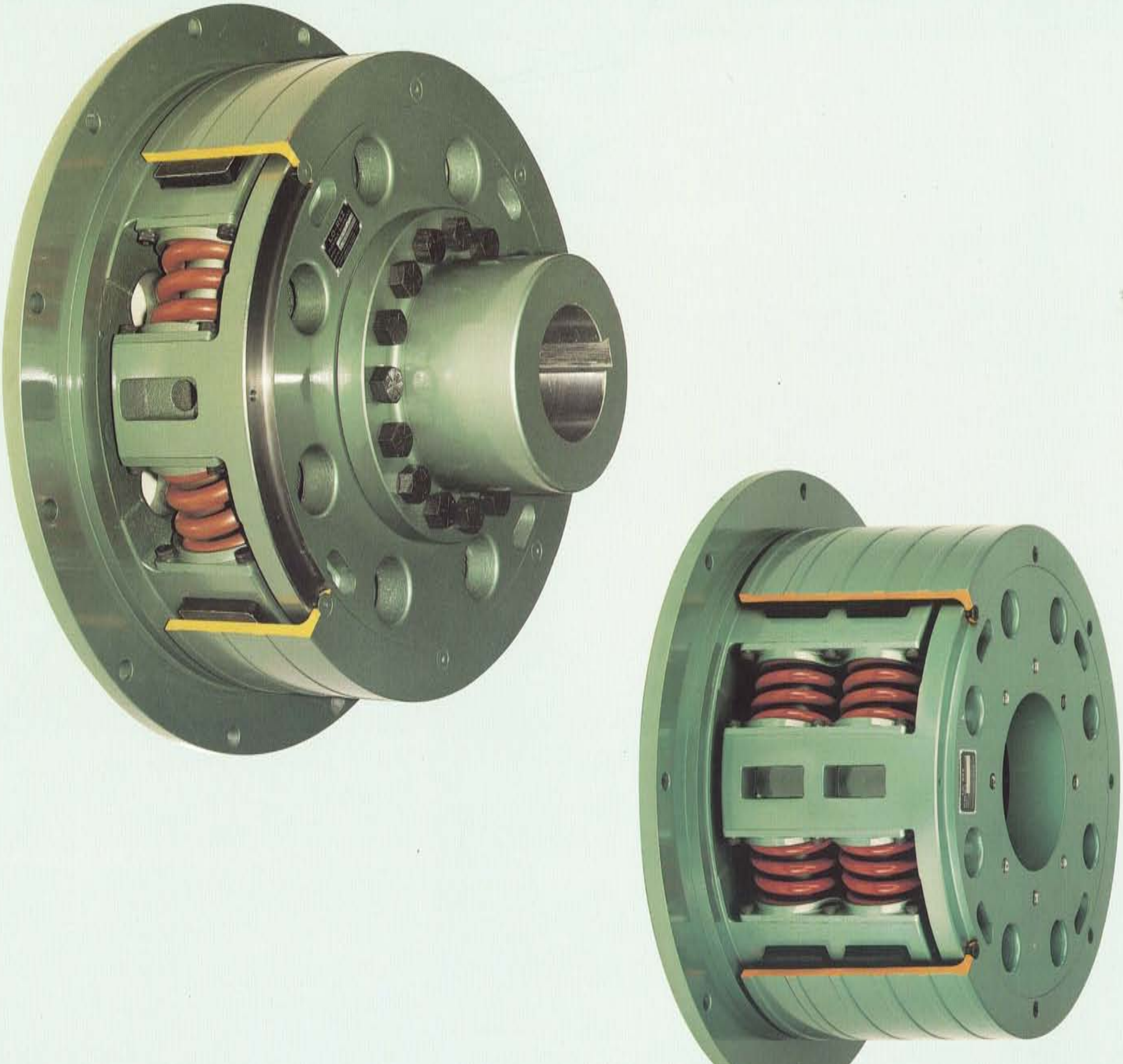
LO·REZ

STEEL-SPRING FLEXIBLE COUPLINGS



LO-REZ VIBRATION CONTROL LTD.

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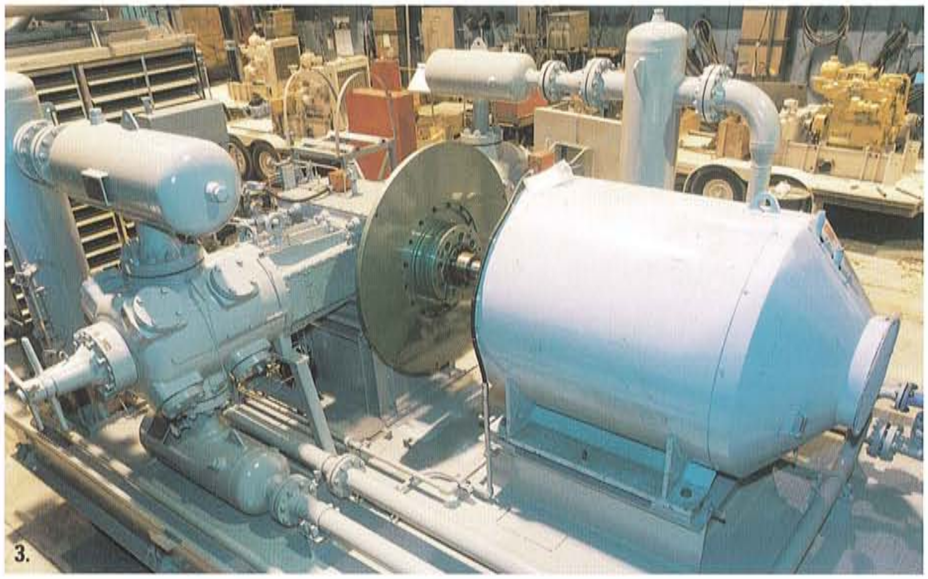
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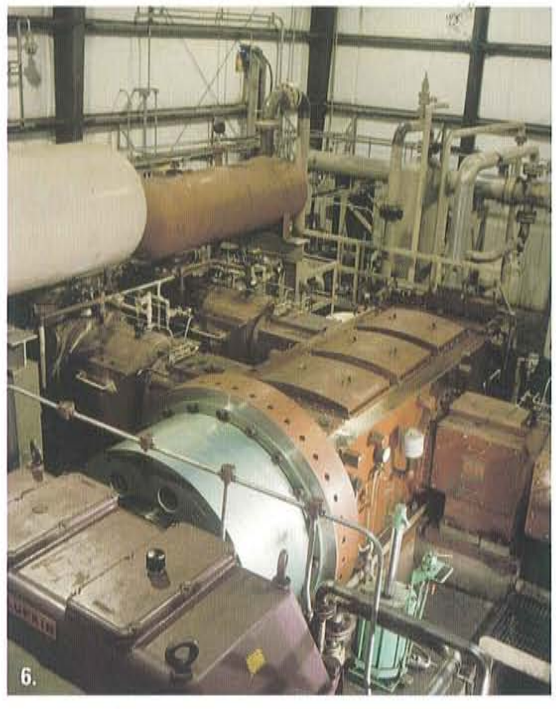
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1. **Lo-Rez F/SF Coupling in Harbour Tug**
2. **Lo-Rez I/SF Coupling in Fisheries Research Vessel**
3. **Lo-Rez E/HF Coupling on motor driven Reciprocating Compressor**
4. **Lo-Rez E/HH Coupling in Locomotive**
5. **Lo-Rez G/HF Coupling in Tug Boat**
6. **Lo-Rez L2/HF Coupling on 7000 HP Reciprocating Compressor**
7. **Lo-Rez DE/SF Coupling in Oceanographic Vessel**

Contents

When Lo-Rez developed its steel-spring coupling concept in the late 40's, torsional vibration was a relatively young science - at least in North America. Having honed his teeth in torsionals and related subjects with a builder of large Diesel engines since 1945, Ted Spaetgens - founder of LO-REZ, became a strong proponent of soft torsional couplings between engines and gear boxes, compressors, etc., as the best solution to the epidemic of early fatigue failures in crankshafts, gears and gear shafts, bearings, clutches, etc. As a torsional consultant to engine builders, shipbuilders and compressor packagers since 1950, a lot of opposition to the soft approach was encountered early on. Fortunately, torsionally-soft couplings have long since become standard and essential equipment for reciprocating power trains. There are a good number of soft coupling manufacturers which turn out good products, and LO-REZ, as well as many of its customers, believe we are among the best of them.

BENEFITS OF THE LO-REZ STEEL SPRING COUPLING

Low and Constant torsional stiffness factor, accurately controlled (±8%) for precise tuning, does not change with age, load or with vibration amplitude, therefore provides constant system torsional characteristics and greatly facilitates torsional analysis and torsiograph evaluation. Springs routinely checked during production; a tolerance of ±4% on stiffness available where required.

Wide selection of torque/stiffness combinations within each of twelve different housing sizes, enables precise selection to be made.

Springs constantly under compression, means that there is no clearance or backlash to affect analytical validity.

Independent friction dampers help control low frequency/large amplitude oscillations — without any change in stiffness. Damping level can be varied. Heat is easily dissipated and does not affect the load-carrying springs nor the torsional stiffness.

Overload limit stops inside the springs prevent excessive spring stresses and coil clash during traversal of major criticals and during shock loading.

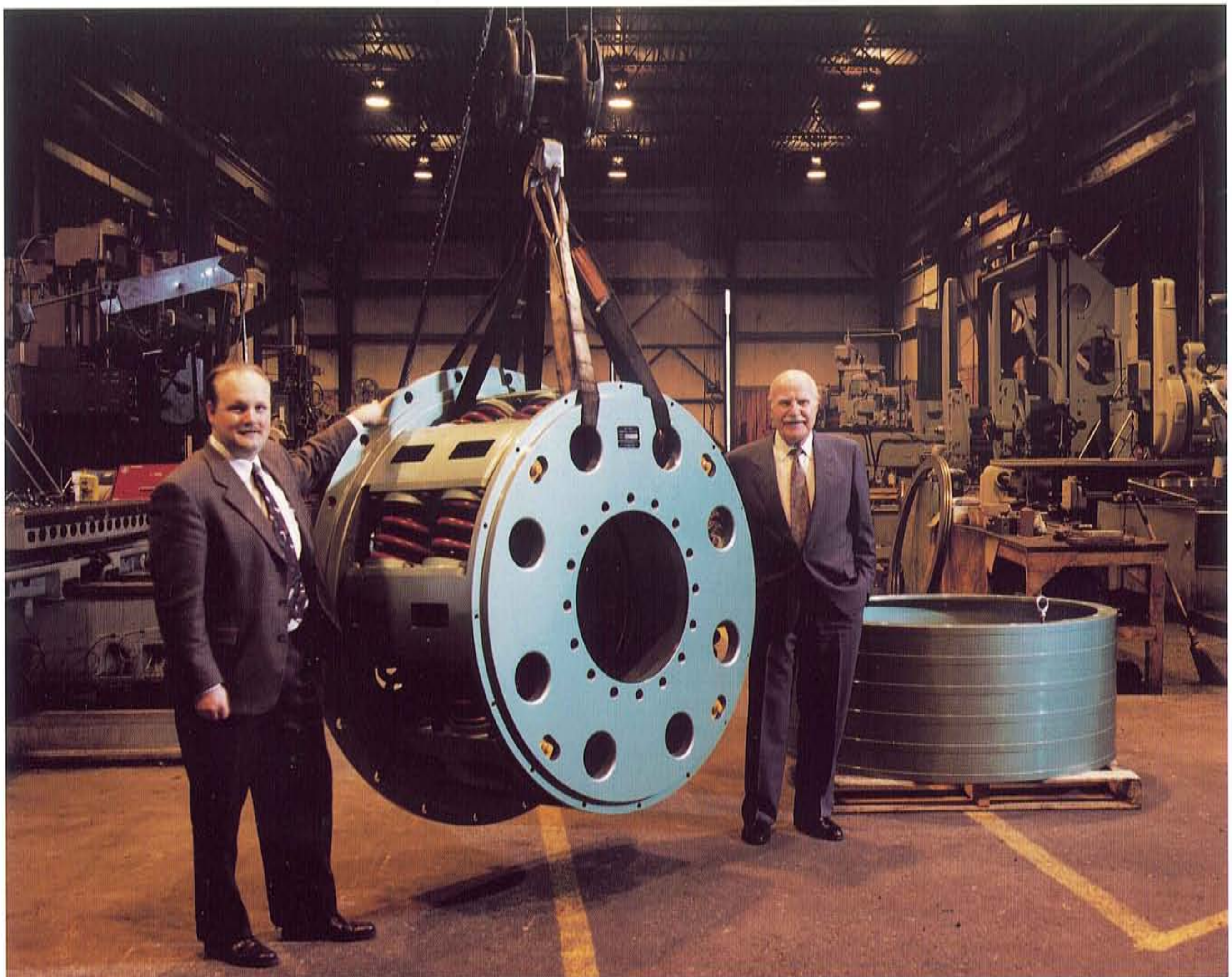
No lubrication required, making the Lo-Rez coupling ideal for retrofitting.

Easy installation and in-place maintenance. Coupling can be inserted or removed without shifting associated equipment, springs can be changed readily with coupling in place.

Ample misalignment capacity with axial, angular and lateral stiffness proportional to torsional stiffness.

Wide variety of arrangements with flanges to suit customer requirement, including pilot bearing, electrical insulation, non-magnetic, special stiffness factors to suit customer requirements.

Double row couplings are suitable for installations where radial space limitations are a problem.



Features

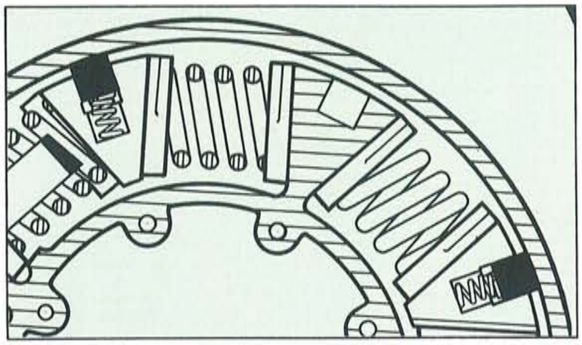
DAMPERS

Each wedge has a cavity in its outer face. Damper pads and springs are located in the cavities in the flange half wedges; there are 4 in all. These dampers are independent of the main coupling springs and can, therefore, be changed to suit particular customer needs. the damper pads act against a cover that is attached to the hub half of the coupling. See Fig. 10.

