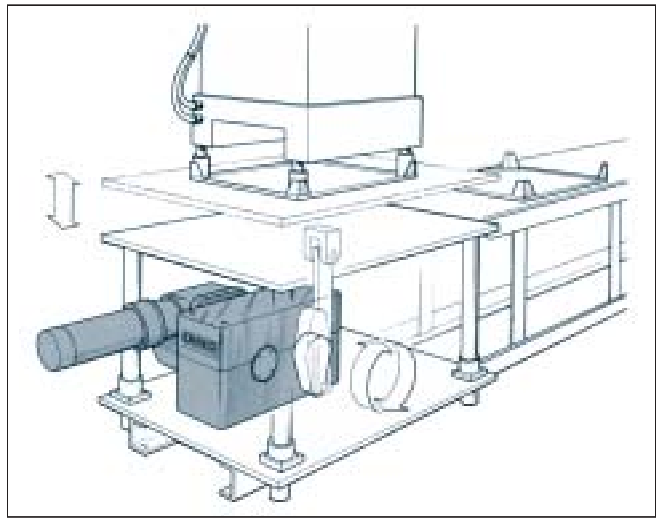
### Conveyor Example Results Screen

## Brief Oscillator Example

IMC software can handle a variety of slider-crank, scotch yoke and 4-bar mechanisms.

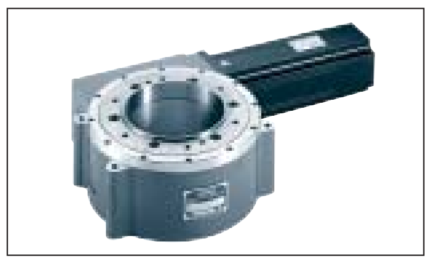
### Slider Crank Linkage Example



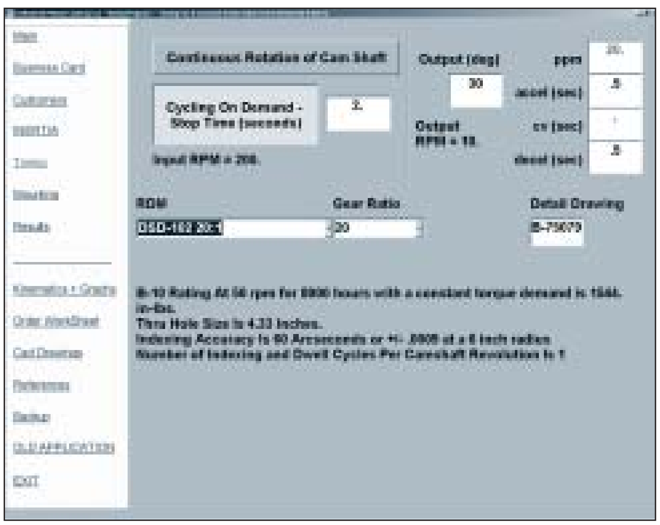



## Servo-Mechanical Example

Camco-Ferguson software can also be used to size servo-driven indexers such as the Flex-i-Dex and Indexers with constant-lead cams. The following illustrations show input and output data for a typical servo-driven application.



**Flex-i-Dex Constant Lead Indexer**




**Servo Application Input Screen**


**Servo Application Output Screen**

# Cycle-On-Demand vs. Continuous Running Applications

If the input shaft (camshaft) of an index drive runs continuously, the ratio of the index time and stop (dwell) time are fixed and a function of the number of degrees on the cam that impart motion to the output shaft (motion period).

If $\mathbf{t_1}$ is the dwell time and $\mathbf{t_2}$ is the index time, total time $t_t = t_1 + t_2$

Typically, the motion period of the cam is rarely less than 90 degrees, due to the geometry constraints of cam design. To illustrate, let's choose a 90 degree index period ($\boldsymbol{\beta_2}$), leaving 270 degrees ($\boldsymbol{\beta_1}$) for dwell time (90° + 270° = 360° total).

Assume 60 RPM camshaft or **N** = 60.

Then

$$\beta_{total} = 360° = 6 \times N \times t_{total} \text{ or}$$

$$t_{total} = \frac{360}{6N} = \frac{360}{6 \times 60} = 1 \text{ sec.}$$

The index time

$$t_2 = \frac{\beta_2}{6N} = \frac{90}{6 \times 60} = 0.25 \text{ sec.}$$

And dwell time

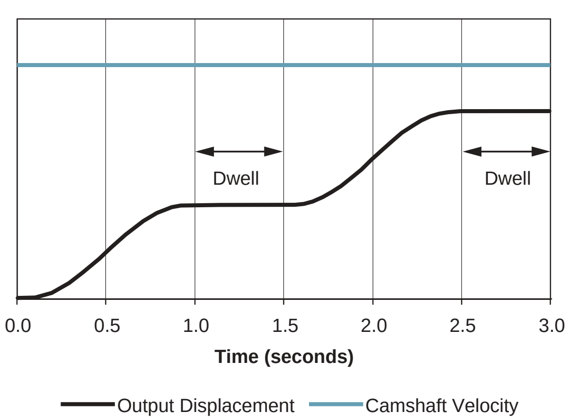
$$t_1 = \frac{\beta_1}{6N} = \frac{270}{6 \times 60} = 0.75 \text{ sec.}$$

Note $t_t = t_1 + t_2 = .75 + .25 = 1.0$ sec.

