VIEWPOINT

Dear Editor:

Thank you for the very good magazine on gears. It is very informational, contains a wealth of trade information, but still maintains a high level of papers, unfortunately not typical for other trade publications.

Eugene I. Rivin
Professor
Wayne State University

Questions From the Industry

Dear Sirs:

We are interested in purchasing a computer program which would enable us to undertake the following:

1. Plot involute gear tooth profile forms.

2. Indicate the interference patterns generated with pinion cutters.

3. Show the action of undercutting for varying numbers of teeth; all produced from mathematical input data.

We shall be pleased to learn if you can be of help to us in this respect.

Llewellin's Machine Co. Ltd.
M.S. Horlick
15 Kings Square
Bristol BS2 8JJ
England
(0272) 424026/9

Gentlemen:

We have a small job shop type gear company. We have recently begun pursuing a computer system to help us in our gear calculations. Due to the time involved in figuring formulas pertaining to various gears on paper, we are looking for programs to use on a computer system.

We are asking if you know of any computer program that

may be available pertaining to gear formulas.

Also, we have a Gould & Eberhardt 48H Gear Hobber and we are interested in a computer program for changing gear calculations for hobbing helical gears non-differentially. This process is often time consuming to do on paper, and we think that somewhere, someone has a program to simplify this process.

Any help or suggestions you can offer would be greatly appreciated.

Bethlehem Gear & Machine Company
Robert A. Schrum
P.O. Box 157
Moundsville, WV 26041
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Gear Couplings

Re the article of Mr. Stan Jakuba: "Give your Gears a Break—Select the Right Coupling!" May/June, '87 issue and the contributions in July/August, '87 issue by Mr. Michael M. Calistrat and Mr. Stan Jakuba.

The following comments of experts in the field of torsional vibration calculation (T.V.C.) will prove that the selection of the right torsionally soft coupling (right elasticity, spring characteristic and damping) is not "black magic" anymore, but an easy task for an engineer having a highly sophisticated computer program and the right input data for all components of the transmission. The right coupling for a geared diesel generator set can be selected in a few minutes. Provided the governor of the engine is stable in the system, the coupling will last for ten or more years. A fluctuating torque monitoring device as developed by the signer can be helpful, especially for larger sized transmissions, as shown below even for rather complicated systems with several branches. The signer has done the T.V.C. for 13 US Navy support vessels with high powered diesel geared installations (300,000. HP total).

As a practical example, Fig. 1 shows a main and auxiliary drive for a large sized shuttle tanker (125,000. tdw). The following components are used. The main diesel two-stroke, slow-speed engine (MAN - B&W, 5 cylinder, 6526 kW at 102 rev) is driving a propeller through a reduction gear, which also includes a PTO (Power Take Off) for a hydro power package (5200 kW at 1200 rev) for discharging the crude oil. Between the main engine and the reduction gear a very soft coupling is arranged. On the forward end of the engine a generator drive (1200 kW at 1800 rev) via a speed increasing gear is attached, and a turbine driven by the exhaust gas (550 kW at 1800 rev) is incorporated. The generator is runnng at constant speed even at variable engine speed using a RCF (Renk Constant Frequency) gearbox. For eight different modes of operation for this vessel, an optimized coupling has to be

