NEW COUPLING APPLICATIONS OR APPLICATION OF NEW COUPLING DESIGNS

by

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Ken Kraemer joined the Allis Chalmer's tool and die department in 1941. During the war he was stationed in Iran, in the combat military police. When the war ended he was in the officer's candidate school in Fort Benning, Georgia. After getting his discharge he attended the University of Wisconsin in Milwaukee, and rejoined Allis Chalmers in 1947, as a field serviceman. In 1952, he transferred to the Allis Chalmer's compressor engineering department, where he was instrumental in the development of high pressure oil seals, high speed rotor designs, precision couplings, pivot shoe bearings, vibration analysis, and lube and seal systems. In 1960, he was transfered to the compressor department marketing group to develop the compressor customer service group, which he supervises today.

A common, underlying objective of such high speed rotating equipment as turbines, motors, gears, pumps and compressors is to cause a fluid to move from one place to another, and to change from one condition to another, for the purpose of refining a raw product into a finished product. All the elements of such a system are supplied to drive the impeller unit to move the raw product. They have no other purpose.

System design initially attempts to select an impeller unit that will do the job with the maximum efficiency possible, and all other components are subservient to that selection.

Feedback is fundamental to the design of rotating system, especially in the selection of its components. Too often the impeller component—for instance a compressor—yields to accumulated feedback information, and designs are modified to insure total system efficiency and reliability. In a control system, feedback signals are transmitted and the system is adjusted in a matter of seconds. In rotating system designs, it is a matter of years.

For many years compressor designs have been limited by the components available for the system. Constant effort has been directed toward eliminating these restrictions, with the most recent success being the technique developed to apply disc type couplings. Disc couplings can be tuned to suit the connected rotors in a more permanent, comprehensive way than gear couplings can be, and the application technique has employed this suitability.

Illustrating an early èffort to design a component for a system, rather than vice versa, was the application in 1955 of a tube-type torque meter which could be incorporated into the coupling spool design. The torque meter manufacturer suggested a disc type coupling design in order to prevent the excessive dynamic forces present in the gear type coupling of the type proposed, which could damage the meter. Since disc type couplings were then of European manufacture, special design efforts were attempted to permit using the gear coupling configuration instead. The coupling was successfully applied, and is still in use today.

Had the equipment been modified to suit the dynamic forces possible with the original coupling design, the driven shaft overhang would have had to be increased in diameter to handle a potential force of 850 pounds, in addition to the weight of the coupling itself. More important, this would be an unbalance force requiring a shaft weight of 17,000 pounds to insure that the unbalance force would be only 10 percent of the rotor weight.

The driver in this case was a 3500 horsepower, 13,500 rpm turbine with a rotor weight of 1200 pounds. A five-inch coupling was employed to suit the torque meter. The coupling weight was therefore high and the effect of component eccentricity was great.

The redesign in this case, was basically the incorporation of a pilot fit to insure concentricity of the coupling hubs and spacer. The user's problem was to understand the need and then properly install and maintain this pilot.

Such design efforts have been applied on gear couplings for the past 15 years with basically the same results. That is to say, suitably designed, properly manufactured and carefully installed gear couplings have been doing the job. Where improper manufacture or installation errors have caused trouble, the errors could be corrected and the coupling made to operate with good reliability.

Unfortunately, generations go by and the technique used to review coupling designs must be taught over and again to Manufacturers, O.E.M.'s, and Users, at the expense of the real objective, production, if this training is not comprehensive.

Production losses are infinitely greater than the expense of developing application specialists and special training programs for users. The problems of education are, however, greater than this expense implies. This feed back, that many errors of omission, manufacturing, or maintenance are continuing, has influenced American Coupling Manufacturers who are now offering to help solve the problem by the development of the disc coupling design and, more important, sound application procedures.

With both gear and disc couplings, the procedure that takes a predesigned catalog item and applies it without modifications to suit the adjacent components is being eliminated.

The Coupling Manufacturers' engineering is becoming more comprehensive to be able to merge with the rotating equipment engineers in adjusting the coupling design after the Compressor and driver have been designed for optimum performance. With this approach, consistent design success is being achieved.

This leaves a burden on the user, because couplings are less standard and have lower tolerance to damage when handled by personnel unfamiliar with the specific design. Due to this feedback, efforts have been made to seek out a coupling which requires less care and experience in the field. This factor is more often considered today and by comparison has supported the development of the industrial flex-disc coupling. An exercise in this relatively new application technique will point this out and illustrate quickly a few of the reasons for the many recent applications of disc couplings.

APPLICATION EXERCISE

Driven Rotor—3 stage centrifugal compressor with a rotor weight of 350 lbs., rated at 12,000 rpm, driven by an electric motor through a gear increaser.

Gear Increaser—Double helical gear transmitting 2500 hp with a ratio of 6.675 to 1, with a pinion static weight of 100 lbs., and a dynamic weight (at rated load) of 2500 lbs.

Motor—1800 rpm synchronous motor drive with a 6000 lb. rotor. Acceleration time to speed is 12 seconds at no-load conditions.

It is immediately apparent that 12 seconds is not a very long time. If any rotating component has an unexpected radial residual force condition, in excess of the strength of the materials, the initial start may be the last one. Every effort must be made to design to eliminate the possibility of high radial forces. Since we are talking about couplings, let's stay with that component.

Starting with the most extreme conditions, the system to be coupled is a compressor shaft end with a static weight of 175 lbs. and a pinion static weight of 50 lbs. Since even at light loads the pinion bearings will see some dynamic load, a weight or resistance of 100 lbs. will be used for the pinion shaft end.