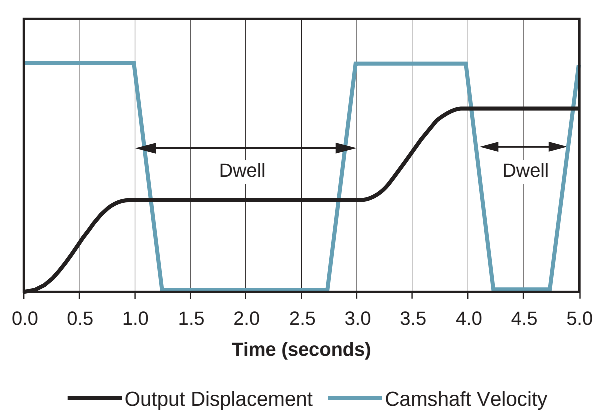
In this example the ratio of dwell time to index time is .75 sec / .25 sec or 3:1.

**Suppose you need more time for dwell, due to the manufacturing process required of the machine. You can then stop the camshaft in dwell for a specified amount of time, and then re-start the camshaft. This is known as cycle-on-demand.** Example: You want to index in 0.25 seconds, but stay in the dwell position for 10 seconds. By using a brake motor (or motor with clutch-brake module), you stop the camshaft and then restart after the required 10 seconds. By rotating the camshaft at 60 RPM when the motor is engaged, you achieve the desired 0.25 index time. These two charts illustrate and summarize the two concepts:

**Continuous Input**
**Fixed Dwell Time**




**Cycle-on-Demand Input**
**Variable or Extended Dwell Time**



## Important Formulas

Refer to page A-16 for Nomenclature

### Torque Due to Inertia

$$T_i = \frac{.09 \times SF \times (Wk^2_{(ext.)} + Wk^2_{(int.)} + Wk^2_{(cl.)}) \times C}{S \times t_2^2}$$

### Torque Due to Friction (dial application)

$T_f = (\frac{1}{2}W_d + W_s) \times R_\mu \times \mu$

Where **$W_d$** = Dial Weight

**$W_s$** = Total weight of the stations

**$R_\mu$** = the radius where rollers or support bearing contact the dial plate

**$\mu$** = coefficient of friction

### Torque Due to Work

$T_w = W_w \times R_w$

Where **$W_w$** = work force or work load

**$R_w$** = radius at which the force is acting, perpendicular to the axis of rotation

### Total Output Torque

$T_t = T_i + T_f + T_w$

### Camshaft Torque

$T_c = T_i \times K_i + (T_f + T_w) \times K_f$

### Relationship between cam angle, time and RPM:

$\beta = 6 \times N \times t \times M$

where $\beta$ (degrees), N (RPM), t (sec.), M (unitless)

or $N = \beta / (6 \times M \times t)$

**Factor for calculating camshaft torque** due to inertia at indexer output, for modified sine motions:

$K_i = 0.56\ K_f$

**Factor for calculating camshaft torque** due to friction and work load at indexer output:

$$K_f = \frac{C_v \times 360 \times M}{\beta \times S}$$

A chart of $K_i$ and $K_f$ factors are listed on page A-19.

### Motion Velocity Factor for Modified Sine Motion
(Moon Velocity Factor)

$$C_v = \frac{1.7596}{1 + (F \times 0.7596)} \quad \text{if } F = 0,\ C_v = 1.7596$$

Where F = % of constant velocity, e.g. F = 0.25 for 25% constant velocity, F = 0 for pure Modified Sine Motion (no constant velocity).

### Motion Acceleration Factor (Moon Acceleration Factor)

$$C_a = \frac{Cv \times \pi}{1 - F} \quad \text{if } F = 0,\ C_a = 5.5280$$

### Constant Velocity Load Adjustment Factor

$$C = \frac{C_a}{5.5280} \quad \text{or } C = \frac{1}{1 - 0.24F - 0.76F^2}$$

### Input Gear Ratio

$G_i \equiv N_{motor} / N_{camshaft} = N_m/N_c$

### Output Gear Ratio

$G_o \equiv D_{driver} / D_{driven} = D_r/D_n$

### Effective Radius of Gyration

$$k = \sqrt{\frac{\sum Wk^2}{\sum W}}$$

K is the theoretical radius at which all of the weight would be concentrated to produce an equivalent weight moment of inertia

### Speed Correction Factor

$$F_s = \left(\frac{50}{N}\right)^{0.3}$$

### Horsepower

$$Hp = \frac{T_c \times N_c}{63025 \times E}$$

## Nomenclature

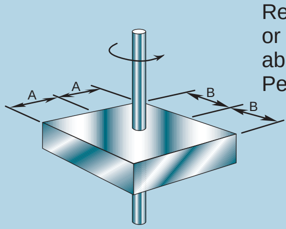
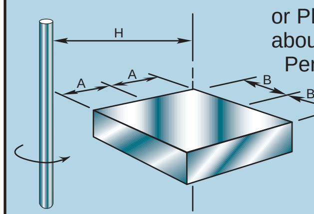
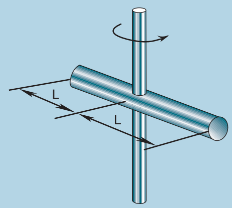
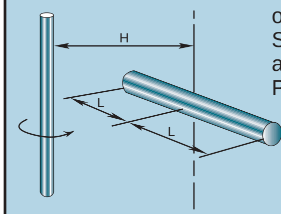
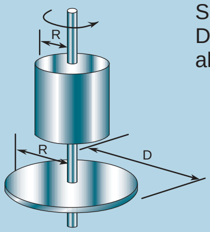
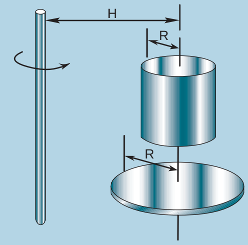
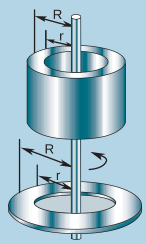
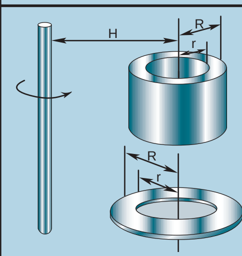
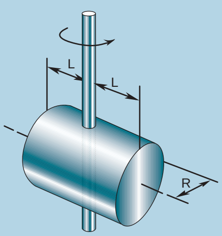
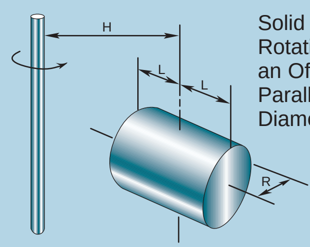
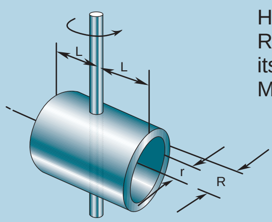
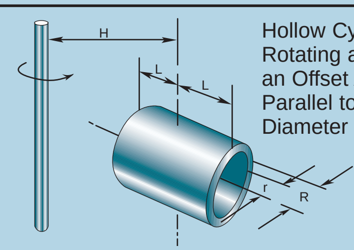
### Terms used for Engineering Calculations

