Ivan J. Garshelis. "Torque and Power Measurement."

Torque and Power Measurement

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Torque, speed, and power are the defining mechanical variables associated with the functional performance of rotating machinery. The ability to accurately measure these quantities is essential for determining a machine's efficiency and for establishing operating regimes that are both safe and conducive to long and reliable services. On-line measurements of these quantities enable real-time control, help to ensure consistency in product quality, and can provide early indications of impending problems. Torque and power measurements are used in testing advanced designs of new machines and in the development of new machine components. Torque measurements also provide a well-established basis for controlling and verifying the tightness of many types of threaded fasteners. This chapter describes the basic concepts as well as the various methods and apparati in current use for the measurement of torque and power; the measurement of speed, or more precisely, angular velocity, is discussed elsewhere in this handbook [1].

24.1 Fundamental Concepts

Angular Displacement, Velocity, and Acceleration

The concept of *rotational* motion is readily formalized: all points within a rotating rigid body move in parallel or coincident planes while remaining at fixed distances from a line called the *axis.* In a perfectly rigid body, all points also remain at fixed distances from each other. Rotation is perceived as a change in the angular position of a reference point on the body, i.e., as its *angular displacement,* $\Delta\theta$, over some time interval, Δt. The motion of that point, and therefore of the whole body, is characterized by its

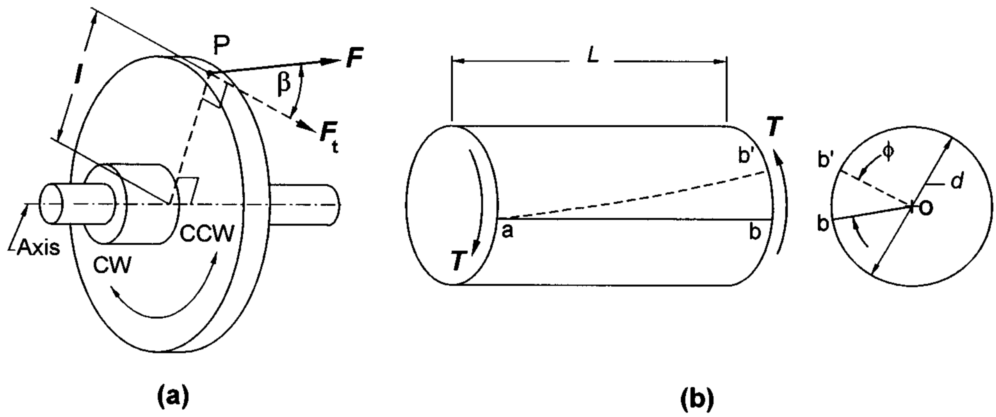

FIGURE 24.1 (*a*) The off-axis force F at P produces a torque $T = (F \cos \beta)l$ tending to rotate the body in the CW direction. (*b*) Transmitting torque T over length L twists the shaft through angle ϕ.

clockwise (CW) or counterclockwise (CCW) *direction* and by its *angular velocity,* $\omega = \Delta\theta/\Delta t$. If during a time interval Δt, the velocity changes by $\Delta\omega$, the body is undergoing an *angular acceleration,* $\alpha = \Delta\omega/\Delta t$. With angles measured in radians, and time in seconds, units of ω become radians per second (rad s^{-1}) and of α, radians per second per second (rad s^{-2}). Angular velocity is often referred to as *rotational speed* and measured in numbers of complete revolutions per minute (rpm) or per second (rps).

Force, Torque, and Equilibrium

Rotational motion, as with motion in general, is controlled by *forces* in accordance with Newton's laws. Because a force directly affects only that component of motion in its line of action, forces or components of forces acting in any plane that includes the axis produce no tendency for rotation about that axis. Rotation can be initiated, altered in velocity, or terminated only by a *tangential force* F_t acting at a finite radial distance l from the axis. The effectiveness of such forces increases with both F_t and l; hence, their product, called a *moment,* is the activating quantity for rotational motion. A moment about the rotational axis constitutes a *torque.* Figure 24.1(*a*) shows a force F acting at an angle β to the tangent at a point P, distant l (the moment arm) from the axis. The torque T is found from the *tangential component* of F as:

$$T = F_t l = \left(F \cos \beta\right) l \tag{24.1}$$

The combined effect, known as the *resultant,* of any number of torques acting at different locations along a body is found from their *algebraic sum,* wherein torques tending to cause rotation in CW and CCW directions are assigned opposite signs. Forces, hence torques, arise from physical contact with other solid bodies, motional interaction with fluids, or via gravitational (including inertial), electric, or magnetic force fields. The *source* of each such torque is subjected to an equal, but oppositely directed, *reaction* torque. With force measured in newtons and distance in meters, Equation 24.1 shows the unit of torque to be a Newton meter (N·m).

A nonzero resultant torque will cause the body to undergo a proportional angular acceleration, found, by application of Newton's second law, from:

$$T_r = I\alpha \tag{24.2}$$

where I, having units of kilogram meter^2 (kg m^2), is the moment of inertia of the body around the axis (i.e., its *polar* moment of inertia). Equation 24.2 is applicable to any body regardless of its state of motion.

When $\alpha = 0$, Equation 24.2 shows that T_r is also zero; the body is said to be in *equilibrium*. For a body to be in equilibrium, there must be either more than one *applied* torque, or none at all.

Stress, Rigidity, and Strain

Any portion of a rigid body in equilibrium is also in equilibrium; hence, as a condition for equilibrium of the portion, any torques applied thereto from *external* sources must be balanced by equal and directionally opposite *internal* torques from adjoining portions of the body. Internal torques are *transmitted* between adjoining portions by the collective action of *stresses* over their common cross-sections. In a solid body having a round cross-section (e.g., a typical shaft), the *shear stress* τ varies linearly from zero at the axis to a maximum value at the surface. The shear stress, τ_m, at the surface of a shaft of diameter, d, transmitting a torque, T, is found from:

$$\tau_m = \frac{16T}{\pi d^3} \tag{24.3}$$

Real materials are not *perfectly* rigid but have instead a *modulus of rigidity, G,* which expresses the finite ratio between τ and *shear strain,* γ. The maximum strain in a solid round shaft therefore also exists at its surface and can be found from:

$$\gamma_m = \frac{\tau_m}{G} = \frac{16T}{\pi d^3 G} \tag{24.4}$$

Figure 24.1(*b*) shows the manifestation of shear strain as an angular displacement between axially separated cross-sections. Over the length L, the solid round shaft shown will be *twisted* by the torque through an angle ϕ found from:

$$\phi = \frac{32LT}{\pi d^4 G} \tag{24.5}$$

Work, Energy, and Power

If during the time of application of a torque, T, the body rotates through some angle θ, mechanical work:

$$W = T\theta \tag{24.6}$$

is performed. If the torque acts in the same CW or CCW sense as the displacement, the work is said to be done *on* the body, or else it is done *by* the body. Work done *on* the body causes it to accelerate, thereby appearing as an increase in *kinetic energy* ($KE = I\omega^2/2$). Work done *by* the body causes deceleration with a corresponding decrease in kinetic energy. If the body is not accelerating, any work done on it at one location must be done by it at another location. Work and energy are each measured in units called a joule (J). Equation 24.6 shows that 1 J is equivalent to 1 N·m rad, which, since a radian is a dimensionless ratio, ≡ 1 N·m. To avoid confusion with torque, it is preferable to quantify mechanical work in units of m·N, or better yet, in J.

