A Designer's Guide to Precision Shaft Couplers

Huco Engineering Industries Ltd
Merchant Drive, Hertford, Herts SG13 7BL, UK
Tel: +44 (0)1992 501900 Fax: +44 (0)1992 500035
e-mail: hei.sales@huco.com
www.huco.com

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Huco is a specialist manufacturer of miniature flexible shaft couplers and related devices. It began operations in the early 1960's with the production of the first plastics U/J, a product that has since become an industry standard for light duty drives.

Setting standards for cost-effective design has become a tradition at Huco. We now manufacture one of the widest product ranges in this specialised area of power transmission.

Mostly, the products find application in positional systems where the criteria for selection can differ considerably from those applied to conventional power transmission. This booklet attempts to alert the designer to the issues that should be addressed and offers guidance on selection from the options available.

Huco products are distributed world-wide and are specified by OEM and Government Agencies in most of the major industrialised nations.

PRODUCT OVERVIEW

Flexible couplers connect two shafts, end-to-end. They transmit rotation from one to the other and accommodate the slight, *unintentional* misalignment that is always present when two independently mounted shafts are presented for connection. There are many kinds of flexible couplers and while they all fulfil their primary purpose, each has certain characteristics that make it more effective in one application than another.

UNIVERSAL JOINTS are a specialised category of mechanical flexible coupler. Because their capacity for misalignment is greater, their main purpose is to transmit rotation through *deliberate* angular and parallel shaft offsets rather than to compensate for *unintentional* misalignment.

DISC & BELLOWS COUPLERS accommodate *unintentional* misalignment through flexure of thin metallic membranes. They have no moving parts and are suited to applications that demand translational accuracy of motion, high speeds and high torsional stiffness. The **Disc couplers** are the more predictable and offer near-infinite life. The **Bellows couplers** are more sensitive to misalignment excursions but offer very high torsional stiffness.

LATERAL DISPLACEMENT COUPLERS operate on mechanical principles that are more suited to general purpose applications characterised by lower speeds, intermittent operation and a propensity for torsional damping. They accommodate misalignment through sliding contact bearings that are under controlled pre-load and this gives them their backlash-free property. Unlike flexural couplers their misalignment capacity is not governed by their overall length and quite severe alignment errors can be accommodated within short space envelopes. When mounted on accurately aligned shafts, lateral displacement

	UNIVERSAL JOINTS			DISC COUPLERS		BELLOWS COUPLERS		
	HUCO-Pol single	HUCO-Pol double	HUCO-Pol teleshaft	HUCO-*flex* M single-stage	HUCO-*flex* M two-stage	HUCO-*flex* B short type	HUCO-*flex* B long type	HUCO-*flex* B stretched type
PARAMETERS								
operating principle	mechanical			metallic flexure		metallic flexure		
sizes in production	7	4	6	6	6	4	4	4
bore range, metric ø mm	3 - 20	3 - 10	3 - 20	3 - 28	3 - 28	3 - 20	3 - 20	3 - 20
bore range, inches dia. in	0.125 - 0.750	0.125 - 0.375	0.125 - 0.750	0.125 - 1.000	0.125 - 1.000	0.125 - 0.750	0.125 - 0.750	0.125 - 0.750
peak torque up to lb.in	95	11.5	95	530	530	88	44	110
torsional stiffness up to lb.in/rad	2025	111	N/A	168000	106000	25490	11594	19870
typical max speed rpm	1000	1000	1000	5000	5000	5000	5000	5000
angular compliance	✓	✓	✓	✓	✓	✓	✓	✓
radial compliance	-	✓	✓	-	✓	✓	✓	✓
axial compliance	-	-	✓	✓	✓	✓	✓	✓
shaft attachment	set screws or cross-pinning			set screws or integral clamp				
keyways, 'D' bore, etc.	✓	✓	✓	✓	✓	✓	✓	✓
electrical isolation	✓	✓	✓	-	-	-	-	-
drive can be de-coupled without detaching hubs	-	-	✓	-	-	-	-	-
backlash	zero (e)	zero (e)	(f)	zero		zero		
satisfies shaft configuration figs.	30, 31	19, 20	19, 20,30, 31	30, 31	19, 20	30, 31	19, 20	19, 20
for data, refer to page								

(a) up to 10mm and 0.375" available at outer hub.

(b) optional with these couplers which can be mounted partially disengaged to provide axial compliance.

(c) set screw attachment only.

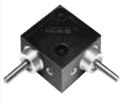
couplers can remain backlash-free for over 10^8 revolutions, depending on the operating conditions. **Uni-Lat couplers** have good damping characteristics while the torsionally stiffer **Oldham** is a 3-part, disengageable coupler with a replaceable wear element.

In general, mechanical couplers are available at lower cost than flexural types and ultra-low cost versions can be manufactured for low duty, high volume applications.

FRICTION CLUTCHES interrupt rotation when the torque being transmitted reaches a pre-determined threshold. **Vari-Tork** clutches are user-adjustable and can function as torque limiters, over-run devices, and brakes. As torque limiters, they protect equipment and personnel from damage or injury in the event of a system malfunction; as over-run devices, they absorb the inertia of a motor driving a load against a limit stop; as brakes, they stabilise rotation of a feedspool, for example, by applying a drag torque.

BEVEL GEARBOXES. Depending on the model and its installation, bevel gearboxes can drive two loads from one input, can introduce a ratio between input and output or simply take the drive through 90°. The **L-Box** is intended for manual or intermittent operation while the **T-Box** is continuously rated and can be specified with 2 or 3 shafts and 1:1 or 2:1 ratios.

HUCO-LOK BORE ADAPTORS are split sleeves that adapt designated bores in a coupler to fit a variety of shaft diameters. They are tensioned by the prevailing fastening system and offer secure fastening without scoring the shafts.

LATERAL DISPLACEMENT COUPLERS			FRICTION CLUTCHES				BEVEL GEARBOXES		BORE ADAPTORS	for more infor-mation on para-meters refer to page
Uni-Lat	Standard Oldham	Oldham X–Y	Vari-Tork concentric	Vari-Tork in-line	Vari-Tork clutch + coupler	Vari-Tork single-plate	L-Box	T-Box	HUCO-Lok	
										
mechanical			mechanical				mechanical			
4	6	6	1	1	1	1	1	1	10	-
3 - 16	3 - 16	4 - 30	6 & 8	6 & 8 ⓐ		4	4 (shaft)	5 (shaft)	2 - 25	-
0.125 - 0.625	0.125 - 0.625	0.157 - 1.125	0.250 & 0.313	0.250 & 0.313 ⓐ		-	-	-	0.125 - 1.000	-
31	150	390	11.7 (max drag torque)			0.7 oz.in	6	6	-	12
2645	10620	23100	-	-	-	-	-	-	-	13
3000	3000	3000	1000 (slipping speed)				250	1500	-	-
✓	✓	✓	-	-	✓	-	-	-	-	4, 5, 6, 10
✓	✓	✓	-	-	✓	-	-	-	-	4, 5, 6, 7
	✓ ⓑ	✓ ⓑ	-	-	✓ ⓑ	-	-	-	-	4, 6, 11
set screws or clamp			set screws or clamp	ⓒ	set screws or clamp	ⓒ	-	-	-	-
✓	-	✓	-	-	-	-	-	-	-	-
✓	✓	✓	-	-	✓	-	✓ ⓓ	-	-	-
-	✓	✓	-	-	✓	-	-	-	-	-
zero ⓔ			2° max			zero	2° max		-	14
19 & 20			-	-	-	-	-	-	-	7 & 10
										-

ⓓ one model only.

ⓔ zero backlash is achieved through pre-load in the bearings.

ⓕ for higher speeds, balancing and special shaft clamping arrangements are necessary.

An Introduction to Miniature Flexible Shaft Couplers

Flexible shaft couplers transfer rotation from one shaft to another and compensate for the slight alignment errors that are a consequence of connecting two independently supported shafts. The means of compensating for misalignment is interposed between the input and output hubs and must also do duty as the medium through which torque is transmitted. While being pliable in the planes of misalignment, the medium must simultaneously resist torsion. It is known as the coupler's ***compliance mechanism*** and its capacity for allowing relative displacement of the hubs is called ***compliance.***

Torsional Stiffness

The measure of torsional resistance is called ***torsional stiffness*** and is significant in closed-loop velocity and motion control systems where it determines the upper limit of performance and stability when the loads are predominantly inertial. The subject is discussed later under 'SELECTION' (page 13). Couplers with low torsional stiffness are said to be ***torsionally damping*** and this property can be beneficial in transmissions that are subject to shock loads. Note however that torsional damping, by definition, impairs translational accuracy.

Backlash

Backlash can be described as torsional free play between input and output hubs and is unlikely to impair steady state, uni-directional drives. When rotation is bi-directional, backlash produces a dwell while the drive takes up the lost motion. In highly dynamic drives, i.e., rapid, short-term movements, backlash can become a source of noise and can cause instability in closed-loop systems. In open-loop systems, backlash can result in a loss of positional accuracy comparable to that of a torsionally damping coupler.

Torque

There are a variety of definitions relating to a coupler's torque capacity, such as ***nominal torque, reversing torque*** and ***peak torque***. The following definition and rationale is specific to the Huco product catalog.

A coupler's capacity for transferring rotation under load is determined by reference to its ***Peak Torque Rating (peak torque).*** This rating is derived from testing and is the ***reversing torque load*** that the coupler has sustained for a minimum of 1 million cycles. It does not necessarily indicate the ultimate limit of endurance. In our terms, the ***application torque*** multiplied by the ***service factor*** should be less than the ***peak torque*** to achieve optimum life. (See page 12).

