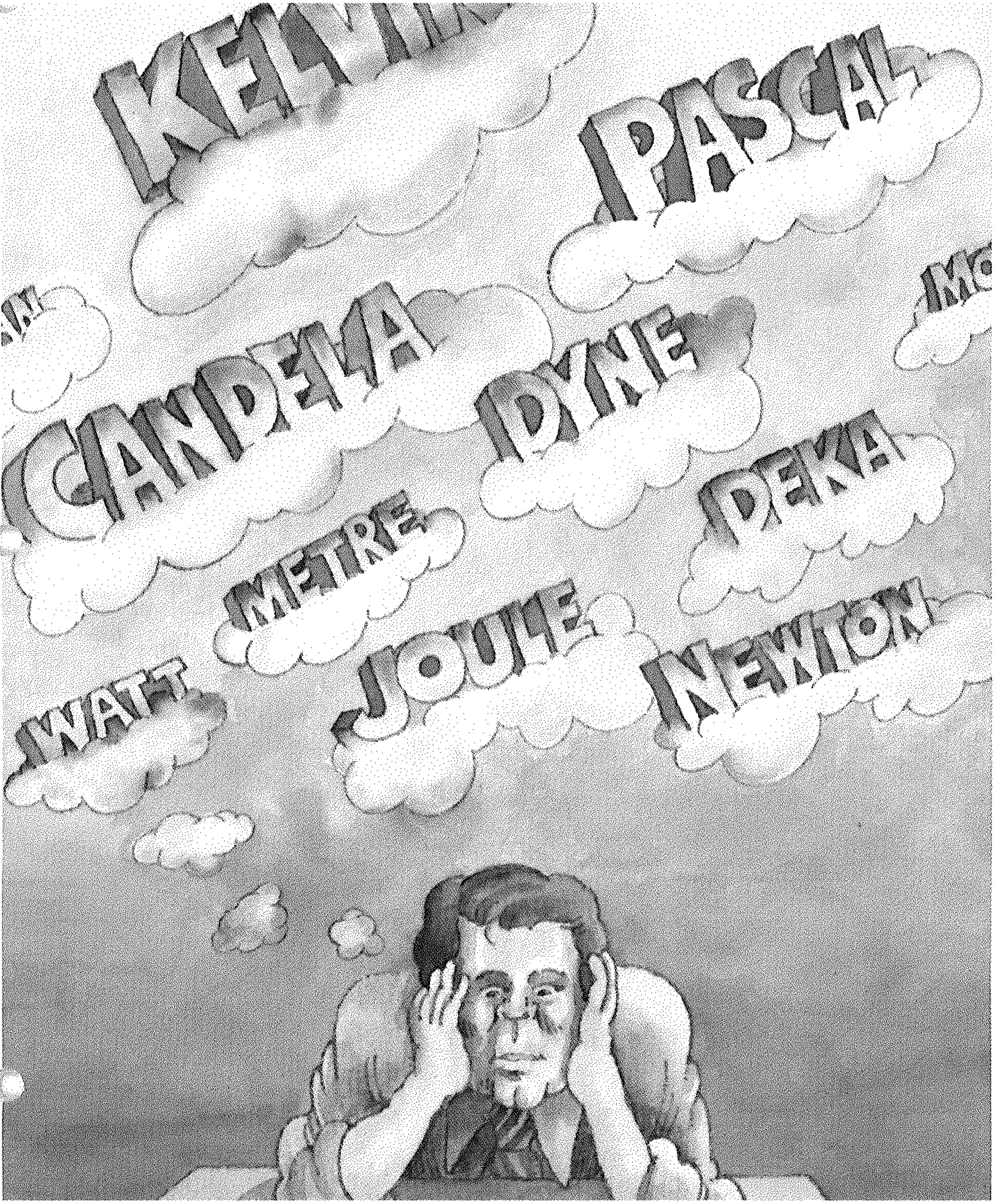
WORTHINGTON PUMP INC. 1977 VOLUME 3 / NUMBER 2

pumpworld



New Yorkers woke to face a cold, dismal morning. At least those New Yorkers who did not depend on electric alarm clocks woke – for there was no electricity. Nor was there heat or hot water. In fact, no water at all issued from the faucets.

There was no gas to cook breakfast, no TV. Transistor radios were of no help because no stations were broadcasting. Telephones were out and no newspapers published.

Hungry, grumpy, bewildered, unshaven men made their way down back staircases to the street (the elevators weren't running) to find no buses, no subways, no traffic at all. Those who had cars could not start them.

While a few hardy souls took off for their offices on foot, most congregated on street corners and asked each other questions. At first, another massive blackout was suspected, but it was obvious it was *more* than that. Before long, all sorts of rumors were flying.

By noon, although the sun was shining brightly, the situation was deteriorating fast. Rioting and looting began to break out. The police were handicapped because all normal communication channels had failed.

In cities and towns around the world, the picture was essentially the same. In the country, however, there was less panic because life was simpler. When farmers found no water in the faucets, they went back to their wells with buckets. There were only a few isolated parts of the world in which no one noticed any difference at all.

Fiction? Of course. But all this could happen if one simple mechanical device – the pump – stopped working. For without pumps, civilization as we know it today could not exist.

THE DAY THE PUMPS STOPPED RUNNING

pumpworld

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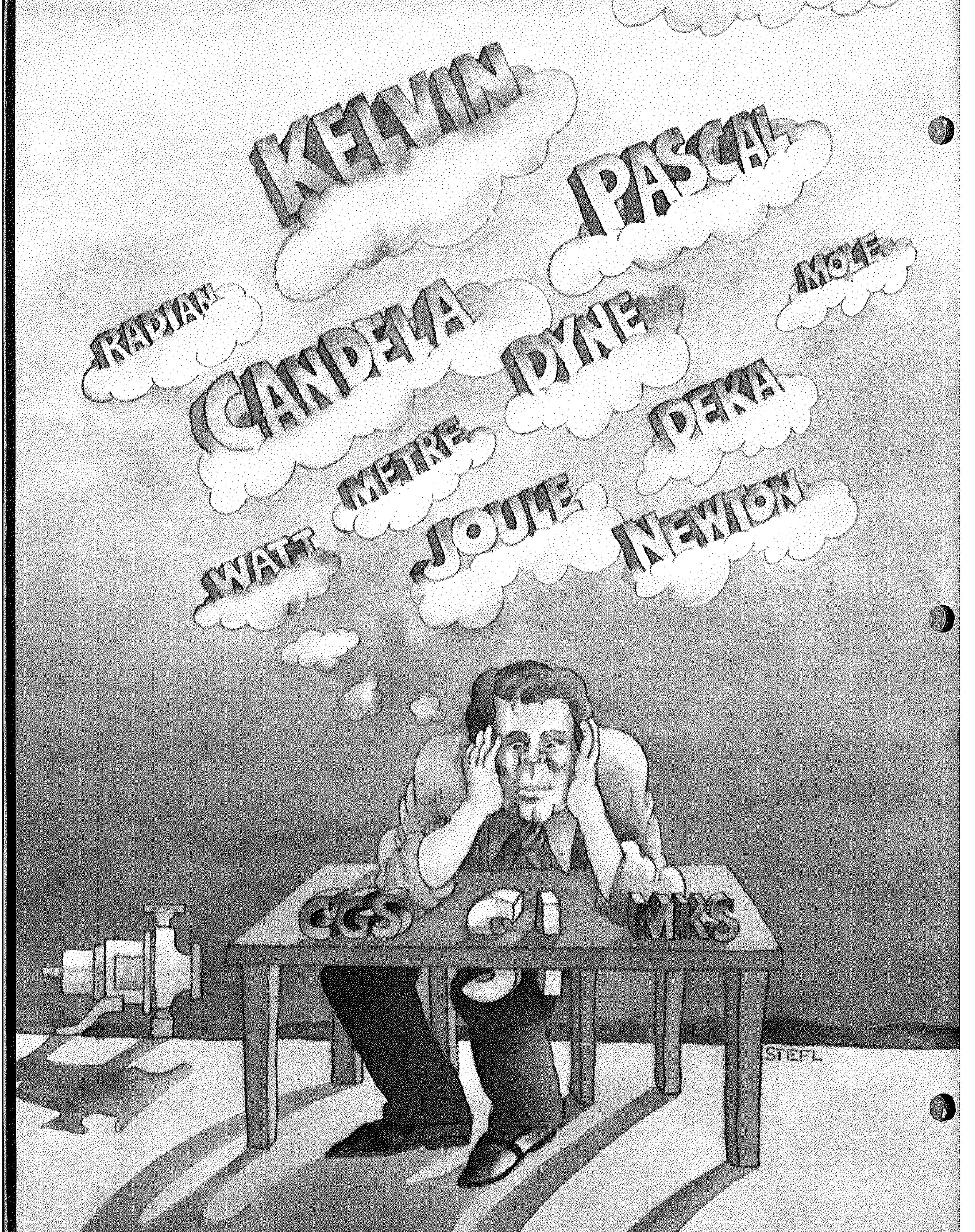
Inside cover:
From "The Fourth Invention"
by I. J. Karassik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Learning to live with S.I. units.

By W. C. Krutzsch

Increasingly, the world seems to be recognizing that it needs a single, unified system of measurement, eliminating the need for conversions between different systems and the opportunities for error which they create. As business has become more and more international in its conduct, the world has been responding to that need, and has been focusing its attention on metric units. Unfortunately the term "metric system" as such cannot fully define a system of measurement, since there are differing systems employing metric units. Historically, scientists have tended to work in the "cgs" (centimetre-gram-second) system, while engineers in metric countries have worked largely in metres, kilograms force, and seconds (the "mks" system). Someday, both of these systems will probably be superseded by the SI system.

The designation SI is the official abbreviation, for any language, of the French title "Le Système International d'Unités," given by the eleventh General Conference on Weights and Measures (sponsored by the International Bureau of Weights and Measures) in 1960 to a coherent system of units selected from metric systems. This system of units has since been adopted by ISO (International Organization for Standardization) as an international standard. The SI system consists of seven basic units, two supplementary units, a series of derived units, and a series of approved prefixes for multiples and submultiples of the foregoing.