The *rate* at which work is performed is termed *power, P.* If a torque T acts over a small interval of time Δt, during which there is an angular displacement $\Delta\theta$, work equal to $T\Delta\theta$ is performed at the rate $T\Delta\theta/\Delta t$. Replacing $\Delta\theta/\Delta t$ by ω, power is found simply as:

$$P = T\omega \tag{24.7}$$

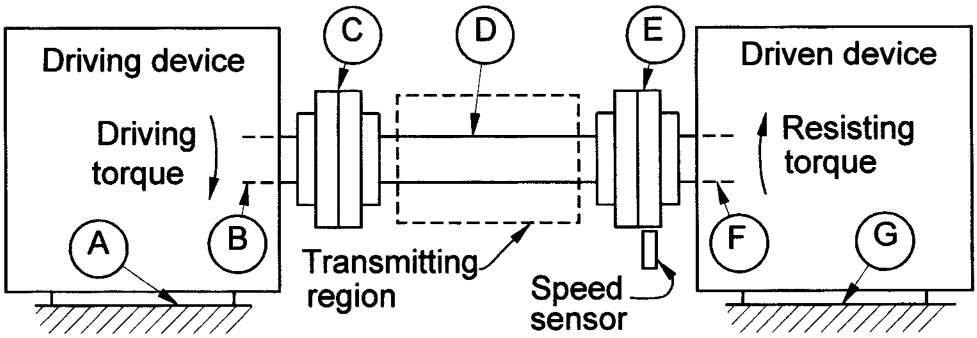

FIGURE 24.2 Schematic arrangement of devices used for the measurement of torque and power.

The unit of power follows from its definition and is given the special name watt (W). 1 W = 1 J s^{-1} = 1 m·N s^{-1}. Historically, power has also been measured in horsepower (Hp), where 1 Hp = 746 W. Rotating bodies effectively transmit power between locations where torques from external sources are applied.

24.2 Arrangements of Apparatus for Torque and Power Measurement

Equations 24.1 through 24.7 express the physical bases for torque and power measurement. Figure 24.2 illustrates a generalized measurement arrangement. The actual apparatus used is selected to fulfill the specific measurement purposes. In general, a driving torque originating within a device at one location (B in Figure 24.2), is resisted by an opposing torque developed by a different device at another location (F). The driving torque (from, e.g., an electric motor, a gasoline engine, a steam turbine, muscular effort, etc.) is coupled through connecting members C, transmitting region D, and additional couplings E, to the driven device (an electric generator, a pump, a machine tool, mated threaded fasteners, etc.) within which the resisting torque is met at F. The torque at B or F is the quantity to be measured. These torques may be *indirectly* determined from a correlated physical quantity, e.g., an electrical current or fluid pressure associated with the operation of the driving or driven device, or more directly by measuring either the *reaction* torque at A or G, or the *transmitted* torque through D. It follows from the cause-and-effect relationship between torque and rotational motion that most interest in transmitted torque will involve rotating bodies.

To the extent that the frames of the driving and driven devices and their mountings to the "Earth" are *perfectly* rigid, the reaction at A will *at every instant* equal the torque at B, as will the reaction at G equal the torque at F. Under equilibrium conditions, these equalities are independent of the compliance of any member. Also under equilibrium conditions, and except for usually minor *parasitic* torques (due, e.g., to bearing friction and air drag over rapidly moving surfaces), the driving torque at B will equal the resisting torque at F.

Reaction torque at A or G is often determined, using Equation 24.1, from measurements of the forces acting at known distances fixed by the apparatus. Transmitted torque is determined from measurements, on a suitable member within region D, of τ_m, γ_m, or ϕ and applying Equations 24.3, 24.4, or 24.5 (or analogous expressions for members having other than solid round cross-sections [2]). *Calibration,* the measurement of the stress, strain, or twist angle resulting from the application of a *known* torque, makes it unnecessary to know any details about the member within D. When $\alpha \neq 0$, and is measurable, T may also be determined from Equation 24.2. Requiring only noninvasive, observational measurements, this method is especially useful for determining transitory torques; for example those associated with firing events in multicylinder internal combustion engines [3].

Equations 24.6 and 24.7 are applicable *only* during rotation because, in the absence of motion, no work is done and power transfer is zero. Equation 24.6 can be used to determine *average* torque from calorimetric measurements of the heat generated (equal to the mechanical work *W*) during a totalized number of revolutions ($\equiv \theta/2\pi$). Equation 24.7 is routinely applied in power measurement, wherein *T* is determined by methods based on Equations 24.1, 24.3, 24.4, or 24.5, and ω is measured by any suitable means [4].

F, T, and ϕ are sometimes measured by simple mechanical methods. For example, a "torque wrench" is often used for the controlled tightening of threaded fasteners. In these devices, torque is indicated by the position of a needle moving over a calibrated scale in response to the elastic deflection of a spring member, in the simplest case, the bending of the wrench handle [5]. More generally, instruments, variously called *sensors* or *transducers,* are used to convert the desired (torque or speed related) quantity into a linearly proportional electrical signal. (Force sensors are also known as *load cells.*) The determination of *P* most usually requires multiplication of the two signals from separate sensors of *T* and ω. A transducer, wherein the amplitude of a *single* signal proportional to the power being transmitted along a shaft, has also been described [6].

24.3 Torque Transducer Technologies

Various physical interactions serve to convert F, τ, γ, or ϕ into proportional electrical signals. Each requires that some axial portion of the shaft be dedicated to the torque sensing function. Figure 24.3 shows typical features of sensing regions for four sensing technologies in present use.