OVERLOAD PROTECTION

Half of the cavities have elastomeric limit stops which are designed to give full face contact at overload and to prevent the main spring from ever becoming close coiled. The couplings have a normal overload capacity of 3-5 times their rated torque and under emergency conditions will accept higher loads.

ADEQUATE COOLING
The heat created by the dampers during passage through critical speeds is dissipated through the cover which, being the outermost part of the coupling, is more than adequately cooled even when the coupling is in close proximity to a diesel engine.



See Fig. 10

NO BACKLASH
The couplings are assembled with all of the main springs under sufficient pre-load to ensure that even during an overload condition, the trailing springs never become unloaded. This eliminates any backlash and the problems associated with backlash.

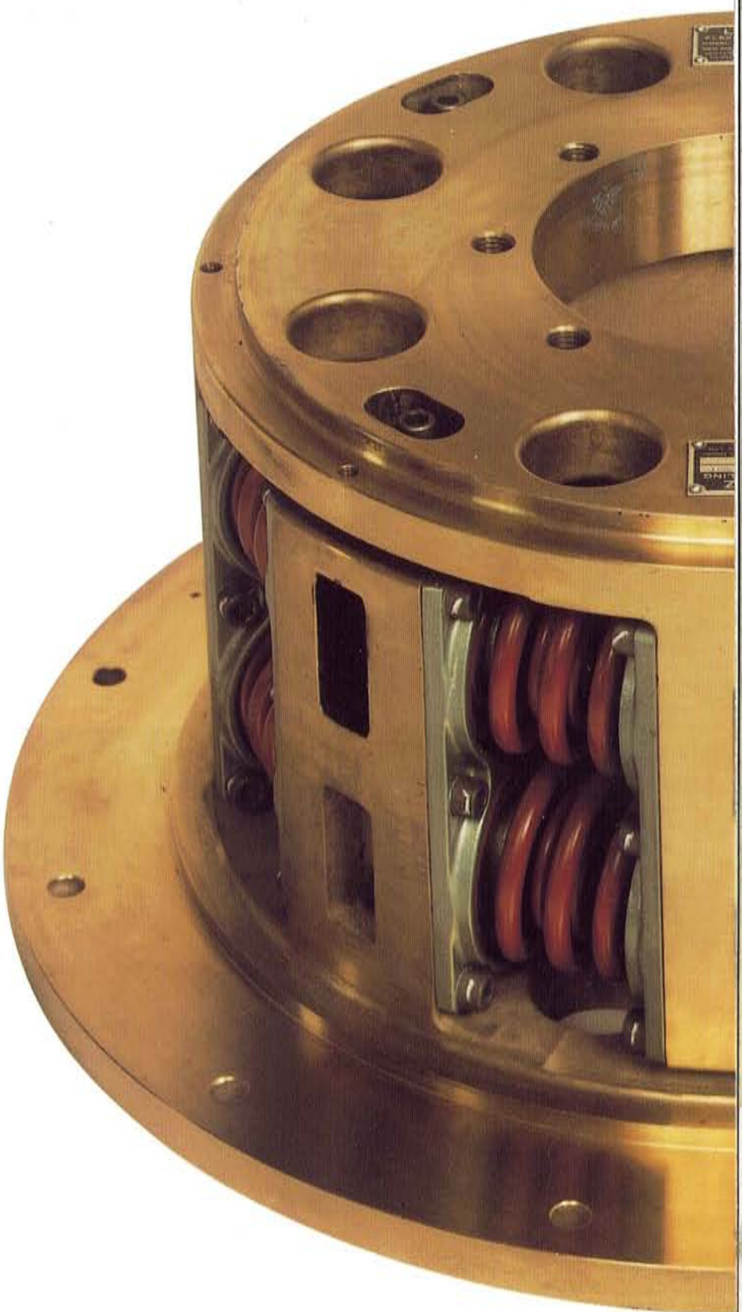
STIFFNESS ACCURACY
The torsional stiffness values for Lo-Rez steel-spring couplings as given in the rating tables are guaranteed accurate to within ±8% with typical error of only ±3%. Lo-Rez units are available with greater accuracy where necessary. The values given are constant, being unaffected by torque, either steady state or vibratory, age or temperature. This enables the torsional vibration engineer to investigate a power train without the need to re-run under changing characteristics and with the knowledge that the coupling data is correct.

VIBRATORY RATING
All of the ratings that appear in the following tables allow for a vibratory rating of ±20% of the rated torque; a value which is representative of many applications. Lo-Rez couplings can be supplied with vibratory ratings as high as ±90% of the rated torque. Lo-Rez should be consulted for further details.

EO SPRINGS
Lo-Rez have developed a spring design known as EO that will reduce the torsional stiffness by 10-15% while maintaining the rated torque. These are available where especially soft couplings might be necessary.

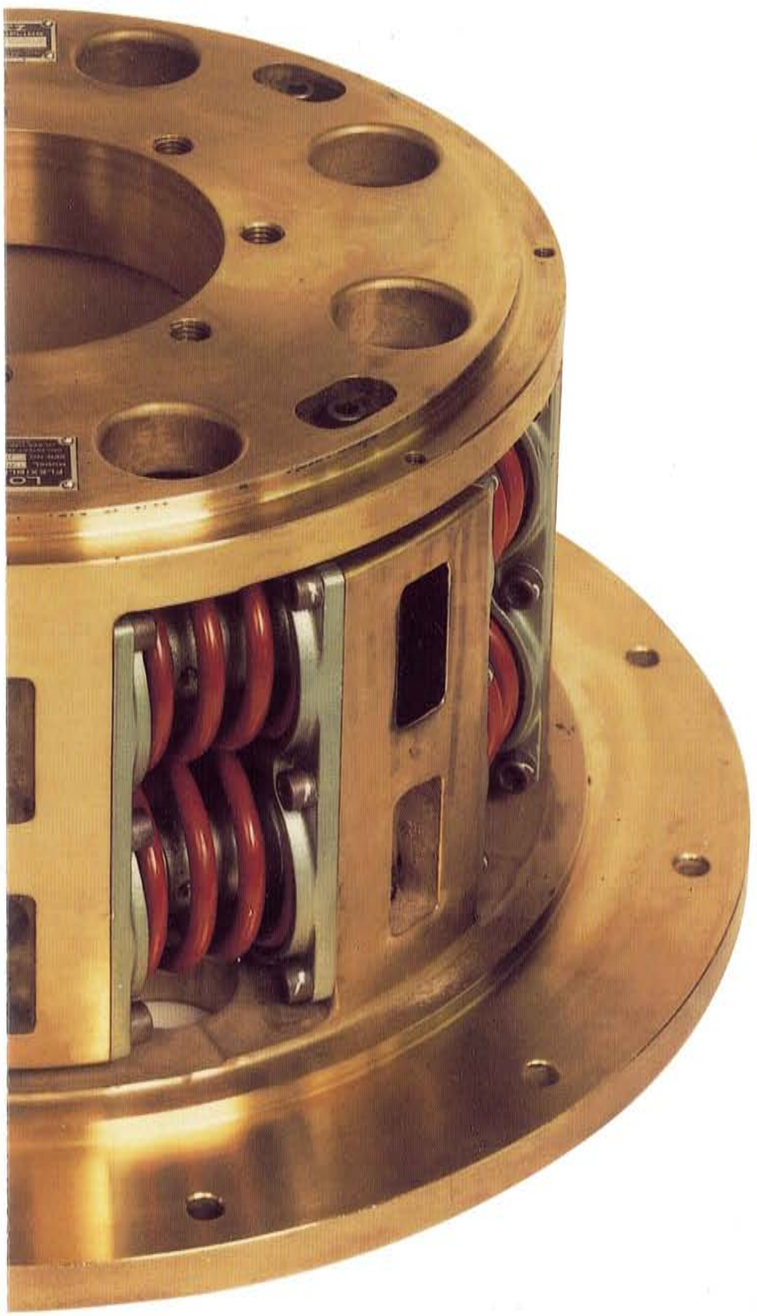
DOUBLE ROW COUPLINGS
Another development is the Lo-Rez double row coupling that effectively doubles both the rated torque and torsion stiffness values, as given in this catalogue. The advantage of this option is to provide a unit that is relatively small in diameter for those applications where radial space is a limitation. See the section on coupling procedure for a typical example (page 8).

TORSIONAL VIBRATION ANALYSIS
The proper selection of torsionally-flexible couplings for systems subject to torsional vibration involves mathematical analysis of the system. Lo-Rez offers its services at nominal cost to customers who do not have their own facilities. In some cases, where the system is simple from the standpoint of torsional vibration, Lo-Rez includes a low-mode torsional vibration analysis without extra charge.



DESIGNATIONS

The basic frame sizes, 12 in all, are designated by a letter D through N. Then come two letters that describe the mounting configuration as shown on page 6. Within any frame size there is an almost limitless number of various torque-stiffness combinations. In the rating charts that follow only a selective number of variations are shown. If a particular combination is required that does not appear, then consult Lo-Rez; it is more than likely that that combination can be made available. Each combination is designated by a series of numbers that represent the rated torque (lb. in.) and the torsional stiffness (lb. in/rad.). Therefore, a typical coupling would be: E/SF 27.2/.440 or F/HF 29.6/.384.



Some designations in the following rating charts appear under the heading STANDARD, while others appear under the heading SPECIAL. The standard unit is as described above, while the special simply means that compound springs (i.e., inner and outer) are used. Positive location of both springs ensures that there is never any interference between the springs.

INSTALLATION AND MAINTENANCE

Each Lo-Rez coupling is supplied with a detailed instruction manual which covers all aspects of installation, running and maintenance.

The four axial compression bolts, supplied with each unit, enable easy insertion or withdrawal of the flexible element from between the driving and driven equipment without disturbing the position of either. For routine inspection, the outer cover can be removed axially to give complete access to damper pads and damper springs, as well as the main springs. In cases where space restrictions prevent the axial movement of the cover, then the Lo-Rez coupling can be supplied with a split cover facilitating radial removal.

With the cover removed, the damper pads and damper springs can be inspected and replaced, if necessary. In addition, using the cavities in the outer faces of the ' lugs ' and a torqueing tool, the relative halves of the flexible element can be rotated to facilitate removal of the main springs. Torqueing tools either mechanical or hydraulic are available from Lo-Rez.

MISALIGNMENT

Each Lo-Rez steel-spring coupling has a high capacity for axial, radial and angular misalignment. The low axial, radial and angular stiffnesses result in relatively low reaction forces being applied to associated machinery. Misalignment stiffness values are given for each unit in the following rating tables and a more detailed explanation of misalignment is given on page 15.

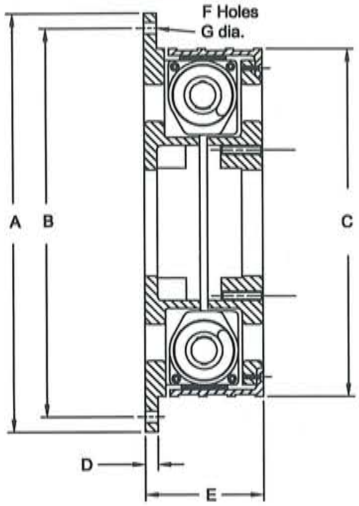
CLASSIFICATION SOCIETIES

Lo-Rez steel-spring couplings are accepted by the leading classification societies.

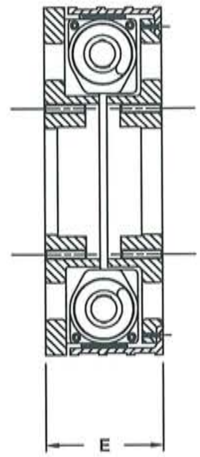
Dimensions

Alternative Arrangements

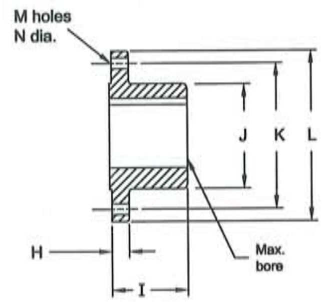
Arrangement SF



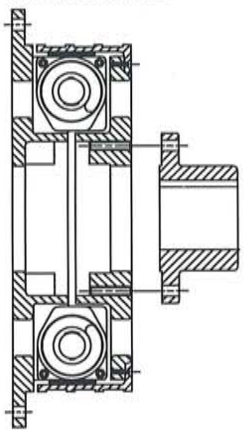
Arrangement SS



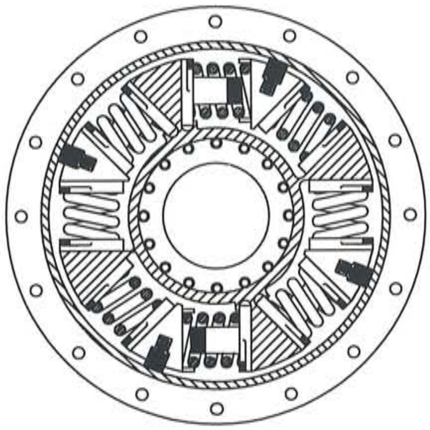
Hub



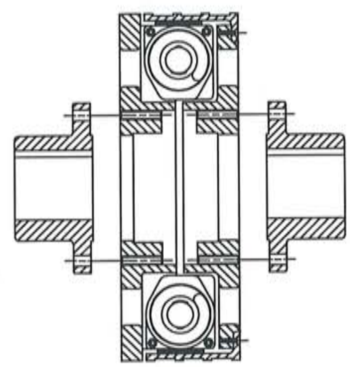
Arrangement HF



Section



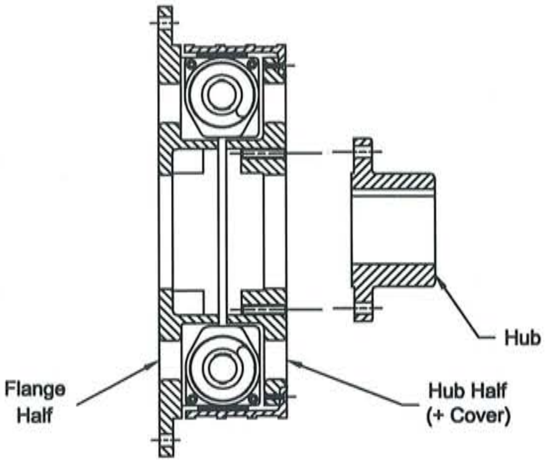
Arrangement HH

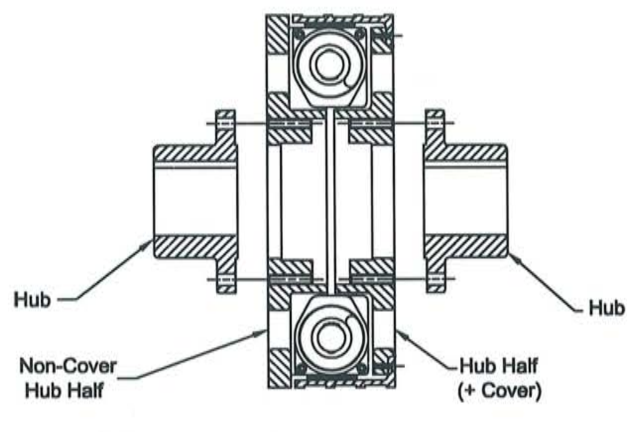


All dimensions are in mm.