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The basic units of the SI system are as follows:

1. Metre[1] (symbol m), unit of length, equal to 1,650,763.73 wavelengths in vacuum of the radiation corresponding to the transition between the levels $2p_{10}$ and $5d_5$ of the krypton-86 atom.
2. Kilogram (symbol kg), unit of mass, equal to that of the international prototype of the kilogram.
3. Second (symbol s), unit of time, equal to the duration of 9,192,631,770 periods of the radiation corresponding to the transition between the two hyperfine levels of the ground state of the cesium-133 atom.
4. Ampere (symbol A), unit of electric current, equal to that which, if maintained in two straight parallel conductors of infinite length, of negligible cross section, and placed one metre apart in vacuum, would produce between those conductors a force equal to 2×10^{-7} newtons per metre of length.[2]
5. Kelvin (symbol K), unit of thermodynamic temperature equal to the fraction 1/273.16 of the thermodynamic temperature of the triple point of water.
6. Candela (symbol cd), unit of luminous intensity, equal to that, in the perpendicular direction, of a surface of 1/600,000 square metres of a blackbody at the temperature of freezing platinum under a pressure of 101,325 newtons per square metre.
7. Mole (symbol mol), unit of substance, equal to the amount of substance of a system which contains as many elementary entities as there are atoms in 0.012 kilogram of carbon-12. (Note that when the mole is used, the elementary entities must be specified and may be atoms, molecules, ions, electrons, other particles, or specified groups of such particles.)

[1]More commonly *meter* in American usage, this spelling having been adopted by the American National Metric Council in 1975. Internationally, *metre* is still officially retained. The same situation exists with respect to the words *liter* and *litre*.

[2]Newton (symbol N), unit of force, equal to that which, when applied to a body having a mass of one kilogram, gives it an acceleration of one metre per second per second.

The two supplementary units of the SI system are as follows:

1. Radian (symbol rad), unit of measure of a plane angle with its vertex at the center of a circle and subtended by an arc equal in length to the radius.
2. Steradian (symbol sr), unit of measure of a solid angle with its vertex at the center of a sphere and enclosing an area of the spherical surface equal to that of a square with sides equal in length to the radius.

For the purposes of this discussion, only those derived units which are of importance in connection with pumps will be covered, but it will be apparent from those listed that some are obvious combinations of the basic and supplementary units, while others, having distinctive names, could be derived only on the basis of accurate knowledge of the physical significance of the quantities they measure. Samples of derived units which are pertinent to subjects covered in this handbook are given in Table 1.

Prefixes for multiples and submultiples of basic and derived units, many of which are already quite familiar, are given in Table 2 for the purpose of identifying those which may not be generally known, and also to indicate the standard SI symbols for the complete range.

A changing system.
As with any such system, SI units will come to be used with a feeling of confidence only after they have been worked with over a period of time. This not only may necessitate the teaching of the system in our educational institutions, or the widespread use of it in industry, but may also require evolutionary changes in the system itself. That some such changes will occur is practically assured by the

mere fact that the international General Conference on Weights and Measures has been meeting, since World War II, on the average of once every three years. That some changes may be desirable will be readily appreciated by considering the magnitude of some of the units encountered in the technology associated with pumps.

Order of magnitude.
Consider, for example, the standard SI unit of pressure, in comparison to the units used commonly today. The pascal, equal to one newton per square metre, is indeed a miniscule value compared to the pound per square inch (1 lb/in^2 = 6,894.757 Pa) or to the kilogram per square centimetre (1 kg/cm^2 = 98,066.50 Pa). In order to eliminate the necessity for dealing with significant multiples of these already large numbers when describing pressure ratings of modern pumps, a serious effort has been made by one interested industry group to obtain acceptance of the bar (1 bar = 10^5 Pa) as an additional standard derived unit. This would indeed accomplish the purpose of reducing the size of the numbers used to describe pressures. The unit is a round multiple of a true SI unit, but it has the disadvantage of being named to infer equality to atmospheric pressure, without being exactly equivalent. In fact, it is equal to neither the standard atmosphere (101,325.0 Pa) nor the so-called metric atmosphere (98,066.50 Pa), but is close enough to be confused with both. It does not present a neat solution.

Order of magnitude is also a problem with the standard SI unit for measuring pump capacity, since in this case it is very large. The cubic metre per second is equal to 15,850.32 U.S. gallons per minute, on the basis of which all but a tiny fraction of pumps now being manufactured would carry a rated capacity value of less than 1.0. Having long recognized the undesirability of dealing with such large units of capacity, pump suppliers in metric countries have generally adopted the practice of rating all but the largest pumps in cubic metres per hour, which gets the numbers up to a more manageable range.

The examples given in the two preceding paragraphs illustrate one more point of considerable interest — the fact that adoption of SI units will require some adjustment of practices in the metric world as well as in the heretofore nonmetric countries.

There will be no country, however, where the impact of this adjustment will be more severe than in the United States. Nevertheless, and in spite of the lack of official government action, the trend here toward adoption of metric measurement, and apparently toward the SI system in particular, is accelerating. Some understanding of the system is thus becoming mandatory, and this commentary is intended to provide at least an introduction to the subject.

In addition, Table 3 will help, where it is necessary, to convert values expressed in U.S. engineering units into the equivalent SI values.

Adapted from a commentary originally prepared by the author for "Pump Handbook," and reproduced here with the permission of McGraw-Hill.

Quantity	Unit (Symbol)	Formula
acceleration	metre/second^2	m/s^2
angular acceleration	radian/second^2	rad/s^2
angular velocity	radian/second	rad/s
area	metre^2	m^2
density	kilogram/metre^3	kg/m^3
energy	joule (J)	N • m
force	newton (N)	kg • m/s^2
frequency	hertz (Hz)	cycle/s
power	watt (W)	J/s
pressure	pascal (Pa)	N/m^2
stress	newton/metre^2	N/m^2
velocity	metre/second	m/s
viscosity, dynamic	newton-second/metre^2	N • s/m
viscosity, kinematic	metre^2/second	m^2/s
volume	metre^3	m^3
work	joule (J)	N • m

Table 1 — Typical Derived Units of the SI System

Prefix	SI Symbol	Multiplication Factor
tera	T	10^{12}
giga	G	10^9
mega	M	10^6
kilo	k	10^3
hecto	h	10^2
deka	da	10
deci	d	10^{-1}
centi	c	10^{-2}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}
femto	f	10^{-15}
atto	a	10^{-18}

Table 2 — Prefixes for SI Multiple and Submultiple Units

Multiply	By	To obtain
atmosphere (normal)	1.013250×10^{5}	pascal
barrel (oil, 42 U.S. gal)	1.589873×10^{-1}	metre^3
BTU (International Table)	1.055056×10^{3}	joule
centimetre of Hg (0 C)	1.333220×10^{3}	pascal
centipoise	1.000000×10^{-3}	newton-second/metre^2
centistoke	1.000000×10^{-6}	metre^2/second
degree (of angle)	1.745329×10^{-2}	radian
degree (Celsius)	$t_K = t_C + 273.15$	kelvin
degree (Fahrenheit)	$t_K = (t_F + 459.67)/1.8$	kelvin
degree (Rankine)	$t_K = t_R/1.8$	kelvin
dyne	1.000000×10^{-5}	newton
fluid ounce (U.S.)	2.957353×10^{-5}	metre^3
foot	3.048000×10^{-1}	metre
foot of water (39.2 F)	2.988980×10^{3}	pascal
foot of water (60 F)	2.986080×10^{3}	pascal
foot/second^2	3.048000×10^{-1}	metre/second^2
foot-pound-force	1.355818	joule
foot-pound-force/minute	2.259697×10^{-2}	watt
foot-pound-force/second	1.355818	watt
foot^2/second	9.290304×10^{-2}	metre^2/second
foot^3/second	2.831685×10^{-2}	metre^3/second
gallon (Canadian liquid)	4.546122×10^{-3}	metre^3
gallon (U.K. liquid)	4.546087×10^{-3}	metre^3
gallon (U.S. liquid)	3.785412×10^{-3}	metre^3
gallon (U.S. liquid)/minute	6.309020×10^{-5}	metre^3/second
horsepower (550 ft-lb f/s)	7.456999×10^{2}	watt
horsepower (electric)	7.460000×10^{2}	watt
inch	2.540000×10^{-2}	metre
inch of mercury (32 F)	3.386389×10^{3}	pascal
inch of mercury (60 F)	3.376850×10^{3}	pascal
kilowatt-hour	3.600000×10^{6}	joule
mil	2.540000×10^{-5}	metre
millimetre of mercury (0 C)	1.333224×10^{2}	pascal
ounce-force-inch	7.061552×10^{-3}	newton-metre
pound-force	4.448222	newton
pound-force-foot	1.355818	newton-metre
pound-force/inch^2	6.894757×10^{3}	pascal
pound-force-second/foot^2	4.788026×10	newton-second/metre^2
pound-mass	4.535924×10^{-1}	kilogram
slug	1.459390×10	kilogram
ton (long, 2,240 lb-m)	1.016047×10^{3}	kilogram
ton (short, 2,000 lb-m)	9.071847×10^{2}	kilogram
volt (international of 1948)	1.000330	volt (absolute)
watt (international of 1948)	1.000165	watt
watt-hour	3.600000×10^{3}	joule

Table 3 — Conversion of U.S. to SI Units

The economics of variable-speed pumping with speed-changing drives.

Part II

By J. R. Bower

There are times when it is beneficial to be able to vary the speed of a centrifugal pump. You might be forced to specify the pump before system conditions can be finalized, or you might anticipate a system change at a later date. In some systems, the conditions of service are continually changing, as in water supply, process, or some automatic-response situations. If you need variable speed, there are many ways to achieve it — some very costly. In the last issue we reviewed the economics of variable speed pumping. Now let's take a closer look at the characteristics of various types of drives to help you make a sound ball-park selection of the best one to meet your needs.

The ranges and main characteristics of all common drives are summarized in the table on the following pages. It shows efficiencies and power factors you can expect, and the applications for which they are best suited. The table has been designed to enable you to make a ball park selection. To determine the best one for your needs, carefully compare details of all drives that seem suitable.