Surface Strain

Figure 24.3(*a*) illustrates a sensing region configured to convert surface strain (γ_m) into an electric signal proportional to the transmitted torque. Surface strain became the key basis for measuring both force and torque following the invention of bonded wire strain gages by E. E. Simmons, Jr. and Arthur C. Ruge in 1938 [7]. A modern strain gage consists simply of an elongated electrical conductor, generally formed in a serpentine pattern in a very thin foil or film, bonded to a thin insulating carrier. The carrier is attached, usually with an adhesive, to the surface of the load carrying member. Strain is sensed as a change in gage resistance. These changes are generally too small to be accurately measured directly and so it is common to employ two to four gages arranged in a Wheatstone bridge circuit. Independence from axial and bending loads as well as from temperature variations are obtained by using a four-gage bridge comprised of two diametrically opposite pairs of matched strain gages, each aligned along a *principal strain* direction. In round shafts (and other shapes used to transmit torque), tensile and compressive principal strains occur at 45° angles to the axis. Limiting strains, as determined from Equation 24.4 (with τ_m equal to the shear proportional limit of the shaft material), rarely exceed a few parts in 10^3. Typical practice is to increase the compliance of the sensing region (e.g., by reducing its diameter or with hollow or specially shaped sections) in order to attain the limiting strain at the highest value of the torque to be measured. This maximizes the measurement sensitivity.

Twist Angle

If the shaft is *slender* enough (e.g., $L > 5\ d$) ϕ, at limiting values of τ_m for typical shaft materials, can exceed 1°, enough to be resolved with sufficient accuracy for practical torque measurements (ϕ at τ_m can be found by manipulating Equations 24.3, 24.4, and 24.5). Figure 24.3(*b*) shows a common arrangement wherein torque is determined from the difference in tooth-space phasing between two identical "toothed" wheels attached at opposite ends of a compliant "torsion bar." The phase displacement of the periodic electrical signals from the two "pickups" is proportional to the peripheral displacement of salient features on the two wheels, and hence to the twist angle of the torsion bar and thus to the torque. These features are chosen to be sensible by any of a variety of noncontacting magnetic, optical, or capacitive techniques. With more elaborate pickups, the relative angular position of the two wheels appears as the amplitude of a *single* electrical signal, thus providing for the measurement of torque even on a stationary shaft (e.g., [13–15]). In still other constructions, a shaft-mounted variable displacement transformer or a related type of electric device is used to provide speed independent output signals proportional to ϕ.

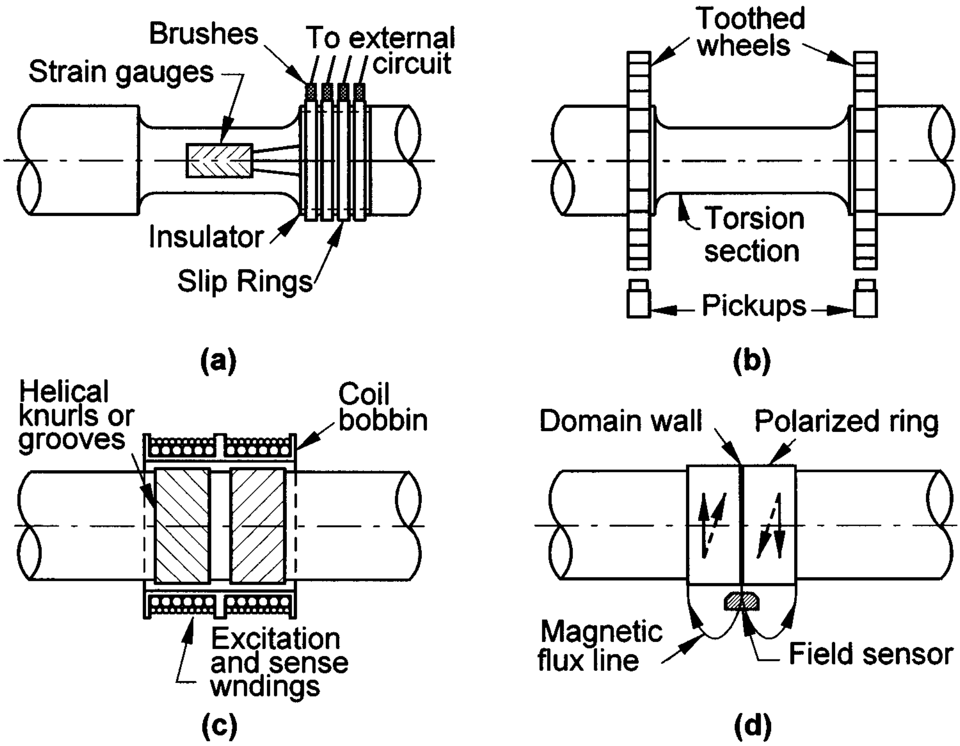

FIGURE 24.3 Four techniques in present use for measuring transmitted torque. (*a*) Torsional strain in the shaft alters the electrical resistance for four strain gages (two not seen) connected in a Wheatstone bridge circuit. In the embodiment shown, electrical connections are made to the bridge through slip rings and brushes. (*b*) Twist of the torsion section causes angular displacement of the surface features on the toothed wheels. This creates a phase difference in the signals from the two pickups. (*c*) The permeabilities of the two grooved regions of the shaft change oppositely with torsional stress. This is sensed as a difference in the output voltages of the two sense windings. (*d*) Torsional stress causes the initially circumferential magnetizations in the ring (solid arrows) to tilt (dashed arrows). These helical magnetizations cause magnetic poles to appear at the domain wall and ring ends. The resulting magnetic field is sensed by the field sensor.

Stress

In addition to elastic strain, the stresses by which torque is transmitted are manifested by changes in the magnetic properties of ferromagnetic shaft materials. This "magnetoelastic interaction" [8] provides an inherently noncontacting basis for measuring torque. Two types of magnetoelastic (sometimes called magnetostrictive) torque transducers are in present use: Type 1 derive output signals from torque-induced variations in magnetic circuit permeances; Type 2 create a magnetic field in response to torque. Type 1 transducers typically employ "branch," "cross," or "solenoidal" constructions [9]. In branch and cross designs, torque is detected as an imbalance in the permeabilities along orthogonal 45° helical paths (the principal stress directions) on the shaft surface or on the surface of an *ad hoc* material attached to the shaft. In solenoidal constructions torque is detected by differences in the *axial* permeabilities of two adjacent surface regions, preendowed with symmetrical magnetic "easy" axes (typically along the 45° principal stress directions). While branch and cross type sensors are readily miniaturized [10], local variations in magnetic properties of typical shaft surfaces limit their accuracy. Solenoidal designs, illustrated in Figure 24.3(*c*), avoid this pitfall by effectively averaging these variations. Type 2 transducers are generally constructed with a ring of magnetoelastically active material rigidly attached to the shaft. The ring is magnetized during manufacture of the transducer, usually with each axial half polarized in an