Upon reaching speed the acceptable *radial* unbalance force of the coupling would be 15 percent of driven and driver shaft resistance, or 26 and 15 lbs., respectively. This indicates the driven shaft end half of the coupling should have less than .1 in.-oz. and the pinion shaft end half .06 in.-oz. residual imbalance, respectively. Quite a target for a gear coupling designer.

If we assume for the moment that the driven shaft and the pinion have no inherent unblance (a poor assumption), the exercise consists of selecting a coupling, suitable for the horsepower, which can be manufactured, and which can be selectively assembled and maintained to operate within these tolerances.

Two types of gear couplings have been applied to to this type of service. One has male teeth integral with the hub, Figure 1, and the other has male teeth integral with the spool, Figure 2. Both have a pilot incorporated into the male tooth form to support the loose member of the coupling in a concentric manner at speed, Figure 3.

Couplings carrying catalog ratings of 4500 horsepower and 16,000 rpm, based on a standard shaft gap, were selected. One coupling catalog wouldn't give the speed rating until the shaft gap was determined, a very good position.

The standard sleeve and spacer weight of the first coupling is 19 lbs. The spool weight for the second type is 4 lbs. One half of these weights is carried by the coupling hubs mounted on each shaft end. Assuming the coupling hubs are mounted without producing any additional eccentric unbalance (another presumption), an eccentricity of .0001 inch on one end of the first spacer would produce .015 in.-oz. of unbalance. The second coupling spool piece would produce .0035 inch-ounces per .0001 inch eccentricity. These pilots, acting at speed can permit only .0004 inch eccentricity and .0015 inches respectively, if we presume the coupling components are perfectly balanced. By their catalog rating both couplings pass.

The next step is to study the operating characteristics of the design to determine if, for instance, the coupling will have an active pilot at operating conditions.

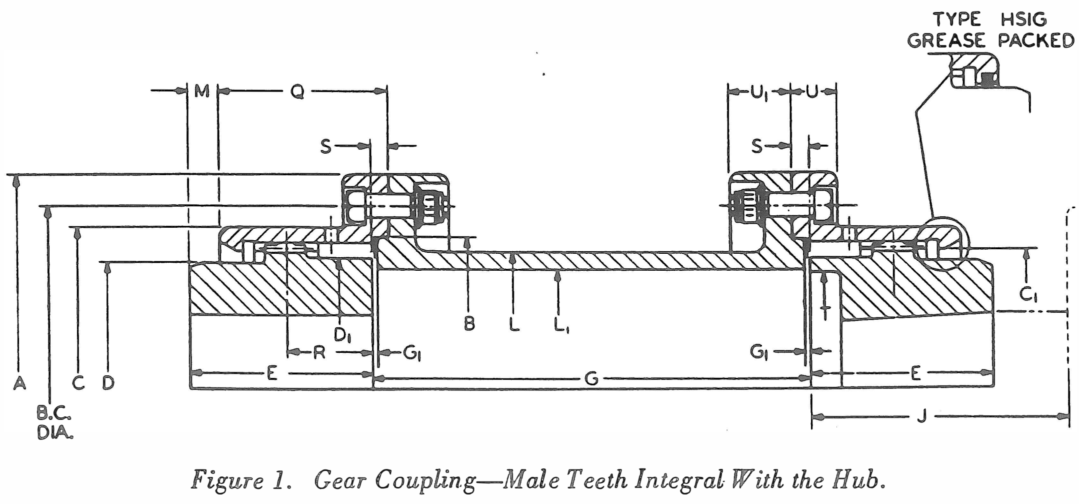


Figure 1. Gear Coupling—Male Teeth Integral With the Hub.

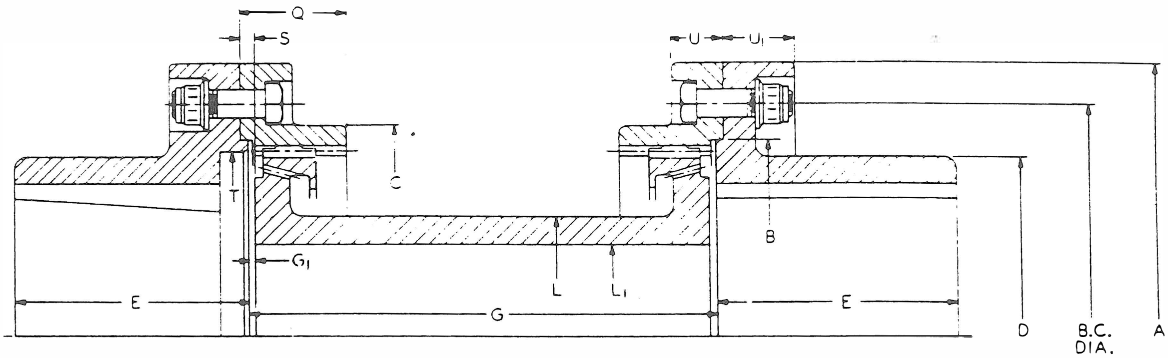


Figure 2. Gear Coupling—Male Teeth Integral With the Spool.

In the first coupling, the heat generated at the teeth, flows differently into the shaft, than it does through the sleeve to the surrounding air. The sleeve will heat up and expand more than the hub. This plus centrifugal force acting on the sleeve will cause it to grow radially as much as .003 to .004 inches more than the hub in proportion to the horsepower being transmitted. Such a differential would permit an eccentricity of .002 inches, or .30 inch-ounces at the rated speed or three times the permissible amount.