COUPLING SIZE	A DIA.	B DIA.	C DIA.	D	E	EE (1)	F	G DIA.	H	I	J DIA.	K DIA.	L DIA.	M (2)	N DIA.	MAX. BORE (3)
D	467	438	355	18	122	186	12	14	20	90	139	159	195	16	14	85
DE	518	489	405	19	143	217	12	14	20	100	155	181	225	16	17	100
E	572	543	460	19	162	248	12	18	25	115	178	203	255	16	17	110
F	673	641	560	21	191	287	12	18	30	140	211	241	290	16	20	125
G	787	737	660	22	216	330	16	18	35	165	264	299	350	16	23	165
H	914	851	760	25	254	387	16	21	45	210	315	356	410	16	26	195
I	1067	1003	900	29	294	457	16	21	47	255	375	419	480	16	30	235
J	1219	1156	1065	30	332	527	16	24	50	285	442	492	560	16	34	275
K	1397	1334	1245	32	375	597	20	27	50	335	508	572	650	16	40	315
L	1600	1537	1450	35	420	690	20	30	55	380	590	711	800	20	40	370
M	1829	1753	1650	38	470	776	20	34	57	430	700	787	900	20	52	430
N	2083	2007	1900	41	520	857	20	42	60	500	810	940	1070	20	52	500

(1) EE is the body width of a double row coupling.
(2) Will be 8 if the coupling is used below 50% of its maximum rating.
(3) Maximum standard bore may be exceeded if special Muff hub is used. Special flange diameters available on request. Dimensions are approximate and not binding.





Single Row Couplings

COUPLING SIZE	MATERIAL	FLANGE HALF (1)		HUB HALF (+ Cover)		NON-COVER HUB HALF		HUB (2)	
		J kg m²	WEIGHT kg	J kg m²	WEIGHT kg	J kg m²	WEIGHT kg	J kg m²	WEIGHT kg
D	ALL	0.27	12	0.20	11	0.17	9	0.04	8
DE	ALUMINIUM	0.56	20	0.40	18	0.35	16	0.07	12
E	ALL	2.26	60	1.36	50	1.36	45	0.12	17
F	FERROUS	4.75	95	3.05	75	3.28	80	0.28	29
G	WITH	10.7	150	7.12	125	7.68	130	0.74	50
H	ALUMINIUM	22.0	227	14.7	190	15.3	195	1.84	89
I	COVER	47.0	330	35.0	275	34.5	275	4.11	146
J		92.0	500	105	575	72.3	455	8.64	225
K	ALL	180	815	185	910	140	725	17.0	340
L		375	1130	385	1270	315	1090	35.3	520
M	FERROUS	620	1700	670	1800	600	1540	77.6	825
N		1410	2400	1470	2400	1200	2130	166	1300

Double Row Couplings

COUPLING SIZE	MATERIAL	FLANGE HALF (1)		HUB HALF (+ Cover)		NON-COVER HUB HALF		HUB (2)	
		J kg m²	WEIGHT kg	J kg m²	WEIGHT kg	J kg m²	WEIGHT kg	J kg m²	WEIGHT kg
D2	ALL	0.33	16	0.30	16	0.23	13	0.04	8
DE2	ALUMINIUM	0.72	27	0.62	25	0.52	23	0.07	12
E2	ALL	2.71	80	1.92	65	1.80	65	0.12	17
F2	FERROUS	5.88	120	4.41	100	4.29	105	0.28	29
G2	WITH	13.6	195	10.4	175	10.3	175	0.74	50
H2	ALUMINIUM	28.2	290	22.6	265	21.5	255	1.84	89
I2	COVER	59.3	450	50.0	410	46.3	400	4.11	146
J2		120	700	158	860	100	650	8.64	225
K2	ALL	225	1090	230	1180	180	1000	17.0	340
L2		475	1500	530	1720	420	1450	35.3	520
M2	FERROUS	900	2360	1050	2630	870	2220	77.6	825
N2		1750	3450	2100	3720	1540	3180	166	1300

(1) Based on standard flange details as given on page 6.
(2) Based on maximum standard bore as given on page 6.
NOTE Weight and J values vary slightly for different torque/stiffness ratings due to differing spring weights. Values shown above are representative of the average.

All of the following selection charts are based on a vibratory torque rating of $\pm 20\%$ which is common for most applications. Each chart shows "standard" and "special" models; the special is simply the standard unit with compound spring arrangement.

SERVICE FACTORS

As the torque ratings given allow for $\pm 20\%$ vibratory torque and a 10% overload condition, which are both typical for most power transmission systems, it is unnecessary to use 'service' factors when making a selection. It is assumed that proper dynamic analysis of the overall system has been used in the arrival of the maximum torque value applied to the coupling. However in extremely severe drives, such as rock or ore crushers, or where very frequent starts and stops of the system occur, a service factor of 1.25 should be used.

SELECTION

There are two basic types of selection: firstly, where a torsional vibration analysis has been carried out and the required stiffness is known, and secondly, where the required stiffness is not known; this is usually the case at an early stage of a project.

Consider the first case:

A compressor drive rated at 148 BHP at 374 rpm, with a $\pm 20\%$ vibratory rating, requiring a coupling stiffness around 50,000 Nm/rad:

$$\text{Torque (Nm)} = \frac{\text{Power (BHP)}}{\text{Speed (rpm)}} \times 7120$$

$$\text{or} \quad \frac{\text{Power (kW)}}{\text{Speed (rpm)}} \times 9549$$

therefore Torque = 2817 Nm

From the rating tables any coupling DE or larger can handle the torque. The relevant stiffness of the DE (model DE 26.0/.523) has a torsional stiffness of 59100 Nm/rad, while that of the E (model E27.2/.440) is 49700 Nm/rad and that of the F (model F29.6/.384) is 43,400 Nm/rad.

It is possible that a further torsional analysis would show that the DE coupling was suitable.

Consider the second case:

A marine propulsion system with a multi-cylinder diesel rated at 1125 BHP at 1225 rpm to a fixed pitch propeller via a reverse reduction gear. Vibratory rating $\pm 20\%$.

$$\text{Torque (Nm)} = \frac{\text{Power (BHP)}}{\text{Speed (rpm)}} \times 7120$$

$$\text{or} \quad \frac{\text{Power (kW)}}{\text{Speed (rpm)}} \times 9549$$

therefore Torque = 6539 Nm

From the rating table the smallest coupling capable of this torque is the E (model E59.5/1.64) which has a torsional stiffness of 185,000 Nm/rad.

Providing that this stiffness is acceptable, then this selection would be correct. If, however, a lower stiffness is envisaged, then the following options are available:

1) Consult Lo-Rez to investigate alternative E coupling ratings, i.e. intermediate values or the use of EO springs.

2) Select from the next frame size up; for this example, select F (model F59.3/1.23) which has a torsional stiffness of 139,000 Nm/rad.

Double Row Coupling

If diametral space is a limitation that prohibits selection of F or E above, then a Double Row coupling may be considered.

For a double row coupling with a torque capacity of 6539 Nm, select a single row unit with a capacity of half this figure (3270 Nm).

From the rating charts a DE (model DE 30.8/.622) can be selected as a double row unit and its designation becomes DE2 61.6/1.24 which has a torque capacity of 6960 Nm and a torsional stiffness of 140,600 Nm/rad.

MODEL D Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS $\frac{\text{Nm/rad}}{10^6}$	SPEED rpm	AXIAL $\frac{\text{N/m}}{10^6}$	RADIAL $\frac{\text{N/m}}{10^6}$	ANGULAR $\frac{\text{Nm/rad}}{10^6}$	AXIAL mm	RADIAL mm	ANGULAR DEG
.950/.006		107	0.00066	2000	0.04	0.04	0.0003	2.5	2.5	0.75
1.25/.008		141	0.00095	2000	0.05	0.06	0.0004	2.5	2.5	0.75
1.61/.012		182	0.00134	2000	0.08	0.08	0.0006	2.5	2.5	0.75
1.94/.016		219	0.00183	2000	0.10	0.11	0.0009	2.5	2.5	0.75
2.78/.022		314	0.00246	2000	0.14	0.15	0.0012	2.5	2.5	0.75
3.17/.033		358	0.00371	2000	0.22	0.23	0.0018	2.5	2.5	0.75
3.44/.037		389	0.00416	2000	0.25	0.25	0.0020	2.5	2.5	0.75
3.98/.045		450	0.00508	2000	0.31	0.32	0.0026	2.5	2.5	0.75
4.87/.061		550	0.00689	2000	0.43	0.43	0.0035	2.5	2.5	0.75
6.40/.096		723	0.01080	2000	0.68	0.68	0.0055	2.5	2.5	0.75
6.70/.106		757	0.01200	2000	0.75	0.75	0.0061	2.5	2.5	0.75
8.05/.138		910	0.01560	2000	0.98	0.98	0.0079	2.5	2.5	0.75
10.7/.222		1210	0.02510	2000	1.57	1.58	0.0120	2.5	2.5	0.75
11.6/.284		1310	0.03210	2000	2.03	2.03	0.0160	2.5	2.5	0.75
	13.5/.340	1530	0.03840	2000	2.18	2.31	0.0180	2.5	2.5	0.70
18.3/.623		2070	0.07040	2000	4.44	4.48	0.0360	2.0	2.0	0.60
20.6/.572		2330	0.06460	2000	3.78	3.81	0.0300	1.8	1.8	0.50
	23.0/.640	2600	0.07230	2000	3.93	4.14	0.0310	1.5	1.5	0.45

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%, typical error being ±3%.

MODEL DE Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS $\frac{\text{Nm/rad}}{10^6}$	SPEED rpm	AXIAL $\frac{\text{N/m}}{10^6}$	RADIAL $\frac{\text{N/m}}{10^6}$	ANGULAR $\frac{\text{Nm/rad}}{10^6}$	AXIAL mm	RADIAL mm	ANGULAR DEG
2.01/.012		227	0.00136	2000	0.06	0.06	0.0007	3.5	3.5	0.90
2.36/.016		267	0.00181	2000	0.08	0.08	0.0008	3.5	3.5	0.90
2.61/.018		295	0.00203	2000	0.09	0.09	0.0010	3.5	3.5	0.90
3.31/.025		374	0.00282	2000	0.12	0.12	0.0014	3.5	3.5	0.90
4.18/.034		472	0.00384	2000	0.17	0.17	0.0019	3.5	3.5	0.90
5.19/.046		586	0.00520	2000	0.24	0.24	0.0027	3.5	3.5	0.90
7.20/.073		813	0.00825	2000	0.38	0.38	0.0043	3.5	3.5	0.90
8.67/.095		980	0.01070	2000	0.50	0.50	0.0055	3.5	3.5	0.90
	11.4/.126	1290	0.01420	2000	0.58	0.62	0.0064	3.5	3.5	0.90
12.8/.177		1450	0.02000	2000	0.95	0.93	0.0100	3.5	3.5	0.90
	15.8/.218	1790	0.02460	2000	1.05	1.09	0.0110	3.5	3.5	0.90
16.5/.254		1860	0.02870	2000	1.36	1.34	0.0140	3.5	3.5	0.90
	20.0/.308	2260	0.03480	2000	1.48	1.54	0.0160	3.5	3.5	0.90
26.0/.523		2940	0.05910	2000	2.80	2.76	0.0300	3.0	3.0	0.85
	30.8/.622	3480	0.07030	2000	2.99	3.13	0.0320	2.8	2.8	0.80
32.1/.771		3630	0.08710	2000	4.13	4.07	0.0450	2.5	2.5	0.75
	37.3/.897	4210	0.10100	2000	4.35	4.53	0.0480	2.3	2.3	0.65
45.6/1.26		5150	0.14200	2000	6.61	6.59	0.0730	2.0	2.0	0.55

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL E Steel-Spring Coupling

DESIGNATION		RATED TORQUE Nm	TORSIONAL STIFFNESS $\frac{Nm/rad}{10^6}$	MAX. SPEED rpm	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL				AXIAL $\frac{N/m}{10^6}$	RADIAL $\frac{N/m}{10^6}$	ANGULAR $\frac{Nm/rad}{10^6}$	AXIAL mm	RADIAL mm	ANGULAR DEG
2.84/.019		321	0.00210	2000	0.08	0.08	0.0011	3.5	3.5	0.80
3.12/.021		353	0.00237	2000	0.09	0.09	0.0012	3.5	3.5	0.80
3.55/.026		401	0.00290	2000	0.11	0.11	0.0015	3.5	3.5	0.80
4.40/.035		497	0.00393	2000	0.15	0.15	0.0020	3.5	3.5	0.80
5.60/.050		633	0.00565	2000	0.22	0.22	0.0031	3.5	3.5	0.80
5.92/.054		669	0.00610	2000	0.24	0.24	0.0033	3.5	3.5	0.80
7.21/0.71		815	0.00802	2000	0.32	0.31	0.0043	3.5	3.5	0.80
9.35/.104		1060	0.01180	2000	0.47	0.46	0.0064	3.5	3.5	0.80
10.3/.122		1160	0.01380	2000	0.56	0.54	0.0076	3.5	3.5	0.80
	12.3/.145	1390	0.01640	2000	0.61	0.62	0.0082	3.5	3.5	0.80
18.0/.222		2030	0.02510	2000	1.01	0.98	0.0130	3.5	3.5	0.80
	20.8/.258	2350	0.02920	2000	1.08	1.10	0.0140	3.5	3.5	0.80
27.2/.440		3070	0.04970	2000	2.01	1.96	0.0270	3.5	3.5	0.80
	30.8/.500	3480	0.05650	2000	2.11	2.15	0.0280	3.0	3.0	0.75
39.0/.782		4410	0.08840	2000	3.55	3.46	0.0470	2.8	2.8	0.70
	44.5/.895	5030	0.10100	2000	3.72	3.79	0.0490	2.5	2.5	0.65
53.5/1.47		6040	0.16600	2000	6.49	6.42	0.0870	2.3	2.3	0.55
	59.5/1.64	6720	0.18500	2000	6.70	6.93	0.0890	2.0	2.0	0.50