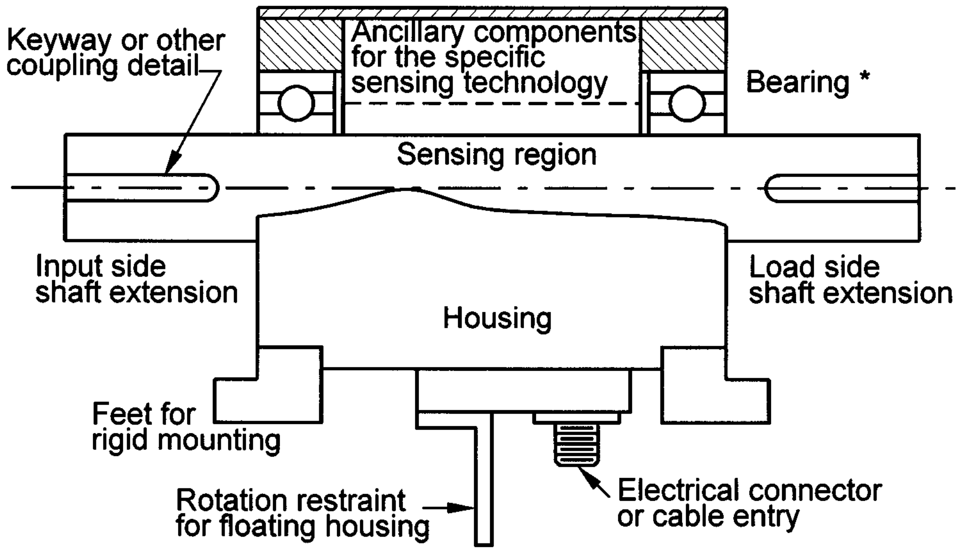

FIGURE 24.4 Modular torque transducer showing generic features and alternative arrangements for free floating or rigid mounting. Bearings* are used only on rotational models. Shaft extensions have keyways or other features to facilitate torque coupling.

opposite circumferential direction as indicated by the solid arrows in Figure 24.3(*d*) [11]. When torque is applied, the magnetizations tilt into helical directions (dashed arrows), causing magnetic poles to develop at the central domain wall and (of opposite polarity) at the ring end faces. Torque is determined from the output signal of one or more magnetic field sensors (e.g., Hall effect, magnetoresistive, or flux gate devices) mounted so as to sense the intensity and polarity of the magnetic field that arises in the space near the ring.

24.4 Torque Transducer Construction, Operation, and Application

Although a torque sensing region can be created directly on a desired shaft, it is more usual to install a preassembled *modular* torque transducer into the driveline. Transducers of this type are available with capacities from 0.001 N·m to 200,000 N·m. Operating principle descriptions and detailed installation and operating instructions can be found in the catalogs and literature of the various manufactures [12–20]. Tradenames often identify specific type of transducers; for example, *Torquemeters* [13] refers to a family of noncontact strain gage models; *Torkducer®* [18] identifies a line of Type 1 magnetoelastic transducers; *Torqstar™* [12] identifies a line of Type 2 magnetoelastic transducers; *Torquetronic* [16] is a class of transducers using wrap-around twist angle sensors; and *TorXimitor™* [20] identifies optoelectronic based, noncontact, strain gage transducers. Many of these devices show generic similarities transcending their specific sensing technology as well as their range. Figure 24.4 illustrates many of these common features.

Mechanical Considerations

Maximum operating speeds vary widely; upper limits depend on the size, operating principle, type of bearings, lubrication, and dynamic balance of the rotating assembly. Ball bearings, lubricated by grease, oil, or oil mist, are typical. Parasitic torques associated with bearing lubricants and seals limit the accuracy of low-end torque measurements. (Minute capacity units have no bearings [15]). Forced lubrication can allow operation up to 80,000 rpm [16]. High-speed operation requires careful consideration of the effects of centrifugal stresses on the sensed quantity as well as of critical (vibration inducing) speed ranges. Torsional oscillations associated with resonances of the shaft elasticity (characterized by its spring constant) with the rotational inertia of coupled masses can corrupt the measurement, damage the transducer by dynamic excursions above its rated overload torque, and *even be physically dangerous.*

Housings either *float* on the shaft bearings or are *rigidly mounted.* Free floating housings are restrained from rotating by such "soft" means as a cable, spring, or compliant bracket, or by an eccentric external feature simply resting against a fixed surface. In free floating installations, the axes of the driving and driven shafts must be carefully aligned. Torsionally rigid "flexible" couplings at each shaft end are used to accommodate small angular and/or radial misalignments. Alternatively, the use of dual flexible couplings at one end will allow direct coupling of the other end. Rigidly mounted housings are equipped with mounting feet or lugs similar to those found on the frame of electric motors. Free-floating models are sometimes rigidly mounted using adapter plates fastened to the housing. Rigid mountings are preferred when it is difficult or impractical to align the driving and driven shafts, as for example when driving or driven machines are changed often. Rigidly mounted housings *require* the use of dual flexible couplings at *both* shaft ends.

Modular transducers designed for zero or limited rotation applications have no need for bearings. To ensure that *all* of the torque applied at the ends is sensed, it is important in such "reaction"-type torque transducers to limit attachment of the housing to the shaft to only one side of the sensing region. Whether rotating or stationary, the external shaft ends generally include such torque coupling details as flats, keyways, splines, tapers, flanges, male/female squares drives, etc.

Electrical Considerations

By their very nature, transducers require some electrical input power or *excitation.* The "raw" output signal of the actual sensing device also generally requires "conditioning" into a level and format appropriate for display on a digital or analog meter or to meet the input requirements of data acquisition equipment. Excitation and signal conditioning are supplied by electronic circuits designed to match the characteristics of the specific sensing technology. For example, strain gage bridges are typically powered with 10 V to 20 V (dc or ac) and have outputs in the range of 1.5 mV to 3.0 mV per volt of excitation at the rated load. Raising these millivolt signals to more usable levels requires amplifiers having gains of 100 or more. With ac excitation, oscillators, demodulators (or rectifiers) are also needed. Circuit elements of these types are normal when inductive elements are used either as a necessary part of the sensor or simply to implement noncontact constructions.