The second coupling also develops the same amount of heat, but the hollow bored spool will accept heat in a manner similar to the sleevs to the extent that no differential growth occurs. Therefore, only centrifugal force should be acting on this pilot, amounting to .001 radial growth permitting only .0005 inches eccentricity, or .02 in.-oz. of force.

By these numbers the first coupling could not do the job, and just as clearly the second could. Note that I said *could* do the job.

If this coupling can be expected to normally produce .02 inch-ounces and .06 is the limit we want, there isn't much left for error in the manufacture or assembly.

The user of this coupling must understand this condition and arrange to component balance the half coupling on the shaft it is to be mounted on, so as to eliminate any accumulated unbalance as a result of balance mandrel or shaft eccentricities, or errors in factory balance of the coupling itself. A balancing plane of tapped holes is needed for that purpose. In the field, where replacement is necessary without dismantling the units, field balance checks are almost mandatory.

There is another area of evaluation—sliding friction coefficent. This produces a resistance to the axial movement necessary as rotors heat and expand. For the same catalog size rating, both coupling types have about the same pitch line diameter, so the unit load on the teeth would be the same. This load and a selected friction coefficient result in a force which must be handled by the thrust bearings and, in this case, one helix of the double helix gear before the teeth will slide and the normal operating position is achieved. A reaction will occur during every thermal change the system goes through.

This is a subject of more interest for the user than it may be for the designer, since the user must contend with tooth surface finishes other than new or as manufactured. This is the reason the words "a selected friction coefficient" were used.

This characteristic of the gear coupling therefore requires a serious study of the capabilities of the driven shaft thrust bearing and the loading service factors used in the gear design "in order to suit the coupling."

Intensive design efforts and complete design changes have been made in recent years to gear couplings to diminish the sliding friction coefficient, with some success, but again the burden is on the user to understand and maintain ideal conditions to insure continuous production.

Closely associated with this latter problem is the matter of lubrication required by gear couplings. A failure in this area results in the infinite sliding friction factor—the locked coupling.

This exercise could continue, for there is much more, but this is sufficient to illustrate the many characteristics users must understand to utilize gear couplings.

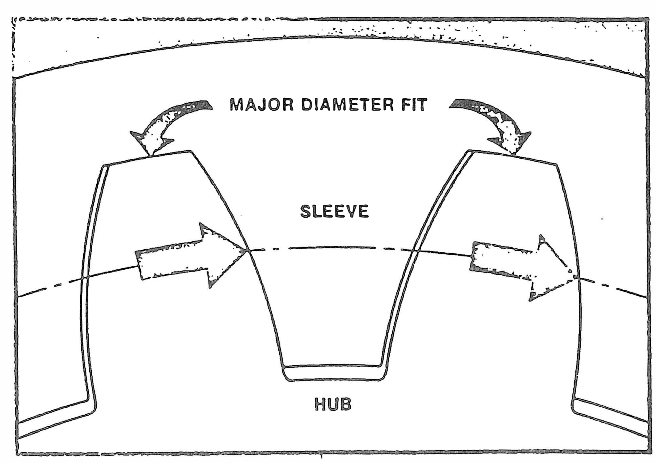


Figure 3. Schematic of Gear Used in Coupling Applications.

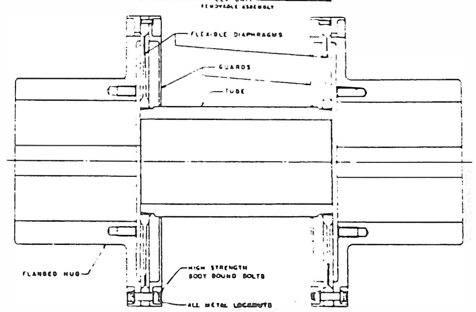


Figure 4. Schematic of a Typical Disc Coupling.

Taking the same conditions, here is a disc coupling to evaluate (Figure 4). The same maximum permissible unbalance values exist, but in this coupling there are only the static pilot fits to consider. There are no changes due to operating temperature or centrifugal forces to cause operating eccentricities. Almost all transient factors can be accounted for by the designer in this selection of the components used to fabricate the coupling, but in this case, he must know and therefore requests shaft characteristics before designing the discs.

The same manufacturing errors that can cause eccentricity in gear couplings can occur in disc couplings, but these can be detected before operating the unit and the chances of them occurring are greatly minimized.

The same assembly eccentricities can occur in the field, but the absence of dynamic radial residuals in the assembly permits more use to be made of the concentricity tolerances allowed.

The one area the user must be aware of is the tolerance permitted for axial growth or axial growth transients shown in Figure 5. This can be a more serious problem with disc couplings than with gear couplings on certain types of units. A rather specific control is placed on the disc deflection range, and the equipment has to be adjusted axially to suit with more accuracy than with gear couplings. Some units are very difficult to move once they have been set. A gas expander, for instance. Where this is a problem, however, arrangements can be made to permit repositioning the coupling by remachining the components for a one time per coupling fit-up, or by adding a spacer plate.

The application of a disc coupling therefore almost completely eliminates the every generation re-education of coupling users in the intricacies of the design itself—once the coupling is designed, because in order to design it, the coupling characteristics must be determined and adjusted to suit the connecting rotors. Furthermore, there are no changes or wear taking place in the coupling characteristics throughout the life of the unit.