The price factor over standard machines uses as its basis pumps with 2-pole totally enclosed, fan cooled induction motor, coupling, guard, baseplate and starter. The 5, 20 and 50 h.p. selections being end-suction centrifugals and the 200 h.p. a split case pump. If the more costly split case pumps are used throughout, the additional cost of a drive will be a lower proportion.

J. R. Bower is project engineer at Worthington-Simpson, United Kingdom.

Single speed change.
Belt drive is the simplest way to change speed between driver and pump. The pump shaft loading from a belt can present problems; above about 4 kw it is usually necessary to use a jack shaft extension to the motor shaft. If using vee belts, the standards on length can allow multibelt drives to have some loose belts while others are the correct tension. Resist the tendency to tension the belts until the slackest is correct, which produces excessive shaft loading.

Flat belt drive overcomes the tensioning problem but is restricted to the same power values as vee belts. Timing belts allow powers up to about 20 kw without jack shaft because of a lower belt tension. Timing belts can be noisy because of an air-squeezing effect where the belt teeth enter the pulley grooves. Compared to other speed-changing methods, the cost of a belt drive is low, but a larger base plate is needed, as well as coupling guard and often a jack shaft. The cost of the pulleys and vee belts may be small compared to these auxiliaries. Flat belts cost about the same as vee belts; timing belts and pulleys about twice as much.

Efficiency is about 90% for vee belts, and higher for timing belts. Maximum ranges for vee belts are up to approximately 75 kw; 8:1 speed ratio. For timing belts the maximum power is 350 kw; 9:1 speed ratio. The power factor will fall at reduced speed due to reduced loading on the induction motor. Single ratio gear boxes are available to cover any power or speed ratio required. Benefits over belt drive include compactness, no loss of speed through the drive and generally improved efficiency. On the other hand, gear boxes are more expensive except at the lowest powers. They also have a higher noise level and generally are likely to require more maintenance.

Infinitely variable-speed drives.
These drives may be mechanically installed between pump and motor, or they may take the form of electrical speed control.

Variable-speed pulleys.
In the simplest form, spacing of the conical surfaces of the driving pulley can be adjusted while stationary to allow small changes of speed ratio: up to about 2:1. The motor must be on a sliding base. A more sophisticated design uses spring-loaded cones which are opened by increasing belt tension, and can only be varied while the drive is in operation.

The inherent problem with variable-speed pulleys is fretting due to cyclical loading of the sliding members — partially solved by use of hardened materials or special lubrication. Belt life is typically 18 months. Efficiency is similar to vee belt drives. The cost of the simplest variable-pulley drive is little more than a standard belt drive installation. Prices range up from there depending on the degree of sophistication involved.

Eddy-current drives.
These high-loss drives permit reduction from the drive motor speed only. They are most suitable for operation at a fairly constant maximum speed with only small variations. Input and output members in the coupling rotate in the magnetic field set up by a dc excitation current in a stationary coil. Eddy currents are generated and produce secondary magnetic fields, resulting in torque transmission between input and output shafts. The input shaft runs at a constant speed, with speed control achieved by increasing the slip by varying the dc excitation current.

For low power applications, belt drive from a top- or side-mounted motor can be used. Up to about 120 h.p., the coupling can be flange-mounted to an induction motor with special shaft extension. Foot-mounted models are available up to 2000 h.p. The eddy-current drive is air-cooled up to 75 h.p. with water cooling at higher powers for improved heat dissipation. The drive offers good flexibility, with such options as braking; tachogenerator on the output shaft to produce a feedback signal for accurate speed control and remote control by signal from an external source. Operating life expectancy is good. The

Variable speed drive applied to raw sewage pumping.

only problems are low efficiency at reduced speed, and the need for water cooling at higher powers.

Fluid couplings.
These are the hydraulic equivalent of eddy-current couplings. Slip between input and output shafts is controlled by varying the amount of oil transmitted through the working circuit of the coupling. Fluid couplings are high-loss drives; heat must be dissipated, and a separate oil cooler is used on the larger units when the mass of the coupling is no longer adequate to remove the heat buildup. The coupling itself can only reduce speed from the input level, but gearing may be used for higher speeds. Speed ratio is restricted to about 4:1. Fluid couplings are available for drives from 20 to 15,000 h.p. or higher using parallel drives to share the load between two couplings. This drive has no rubbing parts, resulting in high reliability.

Positive infinitely variable chain drives.
Originally, this term covered a drive system with a pair of variable valley pulleys expended and compressed simultaneously to allow fixed shaft centers and a positive, slip-free drive with a slatted chain acting in radial grooves on the pulley cones. This system has limitations: correct lubrication is essential to ensure a life of about 10,000 hours; the chains have wearing parts and are heavy, limiting maximum speed. Speed can only be varied while the drive is moving, in a range of 6:1.

A newer type drives by friction between the ends of the chain links and smooth cone faces on the pulleys. While still called positive infinitely variable, the drive is no longer slip-free. A modified, lighter chain increases the power range. Speed ratio is increased to 10:1, and can be varied over the full range in about 5 seconds. Although the speed will only change in operation, required speed can be preset with the drive stationary. This type of drive must be derated by a factor of about .7 for continuous operation near full speed.

(Text continued on page 12)

Ranges and main characteristics of speed changing drives.

TYPE OF DRIVE	POWER RANGE (hp)	SPEEDS		DRIVE EFFICIENCY %		OVERALL EFFICIENCY % INCLUDING MOTOR			
		max rpm	ratio	max. speed 5hp - 200hp	half speed 5hp - 200hp	max. speed 5hp	max. speed 200hp	half speed 5hp	half speed 200hp
'V' Belts or Flat Belts	up to 1,000	5,000 at Limited Power	8:1	'V' Belts 85 - 90 Flat Belts 90 - 95		70 75	80 85	40 45	65 70
Timing Belts	up to 500	6,000 at Limited Power	9:1	95+		80	87	50	75
Gear Box	any	Any but standard units. Usually step down.		95+		80	87	50	75
Variable Speed Pulleys	up to 175	up to 4,500 but at Limited Power	4:1 Std but up to 15:1	85 - 90		70	N.A.	45	N.A.
Eddy Current Coupling	up to 2,000	up to 4,200 with belt drive	above 10:1	96	45 - 50	80	87	25	35
Fluid Couplings	20 to 15,000	3,500 Std.	4:1	95	45 - 50	80	87	25	35
Positive Infinitely Variable Chain Drives	up to 160	up to 5,500 but at Limited Power	10:1	85 - 90		70	N.A.	45	N.A.
Mechanical Variators	up to 100	4,200 at 15HP 2,600 at 100HP	up to 12:1	75 - 90 (depending on speed ratio)		70	N.A.	40	N.A.
D.C. Motor and Thyristor Voltage Control	up to 3,000	1,500 Std. 3,500 avail. higher special	up to 30:1	See overall efficiency.		80	90	45	65
Voltage Control of Induction Motor	up to 70	3,000	4:1	See overall efficiency.		80	N.A.	25	N.A.
Frequency Control of Induction Motor	up to 100	5,000	10:1	See overall efficiency.		80	N.A.	45	N.A.
Schrage A.C. Commutator Motor	up to 500	2,500	4:1 Std. up to 20:1	See overall efficiency.		80	90	45	[illegible]
Rotor-Fed A.C. Commutator Motor	up to 3,000	2,000	up to 10:1	See overall efficiency.		80	90	45	70

Speed changing drives-applications

A. Meeting conditions of service between 2 and 4-pole motor speeds or changing speed to suit an existing non standard speed prime mover.

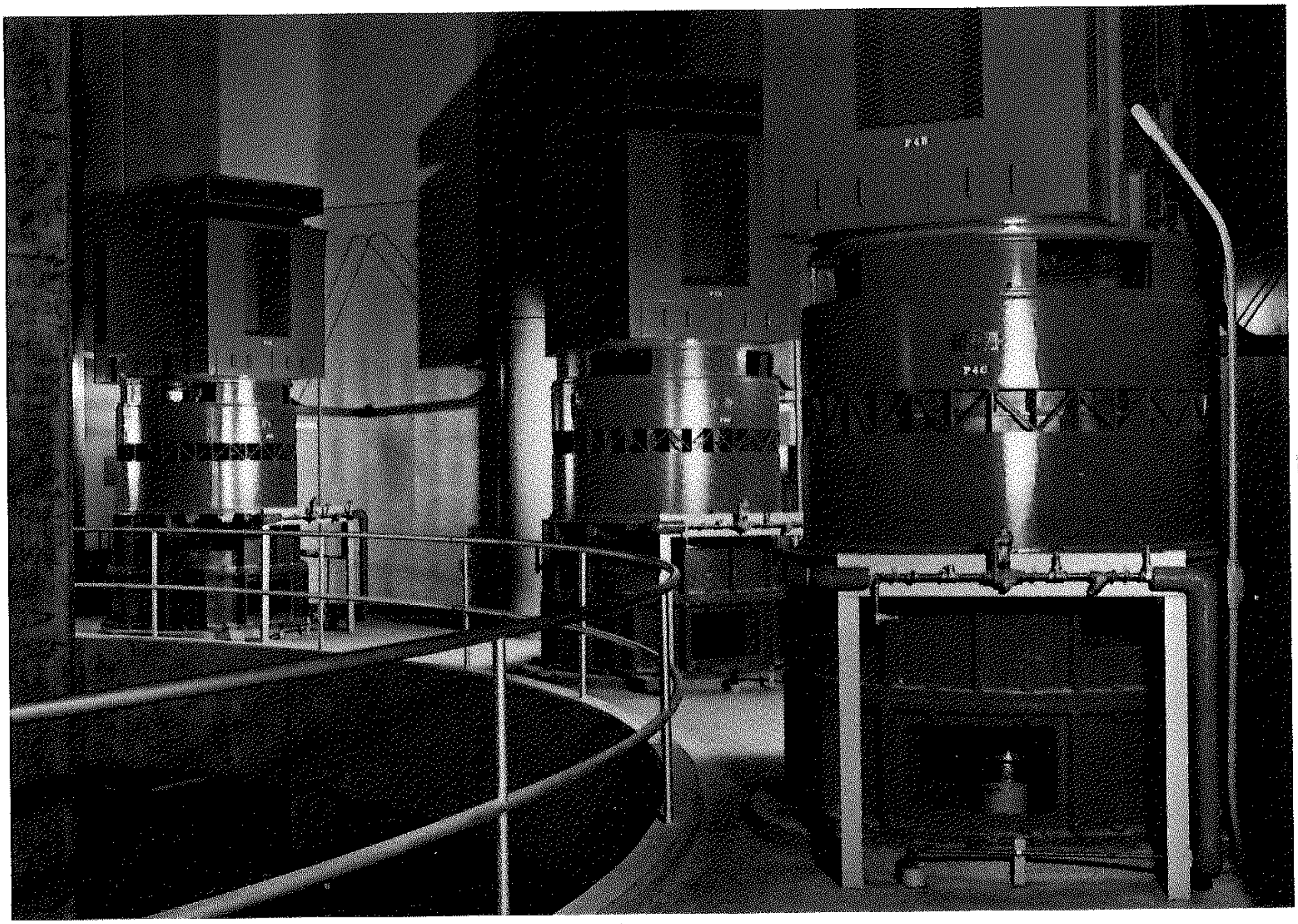
B. To change the pump service after installation.