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL F Steel-Spring Coupling

DESIGNATION		RATED TORQUE Nm	TORSIONAL STIFFNESS $\frac{Nm/rad}{10^6}$	MAX. SPEED rpm	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL				AXIAL $\frac{N/m}{10^6}$	RADIAL $\frac{N/m}{10^6}$	ANGULAR $\frac{Nm/rad}{10^6}$	AXIAL mm	RADIAL mm	ANGULAR DEG
4.73/.033		534	0.00376	1800	0.09	0.09	0.0017	4.0	4.0	0.80
6.12/.044		691	0.00503	1800	0.11	0.12	0.0024	4.0	4.0	0.80
6.50/.049		734	0.00554	1800	0.12	0.13	0.0026	4.0	4.0	0.80
7.72/.063		872	0.00715	1800	0.16	0.17	0.0034	4.0	4.0	0.80
11.1/.108		1250	0.01220	1800	0.29	0.30	0.0060	4.0	4.0	0.80
	14.6/.143	1650	0.01620	1800	0.33	0.37	0.0068	4.0	4.0	0.80
19.2/.210		2170	0.02370	1800	0.57	0.58	0.0110	4.0	4.0	0.80
	23.0/.250	2600	0.02820	1800	0.61	0.67	0.0120	4.0	4.0	0.80
29.6/.384		3340	0.04340	1800	1.04	1.07	0.0210	4.0	4.0	0.80
	34.6/.450	3910	0.05080	1800	1.11	1.21	0.0220	4.0	4.0	0.80
42.5/.710		4800	0.08020	1800	1.96	1.99	0.0390	4.0	4.0	0.80
	48.0/.805	5420	0.09100	1800	2.04	2.18	0.0400	4.0	4.0	0.80
59.3/1.23		6670	0.13900	1800	3.41	3.48	0.0670	4.0	4.0	0.80
	66.8/1.39	7550	0.15700	1800	3.51	3.78	0.0700	3.8	3.8	0.75
80.8/2.09		9130	0.23600	1800	5.74	5.88	0.1100	3.3	3.3	0.70
	88.5/2.30	10000	0.26000	1800	5.89	6.28	0.1100	3.0	3.0	0.65
104/3.49		11800	0.39400	1800	9.50	9.78	0.1800	2.8	2.8	0.55
	116/3.87	13100	0.43700	1800	9.74	10.4	0.1900	2.3	2.3	0.50

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL G Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad 10^6	SPEED rpm	AXIAL N/m 10^6	RADIAL N/m 10^6	ANGULAR Nm/rad 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
11.4/.095		1300	0.01080	1600	0.16	0.17	0.0047	4.0	4.0	0.70
	12.4/.123	1410	0.01390	1600	0.18	0.21	0.0053	4.0	4.0	0.70
20.3/.167		2320	0.01890	1600	0.28	0.30	0.0080	4.0	4.0	0.70
	24.0/.197	2710	0.02230	1600	0.29	0.35	0.0086	4.0	4.0	0.70
30.0/.304		3390	0.03430	1600	0.52	0.56	0.0140	4.0	4.0	0.70
	35.0/.354	3950	0.04000	1600	0.56	0.64	0.0150	4.0	4.0	0.70
43.0/.532		4860	0.06010	1600	0.94	1.00	0.0270	4.0	4.0	0.70
	48.8/.600	5510	0.06780	1600	1.01	1.13	0.0290	4.0	4.0	0.70
59.5/.890		6720	0.10100	1600	1.59	1.68	0.0460	4.0	4.0	0.70
	67.0/1.00	7570	0.11300	1600	1.65	1.83	0.0470	4.0	4.0	0.70
78.0/1.47		8810	0.16600	1600	2.66	2.80	0.0760	4.0	4.0	0.70
	87.5/1.65	9890	0.18600	1600	2.76	3.04	0.0790	4.0	4.0	0.70
105/2.37		11900	0.26800	1600	4.28	4.53	0.1200	3.8	3.8	0.65
	117/2.64	13200	0.29800	1600	4.41	4.88	0.1200	3.6	3.6	0.60
133/3.60		15000	0.40700	1600	6.44	6.82	0.1800	3.3	3.3	0.55
	155/4.18	17500	0.47200	1600	6.68	7.59	0.1900	3.0	3.0	0.50
172/6.00		19400	0.67800	1600	10.6	11.3	0.3000	2.8	2.8	0.45
	194/6.75	21900	0.76300	1600	10.9	12.3	0.3100	2.3	2.3	0.40

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL H Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad 10^6	SPEED rpm	AXIAL N/m 10^6	RADIAL N/m 10^6	ANGULAR Nm/rad 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
28.2/.244		3190	0.02760	1400	0.33	0.35	0.0120	5.0	5.0	0.70
	33.0/.285	3730	0.03220	1400	0.36	0.39	0.0130	5.0	5.0	0.70
40.4/.385		4560	0.04350	1400	0.52	0.55	0.0200	5.0	5.0	0.70
	46.3/.435	5230	0.04910	1400	0.55	0.61	0.0210	5.0	5.0	0.70
56.2/.600		6350	0.06780	1400	0.84	0.87	0.0310	5.0	5.0	0.70
	65.5/.700	7400	0.07910	1400	0.89	0.98	0.0330	5.0	5.0	0.70
74.0/.990		8360	0.11200	1400	1.42	1.45	0.0540	5.0	5.0	0.70
	90.2/1.21	10200	0.13700	1400	1.52	1.68	0.0570	5.0	5.0	0.70
97.5/1.57		11000	0.17700	1400	2.27	2.32	0.0870	5.0	5.0	0.70
	115/1.85	13000	0.20900	1400	2.41	2.62	0.0910	5.0	5.0	0.70
127/2.25		14300	0.25400	1400	3.20	3.29	0.1200	5.0	5.0	0.70
	152/2.70	17200	0.30500	1400	3.39	3.76	0.1200	4.6	4.6	0.65
159/3.24		18000	0.36600	1400	4.65	4.79	0.1700	4.3	4.3	0.65
	187/3.83	21100	0.43300	1400	4.86	5.39	0.1800	4.1	4.1	0.60
197/4.85		22300	0.54800	1400	6.63	6.96	0.2500	3.8	3.8	0.60
	226/5.58	25500	0.63000	1400	6.84	7.68	0.2500	3.6	3.6	0.55
239/7.10		27000	0.80200	1400	10.1	10.4	0.3800	3.3	3.3	0.50
	271/8.05	30600	0.91000	1400	10.4	11.4	0.3900	2.8	2.8	0.45

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL I Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad / 10^6	SPEED rpm	AXIAL N/m / 10^6	RADIAL N/m / 10^6	ANGULAR Nm/rad / 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
52.5/.417		5930	0.04710	1200	0.42	0.44	0.0220	5.5	5.5	0.70
	60.5/.480	6840	0.05420	1200	0.45	0.49	0.0230	5.5	5.5	0.70
70.0/.657		7910	0.07420	1200	0.69	0.70	0.0360	5.5	5.5	0.70
	78.0/.735	8810	0.08300	1200	0.72	0.76	0.0380	5.5	5.5	0.70
91.0/1.00		10300	0.11300	1200	1.07	1.09	0.0560	5.5	5.5	0.70
	113/1.25	12800	0.14100	1200	1.17	1.28	0.0620	5.5	5.5	0.70
116/1.42		13100	0.16000	1200	1.53	1.55	0.0810	5.5	5.5	0.70
146/2.01		16500	0.22700	1200	2.17	2.18	0.1100	5.5	5.5	0.70
	149/1.83	16800	0.20700	1200	1.68	1.85	0.0800	5.5	5.5	0.70
181/2.95		20500	0.33300	1200	3.20	3.22	0.1600	5.5	5.5	0.70
	182/2.48	20600	0.28000	1200	2.34	2.53	0.1200	5.5	5.5	0.70
	215/3.53	24300	0.39900	1200	3.39	3.64	0.1700	5.3	5.3	0.65
224/4.30		25300	0.48600	1200	4.67	4.69	0.2400	5.1	5.1	0.65
272/5.70		30700	0.64400	1200	6.14	6.19	0.3200	4.6	4.6	0.60
	274/5.25	31000	0.59300	1200	4.97	5.40	0.2500	4.3	4.3	0.55
324/8.10		36600	0.91500	1200	8.73	8.78	0.4600	4.1	4.1	0.50
	325/6.80	36700	0.76800	1200	6.45	7.00	0.3300	3.8	3.8	0.50
	380/9.45	42900	1.07000	1200	9.10	9.80	0.4700	3.3	3.3	0.45

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL J Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad / 10^6	SPEED rpm	AXIAL N/m / 10^6	RADIAL N/m / 10^6	ANGULAR Nm/rad / 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
90.5/.735		10200	0.08300	1000	0.53	0.55	0.0390	6.5	6.5	0.70
	111/.900	12500	0.10200	1000	0.58	0.64	0.0420	6.5	6.5	0.70
117/1.04		13200	0.11800	1000	0.75	0.78	0.0550	6.5	6.5	0.70
	147/1.31	16600	0.14800	1000	0.82	0.93	0.0610	6.5	6.5	0.70
147/1.50		16600	0.16900	1000	1.10	1.14	0.0810	6.5	6.5	0.70
	177/1.82	20000	0.20600	1000	1.19	1.32	0.0880	6.5	6.5	0.70
186/2.11		21000	0.23800	1000	1.56	1.61	0.1100	6.5	6.5	0.70
	230/2.61	26000	0.29500	1000	1.69	1.89	0.1200	6.5	6.5	0.70
222/2.83		25100	0.32000	1000	2.10	2.17	0.1500	6.5	6.5	0.70
265/3.84		29900	0.43400	1000	2.88	2.95	0.2100	6.5	6.5	0.70
	266/3.40	30100	0.38400	1000	2.25	2.48	0.1600	6.5	6.5	0.70
318/5.17		35900	0.58400	1000	3.90	3.99	0.2800	6.5	6.5	0.70
	330/4.75	37300	0.53700	1000	3.09	3.44	0.2200	6.5	6.5	0.70
	385/6.25	43500	0.70600	1000	4.14	4.55	0.3000	6.1	6.1	0.65
441/8.75		49800	0.98900	1000	6.58	6.73	0.4800	5.6	5.6	0.60
	515/10.2	58200	1.15000	1000	6.86	7.47	0.4900	5.1	5.1	0.55
600/14.8		67800	1.67000	1000	11.0	11.3	0.8100	4.3	4.3	0.45
	670/17.3	75700	1.95000	1000	11.4	12.6	0.8400	3.8	3.8	0.40

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL K Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad / 10^6	SPEED rpm	AXIAL N/m / 10^6	RADIAL N/m / 10^6	ANGULAR Nm/rad / 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
26.5/.127		2990	0.01430	800	0.06	0.06	0.0056	6.5	6.5	0.60
37.5/.194		4240	0.02190	800	0.09	0.10	0.0089	6.5	6.5	0.60
51.6/.303		5830	0.03420	800	0.14	0.16	0.0140	6.5	6.5	0.60
68.6/.425		7750	0.04800	800	0.20	0.22	0.0210	6.5	6.5	0.60
90.0/.618		10200	0.06980	800	0.31	0.33	0.0310	6.5	6.5	0.60
143/1.20		16200	0.13600	800	0.63	0.66	0.0630	6.5	6.5	0.60
217/2.17		24500	0.24500	800	1.16	1.20	0.1100	6.5	6.5	0.60
	257/2.54	29000	0.28700	800	1.24	1.35	0.1200	6.5	6.5	0.60
312/3.66		35300	0.41400	800	1.97	2.04	0.1900	6.5	6.5	0.60
	369/4.31	41700	0.48700	800	2.10	2.31	0.2100	6.5	6.5	0.60
434/6.30		49000	0.71200	800	3.44	3.53	0.3500	6.5	6.5	0.60
	514/7.46	58100	0.84300	800	3.65	3.99	0.3600	6.5	6.5	0.60
582/10.5		65800	1.19000	800	5.77	5.91	0.5700	6.5	6.5	0.60
	693/12.5	78300	1.41000	800	6.09	6.66	0.6100	5.8	5.8	0.55
765/17.0		86400	1.92000	800	9.36	9.59	0.9300	5.6	5.6	0.50
	880/19.5	99400	2.20000	800	9.83	10.8	0.9800	5.1	5.1	0.45
985/24.8		111000	2.80000	800	13.4	13.8	1.3500	4.6	4.6	0.40
	1142/28.8	129000	3.25000	800	13.9	15.3	1.4100	3.8	3.8	0.35

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL L Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad / 10^6	SPEED rpm	AXIAL N/m / 10^6	RADIAL N/m / 10^6	ANGULAR Nm/rad / 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
67.2/.337		7590	0.03810	700	0.10	0.12	0.0140	7.0	7.0	0.60
87.5/.487		9890	0.05500	700	0.16	0.18	0.0220	7.0	7.0	0.60
140/.882		15800	0.09970	700	0.30	0.34	0.0410	7.0	7.0	0.60
210/1.57		23700	0.17700	700	0.57	0.62	0.0780	7.0	7.0	0.60
302/2.63		34100	0.29700	700	0.98	1.05	0.1300	7.0	7.0	0.60
	370/3.19	41800	0.36000	700	1.06	1.21	0.1400	7.0	7.0	0.60
420/4.19		47500	0.47300	700	1.58	1.68	0.2100	7.0	7.0	0.60
	512/5.10	57800	0.57600	700	1.70	1.94	0.2300	7.0	7.0	0.60
563/6.42		63600	0.72500	700	2.45	2.59	0.3200	7.0	7.0	0.60
	648/7.79	77300	0.88000	700	2.60	2.97	0.3500	7.0	7.0	0.60
735/10.2		83000	1.15000	700	3.92	4.11	0.5300	7.0	7.0	0.60
	891/12.4	101000	1.40000	700	4.16	4.74	0.5600	7.0	7.0	0.60
946/15.7		107000	1.77000	700	6.10	6.38	0.8200	6.9	6.9	0.60
	1110/18.4	125000	2.08000	700	6.40	7.14	0.8700	6.4	6.4	0.55
1189/22.2		134000	2.51000	700	8.55	8.97	1.1200	6.1	6.1	0.50
	1403/26.2	159000	2.96000	700	8.94	10.0	1.2400	5.3	5.3	0.45
1475/33.0		167000	3.73000	700	12.7	13.3	1.6900	4.8	4.8	0.40
	1704/38.0	193000	4.29000	700	13.1	14.7	1.8000	4.3	4.3	0.35