In the design stages, the effects of concentricity, alignment, overhung bending moment during mis-alignment, transient thermal growth and net growths are incorporated. The resonant pattern of the design is then matched to the pattern of the connected rotors.

In the manufacturing stages it is relatively easy to balance the assembly to low residuals as dictated by the shaft design, and the mechanical fits are simple turns, bores and faces.

As with gear couplings, balancing is done on a mandrel, so potentially some eccentricity can occur when the coupling is mounted on the shafts in the field. Since the coupling flex members are rigid radially, field balancing is not complicated by dynamic changes to the pilot fits.

One of the main features of the disc coupling is its known axial deflection load value. When compared to the varying unknown sliding friction factor in the gear coupling, this feature eliminates the greatest concern of the user and to an equal extent the designers of the rotating equipment. In the case of the previous example a specific load can be used in sizing the gear and thrust bearings. This eliminated a big unknown factor in system designs.

In this exercise, both the disc, and the gear couplings can be applied. The difference between them can be noticed, but even though one has advantages over the other, both will succeed as couplings.

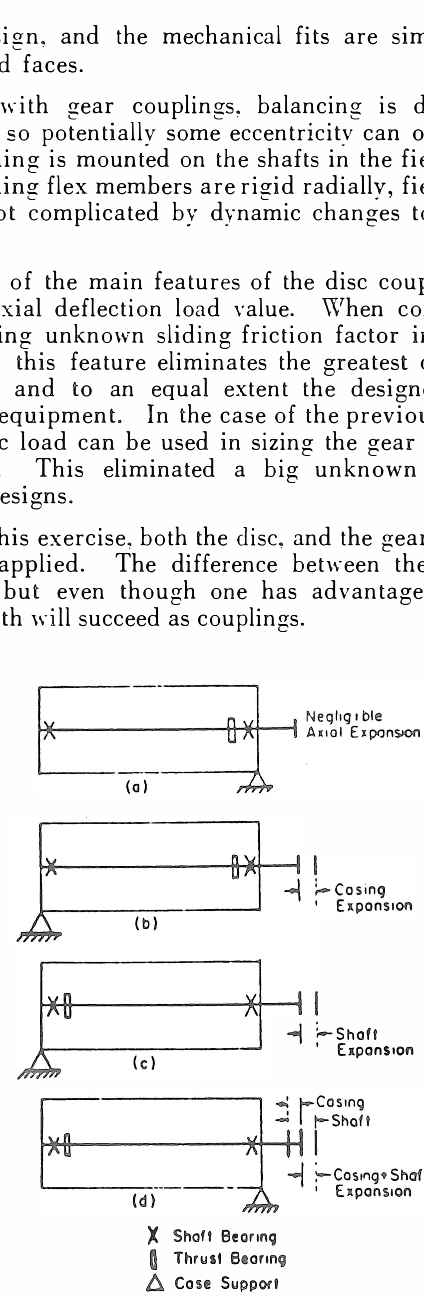


Figure 5. Thermal Movements of Shaft and Casing.

The one, overriding difference to users is the obvious promise that the disc coupling will be less susceptible to usage than the gear coupling, and that it will require less attention.

It should also be very apparent that turbomachinery couplings, whether gear or disc type, should not be simply picked from a catalog.

It is also evident that several American Companies will be offering disc couplings of different designs. Those presently available are shown in the appendix. Each type must be evaluated.

There will be also two phases of coupling applications. One will be new applications on new rotating systems. The second will be the replacement of existing couplings with one of a new type. This latter phase poses the most problems in application.

As mentioned earlier, an intimate knowledge of the rotating equipment is necessary to proper coupling design. If an existing coupling is to be replaced with a new type, there is good justification to review, with the latest techniques, the nature of the rotating system to be coupled. Some installations are very old and some have been revised in other ways in the field. Such engineering reviews are not easy to arrange with busy equipment suppliers.

The tendency is therefore to match the obvious characteristics of the existing coupling and see what happens. With many older designs having relatively heavy and larger diameter shafts the retro-fits have been, as far as we can tell, very successful and trouble free. Part of this success is due to the consideration given to the retrofit by cooperating engineers of the coupling manufacturer and the rotating equipment manufacturer. A large part is due to the dedication of the first companies offering the disc coupling wherein extra efforts to insure success have been made.

If retro-fits and new installations consume the available time of these engineers, the potential for omission increases. Therefore, more time should be allowed for the work.

Until now our applications of disc couplings have utilized the type offered by the Bendix Fluid Power Division. In addition to new unit applications, an increasing number of users of Allis-Chalmers compressors have been replacing the original gear coupling with the Bendix type, either through Allis-Chalmers or directly with Bendix. All of the new unit installations have been successful and, with one exception, all of the retro-fit units have been successful. Characteristically, the one case where it was not, the gear coupling characteristics were simply copied. When the driver and driven shaft characteristics were suitably considered, and the coupling modified, success was achieved. Appendix II contains a list of both new and retro-fit applications on Allis-Chalmers units.

The purpose for changing the coupling varies unit by unit in accordance with the problem stated earlier. In very few cases was it necessary because the gear coupling could not be made to work. In most cases, it was because of the training necessary to make the gear coupling work.

In one application listed, the retro-fit disc coupling has never been installed, because after careful fitting, the gear coupling is still operating after nearly two years of almost continuous duty.

