Failures of Bevel-Helical Gear Units on Traveling Bridge Cranes

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Introduction

Bridge cranes are among the most useful machines in many branches of modern industry. Using standard hooks or other specialized clamping devices, they can lift, transport, discharge, and stack a variety of loads.

Gear technology progress has always been influential to advances in bridge crane design, allowing lighter and more productive cranes. Many bevel-helical gear units are employed in the traveling drives of big industrial bridge cranes, as they form compact packages with couplings, brakes and electric motors that other gear units do not allow for.

A sketch of a typical individual wheel traveling drive is shown in Figure 1, with the electric prime mover (a) and the gear unit (b), with a usual hollow low-speed shaft (c). The gear unit is mounted on a floating base (d), common with the prime mover. All the aggregate pivots are on the low-speed shaft, which is vertically fixed at the other end to the crane framework by means of elastic blocks (e). The flexible coupling between the prime mover main shaft and the gear unit is a high-speed shaft that is usually combined with a drum brake (f).

In a number of cranes, frequent failures of travel bevel gears pose a difficult problem for maintenance. Open discussions have raised questions about the necessary service or application factor to avoid such failures and the associated downtime. Recommendations found in prestigious sources give application factor values from as low as 1.1 to as high as 3.0. In many gear unit catalogs, the crane traveling drive selection refers to the manufacturer, giving no other guidance to crane designers or plant maintenance engineers.

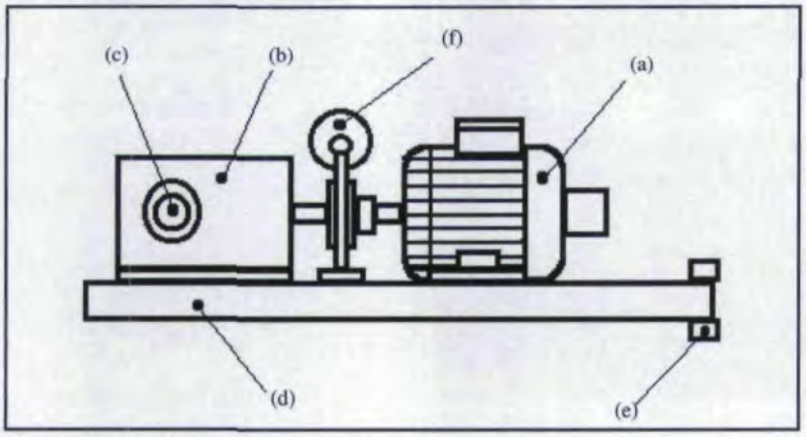

Fig. 1—Typical bridge crane traveling drive.

This paper focuses on the origin of the troubles experienced with the standard, general purpose bevel-helical gear units used in the traveling drives of medium and large size bridge cranes, according to the author's theoretical research and practical experience.

Nature of the Failures

Failures of crane traveling drives are usually of a catastrophic character, with the sudden fracture of one or several teeth in the bevel gear, ordinarily in the high-speed stage in the gear unit.

The above-mentioned failures are very difficult to anticipate, because time between failures (Ref. 2) behaves chaotically. Sometimes the gear works well for a relatively long period, in the order of several weeks, and sometimes the gear breaks down after a few minutes of work.

Such an irregular pattern of failure is usually associated with mechanical resonance. But even a detailed analysis of the bevel gear vibration behavior in crane bridge traveling gear units generally shows no resonance at all in the gear mesh. This fact may be highly misleading to an engineering researcher trying to find the origin of the above-mentioned troubles.

However, the bevel gear mesh is not the sole elastoinertial system related to the high-speed stage of the gear unit. Most important, according to our findings, is the elastoinertial system comprised of gear unit's high-speed shaft and half-coupling, including the brake drum (Fig. 2). For the sake of brevity, the elastoinertial system constituted by the gear unit's high-speed shaft and half-coupling with brake drum is referred to from now on as the shaft/coupling system.

In fact, almost all of the torsional elastic compliances of the shaft/coupling system belong to the shaft. This is because of the much bigger diameter and shorter axial length of the half-coupling and its attached brake drum. Therefore, it can be easily shown that

$$c_s = c_{sh} + c_{hc} \approx c_{sh} \qquad (1)$$

Where

c_{sh} is the shaft's elastic compliance.

c_{hc} is the half-coupling's elastic compliance.