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL M Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad 10^6	SPEED rpm	AXIAL N/m 10^6	RADIAL N/m 10^6	ANGULAR Nm/rad 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
412/3.41		46500	0.38500	600	0.93	1.00	0.1600	7.5	7.5	0.55
	522/4.31	59000	0.48700	600	1.03	1.20	0.1900	7.5	7.5	0.55
553/5.19		62500	0.58600	600	1.45	1.54	0.2500	7.5	7.5	0.55
	693/6.51	78300	0.73600	600	1.58	1.83	0.2900	7.5	7.5	0.55
722/8.15		81600	0.92100	600	2.34	2.45	0.4200	7.5	7.5	0.55
	901/10.1	102000	1.14000	600	2.53	2.88	0.4600	7.5	7.5	0.55
928/11.7		105000	1.32000	600	3.36	3.51	0.6100	7.5	7.5	0.55
	1114/13.9	126000	1.57000	600	3.57	4.00	0.6500	7.5	7.5	0.55
1169/16.3		132000	1.84000	600	4.63	4.90	0.8400	7.5	7.5	0.55
	1406/19.7	159000	2.23000	600	4.90	5.60	0.8900	7.5	7.5	0.55
1445/24.0		163000	2.71000	600	6.96	7.28	1.2400	7.5	7.5	0.55
	1740/29.0	197000	3.28000	600	7.35	8.33	1.3500	7.4	7.4	0.55
1748/32.4		198000	3.66000	600	9.34	9.80	1.6900	7.1	7.1	0.50
	2041/37.8	231000	4.27000	600	9.73	10.9	1.8000	6.6	6.6	0.50
2110/46.3		238000	5.23000	600	13.3	14.0	2.4800	6.4	6.4	0.45
	2420/53.2	273000	6.01000	600	13.8	15.4	2.5400	5.6	5.6	0.45
2518/65.8		284000	7.43000	600	18.9	19.9	3.5000	5.3	5.3	0.40
	2844/74.2	321000	8.38000	600	19.4	21.5	3.5500	4.8	4.8	0.35

(1) See page 15 for full explanation of misalignment.

For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

MODEL N Steel-Spring Coupling

DESIGNATION		RATED	TORSIONAL	MAX.	STIFFNESS			MISALIGNMENT (1)		
STANDARD	SPECIAL	TORQUE Nm	STIFFNESS Nm/rad 10^6	SPEED rpm	AXIAL N/m 10^6	RADIAL N/m 10^6	ANGULAR Nm/rad 10^6	AXIAL mm	RADIAL mm	ANGULAR DEG
570/4.65		64400	0.52500	540	0.88	0.97	0.2200	10.0	10.0	0.55
	660/5.35	74600	0.60400	540	0.92	1.10	0.2300	10.0	10.0	0.55
	710/5.80	80200	0.65500	540	0.95	1.10	0.2400	10.0	10.0	0.55
745/6.80		84200	0.76800	540	1.30	1.40	0.3300	10.0	10.0	0.55
	840/7.55	94900	0.85300	540	1.40	1.60	0.3400	10.0	10.0	0.55
	920/8.40	104000	0.94900	540	1.40	1.70	0.3500	10.0	10.0	0.55
960/9.80		108000	1.11000	540	2.00	2.20	0.5100	10.0	10.0	0.55
	1075/11.0	121000	1.24000	540	2.20	2.60	0.5400	10.0	10.0	0.55
1210/14.0		137000	1.58000	540	2.90	3.10	0.7100	10.0	10.0	0.55
	1435/16.5	162000	1.86000	540	3.00	3.60	0.7600	10.0	10.0	0.55
1500/18.5		169000	2.09000	540	3.90	4.20	0.9700	10.0	10.0	0.55
	1785/23.0	202000	2.60000	540	4.10	4.80	1.0000	9.7	9.7	0.55
	2130/31.0	241000	3.50000	540	6.00	7.10	1.5000	9.1	9.1	0.55
	2530/40.0	286000	4.52000	540	7.80	9.10	1.9000	8.6	8.6	0.50
2650/47.5		299000	5.37000	540	10.0	11.0	2.6000	8.1	8.1	0.50
	3100/56.0	350000	6.33000	540	11.0	13.0	2.7000	7.6	7.6	0.45
	3600/75.0	407000	8.47000	540	14.0	16.0	3.4000	6.9	6.9	0.40
	4170/97.0	471000	11.0000	540	19.0	22.0	4.7000	5.8	5.8	0.35

(1) See page 15 for full explanation of misalignment.

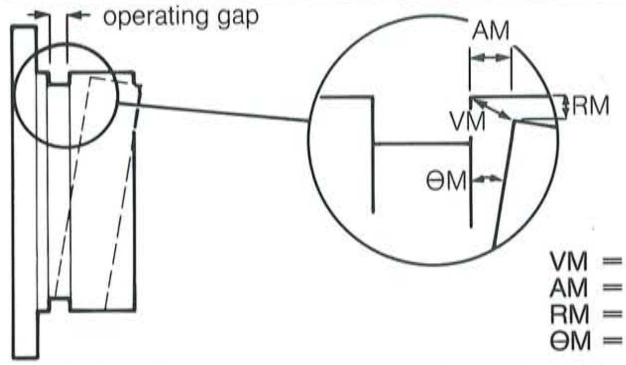
For double row couplings use above misalignment values but double rated torque, torsional stiffness and misalignment stiffnesses.
Above ratings are for ±20% vibratory ratings.
Torsional stiffness values guaranteed ±8%. Typical error being ±3%.

Misalignment

The various misalignment values given in the preceding rating tables are based on the following:

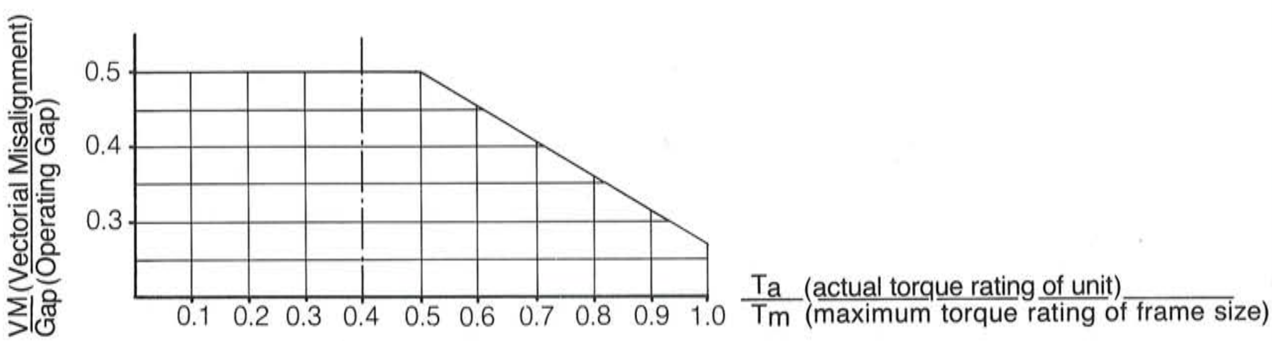
Each coupling has, in the true aligned condition, an operating gap between the cover and the body of the flange half. The value of this operating gap for each frame size is given in the following table:

COUPLING SIZE	OPERATING GAP mm	COUPLING SIZE	OPERATING GAP mm	COUPLING SIZE	OPERATING GAP mm
D	5.00	G	8.00	K	13.00
DE	7.00	H	10.00	L	14.00
E	7.00	I	11.00	M	15.00
F	8.00	J	13.00	N	20.00



VM = Vectorial Misalignment
AM = Axial Misalignment
RM = Radial Misalignment
ΘM = Angular Misalignment

As depicted above, the total allowable vectorial misalignment (VM) is obtained from the following graph and has a radial misalignment component (RM) and an axial misalignment component (AM). The angular misalignment (ΘM) is obtained from the formulae: $\Theta M = \sin^{-1}(AM/R)$ where R is the radius of the coupling body.



Consider a J/SF 265/3.84 which has a rated torque of 29,900 Nm. The maximum torque rating of the J unit is 75,700 Nm.

$$\text{Hence } \frac{T_a}{T_m} = \frac{29900}{75700} = 0.395$$

From the graph (VM/Gap) = .5, and Gap for a J coupling = 13 mm.

Hence VM = 6.5 mm
Maximum Axial misalignment AM = 6.5mm (when RM = 0)
Maximum Radial misalignment RM = 6.5mm (when AM = 0)

$$\Theta M = \sin^{-1} \frac{6.5}{532.5} = 0.70°$$

Any combination of axial, radial and angular misalignment can exist provided that the vectorial misalignment above is not exceeded.

Double row coupling — The misalignment capacities of double row couplings are the same as those listed for single row units. However, the reaction forces must be doubled.

THEORY:

A. GENERAL

Prior to the acceptance of torsionally-soft couplings in the marine and industrial power transmission fields, fatigue failures of engine and compressor crankshafts, gear shafts, gear teeth, bearings, clutches, etc. had reached near-epidemic proportions. Technology was not sufficiently advanced to provide a system to withstand the massive and often unpredictable torsional vibration levels found on many rigidly connected power train systems; those operating across a wide range of speed and employing gearing had the worst 'track record'. With the introduction of torsionally-soft couplings in the early 1950's, these chronic problems began to disappear. Originally there was considerable opposition to the 'soft' concept from several quarters. Fortunately today most critical power train systems employ some form of torsionally-soft coupling.

B. BENEFITS OF TORSIONALLY-SOFT COUPLINGS

When the driving and driven components of a system are connected with a torsionally-stiff coupling, high natural frequencies of torsonal vibration generally result, with the consequence that the entire system is torsionally sensitive to the higher-frequency (and generally more serious) torque harmonics emanating from either or both of the driving and driven equipment. Depending on many factors, including the mass-elastic configuration, the natural frequencies, excitation and damping levels, these exciting harmonics often cause serious critical speeds to fall within the operating range of the system.

Fig. 1 illustrates the level of vibratory torque on the reduction gear pinion in an actual diesel-geared propulsion system when it utilized a torsionally-rigid coupling between the engine and reduction gear. The early and repeated failures of bearings in the reduction unit, teeth breakage, pinion shaft breakage, loosening of gear and clutch hub fits, oscillation of the engine on its bed structure, were continually being reported, all of these being due to the very high level of vibratory torque in the gear box. Tooth separation, with its attendant high noise level, occured below 600 rpm.

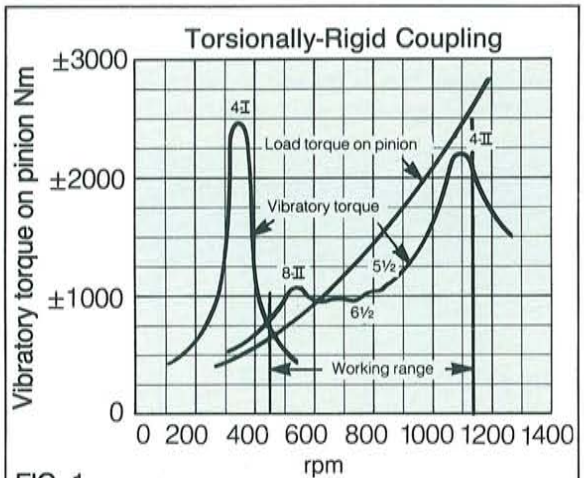


FIG. 1

After refit with a Lo-Rez steel-spring flexible coupling, the torsional vibration characteristics were altered to those shown in Fig. 2. The much lower tuning (or natural frequency) of the engine-gear mode of vibration — the basic 'interactive' mode — shifted the 4-II critical and adjoining peaks down below the minimum operating speed of 450 rpm. The maximum vibratory torque on the pinion was reduced from ±2200 to ±125 Nm at the maximum operating rpm. All difficulties with the system were terminated, including the roughness of the engine on its bed structure.

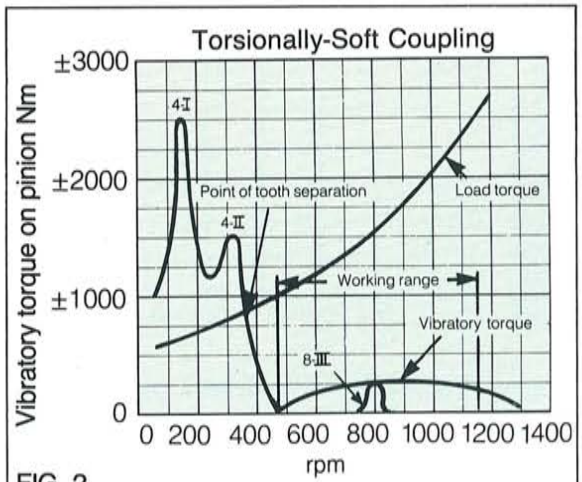


FIG. 2

In addition to the re-tuning of the system to a much lower coupling- or 'interactive'- mode (the one which involves essentially the engine masses in counter phase to the gear box masses), the torsionally-soft coupling effectively isolated the exciting source (the engine) from the remainder of the system. The Lo-Rez steel-spring coupling is a true 'isolating' coupling, depending primarily on torsional flexibility rather than on high levels of torsional damping.

This torsional isolation offered reciprocal benefits. The driven system was 'immunized' against engine vibration. Also the desired natural frequencies of the engine/crankshaft system were not altered by the masses and elasticities of the driven system. In this way, deterioration of the engine's viscous damper performance and resultant increase in crankshaft vibratory stress were effectively prevented.

One important practical aspect of many low-tuned geared and clutched power transmission systems is the fact that since the system is declutched before shutting down or starting up, the major critical, as shown in Fig. 2, exists only in the sense of a forced vibration. It's declutched position and magnitude shift in accordance with the neutral torsionals.

C. BENEFITS OF LO-REZ TORSIONALLY-SOFT COUPLINGS WITH CONSTANT STIFFNESS

Lo-Rez steel-spring couplings maintain constant torsional stiffness independent of load or power torque, vibratory torque (within the rating), vibratory frequency, temperature or age. Together with the accuracy of the specified stiffness, the constancy provides significant simplification of computer evaluation both during the design stage and for any subsequent diagnostic requirement.