Static break torque as applied to Oldham couplers, defines the value at which catastrophic failure (breakage) occurs.

Life

A coupler's ***life requirement*** is specified in operational hours which reflects the life of the machine or the intervals between coupler service or replacement. Its ***life expectancy*** is predicted as the number of revolutions it can sustain and the factors governing life are the nature and extent of the alignment errors and the load. With the predominant speed of rotation known, the prediction can be resolved to hours.

In conditions of perfect shaft alignment and uni-directional rotation, couplers can sustain their peak torque values almost indefinitely. Misalignment warrants more careful control. Even when there is little or no load on the drive, misalignment beyond the recommended limits can induce fatigue failure in flexural devices such as disc and bellows couplers and accelerate wear in mechanical devices such as lateral displacement couplers and universal joints. Torque capacity decreases with misalignment and there is thus a delicate balance between torque, misalignment and life expectancy.

The interdependence of these parameters should be considered in applications where the life requirement in revolutions is either relatively very high or very low. For example, a coupler transmitting periodic incremental rotation over a total life requirement of 5 x 10^3 revolutions will tolerate higher levels of torque and/or misalignment than one transmitting continual reversals at 4,000 rpm over a 24-hour day. Torque and/or misalignment in the first example could safely exceed the recommended values listed for the coupler while in the second, the corresponding values could be optimistic.

Performance values published in catalogs are intended as a guide. They are based on physical simulations that test for optimum combinations of torque and misalignment having regard to the type of application targeted for the coupler. A comprehensive description of what is wanted of the coupler identifies the priorities and these determine the most suitable type and size of coupler.

Attachment Systems

Set screws are an effective, low cost method of shaft attachment provided the screws seat on flats or dimples machined into the shafts. Without these features, the shafts may be difficult to withdraw once the set screws have scored the surface. Set screws are prone to loosen under vibration but can be secured using appropriate semi-permanent adhesives or self-locking set screws.

Clamp fixing does not score the shafts and permits incremental phasing of input to output. Traction increases with shaft diameter and this should be a close fit in the bore. The tensioning screws remain secure under vibration.

High integrity drives are provided by keys fitted to matched keyways in shaft and bore. For smaller shafts, 'D' bores broached into clamp hubs provide high integrity fastening with no possibility of slippage.

Specifying

The process of specifying a coupler should start early on in the evolution of the design. Given sufficient space, almost any operating parameter can be met.

In general terms, transmissible torque increases with coupler diameter. If continual high speeds are a feature of the application, ***metallic flexural*** couplers should be considered and in these, radial misalignment compensation increases with coupler length. For lower speeds, intermittent or periodic rotation, a more compact solution for severe misalignment is often provided by ***lateral displacement*** couplers.

Size for size, ***bellows*** couplers provide the highest levels of torsional stiffness, closely followed by ***disc*** couplers which have the best potential for infinite life. Couplers that incorporate plastics or elastomers in the construction offer torsional damping characteristics and electrical isolation.

Deliberate shaft offsets which are outside the compliance range of misalignment couplers can be resolved by ***universal joints***.

It is sometimes the case that actual values do not match the predictions. This is most common in the prediction of misalignment, possibly because the values are diminutive and difficult to measure, and probably because their importance goes unrecognised. Underestimated misalignment, especially the radial kind, is a frequent cause of premature coupler failure. It can be avoided through a better understanding of the ORIGINS, NATURE and PREDICTION of misalignment which are discussed in the following pages.

MISALIGNMENT

Generally speaking, transmissible torque and/or life expectancy reduce exponentially for linear increases in misalignment. Quantifying the maximum misalignment expected in an application is therefore important and a closer look at the nature and origins of these errors and their prediction will help avoid premature failure.

Origins of Misalignment

Misalignment is the variance between the intended position and attitude of two shafts and the actual. It is generated by the manufacturing tolerances on parts that influence shaft alignment and some of the sources of error are shown below.

Figs.1–4 generate angular errors in alignment and Figs.5–8, move the shafts out radially without changing their inclination. Shaft run-out, Fig.9, gives rise to a conical and/or orbital motion at the shaft termination. Axial displacement, Fig.10, is the longitudinal movement of a shaft caused through thermal expansion/contraction, working clearances (end-float), settlement, etc.

Alignment is affected by thermal imbalances, wear, settlement, and creep and can deteriorate over the lifetime of the machine.

Alignment should be checked cold and again when the machine reaches its operating temperature. A further consideration is whether the alignment accuracy achieved in the factory can be repeated by service engineers replacing parts on site.

The class of tolerance can vary across individual items of connected equipment. For example, the output shaft of a reduction gearbox with a die-cast housing having unmachined mounting surfaces, no register, and clearance holes for attachment purposes will have greater latitude for misalignment than the shaft of a high quality, face mounted servo motor with a machined register. Items mounted on brackets will have greater latitude for misalignment than those mounted direct to precision machined surfaces.

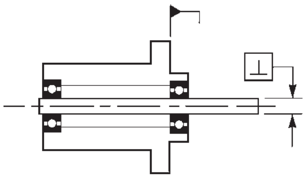


Fig.1 Perpendicularity of shaft axis to mounting datum

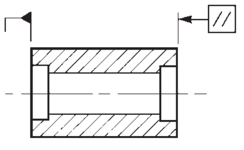


Fig.2 Parallelism of mounting faces

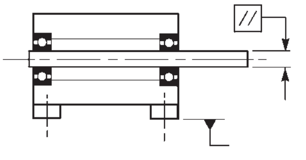


Fig.3 Parallelism of shaft axis to mounting datum

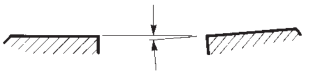


Fig.4 Angular alignment of mounting surfaces

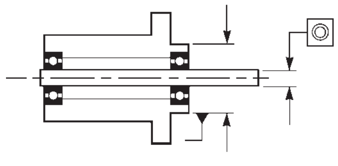


Fig.5 Concentricity of shaft axis and locating register

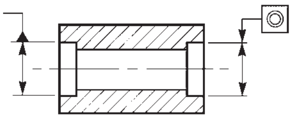


Fig.6 Concentricity of counter bores. Clearances in locating registers

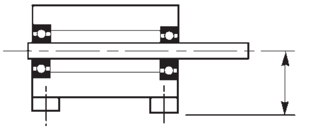


Fig.7 Distance of shaft axis to mounting datum

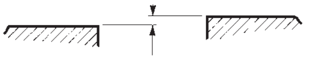


Fig.8 Planar alignment of mounting surfaces

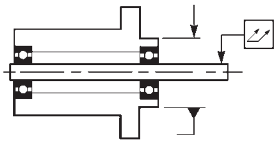


Fig.9 Shaft run-out

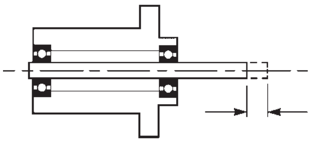


Fig.10 Axial Displacement

Predicting Misalignment

Prediction is aimed at verifying 'worst case' misalignment in order to ensure that it lies within the coupler's capacity.

Shaft misalignment has a parallel, an ***angular*** and a radial component. Each of these is three dimensional. For example. A shaft moved through the extremes of the angular error permitted by the tolerances describes a cone, Figs.11 & 12. While the shaft axes can converge and intersect on the critical plane, Fig.11, the probability is that they will not, thus giving rise to a ***radial*** error which is at a maximum when the axes are tangentially opposed on sphere diameter **r** Fig.12. Since diameter **r** increases with distance **L** and angle α it is necessary to establish these values before the maximum radial error can be calculated. The critical plane lies mid-way between the shaft terminations.

A shaft moved through the extremes of the horizontal and vertical ***parallel*** errors permitted by the tolerances, describes a rectangular figure of section p_1 x p_2 Fig.13. The maximum parallel error occurs when the axes are diagonally opposed on resultant p_3.

'Worst case' ***angular*** error is equivalent to $\alpha_1 + \alpha_2$ Fig.11. 'Worst case' ***radial*** error r_2 Fig.14, is the sum of parallel error resultant p_3 and radial error **r**.

Figs.11 – 14 assume that the errors in the horizontal and vertical planes are the same and apply equally to the LH and RH shafts.