COMPLETE SYSTEM

- ○ principal locations of forced vibration calculation
- ↻ supplementary locations of forced vibration calculation
- ● recommended locations for torque measurement at sea trials
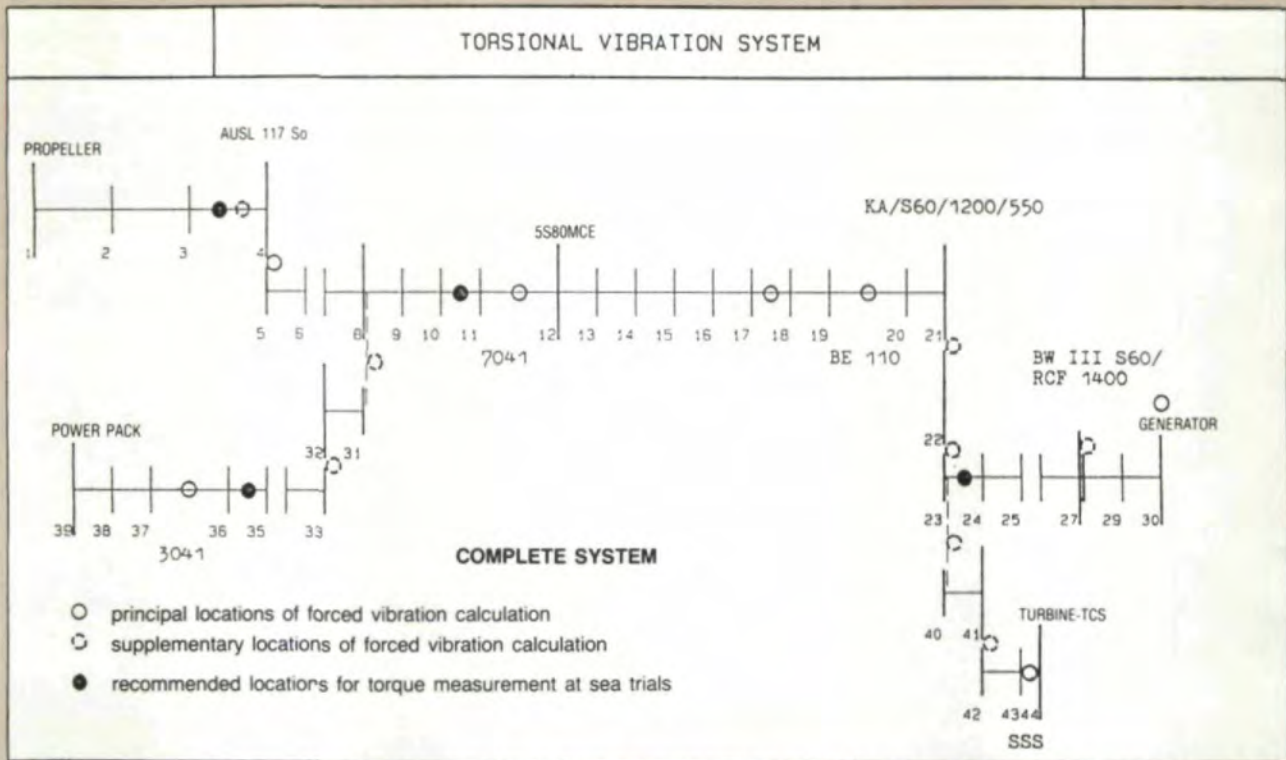
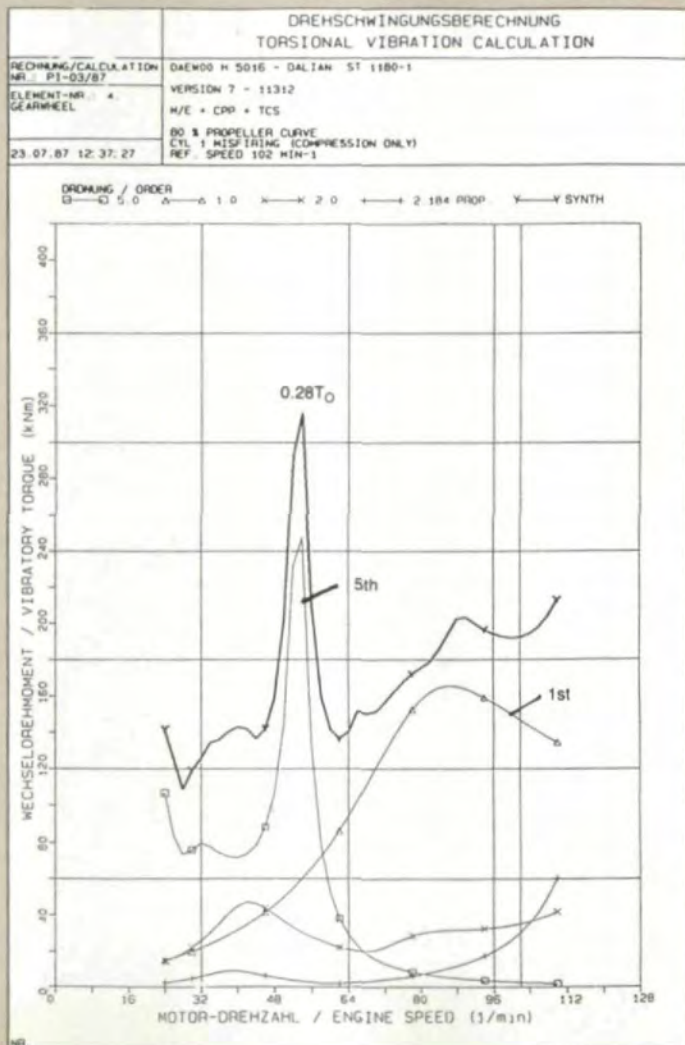Fig. 1 — Scheme of the torsional vibration system for a crude oil shuttle tanker.