C. Increasing capacity of a given pump size by running above 2-pole motor speed.

D. Holding down overall noise level with a low speed prime mover and a speed increasing drive to the pump.

E. Infinitely variable speed to meet changing output demand with minimum running costs.

POWER FACTOR		APPLICATIONS SATISFIED					APPROX. PRICE FACTOR OVER STANDARD PUMP UNIT				MAIN CHARACTERISTICS
max. speed	half speed	A	B	C	D	E	5hp	20hp	50hp	200hp	
0.9	0.3 with same motor	*	*	*	*		1.6	1.5	1.4	1.2	Increased floor space. Not suitable for outside application. Reduced bearing life if jack shaft not used.
0.9	0.3 with same motor	*	*	*	*		1.7	1.6	1.5	1.2	Similar limitations to 'V' belts with reduced shaft loading and greater efficiency.
0.9	0.3 with same motor	*		*			1.6	1.6	2.4	1.5	Compact. Any horsepower. Increased servicing requirements.
0.9	0.3	*	*	*	*	*	3/4	2/3	3/4	N.A.	Limited power and speed ranges. Automatic control difficult. Belt life limited.
0.9	.35	*	*			*	3.6	4.6	4.0	2.5	Low efficiency at reduced speed. Reliability life and automatic control good.
0.9	.35	*	*			*	N.A.	6	5	2	Low efficiency at reduced speed. Automatic control expensive. Heat exchanger required. Expensive for low powers.
0.9	0.3	*	*	*		*	5.3	5.3	7.7	4.7 (160) HP	Limited power range. Life limitations. Careful maintenance important.
0.9	0.3	*	*	*		*	3	3	5.5	N.A.	Limited power range. Life limitations. Correct lubrications and maintenance important.
0.9	0.3	*	*			*	4/5	4/5	5/6	3	Easy speed control. Good range. Low speed efficiency. Remote control convenient. TEFC expensive. Harmonic generation needs consideration.
0.9	0.3	*	*			*	2	2/3	3/4	N.A.	Simple and low cost. Very low efficiency. Motor needs derating.
1.0	1.0	*	*	*		*	5	9/10	10/11	N.A.	Standard induction motor possible. Good efficiency and power factor. High price.
0.9	0.6	*	*			*	3.3	3.1	3.4	4.3	Good efficiency and power factor. Remote control expensive. Brush and gear maintenance required. TEFC expensive.
0.9	0.6	*	*			*	–	3.2	3	3	Improved life over Schrage motor. Remote control expensive. TEFC expensive.



Magnetic drives for centrifugal pump installation.

Up to about 20 kw and full speed, the noise level of these drives is comparable to an induction motor of the same power; larger drives are relatively quieter. Efficiency is approximately 85% for the positive drive type and 90% for the more advanced type, and varies very little with load or speed. At low speed the power factor of the drive induction motor will decrease due to the reduction of load on the motor.

Electrical speed control.

With this type of variable-speed drive, the speed change is introduced through the motor.

DC motor drive.

This is the traditional way to obtain variable speed, because of the ease of control, over a range of 50:1 in some cases, by varying armature or field voltage or both. In one older system the armature voltage is supplied by a variable-voltage dc generator, itself driven by a synchronous or induction motor. This arrangement allows regenerative braking, withstands temporary heavy overloads, and does not feed back harmonics into the supply line.

If a synchronous motor is used, it maintains a good power factor. However, with three rotating machines in the system, the overall efficiency is low, noise is high, considerable floor space is required and initial cost is high. A newer arrangement is thyristor voltage control. Rectification of an ac supply is controlled by varying the thyristor's conducting time during each voltage cycle, and so altering the mean voltage to the motor armature. Efficiency is improved, but this arrangement is likely to generate harmonics. Additionally, dc motors are expensive, involve more maintenance, and are subject to tighter speed limitations. Drives from 1/3 to 3000 h.p. are standard. The reduction of motor cooling at low speed is partially met by the falloff in power requirements, but on large speed reductions, a separate cooling fan is needed on the motor.

Mechanical variators.
Drive is transmitted through balls or conical drive rollers; speed variation is achieved by the input and output drive rings which bear against the balls or rollers at varying diameters, so the peripheral speed at the points of contact varies and changes the driven speed. Speed variation is up to 12:1 at low power and 4:1 at maximum power. Efficiency is maximum at 1:1 speed ratio — about 85-90% — and falls to 75-80% at maximum and minimum output speeds. It is practically independent of load.

Correct lubrication is essential to maintain a hydrodynamic film between the drive components to achieve an adequate life. In a pump drive, motor output will fall with cube of speed and power factor is reduced as a result. Noise level is generally below that of an electric motor of similar power rating.

Voltage-variation controlled induction motor.
It's always advantageous, if you can, to select a system using a standard squirrel-cage induction motor with its high reliability, wide availability, low initial and maintenance costs and totally enclosed and fan-cooled enclosure.

The simplest way to control the speed of an induction motor is by varying line voltage; for example, with a thyristor control unit in which the firing time per cycle is varied to change the mean voltage. Problems? Speed variation is achieved by increasing slip, and at low voltage the magnetic fluxing is poor and rotor currents high, resulting in low efficiency and excessive heat generation. At low speed, fan cooling is also reduced, so it is necessary to derate a standard induction motor from 1.5 to 3 times when used this way. Although initial costs are low, this method has low efficiency at low speed, and is not suitable above about 70 h.p.

Instead of a derated standard induction motor, a special rotor motor can be used to withstand the high currents and heat buildup. However, the special rotor eliminates some of the benefits of this system, which are based on the use of a standard motor.

Frequency-variation controlled induction motor.
If both voltage and frequency are varied with a constant ratio, then speed can be controlled without variation of the maximum torque. The supply can be obtained from a variable-speed alternator or a rotary converter, but the more efficient electrical inverter, particularly the dc-link thyristor inverter, is most common.

Ac supply is rectified to give a controlled dc voltage, which is then "chopped" to produce variable frequency. Because the output waveform is not truly sinusoidal, motor heat is increased — the motor load should be kept about 10% below its normal full rating. With the centrifugal pump power characteristic, reduced cooling from the fan at low speeds is balanced by lower power absorbed. Controller efficiency is constant at about 90%, and under a constant ratio of voltage to frequency, motor efficiency does not fall as rapidly as with voltage control alone. Another significant benefit is that the controller shows a unity power factor to the main supply under all loads.

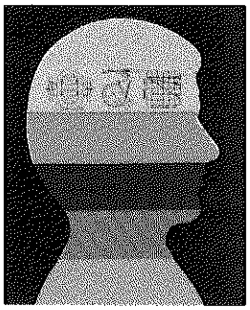
Speeds above and below main synchronous speeds are possible; the drive can be retrofitted on existing installations, and all the benefits of a standard squirrelcage induction motor hold good. Technically, this is the most advanced speed control method. The inverter costs more than other electrical methods, but can be justified in applications where one inverter can be used to control a number of small motors.

Schrage-type ac commutator motor.
In this type of ac motor drive, the line voltage is fed through slip rings to the rotor winding. A voltage is induced in a commutator winding on the rotor. Pairs of moveable brushes on the commutator pick up a variable voltage and feed it to the stator winding, where it adds to the voltage induced in the stator from the rotor field. The resulting voltage and stator field determine the speed.

Maximum power is limited to about 500 h.p. Machine efficiency and power factor are good at all speeds. Speed range is limited to 4:1 as standard, but 20:1 can be achieved with modification. Full-range speed change takes up to half a minute on larger units. The use of brush gear increases maintenance cost.

Stator-fed ac commutator motors.
The motor is lower in cost in the higher power range and requires less maintenance than the Schrage motor.

It is also stator-fed, eliminating slip rings and overcoming the supply voltage limitation of the rotor-fed Schrage motor, and so it has largely replaced the Schrage motor. The stator-fed motors are available up to ratings of about 3,000 h.p. Transfer of voltage to or from the rotor commutator winding is controlled by an induction regulator; this regulator is adjusted by manual methods or by servomotor from a remote signal to control speed. The induction regulator is a separate unit with its own forced cooling on large units. This type of drive is quite widely used for variable speed pumping applications, especially in Europe.



pump refresher

Centrifugal pump operating principles, performance and system curves

Part II

In Part One we reviewed basic centrifugal pump operating principles, performance and system curves. This installment continues the study of performance characteristics and looks further into the effects of changes in speed and impeller diameter.

Speed changes.
In the last issue we saw that if the speed of a centrifugal pump is doubled, head developed by the pump is quadrupled, because head developed is proportional to the square of the velocity.

Doubling the operating speed of a centrifugal pump also doubles the capacity which the pump can handle. This is because the velocity of the fluid through the impeller has doubled. For example, a pump capable of developing 50 ft. total head at 100 gpm when operating at 1750 rpm with a given impeller diameter will be capable of developing 200 ft. total head at a capacity of 200 gpm when operating at 3500 rpm. Let's go back to the formula relating head, capacity, efficiency and brake horsepower:

$$\text{Head, H} = KV^2/g$$

$$\text{Capacity, Q} = VA$$

$$\text{Bhp} = \frac{kQH}{\text{efficiency}}$$

Assuming that the efficiency of the pump remains the same when we change the speed, we see that the horsepower increases by a factor of eight when the speed of the pump is doubled. This is because the capacity doubles when we double the speed, the head quadruples, and these two factors are multiplied together to arrive at the brake horsepower.