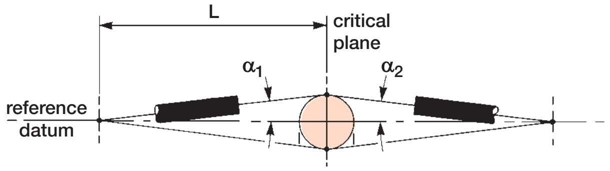


Fig.11 Worst case angular misalignment = $\alpha_1 + \alpha_2$

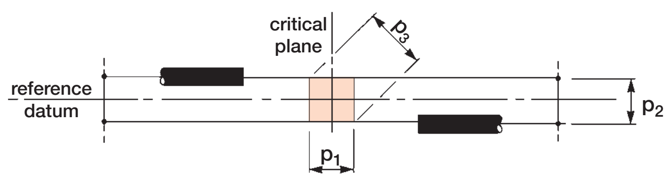


Fig.13 Maximum parallel error = p_3

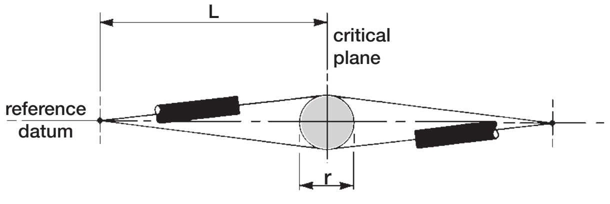


Fig.12 Maximum radial error = r

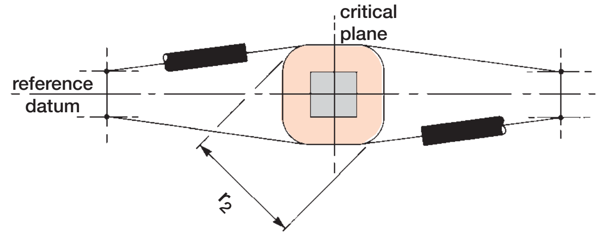


Fig.14 Worst case radial error $r_2 = p_3 + r$

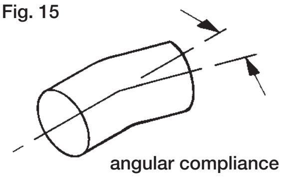
Summary of Misalignment

1. Angular and radial misalignment should be predicted for worst case tolerance accumulations.
2. A coupler registers misalignment mid-way between the shaft terminations. All predictions for misalignment must be on this plane.
3. Apply any recommended correction procedure for simultaneous misalignment.
4. Alignment can deteriorate after a period of operation due to thermal imbalances, creep, wear, and settlement. Make the permissible alignment error known to personnel responsible for inspecting or replacing equipment.

Terminology

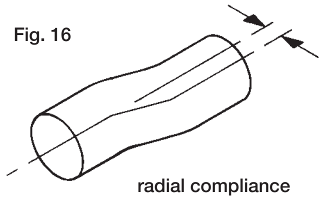
A flexible coupler is a mechanism that holds two hubs in substantially rigid, torsional relationship, yet permits them movement in angular, radial, and sometimes, axial planes. This movement is the coupler's reaction to misalignment (and axial shaft displacement) and is termed ***compliance.***

Fig. 15



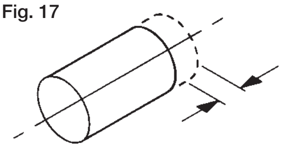
angular compliance

Fig. 16



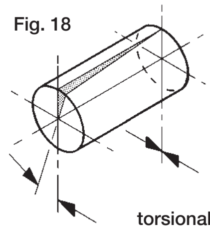
radial compliance

Fig. 17



axial compliance

Fig. 18



torsional compliance

A coupler should offer minimal resistance to compliance in the angular, radial and axial planes since the forces associated with compliance react as bending loads at the shafts and as radial and thrust loads at the bearings. Conversely, the coupler should resist torsional compliance as movement in this plane translates to 'wind-up' and determines positional accuracy in motion control applications.

Axial compliance absorbs longitudinal shaft movements that are due to working clearances, expansion/contraction, and settlement. These movements can damage fragile bearings fitted to sensitive instruments.

Alternatively, the coupler may be required to transmit, rather than absorb, shaft end movement when used in push/pull applications.

Criteria

Couplers are selected on a number of criteria that include:

Angular, radial, and axial compliance. Levels of compliance vary with coupler type, size and length. Available compliance should always exceed 'worst case' misalignment predictions.

Torque. Torque due to start-up, acceleration, deceleration, and braking are the critical values for sizing a coupler.

Cost, duty and life expectancy. These are the fundamental issues in determining the most suitable type and size of coupler for an application. The nature of the driver and the load is pertinent and should always be stated.

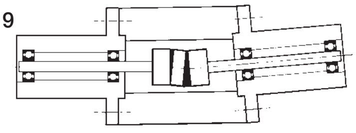
Torsional stiffness (the inverse of torsional compliance). This determines the positional accuracy that can be achieved. In closed-loop systems it influences the upper limits of stability and dynamic performance.

Space limitations. A coupler's torque capacity is related to its diameter and its misalignment capacity is related to its length (lateral displacement couplers excepted). It is good policy to keep the coupler's space allocation flexible until these parameters have been confirmed. The values often turn out greater than predicted.

Selecting for Nominal Misalignment

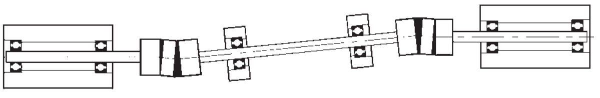
The unintentional misalignment between 2 fixed axis shafts is termed ***nominal*** and can have angular, radial and parallel components, but for practical purposes we need only concern ourselves with the angular and radial elements. Figs. 19 & 20 show the effect of these errors, exaggerated for clarity, on couplers connecting a pair of fixed axis shafts, i.e., shafts supported by 2 bearings.

Fig. 19



In blind assemblies, errors are difficult to verify or correct and the 'worst case' alignment errors should be predicted on the basis of the most adverse combination of tolerance build-up.

Fig.20



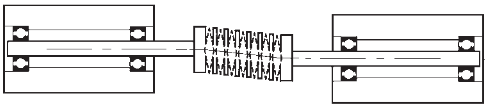
In accessible installations, excessive errors can be measured and corrected.

Misalignment has its origins in the small dimensional variations permitted by manufacturing tolerances. Their random accumulation controls the final position and attitude of the shaft in relation to its reference datum when a motor or gearbox, for example, is bolted into position. If the equipment is accessible as in Fig. 20, the alignment errors can be measured and if excessive, corrected by using shims, straight edges, etc.

If the shaft terminations are inaccessible as in Fig. 19, and the coupler is fitted blind and fastened through access holes in the housing, or if the equipment is on a production line, corrective measures are not usually possible. In these cases, it is necessary to predict the maximum alignment errors that could result from the most adverse combination of tolerance build-up.

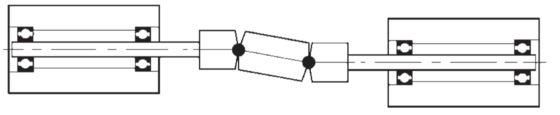
Flexible couplers handle the radial component of nominal misalignment in different ways. The transmission path can be taken through 2 complementary curves as in a flexible shaft, or routed through 2 shallow angles as in a Cardan (double universal joint, 2-stage disc coupler, etc.), or transferred radially through lateral displacement of a disc as in the Oldham. The angular component of nominal misalignment is usually evidenced as unevenly distributed compliance in the coupler, Figs.19 & 20. The enabling mechanisms for misalignment compensation work through sliding contact or material flexure.

Fig. 21



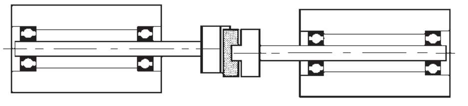
Multi-stage bellows conduct the transmission through two complementary curves.

Fig. 22



Cardanic couplers use pivotal action with defined fulcrums.

Fig. 23



Lateral displacement couplers connect the shafts through a sliding member.

Couplers Based on the Flexible Shaft

Couplers based on the flexible shaft handle radial and angular misalignment through flexure of a variable number of compliant elements. The greater the number of elements, the greater the angular and radial misalignment capacity and the lower the torsional stiffness. A characteristic of these couplers which include the multi-stage bellows, the helical beam and the radial slit concepts, is that the forces needed to effect compliance are broadly proportional to the torsional stiffness. The stiffer the coupler in torsion, the higher the resultant bearing loads.

BELLOWS are manufactured by rolling or hydroforming thin walled tubes fabricated from stainless steel strip. The characteristics of the coupler can be modified by varying the number and/or the wall thickness of the convolutions of the bellows.

Distinctive Property. High torsional stiffness.

Application. Any drive system where motion integrity of a high order is essential. Typically associated with encoder drives in closed-loop servo systems.

Options. Modified spring rates. Manufacture by electro-deposition of nickel. Keywayed and 'D' bores.

Couplers Based on the Cardan

The Cardan handles radial and angular misalignment through a two-stage pivotal action with defined fulcrums. The greater the distance between the pivotal planes, the greater the radial misalignment capacity. Torsional stiffness reduces marginally depending on the length and stiffness of the intermediate member. Angular misalignment capacity cannot be increased beyond the coupler's basic capability, irrespective of the distance between pivotal planes.

UNIVERSAL JOINTS have a mechanical pivotal action controlled by radial bearings. In the Huco-Pol, these are injection molded in acetal and feature controlled pre-load to eliminate backlash.

Distinctive Properties. Large offset capacity. Torsionally damping. Water resistant. Lubrication-free.
Application. General-purpose, typically light duty drives in food, textile, paper handling, packaging, etc., environments.
Options. Gears, pulleys, etc., molded integrally. Square, keywayed and hexagonal bores. 'D' bores fitted with spring clips. All-plastics versions. Bearings without pre-load.

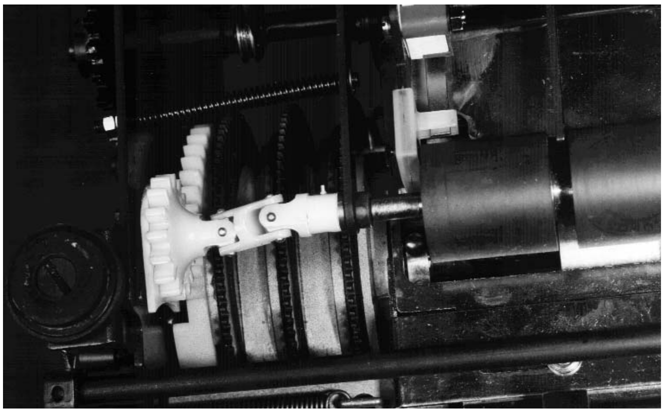


Fig. 24 A special purpose Huco-Pol double universal joint and drive gear supplied as a matched pair for a postal franking machine application. The generous offset capacity of the joint allows envelopes of varying thickness to be passed between the rubber impression rollers which are mounted on sprung radius arms.