| Symbol | Units | Description | Symbol | Units | Description |
|---|---|---|---|---|---|
| $\beta$ | deg. | Index Period | $R_f$ | in. | Friction Force Radius |
| $\mu$ | – | Coefficient of Friction | $R_s$ | in. | Radius to Station Center |
| $B_{10}$ | in.-lbs. | Basic Dynamic Capacity of the Indexer at a Defined Index Rate | $R_w$ | in. | Radius to Point of Work Load Application |
| C | – | Constant Velocity Load Adjustment Factor | S | – | Number of Stops or Stations Per One Revolution of the Output |
| $C_a$ | – | Motion Acceleration Factor | SF | – | Service Factor 1.3 |
| $C_d$ | – | Factor for Calculating Output Torque | $S_x$ | in. | Linear Index Distance |
| $C_v$ | – | Motion Velocity Factor | $T_c$ | in.-lbs. | Camshaft Torque |
| $D_d$ | in. | Dial Plate Diameter | $T_f$ | in.-lbs. | Friction Torque at the Output |
| $D_n$ | in. | Diameter of Driven Pulley or Gear | $T_i$ | in.-lbs. | Inertia Torque at the Output |
| $D_r$ | in. | Diameter of Drive Pulley or Gear | $T_o$ | in.-lbs. | Total Output Torque |
| $D_s$ | in. | Pitch Diameter of Drive Sprocket(s) | $T_s$ | in.-lbs. | Static Torque |
| $D_t$ | in. | Pitch Diameter of Take-up Sprocket(s) | $T_w$ | in.-lbs. | Work Torque at the Output |
| E | – | Reducer Efficiency | t | sec. | Total Cycle Time ($t_1 + t_2$) |
| F | – | Percent of Constant Velocity | $t_1$ | sec. | Dwell Time |
| $F_s$ | – | Index Rate Factor | $t_2$ | sec. | Index Time |
| $G_i$ | – | Input Gear Ratio | $W_c$ | lbs. | Weight of Chain and Fixtures |
| $G_o$ | – | Output Gear Ratio | $W_d$ | lbs. | Weight of Dial Plate |
| k | in. | Radius of Gyration | $W_{ds}$ | lbs. | Weight of Drive Sprocket(s) |
| $K_f$ | – | Factor for Calculating Cam Shaft Torque Due to Friction and Work Load on Output | $Wk^2_{(ext.)}$ | lbs.-in.$^2$ | External Weight Moment of Inertia at Output |
| $K_i$ | – | Factor for Calculating Cam Shaft Torque due to Inertia at Output | $Wk^2_{(int.)}$ | lbs.-in.$^2$ | Internal Weight Moment of Inertia at Output |
| M | – | Type of Cam (Symbol) Integer Number 1, 2 or 3 | $Wk^2_{(cl.)}$ | lbs.-in.$^2$ | Clutch Weight Moment of Inertia |
| N | ind./min. | Index Rate | $W_n$ | lbs. | Weight of Driven Pulley or Gear |
| n | – | Number of Teeth in Conveyor Drive Sprocket | $W_p$ | lbs. | Weight of Total Parts to be Indexed |
| $N_c$ | RPM | Camshaft Rotation per Minute | $W_r$ | lbs. | Weight of Drive Pulley or Gear |
| $N_m$ | RPM | Power Source Rotation per Minute (Motors, Line Shaft, etc.) | $W_s$ | lbs. | Weight of Each Station (Fixture & Part) |
| p | in. | Chain Pitch of Conveyor Sprocket | $W_t$ | lbs. | Weight of Take-up Sprocket(s) |
| | | | $W_w$ | lbs. | Work Load |
| | | | Y | in. | Dial Plate Thickness |

# Inertia Tables

Multiply radius of gyration squared ($k^2$) by weight to get weight moment of inertia for torque demand calculation.