For this reason, a centrifugal pump cannot be arbitrarily speeded up. A pump designed to operate at one speed must be capable of transmitting a great deal of additional horsepower if it is to run at a higher speed. However, we can make use of the speed conversion in slowing a pump down, from 3500 to 1750, or from 1750 to 1150 rpm. To get a rough idea of the pump's capability, we assume the efficiency remains constant and apply the relationship given above.

For a given pump and impeller diameter:

$$\frac{\text{rpm}_1}{\text{rpm}_2} = \frac{\text{gpm}_1}{\text{gpm}_2} = \sqrt{\frac{\text{head}_1}{\text{head}_2}} = \sqrt[3]{\frac{\text{bhp}_1}{\text{bhp}_2}}$$

Or, to put it a different way:

$$\frac{\text{rpm}_1}{\text{rpm}_2} = \frac{\text{gpm}_1}{\text{gpm}_2}$$

$$\left[\frac{\text{rpm}_1}{\text{rpm}_2}\right]^2 = \frac{\text{head}_1}{\text{head}_2}$$

$$\left[\frac{\text{rpm}_1}{\text{rpm}_2}\right]^3 = \frac{\text{bhp}_1}{\text{bhp}_2}$$

A change in speed always results in a change in capacity, head, and horsepower. These formulas give a useful approximation of the final performance capability of the pump — but remember, it is only an approximation, since efficiency does change slightly.

Impeller diameter changes.
A change in impeller diameter of a pump being operated at constant speed has essentially the same effect as a change in pump speed: both result in a change in the speed of the liquid which leaves the impeller. The formulas which apply for the change in capacity, head and horsepower related to impeller diameter changes look exactly the same as they do for changes in speed. The relationships are expressed as follows:

For a given pump and speed:

$$\frac{\text{Imp. dia. 1}}{\text{Imp. dia. 2}} = \frac{\text{gpm}_1}{\text{gpm}_2} = \sqrt{\frac{\text{head}_1}{\text{head}_2}} = \sqrt[3]{\frac{\text{bhp}_1}{\text{bhp}_2}}$$

Or, to put it a different way:

$$\frac{\text{Imp. dia. 1}}{\text{Imp. dia. 2}} = \frac{\text{gpm}_1}{\text{gpm}_2}$$

$$\left[\frac{\text{Dia. 1}}{\text{Dia. 2}}\right]^2 = \frac{\text{head}_1}{\text{head}_2}$$

$$\left[\frac{\text{Dia. 1}}{\text{Dia. 2}}\right]^3 = \frac{\text{bhp}_1}{\text{bhp}_2}$$

Even greater care must be taken in the use of these formulas than the formulas for speed change. A

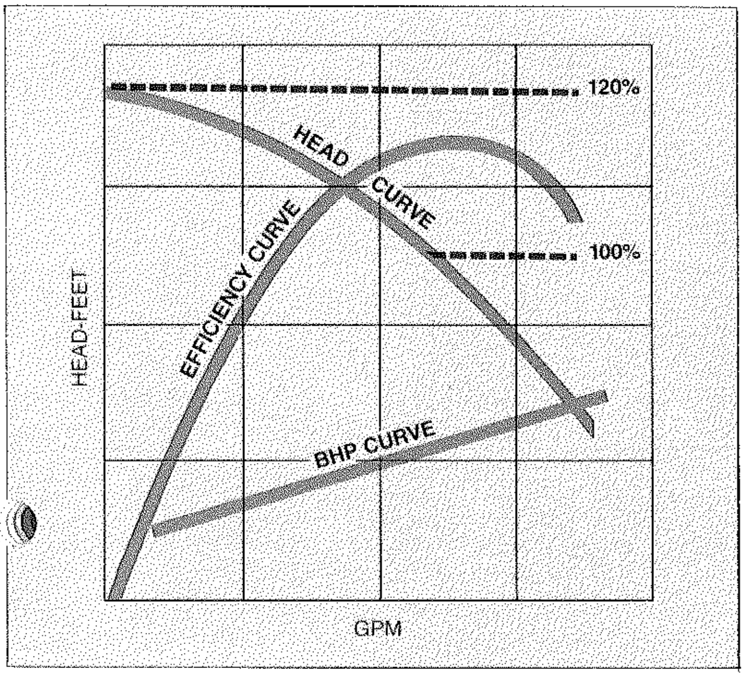


Figure 1 — A "normal" head-capacity curve has a shut off head about 15 to 20 per cent above the bep (best efficiency point).

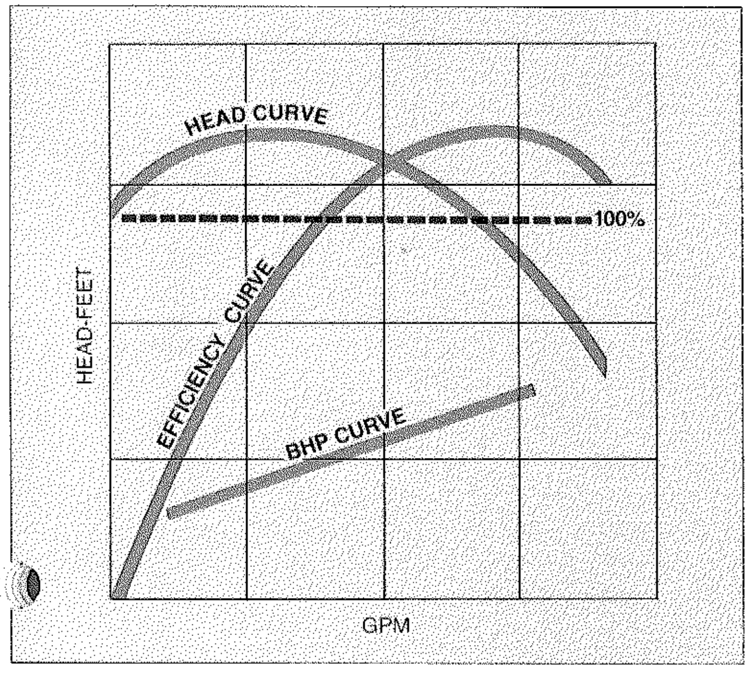


Figure 2 — A "drooping" head-capacity curve has a shut off head close to the bep. For a given head-capacity condition, a pump with a "drooping" head capacity curve will tend to be smaller than a pump with a "normal" head-capacity curve.

change in impeller diameter in a pump affects the basic relationship between impeller and casing, and this alters the design configuration of the pump. For this reason, the impeller change formulas should not be applied when the impeller diameter changes more than about 10%. When an impeller diameter change of more than 10% is indicated, the best practice is to obtain test curves for the new impeller diameter from the manufacturer.

The shape of pump curves.
Published pump head-capacity curves usually apply to a particular impeller design. Manufacturers can sometimes offer pumps fitted with impellers of different characteristics to meet various applications. Three types of head-capacity curves are characteristic: a "normal" rising curve, a "drooping" curve, and a "steeply rising" curve.

In a *normal rising curve*, or rising head-capacity characteristic, the head rises continuously as the capacity is decreased (Figure 1). The rise from best efficiency point to shutoff may be about 10 to 20%. Pumps with curves of this shape are used in parallel operation because of their stable characteristics.

In a *drooping curve*, the head developed at shutoff is about equal to the head at the bep (Figure 2). This characteristic is typical of impellers designed to deliver maximum head per inch of diameter. Efficiency is usually good and the pump may be smaller than normal. When pumps with drooping characteristics are run on throttling systems, they generally operate fine. However, they may cause operating difficulties when the system friction curve is very flat, such as on boiler feed service. These pumps can only be operated in parallel when the operating point is below the shutoff head (head developed with discharge closed).

In a *steeply rising curve,* there is a large increase in head between that developed at design capacity and that developed at shutoff (Figure 3). This pump will operate in parallel over its entire range. It is best suited for operation where minimum capacity change is desired with pressure changes; for example, in batch pumping or filter systems. The bhp curve also tends to be flat, but the head developed per inch of diameter is

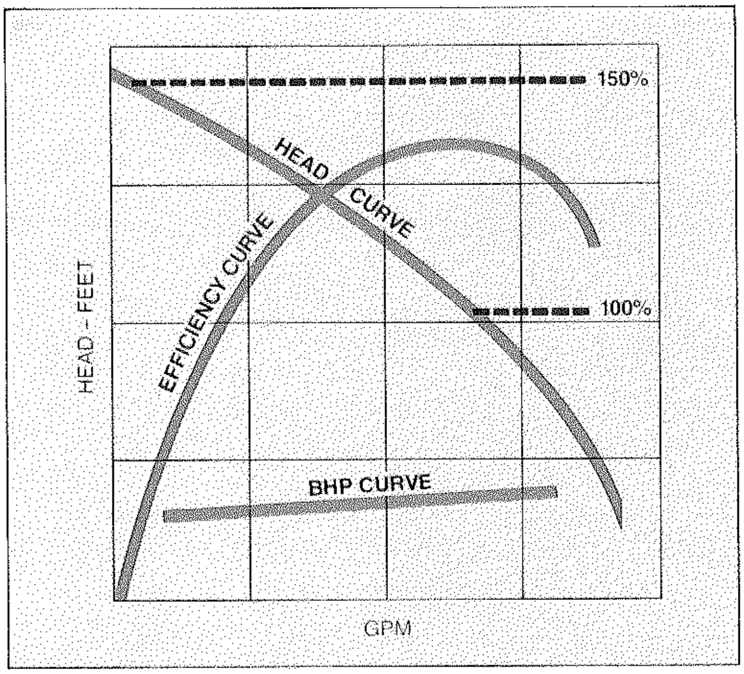


Figure 3—A "steeply rising" head-capacity curve promises good parallel operation, and a small capacity change and a relatively flat bhp curve. However, for a given head-capacity condition, a pump with a "steeply rising" head-capacity curve will be larger than pumps with a "normal" head-capacity curve.

low, so efficiency may be lower, and the pump tends to be larger than normal.