Strain gages, differential transformers, and related sensing technologies require that electrical components be mounted *on* the torqued member. Bringing electrical power to and output signals from these components on rotating shafts require special methods. The most direct and common approach is to use conductive means wherein brushes (typically of silver graphite) bear against (silver) slip rings. Useful life is extended by providing means to lift the brushes off the rotating rings when measurements are not being made. Several "noncontacting" methods are also used. For example, power can be supplied via inductive coupling between stationary and rotating transformer windings [12–15], by the illumination of shaft mounted photovoltaic cells [20], or even by batteries strapped to the shaft [21] (limited by centrifugal force to relatively low speeds). Output signals are coupled off the shaft through rotary transformers, by frequency-modulated (infrared) LEDs [19, 20], or by radio-frequency (FM) telemetry [21]. Where shaft rotation is limited to no more than a few full rotations, as in steering gear, valve actuators or oscillating mechanisms, hard wiring both power and signal circuits is often suitable. Flexible cabling minimizes incidental torques and makes for a long and reliable service life. All such wiring considerations are avoided when noncontact technologies or constructions are used.

Costs and Options

Prices of torque transducers reflect the wide range of available capacities, performance ratings, types, styles, optional features, and accessories. In general, prices of any one type increase with increasing capacity. Reaction types cost about half of similarly rated rotating units. A typical foot-mounted, 565 N·m capacity, strain gage transducer with either slip rings or rotary transformers and integral speed sensor, specified nonlinearity and hysteresis each within ±0.1%, costs about $4000 (1997). Compatible instrumentation providing transducer excitation, conditioning, and analog output with digital display of torque and speed costs about $2000. A comparable magnetoelastic transducer with ±0.5% accuracy costs about $1300. High-capacity transducers for extreme speed service with appropriate lubrication options can cost more than $50,000. Type 2 magnetoelastic transducers, mass produced for automotive power steering applications, cost approximately $10.

24.5 Apparatus for Power Measurement

Rotating machinery exists in specific types without limit and can operate at power levels from fractions of a watt to some tens of megawatts, a range spanning more than 10^8. Apparatus for power measurement exists in a similarly wide range of types and sizes. Mechanical power flows from a *driver* to a *load*. This power can be determined *directly* by application of Equation 24.7, simply by measuring, in addition to ω, the output torque of the driver or the input torque to the load, whichever is the device under test (DUT). When the DUT is a driver, measurements are usually required over its full service range of speed and torque. The test apparatus therefore must act as a controllable load and be able to *absorb* the delivered power. Similarly, when the DUT is a pump or fan or other type of load, or one whose function is simply to alter speed and torque (e.g., a gear box), the test apparatus must include a *driver* capable of supplying power over the DUT's full rated range of torque and speed. Mechanical power can also be determined *indirectly* by conversion into (or from) another form of energy (e.g., heat or electricity) and measuring the relevant calorimetric or electrical quantities. In view of the wide range of readily available methods and apparatus for accurately measuring both torque and speed, indirect methods need only be considered when special circumstances make direct methods difficult.

Dynamometer is the special name given to the power-measuring apparatus that includes absorbing or/and driving means and wherein torque is determined by the reaction forces on a stationary part (the *stator*). An effective dynamometer is conveniently assembled by mounting the DUT in such a manner as to allow measurement of the reaction torque on its frame. Figure 24.5 shows a device designed to facilitate such measurements. Commercial models (Torque Table® [12]) rated to support DUTs weighing 222 N to 4900 N are available with torque capacities from 1.3 N·m 226 to N·m. "Torque tubes" [4] or other DUT mounting arrangements are also used. Other than for possible rotational/elastic resonances, these systems have no speed limitations. More generally, and especially for large machinery, dynamometers include a specialized driving or absorbing machine. Such dynamometers are classified according to their function as *absorbing* or *driving* (sometimes *motoring*). A *universal dynamometer* can function as either a driver or an absorber.

Absorption Dynamometers

Absorption dynamometers, often called *brakes* because their operation depends on the creation of a controllable *drag* torque, convert mechanical work into heat. A drag torque, as distinguished from an active torque, can act only to restrain and not to initiate rotational motion. Temperature rise within a dynamometer is controlled by carrying away the heat energy, usually by transfer to a moving fluid, typically air or water. Drag torque is created by inherently dissipative processes such as: friction between rubbing surfaces, shear or turbulence of viscous liquids, the flow of electric current, or magnetic hysteresis. Gaspard Riche de Prony (1755–1839), in 1821 [22], invented a highly useful form of a friction brake to meet the needs for testing the steam engines that were then becoming prevalent. Brakes of this

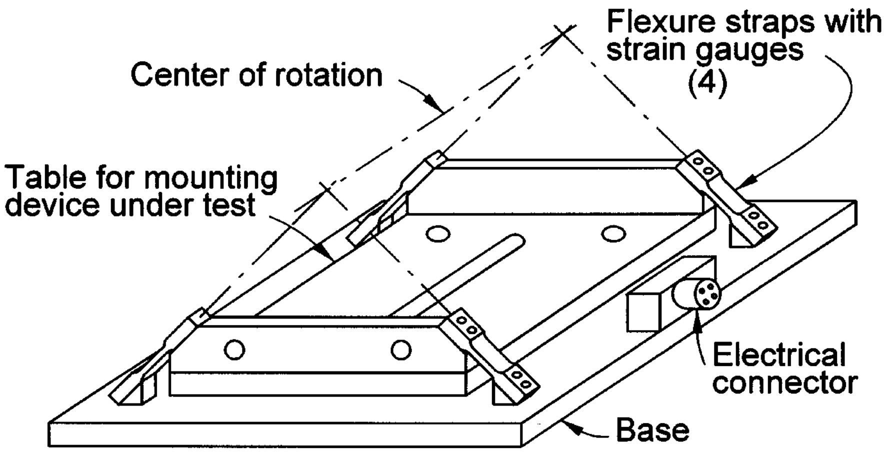

FIGURE 24.5 Support system for measuring the reaction torque of a rotating machine. The axis of the machine must be accurately set on the "center of rotation." The holes and keyway in the table facilitate machine mounting and alignment. Holes in the front upright provide for attaching a lever arm from which calibrating weights may be hung [4, 11].