## Radius of Gyration

| Body With Central Axis of Rotation | $k^2$ | Body with Offset Axis of Rotation | $k^2$ |
|---|---|---|---|
|  Rectangular Prism or Plate Rotating about its Central Perpendicular Axis | $\frac{A^2 + B^2}{3}$ |  Rectangular Prism or Plate Rotating about a Perpendicular Offset Axis | $\frac{A^2 + B^2}{3} + H^2$ |
|  Long Thin Rod of any Cross Section Rotating about its Central Perpendicular Axis | $\frac{L^2}{3}$ |  Long Thin Rod of any Cross Section Rotating about a Perpendicular Offset Axis | $\frac{L^2}{3} + H^2$ |
|  Solid Cylinder or Disc Rotating about its Own Axis | $\frac{R^2}{2}$ or $\frac{D^2}{8}$ |  Solid Cylinder or Disc Rotating about an Offset Parallel Axis | $\frac{R^2}{2} + H^2$ |
|  Hollow Cylinder or Flat Ring Rotating about its Own Axis | $\frac{R^2 + r^2}{2}$ |  Hollow Cylinder or Flat Ring Rotating about an Offset Parallel Axis | $\frac{R^2 + r^2}{2} + H^2$ |
|  Solid Cylinder Rotating about its Diameter at Mid-Length | $\frac{L^2}{3} + \frac{R^2}{4}$ |  Solid Cylinder Rotating about an Offset Axis Parallel to its Diameter | $\frac{L^2}{3} + \frac{R^2}{4} + H^2$ |
|  Hollow Cylinder Rotating about its Diameter at Mid-Length | $\frac{L^2}{3} + \frac{R^2 + r^2}{4}$ |  Hollow Cylinder Rotating about an Offset Axis Parallel to its Diameter | $\frac{L^2}{3} + \frac{R^2 + r^2}{4} + H^2$ |

## Kinematic Calculations

In the "Cam Design Manual for Engineers, Designers, and Draftsmen," Clyde H. Moon developed factors for quickly calculating maximum velocity and maximum acceleration for an application. These are known as "Moon factors." For a Modified Sine Motion, the Moon factors are $\mathbf{C_v = 1.7596}$ and $\mathbf{C_a = 5.5280}$. These factors are unitless and a chart of various Moon factors are listed on page A-19.

If we move an object 12 inches in 0.3 seconds using a Modified Sine Motion, the maximum velocity (at mid-point of index) is:

$$V_{max} = \frac{C_v \text{ x Displacement}}{t_2}$$

$$= \frac{1.7596 \text{ x } 12}{0.3} = 70.4 \text{ inches/second}$$

The maximum acceleration is:

$$A_{max} = \frac{C_a \text{ x Displacement}}{t_2^2}$$

$$= \frac{5.528 \text{ x } 12}{0.3^2} = 737 \text{ inches/sec}^2$$

$$= 1.9 \text{ g's}$$

It can also be used for calculating rotational g force (also known as centrifugal force):

Index a 15-lb. object at a 40 inch radius 90 degrees in 0.5 seconds:

$$\text{Force}_{centrifugal} = \text{Mass x } a_{radialmax}$$

$$= \text{Mass x } \omega_{max}^2 \text{ x R}$$

$$= \frac{15}{386.4} \text{ x } \left(\frac{1.7596 \text{ x } 90° \text{ x } \pi}{180° \text{ x } 0.5}\right)^2 \text{ x } 40$$

$$= 47.4 \text{ lbs.} = \frac{47.4}{15} \text{ g's} = 3.2 \text{ g's}$$

The tangential force component is:

$$\text{Force}_{Tangential} = \text{Mass x } a_{tmax}$$

$$= \frac{15}{386.4} \text{ x } \frac{5.528 \text{ x } 40 \text{ x } 90° \text{ x } \pi}{180° \text{ x } 0.5^2}$$

$$= 53.9 \text{ lbs.} = \frac{53.9}{15} \text{ g's} = 3.6 \text{ g's}$$

# $K_i$ $K_f$ Tables & Moon Factor Tables ($C_v$ and $C_a$)

Values listed are for type 1 units, multiply values by 2 for type 2 units. For motions with constant velocity, multiply K factor by the adjustment factor listed below.

| Number of Stops | Index Period (Modified-Sine Motion) | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 90° | | 120° | | 150° | | 180° | | 210° | | 270° | | 330° | |
| | $K_i$ | $K_f$ | $K_i$ | $K_f$ | $K_i$ | $K_f$ | $K_i$ | $K_f$ | $K_i$ | $K_f$ | $K_i$ | $K_f$ | $K_i$ | $K_f$ |
| 1 | 3.94 | 7.04 | 2.96 | 5.28 | 2.36 | 4.22 | 1.97 | 3.52 | 1.69 | 3.02 | 1.31 | 2.35 | 1.07 | 1.92 |
| 2 | 1.97 | 3.52 | 1.48 | 2.64 | 1.18 | 2.11 | 0.99 | 1.76 | 0.84 | 1.51 | 0.66 | 1.17 | 0.54 | 0.96 |
| 3 | 1.31 | 2.35 | 0.99 | 1.76 | 0.79 | 1.41 | 0.66 | 1.17 | 0.56 | 1.01 | 0.44 | 0.78 | 0.36 | 0.64 |
| 4 | 0.99 | 1.76 | 0.74 | 1.32 | 0.59 | 1.06 | 0.49 | 0.88 | 0.42 | 0.75 | 0.33 | 0.59 | 0.27 | 0.48 |
| 6 | 0.66 | 1.17 | 0.49 | 0.88 | 0.39 | 0.70 | 0.33 | 0.59 | 0.28 | 0.50 | 0.22 | 0.39 | 0.18 | 0.32 |
| 8 | 0.49 | 0.88 | 0.37 | 0.66 | 0.30 | 0.53 | 0.25 | 0.44 | 0.21 | 0.38 | 0.16 | 0.29 | 0.13 | 0.24 |
| 10 | 0.39 | 0.70 | 0.30 | 0.53 | 0.24 | 0.42 | 0.20 | 0.35 | 0.17 | 0.30 | 0.13 | 0.23 | 0.11 | 0.19 |
| 12 | 0.33 | 0.59 | 0.25 | 0.44 | 0.20 | 0.35 | 0.16 | 0.29 | 0.14 | 0.25 | 0.11 | 0.20 | 0.09 | 0.16 |
| 16 | 0.25 | 0.44 | 0.18 | 0.33 | 0.15 | 0.26 | 0.12 | 0.22 | 0.11 | 0.19 | 0.08 | 0.15 | 0.07 | 0.12 |
| 24 | 0.16 | 0.29 | 0.12 | 0.22 | 0.10 | 0.18 | 0.08 | 0.15 | 0.07 | 0.13 | 0.05 | 0.10 | 0.04 | 0.08 |
| 36 | 0.11 | 0.20 | 0.08 | 0.15 | 0.07 | 0.12 | 0.05 | 0.10 | 0.05 | 0.08 | 0.04 | 0.07 | 0.03 | 0.05 |