Effect of viscosity on pump performance.

A change in viscosity can also affect the performance of a centrifugal pump. The published rating curves of a centrifugal pump are based on its performance when handling water, which has negligible viscosity. If the pump is to handle liquids with higher viscosities, its performance changes. A typical curve for viscous performance is shown in Figure 4. The head-capacity curve drops off as viscosity is increased, and brake horsepower increases sharply. The pump manufacturer can supply correction curves to indicate a centrifugal pump's performance when handling viscous fluids.

Effect of entrained air.

The buildup of entrained air also affects the performance of a centrifugal pump. As a result of centrifugal force, the heavier liquid is thrown outward from the impeller eye. The lighter air remains behind and will gradually build into a bubble as big as the impeller inlet ("eye") area which chokes off the suction flow. This is called "air binding." Figure 5 shows the effect of 2 to 6% air entrainment on a typical head-capacity capability curve.

Increased capacity.

It is not always possible to pick a centrifugal pump with a curve which intersects the system head curve at the exact capacity desired. Instead, it may be necessary to use a pump capable of developing higher head through the use of a larger diameter impeller (or increased rotative speed). Let's see what happens when we use a pump capable of developing more head than we need at a given capacity. Starting with a given pump we can plot the performance of the same pump but with a larger diameter impeller, capable of developing more head (Figure 6). This unit can pump a higher capacity because the intersection of its performance curve and the system head curve is at a greater capacity than the original point of intersection. However, note that we have to pay a penalty in additional brake horsepower required.

Analyze system for best selection.

Always plot the system friction curve when selecting a pump. Plot the pump's performance on the same coordinate system in order to determine how the pump will operate in the system. In general, it is desirable to choose a pump to operate within the capacity range of highest efficiency. Smaller, general-service pumps are often designed to operate over a very wide capacity range. If npsh is sufficient to prevent cavitation, the pump will probably turn in a satisfactory performance.

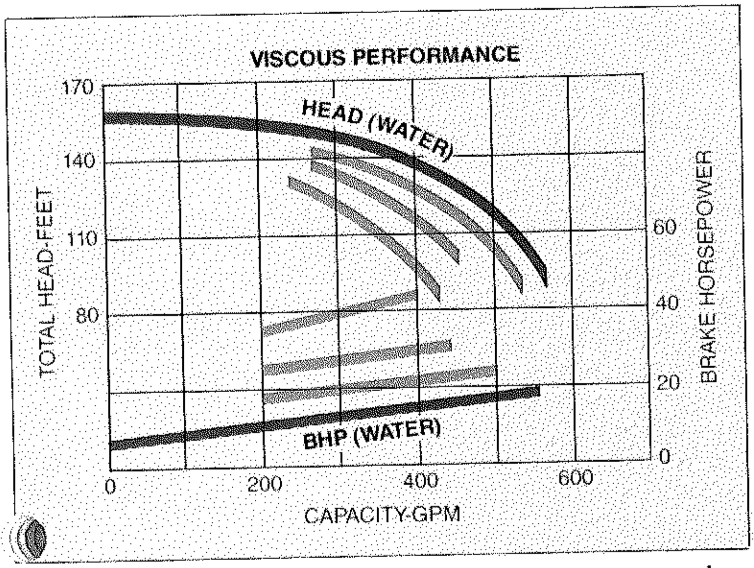


Figure 4—The head-capacity curve drops as viscosity increases, and bhp increases sharply.

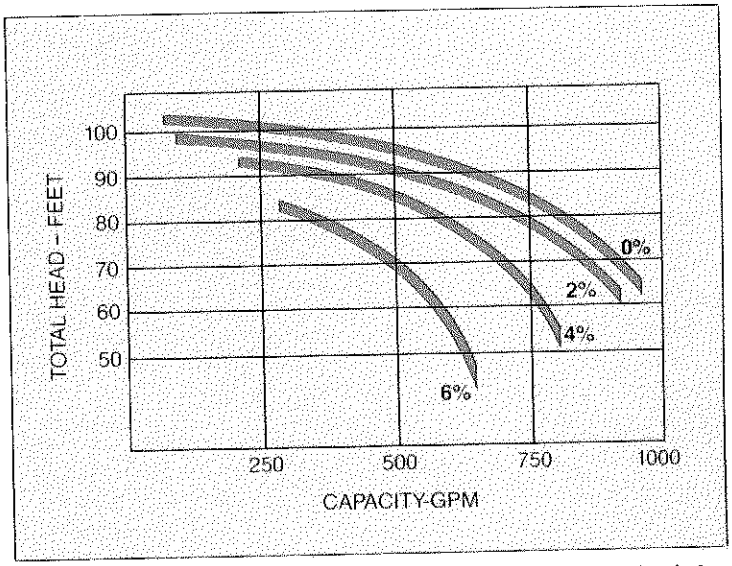


Figure 5—The head-capacity relationship is affected by entrained air, with 6 per cent about maximum before air binding occurs. However, even with less than 6 per cent air, a pump can become airbound if operated near shut off.

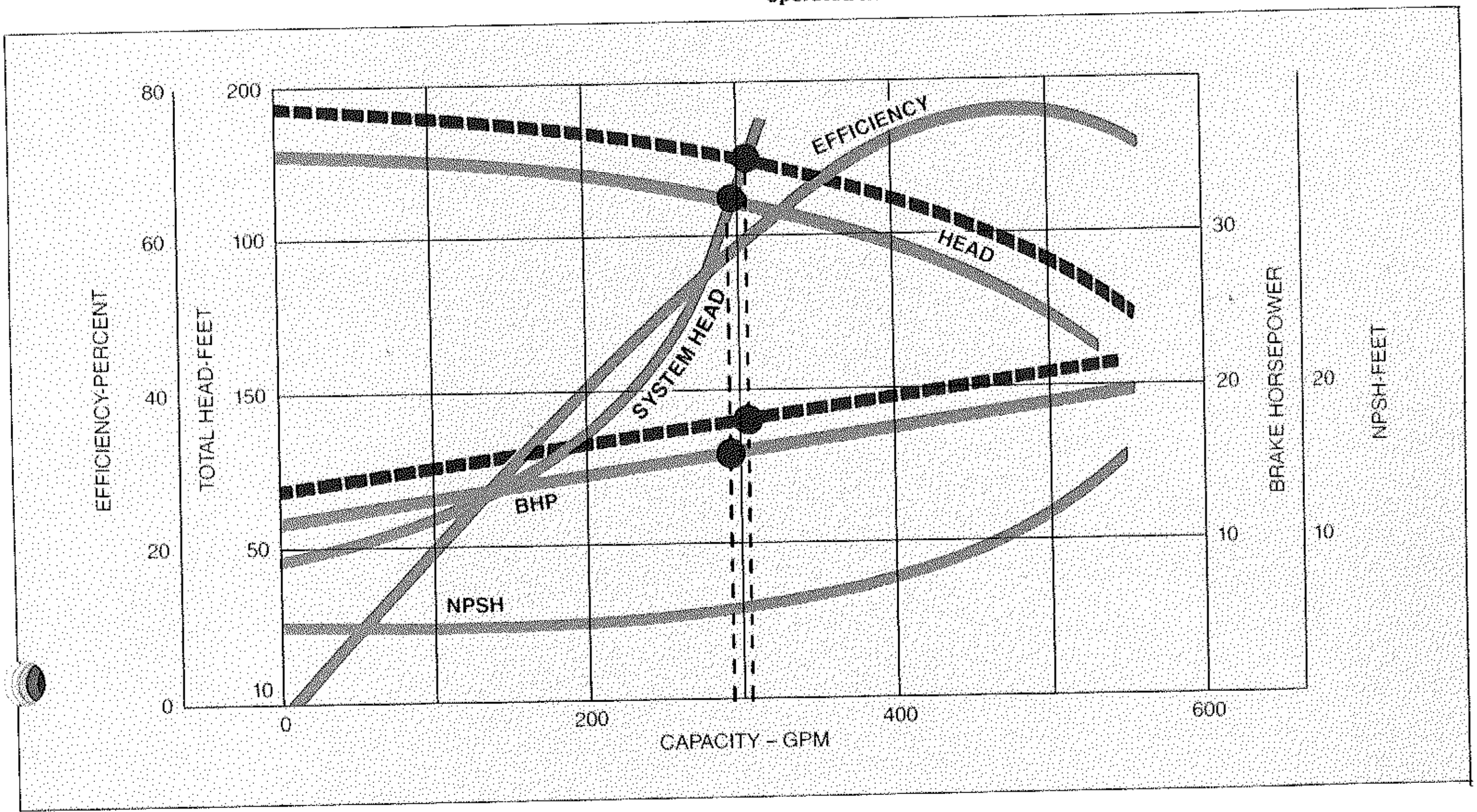


Figure 6—The higher head pump can handle a greater capacity, but at the price of increased horsepower.

Basic terms: Head and NPSH

HEAD is energy per pound of fluid. The term is commonly used to represent the vertical height of a static column of fluid corresponding to the pressure of a fluid at the point in question. Head can also be considered as the amount of work necessary to move a liquid from its original position to the required delivery position — in this case, the term includes the extra work necessary to overcome the resistance to flow in the line.

In general a liquid may have three kinds of energy:
1. POTENTIAL HEAD is energy of POSITION (measured by the vertical height above some plane of reference).
2. STATIC PRESSURE HEAD is energy per pound due to pressure (measured by Bourdon pressure gages or equivalent means).
3. VELOCITY HEAD is the kinetic energy per pound (measured by a pitot tube or calculated from the flow and pipe area).

BERNOULLI'S THEOREM states that energy cannot be created or destroyed. The sum of the three types of energy (heads) at any point in a system is the same at any other point in the system (if there are no friction losses and no extra work is performed).

STATIC SUCTION LIFT is the vertical distance from the free surface of liquid to pump datum line when the source of supply is below the pump (Figure 7).

NET SUCTION LIFT is the sum of static suction lift plus friction losses (a negative figure).