Even the selection of the appropriate flexible coupling for a system is simplified. These aspects are of paramount importance particularly in variable speed systems, the result being much enhanced integrity of the system.

Fig. 3a shows a typical dynamic stiffness/mean torque relationship for both a variable- and a Lo-Rez constant-stiffness coupling. At 100% rated torque the dynamic stiffness of 'B' coupling is typically 3.5 times as high as it is at 25% torque; while the Lo-Rez steel-spring coupling remains constant over the same torque range. Fig. 3b shows the vastly different effect of the two couplings upon the torsional vibration characteristics in a typical variable speed propulsion system where the transmitted power torque varies as the square of the rotational speed, as with a fixed-pitch propeller.

Since both couplings have the same stiffness at 25% of rated torque, the major critical is located at 300 rpm in both cases, which in this case is suitable for 400 rpm idling operation. This critical location remains fixed for the Lo-Rez coupling, and at full load and 800 rpm the vibratory torque on the reduction gear pinion is ±268 Nm. For the variable-stiffness coupling, the 3rd order critical shifts up to 565 rpm when the system runs at 800 rpm, at which point the dynamic stiffness is 1.19×10^6Nm/Rad. The vibratory torque on the pinion becomes ±1342 Nm at 800 rpm, an increase of some 500% over that which occurs with the constant-stiffness coupling. If coupling 'B' had substantial damping, the vibratory torque at 800 rpm would be increased even further. At 800 rpm the vibratory torque on the Lo-Rez coupling itself is ±450 Nm, and on the variable-rate coupling 'B' it is ±4180 Nm, an increase of over 900%.

This condition would be accentuated for a system where the dynamic stiffness of the coupling increased faster than 4/1 or 5/1 with mean torque. The initial low-torque tuning would have to be much lower with coupling 'B' to provide the same effects as the constant stiffness coupling does at 800 rpm. Obviously the question of accuracy in the published torque/stiffness characteristics of couplings is also of paramount importance in these considerations.

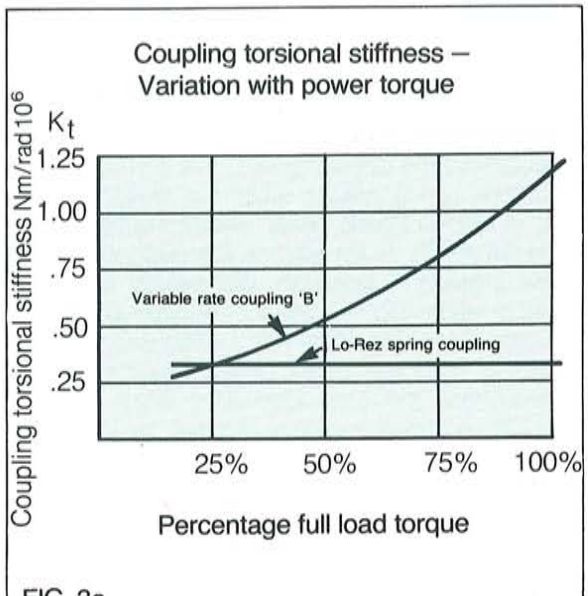


FIG. 3a

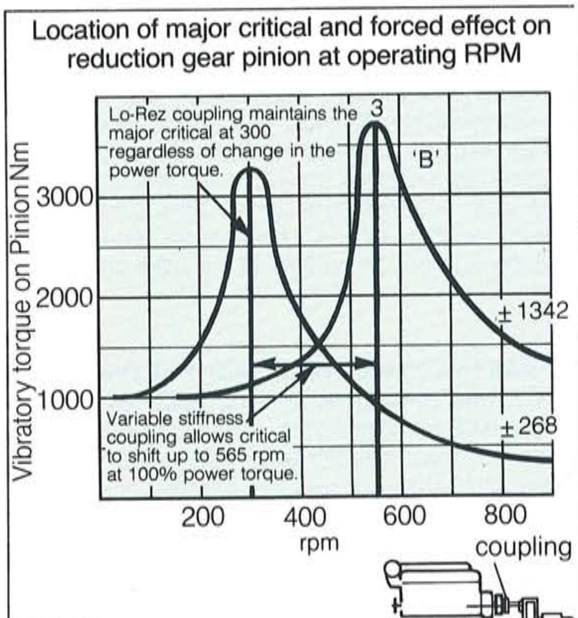


FIG. 3b

D. TRANSIENT AND CONTINUOUS VIBRATORY TORQUE RATING

The Lo-Rez steel-spring couplings shown on pages 9 to 14 are provided with an allowable vibratory rating of 20%, as illustrated in Fig. 4. However, this rating can be increased to 90% where required. This is accomplished by utilizing spring designs which can accept a higher ratio of vibratory/static stress without sacrificing adequate fatigue life. In the process, the torsional stiffness of the coupling is increased. Alternatively the higher vibratory ratings can — in effect — be made available by selecting a coupling which has a rated torque well in excess of the mean power torque requirement. It is recommended that each application, where a higher than 20% continuous value is required, be referred to Lo-Rez for more accurate determination.

Transient torque capacity varies from 3 to 5 times the maximum housing rating, depending upon the system involved and its operational requirements. During traversal of high-amplitude criticals, the springs do not become overloaded because of the cushioned limit stops located inside the main springs. Spring design accommodates any requirement of continuous vibratory torque superimposed at full rated torque — whether 20% or 90%. This variableness is based on conservative fatigue theory allowing for the effect of combined steady and alternating stress components, stress concentration and an adequate fatigue safety factor. Main springs are designed for unlimited life when carrying their mean rated load. An overload factor of 10% is additionally in-built together with a continuously superimposed vibratory load of 20% to 90% of the rated mean load.

E. COMPUTATIONAL TECHNIQUES FOR EVALUATION OF SYSTEM TORSIONAL CHARACTERISTICS

Several different methods are in use for the evaluation of torsional characteristics in power transmission systems. In the initial stages of coupling selection or of system evaluation, Lo-Rez utilizes the energy-balance concept in conjunction with undamped Holzer tabulations for the purpose of calculating resonant amplitudes of motion, stress and torque. The forced (non-resonant) effects are derived generally by forced-undamped Holzer-type tabulations, but occasionally the 'equilibrium amplitude' method is used. The reduction of power train systems to two or three mass equivalent systems for computational simplification is not favored, rather the full unreduced mass-elastic system is preferred.

For final and more rigorous system evaluation, Lo-Rez utilizes a complex forced-damped computer program which permits automatic computation of the resonant and forced effects upon any portion of the system and allows the synthesis or compounding of many different harmonic effects. The program accepts rpm-dependent excitation and damping for any or all masses and shafts in simple or complex branched and geared, multi-engined systems. Fig.5 shows computer-plotted forced-damped results for an actual propulsion system which originally utilized a torsionally-stiff coupling and was

Relationship between peak transient, continuous vibratory, and rated torques in low-tuned power transmission drives

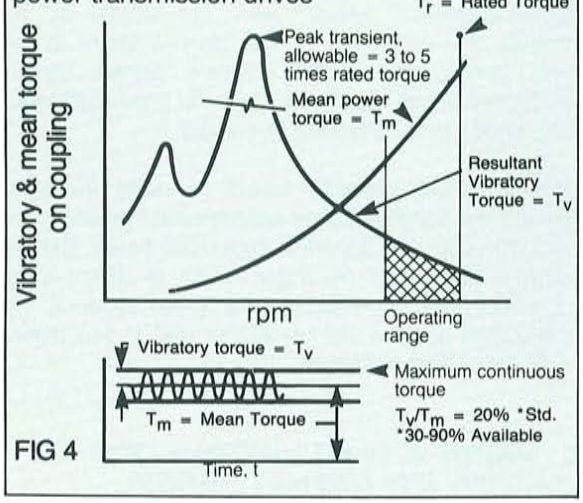


FIG 4

Torsional vibration comparison of torsionally rigid and torsionally soft couplings

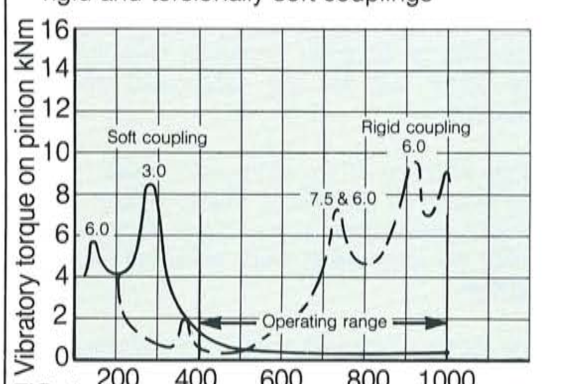


FIG.5

later modified to accept a Lo-Rez steel-spring coupling. In this case 13 different harmonic excitations from half to ninth order are included.

Various techniques for system damping evaluation are utilized by different torsional analysts, but all of these techniques should produce essentially the same end result. It is common to analyze for several different harmonic orders in two or three modes of vibration at least. Single and multiple cylinder-firing inequality, inequality in torque loading of compressor cylinders, inequality of propeller blade (1st order) excitation due to blade damage or tracking error, are some of the analytical refinements considered in the final evaluation.

F. DAMPING THEORY FUNDAMENTALS

1.0 Nomenclature:

		Unit
J	Polar (weight) moment of inertia (Wr^2)	kg m²
I	Shaft sectional polar inertia ($\pi.D^4/32$)	cm⁴
K_t	Torsional stiffness (coupling, shaft)	Nm/rad
T_m	Mean or power torque	Nm
T_r	Rated torque (of coupling)	Nm
T_f	Friction torque of Lo-Rez steel-spring coupling	Nm
T_v	Harmonic vibratory torque	±Nm
T_e	Harmonic (alternating) exciting torque	±Nm
T_d	Harmonic (alternating) damping torque	±Nm
T_o	Maximum value of a harmonic torque	±Nm
θ	Instantaneous value of harmonic motion amplitude	±rad
θ_e	Equilibrium (or static) motion amplitude	±rad
θ_o	Harmonic peak (resonant) motion amplitude	±rad
$\Delta\theta_o$	Harmonic peak twist-motion amplitude (resonant)	±rad
C_v	Viscous damping factor (specific damping torque)	Nm sec/rad
C_h	Hysteresis damping coefficient (metals)	—
H_r	Hysteresis damping coefficient (rubber, etc.)	—
ω_{p_n}	Phase velocity of natural frequency	rad/sec
ω_p	Phase velocity of forcing frequency	rad/sec
ω_v	Vibration velocity ($=\omega_p\,\theta$); $=\omega_{p_n}.\theta_o$ at resonance	rad/sec
F_n	Natural frequency	cycle/min
W	Work or energy	Nm
W_i	Harmonic input work per cycle at resonance	Nm/cycle
W_d	Harmonic damping work per cycle at resonance	Nm/cycle
W_o	Maximum work at resonance ($=W_d$)	Nm/cycle
W_s	Strain energy due to twist (in a coupling)	Nm
DM	Dynamic magnifier ($=\theta_o/\theta_e$)	—
t	Time	sec
ψ	Relative damping ratio	—

2.0 Derivations

A harmonic torque, $\mathbf{T_o.Sin\,(\omega_p.t)}$ . . . **(1)**
acting in conjunction with a harmonic motion $\mathbf{\theta_o.Sin\,(\omega_p.t)}$ . . . **(2)**
either generates or absorbs a maximum amount of work per cycle

$$\mathbf{W_o = T_o.\pi.\theta_o} \quad . . . \text{(3)}$$

For an exciting or energizing harmonic torque T_e, the maximum work input at resonance

$$\mathbf{W_i = T_e.\pi.\theta_o} \quad . . . \text{(4)}$$

And similarly, the maximum work absorbed by a harmonic damping torque, T_d, at resonance

$$\mathbf{W_d = T_d.\pi.\theta_o} \quad . . . \text{(5)}$$

Since the damping torque $T_d = C_v.\omega_v$, and since $\omega_v = \omega_{p_n}.\theta_o$, then

$$\mathbf{T_d = C_v.\omega_{p_n}.\theta_o} \quad . . . \text{(6)}$$

and $$\mathbf{W_d = C_v.\pi.\omega_{p_n}.\theta_o^2} \quad . . . \text{(7)}$$

which is the very well-known viscous damping expression.

C_v is sometimes referred to as the 'damping coefficient', but the term 'factor' or 'specific damping torque' is more appropriate since C_v is not dimensionless. When the damping occurs between two masses of a system (such as in a flexible coupling), θ_o represents the maximum or peak relative amplitude, ie. the twist amplitude across the flexible coupling halves at resonance, so that

$$\mathbf{W_d = C_v.\pi.\omega_{p_n}.(\Delta\theta_o)^2} \quad . . . \text{(8)}$$

Both types of viscous damping (ie. mass-to-mass and mass-to-earth) are present and simultaneously operative in most multi-mass systems. For mass/mass damping, $\Delta\theta_o^2$ from the Holzer tabulation is used and for mass/earth damping, θ_o^2 is used.

The popular 'energy-balance' method of evaluating system torsional vibration characteristics consists of equating the input energy per cycle, W_i, for each mode of vibration to the summation of the damping work per cycle, W_d, for each component of the system which involves damping. The peak vibratory amplitudes (of motion, stress, torque) can thus be determined by utilizing undamped Holzer tabulations. The ramps or flanks of the resonant peaks can generally be determined with fairly sufficient accuracy by using forced-undamped tabulations or by the 'equilibrium amplitude' method, producing a diagram similar to that shown in Fig. 6.

Vibratory stress/rpm diagram for typical marine Diesel propulsion system.

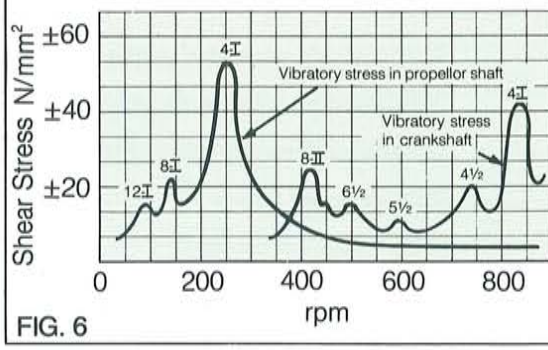


FIG. 6

Many vibration analysts prefer to utilize the dynamic magnifier, DM, in computing the resonant as well as the non-resonant effects. Below is shown the derivation for DM, at resonance only, for a flexible coupling having a stiffness K_t and a viscous damping factor C_v. See Fig. 7.