Fig. 2 — T.V.C. for misfiring situation. Cylinder 1 has only compression. Main engine is driving propeller and turbine. The 5th order and 1st order of engine are dominant.

selected, it being impossible to shift all critical resonances out of the speed range.

Fig. 2 shows a typical example for a misfiring situation (only combustion), which is defining the worst situation for the installation. The first order dominant at misfiring and the fifth order dominant for this five cylinder engine create fluctuating torques which at certain modes of operation may exceed given permissible values for either resilient coupling material or the gears. Twenty years of experience in T.V.C. have built up so much knowledge in selection of the right components for transmisions in marine and industrial application that the points discussed by Mr. Jakuba and Mr. Calistrat are not a serious subject in my daily routine any more.

Dr. Wilhelm Schaeffer
Renk Tacke GmbH
Augsburg, West Germany

KHV PLANETARY GEARING . . .

and the strength of the equal velocity mechanism. Most materials for the CYCLO are high carbon chrominum steel with high hardness, which requires an elaborate heat treatment and an accurate grinding. The KHV has involute teeth with a concave to convex contact (internal gear contacts external gear), which results in a high contact strength. The internal gear has a very strong bending strength, and the external gear is modified, which increases the bending strength. Therefore, the KHV can use conventional gear materials and common accuracy. Usually, the ring gear is carbon steel without grinding, and the planet is either carbon or alloy steel. All the gears of the KHV can be generated by standard gear cutters and conventional equipment.

Compared to the CYCLO, the KHV is easier to manufacture and less expensive. Therefore, it is a very promising gearing.

Many thanks to Professor Ali A. Seireg of the University of Florida and the University of Wisconsin for his review of this article.