STATIC SUCTION HEAD is the vertical distance from free surface of liquid to pump datum, when the supply is above the pump (Figure 7).

NET SUCTION HEAD is the static suction head plus the pressure on the surface of the liquid minus friction losses (the figure may be either positive or negative).

STATIC DISCHARGE HEAD is the vertical distance from pump datum to the free surface of the liquid in the discharge tank or point of free discharge (Figure 7).

NET DISCHARGE HEAD is the sum of static discharge head plus the pressure on the surface of the liquid in the discharge tank and discharge friction losses.

TOTAL HEAD (TH) is the net difference between total suction and discharge heads. It is the measure of the energy increase imparted to the liquid by the pump.
TH = discharge head plus suction lift.
TH = discharge head minus suction head.

NET POSITIVE SUCTION HEAD (NPSH) is the amount of energy in the liquid at the pump datum. To have meaning, it must be defined as either available or required NPSH.

REQUIRED NPSH (NPSHR) is the energy needed for a pump to operate satisfactorily; that is, to fill the pump on the suction side and overcome friction and flow losses from the suction connection to the point in the pump where more energy is added. In a centrifugal pump, NPSHR is the amount of energy (in feet of liquid) required to overcome friction losses from the suction opening to the impeller vanes, and to create the desired velocity of flow into the vanes. NPSHR is a characteristic of the pump. It varies with pump design, pump size, and operating conditions. The figure is determined by test or computation, and is supplied by the pump manufacturer.

AVAILABLE NPSH (NPSHA) is the inherent energy in a liquid at the suction connection of the pump (regardless of the type of pump), over and above energy in the liquid due to its vapor pressure. NPSHA is a characteristic of the system and one you must know to properly apply a pump. It can be calculated or can be obtained for an existing system by taking test readings at the suction side of the pump (Figure 8).

DETERMINING NPSH Since a liquid may have potential, pressure and kinetic energy, and since NPSH is an energy term, the methods of determining NPSHA must take three types of energy into account.

To determine NPSH by calculation, consider the energy at point 1 (Figure 8).

$$\pm Z_1 + P_1 + \frac{V_1^2}{2g} =$$

the sum of potential, pressure, and kinetic energies at the surface of the liquid. Since the area of the surface of the liquid supply is large compared with the area of the suction pipe, the velocity can be considered as negligible and the kinetic energy or velocity head as zero. Total energy at point 1 is then:

$$\pm Z_1 + P_1$$

Although P_1 represents the pressure energy at point 1, in this case atmospheric pressure, in pumping we must be certain the liquid does not vaporize in the suction line. We therefore subtract the vapor pressure (P_V) of the liquid:

$$\pm Z_1 + \frac{(P_1 - P_V)}{\text{Sp. gr.}} \times 2.31$$

The pressure terms, expressed in psia, have been converted to feet of head, the unit in which NPSHA is commonly expressed. Since the energy at point 2, the point at which NPSHA is required, is equal to the energy at point 1 with the exception of losses due to friction, we subtract these losses (h_f) and NPSHA at point 2 becomes:

$$\pm Z_1 + \frac{(P_1 - P_V)}{\text{Sp. gr.}} \times 2.31 - h_f$$

Example 1:
This system uses an open suction tank 10 ft. above pump datum; pipe friction = 8 ft. The pump will handle cold water at 60° F; vapor pressure = 0.256 psia; specific gravity = 1.

$$\text{NPSHA} = 10 + \frac{14.7 - .256}{1} \times 2.31 - 8 = 35.4 \text{ ft.}$$

Example 2:
This system uses an open suction tank 10 ft. above pump datum; pipe friction = 8 ft. The pump will handle hot water at 200° F; vapor pressure = 11.53 psia; specific gravity = 0.965.

$$\text{NPSHA} = 10 + \frac{14.7 - 11.53}{0.965} \times 2.31 - 8 = 9.58 \text{ ft.}$$

Example 3:
This system uses a closed suction tank with 20 in. Hg vacuum (10 in. Hg absolute pressure). Tank is 10 ft. above pump datum; pipe friction = 8 ft. The pump will handle oil at 60° F; vapor pressure = 1 psia; specific gravity = 0.9.

$$\text{NPSHA} = 10 + \frac{10 \times .491 - 1}{0.9} \times 2.31 - 8 = 12.1 \text{ ft.}$$

(Note: 0.491 = 14.7/29.92, required to convert in. Hg. to psi.)

It is sometimes possible to determine NPSHA from test readings. Since point 2 is at datum, the liquid has no potential energy and $Z_2 = 0$. P_2 is the gage reading. By adding atmospheric pressure to the gage reading to obtain absolute pressure head, subtracting vapor pressure, and correcting for the elevation of the suction gage (Y) we obtain NPSHA.

$$\text{NPSHA} = \frac{(P_{gage} + P_{atm.} - P_v)}{\text{Sp. gr.}} \times 2.31 + Y + \frac{V_2^2}{2g}$$

where V_2 is the velocity of the fluid in ft/sec or flow/area.

NPSHA must always be equal to or greater than NPSHR for the pump to fill and deliver the required quantity of liquid.

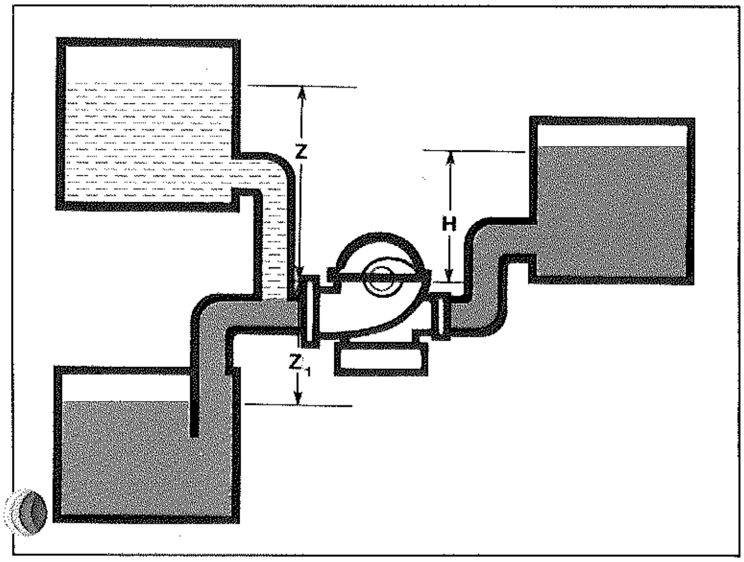


Figure 7 — How head is defined. Z = static suction head. Z_1 = static suction lift. Z minus friction equals net suction head. Z_1 plus friction equals net suction lift. H = static discharge head. H plus friction equals net discharge head. Total head equals net discharge head minus net suction head or plus net suction lift.

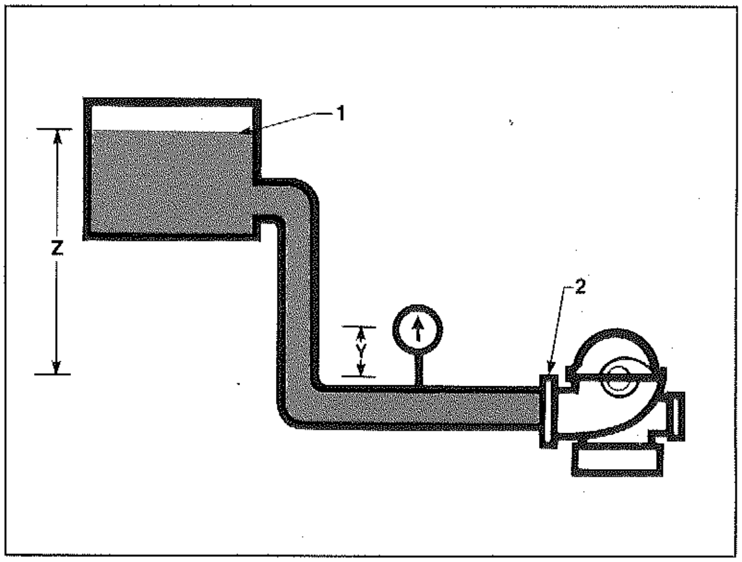
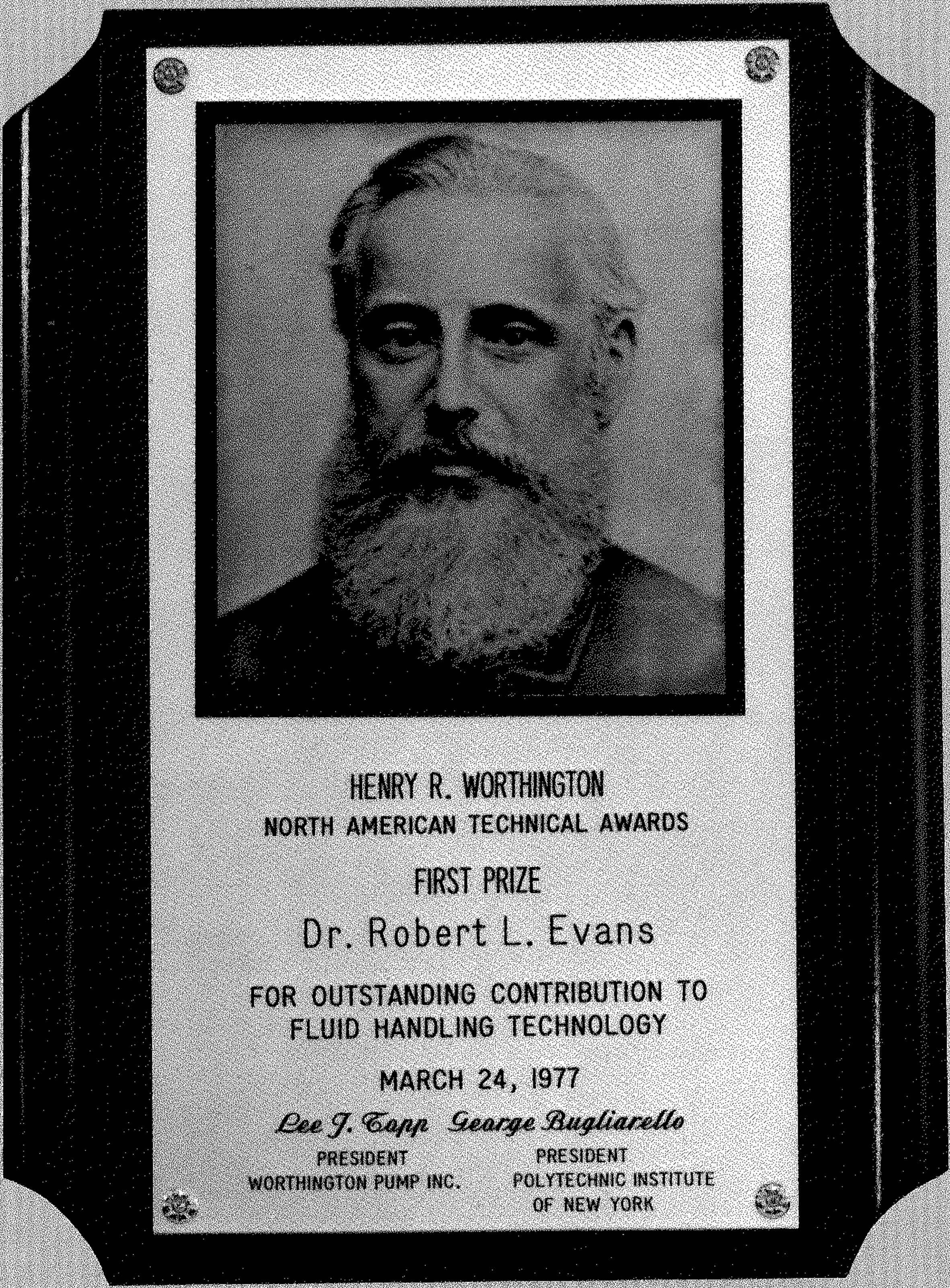