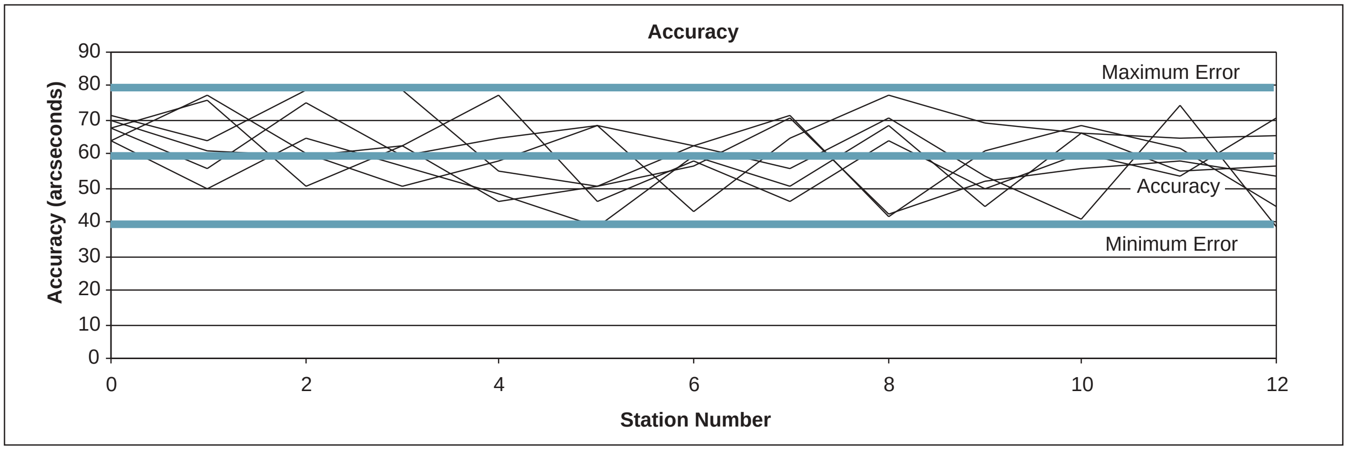
### $K_i$ and $K_f$ Adjustment Factors

| For % Constant Velocity | Multiply K factor by |
|---|---|
| 0% | 1.00 |
| 20% | 0.87 |
| 25% | 0.84 |
| 33% | 0.80 |
| 50% | 0.72 |
| 66% | 0.67 |
| 75% | 0.64 |

### Constant Velocity Adjustment Factor "C", Velocity and Acceleration Factors

| | % Constant Velocity | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 0% | 5% | 10% | 20% | 25% | 33% | 50% | 66% | 75% |
| C | 1.000 | 1.014 | 1.033 | 1.085 | 1.120 | 1.193 | 1.449 | 1.959 | 2.548 |
| $C_v$ | 1.7596 | 1.6952 | 1.6354 | 1.5275 | 1.4788 | 1.4069 | 1.2753 | 1.1720 | 1.1210 |
| $C_a$ | 5.5280 | 5.6060 | 5.7085 | 5.9986 | 6.1943 | 6.5970 | 8.0127 | 10.8295 | 14.0866 |

# Accuracy




IMC intentionally chooses to state the Maximum Error as the indexer's worst possible accuracy. While some index drive manufacturers use the average as their stated accuracy and decline to state the repeatability, IMC takes a more conservative approach.

### Measurement Method

The output angular error of an index drive is measured using a laser collimator mounted to a precision rotary table. The laser collimator is accurate to 2 arc seconds and repeatable to 1 arc second. The indexer will make 3 to 6 complete turns of its output and accuracy measurements are recorded. The accuracy is the mean between the maximum and minimum error. The repeatability is one-half the difference between the maximum and minimum error.

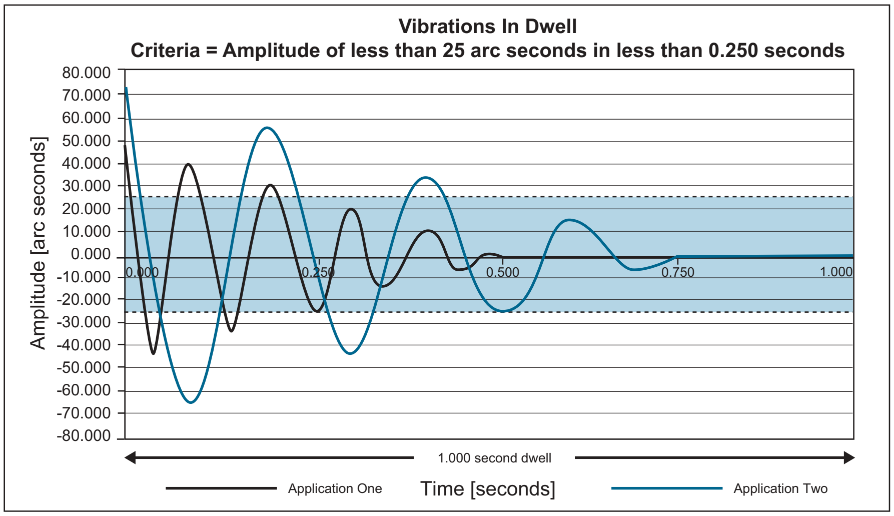
**Upon request, IMC can provide special accuracy reports for a particular indexer.**

# Vibration

Cam-actuated index drives are frequently chosen because of their stability in dwell, especially when operating at high speeds. It is important to ascertain that the entire system is well designed to prevent any unwanted vibrations. Vibration is a function of the index time, index rate, friction (dampens the system), input and output connections, torsional spring rate and the natural frequency of the indexer and driven members.