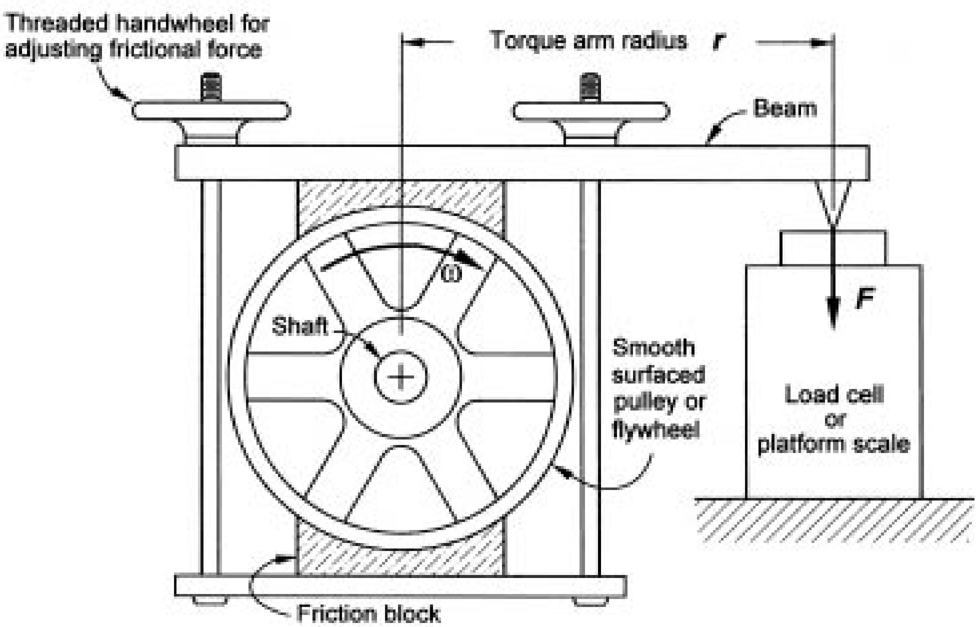

FIGURE 24.6 A classical prony brake. This brake embodies the defining features of all absorbing dynamometers: conversion of mechanical work into heat and determination of power from measured values of reaction torque and rotational velocity.

type are often used for instructional purposes, for they embody the general principles and major operating considerations for all types of absorption dynamometers. Figure 24.6 shows the basic form and constructional features of a *prony brake.* The power that would normally be delivered by the shaft of the driving engine to the driven load is (for measurement purposes) converted instead into heat via the work done by the frictional forces between the friction blocks and the flywheel rim. Adjusting the tightness of the clamping bolts varies the frictional drag torque as required. Heat is removed from the inside surface of the rim by arrangements (not shown) utilizing either a continuous flow or evaporation of water. There is no need to know the magnitude of the frictional forces nor even the radius of the flywheel (facts recognized by Prony), because, while the drag torque tends to rotate the clamped-on apparatus, it is held stationary by the equal but opposite reaction torque Fr. F at the end of the torque arm of radius r (a fixed dimension of the apparatus) is monitored by a scale or load cell. The power is found from Equations 24.1 and 24.7 as $P = Fr\omega = Fr2\pi N/60$ where N is in rpm.

Uneven retarding forces associated with fluctuating coefficients of friction generally make rubbing friction a poor way to generate drag torque. Nevertheless, because they can be easily constructed, *ad hoc* variations of prony brakes, often using only bare ropes or wooden cleats connected by ropes or straps, find use in the laboratory or wherever undemanding or infrequent power measurements are to be made. More sophisticated prony brake constructions are used in standalone dynamometers with self-contained cooling water tanks in sizes up to 746 kW (1000 Hp) for operation up to 3600 rpm with torques to 5400 N·m [23]. Available in stationary and mobile models, they find use in testing large electric motors as well as engines and transmissions on agricultural vehicles. Prony brakes allow full drag torque to be imposed down to zero speed.

William Froude (1810–1879) [24] invented a *water brake* (1877) that does not depend on rubbing friction. Drag torque within a *Froude brake* is developed between the rotor and the stator by the momentum imparted by the rotor to water contained within the brake casing. Rotor rotation forces the water to circulate between cup-like pockets cast into facing surfaces of both rotor and stator. The rotor is supported in the stator by bearings that also fix its axial position. Labyrinth-type seals prevent water leakage while minimizing frictional drag and wear. The stator casing is supported in the dynamometer frame in cradle fashion by *trunnion* bearings. The torque that prevents rotation of the stator is measured by reaction forces in much the same manner as with the prony brake. Drag torque is adjusted by a valve, controlling either the back pressure in the water outlet piping [25] or the inlet flow rate [26] or sometimes (to allow very rapid torque changes) with two valves controlling both [27]. In any case, the absorbed energy is carried away by the continuous water flow. Other types of cradle-mounted water brakes, while externally similar, have substantially different internal constructions and depend on other principles for developing the drag torque (e.g., smooth rotors develop viscous drag by shearing and turbulence). Nevertheless, all *hydraulic dynamometers* purposefully function as *inefficient* centrifugal pumps. Regardless of internal design and valve settings, maximum drag torque is low at low speeds (zero at standstill) but can rise rapidly, typically varying with the square of rotational speed. The irreducible presence of some water, as well as windage, places a speed-dependent lower limit on the *controllable* drag torque. In any one design, wear and vibration caused by cavitation place upper limits on the speed and power level. Hydraulic dynamometers are available in a wide range of capacities between 300 kW and 25,000 kW, with some portable units having capacities as low as 75 kW [26]. The largest ever built [27], absorbing up to about 75,000 kW (100,000 Hp), has been used to test propulsion systems for nuclear submarines. Maximum speeds match the operating speeds of the prime movers that they are built to test and therefore generally decrease with increasing capacity. High-speed gas turbine and aerospace engine test equipment can operate as high as 30,000 rpm [25].

In 1855, Jean B. L. Foucault (1819–1868) [22] demonstrated the conversion of mechanical work into heat by rotating a copper disk between the poles of an electromagnet. This simple means of developing drag torque, based on *eddy currents,* has, since circa 1935, been widely exploited in dynamometers. Figure 24.7 shows the essential features of this type of brake. Rotation of a toothed or spoked steel rotor through a spatially uniform magnetic field, created by direct current through coils in the stator, induces locally circulating (eddy) currents in electrically conductive (copper) portions of the stator. Electromagnetic forces between the rotor, which is magnetized by the uniform field, and the field arising from the eddy currents, create the drag torque. This torque, and hence the mechanical input power, are controlled by adjusting the *excitation* current in the stator coils. Electric input power is less than 1% of the rated capacity. The dynamometer is effectively an internally short-circuited generator because the power associated with the resistive losses from the generated eddy currents is dissipated *within* the machine.