All elastic compliances in (1) and after are given in rad/(N·m), according to the International System of units, SI.

On the other hand, almost all of the moments of inertia of the shaft/coupling system relative to its rotational axis belong to the half-coupling and its attached brake drum. This is due to the very small diameter of the shaft as compared with the brake drum. Therefore, it can be easily shown that

$$I_s = I_{sh} + I_{hc} \approx I_{hc} \qquad (2)$$

Where

I_{sh} is the shaft's moment of inertia.

I_{hc} is the half-coupling's moment of inertia.

All moments of inertia in (2) and after are given in kg·m^2, according to the International System of units, SI.

Being an elastoinertial system with almost lumped (concentrated) parameters, including only one elastic element and only one inertial element, the main proper frequency f_E of the shaft/coupling system can be assessed by the well-known expression

$$f_E = \frac{1}{2\pi}\sqrt{\frac{1}{c_s I_s}} \qquad (3)$$

Proper frequency in (3) and after is given in Hz, according to the International System of units, SI.

The resistive torque at the gear unit's high-speed pinion (Fig. 2) has a pulsation with a frequency equal to the mesh frequency of the high speed gear, given by the relation

$$f_z = n_m z_p \qquad (4)$$

Where

n_m is prime mover's rotational frequency.

z_p is high speed pinion's number of teeth.

Both mesh and rotational frequencies in (4) and after are given in Hz, according to the International System of units, SI. Meanwhile, the number of teeth is considered non-dimensional.

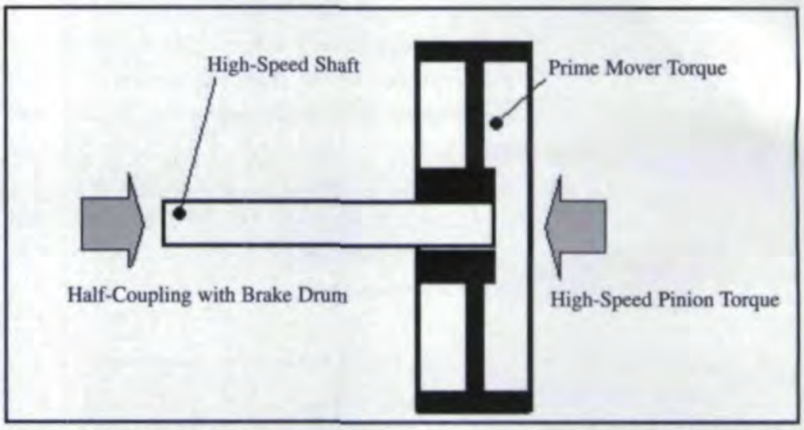

Fig. 2—Elastoinertial system comprised of a high-speed shaft and half-coupling.

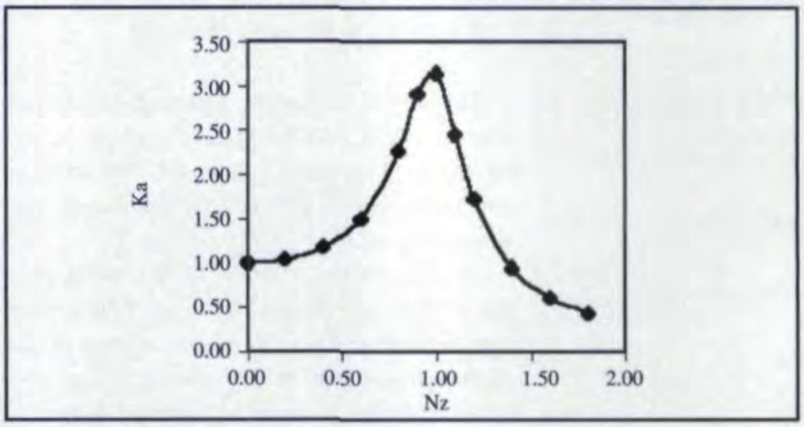

Fig. 3—Application factor for a bevel gear.

Under the excitation of the pulsating pinion torque, the shaft/coupling system develops torsional vibrations, which are superimposed on the otherwise smooth velocity profile of the gear unit high-speed shaft. The severity of such torsional vibrations is higher when the mesh frequency of the bevel gear approaches the proper frequency of the shaft/coupling system.