One simple method to avoid problems is to calculate the ratio of the effective radius of gyration (k) to the cam follower pitch radius. This method does not always produce consistent results. For example, a system with a large effective radius of gyration can be run at slow speed and there are no observable vibrations in dwell. Friction also helps prevent vibration, as in the case of precision link conveyor systems.

*(continued)*



## Vibration

*(continued)*

IMC has developed stability criteria that effectively predict vibration effects. The criteria require that the amplitude of the vibration must be less than 25 arc seconds in 25% of the dwell time. IMC's loading programs automatically check for this prerequisite.

Vibration can be avoided by following the recommended input and output connection methods and confirming the vibration effect of the specific application using IMC's loading software program.

## Emergency Stop

Emergency stops can occur during any part of the index motion. OSHA and other regulatory agencies would like this stop to occur instantaneously. The laws of physics require that the stop occurs within a finite time – and this time cannot be too extended (for it would defeat the purpose of an emergency stop).

Intuition suggests that the worst possible time for an emergency stop is at mid-motion of the index, at peak output velocity. At that moment we have the greatest amount of kinetic energy. The mathematics of motion curves prove otherwise. For a particular type of motion, computer software analysis is the best method for determining the worst case scenario. Upon request, IMC engineering staff can evaluate and calculate the maximum expected stop times for specific applications and also evaluate the resulting stresses on the cam, cam followers, follower wheel and input components (reducer, motor, clutch and brake). Normal forces on the cam follower must not exceed the vendor's recommended maximum and the cam and camshaft stress must not exceed the ultimate yield stress of the material (the cam and camshaft are normally designed for fatigue and not strength).

For an application with an Emergency stop requirement, IMC recommends that the drive package for an indexer should be a low ratio worm gear drive (10:1 or 15:1) along with a helical primary (5:1 or 5:1). This should be coupled to an air or hydraulic clutch-brake. Wet type or Hydro-viscous type clutch-brakes are recommended due to their low inertia of the cyclic parts and high heat dissipation capability. In contrast, dry type clutch-brakes wear quickly. In an Emergency stop mode, the clutch-brake disengages the motor since the low-ratio gear combination (low ratio worm and helical primary combination) will be intentionally back driven. The brake then dissipates the kinetic energy of the Emergency stop. For further details, please contact your IMC sales representative or IMC application engineer.

A

## Overload Protection

IMC offers a wide variety of output overload clutches and input overload clutches designed to protect the indexer drive. Overload clutches are recommended due to the nature of indexing. At the very beginning of an index, the input displacement is large while the output displacement is miniscule. At that precise moment, the instantaneous gear ratio of the drive is extremely high – almost infinite. Small amounts of input torque produce tremendous output torque. If there are any machine components or product parts jamming the mechanism (dial, conveyor belt or other linkage), the tooling or the index drive itself could be damaged. IMC clutches are offered in a wide assortment of geometries to accommodate shaft-to-shaft, flange-to-shaft and dial applications. IMC also offers internal overload clutches on certain models to protect the clutch from foreign contamination. IMC overload clutches for indexing applications have a single position reset point ot ensure accuracy and repeatablility.

Typically, proximity switches are mounted adjacent to the clutch to sense an overload condition (sense the detector plate movement) and shut down the machinery.

## Lubrication

**Indexers**

IMC Index drives are normally shipped without oil to avoid possible leakage during transit. Each particular index drive mounting position requires a different oil level. A "bulls eye" type oil level sight gauge is supplied with each index drive. The unit should be filled with oil until the level reaches the middle of this sight gauge.

Lubricating oils for use in an index drive should be high quality, well-refined petroleum oils or synthetic lubricants with extreme pressure additives. They may be subject to high operating temperatures, so they must have good resistance to oxidation. The lubricant must meet these specifications: MIL-PRE-2105E or SAE 80W-140, ISO 220 or AGMA 5 with EP (extreme pressure) additives.

Some units use grease rather than oil. In this case, the unit will be shipped with the grease. Generally, IMC uses a lithium grease such as Mobilith AW-2.

**Gear Reducers**

Lubricating oils for gear reducers should also be of high quality, well-refined petroleum oils. These oils should meet AGMA 8 or 8A specifications or ISO 680 or 1000 specifications. Oils with EP additives should not be used if the reducer contains bronze parts.

If you have any questions regarding lubricants, please contact IMC's engineering department.

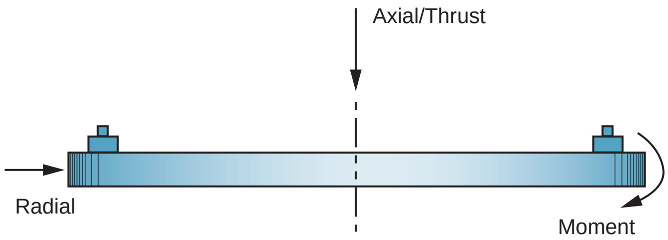
## Axial, Radial & Moment Capacity

In addition to the $B_{10}$ capacity, which is based on the cam follower capacity, an index drive also has a load capacity based on the bearings supporting the output. Several load conditions can be present in an application:

- **Axial or Thrust Capacity** is the maximum balanced load the indexer's output bearing can support. Due to the use of large bearings, this load capacity generally does not need to be addressed in normal applications.
- **Radial Capacity** is the maximum side load of the output bearing, applied through and perpendicular to the axis of rotation.
- **Moment Capacity** is the maximum overturning or unbalanced load capacity of the output bearing.