DISC COUPLERS employ thin metallic membranes pressed from spring quality stainless steel as the pivotal media. They are attached alternately to the drive and driven members and flex in response to misalignment. Torque is resolved to simple tensile stresses in the opposing segments of the membranes which are free of residual stresses as no secondary forming operations are involved in their manufacture.

Distinctive Properties. Near-infinite life, dynamically balanced construction.
Application. Any drive system where motion integrity of a high order is essential, or where high rotational speeds are required. Typically associated with closed-loop servo systems in machine tools, robots, scanners, etc. Centrifuges, turbines, dynamometers, etc.
Options. Modified spring rates. Longer/shorter intermediate member. Keywayed or 'D' bores.

IN THE TWO-STAGE BELLOWS, a plain cylindrical section replaces all but two of the convolutions of the multi-stage version. The objective is to increase torsional stiffness by omitting redundant convolutions and this is achieved at the expense of misalignment capacity. They are intended for high precision, high resolution applications.

Distinctive Property. Very high torsional stiffness.
Application. Main axis drives in closed-loop velocity and position control systems. Encoder, resolver, tachogenerator, etc., drives.
Options. Keywayed or 'D' bores.

Couplers Based on Lateral Displacement

Lateral displacement couplers accommodate radial errors through an intermediate member that slides in a plane perpendicular to the axis of rotation. As the coupler rotates, the member aligns first with one hub, then with the other, straddling the misalignment with an orbital motion. The mechanisms are based on mechanical sliding contact and can accommodate relatively large radial errors without penalising overall length. Radial misalignment capacity is related to coupler diameter.

THE UNI-LAT combines the sliding mechanism of the Oldham with the pivotal action of the U/J to accommodate angular and radial misalignment. Radial pins machined integral with the hubs are slideably and pivotally engaged by a pair of injection molded annular rings that feature controlled pre-load to eliminate backlash.

Distinctive Properties. Generous angular and radial misalignment capacity within a short envelope. Electrically isolating.

Application. General-purpose, light duty stepper (including 1/2 & full step), encoder, resolver, tachogenerator drives.
Light push/pull duties.

Options. 'D' bores or other features machined into hubs. Free-running bearings without pre-load.

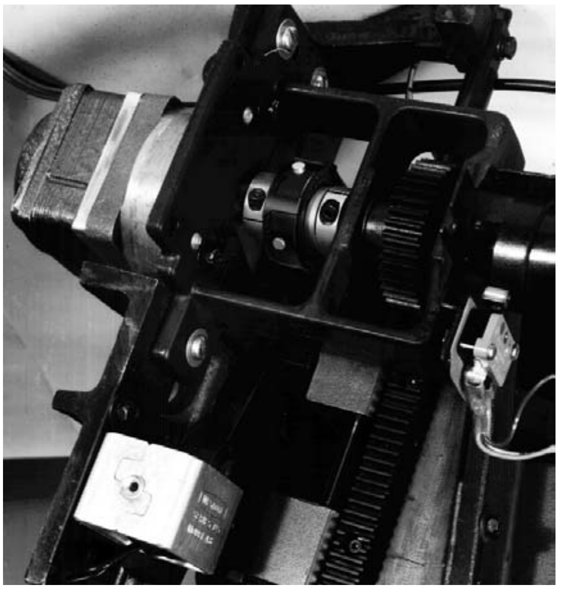


Fig. 25 In this application a Uni-Lat coupler provides the dynamic link between an electric motor and magnification adjustment mechanism in the lens optic sub-system of a Xerox duplicating system.

THE OLDHAM is a 3-part coupler that transmits rotation through a central disc that slideably engages the hubs under controlled pre-load to eliminate backlash. A number of engineering plastics are available to meet a variety of applications.

Distinctive Properties. Unusually large misalignment capability within a short envelope. Drive can be disconnected without removing hubs. Replaceable wear element restores coupler to original specification. Electrically isolating.

Application. General-purpose ranging from incremental control of fluid valves to positional systems in machine tools, robots, slide tables, etc. More suited to microstepper and closed-loop servo systems than 1/2 and full step motor drives.

Options. Keyway or 'D' bores in thru' bored types. Radiation resistant and heat resistant torque discs.
Free-running discs (no pre-load)

Fig. 26

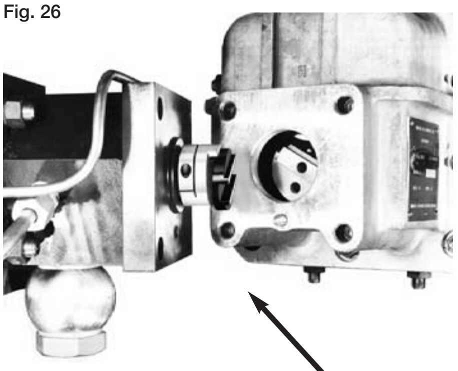
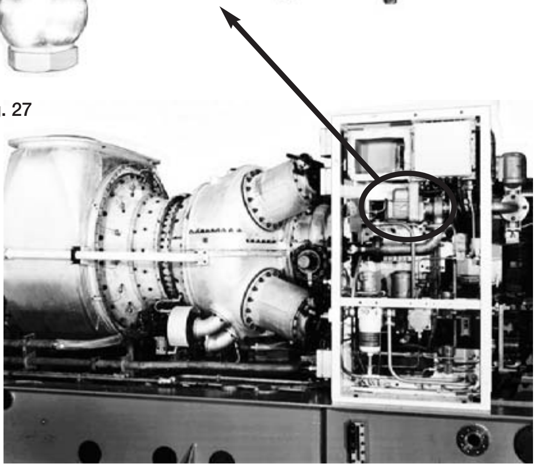


Fig. 27

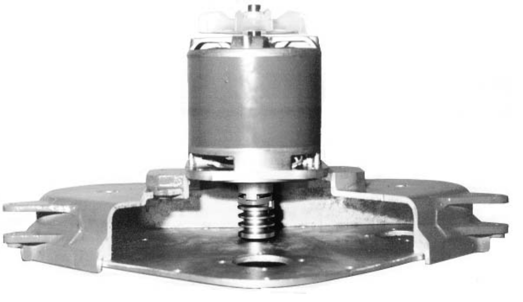


This gas turbine installation provides for single or dual fuel operation. A Huco Oldham coupler is used to connect both the liquid and gas valves to the actuator.

In this application the drive key at one end is machined integral with the customer's piece part. The coupler is designed to provide the positive drive necessary to ensure accurate fuel supply to the turbine under all conditions of load.

Fig. 28

Fig. 29



In this heavy duty mains operated scrubber/drier, one half of an Oldham coupler is mounted on the motor and the other half is located on ball races mounted on a pillar extending from the floor of the casing. This half incorporates 3 vee-belt grooves to drive the 3 scrubbing brushes.

The Oldham was selected because of its superior lateral misalignment capability and its capacity for blind assembly which allows motor replacement without accessing the casing cavity.

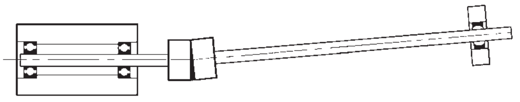
Selecting for True Angular Misalignment

True angular misalignment, known as symmetrical, occurs:

1. When one of the connected shafts is compliantly mounted, for example, when it is located by a self-aligning bearing, Fig. 30.
2. When an unsupported intermediate shaft is placed between the driver and the load, Fig. 31.

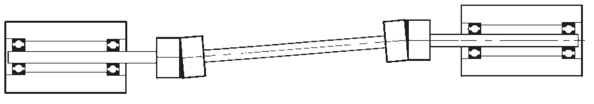
Because these shafts are not rigidly supported by two bearings in the conventional way, they self-align to intersect at the centre of the couplers which function as 'hinges', and in a limited way, as radial bearings. The couplers provide the means of locating these shafts on stable axes of rotation and should therefore be of the ***single-stage*** type. Any radial compliance in the couplers is counter productive. A partially supported shaft as in Fig. 30 is termed ***semi-floating*** and an unsupported shaft, Fig 31, is known as ***fully-floating***.

Fig. 30



A semi-floating shaft located by a self-aligning bearing at one end should be supported with a single-stage coupler at the other.

Fig. 31

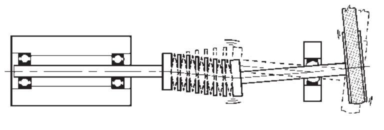


Two single-stage couplers locate a fully-floating shaft on a stable axis of rotation.

Floating shafts align themselves automatically resolving radial errors to true (symmetrical) angular misalignment. Floating shafts should be connected with single-stage Cardanic couplers to provide them with radial support.

Although couplers based on the flexible shaft are sometimes used in these conditions, there is the danger that the system may go into lateral oscillation. The hazard can be demonstrated by visualising the effect of a belt and pulley drive mounted on the compliant shaft. Having a lateral compliance capacity, the coupler would respond to fluctuating tension in the belt by allowing lateral oscillations in the shaft.

Fig. 32



A semi-floating shaft located by a self-aligning bearing at one end and a multi-stage bellows at the other. The coupler reacts to fluctuating tension in the belt by allowing lateral oscillations in the shaft.