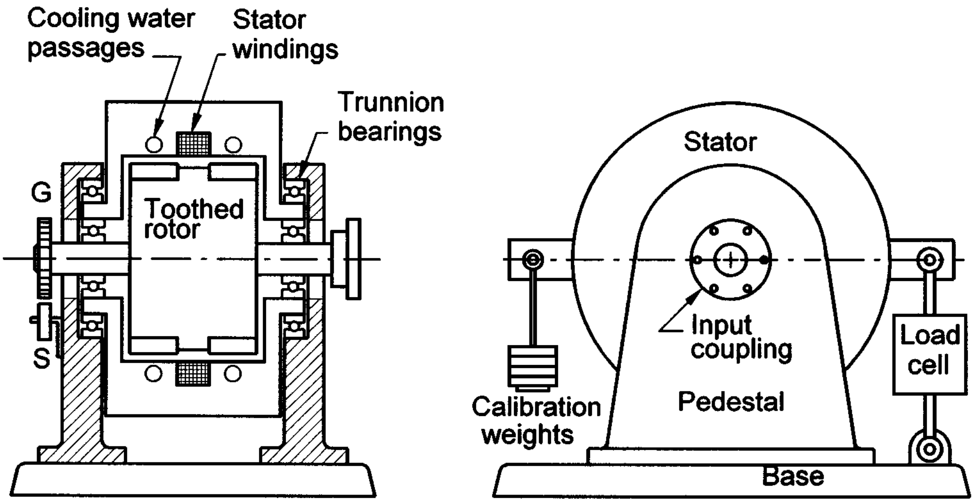

FIGURE 24.7 Cross-section (left) and front view (right) of an eddy current dynamometer. G is a gear wheel and S is a speed sensor. Hoses carrying cooling water and cable carrying electric power to the stator are not shown.

Being heated by the flow of these currents, the stator must be cooled, sometimes (in smaller capacity machines) by air supplied by blowers [23], but more often by the continuous flow of water [25, 27, 28]. In *dry gap* eddy current brakes (the type shown in Figure 24.7), water flow is limited to passages within the stator. Larger machines are often of the *water in gap* type, wherein water also circulates around the rotor [28]. Water in contact with the moving rotor effectively acts as in a water brake, adding a nonelectromagnetic component to the total drag torque, thereby placing a lower limit to the controllable torque. Windage limits the minimum value of controllable torque in dry gap types. Since drag torque is developed by the motion of the rotor, it is zero at standstill for any value of excitation current. Initially rising rapidly, approximately linearly, with speed, torque eventually approaches a current limited saturation value. As in other cradled machines, the torque required to prevent rotation of the stator is measured by the reaction force acting at a fixed known distance from the rotation axis. Standard model eddy current brakes have capacities from less than 1 kW [23, 27] to more than 2000 kW [27, 28], with maximum speeds from 12,000 rpm in the smaller capacity units to 3600 rpm in the largest units. Special units with capacities of 3000 Hp (2238 kW) at speeds to 25,000 rpm have been built [28].

Hysteresis brakes [29] develop drag torque via magnetic attractive/repulsive forces between the magnetic poles established in a reticulated stator structure by a current through the field coil, and those created in a "drag cup" rotor by the stator field gradients. Rotation of the special steel rotor, through the spatial field pattern established by the stator, results in a cyclical reversal of the polarity of its local magnetizations. The energy associated with these reversals (proportional to the area of the hysteresis loop of the rotor material) is converted into heat within the drag cup. Temperature rise is controlled by forced air cooling from a blower or compressed air source. As with eddy current brakes, the drag torque of these devices is controlled by the excitation current. In contrast with eddy current brakes, rated drag torque is available down to zero speed. (Eddy current effects typically add only 1% to the drag torque for each 1000 rpm). As a result of their smooth surfaced rotating parts, hysteresis brakes exhibit low parasitic torques and hence cover a dynamic range as high as 200 to 1. Standard models are available having continuous power capacities up to 6 kW (12 kW with two brakes in tandem cooled by two blowers). Intermittent capacities per unit (for 5 min or less) are 7 kW. Some low-capacity units are convection cooled; the smallest has a continuous rating of just 7 W (35 W for 5 min). Maximum speeds range from 30,000 rpm for the smallest to 10,000 rpm for the largest units. Torque is measured by a strain gage bridge on a moment arm supporting the machine stator.

Driving and Universal Dynamometers

Electric generators, both ac and dc, offer another means for developing a controllable drag torque and they are readily adapted for dynamometer service by cradle mounting their stator structures. Moreover, electric machines of these types can also operate in a motoring mode wherein they can deliver controllable *active* torque. When configured to operate selectively in either driving or absorbing modes, the machine serves as a universal dynamometer. With dc machines in the absorbing mode, the generated power is typically dissipated in a convection-cooled resistor bank. Air cooling the machine with blowers is usually adequate, since *most* of the mechanical power input is dissipated externally. Nevertheless, *all* of the mechanical input power is accounted for by the product of the reaction torque and the rotational speed. In the motoring mode, torque and speed are controlled by adjustment of both field and armature currents. Modern ac machines utilize regenerative input power converters to allow braking power to be returned to the utility power line. In the motoring mode, speed is controlled by high-power, solid-state, adjustable frequency inverters. Internal construction is that of a simple three-phase induction motor, having neither brushes, slip rings, nor commutators. The absence of rotor windings allows for higher speed operation than dc machines. Universal dynamometers are "four-quadrant" machines, a term denoting their ability to produce torque in the same or opposite direction as their rotational velocity. This unique ability allows the effective drag torque to be reduced to zero at any speed. Universal dynamometers [25, 28] are available in a relatively limited range of capacities (56 to 450 kW), with commensurate torque (110 to 1900 N·m) and speed (4500 to 13,500 rpm) ranges, reflecting their principal application in automotive engine development. Special dynamometers for testing transmissions and other vehicular drive train components insert the DUT between a diesel engine or electric motor prime mover and a hydraulic or eddy current brake [30].

Measurement Accuracy

Accuracy of power measurement (see discussion in [4]) is generally limited by the torque measurement ($\pm$0.25% to $\pm$1%) since rotational speed can be measured with almost any desired accuracy. Torque errors can arise from the application of extraneous (i.e., not indicated) torques from hose and cable connections, from windage of external parts, and from miscalibration of the load cell. Undetected friction in the trunnion bearings of cradled dynamometers can compromise the torque measurement accuracy. Ideally, well-lubricated antifriction bearings make no significant contribution to the restraining torque. In practice, however, the unchanging contact region of the balls or other rolling elements on the bearing races makes them prone to brinelling (a form of denting) from forces arising from vibration, unsupported weight of attached devices, or even inadvertently during the alignment of connected machinery. The problem can be alleviated by periodic rotation of the (primarily outer) bearing races. In some bearing-in-bearing constructions, the central races are continuously rotated at low speeds by an electric motor while still others avoid the problem by supporting the stator on hydrostatic oil lift bearings [28].