The Axial, Radial and Moment capacities for most indexers are listed in the appropriate product section.

***Exceeding the capacity of the output bearing with any of these types of forces can cause permanent deformation of the cam, fractured cam followers, or output bearing failure.*** Contact IMC engineering for analysis of application with special requirements regarding any of these conditions.



## Input Considerations

All load calculations are based on a constant velocity input (camshaft speed) during the index. If there are any speed variations on the input shaft, these variations are amplified on the output shaft (velocities are accelerated and accelerations become jerk). It is very important to have a controlled motor speed and a reducer ratio sufficient to dampen any input speed variations. If input belts are used, they must be tightened to prevent any slip or belt jumping when positive torque changes to negative torque (input shafts typically see both positive and negative torque in an indexing application). Pulleys should be maximized to the largest diameter that can fit on the camshaft. Adjustable tensioning idler pulleys are highly recommended. If you have any questions regarding input speed control, please contact your local IMC sales representative or IMC application engineer.

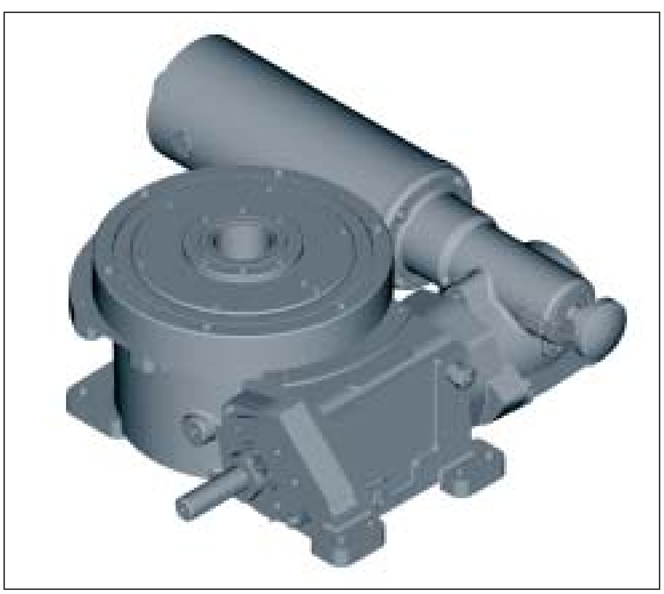
## Output Considerations

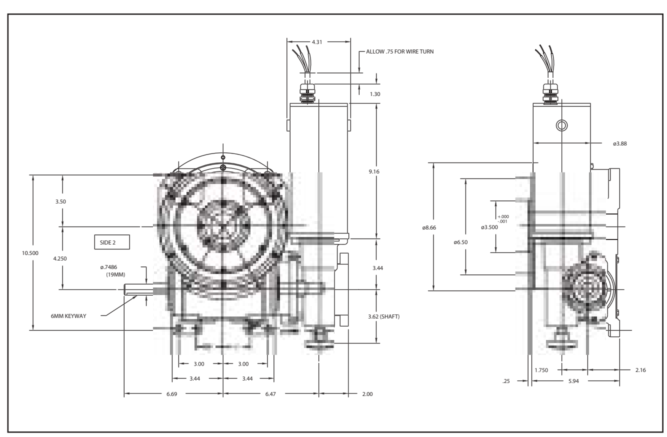
Indexing always imparts positive and negative torques on the driven members. All connections should be tight and doweled whenever possible. Shaft coupling connections should have an interference fit and not depend on the keyway for tightness, as any clearance in the key stock or keyways will eventually cause the connection to loosen.

## IMC Online

The IMC website, **www.camcoindex.com**, features useful tools for those responsible for specifying, applying and servicing Camco and Ferguson products. These include:

- 2-D and 3-D CAD drawings in a variety of formats
- General and Product-specific Service Manuals
- Product Catalogs
- Contact information for local sales representatives





## Weights & Oil Content

| Right Angle | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 301RA | 15 | 7 | C/F | C/F |
| 400RA | 33 | 15 | 1 | 1.0 |
| 401RA | 55 | 25 | 1 | 1.0 |
| 512RA | 80 | 36 | 2 | 2.0 |
| 662RA | 160 | 73 | 6 | 6.0 |
| 663RA | 130 | 59 | 4 | 4.0 |
| 900RA | 220 | 100 | 6 | 6.0 |
| 1200RA | 850 | 386 | C/F | C/F |

| Parallel | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 250P | 18 | 8 | 1 | 1 |
| 387P | 55 | 25 | 2 | 2 |
| 512P | 135 | 61 | 5 | 5 |
| 662P | 430 | 195 | 10 | 10 |
| 900P | 750 | 340 | 20 | 19 |
| 1200P | 1,100 | 499 | 48 | 45 |
| 1800P | 3,000 | 1,361 | 95 | 90 |
| P200 | 20 | 9 | C/F | C/F |
| P325 | 60 | 27 | 2 | 2 |
| P400 | 85 | 39 | 4 | 4 |
| P500 | 110 | 50 | 4 | 4 |
| P600 | 160 | 73 | 6 | 6 |
| P750 | 500 | 227 | 14 | 13 |
| P1050 | 750 | 340 | 24 | 23 |
| P1400 | 1,100 | 499 | 52 | 49 |
| P1700 | 3,000 | 1,361 | 76 | 72 |