Couplers suitable for symmetrical angular misalignment are the single universal joint, the single-stage disc coupler and the single-stage bellows.

SINGLE UNIVERSAL JOINT.
Distinctive Properties. Large offset capacity. Torsionally damping. Water resistant. Lubrication-free.
Options. Gears, pulleys, etc., molded integrally. Square, keywayed and hexagonal bores. 'D' bores fitted with spring clips. All-plastics versions. Bearings without pre-load.

SINGLE-STAGE DISC COUPLER.
Distinctive Properties. Near-infinite life, dynamically balanced construction.
Options. Modified spring rates. Keywayed or 'D' bores.

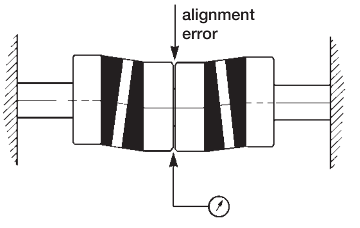
SHORT BELLOWS.
Typically 2 or 3 convolutions.
Distinctive Property. Very high torsional stiffness.
Options. Keywayed or 'D' bores.

In no circumstances should a lateral displacement coupler be used with floating shafts. These couplers have no self-centering action and would permit the floating shafts to orbit in an uncontrolled manner.

Selecting for Zero Misalignment

A zero misalignment condition can be attained by mounting both shafts in self-aligning bearings, Fig. 33. This allows both shafts to float into concentric relationship and permits the use of a solid coupler whose function is to support the shafts in perfect alignment.

Fig. 33



Shafts located by self-aligning bearings can float into perfect alignment for connection with a solid coupler.

Engineers will sometimes attempt to connect fixed axis shafts with solid couplers. There are very real difficulties in bringing, and maintaining, the shafts to the level of alignment necessary for this type of connection. Settlement, creep, thermal expansion and contraction are all contributory factors to relative movement between shafts and precise alignment achieved in factory conditions may not be repeatable by service engineers operating on site.

Solid connections have a better chance of success when both shafts are mounted in a monobloc housing whose bearing locations have been machined in one pass. 'Quill' couplers are used successfully when the prevailing conditions permit. These are purpose-designed for the application and though manufactured in one piece and substantially solid, their proportions permit sufficient compliance to avoid the excessive radial loads that cause shafts to shear and bearings to fail.

Preparatory to installing a solid connector, a revealing test is to couple first with a flexible device, then with the solid coupler. With the machine at normal operating temperature, measure the speed and/or the current taken by the motor in both cases. The difference will be the measure of the losses generated by additional friction at the bearings.

Selecting for Axial Compliance

Longitudinal shaft displacements can be classified as intentional or unintentional. The latter are caused by working clearances, settlement, transit, thermal expansion or contraction, etc. Though small, these movements can result in substantial thrust loads that can damage fragile bearings and in these cases, couplers with capacity for axial compliance should be selected.

Intentional shaft displacements are associated with push/pull applications or with extensible drives where the distance between actuator and load is variable. Teleshafts have a wide range of axial movement and should be considered for these purposes. For push/pull applications, select a coupler capable of resisting the corresponding forces. The relevant values are listed under 'End Loading' for mechanical couplers and under 'Axial Spring Rate' for flexural couplers.

THE HUCO-POL TELESHAFT employs precision drawn nesting tubes of square section brass which are cut to a length appropriate to the intentional shaft displacement.
Distinctive Properties. Can be customised to provide any practical length of stroke. Large offset capacity. Water resistant. Lubrication-free. Electrically isolating.
Options. Slide-locks to lock teleshaft in set position. Gears, pulleys, etc., molded integral with the universal joints. Square, keywayed and hexagonal bores. 'D' bores fitted with spring clips. Bearings without pre-load.

SINGLE UNIVERSAL JOINTS can be end-loaded for push/pull applications at moderate working angles in the range 0° – 10°.
Distinctive Properties. Slight damping characteristic. Electrically isolating.
Options. All-plastics versions. Bearings without pre-load. Special purpose push/pull joints, send for details.

THE UNI-LAT can be end-loaded for light push/pull applications.
Distinctive Properties. Approximately 0.1mm (0.004") axial compliance is evidenced under max permissible end-loads. Electrically isolating.

THE OLDHAM is a de-mountable 3-part coupler that is held in engagement by the shafts. A small amount of axial compliance can be built-in by mounting the hubs slightly out of full engagement.
Distinctive Properties. Resists compression but disengages under tension. Electrically isolating.
Options. Torque disc can be secured to one hub.

All versions of the **DISC** and **BELLOWS** coupler are designed to accommodate unintentional shaft displacements. Axial compliance is highest with the multi-stage bellows and lowest with the single-stage versions of the disc and bellows couplers.
Options. Modified axial spring rates.

Selecting for Torque

Torque is the angular force needed to overcome the resistance of a load.

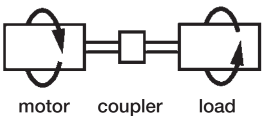
Rotating loads have both a frictional and an inertial component and are classified according to whichever predominates. For example, the resistance of a pump delivering fluid would be classified as a frictional load and the inertial component would be secondary if the pump were accelerated relatively slowly, or if it were started up just once per day. The total ***application torque*** would comprise the frictional + inertial elements. This type of application normally runs at constant speed, producing a ***uniform load*** and the power requirement for the pump would be given in H.P. or kW. These are related to torque by the following formulæ:

$$\text{torque lb.in} = \frac{\text{H.P. x 5250 x 12}}{\text{rpm}} \quad \text{or} \quad \text{torque Nm} = \frac{\text{kW x 9550}}{\text{rpm}}$$

A ball mounted slide table application, on the other hand, would be typified by short cycles consisting of rapid acceleration/deceleration in one direction followed by a similar sequence in the opposite direction of rotation. These loads would be predominantly inertial and being reversing loads would be regarded as non-uniform when assessing the coupler torque rating.

More specifically, the max torque 'seen' by the coupler may depend on whether braking is applied to the load or by the motor. The arrows show the direction of the angular forces due to acceleration, deceleration or braking.

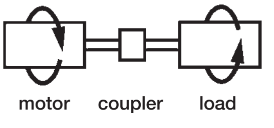
Fig. 34



Mode: Motor accelerates load.

Torque 'seen' by coupler = load inertia + frictional resistance of load.

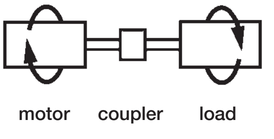
Fig. 35



Mode: Supply to motor discontinued, braking applied to load.

Torque 'seen' by coupler = motor inertia less motor drag.

Fig. 36



Mode: Motor decelerates load.

Torque 'seen' by coupler (in opposite direction) = load inertia less frictional resistance of load. The coupler 'sees' this as a torque reversal although the direction of rotation is unchanged.

These loads determine the maximum torque in the system which need to be assessed in relation to the Peak Torque Rating of the coupler. Select a size where:

Peak Torque ≥ *Application Torque* x *Service Factor*

Note: The service factor for a non-uniform load is 2.
A lower or higher service factor can be incorporated depending on service life required.

Peak Torque Ratings of Huco couplers relate to the static ***reversing torque load*** sustained for a minimum of 1 million cycles under test conditions (zero misalignment).

Selecting for Torsional Stiffness

Torsional stiffness is significant in positional systems. It describes the coupler's resistance to torsional compliance, also known as 'wind-up' or torsional deflection. It is expressed in several different units but is easiest to work with when denominated in lb.in/rad, or Nm/rad if working in metric units. Torsional stiffness is defined as ***torque per unit deflection***.

Torsional deflection is the inverse and is defined as ***deflection per unit torque.*** This too is expressed in many different units but is easier to demonstrate when denominated in degrees/lb.in or degrees/Nm. Use the tables at the end of this publication to convert from other units.

A coupler with a torsional stiffness of ***100 lb.in/rad*** has a torsional deflection rate of:

Fig. 37

$$\frac{1}{100\ \text{lb.in/rad}} \times \frac{180^\circ}{\pi} = \frac{0.573^\circ}{\text{lb.in}}$$

Two such couplers connected together would have a combined deflection rate of:

Fig. 38

$$\frac{0.573^\circ}{\text{lb.in}} + \frac{0.573^\circ}{\text{lb.in}} = \frac{1.146^\circ}{\text{lb.in}}$$

and a combined stiffness of:

Fig. 39

$$\frac{1}{1.146^\circ/\text{lb.in}} \times \frac{180^\circ}{\pi} = \frac{50\ \text{lb.in}}{\text{rad}}$$

If such a coupler were placed between a rotary actuator and a load torque of 1 lb.in, it would absorb 0.573° of angular movement before transmitting rotation to the load. To reverse the load, the actuator would travel through 2 x 0.573° of lost motion while the coupler first 'un-wound', then 'wound-up' in the opposite direction. The positional accuracy of such a system could be no better than ±0.573°. This equates with a resolution of:

Fig. 40

$$\frac{360^\circ}{2 \times 0.573^\circ/\text{lb.in}} = 314 \text{ parts per revolution when the load torque is 1 lb.in}$$

If the torsional deflection rate were 0.40°/lb.in and the load torque 2 lb.in, the coupler would resolve:

Fig. 41

$$\frac{360^\circ}{2 \times 0.40^\circ/\text{lb.in} \times 2\ \text{lb.in}} = 225 \text{ parts per revolution}$$

To find the torsional deflection rate (TDR) needed to resolve 1,000 parts per revolution when the load torque is 4 lb.in:

Fig. 42

$$\text{TDR} = \frac{360^\circ}{2 \times 1000 \times 4\ \text{lb.in}} = \frac{180^\circ}{1000 \times 4\ \text{lb.in}} = 0.045^\circ/\text{lb.in}$$

and to resolve 500 parts per revolution when the load torque is 0.5 lb.in:

$$\text{TDR} = \frac{180^\circ}{500 \times 0.5\ \text{lb.in}} = 0.360^\circ/\text{lb.in}$$

To find the ***torsional stiffness*** (TS) needed to resolve 1,000 parts per revolution when the load torque is 4 lb.in:

Fig. 43

$$\text{TS} = \frac{2 \times 1000 \times 180^\circ \times 4\ \text{lb.in}}{360^\circ \times \pi} = \frac{1000 \times 4\ \text{lb.in}}{\pi} = 1273\ \text{lb.in/rad}$$

and to resolve 500 parts per revolution when the load torque is 0.5 lb.in:

$$\text{TS} = \frac{500 \times 0.5\ \text{lb.in}}{\pi} = 79.6\ \text{lb.in/rad}$$

For metric calculations, substitute Nm/rad for lb.in/rad and Nm for lb.in.