CONCLUSION

The application of couplings is an engineering effort involving the coupling and rotating equipment designers. The user, by the purchasing technique he employes, can aid or hinder this effort. He does have the choice of the basic style of coupling he feels his operations and maintenance people should have to work with.

If his personnel tend to be experienced, the user may not have too much trouble in deciding, but otherwise, all indications are that a choice of a disc coupling is recommended.

In either case, a good purchase spec should recognize that the selection and design of the coupling must follow the rotor design work and should exclude the coupling from becoming involved in competitive bids. It is simply too important an item to risk reliability for initial cost savings.

Finally, the redesign of existing system should be scheduled to suit the available engineering time to insure an adequate review, and no effort to obtain unreasonable liability coverage out of proportion to the coupling being purchased should be made so as to prevent engineering involvement by the rotating equipment firms because of the bad risk factors.

REFERENCES

1. N. B. Rothfuss: "Design and Application of Flexible Diaphram Couplings to Industrial-Marine Gas Turbines." ASME 73-GT-75.

2. John Wright: "Which Flexible Coupling?" Power Transmission Handbook 71/72.

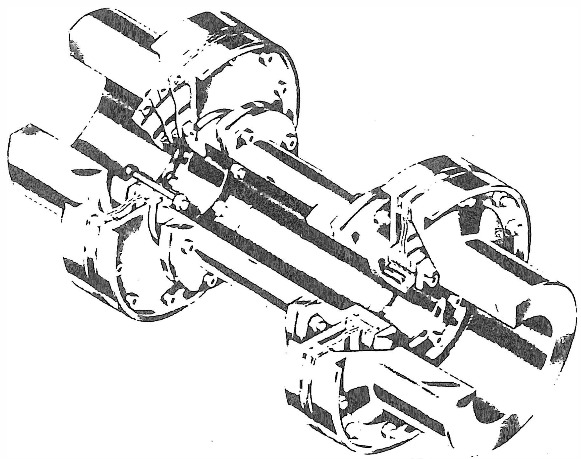
3. P. C. Renso, S. Kaufman, D. E. Derocker: "Gear Couplings." ASME 67-WA/DE-8.

4. L. H. Pray, E. W. Goody: "All Metal Laminated Flexible Element Couplings." ASME-PTG-38.

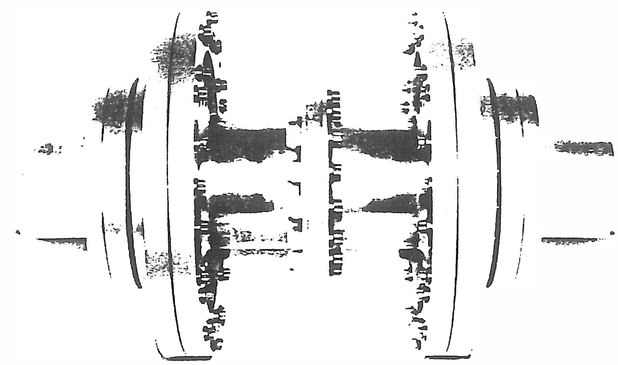
5. J. H. Anderson: "Turbocompressor Drive Couplings." Transactions of the ASME Journal of Engineering for Power.—1962.

APPENDIX I

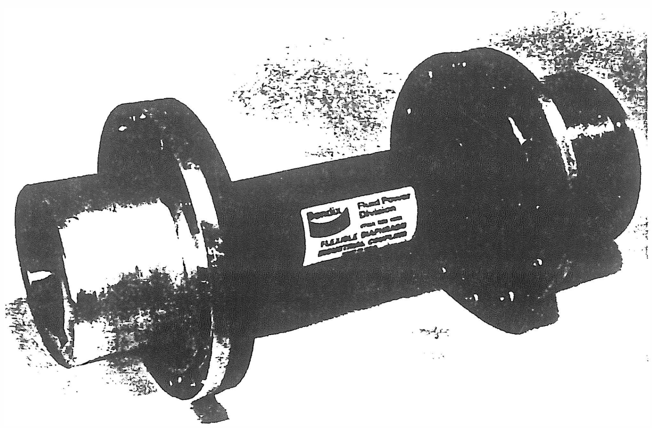
Various Types of Couplings Manufactured



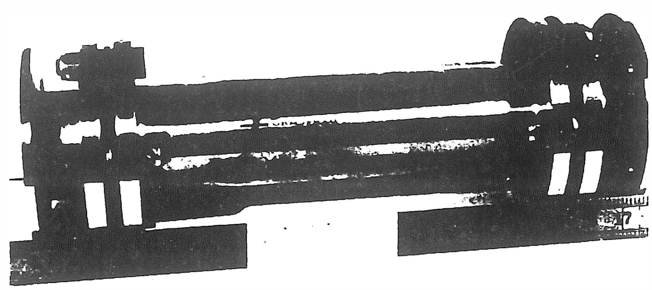
Ameriflex—Manufactured by Zurn Industries, Erie, Pennsylvania.