The degree of mutual approach of the above mentioned frequencies governing the vibratory process can be quantified by the tuning factor, given by the relation

$$N_z = \frac{f_z}{f_E} \qquad (5)$$

The tuning factor in (5) and after is a non-dimensional quantity, as long as both frequencies are given in the same units.

According to widely recognized practice (Refs. 3, 6), an elastoinertial system is in a state of resonance if

$$0.85 \le N_z \le 1.15 \qquad (6)$$

As the traveling drive operates under variable speed, the tuning factor of the shaft/coupling system sweeps a range of values. According to the commands given by the crane operator, the tuning factor stays stochastically in one or another value

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during a certain time.

If enough time is spent under condition (6), the amplitude of the pulsating torque can reach high values. Such high values can develop low cycle volumetric fatigue damage on the teeth of the bevel gear, leading to its quick fracture.

The overload imposed by the resonance effect can be expressed (Refs. 2, 3, 5) by means of an application factor

$$K_A = \frac{1}{\sqrt{(1 - N_z^2)^2 + \left[\frac{\beta N_z}{I_s f_E}\right]}} \qquad (7)$$

Where

β is the factor of viscous damping.

The factor of viscous damping in (7) and after is given in (N·m)/Hz, according to the International System of units, SI. The application factor, as it is well known, is a non-dimensional magnitude.

There is a moderate degree of viscous damping due to oil film in the gear mesh and rolling bearings, as well as from the internal friction of the elastomeric element in the coupling. Therefore, critical damping in the shaft/coupling system is assumed. Such condition is the limit between light and heavy viscous damping. Critical damping is present when relation (8) holds

$$\frac{\beta}{I_s f_E} = \frac{1}{\pi} \qquad (8)$$

The values of the application factor K_A under the assumption (8) are plotted as a function of the tuning factor in Figure 3. It is interesting to note that when the tuning factor equals unity, the application factor reaches its maximum value

$$K_{A\,max} = 3.14 \qquad (9)$$

That is, the torque transmitted by the bevel pair of the gear unit can reach a value more than triple the nominal, enough to fracture its teeth if no ample strength has been left.

In conformity with this result, the service factor of 3.0 according to AGMA 6010-F97 (Ref. 1) appears adequate even when resonance is present. Therefore, to dimension a bridge traveling gear unit for a bulletproof quick design, or for an emergency overhaul, a service factor with the value of 3.0 could be used, but obviously at a cost.

On the Influence of Motor Controllers

Many of the failures in the high-speed bevel gears of crane bridge traveling drives appeared after the advent of the solid-state variable speed controllers for the electric prime movers. These controllers yield an almost constant torque at the prime mover main shaft during the starting period, suppressing the strong saw-tooth torque ripple characteristic of the prime movers under the older magnetic controllers.

Apparently, a number of crane designers shortly after being acquainted with the new high-technology motor controllers, began to discard the old hydrodynamic clutches (the so-called hydraulic couplings) from the traveling drives of new design as an unnecessary piece of hardware. Consequently, the connection between prime mover and gear unit was effected by means of a simple flexible coupling, normally combined with a drum brake.

However, no flexible coupling has the strong viscous damping characteristic of hydrodynamic clutches, which do not allow for high values of resonance loads. Let the increase in viscous damping of the elastoinertial system due to the introduction of a hydrodynamic clutch be estimated conservatively as twofold. Then, as Figure 4 shows, the maximum value of the corresponding application factor will be under 1.7.

This result can justify the service factors from 1.5 to 2.0 given by certain manufacturers for bridge travel gear units, presupposing the use of hydraulic clutches. However, many times very similar values are recommended without any other necessary condition.