In closed-loop position or velocity control systems, the coupler's torsional stiffness is more critical and is a contributory factor in determining the upper limit of dynamic performance and stability. The coupler's stiffness should be such that the torsional resonance frequency exceeds 300 – 600 Hz, depending on the dynamics. Stiffness is most critical when the load inertia is dominant and progressively less so as dominance swings in the motor's favour. The calculations for torsional stiffness and resonant frequency are, respectively:

Imperial Units

Fig. 44

$$C_T \geq \frac{(F_R \times 2\pi)^2}{\left(\frac{1}{J_M} + \frac{1}{J_L}\right)} \times 386$$

Fig. 45

$$F_R = \frac{1}{2\pi} \times \sqrt{\left(\frac{1}{J_M} + \frac{1}{J_L}\right) \times C_T \times 386}$$

where: C_T = torsional stiffness (lb.in/rad) J_M = motor inertia (lb.in^2)
F_R = resonant frequency (Hz) J_L = load inertia (lb.in^2)

Metric Units

Fig. 46

$$C_T \geq \frac{(F_R \times 2\pi)^2}{\left(\frac{1}{J_M} + \frac{1}{J_L}\right)}$$

Fig. 47

$$F_R = \frac{1}{2\pi} \times \sqrt{\left(\frac{1}{J_M} + \frac{1}{J_L}\right) \times C_T}$$

where: C_T = torsional stiffness (Nm/rad) J_M = motor inertia (kgm^2)
F_R = resonant frequency (Hz) J_L = load inertia (kgm^2)

Engineers express torsional stiffness in a variety of units. To help comparison, torsional stiffness and torsional deflection (the inverse) are cross referenced in the tables at the end of this publication.

Selecting for Cost, Duty & Life Expectancy

Huco couplers fall into 2 broad classifications, the mechanical types that work through sliding contact and the flexural types that work through material flexure. In broad terms, selection is influenced by cost, duty and life expectancy and the pertinent issues are summarised below.

Oldham or Uni-Lat couplers should be considered:

1. When cost is the paramount consideration.
2. When the backlash-free life requirement is within the coupler's backlash-free life expectancy.
3. When backlash can be tolerated.
4. When the coupler is required to transmit only incremental or periodic rotation.
5. When the duty is 50% or less, i.e., the coupler is stationary for half of the time or more.
6. When radial misalignment is severe and the available space envelope is short.
7. When radial misalignment is difficult to predict or maintain.
8. When slight torsional damping is beneficial.
9. When a 3-piece coupler is advantageous. With the Oldham the drive can be connected/disconnected with the hubs in place. The wear element is renewable.
10. When electrical isolation of shafts is required.
11. When the coupler is required to transmit longitudinal motion (push/pull).

Uni-Lats have a more pronounced damping characteristic, lower torque capacity, and generally run more quietly than the Oldhams. They have a greater angular misalignment capacity than the Oldhams though this facility is only practicable at relatively low speeds. Uni-Lats are thru' bored.

Oldhams are more robust, can be separated, and feature a replaceable wear element which is also available in both heat and radiation resistant plastics. The standard series hubs are blind bored to a controlled depth, the X–Y series are thru' bored and have 2 to 3 times the backlash-free life. Torque discs are solid but can be supplied bored to permit shafts to pass through though this reduces the overall torsional stiffness of the coupler.

Uni-Lats and Oldhams are general purpose couplers suitable for position control. Uni-Lats are better for full and half step motor drives, Oldhams are better for micro-steppers and closed-loop systems.

Huco-Flex disc or bellows couplers should be considered:

1. When torsional stiffness is a critical parameter.
2. When the backlash-free life requirement is beyond the capacity of the Oldham or Uni-Lat.
3. When speeds are typically in the range 3,000 – 5,000 rpm.
4. When rotation is continuous or the duty cycle exceeds 50%.
5. When a coupler with axial compliance is needed to protect fragile bearings from thrust loads.
6. When there is little risk of the alignment errors exceeding prescribed limits during initial installation or on subsequent replacement of motor, encoder, etc.
7. When the environmental conditions favour an all-metal coupler.

Size for size, Huco-Flex bellows couplers have the highest torsional stiffness ratio and provide a high level of translational accuracy. In general, their life expectancy is not as high as Huco-Flex disc couplers and are therefore more suitable for intermittent applications.

Huco-Flex disc couplers are more reliable and have near-infinite life when used within their torque and misalignment capacity. They provide a high level of translational accuracy and their spring rates can be modified by varying the number and thickness of the stainless steel membranes.

Apply the following tables to equate operating hours with total revolutions at varying speeds of rotation.

operating time per number of revolutions				
r.p.m.	10^6	10^7	10^8	10^9
60	278 hrs	2,778 hrs	27,778 hrs	277,778 hrs
100	167	1,667	16,667	166,667
250	67	667	6,667	66,667
500	33	333	3,333	33,333
1000	17	167	1667	16,667
1500	11	111	1,111	11,111
2000	8.3	83	833	8,333
3000	5.6	56	556	5,556
4000	4.2	42	417	4,167
5000	3.3	33	333	3,333
7500	2.2	22	222	2,222
10000	1.7	17	167	1,667

revolutions per number of operating hours				
r.p.m.	1000 hrs.	2000 hrs.	4000 hrs.	6000 hrs.
60	3.6×10^6	7.2×10^6	1.4×10^7	2.2×10^7
100	6×10^6	1.2×10^7	2.4×10^7	3.6×10^7
250	1.5×10^7	3×10^7	6×10^7	9×10^7
500	3×10^7	6×10^7	1.2×10^8	1.8×10^8
1000	6×10^7	1.2×10^8	2.4×10^8	3.6×10^8
1500	9×10^7	1.8×10^8	3.6×10^8	5.4×10^8
2000	1.2×10^8	2.4×10^8	4.8×10^8	7.2×10^8
3000	1.8×10^8	3.6×10^8	7.2×10^8	1.1×10^9
4000	2.4×10^8	4.8×10^8	9.6×10^8	1.4×10^9
5000	3×10^8	6×10^8	1.2×10^9	1.8×10^9
7500	4.5×10^8	9×10^8	1.8×10^9	2.7×10^9
10000	6×10^8	1.2×10^9	2.4×10^9	3.6×10^9

Failure Modes

Couplers based on flexural systems fail abruptly with little or no warning, discontinuing the drive. The cause is usually metal fatigue due to sustained flexure above the coupler's torque and compliance capacity.

Failure in mechanical couplers is subjective. In zero backlash applications, the coupler is deemed to have failed when backlash is first evidenced. In other applications, the threshold may be 2° of backlash. Useful life therefore varies with individual applications.

APPLICATION PROFILE

Name ____________________ Telephone ____________________

Company ____________________ Fax ____________________

Address ____

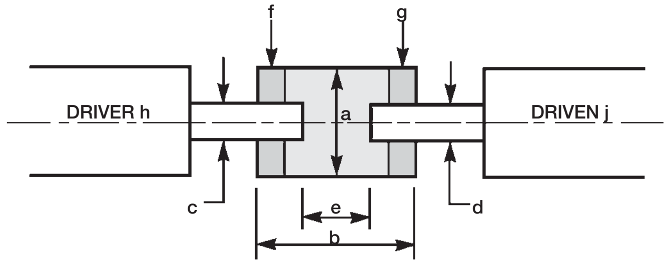
Street / P.O. Box City State / Zip

Please complete this questionnaire on a photocopy & add any additional comments on reverse side.

IMPORTANT
Enter values and mark boxes as appropriate in sections 1 – 10
Ring preferred options at f and g
Ring appropriate denomination (in, mm, etc.). If your values are expressed in other units, please state.

Sections 1 – 6 information is essential
Sections 7 – 9 information is needed if critical to the application
Section 10 – production quantities may offer scope for cost savings

1. Layout and connection



a. Outside diameter
Preferred .... in / mm
Maximum .... in / mm

b. Overall length
Preferred .... in / mm
Maximum .... in / mm

c. Shaft dia. .... in / mm
d. Shaft dia .... in / mm
e. Min gap .... in / mm
Max gap .... in / mm

f. & g. Attachment
set screw at f, at g
clamp at f, at g
other ....

f. & g. Shafts
plain shafts at f, at g
flatted 'D' shaft at f, at g
double flatted shaft at f, at g
keyway at f, at g
details ....
....