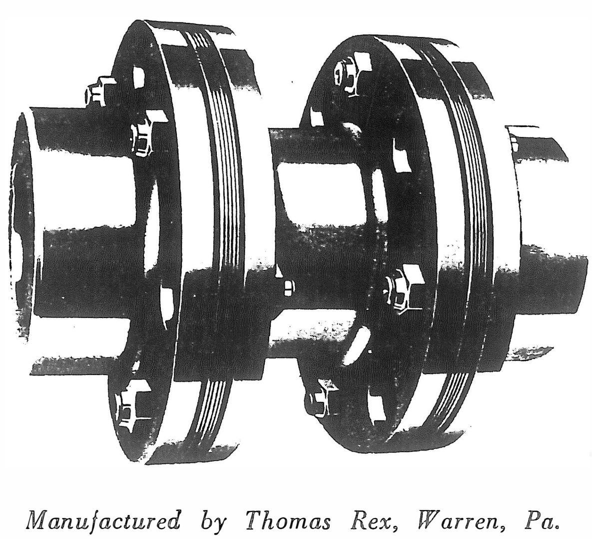
Flexor—Manufactured by Coupling Corp. of America, New Hope, Pennsylvania



Flexible Diaphragm — Manufactured by Bendix Fluid Power Division, Utica, N.Y.



Flange-mounted, 4-bolt Coupling — Manufactured by Formspray Company.



Manufactured by Thomas Rex, Warren, Pa.

APPENDIX II

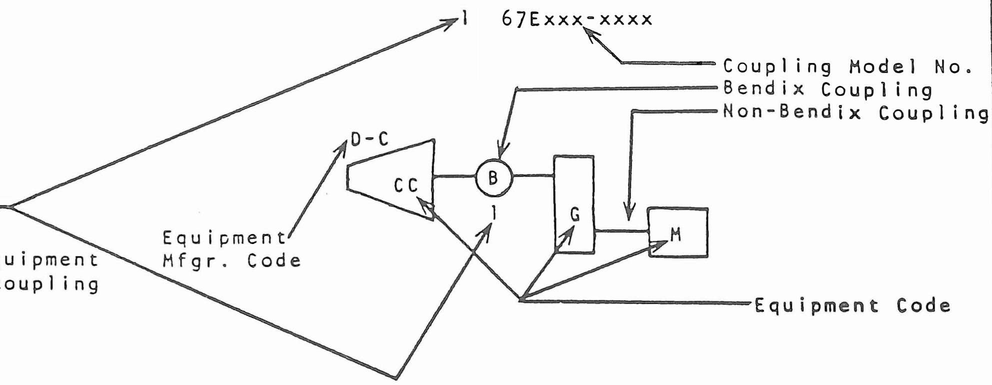
New and Retrofit Applications on Allis-Chalmers Units

EQUIPMENT USER
Equipment Location
Start Up Date
Type of Duty

1 Rated HP Rated RPM

Coupling No. Key

Description of how the equipment is being used. Name of coupling purchaser.

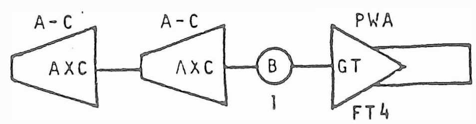


P&WA - WILLGOOS LAB
E. Hartford, Conn.
August - December 1968
Test Facility Duty

1 30,000 HP 3600 RPM

Gas turbine engine driving axial compressors which exhaust an aircraft engine altitude test chamber. Retrofit installation by P&WA.

1 67E322-0042



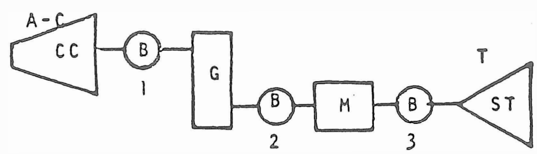
Three Identical Trains

UNION CARBIDE
Whiting, Indiana
February 1970
Continuous Duty

1 4600 HP 7280 RPM
2 4600 HP 3580 RPM
3 715 HP 3580 RPM

Centrifugal Blower pumping Propylene in a chemical plant. The steam turbine is used for starting. New installation by Allis Chalmers.

1 67E408-0058
2 67E410-0059
3 67E306-0086

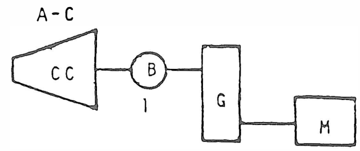


AIRCO INDUSTRIAL GASES
Buffalo, New York
April 1970
Continuous Duty

1 13,500 HP 5300 RPM

Centrifugal compressor providing main air to an air separation plant supplying a steel mill. Retrofit installation by Airco.

1 67E414-0172

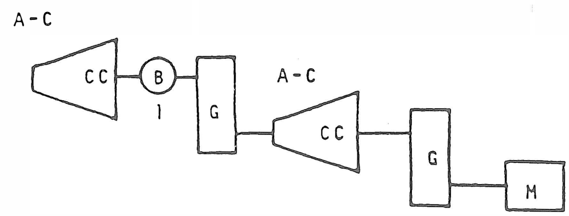


PENNSYLVANIA ELECTRIC CO.
Connemaugh, Pa.
November 1970
Continuous Duty

1 1600 HP 14,600 RPM

Centrifugal compressor for soot blowing air in a steam driven electric generating plant. Retrofit installation by Allis-Chalmers.

1 67E206-0230



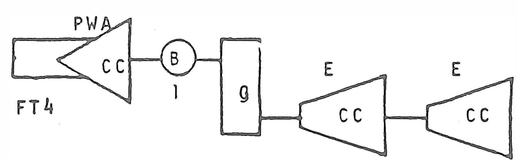
Three Identical Trains

OCCIDENTAL PETROLEUM
Lybia
December 1970
Continuous Duty

1 19,600 HP 3600 RPM

Gas turbine engine driving a centrifugal compressor pumping natural gas. New installation by Turbo Power & Marine Systems.