| Roller Gear | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 350RG | 35 | 16 | 2 | 1 |
| 500RG | 350 | 159 | 5 | 5 |
| 600RG | 390 | 177 | 7 | 7 |
| 700RG | 400 | 181 | 10 | 9 |
| FD-100 | 10 | 5 | 7 oz | 0.2 |
| FD-162 | 50 | 23 | 2 | 2 |
| FD-200 | 115 | 52 | 4 | 4 |
| FD-250 | 125 | 57 | 7 | 7 |
| FD-300 | 300 | 136 | 10 | 9 |
| FD-451 | 560 | 254 | 12 | 11 |
| FD-501 | 900 | 408 | 22 | 21 |

| Overload Clutches | | |
|---|---|---|
| Model | Weight (lbs) | Weight (Kg) |
| 0.39 | 5 | 2 |
| 2.3 | 10 | 5 |
| 4 | 17 | 8 |
| 6 | 25 | 11 |
| 7.8 | 20 | 9 |
| 11 | 40 | 18 |
| 18 | 75 | 34 |
| 35 | 57 | 26 |
| 31 | 123 | 56 |

| Torq Gard Clutches | | |
|---|---|---|
| Model | Weight (lbs) | Weight (Kg) |
| TG3 | 2 | 1 |
| TG6 | 2 | 1 |
| TG20 | 3 | 1 |
| TG60 | 6 | 3 |
| TG200 | 12 | 5 |
| TG400 | 43 | 20 |
| TG800 | 43 | 20 |

| Cambots | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 150RPP | 45 | 20 | 2.5 | 2 |
| 300RPP | 110 | 50 | 4 | 4 |
| 500RPP | 300 | 136 | 10 | 9 |
| 900RPP | 575 | 261 | 48 | 45 |
| WBD-101 | C/F | C/F | These units are grease-filled. Consult the model-specific service manual for lubrication information. | |
| WBD-201 | C/F | C/F | | |
| WBD-301 | C/F | C/F | | |
| WBD-401 | C/F | C/F | | |
| 140LPP | 55 | 25 | | |
| 240LPP | 80 | 36 | | |
| 380LPP | 200 | 91 | | |
| 4120LPP | 340 | 154 | | |
| LPP-101 | C/F | C/F | | |
| LPP-201 | C/F | C/F | | |
| LPP-301 | C/F | C/F | | |
| LPP-401 | C/F | C/F | | |

## Weights & Oil Content

*(continued)*

| Roller Dial | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 601RDM | 70 | 32 | 2 | 2 |
| 902RDM | 130 | 59 | 3 | 3 |
| 1100RDM | C/F | C/F | 8 | 8 |
| 1305RDM | 305 | 138 | 9 | 9 |
| 1800RDM | 1,400 | 635 | 36 | 34 |
| PN300 | 50 | 23 | 4 | 4 |
| PN400 | C/F | C/F | 8 | 8 |
| PN500 | C/F | C/F | 12 | 11 |
| ED200 | 300 | 136 | 6 | 6 |
| ED420 | 750 | 340 | 20 | 19 |
| ED810 | 1,200 | 544 | 20 gal | 76 |
| ED1440 | 3,000 | 1,361 | 30 gal | 114 |
| ED3240 | 9,500 | 4,309 | 35 gal | 132 |
| 425RD | 110 | 50 | 2 | 2 |
| 800RD | 450 | 204 | 5 | 5 |
| 1301RD | 1,000 | 454 | 14 | 13 |
| 1801RD | 2,400 | 1,089 | 9 gal | 34 |
| 122 | 200 | 91 | 16 | 15 |
| 242 | 800 | 363 | 32 | 30 |
| 362 | 1,500 | 680 | 56 | 53 |
| 482 | 6,000 | 2,722 | 33 gal | 125 |
| 722 | 8,750 | 3,969 | 42 gal | 159 |

| Miniature Roller Gear | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 32RG | 3 | 1 | C/F | C/F |
| 40RG | 6 | 3 | 6 oz | 0.2 |
| 50RG | 18 | 8 | 1 | 0.5 |
| 70RG | 25 | 11 | 1.25 - 1.5 | 1.2 - 1.4 |
| 80RG | 65 | 29 | 30 - 38 oz | 1.8 - 2.25 |

| E Series | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (gallons) | Oil Capacity (Liters) |
| 950E | 5,000 | 2,268 | 10 gal | 38 |
| 1150E | 5,500 | 2,495 | 25 gal | 95 |
| 1550E | 6,000 | 2,722 | 40 gal | 152 |
| 2050E | 18,000 | 8,165 | 45 gal | 170 |
| 2750E | 54,000 | 24,494 | 75 gal | 284 |

| Gear Reducers | | | | |
|---|---|---|---|---|
| Model | Weight (lbs) | Weight (Kg) | Oil Capacity (quarts) | Oil Capacity (Liters) |
| 180SM | 10 | 5 | 1 | 1 |
| R225 | 25 | 11 | 1 | 1 |
| R260 | 25 | 11 | 1 | 1 |
| 7300C | 89 | 40 | 1.5 | 1.4 |
| 7350C | 123 | 56 | 3.5 | 3.3 |
| 7400C | 180 | 82 | 4 | 3.8 |
| 7500C | 307 | 139 | 7 | 6.6 |
| 7600C | 433 | 196 | 11 | 10 |
| 7700C | 625 | 283 | 20 | 19 |
| 7800C | 755 | 342 | 26 | 25 |
| 71000C | 1,625 | 737 | 56 | 53 |
| 20CDSF | 35 | 16 | 13 oz | 0.4 |
| 26CDSF | 55 | 25 | 2 | 1.9 |
| 6SF | 85 | 39 | 2 | 1.9 |