Therefore, to avoid unexpected problems, it is suggested that the replacement of hydraulic clutches with ordinary flexible couplings in drives with modern motor controllers should be

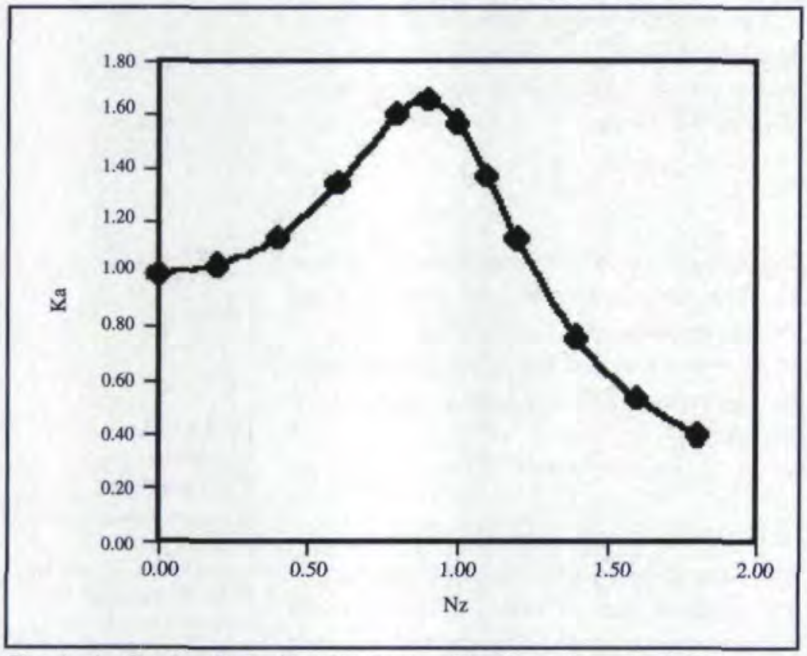

Fig. 4—Application factor for a bevel gear with a hydrodynamic clutch.

undertaken only after a dynamic analysis of the shaft/coupling system.

Suggestions to Manufacturers

AGMA standards point to the application engineer as responsible for an overall system design that avoids operation at resonance. Nevertheless, gear unit manufacturers can also take some basic measures to avoid near resonance operation of speed reducers equipped with standard drum brakes that are typical of crane traveling bridge drives.

There are two complementary characteristics in a non resonant-prone gear unit for crane travel drives: Minimum elastic compliance of the pinion shaft and minimum mesh frequency of the bevel gear.

Both characteristics tend to decrease the value of the application factor as given by (7). A shorter and oversize diameter high-speed shaft, allowed by an improved bearing design, seems to be a practical way to attain the first characteristic. A high-speed pinion with a smaller number of teeth, allowed by a special design of the bevel gear, seems to achieve the second characteristic.

Solutions for Existing Systems

On the user side, the installation of new gear units with a 3.0 service factor may be too costly and beyond the possibilities of existing travel drive systems without a major overhaul. However, there are three complementary modifications that can be done to an existing crane travel drive to improve its resonance behavior:

1. Minimize the compliance of the pinion shaft.
2. Minimize the mesh frequency of the bevel gear.
3. Minimize the moment of inertia of the high-speed shaft half-coupling.

All three modifications tend to decrease the value of the application factor as given by (7). The first and second modifications can be achieved by the same ways suggested in the former section, at only a fraction of the cost of a new gear unit. To minimize the moment of inertia of the shaft/coupling system, a simple solution is to invert the flexible coupling, as shown in Figure 5. This way, the high-speed shaft of the gear unit receives the smaller half-coupling (a), with a minimum moment of inertia, as it lacks the brake drum (b).

A Practical Example

A major industrial enterprise in South America has two special-purpose traveling bridge cranes, each with a total mass of 165 tons, working around-the-clock in a pit-furnace building. The bridge traveling drives for the individual motoring wheels of each crane were composed as follows:

1. A slip ring AC induction electric motor with a nominal power of 25 kW at a rotational frequency of 19 Hz (1,140 min-1), under an operating regime S3 25%.
2. A solid-state electronic controller for the electric motor, which regulates speed and torque through stator tension and rotor resistance.
3. A bevel-helical three-stage gear unit, with a nominal ratio of 1:71, and a main stage center distance of 200 mm.
4. A flexible coupling between motor and gear unit combined with a drum brake. The gear unit half-coupling carried the brake drum.

The crane designers selected these gear units using a service factor of 1.5, as proposed in the

Table 1—Dynamic data of the original drives.