Equating input and damping energies;
$T_e.\pi.\theta_o = C_v.\pi.\omega_{p_n}.\theta_o^2$
actually $\Delta\theta_o^2$ should be used in the damping term, but here $\Delta\theta_o$ and θ_o are synonymous.
$\theta_o = T_e/C_v.\omega_{p_n}$

and $\mathbf{\theta_o = \theta_e.DM}$, by definition . . . **(9)**

and $\theta_e = T_e/K_t$, by definition

Therefore $DM = T_e.K_t/C_v.\omega_{p_n}.T_e$

And $\mathbf{DM = K_t/C_v.\omega_{p_n}}$ . . . **(10)**

The DM value for Lo-Rez steel-spring flexible couplings can readily be calculated from the friction damping torque values provided in Tables I and II, the calculation being considered in section H.

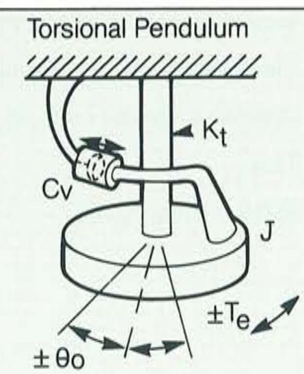


FIG. 7

Another relationship can be derived for the damping energy, the elastic strain energy and the dynamic magnifier, DM, in a flexible coupling, as arranged in Fig. 7.

The strain energy or work involved in twisting the coupling through an angle θ_o is the product of the average torque and the angle of twist;

$$W_s = \frac{K_t.\theta_o \,.\, \theta_o}{2} = \frac{K_t.\theta_o^2}{2}$$

The damping energy at resonance must equal the input energy, which is

$$W_i = W_d = T_e.\pi.\theta_o$$

The 'relative damping ratio' ψ can be defined as

$$\psi = \frac{\text{Damping energy per cycle}}{\text{Strain energy for } \theta_o} = W_d/W_s$$

$$\psi = \frac{T_e.\pi.\theta_o}{K_t.\theta_o^2/2} = \frac{2\pi.\ T_e}{\theta_o.\ K_t}$$

Since $T_e/K_t = \theta_e$ and $\theta_o/\theta_e = DM$

$$\frac{W_d}{W_s} = \frac{2\pi}{DM} = \psi \ldots \textbf{(11)}$$

$$\textbf{ie.} \quad DM = 2\pi/\psi \ .. \ \textbf{(12)}$$

Some coupling manufacturers provide the value of ψ from which the DM can be found.

G. OPTIMUM DAMPING LEVEL FOR TORSIONALLY-SOFT COUPLINGS

A high damping capacity within a torsionally-soft flexible coupling significantly reduces the vibratory torque across the coupling and within the driven system components, at the resonance of the coupling (or what we have called the 'inter-active') mode. However, there is generally an increase in the level of *forced* vibration effects in the coupling and in the driven system components in certain speed ranges above resonance. In some critical systems, very high levels of coupling damping may, therefore, not be desirable nor necessary.

In Fig. 8 the results of several computer runs on a typical geared system are shown. In the first two conditions the flexible coupling has constant stiffness but two different DM values. It is noted that a reduction of the coupling DM from 12 to 3 reduces the 3rd order steady state resonant vibratory torque at the pinion mesh from $\pm$11,500 Nm to $\pm$3500 Nm, while it increases the forced vibratory torque on the pinion mesh at 1000 rpm from $\pm$520 Nm to $\pm$835 Nm, which is not

Effect of coupling damping on resonant and forced vibratory torque on reduction gear pinion

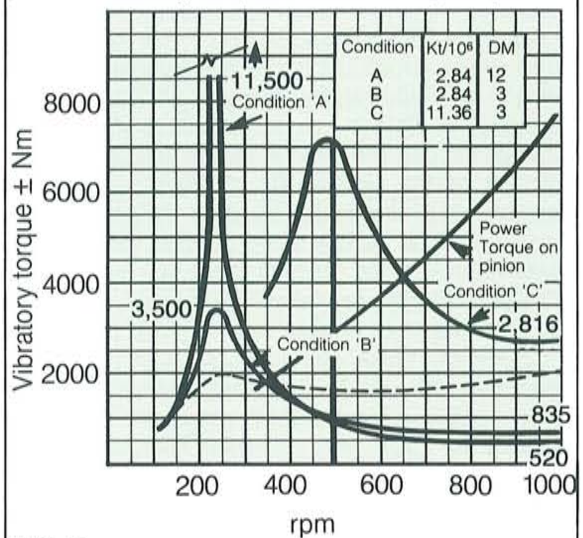


FIG. 8

problem in this particular system. But if the DM is reduced to 1.5 then the forced vibratory torque on the pinion increases to over $\pm$2000 Nm. If at the same time the coupling stiffness were to increase by 400% at full torque, as is typical for some elastomer couplings, then the DM of 3 results in a continually-superimposed vibratory pinion torque of $\pm$2816 Nm at 1000 rpm, while a DM of 1.5 increases it to over $\pm$4000 Nm.

From its inception, therefore, Lo-Rez has adopted an approach for which the objective is a conservative level of coupling damping, sufficient only to control the coupling mode resonance. This approach is discussed in the following section.

H. DAMPING IN LO-REZ STEEL SPRING COUPLINGS

1.0 Damping in Lo-Rez steel-spring couplings results essentially from the friction forces between rubber or composition damper blocks and the inside surface of the cover, these forces and the resulting damping torques being brought into play by the higher-amplitude twisting oscillation of the two coupling halves relative to each other during passage through the low-frequency critical speed(s) of the system. Fig. 9 shows the lug and damper detail. The friction damping available depends upon centrifugal loading of the damper block and the damper-spring force. Both of these can be varied to produce a special, desired damping effect, should the standard damping values, Table 1, not provide the requirements. Small clearances between the damper cavity side walls and the damper block allow the damper to be inactive when the twist amplitudes across the coupling are small, thus permitting improved isolation characteristics for the higher frequency modes.

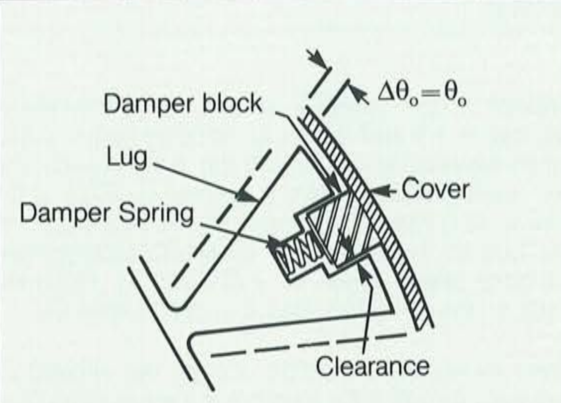


FIG. 9

A significant additional source of damping occurs when the oscillatory twist across the coupling is sufficient — during start up, shut down, or traversal of a major coupling mode critical — to cause contact and compressive flexing of the limit stop bumpers, Fig. 10. Passage through major criticals may involve from 6 to several hundred cycles of maximum peak vibration, the latter case more likely to occur in reciprocating compressor applications. Due to the good thermal conductivity of the coupling cover and the natural air circulation through the coupling, the damper friction heat involved can be dissipated quite adequately.

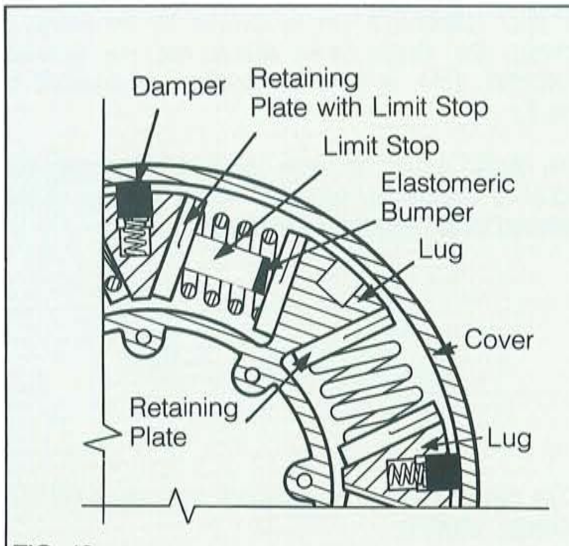


FIG. 10

2.0 Integral Viscous Dampers

Where the requirement exists for attenuation of the vibration levels of high-frequency modes which involve primarily the driven equipment sub-system (such as a reduction gear mode), Lo-Rez incorporates into one half of the steel-spring coupling a relatively small viscous shear damper. Such a damper can provide excellent control for the higher frequencies involved, without introducing mass-to-mass damping across the coupling halves and consequently deteriorating the isolation qualities of the coupling. Such a viscous damper arrangement is shown in Fig. 11.

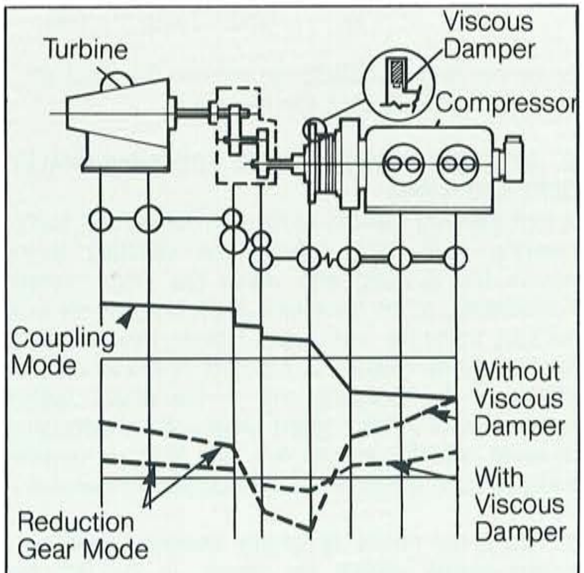


FIG. 11

3.0 Specific Damping Values

Studies have shown that friction damping of the type present in Lo-Rez steel-spring couplings begins to approach the characteristics of viscous damping. Since the friction damping energy in the coupling is but a portion of the overall damping in a system and since the vibratory motions of practical systems is basically sinusoidal, it is possible to derive an equivalent viscous damping factor. This can be done by equating the expression for friction work per cycle to the expression of viscous damping work per cycle, equation (8).

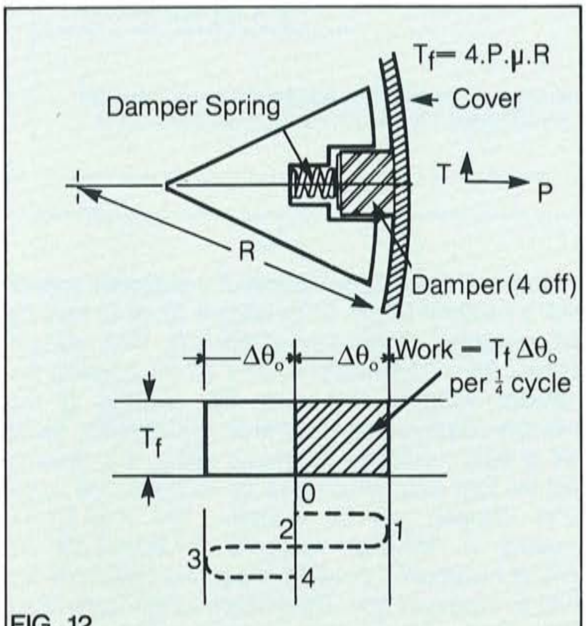


FIG. 12

Fig. 12 illustrates the damping work by friction as $T_f.\Delta\theta_o$ per quarter cycle.

Thus $4.T_f.\Delta\theta_o = C_v.\pi.\omega_{Pn}.\Delta\theta_o^2$

and $C_v = \dfrac{4.T_f}{\omega_{Pn}.\pi.\Delta\theta_o}$

= equivalent viscous damping factor

For passage through major criticals, $\Delta\theta_o$ at resonance will generally build up to the maximum value afforded by contact with the coupling limit stops.
This value of $\Delta\theta_o$ =

$$\frac{\textbf{1.20* × Coupling torque rating}}{\textbf{Coupling torsional stiffness}} \quad \ldots (14)$$

(* for 30% to 90% vibratory rating, this factor varies from 1.30 to 1.90)

By utilizing this maximum value of $\Delta\theta_o$, C_v is smallest and so the approach is conservative, yielding the minimum of coupling damping work when the C_v value is used in the energy-balance (summation) calculations for the entire power train system, or when C_v is used in a forced-damped computer program.

T_f values for Lo-Rez single row couplings are shown in Table I. Double row couplings have T_f values 100% higher, unless otherwise specified. Extra damping (×2, ×3 standard) can be provided for couplings in the 50% and higher torque brackets, where the DM is inherently higher than it is for couplings in the lower torque ranges.

An alternative and useful derivation for DM of the Lo-Rez coupling follows;

$DM = K_t/C_v.\omega_{Pn}$ . . . from (10),
Substituting the value of C_v from (13)
$DM = K_t.\pi.\Delta\theta_o/4.T_f$
Now since $\Delta\theta_o = 1.2\ T_r/K_t$ . . . (14)
$\mathbf{DM = .94\ T_r/T_f}$ **. . . (15)**

Within each coupling housing size (E, F, etc.) the coupling having the lowest torque/stiffness rating will also have the lowest DM value.

For example, coupling F 4.73/.033, when passing through a resonant speed of 300 RPM (regardless of order no.), will have a DM value of
.94 × 534/87 = 5.77 at resonance, whereas coupling F 116/3.87, at the same rotational speed has
DM = .94 × 13,100/87 = 141 at resonance.

During traversal of major critical speeds during start up, shut down, or in coming up to an operating speed range; it is usual for the elastomer limit stop bumpers to be impacted, and it is on this basis that equation (14) and (15) are derived. The compressive flexing of the bumpers results in additional damping, the effect of which has been determined for various combinations of "% maximum rpm" and "% maximum torque" at which the coupling is selected. This additional damping is expressed as a dynamic magnifier adjustment factor, these factors being shown in TABLE II.