1 67E322-0066



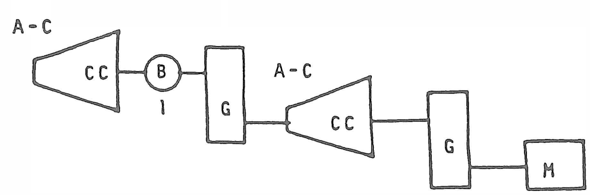
Two Identical Trains

PENNSYLVANIA ELECTRIC CO.
Homer City, Pa.
April 1971
Continuous Duty

1 1190 HP 15,300 RPM

Centrifugal compressor for soot blowing air in a steam driven electric generating plant. Retrofit installation by Allis-Chalmers.

1 67E206-0227

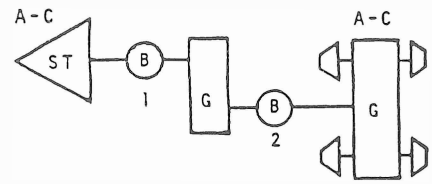


Three Identical Trains

CHEVROLET DETROIT FORGE
Detroit, Michigan
May 1971
1-2 Shifts/Day

1 67E410-0107
2 67E414-0108

1	4000 HP	3600 RPM
2	4000 HP	1200 RPM

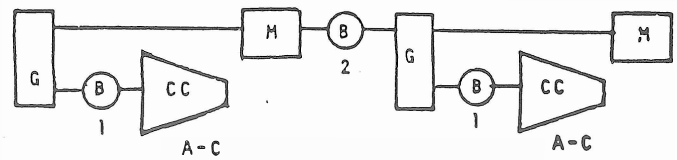


Steam turbine driven four poster compressor supplying plant air to a forging shop. New installation by Allis-Chalmers.

ARNOLD AIR FORCE BASE
Tullahoma, Tenn.
October 1971
Test Facility Duty

1 67E308-0308
2 67E414-0349

1	4000 HP	15,135 RPM
2	4025 HP	1,800 RPM

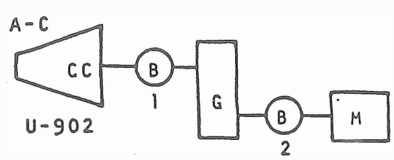


Centrifugal compressors supplying air for wind tunnel testing. New installation by Allis-Chalmers/Phoenix General.

ASARCO-MEXICANA, SA
San Luis Potosi SLP, Mexico
December 1971
Continuous Duty

1 67E406-0291
2 67E410-0292

1	1750 HP	6300 RPM
2	1750 HP	1750 RPM



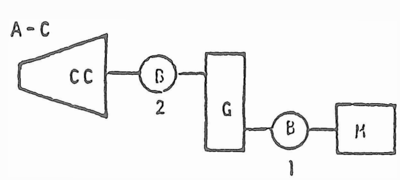
Centrifugal blower supplying air for copper converter plant. New installation by Allis-Chalmers.

DOW CHEMICAL
Stade, Germany

Continuous Duty

1 67E314-0360
2 67E308-0361

1	2850 HP	1500 RPM
2	2850 HP	5500 RPM

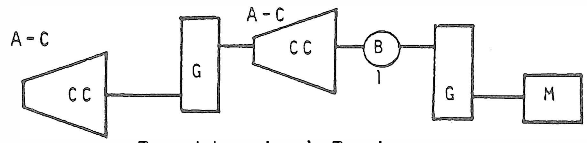


Centrifugal compressor in a chemical plant. New installation by Allis-Chalmers.

COLUMBUS & SOUTHERN OHIO ELECTRIC CO.
Conesville, Ohio
Continuous Duty

1 67E210-0327

1 3450 HP 8832 RPM



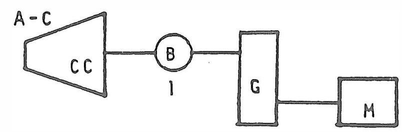
Two Identical Trains

Centrifugal compressor for soot blowing air in a steam driven electric generating plant. New installation by Allis-Chalmers.

AIRCO INDUSTRIAL GASES
Johnstown, Pennsylvania
July 1972
Continuous Duty

1 67E408-0116

1 4500 HP 8433 RPM

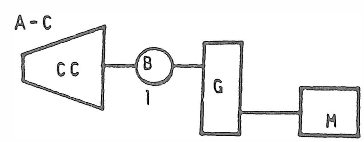


Air separation plant. Retrofit installation by Airco.

AIRCO INDUSTRIAL GASES
Bethlehem, Pennsylvania
Continuous Duty

1 67E414-0182

1 10,600 HP 4850 RPM

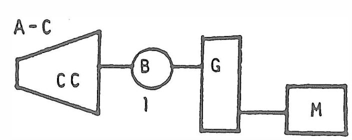


Air separation plant. Retrofit installation by Airco.

AIRCO INDUSTRIAL GASES
Bethlehem, Pennsylvania
Continuous Duty

1 67E310-0183

1 6500 HP 11,311 RPM

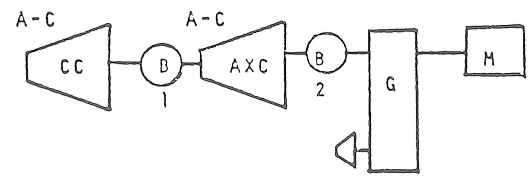


Air separation plant. Retrofit installation by Airco.