Parameter	Value
c_s	$2.11 \cdot 10^{-5}$ rad/(N·m)
I_s	$3.77 \cdot 10^{-2}$ kg·m^2
f_E	178 Hz
n_m	from 0 to 20 Hz
z_p	11
f_z	from 0 to 220 Hz
N_z	from 0 to 1.24
K_A	from 1 to 3.14

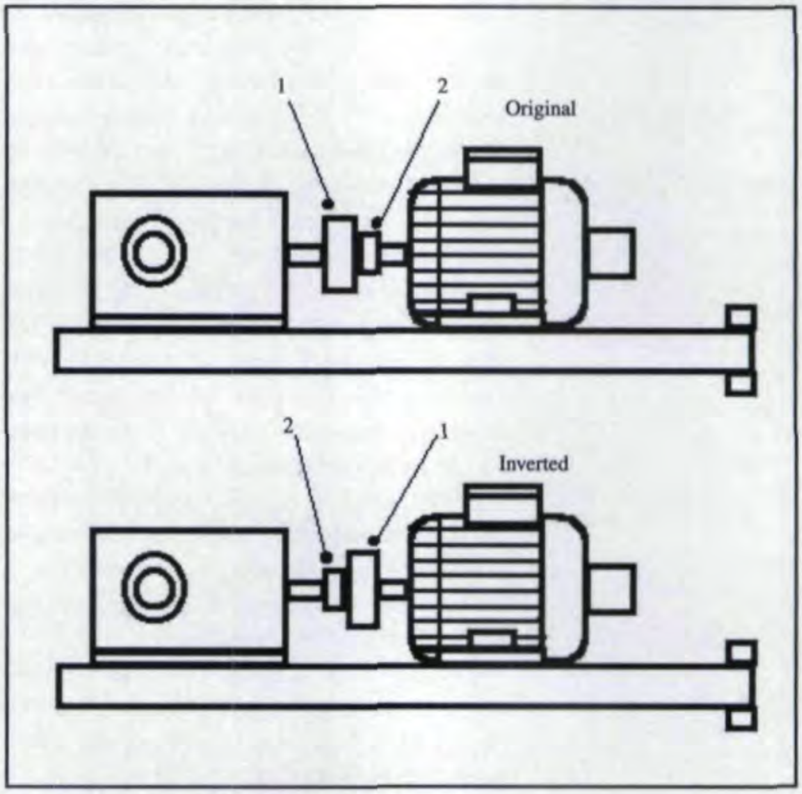

Fig. 5—Inversion of flexible coupling.

Table 2—Dynamic data of the modified drives.	
Parameter	**Value**
c_s	$1.44 \cdot 10^{-5}$ rad/(N·m)
I_s	$7.54 \cdot 10^{-3}$ kg·m^2
f_E	483 Hz
n_m	from 0 to 20 Hz
z_p	9
f_z	from 0 to 180 Hz
N_z	from 0 to 0.373
K_A	from 1 to 1.15

technical catalog of the manufacturer. However, just a few weeks after plant start-up, there were serious troubles with broken teeth in the high-speed stages of the gear units. Time between failures ranged stochastically from 15 days to 15 minutes. It made no difference if spare parts came from the original equipment manufacturer or from other sources.

After a series of attempts by different experts, the problem remained unsolved. The crane manufacturer proposed new gear units sized according to a 3.0 service factor. The author was subsequently called-in by the company as an independent advisor.

Under author's counsel, in-depth research into the dynamics of the original drives was performed. The most important results are given in Table 1. The methods and techniques used in this research are described above.

As can be seen in Table 1, the shaft/coupling system of the original gear units operated well deep in the conventional resonance zone, which covered no more than 24% of all the operating range of travel speeds. Consequently, the bevel gear received strong loads, up to 3.14 times the nominal, according to an unpredictable program that led to tooth fractures after a short period of time. The solution had a five-point strategy as follows:

1. A new bevel gear pair was designed and manufactured, with fewer teeth in the pinion, to lower the mesh frequency.
2. An optimum pinion design allowed a bigger diameter shaft to increase the system's proper frequency.
3. The brake drum was transferred to the motor shaft to increase the system's proper frequency.
4. The new bevel gear pair design was optimized for durability and strength (Refs. 4, 5, 7) using the same materials and within the same space.
5. A stronger taper roller bearing support for the high-speed pinion was devised to fit the same gear unit case without any change.

The dynamic data of the modified shaft/coupling system are found in Table 2, which shows that due to the reduction of the mesh frequency and the sharp elevation of the proper frequency, the tuning factor now covers a much smaller range. Therefore, the maximum value of the application factor has fallen to only 1.15.