2. Driver

- [ ] Stepper motor
  steps per turn ....
- [ ] Manual Operation
- [ ] Rotary solenoid
- [ ] Servo motor

3. Driven

- [ ] Leadscrew / ball screw
- [ ] Pump, type ....

- [ ] Pulse generator
  counts per turn ....
- [ ] Potentiometer
- [ ] Electrical switch
- [ ] Fluid Valve
- [ ] Cam Shaft
- [ ] Gearbox

IF OTHER THAN LISTED, PLEASE DESCRIBE ON REVERSE SIDE

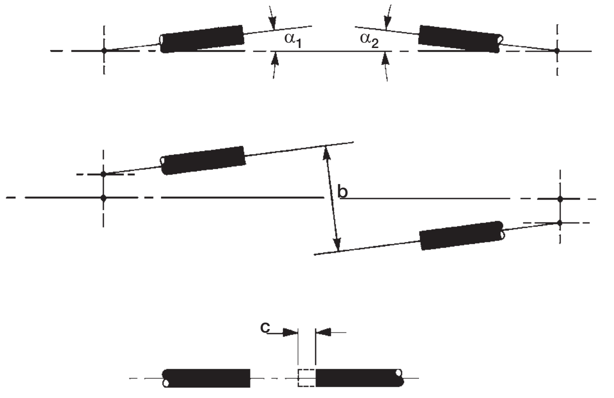
4. Torque

a. Max torque .... lb.in or Nm or other ....
(due to start-up, acceleration, deceleration or braking)

c. Working torque .... lb.in or Nm or other ....

5. Misalignment & axial displacement

a. 'Worst case' angular .... degrees
b. 'Worst case' radial .... in / mm
c. Total axial .... in / mm



Worst case angular= sum of $\alpha_1 + \alpha_2$ (generated by converging shafts)
Worst case radial = max parallel error + max radial error generated by diverging shafts. Errors are measured mid-way between shaft terminations.

6. Duty

a. Incremental / uni-directional / bi-directional. Intermittent / continuous. (Incremental means an operating cycle of less than 1 turn)
b. Max speed .... rpm
c. Revs per cycle .... cycles per day ....
d. Time between regular maintenance intervals ....
e. Total life required if no regular maintenance .... turns/time
f. Can backlash be tolerated ....

7. Torsional Stiffness

min torsional spring rate .... lb.in/rad or Nm/rad or other ....

8. Moment of inertia

Max .... lb.in² or kgm² (mr²) or other ....

9. Environment

a. Temperature min........... max .... °C °F
b. Aggressive liquids / atmospheres or radiation.
....
....
....

10. Predicted annual qty ....
or Predicted order qty ....

Coding of Bore Diameters

For ease of use and avoidance of errors, bore diameters commonly specified with Huco products are coded. These codes are used in all Company sales documentation.

Table 1 lists the *ROUND* bores compatible with Huco products. To specify, state the bore code that corresponds to your shaft diameter.

Bore diameters of 3/8" (or 10mm) and upwards can also be supplied with keyways and are specified by adding the 'P' or 'R' prefix to the appropriate round bore code.

Standard keyways are machined to 2 specifications:

Bore codes prefixed 'P' denote a metric keyway conforming to ISO 773/774 (BS 4235 Pt. 1).

Bore codes prefixed 'R' denote an English keyway conforming to BS 46 Pt. 1.

The key sizes for both standards are detailed in Table 2 together with the key seat depth measured across the bore diameter.

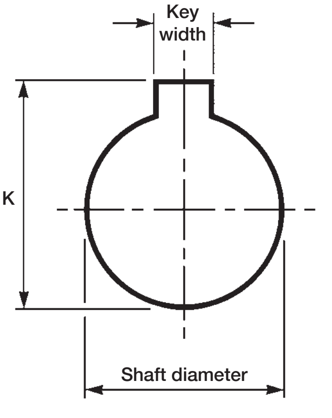
In most cases, keyways prefixed 'R' are compatible with AGMA 9002–A86 but can differ in the depth of the key seat. Shafts fitted with AGMA keys should be measured to determine dimension K and the key width. If these do not conform to the values shown in Table 2, please photocopy this page and enter the required dimensions on the drawing below. Please enter all three dimensions, key width, shaft diameter and dimension K.

We are also tooled to broach 'D' bores in certain components. 'D' bores broached into clamp hubs offer positive, backlash-free fastening of very high integrity.

Table 1 – Round Bore Codes

inch decml	inch fract	metric mm	bore code
0.0787	–	2	11
0.0900	–	2.286	12
0.0938	3/32	2.382	13
0.1181	–	3	14
0.1200	–	3.048	15
0.1250	1/8	3.175	16
*0.1563	5/32	3.969	17
0.1575	–	4	18
0.1875	3/16	4.763	19
0.1969	–	5	20
0.2188	7/32	5.556	21
0.2362	–	6	22
0.2400	–	6.096	23
0.2500	1/4	6.350	24
0.2756	–	7	25
0.2813	9/32	7.144	26
0.3125	5/16	7.938	27
0.3150	–	8	28
0.3438	11/32	8.731	29
0.3543	–	9	30
0.3750	3/8	9.525	31
0.3937	–	10	32
0.4331	–	11	33
0.4375	7/16	11.113	34
0.4724	–	12	35
0.5000	1/2	12.700	36
0.5118	–	13	37
0.5512	–	14	38
0.5625	9/16	14.288	39
0.5906	–	15	40
0.6250	5/8	15.875	41
0.6299	–	16	42
0.6693	–	17	43
0.6875	11/16	17.463	44
0.7087	–	18	45
0.7480	–	19	46
0.7500	3/4	19.050	47
0.7874	–	20	48
0.8661	–	22	49
0.8750	7/8	22.225	50
0.9449	–	24	51
0.9843	–	25	52
1.0000	1	25.400	53
1.1024	–	28	54
1.1250	1-1/8	28.575	55
1.1811	–	30	56

*Not manufactured. Nearest alternative 0.1575" (4mm).



Note that our tooling produces a key seat classified as 'nominal', being a nominal clearance on standard keys.

Table 2 – Keywayed Bore Details & Codes

round bore code	key size width x height inch	key size width x height mm	K bore + keyway inch	K bore + keyway mm	keywayed bore code
31	1/8 x 1/8	–	0.4380	–	R31
32	–	3 x 3	–	11.40	P32
33	–	4 x 4	–	12.80	P33
34	1/8 x 1/8	–	0.5005	–	R34
35	–	4 x 4	–	13.80	P35
36	1/8 x 1/8	–	0.5630	–	R36
37	–	5 x 5	–	15.30	P37
38	–	5 x 5	–	16.30	P38
39	3/16 x 3/16	–	0.6535	–	R39
40	–	5 x 5	–	17.30	P40
41	3/16 x 3/16	–	0.7160	–	R41
42	–	5 x 5	–	18.30	P42
43	–	5 x 5	–	19.30	P43
44	3/16 x 3/16	–	0.7785	–	R44
45	–	6 x 6	–	20.80	P45
46	–	6 x 6	–	21.80	P46
47	3/16 x 3/16	–	0.8410	–	R47
48	–	6 x 6	–	22.80	P48
49	–	6 x 6	–	24.80	P49
50	1/4 x 1/4	–	0.9930	–	R50
51	–	8 x 7	–	27.30	P51
52	–	8 x 7	–	28.30	P52
53	1/4 x 1/4	–	1.1180	–	R53
54	–	8 x 7	–	31.30	P54
55	5/16 x 1/4	–	1.2400	–	R55
56	–	8 x 7	–	33.30	P56

PRODUCT CODING

Examples:

Consult factory for non-catalog variables

Specify variables within min/max availability

205.41.3136 U

Product code – 3 figure serial number

Product size – outside diameter in mm

Bore dia at B1 – derived from std bore code table (31 = 3/8" dia.)

Bore dia at B2 – derived from std bore code table (36 = 1/2" dia.)

*Special metric screws with UNC sockets

This example:
Size 41 Uni-Lat coupler with clamp fastening, bored 3/8" x 1/2" dia. This product is required with blue screws for UNC wrench compatibility.

*With certain products, the standard black screws accept UNC wrenches (as well as metric). With others, the metric screws must be specially formed to accept UNC wrenches. These are colored blue for identification and products fitted with blue screws are suffixed 'U'. A few products are supplied with screws that accept metric wrenches only. Refer to dimensional tables for screw and wrench data.

A master coding list defines the features pertinent to each 3 figure product code. Specials are classified by individual product codes specific to each product sub-group and identified by the product size, followed by 'X' and a serial number.

Specials are typically coded:

Unique drawing number. No variables permitted

520.41.X123

Product code – 3 figure serial number classifies product as a special coupler. Example based on X–Y series coupler. Product codes exist also for special hubs only and special torque discs only.

Product size – nominal outside dia in mm

Unique serial identification of special Oldham X–Y coupler

This example:
Size 41Oldham X-Y special coupler. Deviations from standard are recorded by description or unique drawing.

Other examples of coding

Consult factory for non-catalog variables

Specify variables within min/max availability

707 H 19.2224

Product code – 3 figure serial number

This position is used exceptionally as a discriminator to identify an alternate specification

Product size – outside diameter in mm

Bore dia at B1 – derived from std bore code table (22 = 6mm dia.)

Bore dia at B2 – derived from std bore code table (24 = 1/4" dia.)