Costs

The wide range of torque, speed, and power levels, together with the variation in sophistication of associated instrumentation, is reflected in the very wide range of dynamometer prices. Suspension systems of the type illustrated in Figure 24.5 (for which the user must supply the rotating machine) cost $4000 to $6000, increasing with capacity [12]. A 100 Hp (74.6 kW) *portable* water brake equipped with a strain gage load cell and a digital readout instrument for torque, speed, and power costs $4500, or $8950 with more sophisticated data acquisition equipment [26]. Stationary (and some *transportable* [23]) hydraulic dynamometers cost from $113/kW in the smaller sizes [25], down to $35/kW for the very largest [27]. Transportation, installation, and instrumentation can add significantly to these costs. Eddy current dynamometers cost from as little as $57/kW to nearly $700/kW, depending on the rated capacity, type of control system, and instrumentation [24, 25, 28]. Hysteresis brakes with integral speed sensors cost from \$3300 to \$14,000 according to capacity [29]. Compatible controllers, from manual to fully programmable for PC test control and data acquisition via an IEEE-488 interface, vary in price from \$500 to \$4200. The flexibility and high performance of ac universal dynamometers is reflected in their comparatively high prices of \$670 to \$2200/kW [25, 28].

References

1. Pinney, C. P. and Baker, W. E., Velocity Measurement, *The Measurement, Instrumentation and Sensors Handbook,* Webster, J. G., ed., Boca Raton, FL: CRC Press, 1999.
2. S. Timoshenko, *Strength of Materials,* 3rd ed., New York: Robert E. Kreiger, Part I, 281–290; Part II, 235–250, 1956.
3. S. J. Citron, *On-line engine torque and torque fluctuation measurement for engine control utilizing crankshaft speed fluctuations,* U. S. Patent No. 4,697,561, 1987.
4. Supplement to ASME Performance Test Codes, Measurement of Shaft Power, ANSI/ASME PTC 19.7-1980 (Reaffirmed 1988).
5. See, for example, the catalog of torque wrench products of Consolidated Devices, Inc., 19220 San Jose Ave., City of Industry, CA 91748.
6. I. J. Garshelis, C. R. Conto, and W. S. Fiegel, A single transducer for non-contact measurement of the power, torque and speed of a rotating shaft, SAE Paper No. 950536, 1995.
7. C. C. Perry and H. R. Lissner, *The Strain Gage Primer,* 2nd ed., New York: McGraw-Hill, 1962, 9. (This book covers all phases of strain gage technology.)
8. B. D. Cullity, *Introduction to Magnetic Materials,* Reading, MA: Addison-Wesley, 1972, Section 8.5, 266–274.
9. W. J. Fleming, Magnetostrictive torque sensors—comparison of branch, cross and solenoidal designs, SAE Paper No. 900264, 1990.
10. Y. Nonomura, J. Sugiyama, K. Tsukada, M. Takeuchi, K. Itoh, and T. Konomi, Measurements of engine torque with the intra-bearing torque sensor, SAE Paper No. 870472, 1987.
11. I. J. Garshelis, *Circularly magnetized non-contact torque sensor and method for measuring torque using same,* U.S. Patent 5,351,555, 1994 and 5,520,059, 1996.
12. Lebow® Products, Siebe, plc., 1728 Maplelawn Road, Troy, MI 48099, Transducer Design Fundamentals/Product Listings, Load Cell and Torque Sensor Handbook No. 710, 1997, also: Torqstar™ and Torque Table®.
13. S. Himmelstein & Co., 2490 Pembroke, Hoffman Estates, IL 60195, MCRT® Non-Contact Strain Gage Torquemeters and Choosing the Right Torque Sensor.
14. Teledyne Brown Engineering, 513 Mill Street, Marion, MA 02738-0288.
15. Staiger, Mohilo & Co. GmbH, Baumwasenstrasse 5, D-7060 Schorndorf, Germany (In the U.S.: Schlenker Enterprises Ltd., 5143 Electric Ave., Hillside, IL 60162), Torque Measurement.
16. Torquemeters Ltd., Ravensthorpe, Northampton, NN6 8EH, England (In the U.S.: Torquetronics Inc., P.O. Box 100, Allegheny, NY 14707), Power Measurement.
17. Vibrac Corporation, 16 Columbia Drive, Amherst, NH 03031, Torque Measuring Transducer.
18. GSE, Inc., 23640 Research Drive, Farmington Hills, MI 48335-2621, Torkducer®.
19. Sensor Developments Inc., P.O. Box 290, Lake Orion, MI 48361-0290, 1996 Catalog.
20. Bently Nevada Corporation, P.O. Box 157, Minden, NV 89423, TorXimitor™.
21. Binsfield Engineering Inc., 4571 W. MacFarlane, Maple City, MI 49664.
22. C. C. Gillispie (ed.), *Dictionary of Scientific Biography,* Vol. XI, New York: Charles Scribner's Sons, 1975.
23. AW Dynamometer, Inc., P.O. Box 428, Colfax, IL 61728, Traction dynamometers: Portable and stationary dynamometers for motors, engines, vehicle power take-offs.
24. Roy Porter (ed.), *The Biographical Dictionary of Scientists,* 2nd ed., New York: Oxford University Press, 1994.

25. Froude-Consine Inc., 39201 Schoolcraft Rd., Livonia, MI 48150, F Range Hydraulic Dynamometers, AG Range Eddy Current Dynamometers, AC Range Dynamometers.
26. Go-Power Systems, 1419 Upfield Drive, Carrollton, TX 75006, Portable Dynamometer System, Go-Power Portable Dynamometers.
27. Zöllner GmbH, Postfach 6540, D-2300 Kiel 14, Germany (In the U.S. and Canada: Roland Marine Inc., 90 Broad St., New York, NY 10004), Hydraulic Dynamometers Type P, High Dynamic Hydraulic Dynamometers.
28. Dynamatic Corporation, 3122 14th Ave., Kenosha, WI 53141-1412, Eddy Current Dynamometer—Torque Measuring Equipment, Adjustable Frequency Dynamometer.
29. Magtrol, Inc., 70 Gardenville Parkway, Buffalo, NY 14224-1322, Hysteresis Absorption Dynamometers.
30. Hicklin Engineering, 3001 NW 104th St., Des Moines, IA 50322, Transdyne™ (transmission test systems, brake and towed chassis dynamometers).