References

1. DUDLEY, D. W., et al. *Gear Handbook*, McGraw-Hill Co., 1962.
2. MACMILLAN, R.H. "Epicyclic Gear Efficiencies", *The Engineer*, Dec. 23, 1949, pp. 727-728.
3. MACMILLAN, R.H. "Power Flow and Loss in Differential Mechanisms", *Journal of Mechanical Engineering Science*, Vol. 3, No. 1, 1961, pp 37-41.
4. MOROZUMO, MUNEHARU. "The Efficiency and the Design of Modified Internal Gearing", *Mechanical Researches* (Japanese), No. 5, 1970, pp. 720-726.
5. KUDLYAVTZEV, V.H., "*Planetary Gearing*" (Russian), Machine Building Publishing House, 1966.
6. YU, DAVID. "Problems Arising in the Calculation of Gearing Efficiency", *Liaoning Machine* (Chinese), No. 2, 1980, pp. 33-42.
7. YU, DAVID and N. BEACHLEY. "On the Mechanical Efficiency of Differential Gearing", Transaction of the ASME, *Journal of Mechanisms, Transmissions, and Automation in Design*, Vol. 107, Mar. 1985, pp. 61-67.
8. *Catalog of Equadyne Planetary Speed Reducers*, Ross Industries, 1985.
9. *Catalog of Andantex Reducers & Differentials*, Andantex USA., Inc. 1983.
10. TUPLIN, W. A., *Gear Design*, American Edition, The Industrial Press, 1962.
11. *Catalog of CM-Cyclo*, Sumitomo Machinery Corp. of America, 1987.
12. SPOTTS, M.F. *Design of Machine Elements*. 6th edition, Prentice-Hall, Inc., 1985.
13. OBERY, ERIK, et al. *Machinery's Handbook*. 22nd edition, International Press, Inc., 1984.
14. YU, DAVID, "On the Interference of Internal Gearing", *Proceedings Second World Congress on Gearing*, Paris, Mar., 1986, pp. 317-326.

VIEWPOINT

ERRATA

Editors Note: We apologize that several errors appeared in recent issues of GEAR TECHNOLOGY. We regret any inconvenience these errors may have caused.

In the article **"Longitudinal Load Distribution Factor for Straddle- and Overhang-Mounted Gears"** by Toshimi Tobe, et al, appearing in our Jul/Aug issue, the following errors occurred:

Equation 17 should read

$$k \equiv (P_n \, /b)/[w_1 + w_2 + w^p]_{max}$$
$$k = K_{H\beta}{}^{-0.96}[k] = K_{H\beta}$$

Equation 19 was ommitted. It should read,
When both tooth surfaces just come into contact at $\eta = \eta^*$, the position η^* is obtained from

$$\frac{ds_o(\eta)}{d\eta}\bigg|_{\eta=\eta^*} = 0$$

as follows:

$$\eta^* = \frac{1}{e_1 c_2^2 + e_2 c_1^2}\left\{ e_1(f_1+c_1)\, c_2^2 + e_2\,(f_2+c_2)c_1^2 - \frac{c_1^2\, c_2^2}{2_{b2}}\, e_{eq}\right\}$$

In Equation 20, the figure $e_2\, c_2^1$ should read $e_2\, c_1^2$
The letter ϵ in Figs. 9 and 15b should read ξ

The first equation in the footnote on the bottom of page 16 should read $K_{F\beta} = K_{H\beta}^N$

In the note in Table B.1 the symbol IG/in should read lb/in.

In the caption for Fig. 14, the last equation should read $c_1/b_1 = 0.5$, $e_o^* = 2\,0\mu m$.

In the last two paragraphs on page 46, Fig. B.1 is incorrectly labelled A.1.

In the Sept/Oct issue, on page 50 of Stan Jakuba's **"SI Units — Measurements and Equivalencies"** the figures 1kg-m under "Moment of force . . ." 1kg/dm under "Specific force of gravity" and 1 kg-m/rad under "Spring rate: torsional" all refer to kg force, a symbol sometimes written as kp.

In this same issue, page 47 was incorrectly laid out. The table of terms at the top of the page and Equations 3 and 11 immediately below it both are part of the article, **"Selection of a Proper Ball Size . . ."** by Van Gerpen and Reece, continued from page 34.

Equation 3 should read:

$$PACB = K^*\left[\frac{\pi}{Z} - \left[\frac{(\,BTN}{\cos\,(BHA)\,^*BD)}\right] + K^*Inv(PACP)\right] + PACP) \quad (3)$$

The remainder of the equations on page 47 (Nos. 12-19) belong to Paul Dean's article, **"Interrelationship of Tooth Thickness Measurements as Evaluated by Various Measuring Techniques,"** continued from page 23.