Total Friction Damping Torque, T_f, (Nm)

coupling rpm	D	DE	E	F	G	H	I	J	K	L	M	N
	COUPLING SIZE (SINGLE ROW)											
100	28	37	47	73	88	108	148	212	369	713	1 510	3 400
200	28	38	49	79	102	143	243	445	987	2 280	5 410	12 900
300	29	40	52	87	125	201	394	831	2 020	4 900	11 900	28 700
400	31	44	57	100	150	290	610	1 370	3 440	8 560	21 000	
500	33	48	63	116	197	386	887	2 070	5 310	13 300		
600	36	53	70	136	249	514	1 230	2 920	7 590			
700	39	58	78	159	309	663	1 630	3 930				
800	42	64	87	186	377	853	2 110					
900	46	72	98	217	455	1 040						
1 000	51	80	110	251	542							
1 100	56	90	124	289								
1 200	61	99	138	329								
1 300	67	111	155									
1 400	73											

NOTE: DAMPING IN DOUBLE ROW COUPLINGS = 2 × VALUES SHOWN

Upper rpm values correspond to approximately ⅔ of maximum rpm. Above this point, coupling damping is generally not of concern.

TABLE I

Where preliminary torsional vibration calculations on a system disclose a low-mode resonant torque on the Lo-Rez coupling which is in excess of 1.2× rated coupling torque (for 20% vibratory rating), then the DM values which were derived from equation (15) in conjunction with Table I — and which were used in the preliminary torsional calculations — should be adjusted for the factors shown in Table II, and the calculations refined.

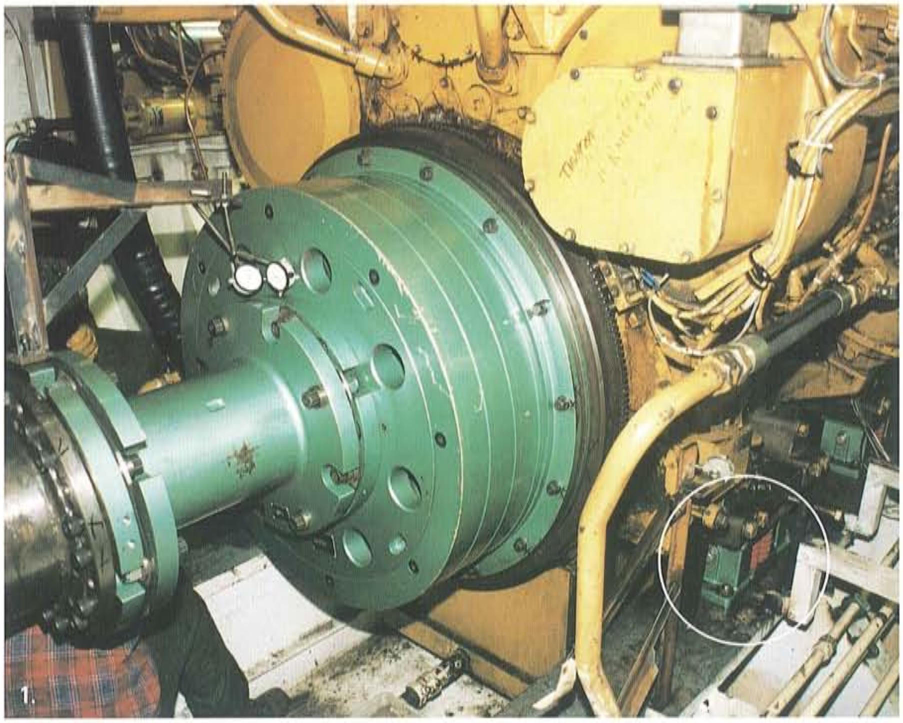
Dynamic Magnifier Adjustment Factors

% OF MAX. TORQUE \ % OF MAX. rpm	10	20	30	40	50	60
10	.80	.83	.86	.89	.92	.95
20	.70	.73	.80	.86	.85	.90
30	.66	.70	.76	.81	.83	.88
40	.62	.67	.72	.77	.82	.86
50	.58	.64	.68	.74	.80	.84
60	.55	.60	.65	.71	.77	.83
70	.52	.58	.64	.69	.75	.81
80	.50	.56	.62	.68	.74	.80
90	.48	.54	.60	.66	.72	.78
100	.46	.52	.58	.64	.70	.76

TABLE II

Example: A preliminary analysis of a proposed system with a Lo-Rez H 97.5/1.57 coupling is found to have its major coupling-mode critical at 400 rpm. Then, using a DM of (.94 × 11,000/290 =) 35.7 for the coupling the resonant vibratory torque on the coupling at the 400 rpm critical is determined to be ±20,000 Nm. Since this is 80% more than the torque rating, it is obvious that the limit stop bumpers will be contacted and some extra damping will be available. The H 97.5/1.57 coupling at 400 rpm represents (11,000/30,600 =) 36% of maximum H coupling torque, and (400/1400 =) 29% of maximum rpm. The DM adjustment factor from Table II is .74 so that a more realistic DM is (35.7 × .74 =) 26.4.

Subsequent adjustment to the system energy calculations, or forced-damped calculations, are found to reduce the vibratory torque across the coupling (and on other components of the system) by 20% to ±16,000 Nm at 400 rpm resonance.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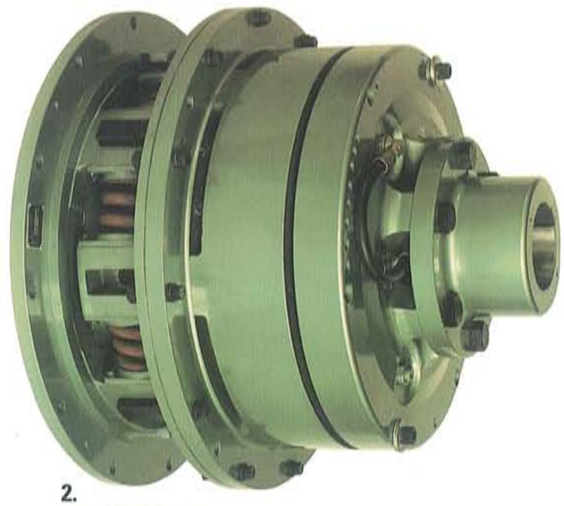
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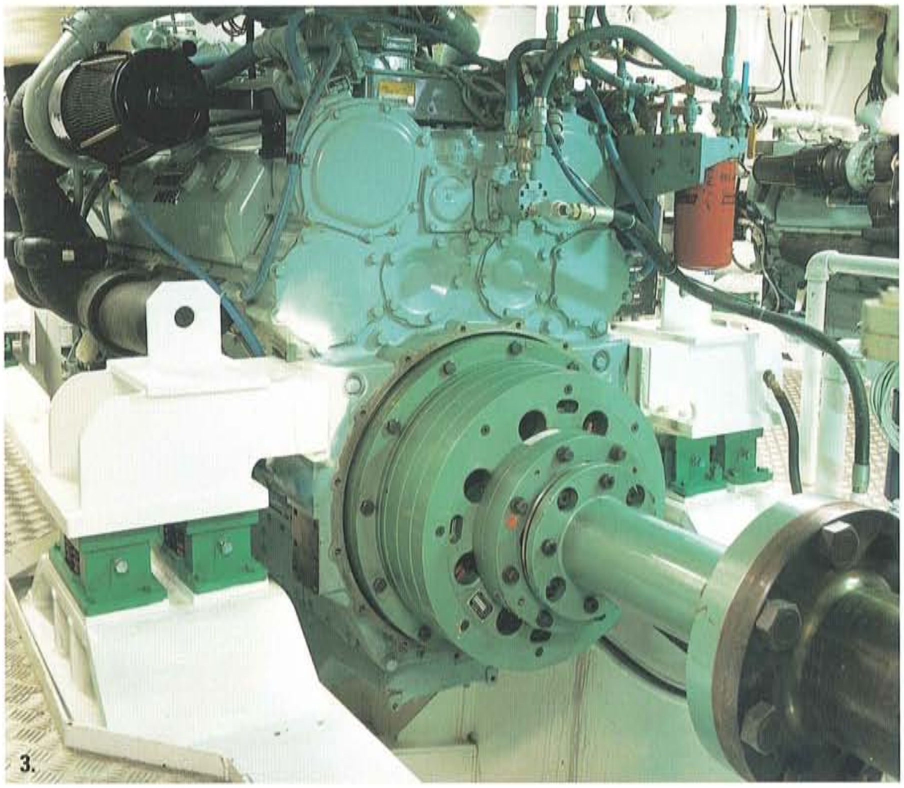
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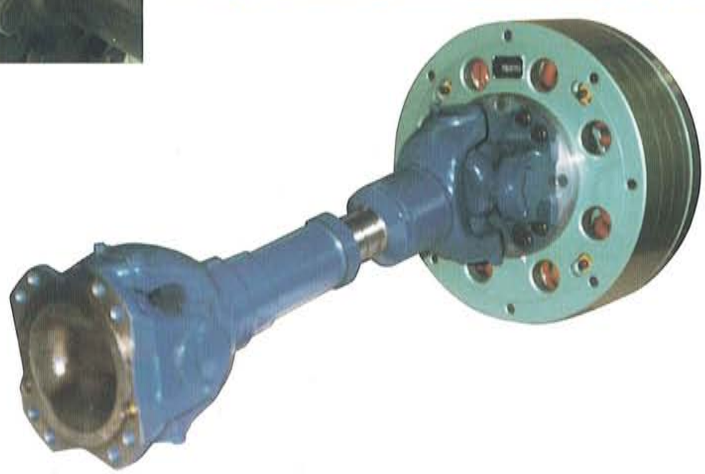

6.

1. Lo-Rez I/SF and 16HLF Couplings
Application: 4000 HP Marine Propulsion Engine Resiliently Mounted on Lo-Rez Isolators

2. Lo-Rez F/FF Coupling with Clutch

3. Lo-Rez F/SF and 8HLF Couplings
Application: 1500 HP Marine Propulsion Engine Resiliently Mounted on Lo-Rez Isolators

4. Lo-Rez I2/HF Coupling and Torque Limiter
Application: 5000 HP Reciprocating Compressor

5. Lo-Rez I2/HFH Coupling
Application: 5000 HP Reciprocating Compressor

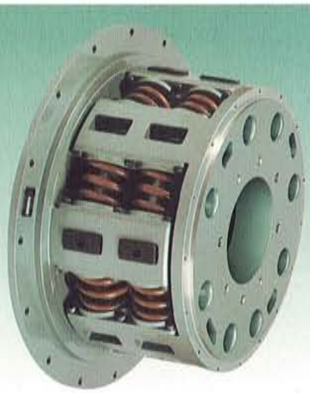
6. Lo-Rez E/HH Coupling with Cardan Shaft
Application: Marine Propulsion Engine

LO-REZ VIBRATION CONTROL LTD. has been dedicated to the design and manufacture of vibration control equipment since the 1950's. Our commitment to research and design as well as high quality and exacting standards in manufacture, means we are meeting the challenge of new technology with systems that are state of the art in the 1990's.
For example the LO-REZ SOFT-MOUNT® SYSTEM, in place in over 375 marine propulsion applications around the world, produces typical vibration isolation efficiency of 97% with noise levels of 62-70dBA.
Committed to system performance LO-REZ provides extensive services and test facilities, providing certification prior and subsequent to overhaul when required.
Comprehensive technical specifications and performance data is available upon request on all LO-REZ systems and components.

BR-T STEEL-SPRING VIBRATION ISOLATORS

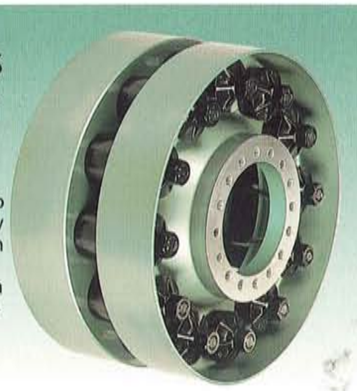
High-efficiency control of vibration produced by diesel propulsion engines in ships, trains and industrial equipment.
A number of models and sizes, as well as a non-propulsion series are available to accommodate specific load requirements.
An integral part of the Lo-Rez Soft-Mount® System.

SS STEEL-SPRING FLEXIBLE COUPLINGS



Providing low, constant and accurate torsional stiffness (±8%) for precise tuning control in geared propulsion, reciprocating compressor and other critical systems. Features include; no lubrication, easy inspection of internal working parts and easily adjustable damping.
A number of models (including a single row series) and sizes are available to suit specific system requirements.
(Shown with cover removed.)

RT STEEL BOLTED RUBBER COUPLINGS

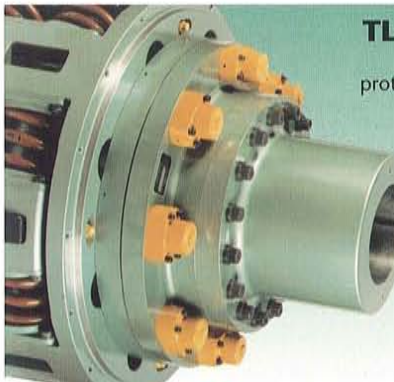


Torsionally flexible and capable of accommodating axial load, these couplings are ideal for any propulsion application. Featuring; reverse thrust capability, noise attenuation, no thrust bearing requirement, low stiffness, non-lubricated and high damping.
Available in various sizes and an integral part of the Lo-Rez Soft-Mount® System.

HLF LAMINATED DISC COUPLINGS

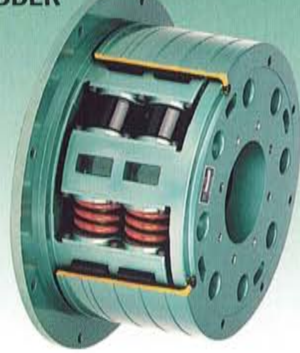
These High Lateral Flexibility disc couplings significantly impede a propulsion engine's linear/torsional vibration and noise from entering any hard mounted gear box.
*(Shown here attached to a Lo-Rez **RT** coupling and its integral spool spacer).*

TL TORQUE LIMITERS



Providing optimum overload protection. With low maintenance and easily reset, these torque limiters keep downtime to a very minimum.
A number of models and sizes are available to suit any system requirement.
(Shown here attached to a Lo-Rez Steel-Spring Flexible coupling/cover removed).

SS-R STEEL-SPRING/RUBBER FLEXIBLE COUPLINGS



A variation of the steel-spring coupling, incorporating a high damping ability while retaining some torsional stiffness accuracy (±15-20%) for tuning control. Features include; no lubrication, easy inspection of internal working parts and easily adjustable damping.
A number of models (including a single-row series) and sizes are available to suit specific system requirements.

VTD VISCOUS TORSIONAL VIBRATION DAMPERS

Tuned, double tuned and non-tuned patented Viscous Dampers. Available in a wide range of sizes, providing optimum damping coefficients, custom designed particularly to suit custom requirements. A vital part of crankshaft, gear train and bearing protection.

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