AIRCO INDUSTRIAL GASES
Chester, West Virginia
March 1972
Continuous Duty

1 4300 HP 4930 RPM
2 15,000 HP 4930 RPM

1 67E410-0261
2 67E414-0266



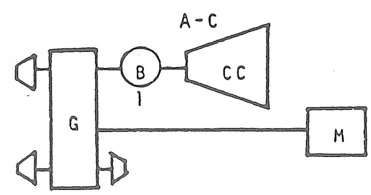
Air separation plant. Retrofit installation by Airco

Two Identical Trains

AIRCO INDUSTRIAL GASES
Chester, West Virginia
March 1972
Continuous Duty

1 7200 HP 12,060 RPM

1 67E408-0267

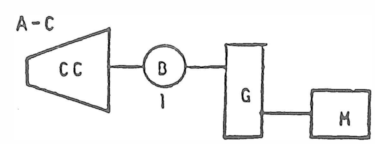


Air separation plant. Retrofit installation by Airco.

LINDE DIVISION UNION CARBIDE
Deer Park, Texas
Continuous Duty

1 3000 HP 8697 RPM

1 67E210-0383

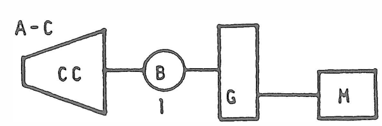


Chemical Plant. New Installation by Allis-Chalmers

DEL MAR VA POWER CORPORATION
Wilmington, Delaware
September 1972
Continuous Duty

1 2250 HP 13,450 RPM

1 67E306-0402

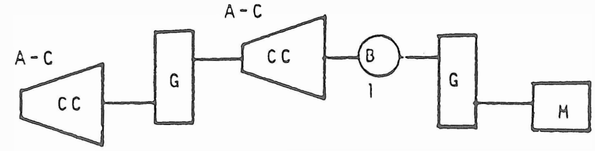


Steam Powered Electric Plant. New installation by Allis-Chalmers and Lotepro

SOUTH CAROLINA POWER AND LIGHT
South Carolina
Continuous Duty

1 3000 HP 8603 RPM

1 67E210-0424



Steam Power Electric Plant.
New Installation by Allis-Chalmers

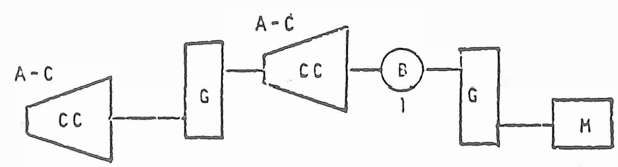
Three Identical Trains

NEW ENGLAND POWER COMPANY
Braydon Point, Rhode Island
Continuous Duty

1 2500 HP 8582 RPM

1 67E210-0448

Steam Powered Electric Plant.
New Installation by Allis-Chalmers.

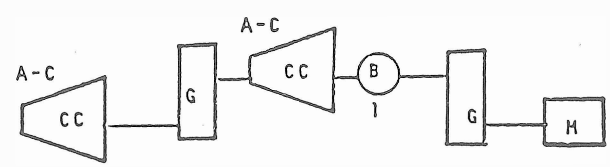


DAYTON POWER AND LIGHT COMPANY
Dayton, Ohio
Continuous Duty

1 2100 HP 8809 RPM

1 67E208-0449

Steam Powered Electric Plant.
New Installation by Allis-Chalmers.

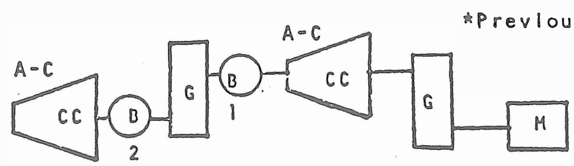


PENN ELECTRIC COMPANY
Connemaugh, Pennsylvania
Continuous Duty

1 1671 HP 8746 RPM
2 1600 HP 14600 RPM

1 67E306-0467
2 67E206-0230*

*Previously Listed



Steam Powered Electric Plant.
Retrofit Installation by
Allis-Chalmers

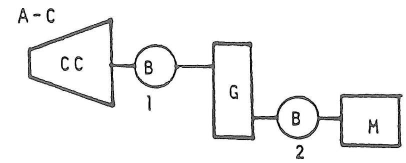
Three Identical Trains

ASARCO
El Paso, Texas
Continuous Duty

1 2250 HP 5100 RPM
2 2250 HP 1780 RPM

Sulfuric Acid Plant. New Installation by Allis-Chalmers.

1 67E408-0482
2 67E410-0483



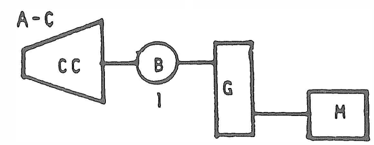
Two Identical Trains

UNION CARBIDE
Deer Park, Texas
Continuous Duty

1 1800 HP 8750 RPM

Chemical Plant. Retrofit Installation by Allis-Chalmers.

1 67E306-0601

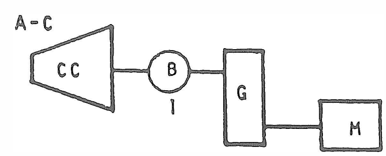


MOBIL OIL COMPANY
Beaumont, Texas
Continuous Duty

1 600 HP 12480 RPM

Refinery. New Installation by Allis-Chalmers and Ford, Bacon, Davis.

1 67E206-0653



Three Identical Trains

NIPAK
Karens, Texas
Continuous Duty

1 4190 HP 8050 RPM

Chemical Plant. Retrofit Installation by Allis-Chalmers and M.W. Kellogg

1 67E212-0732