Thanks to the reengineered high-speed state, the former high rate of bevel gear failures went to zero. This was accomplished at 20% of the cost of new, bigger gear units sized to operate with a 3.0 service factor. Now, after four years of field experience, the solution has proved its effectiveness. ○

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Gleason Technical Support Center, Novi, MI

Gleason Opens New Tech Support Center

Gleason Corporation has announced the opening of its new technical support center at 46850 Magellan Drive, Novi, Michigan. This facility will bring an array of gear manufacturing support services to customers in the Michigan and Western Ohio regions.

The new facility places many gear manufacturing solutions and services in close proximity to the important Detroit marketplace, and helps ensure that Gleason's customers will receive fast, hands-on responses to their needs. Gleason customers will have increased local access to comprehensive training resources; process and application engineering support; tool management including, in some areas, pick-up and delivery; spare parts inventories and on-site service personnel to help reduce repair and maintenance downtime.

West Industries Becomes United Gear and Assembly Inc.

On September 1, 2000, West Industries of Hudson, Wisconsin, a United Stars Company, became United Gear & Assembly Inc. (UGA), a world-class manufacturer of gears, shafts, assemblies and heat-treating operations. The company reputation was built on their responsiveness to customers, satisfying customer requirements and working successfully within narrow shipping windows. UGA meets specific market conditions by utilizing techniques that respond to customer orders with zero rejects and zero turn-around.

UGA has added new facility improvements and services available from a single source. With the ability and experience for additional value-added processes, UGA is committed to continuous improvement with their manufacturing system and with customers' parts and components. UGA offers a totally integrated supply source, including engineering, machining, heat-treating, plating, assembly and a quality assurance structure demanded by the marketplace.

Emerson Consolidates Gearing Brands Under EPT Operations

Emerson Power Transmission (EPT), of Ithaca, New York, has completed the consolidation of Emerson's gearing brands under the EPT organization. Effective September 1, 2000, EPT adds the US Gearmotor brand of gearing products to its already extensive gearing portfolio. Other brands include: Browning brand helical, bevel and planetary speed reducers and gearmotors, and Morse brand worm gear reducers and miter boxes.

According to Bill Boggess, vice president of strategic planning for EPT, "This will enable the US Motors organization to focus all its efforts on electric motors, and EPT to focus on all gearing and other power transmission products. Additionally, it enables Emerson to provide an enhanced support team to serve all of our gearing customers more efficiently."

National Broach & Machine Becomes Nachi Machining Technology

National Broach and Machine Co., the manufacturer of Red Ring® products, a world leader in broaching, gear manufacturing, roll forming equipment and tooling, is changing its name to Nachi Machining Technology Co. The Red Ring® trademark will be retained.

Nachi has worldwide manufacturing facilities and service support offices with over 6,800 employees. Beside broach and gear manufacturing tools and equipment, Nachi is also known for the manufacture of specialty steels, cutting tools, robotics, hydraulics, bearings, heat treatment equipment and specialty machines.

National Broach and Machine Co. has been a part of the Nachi family since 1991. Together, Nachi and National Broach and Machine share over 140 years of processing experience for customer broach and gear manufacturing needs around the world.

New Managing Director for Sumitomo Cyclo Europe

Worldwide power transmission specialist Sumitomo Heavy Industries has appointed Mike McCann as managing director at its European subsidiary, Sumitomo Cyclo Europe. The move is notable because it is the first time a European has taken full control since the Japanese gear giant bought the company from its former German owners in 1993.

McCann originally joined the company in 1998 as sales and marketing director, moving from UK competitor, David Brown Radicon. He takes over from Fuminori "Frank" Miyoshi, who is returning to Japan to take a senior position in Tokyo after seven successful years in Europe and six years previous to that with their American sister company in Chesapeake, Virginia.

Odds and Ends

ASI Machinery Company has been named the exclusive sales rep for Korean hobber manufacturer, Jeil Heavy Industries. • Steve K. Peterson is the new Vice President of Sales, Midwest Region, for SU America, Inc. • Arté Corporation has announced a joint venture with NN Inc. to manufacture plastic gears and components for office automation equipment and industrial applications in Guadalajara, Mexico. • Multi-Arc and Bernex are coming together under one name: IonBond, Inc., which will be a provider of PVD and CVD coating services and equipment. O

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