Figure 8—NPSHA may be determined by calculation of system head or from gage reading.


HENRY R. WORTHINGTON
NORTH AMERICAN TECHNICAL AWARDS
FIRST PRIZE
Dr. Robert L. Evans
FOR OUTSTANDING CONTRIBUTION TO FLUID HANDLING TECHNOLOGY
MARCH 24, 1977
Lee J. Topp
George Bugliarello
PRESIDENT
WORTHINGTON PUMP INC.
PRESIDENT
POLYTECHNIC INSTITUTE OF NEW YORK

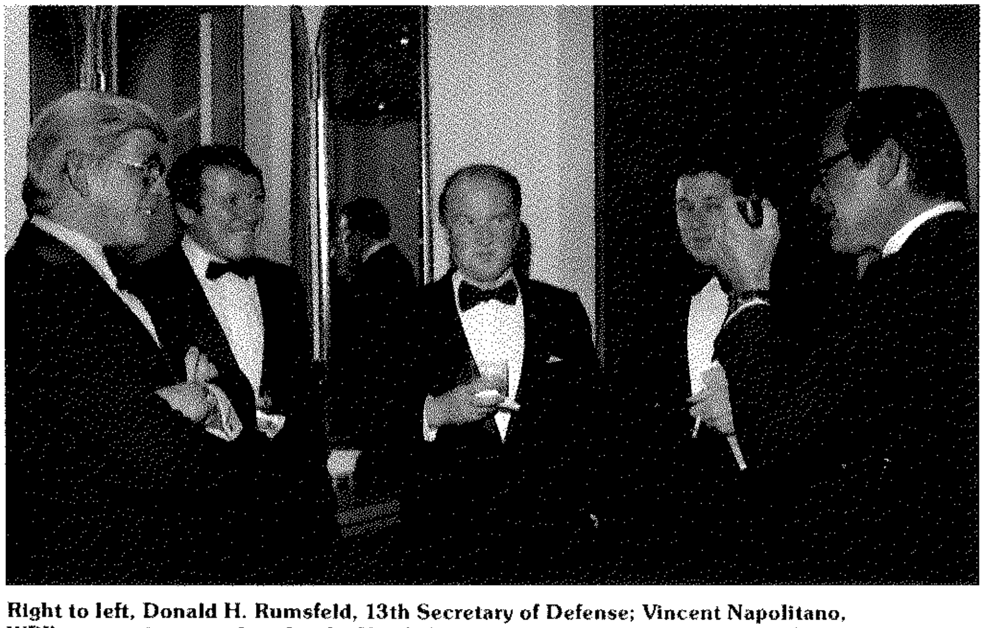
Winners honored in first North American Henry R. Worthington Technical Awards contest.

Winners in the first North American Henry R. Worthington Technical Awards Contest have been honored by representatives of government, the academic world and industry with a black tie dinner at New York's Windows on the World restaurant atop the World Trade Center, where they received nearly $10,000 in cash awards.

Sponsored by the Polytechnic Institute of New York, of which Henry R. Worthington was an original founder, and supported by WPI, the biennial contest seeks to promote and stimulate applied research and innovative thinking in the field of pumps and pump systems.

The North American Awards Program is modeled after the European competition originated by Worthington S.p.A. (Italy) a decade ago, and later expanded to include all of Europe. "The North American Awards Program may be considered the third phase of this research-stimulating contest, and I have the feeling it is far from the last stage," Lee Topp, president of Worthington Pump Inc., told the assembly. "After all, Worthington Pump is a worldwide company, and there are many areas not yet benefitting from the opportunities this sort of contest brings," he added. The most lasting benefit of the competition, Topp stressed, is the application of these principles and findings to improve fluid handling equipment. "There is in these research works a body of important fundamental knowledge. And we urge all companies in the pump industry and related industries to evaluate this information. There is much to be gained here by all those concerned with making equipment for fluid transfer. In a sense, the real competition stimulated by the North American Henry R. Worthington Awards is just beginning. The commercial competition to successfully apply the principles and findings advanced in these papers and in other research, we hope, is the surest route to genuine improvement in equipment made by our industry." All research underlying the winning papers (including incidental patents and rights for industrial use) remain the property of the contestants.

A formal reception in the Governor's Suite was followed by dinner in the Manhattan Prospect room 107 floors above Manhattan overlooking New York Harbor. The ceremonies were attended by Hon. Peter J. McDonough, New Jersey State Senator; John T. Carroll of New York's Municipal Service Administration; Vincent Hindley, special assistant to New Jersey's commissioner of labor and industry, and other government and industry representatives as well as Worthington executives. Speakers included Lee Topp and Dr. Bugliarello, president of the Polytechnic Institute of New York. Keynote speaker was Donald H. Rumsfeld, 13th Secretary of Defense, who spoke on the subject of "Industry and Defense." Paolo Gamboni, WPI Chairman, Lee Topp and Vincent Napolitano, group vice



Right to left, Donald H. Rumsfeld, 13th Secretary of Defense; Vincent Napolitano, WPI's group vice president for the North American region; Paolo Gamboni, chairman and chief executive officer, Worthington Pump Inc.; Lee J. Topp, president, Worthington Pump Inc.; and Peter J. McDonough, New Jersey State Senator gather for the first North American Henry R. Worthington Awards presentation.

president, North American Region, jointly presented the awards.

Papers to be published.
The winning papers, which will soon be published in book form, were selected from more than 90 entries. All papers were reviewed by an international panel of distinguished judges: Professor I. W. Smith of the University of Toronto; Ing. Ignacio A. Alvarez of Mexico's Ministry of Hydraulic Resources; Professor Allan J. Acosta of Cal. Tech; Professor Stephen H. Crandall of MIT; Melvin Hartmann of the NASA-Lewis Research Center, and WPI's chief consulting engineer, Igor J. Karassik. The second Henry R. Worthington North American Technical Awards will be offered in 1979.

First prize: $5,000

Dr. Robert L. Evans, British Columbia Energy Commission, Vancouver. "Boundary layer development on axial-flow turbo machinery blading." A clarification of flow characteristics of fluids in axial-flow machines, with potential application to gas, water and steam turbines and compressors.

Second prize: $2,000

John F. Hurley and *Edward O. Hartel*, AVCO Lycoming, Stratford, Connecticut. "An analytical and experimental investigation of shear force pumps." Design factors of vaneless, bladeless, pistonless, and silent smooth-disk pumps.

Third prize: $1,000

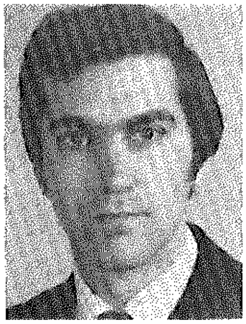
Dr. Budugur Lakshminarayana, Aerospace Engineering Department, Pennsylvania State University. "Fluid dynamic analysis of rocket pump inducers." A recommendation for improvements in inducer blade shape, including detailed analysis of the factors bearing on fluid behavior in high-speed pumping system components.

Honorable mentions: $500

Dr. George F. Round, Faculty of Engineering, McMaster University, Hamilton, Ontario. "Pumping by pulsation."

Dr. Tommy Y. W. Chen, Hydro-Turbine Division of Allis-Chalmers, York, Pennsylvania. "Effect of impeller modification on the performance of shrouded mixed-flow turbo machines."

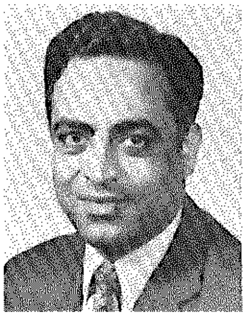
Dr. Jimmy S. Chow, with *A. Y. Hou and L. Landwebber*, Westinghouse Research Labs, Pittsburgh, Pennsylvania. "Calculation of flow through two-dimensional centrifugal impeller by method of hydrodynamic singularities."



Evans

Hurley

Hartel



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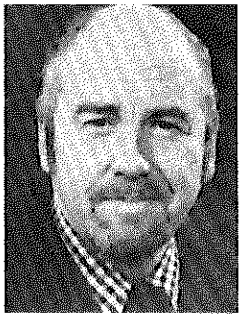
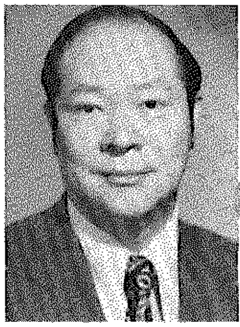


Round

Chen



Chow

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