This example:
Size 19 Multi-Beam 6-beam coupler with clamp fastening, bored 6mm x 1/4" dia.

Consult factory for non-catalog variables

Specify variables within min/max availability

234.25.31 U

Product code – 3 figure serial number

Product size – outside diameter in mm

Bore dia at B1 – derived from std bore code table (31 = 3/8" dia.)

Special metric screws with UNC sockets

This example:
Size 25 Oldham standard hub with clamp fastening, bored 3/8" dia. This product is required with blue screws for UNC wrench compatibility

CONVERSION FACTORS

Moment of Inertia

To convert from A to B, multiply A by entry in table, e.g. 1 kgm^2 = 3417 $lb.in^2$

B / A	kgm^2 (mr^2)	$kgcm^2$	gcm^2	$kgms^2$	$kgcms^2$	$gcms^2$	$oz.in^2$	$lb.in^2$	$lb.ft^2$ (WK^2)	$oz.in.s^2$	$lb.in.s^2$	$lb.ft.s^2$
kgm^2	1	10^4	10^7	1.020×10^{-1}	10.20	1.020×10^4	5.467×10^4	3.417×10^3	23.73	141.6	8.851	7.376×10^{-1}
$kgcm^2$	10^{-4}	1	10^3	1.020×10^{-5}	1.020×10^{-3}	1.020	5.467	3.417×10^{-1}	2.373×10^{-3}	1.416 10^{-2}	8.851 10^{-4}	7.376×10^{-5}
gcm^2	10^{-7}	10^{-3}	1	1.020×10^{-8}	1.020×10^{-6}	1.020×10^{-3}	5.467×10^{-3}	3.417×10^{-4}	2.373×10^{-6}	1.416×10^{-5}	8.851×10^{-7}	7.376×10^{-8}
$kgms^2$	9.807	9.807×10^4	9.807×10^7	1	10^2	10^5	5.362×10^5	3.351×10^4	232.7	1.389×10^3	86.80	7.233
$kgcms^2$	9.807×10^{-2}	980.7	9.807×10^5	10^{-2}	1	10^3	5.362×10^3	335.1	2.327	13.89	8.680×10^{-1}	7.233×10^{-2}
$gcms^2$	9.807×10^{-5}	9.807×10^{-1}	9.807×10^2	10^{-5}	10^{-3}	1	5.362	3.351×10^{-1}	2.327×10^{-3}	1.389×10^{-2}	8.680×10^{-4}	7.233×10^{-5}
$oz.in^2$	1.829×10^{-5}	1.829×10^{-1}	182.9	1.865×10^{-6}	1.865×10^{-4}	1.865×10^{-1}	1	6.250×10^{-2}	4.340×10^{-4}	2.590×10^{-3}	1.619×10^{-4}	1.349×10^{-5}
$lb.in^2$	2.926×10^{-4}	2.926	2.926×10^3	2.984×10^{-5}	2.984×10^{-3}	2.984	16	1	6.944×10^{-3}	4.144×10^{-2}	2.590×10^{-3}	2.158×10^{-4}
$lb.ft^2$	4.214×10^{-2}	421.4	4.214×10^5	4.297×10^{-3}	4.297×10^{-1}	429.7	2.300×10^3	144	1	5.968	3.730×10^{-1}	3.108×10^{-2}
$oz.in.s^2$	7.062×10^{-3}	70.62	7.062×10^4	7.201×10^{-4}	7.201×10^{-2}	72.01	386.1	24.13	1.676×10^{-1}	1	6.250×10^{-2}	5.208×10^{-3}
$lb.in.s^2$	1.130×10^{-1}	1.130×10^3	1.130×10^6	1.152×10^{-2}	1.152	1.152×10^3	6.177×10^3	386.1	2.681	16	1	8.333×10^{-2}
$lb.ft.s^2$	1.356	1.356×10^4	1.356×10^7	1.383×10^{-1}	13.83	1.383×10^4	7.413×10^4	4.633×10^3	32.17	192	12	1

1 kgm^2 (mr^2) = 4 $kgfm^2$ (GD^2)

1 $lb.ft.s^2$ = 1 slug

Torque

To convert from A to B, multiply A by entry in table, e.g. 1 Nm = 8.851 lb.in

B / A	Nm	Ncm	kgm	kgcm	gcm	oz.in	lb.in	lb.ft
Nm	1	10^2	1.020×10^{-1}	10.20	1.020×10^4	141.6	8.851	7.376×10^{-1}
Ncm	10^{-2}	1	1.020×10^{-3}	1.020×10^{-1}	102	1.416	8.851×10^{-2}	7.376×10^{-3}
kgm	9.807	980.7	1	10^2	10^5	1.389×10^3	86.80	7.233
kgcm	9.807×10^{-2}	9.807	10^{-2}	1	10^3	13.89×10^{-2}	8.680	7.233
gcm	9.807×10^{-5}	9.807×10^{-3}	10^{-5}	10^{-3}	1	1.389×10^{-2}	8.680×10^{-4}	7.233×10^{-5}
oz.in	7.062×10^{-3}	7.062×10^{-1}	7.201×10^{-4}	7.201×10^{-2}	72.01	1	6.250×10^{-2}	5.208×10^{-3}
lb.in	1.130×10^{-1}	11.30	1.152×10^{-2}	1.152	1.152×10^3	16	1	8.333×10^{-2}
lb.ft	1.356	135.6	1.383×10^{-1}	13.83	1.383×10^4	192	12	1

Torsional Stiffness

To convert from A to B, multiply A by entry in table, e.g.
1 oz.in / degree = 3.581 lb.in / rad

To convert from B to A, divide B by entry in table, e.g.
1 lb.in / rad = $\frac{1}{3.581}$ oz.in / degree

A \ B	lb.in / rad	Nm/rad	kgm/rad
oz.in / degree	3.581	0.4046	0.04126
oz.in / arcmin	214.9	24.28	2.475
oz.in / arcsec	12 892	1 457	148.5
lb.in / degree	57.30	6.474	0.6601
lb.in / arcmin	3 438	388.4	39.61
lb.in / arcsec	206 264	23 305	2 376
Ncm / degree	5.072	0.5730	0.0584
Ncm / arcmin	304.3	34.38	3.506
Ncm / arcsec	18 256	2 063	210.3
Nm / degree	507.1	57.30	5.843
Nm / arcmin	30 426	3 438	350.6
Nm / arcsec	1 825 590	206 264	21 033
kgcm / degree	49.73	5.619	0.5730
kgcm / arcmin	2 984	337.1	34.38
kgcm / arcsec	179 029	20 228	2 063
kgm / degree	4 973	561.9	57.30
kgm / arcmin	298 383	33 713	3 438
kgm / arcsec	17 902 940	2 022 759	206 264
lb.in / rad	1	0.1130	0.01152
Nm / rad	8.851	1	0.1020
kgm / rad	86.80	9.807	1

Torsional Stiffness / Torsional Deflection

To convert from A to B, divide A into C, e.g.
1 lb.in / rad = $\frac{3.581}{1}$ degrees / oz.in

To convert from B to A, divide B into C, e.g.
1 degree / oz.in = $\frac{3.581}{1}$ lb.in /rad

A	C	B
lb.in / rad	3.581	degrees / oz.in
lb.in / rad	57.30	degrees / lb.in
lb.in / rad	214.9	arcmins / oz.in
lb.in / rad	3 438	arcmins / lb.in
lb.in / rad	12 892	arcsecs / oz.in
lb.in / rad	206 264	arcsecs / lb.in
Nm / rad	0.5730	degrees / Ncm
Nm / rad	57.30	degrees / Nm
Nm / rad	34.38	arcmins / Ncm
Nm / rad	3 438	arcmins / Nm
Nm / rad	2 063	arcsecs / Ncm
Nm / rad	206 264	arcsecs / Nm
kgm / rad	0.5730	degrees / kgcm
kgm / rad	57.30	degrees / kgm
kgm / rad	34.38	arcmins / kgcm
kgm / rad	3 438	arcmins / kgm
kgm / rad	2 063	arcsecs / kgcm
kgm / rad	206 264	arcsecs / kgm

Powers of 10

10^0	=	1	10^0	=	1
10^{-1}	=	0.1	10^1	=	10
10^{-2}	=	0.01	10^2	=	100
10^{-3}	=	0.001	10^3	=	1 000
10^{-4}	=	0.0001	10^4	=	10 000
10^{-5}	=	0.00001	10^5	=	100 000
10^{-6}	=	0.000001	10^6	=	1 000 000
10^{-7}	=	0.0000001	10^7	=	10 000 000
10^{-8}	=	0.00000001	10^8	=	100 000 000
10^{-9}	=	0.000000001	10^9	=	1 000 000 000

Add the powers when multiplying, e.g.:

$(4 \times 10^2) \times (5 \times 10^3) = (4 \times 5) \times (10^{2+3})$
$= 20 \times 10^5$
$= 2\ 000\ 000$

Subtract the powers when dividing, e.g.:

$(4 \times 10^2) / (5 \times 10^3) = (4 / 5) \times (10^{2-3})$
$= 0.8 \times 10^{-1}$
$= 0.08$

Resolve powers before adding, e.g.:

$(4 \times 10^2) + (5 \times 10^3) = 400 + 5\ 000$
$= 5\ 400$

Resolve powers before subtracting, e.g.:

$(4 \infty 10^2) - (5 \infty 10^3) = 400 - 5\ 000$
$= -4\ 600$